限界熱流束の研究

RESEACHES ON CRITICAL HEAT FLUX

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限界熱流束（Critical Heat Flux）は，原子力，超電導等の先端技術に関連しつつ，特にエネルギー問題と関連を持つ重要な現象である。また同時に，たとえば液体燃料ロケット燃焼室の冷却など，他の各種技術と深い関係を持つ現象でもある。

このたび文部省科学研究費・エネルギー特別研究の中で高位有効エネルギーの利用限界に関する研究（Studies on Utilization Limits of Available Energy at High Temperature Levels）の研究班が発足することとなり，当研究室でも，強制流動系における限界熱流束および除熱特性の統一的把握に関する研究を分担協力することになった。

これまでささやかながら当研究室では，限界熱流束に関する基礎的な研究を続けて来ていたが，その結果は各種文献に散在している。そして，このままの状態では，今後，限界熱流束について特に集中的な研究を進めて行くうえで非常に不便を感じる。そこで，上記の研究班内での基礎情報交換．また当研究室で本研究に協力してくれる大学院学生諸君への既存情報伝達，およびこれらに類することを主要な目的として本冊子を準備することとした。いうまでもなく本冊子には，当研究室でおこなって来た諸研究のうちで，限界熱流束に直接関係を持つ研究のみを収録している。

なお，これまでの研究に関係するもので，近く印刷が予定されているものに下記の研究がある。しかし本冊子の作製の目的からいって，これらの原稿を収録するまでのことはないので，そのままにしてある。

1）CHF of forced convection boiling in uniformly heated vertical tubes：Experimental study of HP−regime by the use of Refriqerant 12〔Int. J. Multiphase Flow〕.

2）A study on limiting exit quality of CHF of forced convection boiling in uniformly heated vertical channels〔J.Heat Transfer, ASME〕.

3）一様加熱垂直円管を通る飽和自由流動沸騰の限界熱流束〔日本機械学会論文集〕。

4）一様加熱垂直円管内の強制流動沸騰の限界熱流束：高圧領域で管長径比の小さい場合〔日本機械学会講演論文集〕。

5）一様加熱垂直流路の強制流動沸騰ＣＨＦの極限出口クオリティχ_{ex}°に関する研究〔日本機械学会講演論文集〕：普通講演のみ（印刷の予定なし）。

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目　　次

〔 〕内の文献略記号については目次末尾の（注）参照のこと。

FREE CONVECTION BOILING

EXTERNAL FLOW

I. Submerged state (Pool boiling)

1. Principal mechanism of boiling crisis in pool boiling 〔 IJHMT, 11 (1968), 993–1002 〕 …… 1

2. Mechanism of boiling crisis and transition boiling in pool boiling 〔 4th IHTC, Vol. V (1970), B3.2 〕 …… 16

3. Study of forces acting on a heated surface in nucleate boiling at high heat fluxes 〔 HTJR, 1 (3) (1972), 34–46 〕 …… 27

4. Behavier of a vapor mass in saturated nucleate and transition pool boiling 〔 HTJR, 5 (2) (1976), 45–65 〕 …… 38

INTERNAL FLOW

I. Horizontal clearance

5. Experimental study of nucleate pool boiling in case of making interference-plate approach to the heating surface 〔 3rd IHTC, Vol. 3 (1966), 219–227 〕 …… 59

6. Nucleate and transition boiling in a narrow space between two horizontal, parallel disk−surfaces 〔 JSMEB, 20 (1977), 638–643 〕 …… 68

7. Critical heat flux of saturated natural convection boiling in a space bounded by two horizontal co−axial disks and heated from below 〔 IJMF, 5 (1979), 219–224 〕 …… 74

FORCED CONVECTION BOILING

EXTERNAL FLOW

I. Submerged state (Parallel frow)

8. Critical heat flux of saturated convective boiling on uniformly heated plates in a parallel flow [IJMF, 6 (1980), 575–582] 81

II. Liquid film flow (Parallel and radial flow)

9. Study of the mechanism of burn–out in boiling system of high burn–out heat flux [JSMEB, 16 (1973), 1357–1366] 89

10. Study of mechanism of burn–out in a high heat–flux boiling system with an impinging jet [5th IHTC, Vol. IV (1974), 245–249] 99

11. Burnout in a high heat–flux boiling system with an impinging jet [IJHMT 21 (1978), 295–305] 104

12. Burnout in a high heat flux boiling system with a forced supply of liquid through a plane jet [6th IHTC, Vol. I (1978), 435–440] 115

13. Upper limit of CHF in the saturated forced convection boiling on a heated disk with a small impinging jet[ASMEJ, 101 (1979), 265–269] 121

INTERNAL FLOW

I. Vertical tube

14. A generalized correlation of critical heat flux for the forced convection boiling in vertical uniformly heated round tubes [IJHMT, 21 (1978), 1527–1542] 127

15. A generalized correlation of critical heat flux for the forced convection boiling in vertical uniformly heated round tubes–a supplementary report [IJHMT, 22 (1979), 783–794] 143

16. An analysis of the effect of inlet subcooling on critical heat flux of forced convection boilng in vertical uniformly heated tubes [IJHMT, 22 (1979), 1567–1575] 155
17. Toward the systematic understanding of CHF of forced convection boiling (Case of uniformly heated round tubes) [JUS, (1980), 53–60] 164
18. General features of CHF of forced convection boiling in uniformly heated vertical tubes with zero inlet subcooling [IJHMT, 23 (1980), 493–504] 172
19. Critical heat flux of forced convection boiling in uniformly heated vertical tubes (Correlation of CHF in HP–regime and determination of CHF–regime map) [IJHMT, 23 (1980), 1573–1580] 184
20. On the heat–flux/exit–quality type correlation of CHF of forced convection boiling in uniformly heated vertical tubes [IJHMT, 24 (1981), 533–539] 192
21. On the relation between critical heat flux and outlet flow pattern of forced convection boiling in uniformly heated vertical tubes [IJHMT, 24 (1981), 541–544] 199

Ⅱ. Vertical annulus

22. Generalized correlations of critical heat flux for the forced convection boiling in vertical uniformly heated annuli [IJHMT, 22 (1979), 575–584] 203
23. General features of CHF of forced convection boiling in vertical concentric annuli with a uniformly heated rod and zero inlet subcooling [IJMHT, 24 (1981), 109–116] 213

Ⅲ. Rectangular channel

24. General features of CHF of forced convection boiling in uniformly heated rectangular channels [IJHMT, 24 (1981), 1413–1419] 221

TWO－PHASE FLOW

I. Equations

25. On the construction of a set of equations to describe a compressible two-phase flow 〔NCTAM, Vol. 26 (1976), 401-412〕 …… 229

自 然 流 動 沸 騰

外 部 流

I. 潜水状態（プール沸騰）

1. プール沸騰バーンアウト機構の研究〔機論，34（昭43），345-354〕… 241

2. プール沸騰にバーンアウトはどうしておこるか〔機誌，71（昭43），82-92〕…… 251

3. 高熱流束沸騰蒸気塊の生成・切断挙動の研究〔東機報，第5巻（昭45），137-139〕…… 262

4. プール沸騰におけるバーンアウトおよび遷移沸騰の機構〔機論，37（昭46），535-545〕…… 265

5. 高熱流束プール核沸騰の加熱面近傍に作用する力の研究〔機論，38（昭47），1049-1055〕…… 276

6. 飽和プール核，遷移沸騰における蒸気塊の挙動〔機論，41（昭50），294-305〕…… 283

内 部 流

I. 水平隙間

7. 干渉板を加熱面に近づけた場合の核沸騰の実験的研究〔機論，32（昭41），948-958〕…… 295

8. 平行水平2円板間の狭い空間における核および遷移沸騰〔機論，42（昭51），2854-2861〕…… 306

9. 平行水平2円板間流路の自然流動沸騰の限界熱流束〔機論，45（昭54），1718-1722〕…… 314

Ⅱ. 垂直流路

10. 制限流路内の自然流動沸騰の限界熱流束の無次元整理について〔機論，44（昭53），3908–3911〕…… 319

11. 垂直二重管流路に於ける自然流動沸騰の限界熱流束〔16回シンポ，（昭54），280–282〕…… 323

12. 一様加熱垂直円管流路における自然流動沸騰の限界熱流束〔17回シンポ，（昭55），280–282〕…… 326

強制流動沸騰

外部流

Ⅰ. 潜水状態（平行流）

13. 液流中に平行においた一様加熱平板上の強制流動沸騰の限界熱流束〔機論，47（昭56），545–547〕…… 329

Ⅱ. 液膜流（平行流，放射状流）

14. バーンアウト機構の研究（高バーンアウト熱流束の沸騰系について）〔機論，39（昭48），269–277〕…… 332

15. 衝突噴流沸騰系のバーンアウト機構の研究〔機論，41（昭50），306–314〕…… 341

16. 衝突噴流沸騰系のバーンアウトの研究（第1報，気液の挙動の特性）〔機論，43（昭52），3399–3407〕…… 350

17. 衝突噴流沸騰系のバーンアウトの研究（第2報，バーンアウト熱流束の無次元整理）〔機論，43（昭52），3408–3416〕…… 359

18. 平面噴流による強制液体供給，高熱流束沸騰系のバーンアウト〔機論，44（昭53），2817–2823〕…… 368

19. 衝突小噴流を持つ加熱円板上の強制流動沸騰CHFの上限〔東工紀，A16（昭53），18–19〕…… 375

20. 噴流沸騰系における限界熱流束〔東機報，第14巻（昭54），209–210〕… 377

21. 平面噴流で冷却される一様加熱平板の沸騰CHFに対する速度（ウェーバ数）の影響〔18回シンポ，（昭56），382–384〕…… 379

内 部 流

I. 垂直円管

22. 一様加熱垂直円管内の強制流動沸騰の限界熱流束の無次元整理〔機論，44（昭53），3865–3874〕…… 383

23. 圧力降下を伴う管内強制流動サブクール沸騰の限界熱流束〔16回シンポ，（昭54），277–279〕…… 393

24，一様加熱垂直円管内の強制流動沸騰の限界熱流束の特性について〔機論，45（昭54），859–868〕…… 396

25. 一様加熱垂直円管内の強制流動沸騰限界熱流束の研究（ドライアウト，入口サブクール，沸騰長さなどに関連する事項について）〔機論，45（昭54），1892–1901〕…… 406

26. 一様加熱垂直円管（飽和液流入）内の限界熱流束の全般的特性の観察〔機論，46（昭55），1721–1730〕…… 416

27. 一様加熱垂直円管内の強制流動沸騰の限界熱流束（高圧領域の研究とCHF特性領域線図）〔機論，47（昭56），139–148〕…… 426

28. 一様加熱垂直円管における強制流動沸騰の限界熱流束と流動様式との関係について〔機論，47（昭56），220–222〕…… 436

29. 一様加熱垂直円管内の強制流動沸騰限界熱流束の「熱流束－クオリティ」形式の整理について〔機論，47（昭56），223–226〕…… 439

30. 一様加熱垂直円管内の強制流動沸騰の限界熱流束：R－12によるHP領域の実験的研究〔機講論，No.813–4（昭56），95–102〕…… 443

31. 一様加熱流路で「加熱長さ」／「加熱相当直径」比の小さい場合の限界熱流束〔機講論，No.810–12（昭56），43–50〕…… 451

II. 垂直二重管

32. 一様加熱垂直二重管内の強制流動沸騰の限界熱流束の無次元整理〔機論，45（昭54），1322–1333〕…… 459

33. 内管一様加熱の垂直二重管（飽和液流入）内の限界熱流束の一般特性の観察〔機論，47（昭56），830–835〕…… 471

III. 垂直矩形管

34. 短い矩形流路（一面加熱）内の限界熱流束〔東機報，第13巻（昭53），121–122〕…… 477

35. 一様加熱の矩形流路内の強制流動沸騰の限界熱流束の無次元整理〔機講論，No. 818-1 （昭56），131-137〕…… 479

（注）

ASMEJ	J. Heat Transfer, ASME
HTJR	Heat Transfer－Japanese Research
IHTC	International Heat Transfer Conference
IJHMT	Int. J. Heat and Mass Transfer
IJMF	Int. J. Multiphase Flow
JSMEB	Bulletin of JSME
JUS	Japan－U. S. Heat Transfer Joint Seminar
NCTAM	Proceeding of Japan National Congress for Applied Mechanics

機講論	日本機械学会講演論文集
機　誌	日本機械学会誌
機　論	日本機械学会論文集
シンポ	日本伝熱シンポジウム
東機報	東京大学機械工学研究報告
東工紀	東京大学工学部紀要

FREE CONVECTION BOILING

EXTERNAL FLOW

INTERNAL FLOW

PRINCIPAL MECHANISM OF BOILING CRISIS IN POOL BOILING

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Abstract—High-speed photography has been employed to study the mechanism of the boiling crisis in pool boiling of saturated water at high heat fluxes. Three configurations have been examined: normal free boiling on a heated 10-mm dia. horizontal copper disc and the same system subjected to the interference of a flat plate and a short tube held above the heated disc.

The mechanism of pool boiling crisis is clarified both by the qualitative observations of the fluid behavior and by an analysis based on the experimental facts. It is concluded that the phenomenon of pool boiling crisis is related to the intermittent behavior of vapor masses and to the consumption of liquid film which builds up on the heated surface.

Previous theories of boiling crisis and transition boiling are discussed in relation to the mechanism revealed by the present study.

NOMENCLATURE

A_w, area of heated surface;
A'_w, total sectional area of vapor stems;
D, average diameter of vapor stems;
g, gravitational acceleration;
H_{fg}, latent heat of evaporation;
q, heat flux;
q_c, peak heat flux;
$\bar{q}$, average heat flux in transition boiling;
t, time;
t_c, life of liquid film at the point of peak heat flux.

Greek symbols

δ, height of vapor stem;
δ_0, initial thickness of liquid film;
ρ_l, density of liquid;
ρ_v, density of vapor;
σ, surface tension.

1. INTRODUCTION

AS THE most fundamental boiling crisis, the boiling crisis in saturated pool boiling is studied in the present paper. As is well known, various theories have already been published on this subject. Roughly speaking, we have one group which considers the obstruction of heat transfer due to vapor masses anchored on the heated surface and another which postulates that the crisis is due to the existence of a limit in the state of the fluid system. The latter consists of two groups of theories, namely the geometrical theories of Rohsenow and Griffith [1], and Chang and Snyder [2] based on critical bubble spacing near the heated surface, and the hydrodynamic ones of Kutateladze [3], Chang and Snyder [4], Zuber, Tribus and Westwater [5], and Torikai [6] based on the instability of the interface between liquid and vapor.

However, quite a number of experimental observations on boiling and the boiling crisis cannot be explained by the above-mentioned theories. For example, according to Torikai and Yamazaki [7], and Kirby and Westwater [8] who studied the nucleate boiling on a glass plate coated with a transparent, heated film of metal oxide, in order to observe the state of fluid very close to the heated surface, a liquid film exists on the heated surface even in the vicinity of boiling crisis. Furthermore, Kirby and Westwater reported that dry spots sometimes appear

in the liquid film at high heat fluxes near boiling crisis; that dry spots grow in size but they have a life after which liquid washes across the dry area and again wets the solid; and that dry spots presumably signal the boiling crisis. These observations are not in keeping with the models on which existing theories are based. In addition, it has been known that the properties of the heated surface, such as wettability, influence to some extent the critical heat flux as shown for example, by Costello and Frea [9]. This is also difficult to explain with the existing theories.

The above-mentioned observations would indicate that there are still important fundamental inadequacies in the existing theories. These theories try to explain the boiling crisis by an essentially static model, whereas boiling, especially in the nucleate and transition regions, is a phenomenon characterized by periodical or intermittent processes. If so, the principal mechanism of boiling crisis should be connected with such periodical aspects. In addition, the point of peak heat flux is a limit not only of the nucleate boiling but also of the transition boiling. Consequently, the phenomenon which gives a limit to one region should have a close relation to the mechanism of boiling in the other region.

With such a point of view in mind, observation and analysis of pool boiling at high heat fluxes are carried out in the present study using high-speed photography and adopting particular configurations of boiling which enable us to approach the essential nature of boiling crisis with ease.

2. EXPERIMENTAL APPARATUS

The boiler is illustrated schematically in Fig. 1. This is the same as that used in the authors' preceding study [10], so that description of its details will be omitted. Pure water at the atmospheric pressure is boiled on a horizontal copper disc of 10 mm in diameter facing upward. A vapor mass of comparatively simple shape is generated on the heated surface

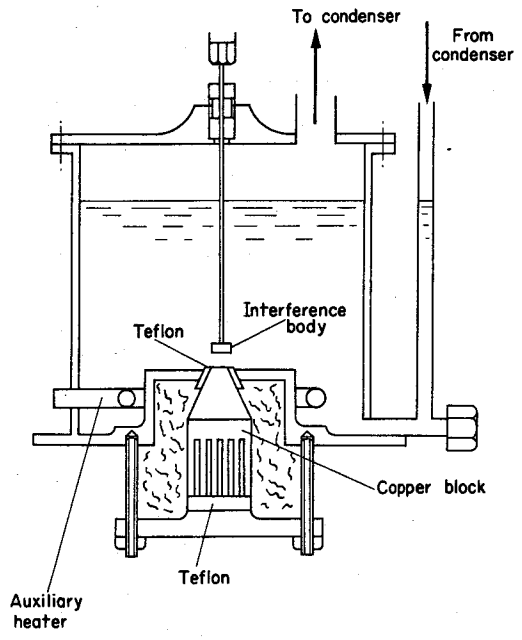

FIG. 1. Boiler and heater.

intermittently at high heat fluxes, and the peak heat flux is around $1{\cdot}6 \times 10^6$ kcal/m^{2}h.

Experiments are carried out for three different configurations of boiling, namely the normal free boiling and the boilings subject to interference originated by holding either an optical assembly (Fig. 2) or a short tube (Fig. 3) vertically above the heated surface. The optical assembly, which is the same as that in the preceding study [10], is used not only to interfere with the behavior of vapor masses by placing the bottom glass-surface parallel to the heated surface, but also to observe the state of the fluid on the heated surface through a circular window. In the last configuration mentioned above, short tubes 3-, 10- and 20-mm long are used, all having outer diameter equal to the diameter of the heating disc. They are made of acrylic resin of good transparency so that the behavior of the fluid inside can be observed with ease.

When the spacing, s, between the bottom surface of each of the interference bodies aforementioned and the heated surface is varied, the peak heat flux, q_c, shows a characteristic

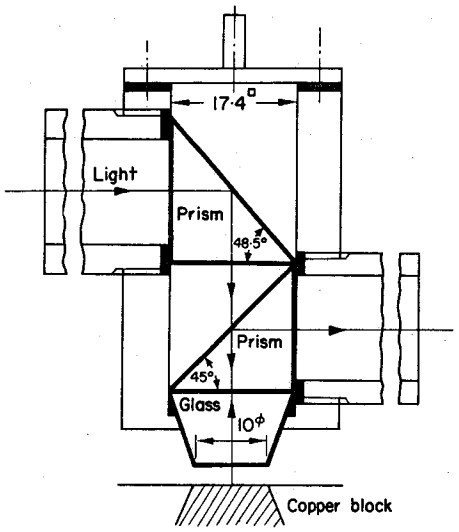

FIG. 2. Optical assembly.

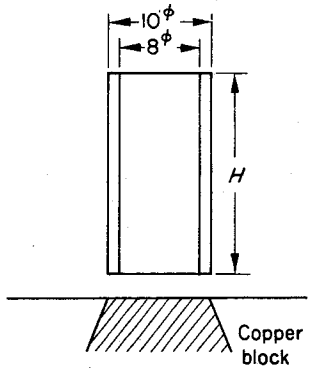

FIG. 3. Tube.

variation such as is shown in Fig. 4. It is particularly of interest to notice that in the case of the tubes there is a spacing range where q_c increases as s is decreased. Of course, if s is sufficiently increased, every curve in Fig. 4 tends to the magnitude of peak heat flux observed in the normal free boiling: $q_c \doteqdot 1{\cdot}6 \times 10^6$ kcal/m^{2}h. It should be noted that most published studies of pool boiling crisis have been made on the normal free boiling, and that they give no reliable explanation of such a variation of q_c as shown in Fig. 4.

3. NUCLEATE BOILING AT HIGH HEAT FLUXES

Before reporting the present results, some of the existing knowledge of nucleate boiling at high heat fluxes will be mentioned for convenience. A typical aspect of nucleate boiling on a heated surface at high heat fluxes is shown schematically in Fig. 5(a). There is a liquid film on the heated surface, and a lot of small vapor stems within the film are supplying vapor to an overlying vapor mass. Gaertner and Westwater [11] have measured the average diameter of the vapor stems, D, in the saturated boiling of water at the atmospheric pressure on a horizontal copper surface to give the solid curve in Fig. 6 showing the variation of D with heat flux q. As for the height of vapor stems, δ, Gaertner [12] has reported, in the discussion on his paper, that the following relation exists independently of q:

$$\delta = 0{\cdot}6\, D. \qquad (1)$$

This is the relation obtained from side-view photographs taken of the boiling of water on a horizontal copper surface at high heat fluxes.

On the other hand, it may be presumed that if an interference plate is pushed downward from above into a vapor mass as shown in Fig. 5(b), the characteristics of boiling heat transfer do not vary until the spacing, s, between the plate and the heated surface approaches the thickness of the liquid film, δ. The critical spacing, s_f, at which the variation of heat-transfer characteristics begins, has been measured by the authors [10]. The results are plotted in Fig. 6 showing the existence of a kind of mutual relation with the result of Gaertner and Westwater.

The authors have also measured the increase of volume of a vapor mass generated just above the 10 mm diameter horizontal copper surface with high-speed photography. According to the results, the rate of increase of vapor within a vapor mass is almost constant through the growth period of the vapor mass indicating

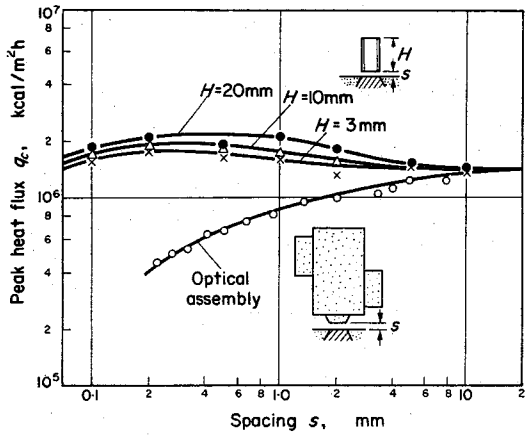

FIG. 4. Variation of peak heat flux with spacing.

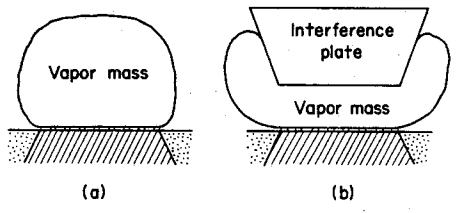

FIG. 5. (a) Nucleate boiling at high heat fluxes. (b) Nucleate boiling with an interference plate.

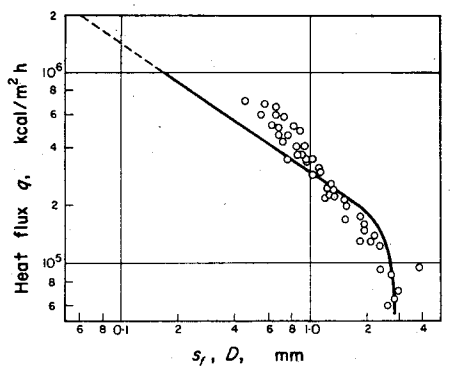

FIG. 6. Variation of critical spacing s_f (indicated by circles) and average diameter of vapor stems D (indicated by solid curve) with heat flux. Broken line is extension of experimental result.

that vapor is generated from the liquid film on the heated surface at a constant rate.

Summarizing the results which have been stated above, it may be concluded that nucleate boiling at high heat fluxes is characterized by nucleate boiling within a liquid film built up on the heated surface and its production of vapor. According to the solid curve in Fig. 6 and equation (1), the thickness of the liquid film decreases as the heat flux is increased, and it is surmized to be only 0·1 mm or so in the vicinity of the boiling crisis of water at atmospheric pressure.

4. GENERATION AND CONSUMPTION OF LIQUID FILM

How such a liquid film as described in the preceding section is built up on the heated surface is a problem of great importance. In order to clarify the mechanism, the behavior of the fluid has been carefully observed in the present study.

4.1. *Free boiling*

As a typical example, Fig. 7 shows successive states of vapor masses in the vicinity of the free heated surface at intervals of 0·84 msec, which were obtained at a heat flux of $1{\cdot}39 \times 10^6$ kcal/m²h. It is noticed that a perfect detachment of vapor mass from the heated surface occurs in frames 9–13 and a new vapor mass has already begun to grow in frame 14 pushing up the overlying liquid. The time elapsed in one detachment of vapor mass is found to be about 3 msec at the most, whereas the average period from one detachment to the next has been found to be around 60 msec in this case. It may be said, therefore, that the heated surface is always covered with a vapor mass except the extremely short period of detachment of vapor mass, occurring nearly at constant frequency.

When a vapor mass grows on the heated surface, it continues a very rapid expansion of volume forcing out the surrounding liquid with a constant rate of increase of vapor, so that the inside pressure should be somewhat higher than

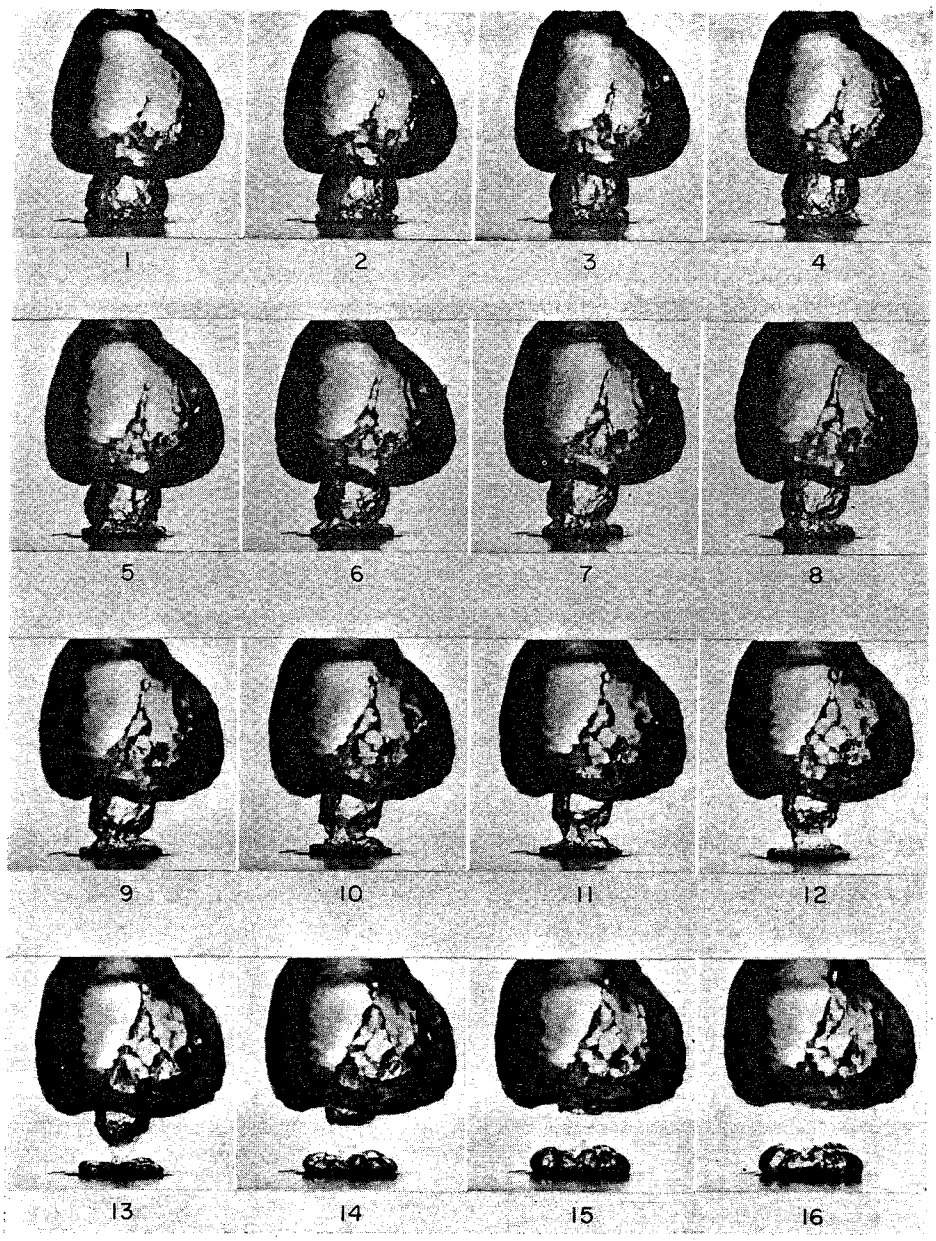

FIG. 7. Successive states of vapor mass in free nucleate boiling ($q = 1{\cdot}39 \times 10^6$ kcal/m^{2}h, interval = 0·84 msec).

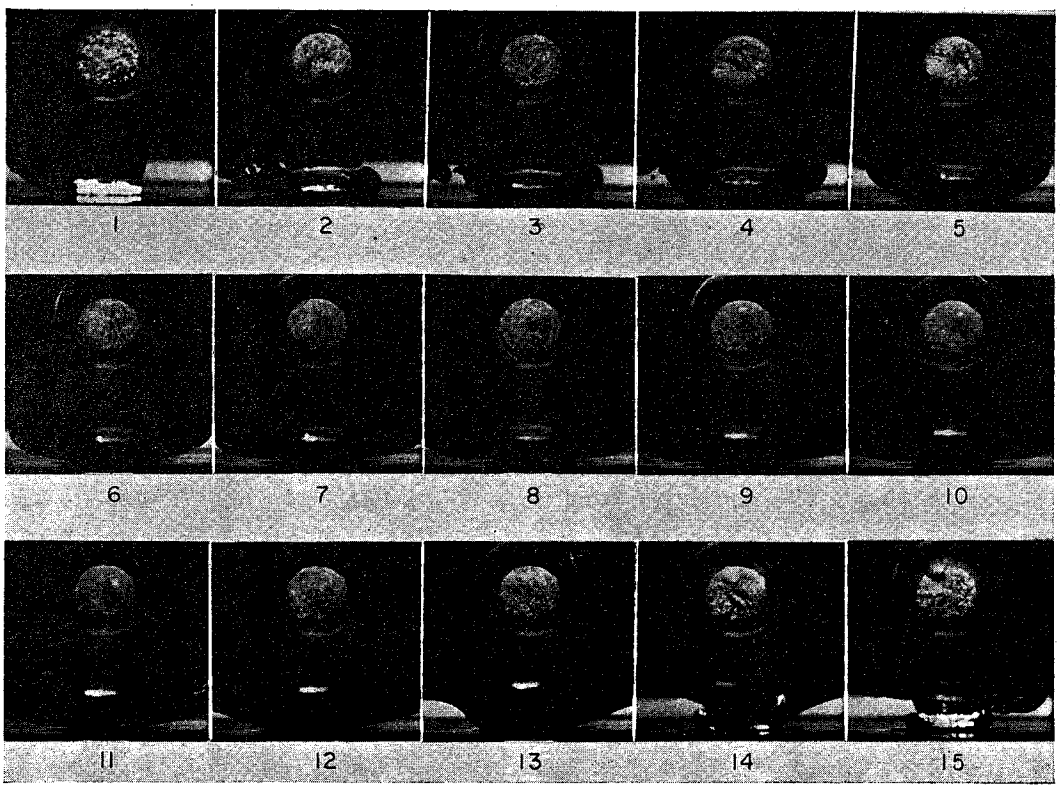

FIG. 8. Nucleate boiling with optical assembly at a heat flux lower than peak heat flux (s = 2 mm, interval = 8·6 msec).

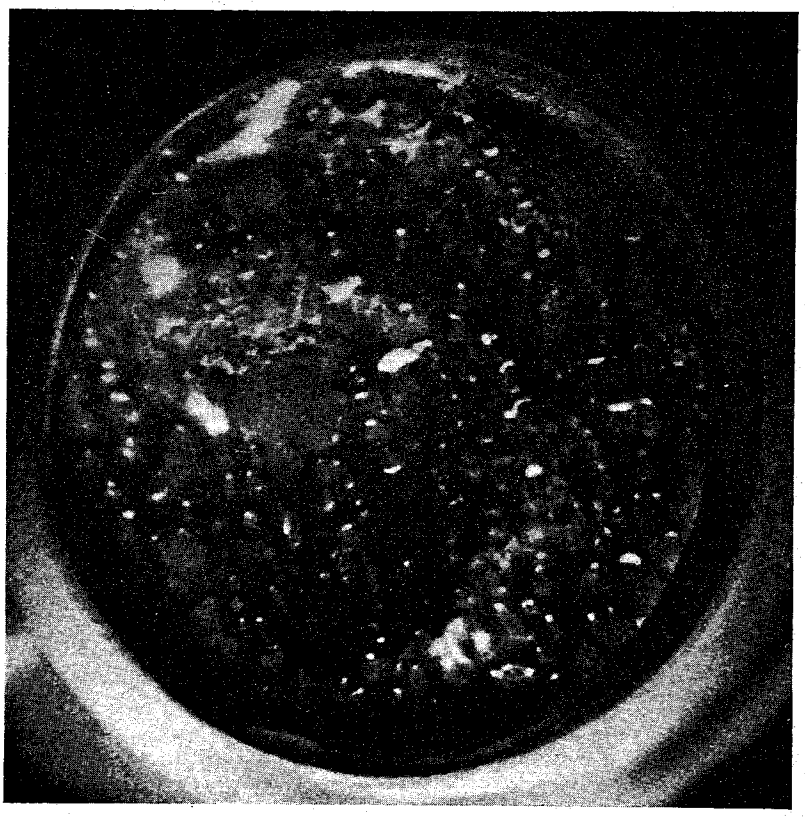

FIG. 9. Liquid film in the late period of a cycle of nucleate boiling with optical assembly very close to peak heat flux ($s = 2$ mm). Liquid is dark, and white points scattered are only reflection of light.

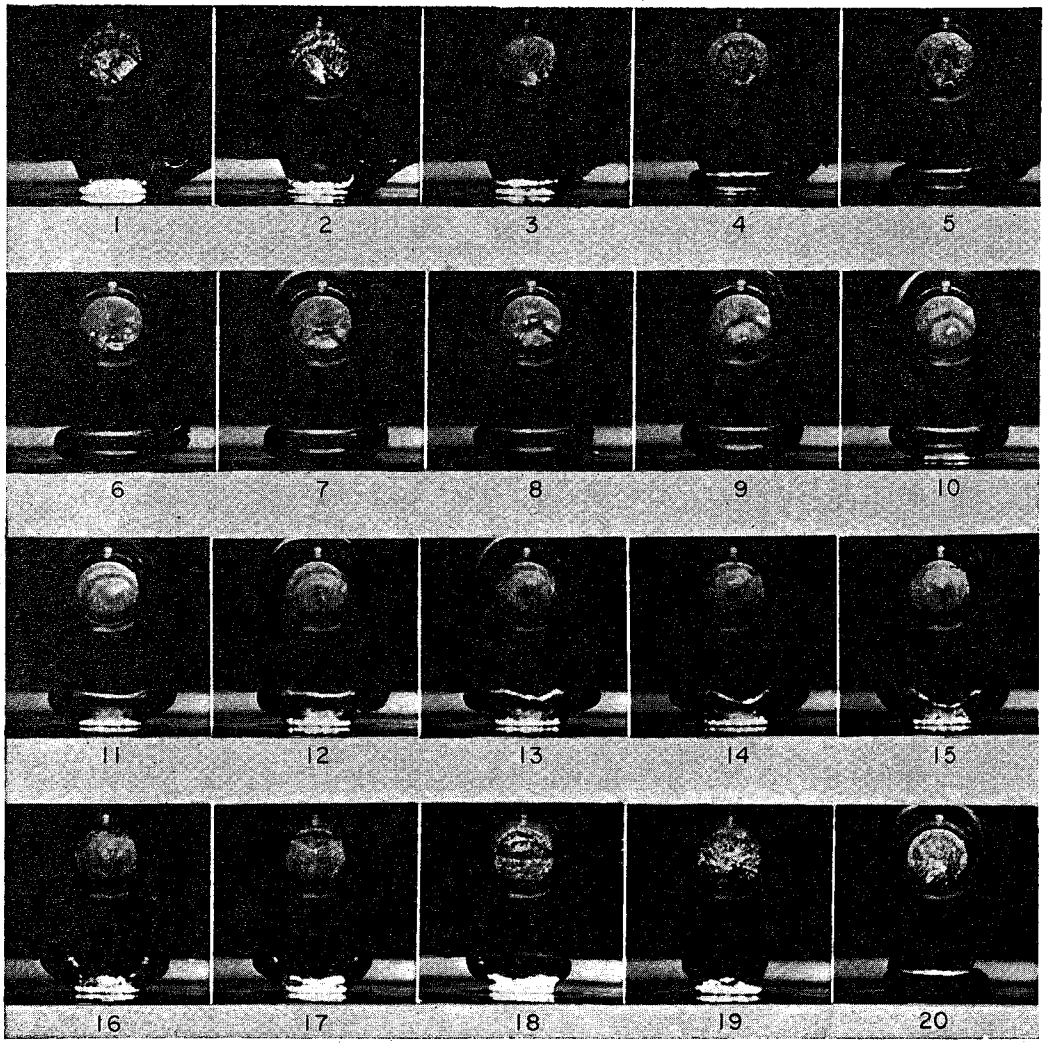

FIG. 10. Transition boiling with optical assembly (s = 2 mm, interval = 5·4 msec). Since a liquid film is built up on the lower surface of glass plate also, meaningless waves are sometimes observed. Due to heat capacity of copper block, quasi-steady transition boiling can be observed.

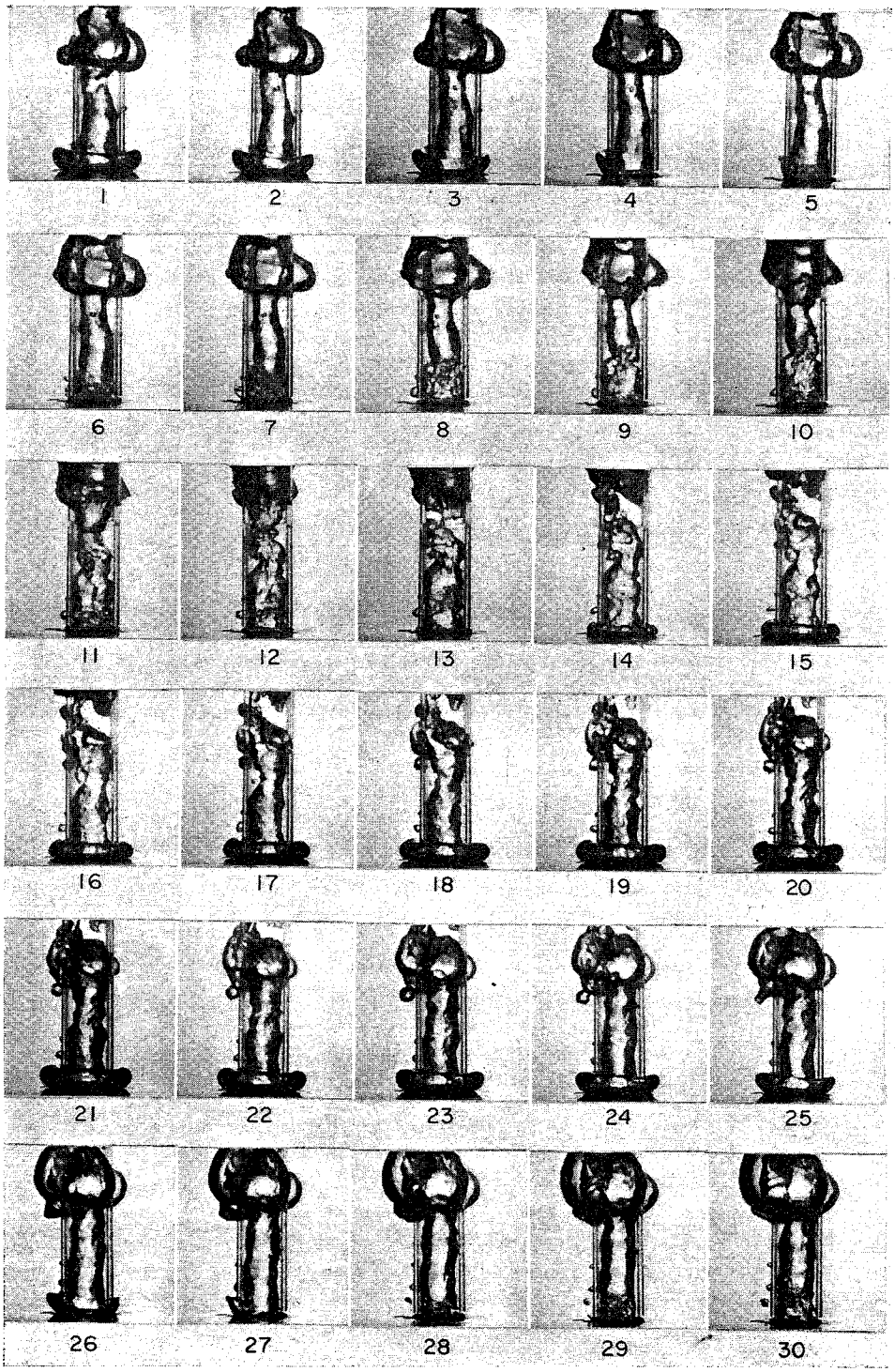

FIG. 11. Successive states of vapor mass in nucleate boiling with tube ($H = 20$ mm, $s = 2$ mm, $q = 1{\cdot}69 \times 10^6$ kcal/m^{2}h, interval = 1·9 msec).

the surroundings. Since this pressure is exerted on the heated surface too, it is almost certain that during the growth period of a vapor mass, liquid cannot be supplied from the surroundings to the heated surface. If so, it is presumed that the supplying of liquid to the heated surface is performed only when the vapor mass detaches from the heated surface, and that thereafter a new vapor mass grows consuming the liquid film thus built up.

4.2. *Boiling subject to interference of optical assembly*

The boiling, subject to interference of the optical assembly with a spacing of $s = 2$ mm, has been studied; the configuration is useful to examine the presumption described in Section 4.1. In this case, photographs were taken simultaneously from the right-hand side of the optical assembly shown in Fig. 2 as well as through the optical assembly itself, thereby not only a side view of boiling near the heated surface but also a horizontal view of the heated surface are caught simultaneously on a film.

First, Fig. 8 shows a typical example of the successive states of the fluid in the case of a heat flux lower than the peak heat flux (interval: 8·6 msec). A circular image on the upper part of each frame is a horizontal view of the heated surface which is observed through the window of the optical assembly. It is noticed that a vapor mass begins to grow on the heated surface in frame 1, and after filling the space over the heated surface, the vapor mass is expanding radially in frame 2. Since the boiling within the liquid film on the heated surface continues ever after, the vapor mass continues to expand more and more until it covers the lower portion of the optical assembly as seen in the frames after frame 7. However, when time has elapsed up to frames 13 and 14, the vapor mass begins a very rapid escape originating an inflow of liquid into the space over the heated surface. Then, reproducing the initial state in frame 15, the same cyclic process as before is repeated.

In the case of Fig. 8, a liquid film remains on the heated surface during the whole period of a cycle. However, if the temperature of the heated surface becomes higher approaching the point of peak heat flux, dry spots sometimes appear and grow on the heated surface near the end of each cycle. This is the consequence of the consumption of liquid film. Since the initial thickness of a liquid film as well as the anchorage period of a vapor mass vary to some extent from cycle to cycle, dry spots appear inevitably near the end of those cycles which happen to have more severe conditions. As an example, Fig. 9, which was obtained at a heat flux close to the peak heat flux, shows several dry areas as well as many dry spots on the heated surface. It should be mentioned with respect to the situations described here that they are very similar to those described in the report of Kirby and Westwater referred to in Section 1.

If the temperature of the heated surface is further increased beyond the point of crisis, a situation arises such that every cycle includes a period during which the heated surface is perfectly dry as shown in Fig. 10. The interval of successive frames is now 5·4 msec and the surface temperature is 5 degC higher than that at the point of peak heat flux. Liquid is flowing into the space between the glass plate and the heated surface creating nucleate boiling in frame 1. Then the same processes as in Fig. 8 (boiling within liquid film and growth of vapor mass) are undergone up to frame 6 or its vicinity where the liquid film is consumed away. Disregarding waves which sometimes appear in a liquid film built up on the lower surface of the glass plate of the optical assembly, and which can be seen in some of the frames, it is noticed that perfect dryness of the heated surface as well as no growth of the vapor mass continue afterwards. In frames 13 to 17, however, an escape of the vapor mass occurs introducing liquid into the space above the heated surface, and the initial state is reproduced in frame 18. Comparing the growth of a vapor mass in Fig. 10 with that in Fig. 8, it may be noticed that the amount of vapor produced during one cycle

is obviously less in Fig. 10 than in Fig. 8. This presumably suggests that the liquid film built up on the heated surface becomes thinner as the temperature of the heated surface is increased.

4.3. *Boiling subject to interference of short tube*

The situation in the case of boiling subject to the interference of a short tube is the same in principle as in the preceding cases. As an example, Fig. 11 shows a result obtained under conditions of $H = 20$ mm, $s = 0{\cdot}8$ mm and $q = 1{\cdot}69 \times 10^6$ kcal/m^{2}h (interval: 1·9 msec). It may be of interest to notice an unforeseen fact that a part of vapor mass protrudes radially through the circumferential narrow opening between the tube and the heated surface as shown in frames 14 to 24. The protrudent part of vapor mass is, however, inhaled finally into the tube when the vapor mass begins to escape and ascend as shown in frames 25 to 27. At the same time, liquid is supplied to the heated surface through the narrow opening following the rapid retreat of vapor. Notice the splash of liquid on the heated surface in frame 28.

5. MECHANISM OF POOL BOILING CRISIS

The observational results reported in the preceding section have already suggested that the point of peak heat flux is a point at which the average life of liquid films on the heated surface coincides with the average detachment period of vapor masses. According to Sections 3 and 4, it may be assumed that the rate of consumption of the liquid film is in proportion to the heat flux, and that the initial thickness of the liquid film decreases with the heat flux. On the other hand, the detachment period of vapor mass is generally finite. Particularly, it should be noted that the detachment period has been found to change very little with the heat flux for the region of nucleate boiling at high heat fluxes investigated in the present study. Consequently, as the heat flux is increased, a transition point must inevitably occur.

Now let us examine the mechanism of boiling crisis quantitatively. If a liquid film is formed with an initial thickness δ_0 on a heated surface of area A_w subject to a high heat flux q, the life of the film, t, is determined as follows:

$$t = \frac{\rho_l \delta_0 (A_w - A'_w) H_{fg}}{q A_w} \tag{2}$$

where ρ_l is the density of liquid, A'_w the total sectional area of vapor stems within the film, and H_{fg} the latent heat of evaporation. A'_w/A_w and δ_0 in equation (2) should be known in order to estimate t. According to the experiments of Gaertner and Westwater [11] of water boiling on a copper surface, the following approximate relation holds independently of q at high heat fluxes:

$$A'_w/A_w = 1/9. \tag{3}$$

As to δ_0, it is reasonable to use equation (1) obtained from the side-view photographs of boiling. Since the circumference of the heated surface is always exposed to the surrounding liquid, δ in equation (1) gives a value rather close to the initial thickness of the liquid film.

Substituting equations (1) and (3) into equation (2) and determining D in equation (1) from the curve in Fig. 6, the life of the liquid film, t, is calculated as a function of q to give the thick solid curve in Fig. 12.

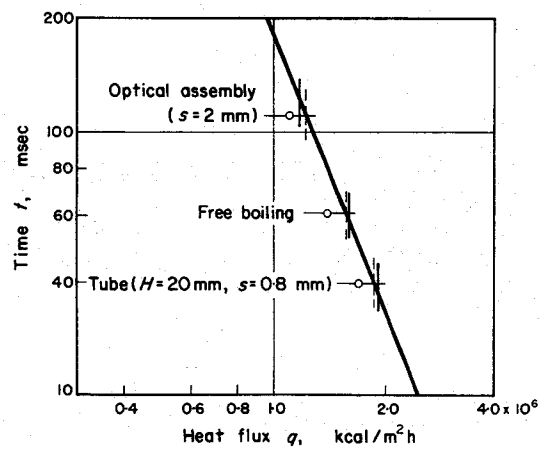

FIG. 12. Comparison of analytical peak heat flux data (indicated by broken vertical lines) with experimental results (indicated by solid vertical lines). Case of optical assembly has a detachment-period about three times greater than case of tube.

Experimental results of average detachment period, t, obtained at a heat flux, q, in the vicinity of boiling crisis are indicated by circles in Fig. 12 for three different kinds of boiling. Then the analytical peak heat fluxes which correspond to these detachment periods respectively should be determined at the positions of the vertical broken lines in Fig. 12. For comparison, the experimental heat fluxes obtained corresponding in the same experiments as above are also shown by the vertical solid lines in Fig. 12. It may be noticed that the agreement between the analytical and experimental results is rather good.

6. DISCUSSION

6.1. *Various features of pool boiling crisis*

The mechanism of boiling crisis which has been revealed in the present paper enable us to explain various features of pool boiling crisis in a unified manner.

Agitation of fluid increases the peak heat flux through the reduction of the period of vapor anchorage. If liquid is supplied by any means to the liquid film lying under a vapor mass, the peak heat flux is increased.

If the period of anchorage of vapor mass is increased, the peak heat flux is reduced. This has already been shown experimentally by a blocking surface attached to a heated surface facing downward in the study of Ishigai *et al.* [12] as well as by the optical assembly in the present study. When for some reason the supply of liquid to the heated surface is obstructed, stable formation of liquid film on the surface is disturbed, reducing the peak heat flux.

Due to scattering of the initial thickness of the liquid film, the detachment period of the vapor mass, and other causes, dry spots and areas sometimes appear on the heated surface in the neighborhood of boiling crisis. Consequently, the temperature of the heated surface fluctuates, to some extent, when the surface is heated at a constant heat flux.

6.2. *Relation to customary theories*

It is an unobjectionable fact that if normal systems of boiling alone are considered, current theories have succeeded considerably in correlating the experimental results of peak heat flux; Kutateladze's correlation, for example, is written as follows:

$$\frac{q_c}{H_{fg}\rho_v}\Big/\sqrt[4]{\left[\frac{\sigma g(\rho_l-\rho_v)}{\rho_v^2}\right]} = \text{const.} \qquad (4)$$

where q_c is the peak heat flux, H_{fg} the latent heat of evaporation, σ the surface tension, g the gravitational acceleration, ρ_v the density of vapor, and ρ_l the density of the liquid. The constant value in the right-hand side of equation (4) is determined experimentally.

Time is not included in equation (4), whereas time has been introduced as an important factor in the present study. In spite of this discrepancy of quality, however, a result similar to equation (4) could be derived easily from the theory of the present paper at least for normal systems of boiling. The range of experiment, though, is so limited in the present study that a general, positive verification is impossible, but it may not be useless to mention the following analysis.

According to the present study, it seems likely that two kinds of vapor velocities play particularly important roles in the physics of boiling crisis, namely the ejection velocity and the escape velocity of vapor from the heated surface. The former is obviously $q/H_{fg}\rho_v$, and the latter is tentatively assumed to be in proportion to $\sqrt[4]{[\sigma g(\rho_l-\rho_v)/\rho_l^2]}$ from an analogy with the rising velocity of bubbles in a liquid. Then, taking account of dimensions, the initial thickness of liquid film, δ_0, and the detachment period of vapor mass, t, may be written respectively as follows:

$$\delta_0 = \text{const.} \times l \left\{\frac{\sqrt[4]{[\sigma g(\rho_l-\rho_v)/\rho_l^2]}}{q/H_{fg}\rho_v}\right\}^{1.5} \qquad (5)$$

$$t = \text{const.} \times \frac{l}{\sqrt[4]{[\sigma g(\rho_l-\rho_v)/\rho_l^2]}} \qquad (6)$$

where l represents a term which has the dimension of length. The index of 1·5 in the right-hand side of equation (5) has been determined so as

to satisfy the relation between δ_0 and q given by equation (1) and the curve in Fig. 6. Substituting equations (5) and (6) into equation (2) and replacing q by q_c, yields

$$\frac{q_c}{H_{fg}\rho_v}\Big/\sqrt[4]{\left[\frac{\sigma g(\rho_l-\rho_v)}{\rho_v^2}\right]} = \text{const.}\left(1-\frac{A'_w}{A_w}\right)^{0.4}\left(\frac{\rho_v}{\rho_l}\right)^{0.1}. \quad (7)$$

In the right-hand side of equation (7), $1-(A'_w/A_w)$ may be regarded as almost unity, and the variation of $(\rho_v/\rho_l)^{0.1}$ is comparatively small; in other words, equation (7) is akin to equation (4).

6.3. *Transition boiling*

In connection with the mechanism of boiling crisis, the transition boiling will be touched. As stated in Section 4, if the temperature of the heated surface is increased beyond the point of boiling crisis, every cycle includes a period of perfectly dry surface. Consequently, the temperature of the heated surface fluctuates considerably in the transition boiling except in the case of surface temperature kept exactly constant. In addition, a very rapid growth of a vapor mass is followed by a period during which the vapor mass ceases to grow, making the explosive expansion of vapor in the initial stage conspicuous. Therefore, some aspects of the behavior of liquid and vapor presumably become different from those in the nucleate boiling region. Furthermore, the properties of the heated surface such as dirtiness and wettability may become particularly influential in the transition boiling because liquid must be supplied to the perfectly dry surface at the end of every cycle. It should be kept in mind that the heated surface is substantially wet in the nucleate boiling region.

In relation to the above observations, it may be worth while mentioning the following analysis of transition boiling. Figure 13 shows the experimental results of boiling of water at atmospheric pressure on a copper surface ob-

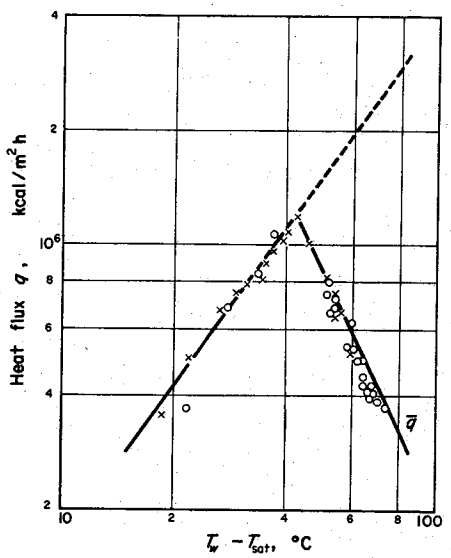

FIG. 13. Heat flux vs. temperature of heated surface in pool boiling of water. Data indicated by circles and crosses from Ishigai *et al.* [14].

tained by Ishigai *et al.* [13], and the thick solid line represents the region of nucleate boiling. Now, for simplicity, three assumptions will be adopted for the transition boiling region: (i) the detachment period of the vapor mass is kept at the same magnitude as that at the point of peak heat flux, (ii) the boiling within the liquid film has the same characteristics as in the region of nucleate boiling, and the extension of the nucleate boiling curve is permitted as shown by the broken line in Fig. 13, and (iii) the heat transfer during the period of perfectly dry surface is negligibly small. Then, if the heat flux to the liquid film is represented by q and the initial thickness of the liquid film by $\delta_0(q)$, equation (2) yields the life of the liquid film, t, as follows:

$$t = \frac{\rho_l \delta_0(q)(A_w - A'_w)H_{fg}}{qA_w}. \quad (8)$$

Similarly, at the point of peak heat flux,

$$t_c = \frac{\rho_l \delta_0(q_c)(A_w - A'_w)H_{fg}}{q_c A_w} \quad (9)$$

where q_c is the peak heat flux. According to the

assumption (i), t_c is equal to the detachment period of the vapor mass in transition boiling. Then, taking the assumption (iii) into account, the average heat flux, $\bar{q}$, for one cycle is deduced to be $\bar{q} = q \times (t/t_c)$; and the substitution of equations (8) and (9) immediately gives

$$\bar{q} = q_c \cdot \frac{\delta_0(q)}{\delta_0(q_c)}. \tag{10}$$

Now, q_c is determined at the point of peak heat flux, and q by the broken line in Fig. 13. $\delta_0(q)$ and $\delta_0(q_c)$ are determined by equation (1) and the curve in Fig. 6. The average heat flux, $\bar{q}$, thus calculated is shown in Fig. 13 as a function of the temperature of the heated surface, and it may be noticed that the agreement with the experimental results is rather good.

Of course, the present analysis has been made with bold simplifications of the phenomenon so that such a correspondence with the experimental results as aforementioned should not be regarded as universally obtainable. However, even if such ambiguities exist, the principal nature of the transition boiling described here is presumed not far from the truth. It may be of interest to mention that the mechanism of transition boiling presumed qualitatively by Ishigai and Kuno [15] from the fluctuation of the temperature of the heated surface agrees well with that in the present study.

7. CONCLUSIONS

(1) Experimental and analytical studies have been carried out in the present study adopting three special configurations of boiling presumed to be particularly effective to attack the mechanism of pool boiling crisis.

(2) Boiling crisis in pool boiling is a phenomenon connected with a balance between the consumption of the liquid film on the heated surface and the supplying of liquid through the intermittent removal of vapor masses.

(3) Physical models upon which existing theories of boiling crisis are based are questionable. However, so far as their final results are concerned, there is a possibility for them to be derived as special cases from the mechanism of boiling crisis revealed in the present paper.

(4) Transition from nucleate to transition boiling does not arise from a perfect metamorphosis of boiling. Roughly speaking, the main cause is only the appearance of a period of perfectly dry surface within every cycle. The nucleate boiling still remains intermittently in the vicinity of boiling crisis at least.

ACKNOWLEDGEMENT

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Résumé—On a effectué des expériences d'ébullition saturée en réservoir de l'eau à des flux de chaleur élevées et des observations des phénomènes qui s'y rapportent à l'aide de la photographie à grande vitesse afin de clarifier le mécanisme de la crise de l'ébullition en réservoir. Trois configurations de l'ébullition sur une surface de cuivre horizontale de 10 mm de diamètre sont étudiées, c'est-à-dire l'ébullition libre normale sur la surface chauffée et les cas soumis à l'interference d'une plaque plane ou d'un tube court tenu au-dessus de la surface chauffée.

Le mécanisme de la crise de l'ébullition en réservoir est alors claifié non seulement à travers l'observation qualitative des comportements de fluide mais aussi à travers l'analyse quantitative qui est basée seulement sur des faits expérimentaux. En résumant les résultats, on en conclut que la crise de l'ébullition en réservoir est un phénomène relié aux comportements intermittents de masses de vapeur aussi bien qu'à la disparition des films liquides qui se forment sur la surface chauffée.

De plus, les théories habituelles de la crise de l'ébullition et de l'ébullition de transition sont discutées également en relation avec le mécanisme de la crise de l'ébullition révélé dans l'étude actuelle.

Zusammenfassung—Es wurden Versuche über Behältersieden von Wasser bei Sättigung und hohen Wärmestromdichten durchgeführt und die zugehörigen Erscheinungen mit Hilfe von Hochgeschwindigkeitsphotographie beobachtet, um den Mechanismus der Krisis beim Behältersieden zu klären: Beim Sieden an einer horizontalen Kupferfläche von 10 mm Durchmesser wurden drei Anordnungen untersucht: gewöhnliches freies Sieden an der Heizfläche und die Fälle, dass Störungen dadurch auftraten, dass entweder eine ebene Platte oder ein kurzes Rohr über die Heizfläche gehalten wurden.

Der Mechanismus der Krisis von Behältersieden wird dann nicht nur durch qualitative Beobachtung des Flüssigkeitsverhaltens erklärt, sondern auch durch eine quantitative Analyse, die allein auf experimentellen Ergebnissen beruht. Aus den Ergebnissen wird geschlossen, dass die Krisis beim Behältersieden eine Erscheinung ist, die ebenso mit dem intermittierenden Verhalten der Dampfmassen zusammenhängt wie mit dem Verschwinden der Flüssigkeitsfilme, die sich an der Heizfläche aufbauen.

Аннотация—Проведено экспериментальное исслование кипения воды при температуре насыщения в большом объеме при больших тепловых потоках и связанных с ним явлений путем исиользования высокоскоростной съемки с целью объяснения механизма кризиса кипения в большом объеме. Изучались три конфигурации кипения на горизонтальной медной поверхности диаметром 10м, включая обычное свободное кипение на нагретой поверхности, а также случаи, отноящеся к интерференции или плоской пластины или короткой трубки, расположенной над нагретой поверхностью.

Механизм кризиса кипения в большом боъеме объясняется не только качественным наблюдением за поведением жидкости, но также с помощью количественного анализа, основанного только на экспериментальных фактах. Суммируя результаты можно сделать вывод, что кризис кипения в большом объеме представляет собои явление, связанное с неустойчивостью паровых образований и жидких пленок, возникающих на нагретой поверхности.

Далее, обсуждаются известные теории кризиса кипения и переходного кипения в связм с механизмом кризиса кипения, которыи выявлен в данном исследовании.

MECHANISM OF BOILING CRISIS AND TRANSITION BOILING IN POOL BOILING

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Abstract

The present study elucidates that boiling crisis and transition boiling are the phenomena closely connected with periodical behaviors of fluid in general. High-speed photography is employed for the study, and experiments are made for saturated boiling of water on a heated flat copper surface mainly 10 mm in diameter under various conditions of orientation, pressure, and depth.

1. INTRODUCTION

Many studies have been published on the mechanism of boiling crisis of saturated pool boiling with proposed theories of various kinds, among which the theories based on hydrodynamic instability of vapor-liquid interfaces seem predominant. It is an undeniable fact that an approximate estimate of peak heat flux can be made in the wide range of liquid conditions by, for example, the correlation of Kutateladze [1] or Zuber et al.[2]. Most existing theories, however, leave the following unsettled problems.

(i) Physical models postulated in the theories do not always correspond to actual fluid behaviors observed in experiments (Kirby and Westwater[3]). (ii) It is difficult to reason out the effects of system geometry, teflon coating and others (Gaertner [4], Hasegawa et al.[5]) on the heated surface of normal sizes. (iii) It is difficult to explain the effects of wettability, contamination (Costello and Frea [6]), heat capacity and thermal conductivity (Carne and Charlesworth [7], Tachibana et al.[8]) of a heated surface particularly noticeable for the surface of comparatively small sizes. (iv) The blanketing process which presumably appears on the heated surface of sizes extremely small compared with a generated bubble (Bernath[9], Carne [10], Ivey and Morris [11], Kutateladze et al. [12]), is beyond the scope. (v) Boiling crisis, which separates transition boiling region and nucleate boiling region, should have close relations with the boiling mechanism in both regions.

Recently, Katto and Yokoya [13] reported the mechanism of boiling crisis conformable to the observation of periodical fluid behaviors in the systems of Fig. 1 (a), (b) and (c). The present study has been made in order to disclose the mechanism of boiling crisis and transition boiling in more detail and extensively.

2. FLUID BEHAVIORS IN SATURATED BOILING AT HIGH HEAT FLUXES

For convenience' sake, the physical model of fluid behaviors postulated by Katto and Yokoya [13] is outlined here. As shown in Fig.2(a), each individual nucleate boiling arising at the active nucleation sites densely distributed on a high heat flux surface supplies vapor to form a vapor mass holding a "nucleate boiling liquid-layer" between the heated surface and the vapor mass. Liquid in the nucleate boiling liquid-layer is consumed at a constant rate due to a violent boiling, while the vapor mass continues to grow as shown in Fig.2(b). The growing vapor mass pushes surrounding liquid away, so that the thin, nucleate boiling liquid-layer is not generally supplied with liquid from the surroundings. As shown in Fig.2(c), however, the vapor mass detaches in due time from the

nucleate boiling liquid-layer while liquid is supplied to the boiling liquid-layer, thereby the initial state is reproduced as in Fig.2(d) to proceed the same cycle again. The time required for the detachment process of a vapor mass is generally very short compared with the period of cycle, τ, from a detachment to the next detachment.

When a nucleate boiling liquid-layer is not wasted out until a vapor mass detaches from the liquid-layer, nucleate boiling continues constantly on the heated surface. As the heat flux, q, is increased, however, an initial thickness of the liquid-layer decreases (cf. equation (2)) and a liquid consumption rate increases whereas the detachment period of vapor mass, τ, varies little. Consequently, a limit, at which the liquid-layer is just wasted out during the detachment period, τ, appears corresponding to boiling crisis. If a surface temperature is raised further, a heat flux higher than peak heat flux wastes up a nucleate boiling liquid-layer, and a vapor mass thus produced stays on a dried out surface for a while reducing a temporal mean value of heat flux to enter the region of transition boiling.

For the mechanism of boiling crisis afore-mentioned, the following equation establishes for peak heat flux, q_c:

$$\tau = \xi \rho_\ell \delta_o H_{fg} / q_c \qquad (1)$$

where τ is the detachment period of vapor mass, ξ the voluminal fraction of liquid within a nucleate boiling liquid-layer, ρ_ℓ the density of liquid, δ_o the initial thickness of a nucleate boiling liquid-layer, and H_{fg} the latent heat of evaporation. According to the experimental results of Gaertner [4], and Gaertrer and Westwater [14], ξ and δ_o may be presumed, for nucleate boiling of water on a normal copper surface at atmospheric pressure, as:

$$\xi = 8/9, \quad \text{and} \quad \delta_o = \delta^* (q^*/q)^{1.5} \qquad (2)$$

where $\delta^* = 0.133$ mm, $q^* = 10^6 \, W/m^2$, and q is the heat flux ($q = q_c$ for boiling crisis).

3. EXPERIMENTAL APPARATUS

The boiler used in the present study is shown schematically in Fig.3. Boiling of distilled water took place on a top round flat surface (10-mm dia. mainly and 35-mm dia.) of a copper block heated with electric heating elements. A conical bakelite ring was used to insulate the copper block from its metallic holder. A fine gap between the ring and the block was filled up with a binding agent, thereby a local boiling was completely suppressed.

A specially designed condenser was connected with the boiler to inhibit an inflow of air for saturated boiling at lower pressures than atmospheric pressure. The boiler could be inclined for orientations other than the upward-facing heated surface. Transition boiling was realized by Isshiki and Tamaki's method [15]: the copper block heated at a constant rate by electric heating elements, was cooled not only by transition boiling on the heated surface but also by forced convection of air flowing through the holes bored in the copper block (see Fig.3). The boiler was equipped with glass windows for the observation of fluid behaviors with a high-speed camera (1000 frames per second).

4. NUCLEATE BOILING AT ATMOSPHERIC PRESSURE

4.1 Observation of fluid behaviors Experiments of nucleate boiling at high heat fluxes were made for the systems shown in Fig.1 (a), (d) and (e).

System of Fig. 1(a): A volumetric analysis of vapor masses with high-speed motion pictures gives Fig.4 showing the vapor production on a heated 10-mm dia. surface near boiling crisis. The vapor production rate, constant independently of time, agrees with the rate of a broken line which corresponds to the measured heat flux, proving the mechanism of Fig.2. Meanwhile, Fig.5 is an instantaneous state of vapor mass on a heated 35-mm dia. surface showing a nucleate boiling liquid-layer beneath the vapor mass, which bears a striking resemblance to the results of Gaertner [4] on a 50.8-mm dia. surface. Taylor

instability complicates the shape of vapor mass as a heated surface becomes large, but no great change of nature appears within the range afore-mentioned.

System of Fig. 1(d): A nucleate boiling liquid-layer on a heated vertical surface is readily observed through an overlying vapor mass. Fig.6 shows successive states on a 10-mm dia. surface near boiling crisis (the bakelite ring has a black look). It may be noticed that a nucleate boiling liquid-layer continues to exist on the heated surface but is considerably wasted in frames 3 and 4 due to the production of vapor.

System of Fig. 1(e): Nucleate boiling curves obtained by changing the depth, H, of a heated face-up 10-mm dia. surface from a free surface of water in the boiler are shown in Fig.7. For H≳2 mm, effects of H almost disappear in the range of high heat flux of $q > 4.6 \times 10^5$ W/m^2, which suggests nucleate boiling within such a thin liquid-layer as shown in Fig.2. According to Fig.8 which shows successive states of boiling for H=3 mm, a vapor mass enclosed with a dome of liquid film grows and then ruptures to make a periodical change of state, while a nucleate boiling liquid-layer continues to exist on the heated surface. The same feature as that of Fig.4 is found as for the vapor production.

4.2 Peak heat flux The observations afore-mentioned support the fluid behavior model of Fig.2. The detachment period of vapor mass (a statistical mean value), τ, near boiling crisis and the peak heat flux, q_c, measured in the afore-mentioned experiments are correlated in Fig.9. The figure also shows the data found in the papers of Gaertner [4], Katto and Yokoya [13], and Ishigai et al. [16] for water boiling on a flat copper surface at atmospheric pressure.

Meanwhile, the relationship between τ and q_c given by equations (1) and (2) appears as a solid line in Fig.9 showing an approximate accordance with the experimental data. Fig.9 indicates at least that q_c may take different values corresponding to variations of τ even for the saturated boiling of the same liquid at the same pressure.

5. NUCLEATE BOILING AT LOW PRESSURES

The type of boiling shown in Fig.2 is possible only when the nucleation site density is great and the "active-site-bubbles" generated in a nucleate boiling liquid-layer are small compared with the heated surface. If the active-site-bubbles become large compared with the heater size, a variation of fluid behavior and peak heat flux ought to appear. This was examined in the present study utilizing nucleate boiling at low pressures.

5.1 Variation of peak heat flux with pressure Fig. 10 shows experimental results of the variation of peak heat flux, q_c, with the relative pressure (ratio of a pressure at the heated surface level to normal atmospheric pressure), $p°$, on a heated 10-mm dia. surface, upward-facing and vertical. The depth of a center of heated surface from a free surface of water is 70 mm for the face-up surface, and 50 mm for the vertical surface. Peak heat flux, q_c, approximately equal for both orientations in this case, decreases with the reduction of $p°$ from unity developing a tendency similar to, for example, the correlation of Kutateladze. In the range of $p° < 0.2$, however, a deviation from the tendency appears to reduce the variation of q_c with $p°$. Within the range of $p°$ tested, effects of hydrostatic pressure change (saturation temperature change) through water are not yet so great.

5.2 An active-site-bubble at low heat fluxes A giant isolated bubble is generated intermittingly in low heat flux nucleate boiling at low pressures. It is absolutely impossible to explain this phenomenon by a high superheat of liquid required for nucleation and a high specific volume of vapor alone. Fig. 11 shows the growth of an isolated bubble initiated at the center of a heated face-up 10-mm dia. surface at $p° = 0.099$, $T_w - T_s = 24.5$ deg and $q = 3.5 \times 10^5$ W/m^2. In frame 1 of no bubble, the heated surface has a white look and the bakelite ring black, being jutted a little (about 0.6 mm) beyond the gray surface of a large round holder. The base of the growing bubble has already spread beyond the heated

10-mm dia. surface in frame 3, and finally extends to a diameter as great as 40 mm.

In Fig. 11, formation of a micro-liquid-film on the heated surface is clearly demonstrated by spreading of a circular dried out area on the heated surface (frames 3 to 8). Time, t, from the start of bubble growth to the appearance of a dried out area r in radius is determined by analyzing high-speed motion pictures as $t=Cr$, where $C=1/657$ h/m. With the same method as Katto and Yokoya [17] in nucleate boiling between a pair of parallel disks, the initial thickness distribution of the micro-liquid-film, s_0, is estimated as $s_0=2.67\times10^{-4}\sqrt{r}$, where s_0 and r in metre. According to the result, s_0 varies from zero to 18.9 μm on the heated 10-mm dia. surface.

Analyses of motion pictures also provide the data of growth of a bubble volume shown in Fig. 12. Fig. 12 also shows, by a light curve, the growth of a bubble volume which is determined theoretically with an assumption that the evaporation on a surface of the micro-liquid-film on the heated surface alone is responsible for the bubble growth after the bubble base has spread over the heated surface. A fairly good accordance is found between the experimental and the theoretical result, indicating that the evaporation on a micro-liquid-layer plays an important role for the growth of the giant bubble.

As is noticed from the experimental data in Fig. 12, a heat insulation period of about 20 ms corresponding to a completely dried out state of the heated surface is included in one cycle consisting of a waiting and a growth time, whose period is 220 ms on the average in the present case. However, appearance of a dried out state on the heated surface alone does not create boiling crisis except a constant heat flux surface with an extremely small heat capacity. Boiling crisis, on a constant temperature surface for strictness, appears under the condition that a heat transfer rate (temporal mean value) begins to decrease as a surface temperature is raised. In the present state of boiling, a heat flux can still increase as a surface temperature is raised, due to a rapid reduction of waiting time as well as an increase of nucleation site density, until boiling crisis appears.

5.3 Boiling crisis Fig. 13, where a black and an outer white ring correspond to the bakelite ring and the painted binding agent respectively, shows successive states of fluid near boiling crisis ($q=5.2\times10^5$ W/m^2). In this case, three large active-site-bubbles grow and then coalesce to form a single vapor mass on the heated surface. Such a nucleate boiling liquid-layer as shown in Fig. 2 is never observed beneath the vapor mass. There are micro-liquid-films which have been made by the active-site-bubbles (frames 7 and 8 suggest their existence), and their evaporation makes the vapor mass grow. Correspondingly, characteristics of vapor production is not so simple as Fig. 4, which can be confirmed immediately by analyzing high speed motion pictures.

Much lower nucleation site density and larger active-site-bubbles compared with the case of Chapter 4 create such a change of fluid behavior as aforementioned, and the waste of nucleate boiling liquid-layer is replaced by that of micro-liquid-film for boiling crisis. It may be worth while mentioning the supposition that if the heated surface were much smaller and had only one nucleation site instead of three sites, boiling crisis would be created by a single bubble blanketing process (no coalescence).

6. TRANSITION BOILING

The mechanism of transition boiling (in the region near boiling crisis at least) is not only of interest but also of importance in connection with boiling crisis. Experimental results are reported below for four different systems of boiling.

6.1 Upward-facing surface A heavy solid curve in Fig. 14 is a boiling curve (nucleate and transition region) obtained by a set of experiments on a heated upward-facing 10-mm dia. surface. A new surface was used for each experiment, particularly in the region of transition boiling. A light broken line in

Fig. 14 is the relationship between temporal mean heat fluxes and temperatures of the heated surface in transition boiling derived analytically with assumptions: (a) a liquid of the quantity given by equation (2) is wasted out with a heat flux given by a broken line (an extension of nucleate boiling region) in Fig. 14 during the period of nucleate boiling, (b) no heat is transferred during the period of dried out state, and (c) the detachment period of vapor mass, τ, is unchanged from the τ at boiling crisis (variations of τ are not great empirically). It is noticed that experimental heat fluxes are higher than analytical values, and the difference increases as a surface temperature is raised, for which the cause must be disclosed.

Variations of the volume of a vapor mass growing on a heated surface in transition boiling, measured at the two different states of 1 and 2 designated in Fig. 14, are shown in Figs. 15 and 16 respectively. A broken line in each figure is a temporal mean vapor production which corresponds to a measured heat flux. Different from such a simple vapor production in the region of nucleate boiling as shown in Fig.4, a voluminal increase of a vapor mass, which takes place in the initial stage, stagnates for a while, and then reopens to continue up to the detachment of the vapor mass. The initial voluminal increase occurs presumably due to the waste of a nucleate boiling liquid-layer established over the heated surface (cf. Fig.2), and the subsequent stagnation corresponds to the dried out state of the heated surface. Horizontal lines drawn in Figs. 15 and 16 respectively indicate the theoretical vapor volumes which ought to appear when a liquid of the quantity given by equation (2) is wasted out with a heat flux given by the broken line in Fig. 14. They show approximate agreements with the measured stagnant volumes.

The voluminal increase in the second stage occurs due to the nucleate boiling of a liquid flowing in from the circumference of the dried out surface covered with a vapor mass. In this case, the higher a heated surface temperature is, the more rapidly the vaporization of the spreading liquid takes place and the more the spreading is confined near the circumference (the inner part of the heated surface is kept dry). Corresponding variations of scale deposition on the heated surface are shown in Fig. 17 obtained by keeping a fresh surface at each given temperature in a tap water boiling: (a) and (b) correspond to Figs. 15 and 16 respectively, and (c) the state nearer a minimum film boiling heat flux.

When a vapor mass breaks off the heated surface finally, a nucleate boiling liquid-layer is reproduced on the surface to proceed the same cycle again. As a surface temperature is raised, contributions of the nucleate boiling liquid-layer to the vapor production decrease very rapidly, while those of the local nucleate boiling of spreading liquid increase (see Figs. 15 and 16). At the same time, the local nucleate boiling is increasingly confined near the circumference of the dried base of overlying vapor mass so as to approach the state of film boiling. The spreading of liquid on a dried solid surface is strongly affected by the affinity between the liquid and surface, and the well-known experience should be recalled here that a boiling curve is heavily changed in the region of transition boiling by a contamination and wettability of the heated surface (Berenson [18], Nishikawa et al. [19]).

6.2 Vertical surface No change of principles from the preceding case appears in all aspects except unsymmetrical modifications in the vertical direction. There is a tendency that the local nucleate boiling of spreading liquid is reduced (the boiling curve approaches to the light broken line in Fig. 14) for a vertical surface presumably due to the buoyant force which never helps a vapor mass to remove perpendicularly from the heated surface. Fig. 18 is a typical example of transition boiling near boiling crisis, which demonstrates the waste of a nucleate boiling liquid-layer leading to a dried out state of the heated surface covered with a vapor mass (cf. Fig.6) and then a minor local nucleate boiling at the lower part.

6.3 Upward-facing surface at very low depthes No fundamental change

from Section 6.1 appears (a 10-mm dia. surface). In most cases of low depth, the local nucleate boiling of inflowing liquid is so readily induced that the circumferential part of the heated surface is subject to boiling through a whole life of a vapor mass. Fig. 19 is an exceptional example (H=3 mm in depth) of transition boiling lacking the local nucleate boiling. No voluminal change of the vapor mass is correspondingly seen in frames 5 to 7 (cf. Fig.8).

6.4 Vertical surface at very low pressures Principal natures of the phenomenon are similar to Section 6.2. As a surface temperature is raised, the difference from the case of atmospheric pressure vanishes due to a rapid increase of nucleation site density and a decrease of the active-site-bubble size.

7. CONCLUSIONS

(1) As far as the fluid behaviors under the conditions of the present study are concerned, they varies continuously over nucleate and transition boiling, and no instability is found at boiling crisis.

(2) The physical model which consists of a nucleate boiling liquid-layer and an overlying vapor mass is applicable for the high heat flux nucleate boiling under normal conditions. When active-site-bubbles have a dimension comparable with, or larger than, the heated surface, variations appear correspondingly in fluid behaviors. In any case, however, the receipt and the consumption of liquid are carried out intermittingly on the heated surface.

(3) When the receipt of liquid loses its reserve power for the increase of liquid consumption, it becomes impossible to raise the heat flux further, then boiling crisis appears.

(4) Under ordinary conditions, τ and δ_o in equation (1) are governed mainly by hydrodynamic behaviors of fluid, and there is a possibility to be able to correlate a peak heat flux approximately with a liquid condition alone.

(5) Other factors can still have effects on peak heat flux. For example, the variation of τ caused by changes of geometrical condition of boiling system; such a loss of the capacity for normal receipts of liquid as brought about by a teflon coating; the wettability, which is particularly effectual for the flat surface of comparatively small sizes because the life of a nucleate boiling liquid-layer is prolonged by the ready supply of liquid from surface sides; the variation of fluid behaviors caused by the size of active-site-bubbles relative to the heated surface; and others.

(6) In the region of transition boiling, three states appear successively in general during a vapor-mass detachment period: the state of wasting a nucleate boiling liquid-layer, the state of dried out condition, and the state of local nucleate boiling of spreading liquid.

(7) The present study has been made for an idealized state of boiling by employing a flat 10-mm dia. surface. For much larger or longer surfaces, Taylor instability may divide a surface into many individual regions.

NOMENCLATURE

- H, depth of a heated surface from a free liquid surface
- H_{fg}, latent heat of evaporation
- p°, relative pressure (ratio of system pressure to normal atmospheric pressure)
- q, heat flux (q_o: peak heat flux)
- r, radius
- s_o, initial thickness of a micro-liquid-layer
- T, temperature (T_s: saturation temperature, T_w: heated surface temperature)
- t, time
- δ_o, initial thickness of a nucleate boiling liquid-layer
- ξ, voluminal fraction of liquid within a nucleate boiling liquid-layer
- ρ_ℓ, density of liquid
- τ, detachment period of a vapor mass

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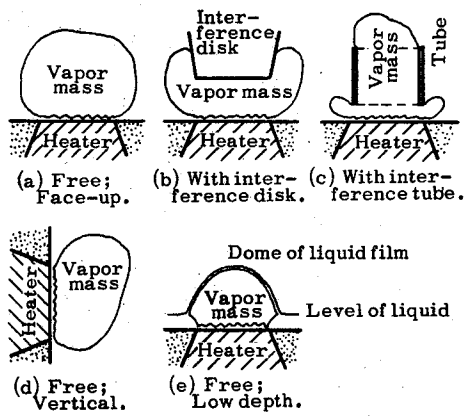


Fig. 1. Various systems of boiling.

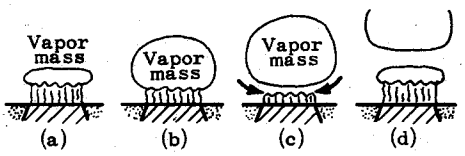


Fig.2. Nucleate boiling at high heat fluxes (thickness of a nucleate boiling liquid-layer beneath a vapor mass is exaggerated).

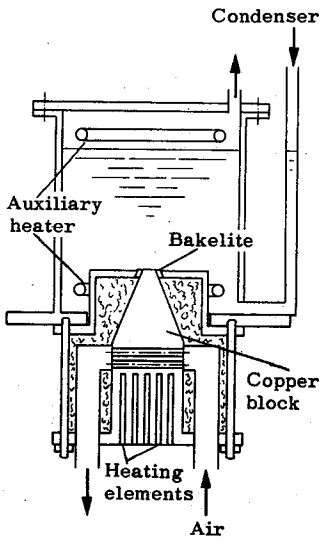


Fig.3. Boiler and heater.

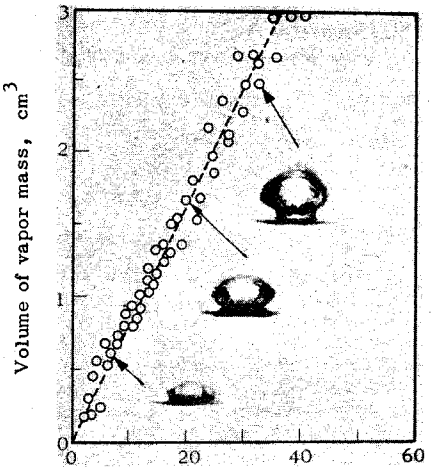
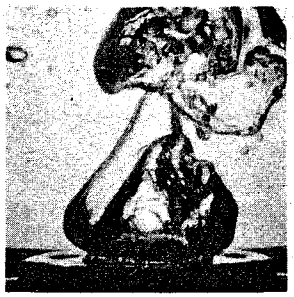


Fig.4. Vapor production in nucleate boiling on a face-up 10-mm dia. surface. ($q=1.39\times10^6 W/m^2$)



Size and position of heated surface.

Fig.5. Nucleate boiling on a face-up 35-mm dia. surface ($q=9.7\times10^5 W/m^2$).

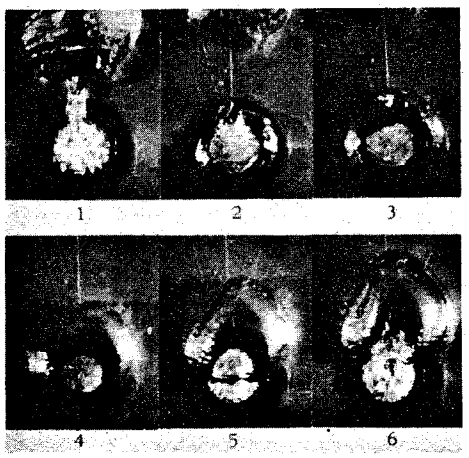


Fig.6. Nucleate boiling on a vertical 10-mm dia. surface ($q=1.14\times10^6 W/m^2$; interval, 10.6 ms). Scratch-marks on the heated surface are not seen due to the existence of a nucleate boiling liquid-layer.

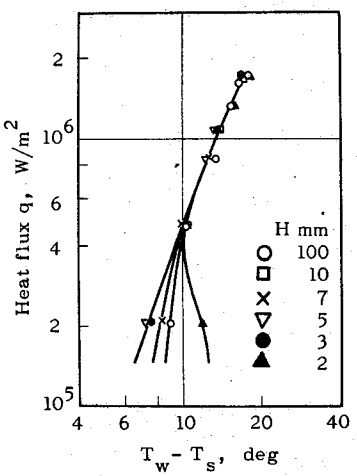


Fig.7. Variation of nucleate boiling curve with depth, H, for a face-up 10-mm dia. surface.

B 3.2

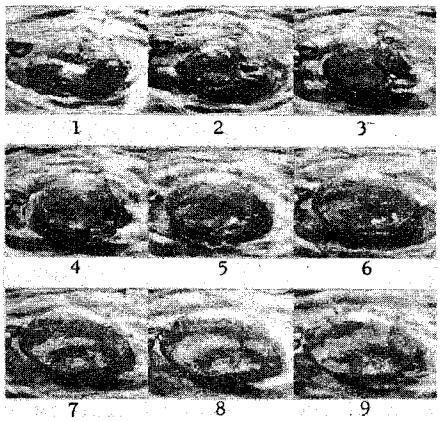
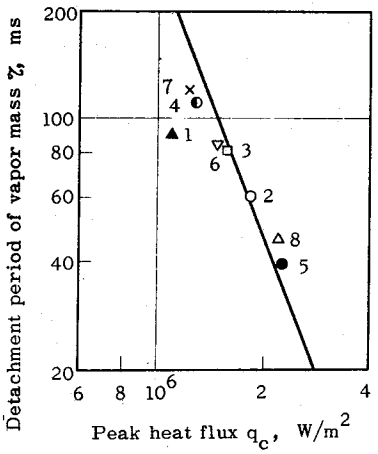


Fig. 8. Nucleate boiling on a face-up 10-mm dia. surface 3 mm in depth ($q=8.6\times10^5 W/m^2$; interval, 8.5 ms).



No.	System	Surface; diameter
1.	Fig. 1(a)	disk; 35mm
2.	ditto	disk; 10mm(Ref. 13)
3.	ditto	disk; 50.8mm(Ref. 4)
4.	Fig. 1(b)	disk; 10mm(Ref. 13)
5.	Fig. 1(c)	disk; 10mm(Ref. 13)
6.	Fig. 1(d)	disk; 10mm
7.	ditto	vertical; 90mm in height(Ref. 16)
8.	Fig. 1(e)	disk; 10mm

Fig. 9. Detachment period of vapor mass (statistically mean value), $\bar{\tau}$, and peak heat flux, q_c.

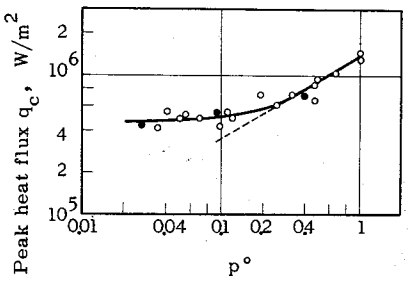


Fig. 10. Variation of peak heat flux with pressure (10-mm dia. surface; ○ face-up, ● vertical).

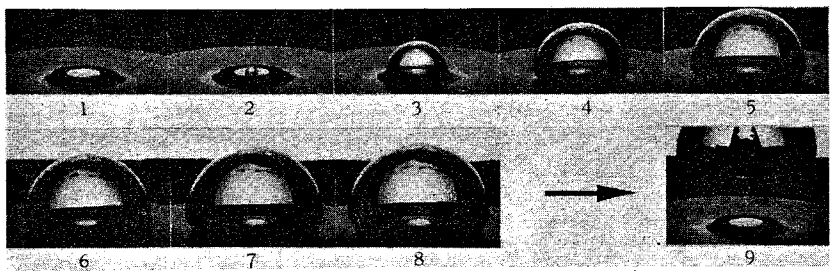


Fig. 11. Growth of an isolated bubble on a face-up 10-mm dia. surface (interval: 3.9 ms for frames 1 to 8 respectively, and 52.7 ms between 8 and 9).

B 3.2

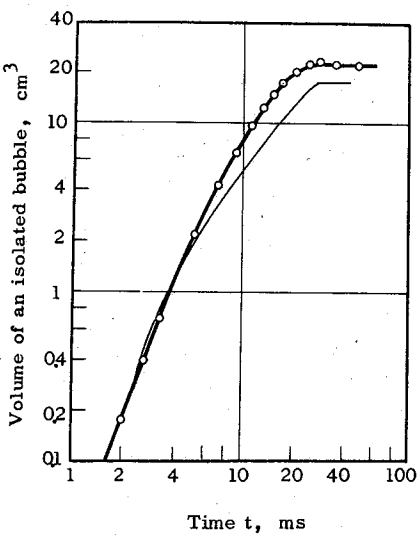


Fig. 12. Variation of volume of an isolated bubble.

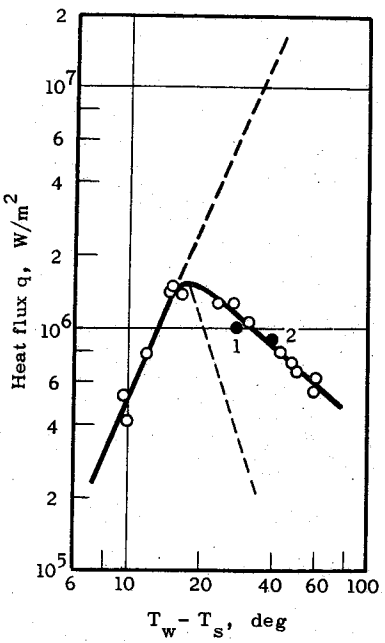


Fig. 14. Boiling curve for a face-up 10-mm dia. surface.

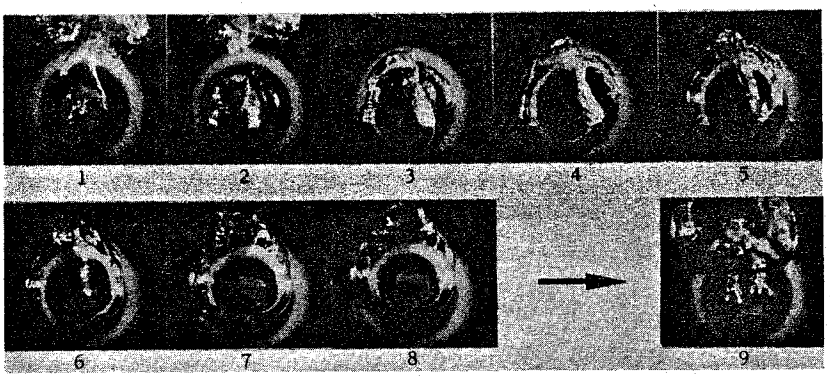


Fig. 13. Nucleate boiling on a vertical 10-mm dia. surface (interval: 5.1 ms for frames 1 to 8 respectively, and 30.5 ms between 8 and 9).

B 3.2

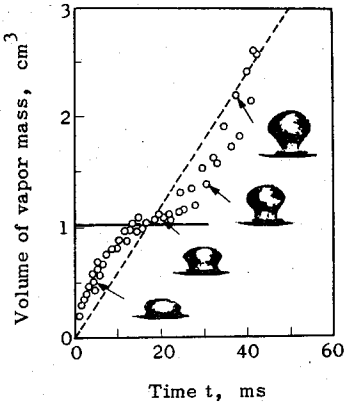


Fig. 15. Variation of volume of vapor mass in transition boiling on a face-up 10-mm dia. surface ($q=1.01\times10^6 W/m^2$).

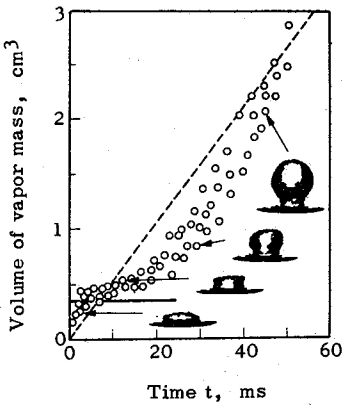
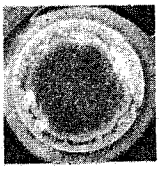
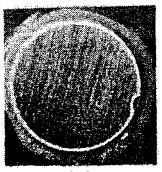


Fig. 16. Variation of volume of vapor mass in transition boiling on a face-up 10-mm dia. surface ($q=8.96\times10^5 W/m^2$).

Fig. 17. Scale deposition on a face-up 10-mm dia. surface in transition boiling. (a) $q=1.01\times10^6 W/m^2$, (b) $q=8.96\times10^5 W/m^2$, and (c) $q=5.10\times10^5 W/m^2$.



(a)



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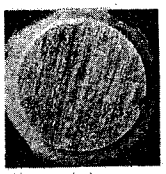

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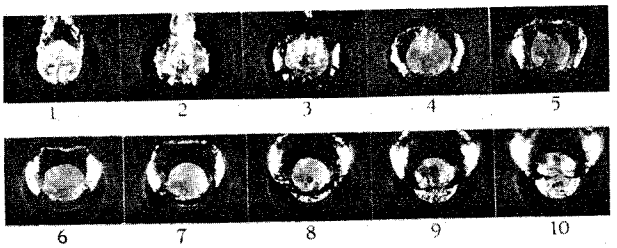

Fig. 18. Transition boiling on a vertical 10-mm dia. surface ($T_w - T_s=25.7$ deg; $q=9.7\times10^5 W/m^2$; interval, 5.4 ms)

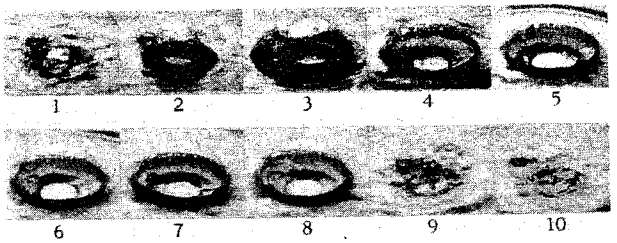

Fig. 19. Transition boiling on a face-up 10-mm dia. surface 3 mm in depth ($T_w - T_s=52.9$ deg; $q=5.9\times10^5 W/m^2$; interval, 27.0 ms).

B 3.2

Study of Forces Acting on a Heated Surface in Nucleate Boiling at High Heat Fluxes

YOSHIRO KATTO AND KOHEI KIKUCHI*

In the region of nucleate boiling at high heat fluxes, vapor masses appear successively on a heated surface due to the accumulation of vapor which is constantly and rapidly produced at many active nucleation sites. The mechanical conditions which dominate the behavior of vapor mass may be one of the most important matters for various phenomena related to high heat flux nucleate boiling.

In the present paper, we report on a heated surface of about 15-mm dia. suspended in a saturated water at atmospheric pressure; a balance was employed to measure the variation of buoyant force due to boiling, and thereby the mean vapor pressure within a vapor mass growing on the heated surface was obtained. High-speed photography was also used to observe the growing behavior of each vapor mass.

A theoretical analysis of the vapor pressure of vapor masses agrees well with the experimental results and suggests that the growth of vapor is governed strongly by the gravitational effects of vapor. The lower limit of the region of high heat fluxes in nucleate boiling was analyzed yielding good agreement with the experimental results. A discussion of burn-out is also presented.

1. Introduction

In the high heat-flux region of nucleate boiling, vapor masses appear successively on a heated surface due to the accumulation of vapor which isconstantly and rapidly produced at many active nucleation sites; it has been revealed by Katto and Yokoya [1] that the periodic behavior of the vapor mass is directly responsible for the occurrence of burnout. Studying the mechanical conditions which dominate the behavior of vapor mass mass may be, therefore, one of the most important matters for a better understanding of the various phenomena related to the high heat-flux nucleate boiling.

Experiments were carried out in the present study with saturated water at atmospheric pressure in nucleate boiling at high heat fluxed on a horizontal heated disk surface of about 15-mm dia. The forces acting on the heated surface were measured and the behavior of the vapor mass during its growth was observed by means of high-speed photography. Experimental results are analyzed to clarify the mechanical conditions such as buoyancy with respect to the behavior of the vapor mass. Burnout heat fluxes are discussed.

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2. Experimental Apparatus

2.1 Experimental Apparatus

As shown schematically in Fig. 1, the main portion of the experimental apparatus consisted of three parts — a water vessel, a heater, and a balance. The heater was suspended by the arm of the balance so that the force acting on the heater could be measured.

The water vessel was divided by plate glass into two chambers, and the heater was suspended within the inner chamber, while an auxiliary heater was set up in the outer chamber in order to keep the water-vessel at saturation temperature. The inner chamber had the shape of a right hexahedron, 150 ×150 mm in cross section and 290 mm in height. The steam generated in the water vessel was condensed in a condenser and then returned to the water vessel to prevent the change of the water level inside the vessel.

The cylindrical heater, 42 mm in diameter and 40 mm in height, was made of stainless steel with a waterproof seal, and its cross section is shown in Fig. 2. The top of a copper block stored in the heater was employed as the heated disk surface with a diameter of 14.5 mm ordinarily and 16. and 13.6 mm for special circumstances. A thin stainless steel plate was brazed to the circumference of the heated disk surface; the plate thickness was less than 0.5 mm. The lower part of the

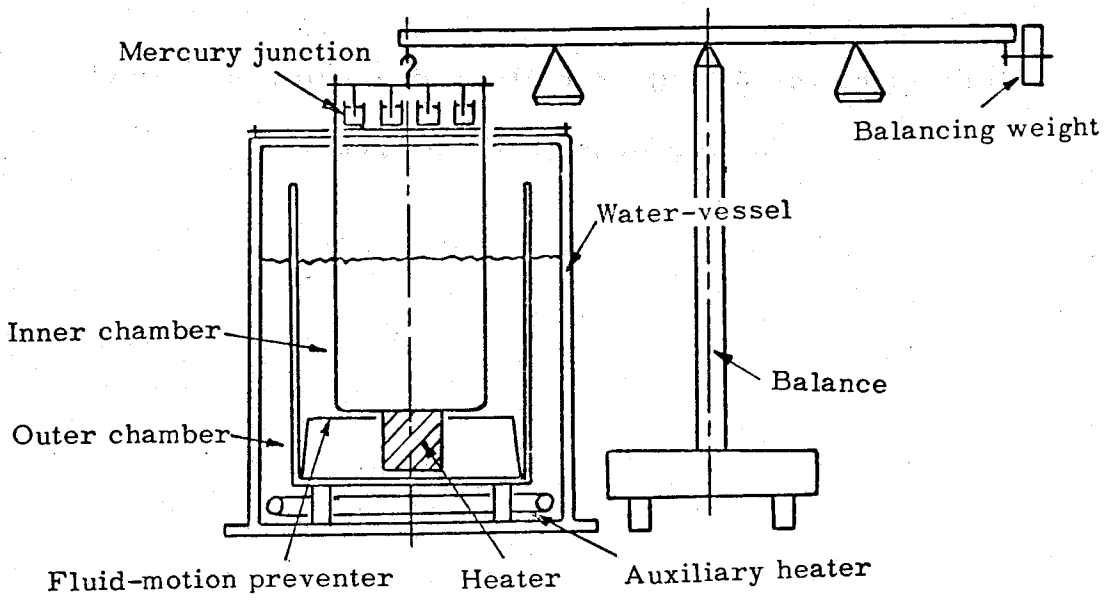

Fig. 1. Main portion of experimental apparatus.

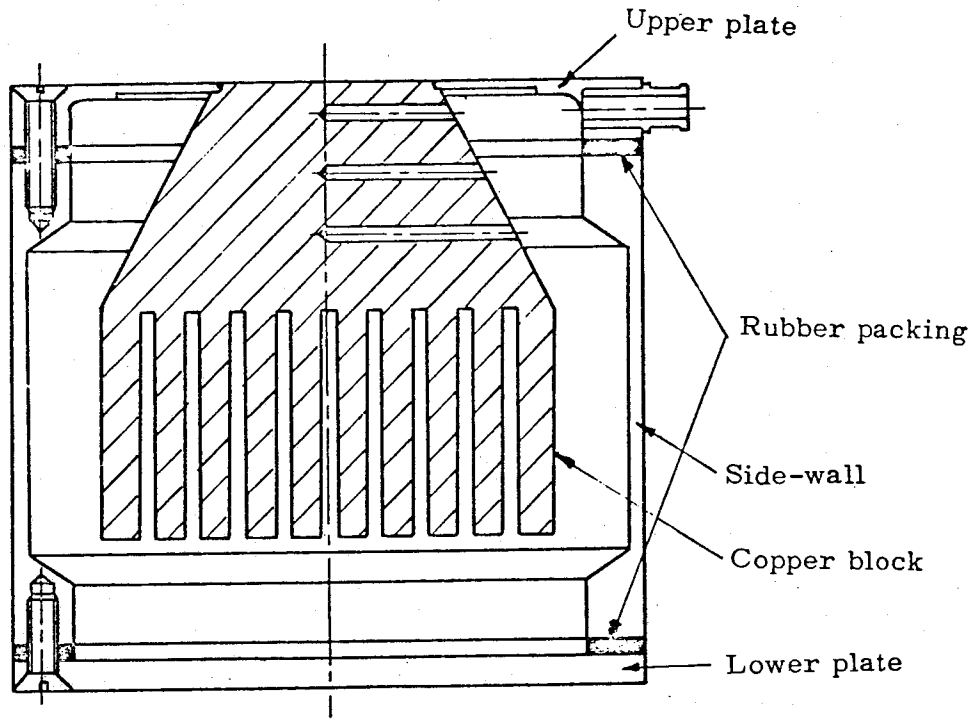

Fig. 2. Cross section of heater (space is filled with asbestos).

copper block, 30 mm in diameter, stored nine electric heating elements. The space between the copper block and the heater wall was filled with asbestos for insulation. Three copper-constantan thermocouples were set up at intervals along the axis of the copper block in its upper part to measure the temperature gradient which is closely related to the heat flux at the heated surface. The relationship between the temperature gradient and the heat flux has been calibrated

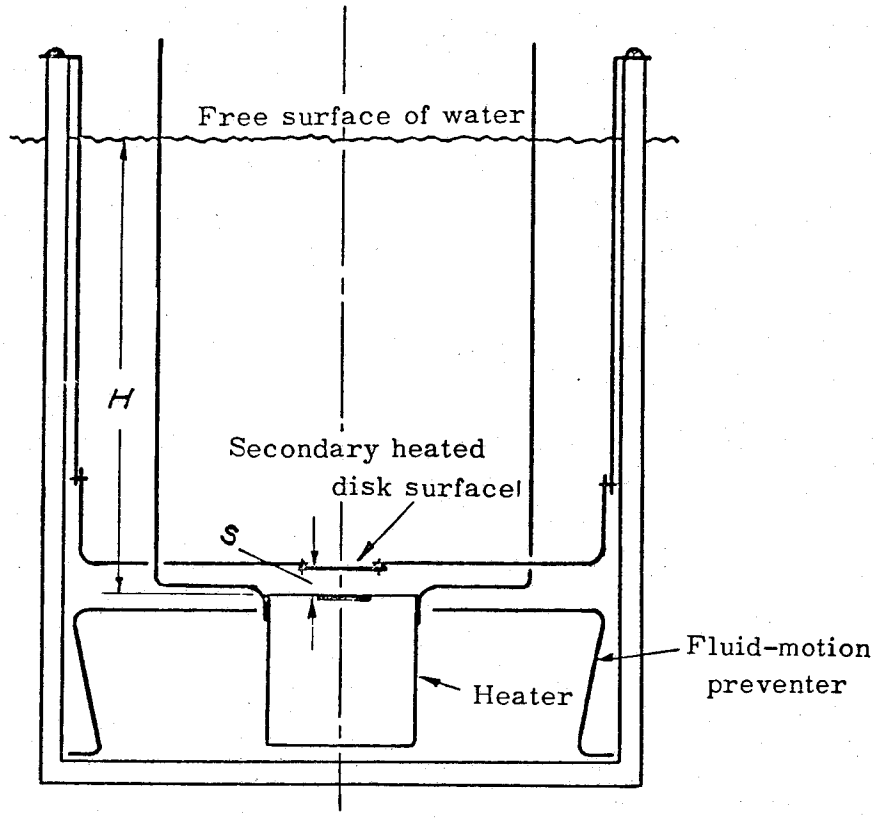

Fig. 3. Orientation of the secondary heated disk surface.

and showed the thermal leakage of input energy to the parts other than the heated surface is almost negligibly small when the heated surface is kept in the nucleate boiling regime with a high heat flux.

The balance was specially designed so as to fit the size of the water vessel. The heater was suspended from the arm of the balance at a position of 200 mm left from the fulcrum, while a balancing weight was put on the right end of the arm. A dish for weights was suspended at each position of 100 mm left and right from the fulcrum. A mercury junction was used for the supply of electric power to the heater as well as for the measurement of the thermocouples.

2.2 Principles of Measurement

When the heater is suspended in still water, a buoyant force appears as the resultant effect of the static pressure which acts on the surface of the heater.

On the other hand, when nucleate boiling at a high heat flux is maintained on the heated disk surface of such a size as used in the present study, a vapor mass of comparatively simple shape (cf. Fig. 7) appears successively on the heated surface repeating the growth, detachment, and rising cycles. Since this is essentially the condition of natural convection in a broad sense, the pressure which acts on the upper surface of the heater becomes lower than that in the case of still water and resulting in an apparent increase of the buoyant force acting on the heater.

In measuring the aforementioned increase of the buoyant force by the balance, a fluid-motion preventer, which is made of a metal plate, was set up in the present study as shown in Fig. 1. The preventer plate had a hold through which the vertical movement of the heater was possible, but the motion of water under the preventer plate was suppressed. If the flow of water along the vertical side-wall of the heater is not prevented, the viscous force acting on the side-wall of the heater may disturb the measurement of the buoyant force.

In the case of nucleate boiling in the present study, the heated surface in the central part of the upper surface of the heater was always

covered by a growing vapor mass, so that it could be assumed that the vapor pressure within the vapor mass is exerted on the heated surface (the nucleate boiling liquid layer which exists on a heated surface is very thin). The vapor pressure within a vapor mass, p_V (temporal mean value), may be assumed to be somewhat different from the liquid pressure near the circumference of the heated surface, p_L, at the level of the upper surface of the heater.

In order to measure, approximately at least, the pressure difference, $p_V - p_L$, the following method was adopted. As shown in Fig. 3, the secondary heated disk surface which had the same diameter as the primary heated surface was set up at a distance s above the upper surface of the heater. Then, electric power was supplied to the secondary heated surface alone with a nucleate boiling on it, and the buoyant force of the heater was measured under such conditions by the balance. The buoyant force at the limit as $s \to 0$ can be assumed by extrapolation of experimental data obtained at different values of s. According to the experiment, the buoyant force undergoes a monotonous change in the vicinity of s = 0, so that it is not difficult to determine the limit, as $s \to 0$, of the buoyant force.

Comparing the limiting state, as $s \to 0$, of boiling on the secondary heated surface with the boiling of ordinary state on the primary heated surface at the same heat load, no difference was found in the circumstances of the production of vapor mass between the two cases. In the former case, however, the primary heated surface is subjected to the pressure of liquid near the heated surface instead of the pressure of vapor, so that the buoyant force which acts on the heater may differ between the two cases at the same heat load. Dividing the difference of the buoyant force by the area of hea heated surface yields $P_L - P_V$.

2.3 Secondary Heated Disk Surface

The secondary heated disk surface mentioned in the preceding section is made as follows. An enameled copper wire of 0.6 mm dia., which has been closely wound with a michrome wire of 0.14 mm dia., is curved like a "circinate incense stick" and is fixed on a Bakelite disk of 1 mm in thickness.

The heated surface thus made differs somewhat from the heated surface proper. However, the present study is not concerned with the active nucleation sites or nucleate boiling on the heated surface, but is concerned with vapor masses which are produced by the accumulation of generated vapor. Besides, it has been confirmed by Katto and Yokoya [1] that when nucleate boiling at a high heat flux is maintained on the heated surface of such a size as adopted in the present study, the transferred heat is entirely consumed in generating the vapor on the heated surface. Consequently, it may be assumed that the circumstances of the production of vapor mass are hardly affected by the slight difference in the form of the heated surface. This assumption is also supported by the high-speed photography of the behavior of the vapor mass.

3. Experimental Results

3.1 Increase in the Buoyant Force of Heater due to Boiling

The increase in the buoyant force of heater due to boiling mentioned in Sect. 2.2 is, of course, affected by the rate of heat transfer. Figure 4 shows, as a typical example, the experimental result obtained on a heated surface of 14.5 mm dia. with a depth of water of H = 150 mm, including data at low heat fluxes as well as those in the region of high heat fluxes. It may be noticed in Fig. 4 that there is a clear distinction in the characteristics of the increase of buoyant force between the isolated bubble region at low heat fluxes and the interference region of high heat fluxes.

The experimental results as shown in Fig. 4 are put together to give Fig. 5, which illustrates the region of high heat fluxes for various values of the depth of water, H. The ordinate of Fig. 5 represents the increase in the buoyant force (with the negative sign) divided by the area of the upper surface of the heater of 42-mm dia., that is, the reduction of the mean pressure on the upper surface of the heater. The abscissa of Fig. 5 represents the heat flux, q, across the heated surface. It may be noticed in Fig. 5 that when the depth of water is constant, the mean pressure on the upper surface of the heater decreases with an increase in heat flux, that is, an increase in the rate of vapor production. This is an obvious result, because the vapor which exists between the heated surface and the free surface of water is nothing else but the origin for the generation of pressure difference within the fluid.

It may be also noticed from Fig. 5 that as H increases, starting from H = 50 mm, the mean pressure on the upper surface of the heater decreases initially, but starts to rise after arriving at the minimum value at H = 150 mm. This phenomenon seems to be attributable to the following circumstances. In the region of low depth of water, the pressure difference within the fluid generated by the existence of vapor increases as the depth of water increases. However, the inner chamber of the water vessel used in the present study has a limited cross section of 150 x 150 mm as mentioned in Sect. 2.1. Consequently, further increase in the depth of water, H, leads to the situation where the side-wall of the inner

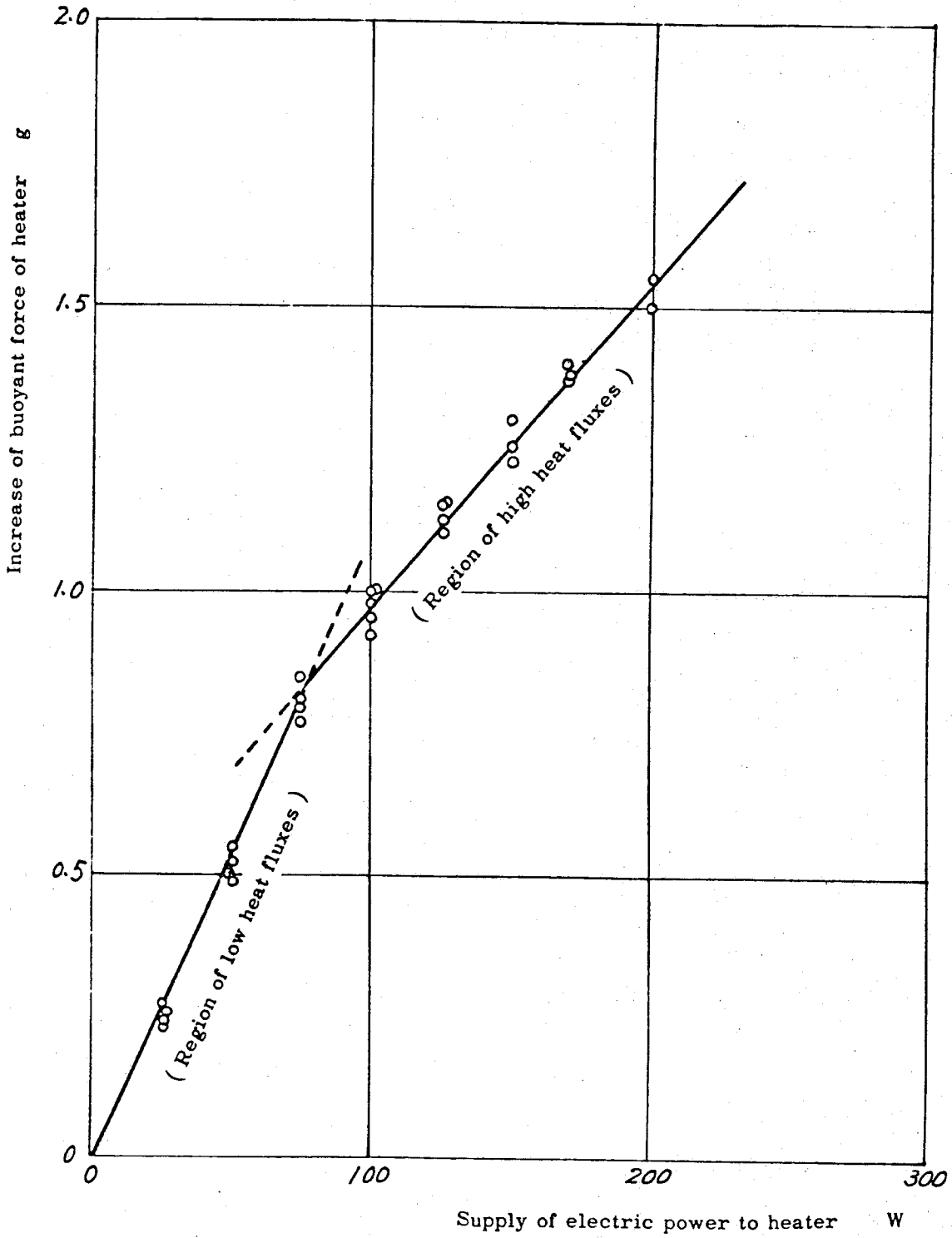

Fig. 4. Increase of buoyant force of heater due to boiling (diameter of heated surface: 14.5 mm, depth of water: H = 150 mm).

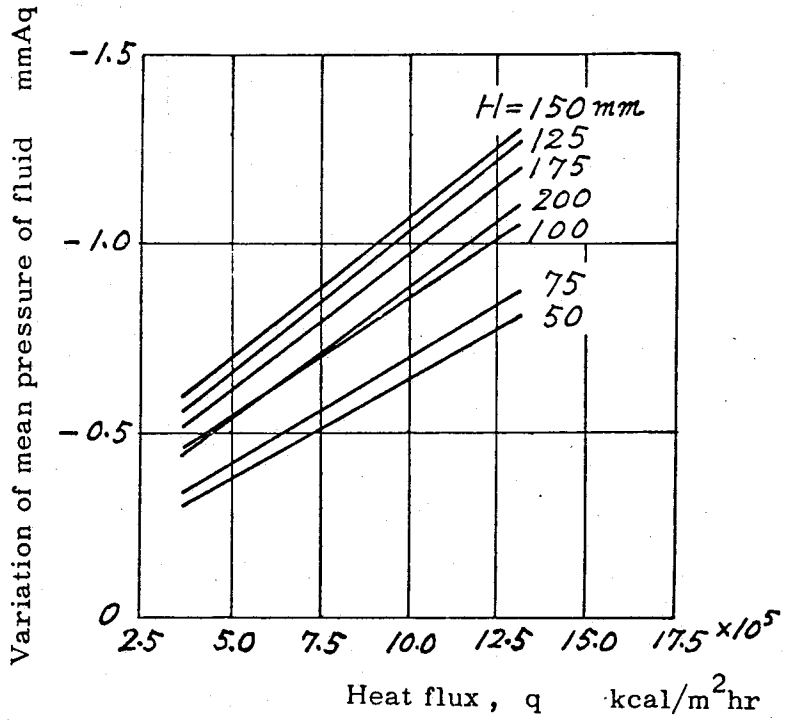

Fig. 5. Reduction of mean pressure of fluid on upper surface of heater (region of high heat fluxes).

chamber produces hindering effects on the flow of the fluid.

3.2 Vapor Pressure within a Growing Vapor Mass

The method mentioned in Sect. 2.2 was adopted for the measurement of pressure difference, p_V-p_L. Three different heated surface diameters, 16.7, 14.5 and 13.6 mm (area ratio, 1.50:1.12:1.00), were preliminarily tested for the case of H = 100 mm, and it was found that such a range of change in diameter had no effect on p_V-p_L.

Figure 6 shows the experimental results of p_V-p_L obtained on a heated surface of 14.5-mm dia. with the depth of water of H ranging from 20 mm to 200 mm and the heat flux ranging from 5×10^5 to 1.3×10^6 kcal/m^{2}hr. According to Fig. 6, p_V-p_L is less than zero, that is, p_V is less than p_L. In addition, it may be found that p_V-p_L becomes constant almost independently of either q and H in the region of high water-depth $H \geqq 100$ mm, while p_V-p_L is affected by q as well as H in the region of low water-depth, $H < 100$ mm.

4. Analysis of Vapor Pressure within a Vapor Mass

4.1 Behavior of a Vapor Mass on the Heated Surface

In order to clear up the reason for the characteristics of p_V-p_L as shown in Fig. 6, we must know the behavior of the vapor masses repeating their growth and departure from the heated surface. As shown in Fig. 7, the vapor which is generated by violent nucleate boiling within a thin liquid-layer on a heated surface inflates a vapor mass, and when the vapor mass is fully grown and detaches from the heated surface, the succeeding vapor mass starts to grow almost immediately. High-speed photography is employed to observe the behavior of vapor mass, and the analysis of recorded pictures of vapor mass discloses the following features:

(i) Detachment period of a vapor mass scatters to some extent, but the statistical mean value, τ, is about 60 ms and the value is hardly changed by the water-depth H or the heat flux q within the experimental ranges of the present study. This value of τ with the aforementioned characteristics agrees with that which has been found in the experiment of Katto and Yokoya [1, 2].

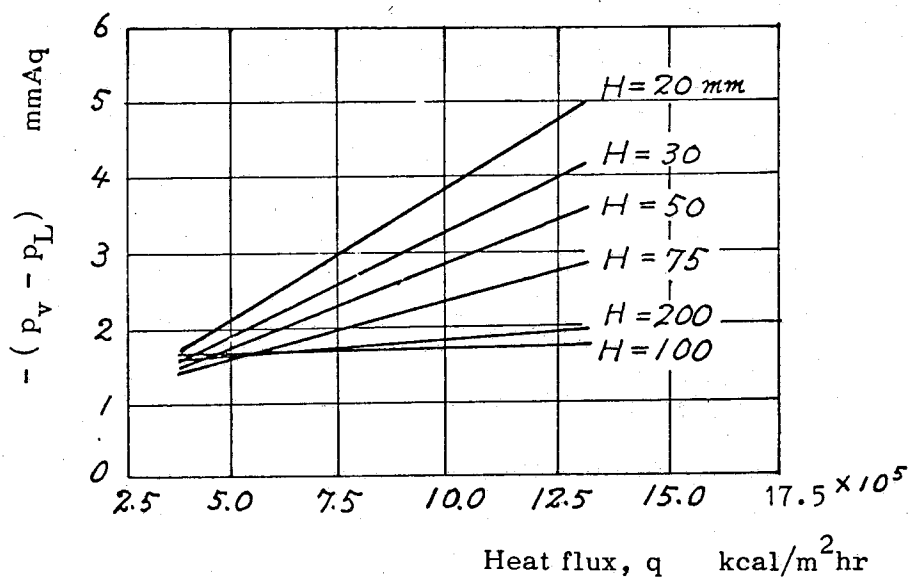

Fig. 6. Vapor pressure within vapor mass.

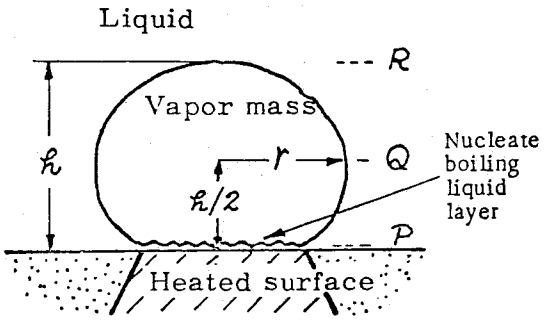

Fig. 7. A vapor mass on heated surface.

(ii) Height of the top surface of a vapor mass, h, shown in Fig. 7, increases with time, t, of course. Figure 8 shows the variation of h with time in the case of comparatively high water-depth of H = 100 mm, and it may be noticed that the rising velocity of top surface of vapor mass, v = dh/dt, which is kept constant during the growth of vapor mass, is not altered by the heat flux, q. On the other hand, Fig. 9 shows the ca case of low water-depth of H = 30 mm, and here the special attention should be paid to the fact that v = dg/dt increases with the heat flux. A broken line in Fig. 9 is the reproduction of the result of Fig. 8 for the sake of comparison. When the depth of water is low, the quantity of liquid which exists above a vapor mass is insufficient so

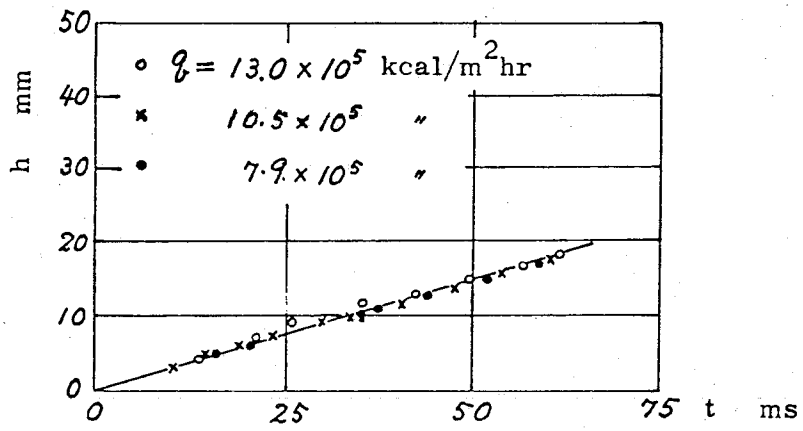

Fig. 8. Rise of top surface of vapor mass (depth of water H = 100 mm).

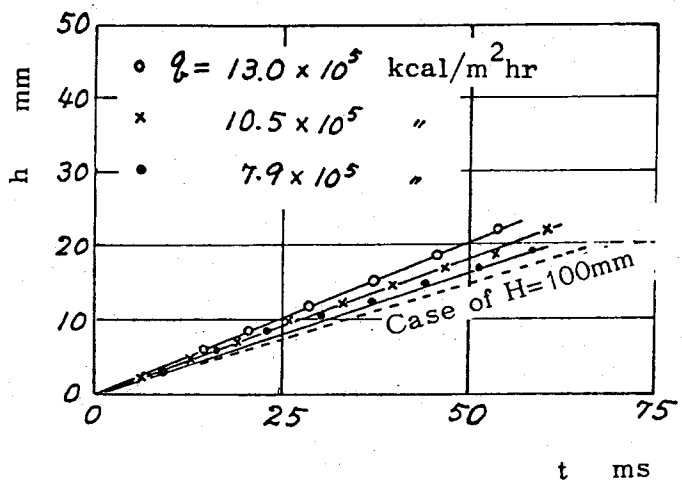

Fig. 9. Rise of top surface of vapor mass (depth of water H = = 30 mm).

that the motion of the top surface of a vapor mass may have special characteristics different from the case of sufficiently high water depth.

4.2 Analysis

A very simple analysis of the pressure within a vapor mass is attempted below using the observed data for vapor mass described in the preceding section. Here, the effect of the inertia of liquid around the vapor mass is ignored for simplicity.

The static pressure of liquid exerted on the interface of the vapor mass varies, of course, with the water level. In Fig. 7, for example, the water pressure is maximum at the level of P and minimum at the level of R (p_L referred to in Sect. 2.2 is the water pressure at the level of P). On the other hand, the vapor pressure within a vapor mass may be assumed, for simplicity, to be approximately uniform taking an instantaneous value of p_V at an arbitrary time.

We consider the mean value of the water pressure, p_m and it is assumed to be approximately equal to the water pressure at a height from the heated surface of h/2 (level of Q in Fig. 7), so that

$$p_m = p_L - \rho_L g \frac{h}{2} \tag{1}$$

where ρ_L is the density of liquid and g is the acceleration due to gravity. Then the vapor pressure within a vapor mass, p_V', ought to be higher than p_m due to the effect of surface tension. It may be assumed without a gross error that for the circumstances of Fig. 7, the difference of pressure, Δp, takes approximately a value of order such as

$$\Delta p = \sigma\left(\frac{1}{h/2} + \frac{1}{r}\right) \tag{2}$$

where σ is the surface tension and r is the radius of the horizontal section of vapor mass at the level of Q in Fig. 7. Equations (1) and (2) yield the vapor pressure within a vapor mass (instantaneous value), p_V', as follows:

$$p_V' = p_m + \Delta p = p_L - \rho_L g \frac{h}{2} + \sigma \frac{1}{h/2} + \sigma \frac{1}{r} \tag{3}$$

With the advance of growth of vapor mass, h and r vary with time, so that the vapor pressure, p_V', is subject to considerable variation during the growth period of the vapor mass, $t = 0 \sim \tau$. In comparison with the variation of p_V', the variation of p_L might be presumed to be so minor that p_L is assumed to be approximately constant in Eq. (3). Then, integrating Eq. (3) through the growth period of $t = 0 \sim \tau$ and dividing the integral by τ, yields p_V, that is, the mean value of p_V

$$p_V = \frac{1}{\tau}\int_0^\tau p_V' dt = p_L - \frac{1}{\tau}\int_0^\tau \rho_L g \frac{h}{2} dt + \frac{1}{\tau}\int_0^\tau \sigma \frac{1}{h/2} dt + \frac{1}{\tau}\int_0^\tau \sigma \frac{1}{r} dt \tag{4}$$

For the h included in the second and third terms in the right-hand side of Eq. (4), the relation of h = vt, where v is a constant rising velocity of top surface of vapor mass, has been given in the experimental results of Figs. 8 and 9. In this case, however, there arises a problem that the definite integral in the third term of the righthand side of Eq. (4) becomes infinitely great. However, the observation by means of high-speed photography reveals that a new vapor mass which succeeds

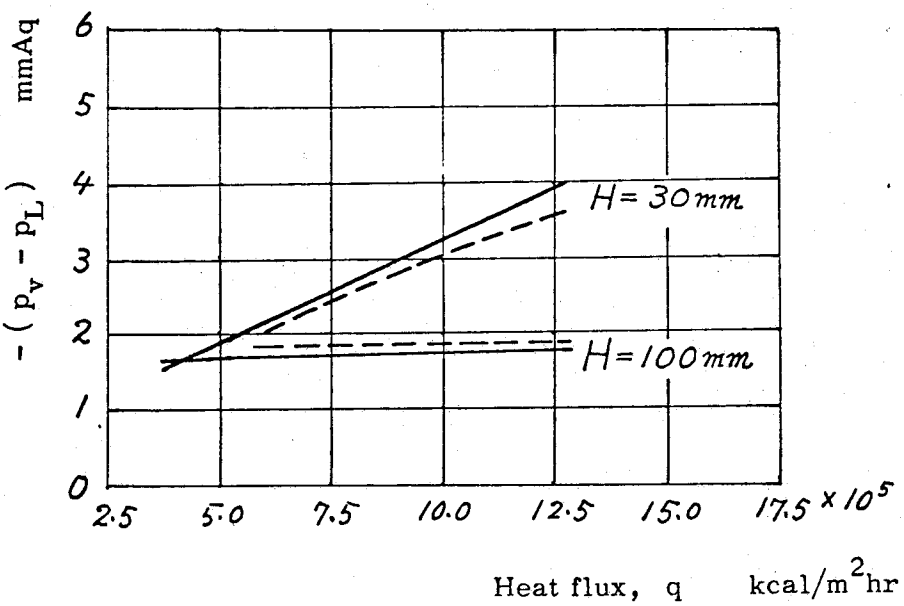

Fig. 10. Vapor pressure within vapor mass — comparison between experiment (solid curves) and analysis (broken curves).

the detachment of the preceding vapor mass always starts to grow with the coalescence of a lot of little bubbles which have existed on the heated surface, and the diameter of the little bubbles is of the order of about 1/10 of h_{max}, where h_{max} is h of the fully grown vapor mass, that is, $h_{max} = v\tau$. Accordingly, if t = 0 is replaced by $t = \tau/10$ for the lower limit of the definite integral in the third term of the righthand side of Eq. (4), it is almost certain that a serious error will not be introduced.

The r included in the fourth term in the right-hand side of Eq. (4) varies with time, but it keeps the same order as the radius of the heated surface (cf. Fig. 7) so that its variation is comparatively small. It will not yield a gross error, therefore, if 1/r is replaced by $1/r_m$, the mean value of 1/r through the growth period of vapor mass, which can be determined easily from the analysis of the recorded film.

Based on the above circumstances, Eq. (4) yields finally the following result:

$$p_V - p_L = -\frac{1}{4}\rho_L g v\tau + \sigma\frac{2\ln 10}{v\tau} + \sigma\frac{1}{r_m} \qquad (5)$$

where v has been determined in the experiment of Figs. 8 and 9, and τ has also been determined to be 60 ms as explained in Sect. 4.1. Two broken curves in Fig. 10 represent $p_V - p_L$ calculated by Eq. (5) for the case of H = 30 mm and H = 100 mm, while the corresponding solid lines in Fig. 10 are the reproduction of the experimental results for H = 30 mm and H = 100 mm which have been shown in Fig. 6. It may be noticed that characteristics of $p_V - p_L$ obtained in the experiment of Fig. 6 are explained successfully by the present analysis. The fact cannot be ignored, of course, that the present analysis has been made on the basis of several assumptions, but the adopted assumptions are physically not so dubious.

5. Discussion

5.1 The Behavior of Vapor Mass

The growth of a vapor mass is maintained on the heated surface being supplied with vapor from the nucleate boiling liquid layer. It may be assumed from the results obtained so far in the present study, especially from Fig. 10, that the growth of a vapor mass takes place while floating up through the liquid rather than while forcing away the liquid that surrounds the vapor mass. According to Fig. 10, p_V is lower than p_L analytically as well as experimentally. In the incipient stage of the growth of vapor mass, h is small in the right-hand side of Eq. (3) and the third term is greater than the second term, so that p_V is higher than p_L. However, after the incipient stage is over, the increase of the volume of vapor mass is accomplished presumably by the cooperative action of the floating up of vapor mass and the supply of vapor from the heated surface. Since the rate of the vapor supply from the heated surface is kept constant in nucleate boiling at high heat fluxes, which has been confirmed by Katto and Yokoya [2] with a heated surface of similar

size as in the present study, the detachment of vapor mass does occur when the rate of growth of vapor mass due to the floating up is accelerated enough to exceed the rate of supply of vapor from the heated surface.

In this connection, it may be of interest to note the rising velocity of top surface of vapor mass, v. As shown in Fig. 8, v = 300 mm/s when the depth of water is sufficiently high, and this value of v is found to be nearly equal to the free-rising velocity of a single bubble in water with an equivalent diameter of about 20 mm (N.B., the vapor mass in the present study appears with a diameter somewhat larger than the diameter of the heated surface) [3]. A simple analogy should not be drawn between the two cases because there is some difference in the mechanical conditions, but the similarity in the two rising velocities may be worth noting.

5.2 Lower Limit of the Region of High Heat Fluxes in Saturated Boiling

It is well-known that in saturated nucleate boiling isolated bubbles are produced successively at each active nucleation site in the region of low heat fluxes, but the aspect changes in the region of high heat fluxes where vapor masses appear due to the accumulation of vapor produced at crowded active nucleation sites.

At high heat fluxes, the heat transferred from the heated surface is almost entirely consumed by the production of vapor on the heated surface, and it is especially true for a heated surface of the size and shape adopted in the present study. Therefore, a uniform outflow of vapor from the heated surface may be approximately assumed to give the outflow velocity, v_s, as follows:

$$v_s = q/\rho_V L \tag{6}$$

where q is the heat flux, ρ_V the density of vapor, and L the latent heat of vaporization.

In order for the outflow of vapor to be able to generate a vapor mass on the heated surface, the outflow velocity of vapor v_s should be presumably greater than the rising velocity of the top surface of the vapor mass v. In view of the discussion in Sect. 5.1 on the behavior of vapor mass, v can be regarded as a quantity rather independent of v_s. Consequently, the heat flux at the lower limit of the high heat flux region may be determined by setting v_s = v in Eq. (6), that is

$$q = v\rho_V L \tag{7}$$

In the present study, v = 300 mm/s for the region of high water-depth as is mentioned in Sect. 5.1, and ρ_V = 0.5977 kg/m^3, L = 538.8 kcal/kg for saturated steam at atmospheric pressure. Substituting these values into Eq. (7) yields

$$q = 3.69 \times 10^5 \text{ kcal/m}^2\text{h}$$

On the other hand, Fig. 4 has been given as a typical example for the variation of buoyant force of heater with heat flux. The point where the data changes abruptly its characteristics in Fig. 4 corresponds to the boundary between the low and high heat flux regions, and the heat flux at the point is found from Fig. 4 to be

$$q = 3.66 \times 10^5 \text{ kcal/m}^2\text{h}$$

Figure 4 is a case of depth of water of H = 150 mm, and it may be of interest to note that almost the same value of q appears in the experiments with H = 100, 125, 175 and 200 mm, respectively, although the buoyant force varies its magnitude with H (cf. Fig. 5). The agreement between the value of q calculated by Eq. (7) and the experimental data is satisfactory.

5.3 The Burnout Heat Flux in Saturated Boiling

Katto and Yokoya [2], who carried out the experimental study of burnout in various systems of pool boiling, reported that burnout takes place when the detachment period of vapor mass on the heated surface becomes equal to the period during which the nucleate boiling liquid layer on the heated surface is consumed, so that the burnout heat flux has the following relation:

$$\tau = \xi \rho_L \delta_0 L/q \tag{8}$$

where ξ = 8/9 (a constant), ρ_L is the density of liquid, δ_0 is the initial thickness of nucleate boiling liquid layer, and L is the latent heat of va vaporization. In addition, δ_0 follows the empirical relation of $\delta_0 \propto 1/q^{1.5}$.

Equation (8) is universal in its application embracing the burnout in special systems of boiling such as the heated surface having an interference from a nearby body, the low depth of liquid, etc. However, only the normal system of boiling is specially considered here, and in this case, it may be assumed from Sects. 5.1 and 5.2 that the fundamental factors which have close relations to the behavior of vapor mass are (i) the rising velocity of top surface of vapor mass v, (ii) the height of top surface of vapor mass at its detachment h_{max}, and (iii) the outflow velocity of vapor from the heated surface $q/\rho_V L$. If it is assumed that the phenomenon is governed by these three factors alone, the following result may be derived.

First, the detachment period of vapor mass is given by

$$\tau = h_{max}/v \tag{9}$$

Next, noting that δ_0 has a dimension of length and follows the empirical relation of $\delta_0 \propto 1/q^{1.5}$,

we get

$$\delta_0 = \text{constant} \times h_{max} \left(\frac{v}{q/\rho_V L} \right)^{1.5} \qquad (10)$$

As mentioned in Sect. 5.1, v included in Eqs. (9) and (10) takes a value near the free rising velocity of a bubble, and it may be assumed without generating a gross error that the free rising velocity of a bubble or a vapor mass of ordinary size encountered in boiling is proportional to $\sqrt[4]{\sigma g(\rho_L - \rho_V)/\rho_L^2}$. Consequently, the following relation is obtained:

$$v = \text{constant} \times \sqrt[4]{\sigma g(\rho_L - \rho_V)/\rho_L^2} \qquad (11)$$

Substituting Eqs. (9), (10) and (11) into Eq. (8), yields

$$\frac{q}{\rho_V L} \Big/ \left[\frac{\sigma g(\rho_L - \rho_V)}{\rho_V^2} \right]^{1/4} = \text{定数} \left(\frac{\rho_V}{\rho_L} \right)^{0.1} \qquad (12)$$

In Eq. (12), the variation of $(\rho_V/\rho_L)^{0.1}$ is very minor due to the exponent of 0.1, so that the right-hand side of Eq. (12) may be assumed to be approximately constant. Equation (12) accordingly can be regarded to agree with Kutateladze's equation for burnout heat flux for the normal system of boiling. In the boiling at extreme low pressure, $\rho_V \to 0$ in the right-hand side of Eq. (12), but such a condition of pressure should be excluded because giant incipient bubbles appear on a heated surface [4] and the aspect becomes different from normal state of boiling.

6. Conclusion

(1) A heater with a horizontal heated disk surface of about 15-mm dia. was suspended in water at atmospheric pressure by the arm of a balance. The buoyant force of the heater was measured with nucleate boiling in order to determine the approximate value of vapor pressure within a vapor mass produced on the heated surface.

(2) The vapor pressure within a vapor mass, p_V, was found to be lower than the water pressure at the level of heated surface, p_L. The pressure difference, $p_V - p_L$, was kept nearly at constant independently of the water depth, H, as well as the heat flux, q, for $H \geqq 100$ mm, but $p_V - p_L$ varies with H and q for $H > 100$ mm.

(3) High-speed photography was employed to observe the characteristics such as the detachment period of vapor mass, τ, the rising velocity of top surface of vapor mass, v, etc. A simple analysis of $p_V - p_L$ was successfully attempted to explain the characteristic aspect of $p_V - p_L$ mentioned in conclusion (2).

(4) The results obtained in the present study suggest that the growth of a vapor mass takes place while floating up through the liquid rather than while forcing away the liquid that surrounds the vapor mass.

(5) The experimental data obtained in the present study support the assumption that when the outflow velocity of vapor from the heated surface v_s exceeds the rising velocity of top surface of vapor mass v the high heat flux region appears accompanying the generation of vapor masses.

(6) A discussion of the burnout heat flux in connection with the present study is made.

Acknowledgment. The authors express their appreciation to M. N. Kim and Mr. H. Koizumi for their assistance to the present study.

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□◆◆◆□

Behavior of a Vapor Mass in Saturated Nucleate and Transition Pool Boiling

YOSHIRO KATTO AND SADAO YOKOYA*

Growth and detachment of vapor mass in the range from nucleate to film boiling was studied by high-speed photography of pool boiling of saturated water at atmospheric pressure on a horizontal 10 mm-dia. heated disk surface. The resulting vapor mass can be classified into three types: A-type, which is not affected by the preceding vapor mass; a-type, which appears immediately after the A-type mass; and B-type, which has features intermediate between A-type and a-type. The lengths of existence of vapor masses, are τ_A = 47 ms, τ_a = 19 ms, and τ_B = = 33 ms.

1. INTRODUCTION

When the temperature of a heated surface is sufficiently high in saturated nucleate pool boiling under usual conditions, the vapor generated at numerous nucleation sites over the heated surface forms vapor masses as the result of coalescence, with the exception in the particular case of boiling with a heated surface such as a very thin wire, or at very low pressure, where the size of the incipient bubble is greater or in the same order as compared with that of the heated surface. The vapor mass aforementioned grown to a certain volume, detaches from the heated surface and moves upward in the liquid, and this behavior of growth and detachment of vapor mass is successively repeated on the heated surface in a cyclic manner.

Then, further rise of the temperature of heated surface leads to the change of the state of boiling from nucleate to transition boiling through burnout. Hence, it would be worthwhile to know the mechanical situation which controls the behavior of the growth and detachment of vapor mass, since the phenomenon of the periodic stay of vapor mass on the heated surface has important effects on the occurrence of burnout and transition boiling (cf. [1,2]).

Honda and Nishikawa [3] carried out experiments of pool boiling of saturated water at atmospheric pressure on a horizontal 8 mm-dia. heated disk surface. From their observation of the behavior of the vapor mass, that the value of fD_d (where f is the detachment frequency of vapor mass, and D_d is the diameter of vapor mass at detachment) as well as the drag coefficient C_D for ascending motion of the vapor mass in the region of nucleate boiling differs from that in the region of transition boiling, they concluded that the hydrodynamical behavior of vapor mass should be different in the two boiling regions. Then, they attempted a theoretical analysis of the transition boiling heat transfer, in which they assumed that in the region of transition boiling, the

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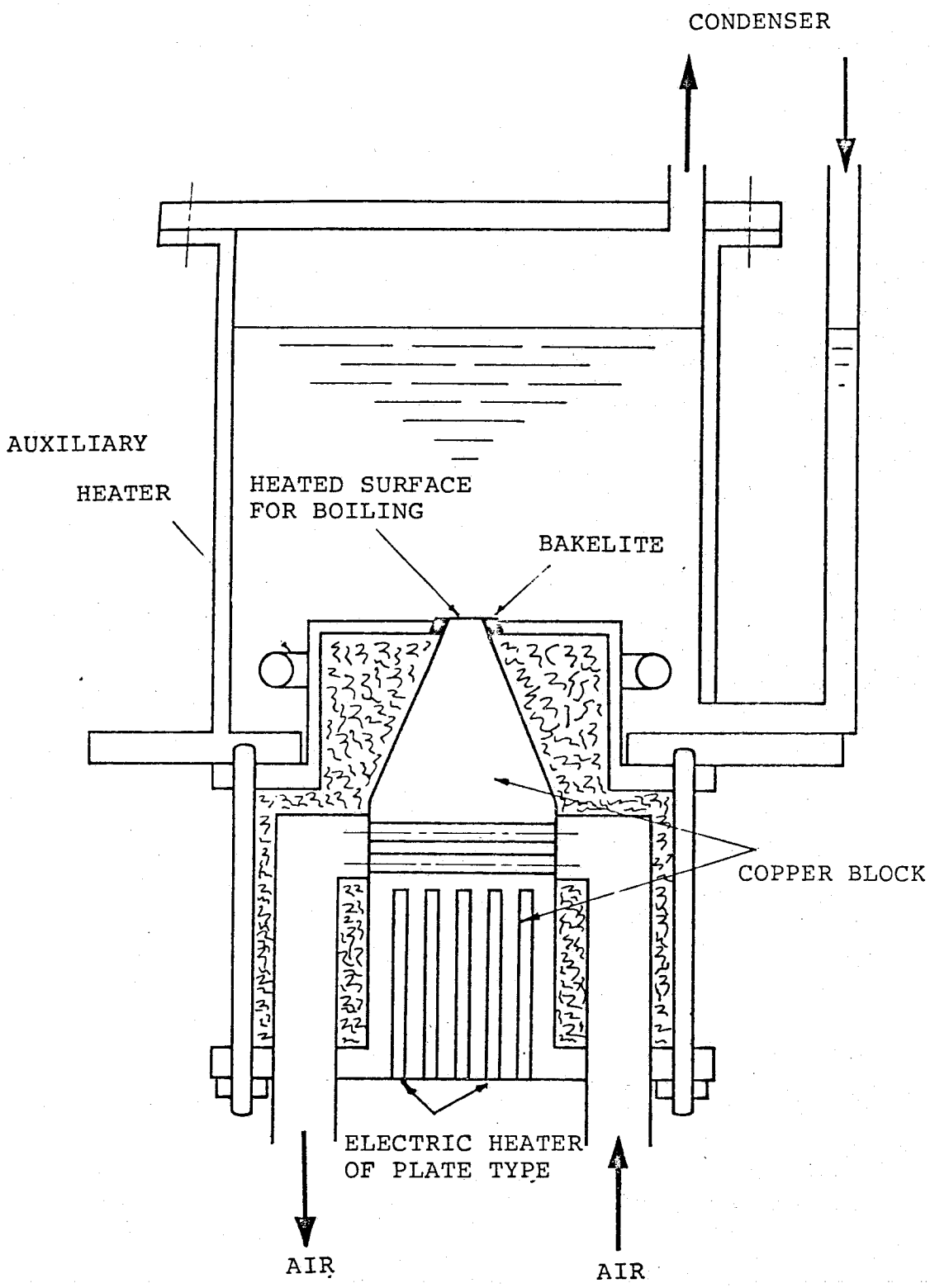

Fig. 1. Main part of experimental apparatus.

mass was different from that in the region of nucleate boiling, because of the dried portion that appeared on the heated surface covered with the vapor mass, provoking adhesive forces due to surface tension as well as the change of pressure at the bottom of the vapor mass. In the experiment of Honda and Nishikawa, the behavior of the vapor mass was observed by a special method employing a camera for the exclusive use of the electromagnetic oscillograph, in which the section of a vapor mass cut at the position 1 mm above the heated surface was continuously recorded on a running sensitive paper of oscillograph through a narrow slit of the camera.

The present paper deals with pool boiling of saturated water at atmospheric pressure on a horizontal 10 mm-dia. heated disk surface, which is almost the same conditions as in the study of Honda and Nishikawa. High-speed photography is employed here to record the instantaneous figure of the vapor masses, thereby the behavior of the growth and detachment of vapor mass is analyzed; a theoretical examination is also attempted on the hydrodynamic condition concerned.

2. EXPERIMENTAL DATA USED IN THE ANALYSIS OF VAPOR MASS

The photographical record used in the present paper to analyze the instantaneous behavior of the vapor mass is obtained in the boiling that takes place in the boiler of the experimental apparatus schematically shown in Fig. 1.

A horizontal 10 mm-dia. disk surface at the top of a conical copper block serves as the heated surface for boiling. The lower portion of the copper block has been designed to be heated by electric heaters of the plate type and at the same time cooled by the flow of air according to the method originated by Isshiki and Tamaki [4] for realizing transition boiling in the system with an

ture and the heat flux at the heated surface can be determined by the temperature distribution measured by the thermocouples located along the center line of the conical copper block. An auxiliary heater is also employed to keep the water in the boiler (distilled water) at saturation temperature. The level of the free surface of water measured from the heated surface is always kept approximately 100 mm high. In recording the behavior of vapor mass on the heated surface, maximum film speed and exposure are 2000 frames per second and 1/5000 second per frame, respectively. Time marks at the interval of 1 ms and the image of a round bar of 6 mm in diameter are simultaneously recorded on the film for the measurement of the velocity and dimensions concerned with the phenomena.

3. OBSERVATIONS OF THE BEHAVIOR OF VAPOR MASS

Figure 2 shows the boiling curves obtained in the experiment conducted with the experimental apparatus shown in Fig. 1. The broken curve that runs through the data denoted by white dots represents the results obtained with a comparatively clean heated surface, while the solid curve that runs through the data of black dots represents the results obtained with a comparatively contaminated heated surface.

In Fig. 2, where T_W-T_S on the abscissa is the superheat of the heated surface (T_W: the temperature of the heated surface, and T_S: the saturation temperature), it may be noticed that nucleate boiling occurs in the range of $T_w - T_s < 20°C$, transition boiling in the range of T_W-T_S between 20°C and 70°C, and film boiling in the range of $T_w - T_s < 70°C$. In addition, Fig. 2 shows that, in the region of transition boiling, the contamination on the heated surface has a remarkable effect upon the characteristics of heat transfer; but it has already been revealed that this phenomenon occurs

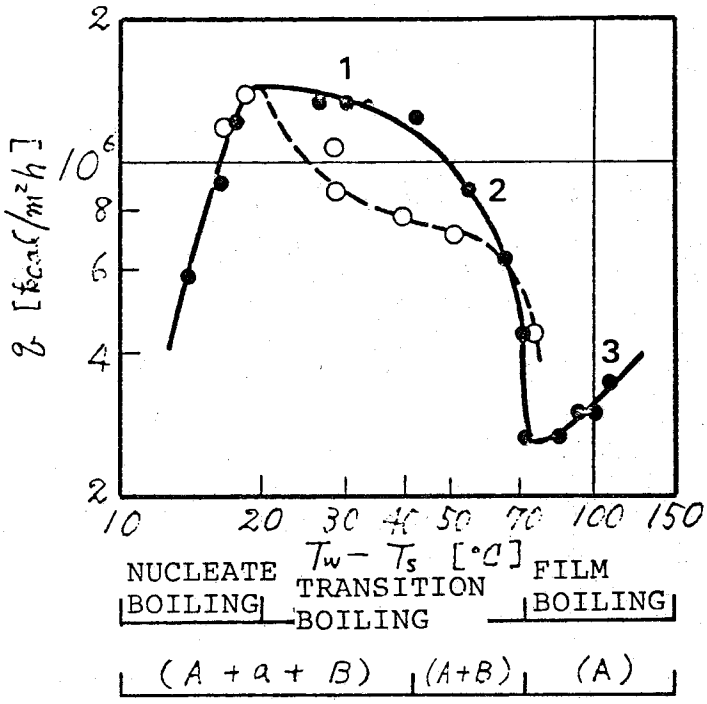

Fig. 2. Boiling curve.

through the appearance of the dried portion on the heated surface in the region of transition boiling [2,5] and the change of affinity of the heated surface with the liquid due to the contamination.

3.1 Classification of a Single Vapor Mass According to the Type of Growth and Detachment

If the behavior of the growth and detachment of vapor mass is examined along the boiling curve shown in Fig. 2, the following facts are readily found. That is, the behavior of vapor mass at the experimental points denoted by 1, 2 and 3, for example, in Fig. 2, are shown in Figs. 3, 4 and 5, respectively, and judging from the apparent features shown in these figures, the vapor mass on the heated surface may be tentatively classified into the following three types.

(I) Vapor mass of type A. Vapor masses shown in frames 1 to 5 in Fig. 3, frames 1 to 3 in Fig. 4, and frames 1 to 5 in Fig. 5 are called type A, with a common feature that a vapor mass grows independently being hardly affected by the preceding vapor mass until it detaches from the heated surface. It may be noticed that the vapor mass of type A is maintained in an approximately spherical shape until it moves up some distance after detaching from the heated surface, and thereafter the vapor mass transforms to a hemispherical shape accompanied by the induced flow at the wake of vapor mass.

(II) Vapor mass of type a. This is the vapor mass which appears as if it immediately follows in the wake of the vapor mass of type A, and grows in the vertical direction without extending its horizontal dimension in such a manner that the tip of the vapor mass penetrates the preceding A-type

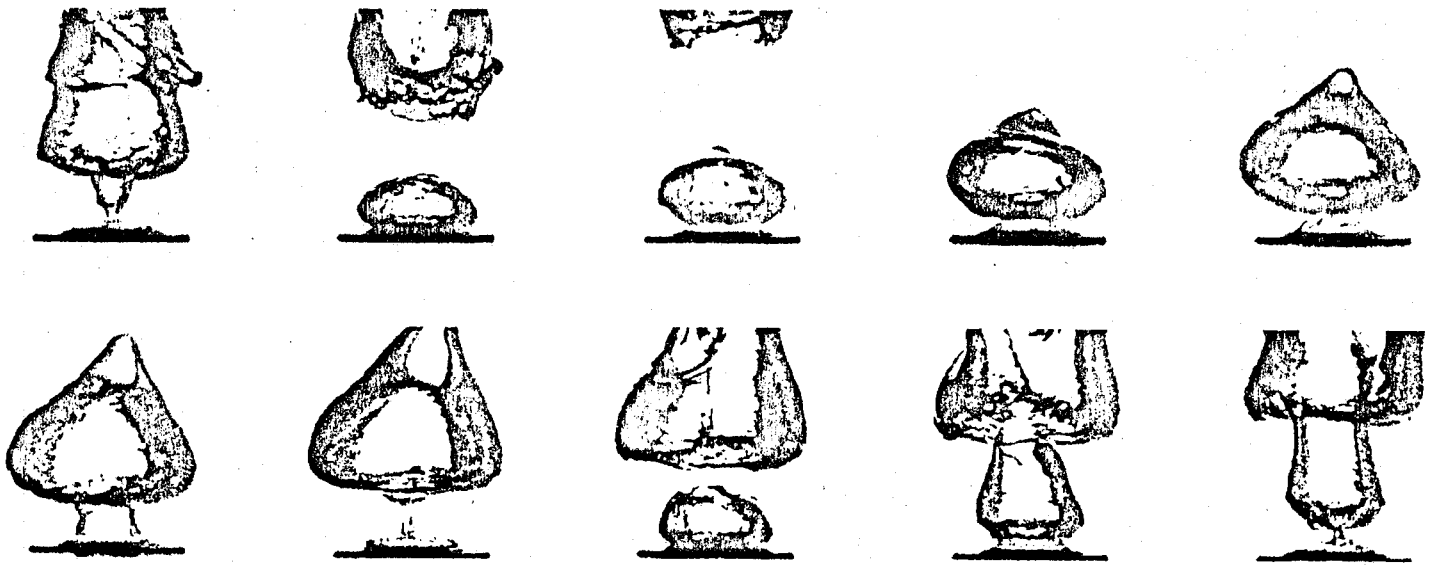

Fig. 3. Behavior of vapor mass at the experimental point 1 in Fig. 2. Time interval between each frame: 11.3 ms for frames 1 to 5, and 8.8 ms for frames 5 to 10.

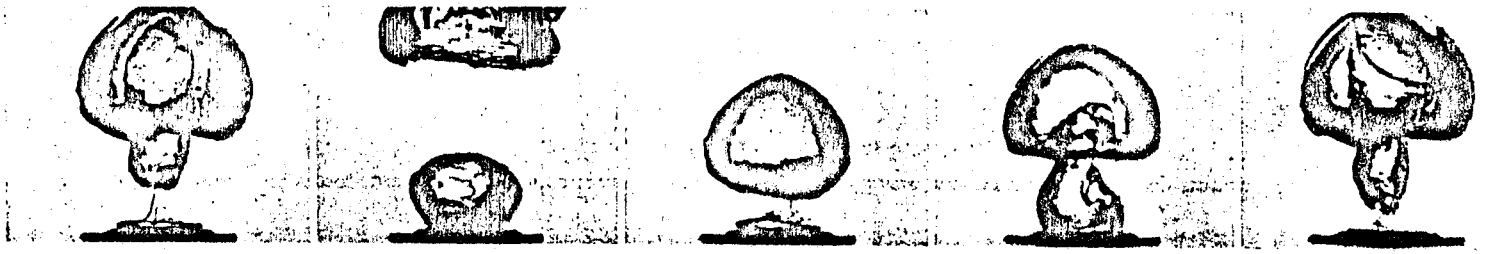

Fig. 4. Behavior of vapor mass at the experimental point 2 in Fig. 2. Time interval between each frame: 26.8 ms for frames 1 to 3, and 15.0 ms for frames 3 to 5.

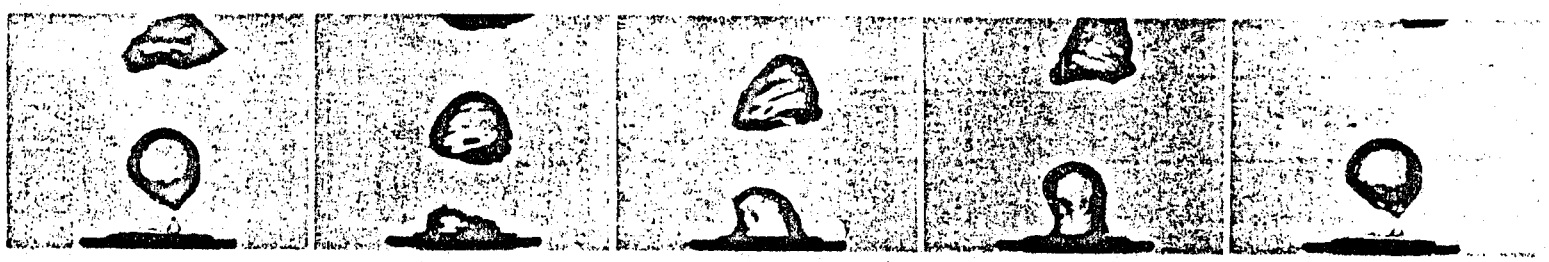

Fig. 5. Behavior of vapor mass at the experimental point 3 in Fig. 2. Time interval between each frame: 14.0 ms for frames 1 to 5.

vapor mass before it finally detaches from the heated surface, as shown in frames 3 to 6 in Fig. 3. After detaching from the heated surface, the vapor mass of a-type in most cases, is swallowed up in the preceding A-type vapor mass. Based on these facts, it is presumed that the behavior of the growth and detachment of a-type vapor mass is strongly affected by the induced flow which is generated in the wake of the preceding A-type vapor mass. Moreover, it should be mentioned here that the detachment of the A-type mass from the heated surface takes place in a vague manner when it is accompanied by a-type vapor mass.

(III) Vapor mass of type B. This is the vapor mass which succeeds to either the a-type vapor mass following A-type vapor mass, or the single A-type vapor mass as shown in frames 7 to 10 in Fig. 3, and frames 3 to 5 in Fig. 4. Vapor mass of type B moves upward being affected by the induced flow of the preceding vapor mass that has moved a considerable distance from the heated surface, and finally detaches from the heated surface. Accordingly, the B-type vapor mass may be said to have intermediate properties between the A-type and the a-type vapor mass.

3.2 Successive Order of Vapor Mass

From the classification of a single vapor mass described in the preceding section, we can denote the order of successively appearing vapor mass in Fig. 3 as (A+a+B), that in Fig. 4 as (A+B), and that in Fig. 5 as (A). The regions where the aforementioned type of succession of vapor mass appears respectively are shown at the lowest part of Fig. 2. It may not be useless to mention the rare experience encountered in the present experiment that, in the region of (A+a+B), the succession type of (A+a) appears lacking the generation of B-type vapor mass, or two succession types of (A+a+B) and of (A+a) appear alternatively at certain intervals. In addition, the vicinity of the boundary between the region of (A+a+B) and that of (A+B), is always subjected to the state that the succession of (A+a+B) and that of (A+B) appear alternatively at certain intervals.

It should be noted that the vapor masses of type A which appear in Figs. 3 and 5, for example, have been classified into the same A-type in Sect. 3.1, but it does not mean that they are in the same mechanical situation. Moreover, the variation of the succession type of vapor mass with $T_w - T_s$ shown at the lower part of Fig. 2, will not occur depending on the magnitude of heat flux q alone. For example, in the region of nucleate boiling of q less than 6×10^5 kcal/m^{2}h, the generation of vapor mass becomes disorderly, and no vapor mass appears in the region of much lower heat flux. On the other hand, in the region of transition or film boiling, the generation of vapor mass occurs in a comparatively regular manner even for the heat flux q less than 6×10^5 kcal/m^{2}h.

3.3 Duration of a Vapor Mass on the Heated Surface

Moving from the nucleate boiling to the film boiling, through the transition boiling, with the rise of superheat of heated surface $T_w - T_s$ in Fig. 2, the pattern of the generation of vapor mass generally changes in the order of (A+a+B) → → (A+B) → (A).

Three typical cases of the results obtained directly from the analysis of the photographic record on the film taken at each experimental point in Fig. 2, are shown in Fig. 6, where the abscissas are the sequence of the vapor masses appearing with lapse of time (● denotes A-type, x a-type, and △ B-type), and the ordinates are the duration of each vapor mass (that is, the period required for the vapor mass from the start of growth on the heated surface to the detachment). The uppermost figure in Fig. 6 shows the state in the region of nucleate boiling with the succession type of (A+a+B), the middle figure shows the state in the

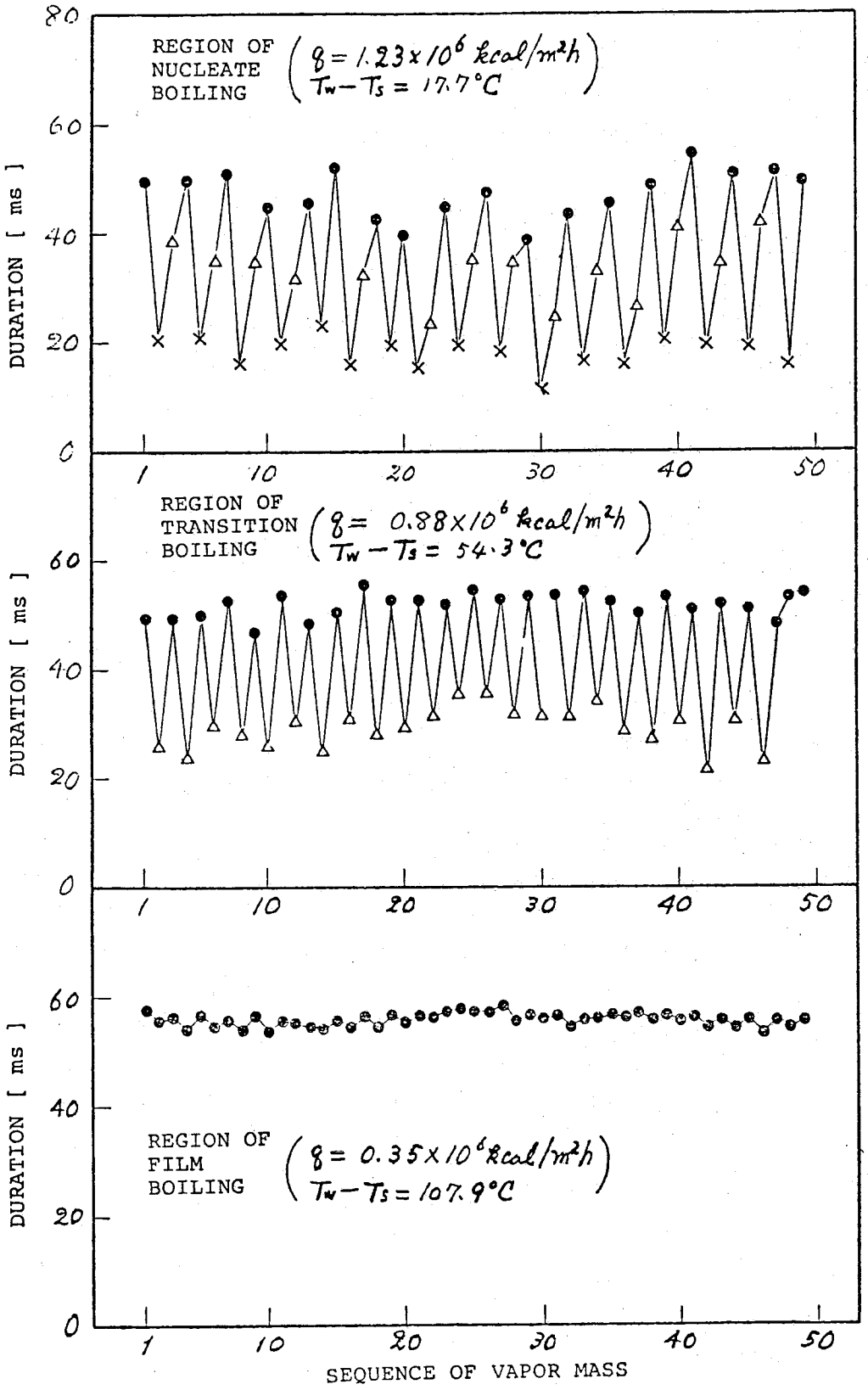

Fig. 6. Successive generation of vapor mass (• A-type vapor mass, X a-type vapor mass, and Δ B-type vapor mass).

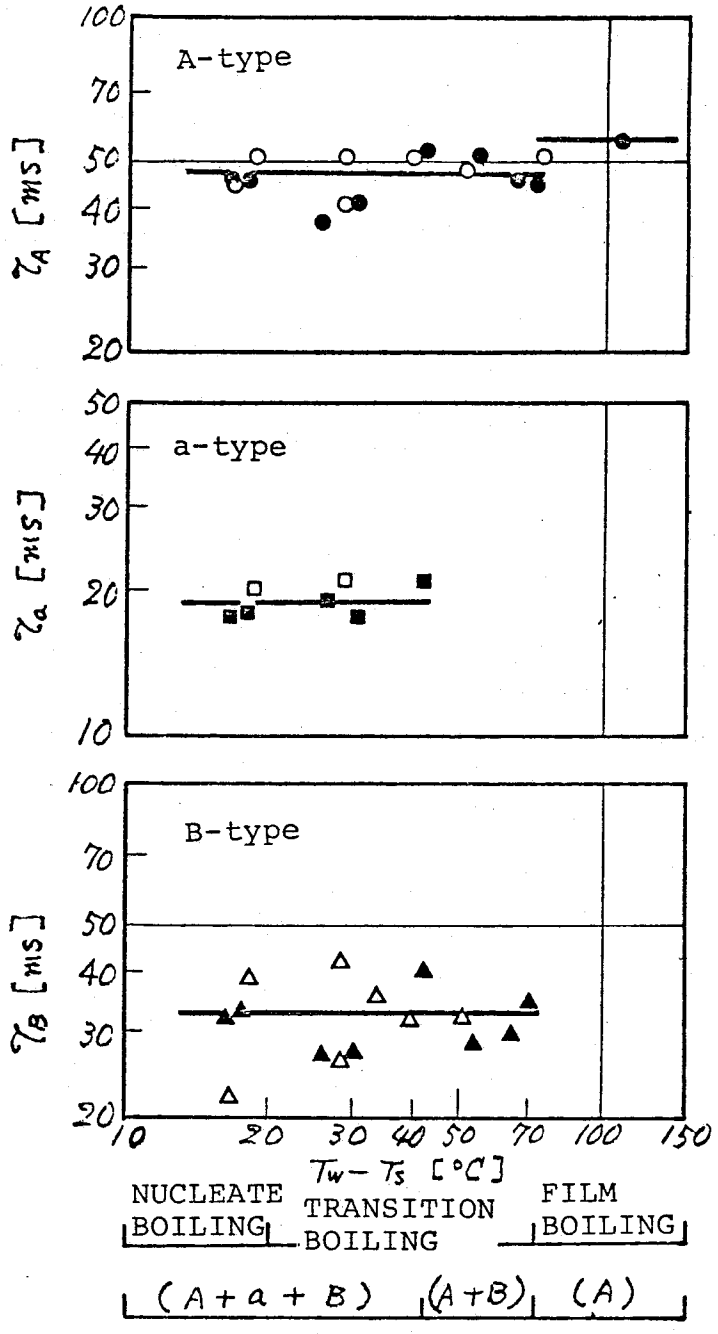

Fig. 7. Duration of vapor mass (measured value).

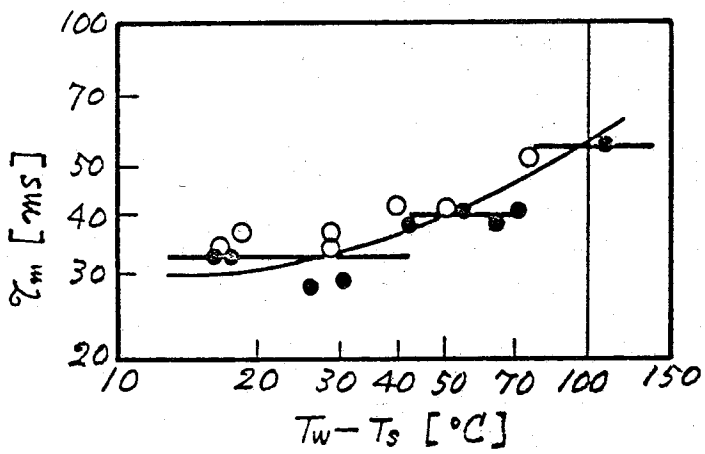

Fig. 8. Overall mean value of duration of vapor mass.

region of transition boiling with the succession type of (A+B), and the lowest figure shows the state in the region of film boiling with the succession type of (A). In the region of film boiling, we can observe the extremely regular generation of vapor mass.

Utilizing the experimental results such as shown in Fig. 6, the duration of vapor mass (the period required from the start of growth to the detachment from the heated surface) τ_A, τ_a and τ_B corresponding to A-, a- and B-type respectively, can be readily obtained at each experimental point in Fig. 2 by computing the average value of the duration for 20 to 40 vapor masses of the same type. The result thus obtained is shown in Fig. 7, where each data has a correspondence to the experimental point in Fig. 2 through T_W-T_S on abscissa as a parameter.

According to Fig. 7, it may be assumed that τ_A, τ_a and τ_B are approximately constant independently of the superheat T_W-T_S, through the regions of nucleate and transition boiling (N.B. the horizontal lines illustrated in Fig. 7 represent τ_A=47 ms, τ_a=19 ms and τ_B=33 ms respectively, and the horizontal line in the region of film boiling represents τ_A=55.8 ms). As for the duration of B-type vapor mass τ_B, a somewhat large scattering of data is observed.

If the types of A, a and B are entirely disregarded, and the overall mean value of duration τ_m is obtained for the vapor masses at each experimental point in Fig. 2, it yields the result shown in Fig. 8. There is an apparent trend in Fig. 8 that τ_m increases with the rise in superheat of the heated surface T_W-T_S, however, this can be calculated from the results of Fig. 7 directly. Namely, the mean value τ_m can be calculated from the result of Fig. 7 as follows:

$$\tau_m=(\tau_A+\tau_a+\tau_B)/3=(47+19+33)/3=33 \text{ ms}$$

for the region of (A+a+B)

and

$$\tau_m=(\tau_A+\tau_B)/2=(47+33)/2=40 \text{ ms}$$

for the region of (A+B)

Among the three horizontal lines illustrated in Fig. 8, the left and the middle line show the values of τ_m aforementioned, while the right line represents τ_A=55.8 ms which has been shown in the region of film boiling in Fig. 7. It is a natural consequence in Fig. 8 that the horizontal lines agree well with the experimental data.

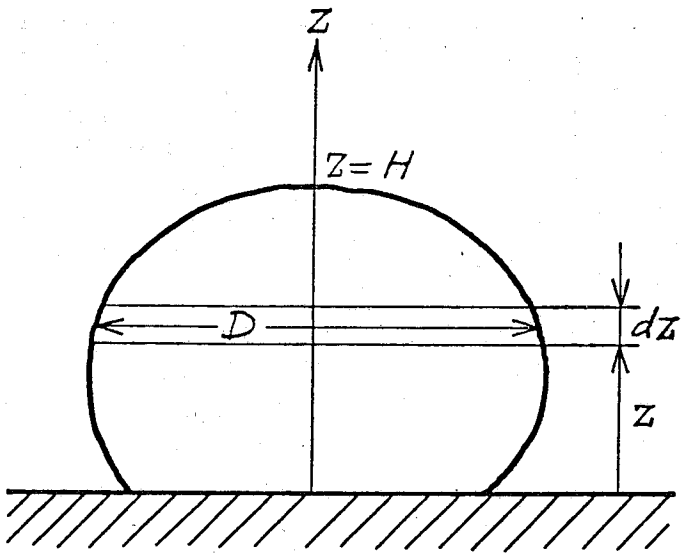

Fig. 9. An image of vapor mass.

3.4 Characteristics of Growing of A-Type Vapor Mass

The A-type vapor mass is fairly independent of the effect of the preceding vapor mass in comparison with the a-type and B-type vapor mass, so that is most adequate for accurate analysis of the mechanical behavior of vapor mass. To make a precise observation of the instantaneous volume of the growing A-type vapor mass on the heated surface, the image of vapor mass recorded on the film is magnified by means of the film analyzer from three to five times the actual size of vapor mass, and the sectional diameter of vapor mass D at the height of z + + dz/2 (see Fig. 9) is measured to determine the instantaneous volume V of the vapor mass by the following integration:

$$V=\frac{1}{a^3}\frac{\pi}{4}\int_0^H D^2dz \qquad (1)$$

where a is the magnifying ratio of the image of vapor mass, H is the height of vapor mass on the heated surface.

In this way, the variation of the volume of a vapor mass with time can be determined, and it reveals the characteristics of growing of vapor mass in each region of boiling. First, in the region of nucleate boiling, the volume of vapor mass V increases at a constant rate with time t as shown in Fig. 10. The reason is that a nucleate boiling liquid layer lying between the vapor mass and the heated surface keeps a constant nucleate boiling in it, which supplies vapor to the vapor mass at a constant rate. On the other hand, an oblique line drawn in Fig. 10 shows the instantaneous volume V determined by the following equation (2) employing the experimental value of heat flux q (= 1.19×10^6 kcal/m^2h) in the present case:

$$V=G_1t, \quad \text{where} \quad G_1=qA/(\rho_V L) \qquad (2)$$

and A is the area of the heated surface, ρ_V the density of vapor, and L the latent heat of evaporation. It may be noticed in Fig. 10 that Eq. (2) agrees with the data obtained from the high-speed photography.

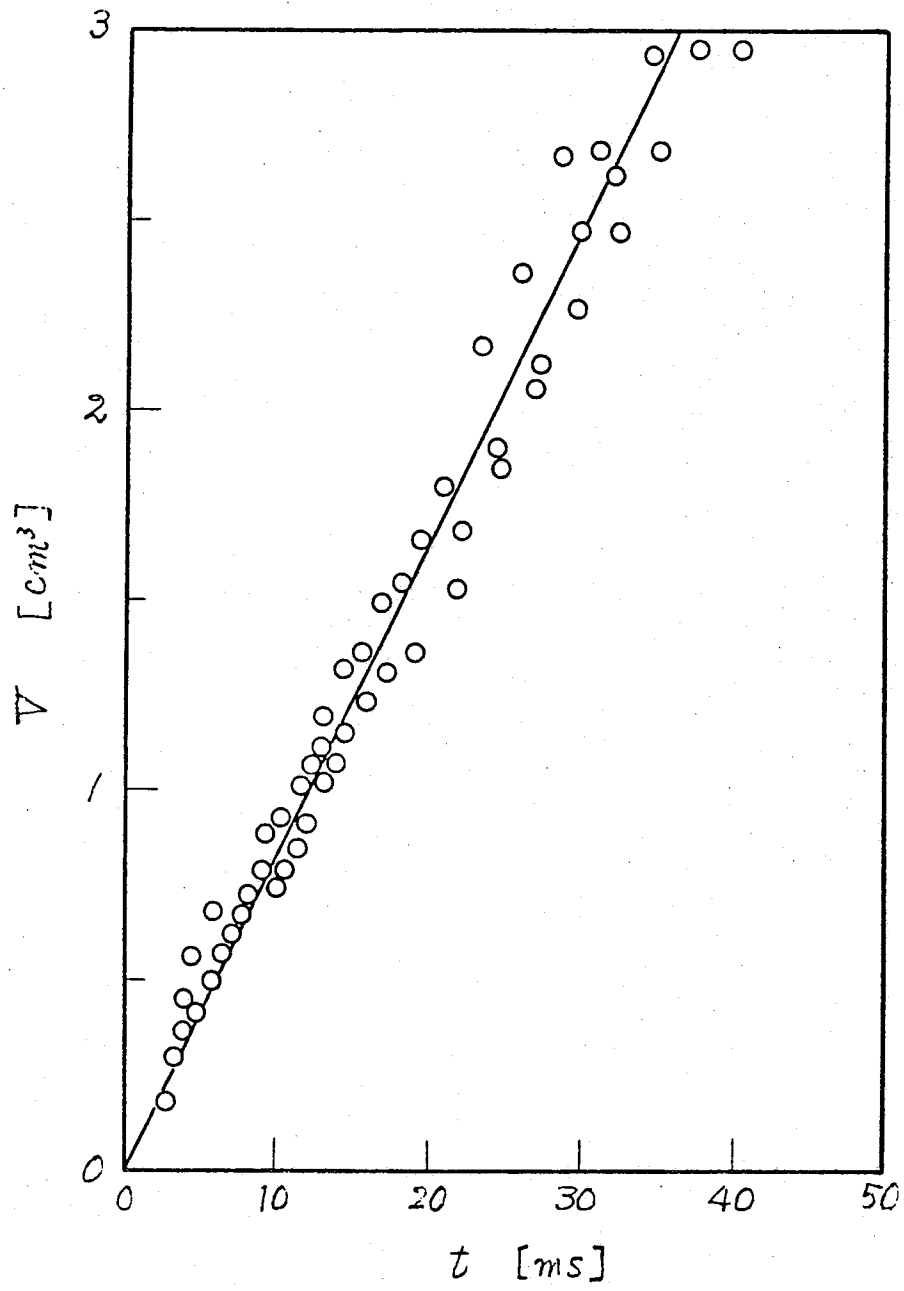

Fig. 10. Variation of the volume of a vapor mass V with time (region of nucleate boiling, $q = 1.19 \times 10^6$ kcal/m^{2}h).

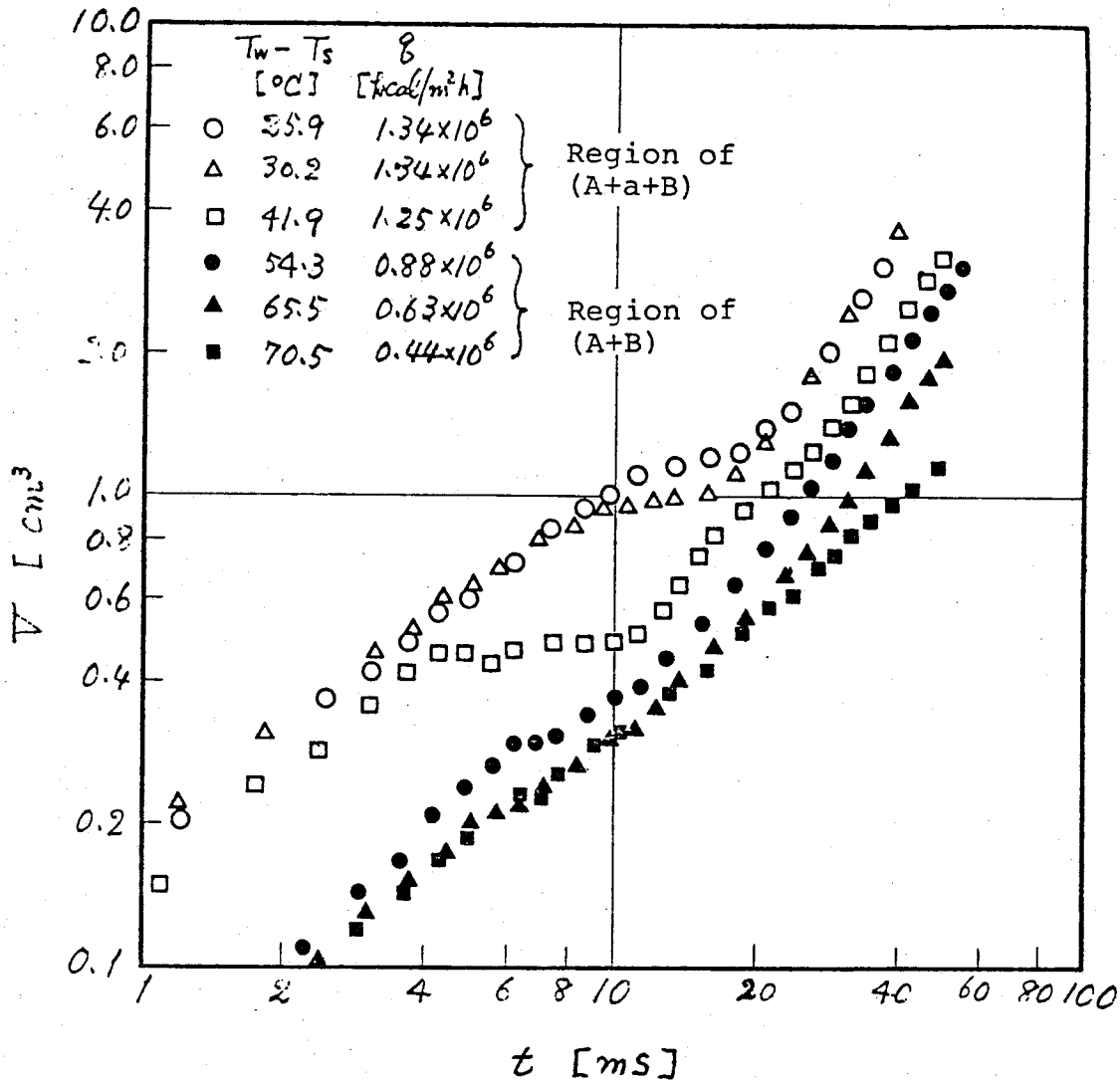

Fig. 11. Variation of the volume of a vapor mass V with time (region of transition boiling).

In the region of transition boiling, the experimental results of V(t) at $T_w - T_s = 25.9°, 30.2°$ and 41.9°C in Fig. 11 (denoted by ○, △, and □) show the remarkable features that the growing of vapor mass is suppressed for a certain period in mid course. The phenomenon of this transient stagnation in the growth of vapor mass has already been reported by the authors [2] and is assumed to have a deep connection with the drying of the heated surface following the waste of the nucleate boiling liquid layer which exists between the vapor mass and the heated surface. It is also presumed that the reopening of the growth of vapor mass after the transient stagnation is caused by the penetration of liquid from the surroundings onto the dried heated surface which generate the nucleate boiling. The region of boiling, where such character of volume variation as aforementioned can be observed, happens to agree roughly with the region of superheat $T_W - T_S$ less than 40°C, that is, the region of (A+a+B).

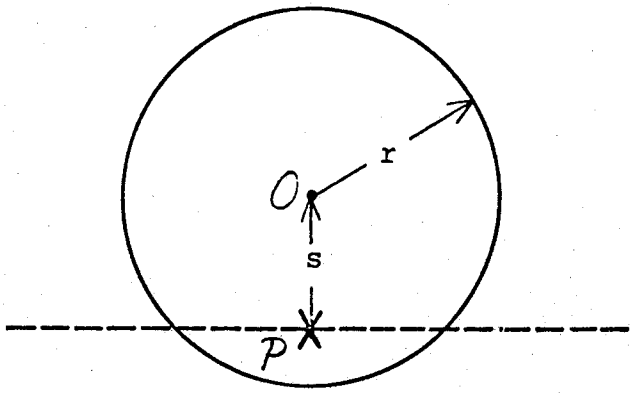

Fig. 12. Increase of volume (increase of r) and ascending motion (increase of s) of a spherical bubble.

With further increase of superheat of heated surface T_W-T_S, it enters the region of (A+B), where the variation of volume with time becomes somewhat monotonic as shown by the experimental data of V(t) at $T_W-T_S = 54.3^\circ$ and $65.5^\circ C$ in Fig. 11 (denoted by ● and ▲). In these cases, it is noted that the first half of the growth period of vapor mass has a growth rate slightly different from that in the latter half of the growth period. Next, the data of V(t) at $T_W-T_S = 70.5^\circ C$ (denoted by ■) shows a very monotonic character, that is, no change of growth rate is found. Anyhow, the region of (A+B) is the region which is strongly characterized by the transition to film boiling.

4. THEORETICAL APPROACH TO THE BEHAVIOR OF VAPOR MASS

A theoretical approach will be attempted to examine the mechanical factors which govern the behavior of growth and detachment of a vapor mass, and A-type vapor mass will be selected for it, since the vapor mass of this type is convenient having a comparatively simple condition. Vapor mass of a-type or B-type, which is affected by the induced flow of the preceding vapor mass, has a condition more complicated than the vapor mass of A-type.

4.1 Fundamental Base For the Analysis

Katto and Kikuchi [6], studying the vapor mass in the high heat flux nucleate boiling of saturated water at atmospheric pressure on the horizontal heated disk surface (standard diameter: 14.5 mm), made the measurement and the theoretical examination on p_V-p_L, where p_V is the vapor pressure inside the vapor mass acting on the heated surface, and p_L is the water pressure in the surroundings at the level of heated surface. According to their study, the pressure difference p_V-p_L is extremely small showing the value from -1.5 to -2.0 mm Aq and at the same time almost independent of heat flux. Therefore, it may be assumed that the growth of a vapor mass is nothing but the natural increase of volume which follows a free rising vapor mass accompanied by the supply of vapor from the heated surface. In other words, the principal motive force governing the behavior of a growing vapor mass is the buoyancy.

On the other hand, Davidson et al. [7,8] indicated that the following simple method of analysis is effective at least for the theoretical estimation of volume of the gas bubble which is produced when the gas is flowing into a liquid through a horizontal orifice or a vertical pipe.

Now, it is assumed that the gas provided from a fixed point source P in the liquid forms a gas bubble as shown in Fig. 12, where the bubble is subjected to not only the increase of volume (the increase of the instantaneous radius r with time), but also the ascending motion (the increase of the instantaneous height s, that is, the vertical distance from P to the center of bubble O). Regarding the ascending motion of bubble, the force of inertia of fluid is assumed to balance the buoyancy of the bubble, and it gives the equation of motion as follows:

$$\frac{d}{dt}\left\{(\xi\rho_L+\rho_V)V\frac{ds}{dt}\right\}=(\rho_L-\rho_V)Vg \qquad (3)$$

where t is the time, ξ the volume of the liquid related to bubble motion (per unit volume of bubble), ρ_L the density of liquid, ρ_V the density of gas, V the instantaneous volume of bubble, and g the acceleration of gravity. As for ξ aforementioned, the analysis of the motion of a solid sphere in fluid well known in hydrodynamics, yields

$$\xi=\frac{1}{2}+\frac{3}{16}\frac{r^3}{b^3} \qquad (4)$$

where r is the radius of sphere, b the distance from the center of sphere to the wall surface placed perpendicularly to the direction of motion. If the sphere is positioned at a sufficiently great distance from the wall surface, b = ∞ so that $\xi=1/2$; but in the present case, since the sphere (vapor mass) is situated in the vicinity of the wall surface (heated surface), b $\fallingdotseq$ r so that $\xi\fallingdotseq 11/16$.

As an example, let us consider a system where gas is discharged at a constant rate into the liquid through a horizontal orifice. In this case, V involved in Eq. (3) is given as follows:

$$V=G_1 t, \text{ where } G_1 = \text{constant} \qquad (5)$$

Therefore, Eq. (3) can be readily integrated, and two integration constants can be determined by the initial conditions [V ds/dt = 0 (zero momentum of fluid) as well as s = 0 at t = 0], so that the variation of height of the center of bubble with time s(t) is determined. On the other hand, the variation of the radius of bubble with time r(t) is readily determined from Eq. (5) and the formula $V\equiv(4/3)\pi r^3$. In the initial stage of growth of bubble, $s(t)<r(t)$ in general, but the ascending motion of bubble is accelerated and finally reaches the following state:

$$s(t)=r(t) \qquad (6)$$

Then, it is assumed that the bubble detaches from the orifice at the moment $t=\tau_d$ which satisfies Eq. (6). This condition of detachment corresponds to the physical situation that the gradually accelerated ascending motion of bubble finally surpasses the increase in size of the bubble held by the discharge of gas. It may be regarded that the detachment of this type occurs naturally as the result of the dynamic flow of fluid, and is entirely different from the type composed from the balance of static forces such as in Fritz's equation of the bubble diameter at departure.

When V included in Eq. (3) is expressed in a more general form as $V = G_m t^m$, where G_m and m are constant respectively, the situation that $s(t)<r(t)$ for $0\leqq t$ is established if $0<m<6$, and the time $t=\tau_d$ satisfying Eq. (6) comes into existence. At the limit of m = 6, according as the value of G_m is

$$\frac{1}{14}\frac{g(\rho_L-\rho_V)}{\xi\rho_L+\rho_V}\leqq\left(\frac{3}{4\pi}G_m\right)^{1/3}$$

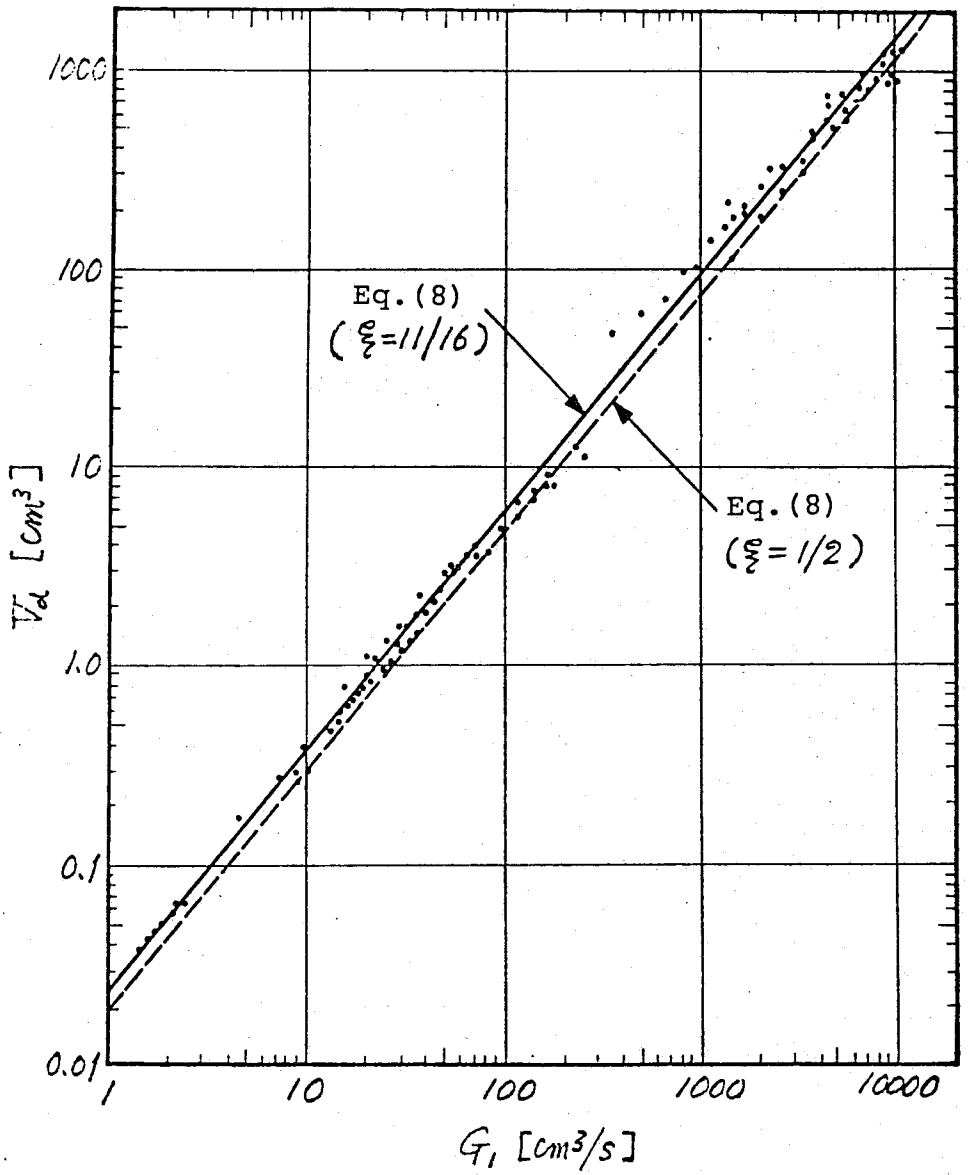

Fig. 13. Comparison of Eq. (8) with experimental results of bubble volume at detachment for constant discharge rate of gas.

or

$$\frac{1}{14}\frac{g(\rho_L-\rho_V)}{\xi\rho_L+\rho_V} > \left(\frac{3}{4\pi}G_m\right)^{1/3}$$

there are two singular states: in the former case, $s(t)<r(t)$ for $0\leqq t<\infty$, that is, the bubble continues to grow unlimitedly without detaching from the surface; and in the latter case, $s(t)>r(t)$ for $0\leqq t<\infty$, that is, no bubble can be formed on the surface.

For the case of Eq. (5) with respect to V, that is, the condition of constant discharge, the duration of bubble τ_d and the volume of bubble at detachment $V_d=G_1\tau_d$ [putting $t=\tau_d$ in Eq. (5)] are determined by the aforementioned method as follows:

$$\tau_d=\left(\frac{3}{4\pi}\right)^{1/5}\left[\frac{4(\xi\rho_L+\rho_V)}{g(\rho_L-\rho_V)}\right]^{3/5}G_1^{1/5} \qquad (7)$$

$$V_d=\left(\frac{3}{4\pi}\right)^{1/5}\left[\frac{4(\xi\rho_L+\rho_V)}{g(\rho_L-\rho_V)}\right]^{3/5}G_1^{6/5} \qquad (8)$$

Equation (8) is shown in Fig. 13 in comparison with the experimental data [7-11] obtained under the condition of constant discharge (combination of air and water in most cases, and diameter of discharge port extending over 0.14 to 25.4 mm). It may be noticed that the analysis agrees fairly well with experimental data over the wide range of experiment ($0.037\ \mathrm{cm}^3< V_d <1300\ \mathrm{cm}^3$, corresponding to the range of diameter of bubble $4.1\ \mathrm{mm}<D_d<135\ \mathrm{mm}$), and the agreement is nearly satisfactory if ξ is assumed to be 11/16 (N.B. the difference between $\xi=1/2$ and $\xi=11/16$ is not so great in Fig. 13).

It may be also noticed in Fig. 13 that the surface tension, which must act on the edge of the discharge port, can be entirely ignored if G_1 is large and the effect of buoyancy is sufficiently great. No term of resistance of fluid is involved in Eq. (3), but it is reasonable in the present case, because we are now dealing with the state of fluid motion such as shown in frames 2 to 3 in Fig. 3, or frame 2 in Fig. 4, and besides we are dealing with a gas bubble with no remarkable shearing force on its surface unlike the case of a solid sphere. Moreover, the application of Eqs. (7) and (8) would lead to the results such as mentioned in Appendixes 1 and 2 for the diameter at detachment and the emission frequency of the vapor mass which appears in the region of high heat flux nucleate boiling, where the situation of Eq. (2), and accordingly that of Eq. (5), is satisfied.

Synthesizing the matters aforementioned, it might be concluded that the theoretical analysis based on Eqs. (3) and (6) is a fairly reliable one.

4.2 Analysis of Duration of A-Type Vapor Mass

The variations of instantaneous volume V of the A-type vapor mass with time observed in the experiment have already been shown in Figs. 10, and 11, and they can be classified approximately to the three types of variation illustrated in Fig. 14.

First, let us consider the case of Fig. 14(c), where the variation of V takes place according to $V = G_n t^n$ in the initial stage $0<t<t_c$, then V = constant in $t_c<t<t_l$, and finally $V = G_m t^m$ in $t_l<t$. The integration of Eq. (3) can be made under the condition that the position of center of vapor mass s as well as the ascending velocity ds/dt is continuously connected at each point of $t = t_c$ and $t = t_l$; and as the result, it yields s(t) in the stage of $t_l<t$ as follows:

$$s(t)=\left\{\frac{1}{2(1+m)}t^2+\frac{m}{1+m}t_l^2\frac{[(t/t_l)^{1-m}-(1+m)/2]}{1-m}\right\}$$
$$-\frac{n}{1+n}t_c t_l\frac{[(t/t_l)^{1-m}-m]}{1-m}+\frac{n}{2(1+n)}t_c^2\left\}\frac{g(\rho_L-\rho_V)}{\xi\rho_L+\rho_V}\right. \qquad \text{for } m\neq 1$$

$$s(t)=\left\{\frac{1}{4}t^2+\frac{1}{2}t_i^2\left(\log\frac{t}{t_i}+\frac{1}{2}\right)-\frac{n}{1+n}t_c t_i\left(\log\frac{t}{t_i}+1\right)+\frac{n}{2(1+n)}t_c^2\right\}\frac{g(\rho_L-\rho_V)}{\xi\rho_L+\rho_V}$$
for m = 1 (9)

On the other hand, the radius of vapor mass r(t) in the stage of $t_i<t$ can be determined from $V=G_m t^m$ and $V\equiv(4/3)\pi r^3$ as follows:

$$r(t)=\left(\frac{3}{4\pi}\right)^{1/3}G_m^{1/3}t^{m/3} \qquad (10)$$

Accordingly, the duration of vapor mass τ_A can be determined by applying the condition of Eq. (6) to Eqs. (9) and (10).

Next, s(t) in the stage of $t>t_c$ in the case of Fig. 14(b) can be readily determined by putting $t_c = t_l$ in Eq. (9), and then s(t) in the case of Fig. 14(a) can be obtained by putting $t_c = 0$ in the case of Fig. 14(b).

The analytical values of τ_A thus obtained through Eqs. (9) and (10) [where $\xi=11/16$] corresponding to the variation of instantaneous volume of A-type vapor mass are shown in Fig. 15. On the other hand, the experimental values of τ_A have been shown in the uppermost figure in Fig. 7. From the comparison between Fig. 7 and Fig. 15, the following remarks are derived:

(i) In the region of nucleate boiling (T_W-T_S < < 20°C), where the relation $V = G_1 t$ is established, the analytical value of τ_A in Fig. 15 almost agrees with the experimental value of τ_A in Fig. 7.

(ii) Moving to the region of transition boiling from that of nucleate boiling, it brings about the change in the behavior of growing of vapor mass, which may be mainly due to the occurrence of drying of heated surface in mid course. However, in view of the fact that τ_A shows nearly the same value in Figs. 7 and 15, it seems likely that there is no remarkable change in the mechanical situation governing the behavior of vapor mass, and the

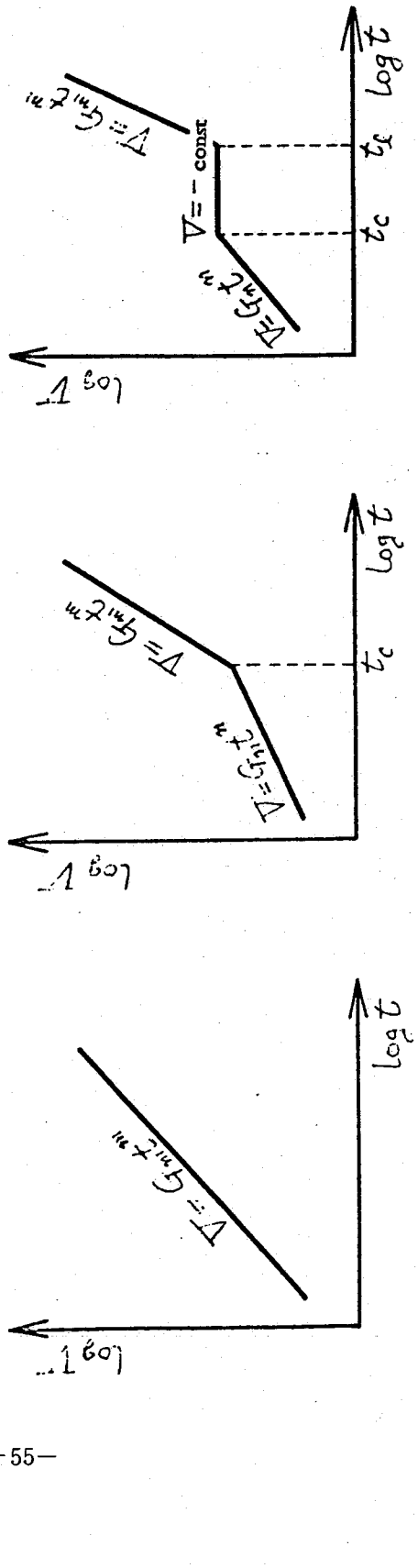

Fig. 14. Typical forms for the variation of volume of vapor mass with time.

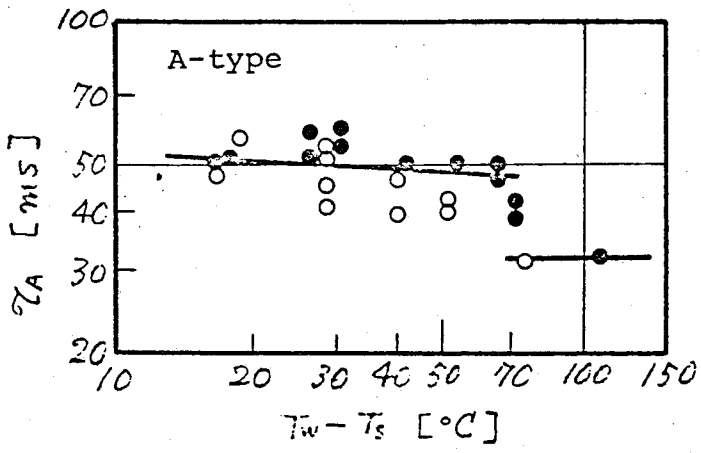

Fig. 15. Duration of vapor mass (analytical value).

assumed mechanical situation, on which Eq. (3) stands, is as yet applicable to the region of transition boiling except the vicinity of film boiling. This presumption differs, however, from that of Honda and Nishikawa, so that further investigations might be necessary to further clarify the matter.

(iii) Entering the region of film boiling, it brings about obvious deviation from the mechanical situation assumed by Eq. (3), leading to the state of emission of bubbles from a wavy interface which is peculiar to the film boiling (N.B. extremely regular emission of bubble in the lowest figure of Fig. 6).

5. CONCLUSION

(1) To study the mechanical situation governing the behavior of growth and detachment of vapor mass through the region from nucleate to film boiling, high-speed photography has been employed observing the vapor mass in pool boiling of saturated water at atmospheric pressure on a horizontal 10 mm-dia. heated disk surface. From the observation aforementioned, vapor mass is roughly classified into three types: A-type vapor mass, which is not affected by the preceding vapor mass, a-type vapor mass, which appears immediately following the A-type vapor mass, and B-type vapor mass, which has intermediate features between A-type and a-type.

(2) The durations of vapor mass τ_A, τ_a and τ_B for A-, a- and B-type vapor mass respectively have been measured from photographic recording, giving the result that $\tau_A = 47$ ms, $\tau_a = 19$ ms and $\tau_B = 33$ ms through the nucleate and the transition boiling region.

(3) The variation of volume with time V(t) of A-type vapor mass has been measured from the photographic recording and it is found that V(t) has special features in each region of nucleate and film boiling.

(4) The theoretical value of τ_A, which should correspond to the V(t) aforementioned, has been calculated on the assumptuon that the vapor mass on the heated surface is subjected to the ascending motion with the balance between the buoyancy of vapor mass and the force of inertia of fluid, and that the vapor mass detaches from the heated

surface when the ascending motion surpasses the growing of volume of vapor mass. From the comparison of the theoretical value of τ_A thus obtained with the measured value of τ_A, it is concluded that there is no remarkable change in the mechanical situation governing the behavior of vapor mass through the region of nucleate and transition boiling. In the region of film boiling, however, distinct change of the mechanical situation is observed.

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APPENDIX 1: f and D_d of vapor mass in the region of nucleate boiling.

Ivey [12] made a study to correlate the existing data regarding the bubble frequency f, and the departure diameter D_d in nucleate boiling. He assumed three types of regions, and obtained the following empirical equation for the hydrodynamic region:

$$fD_d^{1/2}=0.90\,g^{1/2} \tag{A1}$$

This region, with high heat flux and comparatively large bubbles, is regarded to correspond to the region generating the vapor mass dealt with in the present paper; and Eqs. (7) and (8), where

$$\left(f=\frac{1}{\tau_d},\quad V_d=\frac{1}{6}\pi D_d^3\right),$$

immediately yield the following equation for $\rho_V \ll \rho_L$ and $\xi=11/16$:

$$fD_d^{1/2}=0.853\,g^{1/2}. \tag{A2}$$

This theoretical equation is very close to the empirical equation (A1) obtained by Ivey.

APPENDIX 2: fD_d in the region of nucleate boiling.

Ivey [12] compared the measured value of fD_d with the bubble rise velocity u_b, and gave the conclusion that the value of fD_d is fairly close to that of u_b in the hydrodynamic region. Meanwhile, the u_b used by Ivey is the experimental data of Rosenberg and others [13-15], for which we can give an approximate expression $D_d g/u_b^2 = 1.87$; then the following empirical relationship is readily obtained (N.B. the region of bubble diameter greater than 11 mm in Fig. 7. of Ivey's paper):

$$fD_d \fallingdotseq 1.2\,u_b \tag{A3}$$

On the other hand, Eqs. (7) and (8), and $D_d g/u_b^2 = 1.87$ yield the following equation for $\rho_V \ll \rho_L$ and $\xi=11/16$:

$$fD_d=1.17\,u_b \tag{A4}$$

This theoretical equation is also very close to the empirical equation (A3).

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EXPERIMENTAL STUDY OF NUCLEATE POOL BOILING IN CASE OF MAKING INTERFERENCE-PLATE APPROACH TO THE HEATING SURFACE

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The present study was made to obtain information for normal nucleate boiling at high heat fluxes, but it had as a key feature the use of an interference plate that was a solid surface opposed to a flat heating surface with a purpose to give artificial restrictions to the fluid movement near the heating surface. Those matters were particularly studied such as effects of vapor masses on the fluid zone on the heating surface, thickness of the fluid zone, radius of vapor stems, thin-liquid-film vaporization, and a possibility of a boiling combining discrete vapor stems and liquid film vaporization.

Die gegenwärtige Forschungsarbeit wurde durchgeführt, um den gewöhnlichen Kernsiedevorgang bei hohem Wärmefluß zu erklären und beruhte grundsätzlich auf der Methode, einer ebenen, geheizten Fläche eine feste Interferenzplatte gegenüberzustellen, um künstlich die Flüssigkeitsbewegung in der Nahe der heißen Fläche zu verhindern. Insbesondere wurde Folgendes untersucht: Einfluß der Dampfmasse uber die Flüssigkeitsschicht auf der heißen Fläche, Dicke der Flüssigkeitsschicht, Radius der Dampfsäule, Verdampfung in einem dünnen Flüssigkeitsfilm sowie auch die Möglichkeit eines Siedevorgangs, indem man mehrere Dampfsäulen mit der Verdampfung eines dunnen Flüssigkeitsfilmes kombiniert.

1. INTRODUCTION

A large number of investigations have already been carried out on the nucleate boiling, through which the mechanism of its heat transfer has been revealed to a considerable extent. However, not a few ambiguities are still left particularly in the region of high heat fluxes and it is necessary to accumulate more information for this region. The present paper describes the results of an experimental study of saturated nucleate pool boiling attempted with an aim to attack the phenomenon from a somewhat different side than usual.

The nucleate pool boiling differs of course from the normal natural convection in the occurence of nucleation of bubbles on the heating surface. As the heat flux increases, population density of active nucleation sites rises with a trend to generate a zone very close to the heating surface, where the behaviors of fluid are rather spontaneous and comparatively independent from outside. Clarifying what phenomena arise in the spontaneous zone with what mechanism is important for the study of heat transfer. However, formation of irregular masses of vapor above the heating surface makes the appearance of phenomena complicated and obscure.

As a means to investigate the zone important for the heat transfer, a special boiling system is intentionally adopted in the present study consisting of a flat horizontal heating surface of 10 mm in diameter facing upward and a horizontal disk of the same diameter which can be held above the former at various distances. Artificial restrictions and modifications induced by the disk to the movement of liquid as well as vapor over the heating surface are utilized to provide additional information on the behaviors of fluid near the heating surface and their effects on the heat transfer. In addition, though in conditions quite different from normal nucleate boiling, the spreading of a thin flat bubble in a very confined space under a disk surface is studied to get referential data for the heat transfer mechanism of thin-liquid-film vaporization, because this mechanism is considered to be one of the factors important for the heat transfer at high heat fluxes. The heating surface area exposed to the vapor is here so expanded as to permit ready observations of the phenomena on that area.

Being used in such a way, the disk is called interference plate in the present paper. For the purposes not only should the useless staying or accumulation of vapor under the interference plate be avoided as far as possible but also the temperature of the heating surface be kept uniform. In addition, it is desirable in order to simplify the features of boiling that one vapor mass of a simple shape is generated successively above the heating surface. These are the reasons for the adoptation of a comparatively small area of the heating surface as well as of the interference plate.

The present study will give incidentally some of fundamental knowledges as for the boiling in a thin gap, but it is not the main purpose of the study. Phenomena which may arise when the interference plate is much larger in area is also beyond the scope of the present study.

2. DESCRIPTION OF APPARATUS

The boiler is illustrated schematically in Fig. 1. A horizontal, flat, copper heating surface of 10 mm in diameter was set up facing upward at the bottom of a pool of pure water at atmospheric pressure. A fine gap around the edge of the surface was filled with an adequate binding agent so that boiling took place in the 10 mm-diameter area.

Heat was supplied from electric heating elements which were mounted in the slots at the lower cylindrical part of copper block and were energized with single-phase a.c. current. Thermocouples located at five precisely known points on the geometric axis of the conical conducting part of copper block were used to calculate the boiling heat flux by conduction theory and the surface temperature by extrapolation. In order to minimize the heat losses from the part important for measurements, upper half of the furnace was designed to be surrounded by hot water. Careful calibrations and preliminary experiments were made, which assured the sufficient accuracy of the

measuring method as well as the uniformity of the heating surface temperature. The latter was also confirmed with the uniform distribution of nucleation sites over the heating surface. About 14 % of power input was found to escape but it took place mainly from the lower cylindrical part of copper block without any appreciable defect for the experiment.

Characteristics of nucleate boiling on the present heating surface were much the same as usual. For example, as shown in Fig.2, transition from the region of isolated bubbles to the region of interference appears in the range of heat flux of nearly 1.6 to 2.0×10^5 kcal/m^{2}hr agreeing with many past experiences in saturated water boiling at atmospheric pressure, where the terminologies are borrowed from Zuber(1). As can be expected from the size of the heating surface of 10 mm in diameter and the instability theory of a horizontal liquid-vapor interface (critical wave length given by the two dimensional theory: 15.7 mm for water of 100°C), a single vapor mass such as shown in Fig.2 was generated successively above the heating surface throughout the region of interference.

Two types of interference plate made of bakelite of low thermal conductivity, whose dimensions are shown in Fig.3, were used in the initial stage of the study. Type B with a cylindrical hollow of 8 mm in diameter and 5 mm deep is a special one prepared to see the effects of a lump of vapor constantly held above the heating surface on heat transfer.

Next, optical assemblies combining an interference plate and a device to observe the behaviors of fluid over the heating surface are shown in Fig.4. Type A consists of a glass interference plate of conical form, two glass prisms and a water-proof metalic box so as to permit the light entering through the left window to reach the space between the interference plate and the heating surface, and the phenomena in the space can be observed through the right window. Type B is the same as before except having a thin hollow cylinder made of bakelite and stuck at the bottom to give the same shape as that in Fig. 3.

Either an interference plate or an optical assembly was fixed, as shown in Fig. 1, at the lower end of a positioning bar movable vertically with a system of rack and pinion. A dialgauge was used to measure distances over which the bar was moved, thereby the space distance between the interference plate and the heating surface was determined minutely.

3. VARIATIONS OF HEATING SURFACE TEMPERATURE

Experiments were made on the condition of saturation boiling over the range of high heat fluxes including even the region of isolated bubbles with high population density, where the boiling action dominates the heat transfer and natural convection is no longer important. Keeping a heat flux q constant, the heating surface temperature T_w was measured at various space distances s between the interference plate and the heating surface.

A typical result obtained with a simple interference plate of type A is shown in Fig.5, where T_s is the saturation temperature and is constant. Heating surface temperature T_w is invariable while the interference plate is comparatively apart from the heating surface, but it falls as reducing s and thereafter rises. The higher the heat flux, the earlier the boiling crisis appears, and the region where temperature falls and then rises decreases and finally disappears. At very high heat fluxes, as s is reduced, T_w has a tendency to rise gradually before its violent variation appears, but if is not so significant that T_w may be regarded approximately invariable.

As described later, the temperature falling region exhibits aspects as for the behaviors of vapor bubble quite different from normal nucleate boiling due to the effects of interference plate. The space distance s_f where the temperature T_w begins to fall changes with heat flux as shown in Fig.5, and whole experimental results are plotted by black marks in Fig.6 as a function of heat flux. In the figure is also shown a curve representing variation of the average diameter of breakoff bubbles or vapor stems with heat flux in normal nucleate boiling, which has been obtained by Gaertner and Westwater(2) in experiments with water at atmospheric pressure and a flat horizontal copper heating surface. It is of interest to point out that a comparatively good similarity is found between the two independent results aforementioned.

Fig.7 shows a typical result obtained with the interference plate with a hollow (type B) in saturation boiling. The magnitude of T_w on extreme right in the figure where the effects of interference plate is negligible, is compared with that of Fig.5 to be found somewhat higher for the same heat flux, but it depends only on accidental difference in the finishing states of the two heating surfaces. In the present case, T_w is almost kept at the same value as in normal boiling until a very short length of s, due to the existence of a lump of vapor instead of a solid surface above the heating surface.

4. BOILING CRISIS IN SPECIAL SYSTEM USED

Vertical broken lines in Figs.6 and 7 indicate the appearance of boiling crisis where T_w jumps abruptly. Although the main purpose of the present study is not on the boiling crisis, a simple discussion will be described below. Now let us make a quite simple assumption that through the circumferential opening of a cylindrical space between an interference plate and a heating surface, liquid is flowing in at the lower section while vapor is flowing out at the upper section in a continuous and uniform manner. Then assuming that pressure inside the space is equal to outside pressure and that whole transfered heat is consumed to generate vapor on the heating surface, conservation of mass, of momentum and of energy in the space yield the relative velocity in the counterflow of liquid and vapor at the circumferential opening as:

$$\frac{q}{\rho_v h_{fg}} \left(1 + \frac{\rho_v}{\rho_l} \right) \frac{r_o}{s} \qquad (1)$$

where q is heat flux density, ρ_v density of vapor, ρ_l density of liquid, and h_{fg} latent heat of evaporation respectively.

On the other hand, it is well known that a horizontal interface between vapor in the upper side and liquid in the lower side becomes unstable when the two fluids have a relative velocity higher than

$$\sqrt{2 \left(1 + \frac{\rho_v}{\rho_l} \right)} \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right]^{\frac{1}{4}} \qquad (2)$$

where σ is surface tension of liquid. Equating quantity (1) to quantity (2) and replacing q by q_c provide

$$\frac{q_c}{\rho_v h_{fg}} \sqrt{\left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right]^{\frac{1}{4}}} = \sqrt{\frac{2}{1 + (\rho_v / \rho_l)}} \frac{s}{r_o} \qquad (3)$$

where q_c is the critical heat flux which generates boiling crisis in a system of the space distance of s.

Comparisons with experimental results are made in Fig.8 where the oblique broken line expresses equa-

tion (3). Despite the extreme simplicity of assumptions, anyhow the theory shows a rough agreement with experimental data if s/r_0 is small. With increasing s/r_0, however, experimental data deviate from the theory and asymptotically approach the horizontal broken line which represents the results obtained when the interference plate is faraway from the heating surface. It should be noticed that even at a space distance as short as s = 1.4 mm, which corresponds to $\sqrt{2/(1+\rho_v/\rho_l)}\cdot(s/r_0) = 0.4$, critical heat flux does not differ so much from that in normal boiling. This may suggest something to the mechanism of boiling crisis in normal boiling.

Reasons why types A and B of interference plate show somewhat different features in Fig.8 are not clear. Behaviors of fluid inside the space between the interference plate and the heating surface may have some effects on the boiling crisis.

5. OPTICAL OBSERVATIONS OF FLUID BEHAVIORS

Using optical assemblies shown in Fig.4, observations of fluid behaviors in the space between the heating surface and the interference plate were made. In case of the assembly of type B, it was confirmed that the hollow at the bottom of assembly was filled with vapor in such a manner as shown in Fig.4. However, many droplets stuck on the glass surface at the top of the hollow and they interrupted sufficient observations. Therefore, only the experimental results obtained with the assembly of type A will be described below.

Corresponding to Fig.5, typical behaviors of fluid are shown in Fig.9 with instantaneous photographs (shutter speed: 1/1000 sec.) taken of the fluid which is changing its pose over the heating surface constantly with time under conditions indicated in the figure. It can be noticed from the photographs that not only are there three quite different regimes of fluid behaviors but they correspond to the characteristics of T_w vs. s curve which have been described in section 3.

In the first region where T_w is invariable, it is clear that the normal nucleate boiling completely dominates the heat transfer, and the generation of bubbles or vapor stems at isolated active sites indeed characterizes the phenomena even though any form of coalescence of vapor succeeds. When the heat flux is high, the interference plate penetrates into such a large mass of vapor as shown in Fig.2 and the state at the bottom surface of the vapor mass is observed. Transfiguration of the vapor mass due to the interference plate has no appreciable effect on the heat transfer. Since there is no depression of small bubbles on the heating surface, this region might be called the region of undepressed bubbles.

Boiling in the second region where T_w falls as s is reduced may be referred to as a sort of nucleation boiling, but much fewer bubbles than in the first region, not resulting from coalescence of small bubbles, spread on the heating surface to be finally pushed out by penetrating liquid. As the interference plate is very near the heating surface in the present case, volume of the space is considerably limited. Consequently, if one bubble begins to grow, it rapidly spreads on the heating surface and interrupts the generation of other bubbles in the neighbourhood. This is presumably the reason why the number of coexisting bubbles is much less than in the first region though it increases as heat flux is increased. Since bubbles are depressed here with interference plate, this region is called the region of depressed bubbles. It is very interesting that the area of heating surface exposed to vapor is wet with a thin liquid film and some part of vaporization takes place from it. White areas in each photograph of Fig.9 belonging to the present regime are the places where the liquid film has already been dried up.

In the third region where T_w rises as s is reduced, the space over the heating surface is so thin that the rapid growth of bubbles is no longer permitted and hence there is not any type of nucleation of bubbles. Instead, greater part of the space is constantly occupied by continuous vapor, and the area of heating surface exposed to the vapor is dry. Liquid penetrates from the circumference to the dry area and then the vaporization takes place from the narrow front surface of liquid to raise the gradual fluctuation of liquid boundary. Occasionally the penetration of liquid is wide and far-reaching and it leaves behind many small islands of droplets as shown in Fig.9. This region might be called the region of oppressed evaporation.

6. PERSPECTIVE TO NORMAL HIGH FLUX NUCLEATE BOILING

Summarizing the experimental results described so far in sections 3 to 5, it may be possible to derive the following conclusions as for normal nucleate pool boiling at high heat fluxes.

(a) Vapor masses generated above a flat heating surface at high heat fluxes as shown in Fig.2 are regarded as only the results of accumulation of vapor generated steadily at isolated active sites on the heating surface. Perhaps it will not be useless to note here the following additional information. According to the authors' unpublished study measuring the growth of such a vapor mass as shown in Fig.2 with the high-speed photography, the rate of volume increase is kept at constant throughout the whole growth period of a vapor mass.

(b) Vapor masses aforementioned hardly have effects on the behaviors of fluid in the vicinity of the heating surface, which is considered to depend on the low density and the low viscosity of vapor as well as on the self-government in nucleation zone. Even when a vapor mass artificially made through accumulating vapor is constantly held near the heating surface, it has little effects on the heat transfer.

(c) It should be noticed in case of Fig.7 that removal of vapor as well as supply of liquid through a narrow circumferential opening have little effects on heat transfer. This suggests that in normal boiling, there is no difficulty for liquid to spread over the heating surface and underneath the vapor masses.

(d) The data of Gaertner and Westwater(2) as for the average diameter of breakoff bubbles or vapor stems on the flat heating surface, were obtained in an indirect way utilizing the plating so that there may be some questions as to their applicability. However, the result of Fig.6 seems to be one of the points which support the fact that the average diameter of breakoff bubbles or vapor stems becomes smaller with increasing heat flux in the region of interference.

(e) Behaviors of fluid inside the zone which stands on the heating surface with a thickness in the order of the diameter of breakoff bubbles or vapor stems are self-acting and are hardly affected by bulk fluid as far as a strong modification is not imposed there. Phenomena important to the nucleate heat transfer arise at least inside the zone.

(f) Conclusions of items (d) and (e) are in the same direction as Gaertner's observations described in the discussion attached to his paper(3). He made measurements from the side-view photographs of water boiling on a flat copper surface at atmospheric

pressure to reach the results that both the diameter and the length of the vapor stems under the vapor masses decreased by increasing the heat flux, and the length to diameter ratio remained constant at 0.6.

(g) There is a possibility that the generation of boiling crisis strongly depends on the fluid behaviors (perhaps hydrodynamical instability) inside the zone between the vapor masses and the heating surface.

7. FLAT BUBBLE SPREADING AND EXTENSION OF DRY AREA

Being related with the thin-liquid-film vaporization mechanism in nucleate boiling adovocated by Moore and Mesler(4) and intensely emphasized by Torikai(5) who observed boiling from below through a glass heating surface coated with a transparent electrical-conducting material, the spreading of flat bubble in the region of depressed bubbles is of interest since the heating surface exposed to vapor is wet with a thin liquid film as previously described.

Making use of the high-speed photography in the present study, the right-hand side of the optical assembly shown in Fig.4 was put in the camera field of view, thereby not only a horizontal view of the heating surface through the interference plate and the window but also a side view of the interference plate and its surroundings were caught simultaneously on a film. Fig. 10 is an example of results obtained in such a way, showing successive states at intervals of 1.53 msec under the conditions of $q = 1.6 \times 10^5$ kcal/m^{2}hr, $s = 0.4$ mm and $T_w - T_s = 3.0$°C. After a long waiting period, though its greater part is omitted in Fig. 10, a very rapid spreading of a vapor bubble appears in the horizontal views of the heating surface in frames 6 to 8 starting from the center. Then vapor occupies the space over the heating surface for a long time until liquid begins to penetrate into the space from the right-hand side in frame 24. The heating surface exposed to vapor is wet with a thin liquid film and the vaporization from the film is confirmed by gradual spreading of a white circular dried-up area over the heating surface as well as by flowing out of vapor through the circumferential opening of the space.

The spreading of bubble did not always start from the center of the heating surface, but not a few such examples convenient for analysis as shown in Fig. 10 could be collected from a picture film, fortunately, to give Fig. 11 for variation of bubble radius with time: r(t). In spite of a result including many independent spreadings, a surprising regularity exists in Fig. 11 that $r(t) \propto t^{0.51}$ in the early stage of $r < 1.5$ mm and thereafter the index of t increases to reach finally $r(t) \propto t^2$. The velocity of bubble boundary $v_s = dr/dt$ as well as the outflow velocity of liquid at the circumferential opening $v_0 = (r/r_0)dr/dt$ can be calculated from Fig. 11 and are shown in Fig. 12. Since specific volume of vapor and of liquid differ from each other by as much as 1600 times under the present conditions, it is a very close approximation that there is no change of volume in the liquid which is pushed out by the generated vapor. Reynolds number of liquid flow at the front of bubble boundary, defined as $Re_s = 2sv_s/\nu_l$ where s is a space distance between parallel surfaces and ν_l a kinematic viscosity of liquid, is also shown in Fig. 12. Vapor pressure inside a bubble is higher than saturation pressure in order to balance with acceleration and flow friction of liquid, but analyses confirm that it is not so great as to be appreciable in the present conditions.

Similar analysis of a picture film was also made as for the spreading of dried-up area on the heating surface which is a phenomenon much slower than the spreading of bubbles. The results are simple and can be expressed by an empirical formula:

$$t = Cr, \quad C = \frac{1}{270} \text{ hr/m} \qquad (0 < r < r_0) \qquad (4)$$

where t (unit: hr) is time measured from the start of bubble spreading until the drying up of area of r (unit: m) in radius. The liquid film is so thin and the bubble spreads over the heating surface so rapidly that equation (4) can be regarded approximately to give time between the creation of liquid film and the drying-up of that at the position apart from the center of the heating surface with a distance of r. Equation (4) will be used later in an analysis for determining the distribution of liquid film thickness.

8. DISCUSSION ON FLAT BUBBLE SPREADING

As well known, observed moving velocities of the surface of growing bubbles decrease with time in normal boiling. In case of Fig. 12, however, not only is the velocity of bubble boundary v_s nearly constant in the early stage of spreading, but an abrupt increase of v_s begins in mid course to follow a rapid uprising with time.

Although not a few causes may be responsible for it of course, three of them are looked upon as particularly important. First, for simplicity, let us consider a system shown in Fig. 13 (N.B.Space distance is exaggerated to twice the proportion), and assume that liquid has a uniform initial superheat $T_m - T_s$ wherever in the space. For the evaporation which takes place from the bubble-liquid interface consuming superheat stored initially in liquid and supplied through heat conduction within the liquid to the interface, the integral method gives with ease the velocity of bubble boundary:

$$v_s = \frac{2}{3}K^2\left(1 + \sqrt{1 + \frac{3}{2K}}\right)\frac{\kappa_l}{r}; \qquad K = \frac{\rho_l}{\rho_v}\frac{c_{pl}(T_m - T_s)}{h_{fg}}$$

where κ_l : thermal diffusivity of liquid and c_{pl}: specific heat of liquid. The integral method is an approximate analysis but it is at least enough for examining the order of magnitude of v_s. According to this equation, v_s decreases in reciprocal proportion to r. In addition, even if this defect is overlooked, the magnitude of v_s given by the equation is too low compared with the experimental results in the early stage of bubble spreading. For example, theoretical value of v_s is only 0.037 m/sec at $r = 0.5$ mm even when the possible highest superheat of liquid $T_m - T_s = 3$°C is assumed. If agreement with the experimental value of $v_s = 0.66$ m/sec is desired, a superheat as high as $T_m - T_s = 13$°C must be assumed. This situation means at least that local heat transfer from the heating surface to the liquid which is constantly cooled near the bubble boundary is considerably important.

Second, it is noticed in Fig. 12 that the sudden increase of v_s begins near $Re_s = 2{,}000$. It is not clear whether it has a connection with the transition of liquid flow near the bubble boundary from laminar to turbulent, because there is a question about the possibility of transition whithin such a very short time as experienced in the present case. However, if the transition is possible, the superficial increase of thermal conductivity of liquid may affect the heat transfer.

Finally, the effects of evaporation from the thin liquid film on the heating surface is important especially near the final stage of bubble spreading. When the height and the radius of a flat round bubble

are s and r respectively, ratio of the base area to the side area is r/2s. In the present case of s = 0.4 mm, the ratio is greater than unity when r > 0.8 mm and becomes as high as 3.75 at r = 3 mm and 6.25 at r = 5 mm. This suggests the possibility that the effects of vaporization from the bubble base may become significant, but it must not be forgotten that the vaporization does not take place uniformly over the bubble base because thickness of the liquid film varies with r.

Now let us consider the case that there is a liquid film on a heating surface of temperature T_w and evaporation takes place quasi-steadily from the upper surface of liquid of temperature T_s. As well known, if $c_{pL}(T_w - T_s)/h_{fg} \ll 1$ as in the present study, linear temperature distribution across the film is a very good approximation to calculate instantaneous evaporation rate. Consequently, instantaneous thickness of the liquid film δ at time t is obtained with ease as follows:

$$\frac{\delta}{\delta_0} = \sqrt{1 - 2\,\frac{c_{pL}(T_w - T_s)}{h_{fg}}\,\frac{\kappa_L t}{\delta_0^2}} \qquad (5)$$

where δ_0 is δ at t = 0. Putting $\delta = 0$ in equation (5), the time necessary for drying up the liquid film of initial thickness δ_0 is given by

$$\frac{\kappa_L t}{\delta_0^2} = \frac{1}{2}\,\frac{1}{c_{pL}(T_w - T_s)/h_{fg}} \qquad (6)$$

The liquid film created during the bubble spreading is very thin compared with its extension over the heating surface. Moreover, the time necessary for the establishment of quasi-steady temperature in the film is very short compared with time during which equation (4) holds. As such we can substitute equation (4) into equation (6) in order to obtain distribution of the initial thickness of liquid film. Using numerals which correspond to the present experimental conditions, the result is

$$\delta_0 = M\sqrt{r}\,; \qquad M = 1.58 \times 10^{-4}\ m^{\frac{1}{2}} \qquad (7)$$

where unit of both δ_0 and r is meter. It is of interest to note that δ_0 is proportional to $\sqrt{r}$. With equation (7), the magnitude of initial thickness of liquid film is calculated as follows:

r =	0.5	1.0	2.0	3.0	4.0	5.0 mm
δ_0 =	3.54	5.01	7.10	8.70	10.0	11.2 μ

Returning to the bubble spreading, let us assume for simplicity that the vapor produced by vaporization from the bubble base alone is responsible to the spreading of bubble near its final stage. Then conservations of mass and of energy immediately give the velocity of bubble boundary v_s as:

$$v_s = \frac{1}{2\pi r s}\int_0^r \frac{1}{\rho_v h_{fg}}\left(\lambda_L\,\frac{T_w - T_s}{\delta}\right)2\pi r\,dr \qquad (8)$$

where λ_L is thermal conductivity of liquid and δ instantaneous thickness of liquid film. In order to integrate equation (8), the variation of δ with position and time must be known. However, numerical calculations with equations (6) and (7) show that during the rapid spreading under conditions in the present experiment, reduction of thickness of liquid layer due to vaporization is negligibly small compared with the initial thickness of liquid layer only except near the center of the heating surface. This situation may be recognized more easily by a simple calculation that if liquid layer of mean thickness of only 0.25 μ is vaporized over the heating surface, the vapor thus produced fills the space by itself. Hence δ_0 in equation (7) can be substituted into δ in equation (8) as a good approximation. Integrating equation (8) yields

$$v_s = \frac{2}{3}\left(\frac{\rho_L}{\rho_v}\right)\frac{c_{pL}(T_w - T_s)}{h_{fg}}\,\frac{\kappa_L\sqrt{r}}{M s} \qquad (9)$$

With substitution of numerals which correspond to conditions of experiment, equation (9) becomes

$$v_s = 57{,}000\sqrt{r} \qquad m/hr \qquad (10)$$

where r is a bubble radius and its unit is meter. On the other hand, the experimental result given in Fig. 12 is expressed near the final stage of spreading (r > 3 mm) by

$$v_s = 107{,}000\sqrt{r} \qquad m/hr \qquad (11)$$

Equations (10) and (11) are in the same form. Namely the analysis agrees with the experimental result in the point that v_s increases in proportion to $\sqrt{r}$ during the bubble spreading. As for the numerical disagreement, superheat of the heating surface $T_w - T_s = 3°C$ which is assumed in the analysis should be reflected besides the disregard of vaporization from other parts than the bubble base. Although the heater used in the present experiment was thick, the heating surface temperature had to suffer from fluctuations and it might be higher somewhat than the average value when the bubble spreadings started. If $T_w - T_s = 4°C$ is assumed instead of 3°C, the analysis has already given a value which is 71 % of the experimental magnitude. Anyhow, it may be said that this is a very interesting case where vaporization from a thin liquid film has an important effect on the behaviors of bubbles.

9. HEAT TRANSFER DUE TO VAPORIZATION FROM THIN LIQUID FILM

In the experiment described in section 7, heat flux of 1.6×10^5 kcal/m²hr has been obtained with superheat of 3°C. This is rather fantastic because such a high heat flux cannot usually be obtained on a flat heating surface with such a low superheat in normal pool boiling of water at atmospheric pressure. In the present case, of course, the volume increase of vapor in a very confined space induces a convection of liquid with velocities shown in Fig. 12. However, its effects on the heat transfer is not great except the pumping action discharging liquid with a superheat stored during the waiting period, since the period of bubble spreading is very short compared with the whole period of boiling. In Fig. 10, for example, the bubble spreading accomplished rapidly in frames 6 to 8 is followed with a long period where vapor continues to flow out of the circumferential opening of the space under the interference plate. Therefore the cause of the realization of high heat flux with low superheat may also be attributed mainly to the vaporization from the thin liquid film on the heating surface.

If a liquid film with an initial distribution of thickness expressed by equation (7) is vaporized to make a dried-up area of r' in radius, time for the process t' and distribution of the thickness of remaining liquid film δ' are easily obtained with equations (5), (6) and (7) as follows:

$$t' \qquad \frac{1}{c_{pL}(T_w - T_s)/h_{fg}}\,\frac{M^2 r'}{2\,\kappa_L} \qquad (12)$$

$$\delta' \qquad M\sqrt{r - r'} \qquad (r > r') \qquad (13)$$

Equations (7) and (13) permit us to calculate the quantity of liquid vaporized, hence that of heat absorbed by the evaporation from the liquid film over the heating surface of r_0 in radius, while equation (12) yields the time during the period. Then we can calculate mean heat flux $\bar{q}$, over the heating surface area and during the period, due to the evaporation from the thin liquid film as follows:

$$\bar{q} = \frac{8}{5}\frac{c_w(T_w-T_s)}{h_{fg}}\left(\frac{\rho_l k_l h_{fg}}{M\sqrt{r_0}}\right) \times \left(\frac{r_0}{r'}\right)\left\{1-\left(1-\frac{r'^{\frac{5}{2}}}{r_0}\right)-\frac{5}{3}\left(\frac{r'}{r_0}\right)\left(1-\frac{r'^{\frac{3}{2}}}{r_0}\right)\right\} \quad (14)$$

where variation of r' with time is given by equation (12). As shown in Fig. 14, $\bar{q}$ is kept at nearly constant during the period of evaporation due to simultaneous reductions of thickness and surface area of liquid film which have opposing effects on the heat transfer. Generally, the penetration of liquid into the space begins before r'/r_0 reaches unity as shown in frames 24 to 40 in Fig. 10. The magnitudes of $\bar{q}$ calculated with equation (14) under conditions in the present experiment is shown for three values of r' as follows:

r' =	1.25	2.5	5.0	mm
$\bar{q}$ =	2.42×10^5	2.64×10^5	2.52×10^5	kcal/m²hr

These magnitudes of $\bar{q}$ are about 57 % higher than the measured heat flux of 1.6×10^5 kcal/m²hr that is mean heat flux throughout the whole period including the waiting time.

Due to irregular behaviors of fluid which occasionally appeared in the experiment, it is unfortunately rather difficult to estimate average ratio of evaporation time to waiting time from the high-speed motion picture film. Roughly speaking, however, it is about 0.75 and taking this ratio into account gives a conclusion that vaporization from the thin liquid film has a contribution of about 67 % to the total heat transfer. Remaining 33 % of heat transfer is performed through mechanism such as vapor-liquid exchange and others.

10. HEAT TRANSFER IN NORMAL HIGH FLUX NUCLEATE BOILING

With the results obtained in the preceding section alone, we cannot of course derive a hasty conclusion that vaporization from the thin liquid films at discrete bubble bases have important contributions to the heat transfer in normal high flux boiling. However, in connection with equation (14), it may not be useless to mention the following hypothetical analysis attempted for normal boiling.

The mechanism of how such a thin liquid film as expressed in equation (7) is created over the heating surface has not been made clear, so that further work should be undertaken to discover it. At present, however, it seems likely that the thickness of liquid film has little connection with such moving velocity of bubble boundary as shown in Fig. 12. Considering such a situation too, let us give a very bald trial to assume that equation (7) holds for the thickness of liquid film independently of boiling conditions. Then it comes out from equation (14) that $\bar{q}$ increases in proportion to $(T_w - T_s)/\sqrt{r_0}$, that is, when either superheat is high or radius of bubble base is small, $\bar{q}$ can reach very high values. If we take the case, as an example, of saturated water boiling at atmospheric pressure with a superheat of $T_w - T_s = 20$°C, equation (14) yields magnitudes of $\bar{q}$ for three kinds of bubble base radius r_0 as follows:

r_0 =	0.1	0.5	1.0	mm
$\bar{q}$ =	9.8×10^6	4.4×10^6	3.1×10^6	kcal/m²hr

where $r'/r_0 = 0$ has been assumed for simplicity since the $\bar{q}$ does not vary so much as shown in Fig. 14.

On the other hand, let us take experimental heat flux 8×10^5 to 1×10^6 kcal/m²hr for the superheat of $T_w - T_s = 20$°C as comparatively high values among data obtained on flat copper heating surface for saturated water boiling at atmospheric pressure. For this range of heat fluxes, as far as the experimental results available are concerned, the bubble base radius should be presumed to be in the order of 0.1 mm. This comes from not only Gaertner and Westwater's result shown in Fig.6 but also Gaertner's result(3) that the average radius of vapor stems under vapor masses was observed to be 0.1 mm at a heat flux of 8.1×10^5 kcal/m²hr from side-view photographs of boiling. In addition, we know from the data provided in Gaertner and Westwater's paper(2), as an approximate knowledge at least, that total area of bubble bases to heating surface area ratio is nearly constant at about 1/11 throughout the region of interference.

At this stage, if it is attempted to harmonize the analysis with the experimental facts under the assumption that main heat transfer depends on the thin-liquid-film vaporization, it is found that there is scarcely any room to take such a waiting time into account as observed in the region of isolated bubbles. Therefore we reach naturally a model of high flux boiling where vapor generated at each active nucleation site must constitute a vapor stem which lasts for some time and is followed almost instantaneously by a successor which grows from a bubble. It may be of interest in this connection to mention that, according to equations (12) and (14), the liquid film created at a bubble base of 0.1 mm in radius has a capacity for the present conditions to be able to produce vapor of volume sixteen times as large as a hemisphere of 0.1 mm in radius during 0.2 msec.

As such the present presumption shows a possibility of a boiling which combines discrete vapor stems growing from bubbles and the thin-liquid-vaporization, but it is at any rate based on a hypothetical assumption. In addition, there are different views between different investigators as to the behaviors of vapor on the heating surface at high heat fluxes. Torikai and Yamazaki(6), who observed water boiling from below through a glass surface coated with a transparent, heated film of tin oxide, stated that the majority of bubble base area contacting to the surface were in a condition of discrete bubbles at a heat flux of 1.1×10^6 kcal/m²hr. However, Kirby and Westwater(7) who observed carbon tetrachloride boiling in a similar manner stressed the local formation of irregular vapor patches on the heating surface holding thin liquid films between the patches and the surface. Therefore it will be one of the future problems to clarify whether such local vapor patches are really the cause for high rate of boiling heat transfer or it is better to regard them as the results of local collapse of vapor-stems-boiling followed by some reduction of local heat transfer rate.

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NOTATION

c_{pL} specific heat of liquid
g gravitational acceleration
h_{fg} latent heat of vaporization
M dimensional constant (= 1.58×10^{-4} m)
q heat flux density
$\bar{q}$ heat flux density due to vaporization from liquid film
q_c peak heat flux density
Re_s Reynolds number of liquid flow at the front of bubble boundary
r radius
r' radius of dried-up area
r_0 radius of heating surface or interference plate (= 5.0 mm)
s space distance between interference plate and heating surface
s_f s at the beginning of surface temperature falling
T_s saturation temperature
T_w temperature of heating surface
t time
t' time for creation of dried-up area of r' in radius
v_0 outflow velocity of liquid at the circumferential opening
v_g velocity of bubble boundary

δ instantaneous thickness of liquid film
δ_0 initial thickness of liquid film
κ_L thermal diffusivity of liquid
λ_L thermal conductivity of liquid
ν_L kinematic viscosity of liquid
ρ_L density of liquid
ρ_V density of vapor
σ surface tension

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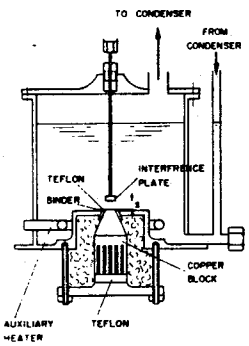

Fig. 1. Boiler and heater.

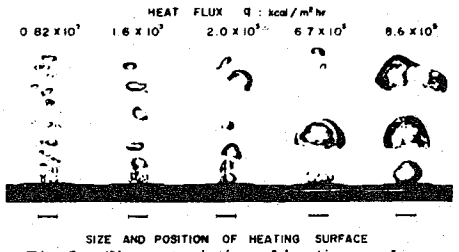

Fig.2. Characteristics of heating surface.

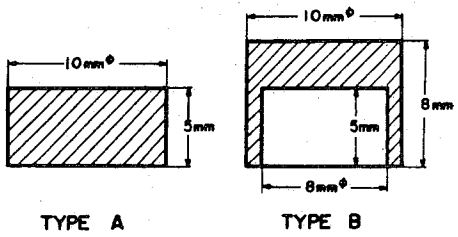

Fig.3. Interference plates.

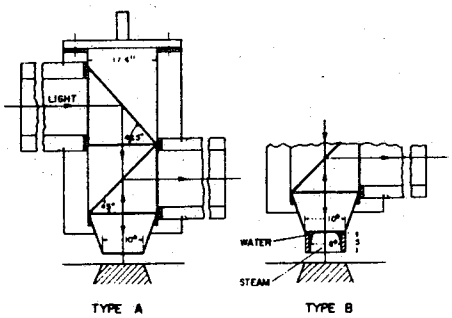

Fig.4. Optical assemblies.

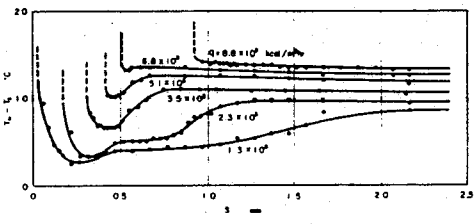

Fig.5. Variation of heating surface temperature with space distance for interference plate of type A.

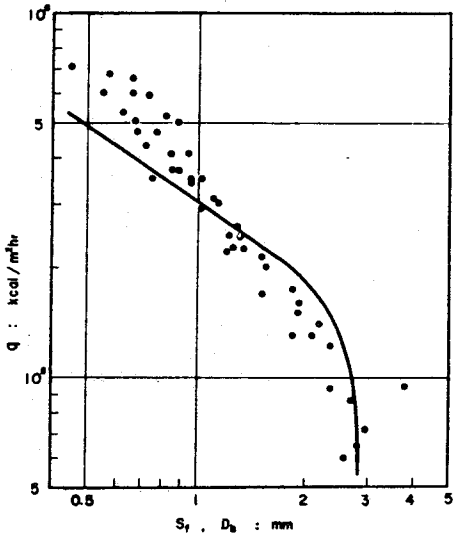

Fig.6. Variation of space distance s_f and diameter of breakoff bubbles or vapor stems D_b (solid line) with heat flux.

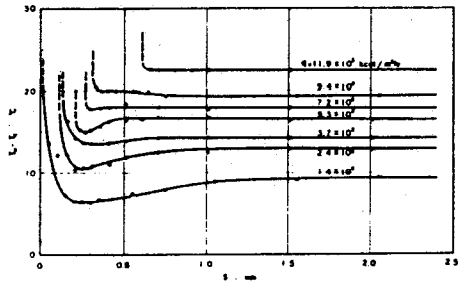

Fig.7. Variation of heating surface temperature with space distance for interference plate of type B.

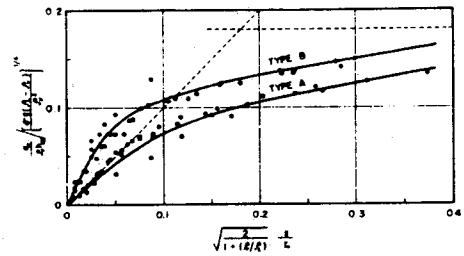

Fig.8. Variation of critical heat flux with space distance.

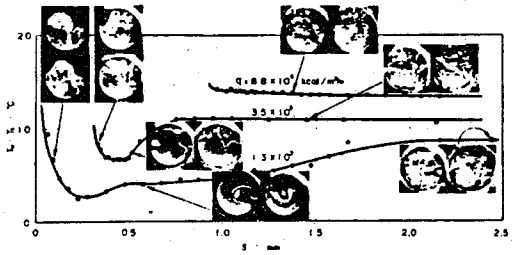

Fig.9. Behaviors of fluid in the space between interference plate and heating surface.

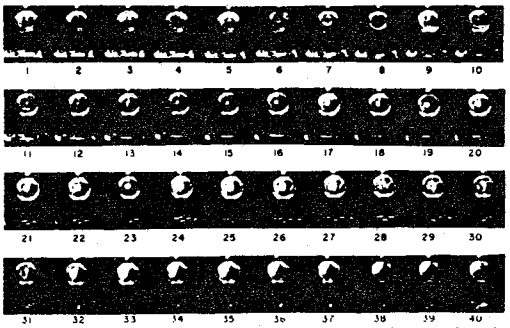

Fig. 10. Successive states of fluid at intervals of 1.53 msec. ($q = 1.6 \times 10^5$ kcal/m²hr; s = 0.4 mm; $T_w - T_s = 3.0$°C)

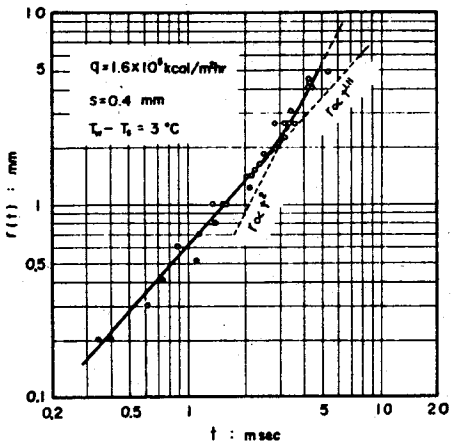

Fig. 11. Variation of bubble radius with time.

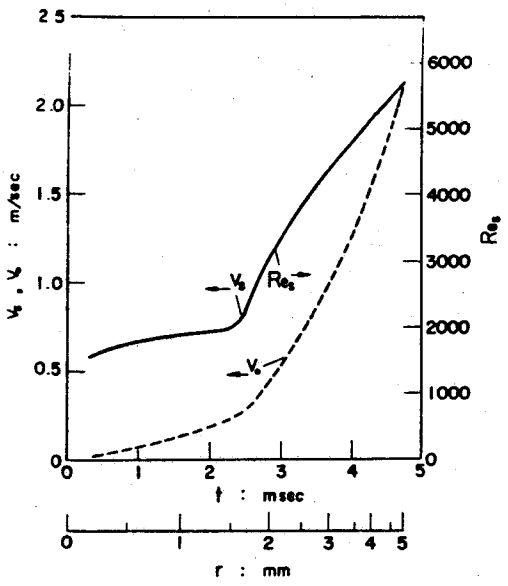

Fig. 12. Velocity of bubble boundary, outflow velocity of liquid and Reynolds number of liquid flow near the bubble boundary.

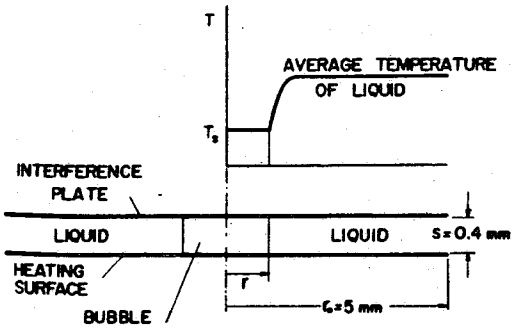

Fig. 13. Flat bubble spreading.

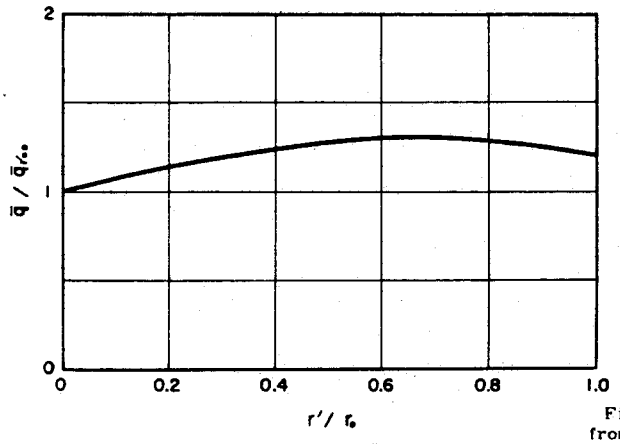

Fig. 14. Variation of heat flux due to vaporization from liquid film on heating surface.

Nucleate and Transition Boiling in a Narrow Space between two Horizontal, Parallel Disk-Surfaces[+]

By Yoshiro KATTO[++], Sadao YOKOYA[+++] and Kozo TERAOKA[++++]

Nucleate and transition boiling within a space gap made by holding a 11-mm dia. glass disc in coaxial position above a heated 11-mm dia. horizontal, upward-facing copper disc has been studied experimentally for saturated water at atmospheric pressure. The distance between the glass disc and the heated surface s is changed within a range from s=∞ to s=0.1 mm, and high-speed photography is employed to record the behavior of liquid and vapor within the narrow space. It is clarified in the present study that, following the change of s, characteristic variations appear in the state of boiling, in the form of supply of liquid to the heated surface, and in the occurrence of burnout, respectively.

1. Introduction

Nucleate boiling in a very narrow space or where conditions are quite similar to it has often become a problem in the related field of industries , and Ishibashi, Nishikawa, et al.[1][2][3][4], Nakajima[5], and others have presented interesting studies on such subject. These studies, however, dealt mainly with heat transfer in the nucleate boiling region, and not with burnout or transition boiling

Regarding basic studies on nucleate boiling and burnout, the authors of the present paper have published reports[6][7] of their experiment on nucleate boiling, including burnout, with the narrow space gap, s, varied from 0.1 to 10 mm which is created by a 10-mm dia. horizontally placed, upward-facing round heated surface and an interference disc of same diameter placed just above and in parallel with the former in the saturated water at the atmospheric pressure. However, behavior of vapor and liquid, especially process of liquid supply into such narrow space, has not been studied in good depth, and experiments on transition boiling are lacking ; therefore, the state at the onset of burnout has not been clarified.

For this reason, experiments were performed from nucleate boiling through transition boiling in the relatively high heat flux region within the narrow space whose condition was almost identical to those described in the reports[6][7]. This paper presents results of the experiment and systematic observation on characteristic changes in heat transfer, behavior of vapor and liquid and burnout by varying the narrow space gap, s.

2. Experimental apparatus

The apparatus used in this experiment was identical to the one which the authors used for other studies[8], and is illustrated in Fig.1. Boiling was generated on a horizontally placed, upward-facing round heated surface (11-mm dia.) located on the top of a conical copper block whose bottom was heated by an electric heater. The liquid used in this experiment was distilled water at the atmospheric pressure (free level of the water contained in the boiling vessel shown in Fig.1 was held at approximately 100 mm above the heating surface), and it was held at a saturation temperature by an auxiliary heater.

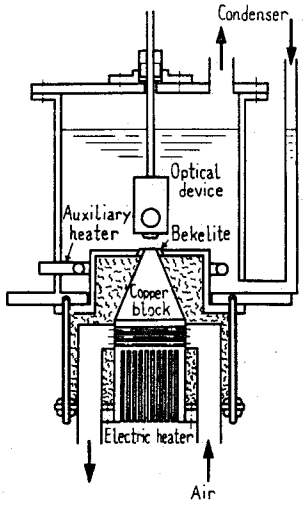

Fig.1 Main part of experimental apparatus

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Internal construction of the optical device located over the heating surface in Fig.1 is shown in Fig.2. Bottom of the conical glass (11-mm dia. at bottom) located at a lower section of apparatus not only provides the necessary space gap between itself and the heating surface, but also permits optical observation of behavior of vapor and liquid in the gap through the glass. This behavior of vapor and liquid was filmed by a high-speed camera. In setting the space gap, s, above the heating surface, as errors caused by thermal expansion in every part of the apparatus needed to be minimized, the temperature in the boiling vessel was held at 100°C for approximately one hour, and the gap, s, was set by the dial gauge whose reading was made to correspond with the amount of movement of the optical device support post.

3. Result of experiment

Boiling curves as shown in Fig.3 were obtained in the relatively high heat flux region (over 10^5kcal/m^2h) from the experiment, using the aforementioned experimental apparatus, with the space gap, s, varied from s=∞ (free boiling) to s=0.1 mm. These curves depict very characteristic changes as the space gap, s, decreases. Causes of such changes in the boiling curves are hereunder analyzed in conjunction with the high-speed photography of behavior of vapor and liquid on the heated surface.

3.1 Observation at s=2.0 mm

As shown in Fig.3, the nucleate boiling at s=2.0 mm showed almost identical heat transfer characteristics to that in free boiling (s=∞). Occurrence of such characteristics at s≧2.0 mm would be due to the following reasons (see references [6] to [8]). In case of boiling of saturated water at the atmospheric pressure, a very thin "nucleate boiling liquid layer" exists on the heated surface in the high heat flux region in the order of 10^5kcal/m^2h or higher. Vapors are generated in this layer and erupt upward, forming a vapor mass on the layer. Moreover, in such a large space gap as s≧2.0 mm, the glass wall is positioned in the vapor mass, and it does not exert any effect on the nucleate boiling liquid layer just below the vapor mass.

Burnout heat flux, on the other hand, is found to be much smaller in value at s=2.0 mm, than in free boiling as shown in Fig.3. The principal reason for it could be that the time duration of the heated surface being covered by the vapor mass becomes longer with a decrease of the space gap, s, as the paper[7] indicates.

In the boiling at s=2.0 mm, it was observed that the liquid penetrates into the space gap as the vapor mass starts to detach from one end of the gap (refer to the paper[9] for studies on dynamical behavior of vapor mass detachment) so as to fill the volume the vapor mass has occupied, as illustrated in Fig.4. Throughout both the nucleate and transition boiling regions, it was observed that the liquid supply to the heated surface took place in the form of vapor and liquid interchange. Let's give an example. Behavior of vapor and liquid in transition boiling at the space gap s=2.0 mm and the superheat T_w-T_s =30.8°C (where T_w is temperature of the heated surface, and T_s the saturation temperature) was photographed by a high-speed camera, and successive changes in the behavior are shown in Fig.5. In the film frames Nos.1 to 4, it is seen that liquid penetrates into the space gap from the left, intense nucleate boiling is taking place in the crescent-shaped liquid penetrated area, and the boundary between the liquid-penetrated area and the vapor-occupied area looks discontinuous. In the frames No.6 and thereafter, it is observed that the vapor created by the nucleate boiling fills again the whole space gap, the heated surface dries up, and the nucleate boiling has disappeared (the waving seen in the frames No.6 and thereafter is

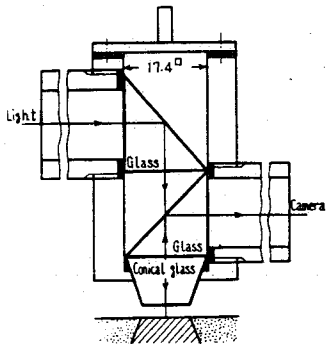


Fig.2 Optical device

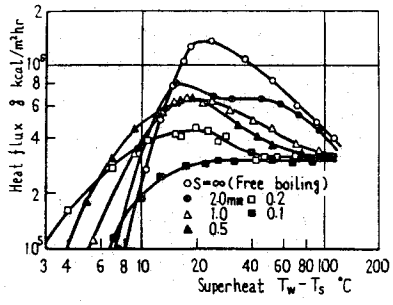


Fig.3 Boiling curve

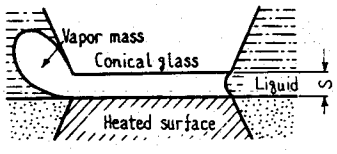


Fig.4 Detachment of vapor mass and invasion of liquid at the heated surface

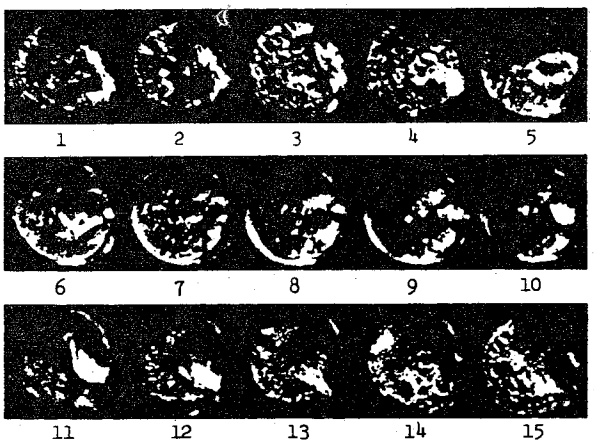
the waving generated on the liquid surface which is continuously in contact with the lower surface of the conical glass of the optical device, and thus the phenomenon is not related to the heated surface). In the frame No.15, liquid is again seen to start penetrating from the left.

3.2 Observation at s=0.2 to 0.5 mm

3.2.1 Nucleate boiling region at low superheat

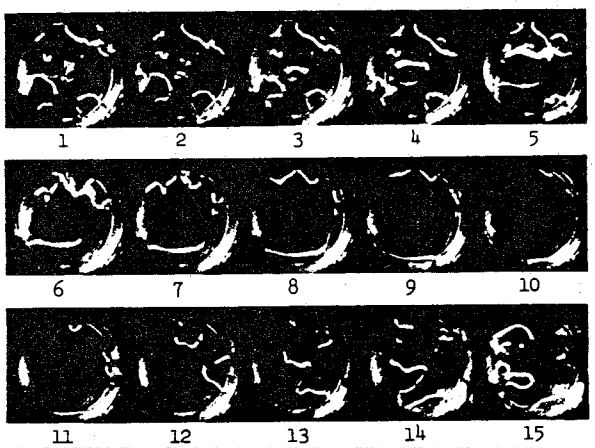
With the decrease of the space gap, s, to 0.2 to 0.5 mm, heat transfer in the low superheat region (see the area $T_w-T_s<$ 10°C) of the nucleate boiling region was found to become higher than that of the free boiling.

This increase in heat transfer is due to the following reason. Behavior of vapor and liquid in nucleate boiling of s=0.5 mm and T_w-T_s=5.6°C was photographed by a high-speed camera, and successive changes in the behavior are shown in Fig.6. As clearly seen in the film frames Nos.1 to 10, the bubble (incipient bubble, and not the coalesced one) which starts at the central portion of the heated surface grows in a special manner spreading within the narrow space gap due to the existence of a glass wall above the heated surface, and an extremely thin static liquid film is formed on the heated surface at the bubble bottom. Rapid evaporation takes place on the upper face of this liquid film, and this causes such an increase in the heat transfer. After this stage, as seen in the frames No. 11 and thereafter, the bubble vapor withdraws at a relatively slow speed from the space gap and the liquid penetrates into the space gap to fill the volume of escaping bubble vapor.



s=2.0 mm, T_w-T_s=30.8°C, time interval between each frame : 3.8 ms for frames 1 to 10, and 6.5 ms for frames 10 to 15

Fig.5 Behavior of vapor and liquid in the space



s=0.5 mm, T_w-T_s=5.6°C, time interval between each frame : 0.75 ms for frames 1 to 10, and 4.7 ms for frames 10 to 15

Fig.6 Behavior of vapor and liquid in the space

3.2.2 Nucleate boiling region at high superheat

When superheat of the heated surface becomes higher in the nucleate boiling region, the heat transfer characteristics become poorer than those in free boiling, as shown by the boiling curves of s=0.2 to 0.5 mm in Fig.3, and burnout heat flux decreases.

In this region, it is noted from the observation of behavior of vapor and liquid by a high-speed camera that the mode of liquid supply to the heated surface has changed to a different mode from that shown in Fig.4. On account of smaller space gap, s, the pressure difference created by the convex meniscus of the liquid front and by the surface tension prevents the liquid from flowing if the liquid tends to penetrate into the space gap in the mode of Fig.4. In place of the Fig.4 mode, a different liquid supply mode as schematically illustrated in Fig.7 appears. This mode is such that a portion of the liquid film that covers the glass wall as seen in Fig.7(a) sometimes approaches the heated surface due to, for example, waving at the interface and supplies the liquid there as seen in Fig.7(b).

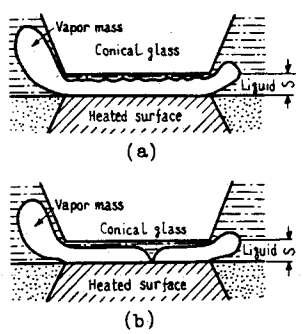
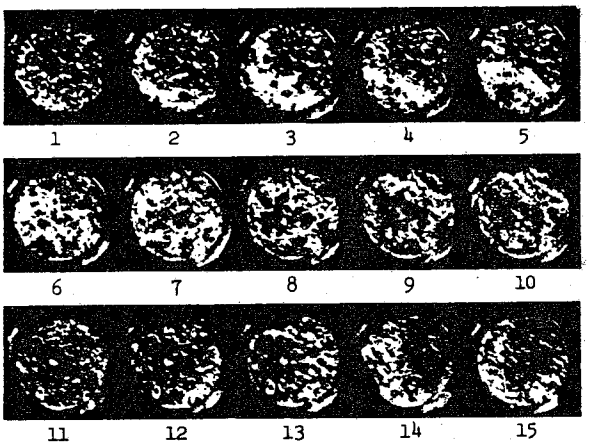


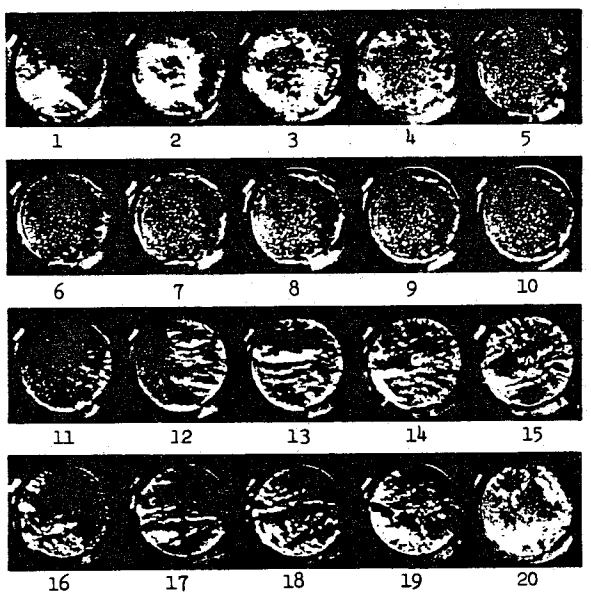
Fig.7 Type of supply of liquid to the heated surface

The state of the boiling with this type of liquid supply mode is illustrated by the actual photographs taken by a high-speed camera as follows. Figure 8 shows successive changes in the behavior of liquid and vapor in the space gap with respect to the nucleate boiling in such a relatively high superheat region as T_w-T_s= 18.4°C in the case of s=0.5 mm. Nucleate boiling is always seen on the heated surface throughout all 15 film frames. Detailed observation of these photographs, however, reveals that a single nucleate boiling liquid layer, as in the free boiling, does not cover the heated surface, but a number of liquid layers spread over the heated surface as in the form of interrupted liquid segments and nucleate boiling is taking place in each of such liquid segments. In other words, the nucleate boiling liquid layer and the dry portion coexist on the heated surface. This can be regarded as the principal reason why heat transfer characteristics at s=0.2 to 0.5 mm are poorer than those in free boiling as seen in Fig.3.



s=0.5 mm, T_w-T_s=18.4°C, time interval between each frame : 0.54 ms for frames 1 to 10, and 3.2 ms for frames 10 to 15

Fig.8 Behavior of vapor and liquid in the space



s=0.5 mm, T_w-T_s=33.9°C, time interval between each frame : 0.63 ms for frames 1 to 10, and 5.0 ms for frames 10 to 20

Fig.9 Behavior of vapor and liquid in the space

In the film frames Nos.1 to 7 of Fig.8, it is observed that the white patterns originate at the lower left end of the heated surface and spread over the heated surface. These white patterns are the waving that takes place on the surface of a liquid film which covers the glass wall. When such waving takes place, liquid supply to the heated surface in the mode of Fig.7(b) often happens, and the liquid which comes in contact with the heated surface is scattered due to the nucleate boiling which takes place in the contacted area. It is presumed that interrupted liquid segments are generated thereby on the heated surface. In fact, in the case of Fig.8, one can notice considerable changes in the liquid distribution on the heated surface before and after the film frames Nos.7 and 8.

3.2.3 Transition boiling region

The liquid supply mode of Fig. 7 is observed to remain the same even when the superheat of the heated surface increases further and enters the transition boiling region, but the mode itself becomes more acute than in the previous case.

Behavior of liquid and vapor in the space gap in the transition boiling at s=0.5 mm and the superheat T_w-T_s=33.9°C was photographed by a high-speed camera, and successive changes in the behavior are shown in Fig.9. In the film frame No.2, it is observed that the liquid at the bottom of the conical glass is in contact with the central portion of the heated surface. In this case, the superheat is very high and the liquid makes contact with the almost dry heated surface, so that very intense nucleate boiling takes place there and fine liquid drops are scattered around. The numerous fine scattered liquid drops which can be seen on the heated surface in the film frames Nos.3 to 11 of Fig.9 are thus created. Since these liquid drops are very small (approximately 0.1 mm in diameter), generation of bubbles does not take place and the liquid

drops vaporize in the static state on the heated surface. The waving seen in the film frames Nos.11 to 19 is the one which traverses the liquid layer at the bottom of the conical glass. In the film frame No.20, again takes place liquid supply to the heated surface and produces the same conditions as seen in the film frames Nos. 2 and 3.

3.3 Observation at s=0.1 mm

Following two characteristics are found from the boiling curve of s=0.1 mm in Fig.3. First, when the value of space gap, s, is decreased from the free boiling (s=∞) with the heat flux kept constant, for example, at $q=2\times10^5$kcal/m^{2}h, the superheat T_w-T_s of the heated surface decreases until s approaches 0.2 mm. However, a rapid return to the high value of T_w-T_s is caused when s approaches 0.1 mm. Second, the boiling curve of s=0.1 mm in Fig.3 no longer has the portion which falls down to the right and accordingly no burnout point appears on the curve; therefore, the boiling at this space gap appears to be entirely different from the normal boiling conditions where nucleate and transition boilings are clearly identifiable.

Behavior of liquid and vapor at s=0.1 mm was observed by a high-speed camera. It was seen that, because of extremely small space gap, s, even the liquid supply mode of Fig.7 could not take place and major portion of the heated surface, except its periphery, was always kept dry. Figure 10 shows an illustration showing the state on the heated surface at s=0.1 mm. When the superheat T_w-T_s is relatively low as in Fig.10(a), the liquid is seen entering or withdrawing irregularly at periphery of the heated surface , but when the superheat T_w-T_s is relatively high, it produces the state such as shown in Fig.10(b) where small liquid drops depositing on the periphery of the heated surface only evaporate. These small liquid drops are probably the splashes made by the nucleate boiling when the liquid occasionally comes in contact with the periphery of the heated surface.

3.4 Chracteristics of heat transfer near film boiling

The trend can be seen in Fig.3 that all the boiling curves within the range of s=∞ ∽ 0.1 mm converge to a same point in the vicinity of the superheat T_w-T_s=150°C.

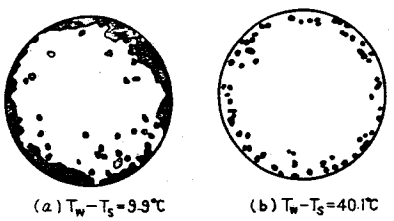


Fig.10 Appearance in the space at s=0.1mm

4. Changes in the state of burnout generation with the change in space gap

Changes in the state of boiling with a decrease of the space gap have been reported in the preceding chapter, and in this chapter the changes in the state of burnout generation with the change in the space gap will be described. Burnout so referred to in this chapter means only the critical heat flux point on the boiling curve.

The burnout within the range of space gaps s=∞ ∽ 2 mm follows the same mechanism and has the same characteristics as those in free boiling. In this space gap region, generally, the whole heated surface is always wetted by the liquid in the range of nucleate boiling, and dryness starts to take place on the heated surface in the transition boiling region.

When the space gap becomes the order of s=0.5 to 0.2 mm, however, in the nucleate boiling region near the burnout point, dry portions have already been in existence with the interrupted liquid segments on the heated surface. The nucleate boiling under such conditions becomes inevitable as far as the liquid supply mode is what is seen in Fig.7. Therefore, the state of boiling changes before and after the burnout point more slowly than in the case of free boiling, and it could be said that the burnout, when compared with that in free boiling, does not take place in a clear-cut manner. Actually, when the space gap becomes as small as s=0.1 mm, burnout itself can no longer be in existence as seen in the boiling curve of s=0.1 mm in Fig.3.

Figure 11 shows result of the measurement performed in the present study on the relation of the space gap, s, and the burnout heat flux, q_c , and the horizontal thin line in the upper right portion shows the measured values of q_c in the free boiling (s=∞). It is interesting to note that there is a slight change in the relation of q_c to s between the two regions of s higher and lower than s=2 mm. However, it may be premature to draw a conclusion on this characteristic change at s=2 mm from this curve alone, and further investigation on this point will be required, because experimental q_c - s data shown in

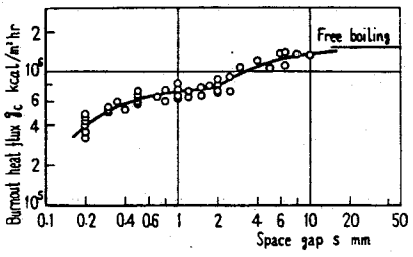


Fig.11 Variation of burnout heat flux with space gap

Fig.4 of the paper[7] almost entirely agree with Fig.11 of this paper but they do not show such a change in the curve at s=2 mm. It may be of some use, however, to mention that the curve in Fig.4 of the paper[7] would have shown such a change as aforementioned if only the two data between s=1 mm and s=2 mm in Fig.4 of the paper[7] were somewhat lower.

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BRIEF COMMUNICATION

CRITICAL HEAT FLUX OF SATURATED NATURAL CONVECTION BOILING IN A SPACE BOUNDED BY TWO HORIZONTAL CO-AXIAL DISKS AND HEATED FROM BELOW

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1. INTRODUCTION

Studies have been made of critical heat flux (CHF) of natural convection boiling in horizontally confined geometries. Katto & Yokoya (1966) performed experiments on boiling of saturated water at atmospheric pressure in a space bounded by two horizontal co-axial disks with lower-disk heating, but no correlations of CHF data were presented. Jensen *et al.* (1976) carried out CHF experiments for natural convection boiling of saturated water at atmospheric pressure in horizontal annular geometries with inside-rod heating, and presented a generalized equation correlating their own data of water together with the data of refrigerant 113 obtained by others. Their correlation was derived on the basis of Reynolds number referenced to viscous flow of vapor in the confined space. The effect of gravitational acceleration g was not taken into account. On the other hand, Smirnov *et al.* (1976) and Smirnov (1976) measured the critical heat flux q_{co} for saturated boiling of water at 0.5–5 bar and that of ethyl alcohol at 1 bar in a horizontal rectangular slot of length b and clearance s. Their data were correlated by the following equation:

$$q_{co} = q^*_{co} \frac{1}{\sqrt{\left(1 + C_3 \frac{\rho_v}{\rho_l}\left(\frac{b}{s}\right)^3\right)}} \qquad [1]$$

where q^*_{co} is the critical heat flux in ordinary pool boiling on an open heated surface, C_3 a constant, ρ_v the density of vapor, and ρ_l the density of liquid. Agreement between [1] and experimental data is not good with noticeable error extending from -33 to $+70\%$.

In the present study, experiments were made to obtain CHF data for natural convection boiling in a space bounded by two horizontal co-axial disks in a pool of saturated liquid at atmospheric pressure. The fluids used were water, R-113, ethyl alcohol and benzene. A generalized correlation equation was derived for the CHF data thus obtained.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

The main part of experimental apparatus employed is shown in figure 1. The test liquid (distilled water, ethyl alcohol of purity higher than 99.5%, R-113 of purity higher than 99.99%, and benzene of purity higher than 95%) stored in a boiling vessel (1) at atmospheric pressure is kept at saturation temperature by means of an auxiliary heater (5). The generated vapor is returned through a condenser to the boiling vessel. The flat, end surface of the upper cylindrical part of a copper block (2), mounted with electric heaters of plate type (4) and insulated by a Bakelite sleeve (5), as well as glass wool (7),

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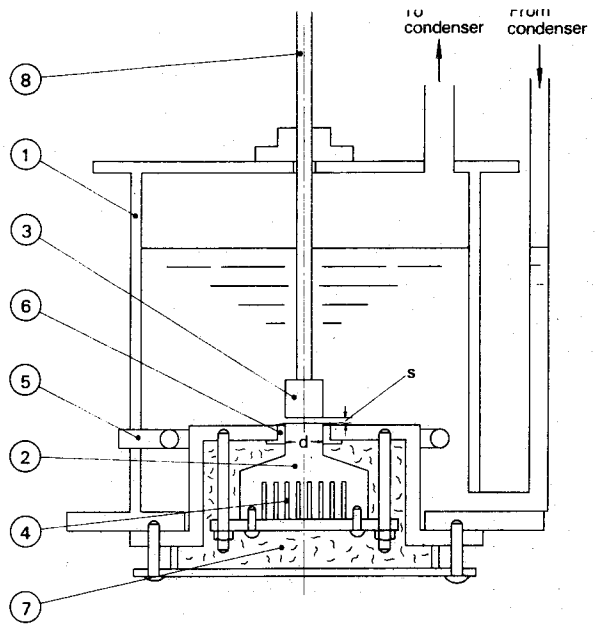

Figure 1. Main part of experimental apparatus.

provides a heated disk surface for boiling. The heat flux across the heated surface is determined by the temperature gradient, measured with three copper-constantan thermocouples placed at regular intervals along the axis of the cylindrical part of the copper block. The thermal conductivity of each copper block employed, which is required for an accurate estimate of the heat flux, has been carefully determined in preliminary experiments. Two diameters of heated surface, 10 and 20 mm, are employed. The axial length of the cylindrical part is 15 mm for the runs with water, which has a comparatively high CHF, and 40 mm for the experiment with the other fluids which have a comparatively low CHF. The unheated Bakelite cylinder (3), which is supported co-axially above the heated disk surface at a distance s, has a thickness and a diameter of the same length as the diameter of the heated surface d. A rod (8) supporting the unheated cylinder (3) is movable up and down by means of a screw system, and the distance s between the heated and unheated disk surface is measured by a dial gauge, not shown in figure 1.

In order to estimate the distance s correctly, the displacement of the heated surface level due to the thermal expansion of the copper block must be taken into account. Therefore, when boiling becomes steady at a heat flux which is a little lower than the presumed value of CHF for a prescribed distance s, the unheated cylinder is rapidly lowered until coming in contact with the heated surface. In this case, the heated surface temperature begins to rise if the surface becomes dry, but it is actually possible to avoid the disturbance by a prompt reading of the dial gauge at the moment of contact. The origin of s effective near the CHF point is thus determined. Immediately after the reading of the dial gauge, the unheated cylinder is lifted and can be fixed at a prescribed distance s above the heated surface. Then, the heat flux is raised stepwise by Δq leading to the corresponding steady state boiling, and thereafter, the procedure is repeated for the increment of heat flux until the occurrence of CHF. In the present study, each increment of heat flux Δq is limited to less than 5% of the preceding heat flux q. When an increment of heat flux brings about the condition where the heated surface temperature cannot have a steady state, critical heat flux q_{co} is determined with an error less than about 5%.

3. CORRELATION OF EXPERIMENTAL DATA

In order to get a clue for correlating the experimental data obtained, dimensional analysis is

utilized. Postulating that the hydrodynamic condition of two-phase vapor-liquid flow is responsible for the occurrence of CHF in the present boiling system, the superficial vapor velocity $q_{co}/\rho_v H_{fg}$ at a critical heat flux of q_{co}, where ρ_v is the density of vapor and H_{fg} the latent heat of evaporation, is governed by physical quantities, such as the density of liquid ρ_l and that of vapor ρ_v—relating to the force of inertia; the surface tension of the interface σ; the gravitational force $g(\rho_l - \rho_v)$ relating to the buoyancy; the diameter of heated disk surface d; and the normal distance between parallel disk surfaces s. At present, the effect of viscous force on CHF is ignored because it is presumed to be negligibly small. Applying dimensional analysis to the physical quantities mentioned above, yields the following relation:

$$\frac{q_{co}}{\rho_v H_{fg}} \Big/ \sqrt[4]{\left(\frac{\sigma g(\rho_l - \rho_v)}{\rho_v^2}\right)} = f\left(\frac{\rho_v}{\rho_l}, \frac{g(\rho_l - \rho_v)d^2}{\sigma}, \frac{s}{d}\right). \qquad [2]$$

Now, in the light of [2], the experimental data for q_{co} is plotted in the following form:

$$\sqrt[4]{\left(\frac{\sigma g(\rho_l - \rho_v)}{\rho_v^2}\right)} \Big/ \frac{q_{co}}{\rho_v H_{fg}} \text{ vs } \frac{d}{s}.$$

Figure 2 is obtained for the case of $d = 10$ mm and figure 3 for that of $d = 20$ mm. In these figures, it is noted that experimental data at $d/s = 0$, which corresponds to the reciprocal of the constant

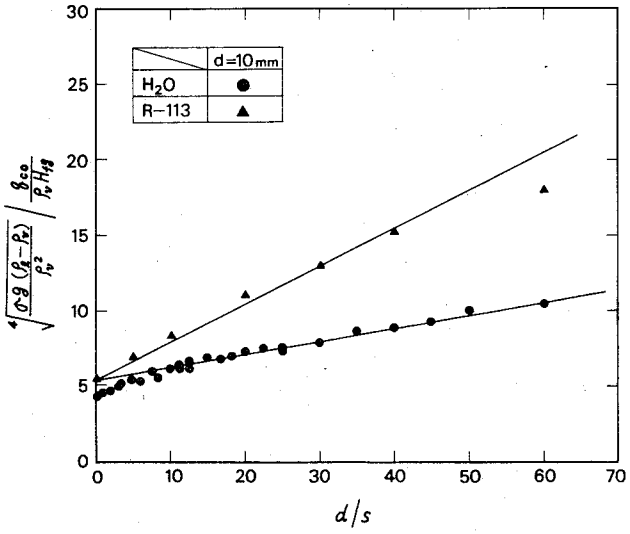

Figure 2. Critical heat flux q_{co} for boiling on a 10 mm dia. heated surface.

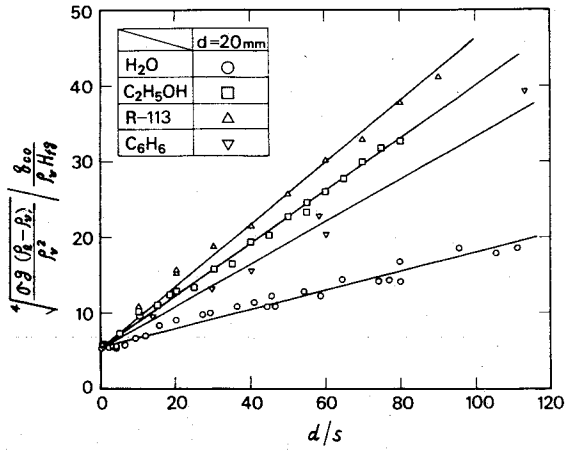

Figure 3. Critical heat flux q_{co} for boiling on a 20 mm dia. heated surface.

C included in Kutateladze (1952) or Zuber (1959) correlation equation of CHF for ordinary pool boiling, shows nearly the same values for all liquids and heated surface diameters. Taking an average,

$$\frac{1}{C} = 5.5 \quad \text{or} \quad C = 0.18. \tag{3}$$

This value of C agrees with the constant in Zuber correlation recommended by Rohsenow (1973).

Next, the following form of CHF correlation may be assumed for the data for $d/s > 0$ in figures 2 and 3:

$$\sqrt[4]{\left(\frac{\sigma g(\rho_l - \rho_v)}{\rho_v^2}\right)} \Big/ \frac{q_{co}}{\rho_v H_{fg}} = \frac{1}{C}\left(1 + \Psi \frac{d}{s}\right) \tag{4}$$

where Ψ/C (that is, 5.5Ψ) denotes the gradient of every straight line drawn for every group of experiments in figures 2 and 3 so as to pass the point 5.5 on the ordinate axis. Of course, Ψ takes a different value for every combination of test liquid and heated surface diameter. Experimental values of Ψ are listed in table 1.

According to [2], Ψ may be regarded as a function of ρ_v/ρ_l and $g(\rho_l - \rho_v)d^2/\sigma$. A test of this possibility is shown in figure 4 where the values of Ψ listed in table 1 are plotted against $g(\rho_l - \rho_v)d^2/\sigma$. It may be noted that there is a close relation between Ψ and $g(\rho_l - \rho_v)d^2/\sigma$, but a slight deviation seems to exist between the group for water for which ρ_l/ρ_v is comparatively high and the group of other liquids for which ρ_l/ρ_v is comparatively low. In fact, as shown in figure 5, improvement is obtained if the effect of ρ_v/ρ_l on Ψ is taken into account and the result of figure 5 is expressed as follows:

$$\Psi = 0.00918\left(\frac{\rho_v}{\rho_l}\right)^{0.14} \sqrt{\left(\frac{g(\rho_l - \rho_v)d^2}{\sigma}\right)}. \tag{5}$$

Substituting C and Ψ on the RHS of [4] by [3] and [5] respectively, yields a generalized correlation equation as follows:

$$\frac{q_{co}}{\rho_v H_{fg}} \Big/ \sqrt[4]{\left(\frac{\sigma g(\rho_l - \rho_v)}{\rho_v^2}\right)} = 0.18 \frac{1}{1 + 0.00918\left(\frac{\rho_v}{\rho_l}\right)^{0.14} \sqrt{\left(\frac{g(\rho_l - \rho_v)d^2}{\sigma}\right)} \frac{d}{s}}.$$

Table 1. Experimental data of Ψ and magnitude of d'

Fluid	d (mm)	Ψ	d'
H_2O	10	1.53×10^{-2}	4.05
H_2O	20	2.45×10^{-2}	8.10
R-113	10	4.55×10^{-2}	9.93
R-113	20	7.31×10^{-2}	19.9
C_2O_5OH	20	5.93×10^{-2}	12.9
C_6H_6	20	4.44×10^{-2}	12.3

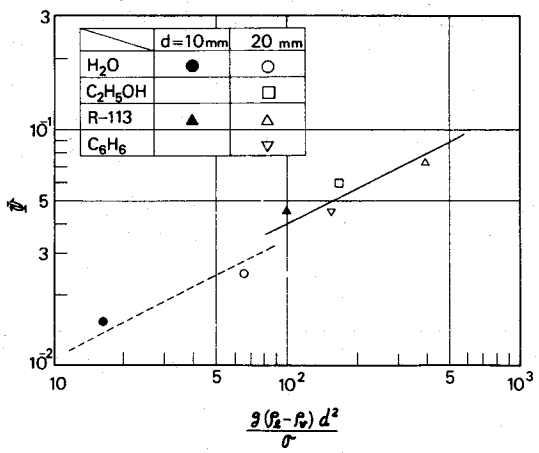

Figure 4. Relation between Ψ and $g(\rho_l - \rho_v)d^2/\sigma$.

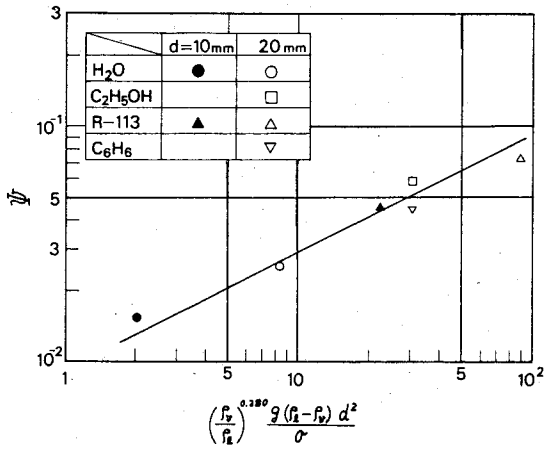

Figure 5. A generalized correlation of Ψ.

Figure 6 shows a comparison of $q_{co}/\rho_v H_{fg}$ obtained experimentally in the present study with the prediction of [6]. It is noted in figure 6 that a part of the water data, indicated by black circles, exhibits somewhat larger deviations than other data, but it must be remembered that these data are in the vicinity of $d/s = 0$ in figure 2. In other words, the above deviation has arisen in connection with the fact that the boiling of water on a 10 mm dia. heated surface exhibits somewhat higher critical heat flux than usual in the vicinity of $d/s = 0$. In this respect, it may be useful to note that for the test liquids employed the magnitude of $d' = d\sqrt{(g(\rho_l - \rho_v)/\sigma)}$ as shown in table 1 is lowest for water for $d = 10$ mm. According to Lienhard & Dhir (1973), when d' is as small as 4.05, there is a possibility of generating such a deviation of CHF as above, that is, to give somewhat higher CHF than usual.

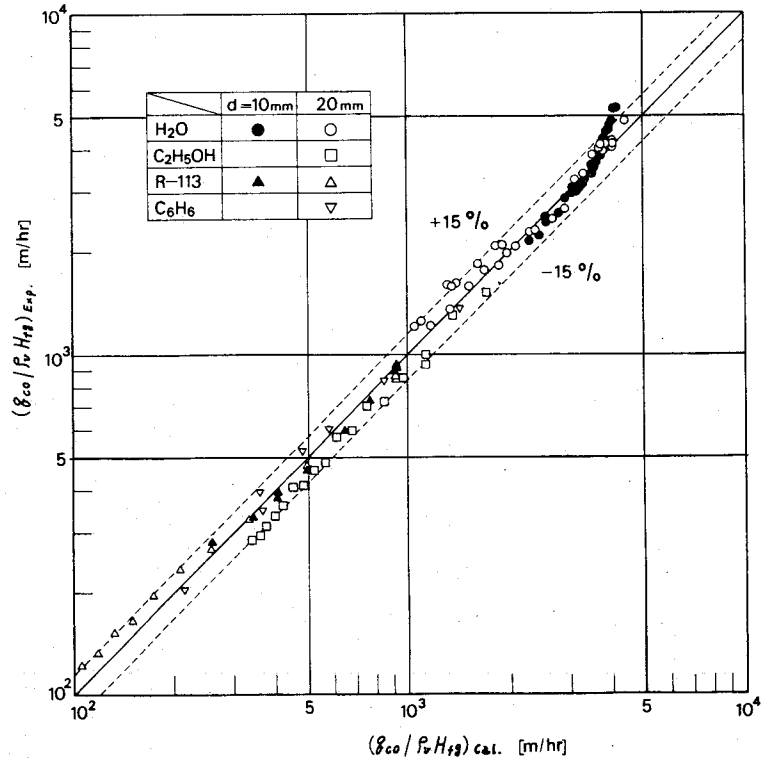

Figure 6. Comparison of $q_{co}/\rho_v H_{fg}$ between prediction of [13] and experimental data.

4. CONCLUSIONS

(i) CHF experiments in boiling at atmospheric pressure have been made for disk diameters of $d = 10$ and 20 mm, the distance between the parallel disks s in a range of $d/s = 0 \sim 120$, and four different fluids (water, R-113, ethyl alcohol and benzene). The CHF data obtained is correlated in a generalized form by [6] with an uncertainty of about $\pm 15\%$.

(ii) In employing [6], if the condition considered is in the vicinity of $d/s = 0$, it is recommended to ascertain that the magnitude of $d' = d\sqrt{(g(\rho_l - \rho_v)/\sigma)}$ is not less than, say, 6.0.

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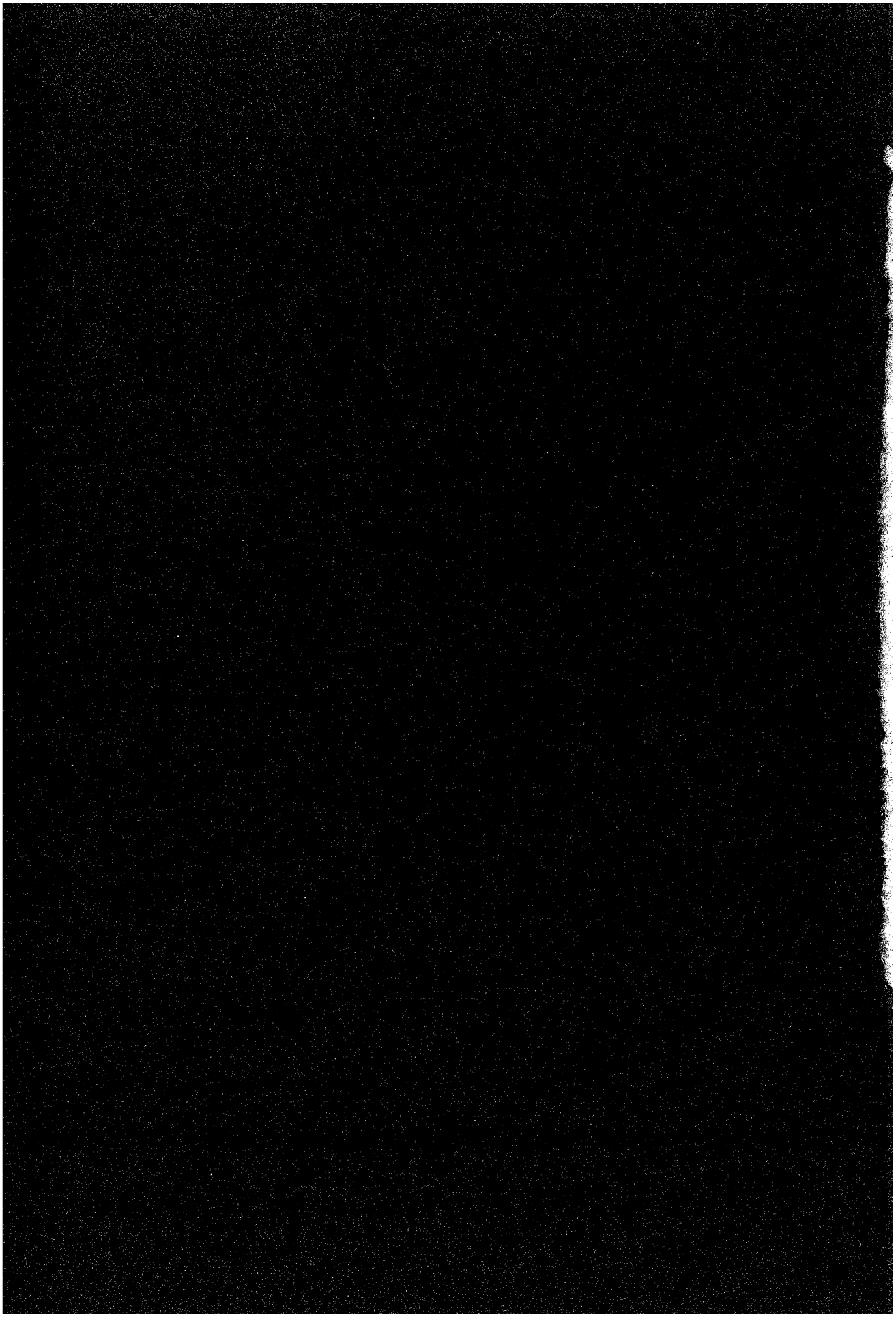
FORCED CONVECTION BOILING

EXTERNAL FLOW



CRITICAL HEAT FLUX OF SATURATED CONVECTIVE BOILING ON UNIFORMLY HEATED PLATES IN A PARALLEL FLOW

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Abstract—Employing saturated water and R-113 at atmospheric pressure, experiments are made for critical heat flux (CHF) on a uniformly heated plate of 10, 15 and 20 mm in length submerged parallel to a uniform liquid flow with velocity of 1.5–10 m/s, and the data of CHF obtained are successfully correlated by a generalized equation. In addition, it is shown that existing data of CHF for acetone, toluene, mono-isopropylbiphenyl and water flowing through internally heated annular channels of very small l/d_{he}, where l is the axial length and d_{he} the heated equivalent diameter, agree well with the above-mentioned correlation.

1. INTRODUCTION

Critical heat flux (CHF) in forced convection boiling ordinarily refers to internal flow in channels, and very few studies have been made on CHF of external flow. However, information about CHF for simple but fundamental boiling systems utilizing external flow are of importance not only for industrial applications (augmentation of heat removal from various heated surfaces with boiling, for example) but also for the potential insight into the basic mechanisms of forced convection boiling.

Among the boiling systems utilizing external flow, one of the simplest is the boiling of a thin liquid film flowing over a uniformly heated surface (exactly, a downward-facing horizontal surface without reentry of splashed liquid to the heated surface). For this type of boiling, Monde & Katto (1978), and Katto & Shimizu (1979) measured CHF for disk heaters cooled by a small round jet of saturated (and subcooled) liquid yielding generalized correlations of CHF data, while Lienhard & Eichorn (1979), and Lienhard & Hasan (1979), analyzed these data theoretically. On the other hand, Katto & Ishii (1978) determined CHF experimentally for forced flow of a plane jet of saturated liquid over a rectangular heater, and presented a generalized correlation for CHF data obtained.

Meanwhile, for forced convection boiling on a heated surface submerged in liquids of sufficiently large volume, Andrews & Mohan Rao (1974) made experiments on CHF for thin horizontal ribbons in submerged vertical liquid jets. The experiments of CHF on a cylinder normal to the liquid flow were also made by McKee & Bell (1969) and others, and recently Lienhard & Eichorn (1976) carried out a theoretical analysis along with an experimental study with saturated liquids, yielding generalized correlations of CHF data in two characteristic regimes.

On the other hand, it is noted that there is a much simpler submerged system, that is, boiling on a uniformly heated plate in a parallel flow, and to the authors' knowledge, experiments have scarcely been made for this system. In the present study, therefore, experiments are made for CHF of this boiling system with saturated water and R-113 at atmospheric pressure.

2. EXPERIMENTAL APPARATUS

Figure 1 shows the test vessel. A copper block is heated by eleven plate heaters inserted into grooves on the l.h.s. Boiling takes place on a rectangular flat surface at the right end of the copper block with a width of 10 mm and three lengths l = 10, 15 and 20 mm. Three Chromel–Alumel thermocouples embedded along the central axis of the copper block are used to

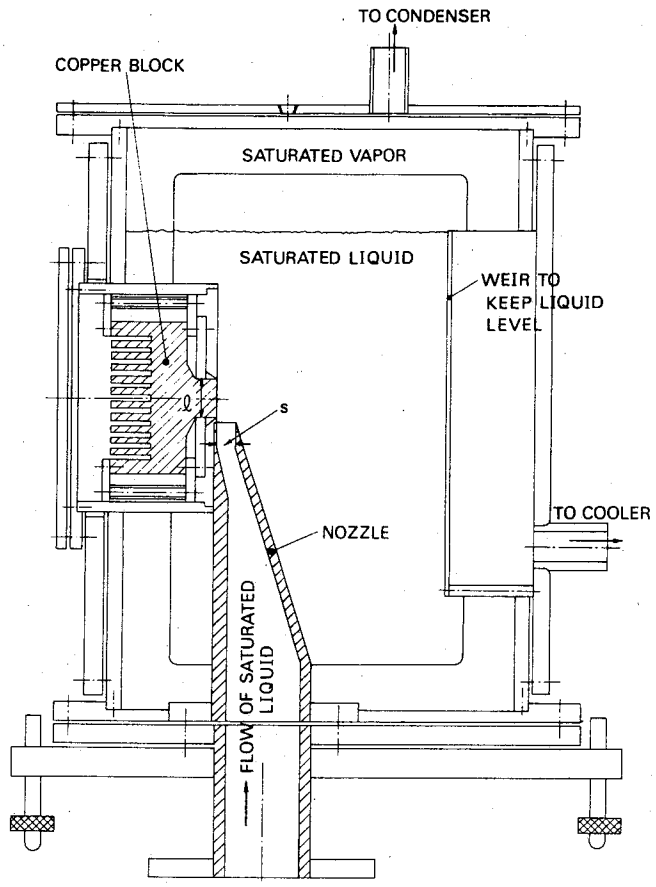

Figure 1. Test vessel.

measure the heat flux across the heated surface as well as to detect the occurrence of CHF indicated by a sharp rise in temperature.

In the test vessel, a nearly constant volume of liquid is maintained at atmospheric pressure by means of a weir on the r.h.s. An auxiliary electric heater, omitted from figure 1, is used to keep the liquid at saturation temperature. Meanwhile, practically saturated liquid with a subcooling of less than 3°C, that is fed through a volute pump, a flow meter and a preheater, flows out of the exit of a rectangular nozzle shown in figure 1 (with a width of 15 mm, and two heights $s = 10$ mm mainly and 5 mm auxiliary) to give a forced convection flow of prescribed velocity along the heated surface. According to the experimental results mentioned in the next chapter, it is found that the difference of CHF is hardly perceived for the change of s from 10 to 5 mm, so that it may be assumed that the experimental condition employed in this study is a very close approximation to that of the heated flat surface parallel to a uniform velocity flow of liquid. Finally, the cooler noted in figure 1 is necessary to remove at least the energy supplied to the system through the dissipation of pump power.

3. EXPERIMENTAL RESULTS OF CHF

3.1 *CHF and flow pattern*

Experiments were made in the range of liquid velocity $u = 1.5$–10 m/s at the nozzle exit for both water and R-113. According to still pictures taken of the two-phase flow along the heated surface, the flow pattern is bubbly when the heat flux q is low, but as q is increased, coalescence of vapor takes place near the heated surface presenting the appearance shown in figure 2(a).

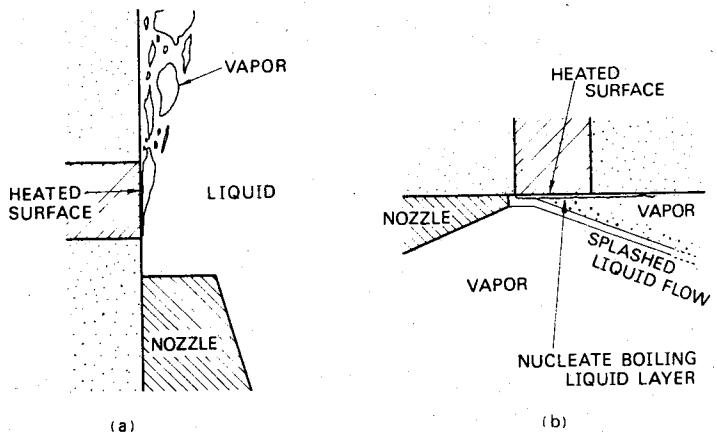

Figure 2. Flow pattern near CHF condition: (a) forced flow of liquid with uniform velocity, (b) forced flow of liquid film.

3.2 *Correlation of experimental data*

All the data of CHF obtained in this study as listed in tables 1 and 2 are plotted in the dimensionless form of q_{co}/GH_{fg} against $\sigma\rho_l/G^2l$ to give figure 3, where q_{co} is the critical heat flux for saturated liquid flow, G the mass velocity of liquid flowing out of the nozzle, H_{fg} the latent heat of evaporation, σ the surface tension, ρ_l the density of liquid, and l the length of heated surface in the direction of flow. In figure 3, data for water and R-113 appear to fall along lines having nearly the same gradient. If a distinction is attempted between the data of $s = 10$ mm and those of $s = 5$ mm in figure 3, the former group of data seem to have a slightly greater gradient than the latter. However, the difference in gradient, if any, is very small. There is not really sufficient data to analyze this effect, so the difference in gradient is neglected in this paper. Then, the situation in figure 3 is quite similar to those encountered in the previous studies (Monde & Katto 1978, Katto & Ishii 1978), suggesting the possibility of unifying the data of water and R-113 by taking into account the effect of vapor/liquid density ratio ρ_v/ρ_l; namely, if $(q_{co}/GH_{fg})/(\rho_v/\rho_l)^{0.559}$ is used instead of q_{co}/GH_{fg} for the ordinate of figure 3, two separate groups of data come together to give the result of figure 4, through which a generalized correlation of CHF in the experimental range of $\sigma\rho_l/G^2l = 10^{-5}$ to 3×10^{-3} is given as follows:

$$\frac{q_{co}}{GH_{fg}} = 0.186\left(\frac{\rho_v}{\rho_l}\right)^{0.559}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.264}. \qquad [1]$$

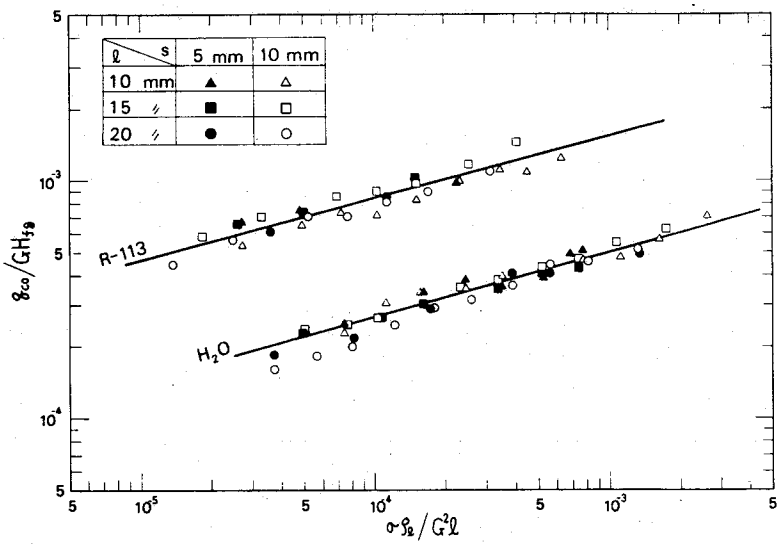

Figure 3. Experimental data of CHF.

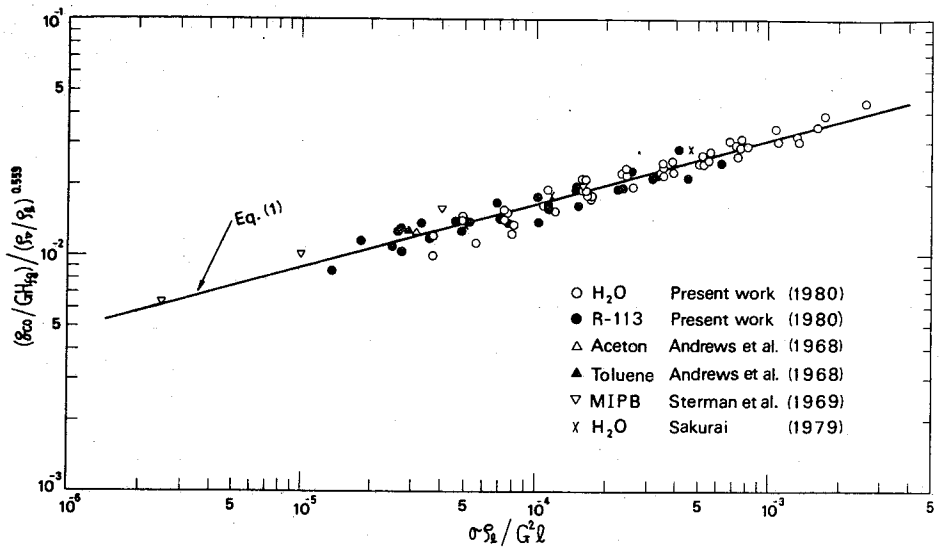

Figure 4. Generalized correlation of CHF data.

Table 1. Experimental data of CHF of water

l = 10 mm				l = 15 mm				l = 20 mm			
s = 5 mm		s = 10 mm		s = 5 mm		s = 10 mm		s = 5 mm		s = 10 mm	
u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2
2.83	3.08	1.53	2.36	2.35	2.15	1.53	2.06	1.51	1.59	1.52	1.69
3.00	3.20	1.95	2.37	2.79	2.42	1.95	2.34	2.35	2.06	1.95	1.97
3.42	2.88	2.35	2.47	3.45	2.67	2.35	2.36	2.83	2.47	2.35	2.24
4.18	3.52	2.84	2.84	5.02	3.29	2.83	2.59	4.18	2.62	2.81	2.19
5.02	4.12	3.42	2.90	6.13	3.50	3.43	2.83	6.14	2.91	3.43	2.33
6.14	4.52	4.19	3.59	9.09	4.47	4.19	3.28	9.09	3.84	4.19	2.59
9.09	4.95	5.02	3.86			5.02	3.30			5.00	2.69
		6.18	4.59			6.17	3.51			6.18	2.66
		7.37	4.90			7.37	3.97			7.37	2.91
		9.11	4.49			9.10	4.64			9.11	3.14

Table 2. Experimental data of CHF of R-113

l = 10 mm				l = 15 mm				l = 20 mm			
s = 5 mm		s = 10 mm		s = 5 mm		s = 10 mm		s = 5 mm		s = 10 mm	
u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2
2.10	0.453	1.26	0.349	2.10	0.472	1.28	0.412	2.10	0.394	1.26	0.306
4.54	0.753	1.49	0.360	3.72	0.599	1.62	0.421			1.70	0.338
6.06	0.897	1.70	0.418	5.00	0.731	2.10	0.454			2.10	0.378
		2.10	0.451			2.55	0.510			2.55	0.399
		2.55	0.456			3.10	0.591			3.10	0.491
		3.13	0.493			3.70	0.606			3.71	0.502
		3.71	0.600			4.48	0.698			4.48	0.554
		4.48	0.647			6.02	0.782			6.02	0.590
		6.02	0.713								

4. CHF FOR ANNULI OF VERY SMALL l/d_{he}

In the case of an annular channel with a uniformly heated rod, if l/d_{he} (where l is the axial length of heated rod and d_{he} the heated equivalent diameter) is extremely small, the condition of convective boiling is closer to that dealt with in this paper than that in annular channels, as pointed out by Katto (1980). From this point of view, therefore, existing data of CHF obtained for internally heated annuli of $l/d_{he} < 10$ (see Katto 1980) will be utilized below to check the applicability of [1]

Table 3. Values of ρ_v/ρ_l for CHF data

Source	Fluid	Pressure	ρ_v/ρ_l x10^2	l/d_{he}
Andrews et al.(1968)	Acetone	1 atm	0.291	1.15
Andrews et al.(1968)	Toluene	1 atm	0.385	1.15
Sterman et al.(1969)	MIPB*	2.0 bar	1.03	7.05
Sakurai (1979)	Water	1.43bar	0.0868	0.0582
Present work	Water	1 atm	0.0624	-----
Present work	R-113	1 atm	0.490	-----

* Monoisopropylbiphenyl.

For annuli of very small l/d_{he} with internal heating, experimental data of CHF have been given as a function of inlet subcooling enthalpy ΔH_i by Andrews *et al.* (1968) for acetone and toluene, and by Sterman *et al.* (1969) for monoisopropylbiphenyl (MIPB), so that the critical heat flux q_{co} for saturated inlet conditions can be estimated by the extrapolation as $\Delta H_i \to 0$. Sakurai (1979) has measured CHF for saturated water at the pressure of 1.43 bar. Table 3 lists the values of ρ_v/ρ_l along with l/d_{he} for the experiments on annuli mentioned above, where physical properties of MIPB are estimated from Core & Sato (1958) and Gambill (1957, 1958, 1959). It also lists the values of ρ_v/ρ_l for the present work. The data for CHF in annuli are shown in figure 4, revealing that it is within the scatter of the data from the present work, though there is a trend of appearing slightly higher than the prediction of [1].

5. DISCUSSIONS

(i) The design of the flow channel near the nozzel exit adopted in figure 1 to give uniform velocity at the nozzle exit, may raise question concerning the velocity component perpendicular to the heated surface. However, according to a simple analysis, the difference in dynamic pressure due to this cause is, at most, a few per cent. In addition, the CHF data for annuli, which are free from the above problem, agree fairly well with [1] as seen in figure 4, suggesting that the design of the flow channel near the nozzle exit does not seriously affect the results.

(ii) The generalized correlation of CHF determined experimentally by Katto & Ishii (1978) for the flow of thin plane jet of saturated liquid over rectangular heated surface is written as:

$$\frac{q_{co}}{GH_{fg}} = 0.0164 \left(\frac{\rho_v}{\rho_l}\right)^{0.133} \left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3} \qquad [2]$$

Comparing this equation with [1] reveals that the exponent of the reciprocal Weber number $\sigma\rho_l/G^2 l$ is nearly the same between the two equations with only a difference of 20 per cent, whereas the difference in exponent of density ratio ρ_v/ρ_l is considerable. In other words, the effect of ρ_v/ρ_l on CHF is much greater in the case of a submerged heated surface than in the case of thin liquid film flow. Meanwhile, if the flow pattern in the former case such as shown in figure 2(a) is compared with that in the latter case such as shown in figure 2(b), which is reproduced from the paper of Katto & Ishii (1978), it may suggest that the escape of generated vapor from the heated surface is much more difficult in the former case due to the interference of a greater mass of liquid. Thus it may be natural to suppose that the high value of exponent of ρ_v/ρ_l in [1] relates mainly to the flow pattern near the heated surface submerged in liquid.

(iii) For CHF on a uniformly heated cylindrical surface in a cross flow, Lienhard & Eichorn (1976) gave generalized correlation equations valid in two regimes, namely the low-velocity and high velocity regimes. According to their observations, both regimes are characterized by two-dimensional flow of escaping vapor. If the present nomenclature is employed, their

equations can be written as follows:

For the low-velocity regime,

$$\frac{q_{co}}{GH_{fg}} = \frac{1}{\pi}\left\{\frac{\rho_v}{\rho_l} + 4^{1/3}\left(\frac{\rho_v}{\rho_l}\right)^{2/3}\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3}\right\} \quad [3]$$

and for the high-velocity regime,

$$\frac{q_{co}}{GH_{fg}} = \frac{1}{\pi}\left\{\frac{1}{169}\left(\frac{\rho_v}{\rho_l}\right)^{1/4} + \frac{1}{18.2}\left(\frac{\rho_v}{\rho_l}\right)^{1/6}\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3}\right\} \quad [4]$$

where D is the diameter of cylinder.

Both [3] and [4] are, however, too complicated in form for the purpose of comparison with [1] and [2]. Therefore, apart from the theoretical analysis of CHF made by Lienhard and Eichorn, [3] and [4] will be replaced by the following empirical equations only for convenience. According to figure 8 in the paper of Lienhard & Eichorn (1976), [3] may be replaced, keeping the same accuracy, by

$$\frac{q_{co}}{GH_{fg}} = 0.530\left(\frac{\rho_v}{\rho_l}\right)^{2/3}\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3} \quad [3']$$

and from figure 7 in the same paper, suggesting a trend that $q_{co}/GH_{fg} \to 0$ as $\sigma\rho_l/G^2l \to 0$, it is found that [4] may be replaced by

$$\frac{q_{co}}{GH_{fg}} = 0.0202\left(\frac{\rho_v}{\rho_l}\right)^{1/6}\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3}. \quad [4']$$

Equations [3'] and [4'] are very close to the respective equations obtained by ignoring the first terms on the r.h.s. of [3] and [4]. In addition, taking into account the results of [3'] and [4'] as well we the criterion assumed by Lienhard and Eichorn for the transition between the low-velocity and high-velocity regimes, it is concluded that [3'] applies to the low-velocity range designated by

$$\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3} > \frac{1}{0.530\pi}\left\{\left(\frac{\rho_v}{\rho_l}\right)^{1/3} + 0.275\left(\frac{\rho_v}{\rho_l}\right)^{-1/6}\right\} \quad [5]$$

and [4'] applies to the high-velocity range designated by

$$\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3} < \frac{1}{0.0202\pi}\left\{\left(\frac{\rho_v}{\rho_l}\right)^{5/6} + 0.275\left(\frac{\rho_v}{\rho_l}\right)^{1/3}\right\}. \quad [6]$$

Now, [3'] and [4'] can be compared with [1] and [2] to reveal the interesting fact that [3'] for CHF on cylinders in the low-velocity regime bears similarities to [1], whereas [4'] for CHF on cylinders in the high-velocity regime is very nearly related to [2]. Thus, in the case of low-velocity, the vapor removel process from the back surface as well as from the front surface of cylinder may be supposed to remain the type of froth flow shown in figure 2(a). For reference, the presumption of flow pattern in this case is shown in figure 5(a). On the other hand, in the case of high-velocity, 2-dimensional vapor sheet develops behind the cylinder as shown in figure 5(b), so that the back surface of cylinder is covered with a nucleate boiling liquid layer in the similar way as in figure 2(b), and the onset of CHF condition is assumed to

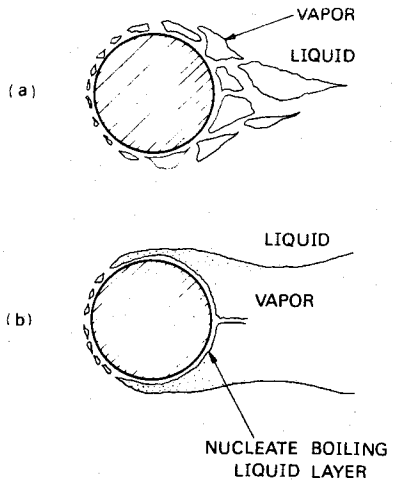

Figure 5. Presumption of flow pattern near CHF condition for nucleate boiling on a heated cylindrical surface in a cross flow: (a) low-velocity regime, (b) high-velocity regime.

occur first at the rear stagnation line. It has been observed by Katto & Ishii (1978) that in case of figure 2(b), the CHF condition occurs through the breakdown of the nucleate boiling liquid layer at the rear edge of heated surface.

5. CONCLUSIONS

Equation [1] has been obtained as a generalized correlation of CHF of saturated convective boiling on heated surfaces submerged parallel to a liquid flow of uniform velocity. Taking into account the data for internally heated annular channels of very small l/d_{he} as well as the data obtained in this study, the experimental range of ρ_v/ρ_l and $\sigma\rho_l/G^2l$ for which [1] is valid is as follows: $\rho_v/\rho_l = 0.000624$–0.0103, and $\sigma\rho_l/G^2l = 2\times10^{-6}$–$3\times10^{-3}$.

Then, on the basis of two typical flow patterns shown in figure 2, presumative discussions are made on the relation between [1] for heated plates in a parallel flow, [2] for heated plates covered with a liquid film flow, and [3′] and [4′] for heated cylinders in a cross flow.

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Study of the Mechanism of Burn-Out in Boiling System of High Burn-Out Heat Flux*

By Yoshiro KATTO** and Masatsugu KUNIHIRO***

Burn-out phenomena are studied for two special systems of boiling which are both realized in the saturated nucleate boiling of water at atmospheric pressure on a horizontal 10mm dia. copper heated surface.

First, it is attempted by mechanical means to cause a forced collapse of the vapor masses which are generated periodically on the heated surface. In this case, burn-out heat flux q_c increases, but the velocity of fluid movement concerned with the collapse of vapor mass is restricted so that the increase of q_c is limited.

Second, the heated surface is supplied with liquid by means of a small liquid jet. In this case, burn-out appears with a mechanism which is different from ordinary pool boiling. Splashing of liquid due to the violent effusion of vapor from the heated surface interrupts the part of supplied liquid in reaching the heated surface so that the quantity of liquid on the heated surface balances finally the heat flux, and burn-out occurs. For the burn-out of this type, q_c can be increased steadily by increasing the velocity of liquid jet.

1. Introduction

According to Katto and Yokoya[1] [2] [3] [4] who carried out the experimental study of burn-out and transition boiling, the burn-out in pool boiling is a phenomenon related to the consumption of liquid within a thin nucleate-boiling-liquid-layer maintained on a heated surface and the periodical behavior of a vapor mass which grows until it detatches from the nucleate-boiling-liquid-layer. The period necessary for complete drying of a nucleate-boiling-liquid-layer decreases with an increase of heat flux, and it becomes finally equal to the detatchment period of vapor mass when burn-out occurs, because the supply of liquid to the heated surface is performed intermittently only through the detatchment of the vapor mass which covers the liquid-layer, preventing the access of the surrounding liquid to the heated surface.

The burn-out in ordinary pool boiling is, therefore, a limit problem which arises out of the balance between the supply and the consumption of liquid on a heated surface which is subject to the existence of vapor masses, that is to say, the burn-out is never a phenomenon inherent in the nucleate boiling itself which is maintained on the heated surface. Such being the case, the artificial control of the behavior of vapor mass as well as the artificial supply of liquid to the heated surface may make it possible to realize the extension of the region of nucleate boiling, raising the burn-out heat flux up to a much higher level. In addition, the experimental study of such special boiling systems as afore-mentioned may be of use not only for yielding further information on the nucleate boiling and burn-out but also for serving as an aid to clarification of the relation between the burn-out in pool boiling and that in forced convection boiling.

The present paper reports the experimental study which has been done from the afore-mentioned view-point, and it is twofold in its scope. The first is concerned with the effect of the control of behavior of vapor mass, and the second with the effect of the control of liquid supply to the heated surface. In each case, such a way has been adopted as to give the effect of artificial control alone by taking care so as to maintain the ordinary state of nucleate boiling near the heated surface as far as possible.

* Received 14th September, 1971.

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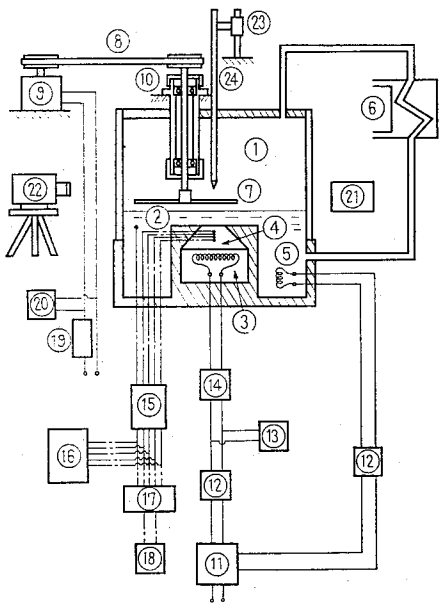
2. Effect of control of behavior of vapor mass

2·1 Experimental apparatus

Experimental apparatus is shown schematically in Fig. 1. The upper surface of a copper block ② (conical shape in upper part and cylindrical shape in lower part) located at the central position of a boiler ① discharges the function of a heated surface of 10mm-dia. The heated surface is polished with emery paper (#0) and then cleaned with acetone and pure water in every experiment. The electric heater ③, which is composed of eleven heating elements of plate type mounted in the grooves of the lower part of copper block, is capable of providing the maximum heat flux of 2×10^7 kcal/m²hr across the upper round heated surface of 10mm-dia. Temperature of the heated surface T_W as well as heat flux q is determined through the temperature distribution which is measured with four copper-constantan thermo-couples ④ set up along the axis of copper block.



① Boiler ② Copper block ③ Electric heater ④ Thermo-couple ⑤ Auxiliary heater ⑥ Cooler ⑦ Rotary cross of rods ⑧ Belt ⑨ Direct current motor ⑩ Bearing box ⑪ Voltage stabilizing apparatus ⑫ Auto-transformer ⑬ Alternating current voltmeter ⑭ Wattmeter ⑮ Ice box ⑯ Recording Thermometer ⑰ Change-over switch ⑱ Millivolt meter ⑲ Sliding resistor ⑳ Direct current voltmeter ㉑ Lighting equipment ㉒ High-speed camera ㉓ Fine adjustment screw ㉔ Water-level measuring needle

Fig. 1 Experimental apparatus (Control of behavior of vapor mass)

Experiment is made on the saturated boiling of pure water at atmospheric pressure, and the saturated condition is kept in the boiler ① with the aid of an auxiliary heater ⑤. Steam generated in the boiler is condensed by a cooler ⑥ and then returned to the boiler to keep a constant water level. The steam, which is generated on the heated surface in the nucleate boiling at high heat fluxes, is accumulated in the preliminary experiment in order to determine the quantity of heat transferred from the heated surface, and by that means the thermal conductivity of copper block is determined. The heat flux determined from the latent heat of generated steam has been found to be nearly equal to the input of the electric heater in the copper block, and the heat loss is only within 5% of the input.

The rotary cross of rods ⑦ in Fig. 1, which is prepared to break vapor masses mechanically, is a cross formed by two metal rods of diameter 2mm and length 50mm. Consequently, four metal rods pass successively over the heated surface in each revolution of the rotary cross of rods, and the revolution speed is regulated by the sliding resistor ⑲. The axle of rotary cross of rods is suspended vertically by the bearing box ⑩, and the distance between the rotary cross of rods and the heated surface can be changed by vertical transfer of the bearing box. Although the clearance between the hole in bearing box and the penetrating axle is sealed by O-ring to prevent the inflow of steam, lubricating oil is not supplied in the least to ball bearings in the bearing box so as to preclude the possibility of the leakage of oil into the boiler.

2·2 Characteristics of experiment

Experiment is made under the condition that the rotary cross of rods is placed above the free surface of water as shown in Fig. 1 and the water level H measured from the heated surface to the free surface of water is kept as low as $H=5$mm and 3mm. Consequently, the rotary cross of rods exerts the force to break vapor masses without any forced stirring of bulk water. In the nucleate boiling of low water levels, a vapor mass which is enclosed with a thin liquid film grows periodically on the heated surface [cf. Fig. 8 in reference (2)]. Due to the surface tension of liquid film, the pressure within the vapor mass is generally a little higher than the atmospheric pressure. Top of the rising liquid film begins to break when it touches the moving metal rod, and then the vapor mass begins to collapse, effusing

vapor. The effusion of vapor which follows the commencement of destruction of liquid film is clearly observed by means of high-speed photography.

As to the boiling curve of nucleate boiling of low water levels, it has been found by Katto and Yokoya [2] [3] that the boiling curve in the region of high heat fluxes (above $q \fallingdotseq 5 \times 10^5$ kcal/m²hr) shows little variation for the water levels above $H=2$mm in the case of saturated boiling of water on the heated surface of 10-mm dia. at the atmospheric pressure. This phenomenon of invariability of boiling curve is ascribed to the fact that the boiling takes place within a very thin nucleate-boiling-liquid-layer which spreads over the heated surface. In addition, no principal difference is generated by the water level in the mechanism of burn-out too.

In the experimental apparatus of Fig. 1, the bearing box ⑩ is fixed, being separated from the boiler ①. However, the clearance between the bearing box and the wall of boiler should be sealed to prevent the leakage of steam, so that the vibration of bearing box is inevitably transferred to the boiler to some extent. As to this problem, however, it has been found in the preliminary experiment that the boiling curve as well as the burn-out heat flux does not differ between the revolving state and the static state of axle of rotary cross of rods.

2·3 Experimental result

Figure 2 shows the variation of burn-out heat flux q_c with revolution speed of rotary cross of rods n on condition that the water level measured from the heated surface to the free surface of water H as well as the distance between the rotary cross of rods and the heated surface s (the rotary cross of rods is located always above the free water surface so that $H<s$) is fixed. The ordinate of Fig. 2 indicates the ratio q_c/q_{co}

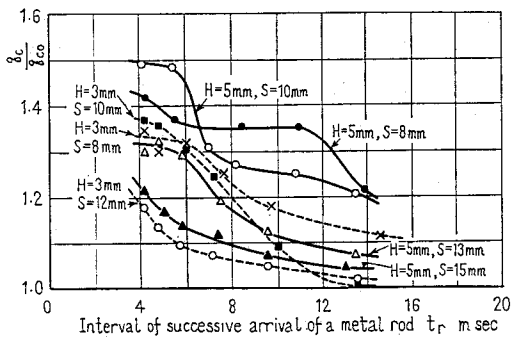


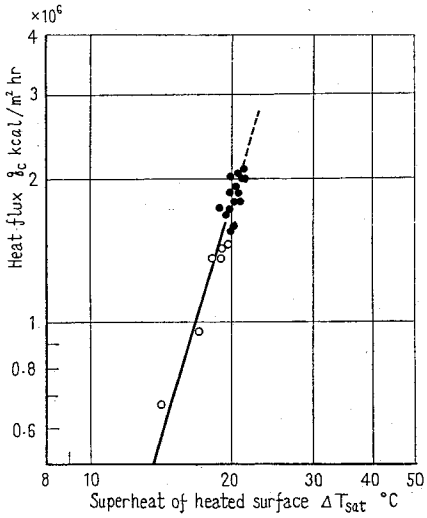
Fig. 2 Variation of burn-out heat flux q_c

where q_{co} is the burn-out heat flux measured under the condition of $n=0$ for the same water-level and heated surface, and the abscissa indicates the interval of successive arrival of a metal rod at the center of heated surface t_r that is a quarter of the period of one revolution.

Figure 2 shows clearly the fact that as compared with the state of free boiling, the burn-out heat flux is increased by the revolution of rotary cross of rods. This increase of burn-out heat flux is accompanied with an extension of nucleate boiling curve toward the region of higher heat fluxes, and it may be noticed in Fig. 3, which shows the case of $H=5$mm and $s=8$ mm as an example, that the nucleate boiling curve in the state of free boiling extends monotonously toward the higher heat flux region.

2·4 Discussion

The data in Fig. 2 shows the general trend that (1) when the distance between the rotary cross of rods and the heated surface s is small, (2) when the interval of successive arrival of a metal rod t_r is short, and (3) when the water level H is high, the burn-out heat flux becomes high. If s and/or t_r are decreased, the duration period of vapor mass becomes shorter, increasing the frequency of liquid supply to the heated surface, and it permits the consumption of liquid to take place in the nucleate-boiling-liquid-layer with higher speed, that is to say, the burn-out heat flux increases. In addition, if H is increased, it may strengthen the flow of liquid to the heated



(Mark of ○: state of free boiling, and mark of ●: extension of the region of nucleate boiling)

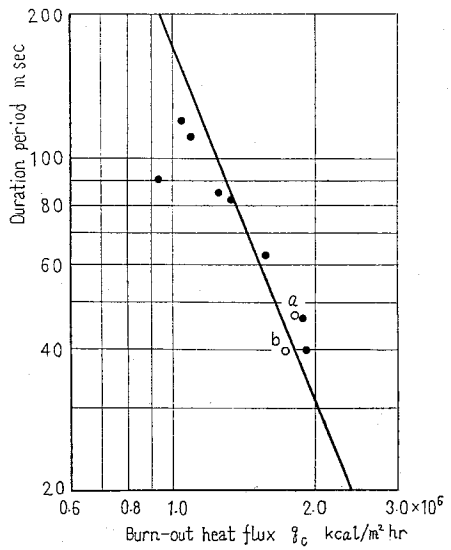
Fig. 3 Boiling curve in the region of nucleate boiling

surface during the period of liquid supply.

As to the mechanism of burn-out, the state afore-mentioned does not differ from the case of ordinary pool boiling except the effect of water level H. As a matter of fact, the variation of burn-out heat flux with duration period of a vapor mass is shown in Fig. 4, where ● indicates the experimental data[2] [3] obtained in the pool boilings of various types and ○ the data in the present experiment. It may be noticed that all data can be correlated approximately within a systematic trend. However, we must not forget the important fact that the increase of burn-out heat flux q_c by means of the present mechanical method is limited to the order of 1.5 times as great as the burn-out heat flux in free boiling. In connection with this problem, two matters should be pointed out as follows.

First, however reduced the interval of successive arrival of a metal rod t_r may be, the distance between the rotary cross of rods and the heated surface s which cannot be smaller than the water level H governs finally the growth period of vapor mass. If H is reduced in order to reduce s, it is accompanied with a disadvantage of a weakened flow of liquid to the heated surface during the period of liquid supply.

Second, the collapse of vapor mass is not accomplished instantaneously when a metal rod touches the top of the growing vapor mass and



Experiment a: H=5mm, s=12mm, t_r=7.15msec
Experiment b: H=3mm, s=10mm, t_r=6.82msec

Fig. 4 Relation between the duration period of a vapor mass and the burn-out heat flux

breaks the liquid film there. Beginning with the effusion of vapor such as mentioned in section 2·2, the collapse of vapor mass advances gradually with various accompanying movements of vapor and liquid, and this is nothing but a phenomenon independent of the rotation of rotary cross of rods since the behavior of fluid has already been left to take its own natural course. This state can be obviously observed by means of high-speed photography. As to the experiments of a and b both indicated by the mark of ○ in Fig. 4, it should be mentioned here that the duration period of a vapor mass is about 40 to 50 msec while the interval of successive arrival of a metal rod t_r is only about 7 msec. After all, if the interval of t_r is reduced, the follow-up speed of fluid behavior is limited and it governs the limitation of burn-out.

In sum, the experiment described in the present chapter is not adequate for the aim to realize a great ascent of burn-out heat flux. The increase of burn-out heat flux is certainly obtained by the artificial destruction of vapor mass, but the rupture of a vapor mass is connected with the behavior of fluid so that the control of vapor mass alone is useless for the great ascent of burn-out heat flux. Accordingly, the forced supply of liquid to the heated surface will be considered next for the afore-mentioned object.

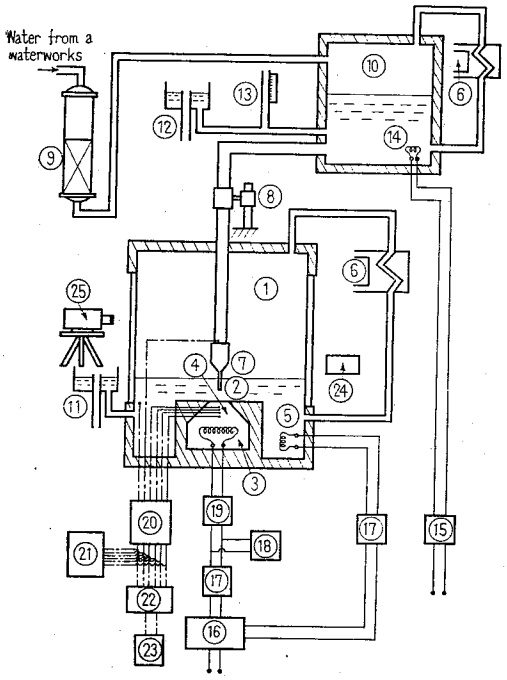
3. Effect of control of liquid supply

3·1 Experimental apparatus

Similarly to the experiment described in the preceding chapter, an experiment is made on the saturated boiling of pure water at atmospheric pressure. Experimental apparatus is shown in Fig. 5, which is particularly different from Fig. 1 in that an equipment to keep a water jet is set up instead of the rotary cross of rods. A nozzle ⑦ is mounted perpendicularly above the center of the round heated surface. The nozzle is made of stainless steal tube of 0.71, 1.165 and 1.60mm in inner diameter, and the distance between the exit of nozzle and the heated surface can be varied by means of the fine adjustment screw ⑧. Saturated pure water, which makes a water jet at the exit of nozzle, is supplied from the boiling water tank ⑩. The duct connecting the boiling water tank with the nozzle is insulated with glass wool so that the water spurting from the nozzle has an exit temperature higher than 97°C even in the worst case. It might be sufficient, therefore, to regard the water approximately to be the saturated water for the purpose of present experiment.

In order to keep the water level in the boiler ① constant, water is drained through an over-flow device ⑪. The saturated condition inside the boiler is secured by means of an auxiliary heater ⑤.

The boiling water tank ⑩ has also an over-flow device ⑫, and thereby the water level in the tank is kept constant. The water jet from the nozzle is maintained by the gravitational difference of water head between the level at the water surface in the tank and the level at the exit of nozzle. Measurement of the flow rate of water jet is performed as follows : under the condition that the supply of water to the tank ⑩ is stopped for a moment and no drainage from the over-flow device ⑫ is confirmed, the descending velocity of the water level in the tank is measured by the water-level gauge ⑬. In this case, the descend of water level in the tank is confined to a range so small as compared with the total head difference between the tank and the nozzle exit, and therefore, the variation of the flow rate of water jet due to the descend of water level in the tank can be neglected.



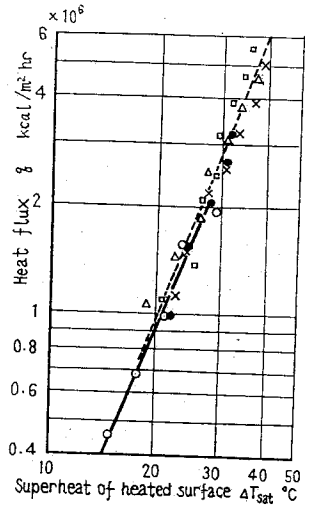
① Boiler ② Copper block ③ Electric heater ④ Thermo-couple ⑤ Auxiliary heater ⑥ Cooler ⑦ Nozzle ⑧ Fine adjustment screw ⑨ Ion exchange resin ⑩ Boiling water tank ⑪ Over-flow device ⑫ Over-flow device ⑬ Water-level gauge ⑭ Electric heater ⑮ Auto-transformer ⑯ Voltage stabilizing apparatus ⑰ Auto-transformer ⑱ Alternating current voltmeter ⑲ Wattmeter ⑳ Ice box ㉑ Recording thermometer ㉒ Change-over switch ㉓ Millivolt meter ㉔ Lighting equipment ㉕ High-speed camera

Fig. 5 Experimental apparatus (Control of supply of liquid)

3·2 Characteristics of boiling curve and others

Experiment is made by altering the water level measured from the heated surface to the free water surface H, the distance between the nozzle exit and the heated surface h, the flow rate of water jet G, and the mean velocity of water jet at the nozzle exit u_e.

In the experiment afore-mentioned, the heated surface is exposed to the forced flow of liquid. However, the violent movement of fluid generated within the nucleate-boiling-liquid-layer on the heated surface governs the heat transfer in the nucleate boiling of such high heat flux as in the present case. The fact, which has been described in section 2·2, that the boiling curve shows little variation for the water levels above $H=2$mm, depends on the same reason. Accordingly, it may be expected that the forced convection can have little effect on the heat transfer for such a low level of velocity as that of a natural falling jet of water. Figure 6 is a typical example of the experiments carried out under various conditions in order to confirm the expectation afore-mentioned. The data indicated by the mark of ○ correspond



○ State of free boiling (H=5mm)
△ : H=5mm, h=3mm, G=58.0g/min, u_e=2.44m/sec
□ : H=5mm, h=3mm, G=169g/min, u_e=2.64m/sec
× : H=5mm, h=3mm, G=246g/min, u_e=2.04m/sec
● : H=0mm, h=3mm, G=62.4g/min, u_e=2.63m/sec

Fig. 6 Boiling curve in the region of nucleate boiling

to the state of free nucleate boiling without a water jet, while all other data correspond to the state of nucleate boiling with a water jet including the data of a special state of $H=0$ mm indicated by the mark of ●. It may be concluded from Fig. 6 that the water jet gives no significant effect on the nucleate boiling curve.

It may be also noticed in Fig. 6 that the nucleate boiling curve in the region of heat flux q (which extends up to $q=6\times10^6$ kcal/m²hr in Fig. 6) higher than the burn-out heat flux of the ordinary free boiling takes a form of monotonous extension of the boiling curve in free nucleate boiling. This characteristic is the same as that in Fig. 3, but the experimental data in the present case cover a range of nucleate boiling extending to much higher heat fluxes, and it may be assumed that no significant variation appears in the mechanism of nucleate boiling up to such high fluxes.

It may not be useless to give here an outline of the behavior of vapor and liquid in the boiling with a small impinging jet at the center of heated surface. In the special case of zero water level ($H=0$mm), where the water of jet alone is supplied to the heated surface without any surrounding bulk water, no vapor mass appears. In the case of $H>0$, however, vapor masses are produced successively on the heated surface by the accumulation of the vapor generated on the heated surface in the same manner as in the state of ordinary free boiling. Roughly speaking, the form and cycle of vapor mass are of the same character as in the case of free boiling.

3·3 Experimental result

Figures 7, 8 and 9 show the burn-out heat flux q_c measured for the case of water level of $H=0$mm, 5mm and 30mm respectively. The marks which are used to indicate data in these figures are classified, as shown in Table 1, by the distance between the nozzle exit and the heated surface h as well as the flow rate of water jet G.

Figure 7 corresponds to the case of $H=0$mm. The water jet which comes out of the nozzle reaches the heated surface without any obstruction in this case, so that all data are correlated by a straight line independently of the distance between the nozzle exit and the heated surface h. It may be also noticed in Fig. 7 that the burn-out is generally governed by the velocity of water jet u_e. The data enclosed by a broken curve in Fig. 7 do not follow exceptionally the correlation afore-mentioned, but they belong to the state of the lowest flow rate of $G=60$ g/min, that is to say, the ordinary value of burn-out heat flux cannot be reached owing to the insufficiency of the

Table 1 Marks used for indicating data in Fig. 7, 8 and 9 with their experimental conditions

	G≒60 g/min	G≒150 g/min	G≒180 g/min	G≒270 g/min
h= 1 mm	○	△	□	◇
h= 3 mm	◑	▲	◨	◆
h=10 mm	●	▲	■	◆
h=30 mm	◇	△	◇	◇

(Notes) Flow rate of jet G is not a primary controlling factor for the burn-out heat flux, so that the troublesome to show every individual data of G is avoided by lumping the data which are in the neighbourhood of the value mentioned in Table 1 under four groups.

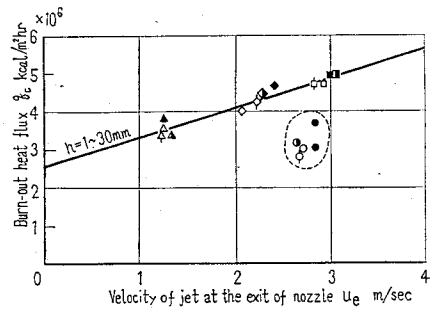


Fig. 7 Burn-out heat flux (Case of $H=0$mm)

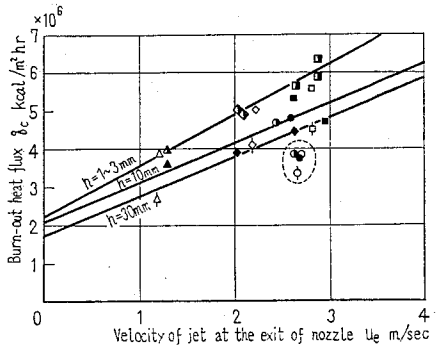


Fig. 8 Burn-out heat flux (Case of $H=5$mm)

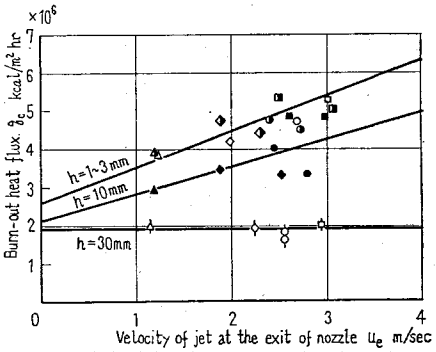


Fig. 9 Burn-out heat flux (Case of $H=30$mm)

supply of water.

Figure 8 corresponds to the case of $H=5$mm that accompanies the periodical generation of vapor mass. When the distance between the nozzle exit and the heated surface h is about 1 to 3mm, the nozzle exit is always kept within the vapor mass, yielding little obstruction to the flow of water jet; but as h increases, the liquid film enclosing the vapor mass begins to interfere with the flow of water jet. The data of $G=60$ g/min enclosed by a broken curve in Fig. 8 belong to a special case deviating from the ordinary trend.

Figure 9 corresponds to the case of $H=30$mm that has a thick layer of liquid on the heated surface. Accordingly, it is presumed that when the distance between the nozzle exit and the heated surface is $h=30$mm, the water jet cannot reach the heated surface owing to the great obstruction of thick liquid layer so that the burn-out heat fluxes are kept at the same value as that of the free boiling (which is about 2×10^6 kcal/m²hr for the present experimental apparatus).

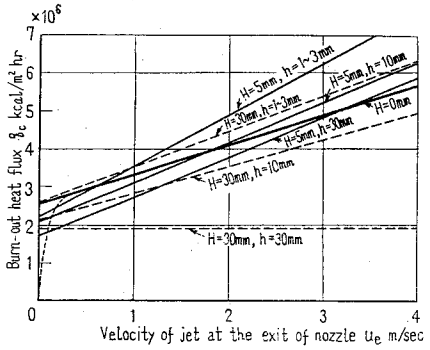


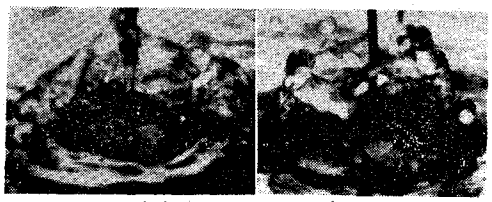
Fig. 10 Burn-out heat flux (Summary of the results obtained in Figs. 7, 8 and 9)



(a) (b) (c)

(Case of $H=0$mm, $h=3$mm, $G=195$ g/min, $u_e=3.04$m/sec)
(a) $q=1.04\times10^6$kcal/m²hr (b) $q=1.87\times10^6$kcal/m²hr
(c) $q=4.02\times10^6$kcal/m²hr

Fig. 11 Variation of the fluid behaviors on the heated surface with the ascent of heat flux



(a) (b)

(a) $H=5$mm, $h=10$mm, $G=185$g/min, $u_e=2.88$m/sec, $q=2.60\times10^6$kcal/m²hr
(b) $H=3$mm, $h=3$mm, $G=64.2$g/min, $u_e=2.71$m/sec, $q=4.16\times10^6$kcal/m²hr

Fig. 12 An instantaneous state of fluid behavior on the heated surface

3·4 Mechanism of burn-out

The correlation lines of experimental data in Figs. 7, 8 and 9 are collected in Fig. 10 in order to make it easy to compare them with each other. Roughly speaking, it may be mentioned that every correlation line in Fig. 10 displays the burn-out heat flux of nearly the same value at the state of $u_e=0$; and it may be concluded that this is a natural result except the case of $H=0$mm, because $u_e=0$ corresponds to the state of free boiling for the case of $H>2$mm where the burn-out heat flux shows little variation for the water levels as mentioned in sections 2·2 and 3·2. In the case of $H=0$mm, however, there is no state of free boiling so that the trend such as shown by a dotted curve in Fig. 10 may be presumed in the vicinity of the state of $u_e=0$.

Now, it may be assumed that the correlation line in the case of $H=0$mm, having no connection with the state of free boiling (the effect of vapor mass etc.), corresponds to the new mechanism of burn-out that is inherent in the boiling with a water jet. In addition, the correlation lines of $H=5$mm and 30mm in Fig. 10, except in the meaningless case of $H=30$mm with $h=30$mm, appear in the neighborhood of the correlation line in the case of $H=0$mm afore-mentioned, and it suggests that the burn-out in these cases is also governed primarily by the new burn-out mechanism concerned with the water jet.

Relating to the new mechanism of burn-out afore-mentioned, an example of the optical observation made of fluid behaviors near the heated surface which is subject to the nucleate boiling is shown in Fig. 11, that is the case of $H=0$mm with the jet velocity of $u_e=3.04$m/sec. When the heat flux is not so high, a radial flow of water is observed almost over the heated surface as shown in Fig. 11 (a) and the formation of vapor bubbles on the heated surface is not confirmed with the naked eye at least. As the heat flux is increased, however, the formation of vapor bubbles takes place noticeably all over the heated surface as

shown in Fig. 11 (b), and accordingly the heated surface comes to look white to the naked eye. In this case, it is noticed from Fig. 11 (b) that the movement of the vapor which escapes upward breaking through the nucleate-boiling-liquid-layer on the heated surface is accompanied with a splashing of droplets of water. It is observed through all the experiments carried out in the present study that the splashing of droplets of water commences apparently at the heat flux of about 1.5×10^6kcal/m²hr (whose corresponding superficial vapor velocity from the heated surface is 1.3m/sec).

As the heat flux is increased further, it generates such a state as shown in Fig. 11 (c) that a great portion of water has already been splashed violently near the central place where the water jet is impinging upon the heated surface since the effusion of vapor from the heated surface is increased proportionately with the heat flux. Then a substantial quantity of water supplied to the heated surface is decreased, and finally, the possible limit of the balance between the supply and consumption of water appears presumably at the circumference of the heated surface. When this state is reached, the burn-out occurs with the distinctive observation that the white part (that is the nucleate-boiling-liquid-layer) disappears from the heated surface and the ground color of copper on the dried out surface bursts upon our sight.

As to the case of $H>0$mm, Fig. 12 shows an instantaneous state of fluid for the cases of $H=5$ mm and $H=3$mm respectively. The variation in the state of fluid on the heated surface with heat flux is found to be nearly the same as that in the case of $H=0$mm afore-mentioned except the formation of such a vapor mass enclosed by a liquid film as shown in Fig. 12.

In sum, the mechanism of burn-out in the case of supplying liquid to the heated surface by a liquid jet is presumed on the basis of observation of fluid behavior on the heated surface to be as follows: the splashing of liquid due to the blowing up of the generated vapor from the heated surface decreases the substantial supplement of liquid on the heated surface up to the limit of just balancing with the consumption of liquid in boiling.

The mechanism of burn-out afore-mentioned is not contradictory to the fact that the burn-out heat flux is correlated as the function of the jet velocity u_e as shown in Fig. 10. It should be mentioned here that the attempt to correlate the burn-out heat flux as the function of the flow rate of jet G has been abandoned because it ends in failure. Under the condition that the blowing up of vapor offers an obstruction to the flow of liquid, the power of liquid to reach the heated surface can be made stronger by increasing the velocity of liquid jet and thereby the burn-out heat flux is raised.

3·5 Additional remark

Mist cooling employing the impingement of droplets upon the heated surface has an appearance similar to the system of liquid jet dealt with in the present study with respect to the forced supply of liquid to the heated surface. Essentially, however, they are under quite different conditions so that we must discriminate between them. In the state of mist cooling, that the droplets of speeds as high as 50m/sec impinge upon the heated surface, nucleate boiling is presumably apt to be suppressed due to the action of dynamic pressure. In addition, the behavior of fluid near the heated surface is affected by the forced control of impinging movement of droplets producing a special state which deviates considerably from the ordinary nucleate boiling.

As to the mist cooling with water droplets of saturation temperature at atmospheric pressure, four data of burn-out heat flux q_c can be found fortunately in the experimental results of Toda[5]. Making use of these experimental data of q_c (mean diameters of droplets of 87.7 to 146μ and impinging velocity range from 42.3 to 72.4m/sec) leads to the presumption that the impinging velocity of droplet should be as high as 100~150m/sec in order to attain the burn-out heat flux of $q_c=6\times10^6$kcal/m²hr in mist cooling. In the case of water jet (cf. Fig. 6), the same burn-out heat flux can be obtained with a surprisingly low velocity less than 3m/sec.

4. Conclusions

Under the condition of preserving the state of ordinary nucleate boiling near the heated surface as far as possible, experimental study of burn-out phenomena has been carried out on two cases: one that the vapor mass generated on a nucleate-boiling-liquid-layer is destroyed by mechanical means, and another that liquid is supplied to the nucleate-boiling-liquid-layer by a small impinging jet, and the following conclusions have been obtained.

(1) In the case of destroying the vapor mass by mechanical means, the burn-out appears with the same mechanism as that in ordinary pool boiling, but the heat flux of burn-out is

raised due to the reduction of obstacle to the supply of liquid to the heated surface. However, the collapse of vapor mass cannot be accomplished instantly and the burn-out is controlled finally by the natural behavior of fluid connected with the collapse of vapor mass giving a limit to the ascent of burn-out heat flux,

(2) In the case of supplying liquid to the nucleate-boiling-liquid-layer by an impinging jet located near the heated surface, the obstacle to the supply of liquid caused by the vapor mass is out of the question. The burn-out appears in this case with the new mechanism that the splashing of liquid due to the blowing up of the generated vapor from the heated surface decreases the supplement of liquid on the heated surface up to the limit of just balancing with the consumption of liquid in boiling.

(3) The obstacle to the supply of liquid owing to the blowing up of vapor, which is mentioned in the preceding item (2), can be suppressed by raising the velocity of liquid jet, and the burn-out heat flux rises steadily as the jet velocity is increased. In the present study employing a liquid jet driven by the gravitational force, the jet velocity is only 3m/sec at the highest, and nevertheless the burn-out heat flux as high as 6.37×10^6 kcal/m^{2}hr is recorded including no effect of forced convection on the heat transfer of boiling.

(4) In the preceding items (2) and (3), as a matter of course, the area of heated surface per a liquid jet and the flow rate of jet should have effects on the heat flux of burn-out. If the heated surface is larger, the condition near the circumference of the heated surface becomes more severe against burn-out for the same velocity of jet; and the similar circumstance appears when the flow rate of jet is reduced below a definite value.

(5) The nucleate boiling in the range of much higher heat fluxes than the heat flux of burn-out in ordinary pool boiling is realized experimentally in the present study, and its boiling curve takes a form of monotonous extension of the boiling curve in ordinary pool boiling. It may be assumed, therefore, that the nucleate boiling yet exists in the state of such high heat fluxes as afore-mentioned, and it is of the same character as that in the nucleate boiling of customary condition.

Acknowledgements

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Discussion

N. Isshiki (Tokyo Institute of Technology):

(1) Destroying a dome of vapor with a rotary cross of rods may yield a chance to supply liquid to the nucleate-boiling-liquid-layer. From what place does the liquid come? Is it not insufficient if the liquid film which encloses the dome of vapor alone is taken into consideration?

(2) As to the case of boiling with a vertical jet, there are a few places in the authors' paper yielding the expression that the splashing of liquid due to the blowing up of the generated vapor from the heated surface decreases the supplement of liquid on the heated surface. Cannot it be explained quantitatively by something like the balance between, for example, the dynamic pressure of jet and the vapor pressure that corresponds to the superheat of heated surface?

Authors' closure

(1) In the case of experiment of nucleate boiling with low water levels such as in the present case, the heated surface is placed within a liquid (cf. Fig. 1, where the water level measured from the heated surface to the free surface of water is H), and liquid enters the heated surface from its circumference when a vapor mass is destroyed. The state of collapse of vapor mass was not shown in the pictures of Fig. 8 in reference (2) since it was regarded as of importance to show the formation of vapor mass. How-

ever, if it is mentioned that the essential characters are the same as those in the pictures of Fig. 19 in reference (2), though Fig. 19 is concerned with transition boiling, it may help the discusser to understand the state afore-mentioned. It depends on this state too that the description such as "if H is increased, it may strengthen the flow of liquid to the heated surface" appears in section 2·4 in this paper. As to the thin liquid film which encloses a dome of vapor, its quantity is too small and additionally there is the splashing so that, in the authors' opinion, it can be neglected without any serious errors for the supply of liquid to the heated surface.

(2) The water jet of velocity of u_e=3m/sec dealt with in this study has a dynamic pressure of about 0.4kg/cm², and an equal rise of saturated steam pressure with this corresponds to the rise of saturation temperature of about 10°C. It should be noticed, however, that the sectional area of jet is less than only about 2.6% of the area of heated surface in this study. Accordingly, the variation of the pressure-temperature condition may hardly appear in the boiling on the heated surface except an extremely limited place near the center of heated surface which is subject to the impingement of jet. As a matter of fact, it is noticed in Fig. 6 that data of u_e=2.04~2.63m/sec lie approximately in the boiling curve obtained for the case of u_e=0 m/sec (the mark of ○), and we are safe in saying that this study is concerned with the saturated boiling at atmospheric pressure.

In this sense, the authors are reluctant to take "the vapor pressure that corresponds to the superheat of heated surface" into consideration. It is true that the authors had the dynamic pressure of vapor effusing from the heated surface as well as the dynamic pressure of jet vaguely in mind when they wrote this paper. However, the flows of vapor and liquid on the heated surface are not of a simple form of mutual impingement so that the use of the expression "dynamic pressure" was intentionally refrained from in this paper. The authors' impression is that it is unreasonable to attempt an analytical explanation before disclosing the details in the mechanical state of flow of vapor and liquid.

B6.2 STUDY OF MECHANISM OF BURN-OUT IN A HIGH HEAT-FLUX BOILING SYSTEM WITH AN IMPINGING JET

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Abstract

Nucleate boiling at very high heat fluxes was created on a heated surface covered with a flowing film of saturated water at atmospheric pressure being maintained by a small circular jet of water held at the center of the heated surface. It was found that increasing the heat flux leaded to a limiting state of flow where the splashing of droplets from the heated surface was no longer increased being kept constant till the burn-out appeared; and that there was a close relation between the burn-out heat flux and the jet velocity. A flow model, which can explain characteristics of this boiling system, is proposed, and it is suggested that the burn-out may be connected with the separation of a liquid flow from the heated surface accompanied with the effusion of vapor.

NOMENCLATURE

G	:	Mass flow rate of splashed droplets, kg/min
G_o	:	Mass flow rate of an impinging jet, kg/min
q	:	Heat flux (time-average value), W/m^2
q_{BO}	:	Burn-out heat flux, W/m^2
q_i	:	Superficial instantaneous heat flux corresponding to alternating current, W/m^2
T_{sat}	:	Saturation temperature, °C
T_w	:	Temperature of a heated surface, °C
u_e	:	Velocity of a liquid jet, m/s

1. INTRODUCTION

Experimental studies on the mechanism of burn-out in saturated pool boiling (including a low liquid-depth boiling) /1/,/2/ showed that in the high heat flux region of nucleate boiling, a comparatively thin "nucleate-boiling-liquid-layer" existed on a heated surface constantly supplying vapor to a "vapor mass" repeating the growth, detatchment, and rising cycles on the liquid-layer. Then, Katto and Kunihiro /3/, employing a heated 10-mm dia. disk surface (copper) and saturated water, made a study of the effect of two factors on burn-out: (i) the reduction of the cycle for supplying liquid to the "nucleate-boiling-liquid-layer" by means of the artificial destruction of the "vapor mass", and (ii) the artificial increase of the supply of liquid to the "nucleate-boiling-liquid-layer" by means of a small impinging circular jet (the cross-sectional area of which was less than 2.6 % of the heated surface area). As for the case of (ii), it was shown that the burn-out heat flux could be increased from 3.5×10^6 to 5.5×10^6 W/m^2 (for the case of zero water-level) by raising the jet velocity from 1 to 3 m/s.

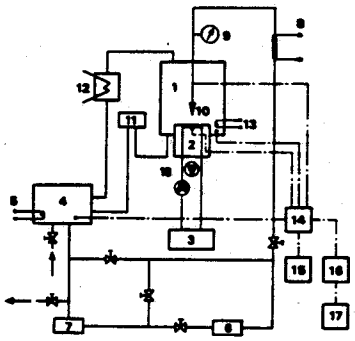
On the other hand, Kopchikov et al. /4/ carried out a study of the saturated nucleate boiling in a nucleate-boiling-liquid-layer maintained on a heated 11-mm dia. disk surface (brass) by supplying liquid through a vertical "spray" nozzle. The liquid flow rate was balanced, in their experiment, with the evaporation on the heated surface, so that there must not have been an overflow of liquid on the heated surface. Accordingly, their boiling system was of a type quite different from that with a small impinging circular jet.

The present study is concerned with a saturated boiling system with a small circular jet, and the extension of the nucleate boiling region up to a very high heat flux without the effect of forced convection is attempted by using a water jet of high velocity from 5 m/s to 60 m/s. At the preparatory stage of the study, the upper limit of heat flux was unknown so that the direct heating with alternating current was adopted for convenience. As explained at necessary points in the present paper, the adoptation of the alternating current heating gave no serious effect on the main purpose of the present study.

2. EXPERIMENTAL APPARATUS AND CHARACTERISTICS

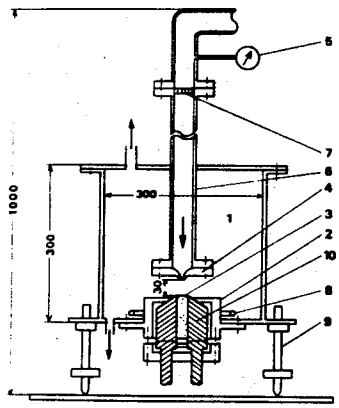
2.1 Experimental apparatus Fig. 1 shows the whole system of the apparatus, where nucleate boiling of saturated water at atmospheric pressure was generated on a heated surface -2- in a boiling-vessel -1-. Water of approximately 97 °C, coming out of a heating-apparatus -4- with an electric heater of 9 kW, was compressed by two pumps of regenerative type -6- and -7-, and was controlled to be kept at 97 °C by a temperature-regulator -8- before flowing out of a nozzle -10-. The piping was thermally insulated. Although the jet water had a slight subcooling of 3 °C, its effect could be neglected for the very high heat-flux nucleate boiling system such as in the present study. Water and steam flowing out of the boiling-vessel -1- were returned to the heating-apparatus -4- through either an overflow-vessel -11- or a condenser -12-.

Fig. 2 shows the main part of the apparatus. Two conducting copper blocks -10- were connected by a Cr-14% stainless steel foil of thickness 80 μm, and the horizontal square part (8 mm x 8 mm) of this foil



1.boiling-vessel, 2.heated surface, 3.low-voltage and high-current apparatus, 4.heating-apparatus, 5.heater, 6.high-pressure pump, 7.low-pressure pump, 8.temperature-regulator, 9.pressure gauge, 10.nozzle, 11.overflow-vessel, 12.condenser, 13.auxiliary heater, 14.ice-box, 15.mV meter, 16.potentiometer, 17.galvanometer, 18.AC meter.

Fig. 1 EXPERIMENTAL APPARATUS



1.boiling-vessel, 2.heating-equipment, 3.heated surface, 4.nozzle, 5.pressure gauge, 6.feed pipe, 7.filter, 8.auxiliary heater, 9.height adjustment screw, 10.conducting copper block.

Fig.2 MAIN PART

-3-, being heated directly by alternating current, worked as a heated surface. The foil was brazed with the copper blocks to reduce the contact electrical resistance, so that more than 95% of the electrical input could be consumed properly at the part of the heated surface -3-. Stainless steel foils on the market were used in the experiment without any processing of the surface except cleaning it with acetone, and the other side of the heated surface -3- was sticked by a binding agent to a bakelite block for thermal insulation (cf. Fig.2).

The nozzle -4- of 2mm I.D. at the end of a feed pipe -6- in Fig.2 was designed to be able to keep a water jet of laminar flow in the experimental range of jet velocity up to 60 m/s, that is, Reynolds number of 4040. The water jet was held vertically at the center of the heated surface -3-, and thereby the heated surface was covered with a radially flowing film of water (the cross-sectional area of the water jet was only 4.9% of the heated surface area). The mass flow rate of water flowing out of the nozzle -4- G_0 was measured by a pressure gauge -5-. Calibrations made with water at temperatures ranging from 30° to 80°C showed that the error was less than a few percent in consideration of the thermal variation of the density of water.

The heat flux across the heated surface q (a time-average value in the present study) was determined by assuming the thermal energy to be equal to the electrical input. It could be assumed that the error in the q thus determined was less than several percent, because the loss due to the contact electrical resistance was less than 5% as afore-mentioned and nucleate boiling heat transfer on the heated surface was very high. In order to confirm this assumption, the vapor generated on the heated surface in nucleate pool boiling at high heat fluxes (5×10^5 to 1×10^6 W/m^2) was specially captured and compared with the electrical input showing the difference to be less than 5%.

The temperature of the heated surface T_w was determined by

$$T_w = T_0 - \frac{1}{2}\frac{q}{\lambda}\delta \qquad (1)$$

where T_0 was the temperature measured by an alumel-chromel thermocouple of 50 μm dia. brazed to the back side of the stainless steel foil, q the heat flux across the heated surface, λ the thermal conductivity of Cr-14% stainless steel, and δ the thickness of the foil. The temperature T_0 was a time-average value obtained through measuring the electro-motive force of the thermocouple by a potentiometer as well as a galvanometer (cf. Fig. 1). As for the order of two terms on the right-hand side of Eq.(1), T_0 was measured with a value near 160°C while $q\delta/2\lambda$ took a value near 16°C in the experimental range of the present study.

2.2 Effect of dynamic pressure of impinging jet

The rise of saturation temperature as well as the rise of pressure appears in the so-called impinging zone near the center of the heated surface in the present study. According to Saad and Antonides /5/, however, in the case of a circular jet impinging on a solid plate, the fluid pressure (gauge) at the surface of the plate falls to about 20% of the stagnation pressure (gauge) at a distance of one and a half times the jet radius, and then about 2.5% at a distance of twice the jet radius, where the dynamic pressure becomes negligibly small for the greater part of experiments in the present study. For the conditions of a jet of radius 1mm standing on a heated surface of 8mm x 8mm, therefore, it can be regarded that more than 80% of the heated surface area is covered with a radially flowing film of liquid at atmospheric pressure.

In addition, simultaneous measurements of the temperature of the heated surface at the stagnation point as well as at a distance of twice the jet radius, revealed that the temperature difference between the two points was always less than 3°C if any. Exact reasons for this phenomenon are unknown, but there may be a possibility that the jet radius is only 1mm so that the effect of circumferencial violent nucleate boiling at a high heat flux can be extended up to the vicinity of the stagnation point. Such being the case at any rate, the temperature of the heated surface T_w in the present paper means that determined at the center of the heated surface for simplicity.

2.3 Developed nucleate boiling region

When the jet velocity u_e was kept constant and the temperature of the heated surface T_w was raised, the heat-flux curve such as shown in Fig.3 was obtained. In the region of a slow gradient on each heat-flux curve, no bubble could be seen on the heated surface with naked eyes at least. However, in the region of a sharp gradient on each heat-flux curve (the mark x in Fig.3 represents a burn-out point), violent nucleate boiling appeared on the heated surface and the splash of

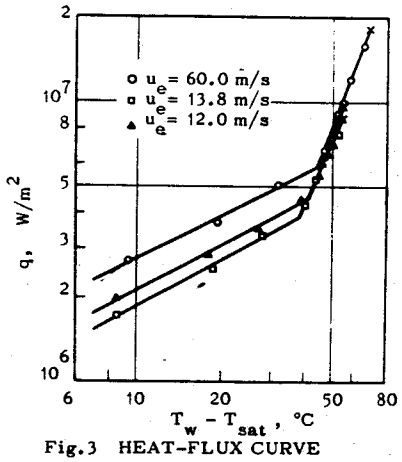


Fig.3 HEAT-FLUX CURVE

droplets from the heated surface was observed. This region of developed nucleate boiling alone will be dealt with below in the present paper.

3. NUCLEATE BOILING CURVE AND NUCLEATE-BOILING-LIQUID-LAYER

3.1 Nucleate boiling curve Since the stainless steel foil was fractured at every burn-out point in the present study, a new stainless steel foil had to be used for each experiment. Collecting the developed nucleate boiling curves such as represented in Fig.3 yields a group of nucleate boiling curves shown in the heat-flux region higher than 2×10^6 W/m^2 in Fig.4. On the other hand, another group of nucleate boiling curves shown in the heat-flux region lower than 2×10^6 W/m^2 in Fig.4 is a similar collection of the data of ordinary pool boiling (no impinging jet) specially obtained in the present experiment.

In spite of the scattering inherent in nucleate boiling, it can be noticed in Fig.4 that all boiling curves fall approximately in a narrow region between two parallel lines; and it may be assumed that the boiling in the present system with an impinging jet appears as a monotonous extension of the boiling curve of ordinary high heat-flux pool boiling.

It may be of interest to mention here that a broken line in Fig.4 represents a typical example of the nucleate boiling curve obtained by Katto and Kunihiro /3/ for their boiling system with an impinging jet of water of velocity u_e = 2.0 to 2.7 m/s including the state mixed with pool boiling. In their case, an end face of a steadily heated copper-block served as a heat transfer surface, while the direct heating with alternating current was used in the present study. The fact that the difference in the type of heating gives little effect on boiling curves in Fig.4 may depend on the circumstances that (i) a slight variation of the surface temperature can be accompanied by a comparatively great variation of the heat flux in nucleate boiling (cf. Fig.4), (ii) a stainless steel foil of thickness 80 μm can have effects of heat capacity, and (iii) nucleate boiling is apt to follow a variation of heat flux keeping a quasi-steady state at every moment.

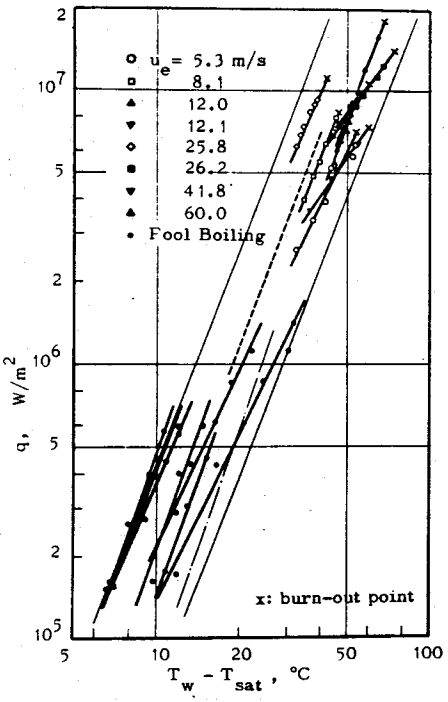


Fig.4 NUCLEATE BOILING CURVE

Finally, a dot-dash-line in Fig.4 represents Nishikawa-Yamagata's correlation equation /6/ for nucleate boiling heat transfer in ordinary pool boiling.

3.2 Nucleate-boiling-liquid-layer In the present boiling system, nucleate boiling appears within a thin film of water covering the heated surface, and according to the result of Fig.4, it may be assumed that the same nucleate-boiling-liquid-layer as in pool boiling at high heat fluxes plays an important role in heat transfer (the effect of forced convection is negligibly small). It may be of use to mention a previously reported result /2/ that the change of water-depth from 100mm to 2mm on the heated 10-mm dia. disk surface had no effect on the boiling curve in saturated boiling at high heat fluxes greater than 5×10^5 W/m^2.

4. FLUID BEHAVIOR IN BOILING SYSTEM WITH AN IMPINGING JET

4.1 Observation by high-speed photography Fig.5 shows instantaneous states of boiling for two cases of the jet velocity u_e = 9.4 m/s and 5.2 m/s. Due to the direct heating with alternating current of 50 Hz, the heating cycle of which is 100 Hz, the superficial instantaneous heat flux corresponding to alternating current q_i varies within a range of zero to 2q where q is the time-average value of heat flux (the amplitude of actual heat flux is less than this range due to the heat capacity of the heated surface). Fig.5 shows instantaneous states at $q_i \fallingdotseq 0$, q, and 2q.

In the state of $q_i \fallingdotseq 0$, the splashing of droplets is limited so that the heated square surface can be clearly seen, but as q_i is increased, droplets are splashed so heavily that the heated surface disappears from view. In addition, th trend may be noticed in Fig.5 that the spalshing of droplets is somewhat suppressed as the jet velocity is increased for the same magnitude of heat flux.



$q_i \fallingdotseq 0$ $q_i \fallingdotseq q$ $q_i \fallingdotseq 2q$

(a) u_e = 9.4 m/s; q = 3.7 x 10^6W/m^2



$q_i \fallingdotseq 0$ $q_i \fallingdotseq q$ $q_i \fallingdotseq 2q$

(b) u_e = 5.2 m/s; q = 3.5 x 10^6W/m^2

Fig.5 INSTANTANEOUS STATE OF BOILING

4.2 Variation of flow rate of splashed droplets with heat flux In order to measure the flow rate of droplets splashed from the heated surface, a droplet-catcher as shown in Fig.6 was installed in the boiling-vessel -1- in Fig.2. In the experiment, splashed droplets had a power enough to fly a considerable distance. The free-liquid overflowing the circumference of the heated surface was independently gathered and discharged from the droplet-catcher along light

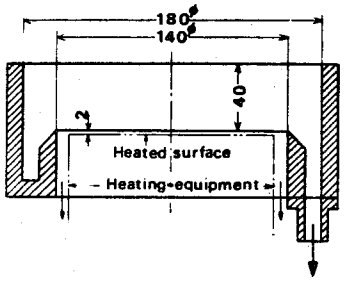


Fig.6 DROPLET-CATCHER

directed lines shown in Fig.6.

Fig.7 represents typical examples of data for the variation of the mass flow rate of splashed droplets G with the heat flux q for a constant mass flow rate of the impinging jet G_0. As q is raised, G is initially increased (with an approximate relation $G \propto q^2$) till the state of $G/G_0 \fallingdotseq 0.4 - 0.5$ is arrived, but afterwards G is kept constant and further increase of q leads to the generation of a burn-out (the mark x in Fig.7 represents a burn-out point).

It should be mentioned here that the evaporated liquid is negligibly small compared with either G_0 or G even at the state of burn-out heat flux. For example, the evaporation of liquid on the heated surface (8mm x 8mm) at a heat flux $q = 6 \times 10^6$ W/m^2 is calculated to be 0.014 kg/min while G_0 has been measured to be

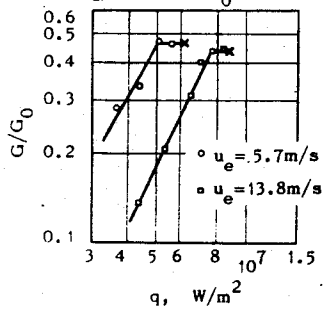


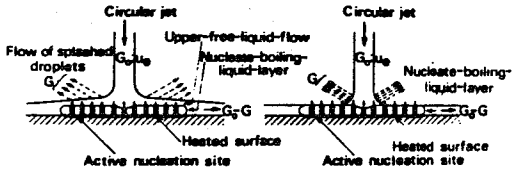
Fig.7 MASS FLOW RATE OF SPLASHED DROPLETS VS. HEAT FLUX (x : burn-out)

1.17 kg/min in Fig.7.

4.3 Flow model Summarizing the observations described in Sections 3.2, 4.1 and 4.2, a model shown in Fig.8 may be assumed for the flow of fluid in the present boiling system.

Fig.8(a) represents a heated surface with an impinging jet of the mass flow rate G_0 and of the velocity u_e, where a "nucleate-boiling-liquid-layer" on the heated surface dominates nucleate boiling. In addition, the nucleate-boiling-liquid-layer is covered with an "upper-free-liquid-flow", from which a "flow of splashed droplets" is produced through the action of vapor jets blowing out of the underlying nucleate-boiling-liquid-layer. The residual liquid being free from the splashing, that is, $G_0 - G$ flows through the nucleate-boiling-liquid-layer, as well as the upper-free-liquid-flow, before it overflows the edge of the heated surface.

4.4 Limiting state of flow It can probably be assumed that Fig.8(b), a limiting state of the flow model of two liquid layers shown in Fig.8(a), corresponds to the final state of flow where G is kept



(a) State of flow (b) Limiting state of flow

Fig.8 A MODEL OF FLOW

constant independently of q which has been observed in Fig.7. In Fig.8(b), the emission of vapor from the nucleate-boiling-liquid-layer has been intensified enough so that the greater part of the upper-free-liquid-flow is splashed immediately near the circumference of the impinging zone disclosing the greater part of the nucleate-boiling-liquid-layer to surroundings. Under these conditions, the increase of the heat flux q is only accompanied by the intensification of nucleate boiling whithin the nucleate-boiling-liquid-layer, that is, a monotonous increase of heat transfer along a boiling curve shown in Fig.4, while the flow rate of splashed droplets G can scarecely be changed.

It is natural to assume that the flow rate of liquid through a thin nucleate-boiling-liquid-layer is so small that $G_0 - G \fallingdotseq 0$, or $G/G_0 \fallingdotseq 1$, is derived in the limiting state of Fig.8(b). As for the fact that $G/G_0 \fallingdotseq 0.4$ to 0.5 was obtained in the present experiment as shown in Fig.7, it should be remembered that an intermission of splashing was included in a cycle of heating due to the direct heating with alternating current (cf. Section 4.1). It may be particularly of use to mention here that the authors /7/ have made experiments on a "steadily" heated disk surface of a size similar to that in the present study, confirming that $G/G_0 \fallingdotseq 0.9$ is established in this steady case.

5. BURN-OUT

5.1 Burn-out heat flux The brun-out heat flux q_{BO} measured in the present study is plotted in Fig.9 against the jet velocity u_e ranging from 5 m/s to 60 m/s. It should be noticed that there is a systematic relation between q_{BO} and u_e over a wide range of u_e. A heavy solid line in Fig.9 has been determined from the data afore-mentioned by the method of least squares.

In addition, Katto and Kunihiro's data /3/ obtained on a steadily heated disk surface of a size similar to that in the present study (within a range of u_e of 1 to 3 m/s) are also shown in Fig.9. Their data occur for three kinds of nozzle diameter (0.7mm, 1.165mm and 1.60mm) suggesting that q_{BO} is not affected by either the nozzle diameter or the mass flow rate of the impinging jet. As for the authors' experiment /7/ mentioned at the

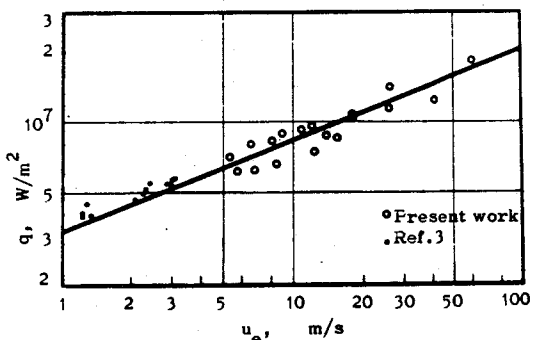


Fig.9 BURN-OUT HEAT FLUX VS. JET VELOCITY

end of Section 4.4, q_{BO} which accords with the solid line in Fig.9 has been given within the experimental range of u_e from 6 m/s to 30 m/s.

The results in Fig.9 show that the difference in the type of heating gives little effect on the burn-out heat flux similarly to the case of the nucleate boiling curve in Fig.4. It may be of use to mention here that a thermally insulated stainless steel foil of thickness 80 μm, subjected to an internal heating corresponding to a heat flux $q = 10^7$ W/m^2, takes about 30 ms for the temperature rise from 160°C to the melting point, while one heating cycle of alternating current of 50 Hz is 10 ms.

The heavy solid line in Fig.9 is expressed by

$$q_{BO} = 3.4 \times 10^6 \, u_e^{0.39} \qquad (2)$$

where q_{BO} in the unit of W/m^2 and u_e in m/s.

5.2 Fracture of stainless steel foil due to burn-out

It was observed in the present study that a fracture line due to burn-out appeared near a side of the heated square surface as shown in Fig. 10, while it appeared always passing through near the center of the heated surface in the case of ordinary pool boiling (no impinging jet). The trend may be seen in Fig. 10 that the position of the fracture moves to a side of the heated surface, though very gradually, as the jet velocity u_e is increased.

5.3 Mechanism of burn-out Judging from the fluid behavior revealed in Chapter 4, it is no longer valid to attribute the occurrence of burn-out in the present system to a simple balance between the consumption and the supply of liquid on the heated surface. The consumption of liquid due to evaporation on the heated surface is of the order of 1/10 at most compared with the quantity of liquid $G_0 - G$ flowing over the heated surface without being splashed (cf. Section 4.2).

In the present boiling system, the burn-out is originated under the conditions that a large quantity of liquid is still overflowing the edge of the heated surface. There are numerous vapor jets blowing out of active nucleation sites on the heated surface, and it may be assumed that the splashing of liquid scarecely occurs from the disclosed nucleate-boiling-liquid-layer in a limiting state as shown in Fig.8(b). Under the interaction between the flow of liquid and that of vapor afore-mentioned, there is a close relation as shown in Fig.7 between the burn-out heat flux (which is proportional to the effusion of vapor from the heated surface) and the jet velocity (which may be assumed to be approximately proportional to the velocity of the liquid flow on the heated surface). In connection with this matter, attention should be paied to the fact that the burn-out heat flux in the present system is not affected by the mass flow rate of the impinging jet (cf. Section 5.1).

Accordingly, the mechanism with which the burn-out is originated in the present system may be tentatively connected with a problem of the separation of a liquid flow from the heated surface accompanied with the effusion of vapor, though this is a presumption at the present stage.

5.4 Appendix In case of a turbulent boundary layer along a porous wall accompanied with the effusion of fluid, the separation of the boundary layer from the wall may be correlated approximately in such a form as

$$\frac{\rho_w v_w}{\rho_\infty u_\infty} \frac{1}{f_o} = \text{const.} \qquad (3)$$

where ρ_w is the density of the effusing fluid, v_w the critical effusion velocity, ρ_∞ the density of the free-stream fluid, u_∞ the free-stream velocity, and f_o the skin friction coefficient in the limit as $v_w \to 0$.



u_e=6.56m/s u_e=12.0m/s u_e=26.0m/s u_e=41.8m/s

(Electrical current flows in horizontal direction)

Fig. 10 FRACTURE OF A STAINLESS STEEL FOIL

Assuming that Eq.(3) can be applied to the present case, the superficial velocity of the effusing vapor at the burn-out heat flux q_{BO} will be written as

$$\rho_w v_w = q_{BO} / h_{fg}$$

where h_{fg} is the latent heat of evaporation. Then, since the single-phase flow is now meaningless, the bubble flow will be assumed to determine f_o. According to the experiment of Malnes /8/ (made within a tube but presumably applicable to a plate because of the state of turbulent flow), f_o for the bubble flow in the limit as the void ratio approaches zero is given as

$$f_o \propto u_\infty^{-0.64}$$

where u_∞ is the velocity of liquid. Substituting $\rho_w v_w$ and f_o afore-mentioned for Eq.(3) yields

$$q_{BO} \propto u_\infty^{0.36} \qquad (4)$$

This analysis is rather simple and imaginative, but it may not be useless to notice that there is a similarity at least between Eq.(2) and Eq.(4).

6. CONCLUSIONS

(1) Nucleate boiling of saturated water at atmospheric pressure has been created at heat fluxes up to 2 x 10^7 W/m^2 on a heated 8mm x 8mm square surface with an impinging 2-mm dia. circular jet.

(2) Boiling in this system seems to appear as a monotonous extension of the boiling curve of ordinary high heat-flux pool boiling.

(3) Increasing the heat flux leads to a limiting state of flow where the splash of droplets from the heated surface is no longer increased.

(4) A close relation exists between the burn-out heat flux and the jet velocity over a wide experimental range of jet velocity from 1 m/s to 60 m/s.

(5) A flow model for this boiling system has been proposed. It is tentatively assumed that burn-out is connected with the separation of a liquid flow from the heated surface.

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BURNOUT IN A HIGH HEAT-FLUX BOILING SYSTEM WITH AN IMPINGING JET

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Abstract An experimental study has been made on the fully-developed nucleate boiling at atmospheric pressure in a simple forced-convection boiling system, which consists of a heated flat surface and a small, high-speed jet of water or of freon-113 impinging on the heated surface. A generalized correlation for burnout heat flux data, that is applied to either of water and freon-113, is successfully evolved, and it is shown that surface tension has an important role for the onset of burnout phenomenon, not only in the ordinary pool boiling, but also in the present boiling system with a forced flow.

NOMENCLATURE

C, constant;
c_{pl}, specific heat of liquid at constant pressure;
D, diameter of a heated surface;
d, inside diameter of a nozzle of impinging jet at the exit;
G_0, mass flow rate of an impinging jet;
G, mass flow rate of splashed droplets;
G', mass rate of evaporation of liquid;
g, gravitational acceleration;
K, constant;
L, latent heat of evaporation;
q, heat flux;
$q_{B.O}$, burnout heat flux;
T_{liq}, temperature of supplied liquid;
T_{sat}, saturation temperature;
T_w, temperature of heated surface;
u_e, exit velocity of liquid jet.

Greek symbols

β, thermal coefficient of volume expansion;
ε_{sub}, correction factor for the effect of subcooling on burnout heat flux;
λ_l, thermal conductivity of liquid;
μ_l, viscosity of liquid;
μ_v, viscosity of vapor;
ρ_l, density of liquid;
ρ_v, density of vapor;
σ, surface tension.

1. INTRODUCTION

As is generally known, burnout in pool boiling has been studied rather extensively, and generalized correlations, that apply to the burnout heat flux data for many kinds of fluid and thermodynamical condition, have been derived by Kutateladze [1], Rohsenow and Griffith [2], Zuber [3], Chang and Snyder [4], Zuber *et al.* [5], and others.

In the case of forced-convection boiling, many studies have also been made for correlating the burnout or critical heat flux data, particularly in boiling with flow inside of tubes, annuli, outside of rod bundles, etc. In this case, however, it is probably close to the truth to say that due to the complexity of included phenomena, no correlation for burnout heat flux has yet been evolved that is applied universally to the data for various kinds of fluid and conditions. As an inevitable consequence, it has been impossible to examine whether the mechanism of burnout in pool and forced-convection boiling is quite different.

On the other hand, one of the authors of this paper, starting from the studies on the mechanism of burnout in pool boiling [6, 7], entered into an attempt [8] to raise the burnout heat flux by holding a small impinging jet through which the liquid is forcibly supplied to the heated surface. Then, it was followed by the authors' preceding study [9], in which the burnout heat flux was observed to vary as a very simple function of the velocity of jet.

As the next step, the present study has been planned to make a more extensive study of nucleate boiling and to search for a possibility of obtaining a generalized correlation of burnout heat flux in the boiling system with an impinging jet, which can be regarded as one of the simplest systems associated with the forced-convection boiling.

2. EXPERIMENTAL APPARATUS

The whole system of experimental apparatus, which is shown schematically in Fig. 1, consists of three cardinal portions; the loop for circulating a test fluid, the electric system for heating the test fluid, and the main part where the nucleate boiling with an impinging jet is generated on a heated surface. The experiment is made in the present study on saturated and subcooled boiling of pure water and freon-113 at atmospheric pressure on the heated surface (2) in a boiling vessel (1). These two fluids with different physical properties have been chosen in order to avert the danger of fire during the experiment. Liquid, comin

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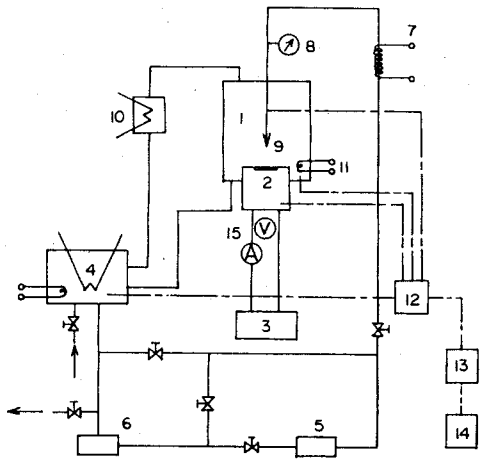

FIG. 1. Whole system of experimental apparatus. 1. Boiling-vessel; 2. Heated surface; 3. Transformer; 4. Heating and subcooling apparatus; 5. High-pressure pump; 6. Low-pressure pump; 7. Temperature-regulator; 8. Pressure gauge; 9. Nozzle; 10. Condenser; 11. Auxiliary heater; 12. Ice-box; 13. Self-registering thermometer; 14. Digital volt meter; 15. AC meter.

out of the heating and subcooling apparatus (4) which consists of a main electric heater, an automatic electric heater for controlling the temperature of liquid and a heat exchanger for subcooling the liquid, is pressurized by two pumps of regenerative type (5) and (6), and is regulated to be kept at a desired temperature by means of a temperature-regulator (7) before flowing out of a nozzle (9). The piping for the liquid is thermally insulated. Liquid and vapor flowing out of the boiling vessel (1) are returned to the heating and subcooling apparatus (4) through either a drain pipe or a condenser (10).

Figure 2 shows the main part of the apparatus in the case of the upward-facing heated surface, and this part can be replaced by another alternative with a downward-facing heated surface in order to make an experiment to check the effect of gravitational force on heat transfer as well as on burnout. The end surface of the conical part of the copper block (3) located at the central position in a boiling vessel (1) is a circular heated surface of 11 mm to 21 mm in diameter. The heated surface is polished with the emery paper (No. 0) and cleaned with acetone and pure water in every experiment. The electric heater (10), which is composed of eleven heating elements of plate type mounted in the grooves of the lower part of copper block, is capable of providing the maximum heat flux of 2×10^7 W/m^2 across the heated surface. The temperature of the heated surface T_w as well as the heat flux across the heated surface q is measured by employing the temperature distribution determined with three Chromel–Alumel thermocouples of 0.1 mm-dia. set up along the axis of copper block.

In the preliminary experiment, the vapor, which is generated on the heated surface in the nucleate pool boiling at sufficiently high heat flux of 5×10^5 to 1×10^6 W m^{-2}, is accumulated in a collector to measure the heat flux across the heated surface, and by this means the thermal conductivity of copper block is determined. When the heat flux, thus determined, is compared with the electric input to the heater mounted in the copper block, it is found that the heat loss from the copper block is less than 10% of the total input.

The nozzle (4) in Fig. 2 has been designed to be able to make a circular liquid jet of laminar flow within the range of jet velocity in the present study. The liquid jet

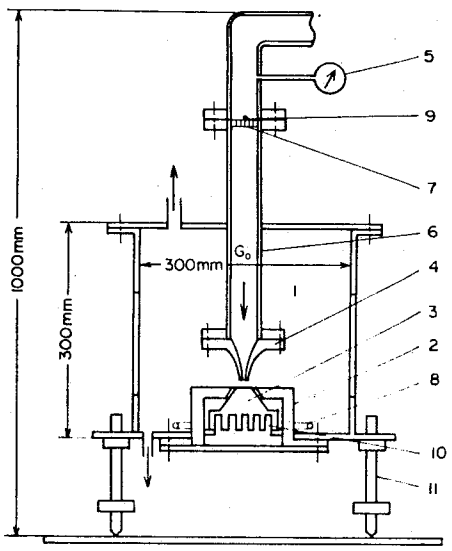

FIG. 2. Main part of experimental apparatus. 1. Boiling-vessel; 2. Heating-equipment; 3. Copper block; 4. Nozzle; 5 Pressure gauge; 6. Feed pipe; 7. Filter; 8. Auxiliary heater; 9. C-A thermocouple; 10. Electric heater; 11. Height adjustment screw.

is held vertically at the center of heated surface (3), and thereby the heated surface is covered with a radially flowing thin liquid film. In this connection, it should be mentioned that the cross-sectional area of the liquid jet used in the present study is only 6–10% compared with the total area of the heated surface, and therefore the rise of saturation temperature as well as the rise of static pressure which appears in the so-called impinging zone near the center of the heated surface, has little effect on the boiling taking place all over the heated surface (cf. [9]). The mass flow rate of liquid flowing out of the nozzle (4) G_0 is determined by means of a pressure gauge (5). Calibrations made with water at temperatures ranging from 30 to 80°C and with freon-113 at temperatures ranging from 20 to 25°C show that the error is less than a few percent if the thermal variation of density of liquid is taken into account. The temperature of liquid flowing out of the nozzle is measured by a Chromel–Alumel thermocouple (9) of 0.3 mm-dia. set up in the feed pipe.

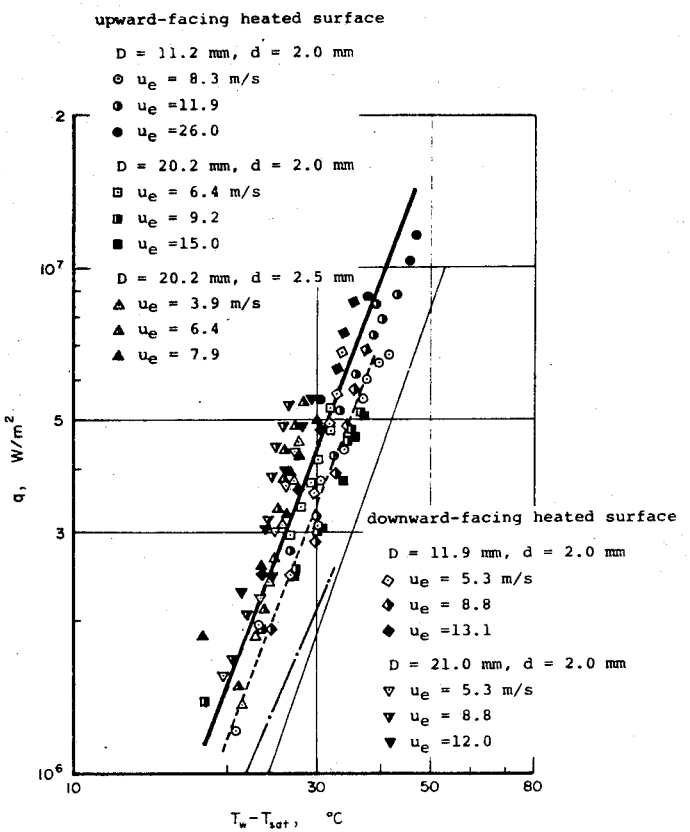

FIG. 3. Saturated nucleate boiling data for water.

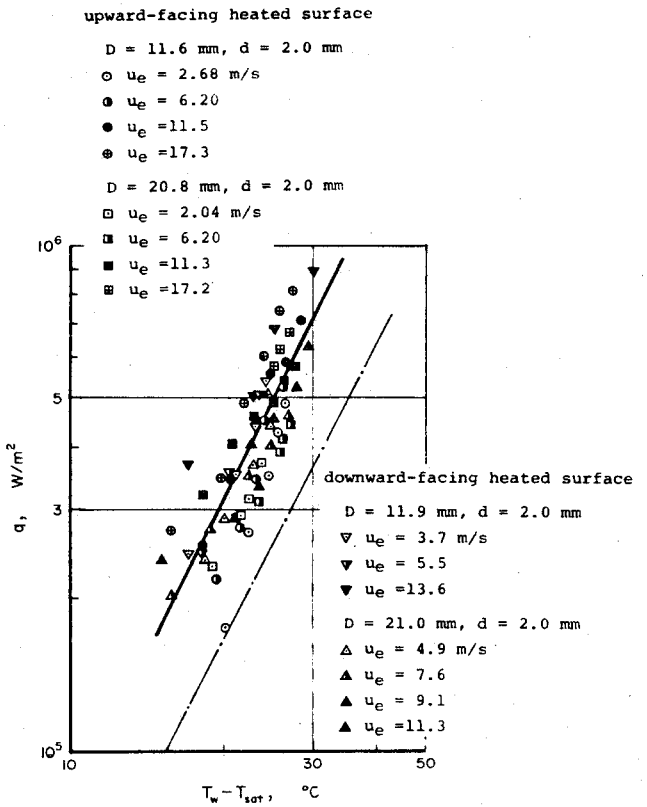

FIG. 4. Saturated nucleate boiling data for freon-113.

3. SATURATED NUCLEATE BOILING HEAT TRANSFER

Figures 3 and 4 show the typical data of fully-developed nucleate boiling obtained in the range of high heat flux for the respective cases of supplying saturated water and freon-113. In Fig. 3, a heavy solid line is the mean nucleate boiling curve, while a broken line is the result of nucleate boiling curve obtained by Katto and Kunihiro [8] for the boiling system with an impinging jet of water of velocity $u_e = 2.0$–2.7 m/s including the state mixed with pool boiling. In this connection, it may be of use to refer the authors' preceding study [9] reporting the empirical fact that approximately speaking, the nucleate boiling curve of the boiling system with an impinging jet is in accord with the monotonous extension of the nucleate boiling curve obtained in the region of high heat flux of the ordinary pool boiling.

Apart from the scattering of data inherent in nucleate boiling, it may be concluded from Figs. 3 and 4 that the boiling curves obtained under different conditions coincide with each other independently of the jet velocity u_e, the ID of nozzle d, the diameter of the heated surface D, and the orientation of the heated surface. Then the following assumption can be made from the afore-mentioned characteristics of boiling curve. Namely, the motion of fluid activated by nucleate boiling on the heated surface is so active in the nucleate boiling at high heat flux, that the state of fluid motion near the heated surface hardly changes even if the forced convection is applied to the bulk fluid. In addition, it can be presumed that almost all the quantity of heat transferred from the heated surface is spent on the latent heat of evaporation at the active nucleation sites distributed on the heated surface in case of the nucleate boiling at high heat flux. Therefore, an active, self-supporting boiling layer may be assumed to cover the heated surface and to govern the nucleate boiling heat transfer at high heat flux. The thin layer above-mentioned will be called nucleate-boiling liquid layer in the present paper.

Finally, for reference, the light solid line in Fig. 3 is the prediction of Nishikawa–Yamagata's equation [10] correlating the nucleate boiling heat transfer of water and some organic liquids in ordinary pool boiling. On the other hand, the respective dot-dash-lines in Figs. 3 and 4 represent the nucleate boiling curves obtained experimentally by Copeland [11] for water and by Ruch and Holman [12] for freon-113 on the boiling system with an impinging jet.

4. BEHAVIOR OF FLUID IN THE PRESENT BOILING SYSTEM

4.1. *Observation of behavior of fluid*

Figure 5, dealing with the case of water supplied with the jet velocity $u_e = 8.8$ m/s from a nozzle of $d = 2.0$ mm to the downward-facing flat surface that includes the heated surface of $D = 11.9$ mm in it, shows the typical aspect of saturated boiling.

Now, Fig. 5(a) represents the special case of heat flux $q = 0$, that is non-boiling, in which the water discharged upwardly from the nozzle impinges on the center of the downward-facing surface and then flows radially over the surface without boiling until it loses kinetic energy resulting in the natural falling of waterdrops at

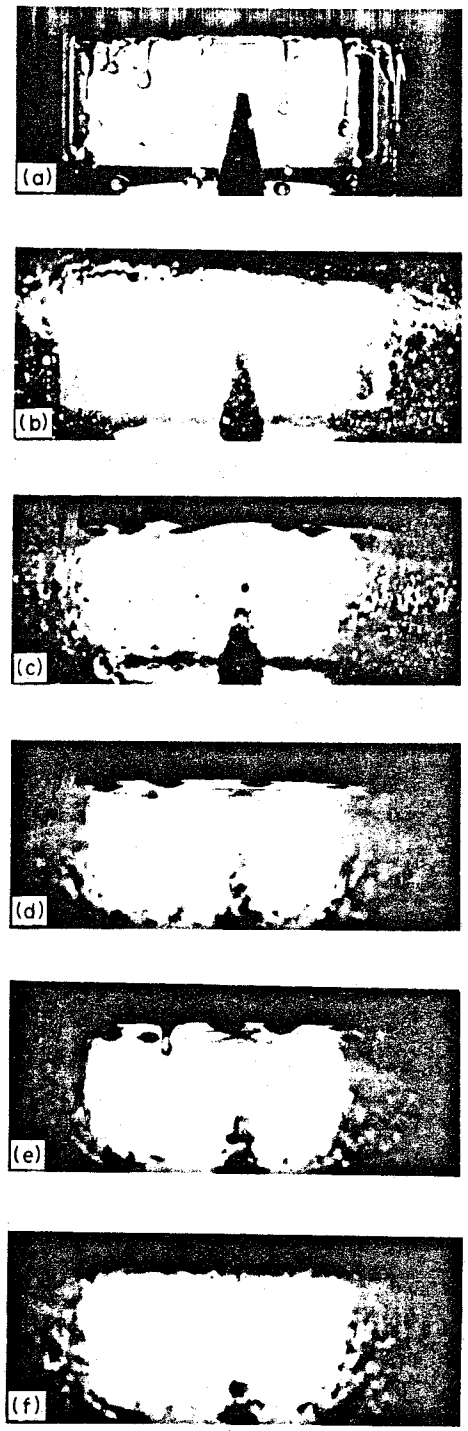

FIG. 5. State of saturated boiling of water downward-facing heated surface ($u_e = 8.8$ m/s, $D = 11.9$ mm, $d = 2.0$ mm).

the position far away beyond the outer edge of the heated surface. Then, Figs. 5(b–e) represent the state of $q = 1.97 \times 10^6$, 3.66×10^6, 5.70×10^6 and 6.80×10^6 W/m^2 respectively. In these cases, the splash of water from the heated surface appears due to the strong ejection of vapor from the nucleate-boiling liquid layer [notice the conical envelop of splashing flow clearly observed in Fig. 5(d) and (e)], but still there is a non-vanishing, residual water flowing over the heated surface, which is verified by the observation of successive formation of waterdrops falling down at the position beyond the outer edge of the heated surface [see Fig. 5(e) for example]. Here, it should be mentioned that the burnout dealt with in the present study, has been observed to appear always with the non-vanishing, residual liquid, flowing over the heated surface and maintaining the nucleate boiling there. Once the burnout has occurred, however, the liquid flowing over the outer edge of the heated surface disappears and all the quantity of liquid impinging on the heated surface is splashed out as shown in Fig. 5(f) [notice no trace of the residual water to make the natural falling of waterdrops].

In the case of the upward-facing heated surface, the same behavior of fluid as afore-mentioned is observed only except that the phenomenon of hydraulic jump appears, instead of the falling of waterdrops, at the position far beyond the outer edge of the heated surface [cf. Fig. 7(b)].

4.2. *Variation of the rate of splashing with heat flux*

In order to measure the mass flow rate of droplets splashed from the heated surface, the droplet-catcher similar to that used in the preceding study [9] is installed in the boiling vessel in case of the upward-facing heated surface. Figure 6 shows typical results thus obtained for the variation of the rate of splashing G/G_0 with the heat flux q, where G is the mass flow rate of droplets

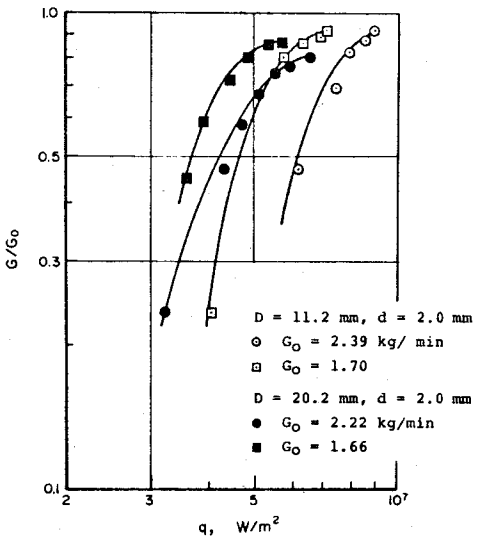

FIG. 6. Rate of splashing vs heat flux (saturated boiling of water).

splashed depending on q, while G_0 is the total mass flow rate of an impinging jet. As q is raised, G/G_0 increases rapidly; and it has been observed in the experiment that when G/G_0 in Fig. 6 reaches its highest value of data shown near $G/G_0 = 0.8 \sim 1.0$ in each experiment, a little increase of q leads to the occurrence of burnout. In this connection, it should be mentioned that the rate of evaporation G'/G_0, where G' is the mass of liquid evaporated on the heated surface per unit time, is readily estimated from q, and G'/G thus determined has been found to be always smaller than $1-(G/G_0)$ at a burnout point. Of course, the residual quantity $G_0-(G+G')$ is nothing but the mass flow rate of the residual liquid flowing over the outer edge of the heated surface which has been mentioned in the preceding section.

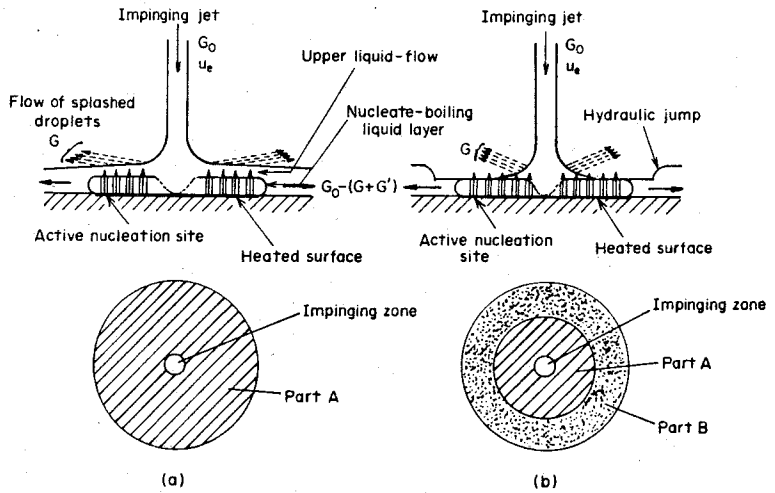

FIG. 7. A model of flow (upward-facing heated surface).

4.3. *A flow model*

On the basis of experimental facts represented by Figs. 3–6, we may assume such a model as shown in Fig. 7 for the flow of liquid and vapor in the fully-developed nucleate boiling at high heat flux with an impinging jet, which is a slight revision of that presented in the authors' preceding study [9]. Self-supporting violent nucleate boiling independent of the external effect is kept in a nucleate-boiling liquid layer on the heated surface, and the thin layer is covered with an upper liquid-flow, from which the splash of droplets is produced due to the ejection of vapor jets from the underlying boiling layer as shown in the part A of Fig. 7(a). As heat flux is far raised, however, the splashing becomes very violent and the upper liquid-flow is splashed out near the impinging zone so that the clear conical envelop of splashing flow such as seen in Fig. 5(d) and (e) is developed, and the downstream part of nucleate-boiling liquid layer is exposed to the surrounding atmosphere of vapor, yielding the state of the part B in Fig. 7(b). Even in the case of Fig. 7(b), the flow rate of the residual liquid flowing over the outer edge of the heated surface $G_0-(G+G')$ does not vanish (cf. Section 4.2), so that the nucleate-boiling liquid layer is maintained on the heated surface.

4.4. *Flow state at the onset of burnout*

According to the observation employing the colour photography and the high-speed photography, the burnout takes place in the state of Fig. 7(b), as far as the boiling of water is concerned. However, in the case of the boiling of freon-113, it seems likely that the flow still remains in the state of Fig. 7(a) when the burnout occurs. Consequently, if there are no particular phenomena inherent to the substance of liquid, it may be concluded that the burnout can occur in either state of Fig. 7(a) and (b). In other words, the change of the state of flow shown in Fig. 7(a) and (b) cannot be the cause for the onset of burnout.

In addition, the observation shows that once burnout takes place, in either state of Fig. 7(a) and (b), the dry-out starts from the outer edge of the heated surface to extend up to the vicinity of impinging zone, being accompanied by the state that all the liquid is splashed out immediately after impinging on the heated surface.

5. BURNOUT HEAT FLUX IN SATURATED BOILING

5.1. *Experimental result*

In the authors' preceding study [9], the burnout heat flux $q_{B.O}$ was measured by varying the velocity of impinging jet u_e alone, to give the empirical and tentative result:

$$q_{B.O} = \text{const.}\ u_e^{0.39}.$$

In the present study, however, not only the jet velocity u_e but also the various factors such as the diameter of heated surface D, the inside diameter of nozzle d, the orientation of heated surface (i.e. the direction of gravitational force), and the kind of boiling liquid are changed, yielding the result of Figs. 8 and 9. Namely, Fig. 8 shows all the experimental data of burnout heat flux obtained for the upward-facing heated surface, and similarly Fig. 9 shows the data obtained for the downward-facing heated surface. In addition, Fig. 8 includes the data of authors' preceding study [9] obtained for the burnout on a square, stainless plate (8 × 8 mm) heated by the conduction of alternating current, for which the diagonal of square plate ($=\sqrt{2}$ × 8 mm = 11.3 mm) is taken as D and the time-averaged value of heat flux observed at burnout is taken as $q_{B.O}$ in Fig. 8.

From Figs. 8 and 9, it may be noticed that $q_{B.O}$ can be correlated as a function of u_e/D, and the solid lines inserted tentatively in both figures represent the relation of

$$q_{B.O} = \text{const.}\ (u_e/D)^{1/3} \tag{1}$$

where the value of constant differs between water and freon-113, and the comparison of Fig. 8 and Fig. 9 shows that there is no effect of gravitational force on $q_{B.O}$ in the present forced-convection boiling system. Besides, it may be noticed from Fig. 8 that there is no effect of d on $q_{B.O}$. The above-mentioned fact that $q_{B.O}$ is not governed by d but is governed by u_e, suggests that $q_{B.O}$ is unaffected by the mass flow rate of impinging jet G_0, because $G_0 = (\pi/4)d^2 u_e \rho_l$ where ρ_l is the density of liquid.

5.2. *Dimensional analysis and correlation of burnout heat flux data*

In order to search for dimensionless groups for correlating the data of Figs. 8 and 9, a dimensional analysis will be attempted on burnout in the saturated boiling with a forced flow in a gravitational field. Since the burnout in the case of saturated boiling can be regarded as a phenomenon independent of heat transfer (i.e. the thermal conductivity and the specific heat of liquid have no effect on burnout), it may be assumed that there are ten physical quantities concerned: that is, $q_{B.O}$: burnout heat flux, L: latent heat of evaporation, ρ_v: density of vapor, ρ_l: density of liquid, μ_v: viscosity of vapor, μ_l: viscosity of liquid, σ: surface tension, $g(\rho_l-\rho_v)$: difference of specific weight between liquid and vapor, u_e: velocity of forced flow, and D: representative length of heated surface. On the other hand, there are four fundamental units (mass, length, time and quantity of heat). Therefore, according to pi theorem, there are six (that is ten minus four) independent dimensionless groups, to give the relation:

$$\frac{q_{B.O}/(\rho_v L)}{u_e} = C\cdot\left(\frac{\rho_l}{\rho_v}\right)^{\alpha_1}\left(\frac{\mu_v}{\mu_l}\right)^{\alpha_2}\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{\alpha_3} \times\left(\frac{\mu_l}{\rho_l\cdot u_e D}\right)^{\alpha_4}\left[\frac{g(\rho_l-\rho_v)D}{\rho_l\cdot u_e^2}\right]^{\alpha_5} \tag{2}$$

where C and the five exponents α_1 to α_5 are constant respectively. All the dimensionless groups included in equation (2) can be regarded as of hydrodynamic character.

Now, it may be of interest to note that equation (2) can also be applied to the burnout in ordinary pool

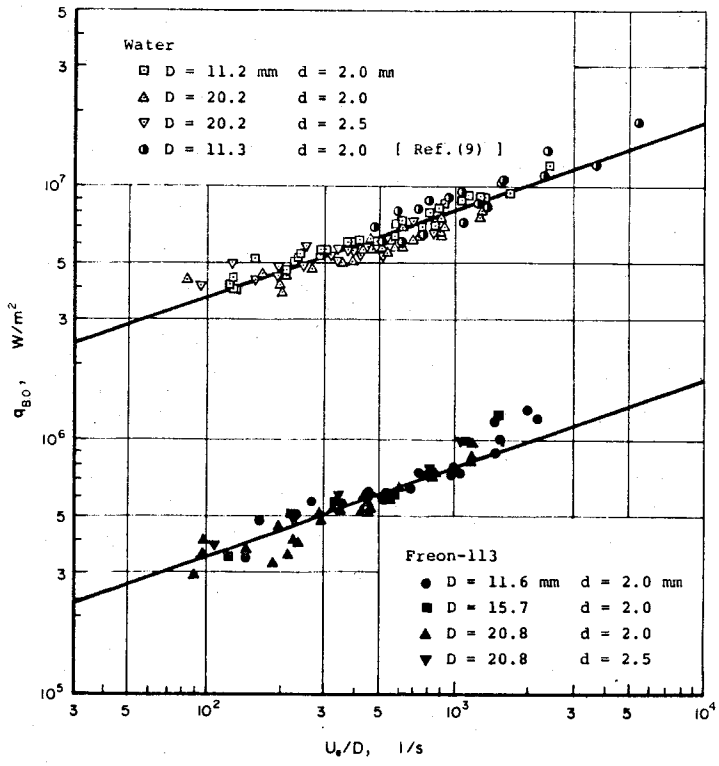

FIG. 8. Burnout heat flux vs u_e/D (saturated boiling, upward-facing heated surface).

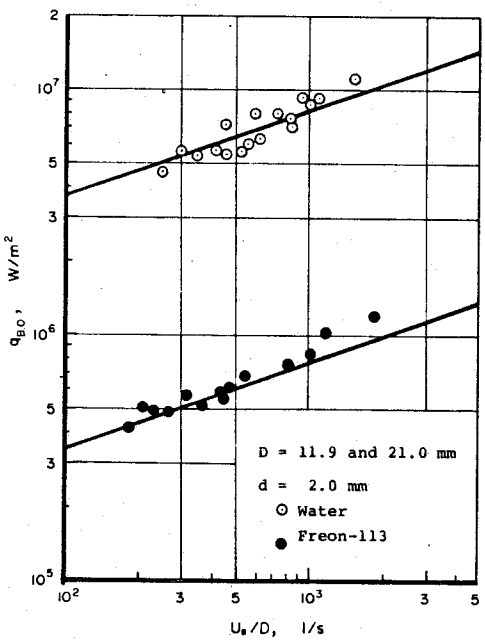

FIG. 9. Burnout heat flux vs u_e/D (saturated boiling, downward-facing heated surface).

boiling if the case of heated surface of extremely small size is excluded. Namely, assume that none of μ_v, μ_l, u_e, and D affects $q_{B.O}$, and all the exponents of equation (2) except α_1 are immediately determined as:

$$\alpha_2 = 0,\ \alpha_3 = 1/4,\ \alpha_4 = 0,\ \alpha_5 = 1/4.$$

Then, equation (2) is rewritten as:

$$\frac{q_{B.O}/(\rho_v L)}{(\sigma g(\rho_l - \rho_v)/\rho_v^2)^{1/4}} = C\left(\frac{\rho_l}{\rho_v}\right)^{\alpha_1 - 1/4} \tag{3}$$

where C and α_1 can be determined, for example, so as to accord with the correlation proposed by Kutateladze [1] to give $C = 0.16$ and $\alpha_1 = 1/4$.

Next, let us apply equation (2) to the burnout in the present study. In this case, presuming that μ_l has no effect on $q_{B.O}$ and recalling the empirical fact that equation (1) holds and g has no effect on $q_{B.O}$ (cf. Section 5.1), it yields immediately:

$$\alpha_2 = 0,\ \alpha_3 = 1/3,\ \alpha_4 = 0,\ \alpha_5 = 0.$$

Then equation (2) is rewritten as:

$$\frac{q_{B.O}/(\rho_v L)}{u_e} = C\left(\frac{\rho_l}{\rho_v}\right)^{\alpha_1}\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3} \tag{4}$$

and the exponent α_1 in equation (4) can be determined so as to combine the two groups of data (water and freon-113) of Figs. 8 and 9 into a single group to give $\alpha_1 = 0.725$. Then, the constant C in equation (4) can be determined from the solid straight line in Fig. 10, correlating the data of Figs. 8 and 9 in the form of equation (4) with $\alpha_1 = 0.725$, to give $C = 7.45 \times 10^{-2}$. As the result, the burnout heat flux for the nucleate boiling with an impinging jet of saturated liquid is

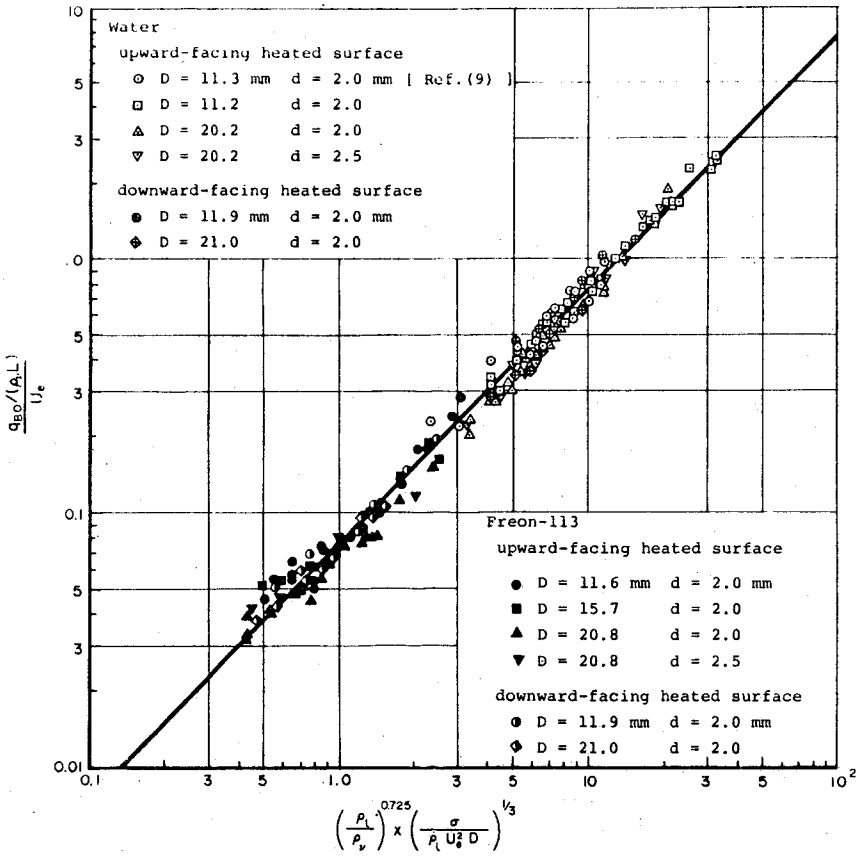

FIG. 10. Nondimensional correlation of burnout heat flux (saturated boiling).

correlated by

$$\frac{q_{B.O}/(\rho_l L)}{u_e} = 7.45 \times 10^{-2}\left(\frac{\rho_l}{\rho_v}\right)^{0.725}\left(\frac{\sigma}{\rho_l \cdot u_e^2 D}\right)^{1/3} \quad (5)$$

keeping such a fine accord with the data as is shown by the heavy solid line in Fig. 10. It should be noted that the burnout is regarded as a phenomenon originated mainly under the effect of the surface tension σ in either case of equations (3) and (5).

5.3 *Appendix*

In connection with the fact that both equations (3) and (4) can be derived from the general equation (2), it may be of use to give the following note. For the conventional heat transfer in a forced flow of single phase in a gravitational field, there are eight quantities concerned (h: heat-transfer coefficient, D: length, λ: thermal conductivity, μ: viscosity, c_p: specific heat, ρ: density, u: velocity of forced flow, $g\rho\beta\Delta T$: difference of specific weight due to thermal expansion) against four fundamental units (mass, length, time, and heat divided by temperature), so that four independent dimensionless groups appear to give the relation:

$$\frac{hD}{\lambda} = C\left(\frac{\mu c_p}{\lambda}\right)^{\alpha_1}\left(\frac{\mu}{\rho u D}\right)^{\alpha_2}\left(\frac{Dg\beta\Delta T}{u^2}\right)^{\alpha_3}. \quad (6)$$

For pure free convection, the exponent of u in equation (6) should vanish, so that $\alpha_2 = -2\alpha_3$, and thereby equation (6) is rewritten as:

$$\frac{hD}{\lambda} = C\left(\frac{\mu c_p}{\lambda}\right)^{\alpha_1}\left(\frac{D^3 g\beta\Delta T}{\nu^2}\right)^{\alpha_1} \quad (7)$$

where ν is the kinematic viscosity. On the other hand, for pure forced convection, the exponent of g in equation (6) should vanish, so that $\alpha_3 = 0$ to give

$$\frac{hD}{\lambda} = C\left(\frac{\mu c_p}{\lambda}\right)^{\alpha_1}\left(\frac{uD}{\nu}\right)^{-\alpha_2}. \quad (8)$$

It is unnecessary to say that either of equations (7) and (8) is a very popular expression in the field of single-phase heat transfer.

6. EFFECT OF LIQUID SUBCOOLING

6.1. *Experimental result*

Experiment of subcooled boiling on the heated surface of 11–21 mm in diameter has been made for the impinging jet of 2 mm in diameter with subcooling ($T_{sat} - T_{liq}$) up to 30°C for water and up to 16°C for freon-113.

As for the heat transfer of nucleate boiling, it is found that when the heat flux q is plotted against ($T_w - T_{sat}$), the data deviate from the saturated nucleate boiling

curve in the range of low $(T_w - T_{sat})$, and the deviation increases as the liquid subcooling becomes higher. As $(T_w - T_{sat})$ is raised, however, the effect of subcooling on heat transfer disappears rapidly and the data begin to fall on the saturated nucleate boiling curves shown in Figs. 3 and 4, indicating that $(T_w - T_{sat})$ is the effective driving force for heat transfer. At the same time, the similar behavior of fluid as described in Section 4.1 can be observed, that is, the splash of liquid takes place all over the heated surface except near the impinging zone. Accordingly, it may be presumed that the heated surface is covered with the self-supporting nucleate-boiling liquid layer, while the upper liquid-flow (cf. Fig. 7) is probably subcooled in this case, so that a part of the vapor rising from the underlying boiling layer may be condensed into liquid.

Then, if $(T_w - T_{sat})$ is raised further, it finally results in the onset of burnout. As for the burnout heat flux $q_{B.O}$ in the subcooled boiling, it is found that $q_{B.O}$ increases as the liquid subcooling $(T_{sat} - T_{liq})$ is increased. Besides, what is of more interest is that the relation of equation (1) still holds in the case of subcooled boiling if $(T_{sat} - T_{liq})$ is kept constant.

6.2. *Correlation of burnout heat flux data*

On the basis of equation (5), let us assume the following expression for the burnout in subcooled boiling:

$$\frac{q_{B.O}/(\rho_v L)}{u_e} = 7.45 \times 10^{-2}\left(\frac{\rho_l}{\rho_v}\right)^{0.725} \times \left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3}(1+\varepsilon_{sub}) \quad (9)$$

where ε_{sub} is the correction factor for the effect of liquid subcooling on the burnout heat flux. Of course, ε_{sub} vanishes for the saturated liquid, while ε_{sub} increases as the liquid subcooling is increased.

Now, it may be assumed that the specific heat of liquid c_{pl}, the subcooling of liquid $(T_{sat} - T_{liq})$, and thermal conductivity of liquid λ_l join the six quantities included in equation (9) but $q_{B.O}$, to exert respective effects on ε_{sub}. Dimensional analysis, such as employed in Sections 5.2 and 5.3, can be applied to the above-mentioned nine quantities to give:

$$\varepsilon_{sub} = K\left(\frac{\rho_l}{\rho_v}\right)^m\left[\frac{c_{pl}(T_{sat}-T_{liq})}{L}\right]^n \times \left(\frac{\sigma}{\rho_l u_e^2 D}\right)^r\left[\frac{\lambda_l(T_{sat}-T_{liq})}{\rho_l u_e L D}\right]^s \quad (10)$$

where K is constant. Then, according to the empirical fact described in the preceding section that equation (1) still holds in the subcooled boiling, the exponents of u_e and D should vanish in equation (10) to give $r = s = 0$, so that equation (10) is written as:

$$\varepsilon_{sub} = K\left(\frac{\rho_l}{\rho_v}\right)^m\left[\frac{c_{pl}(T_{sat}-T_{liq})}{L}\right]^n.$$

Then K, m and n in the above expression can be determined from the experimental result as follows:

$$\varepsilon_{sub} = 2.7\left(\frac{\rho_l}{\rho_v}\right)^{0.5}\left[\frac{c_{pl}(T_{sat}-T_{liq})}{L}\right]^{2.0} \quad (11)$$

and Fig. 11 shows the comparison between the experimental data of $q_{B.O}$ in subcooled boiling and the prediction of equation (9) with the correction factor ε_{sub} of equation (11). In addition, Table 1 shows the rate of increase of burnout heat flux $(1+\varepsilon_{sub})$, calculated by equation (11) for the range of subcooling $(T_{sat} - T_{liq})$ experienced in the present study.

Table 1. Rate of increase of burnout heat flux due to liquid subcooling

Water		Freon-113	
$T_{sat} - T_{liq}$ (°C)	$1+\varepsilon_{sub}$	$T_{sat} - T_{liq}$ (°C)	$1+\varepsilon_{sub}$
3	1.004	3	1.015
10	1.038	10	1.159
20	1.151	16	1.407
30	1.341		

Finally, in connection with the form of equation (11), it may be of interest to note that the correction factor ε_{sub} suggested by Ivey and Morris [13] in the case of subcooled pool boiling is of the following form

$$\varepsilon_{sub} = 0.1\left(\frac{\rho_l}{\rho_v}\right)^{3/4}\left[\frac{c_{pl}(T_{sat}-T_{liq})}{L}\right]^{1.0}$$

which is of the form similar to equation (11).

7. CONCLUSIONS

1. Experimental study has been made on the fully-developed nucleate boiling at very high heat fluxes in a forced-convection boiling system, combining a heated disk surface with a steady, cylindrical liquid jet, which is of a small diameter compared with the diameter of heated surface and impinges on the center of the heated surface at a right angle.

2. As for the flow of vapor and liquid in the above-mentioned boiling system, an idealized flow model may be assumed so as to accord with the observations on the heat transfer as well as on the fluid behavior. However, it seems very difficult to explain the onset of burnout by means of such a superficial flow model alone.

3. It has been found that if a conventional, dimensional analysis is applied to the burnout heat flux in a boiling with a forced flow in a gravitational field, it provides a dimensionless, generalized equation capable of correlating the experimental data, not only in the case of the ordinary pool boiling, but also in the case of the present boiling system with an impinging jet. In addition, surface tension is a common factor for originating the burnout, and it suggests a possibility that there may be a common, hydrodynamic principle for the onset of burnout in the two cases above-mentioned, in spite of the quite different model of flow.

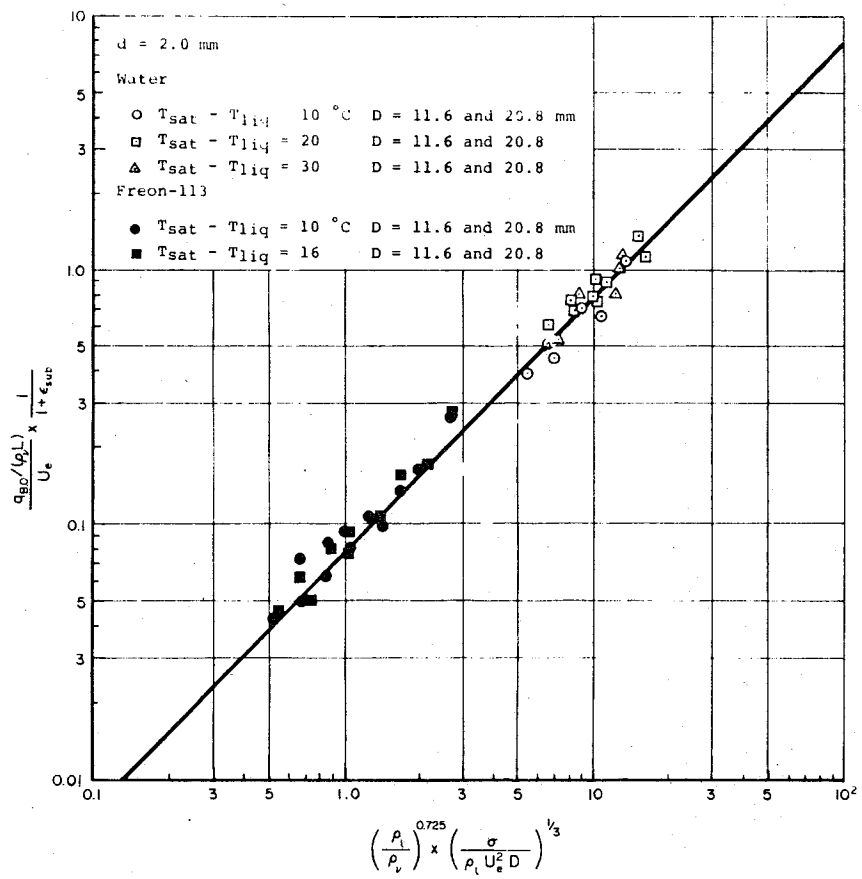

FIG. 11. Nondimensional correlation of burnout heat flux (subcooled boiling).

4. The generalized correlation of the burnout heat flux data in the present boiling system with an impinging jet has been determined as:

$$\frac{q_{B.O}/(\rho_l L)}{u_e} = 7.45 \times 10^{-2}\left(\frac{\rho_l}{\rho_v}\right)^{0.725} \times \left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3}(1+\varepsilon_{sub})$$

where ε_{sub}, the correction factor for liquid subcooling, is given as:

$$\varepsilon_{sub} = 2.7\left(\frac{\rho_l}{\rho_v}\right)^{0.5}\left[\frac{c_{pl}(T_{sat}-T_{liq})}{L}\right]^{2.0}$$

and the above correlation preserves a fairly good agreement with the experimental data.

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BURNOUT DANS UN SYSTEME BOUILLANT A HAUT FLUX THERMIQUE AVEC UN JET INCIDENT

Résume Une étude expérimentale porte sur l'ébullition nucléée pleinement développee a la pression atmosphérique dans un système à ébullition en convection forcée qui consiste en une surface chauffée plane et un petit jet à grande vitesse d'eau ou de fréon 113 frappant la surface. On obtient une formule généralisée, pour le flux de crise, appliquée aussi bien à l'eau qu'au fréeon 113 et on montre que la tension superficielle joue un role important sur la mise en place du phénomène de crise non seulement dans l'ébullition en réservoir ordinaire mais aussi dans le cas présent avec un écoulement forcé.

BURNOUT IN EINEM SIEDESYSTEM HOHER WÄRMESTROMDICHTE MIT EINEM AUFPRALLENDEN STRAHL

Zusammenfassung – Es wurde das voll ausgebildete Blasensieden unter Atmosphärendruck an einem einfachen Siedesystem, das aus einer ebenen Heizfläche und einem, mit hoher Geschwindigkeit aufprallenden, kleinen Wasser- oder R 113-Strahl bestand, experimentell untersucht. Für die Burnout-Wärmestromdichte wird eine allgemeine Beziegung aufgestellt und mit Erfolg auf Wasser und R 113 angewandt. Es wird gezeigt, daß die Oberflächenspannung nicht nur beim gewöhnlichen Behältersieden sondern auch bei dem hier betrachteten Siedesystem mit erzwungener Konvektion von großem Einfluß auf das Einsetzen des Burnout ist.

КРИЗИС КИПЕНИЯ В СИСТЕМЕ С ПАДАЮЩЕЙ СТРУЕЙ ПРИ БОЛЬШОЙ ПЛОТНОСТИ ТЕПЛОВОГО ПОТОКА

Аннотация — Проведено экспериментальное исследование полностью развитого пузырькового кипения при атмосферном давлении и вынужденной конвекции в простой системе, состоящей из нагреваемой плоской поверхности и падающей на неё с большой скоростью струйкой воды или фреона-113. Выведено соотношение, обобщающее данные по критическому тепловому потоку, которое является справедливым как для воды, так и для фреона. Показано, что как и при обычном кипении в большом объеме, поверхностное натяжение оказывает большое влияние на наступление кризиса в рассматриваемой системе с вынужденным течением.

BURNOUT IN A HIGH HEAT FLUX BOILING SYSTEM WITH A FORCED SUPPLY OF LIQUID THROUGH A PLANE JET

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ABSTRACT

Experimental study has been made on burnout in nucleate boiling at high heat fluxes on a heated rectangular copper surface, to which a saturated liquid at atmospheric pressure (water, freon-113, and trichrolethane) is supplied through a plane jet flowing out of a very thin rectangular nozzle set up near the front edge of the heated surface with a prescribed incidence angle. Burnout heat flux (that is, critical heat flux) is measured for the heated surface of 10 to 20mm in length with a plane jet of 0.56 to 0.77mm in thickness, 15° to 60° in incidence angle to the heated surface, and 1.5 to 15m/s in velocity. All the data of burnout heat flux thus obtained are correlated well by a generalized correlation equation, in which the dimensionless burnout heat flux is given as a function of Weber number and the density ratio. Finally, it is shown that this correlation is approximately similar, not only in form but also in quantity, to the correlation equation of burnout in the boiling system with a small round jet vertically impinging on the center of a heated surface.

NOMENCLATURE

C	= constant (dimensionless)
c_{pl}	= specific heat of liquid at constant pressure, J/kgK
D	= diameter (or diagonal) of a heated disk (or square) surface, m
D_o	= diameter of a heated cylindrical surface, m
g_n	= acceleration of gravity, m/s^2
Δh_v	= specific latent heat of vaporization, J/kg
l	= length of a heated rectangular surface in direction of flow, m
$\dot{q}$	= heat flux density, W/m^2
S	= thickness of liquid jet at nozzle exit, m
T	= temperature, K
u_e	= velocity of liquid jet at nozzle exit, m/s
u_o	= velocity of uniform flow of liquid, m/s
η	= dynamic viscosity, kg/ms
ρ	= mass density, kg/m^3
σ	= surface tension, N/m
ϕ	= incidence angle of a liquid jet with respect to the heated surface

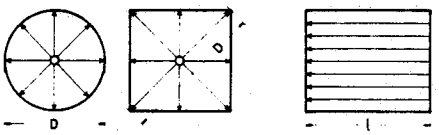
Subscripts

$B.O$	= burnout
l	= saturated liquid
lio	= liquid flowing out of a nozzle
s	= saturation
v	= saturated vapor
w	= heated surface

1. INTRODUCTION

As for pool boiling, Kutateladze (1), Rohsenow and Griffith (2), Zuber, Tribus and Westwater (3), and others have already succeeded in deriving generalized correlations for burnout heat flux in fundamental, simple systems of boiling. For forced-convection boiling, however, the situation is somewhat different from above. The study of burnout in convective boiling with forced flow inside of tubes, annuli, or outside of rod bundles, etc. has been carried on mainly under the necessity of safety of nuclear power plant, but it may be no exaggeration to say that little success has been obtained for the derivation of generalized correlation due to the complex phenomena included. Facing such circumstances, it may not be useless to direct our attention to the burnout in much simpler systems of convective boiling. Recently, Lienhard and Eichhorn (4), being motivated by their interest in finding a relation between pool boiling and flow boiling situations, made a study of burnout on cylinders of diameter D_o in a vertical crossflow of velocity u_o to provide two types of generalized correlations for burnout heat flux, in which the dimensionless group $\dot{q}_{B.O}/(\rho_v \Delta h_v u_o)$ is given as a function of the density ratio ρ_l/ρ_v and the cube root of the reciprocal of Weber number $\sigma/(\rho_v u_o^2 D_o)$.

On the other hand, Monde and Katto (5,6) made a study of burnout in boiling on a heated disk (or square) surface facing upward and downward, which is supplied with a liquid through a small round jet impinging vertically on the center of the heated



(a) BOILING SYSTEM WITH AN IMPINGING JET

(b) BOILING SYSTEM WITH A PLANE JET

Fig.1

surface as shown in Fig.1(a). According to their studies, violent splash of liquid from the heated surface appears, but burnout starts when the radial flow of liquid on the heated surface still remains overflowing the outer edge of heated surface, and burnout heat flux for the heated surface of diameter (or diagonal) D with an impinging jet of velocity u_e is well correlated by the following generalized correlation equation:

$$\frac{\dot{q}_{B.O}}{\rho_v \Delta h_v u_e} = 0.0745 \left(\frac{\rho_l}{\rho_v}\right)^{0.725} \left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3} (1+\varepsilon_{sub}) \quad (1)$$

where ε_{sub}, the correction parameter for the sub-cooling of liquid flowing out of the nozzle, is given as:

$$\varepsilon_{sub} = 2.7 \left(\frac{\rho_l}{\rho_v}\right)^{0.5} \left[\frac{c_{pl}(T_s - T_{liq})}{\Delta h_v}\right]^{2.0} \quad (2)$$

However, the boiling system with an impinging jet mentioned above is a special case on the point that the liquid supplied at the center of heated surface flows radially over the heated surface. Therefore, in order to get a more definite view of the burnout in this type of boiling system, it is at least necessary to gain information about the burnout in case of the boiling with such a parallel, or one-dimensional, flow of liquid on the heated surface as is shown in Fig.1(b).

In the present study, therefore, a series of experiments is made on burnout in the boiling system with a plane jet flowing out of a thin rectangular nozzle set up near the front edge of a heated rectangular surface.

2. EXPERIMENTAL APPARATUS

The whole system of experimental apparatus is shown in Fig.2. Liquid supplied from the tank/4/ through the pump/6/ to the rectangular nozzle/9/ is heated by the heater/5/ as well as the temperature regulator/7/ up to the temperature T_{liq}, which is 3 to 5 K lower than the saturation temperature T_s. Even for the highest temperature difference of $T_s - T_{liq}$ = 5 K, however, ε_{sub} in Eq.(2) is 0.01 to 0.04 at most for the liquids tested (water, freon-113, and trichrolethane), so that it may be allowed to assume the liquid flowing out of the nozzle/9/ to be saturated liquid.

Main part of experimental apparatus, shown in Fig.3, consists of the boiling vessel/1/, the copper block /2/ and the rectangular nozzle/4/. The boiling vessel/1/ has been designed to keep the interior at atmospheric pressure. The under plane surface of the copper block/2/, heated by electric plate heaters mounted in grooves of the block, provides the heated surface facing downward, which is rectangular in shape with the width of 15mm. As for the length of the heated surface in direction of flow, three kinds of length (l = 10, 15, and 20mm) are tested. A plane jet of liquid, flowing out of the rectangular nozzle of the same width as that of the heated surface, has been designed to strike the position of about 2mm before the front edge of the heated surface to avoid the effect of so-called impinging zone. In order to secure the one-dimensionality of flow on the heated surface, a thin plate is fitted

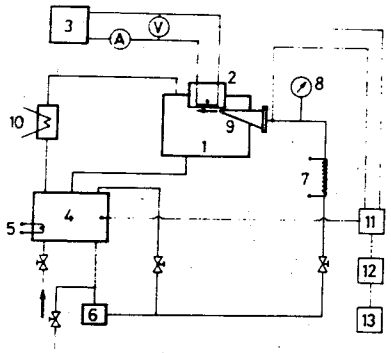


Fig.2 WHOLE SYSTEM OF EXPERIMENTAL APPARATUS

/1/boiling vessel, /2/copper block, /3/constant voltage transformer, /4/tank, /5/heater, /6/pump, /7/temperature regulator, /8/pressure gauge, /9/rectangular nozzle, /10/condenser, /11/ice-box, /12/self-registering thermometer, /13/digital multi-meter

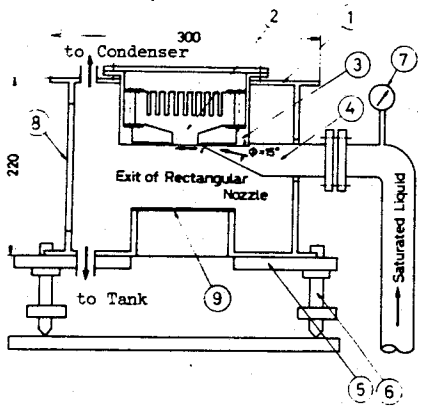


Fig.3 MAIN PART OF EXPERIMENTAL APPARATUS

/1/boiling vessel, /2/copper block, /3/heating capsule, /4/rectangular nozzle, /5/supporting member, /6/height adjustment screw, /7/pressure gauge, /8/window, /9/window

along each lengthwise side of a heated surface. Experiments are made for two kinds of thickness of jet at nozzle exit (S_e = 0.56 and 0.77mm), two kinds of incidence angle of jet with respect to the heated surface (ϕ = 15° normally, and ϕ = 60° to examine the effect of ϕ on $\dot{q}_{B.O}$), and the velocity of jet at nozzle exit ranging over u_e = 1.5 to 15 m/s. It should be noted that Fig.3 shows the normal case of ϕ = 15° with respect to the incidence angle of jet.

The temperature of heated surface T_w as well as the

heat flux density across the heated surface $\dot{q}$ is measured by three Chromel-Alumel thermocouples set up at regular intervals along the center axis of copper block, for which the thermal conductivity of copper block has been determined in the preliminary experiment by measuring the latent heat absorbed by the vapor generated on the heated surface in the saturated nucleate pool boiling of water at high heat fluxes of 10^5 to 10^6 W/m^2 (cf. pool boiling data in Fig.4).

The velocity of liquid jet at the exit of nozzle u_e can be accurately determined by measuring the pressure difference between the pressure gauge/7/ and the inside of boiling vessel/1/ in Fig.3 if the discharge coefficient of the nozzle has been determined in the preliminary experiment.

3. NUCLEATE BOILING OBSERVED

3.1 Nucleate boiling heat transfer

As is mentioned in the preceding chapter, preliminary experiments are made for saturated nucleate pool boiling of water on the heated rectangular surface employed in the present study. The data of pool boiling heat transfer thus obtained in three runs are shown in Fig.4, together with the prediction of the correlation equation proposed by Nishikawa and Yamagata (7) for nucleate pool boiling heat transfer.

Furthermore, Fig.4 shows typical data of heat transfer obtained in the convective boiling system with a forced supply of liquid through a plane jet, together with the mean nucleate boiling curve obtained by Monde and Katto (6) for the convective boiling system with an impinging round jet.

Apart from the scattering of data inherent in nucleate boiling, it may be concluded from Fig.4 that within the experimental range of jet velocity, nucleate boiling can reach the fully-developed state

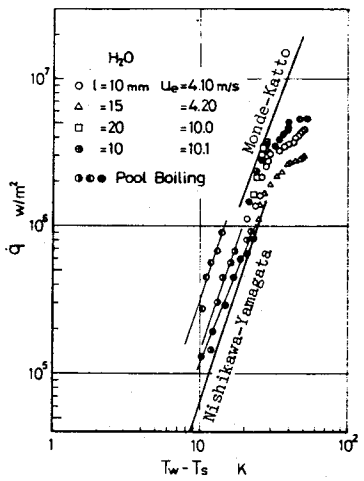


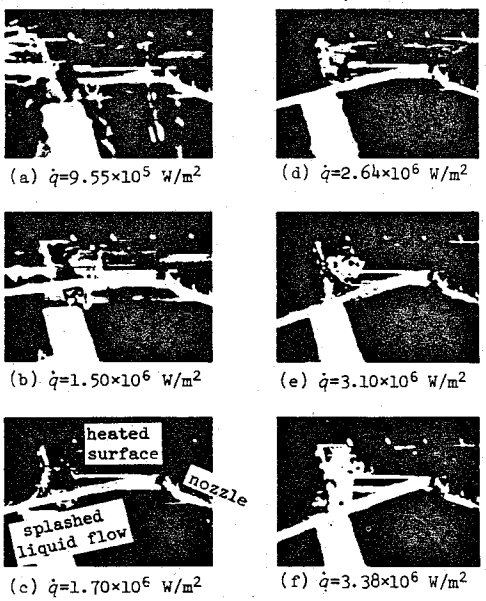
Fig.4 NUCLEATE BOILING DATA FOR WATER

in which the boiling is so violent that it is no longer affected with the forced convection. It is also noticed in Fig.4 that the boiling curve for the convective boiling system with a plane jet tends to change its gradient to some extent before the onset of burnout. However, it should be mentioned that this is the phenomenon observed in the boiling of water alone, and no change of gradient appears in the boiling curves of freon-113 and trichrolethane.

3.2 Variation of fluid behavior with heat flux

Fig.5 shows a typical example for the variation of fluid behavior observed as heat flux is raised in boiling of water on the heated 10-mm length surface with a plane jet of velocity 3.82m/s. In the state (a), nucleate boiling is too week to make a noticeable splash of liquid, so that the liquid is left to flow along the heated surface. In the state of (b), however, one can see a coherent flow of splashed liquid which begins to separate from the heated surface near the rear edge of the heated surface. Thereafter, as heat flux is increased further, the proportion of splashed liquid increases, and the location of separation tends to move toward the front edge of the heated surface being accompanied by the increase of splash angle. Then, in the state of (f), which is near the burnout point, it may be presumed from the study of reference (6) that the greater part of liquid supplied through a nozzle is splashed away near the front edge of heated surface, but there is still a quantity of residual liquid flowing over the rear edge of heated surface.

For freon-113 and trichrolethane, more or less similar features as above are observed, but it is no-



(a) $\dot{q}$=9.55×10^5 W/m^2 (d) $\dot{q}$=2.64×10^6 W/m^2

(b) $\dot{q}$=1.50×10^6 W/m^2 (e) $\dot{q}$=3.10×10^6 W/m^2

(c) $\dot{q}$=1.70×10^6 W/m^2 (f) $\dot{q}$=3.38×10^6 W/m^2

Fig.5 VARIATION OF FLUID BEHAVIOR (Water, ℓ = 10mm, u_e = 3.82m/s)

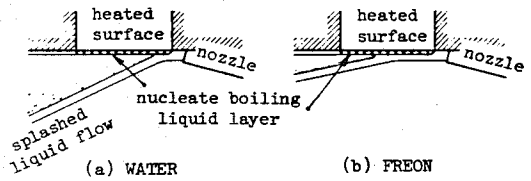
ticed that the splash angle is less than that of water, and the separation of a coherent flow of splashed liquid is not so clear as it is in the case of water. For these liquids, it must be remembered that the apparant velocity of vapor from the heated surface $\dot{q}/(\rho_v \Delta h_v)$ is on a low level, that is, the ejection of vapor from the heated surface is not strong.

4.BURNOUT, AND BURNOUT HEAT FLUX

4.1 Flow state at the onset of burnout

In case of the boiling system with an impinging jet such as studied in references (5) and (6), the radial splash of liquid from the heated surface is apt to obstruct the clear observation of flow state. The present boiling system with a plane jet, however, has the advantage of geoemtrical situation which does not obstruct the observation as is noticed in Fig.5.

Then, based on the photographic observation, flow models such as shown in Fig.6 are assumed for the state of flow just before the onset of burnout.



(a) WATER (b) FREON

Fig.6 FLOW MODEL

In case of water, the greater part of liquid is splashed away near the front edge of heated surface, and almost all part of the heated surface is covered with a layer of nucleate boiling liquid in which vigorous nucleate boiling is maintained being accompanied by the ejection of vapor to the space. In case of freon (or trichrolethane), the separation point of a coherent flow of splashed liquid is rather obscure and does not approach the front edge of heated surface when the burnout takes place; and this situation is particularly noticeable if the length of heated surface is longer and the velocity of jet is lower.

Just after the onset of burnout in either state of (a) and (b) in Fig.6, the dryout starts from the rear edge of heated surface to extend up to the vicinity of the front edge of heated surface, where all the liquid is splashed away making a coherent flow of liquid in the space under the heated surface.

4.2 Burnout heat flux data

All the data of burnout heat flux $q_{B.O}$ obtained for three kinds of liquid under the normal incidence angle $\phi = 15°$ are shown in Fig.7, in which a dimensionless group with respect to the burnout heat flux $\dot{q}_{B.O}/(\rho_v \Delta h_v u_e)$ is plotted against the reciprocal of Weber number $\sigma/(\rho_l u_e^2 l)$. In connection with the thickness of liquid jet at the exit of nozzle S_e, experimental data fall into two groups, corresponding to $S_e = 0.56$ and 0.77mm, nearly half-and-half. However, S_e has no effect on $\dot{q}_{B.O}$ in the experimental range of S_e (cf. Appendix) so that the data in Fig.7 have been shown without discimination of S_e. It is noted in Fig.7 that $\dot{q}_{B.O}/(\rho_v \Delta h_v u_e)$ makes a change in proportion to the cube root of $\sigma/(\rho_l u_e^2 l)$ for every liquid.

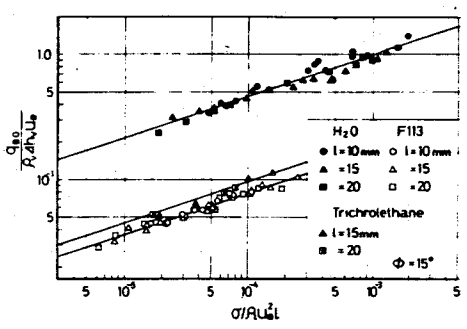


Fig.7 BURNOUT HEAT FLUX DATA FOR WATER, FREON-113, AND TRICHROLETHANE

4.3 Effect of incidence angle of liquid jet

So far the data for the jet of incidence angle $\phi = 15°$ alone are dealt with, but it is important to examine whether the incidence angle of jet has an effect on burnout heat flux or not. Then, Fig.8 indicates the apparatus for the boiling of water supplied through a plane jet of incidence angle $\phi = 60°$; and the data of burnout heat flux thus obtained are shown in Fig.9. A slanting solid line in Fig.9 is the same as that of water drawn in Fig. 7. The data seem to deviate a little from the slanting solid line in Fig.9, but if the scattering of data of water in Fig.7 is taken into account, it may be concluded without serious error that burnout heat flux is hardly affected by the incidence angle of liquid jet.

4.4 Generalized correlation of burnout heat flux

Following the well-known view that burnout is a phenomenon connected with hydrodynamic states, Monde and Katto (6) have presented fundamental dimensionless groups to govern the burnout heat flux in simple systems of boiling as follows:

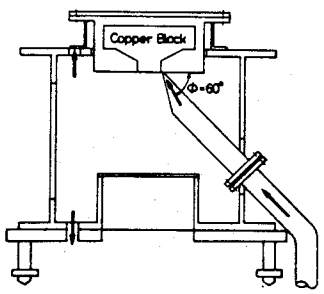


Fig.8 EXPERIMENTAL APPARATUS FOR $\phi = 60°$

$$\frac{\dot{q}_{B.O}}{\rho_v \Delta h_v u_e} = C\left(\frac{\rho_l}{\rho_v}\right)^{n_1}\left(\frac{\eta_v}{\eta_l}\right)^{n_2}\left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{n_3}\left(\frac{\eta_l}{\rho_l u_e l}\right)^{n_4} \times \left[\frac{g_n(\rho_l-\rho_v)l}{\rho_l u_e^2}\right]^{n_5} \qquad (3)$$

where u_e: velocity of forced flow, l: representative length of heated surface, and $g_n(\rho_l-\rho_v)$: difference of specific weight between liquid and vapor. On the right-hand side of Eq.(3), the third dimensionless group (the reciprocal of Weber number) refers to the surface tension, the fourth group (the reciprocal of Reynolds number) to the viscous force, and the fifth group to the gravitational force. The inertia force refers to all the three groups mentioned above.

Now, it may be assumed that burnout in the present convective boiling system is not affected by the gravitational force. In addition, most probably the viscosity of fluid has little effect on burnout heat flux in the case of ordinary fluids. Accordingly, neglecting the effect of gravity and viscosity in Eq.(3), it yields

$$\frac{\dot{q}_{B.O}}{\rho_v \Delta h_v u_e} = C\left(\frac{\rho_l}{\rho_v}\right)^{n_1}\left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{n_3} \qquad (4)$$

where u_e may be regarded as the jet velocity, and l the length of heated surface in the direction of flow.

Eq.(4) can be compared with three slanting lines in Fig.7, fixing the value of n_3 to be 1/3. Then, the data of water, freon-113, and trichrolethane are compared to determine the remaining two constants of C and n_1 in Eq.(4) to give the following generalized correlation equation:

$$\frac{\dot{q}_{B.O}}{\rho_v \Delta h_v u_e} = 0.0164\left(\frac{\rho_l}{\rho_v}\right)^{0.867}\left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{1/3} \qquad (5)$$

Fig.10 shows the comparison of Eq.(5) with all the experimental data obtained in the present study.

5. COMPARISON TO THE BOILING SYSTEM WITH AN IMPINGING JET

Eq.(5), obtained for the boiling system with a plane jet, is similar in form to Eq.(1), obtained for the boiling system with an impinging round jet. Particularly, it is surprising to see that the power of Weber number is identical for both Eqs.(1) and (5).

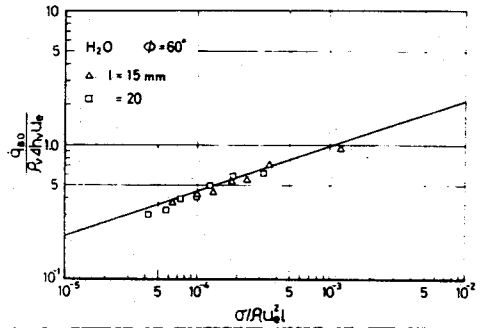


Fig.9 EFFECT OF INCIDENCE ANGLE OF JET ON BURNOUT HEAT FLUX

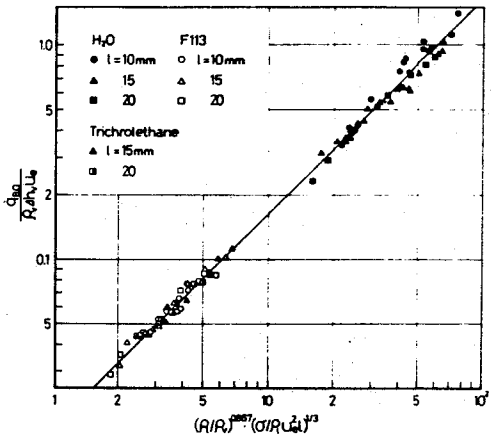


Fig.10 NONDIMENSIONAL CORRELATION OF BURNOUT HEAT FLUX

The above-mentioned matter suggests the the existence of a strong similarity between the two boiling systems for the mechanism to provoke the burnout. In order to make a quantitative comparison between Eqs.(1) and (5), however, D in Eq.(1) should be replaced by $2l$, because l is regarded as the maximum flow length on a heated surface measured from the location of incidence of liquid jet (see Fig.1). Eq.(1) is thus rewritten as:

$$\frac{\dot{q}_{B.O}}{\rho_v \Delta h_v u_e} = 0.0591\left(\frac{\rho_l}{\rho_v}\right)^{0.725}\left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{1/3} \qquad (6)$$

Fig.11 shows the comparison between Eq.(6) and the burnout heat flux data obtained for the boiling system with a plane jet in the present study. Of course, the agreement is not so good as it is in Fig.10, but at the same time, it is noticed that Eq.(6) is not so far from the accurate correlation. To sum up, it may be concluded that approximately

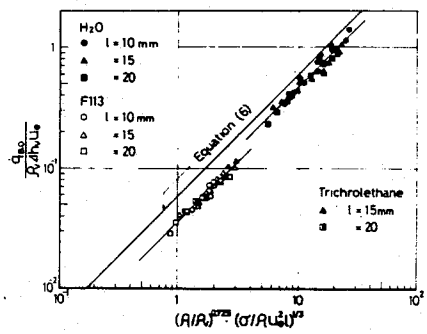


Fig.11 COMPARISON OF EQ.(6) WITH EXPERIMENTAL DATA OBTAINED IN THE PRESENT STUDY

speaking, Eq.(1) is akin to Eq.(5) not only in form but also in quality.

6. CONCLUSIONS

(1) Burnout phenomenon has been studied for the nucleate boiling system in which a saturated liuid (water, freon-113, and trichrolethane) is supplied through a thin plane jet set up near the front edge of a heated rectangular surface.

(2) Burnout takes place under the state of fully-developed nucleate boiling, in which nucleate boiling is so vigorous that the boiling heat transfer is no longer affected by the forced convection.

(3) In the fully-developed nucleate boiling at sufficiently high heat fluxes, part of the liquid supplied from the plane jet is splashed away from the heated surface by the vapor ejected from a layer of nucleate boiling liquid covering the heated surface. However, the layer of nucleate boiling liquid itself is ever maintained on the heated surface until burnout takes place.

(4) As for the plane jet, its incidence velocity alone is important for burnout heat flux. Neither the thickness (cf. Appendix) nor the incidence angle has an effect on burnout heat flux.

(5) As for the heated rectangular surface, its length has an important effect on burnout heat flux, but it should be noted that the burnout heat flux is not affected by the viscosity of liquid.

(6) Burnout heat flux data obtained in the present boiling system can be correlated well by the generalized correlation equation:

$$\frac{\dot{q}_{B.O}}{\rho_v \Delta h_v u_e} = 0.0164 \left(\frac{\rho_l}{\rho_v}\right)^{0.867} \left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{1/3}$$

(7) As far as burnout is concerned, it seems likely that there are affinitites between the boiling system with a plane jet and the boiling system with an impinging round jet.

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APPENDIX

In the present study is excluded a simple kind of burnout which should appear if the flow rate of liquid jet is only deficient as compared with the total heat flux rate applied to the heated surface. This phenomenon has been observed, for example, in the study of Katto and Kunihiro (8). The criterion for not meeting with this kind of burnout at heat fluxes lower than $q_{B.O}$ in the present study, is readily obtained as follows:

$$\rho_l u_e S_e \Delta h_v > q_{B.O}\, l$$

or

$$\frac{S_e}{l} > \frac{\rho_v}{\rho_l} \frac{\dot{q}_{B.O}}{\rho_v \Delta h_v u_e} \qquad \text{(A.1)}$$

where S_e is the thickness of plane jet at the exit of nozzle, and l the length of heated surface. Substituting the left-hand side of Eq.(5) into the right-hand side of inequality (A.1), it yields

$$\frac{S_e}{l} > 0.0164 \left(\frac{\rho_v}{\rho_l}\right)^{0.133} \left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{1/3} \qquad \text{(A.2)}$$

In the present study, the value of the left-hand side of inequality (A.2) is greater than 0.028, while the value of the right-hand side of (A.2) is less than 0.00086 for water, 0.00062 for freon-113, and 0.00069 for trichroethane, because experiments have been made under the condition that $S_e \geq 0.56$mm, $u_e > 1.5$m/s, and 20mm $\geq l \geq$ 10mm. Accordingly, the criterion of (A.2) is sufficiently satisfied in the present study.

Physical properties of the above-mentioned liquids at atmospheric pressure are as follows:

for water (Ts = 100°C),
ρ_l = 958.3 kg/m^3, ρ_v = 0.5977 kg/m^3, Δh_v = 2258 $\times 10^3$ J/kg, and σ = 5.88 $\times 10^{-2}$ N/m;
for freon-113 (Ts = 47.6°C),
ρ_l = 1510 kg/m^3, ρ_v = 7.933 kg/m^3, Δh_v = 199 $\times 10^3$ J/kg, and σ = 1.50 $\times 10^{-2}$ N/m; and
for trichrolethane (Ts = 74.1°C),
ρ_l = 1230 kg/m^3, ρ_v = 4.68 kg/m^3, Δh_v = 223 $\times 10^3$ J/kg, and σ = 1.89 $\times 10^{-2}$ N/m.

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Upper Limit of CHF in the Saturated Forced Convection Boiling on a Heated Disk with a Small Impinging Jet

Critical heat flux (CHF) in the forced convection boiling on an open heated disk being supplied with saturated liquids through a small high-speed jet impinging at the center of disk is studied experimentally employing Refrigerant 12 at comparatively high pressures from 6.0 to 27.9 bars as well as water and Refrigerant 113 at atmospheric pressure. Generalized correlations of CHF are obtained for two characteristic regimes: V-regime where CHF is variable and I-regime where CHF is invariable for the change of jet velocity. Then, the boundaries of each regime are discussed clarifying the aspects for the lower limit of jet velocity capable of generating I-regime as well as the upper limit of CHF.

Introduction

Boiling on an open heated surface being supplied with a liquid by means of a small round or a thin plane high-speed jet may be regarded as one of the simplest systems associated with the forced convection boiling. Studies of critical heat flux (CHF) in such simple systems are important not only for industrial applications but also for obtaining basic information which may be useful in clarifying the mechanism of CHF in forced convection boiling.

Recently, Monde and Katto [1], employing water and Refrigerant 113 at atmospheric pressure, performed an experimental study for CHF in forced convection boiling on open heated disks (both upward- and downward-facing) supplied with liquid through a round jet of very small d/D impinging perpendicularly at the center of the heated disk, and the following correlation was obtained:

$$q_c = q_{co}\left\{1 + 2.7\left(\frac{\rho_\ell}{\rho_v}\right)^{0.5}\left(\frac{\Delta H_i}{H_{fg}}\right)^2\right\} \tag{1}$$

where q_{co}, critical heat flux for $\Delta H_i = 0$, was given by

$$\frac{q_{co}}{\rho_v H_{fg} u} = 0.0745\left(\frac{\rho_\ell}{\rho_v}\right)^{0.725}\left(\frac{\sigma}{\rho_\ell u^2 D}\right)^{1/3} \tag{2}$$

where u is the velocity of liquid jet measured at the exit of the nozzle. Then, CHF in another type of boiling system, where a thin plane jet supplied saturated liquid to just upstream of an edge of rectangular heated surface (downward-facing only), was investigated by Katto and Ishii [2], and their data obtained for water, R-113 and trichloroethane at atmospheric pressure were correlated by

$$\frac{q_{co}}{\rho_v H_{fg} u} = 0.0164\left(\frac{\rho_\ell}{\rho_v}\right)^{0.867}\left(\frac{\sigma}{\rho_\ell u^2 \ell}\right)^{1/3} \tag{3}$$

where ℓ is the length of heated surface in direction of flow. In Chapter 5 of [2], a comparison was made between equations (2) and (3) by replacing D in equation (2) by 2ℓ, because ℓ can be regarded as the maximum flow length measured from the location of incidence of liquid jet; and it indicates that $q_{co}/\rho_v H_{fg} u$ of equation (2) is not so different quantitatively from that of equation (3) within the experimental range. Then, it may be presumed that the difference in the exponent of ρ_ℓ/ρ_v between equations (2) and (3) is attributable to the different type of flow on the heated surface: that is, the radial flow in the former and the parallel flow in the latter.

It is interesting that the experimental data of CHF in the forced convection boiling systems mentioned above can be correlated by equations of simple form such as equations (2) and (3); and in justification of the form of these equations, a discussion based on dimensional analysis has been given in Sections 5.2 and 5.3 of [1]. Also, with respect to the common appearance of $(\sigma/\rho_\ell u^2 D)^{1/3}$ in equations (2) and (3), an explanation based on both hydrodynamic instability theory and vectorial dimensional analysis has been given in Chapter 2 of [14]. However, there is still a problem that equations (2) and (3) imply that q_{co} can increase boundlessly as the velocity of the liquid jet u is raised. Previous studies of boiling suggest that their dependence is unlikely to persist for all u, and therefore it is necessary to determine the upper limit for CHF in the forced convection boiling on open heated surface with a liquid jet, and also to examine the applicability of equations (2) and (3) at different pressures from atmospheric pressure.

In the present paper, CHF in the forced convection boiling of saturated R-12 at the comparatively high pressures of 6.0 to 27.9 bars is studied on a downward-facing heated disk surface of diameter $D = 0.01$ m with a small round impinging jet of diameter $d = 0.002$ m. The jet velocity u cannot exceed 20 m/s in the experiment due to the limit of experimental apparatus employed, but the surface tension of R-12 is smaller than those of water and R-113, and besides, it decreases as the pressure rises, so that the experimental range of $\sigma/\rho_\ell u^2 D$ is extended to much smaller value of $\sigma/\rho_\ell u^2 D$ with the same effect as that of increasing u. Furthermore, experiments of CHF of saturated water and R-113 at the atmospheric pressure are also made to test the reliability of the experimental apparatus in the present

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study as well as to extend the experimental range of ρ_ℓ/ρ_v.

Experimental Apparatus and Measuring Method of CHF

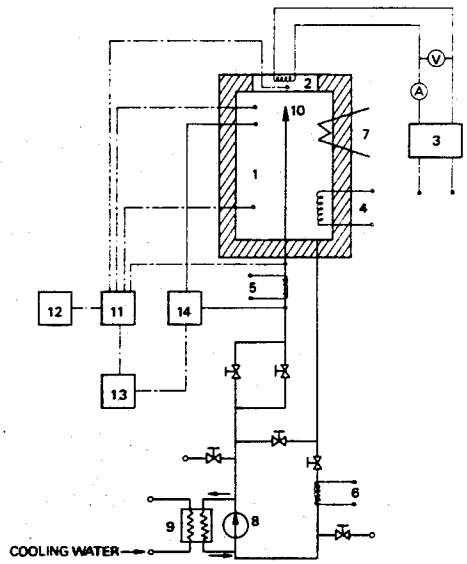
The experimental apparatus is shown schematically in Fig. 1. At the top of a pressure vessel (1) (the details of which are shown in Fig. 2) a copper block (2) is set up with eleven plate-type heaters in the upper conical part, to which electric power is supplied from an a-c transformer (3). The flat, end surface of the lower cylindrical part of the copper block provides a downward-facing heated circular surface of diameter $D = 0.01$ m. The circumferential surface of this cylindrical part is insulated with a ceramic sleeve, and the clearance between the cylindrical part and the insulator is filled with a plastic packing to prevent the contact between Refrigerant vapor and the very high temperature, inner region of the copper block, then avoiding thermal decomposition of the Refrigerant. The test liquid, which is pressurized by a peripheral pump (8) and then adjusted with the nozzle-heater (5) up to the saturation temperature corresponding to the internal pressure of the vessel, flows out of a nozzle (10) of inner diameter $d = 0.002$ m, impinging at the center of the heated disk. The velocity of the liquid jet at nozzle exit u can be determined by measuring the pressure difference between the inlet and the outlet of nozzle by means of a differential pressure transducer (14). The lower half of the inside of the pressure vessel is filled with the test liquid (about 0.4 to 0.5 m deep), which is heated by a vessel-heater (4), while the vapor above the liquid is condensed by a pair of cooling coils (7). The prescribed saturation pressure inside the vessel is maintained by controlling the vessel-heater as well as the cooling coils.

The heat flux across the heated disk as well as the temperature of heated surface is determined by means of three Chromel-Alumel thermocouples set up at regular intervals along the axis of the cylindrical part of copper block. The actual thermal conductivity of the copper block employed, indispensable to an accurate estimate of the heat flux, has been carefully determined in the preliminary experiment in the same way as that of [1, 2].

Critical heat flux is determined by the following means. Increasing the input to heaters in the copper block stepwisely with each increment less than five percent of the preceding heat flux, it finally arrives at the state in which the heated surface temperature cannot have a steady state rising very rapidly, when CHF is determined with an error within five percent. The flow state at the onset of CHF cannot be observed in the present study due to the employment of the pressure vessel, but it has been revealed in the preceding study [1] that once CHF takes place, the dryout occurs at the outer edge of the heated disk to extend toward the central impinging zone.

Experimental Data and Correlation of CHF

Experimental Result of CHF. Fig. 3 represents the experimental variation of CHF for R-12 with the jet velocity u, obtained at pressures $p = 6.00$, 11.6, 17.7, 23.5 and 27.9 bars respectively. It can be seen in Fig. 3 that at the lowest pressure, q_{co} increases with u over nearly the entire experimental range of u. As pressure rises, however, the region in which q_{co} is independent of u extends gradually from the high velocity region toward the low velocity region. In other words, two characteristic regimes can be distinguished for CHF in Fig. 3: one regime of variable q_{co} and the other regime of invariant q_{co}. In this paper, the former regime will be called V-regime, and the latter will be called I-regime for simplicity.



1. PRESSURE VESSEL 2. COPPER BLOCK 3. TRANSFORMER
4. VESSEL-HEATER 5. NOZZLE-HEATER 6. HEATER
7. COOLING COILS 8. PUMP 9. COOLER
10. NOZZLE 11. ICE BOX 12. RECORDER
13. DIGITAL VOLTMETER 14. PRESSURE TRANSDUCER

Fig. 1 System of experimental apparatus

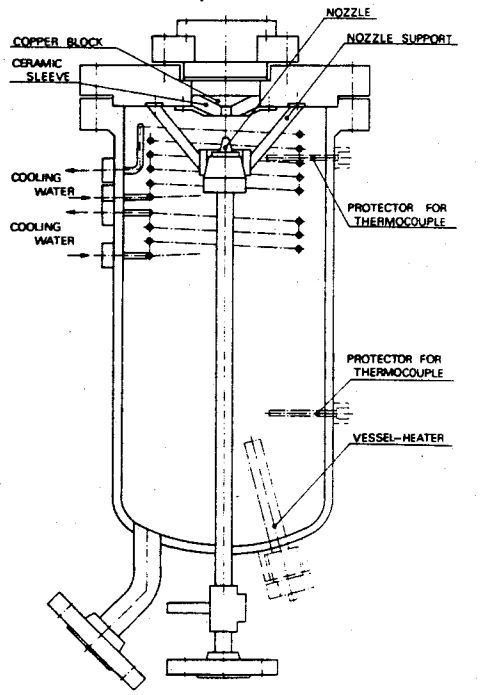


Fig. 2 Pressure vessel

Nomenclature

d = inner diameter of nozzle
D = diameter of heated disk
H_{fg} = latent heat of evaporation
ΔH_i = inlet subcooling enthalpy of liquid
p = absolute pressure
q_c = critical heat flux
$q_{co} = q_c$ for $\Delta H_i = 0$
$q_{co,max}$ = upper limit of q_{co}
u = velocity of liquid jet at nozzle exit
u_{min} = lower limit of u capable of generating $q_{co,max}$
ρ_ℓ = density of liquid
ρ_v = density of vapor
σ = surface tension

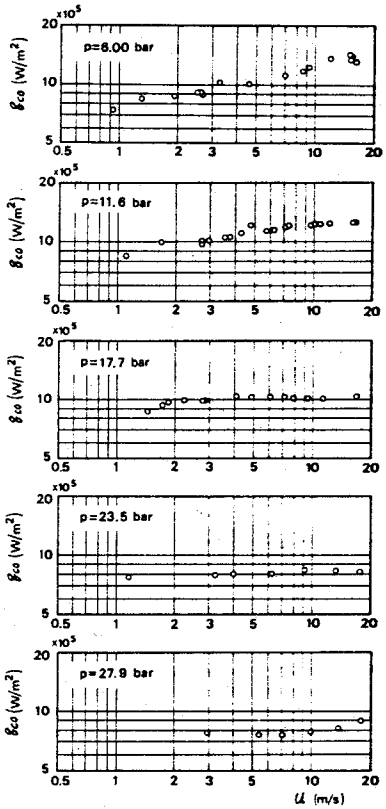


Fig. 3 Relation between CHF and jet velocity obtained for R-12

Analysis of CHF Data in *V*-regime. The relation between $(q_{co}/\rho_v H_{fg} u)/(\sigma/\rho_\ell u^2 D)^{1/3}$ and (ρ_ℓ/ρ_v) suggested by equation (2) for the experimental data classified to be in the *V*-regime is examined in the right side of Fig. 4. It may be noted in Fig. 4 that the *V*-regime data scatter to some extent, but the data of R-12 at pressures from 6.0 to 17.7 bars together with the data of water and R-113 at atmospheric pressure can be correlated by the higher solid line, yielding the following correlation equation:

$$\frac{q_{co}}{\rho_v H_{fg} u} = 0.188 \left(\frac{\rho_\ell}{\rho_v}\right)^{0.614} \left(\frac{\sigma}{\rho_\ell u^2 D}\right)^{1/3} \tag{4}$$

Among the *V*-regime data in Fig. 4, the R-12 data at $p = 17.7$ bars show a somewhat greater deviation from equation (4) than other data, but it should be noted that these data are situated within a limited range near the border with *I*-regime as seen in Fig. 3.

Equation (2) obtained in the experiment of [1] for water and R-113 at atmospheric pressure is also illustrated by the broken line in Fig. 4, and it is noted that the prediction of equation (2) agrees approximately with the present data for water, but falls a little lower than the present data for R-113. Recently, Monde, Kusuda and Uehara [3] carried out experiments for CHF of water and R-113 at atmospheric pressure for the boiling system with plural impinging jets on a heated disk of diameter $D = 0.0252$ m, giving a correlation equation of their experimental data, and if this correlation equation is reduced to the case of a single jet, it yields

$$\frac{q_{co}}{\rho_v H_{fg} u} = 0.170 \left(\frac{\rho_\ell}{\rho_v}\right)^{0.608} \left(\frac{\sigma}{\rho_\ell u^2 D}\right)^{1/3} \tag{5}$$

Equation (5) is illustrated by the chain line in Fig. 4, showing better agreement with the present result.

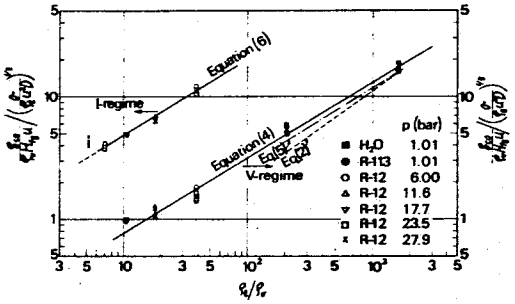


Fig. 4 Nondimensional correlations of CHF in *V* and *I*-regime

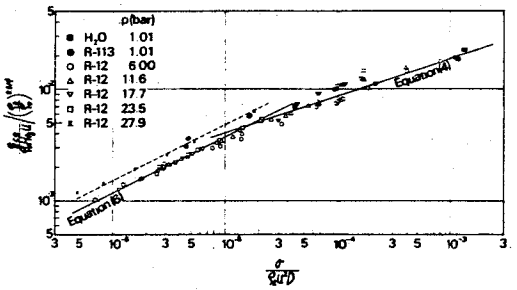


Fig. 5 Comparison of correlation equations (4) and (6) with experimental data

Analysis of CHF Data in *I*-regime. Regarding the experimental data for q_{co} classified to be in *I*-regime where q_{co} is regarded as independent of u, it may not be useless to test the correlation of data employing $(q_{co}/\rho_v H_{fg} u)/(\sigma/\rho_\ell u^2 D)^{1/2}$ and (ρ_ℓ/ρ_v), because $(q_{co}/\rho_v H_{fg} u)/(\sigma/\rho_\ell u^2 D)^{1/2}$ appears to be a dimensionless group which is substantially independent of u; and fortunately it gives acceptable results such as shown in the left side of Fig. 4. It is noted in Fig. 4 that all the *I*-regime data for R-12 except those at the highest pressure $p = 27.9$ bars can be correlated by the upper solid line, to give the following equation:

$$\frac{q_{co}}{\rho_v H_{fg} u} = 1.18 \left(\frac{\rho_\ell}{\rho_v}\right)^{0.614} \left(\frac{\sigma}{\rho_\ell u^2 D}\right)^{1/2} \tag{6}$$

The problem of discrepancy of data at $p = 27.9$ bars will be discussed later relating to characteristic regimes.

Consolidation of Equations (4) and (6). It is of interest to note that generalized correlation equations (4) and (6) derived for *V* and *I*-regime respectively take a common value for the exponent of ρ_ℓ/ρ_v, and owing to such circumstances, Fig. 5 can be presented, showing another type of comparison between all the experimental data obtained in the present study and the correlation equations (4) and (6). The aspect of transition between *V* and *I*-regime can be seen in Fig. 5. As for the data for R-113, however, somewhat dubious features are observed; the reason for this is unknown, leaving a problem to be solved in the future. At the present stage, therefore, it should be understood that the accuracy of the proposed equations (4) and (6) has a tolerance such as indicated by the experimental data in Fig. 5 at least. On the other hand, the singular data for R-12 at $p = 27.9$ bars are discriminated, in Fig. 5, from other data by the broken line, which is parallel to the line of equation (6) yielding the following relation:

$$\frac{q_{co}}{\rho_v H_{fg} u} \propto \left(\frac{\sigma}{\rho_\ell u^2 D}\right)^{1/2} \tag{7}$$

so that q_{co} is independent of u for these singular data too.

Boundaries of Characteristic Regimes

Boundary between *D*-regime and *V*-regime. As has been observed in the experiment by [4] and discussed briefly in the appendix

of [2], there is a simple kind of CHF which takes place only due to the deficiency of liquid supply as compared with the heat load. If it can be assumed for simplicity that all the liquid supplied is vaporized because the fraction of liquid splashing away from the heated surface is so small, and that the jet velocity is high enough to neglect the effects of surface tension and gravity, the critical heat flux q_{co} in this case is related to the liquid supply via the heat balance as follows:

$$\frac{q_{co}}{\rho_v H_{fg} u} = \frac{\rho_\ell}{\rho_v}\left(\frac{d}{D}\right)^2 \tag{8}$$

where d is the diameter of nozzle supplying liquid. The regime responsible to this type of CHF will be called D-regime in this paper, because it is concerned with CHF due to the dryout of heated surface. In many cases, D-regime is situated in the range of low jet velocities.

The boundary between D and V-regime, where q_{co} is presumably continuous, may be determined by eliminating q_{co} from equations (4) and (8) as,

$$\frac{\sigma}{\rho_\ell u^2 D} = 150\left(\frac{\rho_\ell}{\rho_v}\right)^{1.16}\left(\frac{d}{D}\right)^6 \tag{9}$$

In the experiments of the present study, $d/D = 1/5$, and the minimum density ratio is $\rho_\ell/\rho_v = 5.20$ for R-12 at $p = 27.9$ bars, so that equation (9) gives the following possible minimum value of $\sigma/\rho_\ell u^2 D$ for D-regime:

$$\frac{\sigma}{\rho_\ell u^2 D} = 6.50 \times 10^{-2}$$

Since this value is much higher than those value of $\sigma/\rho_\ell u^2 D$ experienced in the present study as shown in abscissa of Fig. 5, it is concluded that the present study has no relation to D-regime.

Boundary between V-regime and I-regime. As in the preceding section, the boundary between V and I-regime is determined by eliminating q_{co} from equations (4) and (6) to give

$$\frac{\sigma}{\rho_\ell u^2 D} = 1.64 \times 10^{-5} \tag{10}$$

Of course, equation (10) corresponds exactly to the value of $\sigma/\rho_\ell u^2 D$ at which two solid lines intersect in Fig. 5.

Possibility of the Fourth Regime. As mentioned before, the R-12 data at $p = 27.9$ bars deviate from the prediction of equation (6), and this fact suggests the possibility of a fourth regime existing at very high pressures. Such a characteristic regime at high pressures is known to exist in forced convection boiling in round tubes and will be discussed in the next chapter. At present, it is recommended that a restriction be placed on the pressure range where equation (6) is applicable as follows:

$$\rho_\ell/\rho_v > 6.0 \tag{11}$$

In case of water, for instance, inequality (11) of the density ratio corresponds to the pressure range: $p < 152$ bars, or $p/p_c < 0.69$ where p_c is the critical pressure.

Upper Limit of CHF

Equation (6) has been so far expressed in the form including the jet velocity u, but u has in fact no effects on q_{co} in I-regime. Therefore, it is preferable that the q_{co}, namely the upper limit of q_{co} be denoted by $q_{co,max}$, and equation (6) be rewritten as:

$$\frac{q_{co,max}}{\rho_v H_{fg}\sqrt{\sigma/(\rho_\ell D)}} = 1.18\left(\frac{\rho_\ell}{\rho_v}\right)^{0.614} \tag{12}$$

Also, the lower limit of the jet velocity u capable of generating $q_{co,max}$ is denoted by u_{min}, when equation (10) gives the following indication for u_{min}:

$$\frac{\sigma}{\rho_\ell u_{min}^2 D} = 1.64 \times 10^{-5} \tag{13}$$

Pressure variation of $q_{co,max}$ and u_{min}, which is predicted by equation (12) and (13), is shown in Fig. 6 for the case of water boiling on heated disks of diameter $D = 0.01$, 0.02 and 0.03 m. Fig. 6 also shows critical

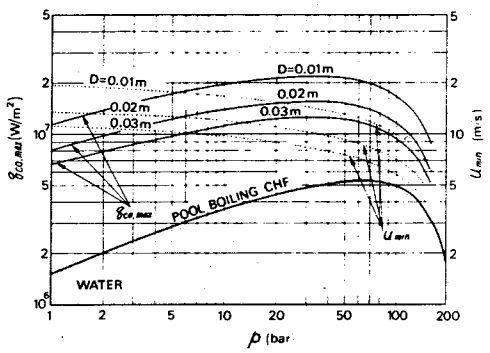


Fig. 6 Upper limit of q_{co} and corresponding lower limit of u for water boiling on heated disks

heat flux q_{co} in saturated pool boiling at the standard gravitational acceleration, which is predicted by Zuber correlation modified by Rohsenow [5]:

$$\frac{q_{co}}{\rho_v H_{fg}} = 0.18\left[\frac{\sigma g(\rho_\ell - \rho_v)}{\rho_v^2}\right]^{1/4}\left[\frac{1}{1+(\rho_v/\rho_\ell)}\right]^{1/2} \tag{14}$$

where g is the gravitational acceleration. It may be of interest to note in Fig. 6 that the prediction of equation (12) for $q_{co,max}$ versus p and that of equation (14) for q_{co} versus p are somewhat similar in character.

Discussion. With respect to the result mentioned above, it may be useful to note the following remarks: Examining CHF in forced convection boiling in vertical uniformly heated round tubes for instance, it is found that there are many data showing CHF saturation trend accompanying the increase of mass velocity, particularly in case of $\Delta H_i = 0$ (cf. Fig. 2(a) of Thompson, et al. [6], Figs. 17–20 of Stevens, et al. [7], Fig. 6–11 of Becker, et al. [8], and Collier [9], Hewitt [10], and Lee [11]). For such a state of CHF, and presumably for CHF in I-regime in the present study too, it may be presumed that, owing to high mass velocity of liquid, the vapor generated on the heated surface is captured by the liquid flow, and CHF takes place due to near wall bulk crowding and vapor blanketing.

In addition, it is well known that when the system pressure is highly elevated in the forced convection boiling in round tubes, characteristics of CHF change to quite different ones from those at low pressures (cf. Becker, et al. [12], Campolunghi, et al. [13], and Katto [14]).

Of course, simple comparison should not be made on CHF between the forced convection boiling on heated disks with only one dimension of radial length and the much more complicated convection boiling in heated round tubes with two dimensions of axial length and inner diameter. However, it may be of use as well as of interest to notice that there are many common characters between the two boiling systems in elemental aspects of CHF.

Conclusions

1 CHF in the forced convection boiling on an open heated disk surface supplied with saturated liquids at various pressures through a small impinging jet has been studied, yielding equations (4) and (6) for the generalized correlation of CHF in V-regime and I-regime, and equations (9), (10) and (11) to predict the boundaries of D, V and I-regime. At present, equations (4) and (6) should to be used with attention to a tolerance of accuracy as indicated by the experimental results, including equations (2) and (5) in Figs. 4 and 5.

2 For the prediction of upper limit of CHF as well as the lower limit of the jet velocity capable of generating the upper limit of CHF, equations (12) and (13) are presented.

3 The possibility is suggested that a fourth regime other than the D, V and I-regimes may exist at elevated pressures.

Acknowledgment

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FORCED CONVECTION BOILING

INTERNAL FLOW

A GENERALIZED CORRELATION OF CRITICAL HEAT FLUX FOR THE FORCED CONVECTION BOILING IN VERTICAL UNIFORMLY HEATED ROUND TUBES

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Abstract—The present study deals with the critical heat flux (CHF) in the conditions that the fluid fed to heated tubes is subcooled (including saturated liquid in the extreme situation) with no entrained vapor. To start with, postulating that there is a state where the hydrodynamic condition is responsible for CHF, a theoretical presumption for the generalized correlation equation of CHF is attempted with the aid of vectorial dimensional analysis. Then, the experimental data of CHF obtained from the literature for fluids of seven different kinds are analyzed, revealing the existence of four characteristic regimes of CHF, called L-, H-, N-, and HP-regime in this paper, and leading to the development of the generalized correlation of CHF data as well as that of CHF-regime correlation.

NOMENCLATURE

- d, I.D. of heated tube [m];
- G, mass velocity [kg/m²s];
- H_{fg}, latent heat of evaporation [J/kg];
- ΔH_i, enthalpy of inlet subcooling [J/kg];
- K, parameter for the effect of inlet subcooling, equation (20);
- l, length of heated tube [m];
- p, absolute pressure, Figs. 5–14 and 16–18 [bar];
- q_c, critical heat flux [W/m²];
- q_{c0}, q_c, for $\Delta H_i = 0$ [W/m²];
- u, inlet velocity [m/s].

Greek symbols

- μ, viscosity [Ns/m²];
- ρ_l, density of liquid [kg/m³];
- ρ_v, density of vapor [kg/m³];
- σ, surface tension [N/m];
- χ_{ex}, exit quality.

1. INTRODUCTION

MANY studies have been made on the critical heat flux (CHF) or burnout for the forced convection boiling in vertical uniformly heated round tubes [1–4]. In the present paper, the discussion will be restricted to conditions where the fluid fed to the heated tube is subcooled with no entrained vapor (including the state of saturated liquid in the limit), and the flow is stable with no oscillations. Up to now, for such conditions as mentioned above, a lot of experimental data of CHF of water have been collected, and many empirical correlation for water have been developed by Tong *et al.* [5], Thompson and Macbeth [6], Bertoletti *et al.* [7], Tong [8], Hewitt *et al.* [9], Becker *et al.* [10], Bowring [11], and others. On the other hand, to make it possible to investigate CHF in nuclear reactors by employing model fluids such as Freons, the study of CHF modelling has been made by Barnett [12], Ahmad [13], etc., based on dimensionless analysis of the problem, and by Stevens and Kirby [14], Dix [15], etc., in pursuit of empirical scaling factors. Theoretical analyses of CHF have also been made by Whalley *et al.* [16], Thorgerson *et al.* [17], etc., evaluating the basic mechanisms of CHF. In addition, it should be noted that about twenty years ago, dimensional analyses of fundamental equations of two phase flow were made for the study of CHF independently by Zenkevich [18] and Griffith [19]. However, in spite of the effort mentioned above, the generalized correlation capable of predicting CHF for any arbitrary fluid, mass velocity, tube diameter and length has not yet been developed, and the present paper will report the result obtained in an attempt to derive a simple but dimensionless correlation of CHF.

2. A CLUE FOR ANALYZING DATA OF CHF

Considering only simple boiling systems of saturated liquid for the present, and postulating that the hydrodynamic condition of two-phase vapor–liquid flow is responsible for the occurrence of CHF, it may be assumed that the superficial vapor velocity $q_{c0}/\rho_v H_{fg}$ at the critical heat flux of q_{c0} is governed by the velocity of liquid u, the length of heated surface l, the densities ρ_l and ρ_v relating to the inertia, the surface tension of interface σ, the viscosities μ_l and μ_v relating to the viscous force, and the gravitational force $g(\rho_l-\rho_v)$ relating to the buoyancy. In this case, applying the dimensional analysis, it yields

$$\frac{q_{c0}}{\rho_v H_{fg} u} = f\left(\frac{\rho_l}{\rho_v}, \frac{\mu_l}{\mu_v}, \frac{\sigma}{\rho_l u^2 l}, \frac{\mu_l}{\rho_l u l}, \frac{g(\rho_l-\rho_v)l}{\rho_l u^2}\right). \quad (1)$$

If it is further assumed that the effect of viscous force

on CHF is negligibly small, equation (1) is simplified to

$$\frac{q_{c0}}{\rho_v H_{fg} u} = f\left(\frac{\rho_l}{\rho_v}, \frac{\sigma}{\rho_l u^2 l}, \frac{g(\rho_l - \rho_v)l}{\rho_l u^2}\right). \quad (2)$$

In the case of pool boiling, u can be eliminated from dimensionless groups in equation (2) to give

$$\frac{q_{c0}}{\rho_v H_{fg}} \Big/ \left[\frac{\sigma g(\rho_l - \rho_v)}{\rho_v}\right]^{1/4} = f\left[\frac{\rho_l}{\rho_v}, \frac{g(\rho_l - \rho_v)l^2}{\sigma}\right]. \quad (3)$$

As is well known, except the case of very small l (cf. [20]), the dimensionless group $g(\rho_l - \rho_v)l^2/\sigma$ in equation (3) has no effects on q_{c0}, when equation (3) agrees with the form of CHF correlation equation obtained by Zuber [21] or Kutateladze [22].

On the other hand, in the case of forced convection boiling, g can be eliminated from equation (2) to give immediately

$$\frac{q_{c0}}{\rho_v H_{fg} u} = f\left(\frac{\rho_l}{\rho_v}, \frac{\sigma}{\rho_l u^2 l}\right). \quad (4)$$

The form of equation (4) is in accord with those correlations obtained experimentally by Lienhard and Eichhorn [23] on heated cylinders in a cross flow, by Monde and Katto [24] on heated disks supplied with a liquid though a small impinging jet, and by Katto and Ishii [25] on rectangular heated surfaces supplied with a saturated liquid through a plane jet. For use afterward in the present paper, the above mentioned correlation of Katto and Ishii [25] is noted here as:

$$\frac{q_{c0}}{\rho_v H_{fg} u} = 0.0164\left(\frac{\rho_l}{\rho_v}\right)^{0.867}\left(\frac{\sigma}{\rho_l u^2 l}\right)^{1/3} \quad (5)$$

where u is the inlet velocity of liquid to the heated surface and l is the length of heated surface measured in the direction of forced flow.

Now, a more detailed analysis will be applied to equations (3) and (4) through the vectorial dimensional analysis. For the infinitesimal wave motion of an interface between the vapor and liquid flowing perpendicularly to gravity, such as illustrated in Fig. 1, the surface tension σ and the gravitational acceleration g exert their influence upon the fluid motion through changing the local pressure of fluid. Then, if Cartesian coordinates x, y, and z are taken as shown in Fig. 1, it is readily found from the well-known hydrodynamic theory of the wave motion [26] that σ and g should have

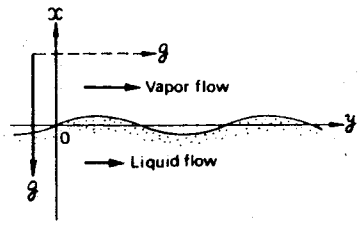

FIG. 1. Infinitesimal wave motion of a liquid–vapor interface.

the following dimensions:

$$[\sigma] = \frac{ML_y^2}{\theta^2 L_x L_z}, \quad [g] = \frac{L_y}{\theta^2} \quad (6)$$

where mass M, time θ, and length L (whose subscripts represent x, y, and z-axis respectively) are fundamental dimensions. In the meanwhile, if the direction of gravity is parallel to the liquid–vapor interface, as shown by a broken line of g in Fig. 1, g is independent of the wave motion so that it has the ordinary dimension $[g] = L_y/\theta^2$, which is the same as that given in equation (6). In other words, equation (6) can apply in either case of the direction of g mentioned above.

Figure 2(a) and (b) show the pool boiling on a horizontal and a vertical heated surface respectively. In both cases, however, equation (6) can equally apply to the liquid–vapor interface that is made very close to the heated surface by the vapor effusing perpendicularly from the heated surface, and besides, it is obvious that $[q_{c0}] = H/\theta L_x L_z$, $[H_{fg}] = H/M$, $[\rho] = M/L_x L_y L_z$ with the fundamental dimension of heat H, so that we obtain

$$\left[\frac{q_{c0}}{\rho_v H_{fg}}\right] = \frac{L_y}{\theta}, \left[\frac{\sigma g(\rho_l - \rho_v)}{\rho_v^2}\right] = \left(\frac{L_y}{\theta}\right)^4, \left[\frac{\rho_l}{\rho_v}\right] = 1. \quad (7)$$

The result of equation (7) means that equation (3) is truly dimensionless for both boiling systems of Fig. 2(a) and (b) provided the horizontal surfaces of extremely small size as well as the very long vertical surfaces are excluded.

Next, Fig. 2(c) shows the forced convection boiling where a liquid is fed to a heated surface of length l with the inlet velocity of u parallel to the heated surface. Taking into consideration that for the liquid–vapor interface that is made very close to the heated surface by the vapor effusing perpendicularly

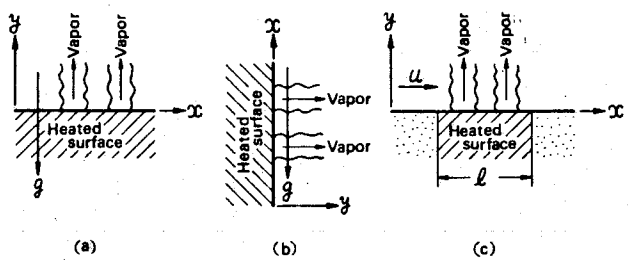

FIG. 2. Three kinds of fundamental boiling systems: (a) pool boiling on a horizontal surface, (b) pool boiling on a vertical surface, and (c) forced convection boiling on a flat surface.

from the heated surface, σ has the dimension given in equation (6), and $[q_{c0}] = H/\theta L_x L_z$, $[H_{fg}] = H/M$, $[\rho] = M/L_x L_y L_z$, $[u] = L_x/\theta$, and $[l] = L_x$, it yields readily

$$\left[\frac{q_{c0}}{\rho_v H_{fg} u}\right] = \frac{L_y}{L_x}\left[\frac{\rho_l}{\rho_v}\right] = 1, \quad \left[\frac{\sigma}{\rho_l u^2 l}\right] = \left(\frac{L_y}{L_x}\right)^3. \quad (8)$$

The result of equation (8) means that in order for equation (4) to be truly dimensionless, it should take the form of $q_{c0}/\rho_v H_{fg} u = f(\rho_l/\rho_v)(\sigma/\rho_l u^2 l)^{1/3}$, which is nothing but the form of those correlations experimentally obtained in [24] and [25], such as that given in equation (5). As for the correlation obtained by Lienhard and Eichhorn [23], which has a slightly different form from above, it should be noted that the vectorial conditions of this boiling system are not so simple as that of Fig. 2(c).

The above results, given for Fig. 2(a), (b) and (c), may be regarded to show that the vectorial dimensional analysis is not useless for the analysis of CHF in the case of simple boiling systems at least.

Now, let us consider a uniformly heated round tube of internal diameter d and heated length l, to which a saturated liquid with no entrained vapor is

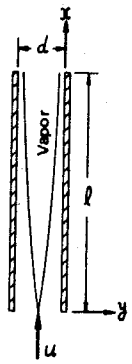

FIG. 3. A uniformly heated tube in the limit as $d \to 0$.

fed with the inlet velocity u. In this case, if the limit as $d \to \infty$ is assumed, the hydrodynamic state is the same as that of Fig. 2(c), so that there is a possibility of obtaining a correlation of CHF similar to equation (5). On the other hand, for the limit as $d \to 0$, it may be presumed that due to the accumulation of vapor in the tube, CHF occurs in such the state as illustrated in Fig. 3, and the liquid–vapor interface parallel to the heated surface is responsible for the occurrence of CHF; for which the vectorical dimensional analysis yields

$$\left[\frac{q_{c0}}{\rho_v H_{fg} u}\right] = \frac{L_z}{L_x}, \quad \text{and} \quad \left[\frac{\sigma}{\rho_l u^2 l}\right] = 1. \quad (9)$$

In this case, therefore, the truly dimensionless correlation is assumed to take the following form:

$$\frac{q_{c0}}{\rho_v H_{fg} u}\cdot\frac{l}{d} = \text{const.}\left(\frac{\rho_l}{\rho_v}\right)^a\left(\frac{\sigma}{\rho_l u^2 l}\right)^b. \quad (10)$$

It is of interest to note that the LHS term of equation (10) multiplied by ρ_v/ρ_l is a quantity proportional to the quality χ_{ex} at the tube exit. Then, trying to invent those correlations which take the form of equation (5) as $d \to \infty$ and that of equation (10) as $d \to 0$, the following equation is found to be one of the equations satisfying the above conditions:

$$\frac{q_{c0}}{GH_{fg}} = \text{const.}\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3} \times \frac{1}{1+\text{const.}\left(\frac{\rho_l}{\rho_v}\right)^m\left(\frac{\sigma\rho_l}{G^2 l}\right)^n\frac{l}{d}} \quad (11)$$

where $G = \rho_l u$, that indicates the mass velocity through the tube because of the premise that saturated liquid is fed to the tube with the velocity of u.

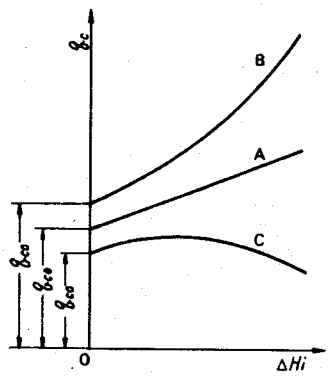

FIG. 4. Relationship between q_c and ΔH_i (type A found in L-, H-, and HP-regime; type B in N-regime, and type C in VL-regime).

3. ANALYSIS OF q_{c0}-DATA (H-REGIME)

In order to test the validity of equation (11), an analysis is attempted by using the esperimental data of q_{c0} which can be obtained in the following way. When the inlet subcooling ΔH_i changes in an experiment keeping the other conditions constant, the relationship between the critical heat flux q_c and the inlet subcooling enthalpy ΔH_i is linear in many cases as illustrated by type A in Fig. 4 but occasionally has the curvature such as illustrated by type B or C in Fig. 4 depending on the conditions. In any case, however, if there are enough data of $q_c - \Delta H_i$ to permit the extrapolation as $\Delta H_i \to 0$, q_{c0} can be determined with accuracy. In the present analysis, q_{c0} is thus determined from the data compiled or presented by Thomson and Macbeth (Tables 1–10) [6], Becker *et al.* (data of inlet subcooling 10°C in Fig. 12) [10], Waters *et al.* [27], Lowdermilk *et al.* (Table IV) [28], Chojnowski and Wilson (Fig. 3) [29], Lee and Obertelli [30], Matzner [31], Weatherhead [32], Barnett [33] and Hewitt *et al.* (Fig. 2.8) [34] for water; Stevens *et al.* [35, 36] for Freon-12; Dix (CHF $\times 10^{-4}$ should be replaced by CHF $\times 10^{-5}$ in Tables of Appendix) [15]

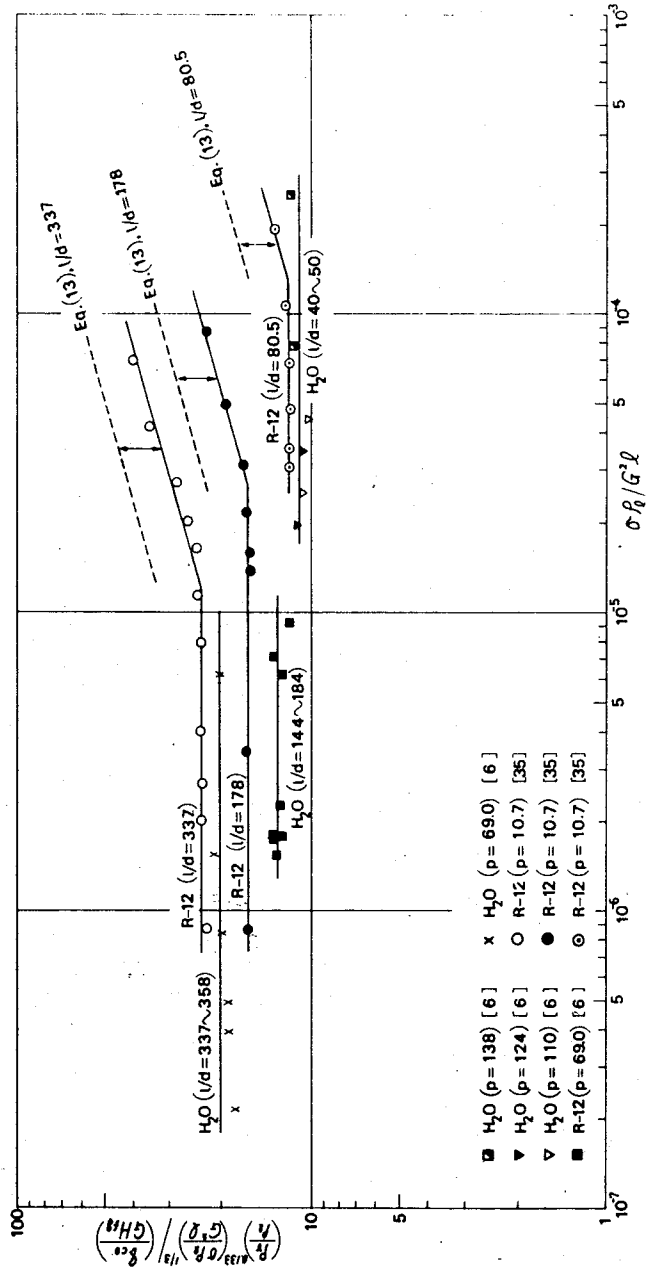

FIG. 5. Characteristics of q_{c0} in H-regime.

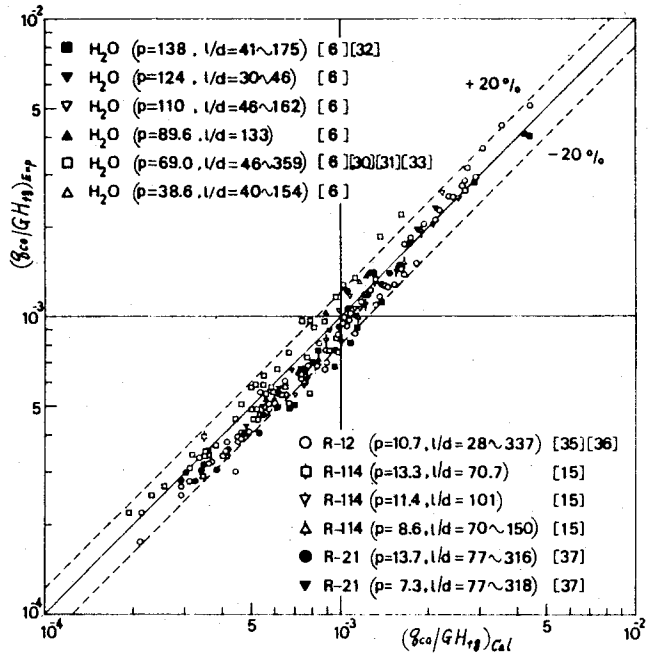

FIG. 6. Comparison of the prediction of equation (12) with the experimental data in H-regime.

for Freon-114; Barnett and Wood [37] for Freon-21; Lewis *et al.* [38] for liquid hydrogen and nitrogen; and Ogata and Sato [39] for liquid helium.

Now, Fig. 5 shows some typical results obtained in the region of $\sigma\rho_l/G^2l = 10^{-7}$ to 10^{-3}, indicating that there is a regime where the value of ordinate is determined as a function of l/d alone independently of the change of $\sigma\rho_l/G^2l$; and the data of q_{c0} belonging to this regime can be correlated fairly well by the following dimensionless equation:

$$\frac{q_{c0}}{GH_{fg}} = 0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\frac{1}{1+0.0031\,l/d} \tag{12}$$

In Fig. 6, all the experimental data of q_{c0}/GH_{fg} in this regime got from the data source cited before are compared with the prediction of equation (12).

Probably, it may be presumed that CHF in this regime is affected by the hydrodynamic instability conditions akin to those assumed in deriving equation (11). Accordingly, this regime will be called H-regime in this paper. As for the effect of inlet subcooling on CHF, the details of which will be described in Chapter 6, such a linear relationship as shown by the type A in Fig. 4 is found to be always established between q_c and ΔH_i in H-regime.

4. ANALYSIS OF q_{c0}-DATA (OTHER REGIMES)

4.1. *L-regime*

Figure 7 as well as a part of Fig. 5 shows typical results obtained in the region of $\sigma\rho_l/G^2l$ relatively higher than that of H-regime; and the data of q_{c0} belonging to this regime can be correlated fairly well by the following dimensionless equation:

$$\frac{q_{c0}}{GH_{fg}} = 0.25\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d} \tag{13}$$

except Freons for which, as shown by broken lines in Fig. 5, equation (13) gives the prediction of q_{c0} higher than the experimental data as much as about 1.36 times, so that the constant of 0.25 in equation (13) must be replaced by 0.34. The reason for this is unknown, but it may depend on the thermal decomposition of Freons on the dried heated surface. In Fig. 8, all the experimental data of q_{c0}/GH_{fg} in this regime which are obtained from the data sources cited in Chapter 3 are compared with the prediction of equation (13) (with the constant as 0.34 for Freons).

It can be presumed that the dryout of an annular liquid-film flowing over the tube wall is mainly responsible for CHF in this regime; because the main part of equation (13) excluding the term of very weak effect of $\sigma\rho_l/G^2l$, that is $q_{c0}/GH_{fg} = 0.25/(l/d)$, is nothing but the condition corresponding to the complete exhaustion of liquid at the exit of the tube with the uniform heat flux of q_{c0}.

Since this regime has the mass velocity G less than that in H-regime, it is called L-regime in this paper. As for the effects of inlet subcooling, a linear relationship is found to be established between q_c and ΔH_i in L-regime (that is, the type A in Fig. 4).

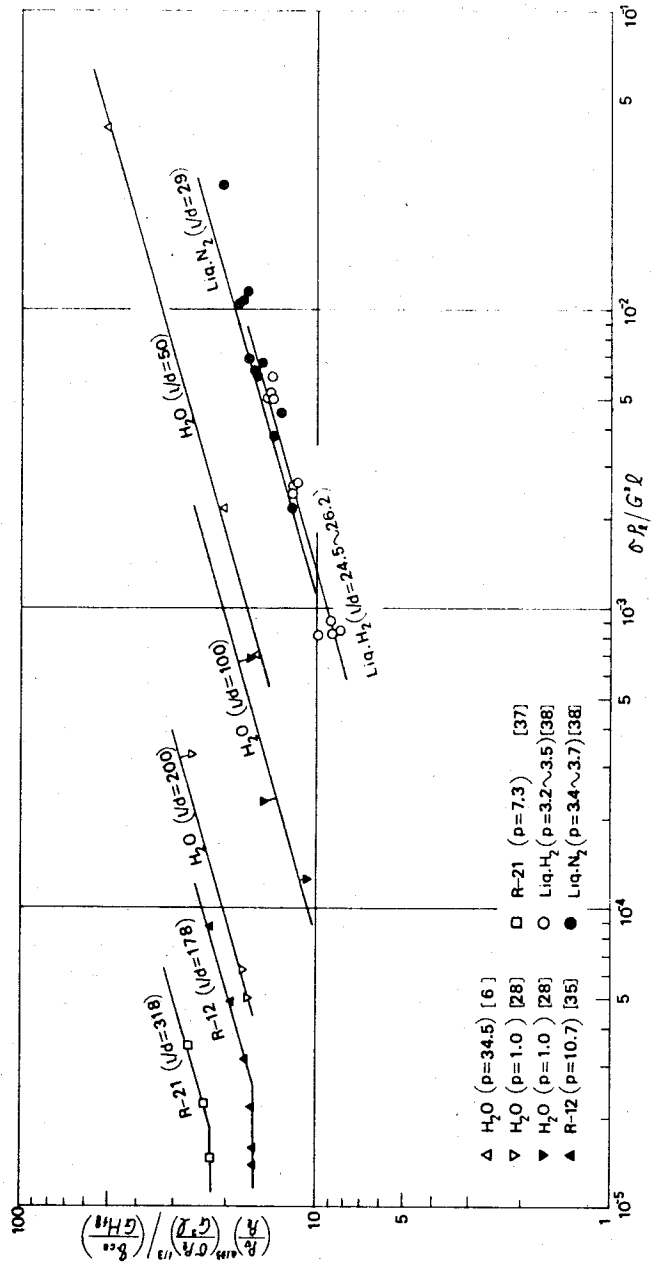

FIG. 7. Characteristics of q_{c0} in L-regime.

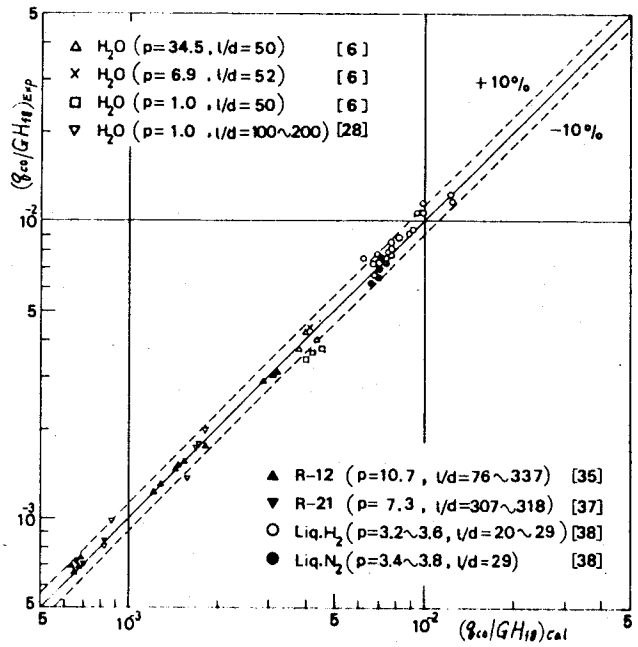

FIG. 8. Comparison of the prediction of equation (13) with the experimental data in L-regime.

Supplementary note: VL-regime. When the mass velocity G is further reduced to low levels so that the inlet velocity of liquid is less than, say, 0.2–0.3 m/s, and when the length of tube is considerably short, there is a possibility of bringing about the deviation from L-regime toward the regime where ordinary pool boiling exerts its influence on CHF. Figure 9 shows two examples of the deviation from L-regime found in the existing experimental data. Because of the very low mass velocity, this regime is called VL-regime in this paper. It should be stressed that when the liquid fed to the tube is subcooled, the non-linear relationship of type C in Fig. 4 appears in VL-regime.

4.2. *N-regime*

As shown in Fig. 10, when $\sigma\rho_l/G^2l$ is reduced from the state of H-regime keeping l/d constant, the value of ordinate tends to increase entering a new region indicated by black symbols of data, where the non-linear relationship of type B in Fig. 4 is found to appear. The data of q_{c0} in this regime can be correlated by the following dimensionless equation:

$$\frac{q_{c0}}{GH_{fg}} = 0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}\frac{(l/d)^{0.27}}{1+0.0031\,l/d} \tag{14}$$

Figure 11 shows the comparison of equation (14) with all the experimental data of q_{c0} in this regime, the

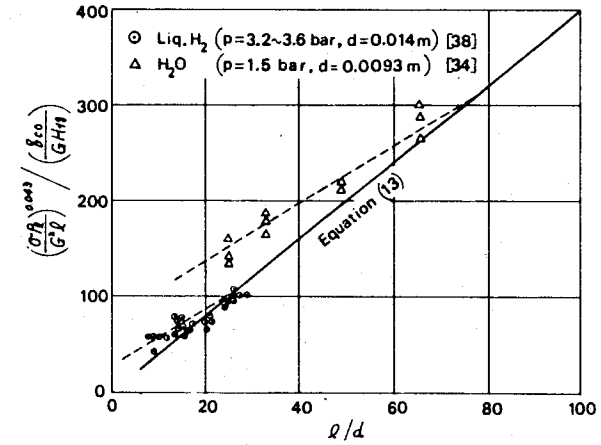

FIG. 9. Deviation from equation (13) in VL-regime.

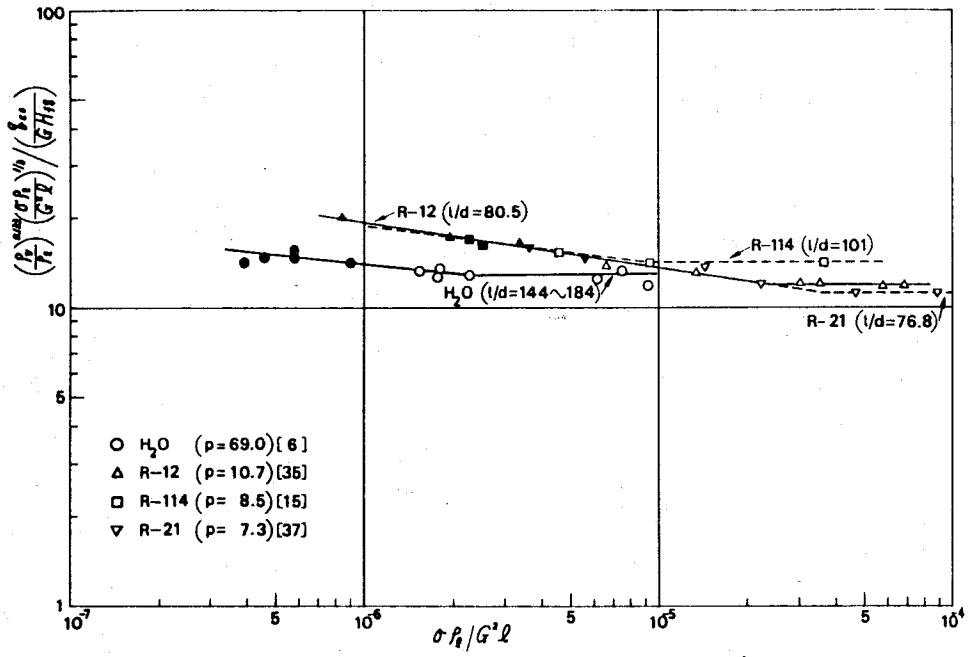

FIG. 10. Characteristics of $q_{i,0}$ in N-regime.

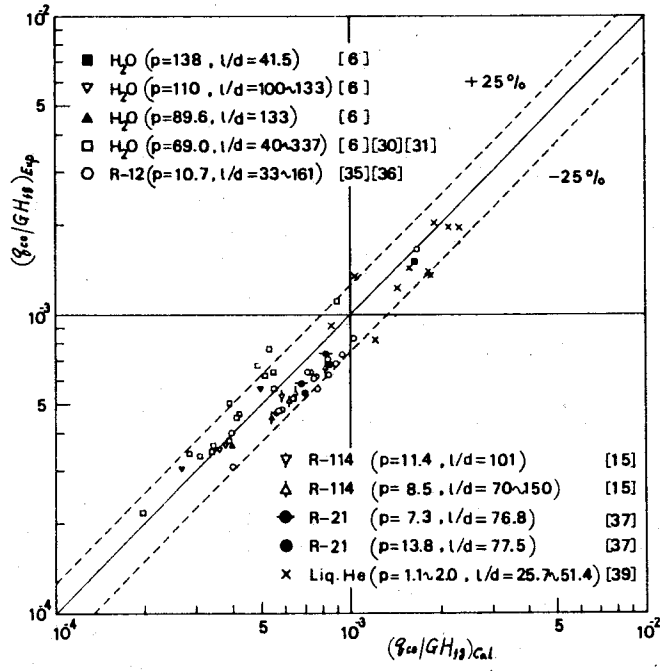

FIG. 11. Comparison of the prediction of equation (14) with the experimental data in N-regime.

error being a little greater than those in other regimes.

This regime is characterized by the non-linear relationship of type B in Fig. 4 between q_c and ΔH_i, so that it is called N-regime in this paper. Generally CHF takes place in this regime with low exit quality in the neighborhood of zero, and accordingly it may be presumed that a bubble layer flows parallel to the tube wall with a liquid core flowing at the center of the tube when CHF appears due to the transition to the film boiling.

Such transitional data as those shown by blank, that is non-black, symbols lying in slant lines in Fig. 10 can be correlated by equation (14), but they are

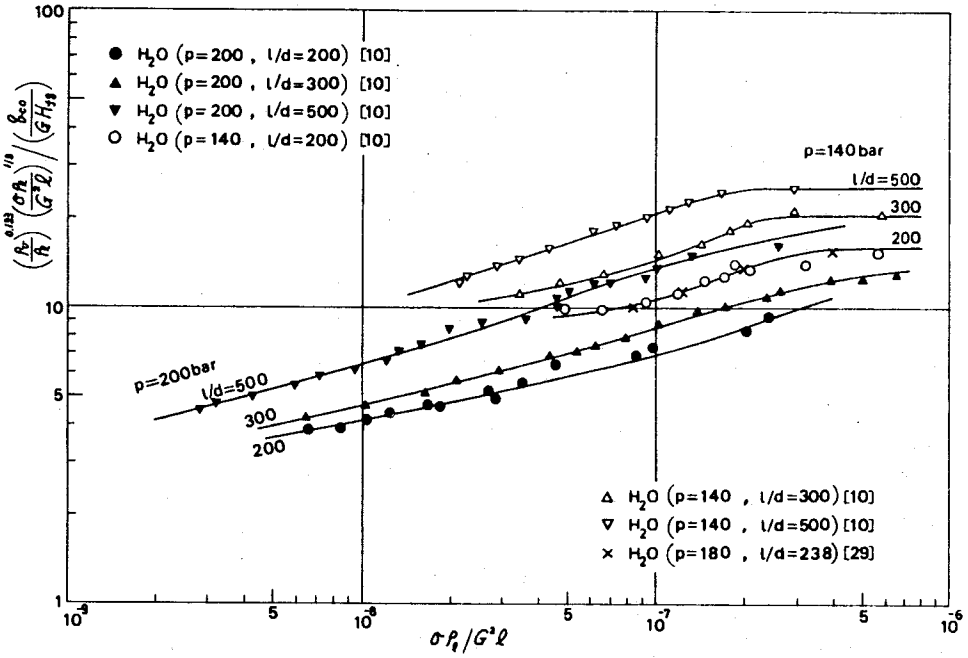

FIG. 12. Characteristics of q_{c0} in HP-regime.

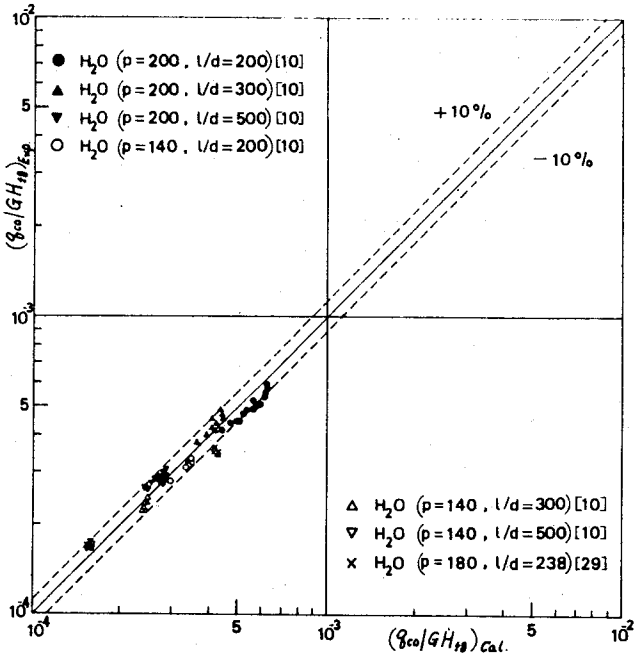

FIG. 13. Comparison of the prediction of equation (15) with the experimental data in HP-regime.

regarded as belonging to H-regime in the point that the linear relationship is established between q_c and ΔH_i. Accordingly, in the present paper, they have been correlated by equation (12) of H-regime in order to avoid the complexity.

4.3. *HP-regime*

In the region of $\sigma\rho_l/G^2l$ less than that of H-regime, it is found that there is another regime where the linear relationship is established between q_c and ΔH_i, and at the same time, such characteristics as shown

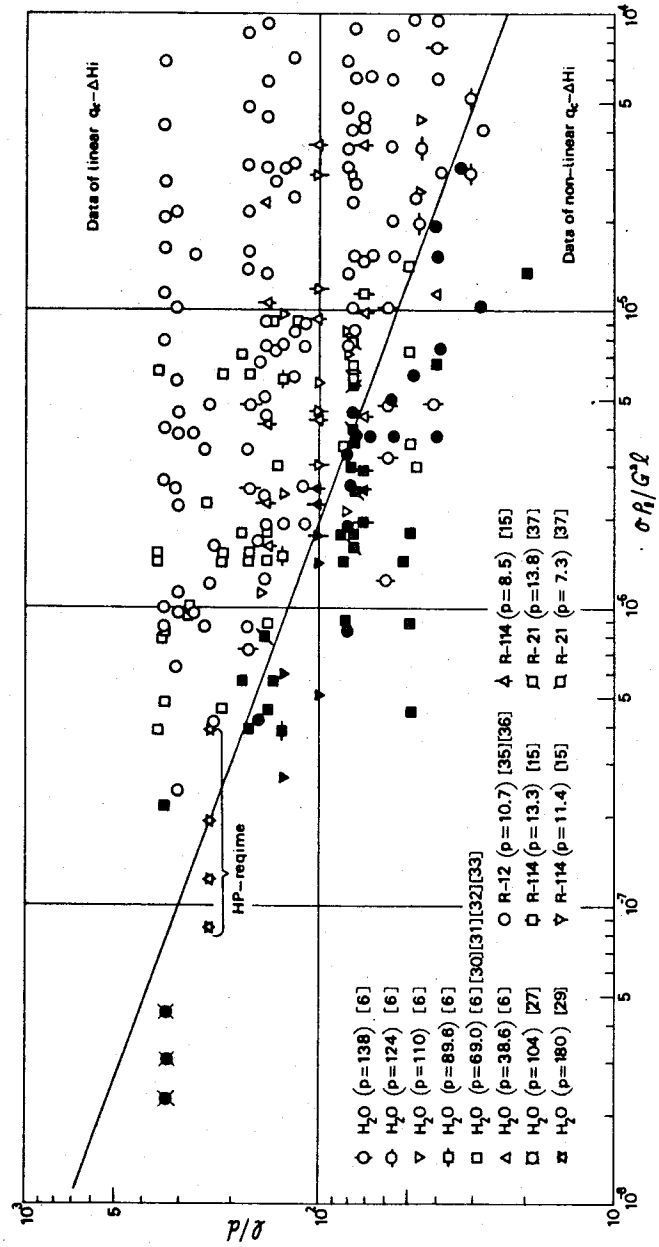

FIG. 14. Boundary between H-regime and N-regime.

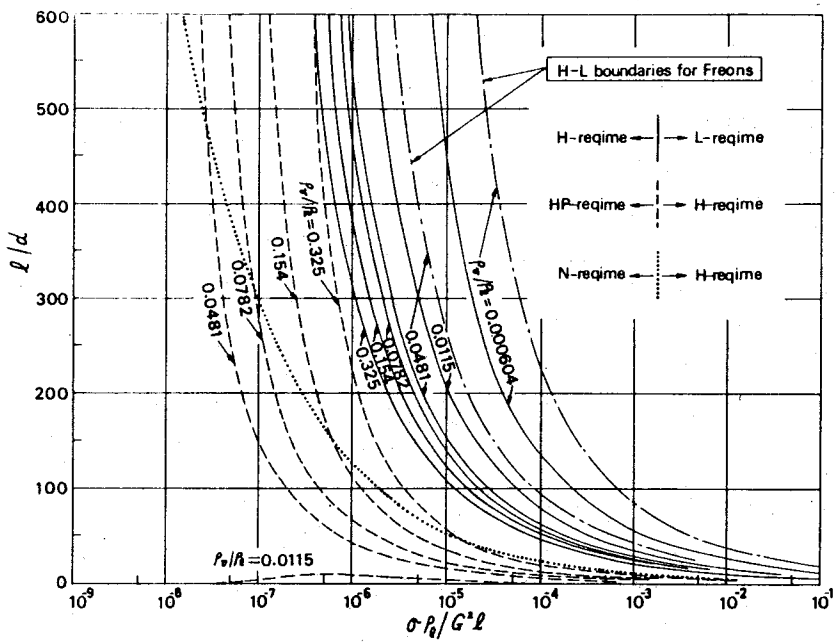

FIG. 15. CHF-regime map.

in Fig. 12 appear for CHF. The data of water only have been obtained in this regime, but they can be fairly well correlated by the following dimensionless equation:

$$\frac{q_{c0}}{GH_{fg}} = 8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.453}\frac{1}{1+107\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.54}\frac{l}{d}}. \tag{15}$$

Figure 13 shows the comparison of equation (15) with all the experimental data of q_{c0} in this regime. As it has been pointed out by Becker *et al.* [10], and as it will be clarified afterward in Section 5 of the present paper, this regime makes its appearance when the pressure is considerably high, so that this regime is called HP-regime in the present paper. It may be of interest to note here that the experimental data compiled by Thompson and Macbeth [6] includes those data belonging to HP-regime, which will be described afterward in Section 6.3.

5. BOUNDARIES OF EACH REGIME

In the existing circumstances of study, it is difficult to extend the discussion to the transition region which ought to lie between the principal regimes mentioned in Sections 3 and 4. Therefore, in the present paper, the boundary of each regime will be determined approximately by neglecting the transition region as follows.

First, by eliminating q_{c0} from equations (12) and (13), the boundary between *L*- and *H*-regime is determined as:

$$\frac{l}{d} = \frac{1}{0.4(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{0.29} - 0.0031} \tag{16}$$

where the constant 0.4 on the RHS is to be replaced by 0.29 for Freons.

Next, for the boundary between H- and N-regime, if q_{c0} is eliminated from equations (12) and (14), it yields

$$\frac{l}{d} = \frac{1.077}{(\sigma\rho_l/G^2l)^{0.37}}, \tag{17}$$

but this does not correspond to the boundary itself as it can readily be seen in Fig. 10. Accordingly, the experimental data are classified into two groups depending on whether the linear relationship of $q_c - \Delta H_i$ holds or not, yielding the result of Fig. 14, where blank, that is non-black, symbols correspond to the experiments having the linear relationship of the type A in Fig. 4 while black symbols the non-linear relationship of the type B in Fig. 4. Then, from the slant boundary line obtained in Fig. 14, the H–N boundary is given as:

$$\frac{l}{d} = \frac{0.77}{(\sigma\rho_l/G^2l)^{0.37}}. \tag{18}$$

Finally, by eliminating q_{c0} from equations (12) and (15), the boundary between H- and HP-regime is determined as:

$$\frac{l}{d} = \frac{82(\rho_v/\rho_l)^{0.517} - (\sigma\rho_l/G^2l)^{-0.12}}{107(\sigma\rho_l/G^2l)^{0.42} - 0.254(\rho_v/\rho_l)^{0.517}}. \tag{19}$$

Boundaries of each regime based on equations (16), (18), and (19) are shown with ρ_v/ρ_l as the parameter in Fig. 15, where H–L boundary lines for Freons are slightly shifted toward right due to the change of a constant in equation (16) as shown by two examples for $\rho_v/\rho_l = 0.000604$ and 0.0481. According to Fig. 15, it is noted that as ρ_v/ρ_l or the

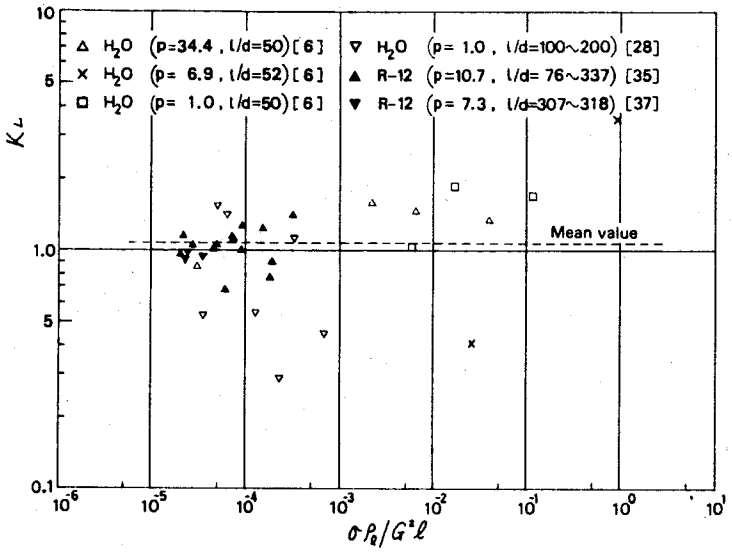

FIG. 16. Correlation of the experimental data of K_L.

pressure increases, L-regime extends to the region of lower $\sigma\rho_l/G^2l$, while H-regime extends to the region of higher $\sigma\rho_l/G^2l$. In consequence of this change, H-regime narrows and it is seen that when $\rho_v/\rho_l = 0.325$ (N.B. the corresponding pressure, 196 bar in the case of water for instance, is very near the critical pressure, 221 bar), H-regime is very narrow. On the other hand, the boundary between H- and N-regime is not affected by ρ_v/ρ_l. In addition, it should be noted that there is no transition between N- and HP-regime; that is, the transitions of H⇄N and H⇄HP alone can take place.

6. EFFECT OF INLET SUBCOOLING ON CHF

When the liquid is subcooled at the entrance of the tubes, and the linear relationship of $q_c - \Delta H_i$ (type A in Fig. 4) holds, one can write

$$q_c = q_{c0}(1 + K\Delta H_i/H_{fg}) \tag{20}$$

where K, that is a non-dimensional parameter independent of $\Delta H_i/H_{fg}$, can be determined from the gradient of the straight line A in Fig. 4.

6.1. K *in L-regime:* K_L

Figure 16 shows K_L *obtained from the data of* water and Freons belonging to L-regime (N.B. K_L cannot be obtained from the existing data of liquid nitrogen and hydrogen [38] because of the experimental range of $\Delta H_i \fallingdotseq 0$). From the mechanism of CHF mentioned in Section 4.1, it may be presumed that $K_L \fallingdotseq 1$ for L-regime; and in fact, the mean value of all the data shown in Fig. 16 is found to be

$$K_L = 1.16 \tag{21}$$

though scattering of the value of K_L is not small as seen in Fig. 16.

6.2. K *in* H-*regime:* K_H

Figure 17 shows the correlation of K_H obtained from all the data of water and Freons belonging to H-regime, and it is formulated as follows:

$$\left.\begin{aligned} &\text{for } \sigma\rho_l/G^2l < 3\times10^{-6}, \\ &\quad K_H = 1.8\left(\frac{l/d}{130}\right)^{-5\rho_v/\rho_l} \\ &\text{for } 3\times10^{-6} < \sigma\rho_l/G^2l, \\ &\quad K_H = 0.075\left(\frac{l/d}{130}\right)^{-5\rho_v/\rho_l}\left(\frac{\sigma\rho_l}{G^2l}\right)^{-1/4}. \end{aligned}\right\} \tag{22}$$

As for Fig. 17, it may be noted that in the region of small $\sigma\rho_l/G^2l$, a few exceptional data lie in the neighborhood of the broken line, but future research will be required to clarify this ambiguous phenomenon.

6.3. K *in HP-regime:* K_{HP}

Among the experimental data belonging to HP-regime, those of Chojnowski *et al.* [29] alone permit us to evaluate K_{HP} directly from the gradient of linear relation of $q_c - \Delta H_i$. The data compiled by Thompson and Macbeth (data* for $l/d = 365$ only in Tables 11–14) [6] and the data of Becker *et al.* (data of inlet subcooling 100°C in Fig. 12) [10] are composed of the independent combinations of q_c and ΔH_i. For these data, therefore, K_{HP} has been determined from equation (20) by employing the value of q_{c0} which is determined by equation (15). Figure 18 shows the correlation of K_{HP} thus obtained, and it is formulated as follows:

* It is revealed by the discrimination by means of Fig. 15 that the data for $l/d = 11.7, 50.0$, and 80.0 do not belong to HP-regime but to N-regime.

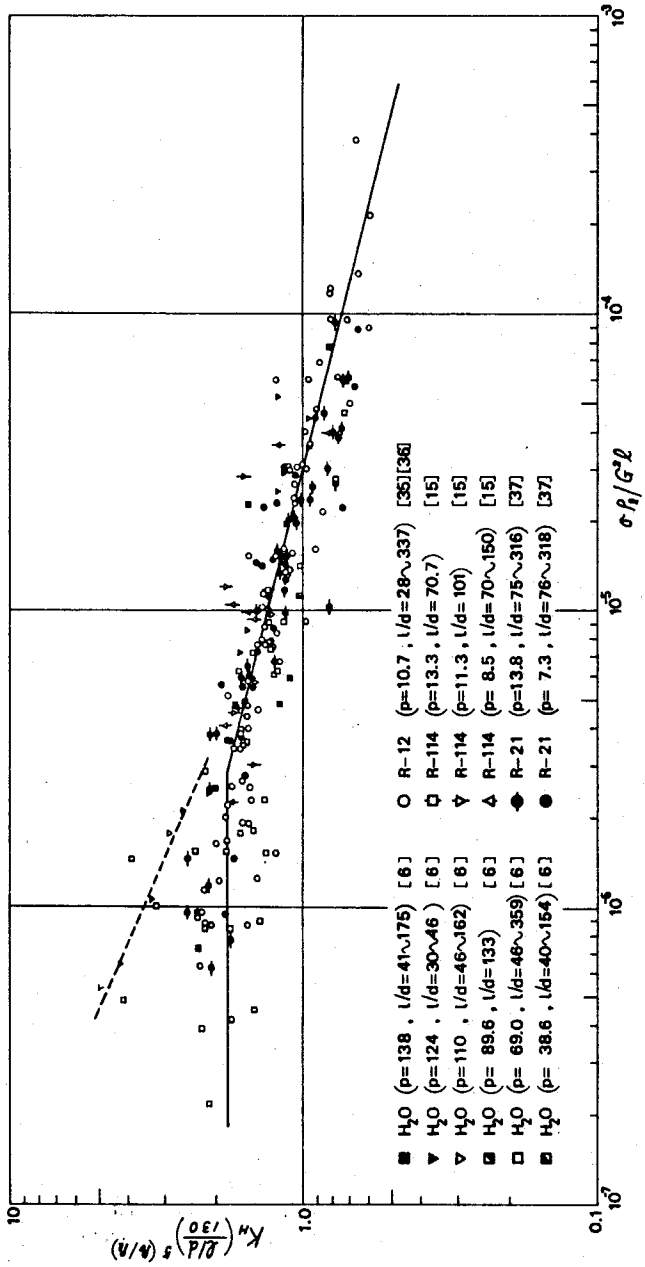

FIG. 17. Correlation of the experimental data of K_H.

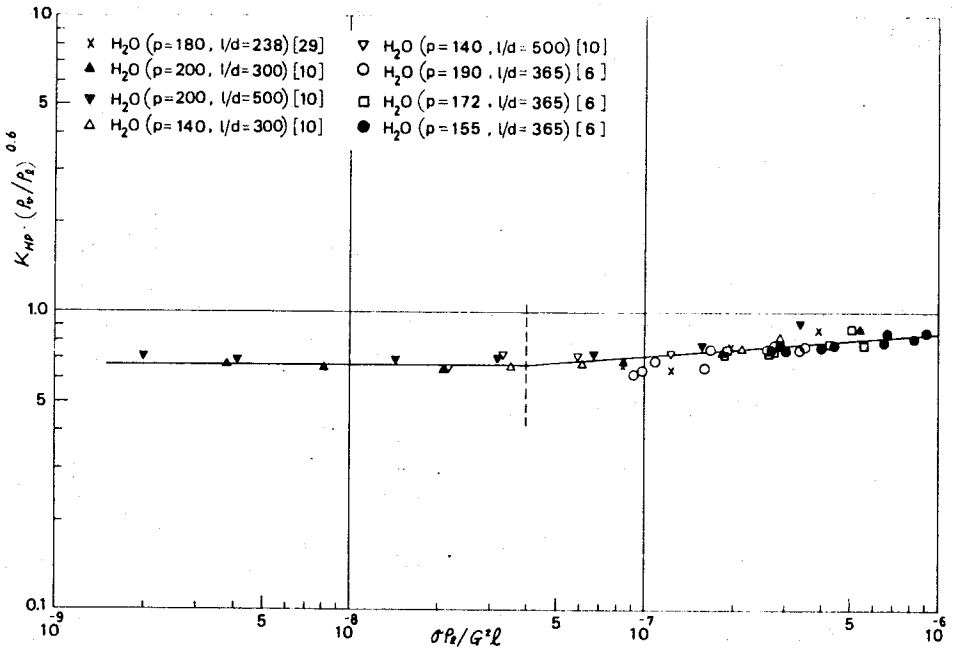

FIG. 18. Correlation of the experimental data of K_{HP}.

for $\sigma\rho_l/G^2l<4\times10^{-8}$,
$K_{HP}=0.664\,(\rho_v/\rho_l)^{-0.6}$
for $4\times10^{-8}<\sigma\rho_l/G^2l$,
$K_{HP}=3.08\,(\sigma\rho_l/G^2l)^{0.09}(\rho_v/\rho_l)^{-0.6}$. (23)

6.4. *Effect of inlet subcooling in N-regime*

For N-regime with the complexity of non-linear relationship for $q_c-\Delta H_i$, there is insufficient experimental data at present to produce correlations for the effect of inlet subcooling.

7. RELATION BETWEEN q_c AND χ_{ex}

For a uniformly heated tube, the heat flux q_c is related to the exit quality χ_{ex} via the heat balance, thus,

$$\frac{4q_c}{GH_{fg}}\frac{l}{d}-\frac{\Delta H_i}{H_{fg}}=\chi_{ex}. \tag{24}$$

Eliminating $\Delta H_i/H_{fg}$ from equations (20) and (24), and rearranging q_{co} by equations (12), (13), and (15) respectively, it yields

for L-regime,

$$\frac{q_c}{GH_{fg}}=0.25\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{K_L\chi_{ex}-1}{\left[K_L\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}-1\right]\frac{l}{d}} \tag{25}$$

where the constant 0.25 is to be replaced by 0.34 for Freons, and K_L is given by equation (21); and

for H-regime,

$$\frac{q_c}{GH_{fg}}=0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\times\frac{1-K_H\chi_{ex}}{1+\left[0.0031-0.40\,K_H\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\right]\frac{l}{d}} \tag{26}$$

where K_H is given by equation (22); and

for HP-regime,

$$\frac{q_c}{GH_{fg}}=8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.453}\times\frac{1-K_{HP}\chi_{ex}}{1+\left[107\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.54}-32.8K_{HP}\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.453}\right]\frac{l}{d}} \tag{27}$$

where K_{HP} is given by equation (23).

It should be stressed that equations (25) to (27) can apply only to the range of χ_{ex} corresponding to the possible range of $\Delta H_i/H_{fg}$ through equation (24).

8. CONCLUSIONS

Regarding CHF for the forced convection boiling in vertical uniformly heated round tubes, the experimental data of CHF for water, Freon-12, Freon-21, Freon-114, liquid nitrogen, liquid para-hydrogen and liquid herium I have been analyzed. As a result,

four regimes of CHF, called L-, H-, N-, and HP-regimes, have been classified, and employing the dimensionless groups of q_c/GH_{fg}, ρ_v/ρ_l, $\sigma\rho_l/G^2l$, l/d, and $\Delta H_i/H_{fg}$, equations (12) to (15) have been obtained for the correlation of CHF at zero inlet subcooling, Fig. 15 for the boundaries of each regime, and equations (21)–(23) for the effect of inlet subcooling on CHF.

It is hoped that through further testing against experiment, the above dimensionless correlation will be improved or modified in the future so as to predict CHF with much higher accuracy.

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UNE FORMULATION GENERALE DU FLUX THERMIQUE CRITIQUE POUR L'EBULLITION EN CONVECTION FORCEE DANS DES TUBES CIRCULAIRES, VERTICAUX ET UNIFORMEMENT CHAUFFES

Résumé—L'étude concerne le flux thermique critique (CHF) dans des conditions telles que le fluide attaquant les tubes est sous-refroidi (incluant le liquide saturé, en situation extrême) sans entrainement de vapeur. Postulant qu'il y a un état où la condition hydrodynamique est responsable du CHF, on essaie une équation de formulation générale du CHF, à l'aide de l'analyse dimensionnelle vectorielle. On analyse ensuite les résultats expérimentaux du CHF obtenus dans la bibliographie pour sept fluides différents, ce qui révèle l'existence de quatre régimes caractéristiques, désignés par régimes L, H, N et H P et conduit au développement d'une formulation générale des données de CHF aussi bien que du régime de CHF.

EINE VERALLGEMEINERTE BEZIEHUNG FÜR DIE KRITISCHE WÄRMESTROMDICHTE BEIM SIEDEN MIT ERZWUNGENER KONVEKTION IN SENKRECHTEN, GLEICHFÖRMIG BEHEIZTEN, RUNDEN ROHREN

Zusammenfassung—Die vorliegende Arbeit befaßt sich mit der kritischen Wärmestromdichte (KWSD) unter der Bedingung, daß das Fluid am Eintritt in die beheizten Rohre unterkühlt ist (eingeschlossen ist als Extremfall gesättigte Flüssigkeit), ohne daß Dampf mitgeführt wird. Zu Anfang wird unter der Voraussetzung, daß es einen Zustand gibt, bei dem die hydrodynamischen Bedingungen für die KWSD verantwortlich sind, eine theoretische Annahme für die verallgemeinerte Korrelationsgleichung für die KWSD mittels der vektoriellen Dimensionsanalyse gemacht. Dann werden experimentelle Daten für die KWSD aus der Literatur für sieben verschiedene Fluide analysiert, wobei die Existenz von vier charakteristischen Bereichen der KWSD offenbar wird, in dieser Arbeit *L*–, *H*–, *N*– und *HP*–Bereich genannt. Die Analyse führt zur Entwicklung sowohl der verallgemeinerten Korrelation von KWSD–Daten als auch der KWSD–Bereiche.

ОБОБЩЕННОЕ СООТНОШЕНИЕ ДЛЯ КРИТИЧЕСКОГО ТЕПЛОВОГО ПОТОКА ПРИ КИПЕНИИ В УСЛОВИЯХ ВЫНУЖДЕННОЙ КОНВЕКЦИИ В ВЕРТИКАЛЬНЫХ РАВНОМЕРНО НАГРЕВАЕМЫХ КРУГЛЫХ ТРУБАХ

Аннотация — В работе исследуется критический тепловой поток при подаче в нагреваемые трубы недогретой свободной от примеси пара жидкости, в том числе насыщенной жидкости, находящейся в экстремальных условиях. На основании предположения о том, что критический тепловой поток определяется гидродинамикой процесса, предпринята попытка обобщить уравнение критического теплового потока с помощью векторного анализа размерностей. Проведен анализ взятых из литературы экспериментальных данных по критическому тепловому потоку для семи различных типов жидкостей и показано наличие четырёх характерных режимов критического теплового потока, названных в данной работе режимами *L*, *H*, *N* и *HP*. Этот анализ даёт возможность обобщить данные по критическому тепловому потоку, а также получить обобщенное соотношение для режима критического теплового потока.

A GENERALIZED CORRELATION OF CRITICAL HEAT FLUX FOR THE FORCED CONVECTION BOILING IN VERTICAL UNIFORMLY HEATED ROUND TUBES—A SUPPLEMENTARY REPORT

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Abstract—Through the comparison with the prediction of a generalized correlation which was developed by the author in the preceding study, the existing data of critical heat flux (CHF) obtained in recent studies, including Milan reproducibility tests and others, are analyzed in the present paper. As the result, not only new information is obtained for the general structure of CHF, but also it is confirmed that the above-mentioned correlation is available as a tool for unifying experiments of various kinds. Besides, the propriety of dimensionless groups with which the generalized correlation is composed is also tested by comparing the prediction of mass flux scaling factor with empirical values obtained in the study of CHF modeling.

NOMENCLATURE

d, I.D. of heated tube [m];
F_G, mass flux scaling factor;
G, mass velocity [kg/m^2 s];
H_{fg}, latent heat of evaporation [J/kg];
ΔH_i, enthalpy of inlet subcooling [J/kg];
K, parameter for inlet subcooling effect, equation (6);
l, length of heated tube [m];
p, absolute pressure [bar];
q_c, critical heat flux [W/m^2];
q_{c0}, q_c for $\Delta H_i = 0$ [W/m^2].

Greek symbols

μ_l, viscosity of liquid [Ns/m^2];
μ_v, viscosity of vapor [Ns/m^2];
ρ_l, density of liquid [kg/m^3];
ρ_v, density of vapor [kg/m^3];
σ, surface tension [N/m];
χ_{ex}, exit quality;
$\chi_{ex,0}$, χ_{ex} for $\Delta H_i = 0$.

1. INTRODUCTION

AMONG recent experimental studies made on the critical heat flux (CHF) of forced convection boiling in vertical uniformly heated round tubes, one should not forget to refer to the study carried out in European laboratories in order to test the reproducibility of CHF data of water providing the Milan reproducibility tests data [1] which may be regarded as very reliable. On the other hand, Becker [2] and Becker and Ling [3] conducted the experiment on CHF of water for two particular cases of a small diameter tube and a tube having a great length. Cumo *et al.* [4] made a comparison of upflow and downflow CHF for Freon-12, Groeneveld [5] conducted an experiment on Freon-12 studying upstream CHF under the condition of not only negative but also positive quality at the inlet, and Campolunghi *et al.* [6] showed experimental results of water indicating a rapid change of CHF in the interval of pressure from 90 to 110 bars. As for investigations of pure fluids other than water and Freons, Alad'yev *et al.* [7] reported CHF data for potassium, and Pokhvalov *et al.* [8] obtained CHF data for benzene. Considerable progress was also achieved in the study of CHF modeling by Ahmad [9] and others. Unfortunately, however, it seems likely that many important studies such as mentioned above have been left separate and accordingly the mutual relations are not necessarily clear.

Recently, the author [10] developed a generalized correlation of CHF for the forced convection boiling in vertical uniformly heated round tubes. Being compared with the prediction of this correlation, in the present paper, the above-mentioned experimental data plus the data of Freon-22 [21] and Freon-11 [23] are analyzed in order to investigate the general structure of CHF, and at the same time, to test the applicability of the correlation.

2. CORRELATION EQUATIONS OF CHF

2.1. *Correlation equations of CHF*

From the generalized correlation developed by the author [10], equations necessary in the present paper are abstracted as follows:

First, for prediction of critical heat flux q_{c0} at $\Delta H_i = 0$, L-regime:

$$\frac{q_{c0}}{GH_{fg}} = C\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.043}\frac{1}{l/d} \tag{1}$$

where C is a constant, 0.25 or 0.34 depending on the conditions.

H-regime:

$$\frac{q_{c0}}{GH_{fg}} = 0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3}\frac{1}{1+0.0031\,l/d} \tag{2}$$

H- and N- regime:

$$\frac{q_{c0}}{GH_{fg}} = 0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.433} \times \frac{(l/d)^{0.27}}{1+0.0031\, l/d} \quad (3)$$

HP-regime:

$$\frac{q_{c0}}{GH_{fg}} = 8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.453} \times \frac{1}{1+107\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.54}\frac{l}{d}}. \quad (4)$$

With respect to equation (3), the boundary between H- and N-regime can be determined by

$$\frac{\sigma\rho_l}{G^2 l} = \left(\frac{0.77}{l/d}\right)^{2.70} \quad (5)$$

Next, in case of $\Delta H_i > 0$, the critical heat flux q_c exhibits the characteristic change such as shown in the region of $\Delta H_i > 0$ in Fig. 1, and for L-, H-, and HP-regime where the linear $q_c - \Delta H_i$ relationship holds, q_c can be predicted by

$$q_c = q_{c0}(1 + K\,\Delta H_i / H_{fg}) \quad (6)$$

where q_{c0} is given by equations (2)–(4), and K is given as follows:

$$K \text{ in L-regime: } K_L = 1 \quad (7)$$

K in H-regime:

for $\sigma\rho_l/G^2 l < 3\times10^{-6}$,

$$K_H = 1.8\left(\frac{130}{l/d}\right)^{5\rho_v/\rho_l}$$

for $3\times10^{-6} < \sigma\rho_l/G^2 l$,

$$K_H = 0.075\left(\frac{130}{l/d}\right)^{5\rho_v/\rho_l}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{-1/4} \quad (8)$$

K in HP-regime:

for $\sigma\rho_l/G^2 l < 4\times10^{-8}$,

$$K_{HP} = 0.664\,(\rho_v/\rho_l)^{-0.6}$$

for $4\times10^{-8} < \sigma\rho_l/G^2 l$,

$$K_{HP} = 3.08\,(\sigma\rho_l/G^2 l)^{0.09}(\rho_v/\rho_l)^{-0.6}. \quad (9)$$

2.2. *On the exit quality at CHF for $\Delta H_i = 0$*

When $\Delta H_i = 0$, the heat flux q_{c0} is related to the exit quality $\chi_{ex,0}$ via the heat balance, thus,

$$\chi_{ex,0} = 4(q_{c0}/GH_{fg})\, l/d \quad (10)$$

q_{c0}/GH_{fg} on the RHS of equation (10) is given by equations (1)–(4), and Fig. 2 shows the result in case of $l/d = 200$ as an example. From Fig. 2, tentative postulation of flow structure at CHF may be made as follows: the annular flow closely allied to dispersed flow in L-regime, where the deficiency of liquid on the heated surface may be responsible for CHF; the annular flow in H-regime, where the hydrodynamic instability of interface may lead to CHF; and the bubble flow in N-regime, where CHF may occur by the transition from nucleate boiling to film boiling. As for HP-regime, the flow structure remains rather vague to the author. However, since the study of CHF mechanism is not the objective of the present paper, the author will only refer to the investigations such as made by Bennett *et al.* [11], Whalley *et al.* [12], Nevstrueva [13], and others which contribute to the understanding of CHF mechanism.

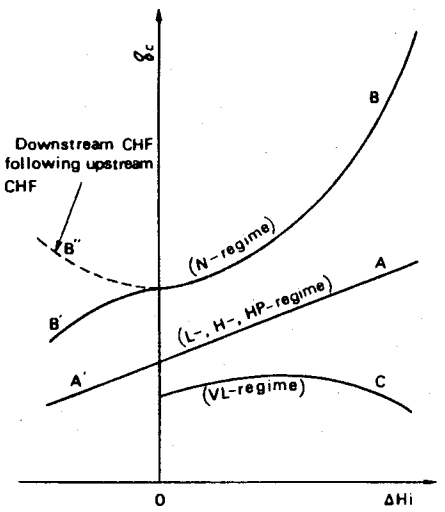

FIG. 1. Relationship between q_c and ΔH_i.

3. L-REGIME

3.1. *Comparison with Milan reproducibility tests and particular cases data.*

With respect to CHF in L-regime, Fig. 3 shows the comparison of equation (1) with the Milan reproducibility tests data of water (N.B. the data used by Becker *et al.* [14] to compare with various empirical correlations have been displayed in Fig. 3), the water data of Becker [2] for a small diameter tube (d = 3.0 mm), and the water data of Becker and Ling [3] for a very long tube (l = 7.1 m), on condition that q_{c0} data shown in Fig. 3 are those derived from the original data (q_c vs ΔH_i) through equations (6) and (7). It may be noted in Fig. 3 that CHF occurs at the state $\chi_{ex,0} = 1$ shown by the broken line in Fig. 3 in case of the very long tubes, whereas all the other data agree well with the prediction of equation (1). One point, however, should be checked, because the constant C in equation (1) has been determined in the author's preceding paper [10] as $C = 0.34$ for Freons and $C = 0.25$ for the other fluids, but by contrast in Fig. 3, $C = 0.34$ adapts for water.

Then Fig. 4 is prepared to examine the data of q_{c0} in L-regime derived through equations (6) and (7) from the experimental data compiled or presented by Thomson and Macbeth (Tables 1–10) [15] for water,

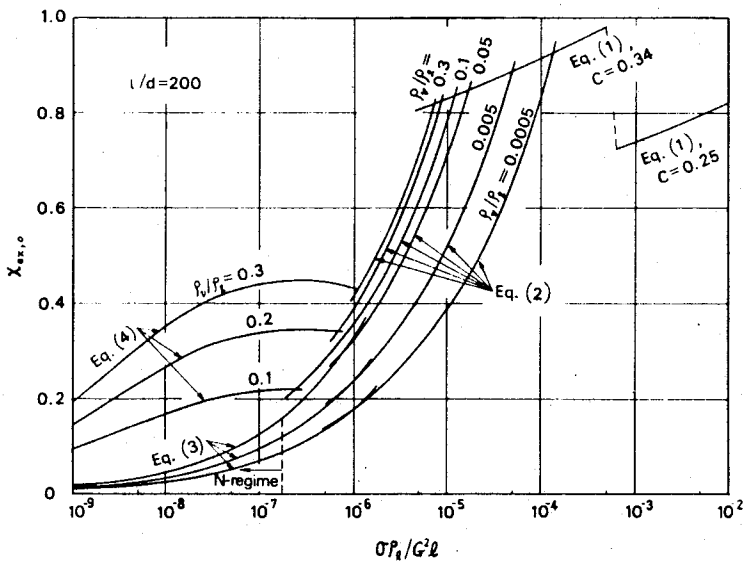

FIG. 2. Exit quality $\chi_{ex,o}$ in case of $\Delta H_i = 0$ and $l/d = 200$.

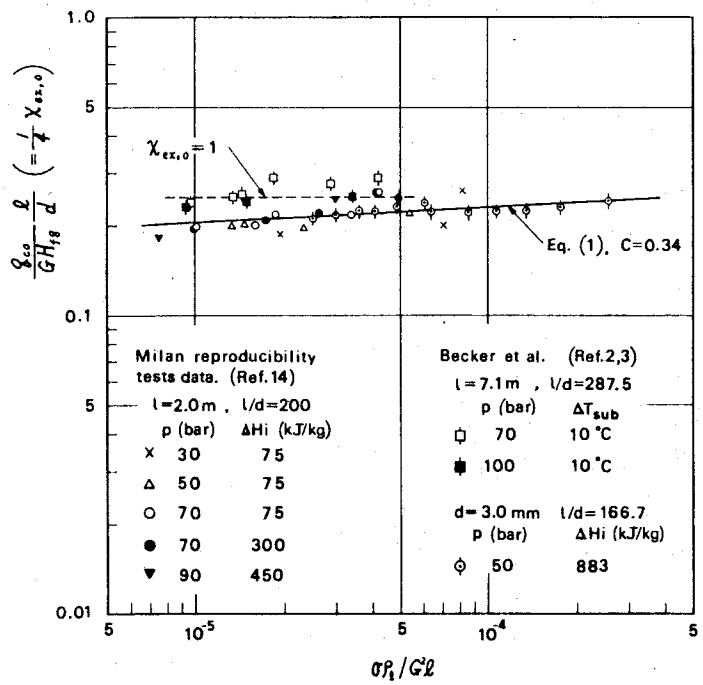

FIG. 3. Comparison between water data in L-regime and the prediction of equation (1).

Lewis *et al.* [16] for liquid hydrogen and nitrogen, Stevens *et al.* [17] for Freon-12 and Barnett *et al.* [18] for Freon-12 and -21. It may be noticed from Figs. 3 and 4 that the value of C seems to change in the vicinity of $\sigma\rho_l/G^2l = 5\times10^{-4}$ independently of the kind of fluid. In the author's preceding study [10] having no already-known information for K in equation (6), q_{c0} had to be derived by means of the extrapolation of q_c as $\Delta H_i \to 0$ along experimental curves such as shown in Fig. 1, and accordingly the available data of q_{c0} was so limited as to accidentally yield the result that the region of $\sigma\rho_l/G^2l < 5\times10^{-4}$ was occupied by the data of Freons alone, erroneously leading to the conclusion that $C = 0.34$ for Freons.

In sum, as for CHF in L-regime, there are usually

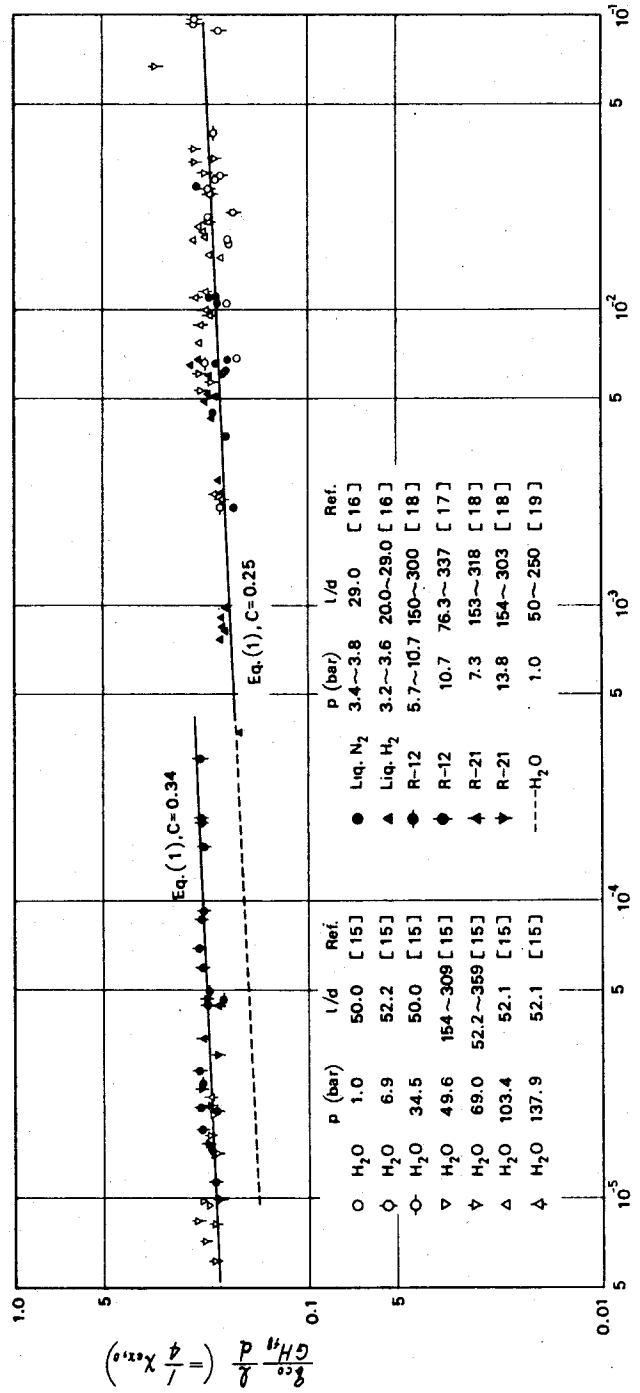

FIG. 4. Experimental data in L-regime and the constant C in equation (1).

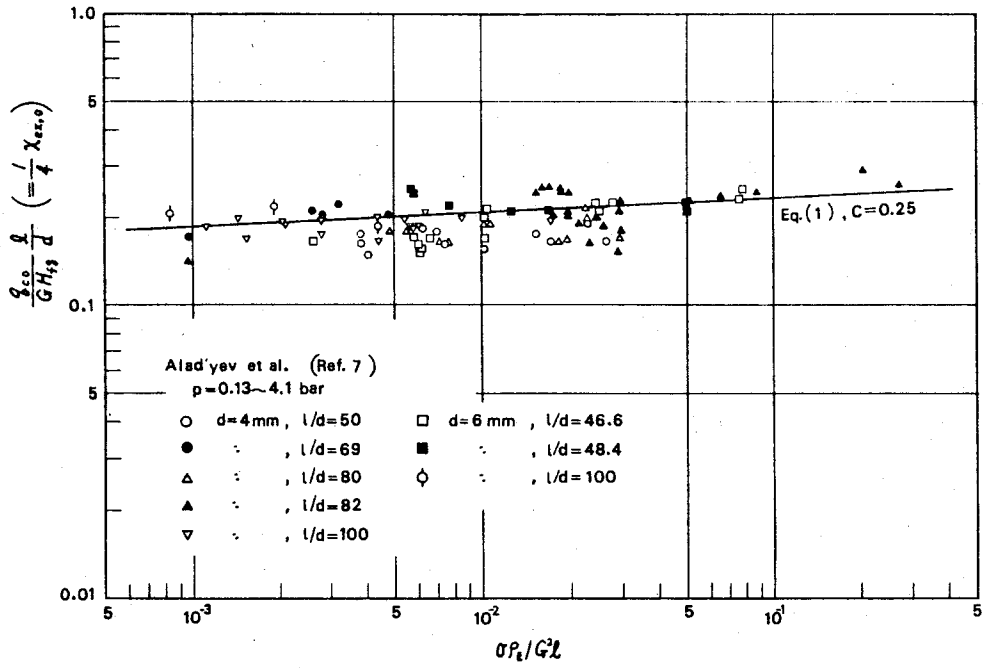

FIG. 5. Comparison between potassium data in L-regime and the prediction of equation (1).

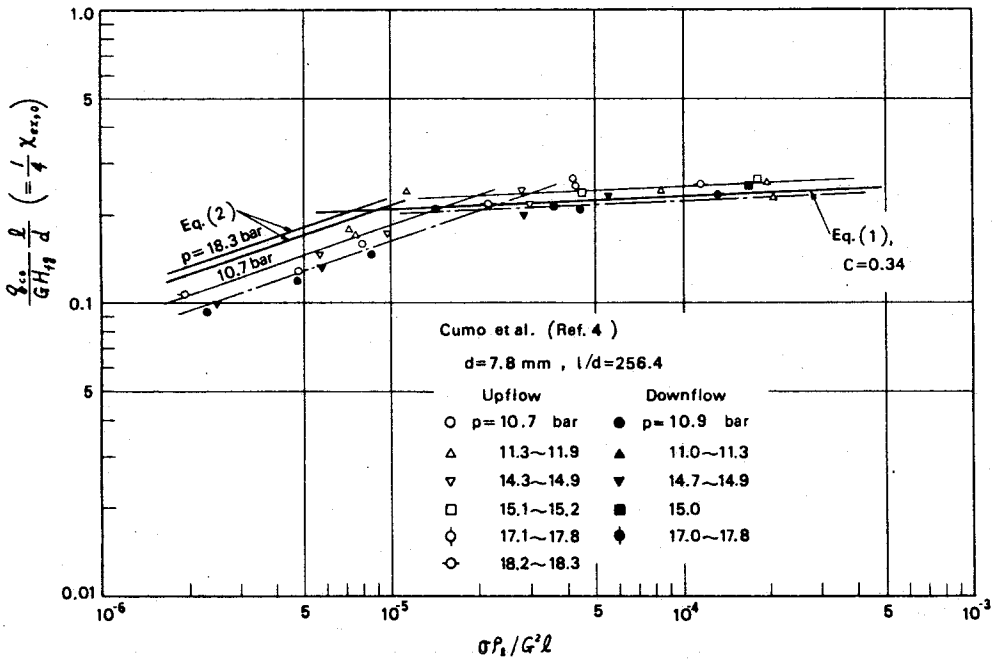

FIG. 6. Upflow and downflow data of Freon-12 in L- and H-regime with the prediction of equations (1), (2) and (5).

two regions of $\sigma\rho_l/G^2l$ characterized by $C = 0.25$ and 0.34, but in the special case of tubes with great length, $\chi_{ex} = 1$ may be the criterion for the onset of CHF. It should be noted, however, that a great many data of water at atmospheric pressure ($d = 1.3$–4.8 mm, $l/d = 50$–250) obtained by Lowdermilk *et al.* (Table III) [19] show the result which adapts to $C = 0.25$ over the whole range of $\sigma\rho_l/G^2l$ in Fig. 4 (see the broken line in Fig. 4). Therefore, further study is still necessary at least before giving the final conclusion for the characteristics of CHF in L-regime.

3.2. *Comparison with potassium data*

Figure 5 shows the comparison of equation (1) with q_{c0} data derived through equations (6) and (7) from the experimental data of Arad'yev *et al.* [7] for potassium ($p = 0.13$–4.1 bars). As for the surface tension of potassium used to prepare Fig. 5, the experimental data obtained by Kiriyanenko (Table 2) [20] at the boundary with potassium vapor has been employed. In addition, the data for $l/d = 30$ have been excluded from the display in Fig. 5, because they were obtained with the conditions including low inlet velocities less than 0.3 m/s, belonging to VL-regime (cf. [10]).

The data in Fig. 5 may be regarded to agree with equation (1) with $C = 0.25$ though some data lie a little lower, and it may be of interest to note that this result is properly in accord with the criterion for CHF determined in the preceding section for the range of $\sigma\rho_l/G^2l > 5\times10^{-4}$.

3.3. *Comparison between upflow and downflow data*

Figure 6 shows the comparison of equation (1) with q_{c0} data which can be obtained by means of the extrapolation of q_c as $\Delta H_i \to 0$ from the experimental data of Cumo *et al.* [4] for Freon-12. The experimental range is in $\sigma\rho_l/G^2l < 5\times10^{-4}$ and it is noted that the data belonging to L-regime show a fairly good agreement with equation (1) with $C = 0.34$ on condition that q_{c0} for upflow is about 11% higher than q_{c0} for downflow. It may not be useless to note that the above data in L-regime are under the condition that $G = 120$ to $325\,kg/m^2s$ and $l/d = 256.4$.

4. H-REGIME

4.1. *Comparison with Milan reproducibility tests data*

With respect to H-regime, Fig. 7 shows the comparison of equations (2) and (3) with q_{c0} data derived through equations (6) and (8) from the Milan reproducibility tests data of water (N.B. the data used by Becker *et al.* [14] have been displayed in Fig. 7). The vertical broken line in Fig. 7 represents the boundary of H- and N-regime predicted by equation (5). The agreement between the data and equations (2) and (3) is fairly good in spite of the difference of pressure as much as three times. If necessary, much better agreement can be realized by a slight modification of constants on the RHS of equations (2) and (3).

4.2. *Comparison with particular cases*

Figure 8 shows the comparison of equations (2) and (3) with q_{c0} data derived through equations (6)

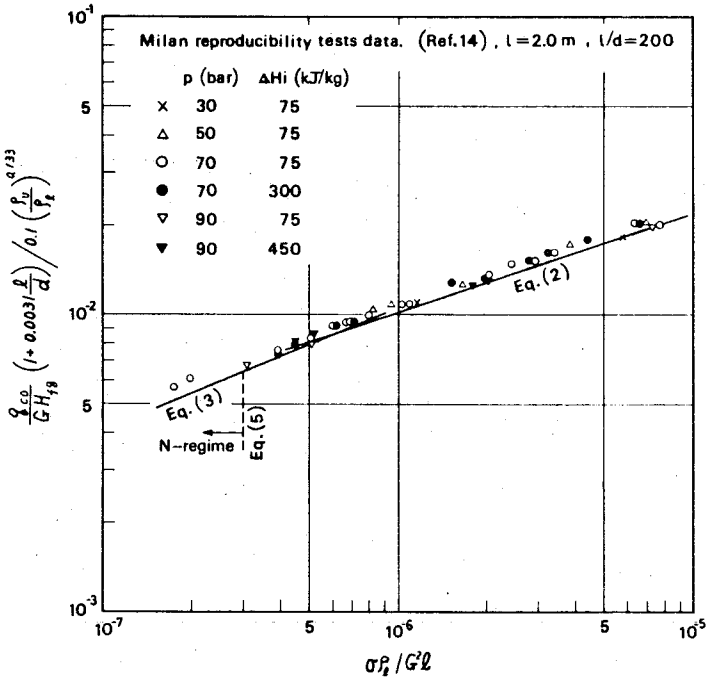

FIG. 7. Comparison between water data in H-regime and the prediction of equations (2), (3) and (5).

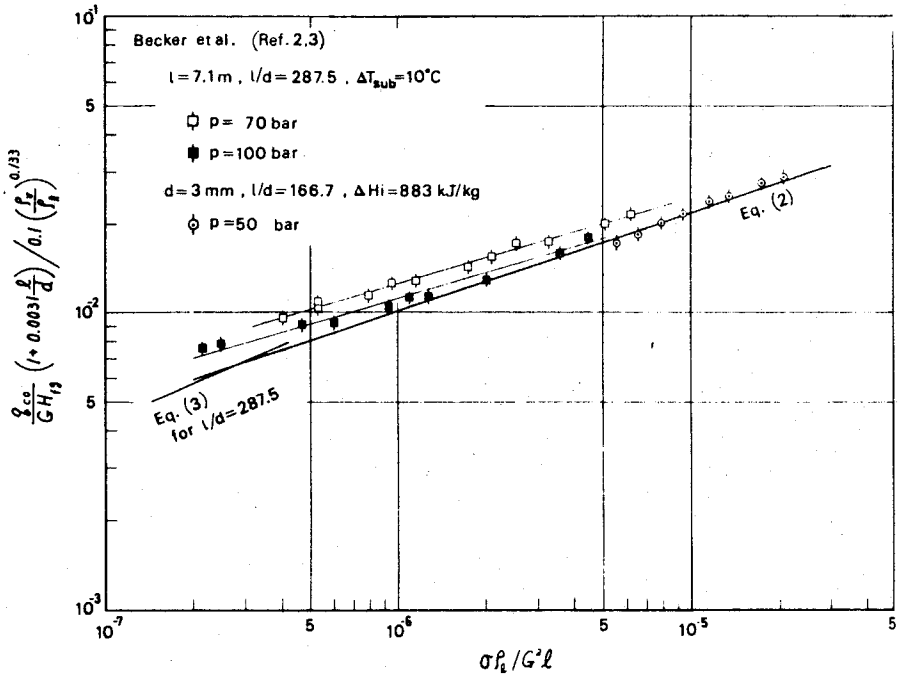

FIG. 8. Comparison between water data in H-regime and the prediction of equations (2) and (3).

and (8) from the experimental data of water in H-regime obtained by Becker [2] and Becker and Ling [3] for two particular cases of $d = 3$ mm and $l = 7.1$ m. The data for $d = 3$ mm show a good agreement with equation (2), whereas the data for the tube having a great length not only lie slightly higher than equation (2) but also show the variation of q_{c0}/GH_{fg} in proportion to $(\sigma\rho_l/G^2l)^{0.29}$ instead of $(\sigma\rho_l/G^2l)^{1/3}$.

4.3. *Comparison between upflow and downflow data*

In Fig. 6 the comparison of equation (2) with the data of Cumo *et al.* [4] in H-regime ($G = 503$–1043 kg/m^2 s, and $l/d = 256.4$) is also shown. It is noted that q_{c0} for upflow is about 11% higher than q_{c0} for downflow as in L-regime. The fact that the experimental data of q_{c0} lie a little lower than the prediction of equation (2) in Fig. 6 will be discussed later in Section 8.

4.4. *Comparison with Freon-22 data*

Figure 9 shows the comparison of equations (2) and (3) with q_c data derived through equations (6) and (8) from the experimental data of Freon-22 obtained by Staub (Table I and III) [21]. Though the column of q_c in Table I of [21] miscopies other data of CHF, q_c can be obtained from the data of $\Delta H_i/H_{fg}$ and χ_{ex} presented in the table via the heat balance [22]. Then, it may be noted in Fig. 9 that the data show a fairly good agreement with equation (2) except that some data lie a little lower than the prediction.

5. N-REGIME

5.1. *Comparison with Freon-12 data*

Groeneveld [5] made an experiment of Freon-12 in the region extending from negative to positive ΔH_i, and q_c at $\Delta H_i = 0$, that is q_{c0}, can be readily obtained from the reported $q_c - \Delta H_i$ relationship such as shown in Fig. 1, to give Fig. 10, where open symbols are the data in H-regime having the linear $q_c - \Delta H_i$ relationship (type A–A′ in Fig. 1) while black symbols represent the data in N-regime having the non-linear $q_c - \Delta H_i$ relationship (type B–B′ or B–B″ in Fig. 1). It is noted in Fig. 10 that the experimental data show a good agreement with the prediction of equations (2), (3) and (5).

5.2. *Comparison with Freon-11 data*

Purcupile *et al.* [23] have presented 37 data of Freon-11 ($l/d = 89.6$ and $p = 10.8$–24.8 bars), but it is found that all the data belong to N-regime without exception. Besides, the data fall in a very narrow range of ΔH_i, so that the direct comparison with equation (3) is impossible, because no correlation equation has yet been developed for the effect of subcooling in N-regime (cf. [10]). However, the imaginary quantity $(q_{c0}/GH_{fg})/(\rho_v/\rho_l)^{0.133}$, which can be derived through equations (6) and (8) from the above-mentioned data, is found to change accidentally in proportion to $(\sigma\rho_l/G^2l)^{0.15}$, so that the experimental value of $(q_{c0}/GH_{fg})/(\rho_v/\rho_l)^{0.133}$ at the boundary between N- and H-regime, for which $\sigma\rho_l/G^2l$ is determined by equation (5) for $l/d = 89.6$, can be readily extrapolated and the value thus

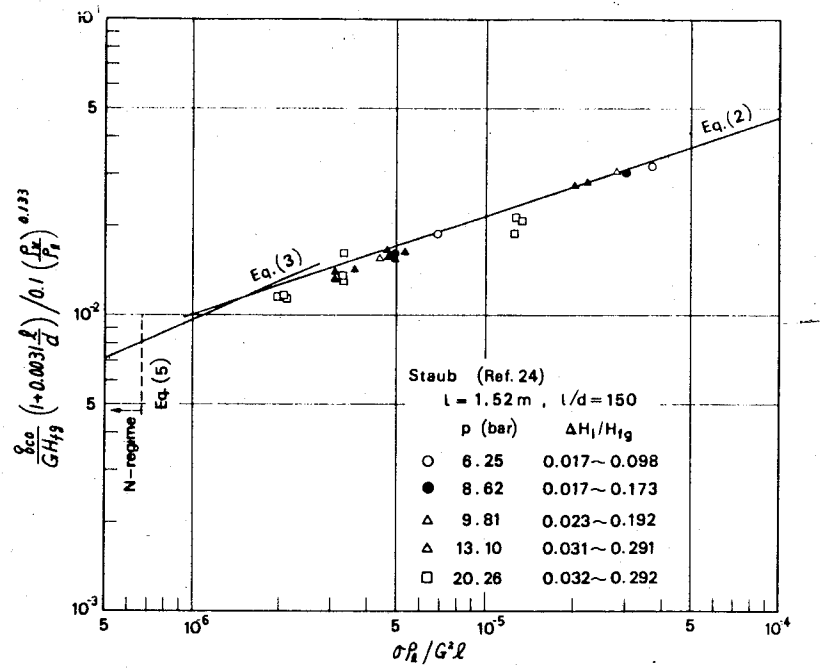

FIG. 9. Comparison between Freon-22 data in H-regime and the prediction of equations (2) and (3).

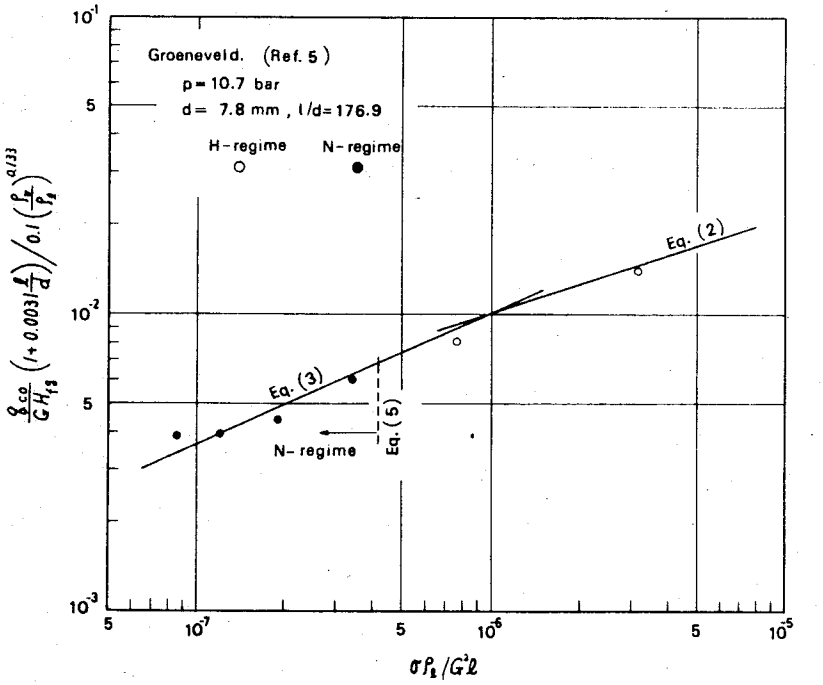

FIG. 10. Comparison between Freon-12 data in H- and N-regime and the prediction of equations (2), (3) and (5).

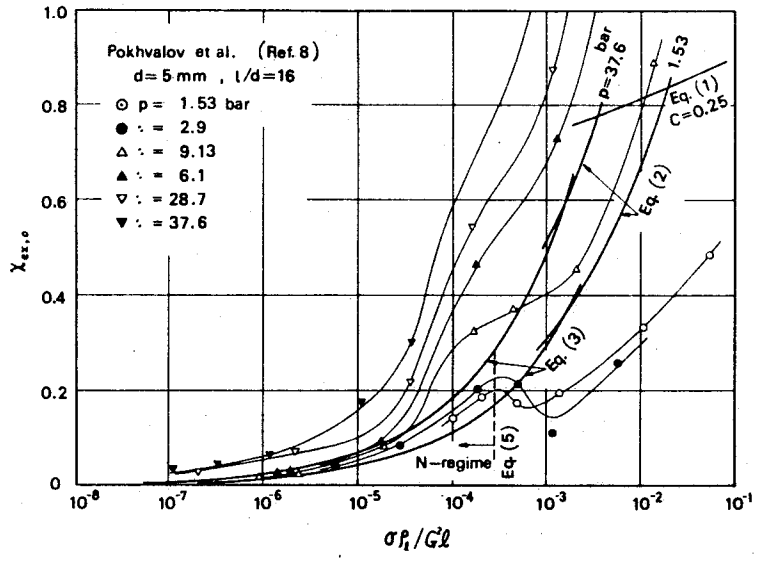

FIG. 11. Comparison between benzene data and the prediction of equations (1), (2), (3) and (5).

obtained is observed to agree well with the prediction of equation (3) at the above-mentioned boundary point.

5.3. *Comparison with benzene data*

Pokhvalov *et al.* [8] gave the experimental data of q_{c0} for benzene ($p = 1.53$ to 37.6 bars). In their experiment, not only is l/d very small as $l/d = 16$, but also it includes the range of very low mass velocities. Under such special conditions, natural convection boiling is apt to prevail rather than forced convection boiling (cf. VL-regime mentioned in the preceding study [10]); and in fact, Pokhvalov *et al.* themselves state that when mass velocity G is low, CHF shows the characteristics akin to those predicted by Kutateladze–Zuber correlation for CHF in pool boiling. According to Fig. 11, in which their experimental data are compared with equations (1)–(3) [transformed to $\chi_{ex.0}$ by equation (10)], it may be noticed that in the range of low G (that is, in the range of high $\sigma\rho_l/G^2l$), natural convection boiling prevails as mentioned above so that the data show a great deviation from the prediction. However, as G is increased (that is, $\sigma\rho_l/G^2l$ is reduced), the data in the range of $p = 1.53$ to 16.1 bars tend to show a good agreement with equation (3) in N-regime. As for the data in the range of higher pressure: $p = 28.7$ to 37.6 bars ($\rho_v/\rho_l = 0.14$ to 0.23), a slight deviation from equation (3) is observed, though the reason is unknown.

6. RELATIONSHIP OF $q_c-\Delta H_i$ IN THE RANGE OF $\Delta H_i<0$

Exactly speaking, if the fluid fed to a tube has negative subcooling ($\Delta H_i<0$) at inlet, it presents a problem that the two-phase flow at the inlet cannot be defined uniquely when only the value of ΔH_i is specified. However, this problem will be left untouched here, and some information of $q_c-\Delta H_i$ relationship found in the analysis of experimental results will be noted below.

According to the experimental result of Cumo *et al.* [4], which has provided the result of Fig. 6, it may be permissible to say that in L- and H-regime, the linear $q_c-\Delta H_i$ relationship (type A–A′ in Fig. 1) holds for $\Delta H_i<0$ as well as for $0<\Delta H_i$. This character is observed in the experiment of Groeneveld [5] too, but when $\sigma\rho_l/G^2l$ is reduced so as to enter N-regime, the $q_c-\Delta H_i$ relationship of type B′ in Fig. 1 begins to appear in the range of $\Delta H_i<0$ keeping the onset of CHF at the tube exit, and as $\sigma\rho_l/G^2l$ is reduced further, the $q_c-\Delta H_i$ relationship changes to the type B″ in Fig. 1 and at the same time the onset of upstream CHF appears.

7. TRANSITION BETWEEN H- AND HP-REGIME

Campolunghi *et al.* [6] reported the experiment of CHF of water for a vertical uniformly heated tube with a great length ($l = 11.25$ m, $d = 12$ mm) connected to an upstream, coiled preheating tube of the same length as above. The inlet subcooling ΔH_i of water entering the vertical tube is not given in their paper, but there is a statement that boiling occurs mainly in the vertical tube, so that $\Delta H_i = 0$ will be tentatively assumed here at the inlet of the vertical tube. The prediction of $\chi_{ex,0}$ obtained under this assumption from equations (2) and (4) [and equation (10)] is compared with the experimental data of Campolunghi *et al.* (given in Fig. 5 of [24]) in Fig. 12.

It may be observed in Fig. 12 that the agreement between the prediction of equation (4) and the data is comparatively good for HP-regime, whereas the data lie considerably higher than the prediction of

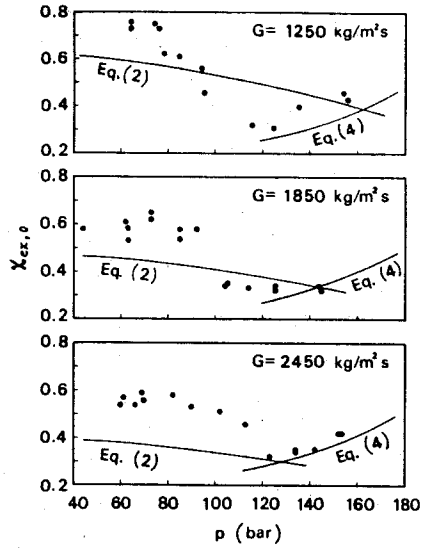

FIG. 12. CHF data of water obtained by Campolunghi *et al.* [6] for a very long, vertical tube ($l = 11.25$ m, $l/d = 937.5$).

equation (2) in H-regime. Taking into account the great length of tube ($l = 11.25$ m), the disagreement in H-regime may be attributed to the same cause as in the case of $l = 7.1$ m shown in Fig. 8. In addition, it should be noted that equations (2) and (4) can predict a rough trend for the transition between H- and HP-regime in Fig. 12, but it seems likely that further study is necessary for uncovering the more accurate feature of CHF in the vicinity of transition.

8. ON THE PROPRIETY OF DIMENSIONLESS GROUPS EMPLOYED

8.1. *Prediction of mass flux scaling factor* F_G

Apart from definite forms of correlation equations (1)–(9), let us only assume in this section that CHF in vertical uniformly heated round tubes is a phenomenon governed by five dimensionless groups q_c/GH_{fg}, $\sigma\rho_l/G^2l$, ρ_v/ρ_l, l/d, and $\Delta H_i/H_{fg}$. In this case, it follows that when the following conditions have been established between two different systems W and F:

$$(\rho_v/\rho_l)_W = (\rho_v/\rho_l)_F, \quad (l/d)_W = (l/d)_F,$$
$$\text{and} \qquad (\Delta H_i/H_{fg})_W = (\Delta H_i/H_{fg})_F, \qquad (11)$$

if the mass velocities $(G)_W$ and $(G)_F$ in the respective systems are selected so as to satisfy

$$(\sigma\rho_l/G^2l)_W = (\sigma\rho_l/G^2l)_F, \qquad (12)$$

the following relation holds

$$(q_c/GH_{fg})_W = (q_c/GH_{fg})_F \qquad (13)$$

so that $(q_c)_W$ in the system W can be known from the experimental data of $(q_c)_F$ in the system F. Now, if the case of $(d)_W = (d)_F$ is considered for simplicity, equation (13) can be rewritten under the condition of equations (11) and (12) as follows:

$$\frac{(q_c)_W}{(q_c)_F} = F_G\frac{(H_{fg})_W}{(H_{fg})_F},$$
$$\text{where} \qquad (14)$$
$$F_G = \frac{(G)_W}{(G)_F} = \left[\frac{(\sigma\rho_l)_W}{(\sigma\rho_l)_F}\right]^{1/2}.$$

Equation (14) is the prediction of mass velocity scaling factor F_G for the phenomenon controlled by the five dimensionless groups in equations (11)–(13).

8.2. *Discussion*

Regarding mass velocity scaling factor $F_G = (G)_W/(G)_F$, Dix [25] compared the experimental data of CHF obtained for Freons-114, -12, -22 and -21 with those for water to give the result that the empirical value of F_G thus obtained is in proportion to $[(\sigma\rho_v)_W/(\sigma\rho_v)_F]^{1/2}$. This result agrees with the prediction of equation (14) under the condition of $(\rho_v/\rho_l)_W = (\rho_v/\rho_l)_F$ given in equation (11).

On the other hand, Table 1 lists F_G given by equation (14) in case of $\rho_l/\rho_v = 20$ with the exception of $\rho_l/\rho_v = 590$ for water-potassium, together with the empirical values of F_G obtained by Stevens and Macbeth [26] for water-Freon-12, Staub [21] for water-Freon-22 and water–potassium, Coeffield [27] for water-Freon-113, and Hauptmann [28] for water-carbon dioxide. Table 1 also lists Ahmad's prediction of F_G [9], which is given by

$$F_G = \left[\frac{(\gamma/\rho_l\mu_l)_F}{(\gamma/\rho_l\mu_l)_W}\right]^{1/3}\left[\frac{(\mu_l/\mu_v)_F}{(\mu_l/\mu_v)_W}\right]^{1/8} \qquad (15)$$

where $\gamma = |\partial(\rho_l/\rho_v)/\partial p|_{sat}$. It is noted in Table 1 that the prediction of equation (14) is always a little lower than the empirical value of F_G in case of Freons. On this point, it may not be useless to note the fact that statistically speaking, the experimental data of q_{c0} in

Table 1. Comparison of the value of F_G at density ratio $\rho_l/\rho_v = 20$ except $\rho_l/\rho_v = 590$ for water–potassium

	System W	H_2O	H_2O	H_2O	H_2O	H_2O
	System F	R-12	R-22	R-113	CO_2	K
F_G	Equation (14)	1.28	1.34	1.21	1.19	1.24
	Empirical	1.40[26]	1.40[21]	1.46[27]	1.13[28]	~1.0[21]
	Ahmad [9]	1.38	1.38	1.34	1.2	1.1*

* The value for $\rho_l/\rho_v = 980$.

H- and N-regime have a trend to be somewhat lower than the prediction of equations (2) and (3) for Freons, which can be seen in Figs. 6 and 11 of the author's preceding study [10] as well as in Figs. 6 and 9 of the present study. On the contrary, Ahmad's prediction of F_G shown in Table 1 is much closer to the empirical value of F_G, and his prediction is primarily based on the framework having the following dimensionless group ψ_{CHF} instead of $\sigma\rho_l/G^2l$:

$$\frac{1}{\psi_{CHF}} = \left(\frac{\sigma\rho_l}{G^2d}\right)^{2/3}\left(\frac{Gd}{\mu_l}\right)^{1/3}\left(\frac{\mu_l}{\mu_v}\right)^{1/5}. \quad (16)$$

This fact seems to imply that the viscosity of fluid can have the secondary effect on the occurrence of CHF of Freons in H- and N-regime at least. However, in order to give a definite answer to this problem, it may also be important to know more reliable data than those known at present for the surface tension of Freons. Besides, the deviation found between the prediction and the empirical value for water-potassium in Table 1 may be regarded to relate to a different cause, because it is concerned with the experimental data of potassium in L-regime shown in Fig. 5.

Finally, it may be of use to note that the propriety of the dimensionless group $\sigma\rho_l/G^2l$, which has not so far been taken into consideration in existing studies, cannot be examined here, because equation (12) is replaced by $(\sigma\rho_l/G^2d)_W = (\sigma\rho_l/G^2d)_F$ automatically under the condition $(l/d)_W = (l/d)_F$ given in equation (11).

8. CONCLUSIONS

(i) The generalized correlation [equations (1)–(9)], showing a fairly wide applicability, may be used as a tool for unifying independent experimental works of various kinds.

(ii) Tentative conclusions are given for CHF in L-regime as follows: usually, the constant C in equation (1) takes the value $C = 0.34$ for $\sigma\rho_l/G^2l < 5 \times 10^{-4}$ and $C = 0.25$ for $5 \times 10^{-4} < \sigma\rho_l/G^2l$ independently of the kind of fluid, and in the special case of very long tubes, CHF seems to be able to occur at the state of $\chi_{ex} = 1$.

(iii) Milan reproducibility tests data show a fairly good agreement with equations (1)–(3). In the special case of very long tubes, experimental data of CHF have a trend to deviate from equations (1)–(3) to some extent.

(iv) For fundamental investigations, the dimensionless groups with which the author's generalized correlation is composed have been examined through the comparison with the results of CHF modeling.

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UNE CORRELATION GENERALE DU FLUX THERMIQUE CRITIQUE EN EBULLITION AVEC CONVECTION FORCEE DANS DES TUBES VERTICAUX ET CHAUFFES UNIFORMEMENT. UN RAPPORT SUPPLEMENTAIRE

Résumé—On analyse les données de flux thermique critique (CHF) obtenues dans des études récentes incluant les tests de reproductibilité de Milan, en les comparant avec une corrélation générale qui a été développée par l'auteur dans une étude précédente. Non seulement de nouvelles informations sont obtenues sur la structure générale du CHF, mais on confirme aussi que la corrélation est valable en tant qu'outil pour unifier des expériences de divers types. La propriété des groupes adimensionnels qui composent la corrélation générale est testée en comparant les prévisions du facteur d'échelle du flux massique avec les valeurs empiriques obtenues sur le CHF.

EINE VERALLGEMEINERTE KORRELATION FÜR DIE KRITISCHE WÄRMESTROMDICHTE BEIM SIEDEN BEI ERZWUNGENER KONVEKTION IN SENKRECHTEN, GLEICHMÄSSIG BEHEIZTEN, RUNDEN ROHREN –ERGÄNZUNGSBERICHT

Zusammenfassung—Durch den Vergleich mit den Ergebnissen einer verallgemeinerten Korrelation, die vom Autor in der vorhergehenden Arbeit entwickelt wurde, wurden in dieser Arbeit vorliegende Daten über die kritische Wärmestromdichte analysiert. Diese Daten entstammen neuesten Untersuchungen, die die Reproduzierbarkeitstests von Milan und anderen beinhalten. Als Ergebnis wurden nicht nur neue Information über die allgemeine Struktur der kritischen Wärmestromdichte erhalten, sondern es wurde auch bestätigt, daß die obengenannte Korrelation ein brauchbares Instrument für die Vereinheitlichung von Untersuchungen der verschiedensten Arten ist. Außerdem wurde die Zweckmäßigkeit der dimensionslosen Gruppen, aus denen sich die verallgemeinerte Korrelation zusammensetzt, durch den Vergleich des berechneten Maßstabfaktors für den Massenstrom mit empirischen Werten geprüft, die man bei Modellversuchen zur kritischen Wärmestromdichte erhalten hat.

ОБОБЩЕННОЕ ВЫРАЖЕНИЕ ДЛЯ КРИТИЧЕСКОГО ТЕПЛОВОГО ПОТОКА ПРИ КИПЕНИИ С ВЫНУЖДЕННОЙ КОНВЕКЦИЕЙ В ВЕРТИКАЛЬНЫХ РАВНОМЕРНО НАГРЕВАЕМЫХ КРУГЛЫХ ТРУБАХ

Аннотация — С помощью обобщенного выражения, полученного автором в предыдущей работе, анализируются новые данные по критическому тепловому потоку (КТП). В результате анализа получены не только новые сведения о характере КТП, но и подтверждено, что предложенное выражение можно использовать для обобщения данных различных экспериментов. Кроме того, при сопоставлении результатов расчёта по обобщенному выражению и эмпирических значений, полученных при моделировании КТП, проверена корректность безразмерных комплексов, входящих в это выражение.

AN ANALYSIS OF THE EFFECT OF INLET SUBCOOLING ON CRITICAL HEAT FLUX OF FORCED CONVECTION BOILING IN VERTICAL UNIFORMLY HEATED TUBES

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Abstract—In this paper, the generalized correlation of critical heat flux (CHF) derived by the author in previous studies is combined with the boiling length concept to make an analysis for obtaining theoretical predictions for the effect of inlet subcooling on CHF. Fairly good agreements are found between the theoretical predictions thus derived and the existing data of experiments. Then the extent of inlet subcooling over which the boiling length method is applicable for correlating CHF data is examined quantitatively by comparing the CHF data obtained in studies of so-called dryout and post-dryout heat transfer with the author's generalized correlation.

NOMENCLATURE

- d, I.D. of heated tube [m];
- G, mass velocity [kg m^{-2} s^{-1}];
- H_{fg}, latent heat of evaporation [J kg^{-1}];
- ΔH_i, inlet subcooling enthalpy [J kg^{-1}];
- K, dimensionless parameter for the effect of inlet subcooling, equation (1);
- l, length of heated tube [m];
- l_b, boiling length, equation (3) [m];
- M, $M = 0.0031l/d$;
- N, $N = 107(\sigma\rho_l/G^2l)^{0.54}l/d$;
- p, absolute pressure, Figs. 3–10 [bar];
- q_c, critical heat flux [W m^{-2}];
- q_{c0}, q_c for $\Delta H_i = 0$ [W m^{-2}];
- r, relative error, equations (9), (15) and (21);
- z, axial coordinate, Fig. 2 [m].

Greek symbols

- ε, $= (1/4)(d/l)(G\Delta H_i/q_c)$;
- ρ_l, density of liquid [kg m^{-3}];
- ρ_v, density of vapor [kg m^{-3}];
- σ, surface tension [N m^{-1}];
- χ, quality (χ_{ex}: exit quality corresponding to critical condition; χ_{in}: inlet quality).

1. INTRODUCTION

FOR CRITICAL heat flux (CHF) of forced convection boiling in vertical uniformly heated tubes, it is well known [1, 2] that plots of q_c against ΔH_i for fixed G often indicate the linear relationship such as is shown in Fig. 1, and it can be written as follows:

$$q_c = q_{c0}(1 + K\Delta H_i/H_{fg}), \tag{1}$$

where K is a dimensionless parameter independent of ΔH_i. Analyzing the experimental data of CHF, the author [3, 4] has shown that CHF data are classified into four regimes called L-, H-, N- and HP-regime. the linear equation (1) applies to regimes other than N-regime, and q_{c0} on the RHS of equation (1) is correlated by equations of the following form:

$$\frac{q_{c0}}{GH_{fg}} = f\left(\frac{\rho_v}{\rho_l}, \frac{\sigma\rho_l}{G^2l}, \frac{l}{d}\right). \tag{2}$$

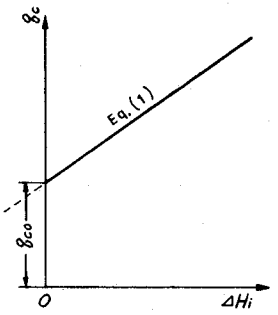

FIG. 1. Linear relationship between q_c and ΔH_i for fixed G.

On the other hand, the idea of the boiling length l_b indicated in Fig. 2(b), that is the tube length minus the length necessary to bring the thermodynamic quality to zero, was devised by Hewitt *et al.* [5] and Bertoletti *et al.* [6] for example, and has since come into use in correlating CHF data (e.g. GEXL correlation [7]). In the special case of $\Delta H_i < 0$ (that is, mixed inlet condition), the boiling length l_b can also be defined as indicated in Fig. 2(c). Then, for a uniformly heated tube, l_b is related to ΔH_i via the heat balance thus,

$$\frac{l_b}{l} = 1 - \varepsilon, \quad \text{where } \varepsilon = \frac{1}{4}\frac{d}{l}\frac{G\Delta H_i}{q_c}. \tag{3}$$

Equation (3) applies to both the case of Fig. 2(b) where $0 < \Delta H_i$ and that of Fig. 2(c) where $\Delta H_i < 0$.

Flow pattern at the point of reaching zero quality in Fig. 2(b) differs generally from that at the entrance of the heated tube in Fig. 2(a) where a saturated liquid is fed with no entrained vapor. However, in the case of $0 < \varepsilon \ll 1$ or $l_b/l = 1$ in equation (3), the boiling front

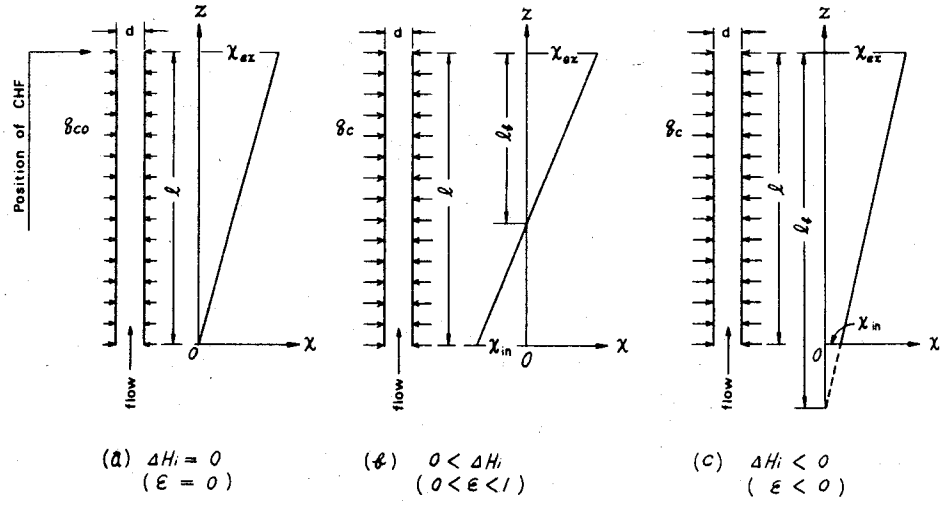

FIG. 2. Linear variation of quality χ along a uniformly heated tube.

can be approximately assumed to agree with the position of zero quality in Fig. 2(b). In other words, the CHF in this case can be correlated with the same function as equation (2), so that it is written as

$$\frac{q_c}{GH_{fg}} = f\left(\frac{\rho_v}{\rho_l}, \frac{\sigma\rho_l}{G^2 l_b}, \frac{l_b}{d}\right). \tag{4}$$

In the case of Fig. 2(c), the point of zero quality becomes an imaginary one, but if $l_b/l \doteqdot 1$, the same assumption of boiling front as above may be laid down without serious error.

Then, q_c in equation (1) with q_{c0} given by equation (2) should coincide with q_c in equation (4) with l_b given by equation (3), providing us with a means to discuss theoretically the parameter K on the RHS of equation (1).

2. PARAMETER K IN L-REGIME

According to the author's correlation of q_{c0} [3, 4], equations (2) and (4) are written for L-regime as follows:

$$\frac{q_{c0}}{GH_{fg}} = C\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.043} \frac{1}{l/d} \tag{5}$$

$$\frac{q_c}{GH_{fg}} = C\left(\frac{\sigma\rho_l}{G^2 l_b}\right)^{0.043} \frac{1}{l_b/d}, \tag{6}$$

where $C = 0.34$ for both $\sigma\rho_l/G^2 l$ and $\sigma\rho_l/G^2 l_b < 5\times10^{-4}$, and $C = 0.25$ for both $\sigma\rho_l/G^2 l$ and $\sigma\rho_l/G^2 l_b > 5\times10^{-4}$. Then, dividing each member of equation (6) by that of equation (5), and substituting equation (3) for l_b, it yields the relation

$$q_c(1-\varepsilon)^{1.043} = q_{c0}. \tag{7}$$

When $|\varepsilon| < 1$, equation (7) can be expanded with respect to ε, and higher order terms than the second term can be ignored to give

$$q_c(1-1.043\varepsilon) = q_{c0} \tag{8}$$

with the relative error r defined as follows:

$$r = \frac{1-1.043\varepsilon}{(1-\varepsilon)^{1.043}} - 1. \tag{9}$$

Equation (9) indicates that the error can remain small for a comparatively wide range of ε. Then, replacing G included in ε of equation (8) by G of equation (5), it yields

$$q_c = q_{c0}\left\{1 + \frac{1.043}{4C(\sigma\rho_l/G^2 l)^{0.043}} \cdot \frac{\Delta H_i}{H_{fg}}\right\}.$$

Comparing this equation with equation (1), it gives K in L-regime (designated by K_L) as follows:

$$K_L = \frac{1.043}{4C(\sigma\rho_l/G^2 l)^{0.043}}, \tag{10}$$

where C takes the value indicated in equation (5).

In Fig. 3, K_L predicted by equation (10) is compared with the experimental data of K_L compiled in Fig. 16 of the preceding study [3]. Though data are limited in their number and show scattering, approximate agreement may be witnessed between predicted and measured K_L.

3. PARAMETER K IN H-REGIME

According to the author's correlation of q_{c0} [3], equations (2) and (4) are written for H-regime as follows:

$$\frac{q_{c0}}{GH_{fg}} = 0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3} \frac{1}{1+0.0031 l/d} \tag{11}$$

$$\frac{q_c}{GH_{fg}} = 0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l_b}\right)^{1/3} \frac{1}{1+0.0031 l_b/d} \tag{12}$$

Dividing each member of equation (12) by that of

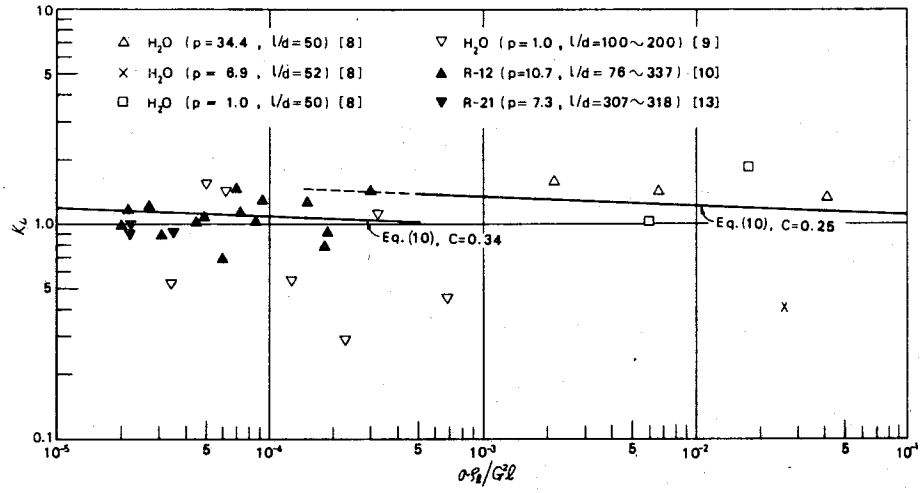

FIG. 3. Comparison between prediction of equation (10) and experimental data for K_L.

equation (11) and substituting equation (3) for l_b yields

$$q_c(1-\varepsilon)^{1/3}\frac{1+M(1-\varepsilon)}{1+M}=q_{c0}, \tag{13}$$

where $M=0.0031l/d$. When $|\varepsilon|<1$, equation (13) can be expanded with respect to ε, and higher order terms than the second term can be ignored to give

$$q_c\left\{1-\left(\frac{1}{3}+\frac{M}{1+M}\right)\varepsilon\right\}=q_{c0} \tag{14}$$

with the relative error r defined as follows:

$$r=\frac{1-\left(\frac{1}{3}+\frac{M}{1+M}\right)\varepsilon}{(1-\varepsilon)^{1/3}\frac{1+M(1-\varepsilon)}{1+M}}-1. \tag{15}$$

Then, replacing G included in ε of equation (14) by G of equation (11), it yields

$$q_c=q_{c0}\left\{1+\frac{5}{6}\frac{0.0124+d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{1/3}}\cdot\frac{\Delta H_i}{H_{fg}}\right\}.$$

Comparing this equation with equation (1) yields K in H-regime (designated by K_H) as follows:

$$K_H=\frac{5}{6}\frac{0.0124+d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{1/3}}. \tag{16}$$

In Fig. 4, K_H predicted by equation (16) is compared with the experimental data of K_H compiled in Fig. 17 of the preceding study [3]. It may be concluded that agreement between predicted and measured K_H is fairly good.

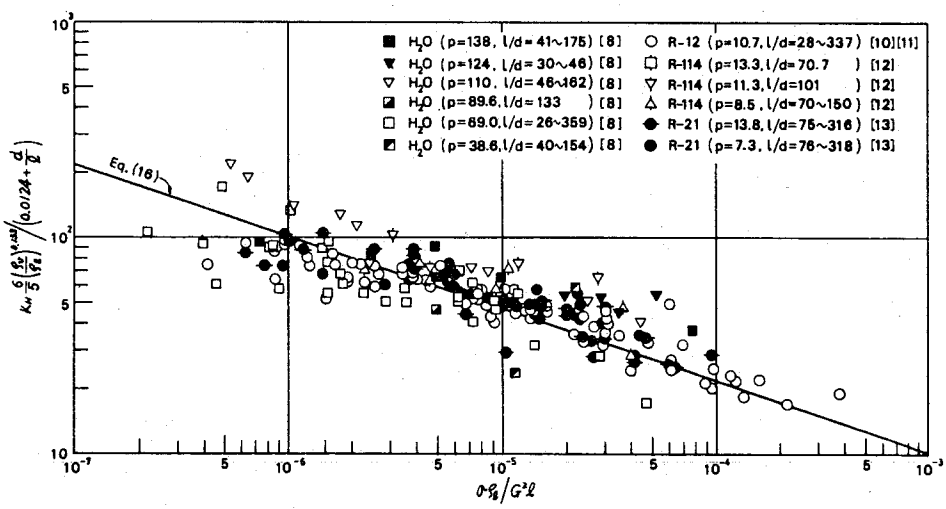

FIG. 4. Comparison between prediction of equation (16) and experimental data for K_H.

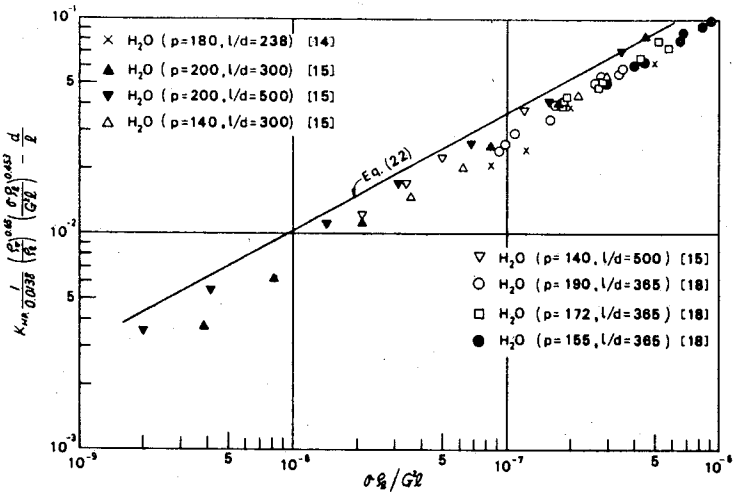

FIG. 5. Comparison between prediction of equation (22) and experimental data for K_{HP}.

4. PARAMETER K IN HP-REGIME

According to the author's correlation of q_{c0} [3], equations (2) and (4) are written for HP-regime as follows:

$$\frac{q_{c0}}{GH_{fg}} = 8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.453} \times \frac{1}{1+107(\sigma\rho_l/G^2 l)^{0.54} l/d} \quad (17)$$

$$\frac{q_c}{GH_{fg}} = 8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2 l_b}\right)^{0.453} \times \frac{1}{1+107(\sigma\rho_l/G^2 l_b)^{0.54} l_b/d}. \quad (18)$$

Then, in the same way as in the preceding section, the following equation is derived from equations (17), (18) and (3):

$$q_c(1-\varepsilon)^{0.453}\frac{1+N(1-\varepsilon)^{0.46}}{1+N} = q_{c0}, \quad (19)$$

where $N = 107(\sigma\rho_l/G^2 l)^{0.54} l/d$, and equation (19) can be replaced by

$$q_c\left\{1-\left(0.453+0.46\frac{N}{1+N}\right)\varepsilon\right\} = q_{c0} \quad (20)$$

with the relative error r defined as follows:

$$r = \frac{1-\left(0.453+0.46\dfrac{N}{1+N}\right)\varepsilon}{(1-\varepsilon)^{0.453}\dfrac{1+N(1-\varepsilon)^{0.46}}{1+N}} - 1. \quad (21)$$

Then, in the same way as before, equations (17), (20) and (1) yield K in HP-regime (designated by K_{HP}) as follows:

$$K_{HP} = 0.0138\frac{215(\sigma\rho_l/G^2 l)^{0.54} + d/l}{(\rho_v/\rho_l)^{0.65}(\sigma\rho_l/G^2 l)^{0.453}}. \quad (22)$$

It may be of interest to note that if d/l is removed from the RHS of equation (22), the equation is written as $K_{HP} = 2.97(\sigma\rho_l/G^2 l)^{0.087}/(\rho_v/\rho_l)^{0.65}$, which is nearly consistent with one of the equations of K_{HP} obtained empirically in the previous study [3], $K_{HP} = 3.08(\sigma\rho_l/G^2 l)^{0.09}/(\rho_v/\rho_l)^{0.6}$. Anyhow, in Fig. 5, K_{HP} predicted by equation (22) is compared with the experimental data of K_{HP} compiled in Fig. 18 of the preceding study [3]. The prediction of equation (22) is slightly higher than the experimental data, and it suggests that the correlation equation (17) may have a little defect with respect to the effect of l/d on CHF. At the present stage, however, the number of available data of q_{c0} in HP-regime is so limited that definite conclusions cannot be obtained.

5. PERMISSIBLE RANGE OF BOILING LENGTH METHOD

On the premise that the author's generalized correlation of q_{c0} [equations (5), (11) and (17)] is substantially valid, and that the linear $q_c - \Delta H_i$ relationship (Fig. 1) holds empirically in L-, H- and HP-regime, the results of Figs. 3–5 are regarded as showing the existence of a range in which the boiling length method for correlating CHF data is applicable with a permissible margin of error. It should be mentioned here that according to the existing CHF data (cf. Ref. [8] and others), the linear $q_c - \Delta H_i$ relationship can be observed to hold to the extent of $\varepsilon > 1$. Besides, logically speaking, it has been unnecessary in the present section to worry about the disagreement of flow pattern between the zero quality point in Fig. 2(a) and that in Fig. 2(b) and (c). Then, in compliance with a marginal magnitude of error, a permissible range of the boiling length method can be determined by equations (9), (15) and (21) for L-, H- and HP-regime respectively.

Numerical results thus obtained are shown in Tables 1–3, and it may be noticed that for the type of

Table 1. Permissible range of boiling length method in L-regime

$\|r\| < 0.025$	$\|r\| < 0.050$	$\|r\| < 0.100$
$-2.70 < \varepsilon^* < 0.59$	$-6.75 < \varepsilon^* < 0.69$	$-28.9 < \varepsilon^* < 0.79$

*r does not become positive for this range of ε.

Table 2. Permissible range of boiling length method in H-regime

M	$\|r\| < 0.025$	$\|r\| < 0.050$	$\|r\| < 0.100$
0.05	$-0.67 < \varepsilon\dagger < 0.40$	$-1.06 < \varepsilon\dagger < 0.52$	$-1.78 < \varepsilon\dagger < 0.65$
0.15	$-0.92 < \varepsilon\dagger < 0.44$	$-1.66 < \varepsilon\dagger < 0.56$	$-4.27 < \varepsilon\dagger < 0.68$
0.30	$-5.08 < \varepsilon < 0.52$	$-6.74 < \varepsilon < 0.63$	$-10.2 < \varepsilon < 0.74$
0.60	$-1.42 < \varepsilon < 0.72$	$-2.24 < \varepsilon < 0.78$	$-3.89 < \varepsilon < 0.85$
1.20	$-0.79 < \varepsilon^* < 0.47$	$-1.29 < \varepsilon^* < 0.62$	$-2.30 < \varepsilon^* < 0.80$
2.50	$-0.60 < \varepsilon^* < 0.35$	$-0.97 < \varepsilon^* < 0.45$	$-1.73 < \varepsilon^* < 0.57$
∞	$-0.45 < \varepsilon^* < 0.26$	$-0.73 < \varepsilon^* < 0.34$	$-1.30 < \varepsilon^* < 0.43$

*r does not become positive for this range of ε.
†r does not become negative for this range of ε.

Table 3. Permissible range of boiling length method in HP-regime

N	$\|r\| < 0.025$	$\|r\| < 0.050$	$\|r\| < 0.100$
0.50	$-0.69 < \varepsilon^* < 0.39$	$-1.12 < \varepsilon^* < 0.50$	$-1.94 < \varepsilon^* < 0.62$
1.00	$-0.79 < \varepsilon^* < 0.41$	$-1.31 < \varepsilon^* < 0.52$	$-2.39 < \varepsilon^* < 0.64$
2.00	$-0.92 < \varepsilon^* < 0.44$	$-1.59 < \varepsilon^* < 0.55$	$-3.09 < \varepsilon^* < 0.66$
5.00	$-1.11 < \varepsilon^* < 0.47$	$-2.01 < \varepsilon^* < 0.58$	$-4.25 < \varepsilon^* < 0.69$
∞	$-1.43 < \varepsilon^* < 0.51$	$-2.37 < \varepsilon^* < 0.62$	$-6.34 < \varepsilon^* < 0.73$

*r does not become negative for this range of ε.

Fig. 2(c), that is $\varepsilon < 0$, the range of ε for a fixed margin of error is comparatively wide in L-regime, whereas for the type of Fig. 2(b), that is $0 < \varepsilon < 1$, the range of ε for a fixed margin of error takes nearly the same value for L-, H- and HP-regime.

Supplementary note: N-regime

According to the preceding study [3], q_{c0} in N-regime is expressed as:

$$\frac{q_{c0}}{GH_{fg}} = 0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.433}\frac{(l/d)^{0.27}}{1+0.0031 l/d},$$

and the same procedure as that in Section 3 based on the boiling length concept provides the following equations instead of equations (15) and (16):

$$r = \frac{1-\left(0.163+\dfrac{M}{1+M}\right)\varepsilon}{(1-\varepsilon)^{0.163}\dfrac{1+M(1-\varepsilon)}{1+M}} - 1$$

$$K = 0.416\frac{(0.0221+d/l)(d/l)^{0.27}}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{0.433}}.$$

On the other hand, however, it has been found in the preceding study [3] that linear $q_c - \Delta H_i$ relationship, that is equation (1), does not hold in N-regime. Therefore, it may be concluded that the boiling length concept is inapplicable in N-regime; and this character seems natural because CHF in N-regime is presumed to occur in a state very near zero quality.

6. TEST OF PERMISSIBLE RANGE OF BOILING LENGTH METHOD

6.1. *Experimental data*

In order to test the permissible range of the boiling length method predicted in Tables 1–3, experimental data of CHF obtained by Ueda *et al.* [16, 17] on R-113, Ueda [18] on R-113, Hynek *et al.* [19] on liquid nitrogen, and Bennett *et al.* [20] on water will be used. These experiments are concerned with the study of so-called dryout and post-dryout heat transfer, but no singular problems arise so long as the length of heated tube l used in this paper is measured as the distance between the entrance of the test tube and the dryout point which appears generally upstream from the exit of the test tube.

Table 4 lists main conditions of the experiments above-mentioned, including the experimental range of ε, l_b/l and M. Classification of experimental conditions into two regimes of L and H in Table 4 has been made in the following way. Namely, if it is tentatively assumed that the boiling length l_b is applicable for correlating CHF over the whole range of these experiments, experimental conditions can be plotted as in Figs. 6–8, where the boundary between adjoining regimes is given by the following equations derived by the author in previous studies [3, 4]: for the boundary between L- and H-regime,

$$\frac{l_b}{d} = \frac{1}{0.10C(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l_b)^{0.29} - 0.0031}, \qquad (23)$$

where C takes the value indicated for equation (6), and for the boundary between H- and N-regime,

$$\frac{l_b}{d} = \frac{0.77}{(\sigma\rho_l/G^2 l_b)^{0.37}}. \tag{24}$$

In Figs. 6–8, experimental conditions located in and close to L-regime are indicated by open symbols, whereas those conditions located in H-regime by black symbols.

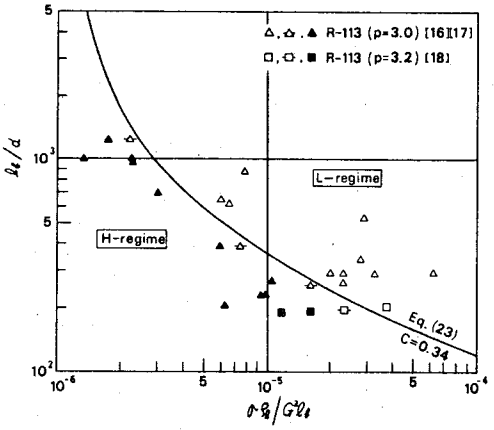

FIG. 6. Experimental conditions for R-113 [16–18].

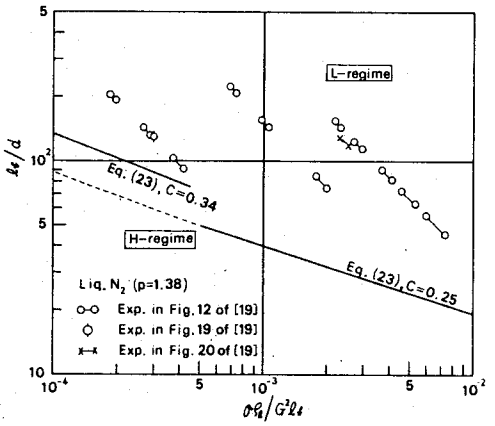

FIG. 7. Experimental conditions for liquid nitrogen [19].

6.2. *CHF in L-regime*

In Fig. 9, data of critical heat flux q_c obtained under the experimental conditions of open symbols are compared with the prediction of equation (6). The experimental data scatter to some extent, but it may be concluded that the data agree with the prediction of equation (6) based on the boiling length concept; and this result is regarded as reasonable because only minor errors are estimated by equation (9) within the experimental range of ε in L-regime indicated in Table 4.

Besides, it may be of interest to note in Fig. 9 that R-113 data ($p = 3.0$ bar) obtained by Ueda *et al.* [16, 17] for the type of Fig. 2(c) agree approximately with equation (6) for values of l_b/l extending from 14.06 to 2.50 (cf. Table 4). This may suggest that the flow in L-regime has a poor memory for the effect of upper stream and is governed by local conditions.

6.3. *CHF in H-regime*

In Fig. 10, data of critical heat flux q_c obtained for the experimental conditions in H-regime in Figs. 6–8 are compared with the prediction of equation (12). It may be noticed in Fig. 10 that the data obtained by Ueda *et al.* [16, 17] for the type of Fig. 2(c) suffer from scattering and have a trend to deviate from the prediction of equation (12), whereas the data obtained by Ueda [18] and Bennett *et al.* [19] for the type of Fig. 2(b) agree well with the prediction of equation (12). These results are consistent with the degree of error predicted by equation (15) for the experimental range of ε and M indicated in Table 4.

7. CONCLUSIONS

(i) The effect of inlet subcooling on CHF in L-, H- and HP-regime is analyzed, yielding equations (10), (16) and (22) to predict the parameter K in equation (1).

(ii) Fairly good agreement between the predicted and the measured K shown in Figs. 3–5 may be regarded as an indirect proof of the substantial validity of the author's generalized correlation of q_{c0}, that is, equations (5), (11) and (17).

(iii) The permissible range of the boiling length method is predicted by equations (9), (15) and (21) corresponding to a required magnitude of error.

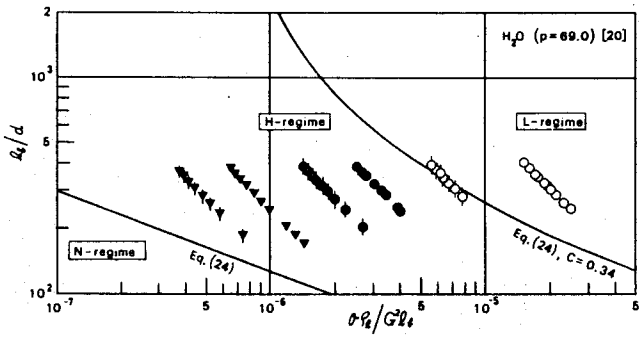

FIG. 8. Experimental conditions for water [20].

Table 4. Experimental range of ε, l_b/l, and M for data compared

Type		$\varepsilon < 0$	$0 < \varepsilon$		
Experiment		Ueda *et al.* [16, 17]	Ueda [18]	Hynek *et al.* [19]*	Bennett *et al.* [20]†
Fluid		R-113	R-113	Liq. N_2	H_2O
p (bar)		3.0	3.2	1.38	69.0
d (m)		0.01	0.01	0.0102	0.0126
L-regime	ε	$-13.06 \sim -1.50$	$0.174 \sim 0.197$	~ 0	$0.094 \sim 0.116$
	$\frac{l_b}{l}$	$14.06 \sim 2.50$	$0.826 \sim 0.803$	~ 1	$0.906 \sim 0.884$
H-regime	ε	$-23.73 \sim -3.95$	$0.216 \sim 0.231$	—	$0.101 \sim 0.166$
	$\frac{l_b}{l}$	$24.73 \sim 4.95$	$0.784 \sim 0.796$	—	$0.899 \sim 0.834$
	M	$0.0338 \sim 0.405$	0.760	—	$0.873 \sim 1.36$

*Data in Figs. 12, 19 and 20 of [19] are employed, excluding not only the data with tape, but also data of $G = 2 \times 10^4$ lb/ft² h in Fig. 20 because of VL-regime (cf. Ref. [3]).

†Data in Table 1 of [20] are employed, excluding data near N-regime ($G = 2.8 \times 10^6$ and 3.8×10^6 lb/ft² h).

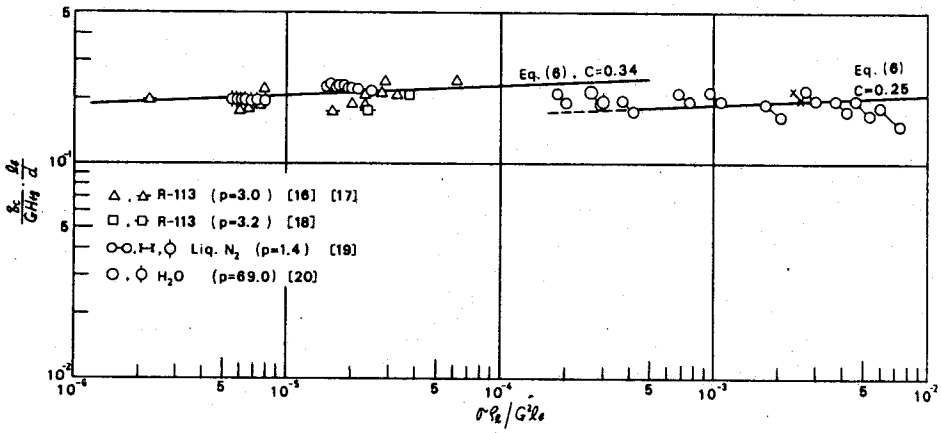

FIG. 9. Comparison between prediction of equation (6) and experimental data for CHF in L-regime.

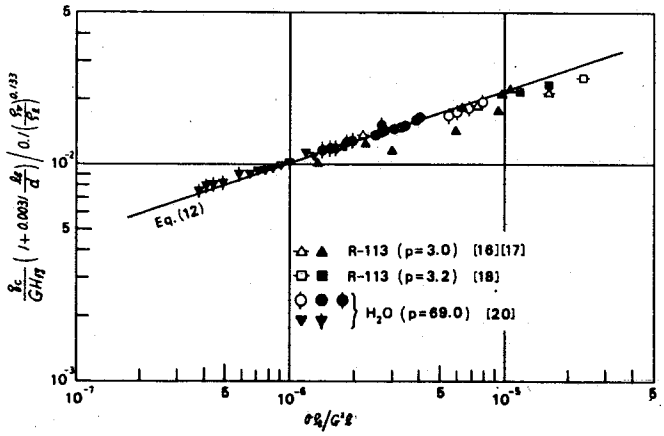

FIG 10. Comparison between prediction of equation (12) and experimental data for CHF in H-regime.

(iv) When the inlet subcooling is within the permissible range of the boiling length method, the regime of CHF can be discriminated by employing the boiling length l_b instead of the heated length l [cf. equations (23) and (24)].

(v) According to the experimental data analyzed in the present paper, it seems likely that the flow in L-regime has a trend to establish it in a proper manner at every local position.

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ANALYSE DE L'EFFET DU SOUS-REFROIDISSEMENT A L'ENTREE SUR LE FLUX CRITIQUE EN EBULLITION AVEC CONVECTION FORCEE, DANS DES TUBES VERTICAUX ET UNIFORMEMENT CHAUFFES

Résumé—La formule généralisée du flux critique (CHF) obtenue par l'auteur dans des études précédentes est combinée avec le concept de longueur d'ébullition pour obtenir le calcul théorique de l'effet du sous-refroidissement à l'entrée sur le CHF. Un très bon accord est obtenu entre les estimations théoriques et les données de l'expérience. On examine quantitativement l'extension du sous-refroidissement d'entrée et de la méthode de la longueur d'ébullition applicable au calcul du CHF en comparant les données de CHF obtenues dans les études de l'assèchement et du transfert de chaleur après assèchement, avec la formule généralisée de l'auteur.

EINE UNTERSUCHUNG DES EINFLUSSES DER EINTRITTSUNTERKÜHLUNG AUF DIE KRITISCHE WÄRMESTROMDICHTE BEIM STRÖMUNGSSIEDEN IN SENKRECHTEN, GLEICHFÄRMIG BEHEIZTEN ROHREN

Zusammenfassung—Zur theoretischen Untersuchung des Einflusses der Eintrittsunterkühlung auf die kritische Wärmestromdichte (CHF) wird hier die vom Autor in früheren Untersuchungen hergeleitete verallgemeinerte Korrelation für CHF mit dem Konzept der Siede-Länge Kombiniert. Die Übereinstimmung der so gewonnenen Ergebnisse mit den vorhandenen experimentellen Daten ist leidlich gut. Ferner wird der Bereich der Eintrittsunterkühlung, in dem die Siede-Längen-Methode zur Berechnung von CHF-Werten anwendbar ist, quantitativ untersucht; hierzu werden CHF-Werte aus Wärmeübergangsuntersuchungen im dryout- und post-dryout-Gebiet mit der verallgemeinerten Korrelation des Autors verglichen.

АНАЛИЗ ВЛИЯНИЯ НЕДОГРЕВА НА ВХОДЕ ВЕРТИКАЛЬНЫХ РАВНОМЕРНО НАГРЕВАЕМЫХ ТРУБ НА КРИТИЧЕСКИЙ ТЕПЛОВОЙ ПОТОК ПРИ КИПЕНИИ В УСЛОВИЯХ ВЫНУЖДЕННОЙ КОНВЕКЦИИ

Аннотация — Для определения влияния недогрева на входе на критический тепловой поток (КТП) автор в данной статье использует понятие длины кипения и полученное им ранее обобщенное соотношение для КТП. Найденное таким образом соотношение хорошо описывает имеющиеся экспериментальные данные. Проведен количественный анализ степени недогрева на входе, для которой применим метод длины кипения, путём сравнения данных по КТП, полученных при исследовании так называемых критических и закритических режимов теплообмена, с обобщенным соотношением автора.

TOWARD THE SYSTEMATIC UNDERSTANDING OF CHF OF FORCED CONVECTION BOILING (CASE OF UNIFORMLY HEATED ROUND TUBES)

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ABSTRACT

The object of this paper is to study the possibility of finding empirically the underlying law which can predict the various trends of critical heat flux (CHF) of forced convection boiling in uniformly heated tubes. The author's generalized correlation of CHF, which classifies CHF into four characteristic regimes, is used for this purpose. The correlation equations concerned are presented in the tabular form for the reader's convenience, with informations about the relations between CHF regimes and flow patterns. The propriety of the prediction for the effects of principal parameters (l, d, G and ΔH_i) on CHF is discussed by comparing the predicted CHF with the existing experimental results. The correspondence of the correlation equations with the scaling laws of fluid modeling is also discussed. Finally, future trends in the study of CHF are discussed with special reference to the role of the systematic understanding of CHF.

NOMENCLATURE

C = constant, Eq.(5)
d = I.D. of heated tube, m
FG = mass velocity scaling factor
G = mass velocity, kg/m^{2}s
H_{fg} = latent heat of evaporation, J/kg
ΔH_i = enthalpy of inlet subcooling, J/kg
K = inlet subcooling coefficient, Eq.(10)
l = length of heated tube, m
p = absolute pressure, bar
q = heat flux, W/m^2
q_c = critical heat flux, W/m^2
q_{co} = q_c for $\Delta H_i = 0$
u_i = inlet velocity of liquid to the system, m/s
μ_l = viscosity of liquid, Pa.s
μ_v = viscosity of vapor, Pa.s
ρ_l = density of liquid, kg/m^3
ρ_v = density of vapor, kg/m^3
σ = surface tension, N/m
x_{ex} = exit quality at CHF condition

1. INTRODUCTION

Critical heat flux (CHF) of forced convection boiling is a phenomenon which is important in energy problems, but is so complicated that we are apt to be as confused as the case of five blind men and the elephant. Systematic understanding of CHF is indispensable to establish proper relations between individual studies. Recently, excellent studies of theoretical analysis of CHF [1-6] have been made for annular steam-water flow with remarkable results for this purpose. In these studies, however, not a few assumptions or approximations must be made regarding two-phase flow and CHF condition, so that the limits of the proper application range of the analytical method are not necessarily clear. At the present stage, therefore, it may be of use to establish empirical methods capable of predicting the whole trend of CHF systematically. In this paper, the discussion is restricted to the most common and simplest CHF which takes place under conditions where a subcooled liquid (or a saturated liquid as a limit) is fed to a uniformly heated tube with a sufficiently high mass velocity and the flow is stable with no oscillations. The generalized correlation recently derived by the author for this CHF [7-11] is used as a tool to test the possibility of systematic understanding.

2. GENERALIZED CORRELATION EQUATIONS OF CHF

2.1 Background of correlation

Fig.1 illustrates two typical boiling systems of saturated liquid flowing parallel to a flat heated surface, for which the author [12,13] has made experimental studies of CHF employing different fluids for the range of l = 10 ∿ 20 mm, $\sigma\rho_l/G^2 l$ = 2 x 10^{-6} ∿ 3 x 10^{-3} (where $G = \rho_l u_i$), and ρ_v/ρ_l = 0.000624 ∿ 0.0103 to give the following generalized correlation equations: for the system of Fig.1(a),

$$\frac{q_{co}}{GH_{fg}} = 0.0164 \left(\frac{\rho_v}{\rho_l}\right)^{0.133} \left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3} \qquad (1)$$

and for the system of Fig.1(b),

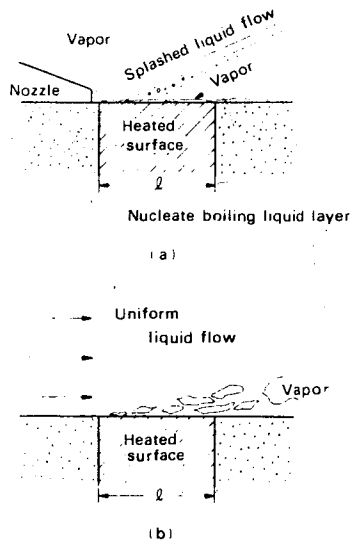

Fig.1 Two Typical Boiling Systems of Forced Convection Flowing Parallel to a Heated Flat Surface: (a) Plane Jet Flow, (b) Uniform Velocity Flow

$$\frac{q_{co}}{GH_{fg}} = 0.186 \left(\frac{\rho_v}{\rho_l}\right)^{0.559} \left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.264} \qquad (2)$$

It is noted that either of Eqs.(1) and (2) is the function of the density ratio ρ_v/ρ_l and the Weber number $G^2 l/\sigma\rho_l$ composed of l, the heated surface length in direction of flow.

On the other hand, Fig.2 shows two limiting cases which may be postulated for forced convection boiling in tubes. The open flow system of Fig.1

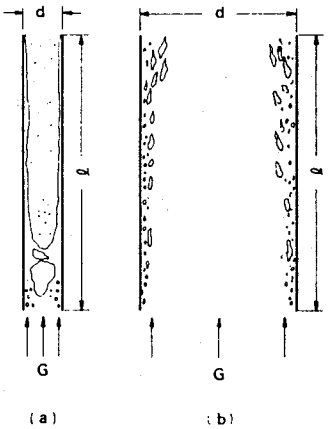

Fig.2 Two Limiting Cases Postulated for Forced Convection Boiling in Uniformly Heated Tubes: (a) Very High l/d or Very Low G, (b) Very Low l/d or Very High G

must be distinguished from the channel flow of Fig.2, but so far as flow patterns are concerned, there are similarities between Fig.1 and Fig.2, suggesting the possibility that the data of CHF accompanying the flow of Fig.2 may be correlated, in an approximate sense at least, by the following function:

$$\frac{q_{co}}{GH_{fg}} = f\left(\frac{\rho_v}{\rho_l}, \frac{\sigma\rho_l}{G^2 l}, \frac{l}{d}\right) \qquad (3)$$

Note: It seems desirable that the analysis of q_{co}/GH_{fg} in Chapter 2 of the author's paper [7] be replaced by the above-mentioned reasoning for the possibility of Eq.(3).

2.2 Correlation equations of CHF in tubes

Basic CHF: q_{co} Analyzing the existing data of 14 different fluids in the range of ρ_v/ρ_l = 0.000270 ∿ 0.413, $\sigma\rho_l/G^2 l = 2.7 \times 10^{-9}$ ∿ 1.0×10^{0}, and l/d = 5.0 ∿ 882, based on the assumption of Eq.(3), the author [7,8,10,11] derived the correlation equations for q_{co}, which are presented in Table 1 being classified into four characteristic regimes (L-, H-, N- and HP-regime). q_{co} is called 'basic critical heat flux' following Macbeth [14].

The numbers of above equations are also tabulated in Table 2 with the naming of sub-regimes (L-1, L-2, H-1 and H-2); and Fig.3 (taken from [10]) shows a comparison with experimental results of various fluids for three nominal values of l/d, while Fig.4 (taken from [11]) represents the boundaries between characteristic regimes for three values of ρ_v/ρ_l. As is noticed in Fig.3, smooth continuous transition between adjoining regimes is neglected for the sake of simplicity.

Table 1. Prediction of q_{co}

L-regime :

$$\frac{q_{co}}{GH_{fg}} = 0.25 \frac{1}{l/d} \qquad (4)$$

$$\frac{q_{co}}{GH_{fg}} = C \left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.043} \frac{1}{l/d} \qquad (5)$$

where $C = 0.25$ for $l/d < 50$, $C = 0.34$ for $l/d > 150$, and

$$C = 0.25 + \frac{(l/d) - 50}{150 - 50}(0.34 - 0.25) \text{ for } l/d = 50 - 150.$$

H- and N-regime :

$$\frac{q_{co}}{GH_{fg}} = 0.10 \left(\frac{\rho_v}{\rho_l}\right)^{0.133} \left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3} \frac{1}{1 + 0.0031\, l/d} \qquad (6)$$

$$\frac{q_{co}}{GH_{fg}} = 0.098 \left(\frac{\rho_v}{\rho_l}\right)^{0.133} \left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.433} \frac{(l/d)^{0.27}}{1 + 0.0031\, l/d} \qquad (7)$$

Boundary dividing Eq.(7) into H- and N-regime is given by

$$\frac{\sigma\rho_l}{G^2 l} = \left(\frac{0.77}{l/d}\right)^{2.70} \qquad (8)$$

HP-regime :

$$\frac{q_{co}}{GH_{fg}} = 0.0384 \left(\frac{\rho_v}{\rho_l}\right)^{0.60} \left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.173} \frac{1}{1 + 0.280\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.233} \frac{l}{d}} \qquad (9)$$

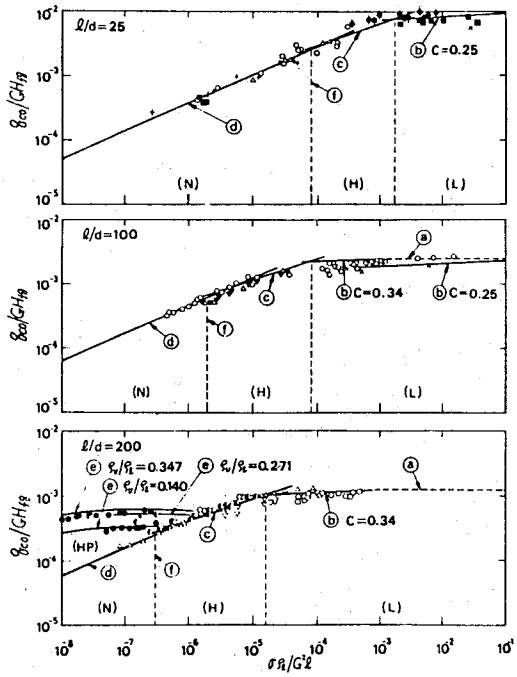

Fig.3 Generalized Representation of q_{co} Data of Various Fluids and Eqs.(4) to (9) for $\rho_v/\rho_l = 0.048$ in H- and N-regime (cf. Table 2)

Effect of inlet subcooling: K Fig.5 illustrates the relationship of q_c vs. ΔH_i observed generally for the existing data of CHF, that is, linear for L-, H- and HP-regime, while non-linear for N-regime in most cases [7]. In connection with this character, Table 3 may be given to compare

Table 2. Characteristic regimes of CHF

Regime	Sub-regime	Basic CHF q_{co}: Prediction of q_{co}	Basic CHF q_{co}: Sign in Fig. 3	Inlet subcooling: Prediction of K	Occurrence conditions (see Figs. 3 and 4)	Flow pattern at CHF condition (see Fig. 6 and [19])	Common term for CHF
L	L-1	Eq.(4)	ⓐ	Eq.(11)	High l/d, or high $\sigma\rho_l/G^2l$	(Annular)	Dryout
	L-2	Eq.(5)	ⓑ	Eq.(12)			
H	H-1	Eq.(6)	ⓒ	Eq.(13)	Intermediate	Spray annular (Wispy annular)	
	H-2	Eq.(7)*	ⓓ	Eq.(14)			
N	—	Eq.(7)*		— **	Low l/d, or low $\sigma\rho_l/G^2l$	Froth or bubbly	DNB
HP	—	Eq.(9)	ⓔ	Eq.(15)	High ρ_v/ρ_l, or high l/d with low $\sigma\rho_l/G^2l$	——	——

* Boundary between H- and N-regime is given by Eq.(8) in Table 1.

** K is unobtainable because of non-linear q_c-ΔH_i relationship in N-regime. In Figs.6 and 10, however, Eqs.(10) and (14) are used tentatively for prediction of q_c in N-regime.

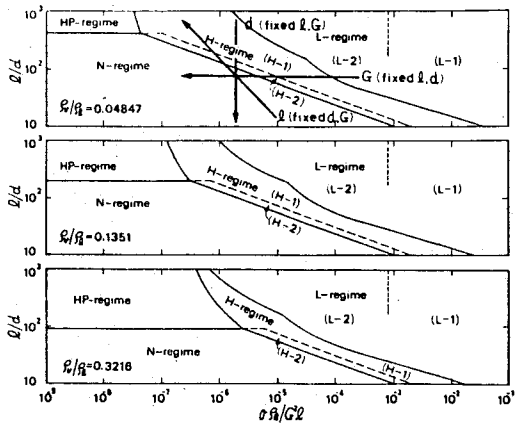

Fig.4 CHF-regime Map (ρ_v/ρ_l = 0.04847, 0.1351 and 0.3216 Correspond to p = 68, 137 and 196 bar Respectively in Case of Water)

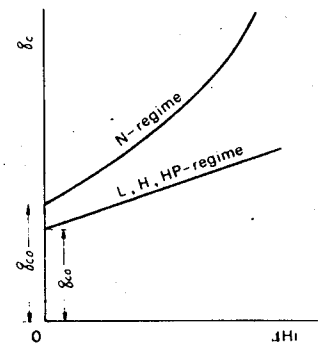

Fig.5 q_c vs. ΔH_i Relationship for Fixed l, d and G

the author's L-, H- and N-regimes with the well-known two regions of Macbeth [14,15] who assumed the linear q_c-ΔH_i relationship for both regions.

From Fig.5, q_c is written as:

$$q_c = q_{co}\left(1 + K\frac{\Delta H_i}{H_{fg}}\right) \tag{10}$$

where the dimensionless coefficient K is independent of ΔH_i for the regimes other than N-regime. In this linear case, if the boiling length concept is assumed to hold exactly near $\Delta H_i = 0$, then K can be derived theoretically from the respective equations of q_{co} in Table 1 (cf. [9],[11]). The results are shown in Eqs.(11) to (15) in Table 4, and the numbers of these equations are also tabulated in Table 2. The prediction of K by Eqs.(11) to (15) shows fairly good agreement with the experimental value of K in L-, H- and HP-regime.

2.3 Relation with flow pattern

For uniformly heated channels, the onset of CHF condition can be assumed to occur first at the

Table 3. Comparison with two regions of Macbeth

G	—— increases ⟶				
Macbeth [14,15]	(Low mass velocity region)		(High mass velocity region)		——
The author	L-1	L-2	H-1	H-2	N-regime
	L-regime		H-regime		
q_c - ΔH_i	linear				non-linear

Table 4. Prediction of K.

From Eq.(4), $K = 1$ (11)

From Eq.(5), $K = \dfrac{1.043}{4C(\sigma\rho_l/G^2l)^{0.043}}$ (12)

From Eq.(6), $K = \dfrac{5}{6}\dfrac{0.0124 + d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{1/3}}$ (13)

From Eq.(7), $K = 0.416\dfrac{(0.0221 + d/l)(d/l)^{0.27}}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{0.433}}$ (14)

From Eq.(9), $K = 1.12\dfrac{1.52(\sigma\rho_l/G^2l)^{0.233} + d/l}{(\rho_v/\rho_l)^{0.60}(\sigma\rho_l/G^2l)^{0.133}}$ (15)

exit end of the channel (cf. Collier [16] for example). Observations of the flow pattern near the exit end of uniformly heated tubes fed with subcooled water were made by Bergles and Suo [17], and Bennett et al. [18]. Examples of the flow pattern map thus obtained are shown in Fig.6 (taken from [19]) in terms of $q/GHfg$ and $\sigma\rho l/G^2 l$. In Fig.6, the thick lines marked L, H and N represent the predictions of q_c made through Eq.(10) and Eqs.(5) to (7) for the prescribed values of $\rho v/\rho l$, l/d and $\Delta Hi/Hfg$, while the short vertical broken line separating H- and N-regime is given by Eq.(8). The results of Fig.6 suggest that H-regime accompanies spray annular flow, while N-regime accompanies froth or bubbly flow. This relation of CHF regimes with flow patterns is recorded in Table 2 along with the common terms of dryout and DNB (departure from nucleate boiling).

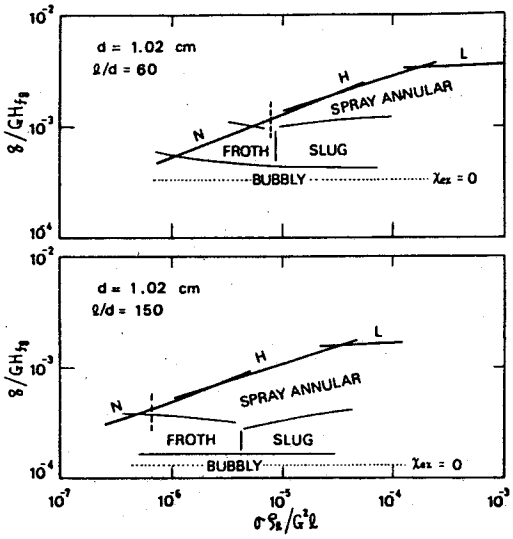

Fig.6 Flow Patterns Near the Exit End of Uniformly Heated Tubes [17] and the Predictions of CHF: Water, p = 69 bar, $\Delta Hi/Hfg$ = 0.0799

3. PARAMETRIC EFFECTS ON CHF

Fixing the fluid substance and the system pressure, there are four principal parameters affecting CHF, namely, l, d, G and ΔHi. According to the author's correlation, l, d and G participate in setting the regime of CHF within the range of ΔHi for which the q_c-ΔHi relationship in Fig.5 holds; and three basic modes for the change of l, d and G are illustrated in Fig.4 by three directed lines respectively.

3.1 Effect of G

For the effect of G on CHF in the case of fixed l, d and ΔHi, typical examples have already been shown in Figs.3 and 6, so that any more explanations will be omitted to save the space.

Next, Fig.7 shows the experimental data of q_c vs. χex obtained by Weatherhead [20] under fixed l and d, changing ΔHi alone at each indicated

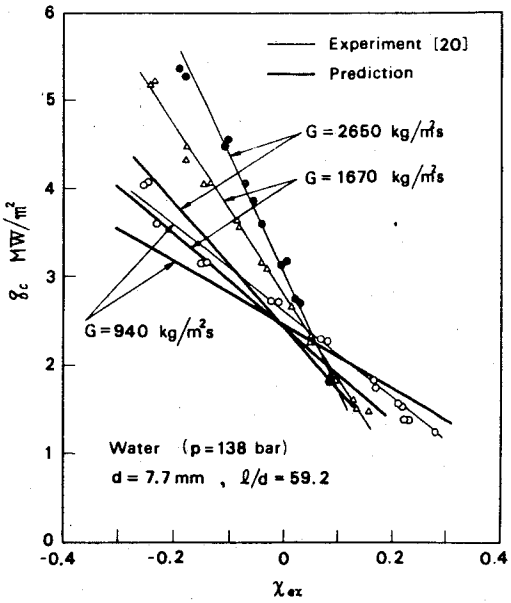

Fig.7 Cross-over of the Mass Velocity Effect on CHF ($\Delta Hi/Hfg \fallingdotseq 0 \sim 0.22$)

value of G. These data are quoted in leading textbooks (Collier [16], Rohsenow [21]) owing to the interesting appearance of cross-over of the effect of G on q_c. Now, the author's CHF regime corresponding to the above experimental condition can be identified by means of Fig.4, and it reveals that the cross-over character in Fig.7 is connected with the experiments made in N-regime. Meantime, the qauntitative evaluation of q_c for $\Delta Hi > 0$ in N-regime is beyond the limits of the author's correlation (see Table 2). The predictions shown in Fig.7 are those made tentatively through Eq.(7) in Table 1, Eq.(10) and the well-known heat balance equation for uniformly heated tubes:

$$\chi_{ex} = 4\frac{q_c}{GH_{fg}}\frac{l}{d} - \frac{\Delta H_i}{H_{fg}} \qquad (16)$$

where K is estimated by Eq.(13) in Table 4 because Eq.(13) is found to be better than Eq.(14) when the predictions are compared with the experimental data of Fig.7.

3.2 Effect of d

Functions of l/d affecting Eqs.(4) to (7) in Table 1 are represented in Fig.8, indicating the trend that in L-regime, q_{co} is subject to the direct effect of d; in H-regime, the effect of d is not noticeable except the range of large l/d; and in N-regime, the effect of d is comparatively small.

Experimental results of Lee [22] shown in Fig.9 are quoted by Lee [23,24] as examples signifying somewhat strange effect of d on q_{co}. The predictions of q_{co} in Fig.9, made by Eqs.(6) and (7) in Table 1, accord with the above experimental results fairly well except minor deviations, suggesting that the comparatively sharp rise of q_{co} with d in the range of small d depends on the

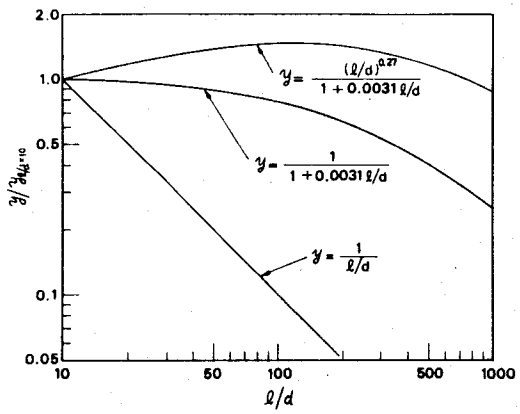

Fig.8 Functions of ℓ/d affecting Eqs.(4) to (7)

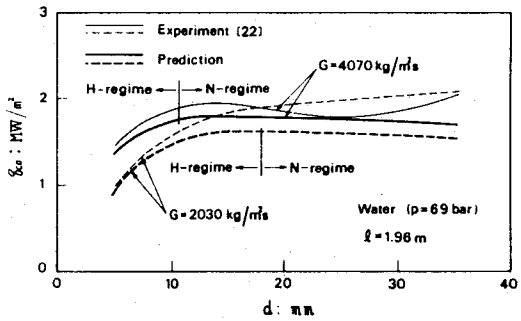

Fig.9 Effect of Diameter on CHF: Water, p = 69 bar, ℓ = 1.96 m

experimental condition of H-regime with large ℓ/d, while the dull effect of d on q_{co} in the range of large d is concerned with N-regime (see Fig.4 for the change of regime due to the variation of d).

3.3 Effect of ℓ

If ℓ varies under fixed d and G, the change of CHF regime takes place in the manner of the slanting directed line in Fig.4. As the system pressure or ρ_v/ρ_l increases, the value of $\sigma\rho_l$ decreases so that the directed line mentioned above moves to the left, making it easy to generate HP-regime in the region of low $\sigma\rho_l/G^2\ell$.

Fig.10 (taken from [25]) is concerned with q_c-χ_{ex} relationship for water flowing through tubes of d = 8 mm with G = 2000 kg/m²s at p = 29.5, 137 and 196 bar. Each short segment in Fig.10 (--- signifies N-regime, —— H-regime, and —·— HP-regime) represents the prediction of q_c-χ_{ex} made by the author's correlation for the prescribed value of ℓ, changing ΔH_i within the ordinary range of 0 to 400 kJ/kg (cf. the note ** of Table 2). Meantime, solid circles plotted in Fig.10 come from the standard table produced by the USSR Academy of Sciences [26] for the q_c-χ_{ex} relation of water in uniformly heated tubes of d = 8 mm. Then, the χ_{ex} indicated by the arrow pointing downward in Fig.10 is called 'the limiting

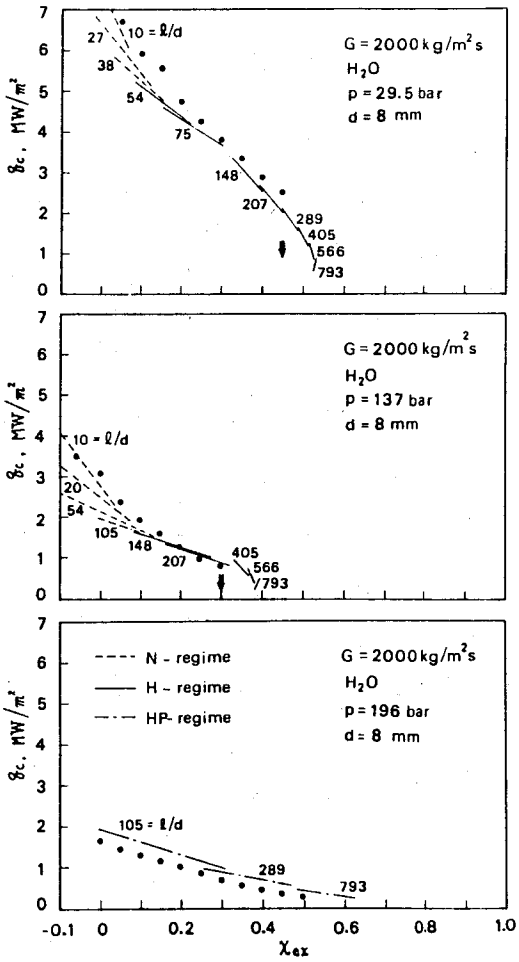

Fig.10 Relationship of q_c vs. χ_{ex} for Water Flowing Through Tubes of d = 8 mm With G = 2000 kg/m²s at p = 29.5, 137 and 196 bar (● and ↓ : USSR Standard Table [26])

quality' in USSR, and it is said that the q_c vs. χ_{ex} relation has a flection point near this position [26]. It may be noted that there is a fairly good agreement between the author's prediction and the USSR standard data (apart from the different interpretations of the form of CHF curve [25]).

Next, Fig.11 represents the experimental data of q_c vs. χ_{ex} obtained by Stevens et al. [27] for R-12 fed to vertical tubes of d = 16.1 mm in the range of $\Delta H_i/H_{fg} \fallingdotseq 0 \sim 0.25$. It is noted that the groups of data for G = 1021, 510 and 203 kg/m²s are characteristic of a sharp bend, a linearity, and a curve being convex to the right respectively. Meanwhile, the prediction of q_c vs. χ_{ex} for the same parametric conditions as above are shown in Fig.12 along with the marks of CHF regimes. In Fig.12, Ⓗ' and Ⓗ-Ⓛ indicate being very near the boundaries between H-1 and H-2, and H and L respectively, while Ⓛ" is fairly close to the state of Ⓗ-Ⓛ. For such conditions as above, the predic-

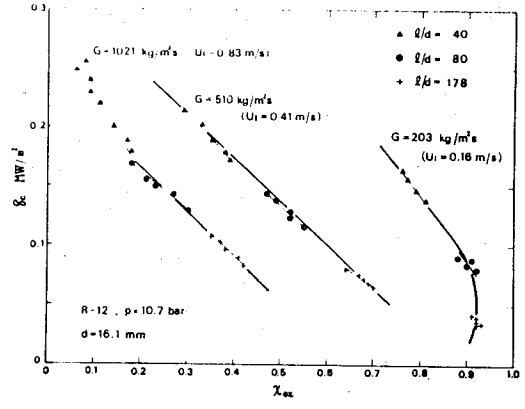

Fig.11 Experimental q_c [27] for R-12 in Tubes of d = 16.1 mm at 10.7 bar ($\Delta H_i/H_{fg} \fallingdotseq 0 \sim 0.25$)

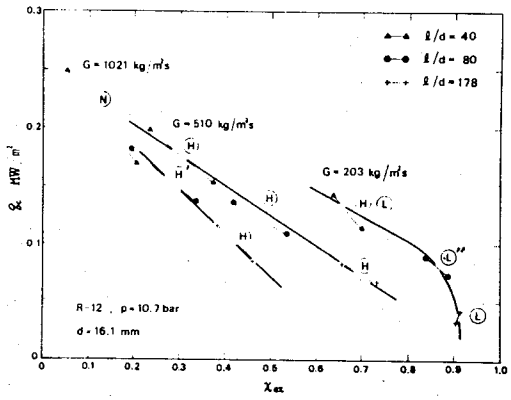

Fig.12 Predicted q_c for R-12 in Tubes of d = 16.1 mm at 10.7 bar ($\Delta H_i/H_{fg}$ = 0 ~ 0.25)

tion of q_c for $\Delta H_i > 0$ is approximately given by averaging the predicted q_c's for two adjoining regimes. Thick lines in Fig.12 accord with those in Fig.11 in respect to the characteristics of curvature. As for the quantitative disagreements noticeable in the case of low G and small ℓ/d, it must be noted that the velocity of liquid at the inlet of tube u_{i} is as low as 0.16 m/s for G = 203 kg/m²s, and 0.41 m/s for G = 510 kg/m²s. It has been mentioned in [7] that when u_{i} is very low and ℓ/d is small for vertical tubes (VL-regime in [7]), gravity is apt to exert its effect on CHF and the author's correlation loses the accuracy.

4. FLUID MODELING OF CHF

The use of model fluids to substitute for water is of practical importance for the lower cost of CHF test, and there exist two approaches to this problem: an approach based on dimensional analysis, and another based on empirical parameter.

4.1 Dimensional analysis approach

The approach, initiated by Barnett [28,29], has been extended skillfully by Ahmad [30], who suggests a scaling law corresponding to the idea that CHF is correlated by

$$\left.\begin{aligned} \frac{q_c}{GH_{fg}} &= f\left(\xi_{CHF}, \frac{\rho_v}{\rho_\ell}, \frac{\ell}{d}, \frac{\Delta H_i}{H_{fg}}\right) \\ \text{where } \xi_{CHF} &= \left(\frac{\sigma\rho_\ell}{G^2 d}\right)\left(\frac{Gd}{\mu_\ell}\right)^{1/2}\left(\frac{\mu_\ell}{\mu_v}\right)^{3/10} \end{aligned}\right\} \quad (17)$$

with empirically derived exponents of 1/2 and 3/10. Eq.(17) is clearly distinguished from the results of other studies [28, 29, 31 etc.] by the fact of being free from the dimensionless groups containing thermal transport properties of fluid.

Eq.(17) signifies that model fluid and water have the same value of q_c/GH_{fg} if the four dimensionless groups ξ_{CHF}, ρ_v/ρ_ℓ, ℓ/d and $\Delta H_i/H_{fg}$ are equal. Then the condition $(\xi_{CHF})_m = (\xi_{CHF})_w$ together with the assumption of equal size gives the mass velocity scaling factor F_G as follows:

$$F_G = \frac{(G)_w}{(G)_m} = \left[\frac{(\sigma\rho_\ell)_w}{(\sigma\rho_\ell)_m}\right]^{2/3}\left[\frac{(\mu_\ell)_m}{(\mu_\ell)_w}\right]^{2/15}\left[\frac{(\mu_v)_m}{(\mu_v)_w}\right]^{1/5} \quad (18)$$

Meantime, on the same principle as above, the author's CHF correlation:

$$\frac{q_c}{GH_{fg}} = f\left(\frac{\sigma\rho_\ell}{G^2\ell}, \frac{\rho_v}{\rho_\ell}, \frac{\ell}{d}, \frac{\Delta H_i}{H_{fg}}\right) \quad (19)$$

gives F_G as follows:

$$F_G = \frac{(G)_w}{(G)m} = \left[\frac{(\sigma\rho_\ell)_w}{(\sigma\rho_\ell)_m}\right]^{1/2} \quad (20)$$

Eq.(18), containing μ_ℓ and μ_v, seems to be of a different nature from Eq.(20). However, in Eq. (18), the effect of μ_ℓ is not large because of the exponent as small as 2/15 (=0.133), and the effect of μ_v is small because the viscosity of saturated vapor does not differ so much between different substances. Therefore, it is not very strange that

Table 5. Comparison of F_G at the state of ρ_ℓ/ρ_v = 20 (taken from [8]).

Model fluid		R-12	R-22	R-113	CO_2
F_G	Empirical	1.40	1.40	1.46	1.13
	Ahmad [30]	1.38	1.38	1.34	1.2
	Eq.(20)	1.28	1.34	1.21	1.19

Eqs.(18) and (20) show almost equal values as shown in Table 5.

4.2 Empirical parameter approach

This approach was initiated by Stevens and Kirby [32], who found that the following relation holds between water and model fluid when ρ_v/ρ_ℓ is equal:

$$\begin{aligned} \chi_{ex} &= f\left\{(G)_w(d)_w^{1/4}\left(\frac{d}{\ell}\right)_w^{0.59}\right\} \\ &= f\left\{F_G\cdot(G)_m(d)_m^{1/4}\left(\frac{d}{\ell}\right)_m^{0.59}\right\} \end{aligned} \quad (21)$$

where the dimensional parameter : $Gd^{1/4}(\ell/d)^{0.59}$ is empirically determined; and Staub [33] extended it to cover the case of $\Delta H_i/H_{fg} > 0$ by replacing ℓ by the boiling length ℓ_b.

Now, according to the author's correlation,

Eq.(21) is presumably the relation concerned with CHF in H- and N-regime, because the following two expressions are obtained for $\Delta H_i/H_{fg} = 0$ from Eqs. (6) and (7) in Table 1 respectively through Eq.(16):

$$x_{ex} = f\left\{\frac{(\rho_\ell/\rho_v)^{0.2}}{(\sigma\rho_\ell)^{1/2}}\, Gd^{1/2}\, \frac{(1 + 0.0031\, \ell/d)^{3/2}}{\ell/d}\right\} \tag{22}$$

and

$$x_{ex} = f\left\{\frac{(\rho_\ell/\rho_v)^{0.15}}{(\sigma\rho_\ell)^{1/2}}\, Gd^{1/2} \times \frac{(1 + 0.0031\, \ell/d)^{1.155}}{(\ell/d)^{0.965}}\right\} \tag{23}$$

Fig.13 shows that either function of ℓ/d on the R.H.S. of Eqs.(22) and (23) may be replaced by $(d/\ell)^{0.59}$ approximately in the practical range of $\ell/d \fallingdotseq 40 \sim 350$. Therefore, Eqs.(22) and (23) may

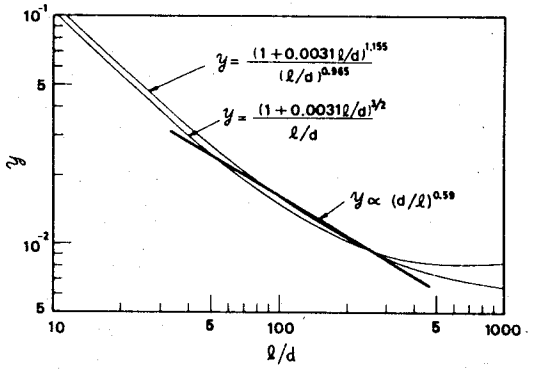

Fig.13 Functions of ℓ/d Related to Eqs.(22) and (23)

be regarded as close to Eq.(21) except a little difference in the exponent of d. It is of interest to refer to Miles et al. [34] who showed that the exponent of d/ℓ in Eq.(21) is diameter dependent.

5. FUTURE TRENDS

The results of this paper suggest that there is room for improvement of the author's CHF correlation equations. Nevertheless, it may be emphasized that it has been clarified that there are great possibilities for the systematic understanding of CHF of forced convection boiling in fundamental boiling systems.

When the systematic understanding of CHF is established for fundamental boiling systems, the current confusions such as deriving disparate conclusions to the same phenomenon may be avoided, and it enables us to make clearly-planned-studies on the details of CHF including the elucidation of mechanims. Thus, in the future, CHF of forced convection boiling will become a fruitful area in the field of heat transfer, aiming the escape from the state of the art to that of the science.

The Weber number $G^2\ell/\sigma\rho_\ell$ composed with ℓ instead of d plays important roles in the author's correlation, but it has never been taken into account in the studies made so far for CHF of forced convection boiling in uniformly heated channels [see Eq.(17) for example]. Studies are needed on the validity or the physical meaning of the use of $G^2\ell/\sigma\rho_\ell$. In addition, the author's correlation will be impoved by taking the effect of viscosity into account, and ξ_{CHF} in Eq.(17) is a suggestive result for this purpose.

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GENERAL FEATURES OF CHF OF FORCED CONVECTION BOILING IN UNIFORMLY HEATED VERTICAL TUBES WITH ZERO INLET SUBCOOLING

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Abstract – In this paper, a graphical method is evolved to give a generalized bird's-eye view of the existing data for critical heat flux (CHF) in vertical tubes with zero inlet subcooling. For this purpose 946 data points obtained from 47 sources are used, including 14 different fluids (water, ammonia, benzene, ethanol, freons, helium I, parahydrogen, monoisopropylbiphenyl, nitrogen, and potassium), tube length from 0.0109 to 8.82 m, tube diameter from 0.00109 to 0.0381 m, and density ratio of vapor/liquid from 0.000270 to 0.413. As a result, a regular nature is elucidated for the CHF under study.

NOMENCLATURE

- C, constant, equation (5);
- d, I.D. of heated tube [m];
- G, mass velocity [$kgm^{-2}s^{-1}$];
- H_{fg}, latent heat of evaporation [Jkg^{-1}];
- ΔH_i, enthalpy of inlet subcooling [Jkg^{-1}];
- K, parameter for the effect of inlet subcooling, equation (1);
- l, length of heated tube [m];
- q_c, critical heat flux [Wm^{-2}];
- q_{co}, q_c for $\Delta H_i = 0$ [Wm^{-2}];
- ΔT_i, inlet subcooling temperature [K].

Greek symbols

- ρ_l, density of liquid [kgm^{-3}];
- ρ_v, density of vapor [kgm^{-3}];
- σ, surface tension [Nm^{-1}];
- χ, quality (χ_{ex}; exit quality corresponding to critical condition; χ_{in}: inlet quality).

1. INTRODUCTION

RECENTLY, Shah [1] and the present author [2–4] attempted developing generalized correlations of experimental data for critical heat flux (CHF) in uniformly heated vertical tubes. However, a bird's-eye view of CHF data cannot be obtained from these studies, because Shah's graphical correlation was made taking quite different forms for two regions of $Y \lesseqgtr 10^5$ where Y is a dimensionless parameter proportioned to $G^{1.8}$, while the author gave the analysis of CHF data separately for four characteristic regimes called L-, H-, N-, and HP-regime*. This is certainly inconvenient for observing the general features of CHF. In the present paper, therefore, a generalized graphical representation of the existing data is evolved on the basis of the author's correlation principle at a little sacrifice of accuracy.

For the author's correlation [2–4], critical heat flux q_c in case of inlet subcooling enthalpy ΔH_i is written as

$$q_c = q_{co}(1 + K\,\Delta H_i/H_{fg}) \tag{1}$$

and q_{co} on the RHS of equation (1) is correlated in the following form:

$$\frac{q_{co}}{GH_{fg}} = f\left(\frac{\rho_v}{\rho_l}, \frac{\sigma\rho_l}{G^2 l}, \frac{1}{d}\right). \tag{2}$$

Then, according to the preceding study [4], a dimensionless parameter K in equation (1) can be evaluated theoretically in many cases only if the function $f(\ )$ on the RHS of equation (2) is known. Therefore, the correlation of q_{co} alone will be dealt with in this paper.

2. COLLECTION OF q_{co} DATA

2.1. *Method of obtaining* q_{co} *data*

Utilizing actual experimental data from a number of sources [5–52], as listed in Table 1, q_{co} data for pulsation-free upflows are obtained, mostly by the following normal methods (i) and (ii), and partly by some exceptional methods (iii)–(v) to extend the range of experimental conditions.

(i) As for the data providing the variation of q_c with ΔH_i for fixed G, if there are enough data points ($q_c - \Delta H_i$), q_{co} can be obtained by the method of extrapolation (or interpolation, occasionally) as $\Delta H_i \to 0$. The same principle also applies when ΔT_i or χ_{in} is used instead of ΔH_i. In the special case of experiment made with ΔH_i which is zero substantially, q_{co} can be obtained immediately from the data.

(ii) As for the data providing the variation of q_c with χ_{ex} for fixed G, the relation of q_c vs ΔH_i can be derived through the well-known heat balance equation for uniformly heated tubes:

* In a rough sense, it seems that Shah's region of $Y < 10^5$ corresponds to Katto's L-regime while $Y > 10^5$ to other three regimes.

Table 1. Summary of the collected data of q_{co}

Source	Fluid	l (cm)	d (cm)	l/d	$(\rho_v/\rho_l) \times 10^2$	$\sigma\rho_l/G^2 l$	No. of Data
Lowdermilk *et al.* [5]	Water	12.9–48.6	0.129–0.243	100–200	0.0623	3.42×10^{-5}–6.82×10^{-4}	7
Aladyev *et al.* [6]	Water	16.0	0.800	20	7.82	3.19×10^{-4}	1
Lee-Obertelli [7]	Water	43.2–366	1.08	40–338	4.81	7.79×10^{-7}–6.60×10^{-6}	5
Matzner [8]	Water	198	0.558–3.75	52–359	4.81	1.45×10^{-6}–1.89×10^{-6}	7
Weatherhead [9]	Water	45.7	0.772	59	13.5	1.26×10^{-6}–1.01×10^{-5}	3
Barnett [10]	Water	178	1.27–3.81	47–140	4.81	3.01×10^{-6}	2
Bergles [11]	Water	1.19–11.4	0.118–0.458	5.0–25	1.24	1.36×10^{-5}–5.22×10^{-4}	12
Thompson–Macbeth [12]	Water	15.2–365	0.304–3.74	26–358	0.0623–13.5	2.14×10^{-7}–9.30×10^{-1}	81
Thompson–Macbeth [13]	Water	8.51–312	0.170–1.00	50–792	0.0623–13.5	7.11×10^{-6}–1.00×10^{0}	139
Matzner *et al.* [14]	Water	244–488	1.02	240–480	4.84	2.90×10^{-8}–2.43×10^{-6}	15
Hewitt *et al.* [15]	Water	121	0.929	131	0.0736–0.122	4.72×10^{-4}–3.16×10^{-3}	3
Lee [16]	Water	63.5–152	1.40–2.82	42–108	6.51–11.2	4.58×10^{-7}–4.67×10^{-5}	38
Bennett *et al.* [17]	Water	144	1.25	290	4.81	4.74×10^{-7}–1.89×10^{-6}	2
Bennett *et al.* [18]	Water	342–416	1.26	271–330	4.81	1.16×10^{-7}–2.38×10^{-5}	36
Barbarin *et al.* [19]	Water	96.0–180	1.20	80–150	0.173	1.33×10^{-4}–1.99×10^{-2}	26
Hewitt [20]	Water	580	0.617	940	4.81	1.33×10^{-7}–2.98×10^{-7}	2
Sterman–Nekrasov [21]	Water	37.0–50.0	0.800	46–62	1.78	1.40×10^{-5}–6.73×10^{-5}	4
Ornatskii *et al.* [22]	Water	8.70–44.0	0.290–0.800	30–150	1.79–7.82	7.80×10^{-4}–1.66×10^{-3}	8
Midler–Shitsman [23]	Water	321	1.00	321	8.05	7.96×10^{-6}	1
Roko *et al.* [24]	Water	360	1.25	288	2.53	5.12×10^{-5}	1
Doroshchuk *et al.* [25, 26]	Water	150	0.800	188	3.19–27.1	1.50×10^{-7}–1.14×10^{-5}	27
Peskov *et al.* [27]	Water	165	0.800	206	36.1	2.73×10^{-8}	1
Becker *et al.* [28]	Water	200–500	1.00	200–500	14.0–34.7	2.78×10^{-9}–6.47×10^{-7}	83
Chojnowski–Wilson [29]	Water	762	3.20	238	24.3	8.43×10^{-8}–3.90×10^{-7}	4
Watson *et al.* [30]	Water	549	3.78	145	27.1	4.68×10^{-8}–1.17×10^{-6}	6

Table 1. – *continued*

Source	Fluid	l (cm)	d (cm)	l/d	$(\rho_v/\rho_l) \times 10^2$	$\sigma\rho_l/G^2l$	No. of Data
Noel [31]	Ammonia*	8.00	0.589	14	5.26	8.20×10^{-7}–3.13×10^{-6}	2
Sterman *et al.* [32]	Benzene	10.0	1.00	10	0.675	1.55×10^{-6}–1.40×10^{-5}	3
Pokhalov *et al.* [33]	Benzene	8.00	0.500	16	0.978–14.3	1.08×10^{-7}–2.65×10^{-5}	10
Tolubinskiy–Matorin [34]	Benzene	6.00	0.400	15	2.37	1.10×10^{-5}	1
Tolubinskiy–Matorin [34]	Ethanol	6.00	0.400	15	1.49	1.17×10^{-5}	1
Giarratano *et al.* [35]	He†	1.09–7.64	0.213	5.1–36	15.1–21.1	2.72×10^{-7}–1.10×10^{-5}	9
Keilin *et al.* [36]	He†	10.0	0.200	50	17.0	6.04×10^{-5}	1
Ogata-Sato [37]	He†	2.80–5.60	0.109	26–51	15.1–41.3	2.90×10^{-6}–5.00×10^{-5}	10
Lewis *et al.* [38]	H_2‡	28.0–40.6	1.40	20–29	6.34–7.10	3.72×10^{-4}–7.74×10^{-3}	21
Sterman *et al.* [32]	MIPD§	10.0	1.00	10	1.03	1.23×10^{-6}–1.10×10^{-5}	3
Lewis *et al.* [38]	N_2	40.6	1.40	29	1.78–1.97	2.16×10^{-3}–2.61×10^{-2}	11
Papell [39]	N_2	30.5	1.28	24	3.00–6.52	1.39×10^{-6}–1.93×10^{-6}	3
Stevens *et al.* [40]	R-12	24.1–357	0.525–1.61	28–337	4.82	2.37×10^{-7}–3.80×10^{-4}	102
Stevens *et al.* [41]	R-12	32.4–259	0.965	33–268	4.82	9.52×10^{-7}–6.10×10^{-5}	11
Barnett-Wood [42]	R-12	43.9–241	0.770–0.965	57–313	2.41–4.81	7.91×10^{-7}–1.00×10^{-4}	31
Groeneveld [43]	R-12	138	0.780	177	4.98	8.47×10^{-8}–3.14×10^{-6}	6
Cumo *et al.* [44]	R-12	200	0.780	256	4.97–9.88	1.96×10^{-6}–1.97×10^{-4}	17
Barnett–Wood [42]	R-21	52.8–331	0.671–1.61	76–318	2.40–4.85	6.17×10^{-7}–9.37×10^{-5}	100
Staub [45]	R-22	152	1.02	150	2.10–4.82	1.99×10^{-6}–3.71×10^{-5}	32
Coffield *et al.* [46]	R-113	76.2	1.02	75	4.89–13.9	1.15×10^{-7}–1.84×10^{-6}	4
Ueda *et al.* [47,48]	R-113	254–882	1.00	254–882	1.47	2.20×10^{-6}–6.17×10^{-5}	9
Dix [49]	R-114	91.4–150	0.998–1.41	65–150	4.85–8.46	1.62×10^{-6}–3.97×10^{-5}	19
Hoffman–Krakoviak [50]	K	37.1–183	0.825–2.21	45–83	0.0820	2.67×10^{-3}–5.11×10^{-3}	6
Bond–Converse [51]	K	87.1–612	1.07–1.94	45–569	0.526–0.953	4.42×10^{-5}–2.89×10^{-3}	16
Akad'yev *et al.* [52]	K	12.0–40.0	0.400–0.600	30–100	0.0270–0.336	1.88×10^{-3}–2.87×10^{-2}	4
							Total 946

* Anhydrous ammonia.
† Helium I.
‡ Para-hydrogen.
§ Monoisopropylbiphenyl.

$$\frac{4q_c}{GH_{fg}}\frac{l}{d}-\frac{\Delta H_i}{H_{fg}}=\chi_{ex}. \tag{3}$$

Therefore, the method (i) is applicable. In this case, however, there are occasionally papers giving q_c only for high degree mixed-inlet-condition ($\Delta H_i < 0$), and the presumption of q_{co} as $\Delta H_i \to 0$ from such data is avoided because of the ambiguity of the flow pattern given at the tube inlet.

(iii) References [18, 24, 35] give the data of sharp wall-temperature rise position for subcooled or saturated inlet condition ($\Delta H_i \geqq 0$). In this case, if the axial length between the above-mentioned temperature rise position and the computed position of $\chi = 0$ is taken as l, the experimental CHF is regarded as q_{co}.

(iv) References [47, 48, 51] give the data of sharp wall-temperature rise position for mixed inlet condition ($\Delta H_i < 0$). In this case, the l determined by means of the method (iii) becomes an imaginary length, but the preceding study [4] has shown that the above-mentionled l can be used to correlate data without serious error so far as L-regime (see Section 3.2) is concerned.

(v) References [13, 45] give the data for independent sets of G, ΔH_i and q_c. Therefore, q_{co} is estimated through equation (1) with semi-theoretical value of K derived in the preceding study [4]. In this case, if the value of $K\Delta H_i/H_{fg}$ is small compared with unity, serious error does not arise for the estimation of q_{co}. To supplement the data in L-regime [13] is used while [45] provides the data of R-22 in H-regime.

2.2. *Note on the data*

The data of [7–10] are those referred to by Collier in his book [59] while the data of [11] are those referred to by Rohsenow in *Handbook of Heat Transfer* [60]. As for [15], the data of thick-walled tube in Fig. 2.9 are used. For [21] the data for $l = 1.57$ m are discarded due to considerable scatter of data, and for [25, 26] actual experimental data shown in graphs are utilized, discarding recommended CHF values which assume non-influence of l and of inlet parameters for $l/d \geq 20$ (N.B. 493 data points from the latter were used in the correlation of Shah [1]). As for [27], only one data at $p = 200$ atm is utilized due to the scarcity of data for fixed G. In [35], the data in case of $\Delta H_i = 0$ alone are used for simplicity. Miscopies found in tabulation of [45] and [49] have been corrected (cf. [2] and [3]).

As was mentioned in [2, 3] as VL-regime, it seems likely that if l/d is small and if the inlet velocity is very low, ordinary pool boiling is apt to prevail. From this point of view, the data of [33] in the range of $\sigma\rho_l/G^2 l > 3 \times 10^{-5}$ (cf. Fig. 10 of [3]) and a part of H_2 data of [38] (cf. Fig. 9 of [2]) have been discarded. For the same reason, among the data of [39], upflow data at the highest velocity (3.5 m/s) alone are adopted for information. The data of Cheng and Ng [53] are not considered, because they are clearly in the regime suffering the effect of pool boiling.

Finally, the data of Borishansky *et al.* [54], Doroshchuk and Lantsman [55], and Smolin [56] cannot be utilized due to the lack of information of l. Eliminations of data other than those mentioned in this Section have not been made in the present study except those cases with clear reasons such as unstable flow, anomalous condition, etc.

3. GRAPHIC REPRESENTATION OF q_{co} DATA

3.1. *Representation of q_{co} data*

The experimental data of q_{co} collected in Section 2 are listed in Table 1 along with the kind of fluid, the range of experimental conditions, and the number of collected data, where physical properties for monoisopropylbiphenyl alone are those estimated by the method of [57, 58]. Then, assuming the possibility of correlating the data in the form of equation (2), 13 graphs are prepared for distinct values of l/d as shown in Fig. 1, and the data of q_{co}/GH_{fg} are plotted against $\sigma\rho_l/G^2 l$ (the division of data by l/d to each graph and the treatment of ρ_v/ρ_l effect will be explained in Section 3.3). Data symbols shown in Table 2 are used in Fig. 1 to discriminate the kind of fluid. When many data gather together to the same place, plotting a part of data is eliminated to avoid the confusion.

3.2. *Representation of the author's correlation equations*

Based on the results of the preceding study [2, 3], the characteristic regimes L, H, N, and HP, the correlation curves (a) to (e), and the boundary (f) between H- and N-regime are shown in Fig. 1 applying the following equations:

Table 2. Symbols used to specify the fluids in Fig. 1

Ref. no.	Fluid	Symbol	Ref. no.	Fluid	Symbol
5–24	Water	○	32	MIPB	-■-
25, 26	Water	◐	38, 39	N_2	■
27	Water	◩	40–44	R-12	△
28–30	Water	●	42	R-21	▽
31	Ammonia	-○-	45	R-22	□
32–34	Benzene	○	46–48	R-113	▲
34	Ethanol	-●-	49	R-114	▽
35–37	He	+	50–52	K	×
38	H_2	●			

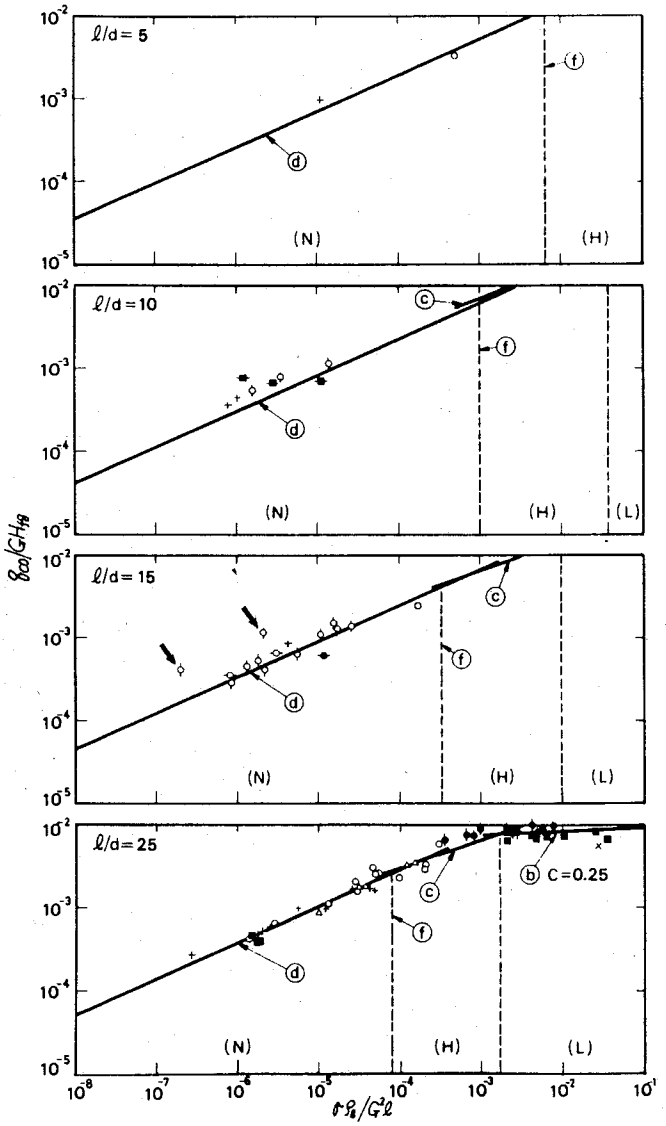

FIG. 1. Generalized graphic representation of q_{co} data. (L): L-regime, (H): H-regime, (N): N-regime, (HP): HP-regime, a: equation (4), b: equation (5), c: equation (6) with $\rho_v/\rho_l = 0.048$, d: equation (7) with $\rho_v/\rho_l = 0.048$, e: equation (8), f: equation (9).

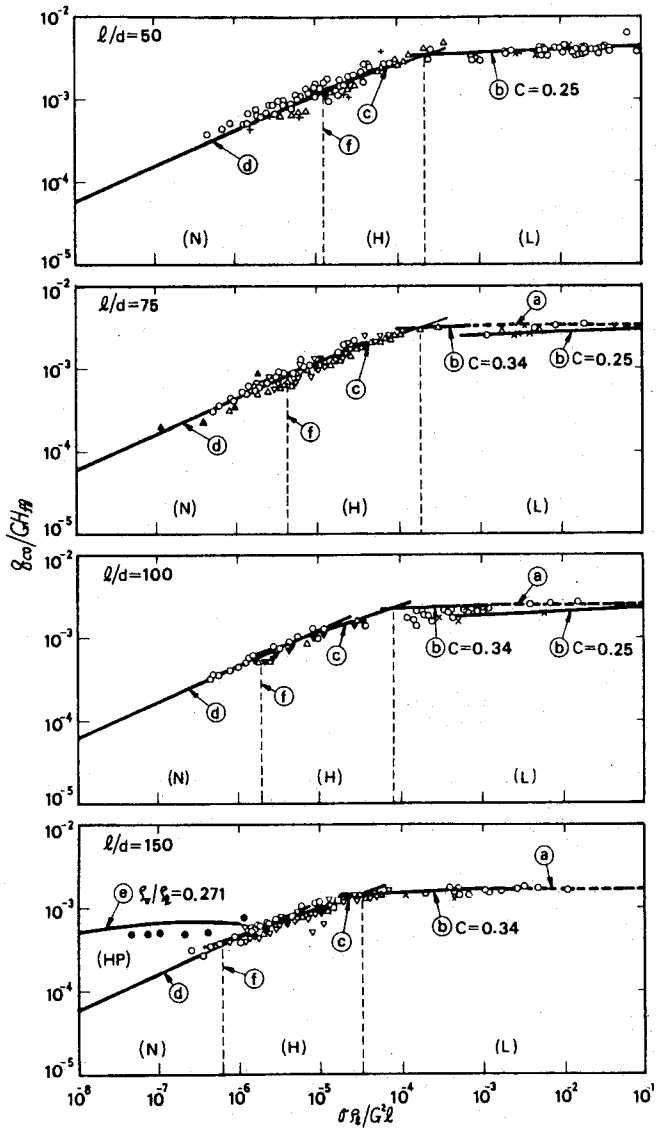

FIG. 1.—continued.

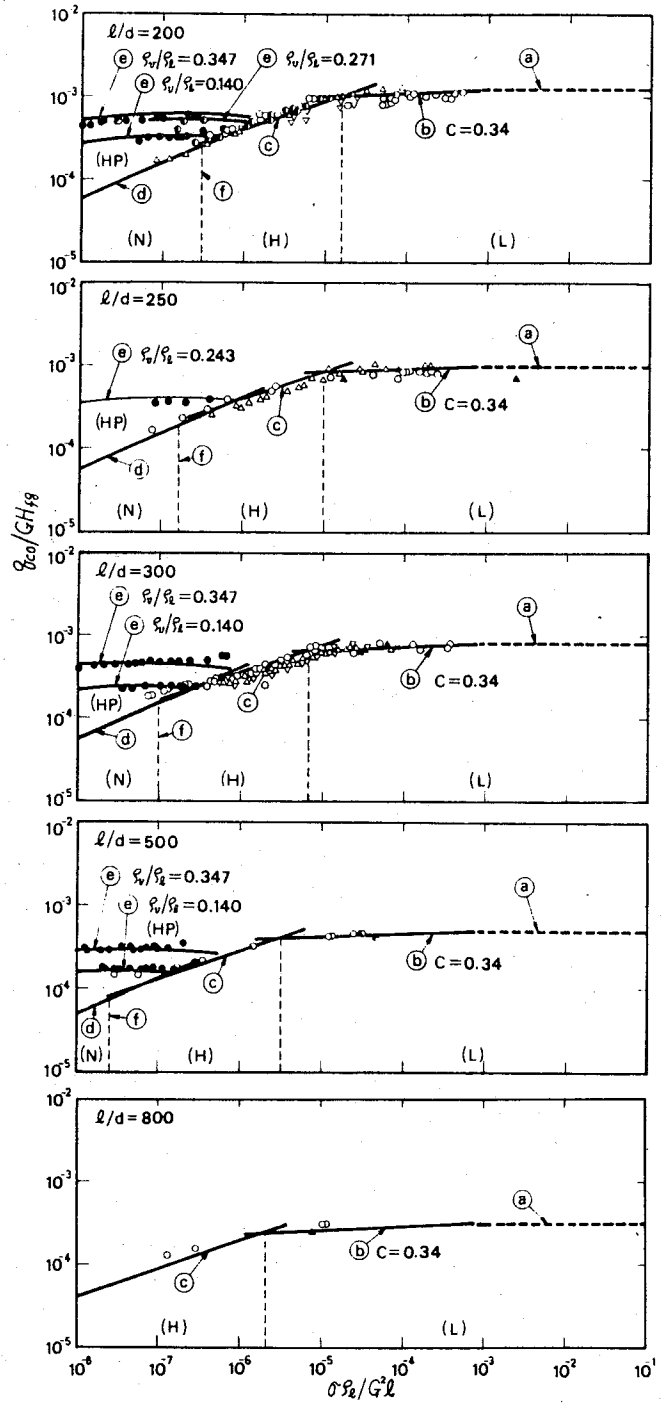

FIG. 1.—continued.

Table 3. Experimental range of l/d and ρ_v/ρ_l in each regime for the data of Fig. 1, excluding the data of Table 4

Nominal $\frac{l}{d}$	HP-regime $\frac{l}{d}$	HP-regime $\frac{\rho_v}{\rho_l} \times 10^2$	H and N-regime $\frac{l}{d}$	H and N-regime $\frac{\rho_v}{\rho_l} \times 10^2$	L-regime $\frac{l}{d}$	L-regime $\frac{\rho_v}{\rho_l} \times 10^2$	Ref. no. *
10	—	—	10.0	1.03	—	—	32
15	—	—	13.6–16.0	0.979–6.42	—	-	31, 33, 34
25	—	—	20.0–33.4	1.79–7.82	20.0–30.0.	0.105–7.10	6, 12, 22, 38, 39, 40, 41, 52
50	—	—	39.2–59.2	1.79–13.5	40.2–60.1	0.0623–13.5	7, 8, 9, 10, 12, 13, 16, 21, 22, 35, 40, 42, 50, 51, 52
75	—	—	67.0–83.5	2.41–11.2	62.5–83.0	0.0820–4.82	7, 8, 12, 16, 19, 21, 40, 41, 42, 46, 49, 50, 52
100	—	—	90.2–112	1.79–11.2	86.8–105	0.0270–2.07	5, 12, 13, 16, 19, 22, 40, 42, 49, 51, 52
150	145	27.1	123–161	1.79–9.28	123–159	0.0736–4.85	7, 8, 10, 12, 13, 15, 19, 22, 30, 40, 41, 42, 45, 49, 51
200	188–206	14.0–36.1	175–214	2.41–13.1	178–224	0.0623–4.82	5, 8, 12, 13, 25, 25, 26, 26, 27, 28, 40, 42, 43, 51
250	238	24.3	229–268	4.84–9.88	249–268	0.404–9.81	12, 13, 14, 29, 29, 41, 44, 47, 48
300	300	14.0–34.7	271–360	2.40–8.05	286–337	0.293–4.86	7, 8, 12, 13, 14, 17, 18, 23, 24, 28, 40, 42, 47, 48, 51
500	500	14.0–34.7	480	4.84	528–569	0.614–2.28	13, 14, 28, 47, 48, 51
800	—	—	940	4.81	792–882	1.47–2.21	13, 20, 47, 48

* Italic numbers indicate the references which give HP-regime data.

L-regime,

(a): $$\frac{q_{co}}{GH_{fg}} = 0.25\frac{1}{l/d}, \quad \text{or} \quad \chi_{ex} = 1 \tag{4}$$

(b): $$\frac{q_{co}}{GH_{fg}} = C\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.043}\frac{1}{l/d} \tag{5}$$

where

$C = 0.25$ or 0.34.

H- and N-regime,

(c): $$\frac{q_{co}}{GH_{fg}} = 0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3} \times \frac{1}{1 + 0.0031\, l/d} \tag{6}$$

where

$\rho_v/\rho_l = 0.048$ for (c).

(d): $$\frac{q_{co}}{GH_{fg}} = 0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.433} \times \frac{(l/d)^{0.27}}{1 + 0.0031\, l/d} \tag{7}$$

where

$\rho_v/\rho_l = 0.048$ for (d).

HP-regime,

(e): $$\frac{q_{co}}{GH_{fg}} = 8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.453} \times \frac{1}{1 + 107\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.54}\frac{l}{d}} \tag{8}$$

Boundary between H- and N-regime,

(f): $$\frac{\sigma\rho_l}{G^2 l} = \left(\frac{0.77}{l/d}\right)^{2.70}. \tag{9}$$

In Fig. 1, the lines (a) and (b) for L-regime are given independently of ρ_v/ρ_l [see equations (4) and (5)]. The intersecting point between (a) and (b) appears at

$$\sigma\rho_l/G^2 l = 7.84 \times 10^{-4} \tag{10}$$

which is determined by eliminating q_{co} from equations (4) and (5) with $C = 0.34$ for $l/d = 75$–800 as seen in Fig. 1. Next, the lines (c) and (d) for H- and N-regime in Fig. 1 show the prediction of equations (6) and (7) in case of $\rho_v/\rho_l = 0.048$ corresponding to saturated steam and water at 68.5 bars for example. It should be noted that l/d does not exert its influence on the lines (c) and (d) so much as on the lines (a) and (b) [cf. equations (4)–(7)]. Finally, the curves (e) for HP-regime of Fig. 1 show the prediction of equation (8) for $\rho_v/\rho_l = 0.140$, 0.234, 0.271, and 0.347.

3.3. *Effects of* ρ_v/ρ_l

The data points plotted in Fig. 1 can be classified according to the regimes of L, H + N, and HP determined in Fig. 1, yielding the result of Table 3 with the range of l/d and ρ_v/ρ_l covered in each data set, on the condition that a small number of data in the regime of H + N with comparative high or low values of ρ_v/ρ_l are omitted from Table 3 to be shown in Table 4.

Now, as for L-regime data in Table 3, it is noted that the value of ρ_v/ρ_l shows a variation as much as 500 times (= 13.5 ÷ 0.0270). However, the data points of CHF plotted in L-regime in Fig. 1 show no mutual discrepancies owing to the difference of ρ_v/ρ_l, and this independence of ρ_v/ρ_l agrees with the character of equations (4) and (5).

Table 4. Data with comparatively high or low values of ρ_v/ρ_l in H and N-regime

Nominal $\frac{l}{d}$	H and N-regime $\frac{l}{d}$	$\frac{\rho_v}{\rho_l} \times 10^2$	No. of data	Ref. no.
5	5.0	0.124	2	[11]
	5.14	15.1	1	[35]
10	7.86–10.9	21.1	2	[35]
	10.0	0.675	3	[32]
15	15.0	0.124	1	[11]
	12.5	0.151	1	[35]
25	25.0	0.124	9	[11]
	21.8–31.2	15.1–21.1	4	[35]
	25.7	15.1–41.3	5	[37]
50	35.9	15.1	1	[35]
	50.0	17.0	1	[36]
	51.4	15.1–41.3	5	[37]
200	188	15.4–20.1	7	[25,26]
			Total 42	

Next, H- and N-regime data in Table 3 change the value of ρ_v/ρ_l in some degree above and below the value of 0.048 (cf. Section 3.2). However, the data points of CHF plotted in H- and N-regime in Fig. 1 appear along by the lines (c) and (d), supporting equations (6) and (7) on the point that the effect of ρ_v/ρ_l on CHF is only in the degree of $(\rho_v/\rho_l)^{0.133}$. It should be noted here that correction of experimental value of q_{co} by $(\rho_v/\rho_l)^{0.133}$ in plotting the data in Fig. 1 has not been done, mainly because of avoiding artificial modification of data as far as possible, and partly because of the difficulty of dealing with the data near the boundary between L- and H-regime with different character as to the effect of ρ_v/ρ_l. However, the correction of multiplying $[0.048/(\rho_v/\rho_l)]^{0.133}$ to q_{co}/GH_{fg} has been made for the small number of data of Table 4, because the difference of ρ_v/ρ_l from 0.048 is comparatively large and the data are sufficiently apart from L-regime.

Finally, HP-regime data in Table 3 are subject to a strong influence of ρ_v/ρ_l as is noticed in Fig. 1, so that the curve (e) representing the prediction of equation (7) is drawn in Fig. 1 for every experimental value of ρ_v/ρ_l on the condition that $\rho_v/\rho_l = 0.361$, which corresponds to only one data of water [27], is omitted for avoiding the confusion of representation.

4. DISCUSSION OF THE RESULT OF FIG. 1

(I) It is of interest to know that a large number of q_{co} data obtained from various independent sources [5–52] indicates such a regular nature as shown in Fig. 1 for a wide range of l/d, suggesting the importance of the dimensionless group $\sigma\rho_l/G^2l$ for this phenomenon, (N.B. the comparative deviation of two benzene data for $l/d = 15$, indicated by thick arrows in Fig. 1, is unaccountable as was mentioned in the preceding study [3]). In connection with this point, it may be of use to add that this type of dimensionless group, which is composed with the length of heated surface measured in the direction of forced convection, can play an important role for correlating CHF in forced flow of a jet over a heated surface [61–64].

(II) It may be said from the result of Fig. 1 that the author's correlation equations (4)–(8) express the general trend of experimental data on the whole.

(III) In the author's preceding study [3], the constant C in equation (5) was tentatively determined as $C = 0.25$ for $5 \times 10^{-4} < \sigma\rho_l/G^2l$ and $C = 0.34$ for $\sigma\rho_l/G^2l < 5 \times 10^{-4}$, and this character is fairly observed in Fig. 1, too. However, a detailed observation of Fig. 1 reveals a better formulation that $C = 0.25$, for $l/d < 50$, $C = 0.34$ for $l/d > 150$, and a transition between $C = 0.25$ and $C = 0.34$ takes place in the range of $l/d \simeq 75$–100.

(IV) Correlation equation (8) for q_{co} in HP-regime agrees with experimental data fairly well as shown in Fig. 13 of [2]. However, there is still an anxious matter found in [4] that the parameter K_{HP} for the effect of inlet subcooling on CHF in HP-regime derived theoretically from equation (8) is slightly higher than the experimental data of K_{HP}. Therefore, it is naturally impossible to derive the ultimate criterion for the onset of HP-regime from the result of Fig. 1 based on equation (8). However, it may not be useless for the present to give the following tentative criterion. If it is assumed that the curve (e) of HP-regime in Fig. 1 can appear only originating itself on the line (c) of H-regime*, the lowest value of ρ_v/ρ_l for permitting the onset of HP-regime occurs when the intersecting point between (e) and (c) [determined by eliminating q_{co}/GH_{fg} from equations (6) and (8)] agrees with the intersecting point between (c) and (d) [determined by eliminating q_{co}/GH_{fg} from equations (6) and (7)] to give

$$\frac{\rho_v}{\rho_l} = \left[\frac{116(l/d)^{-0.134} + 0.976(l/d)^{0.324}}{0.254(l/d)+82}\right]^{1.93}, \qquad (11)$$

and Table 5 shows the computed result of equation (11).

Then, the circumstances for the appearance of HP-regime in Fig. 1 are explained as follows. First, CHF in HP-regime is not observed for $l/d = 25$ and 50 in Fig. 1 in spite of the experimental value of ρ_v/ρ_l as much as 0.413 (see Table 4), but it is natural because the lowest values of ρ_v/ρ_l given in Table 5 for $l/d = 25$ and 50 are much larger than 0.413. On the other hand, HP-regime is observed for $l/d = 150$–500 in Fig. 1, and it is also natural because the values of ρ_v/ρ_l listed in the column of HP-regime in Table 3 are on the whole larger than the respective lowest values of ρ_v/ρ_l in Table 5, though some minor discrepancies are observed.

* Notice that the H–HP boundary defined by equation (19) of the preceding study [2] is kept unchanged.

Table 5. Lowest limit of ρ_v/ρ_l for the appearance of CHF in HP-regime

l/d	5	10	15	25	50	75	100	150	200	250	300	500	800
$\rho_v/\rho_l \times 10^2$	129*	106*	93.8	78.8	59.0	47.8	40.0	29.8	23.2	18.7	15.4	8.31	4.27

* Values of $\rho_v/\rho_l > 1.0$ mean nonexistence of HP-regime.

(V) It is noticed in Fig. 1 that experimental data of freons have a trend to be slightly lower than those of water in H- and N-regime for $l/d \geqq 50$. This trend coincides with the indication given in the preceding paper [3] (see the discussion of mass flux scaling factor). It is also noticed in Fig. 1 that experimental data in H-regime for $l/d \geqq 250$ seem to have slower gradient than the line (c). In addition, it should be mentioned that the data of potassium exist in L-regime alone, while the data for HP-regime are only of water.

5. CONCLUDING REMARKS

Dealing with CHF of forced convection boiling in uniformly heated tubes with zero inlet subcooling, a generalized graphical representation of 946 data points listed in Table 1 has been made giving the result of Fig. 1, and thereby the following quantitative informations are obtained: (a) the behavior of constant C in equation (5) as described in Section 4(III), and (b) the lowest value of ρ_v/ρ_l for the onset of HP-regime as given by equation (11).

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ASPECT GENERAL DU FLUX CRITIQUE EN CONVECTION FORCEE DANS DES TUBES VERTICAUX, CHAUFFES, AVEC UN SOUS-REFROIDISSEMENT NUL A L'ENTREE

Resumé — On développe une méthode graphique qui, en un clin d'oeil, rassemble les données connues pour le flux critique (CHF) dans les tubes verticaux, sans sous-refroidissement à l'entrée. On utilise 946 points expérimentaux provenant de 47 sources et concernant 14 fluides (eau, ammoniac, benzène, éthanol, freons, helium I, parahydrogène, monoisopropylbiphenyl, azote et potassium), une longueur de tube allant de 0,0109 à 8,82 m, un diamètre de tube de 0,00027 à 0,413. On dégage une nature régulière du flux critique.

HAUPTMERKMALE DER KRITISCHEN WÄRMESTROMDICHTE (KWD) BEIM SIEDEN BEI ERZWUNGENER KONVEKTION IN GLEICHFÖRMIG BEHEIZTEN SENKRECHTEN ROHREN BEI EINER EINTRITTSUNTERKÜHLUNG VON NULL

Zusammenfassung – In diesem Aufsatz wird eine grafische Methode entwickelt, um einen allgemeinen Überblick über die vorhandenen Daten für die KWD in senkrechten Rohren bei einer Anfangsunterkühlung von null zu geben. Zu diesem Zweck wurden 946 Versuchsergebnisse aus 47 Quellen verwendet. Diese umfassen 14 verschiedene Flüssigkeiten (Wasser, Ammoniak, Benzol, Äthanol, Freone, Helium I, Parawasserstoff, Monoisopropylbiphenyl, Stickstoff und Kalium), Rohrlängen von 0,0109 bis 8,82 m, Rohrdurchmesser von 0,00109 bis 0,0381 m und Dampf/Flüssigkeit-Dichteverhältnisse von 0,000270 bis 0,413. Als Ergebnis wird ein regulärer Verlauf der untersuchten KWD erhalten.

КРИТИЧЕСКИЕ ТЕПЛОВЫЕ ПОТОКИ ПРИ КИПЕНИИ В УСЛОВИЯХ ВЫНУЖДЕННОЙ КОНВЕКЦИИ В РАВНОМЕРНО НАГРЕВАЕМЫХ ВЕРТИКАЛЬНЫХ ТРУБАХ ПРИ ОТСУТСТВИИ НЕДОГРЕВА НА ВХОДЕ

Аннотация — Предложен графический метод обобщения имеющихся данных по критическому тепловому потоку в вертикальных трубах при отсутствии недогрева на входе. С этой целью использовано 946 экспериментальных точек из 47 источников для 14 различных жидкостей (вода, аммиак, бензол, этанол, фреоны, гелий-I, параводород, моноизопропилбифенил, азот и калий) в трубах длиной от 0.0109 до 8.82 м, диаметром от 0,00109 до 0,0381 м при отношениях плотности пара к плотности жидкости в пределах 0,000270–0,413. Выявлены закономерности для критического теплового потока в исследуемом диапазоне параметров.

CRITICAL HEAT FLUX OF FORCED CONVECTION BOILING IN UNIFORMLY HEATED VERTICAL TUBES (CORRELATION OF CHF IN HP-REGIME AND DETERMINATION OF CHF-REGIME MAP)

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Abstract—Being faced with two important questions arising from the author's previous studies of generalized correlation of critical heat flux (CHF) in tubes, experimental data of CHF obtained in HP-regime (that is, high pressure regime) are re-examined to obtain new equations for correlating CHF in this regime with higher accuracy. Then, the criterion for the onset of CHF in HP-regime can also be revised so as to satisfy all the existing experimental data, and finally a clearly defined map of characteristic regimes of CHF in uniformly heated tubes is determined.

NOMENCLATURE

- C, constant, equation (4);
- d, I.D. of heated tube [m];
- G, mass velocity [$kgm^{-2}s^{-1}$];
- H_{fg}, latent heat of evaporation [Jkg^{-1}];
- ΔH_i, enthalpy of inlet subcooling [Jkg^{-1}];
- K, parameter for the effect of inlet subcooling, equation (1);
- l, length of heated tube [m];
- p, absolute pressure [bar];
- q_c, critical heat flux [Wm^{-2}];
- q_{co}, q_c for $\Delta H_i = 0$ [Wm^{-2}].

Greek symbols

- ρ_l, density of liquid [kgm^{-3}];
- ρ_v, density of vapor [kgm^{-3}];
- σ, surface tension [Nm^{-1}].

1. INTRODUCTION

RECENTLY, the author has made a series of studies [1–4] of generalized equations for correlating experimental data of critical heat flux (CHF) of forced convection boiling in uniformly heated vertical tubes, in which the critical heat flux q_c is expressed as

$$q_c = q_{co}\left(1 + K\frac{\Delta H_i}{H_{fg}}\right) \tag{1}$$

where ΔH_i is the enthalpy of inlet subcooling, and q_{co} (i.e. q_c for $\Delta H_i = 0$) is given in the following generalized form:

$$\frac{q_{co}}{GH_{fg}} = f\left(\frac{\rho_v}{\rho_l}, \frac{\sigma\rho_l}{G^2 l}, \frac{l}{d}\right). \tag{2}$$

In the above correlation, four characteristic regimes are classified, namely L-regime (the regime presumably corresponding to the so-called dryout), N-regime (the regime presumably corresponding to the so-called DNB), H-regime (the intermediate regime between L- and N-regime), and HP-regime (the regime which appears in place of N-regime at high pressure). Then, in all the regimes except N-regime, the linear q_c–ΔH_i relationship such as mentioned in Collier's book [5] holds, so that the dimensionless parameter K on the RHS of equation (1) is determined independently of $\Delta H_i/H_{fg}$.

As for the above-mentioned studies, however, it must be remembered that two important questions remain unsolved regarding HP-regime. First, according to [3], predictions of the parameter K derived theoretically from equation (2) show a small but regular deviation from experimental data in HP-regime, suggesting that the generalized correlation equation of q_{co} given by the author for HP-regime is somewhat defective. Second, according to [4], the criterion for the onset of HP-regime given by the author tentatively agrees with the trend of experimental data, but some quantitative discrepancies are clearly observed. In this paper, therefore, a study is made to solve these questions of HP-regime, and at the same time, to give a CHF-regime map free of such ambiguities as seen in the author's previous studies.

2. SUMMARY OF CORRELATION EQUATIONS OF CHF PROPOSED SO FAR

Correlation equations of CHF proposed through the author's previous studies [1–4] are summarized as in the following.

2.1. *Generalized correlation equations of* q_{co}

Correlation of q_{co} based on equation (2) are illustrated in Fig. 1, where two typical cases of $l/d = 50$ and 200 are shown for $\rho_v/\rho_l = 0.01787$ and 0.3216 (which correspond to $p = 29.5$ and 196 bar respectively in case

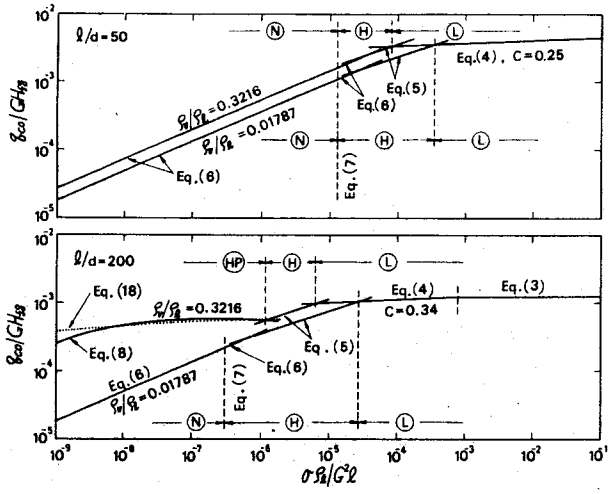

FIG. 1. Examples of the prediction by the author's generalized correlation equations of q_{co}. (L): L-regime, (H): H-regime, (N): N-regime and (HP): HP-regime.

of water), and the equations noted in Fig. 1 are as follows.

L-regime:

$$\frac{q_{co}}{GH_{fg}} = \frac{0.25}{l/d} \tag{3}$$

and

$$\frac{q_{co}}{GH_{fg}} = C\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.043}\frac{1}{l/d} \tag{4}$$

where $C = 0.25$ for $l/d < 50$, $C = 0.34$ for $l/d > 150$, and C changes with l/d in the range of $l/d = 50$–150.

H- and N-regime:

$$\frac{q_{co}}{GH_{fg}} = 0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3}\frac{1}{1+0.0031\,l/d} \tag{5}$$

and

$$\frac{q_{co}}{GH_{fg}} = 0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.433} \times \frac{(l/d)^{0.27}}{1+0.0031\,l/d}, \tag{6}$$

and the boundary between H- and N-regime is given by

$$\frac{\sigma\rho_l}{G^2 l} = \left(\frac{0.77}{l/d}\right)^{2.70}, \quad \text{or} \quad \frac{l}{d} = \frac{0.77}{(\sigma\rho_l/G^2 l)^{0.37}}. \tag{7}$$

HP-regime:

$$\frac{q_{co}}{GH_{fg}} = 8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.453} \times \frac{1}{1+107(\sigma\rho_l/G^2 l)^{0.54}\,l/d}. \tag{8}$$

2.2. *Predictions of K*

Predictions of K which are derived theoretically from the generalized correlation equations of q_{co} for L-, H- and HP-regime are as follows [3].

K for L-regime: K_L

From equation (4),

$$K_L = \frac{1.043}{4C(\sigma\rho_l/G^2 l)^{0.043}} \tag{9}$$

K for H-regime: K_H

From equation (5),

$$K_H = \frac{5}{6}\frac{0.0124 + d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{1/3}} \tag{10}$$

and from equation (6),

$$K_H = 0.416\frac{(0.0221 + d/l)(d/l)^{0.27}}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{0.433}} \tag{11}$$

K for HP-regime: K_{HP}

From equation (8),

$$K_{HP} = 0.0138\frac{215(\sigma\rho_l/G^2 l)^{0.54} + d/l}{(\rho_v/\rho_l)^{0.65}(\sigma\rho_l/G^2 l)^{0.453}}. \tag{12}$$

3. DETAILS OF THE QUESTIONS RELATED TO HP-REGIME

3.1. *The first question*

Figure 2 is the comparison of K_{HP} predicted by equation (12) with the experimental data of K_{HP} collected in Fig. 18 of [1], showing a regular deviation between the predicted and experimental K_{HP}. In the paper [3], however, owing to the apparent high accuracy of equation (8) as compared with experimen-

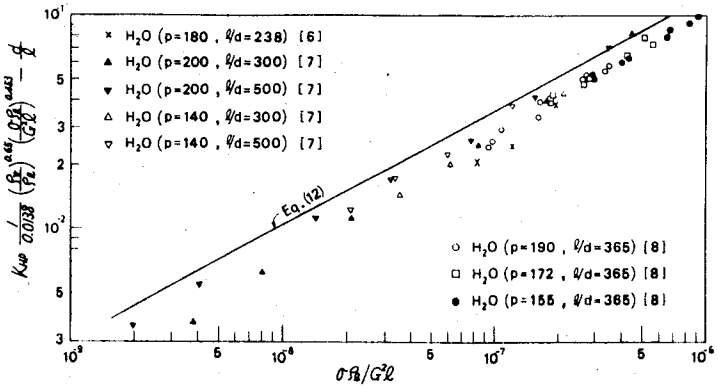

FIG. 2. Comparison between prediction of equation (12) and experimental data for K_{HP}.

tal data of q_{co} (see Fig. 12 of [1]), the cause of the above-mentioned deviation of K_{HP} was regarded as unknown.

3.2. *The second question*

As shown in [3], the boundary between L- and H-regime can be determined by eliminating q_{co} from equations (4) and (5) as follows:

$$\frac{l}{d}=\frac{1}{(0.10/C)(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{0.29}-0.0031}. \quad (13)$$

Next, as shown in [1], q_{co} is eliminated from equations (5) and (6) in H-regime to give the boundary between the realms of these two equations as:

$$\frac{l}{d}=\frac{1.077}{(\sigma\rho_l/G^2l)^{0.37}}. \quad (14)$$

In addition, as shown in [1], the boundary between H- and N-regime is given by equation (7), and the boundary between H- and HP-regime is determined by eliminating q_{co} from equations (5) and (8) as follows:

$$\frac{l}{d}=\frac{82(\rho_v/\rho_l)^{0.517}-(\sigma\rho_l/G^2l)^{-0.12}}{107(\sigma\rho_l/G^2l)^{0.42}-0.254(\rho_v/\rho_l)^{0.517}}. \quad (15)$$

Then, Fig. 3 shows two different examples of CHF-regime map made with equations (7), (13) to (15) for the case of $\rho_v/\rho_l = 0.3216$, where C on the RHS of equation (13) is assumed to change from 0.25 to 0.34 linearly with l/d in the range of $l/d = 50$ to 150.* First, in case of Fig. 3(A), a horizontal line passing through the intersecting point P' of equations (15) and (7) is assumed to separate HP- and N-regime, and this is the idea adopted in [1, 2]. In this case, therefore, the relation between ρ_v/ρ_l and l/d at the point P' is obtained by eliminating $\sigma\rho_l/G^2l$ from equations (15) and (7) as follows:

$$\frac{\rho_v}{\rho_l}=\left\{\frac{79.5(l/d)^{-0.134}+1.08(l/d)^{0.324}}{0.254(l/d)+82}\right\}^{1.93}. \quad (16)$$

Equations (16) gives the lowest limit of ρ_v/ρ_l capable of generating HP-regime for a given l/d, that is, the criterion for the onset of HP-regime.

Now, if experimental conditions, under which the occurrence of HP-regime is clearly observed, are collected from the existing data of CHF, it yields the result of Table 1 with an exceptional condition that l/d for the data of Campolunghi *et al.* [12] alone is an approximate value because of the uncertainty of tube length l corresponding to the condition of $\Delta H_i = 0$. Then, all the combinations of ρ_v/ρ_l and l/d listed in Table 1 are compared with equation (16) in Fig. 4, showing that equation (16) can predict the onset of HP-regime so far as the above data are concerned. However, equation (16) conflicts with two experimental conditions of helium I of Ogata and Sato [13, 14] shown in Fig. 4, which have been confirmed in the preceding study [4] to belong to not HP-regime but N-regime.

In Fig. 3(B), therefore, a horizontal line passing through the intersecting point P'' of equations (15) and (14) is assumed to separate HP- and N-regime, although use of equation (14) for this purpose seems to be rather unnatural from the physical point of view. The criterion for the onset of HP-regime obtained in this way is nothing but the criterion employed tentatively in the preceding study [4], and the relation between ρ_v/ρ_l and l/d at the point P'' is obtained by eliminating $\sigma\rho_l/G_2l$ from equations (15) and (14) as follows:

$$\frac{\rho_v}{\rho_l}=\left\{\frac{116(l/d)^{-0.134}+0.976(l/d)^{0.324}}{0.254(l/d)+82}\right\}^{1.93}. \quad (17)$$

* $C=0.25+\dfrac{(l/d)-50}{150-50}(0.34-0.25)$.

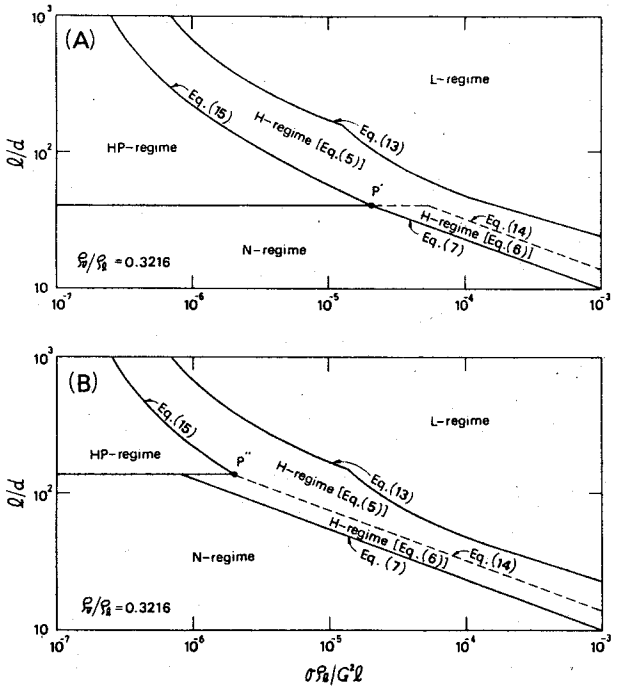

FIG. 3. Determination of the boundary between HP- and N-regime.

However, it is noted in Fig. 4 that the criterion of equation (17) for the onset of HP-regime is satisfactory for the data of helium I, but there are some discrepancies between equation (17) and the HP-regime data.

4. RE-EXAMINATION OF EXPERIMENTAL DATA OF CHF IN HP-REGIME

4.1. *New correlation of q_{co} in HP-regime*

It has been mentioned in Section 3.1 that equation (8) is considerably accurate for correlating q_{co} data. However, two questions explained in the preceding chapter seem to co-operate to suggest weaknesses of equation (8). Therefore, a careful rearrangement of equation (8) is attempted so as to improve the prediction accuracy compared with the existing data of q_{co} in HP-regime, and as the result, the following new correlation equation is obtained

Table 1. Experimental conditions generating CHF in HP-regime

Source	Fluid	Experimental ranges d (mm)	l/d	ρ_v/ρ_l	$\sigma\rho_l/G^2l$	No. of data	Symbol
Thomson–Macbeth [8]	Water	1.9	365	0.170	2.69×10^{-7}–9.15×10^{-7}	9	●
Thomson–Macbeth [8]	Water	1.9	365	0.219	1.86×10^{-7}–5.67×10^{-7}	9	□
Thomson–Macbeth [8]	Water	1.9	365	0.289	9.17×10^{-8}–3.49×10^{-7}	9	○
Peskov *et al.* [9]	Water	8.0	206	0.361	2.73×10^{-8}	1	◩
Becker *et al.* [7]	Water	10.0	200	0.140	4.83×10^{-8}–2.07×10^{-7}	8	-○-
Becker *et al.* [7]	Water	10.0	200	0.347	6.49×10^{-9}–2.40×10^{-7}	14	-●-
Becker *et al.* [7]	Water	10.0	300	0.140	3.40×10^{-8}–2.03×10^{-7}	7	△
Becker *et al.* [7]	Water	10.0	300	0.347	6.41×10^{-9}–2.61×10^{-7}	14	▲
Becker *et al.* [7]	Water	10.0	500	0.140	2.13×10^{-8}–1.28×10^{-7}	10	▽
Becker *et al.* [7]	Water	10.0	500	0.347	2.78×10^{-9}–1.32×10^{-7}	20	▼
Chojnowski–Wilson [6]	Water	32.0	238	0.243	8.43×10^{-8}–1.97×10^{-7}	3	×
Watson–Lee [10]	Water	37.8	145	0.271	4.67×10^{-8}–4.21×10^{-7}	5	■
Doroschuk *et al.* [11]	Water	8.0	188	0.201	1.50×10^{-7}	1	◐
Doroschuk *et al.* [11]	Water	8.0	188	0.271	1.76×10^{-7}–3.14×10^{-7}	2	-◐-
Campolunghi *et al.* [12]	Water	12.0	937*	0.0902–0.176			+

* An approximate value.

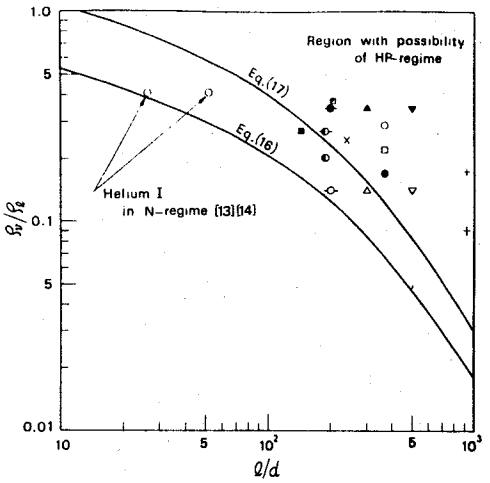

FIG. 4. Criterion for the onset of HP-regime compared with experimental data listed in Table 1.

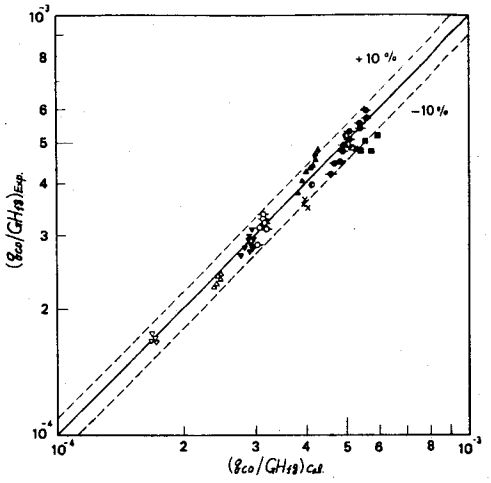

FIG. 5. Comparison of prediction of equation (8) with experimental data in HP-regime (as for the sources of data, see Table 1 through data symbols).

$$\frac{q_{co}}{GH_{fg}} = 0.0384\left(\frac{\rho_v}{\rho_l}\right)^{0.60}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.173} \times \frac{1}{1+0.280(\sigma\rho_l/G^2 l)^{0.233}\, l/d}. \quad (18)$$

According to a comparison between equations (18) and (8) shown in the graph of $l/d = 200$ in Fig. 1, it seems in appearance as if there are no significant differences between the two equations. However, if RMS error of each equation is examined for the experimental data of q_{co} used in the study [1] to derive equation (8), it is 5.58% for equation (18) while 8.0% for equation (8), showing that equation (18) is more accurate than equation (8). Besides, if Fig. 5, which shows the comparison between the prediction of equation (18) and the experimental data, is compared with the similar result of equation (8) given in Fig. 13 of [1], it is readily found that equation (18) is superior to equation (8) in the capability of predicting q_{co} nearer the mean value of q_{co} for almost all independent series of experiment.

4.2. *New prediction equation of K for HP-regime*

If equation (18) is adopted instead of equation (8), new theoretical prediction of K in HP-regime is given by the following equation instead of equation (12) (cf. [3] for the derivation process):

$$K_{HP} = 1.12\frac{1.52(\sigma\rho_l/G^2 l)^{0.233} + d/l}{(\rho_v/\rho_l)^{0.60}(\sigma\rho_l/G^2 l)^{0.173}}. \quad (19)$$

Then, in Fig. 6, K_{HP} predicted by equation (19) is compared with the same experimental data of K_{HP} as used in Fig. 2, showing a marked improvement of accuracy in predicting K_{HP} as compared with Fig. 2.

4.3. *Criterion for the onset of HP-regime*

Eliminating q_{co} from equations (5) and (18), it yields the following equation for the boundary between H- and HP-regime:

$$\frac{l}{d} = \frac{0.384(\rho_v/\rho_l)^{0.467} - (\sigma\rho_l/G^2 l)^{0.160}}{0.280(\sigma\rho_l/G^2 l)^{0.393} - 0.00119(\rho_v/\rho_l)^{0.467}}. \quad (20)$$

Then, if equation (20) is used instead of equation (15), Fig. 3(A) is replaced by Fig. 7, where a horizontal line passing through the intersecting point P of equations (20) and (7) separates HP- and N-regime, and the relation between ρ_v/ρ_l and l/d at the point P is obtained by eliminating $\sigma\rho_l/G^2 l$ from equations (20) and (7) as follows:

$$\frac{\rho_v}{\rho_l} = \left\{\frac{89.3(l/d)^{-0.432} + 21.2(l/d)^{-0.062}}{0.119(l/d)+38.4}\right\}^{2.14}. \quad (21)$$

In Fig. 8, equation (21) is compared with the same experimental data as those in Fig. 4, leading to the conclusion that equation (21) can be a criterion for the onset of HP-regime satisfying all the experimental data shown in Fig. 8.

Lines of constant $\sigma\rho_l/G^2 l$ drawn in the region of generating HP-regime in Fig. 8 are given by equation (20), indicating the highest limit of $\sigma\rho_l/G^2 l$ which enables the appearance of HP-regime at a given point $(\rho_v/\rho_l,\ l/d)$ in Fig. 8. Among the experimental data listed in Table 1, data of Thomson and Macbeth [8] for $l/d = 365$ and $\rho_v/\rho_l = 0.170$ (denoted by ● in Fig. 8) alone are in the experimental range of $\sigma\rho_l/G^2 l$ which is somewhat higher than the limiting value indicated in Fig. 8. However, if the above data are examined carefully, it reveals that there is the possibility of

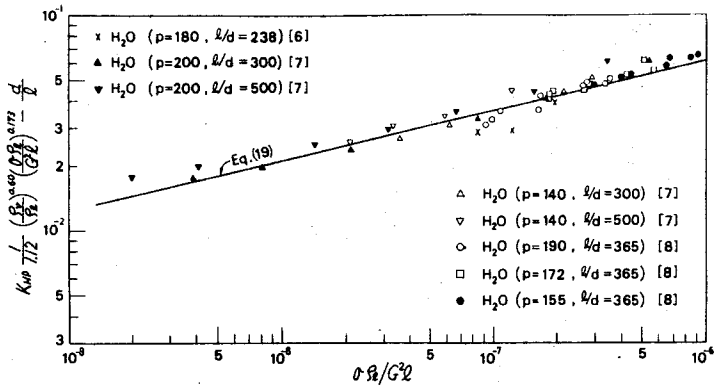

FIG. 6. Comparison between prediction of equation (19) and experimental data for K_{HP}.

retaining the transition character between H- and HP-regime, accordingly the slight deviation such as mentioned above may not be regarded as a serious matter.

Supplementary note. Two data of helium I shown in Fig. 8 are those obtained under the following conditions of $\sigma\rho_l/G^2l$ respectively: $\sigma\rho_l/G^2l = 5.70 \times 10^{-6}$ for the data of $l/d = 25.7$, and $\sigma\rho_l/G^2l = 2.90 \times 10^{-6}$ for that of $l/d = 51.4$.

5. CHF-REGIME MAP

Experiments of CHF made so far in HP-regime are not necessarily sufficient in number as is noticed in Fig. 8. In addition, the existing experimental data for HP-regime are limited to only one fluid, that is water (cf. Table 1). Therefore, further experimental studies of wide range may be necessary in order to give the final approval on the criterion of Fig. 8 for the onset of HP-regime.

However, the criterion mentioned above is regarded as most reliable at the present stage at least. In other words, the means represented by Fig. 7 is most suitable at present for making CHF-regime map. Accordingly, employing the method of Fig. 7, the CHF-regime map is made for each condition of $\rho_v/\rho_l = 0.01787, 0.04847, 0.1351$, and 0.3216 (which corresponds to the pressure $p = 29.5, 69, 137$ and 196 bar in case of water) to give the result of Fig. 9. In every regime illustrated in Fig. 9, the correlation equation of q_{co} that is applicable in the regime is shown together with the prediction equation of K except N-regime, for which the prediction equation of K has not been determined due to non-linear q_c–ΔH_i relationship [1–3]. In addition, a vertical broken line indicated on the upper right side of each diagram of Fig. 9 represents the position of $\sigma\rho_l/G^2l = 7.84 \times 10^{-4}$, which corresponds to the intersecting point of equations (3) and (4) such as shown in the graph of $l/d = 200$ in Fig. 1.

In sum, CHF-regime map of Fig. 9 is a revision of the CHF-regime map proposed in the author's first report (that is Fig. 15 of [1]), taking into account the results obtained in the subsequent studies.

6. CONCLUSIONS

(i) Regarding generalized correlation equations of CHF of forced convection boiling in uniformly heated vertical tubes, the results obtained in the author's

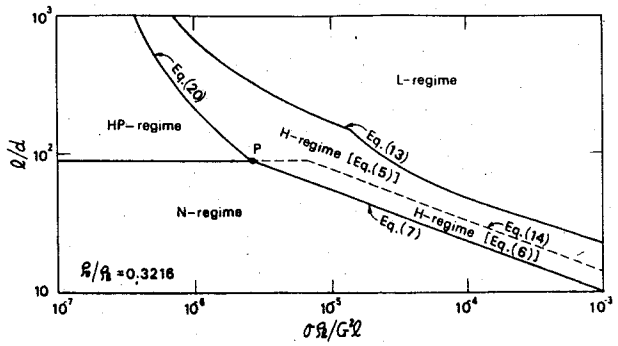

FIG. 7. New method for determining the boundary between HP- and N-regime.

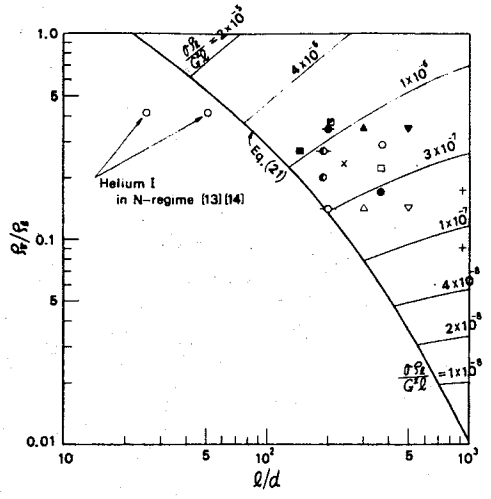

FIG. 8. New criterion for the onset of HP-regime compared with experimental data listed in Table 1.

previous papers [1–4] are summarized in equations (3)–(12) of this paper.

(ii) Through re-examination of existing experimental data of CHF in HP-regime, it is found that equation (8) for the correlation of q_{co} should be replaced by new equation (18), and similarly, equation (12) for the parameter K_{HP} by new equation (19), equation (15) for the boundary between H- and HP-regime by new equation (20), and equation (16) or (17) for the onset of HP-regime by new equation (21). Comparisons of these new equations with experimental data are shown in Figs. 5, 6 and 8.

(iii) A CHF-regime map for forced convection boiling in uniformly heated tubes is determined as shown in Fig. 9, taking into account all the informations obtained so far in the author's studies.

Acknowledgements—The Ministry of Education, Science and Culture is acknowledged for the financial support to this study under Grant in Aid of Special Project Research No. 411002 (1979).

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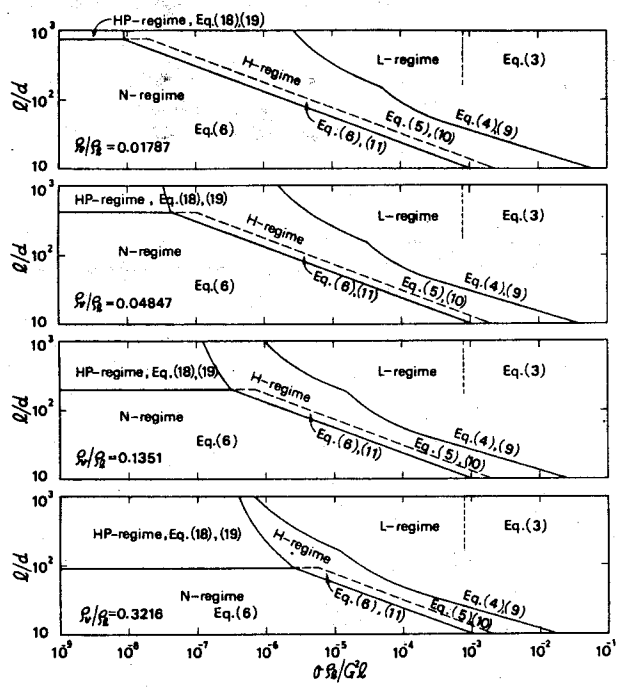

FIG. 9. CHF-regime map with the indication of correlation equations of CHF in each regime (see Fig. 7 for the determination of boundaries of each regime).

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FLUX DE CHALEUR CRITIQUE EN EBULLITION AVEC CONVECTION FORCEE DANS DES TUBES VERTICAUX UNIFORMEMENT CHAUFFES (FORMULATION DU CHF EN REGIME HP ET DETERMINATION DU GRAPHE DES REGIMES CHF)

Résumé—Les précédentes études de l'auteur sur la formulation générale du flux de chaleur critique (CHF) dans les tubes sont prolongées par un nouvel examen des données expérimentales du CHF en régime de haute pression (HP) pour obtenir de nouvelles équations plus précises. D'autre part le critère d'apparition du CHF dans ce régime HP est révisé de façon à satisfaire toutes les données expérimentales connues et ainsi un graphe mieux défini des régimes caractéristiques de CHF dans les tubes chauffés est déterminé.

KRITISCHE WÄRMESTROMDICHTE (KWD) BEIM STRÖMUNGSSIEDEN IN GLEICHMÄSSIG BEHEIZTEN SENKRECHTEN ROHREN (KORRELATION DER KWD IM HOCHDRUCKGEBIET UND AUFSTELLUNG EINER KWD-BEREICHSKARTE)

Zusammenfassung — Wegen der Konfrontation mit zwei wichtigen Fragen, die von den früheren Untersuchungen des Autors über die verallgemeinerte Korrelation der KWD in Rohren herrühren, wurden die experimentellen Daten der KWD im Hochdruckbereich erneut untersucht, um neue Gleichungen mit größerer Genauigkeit für die KWD in diesem Gebiet zu erhalten. Das Kriterium für den Beginn der KWD im Hochdruckgebiet konnte ebenfalls revidiert werden, so daß Übereinstimmung mit allen existierenden experimentellen Daten erreicht wurde. Schließlich wurde eine Karte mit eindeutig bestimmten charakteristischen Bereichen der KWD in gleichmäßig beheizten Rohren aufgestellt.

КРИТИЧЕСКИЙ ТЕПЛОВОЙ ПОТОК ПРИ КИПЕНИИ В УСЛОВИЯХ ВЫНУЖДЕННОЙ КОНВЕКЦИИ В РАВНОМЕРНО НАГРЕВАЕМЫХ ВЕРТИКАЛЬНЫХ ТРУБАХ (КОРРЕЛЯЦИЯ КРИТИЧЕСКОГО ТЕПЛОВОГО ПОТОКА В РЕЖИМЕ ВЫСОКОГО ДАВЛЕНИЯ И СОСТАВЛЕНИЕ КАРТЫ РЕЖИМОВ КРИТИЧЕСКОГО ТЕПЛОВОГО ПОТОКА)

Аннотация — При использовании предложенного ранее автором соотношения для критического теплового потока (КТП) возникло два важных вопроса. Поэтому были пересмотрены экспериментальные данные по КТП в режиме высокого давления с целью получения более точных соотношений для КТП в данном режиме. Уточнен также критерий, определяющий наступление КТП в режиме высокого давления, который удовлетворяет всем имеющимся экспериментальным данным, и, наконец, составлена карта режимов КТП в разномерно нагреваемых трубах.

ON THE HEAT-FLUX/EXIT-QUALITY TYPE CORRELATION OF CHF OF FORCED CONVECTION BOILING IN UNIFORMLY HEATED VERTICAL TUBES

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Abstract—In this paper, it is attempted to use generalized correlation equations of CHF proposed recently by the author for predicting the well-known relation which appears between critical heat flux q_c and exit quality χ_{ex} when tube length l and inlet subcooling enthalpy ΔH_i are varied for fixed pressure p, mass velocity G and tube diameter d. Numerical calculation is made for water under the conditions of $p = 29.5$–196 bar, $G = 750$–$4000\,\mathrm{kg/m^2\,s}$, $d = 0.008$ m, $l/d = 10$–793, and $\Delta H_i = 0$–$400\,\mathrm{kJ/kg}$. Then, the result is compared with the standard table of $q_c - \chi_{ex}$ published recently by Heat Mass Transfer Section, Academy of Sciences, U.S.S.R., for water in uniformly heated tubes of $d = 0.008$ m, and thereby the characteristics of $q_c - \chi_{ex}$ type correlation are discussed.

NOMENCLATURE

- C, constant, equation (3);
- d, I.D. of heated tube [m];
- G, mass velocity [$\mathrm{kg\,m^{-2}\,s^{-1}}$];
- H_{fg}, latent heat of evaporation [$\mathrm{J\,kg^{-1}}$];
- ΔH_i, enthalpy of inlet subcooling [$\mathrm{J\,kg^{-1}}$];
- K, parameter for the effect of inlet subcooling, equation (2);
- l, length of heated tube [m];
- p, absolute pressure [bar];
- q_c, critical heat flux [$\mathrm{W\,m^{-2}}$];
- q_{co}, q_c for $\Delta H_i = 0$ [$\mathrm{W\,m^{-2}}$].

Greek symbols

- ρ_l, density of liquid [$\mathrm{kg\,m^{-3}}$];
- ρ_v, density of vapor [$\mathrm{kg\,m^{-3}}$];
- σ, surface tension [$\mathrm{N\,m^{-1}}$];
- χ_{ex}, quality at the exit end of heated tube.

1. INTRODUCTION

IT HAS long been known that if experimental data of critical heat flux q_c, obtained within a range of inlet subcooling enthalpy ΔH_i for uniformly heated tubes with various lengths, are plotted against exit vapor quality χ_{ex} for fixed pressure p, mass velocity G and tube diameter d, it gives a single line in an approximate sense (cf. Collier [1] for example). From this empirical fact, the concept of local conditions hypothesis, namely $q_c = f(p, G, d, \chi_{ex})$, was born (Barnett [2]), but a little later, it was found that the local conditions hypothesis deteriorated either if l was very short and G was high, where q_c depended not only on χ_{ex} but also on l, or if l was very long, where χ_{ex} was kept nearly constant independently of q_c (cf. Stevens *et al.* [3], Lee and Obertelli [4], Lee [5] etc.). However, CHF/exit-quality ($q_c - \chi_{ex}$) type correlation of experimental data is convenient, so that, even now, it still constitutes the main current of correlation along with the correlation based on the boiling length concept (cf. Hewitt [6]). In addition, Heat Mass Transfer Section, Academy of Sciences, U.S.S.R. [7, 8], surveying CHF data of water (obtained in U.S.S.R. mainly), has recently given a standard table for $q_c - \chi_{ex}$ correlation of water in uniformly heated tubes of $d = 0.008$ m within the range of $p = 29.5$–196 bar and $G = 500$–$5000\,\mathrm{kg/m^2\,s}$.

On the other hand, analyzing CHF data for uniformly heated tubes with subcooled inlet conditions, the author [9–13] presented correlation equations of simple form made up of five dimensionless groups; q_c/GH_{fg}, $\sigma\rho_l/G^2 l$, ρ_v/ρ_l, l/d and $\Delta H_i/H_{fg}$, and it was shown [12] that these equations could outline general features of existing experimental data fairly well. The relation of the boiling length concept to the author's correlation equations mentioned above has been studied in [11, 13], accordingly the $q_c - \chi_{ex}$ type correlation will be investigated in the present paper on the basis of the author's correlation equations.

2. CORRELATION EQUATIONS OF CHF

For uniformly heated tubes, the following equation holds generally via the heat balance:

$$\chi_{ex} = \frac{4q_c}{GH_{fg}}\frac{l}{d} - \frac{\Delta H_i}{H_{fg}}. \tag{1}$$

Then, if q_c on the RHS of equation (1) is written as:

$$q_c = q_{co}\left(1 + K\frac{\Delta H_i}{H_{fg}}\right) \tag{2}$$

q_{co} is the critical heat flux for $\Delta H_i = 0$, and K is the dimensionless parameter for the effect of ΔH_i on critical heat flux.

2.1. *Correlation equations of* q_{co}

From [13], the author's correlation equations of q_{co}

are given as follows, being classified into four characteristic regimes called L, H, N and HP, where L-regime is the regime mainly corresponding to the so-called dryout, N-regime is the regime mainly corresponding to the so-called DNB (departure from nucleate boiling), H-regime is the intermediate regime between L- and N-regime, and HP-regime is a special regime replacing N-regime when the system pressure is extremely high.

L-regime*:

$$\frac{q_{co}}{GH_{fg}} = C\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.043}\frac{1}{l/d} \tag{3}$$

where $C = 0.25$ for $l/d < 50$, $C = 0.34$ for $150 < l/d$, and

$$C = 0.25 + \frac{(l/d) - 50}{150 - 50}(0.34 - 0.25) \text{ for } 50 < l/d < 150.$$

H- and N-regime:

$$\frac{q_{co}}{GH_{fg}} = 0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3}\frac{1}{1 + 0.0031 l/d} \tag{4}$$

and

$$\frac{q_{co}}{GH_{fg}} = 0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.433}\frac{(l/d)^{0.27}}{1 + 0.0031 l/d}. \tag{5}$$

HP-regime

$$\frac{q_{co}}{GH_{fg}} = 0.0384\left(\frac{\rho_v}{\rho_l}\right)^{0.60}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.173} \times \frac{1}{1 + 0.280(\sigma\rho_l/G^2 l)^{0.233} l/d}. \tag{6}$$

2.2. *Equations for evaluating* K

According to [11, 13], K on the RHS of equation (2) is derived theoretically from equations (3)–(6) by taking into consideration the linear $q_c - \Delta H_i$ relationship and the boiling length concept, to give the following equations.

L-regime: corresponding to q_{co} of equation (3),

$$K = \frac{1.043}{4C(\sigma\rho_l/G^2 l)^{0.043}}. \tag{7}$$

H-regime: corresponding to q_{co} of equation (4),

$$K = \frac{5}{6}\frac{0.0124 + d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{1/3}} \tag{8}$$

and corresponding to q_{co} of equation (5),

$$K = 0.416\frac{(0.0221 + d/l)(d/l)^{0.27}}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{0.433}} \tag{9}$$

HP-regime: corresponding to q_{co} of equation (6),

$$K = 1.12\frac{1.52(\sigma\rho_l/G^2 l)^{0.233} + d/l}{(\rho_v/\rho_l)^{0.60}(\sigma\rho_l/G^2 l)^{0.173}}. \tag{10}$$

For N-regime, the linear $q_c - \Delta H_i$ relationship does not hold, accordingly the equation of evaluating K in this regime has not been determined.

2.3. *Characteristic CHF-regime maps*

Figure 1 shows characteristic CHF-regime maps determined in the author's preceding study [13], indicating the extent of L-, H-, N- and HP-regime for the respective cases of density ratio $\rho_v/\rho_l = 0.01787$, 0.04847, 0.1351, and 0.3216 (corresponding to the pressure $p = 29.5$, 69, 137, and 196 bar in the case of water). Equations of q_{co} and K, which are used to estimate CHF in each regime, are denoted by numbers in Fig. 1. As mentioned in Section 2.2, there is no equation of evaluating K in N-regime. However, if it is limited to the range of small ΔH_i within N-regime near H-regime, equation (9) may probably be used to evaluate approximately the effect of subcooling on CHF in N-regime without serious error. In this sense, approximate evaluation of CHF in N-regime will be made in this paper employing equation (9), and this is the reason why equation (9) is denoted in N-regime as well in Fig. 1.

3. PREDICTION OF $q_c - \chi_{ex}$ RELATION

3.1. *Method of calculation*

For fixed p, G, and d, the relation of $q_c - \chi_{ex}$ can be evaluated in the following way for uniformly heated tubes with various lengths within a range of inlet subcooling enthalpy. First, each diagram of Fig. 1 is in the form of l/d against $\sigma\rho_l/G^2 l$, so that if l changes for fixed d, σ, ρ_l and G, it determines an oblique line on the diagram. Three oblique lines for $G = 750$, 2000 and 4000 kg/m^2 s shown in each diagram of Fig. 1 are those thus obtained for water flowing in tubes of $d = 0.008$ m at the pressure indicated in each diagram. It is noted that as the pressure increases, the location of oblique line for the same magnitude of G moves to the left due to the decrease of σ and ρ_l; and even at the same pressure, if G increases, the oblique line moves to the left, making the change of characteristic regime through which the oblique line passes. For example, in the case of $G = 750$ kg/m^2 s in the top diagram of Fig. 1, the oblique line passes through N, H and L successively as increasing l (that is l/d), whereas in the cases of $G = 2000$ and 4000 kg/m^2 s, the oblique lines pass through N and H alone. Then, employing a set of two equations denoted in each characteristic CHF-regime, q_{co} and K can be evaluated for a given value of l (or l/d), and accordingly the relation of $q_c - \chi_{ex}$ is obtained through equations (1) and (2) for a given set of l and ΔH_i.

3.2. *Result of calculation*

In this paper, the relation of $q_c - \chi_{ex}$ for a given value of l/d is evaluated for the range of $\Delta H_i = 0$ to

*$q_{co}/GH_{fg} = 0.25/(l/d)$, which holds in L-regime for $l/d > 100$ and $\sigma\rho_l/G^2 l > 7.84 \times 10^{-4}$, is omitted here because it is unnecessary for the range dealt with in this paper.

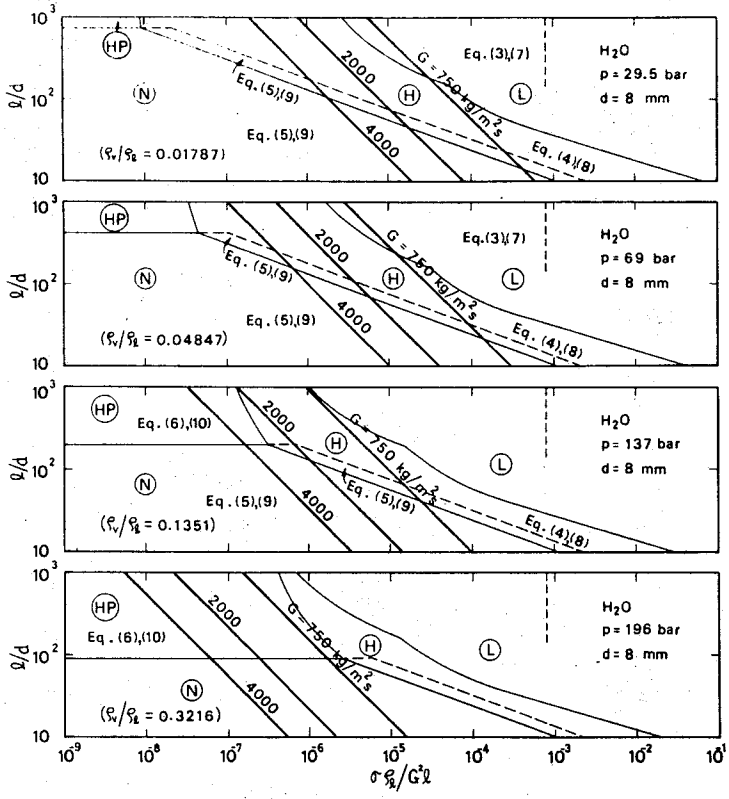

FIG. 1. CHF-regime maps for $\rho_v/\rho_l = 0.01787$, 0.04847, 0.1351 and 0.3216.

400 kJ/kg, with reference to the range of inlet subcooling enthalpy encountered ordinarily in the experiment of water; and the $q_c - \chi_{ex}$ relation thus determined is illustrated in the $q_c - \chi_{ex}$ diagram by a short segment of line for every prescribed value of l/d.

Now, the relations of $q_c - \chi_{ex}$ obtained for discrete values of l/d between 10 and 793 are shown in Figs. 2–5 corresponding to the respective diagrams of Fig. 1, where broken lines show CHF belonging to N-regime, thin lines to H-regime, thick lines to L-regime, and chain lines to HP-regime. In case of $G = 4000\,\mathrm{kg/m^2\,s}$ in Fig. 4, it is noted that there is a discontinuous gap between the predictions of CHF in N-regime ($l/d = 10$–148) and those in HP-regime ($l/d = 207$–793), and it results from the author's correlation equations determined with discontinuous values of q_c and K above and below the horizontal lines separating HP- and N-regime in Fig. 1. On the other hand, in Fig. 5 are shown the prediction of $q_c - \chi_{ex}$ in HP-regime alone, because experimental data of CHF in N-regime have not yet been published for water at $p = 196$ bar.

4. DISCUSSION

(I) As for the relation of $q_c - \chi_{ex}$ predicted by the author's generalized correlation equations, it is noted first in Figs. 2–4 that in N-regime, where l/d is small, q_c is not a single-valued function of χ_{ex}, and the degree of dispersion of $q_c - \chi_{ex}$ relation due to l/d becomes higher as G is increased.* Second, the relation of $q_c - \chi_{ex}$ in H-regime can constitute a continuous curve approximately, and the curve has a trend to bend downward when l/d becomes very large. It may be of interest to note that all the characteristics mentioned above of N- and H-regime agree with those found by Stevens *et al.* [3] in the experiment for boiling of R-12.

Meantime, L-regime has little chance to appear in the range of conditions dealt with in this paper, only taking place when l/d is very large in case of $p = 29.5$–69 bar and $G = 750\,\mathrm{kg/m^2\,s}$.

As for HP-regime, it is noted in Fig. 5 that roughly speaking, q_c has a trend of being approximately a single-valued function of χ_{ex}, and besides, q_c has a very simple relation to χ_{ex} decreasing monotonously as increasing χ_{ex}.

(II) Solid circular symbols plotted in Figs. 2–5 come

*It has been known (cf. Lee and Obertelli [4] for example) that the linear $q_c - \Delta H_i$ relationship does not hold in the range of this dispersion. Meanwhile, the author's definition of N-regime (cf. [9]) is based on the non-linear $q_c - \Delta H_i$ relationship. Thus the two ranges are identical.

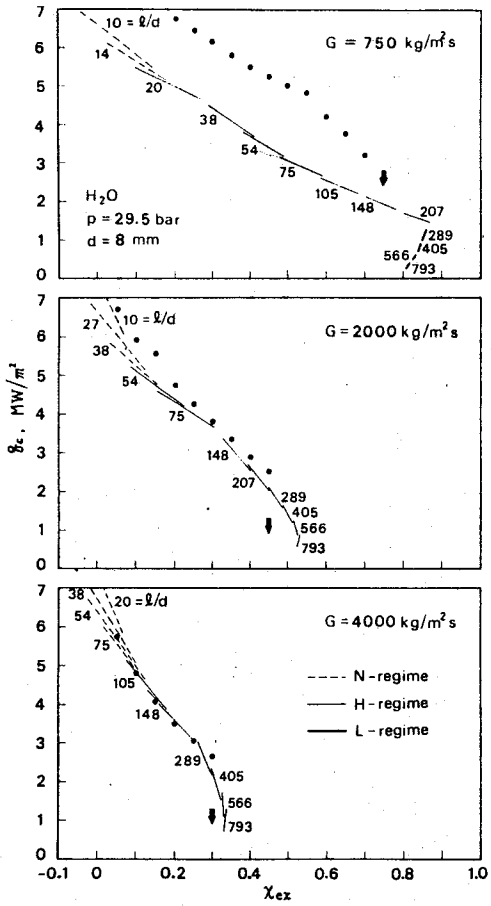

FIG. 2. Relation of $q_c - \chi_{ex}$ for water at $p = 29.5$ bar in tubes of $d = 0.008$ m.

from the U.S.S.R. standard table for the relation of $q_c - \chi_{ex}$ given for water in tubes of $d = 0.008$ m [7, 8], and χ_{ex} indicated by an arrow pointing downward in Figs. 2–4 is called "the limiting quality", depending on the assumption [8] that the relation of $q_c - \chi_{ex}$ has a flection point near this position.

Then, if the relation of $q_c - \chi_{ex}$ given by the U.S.S.R. standard table is compared with the predicted relation of $q_c - \chi_{ex}$, it is firstly noticed that disagreement is noticeable in case of $G = 750$ kg/m² s for both Figs. 2 and 3. In connection with this matter, Milan reproducibility test data for $d = 0.01$ m and $l/d = 200$ [15], which is regarded as very reliable, are shown by crosses in Fig. 3, suggesting the possibility that the standard table may give a little too high values for q_c at $G = 750$ kg/m² s of Figs. 2 and 3. Second, in case of $G = 750$ kg/m² s in Fig. 4, the arrow indicating the limiting quality is located considerably apart from the position where the predicted $q_c - \chi_{ex}$ curve bends downward, but the cause is unknown at present. Third, it is noted in Fig. 5 for HP-regime that the deviation increases to some extent between the predicted and the standard value as increasing G, and the cause is also unknown.

However, excepting the matter mentioned above, we can see a general trend that the author's generalized correlation equations may be a match for the U.S.S.R. standard table for water; and this fact may be regarded as a kind of certification to the availability of the author's generalized correlation.

(III) However, there is a problem whether the change of CHF mechanism takes place or not near the so-called limiting quality indicated by arrows in Figs. 2–4. It is postulated in the papers [7, 8, 14] that CHF which takes place in the range of χ_{ex} less than the limiting quality, shown by solid circles in Figs. 2–4, is of DNB, but near the limiting quality the mechanism of CHF changes to the dryout of liquid film on the tube wall. However, it must be noted in Figs. 2 and 3 (G = 2000–4000 kg/m² s) and in Fig. 4 (G = 2000 kg/m² s) that the q_c–χ_{ex} relation predicted by the author's correlation equations bends downward

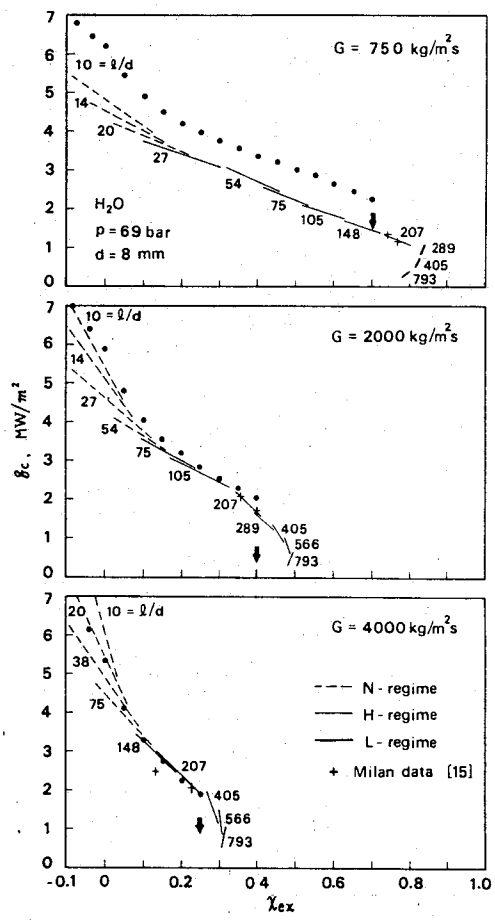

FIG. 3. Relation of $q_c - \chi_{ex}$ for water at $p = 69$ bar in tubes of $d = 0.008$ m.

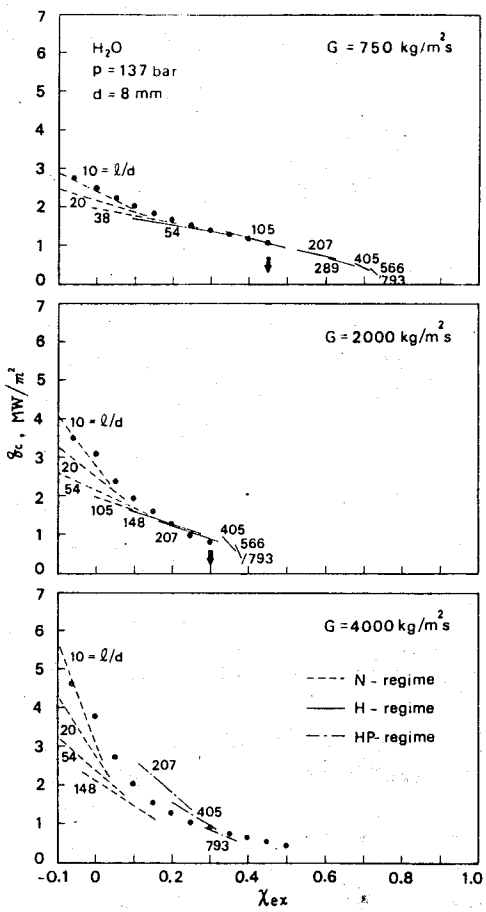

FIG. 4. Relation of $q_c - \chi_{ex}$ for water at $p = 137$ bar in tubes of $d = 0.008$ m.

near the so-called limiting quality, but H-regime remains up to $l/d = 793$ showing no change of CHF-regime. In addition, as mentioned in Section 2, H-regime is the regime lying between N-regime (mainly corresponding to DNB) and L-regime (mainly corresponding to dryout).

Therefore, further studies are necessary on the contradiction mentioned above, but it may not be useless to point out the following fact. Namely, as for the experiments of CHF with subcooled (and saturated in the limit) inlet conditions, there are very few existing data in H-regime for $l/d > 400$ (cf. [12]); and therefore, almost all experiments connecting with the concept of limiting quality are presumed to be made by employing tubes of $l/d < 400$ with mixed inlet conditions. In this respect, Hewitt [6] has mentioned that the generality may be questionable for the conclusions drawn from the experiments of mixed inlet conditions.

(IV) It seems customary to assume that if the length of heated tube l is increased fixing all other conditions, the rise of χ_{ex} takes place along with the fall of q_c so as to enter the state of dryout, and finally the curve of $q_c - \chi_{ex}$ should approach the point of $\chi_{ex} = 1$ and $q_c = 0$. However, there is an empirical fact relating to the study of CHF based on $q_c - \chi_{ex}$ diagram that the high χ_{ex} region such as mentioned above is rarely entered with tubes of practicable lengths in case of the subcooled inlet conditions (cf. Bennett [17], Lee [5] etc.). Besides, according to the prediction of the author's correlation equations, the relation of $q_c - \chi_{ex}$ for fixed p, G and d always, except in HP-regime, has a bend which is convex to the right (see Figs. 2–4). Therefore, there is a possibility that the customary view mentioned at the beginning of this item may not be correct in case of the subcooled inlet conditions. On this point, it should be noted that according to equation (1), when $q_c l \to 0$ for fixed d, G and H_{fg}, the exit quality $\chi_{ex} \to -\Delta H_i/H_{fg}$, where $\Delta H_i/H_{fg}$ does not become negative for subcooled or saturated inlet conditions. Thus the $q_c - \chi_{ex}$ relation predicted by the author's correlation equations in Figs. 2–5 interprets the fact that $q_c l$ tends to vanish as increasing l for L- and H-regime, whereas $q_c l$ does not vanish as increasing l for HP-regime.

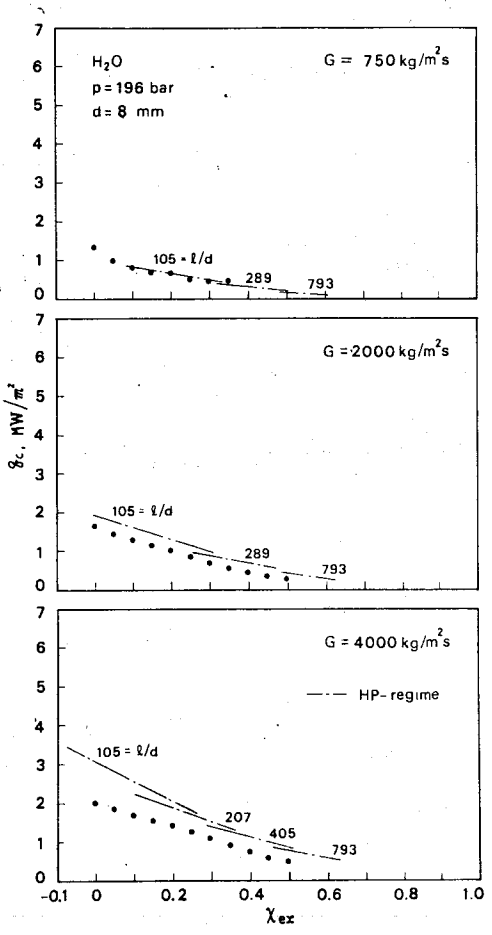

FIG. 5. Relation of $q_c - \chi_{ex}$ for water at $p = 196$ bar in tubes of $d = 0.008$ m.

5. CONCLUSIONS

With respect to the heat-flux/exit-quality type correlation of CHF, the following conclusions follow from the present study.

(i) The author's correlation equations of CHF seem available for outlining the characteristics of $q_c - \chi_{ex}$ type correlation in case of subcooled inlet conditions.

(ii) There are a few contradictions or disagreements between the customary views related to the $q_c - \chi_{ex}$ type correlation and the prediction of the author's correlation equations.

(iii) The $q_c - \chi_{ex}$ type correlation may not be a way adequate for presuming the mechanism of CHF, because it is subject to the ambiguity due to the intrinsic dependence of χ_{ex} upon q_c as seen in equations (1) and (2).

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SUR LA RELATION TYPE DE CHF FLUX THERMIQUE/QUALITE DE LA VAPEUR A LA SORTIE POUR L'EBULLITION AVEC CONVECTION FORCEE DANS DES TUBES CHAUFFES ET VERTICAUX

Résumé — On essaie d'utiliser des formules générales de CHF proposées récemment par l'auteur pour prédire les relations classiques entre le flux thermique critique q_c et la qualité en sortie χ_{ex} quand la longueur du tube l et l'enthalpie de sous-refroidissement ΔH_i à l'entrée varient, la pression p, la vitesse massique G et le diamètre de tube d étant fixée. Un calcul numérique est fait pour l'eau sous les conditions p = 29,5 à 196 bar, G = 750 à 4000 kg/m² s, d = 0,008 m, l/d = 10 à 793 et ΔH_i = 0 à 400 kJ/kg.

Le résultat est comparé avec la table standard de $q_c - \chi_{ex}$ publiée récemment par la Section de transfert de chaleur et de masse de l'Académie des Sciences U.R.S.S. pour l'eau dans des tubes chauffés uniformément avec d = 0,008 m; on discute la relation type $q_c - \chi_{ex}$.

BERECHNUNG DER KRITISCHEN WÄRMESTROMDICHTE BEIM STRÖMUNGSSIEDEN IN GLEICHMÄßIG BEHEIZTEN SENKRECHTEN ROHREN MIT GLEICHUNGEN, BEI DENEN DIE WÄRMESTROMDICHTE VON DAMPFGEHALT AM ROHRAUSTRITT ABHÄNGT

Zusammenfassung—In diesem Aufsatz wird versucht, verallgemeinerte Gleichungen zur Berechnung der kritischen Wärmestromdichte anzuwenden. Die Gleichungen hat der Autor in jüngster Zeit vorgeschlagen, um den bekannten Zusammenhang anzugeben, welcher zwischen der kritischen Wärmestromdichte q_c und dem Dampfgehalt am Rohraustritt χ_{ex} besteht, wenn die Rohrlänge L und die Unterkühlungsenthalpie ΔH_i

verändert werden, bei gleichbleibenden Werten des Druckes p, der Massenstromdichte G und des Rohrdurchmessers d.

Numerische Berechnungen werden für Wasser durchgeführt; die Bedingungen sind $29{,}5 < p < 196$ bar, $750 < G < 4000$ kg/m² s, $d = 0{,}008$ m, $10 < L/D < 793$ und $0 < \Delta H_i < 400$ kJ/kg. Anschließend werden die Ergebnisse verglichen mit den Werten der Tabelle für q_c (χ_{ex}), welche kürzlich von der Wärme- und Stoffübertragungsabteilung der Sowjetischen Akademie der Wissenschaften veröffentlicht wurde. Sie gilt für Wasser in gleichmäßig beheizten Rohren mit $d = 0{,}008$ m. Es werden die Eigenschaften des Gleichungstyps $q_c f(\chi_{ex})$ diskutiert.

О ЗАВИСИМОСТИ ТИПА «ПЛОТНОСТЬ ТЕПЛОВОГО ПОТОКА — ПАРОСОДЕРЖАНИЕ НА ВЫХОДЕ» ДЛЯ КРИТИЧЕСКОГО ТЕПЛОВОГО ПОТОКА В СЛУЧАЕ КИПЕНИЯ ПРИ ВЫНУЖДЕННОЙ КОНВЕКЦИИ В РАВНОМЕРНО НАГРЕВАЕМЫХ ВЕРТИКАЛЬНЫХ ТРУБАХ

Аннотация — Предпринята попытка использовать обобщенные соотношения для критического теплового потока, предложенные ранее автором для расчета зависимости критического теплового потока q_c от паросодержания на выходе χ_{ex} в случае, когда длина трубы l и энтальпия недогрева на входе ΔH_i изменяются при постоянных давлении, массовой скорости и диаметре трубы d. Проведены численные расчеты для воды в следующих диапазонах изменения величин: $p = 29{,}5$–196 бар, $G = 750$–4000 кг/м² сек, $d = 0{,}008$ м, $l/d = 10$–793 и $\Delta H_i = 0$–400 кДж/кг. Дано сравнение полученных значений со стандартной таблицей $q_c - \chi_{ex}$, опубликованной недавно Отделением тепломассообмена АН СССР для воды в равномерно нагреваемых трубах диаметром $d = 0{,}008$ м. Тем самым проведена оценка зависимостей типа $q_c - \chi_{ex}$.

ON THE RELATION BETWEEN CRITICAL HEAT FLUX AND OUTLET FLOW PATTERN OF FORCED CONVECTION BOILING IN UNIFORMLY HEATED VERTICAL TUBES

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Abstract—For critical heat flux (CHF) of forced convection boiling in uniformly heated vertical tubes, the author recently proposed generalized correlation equations being classified into four characteristic CHF-regimes called L, H, N and HP. In the present paper, a study is made on the relation between the above-mentioned CHF-regimes and the flow patterns at the tube exit, by employing existing data of flow patterns for uniformly heated tubes.

NOMENCLATURE

d, I.D. of heated tubes [m];
G, mass velocity [kg m^{-2} s^{-1}];
H_{fg}, latent heat of evaporation [J kg^{-1}];
ΔH_i, enthalpy of inlet subcooling [J kg^{-1}];
l, length of heated tube [m];
p, absolute pressure [bar];
q, heat flux [W m^{-2}];
q_c, critical heat flux [W m^{-2}];
T_i, inlet temperature [°C];
T_s, saturation temperature [°C].

Greek symbols

ρ_l, density of liquid [kg m^{-3}];
ρ_v, density of vapor [kg m^{-3}];
σ, surface tension [N m^{-1}];
χ_{ex}, exit quality.

1. INTRODUCTION

ANALYZING experimental data of critical heat flux (CHF) of forced convection boiling in uniformly heated vertical tubes, the author [1–3] presented generalized correlation equations of CHF being classified into four characteristic regimes called L, H, N and HP. Then, it was followed by a study on generalized graphic representation of many existing CHF data [4], showing that the author's correlation equations of CHF are capable of outlining the general features of CHF concerned.

L-regime mentioned above is the regime that takes place when mass velocity G is low for fixed other conditions, whereas N-regime makes the appearance when G is very high; and H-regime is the intermediate one between L- and N-regime. On the other hand, HP-regime is a special one that replaces N-regime when the system pressure is extremely high. In addition, N-regime has a character distinguishing itself from the other three regimes by showing non-linear relationship between critical heat flux q_c and inlet subcooling enthalpy ΔH_i (cf. [1]).

Then, it is well known (cf. Collier [5] for example) that for uniformly heated channels, the onset of CHF condition can be assumed to occur first at the exit end of the channel. Therefore, relating to the study on the mechanism of CHF, it may not be useless to investigate the connection between characteristic CHF-regimes and flow patterns at the tube exit.

2. FLOW PATTERNS AT THE TUBE EXIT

Bergles and Suo [6] measured flow patterns of water/steam two-phase flow in uniformly heated vertical tubes, changing both the mass velocity G and the heat flux q under the conditions of pressure p = 34.5 bar (with three kinds of inlet temperature T_i = 121, 163 and 204°C against saturation temperature T_s = 241°C) and p = 69 bar (with three kinds of inlet temperature T_i = 149, 204 and 260°C against saturation temperature T_s = 283°C), internal diameter of heated tube d = 0.0102 m, and three kinds of length/diameter ratio of tube l/d = 60, 150 and 240. In the above study, classification of flow pattern was made by recording the variation of conductance between the tube wall and the tip of an electrical resistance probe inserted into the flow at the tube exit. As a result, four kinds of flow regimes called bubbly, slug, froth and spray annular, were classified to give flow regime maps, where flow pattern boundaries were represented in terms of the mass velocity G and the exit quality χ_{ex}.

For uniformly heated round tubes, the following equation holds via heat balance:

$$\frac{q}{GH_{fg}} = \frac{1}{4}\frac{d}{l}\left(\chi_{ex} + \frac{\Delta H_i}{H_{fg}}\right) \tag{1}$$

where H_{fg} is the latent heat of evaporation. Through this equation, therefore, the value of q/GH_{fg} is readily estimated at a point on the flow regime maps mentioned above (Figs. 6–8 in [6]), while the value of $\sigma\rho_l/G^2 l$ at the same point (where σ is the surface tension and ρ_l the density of liquid) can be determined from the experimental conditions. In other words, the

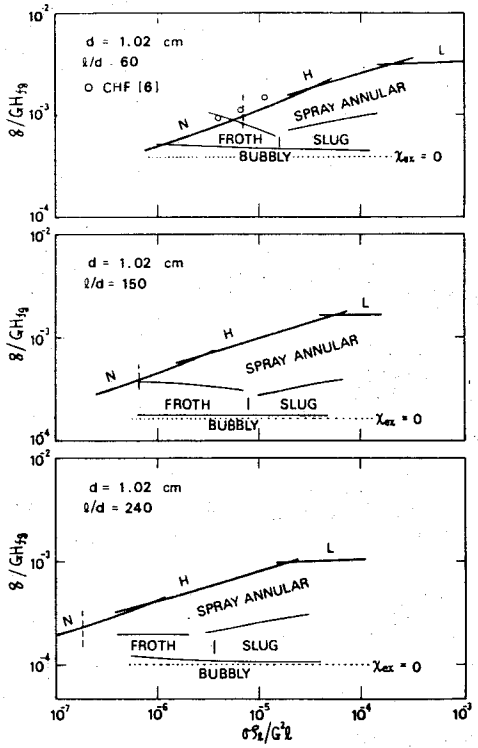

FIG. 1. Flow patterns at the tube exit (Bergles and Suo [6]) and characteristic CHF-regimes. Water, $p = 34.5$ bar, $T_s = 241°C$, $T_i = 204°C$ ($\Delta H_i/H_{fg} = 0.0975$).

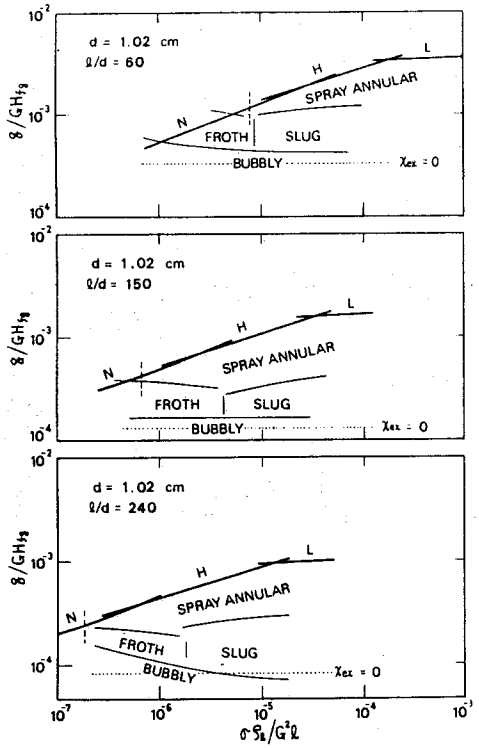

FIG. 2. Flow patterns at the tube exit (Bergles and Suo [6]) and characteristic CHF-regimes. Water, $p = 69$ bar, $T_s = 283°C$, $T_i = 260°C$ ($\Delta H_i/H_{fg} = 0.0799$).

flow regime map of G-χ_{ex} type obtained by Bergles and Suo can be transformed to the diagram of (q/GH_{fg})-$(\sigma\rho_l/G^2l)$ type, and typical results are shown in Fig. 1 for the case of $T_i = 204°C$ at $p = 34.5$ bar, and in Fig. 2 for the case of $T_i = 260°C$ at $p = 69$ bar. A horizontal dotted line in each diagram of Figs. 1 and 2 represents the value of heat flux q to generate $\chi_{ex} = 0$ at the tube exit; and three open circles plotted in the top diagram of Fig. 1 represent experimental data of CHF measured by Bergles and Suo.

On the other hand, Bennett *et al.* [7] made experiments with water under the condition of pressure $p =$ 34.5 bar and 69 bar (inlet subcooling temperature $T_s - T_i \doteqdot 11$ to 17°C in both cases), internal diameter of heated tube $d = 0.0126$ m, and tube length $l = 3.66$ m. In this study, the flow pattern was detected by means of still and high-speed moving pictures of the two-phase flow emerging from the end of the uniformly heated vertical tube, complemented by X-ray photography of the flow in the heated tube, to give flow regime maps of G-χ_{ex} type (Fig. 5.7 for $p = 34.5$ bar and Fig. 5.8 for $p =$ 69 bar in [7]).

Then, a diagram of (q/GH_{fg})-$(\sigma\rho_l/G^2l)$ type corresponding to the above flow regime map in case of 69 bar alone is shown in Fig. 3, for simplicity. A chain line in Fig. 3 is the experimental CHF line obtained by Bennett *et al.* It is noted that experimental conditions giving the result of Fig. 3 are close to those of the bottom diagram of Fig. 2, and it is a matter of no importance that the dotted line of $\chi_{ex} = 0$ in Fig. 3 is located lower than that in Fig. 2, because it only depends on the difference of inlet subcooling. In addition, according to the two reports [6, 7], it may be assumed that the froth flow in Fig. 2 and the wispy annular flow in Fig. 3 designate nearly a similar flow pattern, so that Figs. 2 and 3 are not so different from each other. The boundary between the froth and spray annular flow in the bottom diagram of Fig. 2 is different in position from the boundary between the wispy annular and annular flow in Fig. 3; but following Bergles and Suo, it may be presumed that the discrepancy is due in large part to the different detection techniques of flow pattern.

3. CHARACTERISTIC REGIMES OF CHF

Critical heat flux q_c for a flow with inlet subcooling enthalpy ΔH_i is now written as:

$$q_c = q_{co}\left(1 + K\frac{\Delta H_i}{H_{fg}}\right). \tag{2}$$

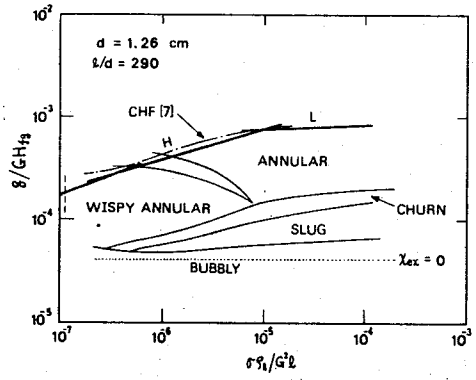

FIG. 3. Flow patterns at the tube exit (Bennett *et al.* [7]) and characteristic CHF-regimes. Water, $p = 69$ bar, $T_s = 283°C$, T_i (average) $= 269°C$ ($\Delta H_i/H_{fg} = 0.0476$).

Then, according to the author's preceding paper [4], q_{co} on the RHS of equation (2) is evaluated as follows:

L-regime*:

$$\frac{q_{co}}{GH_{fg}} = C\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.043}\frac{1}{l/d} \tag{3}$$

where $C = 0.25$ for $l/d < 50$, $C = 0.34$ for $l/d > 150$, and C changes linearly from 0.25 to 0.34 with l/d for $l/d = 50$ to 150.

H- and N-regime:

$$\frac{q_{co}}{GH_{fg}} = 0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3}\frac{1}{1+0.0031\,l/d} \tag{4}$$

and

$$\frac{q_{co}}{GH_{fg}} = 0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133} \times \left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.433}\frac{(l/d)^{0.27}}{1+0.0031\,l/d} \tag{5}$$

where ρ_v is the density of vapor. The boundary between H- and N-regime is given by

$$\frac{\sigma\rho_l}{G^2 l} = \left(\frac{0.77}{l/d}\right)^{2.70}. \tag{6}$$

Next, the author's previous paper [3] has shown that K on the RHS of equation (2) is derived theoretically from equations (3)–(5) by utilizing the linear relationships between q_c and ΔH_i as well as the concept of boiling length as follows:

L-regime: corresponding to equation (3),

$$K = \frac{1.043}{4C(\sigma\rho_l/G^2 l)^{0.043}} \tag{7}$$

H-regime: corresponding to equation (4),

$$K = \frac{5}{6}\frac{0.0124 + d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{1/3}} \tag{8}$$

and corresponding to equation (5),

$$K = 0.416\frac{(0.0221 + d/l)(d/l)^{0.27}}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{0.433}} \tag{9}$$

For N-regime, the linear relationship between q_c and ΔH_i does not hold, so that the equation of evaluating K has not been determined. However, if it is limited to the range of small ΔH_i for N-regime near H-regime, equation (9) may probably be used approximately for predicting the effect of subcooling on CHF in N-regime without serious error. In the present paper, therefore, q_c in N-regime is estimated by this approximate means. Thick lines in Figs. 1–3 represent CHF thus estimated through equations (2)–(9), and the marks of L, H, and N indicate characteristic regimes of CHF, where the vertical broken line separating N-regime from H-regime is given by equation (6).

4. DISCUSSIONS

(i) It is noted in the top diagram of Fig. 1 that three open circles showing the data of CHF measured by Bergles and Suo have appeared near the prediction of CHF by the author's correlation equations, and it is also noted in Fig. 3 that the situation is quite the same for the experimental CHF line obtained by Bennett *et al.*

(ii) Through all the six diagrams of Figs. 1 and 2, it may be assumed approximately that the flow pattern at the tube exit is spray annular when CHF takes place in L- and H-regime, whereas it is froth (or bubbly) when CHF takes place in N-regime. As for Fig. 3, the similar situation as above may probably be assumed in a rough sense at least (see comments on Fig. 3 described in Section 3).

(iii) The result of the preceding item (ii) suggests that the flow pattern near the tube exit is almost identical for both L- and H-regime. Therefore, the reason for the noticeable difference of characters between CHF in L-regime and that in H-regime must be attributed to more detailed structure of flow. Discussions on minute structures of flow are beyond the scope of this study of course, but there may be a possibility, for example, to distinguish the CHF ascribed to the dryout of a liquid film on the heated tube wall (L-regime) from the CHF ascribed to the breakdown of a liquid film flow with a finite thickness on the wall (H-regime).

(iv) The result of item (ii) also suggests the possibility that CHF in N-regime is closely related to the so-called DNB (departure from nucleate boiling). In addition, there is an interesting fact that the non-linear $q_c - \Delta H_i$ relationship, constituting a characteristic feature of N-regime, is related to the flow pattern of froth or bubbly flow.

(v) From the results of Figs. 1–3, it may be con-

* $q_{co}/GH_{fg} = 0.25/(l/d)$, which holds in L-regime for $l/d > 100$ and $\sigma\rho_l/G^2 l > 7.84 \times 10^{-4}$, is omitted here because it is unnecessary for the range dealt with in this paper.

cluded that CHF does not occur under the condition of slug (or churn) flow.

(vi) Due to the lack of experimental informations of flow pattern for CHF in HP-regime, it is unknown at present whether a new type of flow pattern exists or not for CHF in HP-regime.

5. CONCLUSION

Characteristic regimes of CHF in uniformly heated tubes, determined empirically by the author through the analysis of existing CHF data, are compared with flow regime maps obtained by Bergles and Suo, and by Bennett *et al.*, for the two-phase flow in uniformly heated tubes, clarifying the matters mentioned in items (i) to (vi) of Section 4.

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SUR LA RELATION ENTRE LE FLUX THERMIQUE CRITIQUE ET LA CONFIGURATION DE L'ECOULEMENT EXTERIEUR POUR L'EBULLITION AVEC CONVECTION FORCEE DANS DES TUBES VERTICAUX ET CHAUFFES UNIFORMEMENT

Résumé — Pour le flux thermique critique (CHF) de l'ébullition avec convection forcée dans des tubes chauffés verticaux, l'auteur a récemment proposé des formules générales classées en quatre régimes CHF caractéristiques appelés *L*, *H*, *N*, et *HP*. Dans ce texte, on étudie la relation entre les régimes précédents et les configurations d'écoulement pour les tubes chauffés uniformément.

EINE BEZIEHUNG ZWISCHEN DER KRITISCHEN WÄRMESTROMDICHTE UND DER STRÖMUNGSFORM AM ROHRAUSTRITT BEIM STRÖMUNGSSIEDEN IN GLEICHMÄSSIG BEHEIZTEN SENKRECHTEN RÖHREN

Zusammenfassung—Zur Berechnung der kritischen Wärmestromdichte beim Strömungssieden in gleichmäßig beheizten senkrechten Rohren hat der Autor in jüngster Zeit allgemeingültige Gleichungen vorgeschlagen. Die Gleichungen gelten für vier charakteristische Bereiche der kritischen Wärmestromdichte, welche L, H, N und HP genannt werden. In der vorliegenden Arbeit wird untersucht, wie die erwähnten Bereiche der kritischen Wärmestromdichte und die Strömungsformen am Rohraustritt zusammenhängen. Dazu werden Angaben über die Strömungsformen bei gleichmäßig beheizten Rohren verwendet.

О СООТНОШЕНИИ МЕЖДУ КРИТИЧЕСКИМ ТЕПЛОВЫМ ПОТОКОМ И КАРТИНОЙ ТЕЧЕНИЯ НА ВЫХОДЕ ИЗ ТРУБЫ ПРИ КИПЕНИИ С ВЫНУЖДЕННОЙ КОНВЕКЦИЕЙ В РАВНОМЕРНО НАГРЕВАЕМЫХ ВЕРТИКАЛЬНЫХ ТРУБАХ

Аннотация — Для критического теплового потока при кипении с вынужденной конвекцией в равномерно нагреваемых вертикальных трубах автором ранее были предложены обобщенные соотношения для четырех характерных режимов критического теплового потока, обозначенных символами *L*, *H*, *N* и *HP*. В данной работе на основании имеющихся данных по характеру течения в равномерно нагреваемых трубах проведено исследование взаимосвязи указанных режимов критического теплового потока и картины течения на выходе из трубы.

GENERALIZED CORRELATIONS OF CRITICAL HEAT FLUX FOR THE FORCED CONVECTION BOILING IN VERTICAL UNIFORMLY HEATED ANNULI

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Abstract—Following the analytical procedure of the same principle as that applied by the author to the analysis of critical heat flux (CHF) for round tubes, existing experimental data of CHF in annuli with inside heating are analyzed to yield a generalized correlation of CHF proper to this type of heating. As for annuli with outside heating, it is shown that CHF can be readily predicted by utilizing the author's generalized correlation of CHF for round tubes. The prediction of CHF in both cases mentioned above can be established by employing the heated equivalent diameter alone as the representative diameter of annuli. Finally, the case of bilaterally heated annuli is also discussed, and an approximate method for the prediction of CHF in L- and H-regime is presented, while it is suggested that the interference between two walls disappears for CHF in HP-regime.

NOMENCLATURE

- d, inner diameter of round tube [m];
- d_{he}, heated equivalent diameter [m];
- d_i, outer diameter of rod of annulus [m];
- d_0, inner diameter of shroud tube of annulus [m];
- G, mass velocity [kg/m^2 s];
- H_{fg}, latent heat of evaporation [J/kg];
- ΔH_i, enthalpy of inlet subcooling [J/kg];
- K, parameter for inlet subcooling effect, equation (14);
- l, length of heated rod or tube [m];
- p, absolute pressure [bar];
- q_c, critical heat flux [W/m^2];
- q_{c0}, q_c for $\Delta H_i = 0$ [W/m^2].

Greek symbols

- α, fraction of total power added to shroud tube;
- ρ_l, density of liquid [kg/m^3];
- ρ_v, density of vapor [kg/m^3];
- σ, surface tension [N/m];
- χ_{ex}, exit quality.

1. INTRODUCTION

THE PRESENT paper deals with the critical heat flux (CHF) for the forced convection boiling in annular channels in the conditions that the fluid fed to the channels is subcooled (inclusive of saturated liquid in the limit) and the flow is stable with no oscillations. For CHF of water in annuli with inside heating, where the inside rod is heated and the shroud tube unheated, Barnett [1] compiled 724 experimental data at the pressure of 69.0 bars from 8 sources, and later [2] compiled 106 new data at the same pressure from 3 sources. Janssen *et al.* [3] made experiments of water at 41.4–100.0 bars (among their data, those at 69.0 bars were compiled in [1]), while the study of Little [4] included experiments of water at 51.7–69.0 bars. As for pure fluids other than water, the study of Stevens *et al.* [5] included experiments of Freon-12 at 8.1–10.7 bars, while Ahmed *et al.* [6] presented 418 experimental data of Freon-12 at 10.7–16.4 bars. On the other hand, under conditions comprising inside heating (where the rod is heated and the shroud unheated), outside heating (where the rod is unheated and the shroud heated), and bilateral heating (where the rod and shroud are heated), experiments of annuli were made for water, in which Becker *et al.* [7] (p = 8.3–36.8 bars) and Jensen *et al.* [8] (p = 69.0 bars) showed that CHF on one side wall was affected by the heating of the other side wall, whereas Ornatskiy *et al.* [9] (p = 172–216 bars) and Tolubinskiy *et al.* [10] (p = 150–200 bars) found no effects of the other side wall heating on CHF. In another study of Ornatskiy *et al.* [11] were also included experiments of water at 98.1–186.5 bars for outside heating.

As for the correlation of experimental data, Barnett [1], Janssen *et al.* [3], Tong *et al.* [12], Bertolletti *et al.* [13], Becker [14] and Hewitt [15] developed empirical correlations of CHF for water flowing in annuli with inside heating, while Ahmad *et al.* [6, 16] presented a kind of generalized correlation of CHF for inside heating in relation to their study of CHF modeling. In addition, for common use without discrimination between the inside and the outside heating, Tolubinskiy *et al.* [17] presented an empirical correlation of CHF for water, and Ornatskiy [18] a kind of generalized

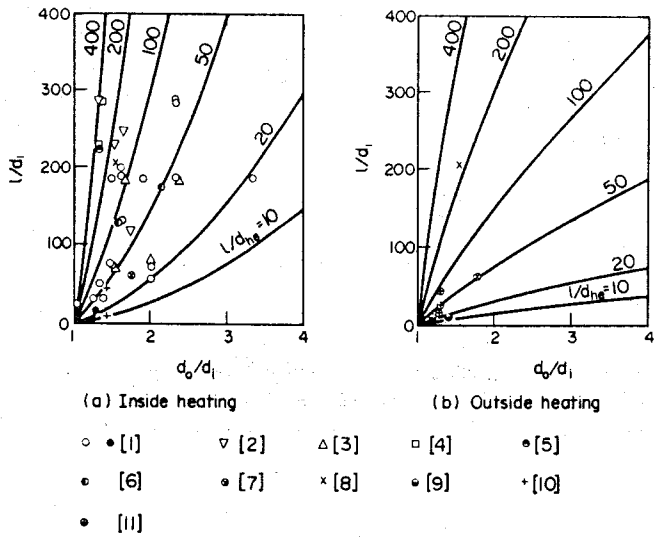

FIG. 1. Geometrical conditions and heated equivalent diameter for annuli.

correlation of CHF, but it must be noted that the former correlation implies CHF of natural convection boiling as a base and the latter correlation of CHF is composed including the gravitational acceleration g to make a dimensionless group.

Recently, the author developed a generalized correlation of CHF for round tubes [19] and showed its wide applicability [20]. The present paper reports the results obtained when the analytical procedure of the same principle is extended to annuli, which have two diameters as against one diameter in round tubes.

2. HEATED EQUIVALENT DIAMETER

Heated equivalent diameter d_{he} plays a dominant role in correlating CHF of annuli as it will be seen later, and a brief description of d_{he} is given here for convenience. Generally, heated equivalent diameter d_{he} is defined for the cross-section of any one-dimensional channel as:

$$d_{he} = (4 \times \text{flow area})/(\text{heated perimeter}), \quad (1)$$

and in case of uniform heat flux, the following heat balance relation holds for channels of length l:

$$\frac{q_c}{GH_{fg}} \cdot \frac{4l}{d_{he}} - \frac{\Delta H_i}{H_{fg}} = \chi_{ex}. \quad (2)$$

On the other hand, from the definition of equation (1), heated equivalent diameter d_{he} for annuli having the rod of diameter d_i and the shroud tube of diameter d_0 is determined as follows:

for inside heating:

$$\frac{l}{d_{he}} = \frac{l}{d_i} \cdot \frac{1}{(d_0/d_i)^2 - 1}; \quad (3)$$

for outside heating:

$$\frac{l}{d_{he}} = \frac{l}{d_i} \cdot \frac{d_0/d_i}{(d_0/d_i)^2 - 1}. \quad (4)$$

Equations (3) and (4) are illustrated in Fig. 1(a) and (b) respectively, and geometrical conditions of annuli employed in the experiments of [1-11] are also shown in Fig. 1. It should be noticed in Fig. 1 that when d_0/d_i is near unity, even a slight change of d_0/d_i causes a great change of l/d_{he}, and besides, that the origin of Fig. 1(a) and (b) ($l/d_i = 0$, $d_0/d_i = 1$) is a singular point corresponding to rectangular channels, where l/d_{he} becomes indefinite in equations (3) and (4) unless the annular width $(d_0 - d_i)/2$ is kept constant.

3. CHF FOR INSIDE HEATING

This chapter deals with CHF in case of annuli with inside heating, where the inside rod is heated and the shroud unheated.

3.1. *Relationship between* q_c *and* ΔH_i

When the relationship between q_c and ΔH_i is surveyed on the experimental data of [1-6], it is found that when the mass velocity G is low, the linear relationship (type A in Fig. 2) is observed, whereas when G is sufficiently high, the non-linear relationship (type B generally, and also type b occasionally in Fig. 2) appears. This phenomenological nature of CHF in annuli with inside heating is the same as that observed in round tubes [19], with the exception of occasional appearance of $q_c - \Delta H_i$ relationship of type b in Fig. 2.

3.2. *The clue for analyzing* q_{c0} *data*

Experiments represented by two black circular symbols near the origin of Fig. 1(a) give rather abnormal data compared with the other experiments, so that only these data are excluded from the analysis in the present paper. The analytical procedure employed in the study of round tubes [19] for correlating q_{c0}, which is obtained by extrapolating q_c

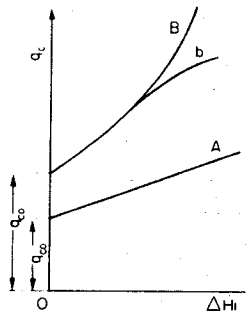

FIG. 2. Relationship between q_c and ΔH_i.

as $\Delta H_i \to 0$, is extended to the case of annuli as follows. Namely, keeping d_i constant in annuli, the limit as $d_0 \to \infty$ is assumed, when there is a possibility of obtaining a correlation of the following form:

$$\frac{q_{c0}}{GH_{fg}} = \text{const.}\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3}. \tag{5}$$

On the other hand, keeping d_i constant, the limit as $d_0 \to d_i$ is assumed, when the correlation equation should take the following form:

$$\frac{q_{c0}}{GH_{fg}} \cdot \frac{l}{d^*} = \text{const.}\left(\frac{\rho_l}{\rho_v}\right)^{a}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{b}, \tag{6}$$

where d^* is a quantity having dimension of diameter.

In many of the existing correlation equations for CHF in annuli, hydraulic equivalent diameter is used either independently or jointly with heated equivalent diameter. However, the hydraulic diameter is primarily a quantity closely related to the friction of viscous flow in channels, so that if CHF is a phenomenon on which the viscosity has only the secondary effect (as assumed in [19] and [20]), the hydraulic diameter cannot be one of the important factors for CHF. On the contrary, the heated equivalent diameter is an indispensable quantity relating to the heat balance of uniformly heated channels [see equation (2)]. Therefore, it may be worthwhile to substitute l/d_{he} in equation (3) for l/d^* in equation (6), and in this case, one can have the following equation which approaches equations (5) and (6) respectively as $d_0 \to \infty$ and $d_0 \to d_i$.

$$\frac{q_{c0}}{GH_{fg}} = \text{const.}\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3} \times \frac{1}{1+\text{const.}\left(\frac{\rho_l}{\rho_v}\right)^{m}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{n}\frac{l}{d_{he}}}. \tag{7}$$

As regards equation (7), it should be noted that when d_i is kept constant in equation (3), l/d_{he} tends to zero in the limit as $d_0 \to \infty$, and l/d_{he} tends to infinity in the limit as $d_0 \to d_i$.

Then, typical examples for the empirical relation between $(\sigma\rho_l/G^2 l)^{1/3}/(q_{c0}/GH_{fg})$ and $\sigma\rho_l/G^2 l$ which is found from the data of q_{c0} for constant ρ_v/ρ_l and l/d_{he} are shown in Fig. 3. It is noticed in Fig. 3 that similarly as in case of round tubes, characteristic regimes can be classified as follows: H-regime where the ordinate value is kept constant, L-regime where the ordinate value increases with $\sigma\rho_l/G^2 l$, and N-regime where the ordinate value decreases with $\sigma\rho_l/G^2 l$, and at the same time, the non-linear $q_c - \Delta H_i$ relationship (a black triangular symbol in Fig. 3 shows the corresponding datum) appears.

3.3. *Analysis of* q_{c0} *data in H-regime*

Consulting equation (7), all the data of q_{c0} in H-regime obtained from [1–6] are correlated as in Fig. 4. The data show scattering somewhat, but almost all the data can be correlated within errors less than $\pm 10\%$ by

$$\frac{q_{c0}}{GH_{fg}} = 0.12\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3} \times \frac{1}{1+0.008\, l/d_{he}}. \tag{8}$$

It is noted that a constant relating to l/d_{he} on the RHS of equation (8) is larger by as much as 2.6 times than that in the H-regime correlation equation for round tubes [equation (A.2) in Appendix].

3.4. *Analysis of* q_{c0} *data in L-regime*

Few data in L-regime can be found in [1–6].

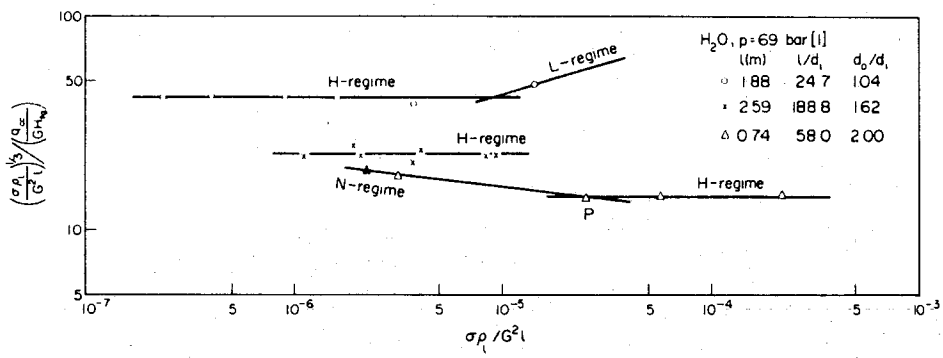

FIG. 3. Characteristics of q_{c0} under the conditions of constant ρ_v/ρ_l and l/d_{he}.

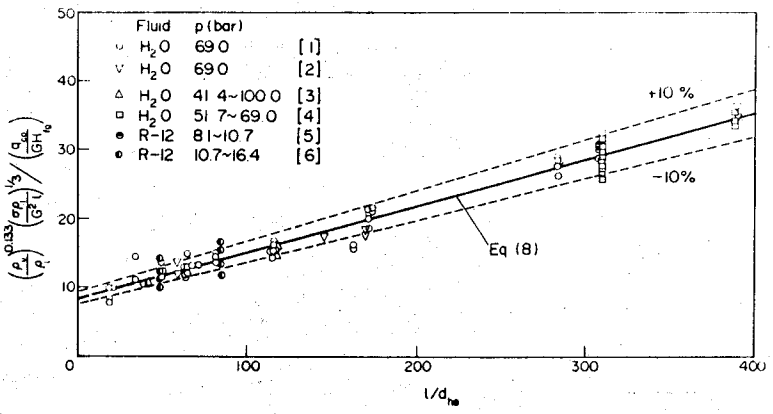

FIG. 4. Nondimensional correlation of q'_{c0} in H-regime.

However, consulting the L-regime correlation equation for round tubes [that is, equation (A.1) in Appendix], and employing l/d_{he} of equation (3) instead of l/d, the data are correlated as in Fig. 5, to give the correlation equation:

$$\frac{q_{c0}}{GH_{fg}} = 0.25\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.043}\frac{1}{l/d_{he}}. \tag{9}$$

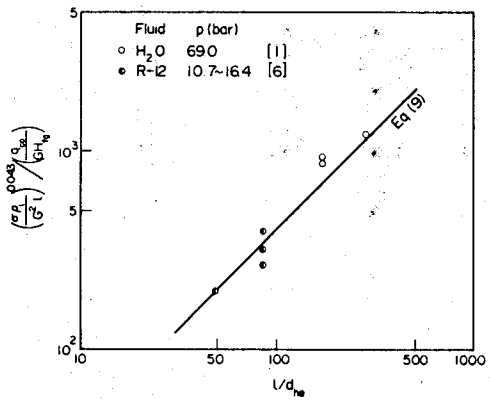

FIG. 5. Nondimensional correlation of q_{c0} in L-regime.

Equation (9) is regarded as a tentative equation due to insufficiency of available data, but having the same form as that of equation (A.1), it may suggest that the deficiency of liquid on the heated surface is responsible for CHF in L-regime similarly as in case of round tubes [20].

3.5. *Analysis of* q_{c0} *data in N-regime*

Strictly speaking, N-regime is the region where the non-linear $q_c - \Delta H_i$ relationship takes place, but as shown by the slant line on the LHS of the point P in Fig. 3, a part of H-regime (that may be called quasi N-regime) shows the same characteristics as those of N-regime for the variation of q_{c0}, so that q_{c0} data in both N-regime and quasi N-regime will be dealt with together. One cannot find sufficiently many data for these regimes in [1–6], but consulting equations (A.2) and (A.3) in Appendix and equation (8), the data are correlated in Fig. 6, to give the correlation equation:

$$\frac{q_{c0}}{GH_{fg}} = 0.22\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.433} \times \frac{(l/d_{he})^{0.171}}{1+0.0081\,l/d_{he}}. \tag{10}$$

It is particularly noticeable that the exponent of l/d_{he} on the RHS of equation (10) is smaller as much as 0.63 times than that in equation (A.3) in Appendix. The data in N-regime show considerable scattering in Fig. 6, but it may be of use to note that similar scattering of data is also observed in N-regime of round tubes (cf. [19]).

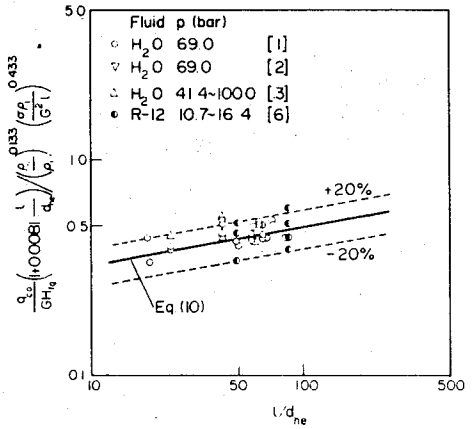

FIG. 6. Nondimensional correlation of q_{c0} in N-regime.

3.6. *HP-regime*

No experiments in HP-regime were made in [1–6], so that the correlation equation of CHF in HP-regime cannot be derived here. Nevertheless, the possibility of HP-regime existing is undeniable, because HP-regime is observed in case of outside heating as shown later in Section 4.3.

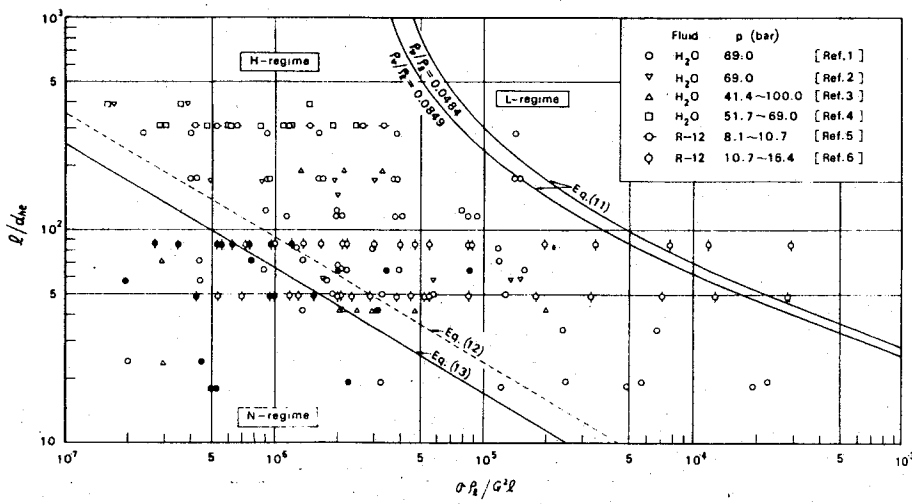

FIG. 7. CHF-regime map for annuli with inside heating.

3.7. *Boundaries of each regime*

By the same means as in case of round tubes, boundaries of each regime are determined. First, the boundary between L- and H-regime is determined by eliminating q_{c0} from equations (8) and (9) as:

$$\frac{l}{d_{he}} = \frac{1}{0.48\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.29} - 0.0081}. \quad (11)$$

Figure 7 shows boundaries predicted by equation (11) for two cases of $\rho_v/\rho_l = 0.0484$ and 0.0849, together with all the data of inside heating which have been so far analyzed in the present paper.

Next, relating to the boundary between H-regime and N-regime, if q_{c0} is eliminated from equations (8) and (10), it yields

$$\frac{l}{d_{he}} = 0.0288\frac{1}{(\sigma\rho_l/G^2 l)^{0.584}} \quad (12)$$

which is represented by the broken line in Fig. 7. However, it is quite apparent that equation (12), corresponding to the point P in Fig. 3, does not indicate the boundary between H- and N-regime. On the other hand, among the data shown in Fig. 7, black symbols represent the data having the non-linear $q_{c0}-\Delta H_i$ relationship (type *B* or *b* in Fig. 2). Since the data of this group are not necessarily sufficient in number, some ambiguities may remain, but if the boundary between H- and N-regime is tentatively determined by the solid, slant line shown in Fig. 7, it yields

$$\frac{l}{d_{he}} = 0.0206\frac{1}{(\sigma\rho_l/G^2 l)^{0.584}}. \quad (13)$$

Equation (13) is the equation which corresponds to equation (A.5) in Appendix for round tubes.

3.8. *Effect of inlet subcooling on CHF*

When liquid is subcooled at the inlet of annuli, critical heat flux q_c in L- and H-regime, where the linear $q_c-\Delta H_i$ relationship holds, can be expressed as follows:

$$q_c = q_{c0}(1 + K\Delta H_i/H_{fg}), \quad (14)$$

where K is a dimensionless parameter independent of ΔH_i.

Utilizing the $q_c-\Delta H_i$ relationship obtained from the experimental data of [1–6], K can be determined, and Table 1 lists K in L-regime (designated by K_L) thus obtained. K_L scatters somewhat, but the mean value is $K_L = 1.18$ (which is nearly the same as $K_L = 1.16$ obtained for round tubes in [19]), and therefore the following approximate expression may be permitted:

$$K_L = 1.0. \quad (15)$$

Table 1. Experimental data of K_L

Fluid	p(bar)	l/d_{he}*	$\sigma\rho_l/G^2 l$	K_L
H_2O	69.0	284.2	1.44×10^{-5}	0.74
H_2O	69.0	173.5	1.48×10^{-5}	1.55
H_2O	69.0	172.5	1.42×10^{-5}	1.08
R-12	10.7	85.5	2.91×10^{-4}	1.34
R-12	10.7	85.5	7.66×10^{-5}	1.52
R-12	16.4	85.5	1.19×10^{-4}	0.91
R-12	10.7	49.4	2.80×10^{-4}	1.11
			Average	1.18

*l/d_{he} is given by equation (3).

Next, Fig. 8 shows K in H-regime (designated by K_H) obtained from the experimental data, being correlated in a similar form as that for round tubes [see equation (A.7) in Appendix], and the straight

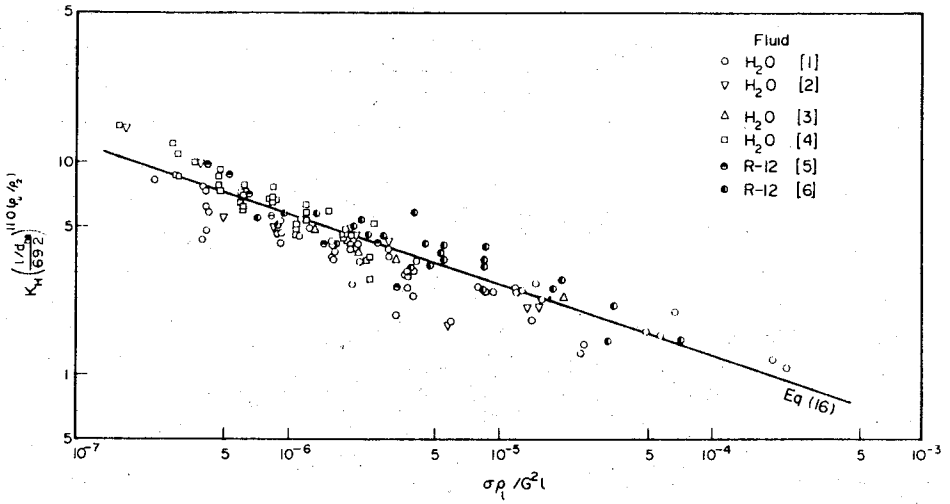

FIG. 8. Correlation of the experimental data of K_{II}.

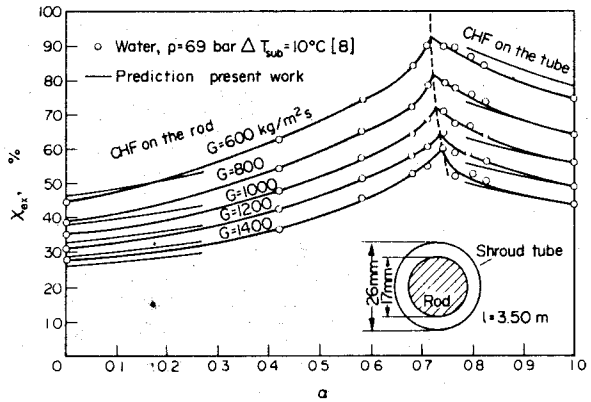

FIG. 9. CHF in bilaterally heated annulus.

line in Fig. 8 is expressed by

$$K_{II} = 0.057\left(\frac{69.2}{l/d_{he}}\right)^{1.1.0(\rho_l/\rho_g)}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{-1/3}. \quad (16)$$

In case of round tubes [19], K_{II} changes its characteristics in the vicinity of $\sigma\rho_l/G^2 l = 3\times10^{-6}$, but no such change is found in case of Fig. 8 for annuli with inside heating.

4. CHF FOR OUTSIDE HEATING

As for CHF in case of outside heating (the rod is unheated and the shroud heated), existing experimental data are few, so that the analysis corresponding to Section 3 cannot be made. Fortunately, however, it seems that only if l/d is replaced by l/d_{he} of equation (4), the correlation equations of CHF for round tubes [equations (A.1)–(A.8) in Appendix] can apply to CHF for annuli with outside heating. Facts illustrative of this rule will be shown below.

4.1. *H-regime*

Figure 9 shows experimental results of exit quality at the state of CHF obtained by Jensen *et al.* [8] for bilaterally heated annuli, where α on the abscissa designates the fraction of total power added to shroud tube, that is

$$\alpha = P_{\text{shroud}}/(P_{\text{rod}} + P_{\text{shroud}}). \quad (17)$$

Therefore, the data for $\alpha = 1$ in Fig. 9 are the data for outside heating, and thereby experimental values of q_c/GH_{fg} for outside heating are readily derived from those data mentioned above through equation (2) [where l/d_{he} is given by equation (4)] to give the result listed in Table 2. On the other hand, according to CHF-regime map given in Fig. 15 of [19], where $C = 0.34$ is now assumed in accordance with the result of [20] for the range of $\sigma\rho_l/G^2 l < 5\times10^{-4}$ (cf. Table 2 in the present text), experiments for $\alpha = 1$ in Fig. 9 are proved to be situated in H-regime.

Table 2. CHF in H-regime for outside heating, corresponding to $\alpha = 1$ in Fig. 9

G (kg/m^{2}s)		600	800	1000	1200	1400
	χ_{ex}	0.746	0.640	0.560	0.488	0.437
	$\sigma\rho_l/G^2l$	1.01×10^{-5}	5.67×10^{-6}	3.63×10^{-6}	2.52×10^{-6}	1.85×10^{-6}
$\frac{q_c}{GH_{fg}}$	Experimental	8.30×10^{-4}	7.17×10^{-4}	6.32×10^{-4}	5.56×10^{-4}	5.01×10^{-4}
	Prediction	8.68×10^{-4}	7.21×10^{-4}	6.25×10^{-4}	5.55×10^{-4}	5.00×10^{-4}

Accordingly, equations (A.2) and (A.7) in Appendix [where l/d is replaced by l/d_{he} of equation (4)] together with equation (14) are used to predict q_c/GH_{fg}, the result of which being listed in the bottom line of Table 2. It is noticed in Table 2 that the agreement between the experimental and the predicted value of q_c/GH_{fg} is very good.

4.2. *N-regime*

Symbols of open circle, black circle and open triangle in Fig. 10 represent experimental data obtained by Ornatskiy *et al.* [11] for outside heating, and it is proved in the similar way as that of the preceding section that the experiments of Fig. 10 belong to N-regime (or quasi N-regime). In addition, as for the data in Fig. 10, it should be noted that most of those data were obtained under conditions of ΔH_i much greater than zero. On the other hand, the symbol x in Fig. 10 represents the prediction of $q_c - \chi_{ex}$ for $\Delta H_i = 0$ by equation (A.3) in Appendix [where l/d is replaced by l/d_{he} of equation (4)] and equation (2). It may be noticed in Fig. 10 that the experimental data for $\Delta H_i < 0$ can be connected to the predicted points for $\Delta H_i = 0$ very smoothly with monotonous curves, and this fact may be regarded to show the applicability of equation (A.3) to predicting CHF in N-regime of outside heating.

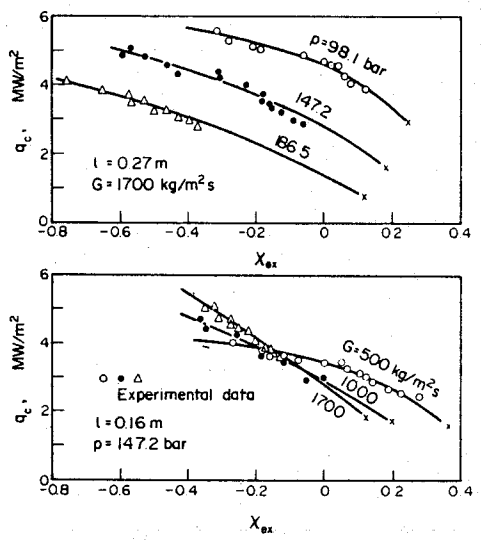

FIG. 10. CHF of water in annuli with outside heating (d_0 = 13 mm and d_i = 10 mm). Comparison between data of [11] and the prediction (x) of equations (A.3) and (2).

4.3. *HP-regime*

Data of water at 216 bars (very near the critical point of water) presented by Ornatskiy *et al.* [9] show the linear $q_c - \Delta H_i$ relationship, and q_{c0} can be obtained by extrapolating q_c as $\Delta H_i \to 0$ to give the data shown in Fig. 11. On the other hand, in the similar way as before, it is proved that the experiments of Fig. 11 are situated thoroughly in HP-regime, and this does not contradict the linear $q_c - \Delta H_i$ relationship mentioned before. Therefore, the prediction of q_{c0}/GH_{fg} by equation (A.4) should be compared with the experimental data in Fig. 11, and it shows a fairly good agreement. For caution's sake, in Fig. 11 are also shown the prediction by equations (A.2) and (A.3) for H- and N-regime respectively, showing noticeable disagreements with the experimental data.

Supplementary note. Ornatskiy *et al.* [9] observed that CHF was not affected by the width of annular channel in their experimental range, while the prediction of equation (A.4) results in a dispersion such as shown by a solid and a broken curve in Fig. 11. In addition, K_{HP} determined from experimental data of Ornatskiy *et al.* has considerably different values from the prediction of equation (A.8). On these discrepancies, further study will be necessary for clarifying causes.

5. CHF FOR BILATERAL HEATING

5.1. *Approximate method for prediction of CHF in L- and H-regime*

When the shroud of an annulus is heated in the ratio of α defined in equation (17), the annular channel may be divided by an imaginary cylindrical surface of diameter d_m as shown in Fig. 12 so as to proportion the cross-sectional area of each sub-channel to the heat load of the wall adjoining the sub-channel, that is

$$(d_0^2 - d_m^2)/(d_m^2 - d_i^2) = P_{\text{shroud}}/P_{\text{rod}}. \tag{18}$$

Then, eliminating $P_{\text{shroud}}/P_{\text{rod}}$ from equations (17) and (18), it yields

$$(d_m/d_i)^2 = \alpha + (1-\alpha)(d_0/d_i)^2. \tag{19}$$

Now, it is boldly assumed that the mass velocity and the axial flow rate of energy are uniform all over the cross-section of annulus, when the transport of mass and energy across the imaginary cylindrical surface can vanish making no alterations to the flow. Then, the inner sub-channel can be regarded as an annulus with inside heating, for which the heated equivalent diameter d_{he} is given from the definition of

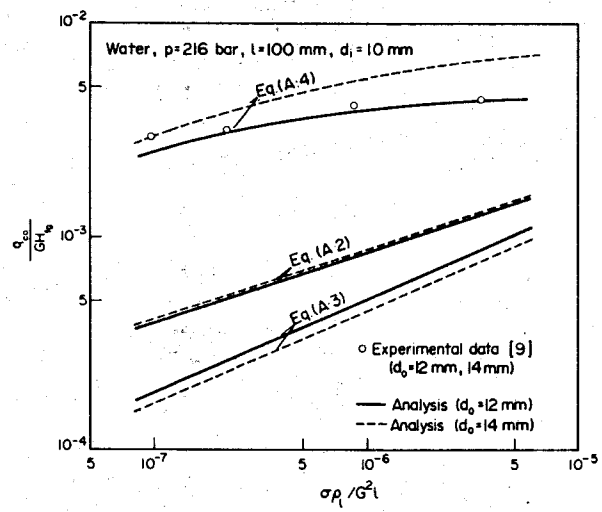

FIG. 11. CHF of water in annuli with outside heating.

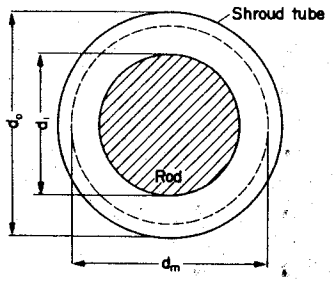

FIG. 12. Division of an annular channel.

equation (1) as $d_{he} = (d_m^2 - d_i^2)/d_i$. Similarly, the outer sub-channel can be regarded as an annulus with outside heating, for which the heated equivalent diameter is given from equation (1) as $d_{he} = (d_0^2 - d_m^2)/d_0$. Consequently, eliminating d_m from these two relations and equation (19), it yields:

for inner sub-channel:

$$\frac{l}{d_{he}} = \frac{1}{1-\alpha} \cdot \frac{l}{d_i} \cdot \frac{1}{(d_0/d_i)^2 - 1}; \tag{20}$$

for outer sub-channel:

$$\frac{l}{d_{he}} = \frac{1}{\alpha} \cdot \frac{l}{d_i} \cdot \frac{d_o/d_i}{(d_0/d_i)^2 - 1}. \tag{21}$$

Equation (20) reduces to equation (3) when $\alpha = 0$, and equation (21) reduces to equation (4) when $\alpha = 1$.

Then, when α is near zero, CHF on the rod may be predicted approximately by applying l/d_{he} of equation (20) to correlation equations (8)–(16). For example, in case of experiments of Fig. 9, they are proved to be situated in H-regime for $\alpha \doteqdot 0$, and therefore χ_{ex} is predicted by equations (2), (8), (14), (16) and (20), yielding thin curves in the range of $\alpha = 0$–0.27 in Fig. 9. Similarly, when α is near unity, CHF on the shroud may be predicted approximately by applying l/d_{he} of equation (21) to correlation equations (A.1)–(A.8) in Appendix. For example, in case of the experiments of Fig. 9, they are proved to be situated in H-regime for $\alpha \doteqdot 1$, and therefore χ_{ex} is predicted by equations (2), (14), (20), (A.2) and (A.7), yielding thin curves in the range of $\alpha = 0.8 - 1.0$ in Fig. 9. Besides, it may be of use to note that similar results as above are also obtained for the experiments of Jensen *et al.* [8] with inlet subcooling temperature $\Delta T_{sub} = 100°C$. In sum, the above-mentioned approximate method for predicting CHF for bilateral heating in the vicinity of $\alpha = 0$ and 1 may be regarded as useful considering its extreme simplicity.

As for the experimental data of Becker *et al.* [7], inlet subcooling enthalpy ΔH_i (or mass velocity G) is not given, so that quantitative comparisons cannot be made. However, their data seem to have the similar trend as that shown in Fig. 9, and moreover, their experimental range extends over L- and H-regime. Therefore, it might be presumed that the above-mentioned method of predicting CHF is effective in L-regime too.

5.2. *Circumstances in HP-regime*

Ornatskiy *et al.* [9], who presented the data shown in Fig. 11, carried out experiments for $\alpha = 0.6$–0.7 at the pressure of 172–216 bars, giving the conclusion that the heating rod had no effect on the CHF on shroud. Tolubinskiy *et al.* [10] also made experiments for $\alpha = 0$–0.5 at the pressure of 150 and 200 bars, giving the conclusion that the heating of shroud had no effect on the CHF on rod. As for the data given by Tolubinskiy *et al.*, those at 150 bars belongs to H-regime, but it can be presumed that the data at 200 bars is situated in HP-regime, because when they are examined by the same method as that

used in Fig. 10, it is found that they belong to neither H-regime nor N-regime. Therefore, according to the above two papers at least, it seems likely that there is no interference between rod and shroud for CHF in HP-regime.

6. CONCLUSIONS

(i) For annuli with inside heating, the existing experimental data of CHF (water and Freon-12) are analyzed to give equations (8)–(10) for the correlation of CHF at zero inlet subcooling, Fig. 7 for the CHF-regime map, and equations (15) and (16) for the effect of inlet subcooling on CHF. Geometrical factors concerned are the heated equivalent diameter of equation (3) and the length of annuli. No correlation in HP-regime is derived for the lack of experimental data.

(ii) For outside heating, it is shown that only if l/d is replaced by l/d_{he} of equation (4), the correlation for round tubes [equations (A.1)–(A.8) in Appendix] applies to the prediction of CHF without any other modifications.

(iii) For bilateral heating, an approximate method is presented for predicting CHF in L- and H-regime in the vicinity of $\alpha = 0$ and $\alpha = 1$. As for CHF in HP-regime, it is suggested that the interference between the rod and shroud may disappear.

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APPENDIX

Correlation equations of CHF derived by the author [19, 20] for round tubes in case of $\Delta H_i = 0$ are as follows:

L-regime:

$$\frac{q_{c0}}{GH_{fg}} = C\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.043}\frac{1}{l/d}, \tag{A.1}$$

where $C = 0.34$ for $\sigma\rho_l/G^2 l < 5\times10^{-4}$ and $C = 0.25$ for $5\times10^{-4} < \sigma\rho_l/G^2 l$.

H-regime:

$$\frac{q_{c0}}{GH_{fg}} = 0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3}\frac{1}{1+0.0031 l/d}. \tag{A.2}$$

H- and N-regime:

$$\frac{q_{c0}}{GH_{fg}} = 0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.433}\frac{(l/d)^{0.27}}{1+0.0031 l/d}. \tag{A.3}$$

HP-regime:

$$\frac{q_{c0}}{GH_{fg}} = 8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.453}\frac{1}{1+107\left(\frac{\sigma\rho_l}{G^2 l}\right)^{0.54}\frac{l}{d}}. \tag{A.4}$$

With respect to equation (A.3), the boundary between H- and N-regime is given by

$$\frac{d}{l} = 0.77\frac{1}{(\sigma\rho_l/G^2 l)^{0.37}}. \tag{A.5}$$

Next, K in equation (14) of the text in case of $\Delta H_i > 0$ is given as follows:

L-regime: $$K_L = 1 \tag{A.6}$$

H-regime:

$$\left.\begin{array}{l}\text{for } \sigma\rho_l/G^2 l < 3\times 10^{-6},\\ \quad K_H = 1.8\left(\dfrac{130}{l/d}\right)^{5.0(\rho_v/\rho_l)}\\ \text{for } 3\times 10^{-6} < \sigma\rho_l/G^2 l,\\ \quad K_H = 0.075\left(\dfrac{130}{l/d}\right)^{5.0(\rho_v/\rho_l)}\left(\dfrac{\sigma\rho_l}{G^2 l}\right)^{-1.4}\end{array}\right\} \quad \text{(A.7)}$$

HP-regime:

$$\left.\begin{array}{l}\text{for } \sigma\rho_l/G^2 l < 4\times 10^{-8},\\ \quad K_{HP} = 0.664(\rho_v/\rho_l)^{-0.6}\\ \text{for } 4\times 10^{-8} < \sigma\rho_l/G^2 l,\\ \quad K_{HP} = 3.08(\sigma\rho_v G^2 l)^{0.09}(\rho_v/\rho_l)^{-0.6}.\end{array}\right\} \quad \text{(A.8)}$$

FORMULATIONS GENERALES POUR LE FLUX THERMIQUE CRITIQUE EN EBULLITION AVEC CONVECTION FORCEE DANS DES ESPACES ANNULAIRES VERTICAUX ET UNIFORMEMENT CHAUFFES

Résumé—Suivant la procédure analytique appliquée par l'auteur au cas du flux thermique critique (CHF) dans les tubes circulaires, on analyse les données expérimentales du CHF dans les espaces annulaires avec chauffage au centre, pour obtenir une formule générale du CHF dans ce type de chauffage. Comme pour l'espace annulaire avec chauffage externe, on montre que le CHF peut être convenablement représenté en utilisant la formation générale de l'auteur concernant le CHF pour les tubes circulaires. Les estimations du CHF dans les deux cas peuvent être établies en employant le diamètre caractéristique de l'anneau. On discute aussi le cas des anneaux chauffés sur les deux frontières et on présente une méthode approchée pour le calcul du CHF dans les régimes bas ou forts, tandis que l'interférence entre les deux parois disparait dans le régime HP.

ALLGEMEINGÜLTIGE BEZIEHUNGEN FÜR DIE KRITISCHE WÄRMESTROMDICHTE BEIM SIEDEN MIT ERZWUNGENER KONVEKTION IN SENKRECHTEN, GLEICHMÄSSIG BEHEIZTEN RINGRÄUMEN

Zusammenfassung—Es wurden vorliegende experimentelle Daten für die kritische Wärmestromdichte (KWD) in Ringräumen mit beheizter Innenseite analysiert, um eine allgemeingültige Beziehung für die KWD bei dieser Art der Beheizung zu erhalten. Dabei wurde ein analytisches Verfahren nach dem gleichen Prinzip angewandt, wie es vom Autor für die Untersuchung der KWD für runde Rohre verwendet wurde. Es wird gezeigt, daß die KWD für Ringräume mit beheizter Außenseite ohne weiteres mit Hilfe der allgemeinen Beziehung des Autors für die KWD runder Rohre berechnet werden kann. Die Berechnung der KWD in den beiden oben erwähnten Fällen kann durchgeführt werden, wenn man den beheizten äquivalenten Durchmesser allein als repräsentativen Durchmesser der Ringräume verwendet. Schließlich wird auch der Fall der beidseitig beheizten Ringräume diskutiert und eine Näherungsmethode für die Berechnung der KWD im L- und H-Bereich angegeben, während beim HP-Bereich zu bedenken ist, daß die Wechselwirkung zwischen den beiden Wänden bezüglich der KWD verschwindet.

ОБОБЩЕННЫЕ ЗАВИСИМОСТИ ДЛЯ КРИТИЧЕСКОГО ТЕПЛОВОГО ПОТОКА ПРИ КИПЕНИИ С ВЫНУЖДЕННОЙ КОНВЕКЦИЕЙ В ВЕРТИКАЛЬНЫХ РАВНОМЕРНО НАГРЕВАЕМЫХ КОЛЬЦЕВЫХ КАНАЛАХ

Аннотация — С помощью того же аналитического метода, который использовался при исследовании критического теплового потока (КТП) для круглых труб, проведен анализ имеющихся экспериментальных данных по КТП в кольцевых каналах с внутренним нагревом и дано обобщенное соотношение для КТП, которое соответствует этому типу нагрева. Для кольцевых каналов с внешним нагревом показано, что КТП можно легко рассчитать с помощью предложенного автором обобщенного соотношения для круглых труб. Для расчёта КТП в обоих случаях достаточно заменить диаметр кольцевых каналов эквивалентным диаметром нагрева. Наконец, рассмотрен двухсторонний нагрев кольцевых каналов и приведен приближенный метод расчёта КТП для режимов L и H в предположении, что взаимное влияние стенок несущественно для КТП в режиме HP.

GENERAL FEATURES OF CHF OF FORCED CONVECTION BOILING IN VERTICAL CONCENTRIC ANNULI WITH A UNIFORMLY HEATED ROD AND ZERO INLET SUBCOOLING

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Abstract—A graphical method is employed to give a generalized bird's-eye view of the existing data for critical heat flux (CHF) in internally and uniformly heated vertical annuli with zero inlet subcooling. 301 data points collected from 25 sources are used for this purpose, including 7 different fluids (water, R-12, R-114, acetone, toluene, monoisopropyl-biphenyl, and sodium), axial length of heated rod from 0.0762 to 8.84 m, outer diameter of heated rod from 0.00500 to 0.0964 m, inner diameter of unheated shroud tube from 0.0127 to 0.101 m, and vapor/liquid density ratio from 0.0000580 to 0.160.

NOMENCLATURE

- d_{he}, heated equivalent diameter [m], equation (3) for inside uniform heating;
- d_i, O.D. of heated rod [m];
- d_o, I.D. of unheated shroud tube [m];
- G, mass velocity [$kgm^{-2}s^{-1}$];
- H_{fg}, latent heat of evaporation [Jkg^{-1}];
- ΔH_i, enthalpy of inlet subcooling [Jkg^{-1}];
- l, axial length of heated rod [m];
- p, absolute pressure [bar];
- q_c, critical heat flux [Wm^{-2}];
- q_{co}, q_c for $\Delta H_i = 0$ [Wm^{-2}];
- ΔT_i, inlet subcooling temperature [°C].

Greek symbols

- ρ_l, density of liquid [kgm^{-3}];
- ρ_v, density of vapor [kgm^{-3}];
- σ, surface tension [Nm^{-1}];
- χ, quality (χ_{ex}: exit quality, χ_{in}: inlet quality).

1. INTRODUCTION

RECENTLY, the author made studies [1–3] of generalized correlation equations for critical heat flux (CHF) of forced convection boiling in uniformly heated vertical tubes, and it was followed by a study [4] in which a graphical method was evolved to give a generalized bird's-eye view of the existing data of CHF in case of zero inlet subcooling.

On the other hand, existing data of CHF in uniformly heated vertical annuli were also analyzed by the author [5], revealing that the data of CHF in annuli with inside heating are correlated by a set of proper correlation equations, while CHF in annuli with outside heating can be correlated by making use of the above-mentioned correlation equations of CHF in vertical tubes. Then, following the similar way as in the case of tubes, an additional study is attempted in the present paper to give a generalized graphic representation of the existing data of CHF in internally heated annuli with zero inlet subcooling.†

2. COLLECTION OF q_{co} DATA

2.1. *Method of obtaining* q_{co} *data*

From the sources [6–31] listed in Table 1, the data of q_{co} are obtained for pulsation-free upflow, mostly by the following normal methods (i) and (ii), and for the rest, by two exceptional methods (iii) and (iv).

(i) As for the data-source providing the variation of q_c with ΔH_i (or ΔT_i, χ_{in} etc.) for fixed G such as shown in Figs 1 and 2, if there are enough data points up to the vicinity of $\Delta H_i = 0$, q_{co} can be estimated by the extrapolation as ΔH_i (or ΔT_i, χ_{in} etc.) $\rightarrow 0$. Knoebel *et al.* [19], who presented the data of q_c including those of Fig. 2, gave the following empirical equation to correlate their data of q_c in the range of $\Delta T_i > 25$°C (that is, excluding data below 25°C subcooling from the correlation):

$$q_c[Wm^{-2}] = 4.844 \times 10^5(1 + 0.169V[ms^{-1}])(1 + 0.12\Delta T_i[°C]) \quad (1)$$

where V is the inlet velocity of water; and the broken line shown in Fig. 2 represents equation (1). However, the author's correlation of CHF of forced convection boiling (cf. [1, 2] for tubes and [5] for annuli) regards the result of Fig. 2 only as belonging to a characteristic regime called N-regime, which is distinguished by non-linear q_c–ΔH_i relationship, and q_{co} is determined by the conventional extrapolation as $\Delta T_i \rightarrow 0$ in Fig. 2.

(ii) As for the data-source providing the variation of q_c with χ_{ex} for fixed G, the relation of q_c vs ΔH_i can be derived from the following heat balance equation for

† The effect of inlet subcooling on CHF can be estimated theoretically in the same way as that shown for CHF in vertical tubes [3], so that the discussion on the effect of inlet subcooling is omitted.

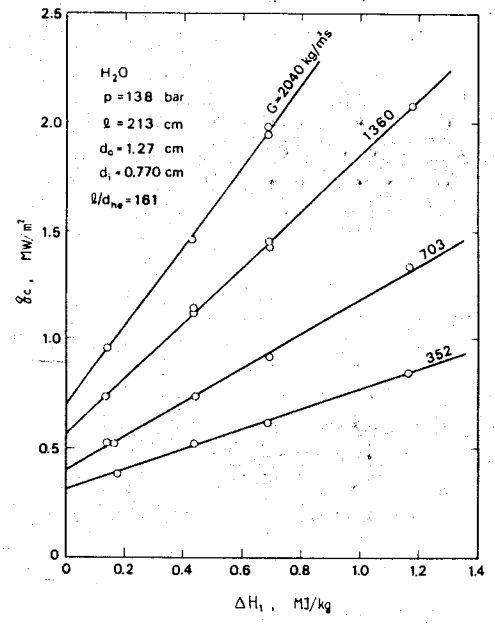

FIG. 1. Examples of the linear relationship between q_c and ΔH_i (data from Table A of Mortimore and Beus [25]).

uniformly heated annuli:

$$\frac{4q_c}{GH_{fg}} \cdot \frac{l}{d_{he}} - \frac{\Delta H_i}{H_{fg}} = \chi_{ex} \qquad (2)$$

where d_{he}, the heated equivalent diameter, is given for annuli with inside heating as follows (cf. [5]):

$$\frac{l}{d_{he}} = \frac{l}{d_i} \cdot \frac{1}{(d_o/d_i)^2 - 1}. \qquad (3)$$

Therefore, the method (i) applies to the relation of q_c vs ΔH_i thus obtained.

(iii) Data-sources [8, 10, 38] present data in the form of (ii), but the data are concerned with the uniform heat flux experiment with mixed-inlet-condition (that is, $\Delta H_i < 0$ and $\chi_{in} > 0$). In this case, if the imaginary length l, determined as $l = l'/[1 - (\chi_{in}/\chi_{ex})]$ for the actual length l' used in the experiment, is assumed, the mixed-inlet-condition of $\Delta H_i < 0$ can be formally transformed to that of $\Delta H_i = 0$. Then, according to the author's study [3] of CHF in tubes, the experimental data of CHF for $\Delta H_i < 0$ can be approximately correlated with the imaginary length l mentioned above if exit quality is in the vicinity of unity. Therefore, taking this fact into consideration in the present paper too, 5 data (Run No. 691–695) with $\chi_{ex} > 0.75$ are adopted from Table II of [8], and 13 data (Run No. 2582–2588 and 2599–2604) with $\chi_{ex} > 0.75$ are adopted from Table 2 of [10], but no data are adopted from [37] because all data are of $\chi_{ex} < 0.75$.

(iv) Data-sources [22–24] give the data for independent sets of G, ΔH_i and q_c. Therefore, q_{co} is estimated, through empirical equations (14) and (16) of the preceding paper [5], from the data at $\Delta T_i = 10°C$, for which $\Delta H_i/H_{fg}$ is restricted to small values so that serious error cannot arise for the estimation of q_{co}.

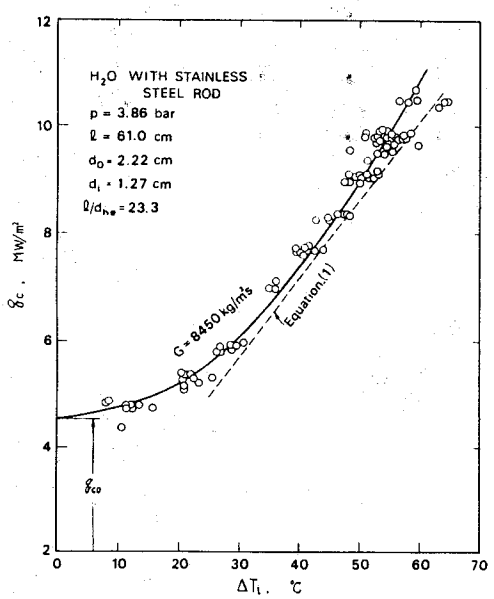

FIG. 2. An example of the non-linear relationship between q_c and ΔH_i (data from Table B-1 of Knoebel *et al.* [19] excluding the data for heaters bowed or with flaws etc.).

2.2. *Note on the data*

[11] reports two groups of experiments carried out at atmospheric pressure and at 59.9 bar respectively with different apparatus, and the data at atmospheric pressure show quite irregular natures, presumably due to the lack of throttling measures to prevent pulsation flows, so that only the data at 59.9 bar are employed. For the experiment of [17], the rod was heated indirectly with a coiled ribbon electric heater set up in the rod, accordingly there is the possibility of some error not only in the uniformity of heat flux distribution but also in the axial length of heated surface of rod. As for experiments reported in [19] for heavy water and aluminum rods, no data were obtained for CHF in the vicinity of $\Delta T_i = 0$ so that q_{co} cannot be determined, and consequently only the data for light water on a stainless steel rod such as exemplified in Fig. 2 are employed. Experimental data of [21] are given with very small figures so that there is the possibility of low accuracies for q_{co} derived from these data. The data for sodium presented in [31] show a considerable scattering, and the data obtained at the pressure of 0.15 bar for the flow rate of 0.0567 kg s^{-1} alone are employed in this study because of comparatively good order. The data of CHF in annuli with inside heating given in [38] cannot be utilized because of lacking the data of mass velocity G.

Table 1. Summary of the collected data of q_{co}

Source	Fluid	l (cm)	d_o (cm)	d_i (cm)	l/d_{he}	$(\rho_v/\rho_l) \times 10^2$	$\sigma\rho_l/G^2l$	No. of data
G.E. [6]	Water	274	2.22	0.952	64.8	4.84	8.93×10^{-7}–8.04×10^{-6}	4
Blackford–Matzner [7]	Water	107	4.43	3.49	50.0	4.84	1.78×10^{-6}–1.24×10^{-5}	4
Bennett *et al.* [8]	Water	471–649	1.42	0.952	402–554	2.13	7.24×10^{-6}–9.96×10^{-6}	5
Janssen–Kervinen [9]	Water	91.4–177	2.22–2.54	0.952–1.27	24.0–42.0	2.61–7.60	2.95×10^{-7}–2.00×10^{-5}	3
Bennett *et al.* [10]	Water	459–884	1.58	0.952	271–522	2.13–4.84	3.85×10^{-6}–6.61×10^{-6}	13
JSME [11]	Water	30.0	1.90	1.00	11.5	4.05	2.05×10^{-6}–3.28×10^{-5}	4
Barnett [12, 13]	Water	73.6–274	1.40–10.1	0.952–9.64	18.4–287	2.61–8.04	1.97×10^{-7}–2.25×10^{-4}	82
Moeck *et al.* [14]	Water	188	7.95–10.1	7.63–9.64	172–287	4.84	4.07×10^{-7}–1.46×10^{-5}	20
Barnett [15]	Water	182–457	2.09–2.67	1.48–1.54	59.0–387	4.84	1.71×10^{-7}–1.51×10^{-5}	11
Little [16]	Water	365–457	2.09	1.58	310–387	3.39–4.84	1.60×10^{-7}–2.46×10^{-6}	40
Moeck [17]	Water	131	2.37	1.97	146	2.11–4.84	5.27×10^{-6}–9.07×10^{-6}	2
Tolubinskiy *et al.* [18]	Water	22.0	1.30	0.500	7.64–11.5	10.3	4.50×10^{-4}–6.75×10^{-4}	2
Knoebel *et al.* [19]	Water	61.0	2.22	1.27	23.3	0.123–0.224	4.59×10^{-7}–1.12×10^{-6}	3
Tolubinskiy *et al.* [20]	Water	10.0	1.30	1.00	14.5	16.0	1.23×10^{-4}	1
Tolubinskiy *et al.* [21]	Water	10.0–100	1.30	0.900	10.2–102	16.0	3.07×10^{-6}–3.07×10^{-5}	7
Andersen *et al.* [22]	Water	350	2.72	1.70	132	4.94	1.55×10^{-6}–9.74×10^{-6}	10
Jensen–Manov [23]	Water	350	2.60	1.70	154	4.84	1.85×10^{-6}–1.01×10^{-5}	5
Becker–Letzter [24]	Water	300	2.13	1.20	116	1.83–4.94	1.90×10^{-6}–3.04×10^{-5}	5
Mortimore–Beus [25]	Water	213	1.27	0.770	161	6.16–13.5	2.61×10^{-7}–4.33×10^{-5}	12
Stevens *et al.* [26]	R-12	365	2.09	1.58	310	3.60–4.82	4.22×10^{-7}–3.31×10^{-6}	8
Ahmad–Groeneveld [27]	R-12	182	2.22	1.04–1.39	49.4–85.5	4.82–8.49	2.67×10^{-7}–7.07×10^{-4}	48
Shiralkar [28]	R-114	183	2.22	1.43	90.3–180	4.81	4.36×10^{-6}–8.72×10^{-6}	2
Andrews *et al.* [29]	Acetone	7.62	2.09	0.604	1.15	0.290	3.13×10^{-5}	1
Andrews *et al.* [29]	Toluene	7.62	2.09	0.604	1.15	0.385	2.90×10^{-5}	1
Sterman *et al.* [30]	MIPB†	11.0	1.60	1.00	7.05	1.03	2.50×10^{-6}–1.60×10^{-4}	4
Noyes *et al.* [31]	Sodium	68.6	1.27	0.635	36.0	0.00580–0.0211	4.22×10^{-4}–7.64×10^{-4}	4
								Total 301

† Monoisopropylbiphenyl.

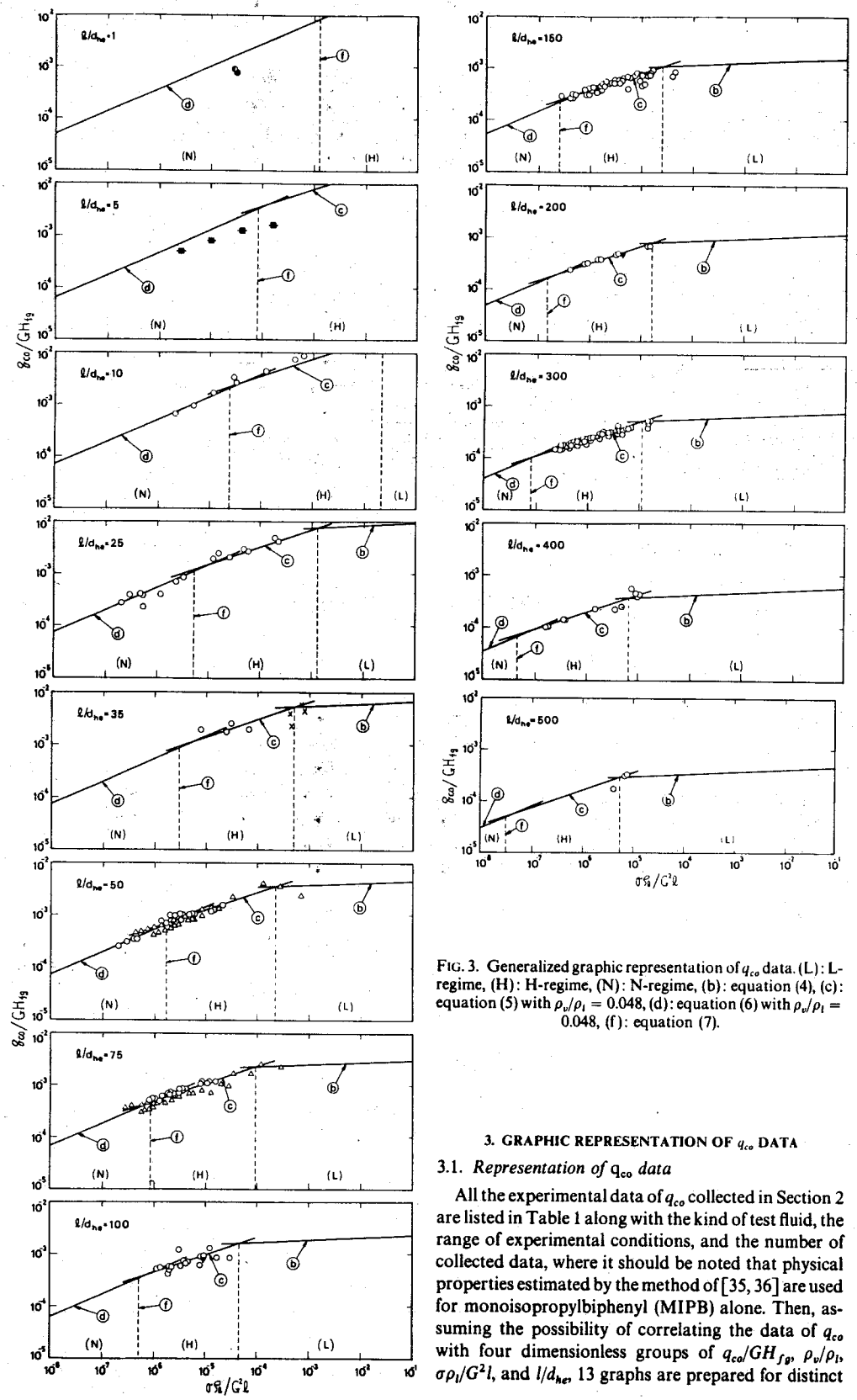

FIG. 3. Generalized graphic representation of q_{co} data. (L): L-regime, (H): H-regime, (N): N-regime, (b): equation (4), (c): equation (5) with $\rho_v/\rho_l = 0.048$, (d): equation (6) with $\rho_v/\rho_l = 0.048$, (f): equation (7).

3. GRAPHIC REPRESENTATION OF q_{co} DATA

3.1. *Representation of* q_{co} *data*

All the experimental data of q_{co} collected in Section 2 are listed in Table 1 along with the kind of test fluid, the range of experimental conditions, and the number of collected data, where it should be noted that physical properties estimated by the method of [35, 36] are used for monoisopropylbiphenyl (MIPB) alone. Then, assuming the possibility of correlating the data of q_{co} with four dimensionless groups of q_{co}/GH_{fg}, ρ_v/ρ_l, $\sigma\rho_l/G^2l$, and l/d_{he}, 13 graphs are prepared for distinct

Table 2. Symbols used to specify the fluids in Fig. 3

Ref. No.	Fluid	Symbol	Ref. No.	Fluid	Symbol
6–25	Water	○	29	Toluene	◑
26, 27	R-12	△	30	MIPB	-■-
28	R-114	▼	31	Sodium	×
29	Acetone	●			

values of l/d_{he} as shown in Fig. 3 and the data of q_{co}/GH_{fg} classified to each graph depending on l/d_{he} are plotted against $\sigma\rho_l/G^2l$ (see Tables 3 and 4 for the division of data by l/d_{he} to each graph and see Section 3.3 for the treatment of the effect of ρ_v/ρ_l). Data symbols shown in Table 2 are used in Fig. 3 to discriminate the kind of fluid.

3.2. *Representation of the author's correlation equations*

Based on the results of the author's study [5], the characteristic regimes L, H and N (cf. [1] for the origin of these names), the correlation lines (b)–(d), and the boundary (f) between H- and N-regime are shown in Fig. 3 applying the following equations:

L-regime,

$$\text{(b)}: \quad \frac{q_{co}}{GH_{fg}} = 0.25\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d_{he}} \tag{4}$$

H- and N-regime,

$$\text{(c)}: \quad \frac{q_{co}}{GH_{fg}} = 0.12\left(\frac{\rho_v}{\rho_l}\right)^{0.133} \times \left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\frac{1}{1+0.0081\,l/d_{he}} \tag{5}$$

where $\rho_v/\rho_l = 0.048$ for (c) in Fig. 3.

$$\text{(d)}: \quad \frac{q_{co}}{GH_{fg}} = 0.22\left(\frac{\rho_v}{\rho_l}\right)^{0.133} \times \left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}\frac{(l/d_{he})^{0.171}}{1+0.0081\,l/d_{he}} \tag{6}$$

where $\rho_v/\rho_l = 0.048$ for (d) in Fig. 3.

Boundary between H- and N-regime,

$$\text{(f)}: \quad \frac{\sigma\rho_l}{G^2l} = \left(\frac{0.0206}{l/d_{he}}\right)^{1.71}. \tag{7}$$

Among the above equations, equation (4) for L-regime is independent of ρ_v/ρ_l, it can be represented by a line (b) in each graph of Fig. 3. On the other hand, equations (5) and (6) for H- and N-regime are subject to the influence of ρ_v/ρ_l, so that the prediction of these equations is given by lines (c) and (d) in Fig. 3 only for the case of $\rho_v/\rho_l = 0.048$, which corresponds to the density ratio of saturated steam and water at 68.5 bar.

3.3. *Effects of* ρ_v/ρ_l

Figure 3 shows that almost all the data points of q_{co} collected in this paper distribute within H- and N-regime. According to equations (5) and (6), the effect of ρ_v/ρ_l on CHF in H- and N-regime may be presumed to be in the mere degree of $(\rho_v/\rho_l)^{0.133}$. Therefore, the data of q_{co} listed in Table 1 are divided into two groups of Tables 3 and 4 depending on the magnitude of ρ_v/ρ_l. As for the data listed in Table 3 with the range of ρ_v/ρ_l between 0.0103 and 0.160, ρ_v/ρ_l is not far apart from 0.048 so that experimental data of q_{co} have been plotted in Fig. 3 without any artificial modification of data. However, as for 9 data listed in Table 4 with the

Table 3. Experimental range of l/d_{he} and ρ_v/ρ_l for the data of Fig. 3, excluding the data of Table 4

Nominal l/d_{he}	$\frac{l}{d_{he}}$	$\frac{\rho_v}{\rho_l}\times 10^2$	Ref. No.
5	7.05	1.03	30
10	7.64–14.5	4.05–16.0	11, 18, 20, 21
25	18.4–26.8	4.84–16.0	9, 12, 21
35	33.8–40.9	4.84–16.0	12, 21
50	42.0–59.0	2.61–8.49	7, 9, 12, 13, 15, 27
75	64.6–85.5	4.82–8.49	7, 12, 27
100	90.3–116	1.83–16.0	12, 13, 21, 28
150	123–172	4.84–13.5	12, 14, 15, 17, 22, 23, 25
200	173–180	4.84	12, 14, 28
300	283–348	2.13–4.84	10, 12, 14, 16, 26
400	316–443	2.13–4.84	6, 8, 10, 15, 16
500	509–553	2.13	8, 10

Table 4. Data with comparatively low values of ρ_v/ρ_l in Fig. 3

Nominal l/d_{he}	$\frac{l}{d_{he}}$	$\frac{\rho_v}{\rho_l} \times 10^2$	No. of data	Ref. No.
1	1.15	0.290–0.385	2	[29]
25	23.3	0.123–0.224	3	[19]
35	36.0	0.00580 0.0211	4	[31]
			Total 9	

range of ρ_v/ρ_l between 0.0000580 and 0.00385, ρ_v/ρ_l is remarkably lower than 0.048, so that the correction of multiplying $[0.048/(\rho_v/\rho_l)]^{0.133}$ to q_{co}/GH_{fg} has been made to be plotted in Fig. 3.

4. DISCUSSION OF THE RESULT OF FIG. 3 WITH CONCLUDING REMARKS

Dealing with CHF of forced convection boiling in internally and uniformly heated annuli with zero inlet subcooling, a graphical representation of 301 data points listed in Table 1 is made to give the result of Fig. 3. Examination of Fig. 3 breeds the following remarks.

(i) It is of interest to note that a number of q_{co} data obtained from various independent sources [6–31] indicate such a regular nature as shown in Fig. 3 for a wide range of l/d_{he}. In addition, these data show a fairly good agreement with the author's correlation equation (5) and (6) in the range of $l/d_{he} = 10$ to 500, but deviation seems to appear when l/d_{he} decreases to the degree of 1–5. On this point, it may not be useless to note that when l/d_{he} is extremely small, boiling is close to that on a heated rod placed in a uniform liquid flow rather than in an annular channel.

(ii) Most data in the range of $l/d_{he} = 10$–500 are those of water, and the other test fluids are limited to R-12 (58 data), R-114 (only 2 data) and sodium (only 4 data). Therefore, further experiments should be made in the future with various fluids other than water.

(iii) It is noted that there are almost no data in L-regime except the vicinity of the boundary between L- and H-regime in Fig. 3, and the reason why data lacks for L-regime is unknown. Such being the case, the correlation equation (4) for L-regime must be regarded as a tentative equation at present.

(iv) In cases of $l/d_{he} = 50$ and 75 in Fig. 3, a statistical trend is observed that the data of R-12 appear slightly lower than those of water in H- and N-regime.

(v) Critical heat flux in HP-regime, such as found for forced convection boiling in tubes or in annuli with outside heating [1–5], should be studied for internally heated annuli too.

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CONFIGURATIONS GENERALES DE LA CONVECTION FORCEE AVEC EBULLITION AU FLUX CRITIQUE, DANS UN ESPACE ANNULAIRE VERTICAL AUTOUR D'UNE BARRE CHAUFFEE UNIFORMEMENT ET AVEC UN SOUS-REFROIDISSEMENT NUL A L'ENTREE

Résumé—On emploie une méthode graphique pour donner une vue d'ensemble des données existantes pour le flux de chaleur critique (CHF) dans des espaces annulaires, verticaux, uniformément chauffés intérieurement et avec un sous-refroidissement nul à l'entrée. 301 données sont tirées de 25 sources et ils concernent 7 fluides différents (eau, R-12, R-114, acétone, toluène, monoisopropylbiphényl et sodium), des longueurs chauffées de 0,0762–8,84 m, des diamètres externes de barre chauffée entre 0,005 et 0,0964 m, des diamètres internes de tube enveloppant entre 0,0127 et 0,101 m, et des rapports vapeur/liquide depuis 0,000058 jusqu'à 0,160.

ALLGEMEINE MERKMALE DER KRITISCHEN WÄRMESTROMDICHTE (KWD) BEIM STRÖMUNGSSIEDEN IN SENKRECHTEN KONZENTRISCHEN RINGSPALTEN MIT GLEICHMÄSSIG BEHEIZTEM STAB OHNE UNTERKÜHLUNG AM EINTRITT

Zusammenfassung — Um einen allgemeinen Überblick über die vorhandenen Daten für die kritische Wärmestromdichte in von innen gleichmäßig beheizten senkrechten Ringspalten ohne Unterkühlung am Eintritt zu erhalten, wurde eine grafische Methode angewandt. Aus 25 Quellen wurden 301 Angaben zu diesem Zweck zusammengetragen. Diese umfassen: sieben verschiedene Fluide (Wasser, R12, R114, Aceton, Toluen, Monoisopropylbiphenyl und Natrium), axiale Längen des beheizten Stabes von 0,0762 bis 8,84 m, äußere Durchmesser des beheizten Stabes von 0,005 bis 0,0964 m, innere Durchmesser des äußeren Mantelrohres von 0,0127 bis 0,101 m und Dampf/Flüssigkeits-Verhältnisse von 0,000058 bis 0,16.

О КРИТИЧЕСКОМ ТЕПЛОВОМ ПОТОКЕ ПРИ КИПЕНИИ В УСЛОВИЯХ ВЫНУЖДЕННОЙ КОНВЕКЦИИ В ВЕРТИКАЛЬНЫХ КОНЦЕНТРИЧЕСКИХ КАНАЛАХ С РАВНОМЕРНО НАГРЕВАЕМЫМ СЕРДЕЧНИКОМ БЕЗ НЕДОГРЕВА НА ВХОДЕ

Аннотация Используя графический метод, проведен обзор имеющихся данных по критическому тепловому потоку в равномерно нагреваемых изнутри кольцевых каналах при отсутствии недогрева на входе. С этой целью была взята 301 точка из 25 публикаций для 7 жидкостей (вода, R-12, R-114, ацетон, толуол, моноизопропил-бифенил, натрий). Длина нагреваемого сердечника по оси составляла 0,0762 8,84 м, наружный диаметр сердечника изменялся от 0,00500 до 0,0964 м, внутренний диаметр ненагреваемой трубы изменялся от 0,0127 до 0,101 м, а отношение содержаний пара и жидкости изменялось в пределах от 0,0000580 до 0,160.

GENERAL FEATURES OF CHF OF FORCED CONVECTION BOILING IN UNIFORMLY HEATED RECTANGULAR CHANNELS

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Abstract—For the critical heat flux (CHF) of forced convection boiling in uniformly heated rectangular channels fed with the subcooled liquid, existing experimental data are analyzed to derive a generalized correlation of the data on the same principle as in the author's preceding studies on CHF of forced convection boiling in tubes and annuli. In addition, experiments made under special conditions of very small l/d_{he}, where l is the heated length of channel and d_{he} the heated equivalent diameter, are also investigated to confirm that the CHF under such conditions is nearly equal to the CHF of forced convection boiling on heated plane surfaces in a parallel flow.

NOMENCLATURE

- b, width of rectangular cross-section [m];
- C, constant, equation (3);
- d_{he}, heated equivalent diameter [m];
- G, mass velocity [kg m^{-2} s^{-1}];
- H_{fg}, latent heat of evaporation [J kg^{-1}];
- ΔH_i, inlet subcooling enthalpy [J kg^{-1}];
- K, inlet subcooling parameter, equation (6);
- l, length of heated plate [m];
- q_c, critical heat flux [W m^{-2}];
- q_{c0}, basic critical heat flux (q_c for $\Delta H_i = 0$) [W m^{-2}];
- s, height of rectangular cross section [m], defined as s = real height for type A, and s = twice real height for type B in Fig. 1;
- w, width of heated plate [m].

Greek symbols

- δ, thickness of heated plate [m];
- ρ_l, density of liquid [kg m^{-3}];
- ρ_v, density of vapor [kg m^{-3}];
- σ, surface tension [N m^{-1}].

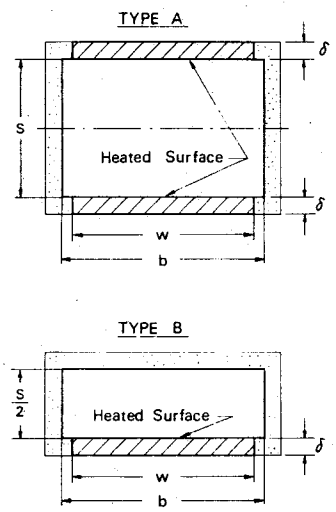

FIG. 1. Two types of rectangular cross-sections.

1. INTRODUCTION

FOR CRITICAL heat flux (CHF) in uniformly heated rectangular channels, Macbeth [1] presented an empirical correlation of water data, while many other studies assumed that the correlation proposed for CHF in round tubes could be used to predict CHF in rectangular channels. Recently, I have made studies of CHF in round tubes [2–7] and annuli [8, 9] to find a method capable of outlining the general framework of the above mentioned CHF. In this paper, therefore, the existing data of CHF in uniformly heated rectangular channels are analyzed to ascertain the general validity of the above method.

For rectangular channels with four side walls, overheating takes place in the corner of the cross-section near the exit end of the channel when all the side walls are heated. Accordingly this paper deals with the case of Fig. 1, where two facing side walls are equally heated in type A, while one side wall alone is heated in type B. The width of heated plate w is designed, in many cases, to be slightly shorter than that of rectangular cross-section b to avoid the danger of overheating in the corner. As for the height of rectangular cross-section, if the value of s is defined as shown in Fig. 1 for types A and B respectively, then the heated equivalent diameter d_{he} = (4 × flow area)/(heated perimeter) is given by the following equation in common for both type A and type B

$$d_{he} = 2sb/w. \tag{1}$$

2. CRITICAL HEAT FLUX

2.1. *Data of* q_{c0}

Figure 2 shows examples of the relationship between

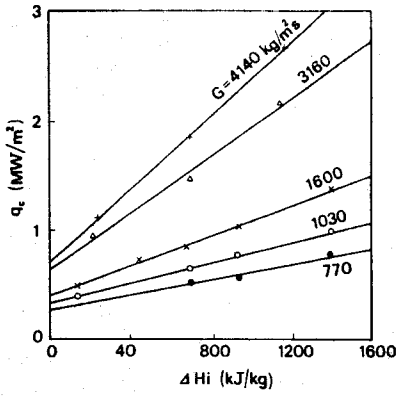

FIG. 2. Relationship between q_c and ΔH_i from the data of Troy [12]: water, $p = 138$ bar, $l = 190$ cm, $s = 0.190$ cm, $b = w = 5.71$ cm, and type A.

the critical heat flux q_c and the inlet subcooling enthalpy ΔH_i from the experimental data of Troy [12]. The basic critical heat flux q_{c0}, that is the value of q_c at $\Delta H_i = 0$, is readily determined from the relation such as shown in Fig. 2, and Table 1 lists the data sources and the experimental conditions for q_{c0} thus obtained. Among the data sources, however, Gambill and Bundy [13] give a data point of q_c for each set of separate values of G and ΔH_i, so that q_{c0} cannot be determined by the above method (see Section 2.4 for the indirect method to determine q_{c0}). The data of Tong, Efferding and Bishop [15] are also in the same form as above, but ΔH_i is kept near zero so that q_c can approximate to q_{c0}. In the experiment of Tippets [14], the thickness of heated plate δ is extremely small as 0.245 and 0.152 mm, accordingly the data for $\delta = 0.254$ mm alone are adopted in Table 1 to avoid the effect of δ on CHF as much as possible. All the experimental studies listed in Table 1 are those made for vertical upward boiling flow except the experiment of Gambill and Bundy which was made for vertical downward boiling flow. The studies of Levy, Fuller and Niemi [16], Chernobay [17], and others based on experiments of the whole perimeter heating are excepted from Table 1.

2.2. *Generalized correlation of* q_{c0}

The data of q_{c0} obtained in the preceding section are classified by nominal values of l/d_{he}, and then plotted in Fig. 3 in the same dimensionless form as for tubes [5] and annuli [9]. According to Table 1, ρ_v/ρ_l takes magnitudes on the order of the value of saturated water at 69 bar (that is, $\rho_v/\rho_l = 0.048$), except the data of Tong, Efferding and Bishop [15], for which ρ_v/ρ_l is very small. Therefore, in the same manner as in the cases of tubes [5] and annuli [9], the data of [15] are corrected by multiplying q_{c0} by $[0.048/(\rho_v/\rho_l)]^{0.133}$ when plotted in Fig. 3.

Meanwhile, the straight lines (a), (b), (c) and (d) in Fig. 3 represent the following equations, respectively

Table 1. Summary of the collected data of q_{c0} and K

Source	Fluid	Type	l(cm)	s(cm)	b(cm)	w/b	l/d_{he}	$(\rho_v/\rho_l)\times 10^2$	$\sigma\rho_l/G^2l$	No. of data	δ(mm)
[11]	Water	A	15.2	0.257	2.54	0.880	26.1	3.67–13.6	2.80×10^{-5}–5.61×10^{-3}	10	1.27
[11]	Water	A	30.6	0.246	2.54	0.880	54.7	13.6	9.43×10^{-7}–3.11×10^{-4}	8	1.27
[11]	Water	A	30.6	0.127	2.54	0.880	106	2.61–13.6	3.06×10^{-6}–2.12×10^{-4}	8	1.27
[11]	Water	A	68.6	0.246	2.54	0.880	122	3.67–13.6	3.00×10^{-7}–8.53×10^{-5}	10	1.27
[11]	Water	A	68.6	0.150	2.54	0.880	201	3.67–13.6	3.02×10^{-6}–3.46×10^{-4}	7	1.27
[12]	Water	A	190	0.190–0.195	5.71	~1.0	487–500	6.16–13.6	1.24×10^{-7}–1.99×10^{-5}	9	—
[13]	Water	A	30.5–30.8	0.109–0.125	2.65–2.73	0.930–0.958	114–132	1.96–2.42	7.45×10^{-7}–1.03×10^{-6}	3*	1.52
[14]	Water	A	94.0	0.635	5.33	1.00	74.0	4.84	3.65×10^{-6}–5.97×10^{-5}	3	0.254
[14]	Water	A	94.0	1.27	5.33	1.00	37.0	4.84	1.43×10^{-5}–5.55×10^{-5}	2	0.254
[14]	Water	B	94.0	2.54	5.33	1.00	18.5	4.84	1.41×10^{-5}	1	0.254
[15]	R-113	B	63.5	1.27	6.03	~1.0	25.0	0.587–0.719	6.97×10^{-6}–1.28×10^{-4}	3	∞†

* Data for nickel test section.
† Very thick heated plate.

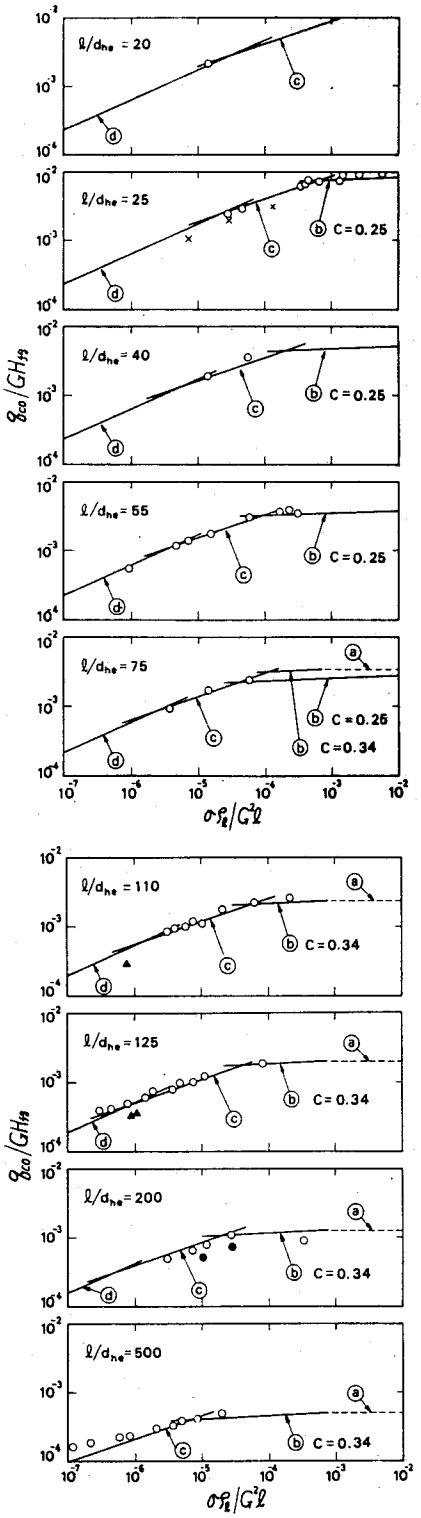

FIG. 3. Generalized graphic representation of q_{c0} data. (a) equation (2), (b) equation (3), (c) equation (4) with ρ_v/ρ_l = 0.048, (d) equation (5) with ρ_v/ρ_l = 0.048.

$$\text{(a)}\quad \frac{q_{c0}}{GH_{fg}} = 0.25\,\frac{1}{l/d_{he}} \tag{2}$$

$$\text{(b)}\quad \frac{q_{c0}}{GH_{fg}} = C\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d_{he}} \tag{3}$$

$$\text{(c)}\quad \frac{q_{c0}}{GH_{fg}} = 0.15\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3} \times \frac{1}{1+0.0077\,l/d_{he}} \tag{4}$$

$$\text{(d)}\quad \frac{q_{c0}}{GH_{fg}} = 0.26\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433} \times \frac{(l/d_{he})^{0.171}}{1+0.0077\,l/d_{he}} \tag{5}$$

on the condition of $\rho_v/\rho_l = 0.048$ for equations (4) and (5). In Fig. 3, it is noticed that there are few data in the regimes represented by the lines (a) and (b). It is almost impossible to derive correlation equations from such few data; and equations (2) and (3) tentatively used above are nothing but the correlation equations obtained for the CHF in tubes [5]. Nevertheless, it may be noted that the line (b) in Fig. 3 agrees fairly well with the existing data if the constant C on the RHS of equation (3) is assumed to be 0.25 for small l/d_{he} and 0.34 for large l/d_{he} in the same manner as in the case of tubes [5].

Next, it is clearly noticed in Fig. 3 that most of the existing data appear in the regime represented by the line (c), and equation (4) giving the line (c) is the one determined so as to fit in with these data. However, there are a few singular data points in Fig. 3, on which the following remarks will be given. Three data points denoted by × for l/d_{he} = 25 in Fig. 3 are those of R-113 obtained by Tong, Efferding and Bishop [15]. The trend of Freon data appearing slightly lower than water data has already been observed for tubes [5] and annuli [9], but it must be noted here that since the experiment of Tong, Efferding and Bishop was made in a channel heated by condensing steam, there might be a possibility of deviating from the uniform heat flux condition. Meanwhile, three data points denoted by a solid triangle for l/d_{he} = 110 and 125 in Fig. 3 are the special ones of Gambill and Bundy [13] which has been mentioned in Section 2.1. According to the result of Fig. 6 in the preceding study [3], for example, there is a trend that q_{c0} for downflow is lower than that for upflow, but the data shown by solid triangles in Fig. 3 seem to be too low, and the cause may be ascribed to the error induced by the indirect determination of q_{c0} (see Section 2.4). On the other hand, two data points denoted by the solid circle for l/d_{he} = 200 in Fig. 3 belong to the total of 43 data of DeBortoli *et al.* [11] listed in Table 1. However, the two data alone show abnormalities not only in Fig. 3 but also in Fig. 4 for the effect of inlet subcooling enthalpy on CHF (see Section 2.4).

Finally, it is noted in Fig. 3 that there are scarcely any data in the regime represented by the line (d).

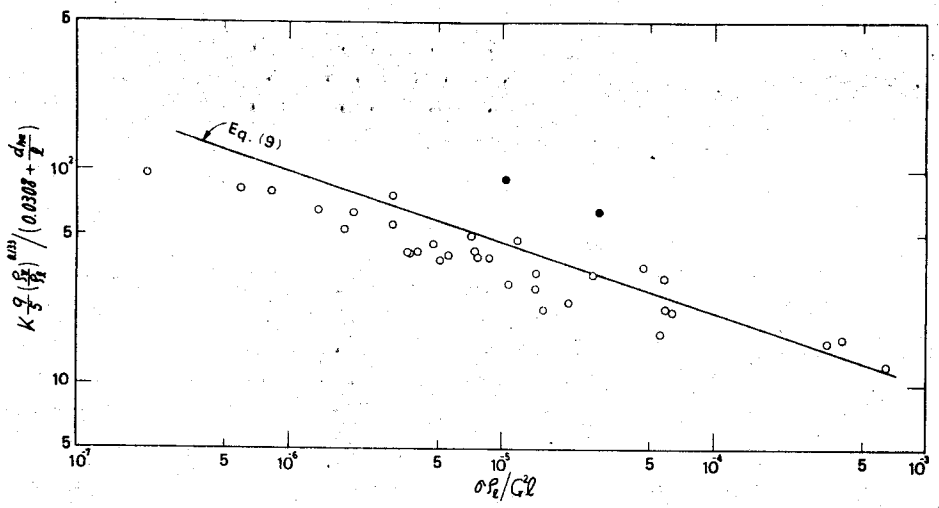

FIG. 4. Comparison between prediction of equation (9) and experimental data for K.

Equation (5), which gives the line (d), is no more than a tentative equation determined by modifying the correlation equation for annuli [8, 9] so as to fit a few data in Fig. 3 for $l/d_{he} = 55$ and 125.

2.3. *Remarks on characteristic regimes of CHF*

In my studies on CHF in tubes [6, 7], CHF has been classified into four characteristic regimes called L, H, N and HP, where L-regime is the one corresponding to low mass velocities, N-regime to very high mass velocities, H-regime is the intermediate one between L- and N-regime, and HP-regime is the regime that replaces N-regime when the system pressure is very high.

For rectangular channels, the boundary between L- and H-regime can be determined by the intersecting point of the lines (b) and (c) in Fig. 3. However, the equation to predict the boundary between H- and N-regime cannot be obtained because of lack of the existing data for N-regime which is characterized by the non-linear relationship between q_c and ΔH_i.

In the case of $l/d_{he} = 500$ in Fig. 3, a special trend is noted for a few data of Troy [12] near the left end to deviate upward from the line (c). For these data, ρ_v/ρ_l is rather high as 0.136, and l/d_{he} is very high as 500, so that the deviation from the line (c) may be regarded as resulting from the generation of HP-regime. If the criterion obtained for the onset of HP-regime in tubes [6] is tentatively applied to this case, it shows that the above-mentioned data certainly satisfy the conditions of entering into HP-regime.

2.4. *Generalized correlation of* K

The experimental data collected in Section 2.1 exhibit a linear relationship between q_c and ΔH_i such as shown in Fig. 2, for which the following equation can be written

$$q_c = q_{c0}\left(1 + K\frac{\Delta H_i}{H_{fg}}\right) \tag{6}$$

where K is the dimensionless parameter for the effect of inlet subcooling enthalpy ΔH_i, and is obtainable from the experimental data. Meanwhile, according to a preceding paper [4], if it is assumed that the concept of boiling length holds for the inlet subcooling condition of $(G\Delta H_i/q_c)/(4l/d) \ll 1$, then K can be derived theoretically from equations (2)–(4), respectively as follows:

from equation (2)

$$K = 1 \tag{7}$$

from equation (3)

$$K = \frac{1.043}{4C(\sigma\rho_l/G^2l)^{0.043}} \tag{8}$$

from equation (4),

$$K = \frac{5}{9}\frac{0.0308 + d_{he}/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{1/3}}. \tag{9}$$

As has been pointed out in Section 2.2, the greater part of the existing data is in the regime represented by the line (c) in Fig. 3 or equation (4). Therefore, the experimental K can be compared with the prediction of equation (9) to give the result of Fig. 4. Two data points denoted by solid circles in Fig. 4 correspond to the singular data points of the same symbol in Fig. 3. The rest of the data points have a similar trend to that of equation (9) for the variation with $\sigma\rho_l/G^2l$, but statistically speaking, their magnitudes are about 20% below the prediction of equation (9) for unknown reasons.

It is noted here that the data of q_{c0} denoted by solid triangles in Fig. 3 are those determined by equation (6) for experimental conditions of q_c, ΔH_i and H_{fg}, assuming K to be 80% of the value predicted by equation (9).

Table 2. Summary of the data of q_{c0} for very small l/d_{he}

Source	Fluid	Type	l(cm)	s(cm)	b(cm)	w/b	l/d_{he}	$(\rho_v/\rho_l)\times 10^2$	$\sigma\rho_l/G^2l$	No. of data	δ(mm)
[18]	Water	B	30.0	2.20	1.70	0.882	6.02	0.0624	8.37×10^{-3}–9.31×10^{-2}	5	∞*
[19, 20]	Water	B	12.7	1.27	0.652	0.896	4.47	0.0624	1.11×10^{-3}–2.79×10^{-4}	4	0.178
[19, 20]	Water	B	12.7	1.27	0.952	0.613	3.07	0.0624	1.11×10^{-3}–2.79×10^{-4}	2	0.178
[21]	Water	B	1.00	0.600	1.80	0.833	0.694	0.0624	1.01×10^{-4}–1.23×10^{-3}	10	∞*
[21]	Water	B	1.50	1.00	1.80	0.833	0.625	0.0624	1.33×10^{-4}–8.24×10^{-4}	9	∞*
[21]	Water	B	1.00	1.00	1.80	0.833	0.417	0.0624	2.38×10^{-4}–4.73×10^{-3}	10	∞*

* Very thick heated plate.

3. CHF FOR VERY SMALL l/d_{he}

In a preceding study [9] on CHF in annuli with inside heating, I showed that when l/d_{he} decreased to the magnitude of about 1 to 5, the experimental data of q_{c0} deviated from the generalized correlation in the form of Fig. 3. Then, Katto and Kurata [10] made an experimental study on CHF of forced convection boiling of water and R-113 on uniformly heated plane surface submerged in a parallel flow to derive the following generalized correlation equation

$$\frac{q_{c0}}{GH_{fg}} = 0.186\left(\frac{\rho_v}{\rho_l}\right)^{0.559}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.264} \tag{10}$$

and showed that equation (10) also agrees with the existing data of q_{c0} for acetone, toluene, monoisopropylbiphenyl and water flowing through internally heated annuli of very small l/d_{he}.

A similar situation is expected for the CHF in rectangular channels too, and Table 2 lists the data sources collected for q_{c0} in the case of very small l/d_{he}. All the studies listed in Table 2 are made with the channels of type B in Fig. 1, and the study of Isshiki *et al.* [18] for horizontal flow with bottom heating, Yücel and Kakaç [19] on the second line in Table 2 for horizontal flow with either bottom or top heating, Yücel and Kakaç on the third line in Table 2 for vertical upflow, and Goto for horizontal flow with top heating (the experiment of Yücel and Kakaç has been confirmed [20] to be made under conditions of l = 127 mm and w = 5.84 mm).

In Fig. 5, the data of q_{c0} mentioned above are compared with two lines (c) and (d) based on correlation equations (4) and (5) for rectangular channels, and with a line based on correlation equation (10) for plane surfaces in a parallel flow. All the data of q_{c0} are those obtained for water at atmospheric pressure (see Table 2), so that ρ_v/ρ_l is taken as 0.000624 for three equations (4), (5) and (10) represented in Fig. 5.

It is noted in Fig. 5 that for l/d_{he} = 6.0, the experimental data (of Isshiki *et al.* [18]) still appear near the line (c). The data for l/d_{he} = 3.0 and 4.5 in Fig. 5 are those of Yücel and Kakaç [19] showing that q_{c0} differs slightly from each other for vertical upflow, horizontal flow with bottom heating or top heating, and Yücel and Kakaç ascribed the difference to the effect of gravity. However, now apart from this problem, it may be noticed that the experimental data for l/d_{he} = 3.0 and 4.5 are much nearer the line of equation (10) than the lines (c) and (d). As is seen in Table 2, these data are those of the experiment made with a very thin heated plate (δ = 0.178 mm), so that there is a possibility of giving somewhat lower values of q_{c0} than usual. However, the experimental data of Goto [21] for l/d_{he} = 0.5 in Fig. 5 are those obtained with a very thick heated plate. To sum up, therefore, it may be concluded that q_{c0} in rectangular channels of very small l/d_{he} takes the value near the prediction of equation (10).

Finally, the result of Fig. 5 suggests a possibility that when l/d_{he} becomes very small, q_{c0} may be affected by

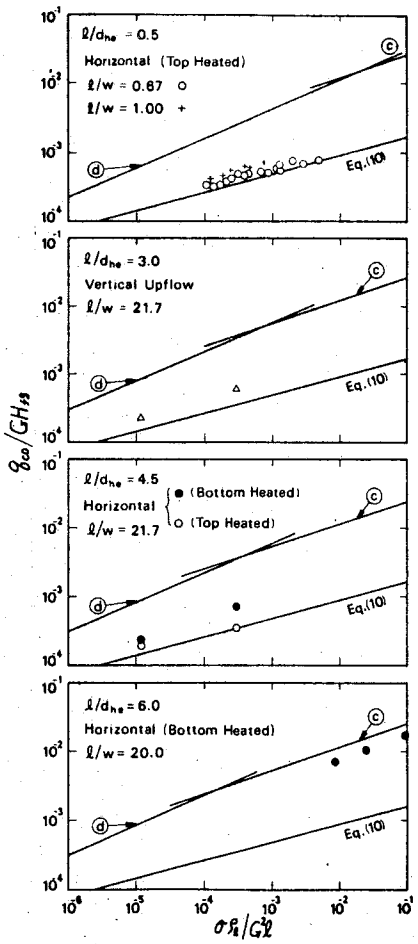

FIG. 5. Generalized graphic representation of q_{c0} data of water at atmospheric pressure (ρ_v/ρ_l = 0.000624) in the case of very small l/d_{he}.

length to width ratio of heated plate l/w. However, further studies are needed on the complicated effects of l/d_{he} and l/w on CHF in this special regime.

4. CONCLUSIONS

(1) For forced convection boiling in uniformly heated rectangular channels, existing data of CHF are analyzed to give the generalized correlation of Fig. 3 for the basic critical heat flux q_{c0} and that of Fig. 4 for the inlet subcooling parameter K. In addition, q_{c0} for very small l/d_{he} is investigated to give the result of Fig. 5.

(2) The existing data of CHF in rectangular channels are very limited in number, experimental condition, and kind of test fluid (see Tables 1 and 2). Further experimental studies are needed to accomplish the generalized correlation of CHF of course. However, the present study has suggested that CHF in rectangular channels is similar in character to CHF in tubes for which general frameworks are comparatively well known [5, 6]. Viewed at this angle, even the results obtained in this paper may be rather useful for estimating CHF in rectangular channels under various conditions as long as high accuracies are not required.

(3) For L-regime of CHF in annuli with inside heating, the author [8, 9] tentatively assumed a single value C = 0.25 on the RHS of equation (3) because of lack of the existing data of annuli in this regime. However, the result of Fig. 3 for rectangular channels seems to cooperate with the result for tubes [5] to suggest that the value of C may change between 0.25 and 0.34 for annuli too according to the magnitude of l/d_{he}.

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CONFIGURATIONS GENERALES DU CHF EN EBULLITION AVEC CONVECTION FORCEE DANS DES CANAUX RECTANGULAIRES UNIFORMEMENT CHAUFFES

Résumé — Pour le flux critique (CHF) en ébullition avec convection forcée dans des canaux rectangulaires alimentés en liquide sous-refroidi, des données expérimentales sont analysées pour obtenir une formule générale sur les mêmes principes que dans les précédentes études de l'auteur pour les tubes et les espaces annulaires. De plus, des expériences faites dans des conditions spéciales de faible l/d_{he} où l est la longueur chauffé du canal et d_{he} le diamètre équivalent chauffé, sont étudiées pour confirmer que, dans ces conditions, le CHF est presque égal au CHF d'ébullition en convection forcée sur des surfaces planes chaudes et un écoulement parallèle.

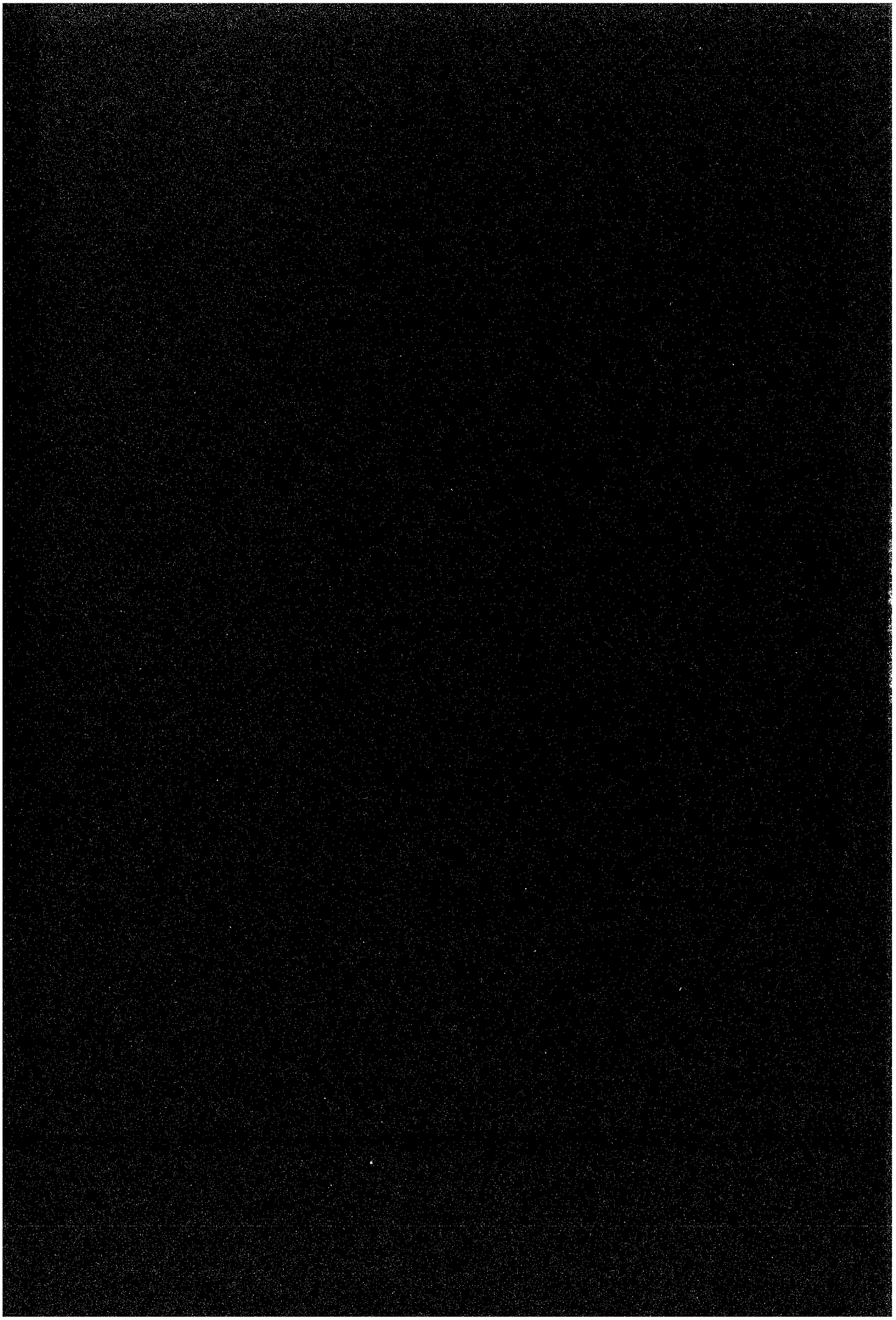
ALLGEMEINE CHARAKTERISTIK DER KRITISCHEN WÄRMESTROMDICHTE BEIM STRÖMUNGSSIEDEN IN GLEICHMÄSSIG BEHEIZTEN, RECHTECKIGEN KANÄLEN

Zusammenfassung—Zur Ermittlung der kritischen Wärmestromdichte beim Strömungssieden von unterkühlter Flüssigkeit in gleichmaßig beheizten, rechteckigen Kanälen werden vorhandene Versuchsdaten mit dem Ziel ausgewertet, eine allgemeine Korrelation der Daten zu ermitteln. Dabei wird dasselbe Prinzip wie in den früheren Arbeiten des Autors über die kritische Wärmestromdichte beim Strömungssieden in Rohren und Ringspalten angewandt. Darüber hinaus werden Versuche unter den besonderen Bedingungen eines sehr kleinen Verhältnisses L/d_{he} durchgeführt, wobei L die beheizte Länge des Kanals und d_{he} der beheizte äquivalente Durchmesser ist. Bei diesen Untersuchungen zeigt sich, daß die kritische Wärmestromdichte unter derartigen Bedingungen nahezu gleich groß ist wie beim Strömungssieden an beheizten, ebenen Oberflächen in einer Parallelströmung.

ОБЩИЕ ХАРАКТЕРИСТИКИ КРИТИЧЕСКОГО ТЕПЛОВОГО ПОТОКА ПРИ КИПЕНИИ В УСЛОВИЯХ ВЫНУЖДЕННОЙ КОНВЕКЦИИ В РАВНОМЕРНО НАГРЕВАЕМЫХ ПРЯМОУГОЛЬНЫХ КАНАЛАХ

Аннотация — Проведен анализ экспериментальных данных по критическому тепловому потоку (КТП) при кипении в условиях вынужденной конвекции в равномерно нагреваемых прямоугольных каналах, в которые поступает недогретая жидкость. Анализ выполнен с целью получения обобщенного соотношения по тому же принципу, что и в ранее проведенных автором исследованиях КТП при кипении в условиях вынужденной конвекции в трубах и кольцевых каналах. Кроме того, проведены эксперименты при малых значениях l/d_{he} (где l — длина нагреваемого участка канала, а d_{he} — эквивалентный диаметр нагрева) для проверки предположения о том, что КТП в рассматриваемых условиях примерно равен КТП при кипении с вынужденной конвекцией на плоских поверхностях нагрева в параллельном потоке.

TWO–PHASE FLOW



On the Construction of a Set of Equations to Describe a Compressible Two-Phase Flow

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The minimum number of physical quantities as well as that of the governing equations indispensable to give a necessary and sufficient description of the evolution of an N-component, separated two-phase flow through a duct is studied in this paper for the respective cases of (i) the evolution in a thermodynamic nonequilibrium state between two phases, (ii) the prescribed evolution such as keeping some physical quantities constant and (iii) another type of prescribed evolution with the condition of thermodynamic equilibrium between two phases. For both types of prescribed evolution, the theorem for the existence of unique solutions in a system of m linear equations in n unknowns is applicable to determine the theoretical conditions which should be satisfied to materialize a given, prescribed evolution.

Nomenclature

a_{ij} = coefficients on the left-hand side of Eq. (2)
b_i = term on the right-hand side of Eq. (2)
C = matrix [Eq.(4)]
D = matrix [Eq.(4)]
h = enthalpy ($\Delta h = h - h_{sat}$, where h_{sat}: saturation enthalpy)
m = number of governing equations
N = number of components
N' = number of insoluble components
n = number of mutually independent physical quantities
p = pressure
q_w = rate of transfer of heat between duct wall and fluid per unit time and unit axial length
T = temperature
S = cross-sectional area of duct
W = overall mass flow rate (constant in z-direction)
w = mass concentration
u = velocity
x = mass flow fraction
z = axial coordinate of duct
α = void fraction
ρ = density
ξ_j = mutually independent physical quantities

Suffix
1 = the first phase
2 = the second phase
A, B, . . ., N = components A, B, . . . , N.

I. INTRODUCTION

For the analytical study of not only critical flow but also other various flow characteristics, the theoretically correct description of the compressible two-phase flow through a duct is necessary. Changes in flow state through a duct are simultaneously governed by several mutually independent physical laws, so that they are expressed by simultaneous equations. Survey of existing studies, however, reveals that the selection of such simultaneous equations differs from researcher to researcher even for the same flow conditions, of which

some examples are illustrated in Table 1. Also, there is a similar problem of the necessary number of physical quantities to be selected for the complete description of kinetic and thermodynamic conditions of fluid in a given two–phase flow. In addition, it is not necessarily clear whether the use of six governing equations (two conservation, two motion, and two energy equations) will always be sufficient to describe the general state of two–phase flow. In this paper, therefore, a study has been made in a general manner of the number of physical quantities and the number of equations necessary to describe the state of fluid in a two–phase flow consisting of N components. To simplify the explanation in this paper, the classical

Table 1. Examples of the set of equations and physical quantities adopted in the foregoing studies of twophase flow.

	Continuity eq.		Momentum eq.		Energy eq.		Physical quantities
	1st phase	2nd phase	1st phase	2nd phase	1st phase	2nd phase	
Ogasawara (1968)[1] Single-component Thermal equilibrium	●		●	●	●		p, u_1, u_2, α
Katto (1969)[2] Single-component Thermal equilibrium	●		●	●	●	●	p, u_1, u_2, x
Ishii-Zuber (1970)[3] Single-component Thermal equilibrium	●	●	●		●		p, u_1, u_2, α
Giot-Fritte (1972)[4] Single-component Thermal equilibrium	●		●	●	●		p, u_1, u_2, α
Giot-Fritte (1972)[4] Two-component Thermal equilibrium	●	●	●	●	●		p, T, u_1, u_2, α
Giot-Fritte (1972)[4] Insoluble two-component Thermal non-equilibrium	●	●	●	●	●	●	p, T_1, T_2, u_1, u_2, α
Sudo-Katto (1974)[5,6] Single-component Thermal equilibrium	●		●	●	●	●	p, u_1, u_2, x
Bouré *et al.* (1976)[7] Two-phase flow with no restriction	●	●	●	●	●	●	p, Δh_1, Δh_2, u_1, u_1-u_2, α
Bouré *et al.* (1976)[7] Two-phase flow under $d\Delta h_1/dz = 0$.	●	●	●	●	●	●	p, h_2, u_1, u_1-u_2, α
Soo (1976)[8] Single-component	●	●	●	●	●	●	———
Yadiagaroglu *et al.* (1976)[9] Two-phase flow with no restriction	●	●	●	●	●	●	———

separated flow model as shown in Figs. 1 and 2 is dealt with, and it is assumed that the flow is steady and that there is no mass tranfer between the duct wall and the fluid (therefore, the mass flow rates W_A, W_B,—of the components A, B,— in Fig. 1 are kept constant through the duct); but the principle can be extended to more complicated conditions without difficulty. In addition, it is assumed in this paper that there are friction and heat transfers between the duct wall and the fluid, and that the flow is one-dimensional where the pressure p is uniform on a cross-section, taking the same value for two phases.

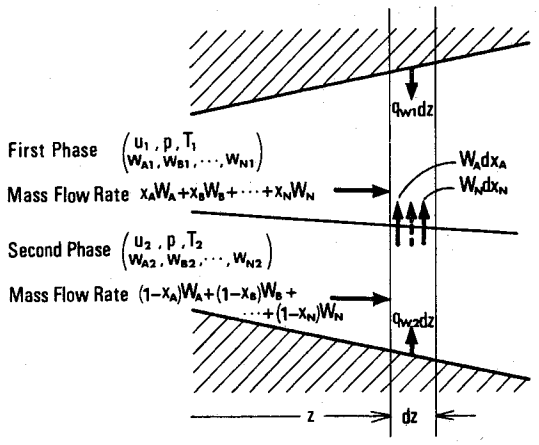

Fig. 1. N-component two-phase flow (Separated flow model)

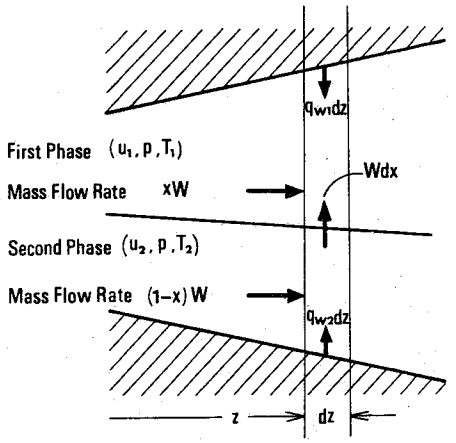

Fig. 2. Single-component two-phase flow (Separated flow model).

II. MATHEMATICAL STATUS

One-dimensional, steady two–phase flow undergoing a continuous change of state through a duct is dealt with, and the axial coordinate of duct is taken as z. Now, it is assumed that n mutually independent physical quantities ξ_j $(j=1, 2, \ldots, n)$ are required for the complete description of physical state of flow at a given position $z=z$ in the duct. The mutually independent physical quantities afore-mentioned are defined as those which are not related to each other by the constitutive laws (the equation of state and other laws regarding the specific character of material such as friction and heat transfer). All the $\xi_j's$ make a change as a dependent variable of the position z, that is $\xi_j = \xi_j(z)$.

From the definition of ξ_j stated above, all the physical quantities f_k (inclusive of ξ_j) possessed or transported by a fluid, such as mass flow rate, momentum and enthalpy flow rate, are generally given as $f_k = f_k(\xi_1, \xi_2, \ldots, \xi_n)$. Accordingly, the infinitesimal increment of f_k is given as:

$$df_k = \frac{\partial f_k}{\partial \xi_1} d\xi_1 + \frac{\partial f_k}{\partial \xi_2} d\xi_2 + \ldots + \frac{\partial f_k}{\partial \xi_n} d\xi_n, \quad (k = 1, 2, \ldots) \tag{1}$$

Now, the change of the state of flow in z-direction within a duct takes place under the simultaneous control of several independent conservation laws (conservation of mass, conservation of component, Newton's second law of motion, the first law of thermodynamics).

Each of these laws is generally described as a linear combination of the df_k ($k = 1, 2, \ldots$) and the compulsive factor acting on a fluid through an infinitesimal interval $z = z \sim z + dz$ (usually assumed, for the sake of simplicity, to act at the wall and the interface). Thus, the simultaneous equations controling the change of state of flow are given as follows:

$$a_{i1}\frac{d\xi_1}{dz} + a_{i2}\frac{d\xi_2}{dz} + \ldots + a_{in}\frac{d\xi_n}{dz} = b_i, \quad (i = 1, 2, \ldots, m) \tag{2}$$

where m is the number of the independent coservation laws afore-mentioned. $a_{ij} = \sum_k (\partial f_k/\partial \xi_j)$ on the left-hand side of Eq. (2) is generally a function of ξ_j, that is $a_{ij} = a_{ij}(\xi_1, \xi_2, \ldots, \xi_n)$. b_i on the right-hand side of Eq. (2) is a term which forces the fluid to change the state of flow, and it is classified into the following two groups:

a) Change in the cross-sectional area S per unit axial length of duct: $b_i = dS/dz$. The b_i in this case is given as a function of z.

b) Frictional force, heat transfer, and mass transfer exerted per unit axial length of duct. These are usually expressed as follows:

$$b_i + \sum_{j=1}^{n} \lambda_j \frac{d\xi_j}{dz} \tag{3}$$

where $b_i = b_i(z, \xi_1, \xi_2, \ldots, \xi_n)$, and
$\lambda_j = \lambda_j(z, \xi_1, \xi_2, \ldots, \xi_n)$.

The first term of Eq. (3) denotes the normal frictional force, heat transfer, and mass transfer. On the other hand, the second term of Eq. (3) appears, for instance, as the momentum transfer effect $\lambda_j d\xi_j$ that is associated with the mass transfer (the quantity of which is assumed to be $d\xi_j$) taking place between the separated two phases. This second term of Eq. (3) can be integrated into the left-hand side of Eq. (2), so that Eq. (2) needs no transformation. In this case, a_{ij} on the left-hand side of Eq. (2) is generally expressed as $a_{ij} = a_{ij}(z, \xi_1, \xi_2, \ldots, \xi_n)$, but it does not create any change in the argument that follows.

Now, if ξ_j at a position $z = z^0$ is given as $\xi_j = \xi_j{}^0 (j = 1, 2, \ldots, n)$, and the value of $d\xi_j/dz$ ($j = 1, 2, \ldots, n$) is uniquely determined from the simultaneous equations (2), all the quantities ξ_j can be determined uniquely in the vicinity of $z = z^0$, and as a result, it provides us the means by which the state of flow can be determined between the inlet and the outlet of the duct. In this case, the number of the unknown quantity ($d\xi_j/dz$), that is n, and the number of equations m, in Eq. (2) are related to the constitutive and the conservation laws respectively, so that one cannot assume $n = m$ *a priori*. According to the theorem on the solution of simultaneous linear equations[10] the necessary and sufficient condition for obtaining a unique set of solution to $d\xi_j/dz$ ($j = 1, 2, \ldots, n$) of Eq. (2) is that the following two matrices C and D have the same rank n.

$$C = \begin{bmatrix} a_{11} \ldots\ldots a_{1n} \\ \ldots\ldots\ldots\ldots \\ a_{m1} \ldots\ldots a_{mn} \end{bmatrix} \text{ and } D = \begin{bmatrix} a_{11} \ldots\ldots a_{1n}\, b_1 \\ \ldots\ldots\ldots\ldots\ldots \\ a_{m1} \ldots\ldots a_{mn}\, b_m \end{bmatrix} \tag{4}$$

In other words, if C and D have the same rank n, Eq. (2) gives a unique solution not only in case of $n = m$ but also in case of $n < m$. Therefore, one needs to examine the relation between n and m from the physical standpoint.

III. CASE OF THERMODYNAMIC NON-EQUILIBRIUM STATE BETWEEN TWO PHASES

1) Mutually Independent Physical Quantities Required for Describing the State of Fluid

The physical quantities dealt with in this paper should consist of those necessary to describe (a) the kinetic state of fluid and (b) the thermodynamic state of fluid.

First, for the description of the kinetic state of fluid, velocity and mass are the physical quantities which should be taken into consideration according to the second law of motion, so that there are the following quantities in the present case:

u_1: velocity of the first phase;

u_2: velocity of the second phase;

$x_A, x_B, \ldots, x_N$: mass flow fraction of each component $A, B, \ldots, N$ in the first phase.

Mass flow fractions of components in the second phase ($1-x_A, 1-x_B, \ldots$) are subject to $x_A, x_B, \ldots$, so that they can be excluded.

Second, for the description of the thermodynamic state of fluid, Gibbs' phase rule[11] is applied to the first phase (N-component single-phase system) and the second phase (N-component single phase-system), respectively, to determine the number of independent physical quantities as follows:

for the first phase: $1 + N$ independent physical quantities (for example, pressure p, temperature T_1 and mass concentration of component $w_{A1}, w_{B1}, \ldots, w_{N-1,1}$), and

for the second phase: $1 + N$ independent physical quantities (for example, pressure p, temperature T_2, and mass concentration of component $w_{A2}, w_{B2}, \ldots, w_{N-1,2}$).

In the first phase, the equation $w_{A1} + w_{B1} + \ldots + w_{N-1,1} + w_{N1} = 1$ holds, so that w_{N1} can be excluded from the set of independent quantities, and similarly w_{N2} can be excluded from the set of independent quantities in the second phase.

The mass concentrations ($w_{A1}, w_{B1}, \ldots, w_{N-1,1}$) and ($w_{A2}, w_{B2}, \ldots, w_{N-1,2}$) have the following relations with the mass flow fraction ($x_A, x_B, \ldots, x_N$):

$$\left.\begin{aligned} w_{A1} &= \frac{x_A W_A}{x_A W_A + x_B W_B + \ldots + x_N W_N}, \\ w_{B1} &= \frac{x_B W_B}{x_A W_A + x_B W_B + \ldots + x_N W_N}, \\ \ldots, \quad w_{N-1,1} &= \frac{x_{N-1} W_{N-1}}{x_A W_A + x_B W_B + \ldots + x_N W_N}. \end{aligned}\right\} \tag{5}$$

$$w_{A2} = \frac{(1 - x_A) W_A}{(1 - x_A) W_A + (1 - x_B) W_B + \ldots (1 - x_N) W_N},$$

$$w_{B2} = \frac{(1 - x_B)\, W_A}{(1 - x_A)\, W_A + (1 - x_B)\, W_B + \ldots + (1 - x_N)\, W_N},$$

$$\ldots\ldots$$

$$w_{N-1,2} = \frac{(1 - x_{N-1})\, W_{N-1}}{(1 - x_A)\, W_A + (1 - x_B)\, W_B + \ldots + (1 - x_N)\, W_N}. \qquad (6)$$

Therefore, if the mass flow fractions ($x_A, x_B, \ldots, x_N$) are chosen as independent physical quantities, the mass concentrations ($w_{A1}, w_{B1}, \ldots, w_{N-1,1}$) and ($w_{A2}, w_{B2}, \ldots, w_{N-1,2}$) become dependent physical quantities. To sum up, the number of mutually independent physical quantities n of a N-component two-phase flow is determied as follows:

$$n = 5 + N \quad (\text{for example}, u_1, u_2, x_A, x_B, \ldots, x_N, p, T_1, T_2) \qquad (7)$$

In the case of one-component two-phase flow shown in Fig. 2, the concentration, and accordingly Eqs. (5) and (6), do not appear, but it can be noted that Eq. (7) is as applicable as ever.

2) *Equations Governing the Change of State of Flow*

The equation of overall mass conservation, including both the first and the second phases, can be readily written as follows:

$$\frac{d}{dz}\left(\frac{x_A W_A + x_B W_B + \ldots + x_N W_N}{\rho_1 u_1} + \frac{(1 - x_A) W_A + (1 - x_B) W_B + \ldots + (1 - x_N) W_N}{\rho_2 u_2}\right) = \frac{dS}{dz} \qquad (8)$$

where ρ_1 is the density of the first phase, and ρ_2 the density of the second phase. Next, the conservation of mass in the first phase (or in the second phase) is concerned with the mass transfer across the interface of the two phases. In this case, it must be noted that (i) the conservation of mass for each component (that is, the conservation of species) needs to be established, and (ii) a formula governing the mass transfer exists for each of N components A, B, ..., N. Therefore, there exist N equations, which govern the respective changes of mass flow rate for N components, $dx_A, dx_B, \ldots, dx_N$, caused by the mass transfer across the interface. Since it has been assumed in the present case that two phases are theremodynamically in the non-equilibrium state, mass transfer takes place naturally between the two phases in consequence of the non-equilibrium condition.

The law of motion and the conservation law of energy need to be satisfied respectively by both the first and the second phases (or alternatively, by either of the two phases and the combined phase). These two laws are concerned with the physical quantities of universal nature such as momentum, force, and energy; therefore, they need not be applied to each component. By summing up the above presentation, the number of the equations governing the N-component two-phase flow is determined as follows:

$m = 5 + N$ (an equation of overall continuity, N equations of conservation of each component, two equations of motion, and two equations of energy). (9)

In the case of one-component two-phase flow, where the discrimination of component is unnecessary, Eq. (9) is as applicable as ever, for which the equation of conservation is nothing but the equation of continuity for one phase. In addition, the mass transfer in this case takes plase as the result of heat transfer with phase change.

From the results of Eq. (7) and Eq. (9), it is concluded that the relation of $n = m$ holds in the present case.

3) Case of Flow Including Insoluble Components

When N' components A, B, ... , N′, among the N components A, B,. . . . , N ($N' \leqq N$), exist in either the first phase or the second phase alone, and are insoluble in the other phase, it establishes the state of $dx_A = dx_B = \ldots = dx_N = 0$; in other words, $x_A, x_B, \ldots, x_N$, are no longer variables. Thus, Eq. (7) becomes $n = 5 + N - N'$. Also, the equations governing the mass transfer of N' components disapperar, so that Eq. (9) becomes $m = 5 + N - N'$. Therefore, the following relation holds in the present case:

$$n = m = 5 + N - N'. \tag{10}$$

Appendix: When $N = 2$ and $N' = 2$ (both components are insoluble with each other in a two-components two-phase flow), Eq. (10) gives $n = 5$ (u_1, u_2, p, T_1, T_2) and $m = 5$ (an equation of overall continuity, two equations of motion, and two equations of energy). For the same condition, Giot and Fritte[4] gives $n = 6$ ($u_1, u_2, p, T_1, T_2, \alpha$) and $m = 6$ (two equations of continuity, two equations of motion, and two equations of energy). The reason for this can be explained as follows. In the present case, one can assume $x_A = 1$ and $x_B = 0$ because of the insolubility of both components, and then the void fraction α is given as follows:

$$\left.\begin{aligned} \alpha &= W_B/(\rho_2 u_2 S) \\ \text{where} \quad \rho_2 &= \rho_2(p, T_2). \end{aligned}\right\} \tag{11}$$

If this void fraction α is adopted as one of the physical quantity to be added to the aforementioned system of $n = m = 5$, and at the same time, if Eq. (11) is adopted as an equation of continuity to be added to the set of equations, it yeilds Giot and Fritte's stystem of $n = m = 6$. Therefore, the system of $n = m = 5$ obtained in this paper and the system of $n = m = 6$ given by Giot and Fritte are essentially identical with each other. However, it should be noted that as seen in Eq. (11), the void fraction α at the position $z = z$ is determined depending on the physical quantities u_2, p, and T_2 (and the crosssectional area of duct S) at $z = z$.

IV. PRESCRIBED EVOLUTION WITH A FIXED PHYSICAL QUANTITY

As for the prescribed evolution, one can assume, as the simple case, the case where one of the physical quantities is designated to be kept constant for eigher of two phases, such as isentropic or isothermal change. In this case, one can begin the analysis by setting up Eq. (2) including the designated physical quantity ξ_j in n mutually independent physical quantities. Then, taking into account that ξ_j is kept constant through the duct, $d\xi_j/dz$ can be deleted from Eq. (2); that is, the number of unknown quantities is reduced by one. Thus, Eq. (2) becomes a system of m equations in $n = m - 1$ unknown quantities. In this case, the necess-

ary condition for the matrices C and D of Eq. (4) to have the same rank n, is readily given as follows:

$$\begin{vmatrix} a_{11} \cdots\cdots\cdots a_{1n}\ b_1 \\ \cdots\cdots\cdots\cdots\cdots\cdots \\ a_{m1} \cdots\cdots\cdots a_{mn}\ b_{m1} \end{vmatrix} = 0, \quad \text{where } n = m - 1. \tag{12}$$

Eq. (12) provides the limitation to be imposed on $b_1, b_2, \ldots, b_m$ in order to maintain the designated condition $\xi_j =$ constant. Should any possible value of $b_1, b_2, \ldots, b_m$ be unable to satisfy Eq. (12), the prescribed evolution is impossible. When Eq. (12) is satisfied, the change of the state of flow through the duct can be determined uniquely from any set of $m - 1$ simultaneous equations arbitrarily selected from Eq. (2) cosisting of m equations in $m - 1$ unknowns.

Appendix: If $d\xi_j/dz$ is solved from the simultaneous equations of Eq. (2) which still contains the term of $d\xi_j/dz$ (that is, m equations in m unknown quantities), we obtain $d\xi_j/dz = N_j/\Delta$, where Δ is the determinant of a_{ij} which are the coefficients of the left-hand side of Eq. (2), and N_j is the determinant obtained from Δ by replacing in Δ the jth column by the column with the elements $b_1, b_2, \ldots, b_m$. Then, assuming $d\xi_j/dz = 0$, it yields $N_j = 0$ ($\Delta \neq 0$), which is identical with Eq. (12); this is the method employed by Bouré and others.[7] However, it should be noted that the method utilizing the theorem of the same rank n for the matrices C and D of Eq. (4) is in more universal form.

V. CASE OF THEMODYNAMIC EQUILIBRIUM STATE BETWEEN TWO PHASES

1) Prescribed Evolution

Change of state of flow keeping two phases in the state of themodynamic equilibrium at any position $z = z$ is also a kind of prescribed evolution. However, the method presented in Section IV is inconvenient to apply to the case of thermodynamic equilibrium state. Therefore, the fundamental analysis will again be attempted for examining the number of mutually independent physical quantities as well as the number of governing equations.

The quantities required to describe the kinetic state are the same as in Section III–1 to give

$$u_1, u_2, x_A, x_B, \ldots, x_N.$$

As for the description of the thermodynamic state of fluid, Gibbs' phase rule[11] is applied to the whole system combining the first and the second phases (which is an N-component two-phase system in thermodynamic equilibrium state), to give the mutually independent physical quantities as follows:

N physical quantities (for example, pressure p, and mass concentration of each component $w_{A1}, w_{B1}, \ldots, w_{N-1,1}$).

Because of the thermodynamic equilibrium state in the present case, the mass concentrations in the second phase $w_{A2}, w_{B2}, \ldots, w_{N-1,2}$ are determined depending on N independent physical quantities afore-mentioned ($p, w_{A1}, w_{B2}, \ldots, w_{N-1,1}$).

On the other hand, Eqs. (5) and (6) still hold in the present case, and it means that each of $w_{A1}, w_{B2}, \ldots, w_{N-1,2}$ can be determined depending on $w_{A1}, w_{B1}, \ldots, w_{N-1,1}$, through the parameters of $x_A, x_B, \ldots, x_N$. Therefore, from the double relations aforementioned, N-1 concentrations $(w_{A1}, w_{B1}, \ldots, w_{N-1,1})$ are determined depending on the pressure p, so that they can be deleted from the set of independent physical quantities. Then, combining this fact wtih Eq. (5) reveals that N-1 quantities $x_A, x_B, \ldots, x_{N-1}$ are determined depending on the value of x_N. Thus, in conclusion, the number of mutually independent physical quantities n is determined for a N-component two-phase flow with thermodynamic equilibrium state as follows:

$$n = 4 \quad \text{(for example, } u_1, u_2, x_N, p\text{)}. \tag{13}$$

Next, the number of governing equations is examined in the following manner. Since two phases are now in the state of thermodynamic equilibrium, mass transfer due to non-equilibrium does not take place. In other words, there is no law of mass transfer between the two phases in the state of thermodynamic equilibrium, and accordingly, N equations of conservation of component in Eq. (9) vanish. On the other hand, the equation of motion and that of energy make no change in Eq. (9), because the second law of motion as well as the first law of thermodynamics must be satisfied by the first phase and the second phase respectively. On the equation of energy, it may be of use to note that (i) there is no heat transfer across the interface between two phases because of the state of thermodynamic equilibrium, but there can exist heat transfer between the duct wall and the fluid of the first and second phases, and (ii) the equation of energy is nothing but the conservation law of energy, which has no relation to the condition of thermodynamic equilibrium. Therefore, it is an incorrect idea to assume that the number of energy equations reduces to one (an overall energy equation alone exists) due to the state of thermodynamic equilibrium. Thus, in conclusion, the number of equations governing a thermodynamic equilibrium N-component two-phase flow is given as follows:

$$m = 5 \quad \text{(an overall equation of continuity, two equations of motion, and two equations of energy)}. \tag{14}$$

From the result of Eqs. (13) and (14), it shows that Eq. (2) becomes a system of five equations in four unknown quantities ($n = m - 1$), for which the condition to be able to get a unique solution is obtained in the same manner as in the case of Eq. (12) to give

$$\begin{vmatrix} a_{11} \cdots\cdots\cdots a_{14} & b_1 \\ \cdots\cdots\cdots\cdots & \\ a_{51} \cdots\cdots\cdots a_{54} & b_5 \end{vmatrix} = 0. \tag{15}$$

Eq. (15) gives the necessary condition for creating the change of state of flow under the condition of thermodynamic equilibrium between two phases. Equation (2) in the present case includes two energy equations, and accordingly, the respective terms of b_4 and b_5 in Eq. (15) are concerned with the rate of heat transfer (per unit axial length) q_{w1} between the duct wall and the first phase and q_{w2} between the duct wall and the second phase, at the interval $z = z^0 \sim z^0 + dz$. Namely, in order to maintain the state of thermodynamic equilibrium

between two phases at the interval $z = z^0 \sim z^0 + dz$, q_{w1} and q_{w2} need to have a definite relation with each other, and the relation needs to satisfy Eq. (15). When Eq. (15) is satisfied the evolution of flow through a duct is uniquely determined by any set of four simultaneous equations arbitrarily chosen from the five equations.

Appendix: (i) In the case of single-component two-phase flow ($N = 1$), Eq. (13) can be written as $n = 4\,(u_1, u_2, x, p)$, where x is the mass flow fraction of the first phase (*cf.* Fig. 2). The void fraction α in the present case is given as:

$$\alpha = (1 - x)W/(\rho_2 u_2 S), \text{ where } \rho_2 = \rho_2(p) \tag{16}$$

Therefore, one can select α instead of x as an independent physical quantity, to substitute $n = 4\,(u_1, u_2, \alpha, p)$ for $n = 4\,(u_1, u_2, x, p)$.

(ii) In the case of two-component two-phase flow ($N = 2$), Eq. (13) can be written as $n = 4\,(u_1, u_2, x_A, p)$ or $n = 4\,(u_1, u_2, x_B, p)$. According to the Gibbs' phase rule, the thermodynamic state of two-component two-phase equilibrium system is determined by two physical quantities. Therefore, if p and T are employed for the two quantities, it yields $w_{A1} = w_{A1}(p, T)$ and $w_{A2} = w_{A2}(p, T)$, so that one can obtain $x_A = x_A(p, T)$ and $x_B = x_B(p, T)$ through Eqs. (5) and (6). Accordingly, Eq. (13) in the present case can be written as $n = 4\,(u_1, u_2, p, T)$. For this condition, Giot and Fritte[4] give $n = 5\,(u_1, u_2, p, T, \alpha)$. In the present case, however, the void fraction α is given as

$$\left.\begin{aligned} \alpha = [(1 - x_A)W_A + (1 - x_B)W_B]/(\rho_2 u_2 S), \\ \text{where} \quad \rho_2 = \rho_2(p, T) \end{aligned}\right\} \tag{17}$$

and it is readily found that the same circumstances as described in the appendix at the end of Section III hold here.

2) Application to the Case of Approximate Thermodynamic Equilibrium

The analysis in the preceding section is concerned with the case where two phases are kept exactly in the state of thermodynamic equilibrium. There may be, however, another situation to be considered. Namely, it is well known that when two phases in the state of non-equilibrium are brought into contact, two phases change toward the state of thermodynamic equilibrium through the heat and mass transfer between them. In this case, infinite time is required in principle in order to reach the state of thermodynamic equilibrium, but if the evolution of non-equilibrium two-phase flow through a duct is taking place very slowly, it may be assumed that two phases are approximately in the state of thermodynamic equilibrium at any cross-section of the duct. To make the analysis in such a case, it is sometimes simpler to utilize the equilibrium analysis described in Section V–1 than to utilize the non-equilibrium analysis given in Section III. Now, it should be noted that unlike the case of Section V–1, the present case should be governed by actually given thermal boundary conditions q'_{w1} and q'_{w2}, which are in different values from q_{w1} and q_{w2} satisfying Eq. (15). However, we can still have an allowance to assume the following relation:

$$q'_{w1} + q'_{w2} = q_{w1} + q_{w2} \tag{18}$$

where q_{w1} and q_{w2} are determined so as to satisfy Eqs. (15) and (18). To assume the state of

approximate equilibrium between two phases, means that the quantity of heat $q'_{w1} - q_{w1}$ $(= q_{w2} - q'_{w2})$ is transferred across the interface between two phases due to the phenomenon of heat transfer. Particularly, if the duct wall is thermally insulated, $q'_{w1} = q'_{w2} = 0$, so that the quantity of heat $-q_{w1}$ $(= q_{w2})$ is transferred across the interface between two phases.

After all, the situation above-mentioned means that there is a temperature difference ΔT between the two phases (which is actually kept in the thermodynamic non-equilibrium state) and the ΔT has a relation with the rate of heat transfer above-mentioned as follows:

$$\Delta T = (q'_{w1} - q_{w1})/(h_{12}l) \tag{19}$$

where h_{12} is the heat transfer coefficient between the two phases (N.B. two phases are in the state of thermodynamic non-equilibrium), l is the length of the intersection of the cross-sectional plane of the duct and the interface of two phases, q'_{w1} is the actual rate of heat transfer between the duct wall and the first phase ($q'_{w1} = 0$ when the duct wall is insulated), and q_{w1} is determined by Eq. (15) under the condition of Eq. (18). ΔT in Eq. (19) is regarded as a measure of the discrepancy from the thermodynamic equilibrium for the flow with a given condition of q'_{w1} and q'_{w2}. When the calculated value of ΔT is large, we must avoid this method based on the assumption of thermodynamic equilibrium, and the analysis described in Section III should be employed. The most interesting in practice is the case of the thermally insulated duct wall ($q'_{w1} = q'_{w2} = 0$), in which ΔT is in proportion to the absolute value of q_{w1}, as is evident from Eq. (19). Then, judging from the form of Eq. (15), it may be presumed that $|q_{w1}|$ increases as the rate of change of the sectional area $|dS/dz|$ becomes high. This means that the two-phase flow in a short nozzle with a rapid change of sectional area is very far from the state of thermodynamic equilibrium, while the two-phase flow which flows through a long constant-area tube ($dS/dz = 0$) with small friction, is apt to realize the state of approximate thermodynamic equilibrium.

VI. CONCLUSIONS

The following conclusions follow from the present study for an N-component two-phase flow with no mass transfer on the duct wall.

(1) Change of state of flow along the duct is determined by $m = 5 + N$ equations (an overall equation of continuity, N equations of conservation of each component, two equations of motion, and two equations of energy) in $n = 5 + N$ independent physical quantities. Especially, when $N = 1$ there is no necessity of discrimating component, and it yields the result of governing equations as $m = 6$ (two continuity, two motion, and two energy equations).

(2) When N' ($N' \leqq N$) insoluble components are contained, the change of state of flow is determined by $m = 5 + N - N'$ equations (an overall equation of continuity, $N - N'$ equations of conservation of component, two equations of motion, and two equations of energy) in $n = 5 + N - N'$ physical quantities.

(3) The prescribed change of state of flow with a constant physical quantity, for instance, is governed by m equations in $m-1$ physical quantites, and the evolution is determined by any set of m–1 equations arbitrarily chosen under the condition of Eq. (12).

(4) The prescribed evolution to keep two phases in the state of thermodynamic equilibrium is governed by five equations (one overall equation of continuity, two equations of

motion, and two equations of energy) in four physical quantities, and is determined by any set of four equations arbitrarily chosen under the condition of Eq. (15).

(5) Even in a thermodynamic non-equilibrium two-phase flow, when the change of state of flow takes place very slowly the equilibrium analysis mentioned in item (4) can be employed by assuming the approximate equlibrium state, for which the deviation from the thermodynamic equilibrium state is estimated by Eq. (19).

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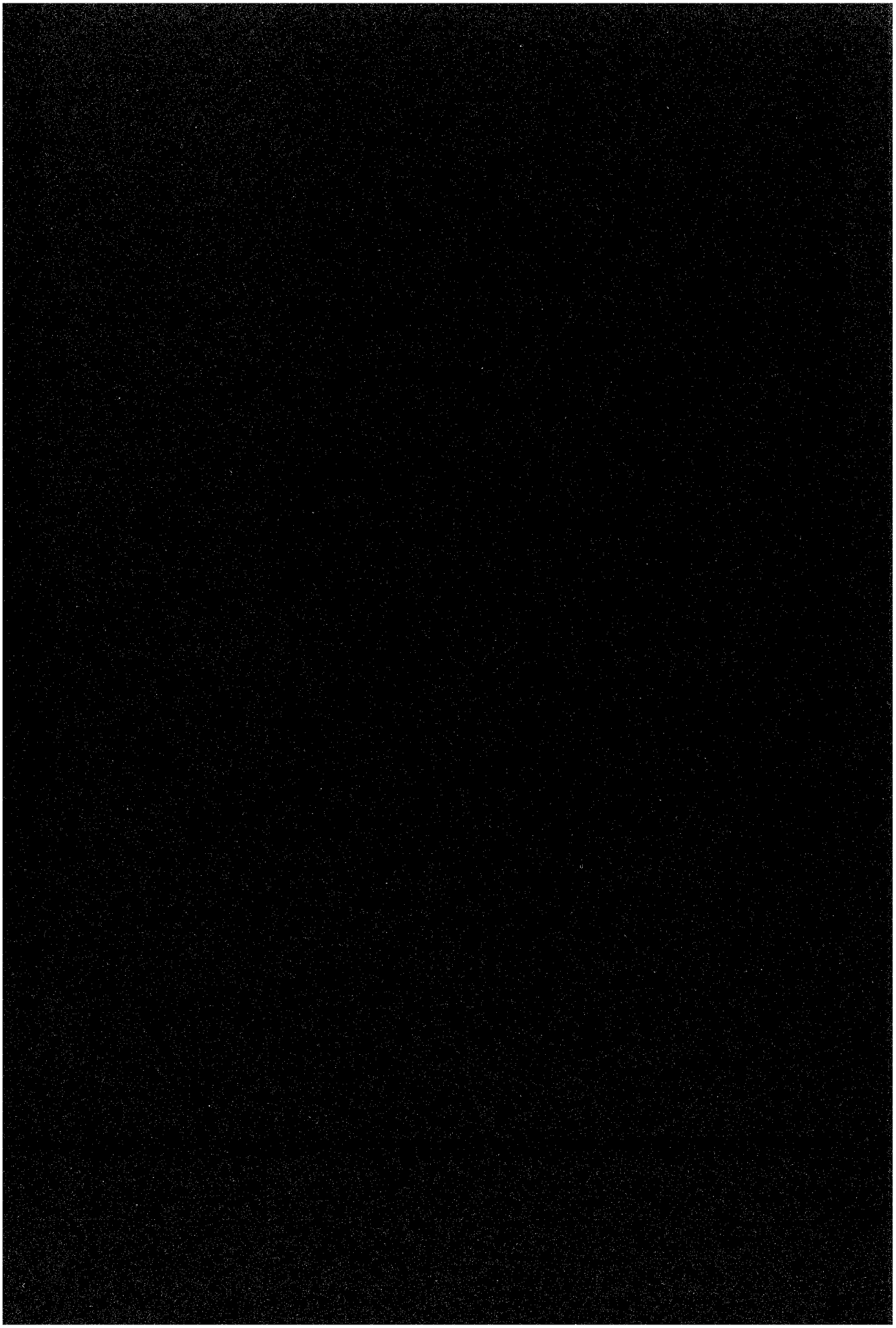
自然流動沸騰

外部流

内部流



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プール沸騰バーンアウト機構の研究*

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1. 序　論

最も基本的なバーンアウトとして，本報では飽和プール沸騰におけるバーンアウトの機構を論ずる．周知のように，従来この現象について各種の考え方や説が提出されている．代表的なものとしては，(I) 伝熱面に付着する蒸気塊の伝熱妨害によるとするもの，(II) 核沸騰の成立する流体状態に限界が来るとするものなどがある．そして後者はさらに二つの考え方にわかれる．すなわち第一は，いわば幾何学的理論で，Rohsenow と Griffith[1], Chan と Snyder[2]などのごとく伝熱面近傍での蒸気ほうの配置の限界を考える．第二は，いわば流体力学的不安定理論で，Kutateladze[3], Chan と Snyder[4], Zuber, Tribus および Westwater[5], 鳥飼[6]などのごとく対向して流れる気液流の界面の不安定を考える．

しかし，これらのバーンアウト機構モデルは，現実の沸騰やバーンアウトの実験事実と自然な形で結びつかぬ面が多い．たとえば導電ガラスを加熱面として用い，加熱面上の状況をガラスを通して裏側から観察した鳥飼と山崎[7], Kirby と Westwater[8] によれば，バーンアウト近傍でも加熱面（大きな蒸気塊におおわれている）上に液膜が存在する．しかも Kirby と Westwater は次のことがらを述べている．バーンアウト近傍の高熱流束になると乾燥はん点がときどき加熱面上に出現する．これらは拡大生長し，かつ隣どうしで結合したりするが，ある寿命を持ち，それがすぎると乾燥面は液体に洗われ，ふたたびぬらされる．そして彼らは，上記の乾燥はん点がたぶん，バーンアウトの前兆になるといっている．これらの観察結果は，前述のバーンアウト機構モデルとは相当に異質なものである．また，たとえば Costello と Frea[9]が示しているように，加熱面のぬれやすさなど加熱面性状がバーンアウト熱流束に影響を持つことが知られている．これもまた，前述の機構モデルの大部分で理解に苦しむところである．

上述の事情は，これまで提出されているバーンアウト機構の説には，まだ現象の本質についてかなり重要な見落しがあることを示している．これらの説では多かれ少なかれ，いわば静的な状態でバーンアウト現象を説明しようとしている．しかし沸騰，特に核沸騰や遷移沸騰は種々の面で時間的に急激な変動，つまり周期的ないし断続的過程を含む現象である．そして，もしそうなら，こうした沸騰機構と密接に結びついた形でこそ，バーンアウトの主要機構は求められるべきである．また一般にバーンアウト点は核沸騰領域の限界であると同時に，また遷移沸騰領域の限界でもある．したがって一方の領域に限界を与える現象は，他方の領域の沸騰機構と密接な関連があるべきものである．

それにしても一方，在来のバーンアウト理論が実験結果の整理に相当程度，成功している事実も忘れてはならぬことがらである．したがって本研究の開始時に著者らが当面していた事情はかなり複雑怪奇なものであった．ここで一つの逃げ道は，バーンアウトにいろいろの機構があるとすることである．しかし特殊な条件下に現われる副次的なことがらを除き，少なくとも本質的な機構がそれほど多岐にわたるとは考えられない．なにか，まだ見失われている知恵の輪を見出すことが必要である．

こうした見地から本報では，バーンアウト機構を従来よりもっと確実な形で解明するのに適切と考えられる特殊な沸騰形態を選び，高速カメラを用いて高熱流束プール沸騰の観測をおこない，かつ沸騰の観測事実だけを基礎にした解析をもおこなう．そして定性的観測，定量的解析の両者からバーンアウト機構の解明を試みると同時に，在来の説との関連，および遷移沸騰との関連についても検討を加える．

2. 実験装置

使用した沸騰装置の主要部を図式的に図1に示す．これは著者らの前報[10]に使用したものと同じなので詳細の説明は省く．加熱面は銅ブロックの上面で直径10 mm の水平面である．液体は大気圧の純水を使用した．高熱流束域では，この加熱面上に比較的単純な形状の蒸気塊が次々に生長し，バーンアウト熱流束はほぼ 1.6×10^6 kcal/m^{2}h の値を示す．

* 昭和 42 年 5 月 19 日　第 4 回日本伝熱シンポジウムにおいて講演，原稿受付　昭和 42 年 8 月 11 日．

** 正員，東京大学工学部（東京都文京区本郷 7-3-1）．

さて本報では3種類の相異なる沸騰形態について実験をおこなう．すなわち通常の沸騰のほか，図2の光学装置，図3の短い円管をそれぞれ垂直に，加熱面近くに保持した場合である．上記の光学装置は，前報[10]に使用したものと同じで，加熱面と同径のガラス底面を加熱面に近づけ，沸騰に干渉を与えつつ同時に加熱面上の流体の挙動が右側の窓を通して観測できる．他方，短い円管のほうは，外径が加熱面と同径，かつ高さは $H=3$, 10, 20 mm の3種類を用いた．これらは透明のアクリル樹脂製なので管内の流体の挙動が容易に観測できる．

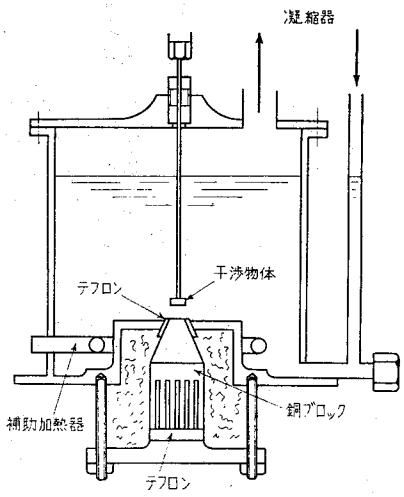


図1　実験装置主要部

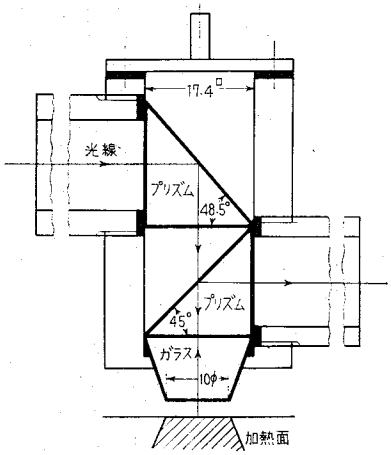


図2　光学装置

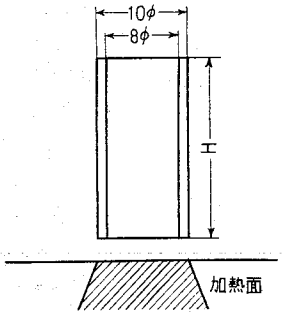


図3　円管

これらの干渉物体の底面と，加熱面との間のすきま距離 s を変えるとき，バーンアウト熱流束 q_c は図4に示すような特徴的変化を示す．特に円管を加熱面に近づけた場合，s の減少とともに q_c がかえって増加する領域があることは興味深い．他方，当然のことながら，s の値が十分大きくなれば，図4のどの曲線も自由加熱面のバーンアウト熱流束：$q_c \fallingdotseq 1.6\times10^6$ kcal/m²h に近づく．従来のバーンアウトの研究はほとんど自由加熱面についておこなわれているばかりでなく，上述のような q_c の変化の原因に対し，信頼し得る説明を下し得ない．また本報では故意に，そうした沸騰形態を選んだのである．

3.　高熱流束域における核沸騰

本報の結果について述べる前に，便宜上，高熱流束域における核沸騰について，従来の実験的知識のいくつかについて述べておく．

図5(a)は，自由加熱面上の高熱流束核沸騰の様相

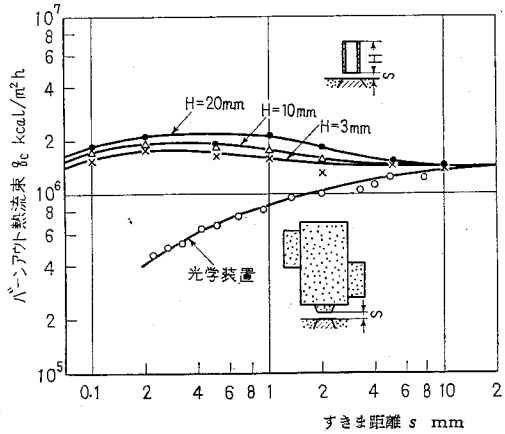


図4　すきま距離によるバーンアウト熱流束の変化

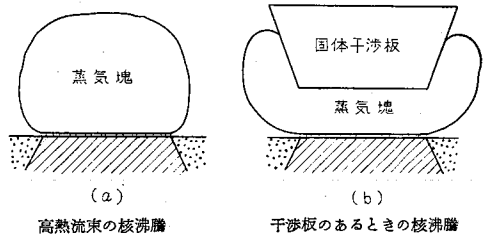


図　5

を模型的に示すもので，加熱面上に液膜があり，その液膜内にある多数の小蒸気柱から発生する蒸気が，上方に横たわる蒸気塊に供給されている．これに関し，Gaertner と Westwater[11] は，大気圧の水中においた銅製水平加熱面上の飽和沸騰の場合，上記の蒸気柱の平均直径 D が熱流束 q によって変化する模様を測定（めっき法による間接測定），図6に示す曲線を得ている．また Gaertner[12]は，彼の論文に付随する討論において次のことがらを述べている．すなわち，小蒸気柱の高さ δ と直径 D の比は熱流束 q に関係なく一定で次の関係がある．

$$\delta=0.6D \quad \cdots\cdots(1)$$

これは水平銅板上の大気圧水の高熱流束沸騰の側面写真の観測から求めたものである．

他方，図5(b) に示すように，蒸気塊の中へ固体干渉板を上方からそう入しても，干渉板と加熱面間の距離 s が液膜の厚さ δ にかなり近づくまでは，沸騰伝熱特性に変化が現われぬと推定される．図6には，伝熱特性に変化が現われ始める限界距離 s_f を，著者らがすでに前報[10]で測定した結果に基づいて示すが，前記の Gaertner と Westwater の結果と相当の相関関係があることを示している．

なお著者らはすでに，本報と同じ直径 10mm の自由加熱面上の蒸気塊の体積膨張速度を高速カメラで測定している[10]．それによれば，気塊生長の全期間にわたり蒸気増加速度はほぼ一定で，ばらつきを除いて時間的変化は認められなかった．これは，加熱面上の液膜からの沸騰蒸発が蒸気塊に蒸気を供給，かつそれが一定速度でおこなわれることを示す．

以上を総合するに，高熱流束核沸騰域では，加熱面上の液膜内の核沸騰と，その蒸気発生，および蒸気塊への蒸気供給という機構がほぼ確かである．なお図6

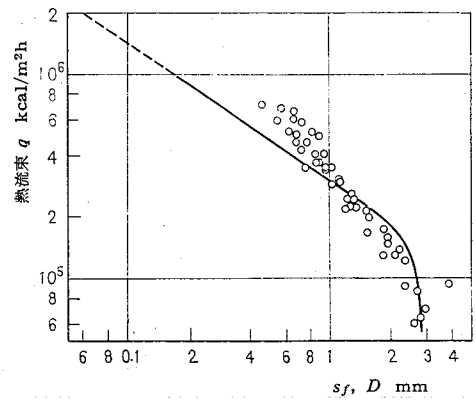


図6　限界すきま s_f（○印）および蒸気柱の平均直径 D（実曲線）の熱流束による変化．破線は実験結果の外そう

の曲線および式(1)の関係によれば，液膜の厚さは熱流束の増加とともにかなり薄くなり，大気圧の水の沸騰ではバーンアウト近傍でわずか0.1mm程度の厚さになることが推定される．ただし，ここで問題にしている液膜は，いわゆる薄液膜蒸発機構のそれではなく，後者に比べればまだ非常に厚いものである．

4. 液膜の形成と消耗

さて，前節に述べたような液膜がどんな機構で加熱面上に形成されるのか――これが重要な問題である．それを明らかにするため，流体挙動の忠実な観察をおこなった．

4·1 自由沸騰　代表的な例として図7を掲げるが，これは熱流束 $q=1.39\times10^6$ kcal/m²h の自由加熱面の近傍における，時間間隔 0.84 ms ごとの蒸気塊の様子を示すものである．ここで，ほぼコマ9～13において蒸気塊が加熱面から完全に切断し，コマ 14 に至るとすでに新しい蒸気塊が上方の水を排除しつつ生長をはじめている．したがって蒸気塊切断の期間はせいぜい 3 ms 程度であるが，一方，蒸気塊の完全切断の発生から次の完全切断の発生までの平均周期は，この場合約 60 ms であることが観測された．したがって，加熱面は常に蒸気塊でおおわれ，ただその間にきわめて短い切断期間が比較的等時間間隔でそう入されているということができる．

ところで加熱面上の蒸気塊は，周囲の液体を排除しつつ急激に膨張を続ける（前章に述べたように加熱面からの蒸気生成速度はほとんど常に一定）もので，したがって多かれ少なかれ蒸気塊近傍の圧力は周囲より高いはずである．そしてこの圧力は加熱面上にも作用するから，蒸気塊が生長しつつある期間に，周囲液体が加熱面上に流入（気塊と加熱面間の非常に薄い空間を通して）することは，ほとんどあり得ない．とすれば，加熱面上に液体が供給されるのは，加熱面から蒸気塊が切断する時だけであり（切断は蒸気と液体の置換があればこそ生ずる），それ以後は，その液膜を消費しつつ蒸気塊の生長がふたたびおこなわれることになる．

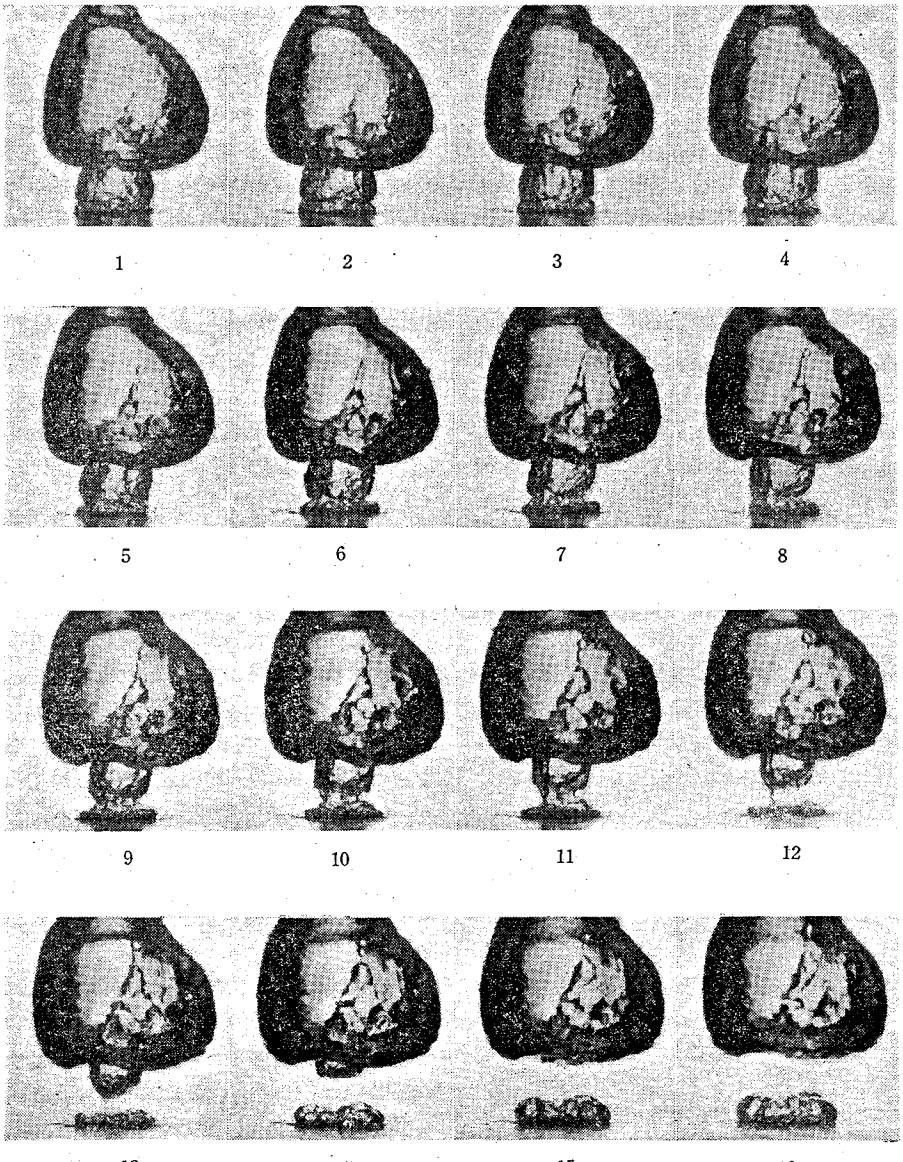
4·2 光学装置の干渉を受ける沸騰　図2の光学装置を加熱面上にすきま $s=2$ mm をへだてて保持した場合の沸騰を観察した．これは前節に述べた推定の検証にも役だつ．なお $s=2$ mm というすきまは，図6の結果から考えて，現在問題とする高熱流束域では加熱面上の液膜になんらの変化も与えないと考えられる．

実験にあたっては図2の光学装置の右側面をカメラ

の視野に入れる．これにより，加熱面近傍の沸騰の側面写真，および加熱面上の状況とが同時にフィルム上に記録される．

まず図8は，バーンアウト熱流束より低い熱流束での代表的な例を示す．時間間隔 8.6 ms ごとの液体の状況である．各コマの写真において，上方の円形の像が加熱面上の液膜の状況であり，下方部分には加熱面近傍の蒸気の挙動が側面から撮影されていることになる．さてコマ1では，加熱面上に新しい蒸気塊が生長を開始しており，コマ2では，この蒸気塊が加熱面上のすきまをみたしたうえ，周囲へリング状に押出されて来ている．なお加熱面上の液膜内沸騰は以後もずっと続くため，蒸気塊はどんどん膨張し，コマ7以降になると光学装置の外側壁に沿ってひろがる部分も生じている．しかしコマ 13，14 に至ると，上記の蒸気塊が急速な離脱を開始する．そしてすきま内に水が導入されるとともに，コマ 15 で最初の状態が再現され，以後同様なサイクル過程をくりかえす．

図8では，1サイクルの全期間中，加熱面上に沸騰液膜が残存する．しかしながら，加熱面温度をもっと



$q=1.39\times10^6$ kcal/m²h，時間間隔＝0.84 ms

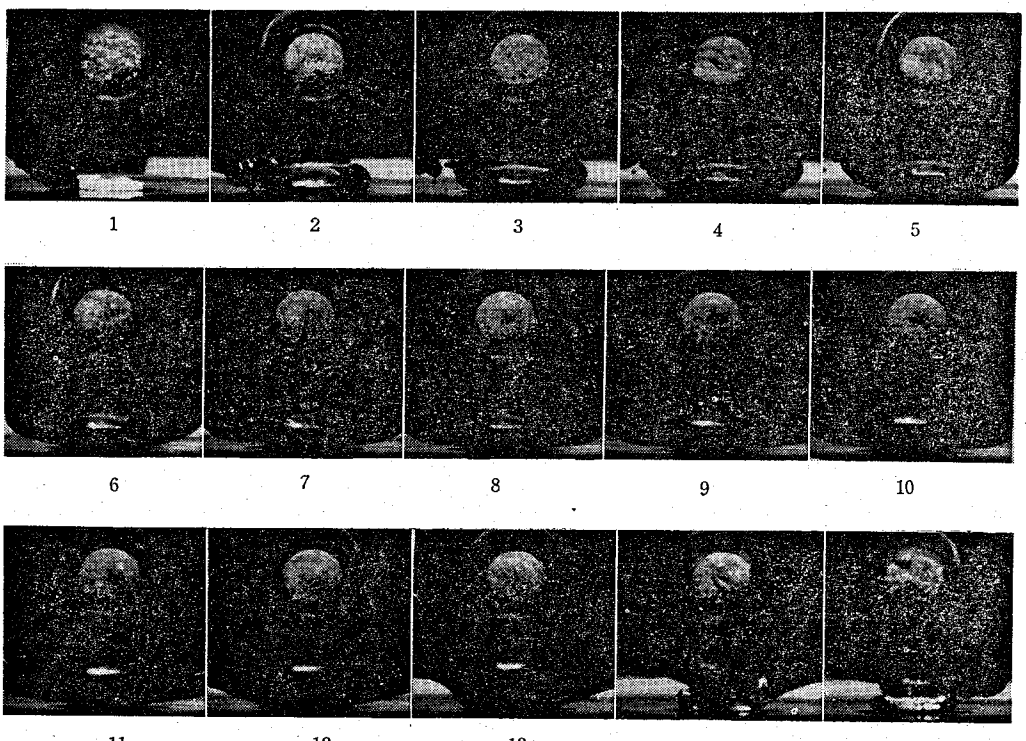
図 7　自由沸騰（核沸騰）における蒸気塊の変化

高くしてバーンアウト点に近づくと，時々サイクルの終わり(すなわち蒸気塊が離脱する前)あたりで，加熱面上に乾燥はん点が出現する．これは液膜の消耗の結果である．液膜の初期厚さ，および蒸気塊の滞留時間がサイクルごとに多少ばらつくため，たまたま液膜条件がきびしいサイクル過程の終段階で上記の現象を生ずると考えられる．図9は，バーンアウト近傍の加熱条件下で生じた一例を示すもので，多数の乾燥はん点のほか，やや広い乾燥部もみられる．そして，ここに述べた状況は，1章に引用した Kirby と Westwater の報告と非常に似ていることに注意すべきである．

さらに加熱面温度をバーンアウト点より高くすると，図10(各コマの時間間隔は 5.4 ms) のごとくなり，かならず加熱面の完全乾乾燥期間を1サイクル中に含む．図 10 は加熱面温度がバーンアウト点より 5°C 高い場合である*．なお光学装置を使用した場合，当然ガラス底面にも液膜が構成される．そして，しばしばこれに波が発生するのが観察されるが，現在は無用なのでこれを無視して図 10 をみてみよう．すなわちコマ1では加熱面上のすきまに水が浸入しつつ核沸騰が開始されている．そして以後コマ6くらいまでは，図8と同じ過程（液膜内の沸騰，蒸気塊の生長）を経る．しかし，そこで液膜が消耗しつくされ，その



図9　加熱面上の液膜．光学装置使用（$s=2$mm），バーンアウト近傍の核沸騰におけるサイクルの末期．液体は黒くみえ，また多数の白点は単に光の反射



$s=2$ mm，時間間隔 $=8.6$ ms

図 8　光学装置使用．バーンアウト熱流束より低い熱流束における核沸騰

* 本報のような実験装置でも銅ブロックの熱容量のため加熱面温度は比較的ゆっくり変化し，高速カメラのごく短い撮影期間中はほぼ定常的な遷移沸騰を保持する．

結果その後は，加熱面が完全に乾燥，かつ蒸気塊の生長も停止したままであるのがみられる．コマ 13～17 付近で蒸気塊の離脱がおこり，同時に水がすきま内に導入されてコマ 18 で最初の状態が再現される．なお蒸気塊の生長について図 8，10 を比較すると，図 10 のほうが 1 サイクル中に発生する蒸気量が明らかに少ない．これは，加熱面温度が高くなると，加熱面上に形成される液膜の初期厚さが薄くなることを意味する．

4・3 円管の干渉を受ける沸騰　図 3 の中空円管を加熱面上に垂直に保持する場合も，状況は本質的にまったく同様である．一例として図 11 は，$H=20$ mm, $s=0.8$ mm, $q=1.69\times10^6$ kcal/m²h の条件下で時間間隔 1.9 ms ごとの状況を示す．ここで興味深いのは，コマ 14～24 にみられるように，生長する蒸気塊の一部は，円管底部と加熱面間のすきまから周囲へリング状にはみ出すという，ちょっと予想外の事実である．しかし，このはみ出した蒸気も，コマ 25～27 にみられるごとく，蒸気塊の上昇離脱に際しては，きわめて急速に円管内部に吸いこまれて行く．そしてこのときに，はじめて加熱面上に水が供給される（円管と加熱面のすきまから）のであり，コマ 28 では加熱面上に水のしぶきがみられる．

5. プール沸騰バーンアウトの機構

前章に述べた諸観察結果はすでに，次のバーンアウト機構を暗示している．すなわち加熱面上の液膜の平均消耗時間と，蒸気塊の平均切断周期がひとしくなる点がバーンアウト点である．3，4 章に述べたところによると，加熱面上の液膜の消耗速度は熱流束に比例し，かつ液膜の初期厚さは熱流束とともに減少すると考えてよいようである．一方，蒸気塊の切断周期は一

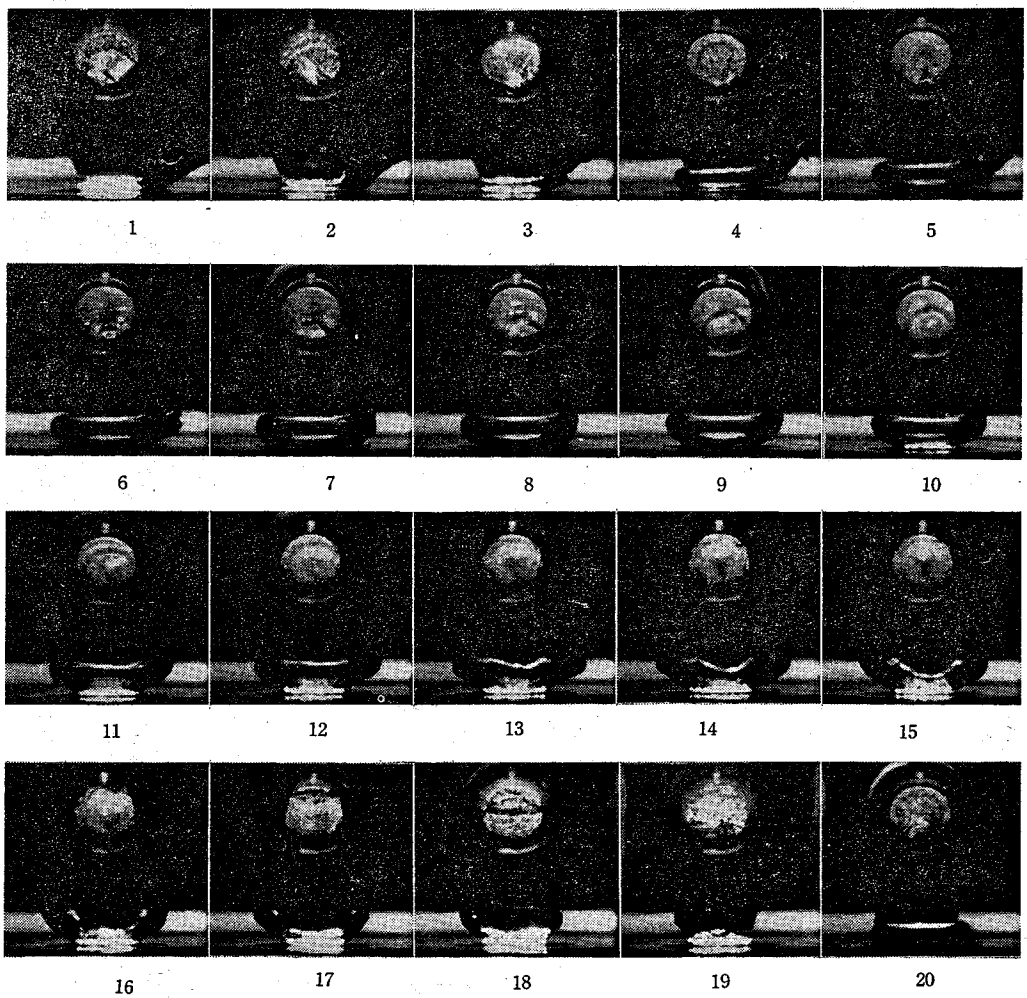
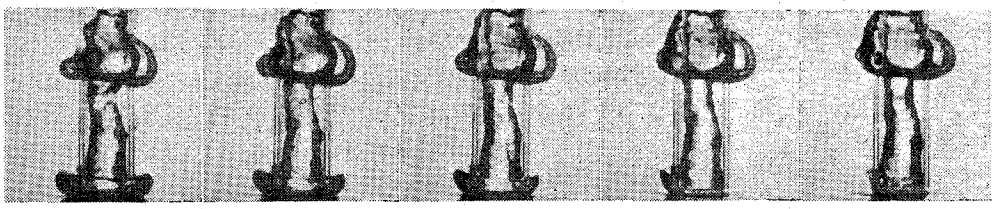
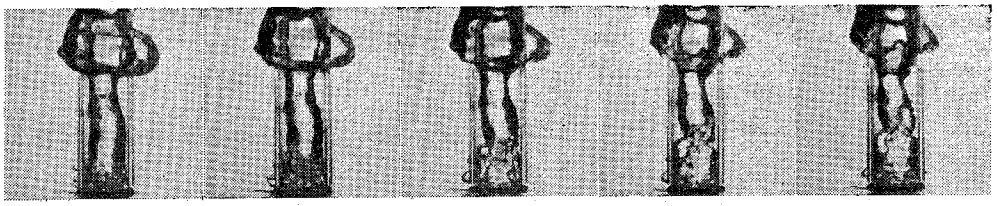


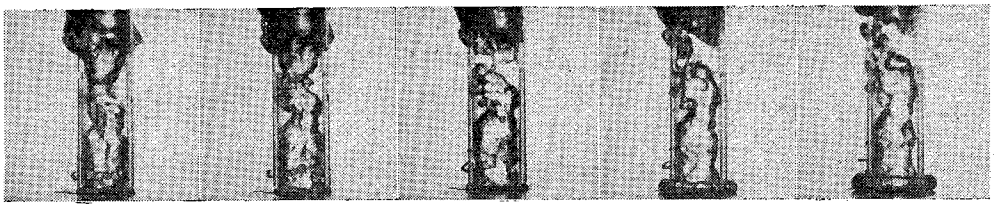
図 10　光学装置を使用（$s=2$ mm）としたときの遷移沸騰．時間間隔＝5.4 ms



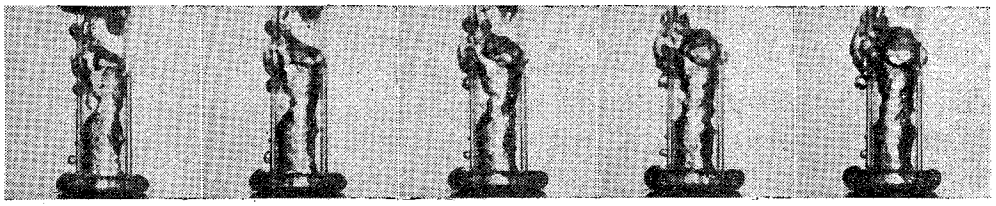
1 2 3 4 5



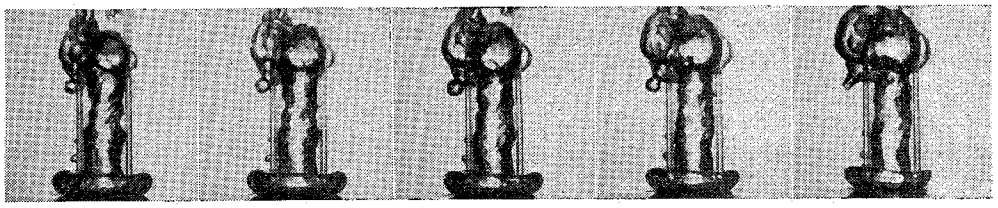
6 7 8 9 10



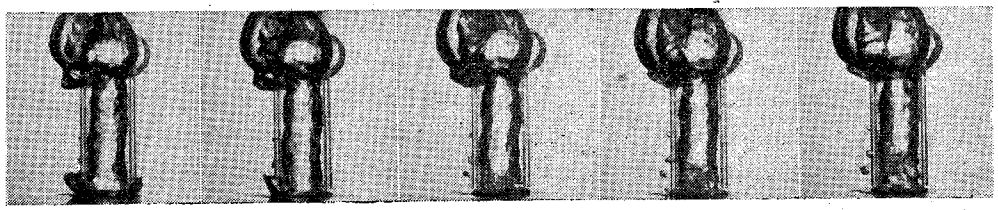
11 12 13 14 15



16 17 18 19 20



21 22 23 24 25



26 27 28 29 30

H=20 mm, s=0.8 mm, q=1.69×10^6 kcal/m^{2}h, 時間間隔=1.9 ms

図 11　円管使用の核沸騰における蒸気塊の変化

般に有限である．特に本報の実験の場合などは，その周期は高熱流束核沸騰領域で熱流束による変化はほとんどみられなかった．したがって熱流束を増加して行くと，上記の限界点が不可避的に出現することになるわけである．

いま，上記のバーンアウト機構を定量的な形で検討してみよう．面積 A_W の加熱面上に初期厚さ δ_0 の液膜が形成され，熱流束 q が保たれるとき，その液膜の消滅時間 t は

$$t=\frac{\rho_L\delta_0(A_W-A_W')L}{qA_W} \quad \cdots\cdots(2)$$

ここに ρ_L: 液体の密度，A_W': 液膜内の小蒸気柱の全断面積，L: 蒸発熱である．ここで t を計算するためには，A_W'/A_W および δ_0 を知らねばならない．A_W'/A_W の影響は大きいものではないが，Gaertner と Westwater[11] は，銅板上の大気圧の水の高熱流束沸騰で次の関係が成立することを観測している．

$$A_W'/A_W \fallingdotseq 1/9 \quad \cdots\cdots(3)$$

これは熱流束に関係しない．他方，δ_0 については式 (1) の δ を用いても，あまり無理はない．式 (1) は蒸気塊と加熱面間のすきま距離 δ を側面写真から求めたものであるが，加熱面周縁は常に周囲の液体に接し，したがってそこでの液膜の厚さは，液膜の初期厚さにかなり近いと考えられる．

式 (1), (3) を式 (2) へ代入，かつ D の値を図 6 の曲線から求めると，液膜消滅時間 t が熱流束 q の関数として図 12 の太い斜線のように定まる．

他方，3 種類の相異なる沸騰形態のそれぞれに対し，蒸気塊の平均切断周期 t をバーンアウト近くの熱流束 q で実測した結果を ○ 印で図 12 上に示してある．したがって，これらの切断周期に対応するバーンアウト熱流束の理論値は，図 12 の縦の破線位置で与えられることになる．一方，上記の切断周期測定と同じ実験で求めたバーンアウト熱流束実測値は図の縦の実線で示される．上記の理論値，実測値を比較するに，両者は相当によく一致し，3 種類のはなはだ状況の異なる沸騰形態に対してバーンアウト熱流束の相違の原因を統一的に説明することになる．ここで光学装置の場合の蒸気塊切断周期は，円管の場合の約 3 倍の大きさであり，これに対応して前者のバーンアウト熱流束は後者の約 1/2 になっている．

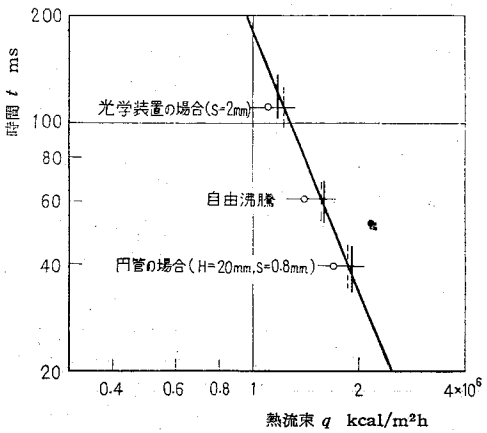


図 12 バーンアウト熱流束の解析値（破垂直線）と実験値（実垂直線）の比較

6. 検　　討

6·1 プール沸騰バーンアウトの種々の様相　これまで明らかにしたバーンアウト機構によれば，プール沸騰バーンアウトの種々の様相を統一的な形で説明することができる．液体のかくはんなどによって蒸気塊の居すわり時間を短くすれば，バーンアウト熱流束は増加する．また，蒸気塊の下に横たわる液膜に，なんらかの方法で液体を強制供給すればバーンアウト熱流束が上昇する．

逆に蒸気塊の切断周期をのばせばバーンアウト熱流束が減少する．これは本報の光学装置の場合にみられるのみならず，石谷ら[13]の実験でもすでに経験されている．すなわち，そこでは下向きの加熱面に，さらに蒸気塊の離脱を妨げる面を付加した場合，バーンアウト熱流束の低下することが示されている．加熱面が液体でぬれ難いときは，加熱面上への液膜の安定な形成が乱され，バーンアウト熱流束を低下させる．

液膜の初期厚さ，蒸気塊の滞留時間，その他はサイクルごとにばらつくため，バーンアウト点に近づくと加熱面上にときどき図 9 のような状況を生ずる．したがって熱流束一定の条件下などでは加熱面温度にある程度の脈動を生ずる．またバーンアウト近傍ではこうした状況下にあるため，最高熱流束点の近傍で伝熱特性が急激に変化するとは限らず，比較的ゆるやかな変化をみせる場合もあり得る．

なお熱流束を時間的に急激に増加させる過渡沸騰の場合，ここでは種々変わった現象も付随して生ずると考えられるが，少なくとも定常沸騰よりバーンアウト熱流束が高くなることの定性的説明は容易である．

6·2 在来のバーンアウト理論との関連　適用できる沸騰形態が普通の自由沸騰に限られるにしても，在来のバーンアウト理論から導かれる諸式が，バーンアウト熱流束 q_c の実験結果の大まかな整理に役だっているのは否定し得ない事実である．そして，たとえば Kutateladze の式は次の形に書かれる．

$$\frac{q_c}{L\rho_V}\Big/\sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}}=\text{const.} \quad \cdots\cdots(4)$$

ここに L: 蒸発熱, σ: 表面張力, g: 重力加速度, ρ_V: 蒸気密度, ρ_L: 液体密度である. 式 (4) 右辺の定数は, 実験によってかなり相違するが Kutateladze は平均的に 0.16 としている.

さて式 (4) は時間を含んでいないのに対し, 本報では時間が重要な因子として導入されている. しかし最初に述べたような事情がある以上, 普通の沸騰系だけを考えるならば本報の立場からも式 (4) に類似の結果が導かれなければならない. 不幸にして本報の実験範囲は非常に限られているから, 一般的かつ実証的な証明は不可能である. しかし次の解析を記しておくのは無駄でないと思われる.

本報の研究結果によれば, バーンアウト機構では特に二つの蒸気速度が重要な働きをしていると考えられる. すなわち一つは, 加熱面からの蒸気発生速度, 他は, 加熱面からの蒸気脱出速度である. そして前者は明らかに $q/L\rho_V$ であり, また後者は液中の気ほう上昇速度との類推からとりあえず $\sqrt[4]{\sigma g(\rho_L-\rho_V)/\rho_L^2}$ に比例すると仮定しよう (これらの項はともに速度の次元をもつ). この場合, 次元を考慮すると, 液膜の初期厚さ δ_0, および蒸気塊の切断周期 t は次のように書かれる.

$$\delta_0=\text{const.}\, l\left(\frac{\sqrt[4]{\sigma g(\rho_L-\rho_V)/\rho_L^2}}{q/L\rho_V}\right)^{1.5} \quad \cdots\cdots(5)$$

$$t=\text{const.}\,\frac{l}{\sqrt[4]{\sigma g(\rho_L-\rho_V)/\rho_L^2}} \quad \cdots\cdots(6)$$

ここに l は, 長さの次元を有する, ある項をあらわす. 式 (5) 右辺の指数 1.5 は, 式 (1) および図 6 の曲線の関係を満足するような δ_0 と q の関係からとるべき値である.

さて式 (5) の δ_0, および式 (6) の t を式 (2) に代入し, q を q_c と書けば

$$\frac{q_c}{L\rho_V}\Big/\sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}}=\text{const.}\left(1-\frac{A_W'}{A_W}\right)^{0.4}\left(\frac{\rho_V}{\rho_L}\right)^{0.1} \quad \cdots\cdots(7)$$

上式右辺で $1-(A_W'/A_W)$ はほとんど 1 とみなしてよい. また $(\rho_V/\rho_L)^{0.1}$ は比較的変化が少ない. したがって式 (7) は式 (4) にかなり近い性格を持つ. もちろん上述の解析は, とりあえずのものであるが, それにしても本報で明らかにしたバーンアウト機構と, 在来の理論結果 (その物理モデルは疑わしいが) の間になんらかの関係があり得ることを示しているといえよう. なお上の解析は単なる次元解析的なものではなく, 式 (5) 右辺の指数 1.5 が少なくとも一つ実験事実として考慮にはいっていることに注意すべきである.

6·3 遷移沸騰 バーンアウト機構との関連において遷移沸騰についてふれておこう. 4 章で述べたように, 加熱面温度をバーンアウト点より高くしたとき, 1 サイクルの途中で液膜が消耗し, 完全な乾燥期間が含まれるようになる. したがって厳密な一定加熱面温度条件以外の実験では, 加熱面温度がかなり変動する. また蒸気塊は初期の急激な生長ののち, 生長停止期間を伴うようになる (これは初期段階の蒸気の爆発的挙動をきわだたせる). そのため液体や蒸気の挙動に, 核沸騰領域との差異が現われて来るであろう. たとえば蒸気塊の切断時間など, 多少変化する. さらに高温の乾燥加熱面上への液体の流入に際しては, 面のよごれ, ぬれやすさなど加熱面性状が特に影響を持つことが考えられる. ここで, 核沸騰領域の加熱面は, 実質上, 常にぬれた状態に保たれていることに注意すべきである.

しかし次の簡単な解析を遷移沸騰に対して記しておくのは無駄でないと思われる. 図 13 は, 石谷ら[14]が銅加熱面上の大気圧の水の沸騰に対し, 加熱面温度と熱流束の関係を実測した結果 (○ 印および × 印) を示す. 太い斜線が核沸騰域である. ここで遷移沸騰領域を考えるに際し, 簡単のため次の三つの仮定をおこう. (I) 蒸気塊の切断周期はバーンアウト点のそれと同じである. (II) 液膜内の沸騰は核沸騰領域と同性格で, 核沸騰曲線の延長線 (図 13 に破線で示す)

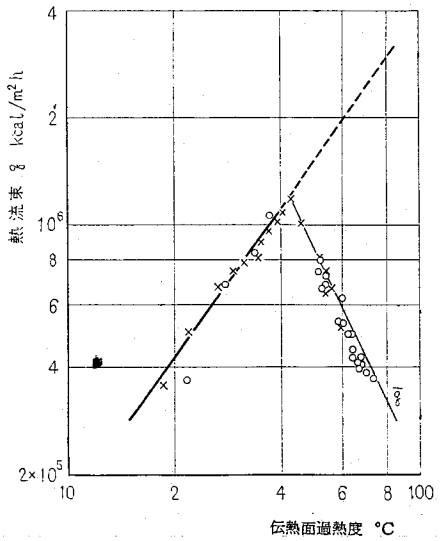


図 13 水のプール沸騰における伝熱面温度と熱流束. 実験値 (○, × 印) は石谷ら[14]による.

上の状態で生ずる，（Ⅲ）乾燥期間中の伝熱は無視できる．さて，いま液膜沸騰時の熱流束を q，これに対する液膜初期厚さを $\delta_0(q)$ とすれば，液膜の消耗時間 t は式（2）より

$$t=\frac{\rho_L\delta_0(q)(A_W-A_W')L}{qA_W} \quad \cdots\cdots(8)$$

同様に，バーンアウト点では

$$t_c=\frac{\rho_L\delta_0(q_c)(A_W-A_W')L}{q_cA_W} \quad \cdots\cdots(9)$$

ここに q_c はバーンアウト熱流束である．仮定（Ⅰ）により t_c は遷移沸騰域での蒸気塊の切断周期にひとしい．したがってさらに仮定（Ⅱ）を考慮すれば遷移沸騰における1サイクルの平均熱流束 $\bar{q}$ は $\bar{q}=q\times(t/t_c)$ であり，これに式（8），（9）を代入すれば直ちに次の関係を得る．

$$\bar{q}=q_c\frac{\delta_0(q)}{\delta_0(q_c)} \quad \cdots\cdots(10)$$

ここで q_c は図 13 のバーンアウト点の熱流束，また q は図 13 の破線から加熱面温度の関数として定められる．$\delta_0(q)$，$\delta_0(q_c)$ は式（1）および図6の曲線から定められる．このようにして計算した遷移沸騰域の熱流束 $\bar{q}$ は，図 13 に示すように実測値とかなりよく合っている．

もちろん上記の解析は，大胆な簡単化を含んだ形でおこなわれている．したがって，上記のような実験との対応のよさが普遍的にいつでも成立すると考えるべきではない．しかし少なくとも，ここで述べた遷移沸騰機構の本質は真実からそう遠くないと推定される．なお石谷と久野[15]が，伝熱面上の熱電対の温度変動から定性的に推定した遷移沸騰の機構は，本報のそれときわめてよく一致している．

7. 結　　論

（1）本報ではバーンアウト機構の確実性ある解明に特に有効と考えられる3種類の沸騰形態を選び，高速カメラによる現象の忠実な観察観測，および沸騰の実験結果のみに基礎をおいた解析をおこなった．

（2）プール沸騰バーンアウトは，加熱面上の液膜の消耗と，蒸気塊の周期的断続的挙動に関連した現象である．

（3）在来のバーンアウト理論が仮想している物理モデルは現実に非常に疑わしい．しかし，その最終結果に関する限りは，本報で明らかにしたバーンアウト機構から，その中に含まれる特殊ケースとして導出し得る可能性がある．

（4）核沸騰から遷移沸騰への移行は，沸騰形態の完全な変容によって生ずるものではない．大まかにいって主原因は単に，加熱面の完全乾燥期間が各サイクルに必ず含まれるようになるためで，核沸騰は依然として断続的な形で残存する（少なくともバーンアウト近傍で）．

終わりにのぞみ，本研究に協力された土屋孝夫君に深い謝意を表する．

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解説

536.423.1.01:532.5

プール沸騰にバーンアウトはどうしておこるか*

甲　藤　好　郎**

1. はしがき

沸騰はだれにもごく身近かな自然現象であると同時に，ボイラや沸騰水形原子炉などにみられるように工業的にもきわめて重要な役割を果たしている．ところで，この沸騰はたいへんおもしろい特異性格をもっている．つまり，流体と，それに接する加熱面の間に温度差があるとき，普通なら温度差を増せば増すほど伝熱量が増加していくのに，沸騰加熱面には，その一般常識のくずれる特殊状態があらわれる．

いまかりに容器の中に大気圧の水があり，飽和温度，つまり 100°C の水蒸気と平衡を保たせておくとしよう．この水の中に加熱面を入れ，その表面温度 T_W を飽和温度 T_S 以上にあげる．このとき，温度差 T_W-T_S が 2～3°C 程度までは水に普通の自然対流が生ずるのみだが（水全体が多少過熱され上部の自由水面から蒸発がおこる），T_W-T_S をもっとあげると，加熱面上の発ぽう点で蒸気を発生して，いわば普通の沸騰がはじまる．これを「核沸騰」という．なお加熱面温度をずっと高くすると，加熱面全体が薄い蒸気膜におおわれ水との直接接触がなくなる．しかし高温加熱面から薄い蒸気膜をとおして気液界面に伝えられる熱のため，そこの液面に蒸発を生じ，核沸騰とはまったく別の形の沸騰をおこす．これを「膜沸騰」という．ストーブの高温面上に水滴が静かにのっているときと同じ状況で，ただストーブの場合は水滴底面より発生する蒸気が水滴底部周辺から大気中に流出を続け沸騰の形態をとらぬだけである．ともあれ膜沸騰では，熱伝導率のきわめて低い蒸気の膜が加熱面をおおうため，伝熱特性は非常に悪くなる．

さて上記の装置（このように容器内の液体に関する沸騰をプール沸騰という）で，加熱面から単位面積，単位時間あたり伝わる熱量 q（これを以下，熱流束と呼ぶ）を測定してみると，前述の説明に対応して，温度差 T_W-T_S と熱流束 q の間に図1のような関係が得られる．普通，簡単のためこれを沸騰曲線といっている．抜山[1]によって発見されて以来，沸騰曲線が一般に図1の形をもつことは周知のことがらだが，核沸騰領域 BCD において T_W-T_S の増加とともに増して行く q は D 点で頭打ちになり，さらに T_W-T_S が増すとかえって q が減少する領域 DE を伴うのである．この特異領域の沸騰を一般に「遷移沸騰」と呼んでいる．

ここに D 点の存在は工業上，非常に重要な問題を提起する．すなわち図1からもわかるように核沸騰ではごくわずかの温度差 T_W-T_S で，一般にきわめて大きな熱流束が得られる．そしてわれわれが，やかんに水を入れて強く加熱しても，やかんが焼けないのはこの恩恵に浴しているわけだが，その恩恵にも限界があるということである．やかんはともかく，ボイラや原子炉となると焼けてはたいへんなことになる．この重要な限界点 D のことを本邦ではバーンアウト点ということが多く，その点における熱流束をバーンアウト熱流束という．本稿でもその用語に従っておくが，いかなる機構でこの限界点が出現するのかということ

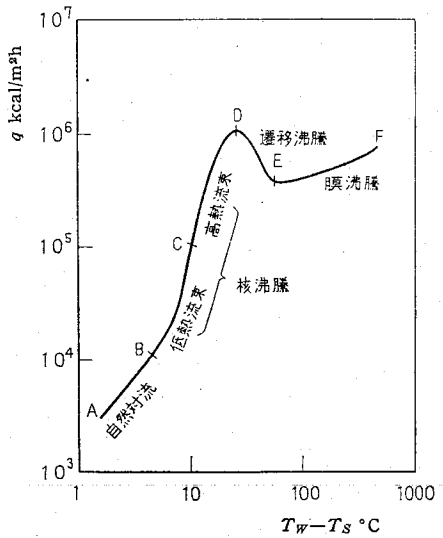

図1　沸　騰　曲　線

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が，知的興味からも，また工業上の必要からも問われることになるのは当然である．

2. 原因は果たしてなにか

もちろん読者のなかには，加熱面温度を高くすれば，いつかは核沸騰から膜沸騰に遷移せざるを得ず，しかも膜沸騰では伝熱特性がきわめて悪いから，図1のDEのような熱流束低下領域があらわれるのは当然と考えられる方もあろう．もちろん，これは間違ってはいない．しかし，これだけでは河川の水が必ず海へ流れこんでいる以上，地球上の陸地に降る雨量は陸地での蒸発量より多いというようなもので，真理ではあっても空々漠々としすぎている．バーンアウト機構という以上，水と蒸気がどんな状況になったとき発生するのかが問題である．

さてちょっとの間，話が固苦しくなるのを許していただきたいが，沸騰を生じている容器内にあるのは，液体と蒸気の動きにすぎず，この流体運動はあくまで質量保存，およびニュートンの力学法則（慣性力，静圧，重力，摩擦力，表面張力のつりあい）のもとに生じている．そして流体運動をおこすもとは加熱に伴う蒸気発生である．したがって原理は，連続，運動量，エネルギの各方程式ということであるが，非定常現象，また多体問題ともいえる不均一性，さらに表面張力を含むこと，加熱面の性状と考えてくると，たいへんな「流れ学」である．必然的に沸騰（ただし膜沸騰は比較的単純）は非常に複雑な現象で，事実，その伝熱のはっきりした機構はまだよくわかっていないといってよい．特にバーンアウト点に近い高熱流束域の核沸騰になるとそうである．バーンアウト機構もなかなか難問題で，従来いろいろなモデルに従って多くの学説や考えが提出されている（詳細は沸騰熱伝達[2]など伝熱関係の専門書を参照されたい）．そしてどの説もいいかげんに出されたものではなく，それぞれ一応の根拠の上にたっている．ただ，その根拠が本当に真実確実なものかということになると，これはかなり問題である．

むかしニュートンは，科学者が仮説をたてるにあたって守るべきことがらの一つとして，「われわれは観測された事実の説明に十分にして，かつ必要である以上の原因を仮定してはならない」といっている．必要にして十分な原因とは結局，唯一の基本真実になろうが，そこまでいかぬにしても，この見地から前述の状況を考えてみると，どうも必要にして十分という条件に欠けているようである．少なくともバーンアウトの原因根拠として考えるモデルが，現実の沸騰において確実に生じていると観察実証されない以上，いかにモデルから導かれる結果が都合のよいものであっても根拠薄弱ということになる．

3. 水の消費と供給

さて前章に沸騰や，その熱伝達は複雑な現象であると記した．とすると，われわれはバーンアウトの正確な機構に挑戦することをあきらめるべきだということになりかねない．だが現在，われわれが当面しているのは，かくかくの形の沸騰がなぜ生ずるのかという問題ではなく，その沸騰の存在を前提とした上で，それになぜ限界が生ずるかという問題である．もちろん両方の原因が一致するときは問題だが，無関係ならば沸騰自体は現象論的な立場にとどめておいてよい．遠心式や軸流式流体機械のサージング発生が，その圧力-流量の性能特性だけで論ぜられること[3]など，これに類するものといえよう．

いま，この立場に立って先に進んでみよう．この際まずたいせつな現象が一つある．すなわち，沸騰加熱面から伝えられる熱は，加熱面上の気ほう生成（蒸気生成）にすべて消費されているように一般の人はとかく考えやすい．しかし，これは事実に反し，孤立した気ほうが，加熱面上の各発ぽう点から上昇する低熱流束の核沸騰ではまだ自然対流的要素が残っているため，容器内の水自体も熱を受けるのである（それは，水中を上昇している気ほう界面，上方の自由水面の蒸発で吸収される）．もっとも熱流束があがるに従い，いわば真の沸騰になってくるとでもいえようか，とにかく上記の現象は消えてくる．そして高熱流束領域の核沸騰（ここでは加熱面上に密に分布した多数の発ぽう点から発生を続ける多量の蒸気があつまって，上方に大きな蒸気塊を構成するようになる）に深くはいると，全伝熱量が加熱面上での蒸気生成に消費されると考えてもよくなるのであって，これは高速カメラで蒸気塊の体積膨張速度を観測することで確められる．したがってバーンアウト点近傍では，熱流束 q に対応して，加熱面から単位面積，単位時間あたり生じてくる蒸気体積がはっきり定まる．あるいは，その蒸気が加熱面から一様に吹出すようにみなすと，その吹出し速度は

$$\text{吹出し速度}=q/L\rho_V \quad \cdots\cdots\cdots\cdots\cdots\cdots (1)$$

ここに L は蒸発熱，ρ_V は蒸気の密度である．

こうして熱流束 q が増加すると，それに比例して加熱面からの蒸気発生量も増すことになるが，蒸気が発生するためには，少なくともそれだけの水ないしそれ以上の水が加熱面に供給できる状況になっていなけれ

ばならない．しかも蒸気は加熱面から出ていくものであり，逆に水は加熱面に供給されるものである．したがってここに水の消費と供給の問題があり，しかも両者の間の相互干渉が熱流束の増加とともにひどくなってくることが容易に予想される．ところで，このように書いてくると，気のはやい読者のなかには，だからバーンアウトが現われるのは理の当然だといわれる方がおられるかも知れない．しかし，もう少し待っていただきたい．すなわち冷静にいって上記のことがらは，限界の出現に少なくともこうした可能性が一つあるということを記したにすぎない．ただ，それにしてもかなり大きな可能性を持っているといえよう．

そこで，とりあえず，この線に沿ってもう一歩進んでみることにしよう．この場合，水の消費と供給というばく然とした表現のなかに含まれ得るシステムの多様性が実は問題である．したがって，まず最も基本的な論理モデル（具体的なことにはふれず，主として論理的形式を問題にするので論理モデルと名づけておく）を考えてみるに，つぎの二つがあげられる．すなわち図2(a)は，交通規則のない一つの道を往復している人間群のごとく，まじり合った形で，しかも蒸気は上方へ，水は下方へ動いているという形である．つぎに図2(b)は，交通規則のある一つの道を往復している人間のごとく，蒸気と水が整然と通路をわけて流れている形である．そして人間の論理というものは誰でも同じだからであろう，従来の主要なバーンアウト説は，このどちらかに属しているともいえるほどである．

図2(a)の形式のものは，比較的初期に出たもので，Rohsenow と Griffith[4]，その他が考えた．歴史的にみても初期には，低熱流束の核沸騰の研究が多くおこなわれ，沸騰はとにかく小さな気ほうのできるものだという気分が強かったのであろう．そして加熱面から離脱上昇する孤立気ほう群の空間密度が，ある値になると限界になると考える．しかし，そう考えるというまでのことで，第一，バーンアウト近傍の沸騰は決してこのような形にもなっていないし，説得力がない．

これに対し図2(b)の形式のほうは非常に有力なもので，Kutateladze[5]，Zuber[6]，鳥飼[7]などが属している．もっとも，このなかで Kutateladze は，こうしたモデルはいっさい考えず，流体の運動という見地から次元解析的に次の関係を導いた．

$$\frac{q_c}{L\rho_V}\Big/\left[\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}\right]^{1/4}=\text{定数(無次元)} \qquad \cdots\cdots(2)$$

ここに q_c：バーンアウト熱流束，L：蒸発熱，ρ_V：蒸気密度，ρ_L：液体密度，σ：表面張力，g：重力加速度．けれども彼は，Zuber などに共鳴するようなことをいっているから，このグループに入れてよい．そしてこのグループでは，図2(b)にみるように逆方向に流れる気液二相をとにかく考え，その界面に不安定性の出現（風の強い日，海面が不安定になり，その結果としてしぶきを生ずるようなもの）などをもって限界になるとして式(2)と同じような結果を導く，式(2)の強味はなんといっても，ある程度おおまかながら，バーンアウト熱流束 q_c に関する実験結果を広い条件範囲にわたって説明できることである．

4. 時間の導入

けれども筆者らにとって図2(b)のモデルだけは，どうしても納得がいかない．なぜなら，いくら高速写真をとって水や蒸気の模様をみてみても，実際の沸騰にこうした状況は絶対にみられない．また後述の実験のように加熱面近くで気液が向流を生ずる必要のない系で特に実験してみても，バーンアウトの発生状況はあまり変わらない．つまりこのモデルは仮想的なものと断わぜざるを得ないのである．

そこでもう一度，水の消費と供給という基本に立ちもどって考えてみることにしてみたい．この場合，基本論理モデルはなにも図2(a), (b)だけが専売特許とはいえない．もちろん，これらは定常的な流体移動を考える場合の最も基本的な形だが，このほかに非定常的な基本形があるはずである．つまり間欠的な形で水が加熱面上に供給され，それら間欠的供給の間に，その水が加熱面上で消費されるとすれば，定常的な流動を考えなくてもよい．そして，これに対応して図3に示

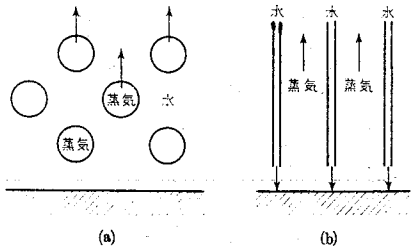

図2　定常・論理モデル

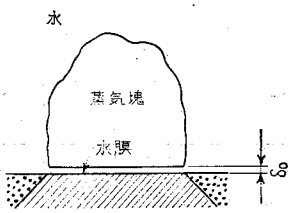

図3　非定常・論理モデル

す論理モデル（誰も無限に広い加熱面での沸騰など経験したことがないから，加熱面の広さは有限でよい）が得られる．ここで加熱面上の水膜内に生ずる沸騰（加熱面上には多数の発ぽう点あり）が，水膜を「消費」しつつ蒸気を出して蒸気塊を生長させ，この蒸気塊がある時間ののち，上方へ離脱すれば加熱面へ水の「供給」がおこなわれることになる．もし水膜が消耗してしまうより短い周期で蒸気塊が離脱するなら，加熱面上の水膜内には絶えず沸騰が続く．そして，その発生蒸気は直ちに次の蒸気塊を作りはじめ，上述のサイクルを継続する．

この非定常モデルの場合，水の消費と供給に限界を与えるのは，もちろん，水膜の消耗時間が，蒸気塊の離脱周期と一致したときである．いま新しい蒸気塊が構成された直後の水膜の厚さを δ_0 とすれば，熱流束 q（この q は水膜内の沸騰で吸収されるもので，水膜を通しての単純な熱伝導で伝わるものではないことに注意）のとき，その水膜が消耗してしまうまでに要する時間 t は容易に

$$t=\rho_L\delta_0 L/q \quad \cdots\cdots(3)$$

右辺の分子は，上記の水膜が加熱面単位面積あたり吸収し得る熱量である（水膜体積内には沸騰あり，少量の蒸気を含むが，本稿では簡単のため無視してある）．上式によれば，熱流束 q が大きくなるほど水膜の消耗時間 t が短くなりそうである．一方，蒸気塊は当然はやく生長するが，必ずしもそれだけはやく離脱するとはいえない．なぜならば蒸気塊の生長は周囲の水の排除を必要とし，これを逆にいえば水は蒸気塊を加熱面上にとどめておく作用をもつからである．したがって，熱流束の増加とともに，いつかは上述のサイクルに限界をきたす可能性があり得る．

もちろん，以上に記してきたことがらは頭の中だけの論理であるが，なお式 (2) と式 (3) の対応について付記しておく必要があろう．すなわち式 (2) は時間を含まないのに，式 (3) は時間を含み，両者はきわめて異質にみえる．しかし図3のモデルにおいて長時間の時間的平均（沸騰している状態をシャッタ解放でとった写真をみるようなもの）を考えてみると，これはある種の気液二相の流れともいえよう．したがって必ずしも式 (2) は式 (3) と異質なものではないといえるし，しかも Kutateladze はまったくモデルを想定せず，むしろそれは自然の選択にまかせて式 (2) を導いていることを忘れてはならない．このように考えると式 (3) は，式 (2) を特殊な場合として含むもっと一般的な形になる可能性をもつかも知れないのである．

5. 高熱流束領域の核沸騰

さてここで，とりあえず実際の高熱流束核沸騰がどんな姿なのかをみてみなければならない．一例として図4の写真は，大気圧の飽和水のなかに水平においた直径 10 mm の銅の加熱面上に大きな蒸気塊が発生している状況を示す．加熱面上方には，すでに以前，加熱面を離脱した蒸気塊が浮上している．加熱面の直径が 30～50 mm くらいに大きくなっても，蒸気塊の形が複雑になるだけで本質はあまり変わらない．ともあれ図4に例示される実際の沸騰状況にくらべ，図2(b)のモデルはまったく異質なものであることがわかるであろう．

なお図4のように蒸気塊でおおわれていても，その下の加熱面上には常に水が存在することは，いろいろの方法で確かめられているし〔鳥飼・山崎[8]，Kirby－Westwater[9]，甲藤・横谷[10]〕，またその水膜内の沸騰なら蒸気が供給されるからこそ，蒸気塊が体積膨張を続けているのである．ついでながら，こうした水膜の初期厚さ δ_0 は熱流束 q が高くなるほど薄くなり，ほぼ

$$\delta_0 \propto 1/q^{1.5} \quad \cdots\cdots(4)$$

の関係をもち，また大気圧の水での沸騰の場合，バーンアウト点近くで 0.1 mm 程度の厚さ*である[10]．つまり，かなり薄いものだが，薄液膜蒸発機構（核沸騰の伝熱機構として考えられている説の一つ）で問題にしている厚さ数 μ の薄液膜にくらべれば，はるかに厚

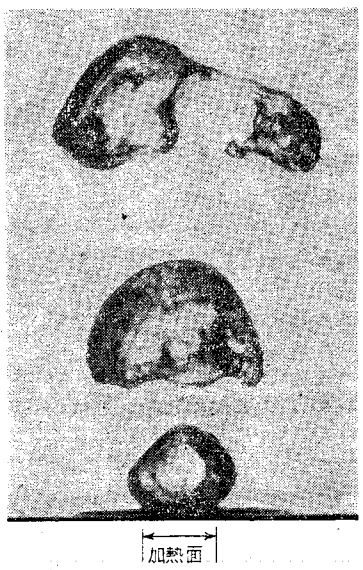

図 4　高熱流束核沸騰の蒸気塊

* したがって加熱面の寸法が極端に微小（極細線など）のときを除き，本文に述べた沸騰形態が成立する．

いものである．したがってまた，そのなかに核沸騰がおこり得るのである．

蒸気塊が大きくなって加熱面から離脱する平均周期は，同一の加熱面に対し熱流束によってあまり変わらない．詳しくいうと熱流束の増加とともに周期が多少のびる傾向がみられるようであるが，たいしたものではない．ちなみに，われわれがいま取扱っている蒸気塊は，低熱流束域の核沸騰で加熱面上の発ぽう点に生ずる小さな孤立気ほうとはあらゆる面で異なるものである．たとえば加熱面に付着した孤立気ほうでは周知のように付着したところの表面張力に，気ほうの浮力がつりあうところで離脱するというようなことを考える．しかし図5に示すように，加熱面上の水膜上にのっている形の蒸気塊では，その浮力が表面張力をとおして加熱面と結合することはなく〔甲藤・菊池[11]は加熱面にかかる力を測定して確かめている〕，したがって離脱の機構も異なる．むしろ離脱というより，切断ないし浮上に近いものであるといえよう．

なお参考までに，水中にある気体の挙動について次のことを注意しておこう．図6(a)は水中にある固体球表面に作用する水圧分布を示したもので，周知のように，その合成力が浮力として固体球に作用し浮上させようとするだけである．一方，図6(b)は水中に球状気体がある一瞬間を考えたもので，その表面には前と同じ水圧分布が作用する．しかし気体の圧力は一様だから平均として Q の水位の水圧に近く，したがってたとえば P の水位では水圧が気体圧力より高く，R の水位では水圧が気体圧力より低くなる．当然，その圧力差に従い流体に流動が生じ，図に破線で示すような気液界面の変形がおこり，これが同時にいわゆる浮上になる．つまり，気体の浮上現象は，それをも含めた流体全体の流れ（気体体積の保存という拘束下に）の問題である．

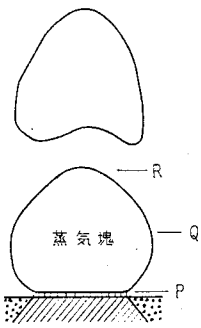

図 5　高熱流束核沸騰の蒸気塊

ただ，上にはわざとことわり抜きで説明をしたが，いつでもこうなるわけではない．すなわち球状気体が小さくなると，気ほう表面に働く水圧分布はほとんど一様になる一方，表面張力の働きが大きくなる．したがって，ある限界以下の小気ほうは球形からわずかにみずからを変形することにより，表面上のすべての点で力のつりあいを保ち，その形を保存できるようになる．ビールやサイダのなかに出てくる小気ほう，低熱流束域の核沸騰において生ずる孤立気ほうなどみなこれである．なお Peebles と Garber[12] の実験によって，直径 8 mm 程度以下の気ほうが液中を浮上する速度 u_b は次のようになる．

$$u_b=1.18\left[\frac{\sigma g(\rho_L-\rho_V)}{\rho_L{}^2}\right]^{1/4} \quad \cdots\cdots\cdots\cdots(5)$$

すなわち u_b は，表面張力 σ，重力加速度 g，液体密度 ρ_L，気体密度 ρ_V に関係し，気ほうの大きさなどには関係しない．

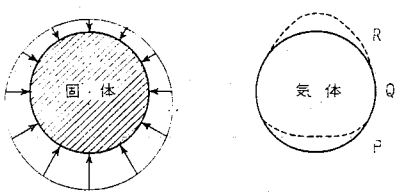

図 6　水中にある固体球と球状気体

6. 自然の論理

前章に記した高熱流束の核沸騰の現実の姿を考えてみるのに，これは加熱面での水の消費と，それへの供給について考えられる3種類の基本論理モデルのうち，まさに図3の非定常モデルの形式をとっているといえる．もちろん，いろいろなバリエーションはあるにしても，少なくともこの基本線は明らかであろう．またさらに次の一般常識：

(1)　液体中に気体ができるとき，それは気ほうにせよ気塊にせよ，とにかく液中でかたまりになる．

(2)　沸騰の場合，気体は加熱面上で生成される．

を考えてみるとき，自然が非定常モデルを採用するのは自然構造に合致したものであると考えられる．すなわち，この場合加熱面には気体のかたまりでおおわれる部分が必ずでき，その部分の加熱面への液体の供給は，(イ) 気塊を取除いておこなうか，あるいは (ロ) 気塊底面と加熱面の間に薄い液体通路があるかである．

さて読者は，この(イ)と(ロ)のどちらも可能だと考えられるであろう．たしかに，これだけではそういわざるを得ない．しかし加熱面上の水膜から蒸気をうけて生長を続けている蒸気塊は周囲（特に上方）の水を排除している．いま図5について考えるのに，蒸気塊が加熱面上に底面をおいていすわっている以上，その内部の蒸気圧力は少なくとも P の水位の水圧にひとしい．つまり蒸気塊はこの圧力で加熱面上にふんばり，上方の水を押しあげて行くのである．したがって図5で薄い水膜内の圧力は P 点の圧力にほとんどひとしく，水が定常的に外部から蒸気塊下部へ供給されることはない．結局，前述の(イ)，(ロ)のうち，(イ)だけが残ることに

なる．

一般に自然対流は一種の熱機関であり，たとえばわれわれは地球上に生ずる自然対流から多くの動力を得ている．そして単相流体に生ずる自然対流は，無限に微小な熱機関が場全体に無数に分布した形式に近い．しかし沸騰の場合は，加熱面上の気ほうや気塊を生ずる各箇所に，かなり熱機関が集中しており，しかもそれは一種の容積形熱機関（吸気と排気が間欠的）であるといえようか．容積形熱機関は単純にシリンダとピストンだけで構成できるのに対し，速度形熱機関は圧縮，加熱，出力の三系統を直列に組合わさねばならず，かなり技術を要する．したがって沸騰に出現するにはいささか無理であろう．

なお少しあいまいだが次の重要なことがらを記しておこう．すなわち，これまで記してきた水の消費，供給の機構によれば，消費に対して供給がぎりぎりになる限界の出現に対し，特に二つの蒸気速度が重要な働きをするように考えられる．すなわち一つは，加熱面からの蒸気発生速度，他は，加熱面からの蒸気脱出速度である．そして前者は式(1)であらわされ，また後者はとりあえず式(5)の値に比例すると仮定しよう．この二つの量（ともに速度）で現象が支配される場合，次元を考慮するならば，水膜の初期厚さ δ_0（長さ），蒸気塊の切断周期 t（時間）は次のように書かれる．

$$\delta_0 = \text{定数} \times l\left(\frac{\sqrt[4]{\sigma g(\rho_L-\rho_V)/\rho_L^2}}{q/L\rho_V}\right)^{1.5}$$

$$t = \text{定数} \times \frac{l}{\sqrt[4]{\sigma g(\rho_L-\rho_V)/\rho_L^2}}$$

ここに l は，長さの次元をもつ量（ただし物体の寸法などである要はない）である．また第一式右辺の（ ）につけた指数1.5は，δ_0 と q が式(4)の経験的関係を満たすためとるべき値である．さて上記の δ_0, t を式(3)に代入し，q を q_c と書けば

$$\frac{q_c}{L\rho_V}\Big/\left[\frac{\sigma g(\rho_L-\rho_V)}{\rho_L^2}\right]^{1/4} = \text{定数} \times \left(\frac{\rho_V}{\rho_L}\right)^{0.1} \quad \cdots\cdots(6)$$

ここで右辺の $(\rho_V/\rho_L)^{0.1}$ は指数がわずか0.1なので，その変化はかなり少なく，したがって上式は式(2)に非常に近い性格のものである．つまり，われわれは非定常モデルを基盤に，しかも式(2)に近い関係すら導き出せる．もちろん多少おおまかな話だが，それでも上述の解析で物理的な面はそれほどいい加減なものではない．それを思えば，われわれは真理からあまり遠くないところにいるといえるだろう．

7. バーンアウトの諸相

水の消費と供給を前提とした場合，バーンアウト機構はどうあるべきか，いわばその物理的な哲学について考え，また関連した事項を記してきた．では，バーンアウト法則式(3)は現実のバーンアウトと対応するであろうか．もちろん式(3)で，t はバーンアウト点における蒸気塊の離脱周期，q はバーンアウト熱流束にとる．この問題に対し，甲藤・横谷は既報[10]およびその後の研究[13]における各種の実験で定量的な検討をしており，少なくともそれによれば上記の質問に対する解答は完全に肯定的である．また式(3)によれば，従来の考え方では説明も解釈もできなかったいろいろの現象が統一的に説明されるだけでなく，これまで手をふれるのがためらわれたような多様な沸騰条件の場合にまで一挙にわれわれの視野をひらいてくれるのである．以下，そうしたことを含めてバーンアウトのいくつかの様相について記してみよう．

7·1 加熱面の配置，形状，大小などによる q_c の変化　たとえば「大気圧の飽和水のプール沸騰」というように流体側の条件を固定したとき，蒸発熱 L，表面張力 σ，液体密度 ρ_L，蒸気密度 ρ_V など一定になる．したがって式(2)によれば加熱面の配置（水平とか垂直とか），形状，大小，その他いっさいにおかまいなく q_c＝一定である．しかし現実のバーンアウトはそうはならない．これは従来，周知の事実であるが，その原因は不明のままに（あるいは二次的なものと考えて），式(2)の定数も多くのデータの平均値をとって定められるのである．

これに対し式(3)は，式(4)を考慮して次の関係を与える．

$$q_c = \text{定数} \times (1/t^{0.4}) \quad \cdots\cdots(7)$$

つまり流体側の条件が同じでも，加熱面の状況その他で蒸気塊の周期 t が変わると，バーンアウト熱流束 q_c が変化することを示す．そして事実，高速カメラで t を測定してみると，上式の関係がほぼ成立することが知られる．ちなみに大気圧の水中に直径 10 mm の円形加熱面を水平，上むきにおいたとき，$q_c = 1.6 \times 10^6$ kcal/m²h，$t = 0.06$ s 程度であることを付記しておこう．

7·2 蒸気塊の脱出を妨害した場合　石谷ら[14]は，図7(a)に示すように下向きの加熱面に，さらに蒸気塊の離脱を妨げる面を付加した実験をおこない，この妨害面を大きくするほどバーンアウト熱流束が低下することを示している．この場合，図に示すように蓄積されていく蒸気塊の先端が，妨害面の端部をはみ出た部分で蒸気塊は急速にその方向に流出，一方に逃げ去る．したがって妨害面が大きいほど蒸気塊のいすわり時間は長くなり，式(7)の t の増加とともに q_c が

減少する．

同じような例として図7(b)は，上向きの加熱面の上方に妨害物体をおいた場合[10]で，蒸気塊は図示のような形で生長，ある大きさになると，やはり一方向に急速に流出離脱する．妨害物体にガラスを使ったのは，その下の蒸気塊（透明）を通して加熱面上の水膜の状況を観察できるようにしたもので，その結果についてはあとで述べることにしよう．ともあれ，妨害物体を加熱面に近づけて行くほど q_c が減少していく．そして蒸気塊の平均離脱周期 t を測定すると，式(7)が検証される．

7·3　加熱面に垂直に円筒を保持した場合　3章に記したように，図2(b)に示すモデルを基本とするバーンアウト説では，互に逆方向に流れる気液二相の不安定現象に問題を帰そうとしている．しかし，そうした安定問題がバーンアウト現象に果たしてからむものか否か．考えてみると，これはかなり重要な問題である．なぜなら，水の消費と供給という立場を同じくしても，不安定説ではバーンアウト点で水の供給系統が破損してだめになるという考えであり，これに対しわれわれのほうは，なにも供給系統に異常がおこっているわけではなく，ただ消費に対し供給がぎりぎりになる（まだ供給はちゃんとおこなわれている）だけだという考えである．

さて図7(c)は円形加熱面とわずかすきまをおいた形で円筒を上部に保持した場合[10]である．こうすると水は下部のすきまから供給される一方，蒸気は円筒内を上昇する．つまり，加熱面にむかう水と，加熱面からたちのぼる蒸気が逆方向にすれちがって動く必要はなくなる．したがって不安定現象説によるなら，バーンアウト熱流速はずいぶん上昇していいはずである．けれども実験してみるとそうはならない．のみならず流体の挙動を観察すると，つぎつぎにできる蒸気塊が図に示すようなおもしろい形にまで生長した後，急速に円筒内を上昇離脱する．このとき，すきまからはみ出していた部分の蒸気も円筒内にすっと吸いこまれ，それを追う水がすきまを通って加熱面上に供給される．

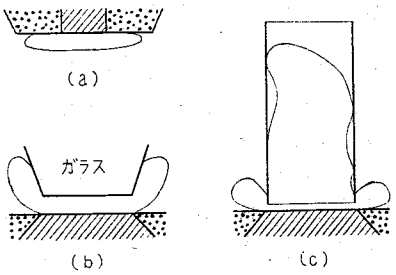

図 7 · 各種の沸騰系

つまり水の供給はやはり間欠的であり，しかも蒸気塊の離脱周期を測定すると，式(7)の関係でバーンアウト熱流束と合致するのである．

7·4　バーンアウト点近傍における加熱面の状況

バーンアウト点とは要するに，加熱面上の水膜が蒸発消滅するまでの間，加熱面上に蒸気塊がちょうどいすわっている点である．つまり加熱面が常に水でぬらされているぎりぎりの点ということになる．そしてそれに間違いないが，ただもっと詳しくいうと空間的および時間的平均でいってそうだということである．これについて以下に説明しよう．

第一に，水膜といっても単純一様に平均化されて加熱面上に静止してのっているのではなく，そこに沸騰を生じている．薄い水層内の沸騰であるから，蒸気ドームなどを発生する部分がある．なおバーンアウト点近傍では，一つのサイクルの終わりごろ，水膜の平均厚さが極端に薄くなるため，沸騰が停止した状況も出現する．しかし，この場合にも加熱面が一様に水でおおわれるわけではない．すなわち加熱面は図8に示すように，それまで発ぼうしていた多数の点を中心にした大小さまざまの乾燥部分と，それ以外の水におおわれた部分とになる．こうしたことがらはすべて，図7(b)の方法で明らかに観察される．したがって鳥飼・山崎[8]のように，加熱面(導電ガラス使用)を裏側から透視し，ぬれている部分と，そうでない部分の割合の測定を試みると，加熱面の何割かは乾いているという結果になる．ついでながら上記論文には，バーンアウト点で加熱面は100%ないしこれに近く乾いていると「従来」は予想されたという意味の記述がある．これに対し「現在」の立場では，加熱面が100%ぬれている状態の限界がバーンアウト点である．同一の自然現象を考えながら，天と地ともいうべき違いがでているのはおもしろい．なお Kirby と Westwater[9] は，同様に加熱面を裏側から透視し，つぎの観察結果を述べている．

“バーンアウト近傍の高熱流束になると乾燥はん点がときどき加熱面上に出現する．これらは拡大生長し，かつ隣どうし結合したりするが，ある「寿命」をもち，それがすぎると乾燥面は液体で洗われ，ふたたびぬらされる”（ここで「　」は筆者がつけたもの）．この結果にもとず

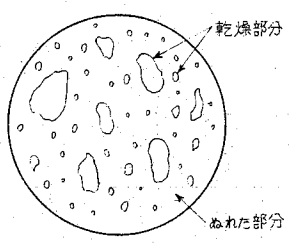

図 8　加熱面の状況

き彼らは，上記の乾燥はん点がたぶん，バーンアウトの前兆になるといっているが，それはともかく上の観察結果はまさに，われわれの考えてきた機構そのものなのである．

さて第二の問題は，サイクルごとに蒸気塊の離脱の模様，周期，あるいは水膜の初期厚さなどにある程度のばらつきのあることである．特にこのばらつきが大きい場合は，バーンアウト点というはっきりしたものではなくなり，その近傍で伝熱特性が比較的ひろがりのある変化をみせたり，また加熱面温度にある程度の脈動を示したりする．

7·5 バーンアウト熱流束に対する加熱面性状の影響 この問題は従来，バーンアウト機構の論争において両刃の剣であった．すなわち式(2)によればバーンアウトは，あらさその他の加熱面性状から無縁なものであり，したがって流体力学的な原因のみで生ずるものということになる．また実際，多くの場合そうであって，流体力学的不安定説の支持者にとり，きわめて大きな拠点となるものであった（たとえば Berenson[15]）．ところが一方，加熱面性状にまったく無縁かというとそうではない．液体に対する加熱面のぬれにくさなどは確かに影響する．そこで，たとえば Costello-Frea[16] などは，逆にこれを根拠として上記の説に疑念を表明していたのである．

さて加熱面上の水膜からの蒸気発生，それに伴う蒸気塊の生長，周囲の水の排除，ついで蒸気塊の離脱による給水といった機構は，流体の挙動に関連したものである．したがって加熱面上に水膜が普通に構成されるものならば，加熱面の性状は現象に無関係である．たとえば加熱面のあらさなどは，加熱面が水を受け入れる難易さ自体には，影響してもごくわずかである．もちろんそれは水膜内の発ぽうには大きく影響するから，伝熱特性を非常にかえるけれども，バーンアウト熱流束はほとんど変えないことになる．これが，いわば普通一般の場合である．

しかし，たとえば加熱面が水を非常に受け入れ難い（加熱面上にテフロンの薄層が被覆されているときなど）といった場合はどうなるであろうか．このときは式(3)に含まれる水膜の初期厚さ δ_0 が，いわば減少した形の効果をもつことになる．なお流体側の条件は不変だから t は変わらないと考えてよい．したがってバーンアウト熱流束 q_c は減少せざるを得ないのである．

7·6 その他 以上，バーンアウトに関連するいろいろな問題の説明をしてきた．なお，低水位の場合[13]（加熱面が非常に浅いところにあるとき），加熱面とは別熱源からも多量の蒸気を発生させた場合[13]，あるいはまたサブクール沸騰[13]（容器内の大部分の水の温度が飽和温度より低いときの沸騰），過渡沸騰（加熱面からの熱流束が時間的に急速変化しているときの沸騰）など，述べるべき興味あることがらも多いが，紙数の関係もあるので割愛する．

8. 遷移沸騰への道

これまで，加熱面上の水の消費と給水とがぎりぎりにつりあう点としてバーンアウト点を説明してきた．しかし実は，これだけではバーンアウトを完全に論じたことにならない．図9は，バーンアウト点近傍の沸騰曲線を，わかりやすさのため多少模型化して CDE と示すものであるが，ここで D 点が上記の状況の点であるにしても，それでは一体なぜ沸騰曲線が，ここで上に凸の形で折れまがり（なだらかにまがってもいい），D 点を熱流束の極大点とするのであろうか．安定な沸騰領域は，いわゆる核沸騰，膜沸騰の両領域のみで，遷移沸騰は存在しないというのなら話は別だが，その存在はいろいろの実験で示されている．したがって D 点の折れまがりの正当な説明が与えられねばならない．

さて図9をみて，つぎのことが直ちにいえる．すなわち遷移沸騰領域 DE にはいると，核沸騰領域 CD とはなにか変った現象が生じている（同一の熱流束 q の値に対し両領域は異なる温度差 T_W-T_S をもつ）．しかしここにたいせつなことは，領域 CD と領域 DE が D 点で連続していることである．つまり，D 点は核沸騰領域 CD の限界点であり，同時に遷移沸騰領域 DE の起点である．このことは，上述の「なにか変わった現象」の発生原因が，バーンアウト D 点を規定した現象と密接な関連をもつことを示す．

ところで図10(a)は，バーンアウト以前の核沸騰領

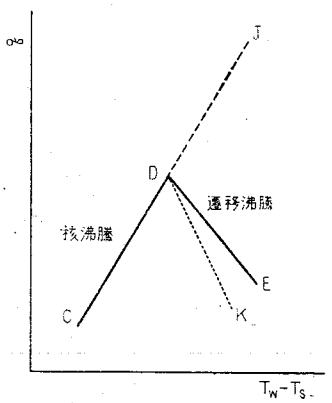

図 9 バーンアウト点近傍の沸騰曲線 CDE

域における加熱面上で沸騰蒸発のため減少していく水膜の厚さの時間的変化を多少模型化して示す．蒸気塊の離脱周期 τ（本章と次章では便宜上 τ であらわす）ごとに，水膜は水の供給をうけて初期厚さに回復し，加熱面上には絶えず水が存在する．ここで加熱面温度をあげると，当然図9の核沸騰曲線CDに沿い熱流束 q が増加する．そのため，水膜の初期厚さ δ_0 が減少〔式（4）参照〕するとともに，水膜の消耗速度がまし〔式（3）参照〕，結局図10(b)の状態すなわちバーンアウト点に到達するわけである．

さて，さらに加熱面温度をあげ，図9の遷移沸騰DEの領域へ入れたらどうなるであろうか．まず実際の沸騰について，図4のような写真をうつし，しらべてみよう．すると核沸騰領域と同様に蒸気塊が周期的に加熱面からたちのぼるのであって，外見上は核沸騰領域ときわだった差異がない．特に蒸気塊が不安定になるようなこともまったくなく，かつ蒸気塊の離脱周期も，バーンアウト点より少し減少する傾向にあるが同じオーダである．したがって，こうした沸騰の外見写真だけをみる限り，核沸騰も遷移沸騰もほとんど同性格で，特に図9の沸騰曲線のバーンアウト点における折れまがりのような特異な変化がどうして出るのか，疑わしくさえなる．きつねにつままれたような気持というのは，まさにこんなときに感ずるものをいうのだと思う．と同時にバーンアウトに関する流体力学的不安定説を信じようにも，これではあまり不思議で，どうにもならないのである．

では，加熱面上の水膜の状況はどうなっているのだろうか．というより，そこにしか主な原因はあり得ないはずである．そこで図7(b)の装置で遷移沸騰における状態をみてみると，図10(c)に模型的に示すような水膜の時間変化状況がただちに観察されるのである．すなわち各サイクルの初期には，加熱面上に水があつて「核沸騰」を生じており，これが蒸気塊を作るが，やがて水が消耗して，サイクルの終期には加熱面の乾燥した状態（加熱面はまだ蒸気塊におおわれているので水が供給されない）が続く．このようにサイクルごとに乾燥期間を含むと，その間，加熱面から流体への伝熱がほとんどおこなわれないから，遷移沸騰で測定される熱流束はバーンアウト点のそれより低下する（定量的な詳細は次章参照）．なおまた，加熱面の瞬間熱流束は，きわめて大きな値と零の間で変化していくから，加熱面温度を完全に一定に保とうとしても現在の技術では無理で，加熱面温度の時間的変動がどうしても生ずる．石谷と久野[17]は，温度変動の測定から遷移沸騰やバーンアウトの機構を定性的に推察しているが，本章に述べたものに実によく一致する．また遷移沸騰では，断続的な加熱面上の水の完全乾燥のため，水中の不純物が加熱面全体に析出しやすい*．西川・長谷川ら[18]は不純物の析出が，核沸騰領域では発ぽう点を中心に，また遷移沸騰領域ではほぼ加熱面全面に生ずるという貴重な観察をしているが，まったく当然なことである．

ともあれ，本章のはじめに記した「なにか変わった現象」というのは，周期的な加熱面乾燥期間の出現だったわけである．そして水の消費に対するぎりぎりの給水こそ，バーンアウト点を規定すると同時に，また遷移沸騰の開始を告げるものなのである．

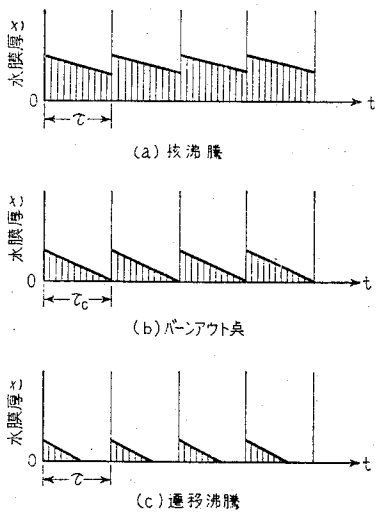

図10　水膜厚さの時間的変化

* 不純物の点から考えると，核沸騰領域はまさに自然循環ボイラ，また遷移沸騰領域はまさに貫流ボイラの状況に相当する．

9. 遷移沸騰の機構

われわれは遷移沸騰のなかにも「核沸騰」が断続的に含まれているのを知った．しかし加熱面温度はバーンアウト点より高くなっているのだから，その「核沸騰」はもはや図9の領域CDのものではない．それにしても明敏な読者はすぐに想起されるであろう．すなわち領域CDがなぜバーンアウトD点でストップしたかを思い出すと，それは単に水の供給がぎりぎりになったためにすぎず，加熱面上の水膜内の核沸騰が不可能になったためでは決してなかった．したがって，核沸騰だけを純粋に考えるなら，それは図9に破線DJで示す延長（領域CDの）線上をなお進み得るものであり，かつ進むのが当然であって，遷移沸騰に断続的に含まれる「核沸騰」はこれである（ひかえ目にみても近似的にそういえる）．そして遷移沸騰で前記

の「核沸騰」が生じている期間，加熱面はバーンアウト熱流束 q_c より高い熱流束 q で熱を伝えているのである．

さて遷移沸騰において，この熱流束 q の「核沸騰」の一継続期間 t は，式 (3)，(4) から

$$t=\text{定数}\times(1/q^{2.5}) \quad \cdots\cdots(8)$$

このときの蒸気塊離脱周期を τ とすれば，τ のうち時間 t の間は上記の伝熱（熱流束 q）がおこなわれるが，残りは乾燥期間で伝熱はないにひとしい．こうした伝熱の時間的平均として，遷移沸騰の熱流束 $\bar{q}$ は，$\bar{q}=qt/\tau$．この右辺に式 (8) を入れれば，$\bar{q}=$定数$\times(1/\tau q^{1.5})$．この式はバーンアウト点（$\bar{q}=q=q_c$）でも成り立つので，この点の蒸気塊離脱周期を $\tau=\tau_c$ とすれば定数の値が，定数$=\tau_c q_c^{2.5}$ と定まる．したがってこの値を前の式へ入れると結局，次の結果となる．

$$\bar{q}=q_c\cdot\frac{\tau_c}{\tau}\cdot\left(\frac{q_c}{q}\right)^{1.5} \quad \cdots\cdots(9)$$

すでに前に述べたように τ_c/τ はおおまかにいって，ほとんど1のオーダに近く，したがって上式から $q>q_c$ のとき $\bar{q}<q_c$．つまり遷移沸騰の熱流束 $\bar{q}$ はバーンアウト熱流束 q_c より小さくなるのである．

図9の破線 DJ 上の任意の点（その核沸騰の熱流束は q に対応する温度差 T_W-T_S における遷移沸騰熱流束 $\bar{q}$ は式 (9) で計算できる．もっとも式 (9) で計算した結果は，一般には多くの場合，図9に点線 DK に示すようになる．つまり実際に測定される遷移沸騰 DE は式 (9) による計算結果 DK よりもっと多くの熱を伝えていることになる．これは一体なぜであろうか．かいつまんでいうと，次のような現象[13]によるのである．すなわち各サイクルの最初，加熱面上の水膜の激しい沸騰からの蒸気をうけて蒸気塊が，多少爆発的なような姿で生長する*．しかし水が消耗してしまうと，蒸気の供給がとだえ，蒸気塊は乾燥加熱面をおおったまま暫時静かになる．前述の解析では簡単のため，蒸気塊がそのまま変化せず離脱まで保たれると考えた．だが，第一それではいつ離脱できるのかさえわからなくなってしまう．実際には，やがて乾燥加熱面の周辺付近に水が浸入し（振動的で，それはさきに甲藤・横谷[19]が，せまい空間内の沸騰に関連し抑圧蒸発と名づけた一形式に似る），それが沸騰や蒸発を続けて蒸気塊をさらに成長させ，離脱に到るのである．上記の水の浸入には，加熱面のぬれやすさなどが特に影響しよう．そしてこの伝熱が上記の差異を生じ，場合によってはその役割が大きい．

それにしても遷移沸騰領域では，上記の説明からわかるように，乾燥固体面と水の親和性が伝熱特性に相当な影響をもつ．しかも，前章にのべたように遷移沸騰では水中の不純物が加熱面上に析出しやすく，これは上記の親和性をかえ，伝熱特性をかえる．また加熱面温度をどこまで一定に保ち得るか，実験方法が影響を持ったりして，核沸騰領域（ここでは加熱面がつねにぬれている）ではあまり問題にならなかったことがあらわれてくる．

10. むすび

水（ないし液体）に接する加熱面の温度を飽和温度より高くあげる．考えてみれば，たったそれだけのことである．しかしそこで，水と蒸気が，否，ひろくいえば自然が，われわれの前に演じてみせる躍動的な姿は，なんと美しく，また多様なものであろうか．しかもそれは，われわれの生活，新鋭の技術，最新の工業とも深く結びつき，永遠の伴りょとして人間の前にあらわれるのである．複雑な現象ではある．しかしそこになにがおこっているのか．またそれは，自然の基礎法則とどう結びついて生じているのか．それを知りたい，また知らねばならぬという要請は，多くの人々の興味や関心をこの現象に引きつけてきたし，またこれからも引きつけていくであろう．

本稿はバーンアウトを主軸に，水と蒸気が沸騰でどんなふるまいをしているかを説明したものである（筆者と横谷定雄君との最近の協同研究を背景にしている）．あえていえば従来は，バーンアウト点を，あたかもそこで一つの沸騰形態が断絶してしまう特殊状態，極限状態であるかのようなとらえ方をしていたことが多いと思う．またそうした気分は，必然的に遷移沸騰の理解にも無用な混迷を与えた．しかし現実の沸騰はそうしたものではないようである．すなわち核沸騰と遷移沸騰を通して一本のレールが敷かれ，水と蒸気の挙動はむしろ連続的に変化していく．しかもその連続的な変化こそが，バーンアウトという特殊な現象を生ずる原因なのである．こうしてわれわれは，ある意味でバーンアウトがながく持っていた権威（その動的な姿をみようとしなかったために生じた権威）をはく奪してしまった形である．しかし真理はいつも，みかけの権威が倒れたあとの明るい空に，その姿をあらわすものではないであろうか．

それにしても，沸騰について，バーンアウトについて，われわれはまだ探求すべき多くのことがらをもち，それらについて一歩一歩，着実な分析と解明を続けていかねばならないであろう．そして筆者が特に痛

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* サイクルの初期，乾燥加熱面へ周囲から水が浸入しつつ沸騰をはじめていくので，核沸騰領域のときと少し状況がちがってくる．

感することは，いかなる熱伝達（流体の動きを伴う伝熱）も，その基盤をなすものはあくまで流体の動き，流体に関する力学である．力学を忘れた熱伝達，伝熱だけの熱伝達は，現象論の水準にしか立ち得ないであろうということである．

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M16　高熱流束沸騰蒸気塊の生成・切断挙動の研究

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1. 序

核沸騰において熱流束が低いうちには，気泡はばらばらに発生する。しかし熱流束が高くなると伝熱面から発生した気泡は合体しはじめ，更に熱流束が高まると，伝熱面近くに合体蒸気塊を形成する。この高熱流束核沸騰の機構は，気泡の相互干渉，合体などの複雑な因子を含むもので，理論的な解析は非常に困難である。従って未だに未解明な点が少なくない。本研究は発泡核から発生する蒸気を多孔板の孔または金網の目から噴出する空気におきかえ，そのときに形成される空気塊の挙動を研究し，この結果から高熱流束核沸騰における合体蒸気塊の生成離脱の機構を純流体力学的に解明せんとするものである。

2. 実　験

実験装置の概略は図1のとおりである。口径8φ～36φの黄銅製オリフィスの出口表面に孔径0.25φの多孔板または50～400メッシュの金網を接着剤ではりつけたものを，水またはメチルアルコールを満たした水槽の底に取りつけ，そこから空気を吹出させ，液中に気泡を発生させる。この気泡の挙動を高速度写真等を用いて観察し，またオリフィス面に気泡を形成するに要する圧力をオリフィス前の蓄気室内圧力P_3，水槽内の圧力P_1，水位hを測定して求める。

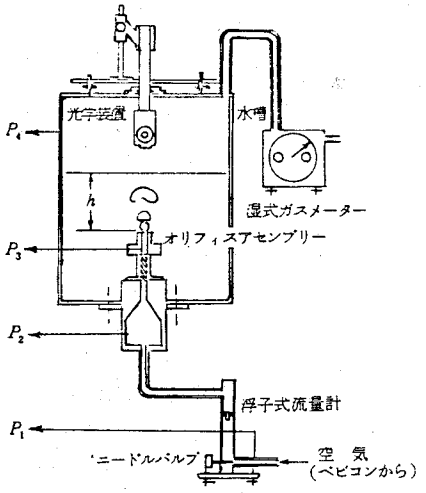

図1　実験装置概略図

3. 実験結果と考察

3.1 流量による気泡形成の変化

低流量範囲（G/A=0～38cc/cm²s）においては少数の発泡点から一つ一つ発生していた気泡は，流量が増して発泡点の密度が高まるにつれて相互干渉によって合体しはじめ図2に示すような合体泡をオリフィス面上に形成する。このような過程はGaertner(1)による核沸騰の写真的研究と一致している。

3.2 高流量範囲（G/A=38～180cc/cm²s …相当熱流束q_e=0.6×10⁶～1.8×10⁶kcal/m²h）における合体泡の挙動

この範囲における気泡生成の写真の代表的な例を図2に示す。合体泡は写真のような一連のサイクルを繰り返して形成されている。これらの各サイクルは一つの大気泡（A）とそれに続き後流の影響で大気泡に吸いこまれてゆくいくつかの小気泡（B_1，B_2，…）から成り立っている。オリフィスの周囲の流れは乱れているので，一サイクル中に含まれるBの数はかなりばらつく。またAが成長しはじめてから離脱する迄の周期もかなりばらついているが，その平均値は流量によってあまり変化しない。（わずかに増加の傾向はもつ。）

このような合体泡の挙動は甲藤・横谷によって撮影された高熱流束沸騰蒸気塊の写真と殆んど変るところが無い。また従来の単孔オリフィスからの気泡発生の実験

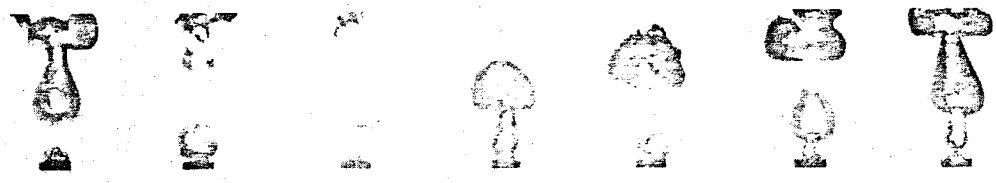

図2　合体泡の一例（10φオリフィス，0.25φ多孔板），G/A=153cc/cm²s

(Davidson と Schüler(2), Walters と Davidson (3) など)ともきわめて類似している。これらの事実から金網または多孔板による合体泡，高熱流束核沸騰の蒸気塊は流体力学的に全く同一の機構によっており，それらの挙動は，単孔オリフィスから一定流量で空気を吹き出させた場合の発生気泡と本質的に変らないことが考えられる。

3.3 合体泡の底部の液層について

気泡を形成するに要する圧力は，オリフィス前後の圧力差からオリフィスの流出損失を引くたとによって得られる。この値は合体気泡生成の機構に密接に関連している。この圧力を種々の多孔板，金網に対し測定した結果を図3に示す。これより気泡成形圧 Δp は表面張力による圧力 $\frac{4\sigma}{\delta}$ (δ は金網の目の対角線長さ，または多孔板の孔径) にほぼ等しいことがわかる。

次に，干渉ガラス板を底に取り付けた光学装置をオリフィスに近づけ，合体泡をおしつぶして干渉を与えながら気泡底部の様子を観察した。干渉板とオリフィス面との間隔 s による Δp の変化を示したのが図4である。これより Δp は s と流量の両方により変化するが，s が 5.0 mm 以上になると $s=\infty$ (干渉板の無い場合) とほとんど等しくなり，干渉板の影響がほとんど無くなることがわかる。同時に撮影した高速度写真は合体泡の底部には，液層が常に存在しており，この液層には液が周期的に供給されているが，間隔 s が小さくなると，液が供給されにくくなり，液層は厚さを減じ，吹き抜けがおこることを示しており，これにより Δp が低下することがわかった。次にこの液の形成を更によく観察するために，特殊な方法で気泡の断面撮影を行ない，液層は合体泡が離脱した際にオリフィス面上に入りこんだ液が，合体泡がその上に成長するときにも面上に残存することにより形成され，液はそこには，合体泡が成長中には決して供給されることが無いことを確認した。

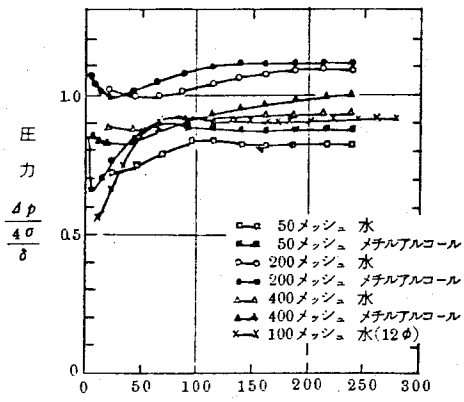

図3 気泡形成圧力

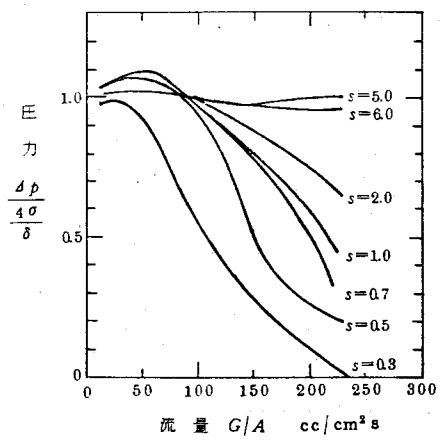

図4 干渉板による気泡形成圧の変化

4. 理論と考察

3.2 の結果から Davidson と Schüler によって単孔オリフィスからの気泡形成 (流量一定) に対して得られている理論が，この場合にも適用できることを知り，これを用いて理論解析を行なった。成長中の気泡に対して

$$(\rho_L-\rho_G)V_g=\frac{d}{dt}\left\{\left(\frac{1}{2}+\frac{3}{16}\frac{r^3}{b^3}\right)\rho_L V\frac{du}{dt}+\rho_G V\frac{dy}{dt}\right\}$$

$r=b$ とおくと，$\rho_G\approx 0$ から $V_g=\frac{d}{dt}\left(\frac{11}{16}V\frac{dy}{dt}\right)$ が得られる。次に流量=const. から $V=Gt$ とおき，初期条件 $t=0$ にて $y=0$, $\frac{dy}{dt}=0$ を用いて積分し，気泡が $y=r$ になると離脱すると考えると，離脱気泡の体積は $V=1.378\frac{G^{6/5}}{g^{3/5}}$ になる。これから気泡の周期として

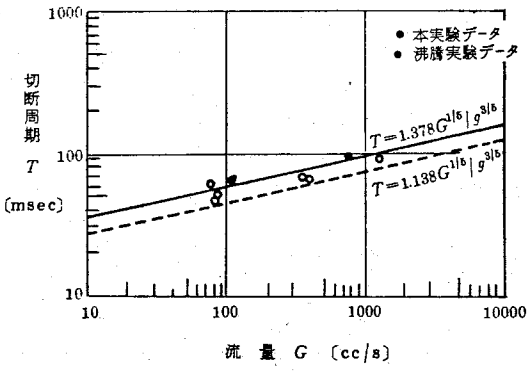

図5 合体泡周期 (理論と実験の比較)

$T=\frac{V}{G}=1.378\frac{G^{1/6}}{g^{3/6}}$ が得られる。これはオリフィスが水槽の底に近い場合を考えているが，底から十分に離れている場合には $T=1.138\frac{G^{1/6}}{g^{3/6}}$ になる。両式を本実験の高速度写真解析結果と比較すると（図5）かなり良く一致し，また，甲藤・横谷による沸騰の実験データとも一致することがわかる。

5. 結　　論

以上のことから，高熱流束沸騰蒸気塊に対して次のことがいえる。

1）合体蒸気塊の挙動は純流体力学的に取り扱うことができ，近似的な理論から得られた $T=1.378\frac{G^{1/6}}{g^{3/6}}$ は（$G=\frac{Aq}{\rho_V L}$ とおきかえることによって）蒸気塊の離脱周期の近似式に用い得る。

2）蒸気塊の底部の液層は気泡が離脱した際に伝熱面上に入った液が，蒸気塊の形成中にも伝熱面上に残存することにより形成されるが，蒸気塊の成長中には液の供給を受けないのでバーンアウト及び遷移沸騰現象をひきおこす。

記　　号

ρ_L, ρ_G, ρ_V：液体，気体，蒸気の密度，r：気泡を球形と仮定した場合の半径，b：壁と気泡との距離，V：気泡体積，y：オリフィス面と気泡の中心の距離，A：オリフィス断面積あるいは伝熱面面積，q：熱流束，L：蒸気潜熱

文　　献

（1）ASME Feb. 1965, pp. 17～

（2）Trans. Inst. Chem. Enges. 38, 335

（3）J. of Fluid Mechanics 17, 321

536.423.1 : 532.5

プール沸騰におけるバーンアウトおよび遷移沸騰の機構*

甲藤好郎**, 横谷定雄**

1. 序　論

プール飽和沸騰におけるバーンアウトの機構について多くの研究があり，各種の説や理論が提出されているが，そのなかで，気液界面の流体力学的不安定理論が有力なようである．たとえばKutateladze[1]や，Zuberら[2]の関係式が，広い範囲の液体条件に対して，バーンアウト熱流束のだいたいの値を指示しうることは否定し難い事実である．しかし，多くの現存理論は，次のような点に多くの問題を残している．

（i）現存理論で仮定される物理モデルは，実際に観察される流体挙動と必ずしも対応しない（Kirby-Westwater[3]など）．(ii) あまり小さくない普通寸法の加熱面では，その寸法，形状，方位，構成，あるいはテフロン・コーティングその他の影響が説明しにくい（Gaertner[4]，長谷川ら[5]）．(iii) 加熱面寸法が比較的小さいときに特に著しい，よごれ，ぬれやすさ（Costello-Frea[6]），加熱面熱容量，熱伝導率（Carne-Charlesworth[7]，橘ら[8]）の影響が説明しにくい．(iv) きわめて寸法の小さい加熱面，あるいは発生気ほうに比し加熱面寸法の小さい場合には生じていると推定される，いわゆる気ほう被覆作用（Bernath[9]，Carne[10]，Ivey-Morris[11]，Kutateladzeら[12]）への変化機構が説明しにくい．(v) バーンアウト点は，核沸騰領域および遷移沸騰領域の限界であり，したがって両領域の沸騰機構と密接な関係をもつはずだが，遷移沸騰の解明に現存理論が無力なことは奇妙である．

最近，甲藤・横谷[13]は，直径10 mmの上向き円板加熱面に対し，図1（a）～（c）に示す三つの系について大気圧下の水のプール飽和沸騰の実験をおこない，周期的な流体挙動の観察から結論されるバーンアウト機構について報告した．本報は，さらに詳細かつ広範囲にわたって，バーンアウト機構および遷移沸騰機構を明らかにするべくおこなった研究結果について記す．

* 昭和43年5月16日　第5回日本伝熱シンポジウムおよび昭和44年5月29日第6回日本伝熱シンポジウムにおいて講演，原稿受付　昭和44年11月24日．

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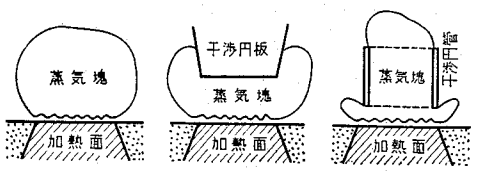
2. 高熱流束飽和沸騰における流体挙動

便宜上，甲藤・横谷[13]が想定した，高熱流束飽和沸騰の流体挙動機構をここに概説する．高熱流束では，図2（a）に示すごとく，加熱面上に密に分布する各発ぽう点で，それぞれ生じている核沸騰が蒸気を発生，その蒸気は上方で蒸気塊を形成，この蒸気塊と加熱面の間には，その内部で核沸騰を続ける液層（以下，これを核沸騰液層と呼ぶ）がはさまれる．この核沸騰液層では，図2（b）のごとく，激しい沸騰のために液体が一定速度で消耗すると同時に，蒸気塊は蒸気の供給をうけて生長を続け，この蒸気塊は周囲液体を排除する．しかし蒸気塊は，図2（c）のごとく，ある周期で核沸騰液層から切断浮上すると同時に，このとき当然，液体が周囲から核沸騰液層に供給され，図2（d）のごとく最初の状態が再現されて，以下同じサイクルを再びくりかえす．上記の蒸気塊切断プロセスに要する時間は，蒸気塊の一切断から次の切断ま



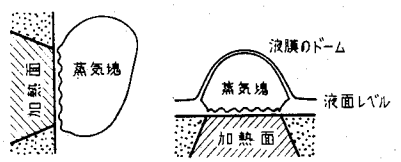


図1　種々の沸騰系

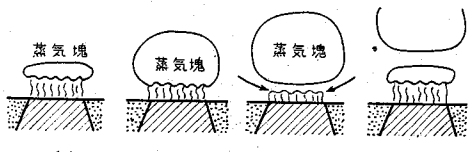


図2　高熱流束，飽和核沸騰（核沸騰液層の厚さは誇張した寸法で示されている）

での1サイクル周期 τ にくらべると一般にきわめて短い．

核沸騰液層上にある蒸気塊が切断浮上するまでに，この液層が消耗しきらない状況下では，核沸騰が加熱面上に絶えず保持される．しかし熱流束 q が上昇すると，蒸気塊切断周期 τ 内に，核沸騰液層がちょうど消耗しつくされる極限状態に達し，これがバーンアウトに対応する．さらに加熱面温度が高くなると，バーンアウト熱流束より高い熱流束によって核沸騰液層が消耗，これによって発生した蒸気塊はその後もしばらく加熱面上に停滞するから加熱面の乾燥期間を生じ，このため周期 τ 内の時間的平均熱流束は低下して遷移沸騰領域へはいる．

上記の機構では，バーンアウト熱流束 q_c に対して次式が成立する．

$$\tau=\xi\rho_l\delta_0 H_{fg}/q_c \quad \cdots\cdots(1)$$

ここに τ：蒸気塊切断周期，ξ：核沸騰液層内に含まれる液体の体積割合（液層内には気ほう蒸気も存在する），ρ_l：液体密度，δ_0：核沸騰液層の初期厚さ，H_{fg}：蒸発熱である．Gaertner[4] および Gaertner-Westwater[14] の実験を参照すると，平らな銅加熱面上の大気圧の水の核沸騰に対し，ξ および δ_0 は次のごとく推定される．

$$\xi=8/9, \text{ および } \delta_0=\delta^*(q^*/q)^{1.5} \quad \cdots\cdots(2)$$

ここに $\delta^*=0.133$ mm，$q^*=10^6$W/m^2，また q は熱流束（バーンアウト点では $q=q_c$）．

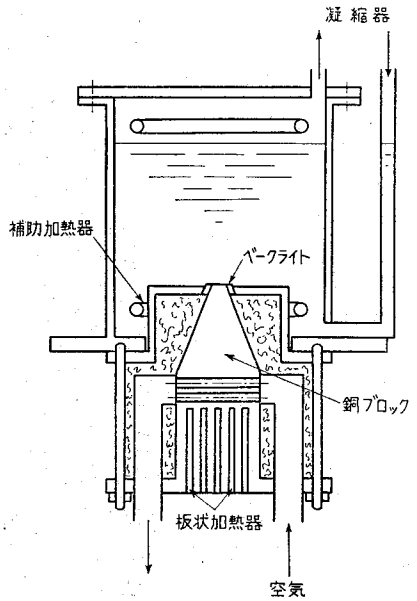


図 3　実験装置の主要部

3. 実験装置

実験装置の主要部を図3に概念的に示す．下部を電気加熱器で加熱される銅ブロックの上端の円形加熱面（本報では主として直径 10 mm，なお 35 mm も使用）上で蒸留水の沸騰を生じさせる．加熱面温度および熱流束は，銅ブロック内の熱電対で定められる温度分布から測定され，この測定には慎重な検定をおこなっている．

本装置は，低圧下の飽和沸騰に対し空気の流入を防止するように設計された凝縮器に結合され，大気圧以下の低圧沸騰の実験が可能である．また装置全体を自由な角度に傾けられるので，上向き加熱面のほか垂直

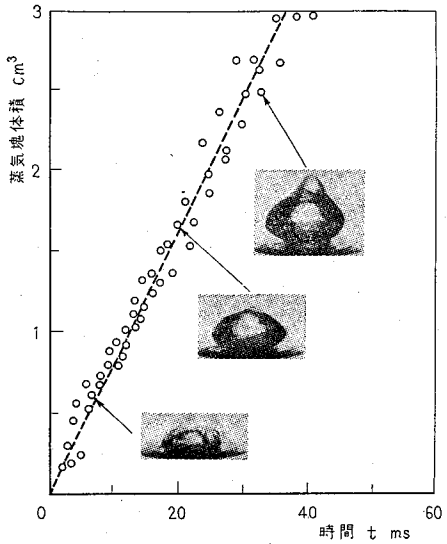


図 4　直径 10 mm，上向き加熱面上の核沸騰における蒸気発生状況（$q=1.39\times10^6$W/m^2）

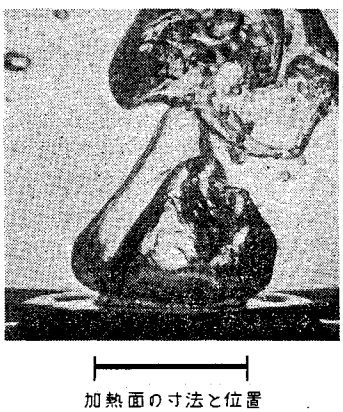


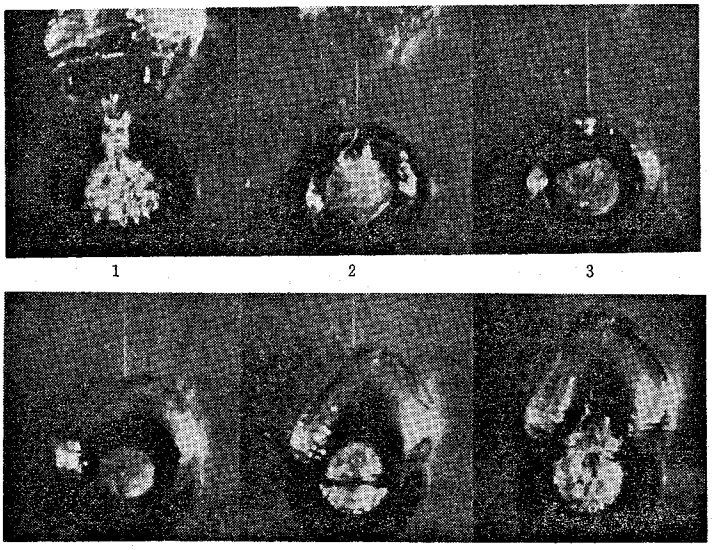
図 5　直径 35 mm，上向き加熱面上の核沸騰（$q=9.7\times10^5$W/m^2）

加熱面などの実験ができる．なお遷移沸騰の実現に際しては，一色・玉木[15]によって考案された方法を用いた．また装置にはガラス窓があり，高速カメラ（1000こま/秒）によって流体挙動が観測できる．

4. 大気圧下の飽和核沸騰

4・1 流体挙動の観察 図1(a)，(d)，(e)の3個の系について，大気圧下の核沸騰における流体挙動を観察，分析した結果を記す．

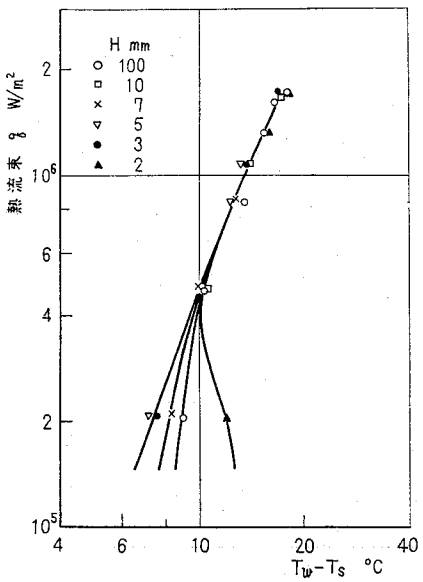
図1(a)の形式（上向き加熱面）：図4は，バーンアウトに近い核沸騰において，直径 10 mm の加熱面からの蒸気生成量を，高速カメラ写真の解析から測定した一例（参考までに蒸気塊の瞬間状況写真も掲げる）である．蒸気発生率は時間に独立に一定，かつ測定熱流束 q から計算した蒸気生成(図4に破線で示す)と一致しており，これは図2の蒸気発生機構を示唆する結果である．また図5は直径 35 mm の加熱面上の蒸気塊の一瞬間状況を示すものであるが，蒸気塊の底部に核沸騰液層の存在が明らかにみられ，これは Gaertner[4] の観察結果（直径 50.8 mm の加熱面で実験）ときわめてよく似ている．加熱面寸法が大きくなると，Taylor 不安定のため蒸気塊の形状は複雑な形になる傾向を持つが，上述の寸法範囲の加熱面では，現象の本質にまだ大きな相違はあらわれない．



$q=1.14\times10^6$ W/m², 時間間隔 10.6 ms
核沸騰液層が絶えず存在するため，加熱面上のきずあとが見えない

図 6 直径 10 mm，垂直加熱面上の核沸騰

図1(d)の形式（垂直加熱面）：垂直加熱面では，加熱面上の核沸騰液層の状況が，蒸気塊を通して直接観察しやすい．図6（ここで黒いリング状のものは熱絶縁用のベークライト・リング）は，直径 10 mm の加熱面で，バーンアウト近傍の核沸騰における流体挙動の時間的経過を示す 加熱面上には常に核沸騰液層が存在し続けること，およびこま3，4では蒸気生成のため核沸騰液層が相当に消耗しているのがみられる．



直径 10 mm，上向き加熱面

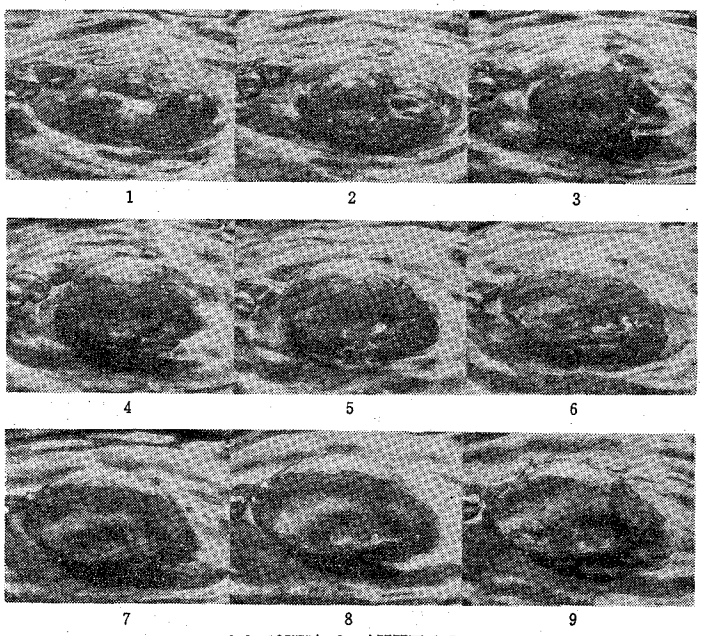
図 7 水位 H による核沸騰曲線の変化

図1(e)の形式（低水位）：沸騰装置内の水の自由水面から，上向き水平加熱面（直径 10 mm）までの深さ H を変えて，核沸騰領域の沸騰曲線を求めた結果を図7に示す．同図において $q>4.6\times10^5$ W/m² の高熱流束域になると，沸騰曲線に対し水位 H の影響はほとんど消滅している（少なくとも $H=2$ mm の低水位まで）．これは，高熱流束核沸騰が，図2に示したような薄い核沸騰液層で生じていることを示唆する．さて，図8は $H=3$ mm の低水位の場合における流体挙動の時間的経過を示す．蒸気塊が，液膜ドームに包まれた形で生長し，やがて破壊するという周期的なサイクル挙動を形成し，なお加熱面上には核沸騰液層が絶えず存在することが見られる．この核沸騰液層から発生する蒸気量を，液膜ドームの体積変化から計測する

と，図4の場合とまったく同じ傾向が得られる．

4·2 バーンアウト熱流束　前節に述べた流体挙動の観察は，すべて図2に示した流体挙動モデルを支持する．さて，上述の実験のおのおのにおいて，バーンアウト近傍における蒸気塊切断周期 τ（統計的平均値），およびバーンアウト熱流束 q_c が測定された．その結果，τ と q_c の関係が図9に示される．なお同図には，Gaertner[4]，甲藤・横谷[13]，および石谷ら[16]の報告から見出した τ と q_c に関するデータをも掲げてある．

一方，式（1），（2）から解析的に導かれる τ と q_c の関係は図9に示す直線となり，実験データとだいたい一致しているのがみられる．



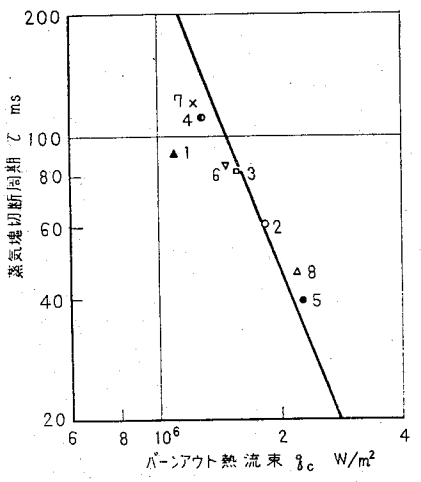
$q=8.6\times10^5$ W/m², 時間間隔 8.5 ms

図 8　直径 10 mm，上向き加熱面，水位 $H=3$ mm における核沸騰

5. 低圧下の飽和核沸騰

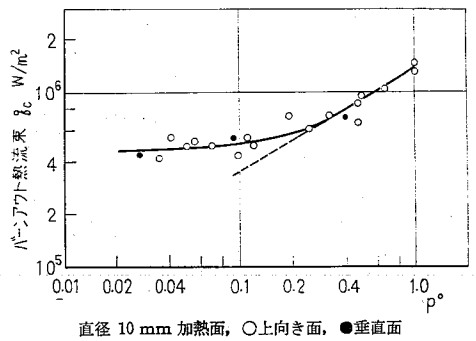
図2に示す形式の沸騰は，発ぼう点密度が大きく，かつ核沸騰液層内に生ずる気ほう（以下，発ぼう点気ほうと称する）の大きさが加熱面寸法に比して小さいときにのみ成立する．もし加熱面寸法に対する発ぼう点気ほうの相対的大きさが大になると，流体挙動およびバーンアウトに変容が当然あらわれるはずである．これを確かめるため本報では，低圧下の沸騰を利用した．実験は直径 10 mm の上向き，および垂直加熱面でおこなう．

5·1 圧力によるバーンアウト熱流束の変化　図10は，上向き加熱面および垂直加熱面において，圧力（加熱面水位における圧力を，標準大気圧に対する比であらわす）$p°$によるバーンアウト熱流束 q_c の変化を測定した結果を示す．この場合，加熱面の両形式（上向き，および垂直）に対し，q_c はほぼ同じ値をとっているが，いずれにせよ $p°$ が1より減るに従い q_c は減少し，その変化傾向は，たとえばKutateladzeの関係



沸騰系	加熱面；直径
1：図1(a)	円板；35 mm
2：図1(a)	円板；10 mm（文献 13）
3：図1(a)	円板；50.8 mm（文献 4）
4：図1(b)	円板；10 mm（文献 13）
5：図1(c)	円板；10 mm（文献 13）
6：図1(d)	円板；10 mm
7：図1(d)	垂直面；高さ 90 mm の円柱（文献 16）
8：図1(e)	円板；10 mm

図 9　蒸気塊切断周期 τ とバーンアウト熱流束 q_c



直径 10 mm 加熱面，○上向き面，●垂直面

図 10　圧力によるバーンアウト熱流束の変化

式が指示する傾向に似たものである．しかし $p^\circ<0.2$ の圧力範囲になると，上記の傾向からのかたよりが明らかにあらわれ，p° による q_c の変化が減少している．もちろん，p° が非常に小さくなると，自由液面からの深さに伴う静水圧変化（飽和温度変化）が水中に生ずるが，本報での p° の実験範囲では，その影響はまだ少ない．

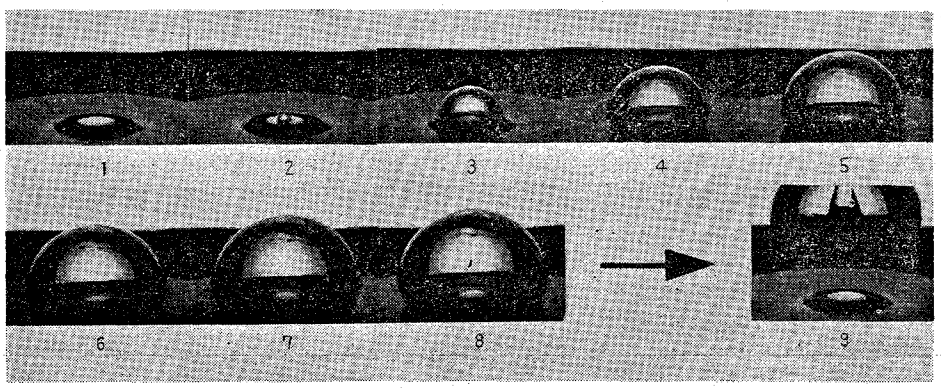
5・2 低熱流束域における孤立気ほう バーンアウトについて記す前に，低熱流束域における状況を述べておかねばならない．低圧下の低熱流束核沸騰では，巨大な孤立気ほうが発生する．そして，この現象は，発ぼうに要する高い過熱度や，蒸気の比容積が大きくなることだけではけっして説明がつかぬものである．図11は，圧力 $p^\circ=0.099$，過熱度 $T_w-T_s=24.5$ deg，熱流束 $q=3.5\times10^5$ W/m^2 における上向き加熱面（直径 10 mm）の中心から発生した孤立気ほうの生長を示す．気ほう発生前のこま1にみられるように，加熱面は白く，熱絶縁用ベークライト・リング面は黒く写り，なお両面は灰色の大きな円形のホールダー面よりわずかに（約0.6 mm）高くつき出ている．図11において，孤立気ほうの底面は，こま3ですでに加熱面をこえて広がっており，最後には直径 40 mm の大きさにまで広がる．これは巨大な気ほうである．

さて図11で注目すべきは，加熱面上にいわゆる薄液膜が形成されていることで，それは加熱面中心から乾燥部分がひろがっていくこと（こま 3～8）によって証される．そして，気ほう生長開始時刻から測って，半径 r の乾燥部分が生ずるまでの時間 t は，高速撮影フィルムの分析から，$t=Cr$（ここに $C=1/657$ h/m）とあらわされた．甲藤・横谷[17]が，平行2円板間の核沸騰気ほうの場合に用いたと同じ方法で，上記の薄液膜の乾燥速度から薄液膜の初期厚さ s_0 の分布を算定すると，$s_0=2.67\times10^{-4}\sqrt{r}$（ここに s_0, r はともにメートル）を得る．

高速撮影フィルムの分析はまた，上記の孤立気ほうの体積変化について図 12 の結果を与える．また同図に示す細い曲線は，気ほう底面が加熱面上にひろがった以後，上記の加熱面上の薄液膜からの蒸発だけで気ほうが生長したと仮定したときの体積変化（甲藤・横谷[17]と同じ方法で計算）を示すものである．この解析結果は実験結果にかなり近く，少なくとも加熱面上の薄液膜からの蒸発が，上記のような巨大な気ほうの生長に対して非常に重要な役割を果たしていることを示している．なお以上の結果は，巨大気ほうの発生原因に関する秋山ら[18]の推測を明確な形で実証したものになっている．

上記の核沸騰では，気ほうの生長期間に比し休止期間（加熱面上に気ほうの存在しない期間）が長く，両期間の和から成る1サイクルの周期は平均 220 ms であるが，そのなかには，図 12 の実験データにみられるごとく約 20 ms 程度の断熱期間（加熱面全面の完全な乾燥期間）が含まれている．しかしながら，乾燥断熱期間の出現だけではバーンアウトの原因にはならない（極端に熱容量の小さな加熱面で，熱流束一定加熱の場合を除く）．あくまでバーンアウトは，加熱面温度をあげるとき熱流束（時間的平均値）が低下をはじめる状態である．そして現在の沸騰系では，加熱面温度をあげると，休止期間の急激な減少，および発ぼう点密度の増加によって熱流束がまだ増加できるのであって，それを経てはじめてバーンアウトを生ずる．

5・3 バーンアウト 図13は圧力 $p^\circ=0.108$，垂直加熱面においてバーンアウト近傍の状態（$q=5.2\times10^5$W/m^2）での気液挙動の時間的経過を示す．ここで，加熱面周囲に黒くみえるリングは熱絶縁ベークラ



時間間隔：こま1～8 の間は 3.9 ms ごと，こま8～9の間は 52.7 ms

図 11 直径 10 mm，上向き加熱面の中心から発生した孤立気ほうの生長

イト・リングであり，その外側の白いリングは，塗布された接着剤である．この場合，最初3個の比較的大きな発ぼう点気ほうが生長し，次いでこれらが合体して加熱面上に単一の蒸気塊を形成するのがみられる．この合体蒸気塊の底部には，図2に示したような核沸騰液層はもはや観察されない．そこにあるのは，最初の発ぼう点気ほうのそれぞれの底部に形成された薄液膜（こま7，8には特にその存在が観取される）で，この薄液膜の表面からの蒸発が蒸気塊を生長させる．事実，加熱面から発生する蒸気量を測定してみても，図4のような単純な性格は示さない．

4章の場合に比べ，発ぼう点密度がはるかに小さく，かつ発ぼう点気ほうがはるかに大きいため，上述のような機構変化があらわれているのであって，バーンアウトの原因としては，核沸騰液層の消耗の代わりに，上記の薄液膜の消耗でおきかえられる．そしてこれは，バーンアウトを越えた遷移沸騰における気液挙動を観察することによっても確かめられた．

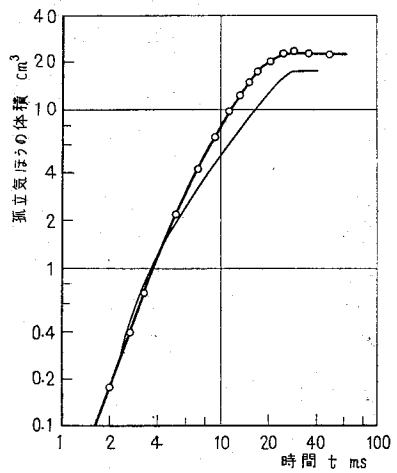


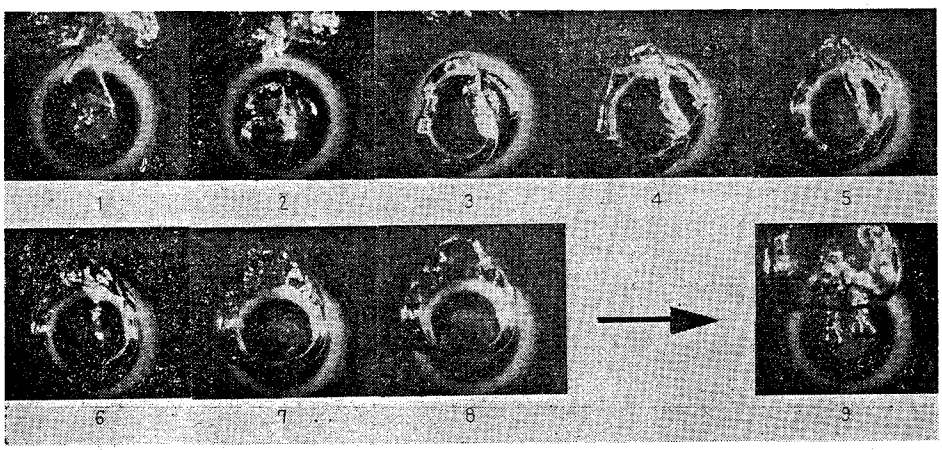
図 12　孤立気ほうの体積の変化

6. 遷 移 沸 騰

遷移沸騰の機構（少なくともバーンアウトの近くの領域における）は，バーンアウトとの関連において興味あるのみならず重要なことがらである．以下には，3個の異なる沸騰系に対する実験結果を記す．

6·1　上向き加熱面　図14の太い実曲線は，直径10 mm の上向き加熱面に関する一連の実験から求めた沸騰曲線（核沸騰および遷移沸騰領域）である．各実験ごとに，特に遷移沸騰領域において，よごれのない新鮮な加熱面を用いた．さて，甲藤・横谷[13]が前報でおこなったように，ある加熱面温度の遷移沸騰に対し，（a） 核沸騰期間中は，式（2）で与えられる液量が，図14の太い破線（核沸騰領域の延長）で与えられる熱流束で消耗，（b） 乾燥期間中は熱流束が零，（c） 蒸気塊切断周期 τ は変化が少ないので，常にバーンアウト点における τ にひとしいと仮定して，上記両期間内の時間的平均熱流束を解析的に求めると，図14の細い破線を得る．この解析結果に比し，実測熱流束の値は高くあらわれ，かつ両者の相違は加熱面温度の上昇とともに大きくなっている．この原因が明らかにされねばならない．

さて遷移沸騰における加熱面上に生長する蒸気塊の体積に関し，その時間的変化を測定した結果を図15，16に示す（両図は，それぞれ図14の点 *1* および点 *2* の実験状況に対応する）．なお図15，16で，破線は熱流束の実測値から計算したところの時間平均的な蒸気増加である．前に図4に示した核沸騰領域における単純な蒸気発生傾向とは異なり，蒸気塊の体積増加がは



時間間隔：こま1～8の間は 5.1ms ごと，こま8～9の間は 30.5 ms
図 13　直径 10 mm，垂直面上の低圧下核沸騰

じめ急激に生じたのち一時停滞し，その後ふたたび体積増加が続き，加熱面からの蒸気塊切断にまで至るという特徴がみられる．

ここで初期の体積増加は，加熱面上に形成された核沸騰液層（図2参照）の消耗で生じ，次いで加熱面が乾燥して体積変化が停滞すると考えられる．図15，16に示す水平線は，式（2）で与えられる液量が，図14の破線（核沸騰の延長）で示される熱流束で消耗したと仮定して計算したときの発生蒸気体積で，前記の実測停滞体積にだいたい一致している．

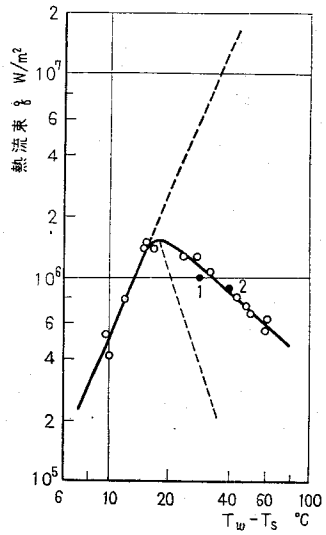


図 14　直径 10 mm，上向き加熱面の沸騰曲線

次に，加熱面がこのようにして乾燥した以後の蒸気塊体積増加は，蒸気塊でおおわれている乾燥加熱面の周辺から液体が侵入しながら核沸騰して蒸気を発生することによると考えられる．この際，加熱面温度が高いほど，侵入した液体の気化は急速に生ずるから，液体の侵入はそれだけ加熱面の周辺部分だけに局限される（加熱面の中心部は乾燥状態に保たれる）ようになる．このことがらを実証するため図17（a），（b）は，新鮮な加熱面をそれぞれ図15，16に対応する状態，図17（c）はさらに膜沸騰に近い高温状態で一定時間沸騰を保持した後の加熱面を示すもので，加熱面温度が高いほどスケール析出が加熱面周辺に局限されて現われている．すなわち，これらの部分に特に沸騰蒸発が多かったことを示すものである．

このようにして蒸気塊が十分に生長し，加熱面から切断浮上すると，加熱面の全面に核沸騰液層がふたたび形成され，上記と同じサイクルをくりかえす．図15，16に明らかなように，加熱面温度の上昇とともに，伝熱に対して核沸騰液層の役割（サイクル初期の蒸気発生）はきわめて急速に減じ，局部的な侵入液体の核沸騰（サイクル後期の蒸気発生）の役割が増加する．同時に，液体侵入とその核沸騰は加熱面周辺に局限されていくから，気液挙動は結局，しだいに膜沸騰の状況へ接近していく傾向を持つことになる．蒸気塊でおおわれた乾燥加熱面へ，その周辺からの液体侵入は，固体面上の液体ひろがり現象の性格をもち，したがって加熱面と液体の親和性によって特に大きな影響

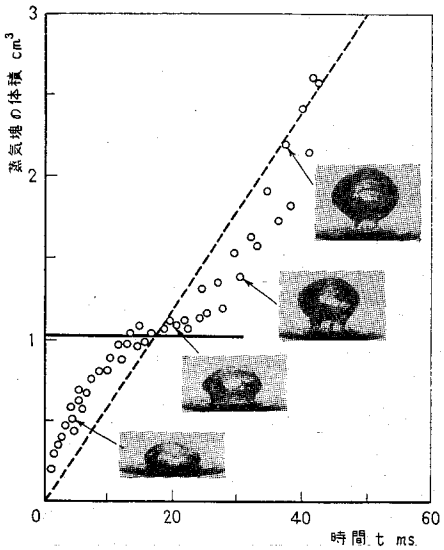


図 15　直径 10mm，上向き加熱面上の遷移沸騰における蒸気塊体積の変化（q=1.01 ×10⁶W/m²）

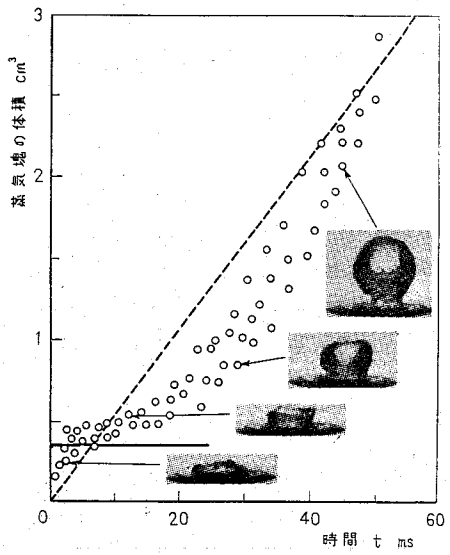
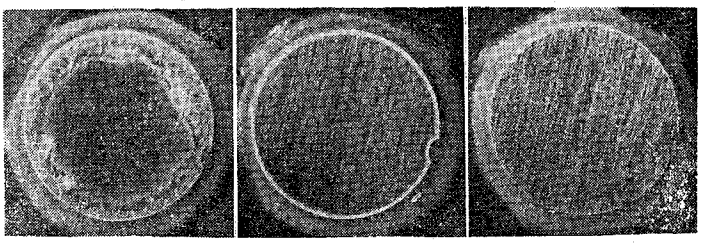


図 16　直径 10 mm，上向き加熱面上の遷移沸騰における蒸気塊体積の変化（q=8.96 ×10⁵W/m²）

を受けるはずである．すでに Berenson[19]，西川ら[20]の研究などでよく知られているごとく，遷移沸騰領域の沸騰曲線（伝熱特性）が，他の条件は同じでも，加熱面のよごれやぬれやすさで大きく変えられる事実，これは上述の事情に起因すると考えられる．

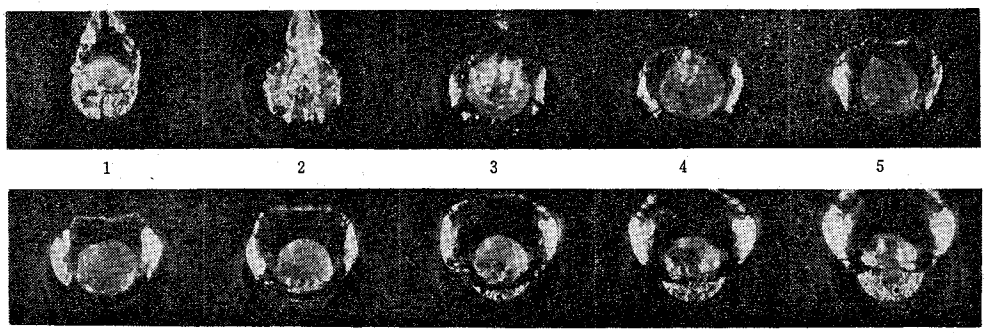
6·2 垂直加熱面　垂直加熱面でも上下方向に非対称性があらわれるだけで，現象の本質は前節に述べたことと変わらないことが見出された．ただ上向き水平面の場合と異なり，加熱面から垂直方向に蒸気塊を移動させるようには浮力が働かぬ（蒸気塊は加熱面に沿って上方へ移動しながら生長する傾向あり）ため，液体侵入による局所核沸騰が減ずる傾向をもち，したがって沸騰曲線も図 14 の細い破線（遷移沸騰）に近づく傾向をもっている．図 18 はバーンアウト近傍における遷移沸騰の代表例である．核沸騰液層が消耗後，それによって生じた蒸気塊の下に加熱面は乾燥状態となり，それから下方に小規模の局所核沸騰が生ずる状況が見られる．

6·3 低水位，上向き加熱面　低水位の場合も，6·1 節の場合と原理的に同じ状況が見出された．ただ低水位では多くの場合，侵入液体の局所核沸騰が容易のため加熱面周辺部分は絶えず沸騰の状況におかれる．つまり 6·1 節に述べたサイクル前期と後期の間が比較的連続してつながる状況になる．ただし例外的な例として図 19（水位 $H=3$ mm）を示しておく．ここでは，核沸騰液層が消耗し加熱面が乾燥した後，周辺からの液体侵入をまったく欠いており，そのためこま 5～8 の間では蒸気塊の体積変化もほとんど停止している．



(a)　(b)　(c)

図 17　直径 10 mm，上向き加熱面上に遷移沸騰で生じたスケール付着

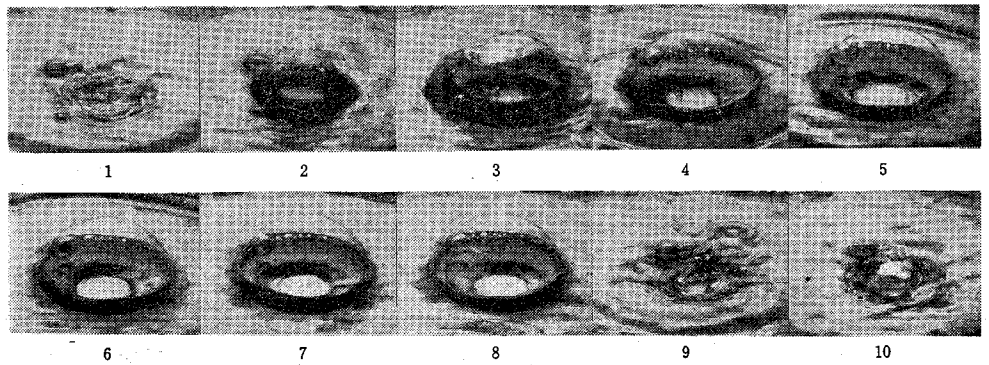


1　2　3　4　5

6　7　8　9　10

$T_w-T_s=25.7°C$，$q=9.7\times10^5$ W/m²，時間間隔 5.4 ms

図 18　直径 10 mm，垂直加熱面上の遷移沸騰



1　2　3　4　5

6　7　8　9　10

$T_w-T_s=52.9°C$，$q=5.9\times10^5$ W/m²，時間間隔 27.0 ms

図 19　直径 10 mm，上向き加熱面，水位 $H=3$ mm における遷移沸騰

7. 討論および結論

（1） 主として直径 10 mm の平らな加熱面（上向き，垂直，低水位）における，大気圧および低圧下の水の飽和プール沸騰について，高速カメラによる気液挙動の観察と，その分析を通して，バーンアウトおよび遷移沸騰の機構を研究した．

（2） 少なくとも本研究の条件下の流体挙動は，核沸騰，遷移沸騰領域にわたって連続的に変化し，バーンアウト点において流体挙動が不安定になるとか，不連続になるようなことはみられない．

（3） 核沸騰液層と，その上に停滞する蒸気塊からなる物理モデルは，普通の条件下の高熱流束核沸騰の場合に妥当である．発ぼう点気ほうが，加熱面と同程度ないしそれ以上の大きさになるときは，これに対応して流体挙動も上述とは違ったものになる．しかし，いずれの場合でも，加熱面上での液体受入れと液体消費は断続的な形で生ずる．

（4） 熱流束増加に対応する加熱面上の液体消費（蒸気発生）増加に対し，加熱面の液体受入れにもはや余裕がなくなったとき，熱流束の増加は不可能になってバーンアウトを生ずる．

（5） 通常の条件下では，式（1）に含まれる τ や δ_0 は流体の流体力学的挙動によって支配される面が大きいから，バーンアウト熱流束が流体の種類，圧力など流体条件だけで大まかに指示できる可能性をもつ．バーンアウト熱流束に関し提出されている Kutateladze の式その他は，この形式である．

（6） けれども他の因子もバーンアウト熱流束に影響を持ちうる．たとえば，沸騰系の幾何学的条件で τ は変化し，特に加熱面の近くに干渉物体をおいたりすれば τ の変化はいちじるしく，したがってバーンアウト熱流束が大きく変わる．テフロン被覆をした加熱面などでは，非常なぬれにくさのため，液体受入れや発ぼうなどの点で通常の形態を保ちえず，この面からバーンアウト熱流束を変化させる．また加熱面のぬれやすさは，比較的小さな寸法の平らな加熱面などのバーンアウトに対し特に影響をもつが，これは加熱面端部からの液体補給が容易となって核沸騰液層の寿命が延ばされるためである．また発ぼう点気ほうの，加熱面に対する相対的な大きさが変われば，流体挙動も変わり，バーンアウトを変化させる．こうしたことがらはすべて，本報に明らかにしたバーンアウト機構から無理なく説明される．

（7） 遷移沸騰領域では一般に，蒸気塊の発生から切断に至る1サイクル内に3個の状態が継続しておこる．すなわち，核沸騰液層を消費する状態，次いで加熱面が乾燥する状態，そして最後に液体侵入による局所核沸騰の状態である．

（8） 本研究は主として直径 10 mm の平らな加熱面を用い，蒸気塊について一つの理想化した状態でおこなわれている．もっと大きな加熱面，あるいは長い加熱面などの場合には，Taylor 不安定から容易に想像されるように，加熱面がいくつかの個々の領域に分割されることになろう．そして各領域間で相互干渉その他も生じうるが，一つ一つの領域についてみれば本質的には本報に記したような状況が生ずるものと考えられる．

終わりにのぞみ，本研究に協力された安仲正好，石津尚澄，河東杉雄，国広昌嗣，高橋直樹，中村隆茂の諸君に謝意を表する．

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討　　論

〔質問〕　西川兼康（九州大学工学部）

（1）図7において低熱流束域で水位の影響がでており，しかもその影響が水位に関して順序正しくあらわれていないのはいかなる理由によるのであろうか．質問者らが 5×10^4 kcal/m²h 程度の低熱負荷で行なった実験（機論，34巻261号935ページ）では水位が低くなるほど熱伝達係数は順序正しく上昇しているが，著者が行なわれた高い熱流束域と質問者の行なった低い熱流束域では現象に根本的差異があるのであろうか．この点について著者のご意見を伺いたい．

（2）バーンアウト近傍および遷移沸騰において，気ほう塊の下に薄液膜が存在し，ここで核沸騰曲線延長上の熱流束で核沸騰が行なわれるというモデルが本研究の根本思想になっているように思う．事実図14の *1* および *2* 点について，液膜の消耗時の熱流束を著者の理論で，液体浸入時の熱流束を質問者らの研究（機講論，No. 700-6，101ページ）から計算して全熱流束を求めてみると，*1* 点については $q_{本実験}=8.7\times10^5$ kcal/m²h，$q_{計算}=9.2\times10^5$ kcal/m²h，点 *2* については $q_{本実験}=7.7\times10^5$ kcal/m²h，$q_{計算}=8.1\times10^5$ kcal/m²h となり，上述の思考モデルの妥当性を示すように思われる．薄液膜内で核沸騰曲線の延長上の熱流束で核沸騰が行なわれるという仮定は，他に基準が求められない現状ではやむをえないことと思うが，図15および16から薄液膜の厚さを推定してみると，それぞれ0.012および0.004 mmと非常に薄くなる．このような薄液膜内でいわゆる核沸騰が起こりうるであろうか．また，図15および16において初期液膜の蒸発による成長期間と考えられる部分の気ほう成長速度が核沸騰曲線延長上の熱流束を伝熱面で伝達すると仮定したときの成長速度より小さいことから考えると，薄液膜の存在は妥当としても，そこで核沸騰曲線延長上の熱流束の核沸騰が行なわれているという点には特に遷移沸騰における三重界面の性格の複雑性とも関連して今後究明すべき重大な問題を含んでいるように思う．この点についてご教示いただきたい．

〔回答〕　（1）低熱流束域での熱伝達特性に対する水位の影響は，西川氏らのご研究とまったく傾向が同じと考えている．すなわち図7において，熱流束一定で T_w-T_s を比較すると，$H=100\sim10$ mm でほぼ一定（簡素化のため $H=10$ mm の測定値の記入を省略してあるが），そして H が10 mm以下に低下するにつれて T_w-T_s が低下している．ただ $H=2$ mm まで下がると，水膜ドームの1サイクルの間に，加熱面が乾燥（その上に小さな水滴がまるまって残る）する場合がときどきはいり，そのため平均として T_w-T_s が急に上昇している．そして，$H=2$ mm 以下では定常状態の測定が不可能であった．以上の次第で，西川氏らの結果と相違してくるのは，大まかにいえば，水位 H がまだ2～3 mm 程度ではやくも熱伝達率最高点を経過してしまうということだけであるが，これも筆者らの加熱面直径が西川氏らのよりかなり小さく，したがって水膜ドームの圧力も高くなるから，自然なことと思う．要するに筆者らの結果は，西川氏らのご研究を全面的に支持するものと思っており，したがって本件に関する研究は当面不要であり，かつ本論文の主題とは直接の関係がないため図7を簡単に示すにとどめた次第である．なお，そういう次第で，この領域のデータは定量的に西川氏らのものほど精密さに努力を払ったものではないことを付記する．

（2）非常に貴重なご意見で，実はこうした問題とも関連して，従来とは別種の面から研究も進めている段階である．ただ本論文に関連して下記のことがらを回答申しあげたい．蒸気塊が加熱面から離脱する際，これは相当にマクロ的な現象である．したがって蒸気塊に置きかわる形で蒸気塊と加熱面の間に流入してくる液体は相当な厚さを持ち，それが加熱面に触れたところで微細な多数の発ぼうを開始，その吹出し蒸気が加熱面よりやや上部（高熱流束なので微細な発生気ほうの直径くらいのオーダの寸法と想像する）で合体しつつ，さらにこの合体蒸気層が上方の液体（さらにその上方には離脱した蒸気塊あり）を排除していく形になる．つまりご指摘のような薄い核沸騰液層自体が，蒸気塊離脱の瞬間にすぐさま機械的にその場に作られるというものではない．相当厚い液体の底部（加熱面）に，極端にいえば，少なくとも1回の核沸騰を経て，はじめて加熱面上に液層（気ほうと気ほうの間などには液体が残る）と，それをおおう合体蒸気層ができるという考えである．そして加熱面温度が非常に高くなると，発ぼうは密に，かつ気ほうは小さくなるから，いわゆる核沸騰液層は薄くなり，論理上の極限としては，前記の1回の核沸騰をした後，核沸騰がとだえ，残った液体は液膜の純蒸発で気化するという状況にまで立ち至るかもしれない．以上は少し極端に記した

が，かりにこうした状況に近いとすれば，たとえば4 μ程度の薄い液層も絶対不可能というわけではなくなる．それに，このような状況下では，図16からもわかるように，液層が気化消滅するまでの期間はきわめて短く，液層内部で何回も核沸騰が継続して生ずるものではないようである．要するに筆者らの考えは，加熱面上に液層（その上部は蒸気で分離された）ができてくる以上は少なくとも核沸騰がなければ不可能ということである．ただし，核沸騰曲線の延長上の熱流束という点は，ご指摘のとおり仮説である．また蒸気塊の離脱に伴い，加熱面の周辺のほうから中心のほうへ，前記の核沸騰と液層形成現象が進んでいくもので，瞬間的に加熱面上で一様かつ同時に生ずるわけではない．事実，図15，16でもサイクル初期の蒸気塊は加熱面周辺から生じ，ためにその形はいわばドーナツ状で，その内径がしだいに減少し，最後に内部がくっついてくるという経過を示す．したがって三重界面の問題も存在しよう．しかし，この場合も，ある厚さの薄液層がそれ自体でひとりでに構成されるのではないということに一考を払う必要もあろうかと思う．

536.423.1 : 532.5

高熱流束プール核沸騰の加熱面近傍に作用する力の研究*

甲藤好郎**，菊池浩平***

1. 序　論

核沸騰において高熱流束領域にはいると，加熱面上の多数の発ぼう点から発生し続ける蒸気が集合しつつ，大きな蒸気塊が加熱面上に形成され，ある大きさまで生長すると浮上していく．そしてプール沸騰では，この蒸気塊の周期的な生長，離脱と，バーンアウト発生機構の間に関係があることを著者のひとりと横谷が報告しているが[1]，このような面からみても，蒸気塊の挙動を支配する力学的状況を調べておくことは，高熱流束核沸騰に関連する諸現象の理解のため重要なことがらの一つと考えられる．

本報は，大気圧の飽和水内で高熱流束核沸騰を生じている円形水平加熱面に作用する力を測定するとともに，蒸気塊挙動を高速カメラで観測，それらの解析を通して蒸気塊に関する力学的状況の探究を試みた結果，および二，三の問題の考察について報告するものである．なお，上記のような測定は従来おこなわれていないが，蒸気塊の形状が比較的単純なほうが取扱いに便利なので，加熱面としては，直径 15 mm 程度の大きさのものを用いて実験をおこなった．

2. 実験装置

2·1 実験装置　図1に図式的に示すように，実験装置の主要部は水そう，加熱器，てんびんの3部分からなるもので，てんびんの片腕から水中につり下げた加熱器に働く力をてんびんで計測するシステムになっている．

水そうは，外部への熱損失をおぎないつつ水そう内の水を飽和温度に保たせるために補助加熱器を有するが，その沸騰による水の運動じょう乱の影響をしゃ断する必要がある．そのため水そうをガラス板で分割して二重水そうとし，補助加熱器は外そうのほうへ置き，加熱器は内そう内へつり下げるようにした．ここで内そうの寸法，形状は，断面 150×150 mm，高さ 290 mm の直方形である．なお水そう内の発生蒸気は復水器に導いて復水後，水そうにもどすようにし，水そう内の水位の時間的変化を防止した．

加熱器は外径 42 mm，高さ 40 mm の円柱形，ステンレス製で，水中に浮遊させるため防水式のものである．その内部断面は図2に示すごとくで，中央に内蔵された銅ブロックの上端面が沸騰を生ずる円形加熱面〔直径 14.5 mm（標準）のほか，16.7 mm，13.6 mm の3種類を製作使用した〕である．上記の加熱面の周縁に銀ろうで固着される部分のステンレス板は特に薄くし，厚さ 0.5 mm 以下である．銅ブロックの下部は直径 30 mm で，この部分にそう入した9枚の電気加熱板によって，バーンアウトまでの高熱流束加熱が十分におこなえるようにしてある．銅ブロックのテーパ部分の中心軸上には，3個の銅・コンスタンタン熱電対がある間隔をおいてそう入され，それによって測定される温度こう配から加熱面熱流束が定められる．その検定は，前もって発生蒸気を捕集する方法で精密におこなわれた．なお加熱器内において，銅ブロックと加熱器外壁間のすきまは石綿を充てんして断熱をおこなっているが，検定の結果によれば，加熱面上に高熱

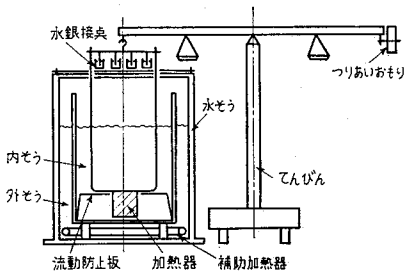

図 1　実験装置の主要部

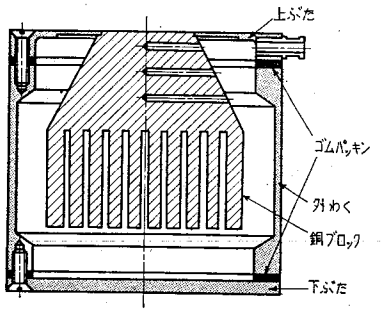

図 2　加熱器の構造（空げきには石綿充てん）

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流束核沸騰を生じているとき，加熱器への電気入力のうち加熱面以外の部分へ逃げる割合は非常に少なく，ほとんど無視しうる程度であった．

てんびんは，水そうの大きさに合わせたものを特別に製作して使用した．その腕には，中央支点から左右それぞれ 100 mm の位置に測定用分銅さらがさげられる．また支点から左方 200 mm の位置に加熱器をつり下げ，これに対し腕の右方端部にはつりあいおもりが設けられている．なお加熱器への電力供給，および温度測定用熱電対のための配線は，加熱器のつり下げわくに沿って設けてあるが，これらと外部配線との結合はそれぞれ水銀接点（図1参照）をもっておこない，なんらてんびんには力がかからぬようにした．

2・2 測定原理 静止した水中に加熱器がつり下げられているとき，加熱器の上下面に働く圧力の差に基づいて，加熱器に浮力が作用することはいうまでもない．

これに対し，加熱器上面中央にある加熱面で高熱流束核沸騰を生じさせると，本報で使用した程度の寸法の加熱面上には，比較的単純な形状の単一蒸気塊が周期的に生長，離脱，浮上を続ける状態となる（後出の図7参照）．そして，これによって広義の自然対流が生じ，加熱器上面に働く流体圧力は前記の静止水の場合より一般に低下することになるから，加熱器に働く浮力（時間的平均値）はそれだけ増加する．

この浮力増加はもちろんてんびんで容易に測定される．ただし，浮力の変化の大きさは一般に小さな値であるから精密，かつ慎重な測定を要することはいうまでもない．なお，これに関連し，本報では図1に示すごとく流動防止板を設けてある．この流動防止板は，加熱器断面よりわずかに大きい直径の穴を持ち，その穴のなかで加熱器が自由に上下移動できるが，流動防止板の下方にある水を静止させる目的のものである．もし加熱器の側壁に沿う水の流れ（上方の自然対流によって誘起される）を放置すると，水の粘性力が加熱器を上向きに多少ひきあげるように働き，測定に影響を与えるからである．

さて，上記の核沸騰状態の場合，加熱器の上面中央にある加熱面だけに注目してみると，その上には絶えず蒸気塊が生長を続けており，蒸気塊内の蒸気圧力が加熱面上に作用しているとみてよい（蒸気塊と加熱面の間には核沸騰をしている液層があるが，それはきわめて薄い）．そして，この蒸気圧力 p_V(時間的平均値)と，加熱面周縁近傍の液体圧力 p_L(加熱器上面の水位における）との間には，なんらかの差があるはずである．

この圧力差 p_V-p_L は次のような方法によって，少なくとも近似的に測定ができる．すなわち図3に示すように加熱面の上方に距離 s をへだてて，加熱面と同直径の薄い第2の円形加熱面をおく．そして加熱器には電気入力を加えず，上記の第2加熱面（てんびんには結合していない）だけに電力を加えて加熱，沸騰させた状態で，加熱器（てんびんに結合）に働く浮力を測定し，高さ s を変えた実験から $s\to 0$ の極限における浮力の値を求める．実際に実験をした結果では，$s=0$ の近傍で s を変えた場合，浮力は単調な変化を示し，上記の外そう極限値を求めるのは容易であった．

ところで，この $s\to 0$ の極限状態と，前に記した普通の沸騰状態（第2加熱面を使わない）とを比較するに，加熱量がひとしいとき蒸気塊の生成状況は両者，同じであり，したがって流体の流動状況は両者でほとんど変わらない．ただ加熱面上に，前者では加熱面近傍の液体圧力が作用しており（蒸気塊は第2加熱面上に生じ，加熱器の加熱面とはしゃ断されている），後者では加熱面上に蒸気圧力が作用している．したがって前者および後者において，同じ加熱量に対し，それぞれ加熱器にかかる浮力を測定，両者の浮力差を加熱面面積でわれば p_L-p_V が得られることになる．

2・3 第2加熱面について 前節に記した第2加熱面を実現するにあたり，本報では，0.6ϕ のエナメル線を心にして，その上に 0.14ϕ のニクロム線をらせん状に密に巻いたものを，厚さ 1 mm のベークライト円板上にかとり線香のごときうず巻形に固着する方法によった．これは，もちろん完全な円形加熱面とは多少相違する．

しかし現在問題になるのは，加熱面上の発ぼう点や核沸騰ではなく，発生蒸気が集合して生ずる蒸気塊である．そして，少なくとも本報で使用した程度の大きさの加熱面の高熱流束核沸騰では，加熱量のすべてが加熱面上で蒸気発生に消費されるとみてよいことを，著者のひとりは前に確かめている[2]．したがって，上

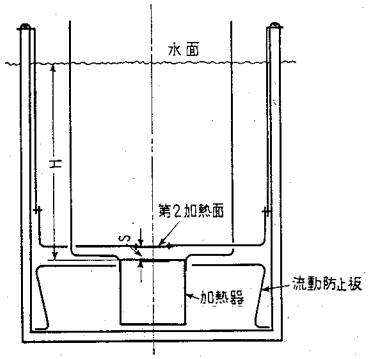

図 3 第2加熱面の配置

記の程度の加熱面の形式の相違は蒸気塊の生成状況に対し，ほとんど影響しないはずである．また実際に本研究において高速カメラで検討した結果もそれを支持するものであった．

3. 実 験 結 果

3·1 沸騰による加熱器浮力の増加　本報の主題ではないが，2·2 節のはじめに記した核沸騰時の加熱器浮力増加特性について述べておく．この浮力増加は沸騰の状態，すなわち加熱面からの伝熱量によって変化するが，図4は一例として，自由水面から加熱面までの水深 $H=150$ mm の場合の測定結果（加熱面直径 14.5 mm）を示すもので，低熱流束域の測定をも含む）同図にみられるごとく，低熱流束の孤立気ほう領域と，高熱流束の合体蒸気塊領域とで，浮力の増加特性に相違があらわれる．なお，他の条件下の実験結果もすべて，図4とまったく同じ特性を示した．

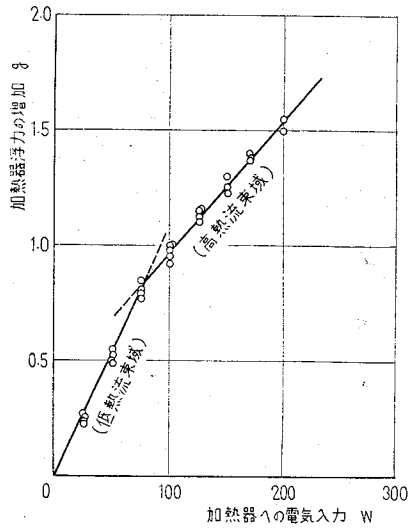

図 4　沸騰による加熱器浮力の増加
（加熱面直径 14.5 mm，水深 $H=150$ mm）

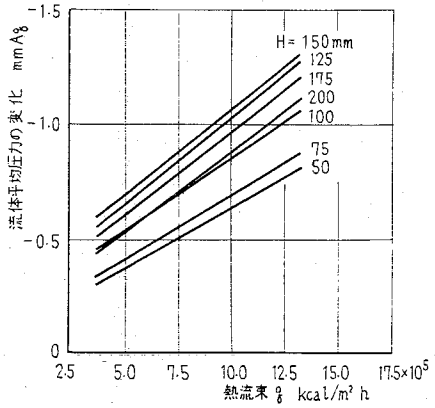

図 5　加熱器上面に働く流体平均圧力の低下
（高熱流束域）

さて図5は，図4のような測定結果を総合して，種々の水深 H における測定結果を特に高熱流束域について示すものである．ただし図5の縦軸は，上記の浮力増加を加熱器上面（直径 42 mm）面積で除して負号をつけたもの，すなわち加熱器上面に働く流体平均圧力の低下の形としてある．また図5の横軸は，加熱面からの熱流束 q である．

図5によれば，水深 $H=$一定 の場合，加熱器上面に働く流体平均圧力は，熱流束がまし蒸気発生速度が増加するにつれて低下（同水位の遠方静止液体圧力より低下）する．加熱面から自由水面の間の液体内に存在する蒸気が，流体内の圧力差発生の素原因（この非平衡圧力差を打ち消す方向に自然対流が生じている）だから，上記の結果は自然な傾向といえよう．

また図5によれば，水深 H が $H=50$ mm から増すに従い流体圧力低下も増大していくが，$H=150$ mm で圧力低下が最大となり，それ以上の水深では再び圧力低下が小になる傾向を示している．これは次の理由によると考えられる．すなわち，最初，水深が小なる範囲では水深が増すほど前記の非平衡圧力差が大となり，いわば煙突などの場合に似て流体圧力低下が大きくなる．しかし本実験に用いた水そうの内そう断面の大きさは，2·1 節に記したごとく 150×150 mm に定まっている．したがって水深 H が，ある程度以上大きくなると，内そうの側壁面が流体のフローパターンに影響を持つような特殊な状態になる．このときの状況は底部の閉じた熱サイフォンのような形式に近づくものであり，流体の流動をしだいに阻害する方向に変化するものである．

3·2 加熱面上に生長する蒸気塊内の蒸気圧力

2·2 節の後半に述べた方法で，加熱面上に生長する蒸気塊内の蒸気圧力の時間的平均値 p_V と，加熱面近傍の液体圧力 p_L（加熱面の水位における）との差 p_V-p_L を測定した．図6は直径 14.5 mm の加熱面について，

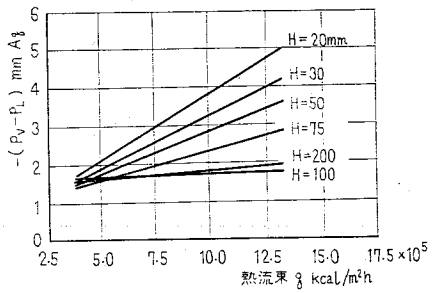

図 6　蒸気塊内の蒸気圧力

水深 $H=20\sim200$ mm で測定した p_V-p_L を，熱流束 q の関数として示すものである．これによれば，$p_V-p_L<0$ すなわち $p_V<p_L$ である．また p_V-p_L は，水深が $H\geqq100$ mm の深い領域では熱流束 q および水深 H にほとんど独立の一定値をとるが，水深が $H<100$ mm の浅い領域では q および H によって変化するという特徴をもっている．

なお，直径 16.7，14.5，13.6 mm の3種類の大きさの加熱面（面積比，1.50：1.12：1.00）のそれぞれについて，水深 $H=100$ mm の場合の p_V-p_L を測定比較したが，少なくともこの範囲の加熱面面積の変化は結果になんら影響がなく，p_V-p_L は3種類の加熱面に対し同じ値を示した（ここで熱流束の検討範囲は $q=5\times10^5\sim1.3\times10^6$ kcal/m²h である）．

4. 蒸気塊内の蒸気圧力に関する解析

4·1 加熱面上の蒸気塊の挙動　図6に示すような特性が p_V-p_L に現われる原因を明らかにするためには，まず加熱面上に生成，離脱をくりかえす蒸気塊のようすを知らねばならない．そのため高速カメラによる蒸気塊挙動の観測をおこなった．図7に示すように，加熱面上に存在する薄い液層内の核沸騰から生ずる蒸気が集合し，蒸気塊が液層上に生長，それが大きくなると加熱面から切断されたあと，ほとんど直ちに次の蒸気塊が生長を開始するのであるが，高速カメラの撮影フィルムの解析から次の結果を得た．

（i）蒸気塊が加熱面から切断される周期には多少ばらつきがあるが，統計平均周期 τ は約 60ms で，これは水深 H，熱流束 q によってほとんど変化しない（本報で扱う H と q の範囲で）．この τ の値，およびその特性は，著者のひとりと横谷が本報とほぼ同程度の寸法の水平円形加熱面上の実験で前に求めた結果[1][2]に合致している．

（ii）図7に示すごとく蒸気塊の上部界面の高さ（加熱面からの）を h とするとき，もちろん h は時間 t とともに増加する．さて図8は，水深 $H=100$ mm の場合の h の時間的変化を示すものであるが，蒸気塊上部界面の上昇速度 $v\equiv dh/dt$ は蒸気塊生長中ほぼ一定に保たれ，かつ熱流束 q にも独立である．これに対し図9は，水深 $H=30$ mm の場合であって，このように水深が浅くなると $v\equiv dh/dt$ が熱流束 q の影響を受け，q の増加とともに v が大きくなる傾向を示している．なお図9の破線は，図8の結果を比較のために示したものである．水深が浅くなると蒸気塊の上方に存在する液体の量が少なくなるため，上記のような特性が現われるものと考えられるが，ここでは現象論的に上記の結果を認めるにとどめる．

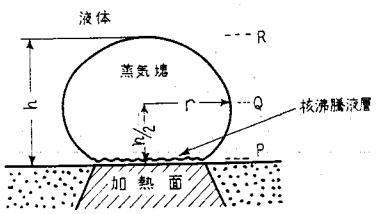

図 7　加熱面上の蒸気塊

4·2 理論的解析　前節に記した蒸気塊挙動の観測結果をもとに，蒸気塊内圧力について，簡単な解析をおこなうが，蒸気塊の周囲にある液体の慣性力の影響を無視した場合の結果を実験結果と比較してみることにする．

蒸気塊の周囲の液体が蒸気塊界面におよぼす静圧は，もちろん界面の水位で異なり，たとえば図7で P, Q, R の水位における水圧を考えると，P の水位の水圧が最も大きく，R の水位の水圧が最も小さい（なお 2·2 節で p_L と記した圧力は P の水位の水圧である）．他方，蒸気塊内部においては蒸気密度もきわめて小さく，したがって圧力は近似的にいって蒸気塊内でほぼ一様な値 p_V'（任意の時刻における瞬間値）をとると考えてよいであろう．

さて上述のごとく，蒸気塊界面に働く液体圧力は界面上の位置によって異なるが，簡単のため，その平均圧力 p_m について考えるに，それは近似的にいって，

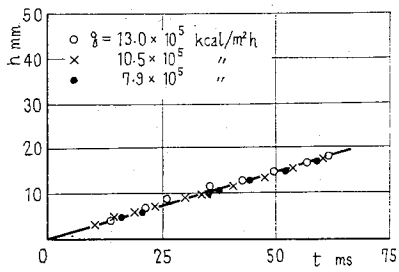

図 8　蒸気塊上部界面の上昇
（水深 $H=100$ mm の場合）

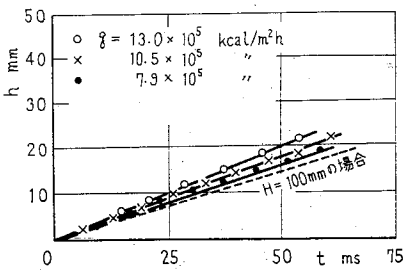

図 9　蒸気塊上部界面の上昇
（水深 $H=30$ mm の場合）

加熱面からの高さ $h/2$ の水位（図7のQ点）の水圧にほぼひとしい．したがって図7でPの水位の水圧を p_L とするとき，p_m は次のごとく与えられる．

$$p_m=p_L-\rho_L g\frac{h}{2} \quad \cdots\cdots(1)$$

ここに ρ_L：液体密度，g：重力加速度である．一方，蒸気塊界面は曲率をもち表面張力が働くから，蒸気塊内の蒸気圧力 p_V' は，上に求めた周囲液圧 p_m より高くなるはずである．いま，この圧力差を Δp とするとき，図7からみて，近似的に次のごとき程度の値になるとして大きな誤りはないであろう．

$$\Delta p=\sigma\left(\frac{1}{h/2}+\frac{1}{r}\right) \quad \cdots\cdots(2)$$

ここに σ は表面張力，r は図7のQの高さにおける蒸気塊の水平断面の半径である．そして式（1），（2）から結局，蒸気塊内の蒸気圧力（瞬間値）p_V' は次式で与えられる．

$$p_V'=p_m+\Delta p=p_L-\rho_L g\frac{h}{2}+\sigma\frac{1}{h/2}+\sigma\frac{1}{r} \quad \cdots\cdots(3)$$

ところで蒸気塊の生長に伴い，特に h および r は時間的に変化する．したがって蒸気塊が加熱面上で生長する期間 $t=0\sim\tau$ の間に，蒸気圧力 p_V' は相当に変化する．なお p_V' の変化にくらべれば，液体圧力 p_L の変化は非常に小さいと考えられ，式（3）右辺で p_L は近似的に一定とみなしてよいであろう．このとき式（3）から，時間 $t=0\sim\tau$ 内の p_V' の平均値 p_V を求めると

$$p_V=\frac{1}{\tau}\int_0^\tau p_V'dt=p_L-\frac{1}{\tau}\int_0^\tau\rho_L g\frac{h}{2}dt+\frac{1}{\tau}\int_0^\tau\sigma\frac{1}{h/2}dt+\frac{1}{\tau}\int_0^\tau\sigma\frac{1}{r}dt \quad \cdots\cdots(4)$$

さて式（4）右辺第2，3項に含まれる h については，図8，9の実験結果から $h=vt$（ここに v は蒸気塊上部局面の上昇速度で一定値）の関係が与えられている．この場合，式（4）右辺第2項の積分についてはなんら問題を生じないが，式（4）右辺第3項については，その積分値が無限大になってしまう．しかし表面張力が無限大の圧力を生ずることはありえず，また実際，撮影フィルムの観察によれば，加熱面から蒸気塊が切断したあとに続く新しい蒸気塊は，加熱面上に存在する多数の分断気ほう群が合体して生長を開始する様相を呈している．そして，この気ほうの直径は，いつも蒸気塊の全生長時の h の値（つまり $h=v\tau$）の約 1/10 程度の大きさであることが観察された．したがって式（4）右辺第3項の積分にあたり，簡単のため積分下限を $t=0$ の代わりに $t=\tau/10$ とおきかえても，それほど重大な誤差が導入されることはないであろう．なお最後に，式（4）右辺第4項に含まれる r は，$t=0\sim\tau$ の期間を通して加熱面の半径と同程度の大きさの範囲内で変化するものであるから，時間的変化も比較的少なく，かつ式（4）右辺第3項にくらべ第4項の影響は一般に小さいものである．したがって式（4）右辺第4項においては，$1/r$ の代わりに，蒸気塊生長期間中の $1/r$ の時間的平均値（それを $1/r_m$ と記すことにする）を代入しても大きな誤差は生じない．

以上に記したことがらを考慮すると，式（4）は次の結果を与える．

$$p_V-p_L=-\frac{1}{4}\rho_L g v\tau+\sigma\frac{2\ln 10}{v\tau}+\sigma\frac{1}{r_m} \quad \cdots\cdots(5)$$

ここに v の値は図8，9の実験結果から与えられ，また τ の値は4·1節に述べたごとく $\tau=60$ ms である．また $1/r_m$ については，蒸気塊生長の撮影フィルムの解析から，$t=0\sim\tau$ の間における $1/r$ の時間的平均値が容易に求められるゆえ，その実測データを使用する．

さて，このようにして式（5）から計算した p_V-p_L を，水深 $H=30$ mm および 100 mm の場合に対し図示したものが図10の破線である．また，図10の実線は，すでに図6に示した p_V-p_L に関する実験結果のうち水深 $H=30$ mm および 100 mm の場合を再び掲げ比較したものである．図10によれば，前記の解析結果は少なくとも実験結果の特徴をかなりよく説明しているといえよう．もちろん本報でおこなった解析には，いくつかの仮定が含まれ，かなりおおまかな解析であることは否定しえない．しかしながら，物理的に真偽不明の仮定を多数含んでいるというほどでもなく，したがって実験結果のよってきたる原因を相当程度，明らかにしえているのではないかと考えられる．

5. 討　　論

以上，本報でおこなった実験，およびそれに関連す

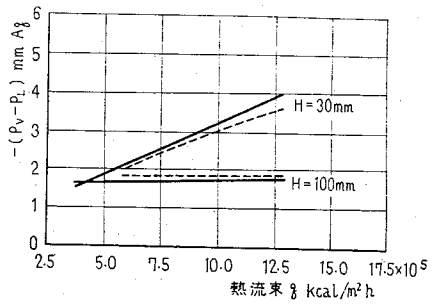

図 10　蒸気塊内の蒸気圧力—実測（実線）と解析（破線）の比較—

る簡単な解析的考察について述べたが，これらの結果と関係する二，三の重要なことがらを記しておくことにする．

5・1 蒸気塊の挙動について 加熱面上の蒸気塊の生長は，加熱面上の核沸騰液層から蒸気を供給されつつ，液体内で蒸気体積を増してゆく形で生ずる．ところで前章までに得た結果，なかでも特に図10の結果などによれば，蒸気塊の生長期間全体の平均的傾向としては，蒸気は周囲液体を押しのけながら拡大してゆくというより，液中を浮上しながら生長していく性格のほうがはるかに強いものであるといえるであろう．図10によれば，実験的にも，また式（5）からも $p_V < p_L$ である．ただ式（3）によれば，h が小なるとき右辺第2項より第3項が大になるから $p_V' > p_L$ である．したがって蒸気塊生長の初期段階では，周囲液体を排除してゆく作用もありうると考えられるが，その段階をすぎた後は浮力で上昇しつつ，なお加熱面上の核沸騰液層から供給される蒸気を受けて体積を増大させてゆくものと考えられる．そして，この加熱面から蒸気塊への蒸気供給速度は時間的にほぼ一定に保たれる（著者のひとりが本報と同程度の寸法の加熱面の実験で前に確かめている[2]）から，やがて浮上による蒸気塊体積増加速度のほうが大になると，それに追いつけなくなり，蒸気塊の切断を生ずるものと推定される．

なお，上記のことがらに関連して興味あることは，蒸気塊上部界面の上昇速度 v である．これについて，水深がある程度以上深い場合をみるに，図8（ここで $v \equiv dh/dt$ である）が示すごとく，ほぼ $v = 300\,\mathrm{mm/s}$ という結果が得られている．ところで，この v の値は，相当直径 20 mm 程度の単独気ほう（本報の蒸気塊は直径約 15 mm の加熱面上に，それと同程度の大きさをもって生ずるものであることに注意）が水中を自由浮上するときの浮上速度―たとえば Ivey[3] の論文参照―にほぼ近いものである．もちろん，上記の二つの場合は，同じ浮上といっても条件に相違もあるから，単純に相似させることはできないにしても，一応，注目してよいことがらだと思われる．

5・2 飽和沸騰における高熱流束域の下限熱流束 周知のごとく飽和核沸騰では，低熱流束時には各発ぼう点から孤立気ほうが生長，離脱をおこなうが，高熱流束になると，加熱面上に密に分布する多数の発ぼう点で生成される蒸気が集合して，大きな蒸気塊を形成するようになる．そして後者の状態では，多くの場合，加熱面から伝えられる熱量は，ほとんどすべてが加熱面上での蒸気生成に消費されるとみてよい．特に本報で用いたような寸法，形式の加熱面では，そのとおりである．したがって，この場合，蒸気は加熱面上から一様に吹出すごとき状況に近く，その吹出速度 v_s は次のごとくなる．

$$v_s = q/\rho_V L \quad \cdots\cdots (6)$$

ここに q：熱流束，ρ_V：蒸気密度，L：蒸発熱である．

ところで，こうして加熱面から吹出してくる蒸気が，加熱面上に蒸気塊を形成しているのであるから，少なくとも蒸気吹出速度 v_s が，蒸気塊上部界面の上昇速度 v（これは5・1節に記したごとく浮上現象的で，v_s とは一応独立なもの）より高くなっているはずである．したがって，この高熱流束域（いわゆる合体気ほう域）の下限熱流束は，式（6）の v_s を v にひとしいとおいて求められることになり

$$q = v\rho_V L \quad \cdots\cdots (7)$$

さて本報の実験では，5・1節に記したごとく，水深がある程度以上深い場合，$v = 300\,\mathrm{mm/s}$ である．また大気圧の飽和水蒸気に対し，$\rho_V = 0.597\,7\,\mathrm{kg/m^3}$，$L = 538.8\,\mathrm{kcal/kg}$ であるから，これらの値を式（7）に代入すると，高熱流束域の下限熱流束として次の値が得られる．

$$q = 3.69 \times 10^5\,\mathrm{kcal/m^2 h}$$

他方，浮力に関する実験結果の一例として示した図4において，浮力特性に変化を生じている箇所は，低熱流束域と高熱流束域の境をなしている．そして，この位置における実測熱流束は次の値である．

$$q = 3.66 \times 10^5\,\mathrm{kcal/m^2 h}$$

なお図4は，水深 $H = 150\,\mathrm{mm}$ の実験であるが，$H = 100, 125, 175, 200\,\mathrm{mm}$ のそれぞれについておこなった実験でも H によって浮力の大きさには相違（たとえば図5参照）が現われたにかかわらず，浮力特性に変化を生ずる位置の熱流束は，上記とほとんど同じ値であった．ともあれ，これらの結果は，前述の式（7）による計算結果に近いものである．もちろん上記の比較において，両者の限界熱流束値があまりにもよく合いすぎている点には，偶然の要素がからんでいることもあるかもしれないが，しかし，式（7）の導出に際し，特に大きな仮定は使用していないことを記しておくのはむだでないであろう．

5・3 飽和沸騰におけるバーンアウト熱流束 さきに著者のひとりは，各種のプール沸騰系のバーンアウトについて実験的研究をおこない，加熱面からの蒸気塊切断周期と，加熱面上の核沸騰液層の消耗時間がひとしくなる点でバーンアウトが生ずる（この場合，加熱面上への液体補給は蒸気塊切断時にのみ自動的に生じ，それに要する時間は蒸気塊切断周期に比べれば小

さい）ことを報告，そしてバーンアウト熱流束 q について次式を提出している[2]．

$$\tau=\xi\rho_L\delta_0 L/q \quad \cdots\cdots(8)$$

ここに $\xi=8/9$（定数），ρ_L：液体密度，δ_0：核沸騰液層の初期厚さ，L：蒸発熱である．なお，上記の δ_0 について $\delta_0\propto 1/q^{1.5}$ の経験的関係がある．

さて式（8）は，加熱面の近くに干渉物体を置くとか，低水位の場合などの特殊な沸騰系にも成立するものであるが，以下には特に通常の沸騰系のみを考えることにする．このとき 5·1，5·2 節に記したことがらから考えて，蒸気塊の挙動にかなり関連の深い基本因子として，（i）蒸気塊上部界面の上昇速度 v，（ii）蒸気塊切断時の上部界面の高さ（加熱面からの高さ）$h_{\max}$，および（iii）加熱面からの蒸気吹出速度 $q/\rho_V L$ をあげることができる．そして，この 3 因子だけによって現象が大きく支配されていると仮定した場合，つぎのような結果が導かれる．

まず，蒸気塊切断周期 τ に対し次式が成立する．

$$\tau=h_{\max}/v \quad \cdots\cdots(9)$$

つぎに核沸騰液層の初期厚さ δ_0 については，これが長さの次元をもつことを考慮し，かつ現象が前記 3 因子だけに支配されるという仮定，および $\delta_0\propto 1/q^{1.5}$ の経験的関係があるという事実を配慮すると次の関係が導かれる．

$$\delta_0=\text{定数}\times h_{\max}\left(\frac{v}{q/\rho_V L}\right)^{1.5} \quad \cdots\cdots(10)$$

なお．式（9），(10) に含まれる v は，5·1 節に記したごとく，液中の気ほうの浮上速度に近い値をとるが，周知のように沸騰で普通生ずる大きさの気ほうや気塊の浮上速度は，おおまかには $\sqrt[4]{\sigma g(\rho_L-\rho_V)/\rho_L{}^2}$ に比例するとして大きな誤差は生じない．したがって近似的に次のごとく書いて十分である．

$$v=\text{定数}\times\sqrt[4]{\sigma g(\rho_L-\rho_V)/\rho_L{}^2} \quad \cdots\cdots(11)$$

さて，式（9），(10)，(11) の関係を式（8）へ代入すれば，ただちに次式が導かれる．

$$\frac{q}{\rho_V L}\Big/\left[\frac{\sigma g(\rho_L-\rho_V)}{\rho_V{}^2}\right]^{1/4}=\text{定数}\left(\frac{\rho_V}{\rho_L}\right)^{0.1} \quad \cdots(12)$$

ここで右辺の $(\rho_V/\rho_L)^{0.1}$ は，指数が 0.1 のため，変化が非常に小さい．したがって，その影響を無視すると，式 (12) はバーンアウト熱流束に関する Kutateladze の式に一致する．なお，非常な低圧になると，$\rho_V\to 0$ のため式 (12) 右辺の値も零に近づくという結果になってしまうが，そうした圧力下では，加熱面上に生ずる一次気ほうが巨大なものとなり（秋山らの研究[4]参照），普通の沸騰の場合とは状況が異なってくるから議論から除外すべきことを付記しておく．

本節において，式（8）から式 (12) を導く過程のすじ道自体は，さきに著者のひとりが発表したもの[1]と同形式である．しかし本報の実験的事実をふまえている点で，前報とは相当に異なるものといえる．ただし式 (10) の関係については，けっして自明のものでなく，ここに大きな問題がまだ残されているといえるであろう．

6. 結　　論

（1） 直径 15 mm 程度の大きさの円形水平加熱面を持つ加熱器を，てんびんによって大気圧水中につり下げ，高熱流束核沸騰時の浮力，および蒸気塊内圧力の測定をおこなった．

（2） 蒸気塊内圧力 p_V は，加熱面水位の水圧より低い値を示した．また p_V は，加熱面水深 $H\geqq 100$ mm では水深 H，熱流束 q に関係なく，ほぼ一定値を示したのに反し，$H<100$ mm では H および q によって値が変化した．

（3） 高速カメラによって蒸気塊の挙動を観測し，蒸気塊切断周期 τ，蒸気塊上部界面の上昇速度 v などについての特徴をはあく，それに基づいて蒸気塊内圧力 p_V の簡単な解析をおこなったところ，上記（2）項の特性をよく説明する結果を得た．

（4） 以上の結果からの判断によれば，蒸気塊の生長は，蒸気塊が液中を浮上しつつ，加熱面からの供給蒸気を受けて自然に体積を増加させてゆくという性格を強く持つものである．なお蒸気塊上部界面の上昇速度 v の実測値も，同程度の大きさの単独気ほうの自由浮上速度にほぼ近い値を示した．

（5） 加熱面からの蒸気吹出速度 v_s が，蒸気塊上部界面の上昇速度 v を越えるとき，蒸気塊を伴う高熱流束域が生ずるとの考えに基づき検討をおこなったところ，少なくとも本報の実験結果は非常によく合致した．

（6） 本報の実験結果に基づきバーンアウト熱流束に関する一考察を示したが，核沸騰液層の初期厚さを支配する現象がまだ明確でない．

終わりにのぞみ，本研究に協力された金信行，小泉英夫の両君に謝意を表する．

文　　献

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（2） 甲藤・横谷，機論，**37**-295（昭 46-3），427．
（3） Ivey, H.J., *Int. J. Heat & Mass Transf.*, **10** (1967), 1023.
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536.423.1 : 532.5

飽和プール核，遷移沸騰における蒸気塊の挙動*

甲藤好郎**，横谷定雄**

1. 序

たとえば電流加熱細線のように，発生一次気ほうの大きさに比べて寸法の小さな加熱面とか，あるいは一次気ほうが加熱面と同程度の大きさになる低圧沸騰などの特殊な状況は除くこととし，普通の条件下の飽和プール核沸騰では，加熱面温度が高くなると，一般に加熱面上の薄い核沸騰液層内で生成される蒸気が集合して蒸気塊を形成するようになる．そして，この蒸気塊は，ある大きさまで生長すると加熱面から切断，浮上し，その後には，また新しい蒸気塊の生長と切断とが継続する．

そして，さらに加熱面温度が上昇するに伴い，沸騰の状況が核沸騰からバーンアウト点を経て遷移沸騰（さらには膜沸騰）に移行するわけであるが，ここで蒸気塊の生長，切断挙動を支配する力学的状況を知ることは重要な問題といえるであろう．なぜなら，多くの場合，加熱面近傍における蒸気塊の周期的滞留現象が，バーンアウトや遷移沸騰の発生に大きな影響を持っているからである［文献(1)，(2)など参照］．そして本田・西川[3]は，直径8 mmの水平円形加熱面上の大気圧の飽和水のプール沸騰の実験をおこなって，蒸気塊の挙動を観察，蒸気塊の fD_d 値（ここに f：蒸気塊の切断ひん度，D_d：切断蒸気塊直径），および蒸気塊の浮上運動に関する抵抗係数 c_D が，核沸騰域と遷移沸騰域とで性格を異にすることから，両沸騰域において蒸気塊の流体力学的挙動に相違があるのだとしている．そして彼らは，核沸騰域と遷移沸騰域の間で，蒸気塊が加熱面から受ける力に相違を生ずる（遷移沸騰域では蒸気塊に覆われた加熱面上に乾燥部が発生，そのため「表面張力による付着力」および「底面圧力の変化」が生ずる）ものと考え，その想定下に蒸気塊の運動方程式を導いて遷移沸騰熱伝達の解析をおこなっている．なお本田らは，蒸気塊の挙動観察に際し，次の特殊方法によっている．すなわち，電磁オシログラフの暗箱を撮影機として使用し，蒸気塊の像のうちで加熱面の上方1 mmの位置にある部分（換言すれば，その位置で切断した蒸気塊の横幅だけ）を，幅1 mmのせまいスリットを介してオシロ紙に連続撮影して行く方法である．

さて本報は，前記の本田・西川の研究とほとんど同じ条件，すなわち直径 10 mm の水平円形加熱面上の大気圧の飽和水のプール沸騰を扱っているが，高速カメラによる撮影結果から蒸気塊の生長，切断挙動を調べ，かつこれに関する流体の力学的状況について理論的検討をおこなった結果について報告するものである．

2. 蒸気塊の挙動分析に使用した実験データ

本報において，蒸気塊の挙動の観察，分析に使用した高速撮影フィルムは，図1に概念的に示す実験装置における沸騰について求められたものである．ここに円すい形をしている銅ブロック上端の円形水平面（直径 10 mm）が沸騰加熱面となり，なお銅ブロック下部には，遷移沸騰の実現のため一色・玉木[4]によって考案された方法により，板状電気加熱器による加熱，および空気流による冷却ができるようにしてある．加

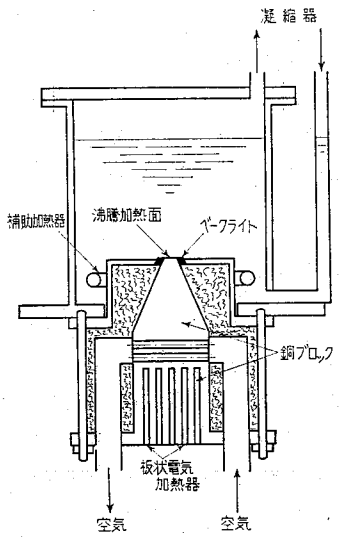

図1 実験装置主要部

* 昭和45年5月21日 第7回日本伝熱シンポジウムおよび昭和48年5月31日 第10回日本伝熱シンポジウムにおいて講演，原稿受付 昭和48年7月14日．

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熱面温度および熱流束は，銅ブロック軸心上に配置した熱電対で定められる温度分布から測定される．液体としては蒸留水を用い，補助加熱器によって飽和温度に保つが，加熱面から自由水面までの高さは約 100 mm にしている．加熱面上に発生する蒸気塊を高速カメラで撮影するに際し，撮影速度および露光時間は，最大約 2 000 こま/秒 および 1/5 000 秒/こま である．フィルム上に記録される現象について，その速度，寸法などの決定のため，フィルム上に 1 ms 間隔のタイミングマークおよび直径 6 mm の丸棒を現象と同時撮影するようにしている．

3. 蒸気塊の挙動の実験的観察

図2は，図1の実験装置でおこなわれた一連の実験において求められた沸騰曲線を示すもので，ここに破

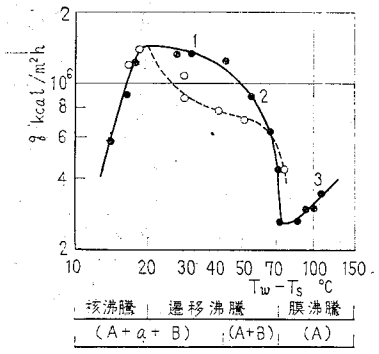
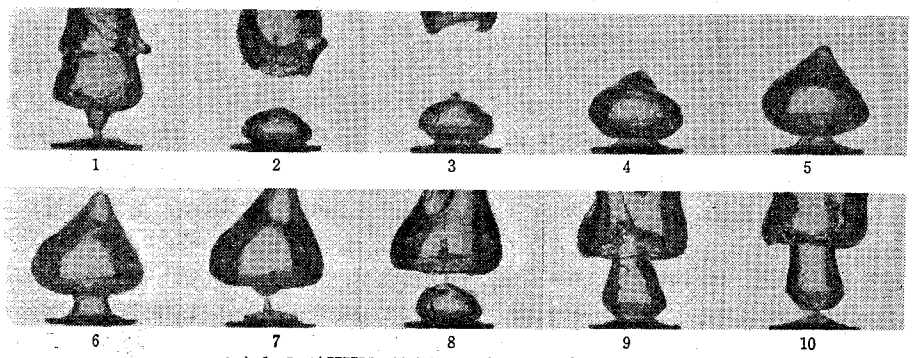


図 2. 沸騰曲線

曲線（記号 ○ で示すデータを結ぶ）は比較的に新鮮な加熱面を用いた実験，また実曲線（記号 ● で示すデータを結ぶ）は比較的に汚れた加熱面を用いた実験の結果である．図 2 に明らかなように，加熱面の過熱度 T_w-T_s（ここに T_w：加熱面温度，T_s：飽和温度）に対して，だいたい $T_w-T_s<20°C$ が核沸騰域，$20°C<T_w-T_s<70°C$ が遷移沸騰域，$70°C<T_w-T_s$ が膜沸騰域という状況になっている．なお，特に遷移沸騰域において，加熱面の汚れ（それに伴い加熱面と液体との親和性が変化）が伝熱特性に著しい影響を持つという結果を図2は示しているが，これは遷移沸騰域で加熱面上に乾燥部の発生があることに関係した結果であることが，従来すでに明らかにされている[2][5]．

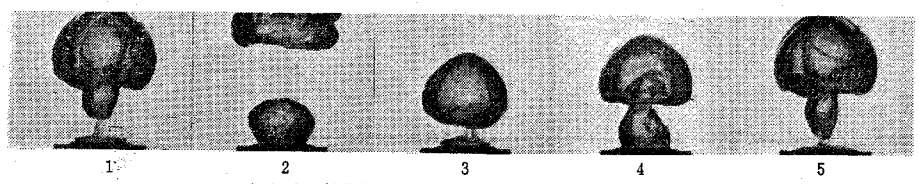
3・1 生長，切断形式による蒸気塊の分類 図2に示した沸騰実験において，蒸気塊の発生，切断挙動を観察してみると，次のことが見出される．すなわち，いま図 2 において *1*, *2*, *3* の番号を付した実験点を代表に選び，そこでの蒸気塊の生長，切断状況を示すことにすると，実験点 *1* の状況は図 3 の写真，実験点 *2* の状況は図 4 の写真，実験点 *3* の状況は図 5 の写真のごとくになる．そして，この図 3～5 にあらわれている蒸気塊の挙動形態について通覧し，ごく外見的な特徴から，便宜上，蒸気塊を下記の 3 種類の形に分類しても差し支えないであろう．

（I） A形蒸気塊 これは，図 3 のこま 1～5，図 4 のこま 1～3，図 5 のこま 1～5 にみられるような蒸気塊を呼ぶもので，先行蒸気塊にあまり影響されず，



こま 1～5：時間間隔＝11.3 ms，こま 5～10：時間間隔＝8.8 ms

図 3 蒸気塊の挙動（図 2 の実験点 *1*）



こま 1～3：時間間隔＝26.8 ms，こま 3～5：時間間隔＝15.0 ms

図 4 蒸気塊の挙動（図 2 の実験点 *2*）

かなり独立的に大きく生長し，やがて加熱面から切断する．なおこのA形蒸気塊は加熱面から切断し，ある距離を上昇すると球冠状に変形し，底部に誘起流れ（伴流）を生ずるが，少なくともそれ以前は球形に近い形を保持する．

（Ⅱ） a形蒸気塊　これは図3のこま 3～6 にみられるように，A形蒸気塊の下部にすぐ接続するかのような形で生ずる蒸気塊である．そして水平方向への生長がほとんど無く，蒸気塊頂部先端が，上方の先行A形蒸気塊内部へ突き進む形をとって上昇しつつ生長し，やがて加熱面から切断する．なお多くの場合，a形蒸気塊は加熱面から離脱後，先行A形蒸気塊の内部へ吸いこまれてしまう．これによってみても，a形蒸気塊の生長，切断挙動は，先行A形蒸気塊の底部付近の誘起流れ（伴流）によって直接，強く支配されていると推定される．また，a形蒸気塊を随伴するA形蒸気塊では，加熱面からのA形蒸気塊の切断が外見上，不めいりょうな形になる．

（Ⅲ） B形蒸気塊　これは，図3のこま 7～10，図4のこま 3～5 にみられるように，A形に随伴するa形蒸気塊，あるいは単独A形蒸気塊のあとに現れる蒸気塊で，比較的に独立性を持つけれども，先行蒸気塊（加熱面からすでに相当離れて浮上中の）の誘起流れの影響を受けて浮上，やがて加熱面から切断する．従って，B形蒸気塊は，A形蒸気塊とa形蒸気塊の間の中間的な性格のものである．

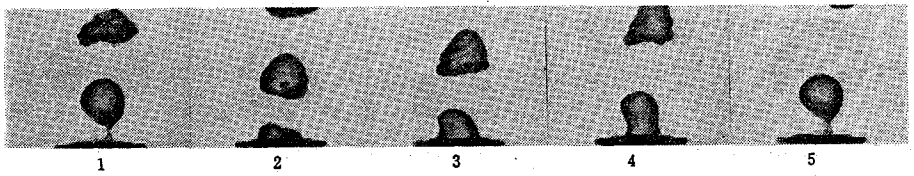
3・2 蒸気塊の連鎖形式とその領域　蒸気塊の形を一応，上記3・1節のごとく分類すると，図3では（A+a+B）形の連鎖形式，図4では（A+B）形の連鎖形式，図5では（A）形の単独形式によって蒸気塊の発生が続くことになるわけである．そして蒸気塊の発生に関し上記各形式が成立する領域は，およそのところ，図2の下方に示すように区分されるものであった．ただ（A+a+B）の領域において，ときとするとBの発生を欠き，（A+a）の連鎖形式になったり，あるいは時間の経過中に（A+a+B）と（A+a）が，ある期間をおいて発生を交替して行くものがあった．また（A+a+B）の領域と，（A+B）の領域の接続境界付近では，時間の経過中に（A+a+B）と（A+B）が，ある期間をおいて交替しながら生ずるものがあった．

なお，ここで次のことがらを付記しておくべきだと思われる．すなわち上記の分類で，同じA形蒸気塊と称しても，たとえば図3と図5のそれが同じ力学的状況にあると考えているのではない．また図2の下方に示した蒸気塊の連鎖形式の変化は，熱流束 q の大小だけに支配されて生ずるものではない．たとえば核沸騰域で $q<6\times10^5$ kcal/m^{2}h では，蒸気塊の発生が無秩序的になり，もっと低熱流束では発生しなくなるのに対し，遷移沸騰や膜沸騰では，$q<6\times10^5$ kcal/m^{2}h の熱流束でも蒸気塊の発生が秩序的である．

3・3 蒸気塊の切断周期　前述のごとく図2において加熱面過熱度 T_w-T_s の上昇に伴い，核沸騰域から遷移沸騰域を経て膜沸騰域まで移行する過程において，蒸気塊の発生形式が一般に（A+a+B）→（A+B）→（A）と変化して行く．そして図6は，高速写真フィルム（図2の各実験点で撮影）の分析観測から直ちに得られる結果の一例で，時間の経過中に順次あらわれて来る蒸気塊の順番を横軸にとり，蒸気塊の形式（● 印がA形，× 印がa形，△ 印がB形）および各蒸気塊の持続周期（その蒸気塊が加熱面上に生長を開始してから離脱するまでの時間）を縦軸に示している．なお図6のうち，上図は核沸騰域（A+a+B）形，中図は遷移沸騰域だが膜沸騰に近い領域で（A+B）形，下図は膜沸騰域（A）形である．膜沸騰域では，きわめて規則正しい振動的な蒸気塊の発生がみられる．

さて図6のような結果から，各実験点において，そこで生ずるA形，a形，B形の各蒸気塊の持続周期 τ_A, τ_a, τ_B（それぞれ 20～40個の同形蒸気塊の持続周期の平均値で定める）を求めることができる．そして，その結果を図7に示すが，図7の各データは横軸の T_w-T_s を介して図2の各実験点と対応している．

図7によれば，核沸騰域，遷移沸騰域において，τ_A, τ_a, τ_B は過熱度 T_w-T_s に独立にほぼ一定（図7に記入した水平線はそれぞれ $\tau_A=47$ ms, $\tau_a=19$ ms, $\tau_B=33$ ms であり，なお膜沸騰域の水平線は $\tau_A=55.8$ ms）とみてよいようである．ただし，B形蒸気塊の持続周期 τ_B については，ばらつきがやや大きい傾向



こま 1～5：時間間隔＝14.0 ms

図 5　蒸気塊の挙動（図2の実験点 3）

がみられる．なお3·1節において，B形蒸気塊はA形，a形両蒸気塊の中間的な性格と述べたが，図7においてたまたま $\tau_B=(\tau_A+\tau_a)/2$ の関係があることを付記しておく．

なお図2の各実験点で現れる蒸気塊について，上記のようなA，a，B形の区別を全くせずに，全体の平均持続周期 τ_m（たとえば図6で蒸気塊の形の区別を考えずに平均周期を計算）を求めてみると，図8の結果を得る．そして同図に記入した曲線にみられるように，遷移沸騰域で τ_m が加熱面過熱度 T_w-T_s と共に上昇するといった外見的傾向が観察される．しかし，その背景には図7の内容が存在していることを忘れてはならない．すなわち図7の結果から τ_m の値は次のごとく推定できる．

（A+a+B）領域：

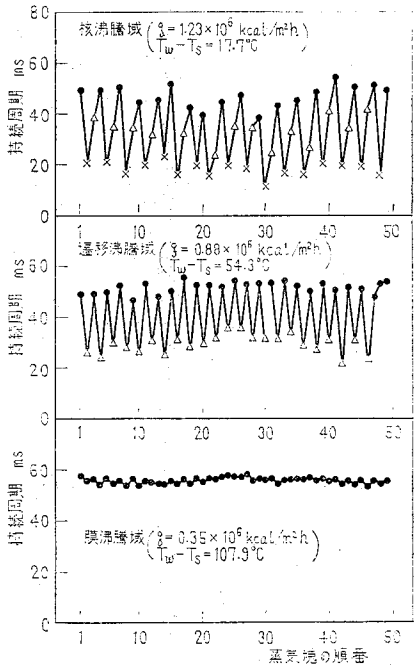
$\tau_m=(\tau_A+\tau_a+\tau_B)/3=(47+19+33)/3=33$ ms

（A+B）領域：

$\tau_m=(\tau_A+\tau_B)/2=(47+33)/2=40$ ms

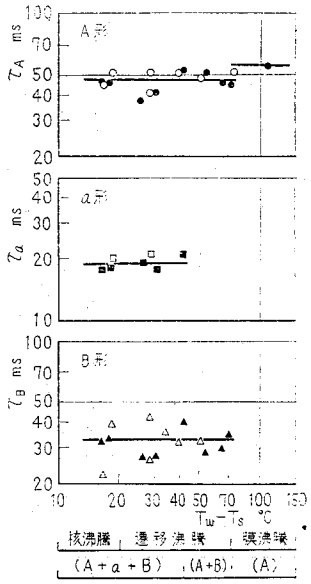
そして図8に記入した3水平線のうち，左端および中間のものは上記の τ_m の値を示し，また右端の水平線は図7の膜沸騰域の $\tau_A=55.8$ ms を τ_m にとったものである．当然ながら図8で3水平線が実験データとよく合っているのが見られる．

3·4 A形蒸気塊の体積生長特性 A形蒸気塊は，a形，B形の蒸気塊に比べると，先行蒸気塊の影響から比較的に独立していて，蒸気塊の挙動状況の分析観測がおこないやすく，かつ精度もよい．そこで加熱面上で生長するA形蒸気塊について，高速フィルムに撮影された蒸気塊をフィルム解析装置で蒸気塊実寸法の3～5倍に拡大した上で，図9に示すごとく加熱面からの高さ $z\sim z+dz$ の間の部分の直径 D を測定，次式により区分求積法で蒸気塊の瞬間体積 V を求めた．



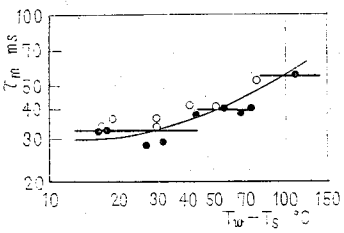
●：A形蒸気塊，×：a形蒸気塊，△：B形蒸気塊

図 6 蒸気塊の継続発生状況



○□△は図2の破曲線，●■▲は図2の実曲線に対応するデータ

図 7 蒸気塊の持続周期（実測値）



○は図2の破曲線，●は図2の実曲線に対応するデータ

図 8 蒸気塊の平均持続周期

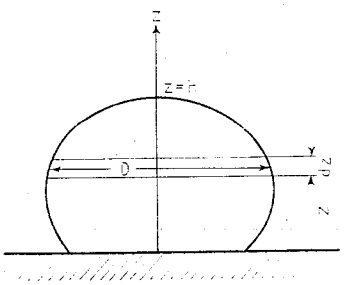


図 9 蒸気塊の撮影像

$$V=\frac{1}{a^3}\frac{\pi}{4}\int_0^H D^2dz \quad \cdots\cdots(1)$$

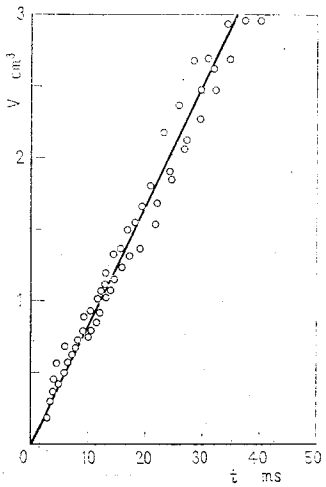
ここに a：蒸気塊の像の寸法倍率，H：蒸気塊の頂点の高さである．

さて，このようにして蒸気塊体積Vの時間変化を求めることができるが，その変化特性は沸騰領域によって次のようになる．まず核沸騰域では，図 10 に示すように時間 t と共に一定速度で増加する．図 10 には 3 個の蒸気塊の観測例を代表に掲げてあるが，他の観測値もこのばらつきの範囲内におさまる．これは，蒸気塊の底部の加熱面上に存在する核沸騰液層（加熱面に接し激しい核沸騰を続けている）から，絶えず一定速度で蒸気の供給がおこなわれるからであり，なお図 10 の斜線は，この実験における実測熱流束 q から次式によって求めた蒸気塊体積Vの時間変化であって，高速フィルムによる測定データとよく一致する．

$$V=G_1t, \quad \text{ここに} \quad G_1=qA/(\rho_V L)\cdots\cdots(2)$$

ここに A：加熱面面積，ρ_V：蒸気密度，L：蒸発潜熱である．

しかし，バーンアウト点を越えて遷移沸騰域へ入ると，図 11 において加熱面過熱度 $T_w-T_s=25.9°$, 30.2°, 41.9°C の場合の各データ（記号 ○, △, □ であらわす）にみられる特性となり，蒸気塊体積Vの増加が，蒸気塊生長途中でいったん停滞し，しばらく後にVの増加が再開されるといった形のものとなる．この蒸気塊体積Vの増加の一時停滞という特徴は，すでに著者ら[2]が報告しているところのもので，蒸気塊底部の核沸騰液層の消耗のため加熱面が乾燥することに関係すると考えられている．またVの増加が一時停滞後，再開されるのは，蒸気塊に覆われている乾燥加熱面上へ周辺から液体が侵入しつつ核沸騰蒸発を生ずるものと推測されている．なお，上記の体積生長特性がみられる沸騰領域は，少なくとも本報の実験では，加熱面過熱度 T_w-T_s がほぼ 40°C までの領域，すなわち蒸気塊の連鎖形式が（A+a+B）の領域はほぼ一致するという結果になっている．



核沸騰域，$q=1.19\times10^6$ kcal/m²h

図 10　蒸気塊体積Vの時間変化

さて，さらに加熱面過熱度 T_w-T_s が高くなり，（A+B）の領域に入ると，図 11 の $T_w-T_s=54.3°$, 65.5°C の場合のデータ（記号 ●, ▲ であらわす）にみられるごとく，体積Vの時間変化状況は比較的に単調なものとなる．ただし，蒸気塊の生長期間の前半と後半とで，生長特性にやや相違がみられる．なお加熱面過熱度 $T_w-T_s=70.5$°C の場合のデータ（記号■であらわす）に至ると，非常に単調な性格，つまり生長特性がほぼ一様という特性となる．ともあれ，この（A+B）の領域は，膜沸騰への遷移性格の顕著な領域ということができるであろう．

4.　蒸気塊の挙動に関する理論的考察

前章に述べたような蒸気塊の生長，切断の挙動に関し，これを支配している主要な力学的状況について，多少，理論的な面から検討を試みてみることにしよう．この際，A形蒸気塊が比較的に単純な状況にあるので，以下これについて力学的状況の検討をおこなうことにする．なおa形，B形の蒸気塊は，先行蒸気塊の浮上運動に伴う誘起流れ（伴流）の影響を受け，その挙動状況がA形蒸気塊より複雑になるが，それだけのことで，特に新奇な力学的条件が加わっているようには考えられない．

4·1　蒸気塊の挙動解析の基礎　さきに甲藤・菊池[6]は，大気圧の飽和水内に保持した水平円形加熱面（標準直径 14.5 mm）上の高熱流束核沸騰において発

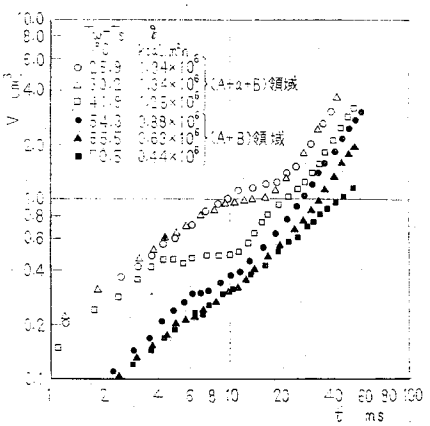


図 11　蒸気塊体積Vの時間変化（遷移沸騰域）

生する蒸気塊について，加熱面上に作用する蒸気塊内蒸気圧力 p_V と，加熱面の水位における周辺水圧 p_L との差 p_V-p_L（時間平均値）の測定と解析的検討をおこなっている．そしてそれによれば，加熱面から自由水面までの高さが 100mm 以上の場合，上記の圧力差 p_V-p_L は -1.5～-2.0 mmAq 程度の非常に小さい値（熱流束にほとんど独立）であり，その結果彼らは次の結論を記している．蒸気塊の生長は，蒸気塊が液中を浮上しつつ，加熱面からの蒸気供給を受けて自然に体積を増加させて行くという性格を強く持つ．これを換言すれば，蒸気塊の生長挙動にかかわる主要な力は浮力（近似的に蒸気塊周囲の静水圧で定められるところの）であるということに他ならない．

一方 Davidson ら[7][8] は，水平オリフィスないし垂直管出口孔を通して気体を液体内に流出させるようにしたときに発生する気ほうに対し，少なくとも生成気ほうの体積の理論的推定などについて，下記のような形式の簡単な解析が有効であることを示している．

いま，簡単のため図 12 のごとく液内に固定された点Pから供給される気体が球形気ほうを形成するとする．そして，この気ほうは供給気体をうけて体積増加（気ほうの瞬間半径 r が時間と共に増加）をしつつ，かつ気ほうは上昇運動（気ほう中心Oの点Pからの高さ s が時間と共に増加）をするものとする．ここに気ほうの上昇運動については，それに関与する流体の慣性力が気ほうの浮力につりあうとして，次の運動方程式が与えられる．

$$\frac{d}{dt}\left\{(\xi\rho_L+\rho_V)V\frac{ds}{dt}\right\}=(\rho_L-\rho_V)Vg \quad \cdots\cdots(3)$$

ここに t ：時間，ξ ：気ほう運動に随伴する液体容積（気ほうの単位体積あたり），ρ_L ：液体密度，ρ_V ：気体密度，V ：気ほうの瞬間体積，g ：重力加速度である．お上記の ξ については，流体力学で流体内の球の運動の解析から周知のごとく次式で与えられる．

$$\xi=\frac{1}{2}+\frac{3}{16}\frac{r^3}{b^3} \quad \cdots\cdots(4)$$

ここに r は球の半径，b は球中心と壁面（球の運動方向に垂直）の間の距離である．球が壁面から十分に離れていれば，$b=\infty$ により $\xi=1/2$ であるが，本報の問題では球（蒸気塊）が壁面（加熱面）上にあるといった状況であるから，$b=r$ として $\xi=11/16$ の程度である．

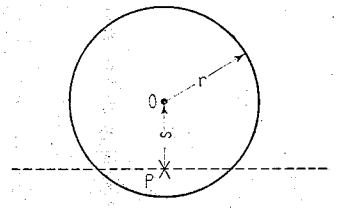


図 12　球形気ほうの体積増加と上昇運動

さて，一例として水平オリフィスを通し液体内へ気体が一定速度で供給される系を考えると，このとき式 (3) に含まれる V は

$$V=G_1t, \quad \text{ここに } G_1 \text{ は定数} \quad \cdots\cdots(5)$$

従って式 (3) は直ちに積分され，2個の積分定数は初期条件〔$t=0$ において $Vds/dt=0$（流体の持つ運動量零）および $s=0$〕から定まり，気ほう中心の高さ s の時間変化 $s(t)$ が確定する．一方，式 (5) と $V\equiv(4/3)\pi r^3$ から気ほう半径 r の時間変化 $r(t)$ も確定している．そして，図 12 にみるように，普通，気ほうの生長初期は $s(t)<r(t)$ であるが，やがて上昇運動が加速されて行く結果，次式の状況に達する．

$$s(t)=r(t) \quad \cdots\cdots(6)$$

そして式 (6) を満足する時刻 $t=\tau_d$ に，気ほうはオリフィスから離脱するものとする．この離脱条件の物理的意味は，「次第に加速される気ほうの上昇運動」が，「気体供給に伴う気ほうの寸法増加」に打ち勝つというもので，ここでの離脱は気液の流動状況だけに従い自然に生ずる流体力学的形式のものである．核沸騰の一次気ほうの離脱に関する Fritz の式のごとく，静力学的な力の平衡による形式とはまったく異なる．

なお，式 (3) に含まれる V が，一般に $V=G_mt^m$（ここに G_m, m はそれぞれ定数）のとき，$0<m<6$ ならば，$0\leqq t$ で $s(t)<r(t)$ の状況が成立し，そして式 (6) を満足する時刻 $t=\tau_d$ が存在する．限界 $m=6$ では

$$\frac{1}{14}\frac{g(\rho_L-\rho_V)}{\xi\rho_L+\rho_V}\leqq\left(\frac{3}{4\pi}G_m\right)^{1/3}$$

$$\text{または} \quad \frac{1}{14}\frac{g(\rho_L-\rho_V)}{\xi\rho_L+\rho_V}>\left(\frac{3}{4\pi}G_m\right)^{1/3}$$

により，前者では $0\leqq t<\infty$ で $s(t)<r(t)$，つまり気ほうはいつまでも離脱せずに生長を続け，後者では $0\leqq t<\infty$ で $s(t)>r(t)$，つまり気ほうの寸法増加作用が弱すぎて気ほうの形成ができないことになる．

ともあれ，V に関し式 (5) の状況の場合，上記の方法により，気ほうの持続周期 τ_d，および離脱気ほう体積 $V_d=G_1\tau_d$〔式 (5) で $t=\tau_d$ とおく〕は次のごとく求められる．

$$\tau_d=\left(\frac{3}{4\pi}\right)^{1/5}\left[\frac{4(\xi\rho_L+\rho_V)}{g(\rho_L-\rho_V)}\right]^{3/5}G_1^{1/5} \quad \cdots\cdots(7)$$

$$V_d=\left(\frac{3}{4\pi}\right)^{1/5}\left[\frac{4(\xi\rho_L+\rho_V)}{g(\rho_L-\rho_V)}\right]^{3/5}G_1^{6/5} \quad \cdots\cdots(8)$$

そして図 13 は，式 (8) の結果を，一定速度気体流出のいろいろな実験[7]～[11]（主として空気と水によるも

ので，水平オリフィスや垂直円管出口孔など気体流出孔の直径は 0.14 mm～25.4 mm の範囲におよぶ）の結果と比較したものである．広い実験範囲（0.037 cm³ $<V_d<$ 1 300 cm³，これに対応する離脱気ほう直径は 4.1 mm $<D_d<$ 135 mm）にわたって，解析と実験は少なくとも近似的に合致し，かつ $\xi=11/16$ ととればだいたい十分（なお図 13 にみるごとく $\xi=1/2$ と $\xi=11/16$ の相違は余り大きくない）であることがわかる．

図 13 に掲げる実験では，もちろん気ほう運動に対し，オリフィス縁や円管口周辺に沿う表面張力も作用するはずだが，G_1 が十分に大で，浮力の効果が大きいときは，これに比べ表面張力の作用は無視できることを図 13 は示している．また式 (3) には，いわゆる流体抵抗力が含まれていないが，現在扱っているのは，図 3 のこま 2，図 4 のこま 2 のような状況の流体運動であり，かつ気ほう（その界面には固体球のようなせん断力は生じない）にかかわるものであるから，通常の液体では妥当なことと考えられる．なお式 (7)，(8) を利用すると，高熱流束核沸騰域で生ずる蒸気塊［ここで式 (2) の状況は式 (5) と同じ］の大きさと発生周期に対し，付録 1，2 のような結果を導くことも可能である．

ともあれ，以上のことがらから判断して，式 (3)，(6) による解析は，かなり信頼性のおけるものと考えてよいであろう．

4・2　A 形蒸気塊の持続周期 τ_A の解析　本報の沸騰実験に現れる A 形蒸気塊の瞬間体積 V の時間変化は，すでに図 10，11 に例示してある．そして，これらの V の時間変化は，図 14 の 3 種類の形式のどれかによって近似的に表現することができる．

さて，まず図 14(c) について考えるに，これは $0<t<t_c$ の期間 $V=G_n t^n$，$t_c<t<t_l$ の期間 $V=$一定，$t_l<t$ において $V=G_m t^m$ の形の体積変化をする．この V の時間変化に応じ，式 (3) を積分するに際し，$t=t_c$ および $t=t_l$ の接続時点で，蒸気塊の中心位置 s，および上昇速度 ds/dt の値を連続させればよい．その結果，$t_l<t$ における $s(t)$ は次のごとく求められる．

$$\left.\begin{aligned} s(t)&=\left\{\frac{1}{2(1+m)}t^2+\frac{m}{1+m}t_l^2\frac{[(t/t_l)^{1-m}-(1+m)/2]}{1-m}\right\}\\ &\quad-\frac{n}{1+n}t_c t_l\frac{[(t/t_l)^{1-m}-m]}{1-m}+\frac{n}{2(1+n)}t_c^2\Big\}\frac{g(\rho_L-\rho_V)}{\xi\rho_L+\rho_V}\\ &\qquad(m\neq 1\ \text{の場合})\\ s(t)&=\left\{\frac{1}{4}t^2+\frac{1}{2}t_l^2\left(\log\frac{t}{t_l}+\frac{1}{2}\right)\right.\\ &\quad\left.-\frac{n}{1+n}t_c t_l\left(\log\frac{t}{t_l}+1\right)+\frac{n}{2(1+n)}t_c^2\right\}\frac{g(\rho_L-\rho_V)}{\xi\rho_L+\rho_V}\\ &\qquad(m=1\ \text{の場合}) \end{aligned}\right\}\quad\cdots\cdots\cdots(9)$$

一方，$t_l<t$ における蒸気塊の半径 $r(t)$ については，$V=G_m t^m$ および $V\equiv(4/3)\pi r^3$ から

$$r(t)=\left(\frac{3}{4\pi}\right)^{1/3}G_m^{1/3}t^{m/3}\quad\cdots\cdots\cdots(10)$$

そして，式 (9)，(10) に対し，式 (6) の条件によって蒸気塊の持続周期 τ_A が求まる．

なお，図 14(b) の場合は，式 (9) で $t_c=t_l$ とすればよく，さらに図 14(a) の場合は $t_c=0$ とすればよい．

さて，以上のようにして，図 10，11 に例示される A 形蒸気塊の体積変化を図 14 の形式に近似表現し，

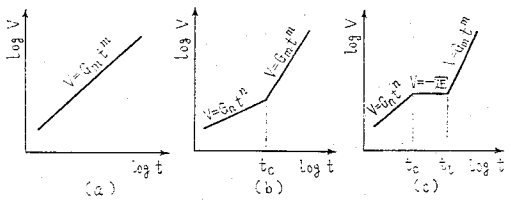
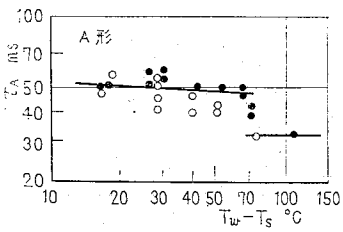
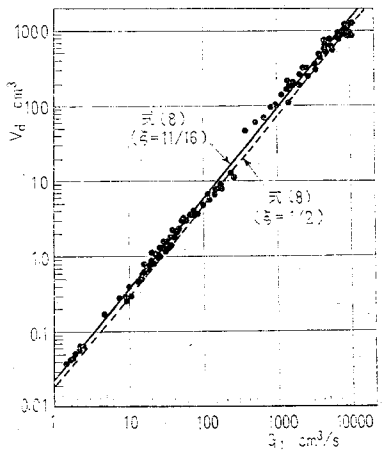


図 14　蒸気塊体積の時間変化の代表形式



○は図 2 の破曲線，●は図 2 の実曲線に対応する解析結果

図 15　蒸気塊の持続周期（解析値）



実験結果については，文献 (8) の図 5 に，文献 (7) のデータの一部（$G_1=1.5\sim3.0$ cm³/s）を追加して作成したもの

図 13　離脱気ほう体積（一定速度気体流出）実験結果と式 (8) の比較

式(9)，(10)〔ここで $\xi=11/16$ を使用〕によって求めた持続周期 τ_A（解析値）は図15の○および●のような結果になる．そして，これに対応する τ_A の実測値は，すでに図7の最上図に示されているのであるが，この両図の比較から，次のようなことをいってもよいように思われる．

（i） まず核沸騰域（$T_w-T_s<20°C$）では，$V=G_1t$ の関係が成立しているが，ここで τ_A の解析値と実測値がほぼ一致している．

（ii） 次に，核沸騰から遷移沸騰に入るに従い，加熱面乾燥などに伴う蒸気発生状況に変化（図10，11参照）を生ずるが，図7，15の τ_A の値，および傾向ともにほぼ一致していることから考えて，蒸気塊の挙動を支配する力学的状況に大きな変化はなく，だいたい，式(3)が想定する状況に近いとみてよいのではないだろうか．ただし，この結論は1章に述べた本田・西川[3]の結論と相違しており，今度，さらに慎重な検討が望まれる．

（iii） 膜沸騰域になると，式(3)の想定する力学的状況から明らかに偏差を示し，膜沸騰独特の界面波動形式的な蒸気塊放出の状況（図6の最下図にみるごとく蒸気塊放出がきわめて規則的）になると考えられる．ただし，本報のような直径10 mm程度の小さな円形加熱面の上での膜沸騰（図5参照）は，いかなる伝熱と蒸気発生形式になるのか，著者らには不明な点が多いことを付記しておきたい．

5. 結　　論

（1） 核沸騰域から膜沸騰域にわたり，蒸気塊の生長，切断挙動を支配する力学的状況を検討する目的で，水平円形加熱面（10 mmϕ）上の大気圧，飽和水のプール沸騰について，高速カメラで蒸気塊挙動を観測した．蒸気塊は，その挙動の特徴から，A形（先行蒸気塊に影響されず独立的），a形（A形蒸気塊にすぐ接続する形で発生），B形（A，a両形の中間的性格）の3種類の形に大別された．

（2） 蒸気塊をA，a，B形にわけて，それぞれの蒸気塊持続周期（加熱面上に滞留する周期）τ_A，τ_a，τ_B を高速フィルムから計測した．核沸騰域から遷移沸騰域にわたって，$\tau_A=47$ ms，$\tau_a=19$ ms，$\tau_B=33$ ms 程度で，かつほぼ一定に保たれる．

（3） A形蒸気塊について，その生長時の気塊体積 V の時間変化 $V(t)$ を高速フィルムから計測した．$V(t)$ の特性は，核沸騰域と遷移沸騰域で異なるものとなり，かつ遷移沸騰域のなかでも，核沸騰に近い領域と膜沸騰に近い領域とでは，相当に性格の異なるものとなる．

（4） 加熱面上で蒸気塊は，その浮力が流体慣性力につりあう形で上昇運動をおこない，かつ，その上昇運動が蒸気塊の体積増加にまさった状況で加熱面から切断浮上するとして，前記(3)項で計測した $V(t)$ に対応すべき τ_A の解析値を求めた．そして，これを前記（2）項の τ_A の実測値と比較し，核沸騰域，遷移沸騰域を通して，蒸気塊の挙動を支配する力学的状況自体に大きな変化はないらしいという結果を得た．ただ膜沸騰域に入ると，力学的状況は明らかに変化する．

本研究について，川西　洋君から理論解析に貴重な教示を得たこと，また研究に協力された詫間幸義君に謝意を表する．

付録 1：核沸騰域の蒸気塊の f と D_d

Ivey[12] は，核沸騰における気ほう発生サイクル f と，離脱気ほう直径 D_d について従来の測定値の整理を試みている．そして彼は，それを3種類の領域に分類し，そのなかで流体力学的領域と称するものについて次の経験式を得ている．

$$fD_d^{1/2}=0.90\,g^{1/2} \quad \cdots\cdots(11)$$

この領域は，熱流束が高く，気ほう径も大きく，本報で扱うような蒸気塊の発生領域に対応していると考えられる．ところで，式(7)，(8)によれば

$$\left(f=\frac{1}{\tau_d},\quad V_d=\frac{1}{6}\pi D_d^3\right),\qquad \rho_V\ll\rho_L$$

のとき直ちに次式を得る．

$$fD_d^{1/2}=0.853\,g^{1/2} \quad \cdots\cdots(12)$$

これは上記 Ivey の求めた経験式(11)にきわめて近い結果である．

付録 2：核沸騰域の蒸気塊の fD_d 値

核沸騰では気ほうの fD_d 値がよく問題になるが，Ivey[12] は fD_d 値の測定値と，液体内の気ほうの自由浮上速度 u_b との比較を試み，流体力学的領域では fD_d 値が u_b に近い値をとると述べている（これが流体力学的領域の命名の理由）．ところで，Ivey が用いた u_b はRosenberg など[13]~[15]の実験結果であり，それは $D_dg/u_b^2=1.87$ とあらわすことができる．そして，この u_b に対して次の経験関係（Ivey の論文の図7で気ほう径がほぼ11 mm以上の領域から得る）が与えられる．

$$fD_d\fallingdotseq 1.2\,u_b \quad \cdots\cdots(13)$$

ところで，式(7)，(8)および $D_dg/u_b^2=1.87$ から，$\rho_V\ll\rho_L$ のとき直ちに次式を得る．

$$fD_d=1.17\,u_b \quad \cdots\cdots(14)$$

これも上記の経験関係式(13)にきわめて近い．

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討　論

〔質問〕　**一色尚次**（東京工業大学）

（1） 従来のいわゆる蒸気ほう（一次気ほう）ではなく，蒸気塊（二次気ほう）に注目されて，その力学的挙動を研究された点に敬意を表する．このさい「蒸気塊」なる名称を核，遷移，膜沸騰の領域全般にわたって使用されているが，この名称の定義と領域はいかに定めておられるか．もし「伝熱面と切り離されているもの」という条件が入るならば式（3）にて表面張力の項を用いないことの妥当性が生ずると思う．

（2） 図 14，C形のように一度成長停止期がある場合には，式（3）では右辺が優勢となり，直ちに離脱の契機となると思われるが，それが離脱しないのはなぜか．

またこのさい底面がドライとなるという説明では，このような時には気塊と底面との間に接続点が生じて，そのため急に表面張力による引張力が生じて停留するのではないか．

（3） 本論文の後半では主としてA形気ほう力学について述べていると思われるが，前半で示されたように，核→遷移，→膜沸騰と領域が変化するにつれて形式が〔A+a+B〕→〔A+B〕→〔A〕と変化する理由について深く論ぜられていないが，これにも底面における表面張力の影響の変化が強く作用しているからではないか．

この意味で上の質問（1）の蒸気塊の定義も難しくなってくると思う．

〔**回答**〕　（1） 本報では，高熱流束核沸騰域における蒸気塊を基本としつつ，遷移，膜沸騰にまで概念を拡張する形をとっているが，ここでの蒸気塊とは要するに，ある程度以上の広さを持つ加熱面上に連続分布，ないしそれに近い形で発生する蒸気の集合によってできる蒸気の塊の呼称である．そして質問者のいわれるような「伝熱面と切り離されているもの」という条件は含んでいないし，含めなければならぬ必要も別に存在しない．

式（3）に表面張力が関係しない理由については，すでに本文（300 ページ左欄 10～14 行め）に記してあるが，もう少し詳しく述べることにすれば次のごとくである．すなわち水平オリフィスや垂直管出口孔を通して気体を液体内へ流出させ，気ほうを発生させる実験において，二つの特性領域があることが従来からよく知られている（文献[7][10]など参照）．第1領域は，流量 G_1 が非常に小さい領域で慣性力の作用が微弱であり，ここでの気ほう挙動は流出孔周辺に沿って働く表面張力に強く支配される．これに対し流量 G_1 が大きい第2領域では表面張力の作用が無視できる（浮力と慣性力の作用が優位に立つ）．このことは，実際の気ほう離脱時の浮力と表面張力の大きさを比較してもわかるし，また事実，図 13 の実験データは，0.14～25.4 ϕ という広範囲の流出孔寸法にわたるものであるが，流出孔に沿う表面張力の作用による相違は現れていない．そして本報の実験（図 10，11 参照）では，蒸気発生速度が $G_1=20\sim100\ \mathrm{cm^3/s}$ 程度の範囲にある（図 13横軸の範囲に入る）ことを考えれば，遷移沸騰において加熱面上に乾燥部分があったとしても，蒸気塊の離脱に対し表面張力の作用が十分に無視できる状況にあると考えてよい．

（2） 式（3）の両辺は常につりあっているもので，ある瞬間から右辺が優勢になるというようなことは生じない．また離脱は式（3）ではなく，式（6）の条件で定まるものである．そして一般には，図 14（c）の場合，時刻 $t=t_e$ において「$V=$一定」の期間に入ったからといって，そこで直ちに離脱条件に達するというものではない．しばらく蒸気塊は，$s(t)<r(t)$ の状態，すなわち加熱面に底部を接触した状態での加速上昇運動を依然として持続する．そして時刻 $t=t_l$ に至っても，まだ $s(t_l)<r(t_l)$ の状態にあれば，さらに次の生長過程に入り〔式（9），（10）に従う〕，やがて式

(9) の $s(t)$ と式 (10) の $r(t)$ の間に，式 (6) の関係が成立するに及んで初めて蒸気塊は加熱面から離脱することになる．なお実際に実験において観察されている図 11 の V の時間変化特性（実測値）などで，図 14(c) のタイプに属するものについて，上記の理論による解析を試みてみると，$t_c<t<t_l$ の期間には，まだ式 (6) の成立には至らぬ条件におかれていることが示し得ることを記しておきたい．

（3） ここで同じA形といっても，本文 296 ページ右欄 3～5 行めおよび 296 ページ左欄下 14～13 行めに記したごとく，膜沸騰におけるものは，核，遷移沸騰のものとは力学的状況が非常に違うものであること，および膜沸騰では液体は加熱面に触れないから「質問者の言われる意味での表面張力，すなわち底面における表面張力」の作用は明らかに消滅，存在しなくなることに留意する要がある．また図 2, 7, 11 をみていただくと明らかなように，(A+a+B) の形式は，高熱流束核沸騰から，さらに遷移沸騰の相当な領域（そこでは加熱面上に乾燥部分の発生をみる）にまでわたって成立するものである．そして以上のことは，質問者の立場に立って考えてみるとき，むしろ表面張力の影響が重要なものではないことを示すものになってはいないだろうか．そのうえ，表面張力に関し前記（1）項に述べた事情あり，従って著者らとしては本文 (298 ページ右欄下 8～4 行) に述べたことがらでほぼ十分のように思う．また前記（1）項に記した蒸気塊の定義に対し，別に困難が加わるようなことも生じない．

〔質問〕 **本田博司**（九州大学生産科学研究所）

（1） 本論文では生長蒸気塊の挙動について，一般性のある解析がなされ，蒸気塊持続周期 τ_A の解析値と実験値とが比較されている．いま図 15（解析値）を図 2 の沸騰特性曲線と比較すると，○, ● の各実験について τ_A（解析値）と熱流束 q の変化傾向に一致が認められる．これは一般的に生長速度（したがって q）が減少すると解析値 τ_A も減少する性質を持つことを示しているものと思われる．また理論式 (9), (10)を見ても，指数 m が 6 より小である限り図 14 に示されている個々の蒸気塊体積変化形式について同様のことがいえるように思われる．他方図 7 によれば τ_A の実験値は T_w-T_s とともに多少増大（● のデータ），またはほぼ一定（○ のデータ）と見ることができる．この実験値と理論値との傾向的な差異はどのような原因によるものか．

（2） 図 13 の気体流出速度 G_1 と離脱気ほう体積 V_d の関係を文献 (8) によってみると，個々の気体流出孔形状のデータについては G_1 が大なる程実験と理論の一致が良く，G_1 が小さくなると実験値 V_d は理論値よりも大きくなる傾向があるようである．これは (1) に述べた本実験における τ_A の実験値と解析値の関係と同性質のもののように思われるが，もしそうであれば理論において気体流出速度以外の因子を考慮する必要のある領域があるのではないか．

〔回答〕 （1） 膜沸騰の近傍は特異ゆえに除き，$T_w-T_s<65.5°C$ の範囲について考えることにする．さて図 7 の τ_A の実験値に関しては，限られたデータ数，およびばらつきの存在の点からみて，大まかに τ_A はほぼ一定とみるのが妥当（図 7 に関する質問者の観察はやや行きすぎ）だと思う．他方，図 15 の τ_A の解析値において，○ のほうのデータが T_w-T_s の上昇と共に τ_A のわずかな減少傾向を示していることは事実である．しかしながら，図 15 について質問者が表明されているご意見は，必ずしも一般性を持ち得ない．なぜなら図 2 で $T_w-T_s=65.5°C$(実曲線と破曲線の交点で，○● 共通) における q の値は，$T_w-T_s\fallingdotseq20°C$ における q の値の約 1/2 の大きさに低下しているにかかわらず，図 15 で，● のほうのデータは，τ_A がほぼ一定という性格を示している．換言すれば，質問者は，τ_A に対する G_1 の影響〔近似的には式 (7) にみるごとく $\tau_A\propto G_1^{1/5}$ であり，G_1 の変化による τ_A の変化は非常に小〕だけを考えておられるところに問題があり，図 11 にみるような生長特性 $V=V(t)$ の変化もまた同程度の影響を τ_A に対し持ち得ることを見落としてはならない．

従って図 7, 15 の間に存在する差異といえば，○ のほうのデータに多少，その傾向があるというべきものであろうが，本報の実験および解析の精度範囲を越えて，その差異の原因にまで立ち入ろうとするのは科学的でないと思う．ただし，この差異の原因が表面張力によるものでないことは，かなり確かである．すなわち図 10, 11 にもみるごとく離脱時の蒸気塊体積は $V_d=3\,cm^3$ 程度の大きさであるが，大気圧の水の飽和状態で，この大きさの蒸気塊の浮力 $(\rho_L-\rho_V)V_dg$ は，2.87 g である．一方，仮に加熱面（直径 $d=10\,mm$）が全面乾燥し，接触角 90° で蒸気塊が離脱するとする時の表面張力の保持力（これは論理的な最大値で，実際の離脱状況下には少なくともその数分の 1 以下にはなるはず）$\sigma\pi d$ は，0.19 g と計算されるが，これでも前記の浮力の数%の大きさにしかすぎない．

（2） 感覚的にではなく，もっと科学的に文献 (8) 図 5 をみる要があると思う．まず，ここで質問者が「理論」と言っておられるものは，本報の図 13 の破線〔式(8), $\xi=1/2$〕のことである．そして Walters-

Davidson が，これを実験結果と比較したのは，もともと彼らの論文の主体が，無限に広い液体内での気ほう（はじめ静止）の初期運動の解析にかかわるものだったからということは明白である．なお彼らは，質問者の言われるような傾向について，なにも語っていない．事実，文献 (8) 図 5 で，そういう傾向をみせるのは，「○印」および「破線表示」のデータだけといってよい．

ところで他方，本報の図 13 の実線〔式 (8)，$\xi=11/16$〕を描いてみると，図 13 にみるごとく，これまた G_1 の広い範囲にわたって実験結果とよく合うと言って，なんら差し支えない．なお図 13 で，$G_1=1.5\sim3.0$ cm³/s の実験データは，文献 (7) 図 1 で Davidson-Schüler が，理論〔式 (8)，$\xi=11/16$〕とのよい一致を述べているものである．また，前に「破線表示」のデータと言ったものは，文献 (7) 図 2 に示されるデータ（$G_1=15\sim37$ cm³/s）に一致するもので，これは図 13 で G_1 の増加と共に実線のほうから破線のほうに移って来る性格を持っているが，ここで，Davidson-Schüler は，質問者とは正反対に，「G_1 が大なるほど実験と理論の一致が悪くなる」と考え，その原因として，「気ほうの連続浮上運動に伴う誘起流れ」ならびに「気ほうの変形」を推測的にあげている．

以上の研究現状からみて，図 13 の実線，破線のどちらを優位にするかの決定は難しく，しかも仮に一方に偏しても V_d にわずか 17% の偏差（現実の個々の蒸気塊の体積は図 10 にもみるごとく ±10% くらいの偏差あり）しか生じない．ということは，本報の研究のような場合，現状として図 13 に関し，本文 300 ページ左欄 4～9 行程度に考えてよく，またそれが妥当であり，また文献 (8) 図 5 に対する質問者のご意見は，それを支持する根拠をまだ欠くものと言わざるを得ないと思う．

〔質問〕　本田博司

前質問 (2) の意図が不明確であったので，再質問させていただく．

単一オリフィスから発生する気ほう体積に関しては文献(付1)の図 5.2 (8.236) に Davidson らを含めた 6 研究者のデータ（空気･水系，空気流量一定の場合）が離脱気ほう体積 V_{B0}（本論文の V_d）対オリフィスレイノルズ数 $R_{e0}=u_gD_0/\nu_g$（u_g：オリフィス通過空気速度，D_0：オリフィス直径，ν_g：空気の動粘性係数）の関係でまとめられている．これによると個々のオリフィスについて $R_{e0}<10\sim30$ では V_{B0} がほぼ一定値をとり，$R_{e0}>200$ では V_{B0} が R_{e0} にほぼ比例し，$30<R_{e0}<200$ はその遷移領域になっている．またこれらのデータを浮力･表面張力モデルで整理し直した同文献の図 5.5 から，$R_{e0}<10\sim30$ は浮力と表面張力が決定的な領域，$R_{e0}>200$ は表面張力がほとんど関与しない領域と判断され，R_{e0} は気ほうの力学的挙動を決定する重要な物理量であると考えられる．したがって Davidson らの理論には適用限界があり，また本論文の主題である沸騰気ほう塊の場合にも表面張力の寄与の有無についてはオリフィスレイノルズ数に相当する量によって検討する必要があるのではないか．

〔回答〕　質問者は，「気ほう生成に対しオリフィスレイノルズ数 R_{e0} が重要な支配物理量である」と考えておられる．しかしこれに対し，その根拠として引用された文献(付1)をも含めて，以下のごとき重大な批判を呈せざるを得ない．

まず全般的にいって，ここで問題にしているような状況下の気ほう生成現象に対しては，気体の粘性 μ_g がほとんど影響を持たず，また気体密度 ρ_g の影響は，普通は浮力を通し $\rho_l-\rho_g$ の形で関係するのが主体で，なお通常の条件 $\rho_g\ll\rho_l$ のときは無視できる．これらは気液の力学的状況から自明といってもよいほどのことがらである．それなのに，文献(付1) 5･1 節では，実にすべての実験データ表示が，オリフィスレイノルズ数 R_{e0}（ν_g を通し μ_g，ρ_g が μ_g/ρ_g の形で影響を持つところの因子）を横軸にとっておこなわれている．なお Ramakrishnan らの解析（その解析中どこにも μ_g や ρ_g が現れない）の結果を掲げた図 5･8 の横軸にさえ，わざわざオリフィスレイノルズ数 R_{e0} を取ってある．以上のことがらは要するに，「現象の物理的はあくの完全な忘失」を示すものと言わざるを得ない．

次に文献(付1)の図 5･5 に固有の問題に移る．ここで左方領域は，近似的に式 (5･6) であらわされるもので，従来から周知のことがらゆえ論ずる要はない．さて図 5･5 の右方領域は，近似的には 45° の斜線で表示でき，従って数式で記せば，

$$g(\rho_l-\rho_g)D_{B0}^3/(\sigma D_0)\fallingdotseq\text{const.}\ u_gD_0/\nu_g$$

ここで離脱気ほう体積 $V_{B0}=(\pi/6)D_{B0}^3$，および気体の体積流量 $G_1=(\pi/4)u_gD_0^2$ を使えば次式となる．

$$V_{B0}\fallingdotseq\text{const.}\frac{2}{3}\frac{1}{g(\rho_l-\rho_g)}\left(\frac{\sigma}{\nu_g}\right)G_1 \quad \cdots\cdots\cdots\cdots(\text{i})$$

ところで図 5･5 は，実は常温近傍の空気-水系のデータだけの整理ゆえ，式 (i) において ρ_l, σ, ν_g などの物性値が近似的に不変という条件下にあり，従って

(付1)　世古口，伝熱工学の進展，Vol.1，気液二相流 (1)，(昭 48)，236，養賢堂．

図5·5の右方領域は，V_{B0}≒const.G_1の関係と解してこそ正当なものである．そして本報の式(8)の場合は通常の$\rho_L \gg \rho_V$の状況下に気液の種類によらず一般的にV_d=const.$G_1^{6/5}$となり，前記のV_{B0}≒const.G_1は，これとほとんど同じ結果になっているのであり，またこれが図5·5で右方領域の整理が偶然できた理由である．ついでながら，図5·5の場合，水以外の液体（普通σの値が水と大きく異なる）のデータを目盛ってみると，図5·5に示される水の場合の整理結果からずれたところに現れる．図5·5が，σやν_gの変化をも含み，あたかも一般的に成立するかのような形の表示になっていることは，読者に対して重大な誤解を与えるものである．また質問者が再質問において，図5·5の右方領域は，「表面張力がほとんど関与しない領域と判断される」と言っておられるのは，図5·5（縦軸に表面張力σを含む）の結果を信奉される限りは矛盾した発言となる．

要するに以上の状況にあり，従って，質問者の言われる意味での「Davidsonらの理論には適用限界がある」とか，「表面張力の寄与の有無においてはオリフィスレイノルズ数によって検討する必要がある」というようなことを科学的に支持する根拠はまったく存在しない．

536.423:536.24

干渉板を加熱面に近づけた場合の核沸騰の実験的研究*

甲藤好郎**, 横谷定雄***

1. 序　論

核沸騰については多くの研究がおこなわれ，その伝熱機構もかなり明らかになって来ている．しかし高熱流束領域では特に，基本的に不分明な点が多く，いろいろの面から具体的実証を通し着実に検討のメスを加えて行くことが必要と思われる．

本報では，プール飽和核沸騰状態にある円形水平加熱面（直径 10 mm）の上方から同径の水平円板（干渉板と呼ぶ）を故意に近づけ，液体や蒸気の挙動に与える人工的な制約とか変容を通して，普通の核沸騰伝熱機構に照明をあてることを試みる．なお主目的ではないが，狭いすきま内での沸騰に生ずる各種現象に関し必然的に基礎的知識を提供することにもなろう．

そもそも飽和沸騰では，相変化に起因し大部分の流体はほぼ飽和温度に保たれ，加熱面の近傍のみに温度差があらわれる傾向を持つ．特に熱流束の増加とともに加熱面上の発ぽう点密度や発生蒸気量が増すと，ほとんど完全にこの傾向となる．したがって伝熱の立場からいうと，加熱面のごく近傍の流体挙動だけが重要となる．

しかも，この流体挙動に対する外部流体運動の影響はどうか．普通の対流伝熱では，壁面近傍の温度境界層内の流体挙動も系全域の流動で一般に規定されるから，結局は後者をぬきにして有用な議論をすることができない．しかし核沸騰では加熱面上に発ぽうというかなり自律的な流体挙動を生じ，またその影響が大きい．もちろん低熱流束範囲，すなわちほぼ孤立気ほう域ではまだ加熱面からの気ほうの離脱に対する浮力の影響（Fritz[1]），上昇気ほうに随伴する過熱液体の流れ（一色・玉木[2]，原[3]，Zuber[4]）など自然対流的要素が強い．けれども高熱流束になると，気ほう生長に伴う流体慣性（一色・玉木[2]，小竹[5]）や気ほう間相互干渉などのため，加熱面上に分布する気ほうの離脱も自発性をもち，したがって伝熱特性にも浮力や重力加速度に伴う流れがあまり影響しなくなるようである（Merte-Clark[6]，Usiskin-Siegel[7]，Costello-Tuthill[8]）．

したがって高熱流束のプール核沸騰熱伝達は少なくとも第一近似として，加熱面およびそのごく近傍の流体に内在する条件とそれに伴う流体挙動に原動力があり，それより外部の流体の挙動は二次的なものと考えてよいであろう（この見地から本報では意識的に干渉板の使用を企図した）．そしてこの場合，上述の自律的領域が加熱面からどのような範囲内にあるかは一つの重要な問題である．また高熱流束では加熱面近くに大きな合体気ほうを生じ，見かけ上沸騰の様相を非常に変えるが，これが伝熱に大きな影響を持つか否か．またこの合体気ほうが，加熱面上の蒸気発生とどんな機構で結びつくか．本報ではまず，こうした問題について検討を加えてみよう．直径 10 mm という比較的小さな加熱面，干渉板の使用は，上記の目的に対し滞留蒸気による悪影響の防止，ならびに合体気ほう形状の単純化（次章参照）などのためである．

次いで核沸騰熱伝達には，薄膜蒸発機構（micro-layer vaporization mechanism）による伝熱が大きいとの説がある．すなわち，加熱面上の気ほう底部にごく薄い液膜があり，その急速な蒸発によって多量の熱が気ほう内に吸収されるとするもので，Moore と Mesler[9] に始まり，特に鳥飼[10]は光学的方法による薄液膜存在の観察を基礎にこれを強く主張している．この説の当否は，高熱流束における核沸騰伝熱機構の重要な問題といえるが，加熱面へ干渉板をずっと近づけると気ほうが押しつぶされ，寸法の確定したすきま内を平たく生長する場合があり得る．したがって，かなり特殊な形式ではあるが，発生蒸気に排除されて液体が急速に後退するあとの加熱面上の現象が比較的単純な形でとらえやすい．本報ではこの現象の観測と検討についても記す．

以上要するに本報は，従来試みられなかった特殊な形式をとりつつ，複雑な高熱流束核沸騰の伝熱機構に対して現象の本質に接近する道をひらこうとするものである．以下，順を追って述べて行くことにしよう．

記　号

c_{pL}: 液体の比熱

* 昭和 40 年 5 月 22 日　第 2 回日本伝熱シンポジウムおよび昭和 40 年 12 月 2 日　第 746 回講演会において講演，原稿受付　昭和 41 年 1 月 25 日．
** 正員，東京大学工学部（東京都文京区本郷 7-3-1）．
*** 准員，東京大学工学部．

g： 重力加速度
h_{fg}： 蒸発潜熱
M： 有次元定数（$=1.58\times10^{-4}\ \mathrm{m}^{1/2}$）
q： 熱流束（単位面積，時間あたり熱流量）
$\bar{q}$： 薄膜蒸発に基づく熱流束
q_c： 臨界熱流束
R_{e_δ}： 気ほう境界前面の液流のレイノルズ数
r： 半径
r'： 乾燥領域の半径
r_0： 加熱面の半径（$=5$ mm）
s： 干渉板と加熱面のすきま間隔
s_f： 加熱面温度低下開始点の s
T_s： 飽和温度
T_w： 加熱面温度
t： 時間
t'： 半径 r' の乾燥領域の生成時間
v_0： 周縁開口における液体流出速度
v_s： 気ほう境界の進行速度
δ： 液膜の瞬間厚さ
δ_0： 液膜の初期厚さ
κ_L： 液体の温度伝導率（$=\lambda_L/c_{pL}\rho_L$）
λ_L： 液体の熱伝導率
ν_L： 液体の動粘性係数
ρ_L： 液体の密度
ρ_V： 蒸気の密度
σ： 表面張力

2. 実 験 装 置

使用した沸騰装置の主要部を図1に図式的に示す．加熱面は銅，直径 10 mm の水平面で，これを大気圧の純水プールの底に設置する．加熱面周縁と断熱用テフロン間に生ずる微細なすきまには適当な接着剤を満たし，これによって沸騰が直径 10 mm の加熱面内のみに生ずるようにした．

熱は，銅ブロック下部の円柱状部分に設けたみぞ内に電気加熱器をそう入，単相交流で加熱，熱伝導で加熱面へ供給した．銅ブロック上部の円すい状部分には，その中心軸上に5箇所，正確な位置を決めた点に熱電対を設置し，熱伝導計算によって加熱面通過熱流束を求め，また外そうによって加熱面温度を求めた．この計測方法については，沸騰装置に銅ブロックを実際に取りつけた状態で注意深い検定をおこない，十分な精度を持つことを確かめている．なお本装置では電気入力の約 14% が熱伝導で周囲へ失われるが，ほとんど銅ブロック下部の円柱状部分からの損失であり，計測にあまり問題ないことを付記する．

本実験に使用した加熱面の核沸騰特性については，普通の場合とほとんど同じであることを確認している．たとえば図2に示すように熱流束の増加とともに加熱面上の気ほうが独立している孤立気ほう域から隣

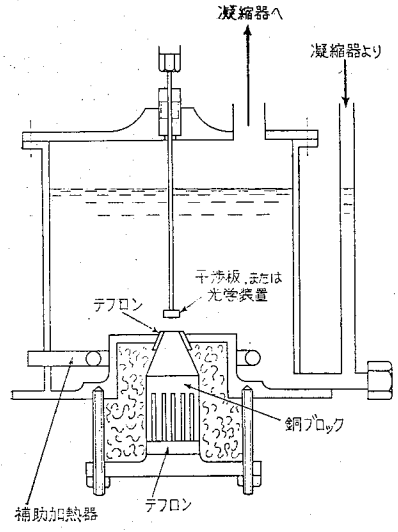


図1 沸 騰 装 置

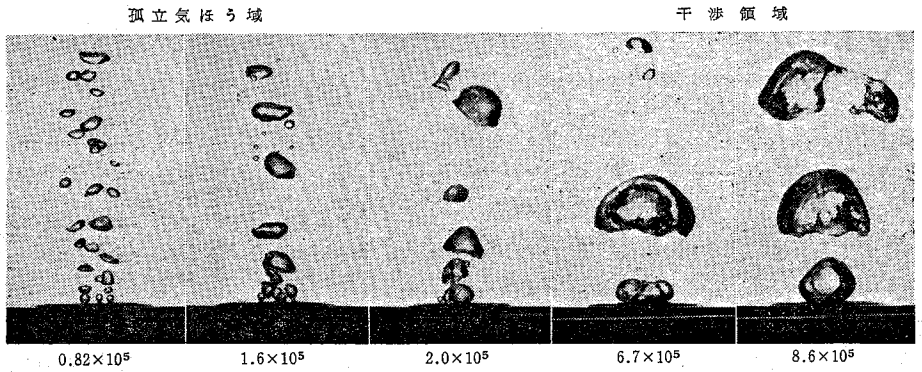


図2 加熱面の沸騰特性（気ほう状況）

接気ほうどうしが影響し合う干渉領域(合体気ほう域)への遷移はほぼ熱流束 $q=2\times10^5$ kcal/m^{2}h 前後で生じ，大気圧の水の飽和沸騰に関する従来の実験結果と一致する．また加熱面直径が 10 mm という寸法であること，および液体が上側にある水平気液界面の不安定理論（二次元理論による臨界波長は 100°C の水で 15.7 mm）から予想されるように，この加熱面上には干渉領域では常に図2にみられるごとき単一合体気ほう，すなわち比較的単純な形状の合体気ほうが次々に発生する．なお本報で以下，合体気ほうと称するのはだいたい，こうしたいわゆる二次合体ほうのことである．

使用した干渉板は図3に示すA形，B形の2種類で，ともに低熱伝導率のベークライト製，かつ外径は加熱面と同じ 10 mm の円板形である．ただしB形は底面に内径 8 mm，深さ 5 mm の円筒状くぼみをもつが，これは加熱面上に蒸気塊が定常的に保持されるとき熱伝達におよぼす影響をみるべく準備されたものである．

また干渉板を兼ねると同時に，加熱面上の流体挙動を観測するために用いた光学装置を図4に示す．A形装置は1個のガラス製干渉板と2個のガラス製プリズムおよび防水金属箱からなり，左方の窓からはいる光線によりガラス製干渉板と加熱面間の空間を照明するとともに，右方の窓を通して観測ができるようになっている．左右の窓部にそれぞれ設けた円筒は，下方から上昇して来る気ほうや蒸気が観測窓をさえぎるのを防ぐためのものである．また図には光線径路を単純な形で示してあるが，実際は光線が垂直でなく，ある傾きで加熱面に入射するようプリズム角度が設計されている．これは加熱面が薄い液膜でぬれた部分とかわいた部分にわけられるとき（後述の薄膜蒸発機構の検討の際など），両部分が良好なコントラストで区別できることを予備実験で確かめたからである．図4のB形のほうは，A形の底部にベークライト製薄肉中空円筒をとり付け円筒状くぼみ（図3のB形と同じ）を作っただけの相違である．

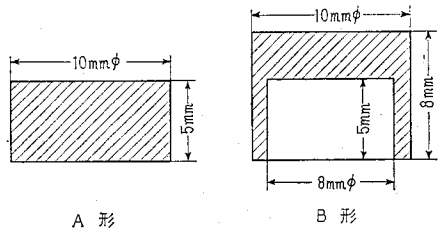


図3　干渉板（ベークライト製）

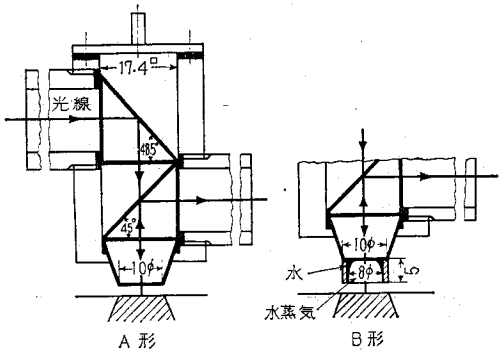


図4　光学装置

上述の干渉板や光学装置は，図1に示すように沸騰装置中心の位置ぎめ支持棒下端にとりつけられ，この棒はラック・ピニオンにより垂直に動かされる．棒の移動距離はダイヤルゲージで測定，これにより干渉板と加熱面の間のすきま間隔が精密に決定される．干渉板と加熱面の平行度については十分に留意してある．

3. 加熱面の温度変化の実験

実験はまずベークライト干渉板を用い飽和沸騰の条件下に，発ぽう点密度の高い孤立気ほう域をも含めた高熱流束領域にわたっておこなわれた．これは沸騰運動が伝熱を支配し，自然対流がもはや重要でない領域である．そして加熱面を通過する熱流束 q を一定に保持し，干渉板と加熱面のすきま間隔 s をいろいろに変えたときの加熱面温度 T_w を測定した．

単純なA形干渉板を使用した際に得られた結果の代表的なものを図5（縦軸 T_w-T_s の T_s は飽和温度で一定）に示す．図から明らかなように加熱面温度 T_w は一般に，干渉板が加熱面から比較的離れている間は一定に保たれている．しかしすきま間隔 s を減ずるに従い低下する領域があらわれ，その後上昇するのがみられる．ただし熱流束が増すと，それだけ早くバーンアウトが発生するため，加熱面温度が低下および上昇する二つの領域のひろがりは減じ，ついには消滅する．なお熱流束 q が高いとき，T_w の急上昇以前に s の減少とともに T_w が徐々に上昇して行く傾向がみら

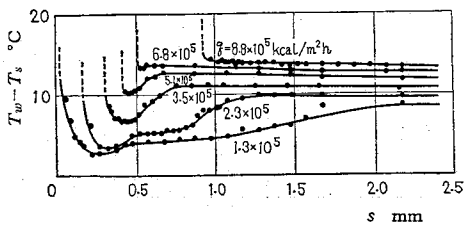


図5　すきま間隔 s による加熱面温度 T_w の変化（A形干渉板）

れるが，顕著ではなく近似的に T_w 一定とみてよいであろう．

さて上述の加熱面温度低下域の出現は常識に反するような結果だが，ここには後述するように興味ある現象が秘められているのである．それはともかく後述のごとく，この領域にはいると気ほうは自由に生長せず干渉板のため平たく押しつぶされる効果がではじめる．そしてこの加熱面温度低下が始まる所のすきま間隔を s_f とするとき，これは熱流束 q とある関係があり図6にその測定結果を黒点で示す．

なお同図に示す曲線は，Gaertner と Westwater[11] が本報と同じく大気圧の水中においた水平の銅製加熱面上で，干渉板なしの普通の核沸騰に対し離脱気ほう直径 D_b が熱流束 q によって変化する模様を測定した結果（水にニッケル塩をとかし，加熱面上へのめっきを気ほうがさまたげた領域から間接的に求めたもの）である．彼らは同時に発ぽう点密度も測定しているから，加熱面上の気ほう付着面積割合が計算できるが，最近鳥飼と山崎[12]がガラス面上に蒸着した酸化すずに通電して生じさせた大気圧水の沸騰をガラス裏面から光学的に観測した付着面積割合に比べると値が小さく，かつ熱流束 q による変化傾向も異なる．ただここでは，Gaertnerらの実験では時間的に持続性ある付着部分のみが特に強調されることを考え，定性的には鳥飼らと矛盾しないとしておこう．そしてこのような問題があるにしても図6において，筆者らと Gaertner らの互いに独立な実験結果が比較的良い相似性をみせることを指摘しておきたい．

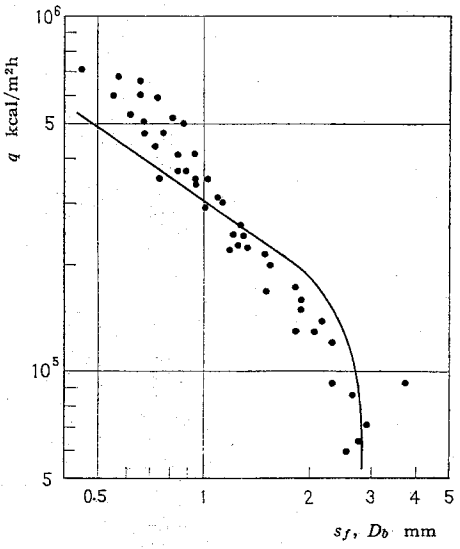


図6　限界すきま間隔 s_f と離脱気ほう直径 D_b の熱流束による変化

次に図7は，底面にくぼみを持つB形干渉板を使用した場合の代表的な結果である．同図で，干渉板の影響が少ないと考えられる右端近傍の T_w の値を図5のそれらと比較すると，同じ熱流束の場合に対し図7の値のほうがやや高くでている．しかしこれは，両実験でたまたま加熱面の仕上状態が一致しなかっただけのものである．そして現在の場合，加熱面上に固体干渉面でなく蒸気塊が存在するため，きわめて小さなすきま間隔 s まで，加熱面温度 T_w が普通の沸騰におけるとほとんど同じ値に保たれることがみられる．

4. 干渉板をもつ系のバーンアウト

図5，7の垂直破線位置はバーンアウトの発生を示し T_w の急激な上昇を生ずる．バーンアウト現象は本報では主題としていないが，簡単に触れておきたい．

いま加熱面上方にすきま間隔 s をへだてて干渉板があるとき，その周縁開口の下部から液体がすきま内部へ流入，発生蒸気は開口上部から流出，かつこれらの流れはともに連続一様なものと簡単に考えてみる．このときすきま内部と外界の圧力がひとしく，また全伝熱は加熱面上で蒸気生成に消費されると仮定すると，すきま内の質量，運動量，およびエネルギの保存から容易に，半径 r_0 の加熱面周縁開口部における液体蒸気間の相対速度が次のように与えられる．

$$\frac{q}{\rho_V h_{fg}}\left(1+\frac{\rho_V}{\rho_L}\right)\frac{r_0}{s} \qquad \cdots\cdots(1)$$

ここに q: 加熱面熱流束，ρ_V: 蒸気密度，ρ_L: 液体密度，h_{fg}: 蒸発潜熱である．

他方，周知のように，液体下側の水平気液界面は2流体が次の値より大きな相対速度を持つとき不安定になる．

$$\sqrt{2\left(1+\frac{\rho_V}{\rho_L}\right)}\left[\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}\right]^{1/4} \qquad \cdots\cdots(2)$$

ここに σ: 液体の表面張力，g: 重力加速度である．量 (1) と量 (2) を等置し，かつ q を q_c で置きかえ

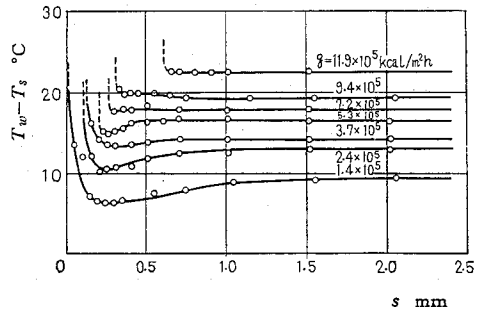


図7　すきま間隔 s による加熱面温度 T_w の変化（B形干渉板）

ると

$$\frac{q_c}{\rho_V h_{fg}}\Big/\left[\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}\right]^{1/4}=\sqrt{\frac{2}{1+(\rho_V/\rho_L)}}\frac{s}{r_0} \quad \cdots\cdots(3)$$

ここに q_c は，すきま間隔が s のときバーンアウトを生ずる臨界熱流束である．

この解析を実験結果と比較したのが図8で，斜めの破線が式(3)をあらわす．理論の仮定が非常に簡単であるにかかわらず，s/r_0 が小さいとき解析結果はとにかく実験とおよそは一致する．また s/r_0 が増すと，実験結果は理論からかたより漸近的に水平破線に近づいて行く．この水平破線は，普通沸騰のバーンアウトに対する Zuber の式（Rohsenow-Choi の書物[13]による）をあらわすもので，本実験では平渉板を加熱面から非常に遠ざけるときだいたいこの式に一致した．ともあれ，すきま間隔がわずか $s=1.4$ mm（これは $\sqrt{2/[1+(\rho_V/\rho_L)]}(s/r_0)=0.4$ に対応する）ですでに，臨界熱流束が普通沸騰のそれとあまり相違しない値になることに注目すべきである．これは普通沸騰のバーンアウト機構に，なにかを暗示するかも知れないが，

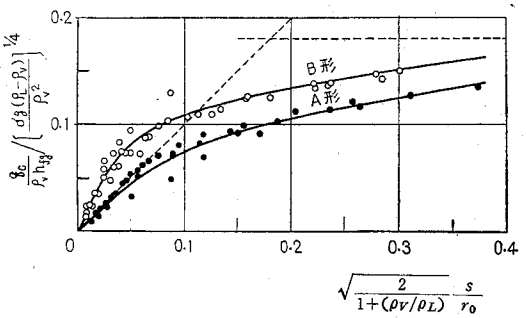


図8　すきま間隔 s による臨界熱流束 q_c の変化

いまはその指摘にとどめよう．

なお図8において，A形とB形の干渉板が，やや違った傾向を示す理由は明らかでない．

5. 流体挙動の光学的観察

干渉板と加熱面間のすきまにおける流体挙動は，図4に示す光学装置を用いて観察された．B形装置では図4に図示する形で，予想どおり底部くぼみに蒸気が保持されることが判明した．しかし，くぼみ上部のガラス面に多数の水滴が付着したりして明確な観測が困難であったため，ここにはA形装置による実験結果のみを記す．

すなわち加熱面温度変化に関する図5の実験結果に対応し，すきま内の流体挙動の代表的なようすを示せば図9のごとくである．これは同図 T_w-s 曲線に矢印で示す各条件下で，絶えず変動しつつある流体の二，三の瞬間状態である．これらの写真から，流体挙動に関し三つの相異なる領域があり，しかも3章で述べた T_w-s 曲線の特性にそれぞれ対応することがわかる．

まず加熱面温度 T_w が不変である第1の領域では，普通の核沸騰が熱伝達を支配し，加熱面上での小気ほうの発生こそとにかく現象を特色づけるものである．ここで高熱流束域では，干渉板が合体気ほうの中へ押しこまれて行くわけで，干渉板下面に沿う蒸気膜を通して合体気ほう下面の状況が観測されているのである．そして普通の沸騰でも，高熱流束では，加熱面上に定常的に発生する蒸気が上方で集合して，図2に示すような大きな合体気ほうを次々に生じていると判断される．ともあれ現在の場合，干渉板による気ほうの押しつぶしがないから，この領域を「非押しつぶし気

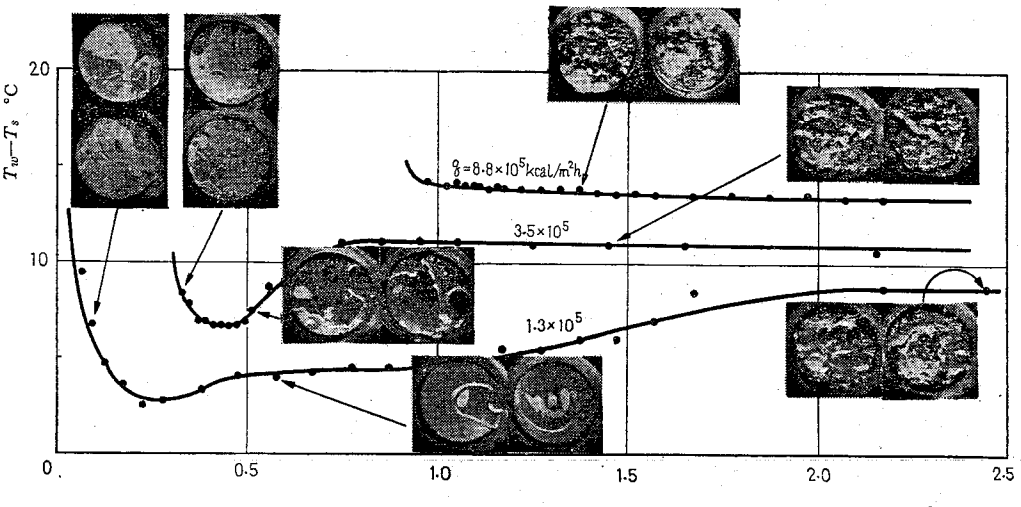


図9　すきまにおける流体の挙動

ほう域」と名付ける．

次に，すきま間隔 s の減少とともに T_w が低下する第2の領域での沸騰は，一種の核沸騰であるが，前領域に比べればはるかに数少ない気ほう（ただし小気ほうの合体によるものではない）がすきま内にひろくひろがり，最後に浸入して来る液体に押出される形になる．現在の場合のように干渉板がきわめて加熱面に近い時は，すきま容積が相当に小さく，したがって1個の気ほうが生長を開始すると急速に加熱面上にひろがり，その近傍で他の気ほうが発生するのをさまたげるものと考えられる．ただし当然ながら，熱流束の増加とともに共存気ほう数は増す傾向がある．この領域では干渉板によって気ほうが押しつぶされているから「押しつぶし気ほう域」と名付ける．興味あるのは，気ほうの生長時，蒸気にさらされる加熱面が薄い液膜でぬれており，しかも明らかにこの液膜からも蒸発が生じていることである．図9において本領域に属する各写真に白いひろがりをみせる小部分は，液膜がすでにかわいた箇所である．

最後に，すきま間隔 s の減少とともに T_w が上昇する第3の領域では，加熱面上の空間がきわめて薄く，気ほうの急速な生長がもはや許されぬため，いかなる形の核生成もみられない．そして干渉板，加熱面間のすきまの大部分は定常的に蒸気に占有され，かつその蒸気にさらされた加熱面はかわいているのである．液体は周縁開口からこの乾燥領域へ浸入，蒸気に面する狭い液面で蒸発するため，気液界面は比較的ゆるやかな前進と後退をくりかえす．まれに液体の浸入が広くかつ深く生じ，そのあとには多くの小液滴の孤島を残すが，図9にもそれらがみられる．この領域を「抑圧蒸発域」と呼ぶことにする．

上述の第2，第3の領域は常に存在するわけではなく，バーンアウトの発生によって制限されることに注意せねばならない．すなわちすきま間隔の減少に伴いバーンアウトが十分早く生ずれば，上記の2領域はみられなくなる．さらに熱流束が十分高ければ，第1領域をも含めた3領域が生じなくなることはいうまでもない．

6. 普通の核沸騰に対する考察

3章および5章の実験結果を総合すれば，干渉板のない普通の高熱流束プール飽和核沸騰に対し，次のような推論考察が導かれる．

（a） 図2に示すように，ある程度以上のひろさを持つ加熱面上に生ずる合体気ほうは，加熱面に付着しているようにみえるが，やはり加熱面上に定常的に生ずる小気ほうから供給される蒸気の単なる蓄積結果である．なお高速カメラを用い，この合体気ほうの生長を測定した著者らの未発表研究によると，その生長の全期間にわたり蒸気増加速度が常に一定で時間的変化はみられず，上記結論に適合する．

（b） 上記の合体気ほうは，少なくともそれが加熱面への液体の供給をさまたげたりしない限り，沸騰熱伝達にはほとんど影響しない．また人工的に大きな蒸気塊を加熱面近くに絶えず保持しても，加熱面上で自由な気ほう生長が許される限り，熱伝達特性はあまり変わらない．

（c） 図7の実験結果において，干渉板と加熱面の狭い周縁開口を通じておこなわれる蒸気除去および液体供給が，バーンアウトにはともかく，熱伝達にはあまり影響しないことも注目すべきである．そして前記(a),(b)の結論とあいまち，加熱面近傍の流体挙動の自律性，および熱伝達への強い影響を暗示する．

（d） Gaertner と Westwater[11] の実験には多少問題があるにしても，図6の結果からみてやはり，干渉領域では熱流束とともに図6のような傾向で離脱気ほう直径が小さくなると考えられる．もちろんこの結論の背後には，干渉板を用いたとき「非押しつぶし気ほう域」の限界すきま間隔が，干渉板のない時の離脱気ほう直径におよそ等しいという思想があるが，大まかな傾向としては妥当であるまいか．特に高熱流束では加熱面上の小気ほう群のすぐ上空に，密度も粘性も低い蒸気雲（合体気ほう）がひろがっているから，その内部に干渉板を入れても加熱面上の状況はあまり変わらないはずである．

（e） 加熱面から離脱気ほう直径程度の距離内の流体挙動は自律的であり，外部から強い影響を受けぬ限りあまり変わらない．しかも核沸騰熱伝達に重要な現象は，それがいかなるものであれ少なくとも，この領域内で生じていると考えられる．そしてもし前記（d）項の結論が正しいとすれば，図6より加熱面の近傍，特に高熱流束では加熱面上の非常に薄い領域のみが重要だということになる．

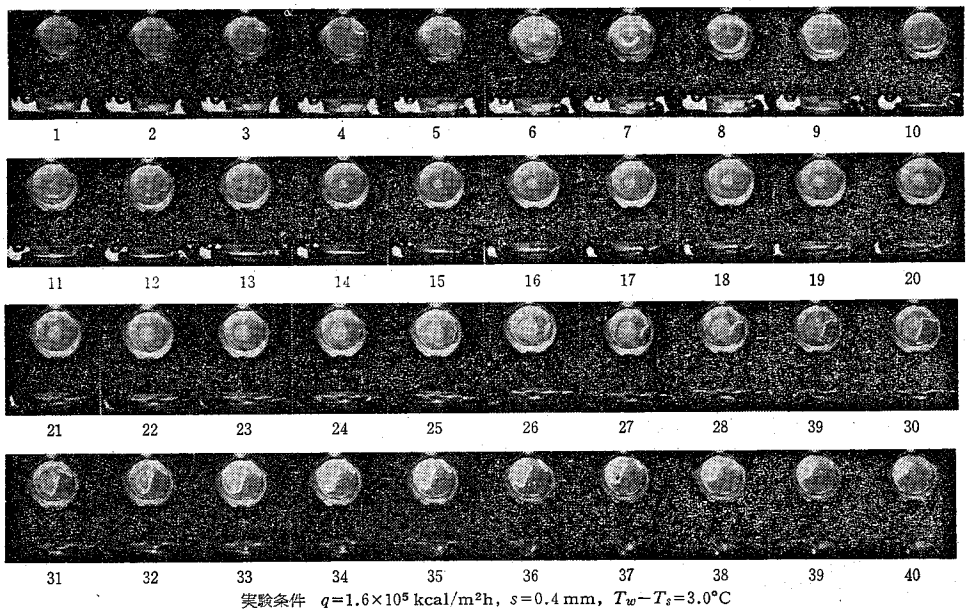
7. 平らな気ほうのひろがり

核沸騰における薄膜蒸発機構に関連し，押しつぶし気ほう域の平らな気ほうのひろがりは興味がある．なぜなら蒸気にさらされた加熱面が，5章で述べたように薄い液膜でぬれているからである．

図4のA形装置を右方から高速力カメラで写し，加熱面上の流体挙動をとらえた一例を図10（写真下部には干渉ガラス板が側面からも写る）に示す．これは熱

流束 $q=1.6\times10^5$ kcal/m²h, すきま間隔 $s=0.4$ mm, および加熱面過熱度 $T_w-T_s=3.0$°C の条件下に時間 1.53 ms ごとの状況を与えるものである. 図には省いてあるが長い休止時間の後, 加熱面中心からはじまる非常に急速な気ほうのひろがりが第6こまから第8こまにかけて現われる. 次いで蒸気は比較的長時間, 加熱面上のすきまを占有しているが, ついに第 24 こまに至り右側から液体の侵入が始まっている. 蒸気にさらされている加熱面は薄い液膜でぬれており, その液膜からの蒸発は白い円形乾燥域が加熱面上にひろがる（特に第 14 こま付近からはっきり見える）ことで確認される.

もちろん気ほうのひろがりは加熱面中心から始まるとは限らないが, この場合は幸いに1本の撮影フィルム中から図 10 のように解析に便利な形のものが少なからず得られ, その分析から気ほう半径 r の時間 t による変化 $r(t)$ に対し図 11 の結果を得た. 多数例のプロットにもかかわらずこれには驚くほどの規則性あり, $r<1.5$ mm の初期段階では $r(t)\propto t^{1.11}$, それ以後 t の指数は増し, ついには $r(t)\propto t^2$ となっている.



実験条件 $q=1.6\times10^5$ kcal/m²h, $s=0.4$ mm, $T_w-T_s=3.0$°C

図 10　流体状態の変化（時間 1.53 m s おき）

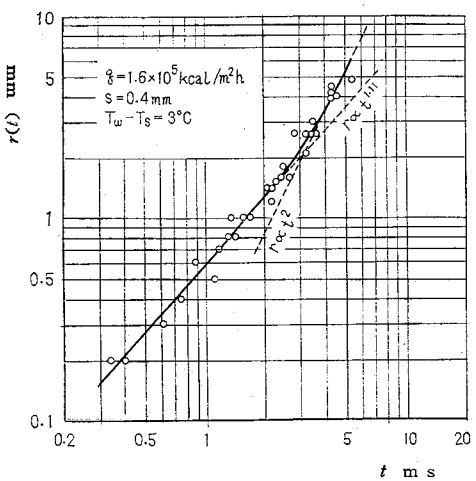


図 11　気ほう半径 r の時間変化

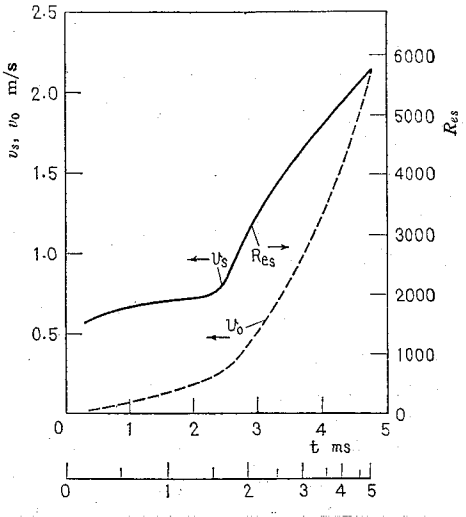


図 12　気ほう境界進行速度 v_s, 液体の周縁流出速度 v_0, 気ほう境界前面の液流のレイノルズ数 R_{es} の変化

とにかく図 11 の結果から気ほう境界の進行速度 $v_s \equiv dr/dt$, また半径 r_0 の加熱面周縁開口における液体の流出速度 $v_0 \equiv (r/r_0)dr/dt$ が計算できて図 12 に示す. 後者の v_0 は単に参考までに示したものであるが, その計算にあたり, 現在の実験条件では蒸気と液体の比容積が 1 600 倍も異なるから, 蒸発にかかわらず液体の体積変化はないとみなしてさしつかえない. また気ほう境界前面, すなわち気ほう境界に押されて流れる液体のレイノルズ数は, 干渉板と加熱面のすきま間隔 s および液体の動粘性係数 ν_L に対して $R_{es} \equiv 2sv_s/\nu_L$ (平行板流路の平均水力直径は $2s$ である) となり, これも図 12 に示す. なお気ほうのひろがりに伴う液体の加速や流動摩擦抵抗につりあうため, 気ほう内圧力は飽和圧力より高い. しかし解析的検討の結果, 本実験条件下では問題にする必要のない大きさであることを付記しておく.

加熱面上の乾燥領域のひろがりは, 図 10 からも明らかなように, 気ほうのひろがりより, はるかにゆっくりした現象である. これについても撮影フィルムから同様な統計的分析をした. そして, その結果は簡単で次の経験式表示ができる.

$$t = Cr, \quad C \equiv 1/270 \ \text{h/m} \quad (0 < r < r_0) \cdots\cdots (4)$$

ここに t (単位: h) は, 気ほう生長開始から半径 r (単位: m) の領域が乾燥するまでの時間である. 液膜は非常に薄く, かつ気ほうは発生後, きわめて迅速に加熱面上にひろがってしまうゆえ, 式 (4) は加熱面中心から半径 r の位置で生成された液膜が乾燥するまでの時間と近似的にみなしてもさしつかえない. 式 (4) は次章において, 液膜の厚さの分布の解析に用いられる.

8. 気ほうのひろがりに関する考察

周知のごとく, 普通の沸騰では生長しつつある気ほうの表面移動速度は時間とともに減少する (たとえば秋山[14]参照). ところが図 12 の場合, 気ほう境界の速度 v_s はひろがりの初期段階でほぼ一定であるのみならず, 途中で急に増し始め, 以後時間とともに急激に上昇して行くのである. したがってその原因について考えてみる必要がある.

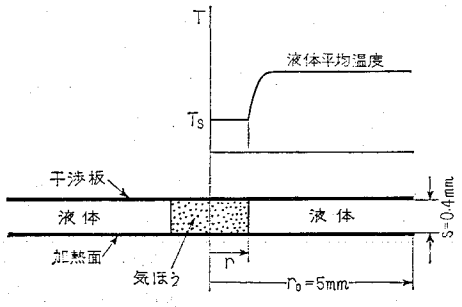


図 13 平らな気ほうのひろがりに関する模型

この場合, 種々の因子が関係していようが, 以下に記す三つのものが特に重要と思われる. 第 1 に, 簡単のため図 13 (見やすくするためすきま間隔を 2 倍に拡大して示す) の系を考え, すきま内の液体はいたるところ一様な初期過熱度 $T_m - T_s$ を持っていたと仮定してみよう. こうして, はじめ液体内にたくわえられ, 熱伝導で気液界面へ供給される過熱エネルギを消費しつつ界面に生ずる蒸発に対し, プロフィル法 (液体内温度分布は二次曲線で近似, 界面は飽和温度とする) で解析すると容易に次の気ほう境界移動速度が得られる.

$$v_s = \frac{2}{3}K^2\left(1+\sqrt{1+\frac{3}{2K}}\right)\frac{\kappa_L}{r},$$

ここに

$$K \equiv \frac{\rho_L}{\rho_V}\frac{c_{pL}(T_m - T_s)}{h_{fg}}$$

κ_L: 液体の温度伝導率, c_{pL}: 液体の比熱である.

プロフィル法は近似法であるが, 普通の沸騰の気ほう生長に適用しても従来の解析とオーダがよく一致するし, 下記のようにここではオーダの比較に使用されるだけだからこれで十分である. さて上式によれば v_s は気ほう半径 r の増加とともに減少してしまう. またかりにそれは問わぬとしても, 上式の与える v_s の大きさは, 気ほうひろがりの初期段階における実験結果に比べてあまりに小さすぎる. 考え得る最大初期過熱度 $T_m - T_s = 3°C$ をかりにとっても, たとえば $r = 0.5$ mm で v_s はわずか 0.037 m/s (実験結果は約 0.66 m/s). これを実験に合わせようとすると液体の平均過熱度 $T_m - T_s$ として, 約 13°C という途方もない値を考えねばならない. この間の事情から本報のような系では少なくとも, 気ほう境界近傍で蒸発のため冷やされつつ流れている液体へ加熱面からおこなわれる熱伝達が相当に重要であると思われる.

第 2 に, 気ほうひろがりの途中で v_s が急に増しはじめる原因は判然としないが, たまたま $R_{es} = 2\,000$ の近傍で生じている. したがってここではいちおう, 気ほう境界近傍の液体の流動が層流から乱流へ遷移し, したがって加熱面から液体への熱伝達が増すためとも考えられる. ただし非常な短時間に, こうした遷移が可能か否か, また平行平板間の放射状流れの臨界レイノルズ数 (平行流れの値 2 400 よりは低いと考えられるが) も確定しているわけでない問題もある. ともあれ次に述べる事情で, v_s は少なくともどこかで急に増

加せねばならぬ運命をもっていることだけは確かである．

すなわち第3の重要因子として，加熱面上の薄液膜からの蒸発の影響があり，これは気ほうひろがりの終段階で特に重要である．いま平らな円形気ほうの高さを s，半径を r とするとき，その底面と側円筒面との面積比は $r/2s$ となる．この面積比は現在の場合，気ほう半径 $r>0.8$ mm で1より大きくなり，$r=5$ mm では 6.25 の大きさにまで達する．したがって気ほう底面からの蒸発効果が顕著になる可能性があるが，液膜の厚さは加熱面中心からの距離 r による分布をもつゆえ，気ほう底面から一様な蒸発を生ずるわけではない．

いま液膜底面が温度 T_w の加熱面に接し，温度 T_s の液膜表面から準定常的に蒸発が生じている場合を考えてみよう．本研究のように $c_{pL}(T_w-T_s)/h_{fg}\ll 1$ のときは，周知のごとく液膜を通し直線温度分布が成り立つとしてさしつかえなく，この温度こう配による伝導熱量から液膜表面での瞬間蒸発速度が直ちに求められる．したがって時刻 t における液膜の瞬間厚さ δ も容易に求められ

$$\frac{\delta}{\delta_0}=\sqrt{1-2\frac{c_{pL}(T_w-T_s)}{h_{fg}}\frac{\kappa_L t}{\delta_0^2}} \quad \cdots\cdots(5)$$

ここに δ_0 は $t=0$ における δ をあらわす．上式で $\delta=0$ とおけば，初期厚さ δ_0 の液膜が乾燥してしまうまでの所要時間が次のように与えられる．

$$\frac{\kappa_L t}{\delta_0^2}=\frac{1}{2}\frac{1}{c_{pL}(T_w-T_s)/h_{fg}} \quad \cdots\cdots(6)$$

さて気ほうひろがりに際し創造される液膜は，その加熱面上のひろがりに比べ非常に薄い．また液膜内に準定常的な温度分布が成り立つまでの時間もきわめて短い．したがって (4)，(6) 両式から t を消去すれば液膜の初期厚さ δ_0 の分布が得られる．現実験条件における各数値を入れて

$$\delta_0=M\sqrt{r}, \quad M\equiv 1.58\times 10^{-4}\ \mathrm{m}^{1/2} \quad \cdots\cdots(7)$$

ここに δ_0 および加熱面中心からの距離 r の単位はともに m である．興味あるのは δ_0 が $\sqrt{r}$ に比例した分布を示すことであり，式 (7) から δ_0 の大きさを計算して示せば

$r=$	0.5	1.0	2.0	3.0	4.0	5.0 mm
$\delta_0=$	3.54	5.01	7.10	8.70	10.0	11.2 μ

ここで気ほうのひろがり速度の問題にもどろう．簡単のため，ひろがりの終段階近くでは，気ほう底面の液膜からの蒸発だけを考えることにする．この蒸発については式 (5) を導く際と同じ事情にあり，それから直ちに気ほう境界の進行速度 v_s が次のように得られる．

$$v_s=\frac{1}{2\pi r s}\int_0^r\frac{1}{\rho_V h_{fg}}\left(\lambda_L\frac{T_w-T_s}{\delta}\right)2\pi r dr \quad \cdots(8)$$

ここに λ_L: 液体の熱伝導率，δ: 液膜の瞬間厚さである．さて式 (6)，(7) を用いて検討すると，本実験条件下では気ほうのひろがり期間中の蒸発による液膜厚さの減少は，初期厚さに比べて無視できるくらい小さい（加熱面中心付近は除く）．なおかりに平均厚さわずか 0.25 μ の液膜が加熱面全面で蒸発したとすると，干渉板と加熱面のすきまはその蒸気だけで満たされる大きさであることも付記しておこう．したがって式 (7) の δ_0 を式 (8) の δ に代入しても近似的にさしつかえない．そして式 (8) の積分をおこなえば

$$v_s=\frac{2}{3}\left(\frac{\rho_L}{\rho_V}\right)\frac{c_{pL}(T_w-T_s)}{h_{fg}}\frac{\kappa_L\sqrt{r}}{Ms} \quad \cdots\cdots(9)$$

右辺へ現実験条件における数値を入れると

$$v_s=57\,000\sqrt{r} \quad \mathrm{m/h} \quad \cdots\cdots(10)$$

ここに気ほう半径 r の単位は m である．他方，気ほうひろがりの終段階近くの実験結果は図 11 より

$$v_s=107\,000\sqrt{r} \quad \mathrm{m/h} \quad \cdots\cdots(11)$$

式 (10)，(11) を比較するに，気ほうが加熱面上をひろがるにつれて，v_s が $\sqrt{r}$ に比例して増加するという点で解析結果は実験に一致する．ただ前者は後者の 53% 程度の大きさになっているが，解析では気ほう底面以外からの蒸発を無視したほか，特に加熱面過熱度として実測値 $T_w-T_s=3$°C を用いていることを忘れてはならない．実測では時間的平均温度が，しかも加熱面から離れた所の温度分布を利用して計測されるが，実際には加熱面近傍の温度に多少の変動があるはずである（銅ブロック上に加熱面が構成されているが）．そして気ほうの生長開始前の加熱面温度は，その平均温度よりは高くなっており，しかも気ほうのひろがり完了までわずか 4.7 ms の短期間くらいは，まだそれに近い温度に保たれよう．かりに $T_w-T_s=4$°C としても，すでに解析結果は実験の 71% の大きさとなる．いずれにせよ上例は，薄液膜からの蒸発をもととした解析が気ほうの挙動をよく説明し得る興味ある場合といえよう．

9. 薄膜蒸発による熱伝達

前々章に述べた実験では，わずか 3°C の加熱面平均過熱度で 1.6×10^5 kcal/m² h もの大きな熱流束を得るという驚くべき結果である．そしてこれまた加熱面上の薄液膜からの蒸発に帰せらるべきものである．事実，図 10 において気ほうの加熱面上ひろがりが第6～8こまの短時間に完了した後も，干渉板と加熱面の

周縁開口から蒸気が長期間にわたって流出を続けているのがみられる．

さて式 (7) であらわされる初期厚さ分布を持つ液膜が蒸発して，加熱面中心から半径 r' 以内の領域が乾燥するとき，それまでの経過時間 t'，および残存液膜の厚さの分布 δ' は式 (5), (6), (7) から容易に次のように求められる．

$$t'=\frac{1}{c_{pL}(T_w-T_s)/h_{fg}}\frac{M^2r'}{2\kappa_L} \quad \cdots\cdots(12)$$

$$\delta'=M\sqrt{r-r'} \quad (r>r') \quad \cdots\cdots(13)$$

式 (7), (13) から，加熱面上の各点で蒸発した液体の量，したがってそれによって吸収された熱量が容易に計算できる．一方，式 (12) はその間の経過時間を与える．したがって薄液膜の蒸発に基づく平均熱流束 $\bar{q}$ (加熱面面積および経過時間に対する平均) が次のように定まる．

$$\bar{q}=\frac{8}{5}\frac{c_{pL}(T_w-T_s)}{h_{fg}}\left(\frac{\rho_L\kappa_L h_{fg}}{M\sqrt{r_0}}\right)\left(\frac{r_0}{r'}\right)\times\left\{1-\left(1-\frac{r'}{r_0}\right)^{5/2}-\frac{5}{3}\left(\frac{r'}{r_0}\right)\left(1-\frac{r'}{r_0}\right)^{3/2}\right\} \quad \cdots\cdots(14)$$

ここで r' の時間変化は式 (12) で与えられる．本式によれば $\bar{q}$ は図 14 に示すように，蒸発期間中あまり変化しない．これは，蒸発に伴い液膜厚さおよび液膜表面の減少が同時に生じ，これらは伝熱に対し相反する影響を持つからである．なお一般に，$r'/r_0=1$ に達する前にすきまへの液体の侵入が開始される (図 10 参照)．現実験条件下に式 (14) から計算される $\bar{q}$ の値は

$r'=1.25$　　2.5　　5.0　　mm

$\bar{q}=2.42\times10^5$　2.64×10^5　2.52×10^5 kcal/m² h

これらは，休止期間をも含む全期間に対する平均熱流束 (実測値) 1.6×10^5 kcal/m² h より約 57% 大きい値である．

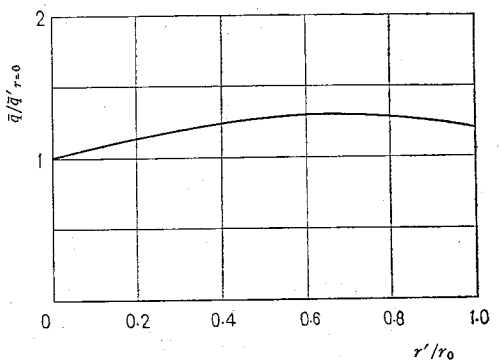


図 14　薄膜蒸発による熱流束の変化

高速カメラ撮影フィルムから，蒸発時間と休止時間の比の平均値を求めるのはかなりむずかしい．実験では不規則な流体挙動がしばしば現われるためである．しかし大まかにいえば 0.75 程度の値である．これらを考慮すると本実験の場合，薄液膜からの蒸発は全伝熱の約 2/3 を受けもっていることになる．残りの 1/3 は気液交換機構その他による伝熱である．

もっとも，以上の結果から直ちに，普通の核沸騰でも薄膜蒸発が常に気ほうの挙動や伝熱に大きく関与すると速断してはならない．少なくとも低熱流束の孤立気ほう域では従来，気ほう生長に関し薄膜蒸発を考えぬ理論が大まかには実験と傾向一致をみせているし，また伝熱自体も加熱面上の気ほう内蒸発に相当する伝熱量だけではことがすまぬことが常識である．問題にするとすれば，やはり高熱流束領域についてである．そして式 (14) に関連し，次の試論を付記しておくのは無駄でないと思われる．

式 (7) にあらわされたような非常に薄い液膜の創成機構はまだはっきりしない．しかし少なくとも気ほう境界の進行速度 (図 12) などとあまり深い関係がないようにもみえる．こうした事情もあるので，大胆ではあるがかりに沸騰条件に関係なく液膜の厚さが式 (7) であらわされ定数 M の値も変わらぬとすれば，式 (14) から $\bar{q}$ は $(T_w-T_s)/\sqrt{r_0}$ に比例して増加，つまり加熱面過熱度が高く，気ほう底面半径が小さいとき $\bar{q}$ は非常に高い値に到達し得ることになる．

試みに過熱度 $T_w-T_s=20$°C で沸騰している大気圧の水 (熱流束の実測値は普通 10^6 kcal/m² h 程度) を考え，式 (14) で $\bar{q}$ を計算すると ($\bar{q}$ は r'/r_0 であまり変わらぬから簡単のため $r'/r_0=0$ の場合をとる)，下記の気ほう底面半径 r_0 の値に対してそれぞれ

$r_0=0.1$　　0.5　　1.0　　mm

$\bar{q}=9.8\times10^6$　4.4×10^6　3.1×10^6 kcal/m² h

ここで気ほう付着面積と加熱面面積の面積比，休止時間，気ほう底部以外の蒸発などの影響を考えると，実測熱流束 10^6 kcal/m² h 程度の核沸騰では，気ほう半径はどうも 0.1 mm のオーダにあるのではないかということになる．たまたま，この結果は図 6 の結果と少なくとも傾向的に一致するのは興味深い．ただし以上は，あくまで試論であることを念頭におきつつ記したものであることをことわっておく．

10. 結　　論

本報では高熱流束域の核沸騰伝熱機構の問題を背景にしつつ，干渉板を加熱面に近づけた場合の沸騰実験をおこない，いろいろのことがらを明らかにしたが，

以下に要約して結論とする．

（1） すきま間隔の減少とともに一般には，非押しつぶし気ほう域，押しつぶし気ほう域，抑圧蒸発域の3領域があり，図9に示すように伝熱や流体挙動にそれぞれ興味ある特色がある．干渉板をもつ系のバーンアウトについては簡単な解析との比較を示した．

（2） 非押しつぶし気ほう域では，普通の核沸騰が支配的である．そしてその限界すきま間隔は，Gaertner ら[11]の求めた普通沸騰離脱気ほう直径にほぼ等しく，また蒸気塊保持くぼみをもつ干渉板では小さなすきま間隔まで伝熱特性の変化があまりみられないなどの結果を得る．流体挙動の観察などと総合して，高熱流束普通沸騰を考えてみると，合体気ほうは加熱面上の小気ほうからの蒸気の単なる集積結果で伝熱にはほとんど影響しないこと，少なくともある程度以上の広さをもつ加熱面上の小気ほうの直径は熱流束とともに連続的に減少するらしいこと，伝熱に重要な現象は加熱面から小気ほう直径程度の非常に薄い範囲内にあり自律的傾向をもつものであることなどが結論される．

（3） 押しつぶし気ほう域の現象と関連しては，本報で特に重要なものと考えられるが，薄膜蒸発機構の問題がある．すなわち，すきま空間における気ほうのひろがり速度が生長とともに増加したり，加熱面過熱度に比し熱伝達がはなはだ大きいという特異な現象が，薄膜蒸発を考えることによってよく説明される．しかもこれは薄液膜の厚さ分布の実測と，それに基づく解析に裏づけされたもので，従来，推論的な段階にとどまっていた薄膜蒸発機構を，特殊な系においてではあるが具体的な姿でとらえ得た――．換言すれば，この機構が大きな役割をもつ沸騰形式が少なくとも存在することが現実に見出されたわけである．しかし普通の沸騰での役割が大きいか小さいかについては，まだ具体的に明示する段階に達していないので，本報では終わりに薄膜蒸発機構を支持する立場からの一つの試論を付記するにとどめた．薄液膜の創成機構とともに今後確定すべき重要事項である．

（4） 抑圧蒸発域では核生成なく，特殊な蒸発形式をとる．

終わりにのぞみ，本研究の一部に協力された岡林慶一，谷　紘一の両君に感謝の意を表する．

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平行水平2円板間の狭い空間における核および遷移沸騰*

甲藤好郎**, 横谷定雄**, 寺岡幸三***

1. はしがき

狭い空間，ないしそれに近い状況下での核沸騰が，工業上問題になることがあり，そのため石橋・西川ら[1]~[4]，あるいは中島[5]などの興味ある研究がある．ただし，これらの研究は核沸騰域の熱伝達の研究を主体とするものであり，バーンアウトや遷移沸騰は扱われていない．また，核沸騰とバーンアウトの基礎的研究に関連して，著者ら[6][7]は大気圧の飽和水中に設けた直径 10 mm の水平上向き円形加熱面の上方に，同直径の干渉円板面を平行に保持し，両面間のすきま幅 $S=0.1\sim10$ mm 程度の範囲に対し，バーンアウトを含んで核沸騰の実験結果を報告している．しかし気液の挙動，特に狭いすきま内への液体供給のプロセスなどが，必ずしも系統的に調査されているわけではない．またごく一部を除いて遷移沸騰の実験を欠き，従ってバーンアウトの発生状況が明白でない〔文献(6)の研究では現実の気液挙動とは無関係な流動形態の仮定のうえに，バーンアウト発生の考察をしている〕などの問題がある．

そのような状況下に本報では，前記文献[6][7]とほぼ同じ条件（すなわち加熱面は直径 11 mm の円形，水平上向き）の狭いすきま内において，核沸騰から遷移沸騰にわたる比較的に熱流束の高い領域の実験をおこなった．そしてすきま幅 S の変化によって，熱伝達，気液挙動，およびバーンアウトがどのような特性変化を示すかについて，多少とも体系的な観察をおこない，その結果を報告するものである．もちろん加熱面の寸法，オリエンテーションなどが，きわめて限られた条件下の実験にすぎない．しかしそこに生ずる特異な気液の挙動と，それに伴う伝熱特性の変化の観察結果は，報告しておくだけの価値があるように思われる．また同時に，このような状況下に生ずる特殊なバーンアウトの性格に関する知見は，一般のバーンアウト現象の理解のためにも役立つはずである．

2. 実験装置

本報の実験装置は，さきに著者らが別の研究[8]で用いたものと同じで，その主要部を図1に示す．下部を電気加熱器で加熱される円すい形銅ブロックの上端にある，水平上向き円形加熱面（直径 11 mm）上で沸騰を生じさせる．なお，この銅ブロックの下部においては，電気加熱と同時に空気冷却をおこなわせることができ，この方法によって加熱面上に遷移沸騰が安定に実現できる．加熱面温度および加熱面熱流束は，銅ブロック内の熱電対で定められる温度分布から測定される．

使用した液体は，大気圧の蒸留水（実験に際し図1の沸騰容器内で自由水面の水位は加熱面上，約 100 mm の高さに保つ）で，補助加熱器によって飽和温度に保たれている．

図1で，加熱面上方に保持されている光学装置（著者らが前の研究[6][7]で使用したものと同じ）は，図2に示すような内部構造のもので，下方の円すい状ガラスの下面（直径 11 mm）が，加熱面と対になってすきまを形成すると同時に，このガラスを通してすきま内の気ほう挙動が光学的に観察（光学装置の左側の窓から入る光線で加熱面上を照明し，右側の窓を通して

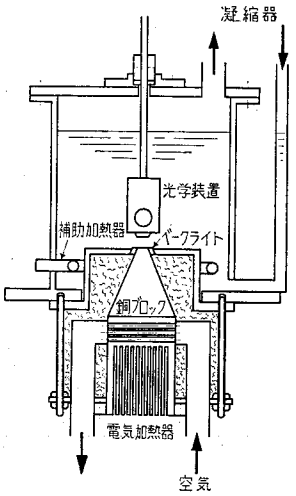

図 1 実験装置主要部

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観察）できるものである．なお本報では高速カメラを用い，上記の気液挙動を観測記録した．

実験において，加熱面上のすきま幅 S を設定するに際しては，実験装置各部の熱膨張による誤差を避けるため，約1時間沸騰容器内の温度が 100°C の状態に保持した後，光学装置の支持棒を移動させ，移動量をダイヤルゲージで測定するという方法で S を所定の値に設定した．なお銅ブロックは，加熱時，下方にむかって伸びるように取付けてあるので，加熱面温度を変える場合も加熱面の移動はなく，従って S の設定値に大きな誤差は生じないはずである．

3. 実験結果

前章に述べた実験装置で，加熱面直径 11 mm に対し，すきま幅 S の値を，$S=\infty$ mm（すなわち自由沸騰）から $S=0.1$ mm まで変えた場合，比較的高い熱流束域（10^5 kcal/m²h 以上）において，図3に示す沸騰曲線が得られた．これらの沸騰曲線は，S の減少と共に特徴的な変化をみせている．そこで以下，高速カメラによる加熱面上の気液の挙動観察と関係させながら，この沸騰曲線の変化の原因について記すことにしよう．

3·1 $S=2.0$ mm 付近における状況　図3にみるごとく，すきま幅 $S=2.0$ mm の場合の核沸騰は，まだ自由沸騰（$S=\infty$）の場合の核沸騰とほとんど同じ伝熱特性を示している．なおこの結果は，表示形式が非常に異なるが，文献(6)の図5および9（ここで $S\geqq 2.0$ mm の状況を考える）に報告した結果と同内容である．ともあれ $S\geqq 2.0$ mm のすきまの沸騰において，このような特性がみられる原因は，すでに文献(6)～(8)に記したごとく次の事情によると考えられる．すなわち大気圧の飽和水の沸騰の場合，10^5 kcal/m²h 程度以上の高熱流束域においては，加熱面に接して非常に薄い核沸騰液層があり，この液層内で激しい核沸騰を持続しつつ蒸気を上方に噴出，その液層に接して上方に蒸気塊を形成する状況となる．従って，$S\geqq 2$ mm 程度の比較的にすきまのひろい範囲では，ガラス壁は蒸気塊内にあり，その下方の核沸騰液層に対しガラス壁がほとんど影響を与えないからである．

ただしバーンアウト熱流束については，図3にみるごとく，$S=2.0$ mm の場合自由沸騰の場合のバーンアウト熱流束よりかなり低くなっている．そしてその主原因としては，すでに文献(7)の研究に示した状況，すなわち S の減少と共に，蒸気塊が加熱面を覆っている時間が長くなり，加熱面への液体供給と次の液体供給までの時間が延長されることによると考えてよいと思われる．

なお $S=2.0$ mm 程度のすきまにおける沸騰ではまだ，図4に模式的に示すごとく，すきまの一方から蒸気塊が離脱を開始〔その力学的挙動については文献(9)の研究を参照〕したとき，その蒸気塊を追尾する形で液体がすきま内に侵入する．そして核沸騰および遷移沸騰の両領域を通じて，この気液交換方式による加熱面への液体供給がおこなわれていることが観察される．たとえば一例として図5は，本報の実験において，すきま幅 $S=2.0$ mm，過熱度 $T_W-T_S=30.8$°C（ここに T_W：加熱面温度，T_S：飽和温度）の遷移沸騰時，すきま内における気液挙動を高速カメラで記録した結果を示すものである．すなわち図5のこま 1～4 において，左方から液体がすきま内に侵入しており，三日月状の液体侵入部分には激しい核沸騰を生じ

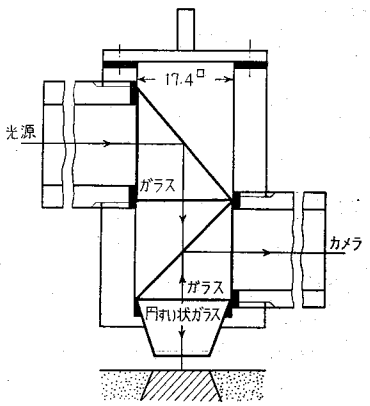


図 2　光学装置

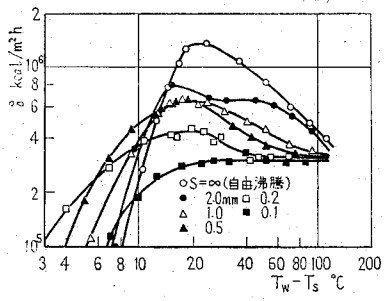


図 3　沸騰曲線

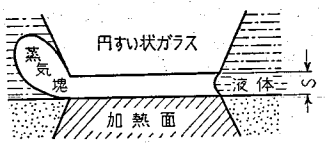


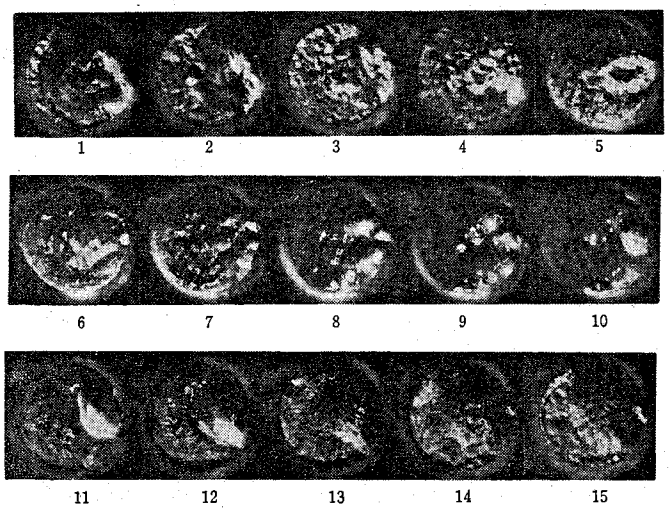
図 4　蒸気塊の離脱と液体の侵入

つつ，残りの蒸気占有部分との間に不連続な境界を出現しているのがみられる．なお図5のこま6以降になると，上記の核沸騰で生じた蒸気によって，すきま内全体が再び満たされ，かつ加熱面上は乾燥して核沸騰が消滅していることが観察される（こま6以降の写真にみられる波紋は，加熱面とは関係のない現象で，光学装置の円すい状ガラスの下面を連続的にぬらしている液面に生ずる波動である）．そして図5のこま15に至ると，再び左方から液体の侵入が開始するのがみられる．なお上記の図5に関する観察結果は，文献(7)の図10についての観察結果とも合致するものである．

3・2 $S=0.2\sim0.5$ mm 付近における状況

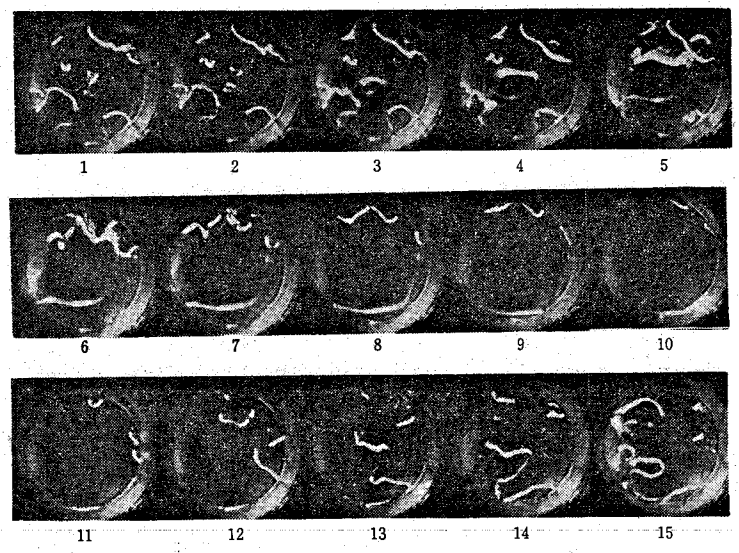
3・2・1 低過熱度の核沸騰域　すきま幅が $S=0.2\sim0.5$ mm 程度に減少すると，図3にみられるように，まず核沸騰域のうち過熱度の低い領域（$T_W-T_S<10°C$ の部分参照）の伝熱が，自由沸騰の場合よりかえって良好になるという現象を生ずる．

この伝熱上昇は，すでに文献(6)にも報告したことと一致するが，次のような現象に起因して生ずるものである．すなわち図6は，本実験で $S=0.5$ mm，$T_W-T_S=5.6°C$ の核沸騰に対し，すきま内の気液挙動を高速カメラで記録した結果であるが，この場合こま1～10にみるごとく，加熱面の中心付近に発生した気ほう（初気ほうであって，合体気ほうではない）が，上部のガラス壁の存在のため，すきま内を平らに広くひろがる形で生長し，このとき気ほう底部の加熱面上に，きわめて薄い静止液膜を形成，この薄液膜上面で急速な蒸発（図6の場合は薄液膜が乾くまでには至っていない）を生ずるためである．なおその後，図6のこま11以降にみるごとく，比較的ゆっくりした速度で気ほう蒸気がすきまから抜け出しつつ，その後に液体が侵入する方式で液体供給が生じている．



$S=2.0$ mm，$T_W-T_S=30.8°C$，時間間隔＝3.8 ms（こま 1～10），6.5 ms（こま 10～15）

図 5　すきま内の気液の挙動



$S=0.5$ mm，$T_W-T_S=5.6°C$，時間間隔＝0.75 ms（こま 1～10），4.7 ms（こま 10～15）

図 6　すきま内の気液の挙動

3・2・2 高過熱度の核沸騰域

加熱面の過熱度がもっと高い核沸騰の領域になると，図3の $S=0.2\sim0.5$ mm の沸騰曲線にみるごとく，その伝熱特性が自由沸騰の場合より不良となり，かつバーンアウト熱流束の値も，すきま幅 S の減少と共に低下する傾向がみられる．

さてこの領域において，すきま内の気液の挙動を高速カメラで観察してみると，加熱面上への液体の供給形式が，図4のものとは違う形式に変わっていることが注目される．すなわち，ここではすきま幅 S がかなり小さくなっている結果，図4の

液体侵入形式では，液体前面の凸面メニスカスと表面張力の作用によって発生する圧力差のため，液体の流動がさまたげられるといった事情によるのではないかと思われるが，図4の形式の代わりに，図7に模式的に示すような液体供給方式が出現する．すなわち，図7(a)のごとくガラス面をぬらしている液膜の一部が，界面の波動などに起因し，ときどき図7(b)のごとく加熱面のほうへ近づき，そこへ液体を供給するという方式である．

この液体供給方式を含む沸騰の状況を，実際の写真観察について説明すると，次のごとくである．すなわち図8は $S=0.5$ mm の場合，$T_W-T_S=18.4°C$ という比較的に過熱度の高い領域の核沸騰について，すきま内の気液挙動を高速カメラで記録した結果である．まず図8のこま1からこま15に至る全写真において，常に加熱面上に核沸騰が分布しているのが見られるが，詳細に観察すると，自由沸騰の場合のごとく単一の核沸騰液層が加熱面全面を覆っている状況にはなく，いわば分断された液片の形で多くの液層が加熱面上に分布しており，その分断液層内に核沸騰を生じていることがわかる．換言すれば，加熱面上に核沸騰液層部分と乾燥部分とが共存していることにほかならず，そして図3にみるごとく，$S=0.2$～0.5 mm の場合の伝熱特性が自由沸騰の場合より悪化する主原因はここにあると考えられる．なお，たとえば図8のこま1～4の写真で，加熱面の中心付近にみられるような，薄い液膜に包まれた形の比較的大きな気ほう（このような気ほうは大きく生長すると破裂することも多い）がよく観察されるが，この形式の気ほう発生も，分断液片内の発ぼうと密接に関係する現象である．

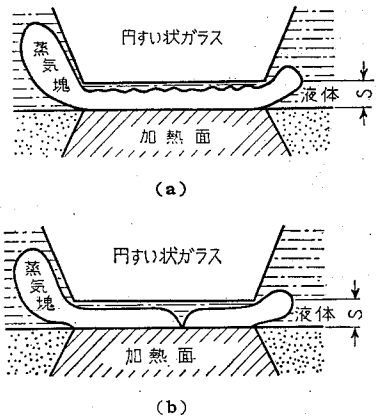
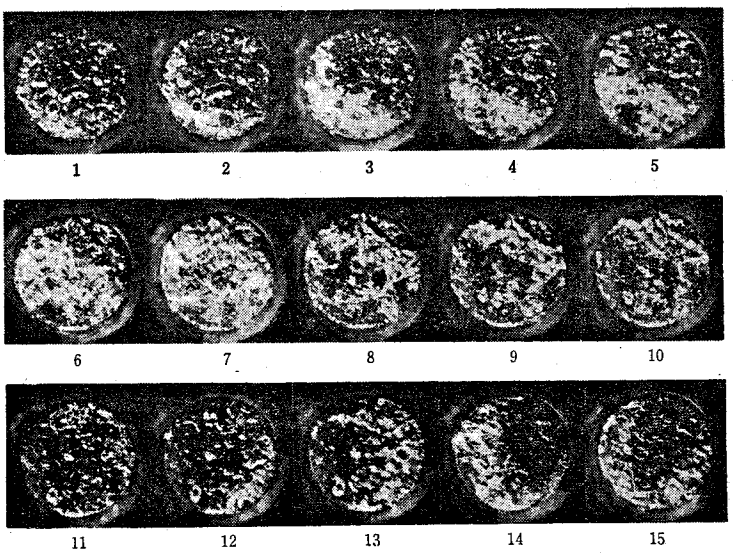


図 7　加熱面への液体供給の一形式

さて図8のこま1～7において，白い模様が加熱面の左下方から始まり，加熱面上にひろがって行く状況がみられる．そしてこの白い模様は，ガラス面をぬらしている液膜表面に生じた波動が観察されているものであるが，このような状況の場合にしばしば，図7(b)の形で加熱面への液体供給が生じ，このとき加熱面に触れた液体は，その接触部に生ずる核沸騰（なお液体は加熱面の乾燥部に触れることも多い）などのため飛散を生じ，その結果，付近の加熱面上に分断液片を供給するもののようである．実際図8の場合，こま7，8を境にして，その前後において加熱面上の液体の分布状況に相当な変化を生じている．そして上記のようなガラス面液膜の波動と，加熱面上の液体分布状況の変化の結びつきは，本報の実験において一般的に観察される現象であった．



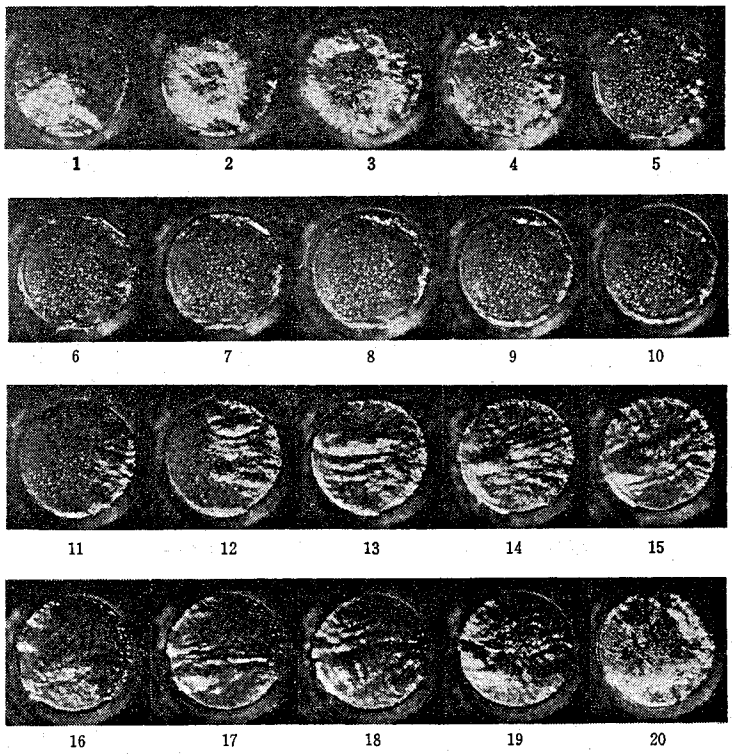
$S=0.5$ mm，$T_W-T_S=18.4°C$，時間間隔＝0.54 ms（こま 1～10），3.2 ms（こま 10～15）

図 8　すきま内の気液の挙動

3・2・3　遷移沸騰域　加熱面の過熱度がさらに高くなり，遷移沸騰域に入っても，図7の液体供給形式は保持され，しかもその供給形式が前節の場合より，もっと顕著な形で観察される．

すなわち図9は $S=0.5$ mm の場合，過熱度 $T_W-T_S=33.9°C$ の遷移沸騰について，すきま内の気液挙動を高速カメ

ラで記録した結果である．図9のこま2あたりで，円すい状ガラス下面の液体が加熱面の中心付近に接触しており，この場合過熱度が高く，かつほとんど乾燥した加熱面との接触のため，そこに非常に激しい核沸騰を生じ，細かい液滴を飛散させるのであって，図9のこま3～11において加熱面上にみられる多数の細かな分布液滴は，このようにして生じたものである．なおこれらの液滴は直径 0.1 mm 程度で小さく，従って気ほうの発生はみられず，加熱面上に静止した状態で蒸発している．また図9で，こま 11～19 にみられる波紋は，円すい状ガラス下面の液層を伝わる波動であって，こま 20 に至って再び加熱面への液体供給を生じ，前のこま 2，3 の近傍と同様な状況が再現されている．

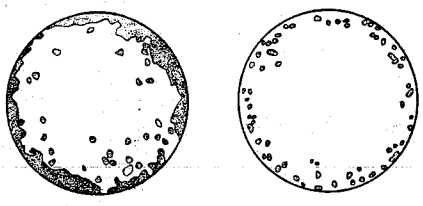


S=0.5 mm，T_W-T_S=33.9°C，時間間隔=0.63 ms（こま 1～10），5.0 ms（こま 10～20）

図 9　すきま内の気液の挙動

3·3　S=0.1 mm 付近における状況　図3において，S=0.1 mm の場合の沸騰曲線について，次の二つの特性が見いだされる．すなわち第1は，図3において熱流束をたとえば $q=2\times10^5$ kcal/m²h の値に保ちながら，自由沸騰（$S=\infty$）から次第に S の値を減少させる場合を考えると，S=0.2 mm に至るまでは S の減少と共に，加熱面の過熱度 T_W-T_S は低下するのであるが，さらに S=0.1 mm に至ると，T_W-T_S の値が再び高い値に急激にもどるということである〔これは文献（6）の図5において $q=1.3\times10^5$～2.3×10^5 kcal/m²h の曲線で示す結果とほぼ同性格である〕．第2に，図3で S=0.1 mm の場合の沸騰曲線には，もはや右下がりの部分がなく，いわゆるバーンアウト点の存在をみない状況になっており，従って核沸騰，遷移沸騰の両者が明確に区別できるような通常の沸騰状況とは，かなり異なる状況にあるといえる．

さて，S=0.1 mm のすきま内の気液挙動を高速カメラで観察した結果，ここではすきま幅 S が非常に小さいため，図7の形式の液体供給も生じ得なくなり，かつ加熱面はその周辺部を除き大部分が常に乾燥状態に保たれていることが見いだされた．すなわち，図 10 は S=0.1 mm における加熱面上の状況のスケッチである．そして，図 10（a）のごとく過熱度 T_W-T_S が比較的に低い状況では，加熱面周辺で不規則に液体が出入するのがみられる〔この状況は文献（6）の図9中にそう入した写真のうち，左端の2枚と同性格である〕が，図 10（b）のごとく過熱度 T_W-T_S が相当に高い状況になれば，加熱面周辺は小液滴が付着して蒸発するだけの状態になる．この小液滴は，加熱面外縁にときどき液体が触れ，そこに生ずる核沸騰が飛散させたものと考えられる．



（a）T_W-T_S=9.9°C　　（b）T_W-T_S=40.1°C

図 10　すきま（S=0.1 mm）内の状況

なお S=0.1 mm の場合では，円すい状ガラス下面もまた乾燥状態になっていると判断される．なぜな

ら，図 5，7，8 などにみられた形の波紋が，$S=0.1$ mm の実験ではまったく観察されないからである．

3・4 膜沸騰の近傍の状況 図 3 をみるに，$S=\infty\sim0.1$ mm の範囲のすべての沸騰曲線が，過熱度 $T_w-T_s=150°C$ 付近で，ほぼ同一の値にむかって集合して来る傾向をみせている．そして，これは次の事情に関係していると考えられる．

ここで，まず自由沸騰（$S=\infty$）における膜沸騰を考えるに，本報に用いた寸法の加熱面（直径 11 mm であって，これは 100°C の飽和水-水蒸気の気液界面の臨界波長より小）の場合は，主として加熱面外縁付近だけで伝熱を生ずる状態にあることが観察される．なぜならこの膜沸騰では，すでに文献 (9) の図 5（写真）と図 6 に報告したごとく，加熱面を全面的に覆う形で生長する単一気ほうが，非常に規則正しい振動的周期をもって加熱面から離脱する状態にあるのであって，従って 1 周期の大部分の間，加熱面は常に気ほう蒸気に厚く覆われながら，しかもその間に大きく気ほうが生長して行くのであり，この気ほう生長に必要な蒸気は，加熱面外縁の近くの薄い蒸気層を通ずる伝熱以外に供給源が存在し得ないのである．そしてこのような状況の膜沸騰では，加熱面上にそれと同寸法のガラス壁を置き，かつそれを加熱面に近づけて行っても，加熱面外縁の伝熱については大きな変化なく，そのため図 3 のような結果になっているものと推定される．

4. すきま幅による状況変化

前章に実験，観察結果を記したが，それらを通覧し，すきま幅 S の変化によって沸騰およびバーンアウトに状況変化が生ずる様子について，整理した形の考察を加えておくのが有用と考えられる．ただし以下において遷移沸騰と言う場合，膜沸騰の近傍領域は除くものとする．

4・1 沸騰状況の変化 前章の結果によれば，加熱面直径 11 mm の場合，すきま幅 $S=\infty\sim2$ mm 程度の範囲内の沸騰状況は，まだ自由沸騰（$S=\infty$）の場合とほぼ同じ特性を保持していると考えてよいように思われる．すなわち，核沸騰域では核沸騰液層が加熱面上を全面的に覆う状況にあり，一方遷移沸騰域では周期的に乾燥状態が加熱面上に出現する状況にある．また加熱面上への液体の供給は，核沸騰，遷移沸騰いずれの領域でも，蒸気塊が加熱面から離脱する際に気液交換形式でおこなわれるとみてよいようである．

しかし，$S=0.5\sim0.2$ mm 程度のすきまになると，沸騰状況にかなり大きな本質的変化が現れる．まず，加熱面の過熱度がかなり低い核沸騰域において，すきま内を広くひろがる平らな気ほうの発生，および薄液膜を通しての激しい伝熱現象がみられる．一方加熱面の過熱度が高くなると，自由沸騰の場合に近い沸騰状況が加熱面上にみられるようになるが，ただし加熱面上に存在する核沸騰液層は，自由沸騰の場合とは異なり，分断された液片状であって，それらの液片の間に加熱面乾燥部分を混在させた形になっている（沸騰曲線上の核沸騰の領域にあるにかかわらず）．しかもこの加熱面上の分断液片の存在は，図 7 に示す液体供給形式と結びついて生ずる現象のごとくである．そして，さらに加熱面の過熱度が上昇するに従い，加熱面上に飛散する液片の寸法が小となりつつ，乾燥部分の増加などによる熱流束低減効果が打ち勝つようになり，やがて沸騰曲線上に限界熱流束の状況の発生をみるに至るのではないかと考えられる．

最後にすきま幅が $S=0.1$ mm 程度になると，沸騰ないし蒸発は常に加熱面外縁付近で生ずるのみで，すきま内はほとんど蒸気のみに占められ，加熱面およびガラス壁とも，そのほとんど大部分が乾燥状態に保たれるという状況になる．そして，このような状況の加熱面は，通常の意味での核沸騰や遷移沸騰の状況とはまるで無縁なものである．

4・2 バーンアウトの発生状況の変化 すきま幅 S の減少に伴う沸騰状況の変化について前節に記したが，それに付随するバーンアウトの発生状況の変化は下記のごとくである．ただし本節でバーンアウトというのは，沸騰曲線上の限界熱流束点を形式的に指すだけの言葉としておくことにしよう．

まず $S=\infty\sim2$ mm 程度の範囲内でのバーンアウトは，自由沸騰におけるバーンアウトと同じ機構に従うものである．ここでは一般に，核沸騰域の加熱面は全面が常に液体にぬらされた状況にあり，そして遷移沸騰域に入ると加熱面上に乾燥状態が生じ始めるといった形式を持っている．

しかし，$S=0.5\sim0.2$ mm 程度のすきまになると，前節に記したごとく，バーンアウト点近くの核沸騰域において，すでに加熱面上の分断液片間に乾燥部分が混在しており，しかもこのような形態の核沸騰こそが，現在の場合の液体供給方式（図 7 参照）と関連する必然的状況である．従ってこの場合，バーンアウト点の前後で，加熱面上の沸騰状況は自由沸騰の場合よりもっとゆるやかな変化を示すこととなり，それだけバーンアウトというものが自由沸騰の場合に比し，あいまいな性格をもって発生していると言ってよいであ

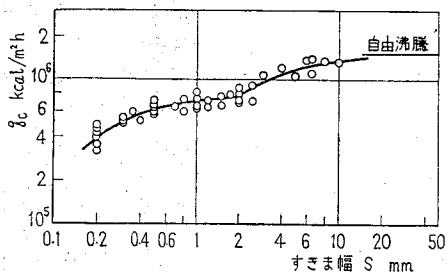


図 11　すきま幅 S によるバーンアウト熱流束 q_c の変化

ろう．そして実際，さらにすきまがせまくなり，S=0.1 mm 程度になると，図3の沸騰曲線にみるごとく，もはやバーンアウトというもの自体が存在しなくなってしまうことになる．

なお図 11 は，本報の実験において，すきま幅 S とバーンアウト熱流束 q_c の関係について測定した結果を示すもので，かつ同図右上部に掲げた水平細線は，自由沸騰（$S=\infty$）の場合の q_c の測定値である．そして少なくとも図 11 のデータに関する限り，S=2 mm の付近を境に q_c-S の関係が多少の特性変化を示しているようにみえるのは興味深い．ただしこのことについては，図 11 の結果だけから真実と断定するのは危険であり，なお今後の検討が必要であろう．なぜなら文献（7）の図4に掲げた q_c-S の実験データは，本報の図 11 のデータと値がほぼ一致しているものの，S=2 mm 付近を境にする特性変化は示していないからである．もっとも文献（7）の研究の段階では，S=0.5～0.2 mm 付近の沸騰状況が，$S=\infty$～2 mm の領域のそれと相違することが留意されておらず，従って図 11 に比べてはるかに実験データの数が少なく，q_c の測定にかかわる統計的精度が本報よりまさっているわけのものではない〔仮に文献（7）の図4において S=1～2 mm の間の2個の ○ 印データがもう少し低い値であったとすると図 11 の傾向に近くなる〕

5. むすび

本報では，直径 11 mm の水平上向き円形加熱面の上方に，直径 10 mm のガラス壁を保持し，すきま幅 $S=\infty$～0.1 mm の範囲で，大気圧の飽和水について核沸騰および遷移沸騰の実験をおこなった．そして S の変化に伴う沸騰状況，液体供給方式，バーンアウト発生状況などの変化の特徴を明らかにすると同時に，著者らの従来の研究報告[6]～[8]に含まれている各種状況が占める位置と相互関係を決定することができたと思う．

なお本報に記したごとき，すきま内の気液挙動の観察は，一見非常に容易におこなわれたもののようにみえるかもしれない．しかし実際には，加熱面上の状況とガラス壁下面の状況が同時に無差別に観測されるうえ，非常に不規則な変動現象を含んだりするため，観察された状況に矛盾なく合致し，かつ信頼できる流動モデルを導くためには，きわめて多数の高速写真記録について試行錯誤的な分析のくりかえしが必要である．すきま内の気液挙動の観察には，少なくとも上記のような困難さがあることを記しておきたい．

またせまいすきま内の沸騰と，管内流の沸騰とは，まったく別の現象であり，両者の間に安易な類似を考えるのは危険である．しかし，流体を含む空間が固体壁で拘束されている点に共通性があるためか，バーンアウトの発生状況の変化について，両沸騰の間には多少の類似性があるように思われる．

文　献

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（2） 石橋，機論，**34**-257（昭 43-1），150.
（3） 石橋，機論，**34**-260（昭 43-4），743.
（4） 西川・ほか2名，機論，**34**-264（昭 43-8），1455.
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（6） 甲藤・横谷，機論，**32**-238（昭 41-6），948.
（7） 甲藤・横谷，機論，**34**-258（昭 43-2），345.
（8） 甲藤・横谷，機論，**37**-295（昭 46-3），535.
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討　論

〔質問〕　赤川浩爾（神戸大学工学部）

狭い空間における沸騰現象が分類された各様式について，実験結果を物理的に明確に説明されており，沸騰伝熱の本質を明らかにされていることに敬意を表する．本報のみにかぎらず，この一連のご研究に関連する事柄と思われるが，次のことについてご教示をねがいたい

文献（7）で述べられているように「沸騰…は種々の面で時間的に急激な変動，つまり周期的ないし断続的過程を含む現象である」ので，バーンアウトあるいは遷移沸騰が同文献，図 12, 13 のように時間的因子を含めて解明されている．このようなことに関係して，本報で各沸騰状況の現象的説明はめいりょうになされているが，液膜供給に関係する狭い空間における気ほう

の成長，離脱，液供給などの時間的因子についてはあまり述べられていない．これらの値は図3の沸騰曲線を理解する上からも重要であるように思われる．したがってこれらの時間と，S，q とのおおよその関係をご教示下さるとありがたい．なお文献 (6) 957 ページで「蒸発時間と休止時間の比の平均値を求めることはむずかしい」と述べられており，統計的にかなりばらつきのあるものと思われるが，上記の値のおおよその範囲と傾向でも教示くださることを望む．

〔回答〕　気ほうの時間的因子と q の関係という場合，(1) 気ほうが初気ほうをいうのか蒸気塊をいうのか，また (2) q が沸騰曲線のそれ (T_W-T_S との関係) をいうのか，限界熱流束のそれ (q の限界値) をいうのかを明確に区別した上で議論する要があり，なお (3) 一般に沸騰曲線の q に対しては特に時間以外の因子 (発ぽう点密度，面の汚れなど) が強く影響する．なお上記 (3) 項に関連し，文献 (7) の図 13 だけでなく，その後の研究による文献 (8) の図 14 をも参照していただければ幸じんである．

さて端的に言って，著者らが従来の研究でとりあげた時間的因子としては，通常のプール沸騰の「限界熱流束」(およびそれに関連する遷移沸騰の解釈) に対する「蒸気塊」の時間因子の影響が主であり，それ以外では特に時間的因子を主役にするようなことをしていないことをご了解いただけないであろうか．そして少なくとも，本報の図3の沸騰曲線に対し，時間と S，q のおおよその関係を考えるという問題に対し，著者らはまず上記のように回答申しあげるべき立場にあるように感じる．

次に，本報の実験範囲の狭い空間での気液の挙動に関し，その時間的因子については，文献 (7)～(9) および本報内に，だいたいのデータは掲示してあるので，それによりおおよその値は見ていただけるのではないかと思う．すなわち

(i)　$S\fallingdotseq\infty\sim2$ mm．蒸気塊の離脱周期 (切断周期) の変化範囲については文献 (7) の図 12 を参照されたい．なおこの領域では，核沸騰液層 (3·1 節参照) 内の沸騰状況は S によって変わらず，また遷移沸騰も文献 (8) の図 14 の状況に近い性格を保持すると考えられる (図5の説明参照)．

付表 1

T_W-T_S	液体供給の平均周期
33.7°C	72.8 ms
52.1°C	121.8 ms

(ii)　$S\fallingdotseq0.5\sim0.2$ mm の低過熱度核沸騰域．ここで生ずる平たい初気ほうの生長には，加熱面上の発生位置 (および q による発ぽう点密度の変化) の問題があるが，気ほうが加熱面の中心付近から発生する場合，生長時間 (すきま内の気ほう生長期間，およびそれに続く長い薄液膜蒸発期間) のオーダは文献 (6) の図 10 (これは S や T_W-T_S に関し図6とほぼ同じ) から知ることができる．なおこの場合，気ほうの蒸発時間と休止時間の比は，文献 (6) 957 ページに記したごとく 0.75 程度のオーダである．

(iii)　$S\fallingdotseq0.5\sim0.2$ mm の遷移沸騰域．この領域では，本報の図 7(b) の形式の液体供給の周期が比較的めいりょうに計測できる．例えば本報の図9のこま 1～20 の間の時間が，それに相当するものであり，同図から 50.0 ms と計算される．なお，この液体供給周期は，T_W-T_S の上昇と共に長くなって行く傾向があり，例えば $S=0.5$ mm の場合，計測例として付表1のようなデータがあるので付記する．

このように T_W-T_S の上昇と共に周期が長くなるのは，液体供給時および飛散液滴の加熱面上での蒸発がより急激となり，蒸気圧力が高くなって液体の侵入をさまたげる方向に働くなどのためかもしれない．

(iv)　$S\fallingdotseq0.1$ mm．ここでは，いかなる形にせよ気ほうの時間的因子と q との関係といった問題は無くなるであろう．

536.423.1：532.542

平行水平2円板間流路の自然流動沸騰の限界熱流束*

高庄幸孝**，甲藤好郎***

1. 序

制限流路（confined channel）内の自然流動沸騰の限界熱流束（CHF）の問題は，工業上しばしば重要になり，またCHF現象の基礎研究の面からも大切な問題である．

さて制限流路のなかで水平流路系を考えると，甲藤・横谷[1]は平行2円板間流路（下円板加熱）に関し，大気圧の水の実験をおこなったがCHFデータの整理式は提出していない．またJensenら[2]は同心二重管（内管加熱）の形の水平流路に関し，大気圧の水でCHFの実験をおこなうと共に，R-113の1データ（バブコック・ウイルコックスで得られたもの）をも考慮して一つの無次元整理式を発表している．しかし，その整理式は，蒸気の流れに関するレイノルズ数（蒸気の粘性による）を主因子とし，重力の加速度gを欠く形のものである．一方，Smirnovら[3][4]は，幅b，すきまsの平行スロット（幅bに直角方向には非常に長いスロットと想像される）に関し，水（$p=0.5\sim5$ bar）およびエチルアルコール（$p=1$ bar）の限界熱流束q_{co}を測定，その結果を次式で整理している．

$$q_{co}=q_{co}^{*}\frac{1}{\sqrt{1+C_3(\rho_v/\rho_l)(b/s)^3}} \quad \cdots\cdots\cdots\cdots(1)$$

ここにq_{co}^{*}は通常のプール沸騰（$s\to\infty$）でのCHF，C_3は定数，ρ_vは蒸気密度，ρ_lは液体密度である．しかし式（1）と実験データの一致はあまり良好ではなく，−33〜+70%程度の誤差がみられる．

ところで著者の一人は最近，一様加熱垂直円管内の強制流動沸騰のCHFに対し気液界面の不安定に基づく考えを軸にして無次元整理式を導き[5][6]，そしてそれと同様な手法をいくつかの制限流路内の自然流動沸騰系のCHFに適用した場合の見通しについても報告した[7]．

さて，上記の報告[7]において，平行水平2円板間流路（下円板加熱）のCHFも扱われているが，実験データ不足のため漸定的な結果を与えるだけに止まっている．そこで本報では，この沸騰系について，新たに2種類の加熱面径，および4種類の流体を用いて大気圧下の飽和プール沸騰の実験をおこない，CHFの無次元整理式を導いた結果について報告する．

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記　号

d：加熱面直径　m または mm
g：重力の加速度　m/s²
H_{fg}：蒸発潜熱　J/kg
q_{co}：飽和沸騰の限界熱流束　W/m²
s：加熱面と非加熱円板との間のすきま距離　m または mm
ρ_l：液体密度　kg/m³
ρ_v：蒸気密度　kg/m³
σ：表面張力　N/m

2. 実験装置および実験方法

実験装置の主要部を図1に示す．沸騰容器①の中にある大気圧の液体（蒸留水，純度99.5%以上のエチルアルコール，純度99.99%以上のR-113および純度95%以上のベンゼン）は，補助加熱器⑤によって飽和

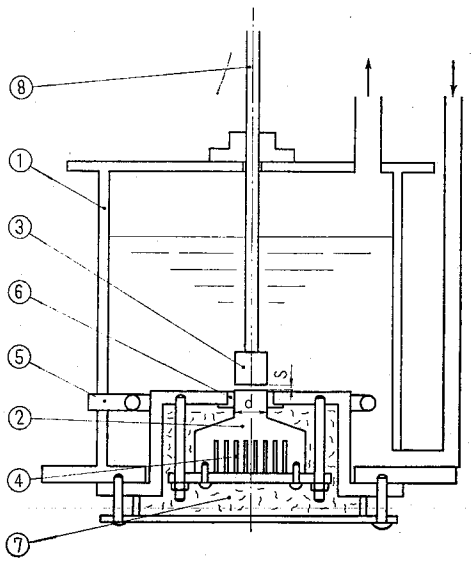

図1　実験装置主要部

温度に保たれ，発生蒸気は凝縮器を通して沸騰容器にもどされる．沸騰を生ずる加熱面(直径d)は，底部を板状電熱器により加熱される銅ブロック②の上部の円柱部端面を用い，この上向き加熱面を通る熱流束の測定は，円柱部の中心軸上に等間隔に配置した3個の銅-コンスタンタン熱電対が示す温度こう配から求めた．この際，熱流束の算出に必要な銅の熱伝導率については，使用銅ブロックごとに予備実験で伝熱量を慎重に検定して求めたところの値を用いている．加熱面直径dは 10 mm および 20 mm の2種類，また円柱部の軸方向長さは 15 mm（限界熱流束の値の大きい水の実験に使用）および 40 mm（限界熱流束の値の小さい他の液体の実験に使用）の2種類である．加熱面と同軸上方に，すきま距離sをもって保持される非加熱水平円板③はベークライト製のもので，その直径と厚さは共に加熱面直径dにひとしい．非加熱水平円板③の支持棒⑧は，ねじで上下に移動できるもので，加熱面と非加熱円板間のすきま距離sはダイヤルゲージで測定した．

実験は次の手順でおこなった．沸騰容器内の液体を飽和温度に保った状況下に，加熱面上の沸騰の熱流束をCHFの予想値よりやや低めの状態にしてから，水平円板③を下ろして加熱面に接触させる．このとき加熱面は乾き，温度上昇をはじめるが，迅速にダイヤルゲージを読むのは容易で，これによってすきま距離sの零点が決定される．このような方法をとったのは，銅ブロックの熱膨張による加熱面の位置の移動誤差を極力防ぐためである．次いで水平円板③を引きあげて所定のすきま距離sに設定してから，銅ブロックの加熱を段階的に上昇（CHFの測定誤差が5％以内になるよう上げ幅を設定）させ，加熱面温度が定常値を保ち得ず急上昇を開始する状態で限界熱流束q_{co}を求めた．

3. 限界熱流束の実験結果

さきに著者の一人[7]は，本報で扱っている飽和沸騰系のCHFが漸定的に

$$\frac{q_{co}}{\rho_v H_{fg}}\Big/\sqrt[4]{\frac{\sigma g(\rho_l-\rho_v)}{\rho_v^2}}=K\frac{1}{1+Cd/s} \quad \cdots\cdots(2)$$

の形でまとめられること（ここにK, Cは無次元定数），およびCは実は定数ではなく$g(\rho_l-\rho_v)d^2/\sigma$の関数になり得る可能性があることを記した．そこで本研究で求められたCHFの実験データを示すにあたり，式（2）を参考にして，まず

$$\sqrt[4]{\frac{\sigma g(\rho_l-\rho_v)}{\rho_v^2}}\Big/\frac{q_{co}}{\rho_v H_{fg}} \text{ 対 } \frac{d}{s}$$

の形に図示してみたところ，d=20 mm の場合に対し図2，d=10 mm の場合に対し図3の結果が得られた．これらの結果によれば，まず式（2）右辺の定数Kの値（図2,3において$d/s=0$における縦軸の値が$1/K$である）については，各液体および加熱面径に対して，ほぼ同じ値をとり，それらの平均値を求めると

$$\frac{1}{K}=5.5 \quad \cdots\cdots(3)$$

次に，図2,3の各斜線は，$d/s=0$において縦軸の値5.5［式（3）の値］を通るごとく引いたもので，従って式（2）右辺分母のCに対し各斜線のこう配が$5.5C$をあらわしている．すなわちCの値は，加熱面径と液体の種類の組合せごとに異なるわけで，図2,3から求めた結果は表1のごとくである．

4. 限界熱流束の実験データの無次元整理

3章に記したCHFの実験データに関し，それらの無

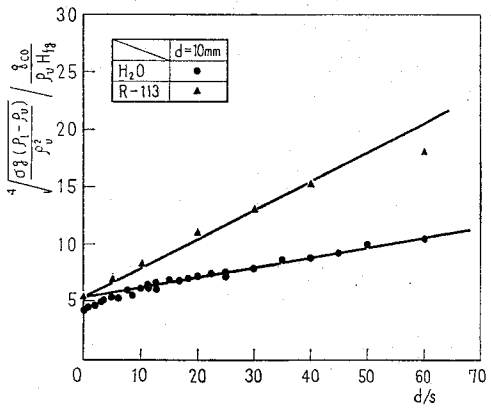

図 2 限界熱流束 q_{co} の実験結果

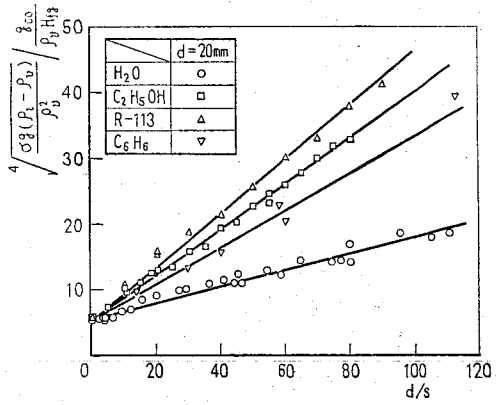

図 3 限界熱流束 q_{co} の実験結果

次元整理式を導くためには，表1に示すCの値に関し一般的に成立する規則性を見いだす要がある．そこで3章のはじめに記したところのCに関する可能性をテストすべく，Cと$g(\rho_l-\rho_v)d^2/s$の間の関係をみた結果が図4である．そして図4によると，Cと$g(\rho_l-\rho_v)d^2/s$の間に密接な関係がみられるが，なお水（ρ_l/ρ_vの値が比較的に大）と他の液体（ρ_l/ρ_vの値が比較的に小）のデータ間にわずかな偏りが存在するようにみえる．もちろん現状では，この偏差が本質的なものか，あるいはデータのばらつきにすぎないものか，確定はできない．しかし，いまは現象論的に扱うこととし，上記の偏差の調節のためρ_l/ρ_vの影響を加えて整理してみた結果が図5である．図5では，各種の実験のCの値が図4よりよく整理され，これから次の無次元整理式が得られる．

表 1　Cの実験値

流　体	d mm	C
		$\times 10^{-2}$
H_2O	10	1.53
H_2O	20	2.45
C_2O_5OH	20	5.93
R-113	10	4.55
R-113	20	7.31
C_6H_6	20	4.44

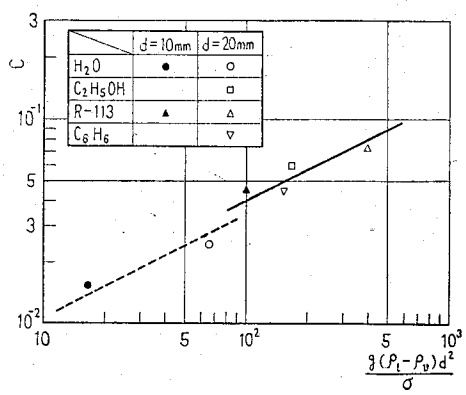

図 4　Cと$g(\rho_l-\rho_v)d^2/\sigma$の関係

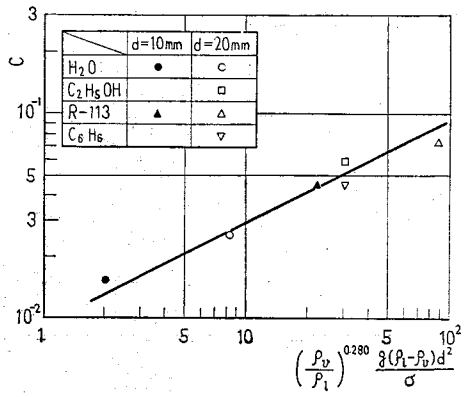

図 5　Cの無次元整理

$$C=0.009\,18\left(\frac{\rho_v}{\rho_l}\right)^{0.14}\sqrt{\frac{g(\rho_l-\rho_v)d^2}{\sigma}} \quad \cdots\cdots(4)$$

従って，式（3）に得られたK，および式（4）に得られたCを，式（2）右辺に代入すれば，本報の実験で求めた CHF の無次元整理式が次のごとく定められる．

$$\frac{q_{CO}}{\rho_v H_{fg}}\Big/\left[\frac{\sigma g(\rho_l-\rho_v)}{\rho_v^2}\right]^{1/4}=0.18\frac{1}{1+0.009\,18\left(\frac{\rho_v}{\rho_l}\right)^{0.14}\left[\frac{g(\rho_l-\rho_v)d^2}{\sigma}\right]^{1/2}\frac{d}{s}} \quad \cdots\cdots(5)$$

本式は$s\to\infty$（すなわち$d/s\to 0$）の極限において，通常の圧力下の飽和プール沸騰における CHF の Zuber の式（同式に対し定数の値は 0.18 が Rohsenow[8]によって推奨されている）に一致する．他方，図2,3の結果からみて，少なくとも$0\leqq d/s<120$の範囲において式（5）は適用可能としてよいであろう．この条件は，たとえば$d=20$ mm のとき，0.18 mm $<s\leqq\infty$に対応し，sについて非常に広い範囲を与えるものである．

最後に，図6は，本報で得られた$q_{CO}/\rho_v H_{fg}$の全実験値を，同じ条件下に式（5）が与える$q_{CO}/\rho_v H_{fg}$の計算値と比較したもので，計算値は実験値に対し，

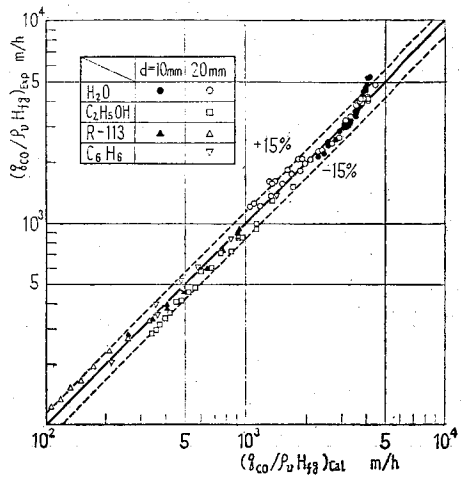

図 6　限界熱流束の実験値と無次元整理式（5）による計算値の比較

ほぼ±15%の誤差範囲で一致することが見られる．なお図6で，$d=10$ mm の水のデータ（●印）の一部が，他の諸データに比し誤差が大きいが，これは図3において $d/s=0$ の近傍のデータのものである．すなわち $d=10$ mm の水の実験では，通常のプール沸騰時（$s=\infty$）の CHF が，Zuber の整理式より多少高い値をとることが上記の偏差の原因になっている．そこで試みに次の値 d'：

$$d'=d/\sqrt{\sigma/g(\rho_l-\rho_v)} \quad \cdots\cdots(6)$$

を求めてみると，大気圧の飽和水で，$d=10$ mm の場合 $d'=4.05$，$d=20$ mm の場合 $d'=8.10$ となる．そしてプール沸騰 CHF に対する加熱面寸法の影響を研究した Lienhard らの論文[9]などを参照すると，$d'=4.05$ の場合は上記のような偏差を生ずる可能性が十分にみられるのであって，従って当然の結果と解してよいようである．なお図3に掲げる，もう一つの流体 R-113（大気圧）については，$d=10$ mm のとき $d'=9.93$ であり，上記のような偏差の発生の可能性がないことを付記する．

5. 結　　論

（1） 液体内にある水平上向き円形加熱面上にすきま s を隔てて同径の非加熱水平円板を保持した流路における大気圧下の飽和自然流動沸騰に関し，加熱面径 $d=10$ mm および 20 mm について，水，エチルアルコール，R-113，ベンゼンによる限界熱流束 q_{co} の実験（$d/s=0\sim120$ の範囲）をおこない，無次元整理式（5）を得た．

（2） 本報の実験範囲は，特に加熱面径 d が 10 mm，20 mm に限られ，かつ4章に記したような問題を含んでいる．従って，将来，もっと大きな加熱面径による実験がおこなわれ，それによる式（5）の検討がおこなわれることが望まれる．

（3） 本研究の結果は，他の形状の制限流路内の自然流動沸騰の CHF について無次元整理式を導く際に参考になるのではないかと考えられる．

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討　　論

〔質問〕　**武山斌郎**（東北大学工学部）

（1） $d=10$ mm の実験でも，d/s が零に近いほど，沸騰実験では，測定値が安定で，精度も高くばらつきも少ないだろうと考えられる．しかし，整理式からみると，他の諸データに比し誤差が大きいという表現が文中にあることに，少しく抵抗を感じる．無次元整理式にはデータの重みは入らないものか．

（2） 結論に，もっと大きな加熱面による式（5）の検討が望まれるとあるが，予想して，$d=10$ mm をも含む式になるのか，そうでないのか，考察があればお教え願いたい．

（3）「銅の熱伝導率については使用ブロックごとに予備実験で伝熱量を慎重に検定して求めたところの値」とあるが，その方法と値を，もしよろしければお教え願いたい．

〔回答〕　（1） 1721 ページ左欄 1～3 行めをみていただくとおわかりのように，ここで誤差が大きいというのは，実験データを式（5）と比較した場合の話である．実験誤差の意味ではない．そして実は，$d'=4.05$ の場合，$d/s=0$ の近傍で式（5）の方が成立しなくなると見るべきだろうと述べてさえいるのである（1721 ページ左欄 12～15 行参照）．

（2） 本件については現在のところ何もわからない．それだけに特に実験による検討を必要とするともいえる．

（3） 大きく安定な飽和液プール内に銅ブロック加熱面をおき，限界熱流束に比較的近い高熱流束領域（ここでは伝熱量のほとんど全部が加熱面近傍で蒸気生成に消費される）で核沸騰をさせる．そして発生蒸気を補そくし凝縮器に導き，これによって伝熱量を求める．周知のように金属では，純度や製造過程によって熱伝導率が変わる上に，同じ材料の銅ブロックでも

ブロック内にそう入する熱電対と銅の熱的接触状況などにより温度こう配の測定値に相違を生じ，それから来るみかけ熱伝導率の相違の問題もある．なお温度こう配の大きさを数点かえて伝熱量を測定するが，普通両者は比例関係を示し，その比例定数として熱伝導率（みかけの）が定められる．本報の実験の場合，銅の熱伝導率の測定値は 342 kcal/mh°C とか 337 kcal/mh°C などといった大きさであった．

〔質問〕　**伊藤猛宏**（九州大学工学部）

ささいなことだが，図6の●印の点に関する記述に関連して伺う．「他の諸データに比し誤差が大きい」ことの理由が述べてあるのだが，

（1）式（3）の $1/K=5.5$ が，つまり $K=0.18$ が疑念をはさむ余地がない程度に確立された値であると考えてよいか．

（2）文献（9）や他の Lienhard らの研究(付1)を参照しても，d' が4程度ならば加熱面の寸法効果が顕著であるが，8や10程度であれば問題にしなくてもよいとは理解されない．以上の2点に関してご意見をいただきたい．

〔**回答**〕　（1）本報の内容は，あくまで本報で得られた実験データとその整理のみにかかわるものである．図6に関し，±15% の誤差範囲とか，データの一部がそれより誤差が大きいという記述もすべて，その意味で使われているところの事実である（1720 ページ右欄下3～1721 ページ左欄3行め）．また式（3）の $K=0.18$ も本報の実験データから得られた数値であり，かつこの $K=0.18$ に対し質問者の言われるような性格づけはしていない．疑念のない値が厳存する場合には，推奨値というようなものは使用する必要がない（1720 ページ右欄下 12～9 行め）．

（2）（i）質問者の引用文献（付1）は現在の問題に対し無意味である．実際，文献（付1）は無限大の平板加熱面上の限界熱流束の研究が主目的であり，そのため故意に有限加熱面の周囲に側壁，また加熱面上にしゃ断板などを置いた人為的状況下の実験をおこなっている．従って，そのデータは現在の問題（自然的状況下の d' の限界の定量的推定）には使用できない．実際，Lienhard らがその研究を集大成した形の文献（付2）では，文献（付1）の研究は無限大加熱面の問題として2章におかれ，次いで3章においてはじめて文献（9）を主体とした有限加熱体の限界熱流束が扱われる形になっている．

（ii）文献（9）の図6は，4個の形状の有限加熱体の限界熱流束の多くの実験データの統一的表現である．水平円形加熱面のデータが見当たらないので，現在のところ一応上図の結果からの推測にたよらざるを得なかった．それにしても上図の結果では，加熱面形状による相違はあるが $L'=L\sqrt{g(\rho_l-\rho_v)/\sigma}$ が2～～6以上で限界熱流束の値が L' に独立に一定となる結果を与えている〔上図の原データについては文献（9）の図4,5，または文献（付3）の図7などを参照〕．

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寄　書

536.423.1 :: 532.542

制限流路内の自然流動沸騰の限界熱流束の無次元整理について*

甲　藤　好　郎**

1. 序

通常の単純なプール沸騰，および少数の沸騰系〔たとえば文献（1）～（3）など〕を別にすれば，多くの沸騰系の限界熱流束について無次元整理式がまだ得られていない．著者は最近，円管内の強制流動沸騰に関し広はんな実験データ（流体は7種類）を分析，4個の特性領域を区別した上で各領域に対し限界熱流束の無次元整理式を導いた[4]．本報は，前記分析に用いたと同じ手法を，制限流路（confined channel）内の自然流動沸騰に適用してみた場合の暫定的結果（実験データが少ないので）について報告するものである．

2. データ分析の手がかり

本報では一様加熱（等温）加熱面を持つ制限流路内の飽和沸騰に対し，図1に示す場合を扱う．図1（a）は，水平上向き円形加熱面（直径 l）上にすきまの厚さ s を持って非加熱水平円板（直径 l）が同軸に置かれる場合，図1（b）は，水平円柱加熱面に対し，半径すきま s をもって非加熱垂直平板（厚さ l）が置かれる場合，図1（c）は，垂直加熱面（高さ l）に沿い垂直流路（流路厚さ s）がある場合，図1（d）は，（c）の流路の下端を閉じた場合であり，上記4者とも十分に大きな容器内の飽和液に沈められているものとする．

いま，これらの飽和沸騰系の限界熱流束 q_{c0} が，加熱面寸法 l，流路厚さ s，蒸発潜熱 h_{fg}，液体密度 ρ_L，蒸気密度 ρ_V，表面張力 σ，および重力の作用 $g(\rho_L-\rho_V)$ に関係して定まると仮定（流体粘性の影響は強制流動沸騰の取扱い[4]でも無視した）すると，通常の次元解析から

$$\frac{q_{c0}}{\rho_V h_{fg}}\Big/\sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}} = f\left(\frac{\rho_L}{\rho_V},\ \frac{g(\rho_L-\rho_V)l^2}{\sigma},\ \frac{s}{l}\right) \cdots\cdots (1)$$

さて図1（a）～（d）ともに，$s\to\infty$ の極限を考えると，それぞれ通常の単純なプール沸騰（水平加熱面，水平円柱加熱面，垂直加熱面）の状況となる．この時，加熱面寸法 l が非常に小さい場合を除けば，一般に式（1）右辺 f（　）内の第2項はほとんど影響せず，かつ臨界点近くを除けば ρ_L/ρ_V の影響も小さく，次の形の整理式が成立する．

$$\frac{q_{c0}}{\rho_V h_{fg}}\Big/\sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}} = C\ (\text{一定}) \cdots\cdots (2)$$

次に $s\to 0$ の極限を考えると，図1に示すごとく合体蒸気塊が制限流路内を一方向に流れる状況（加熱面に沿う液体の流れは蒸気塊の流れの方向に一致する要はない）になる．このとき，二相流の安定性に重要な気液界面は蒸気塊の界面で，それは $s\to 0$ の極限で加熱面に平行となる．いま前報[4]に記した方向性次元解

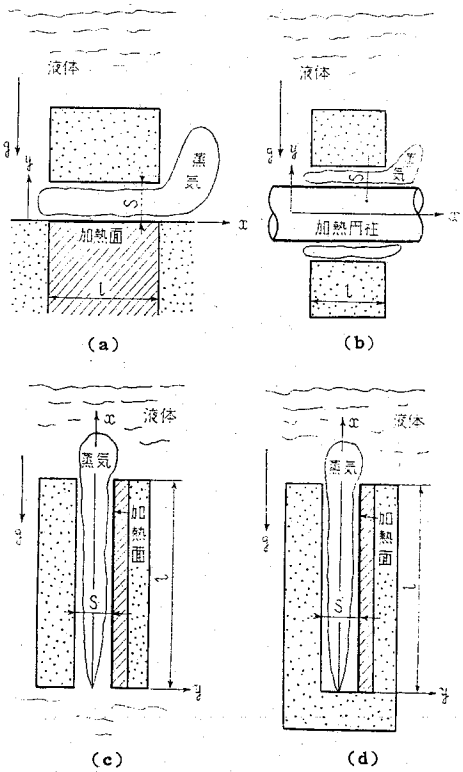

図 1　自然流動沸騰系

* 原稿受付　昭和52年11月22日．
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析を，この気液界面に適用すると，x, y, z 方向の長さの次元を区別した上でなお式（1）が無次元なるためには次式の形が要求される．

$$\frac{q_{co}}{\rho_V h_{fg}} \Big/ \sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}} = \frac{s}{l} F\left(\frac{\rho_L}{\rho_V}, \frac{g(\rho_L-\rho_V)s^2}{\sigma}\right) \cdots\cdots(3)$$

さて，$s \to \infty$ の極限で式（2），$s \to 0$ で式（3）となり得る簡単な形として，次式はその一つになり得る．

$$\frac{q_{co}}{\rho_V h_{fg}} \Big/ \sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}} = C \frac{1}{1+\frac{l}{s}\frac{1}{F[(\rho_L/\rho_V),\ g(\rho_L-\rho_V)s^2/\sigma]}} \cdots\cdots(4)$$

なお，ここで次のことを記しておく要がある．すなわち尾形・佐藤[5]はすでに，図1（c）の系に対し大気圧下の液体ヘリウムの限界熱流束 q_{co} の実験をおこない，そのデータを次の形の実験式で整理している．

$$q_{co} = \frac{1}{K_1 + K_2(l/s)}$$

ここに K_1, K_2 は定数である．本式は，式（4）で $F(\ ) =$ 定数とした場合と同形式である．

3. 式(4)と実験データの比較

3·1 図1（a）の沸騰系　図2は，$l=10\sim11$ mm の円形加熱面に対する大気圧の飽和水の q_{co} の実験データ〔文献（6）の図8のA形，および文献（7）の図11〕を示す．実験データは $l/s=0$ の近くでやや複雑な挙動を示すように見える（今後の研究の検討を要する）が，近似的には同図の直線，すなわち次式であらわされる．

$$\frac{q_{co}}{\rho_V h_{fg}} \Big/ \sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}} = 0.17 \frac{1}{1+0.040\, l/s} \cdots\cdots(5)$$

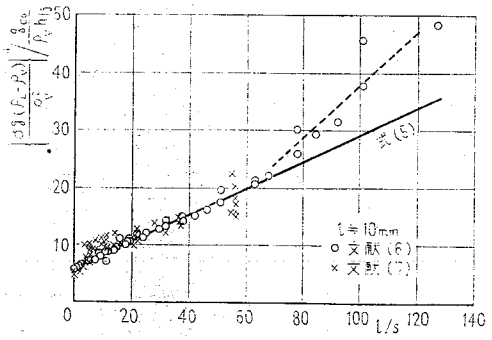

図2　図1（a）の沸騰系の限界熱流束

なお図2において，破線でだいたいの傾向を示すごとく，$60<l/s$ の範囲で実験データが式（5）から偏るのが見られる．しかし，これはすきま厚さ s の絶対値（現在 $l=10$ mm のため $s<0.17$ mm）が非常に小さい範囲で，水がすきま内に流入し難くなるため〔文献（7）参照〕，および恐らく s の測定誤差のためであり，特に無次元量 $l/s=60$ の値に意味があるわけではないと推定される．

〔付記〕　図2，および式（5）は $l \fallingdotseq 10$ mm だけに対する結果である．図1（a）の沸騰系では，l が大になるとすきまから流出する蒸気塊がいくつかに分断〔そのピッチは $\sqrt{\sigma/g(\rho_L-\rho_V)}$ に比例するであろう〕して離脱し，その影響が q_{co} に及ぶ可能性がある．従って式（5）右辺分母に含まれる定数 0.040 は，実は無次元量 $\sigma/g(\rho_L-\rho_V)l^2$ の関数として変化するものであるかも知れないので，今後の検討を要する．

3·2 図1（b）の沸騰系　図3は，外径 12.7 mm の円柱加熱面，$l=12.29$ mm（ただし $l/s=13.9$ の1データのみ $l=6.33$ mm），$s=1\sim0.08$ mm および ∞，大気圧の飽和水の q_{co} の実験データ（Jensen らの論文[8]の図8）を示し，同図の直線は次式で表される．

$$\frac{q_{co}}{\rho_V h_{fg}} \Big/ \sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}} = 0.20 \frac{1}{1+0.068\, l/s} \cdots\cdots(6)$$

なお上記の Jensen らは一つの無次元整理を提示しているが，それは q_{co} が蒸気粘性の全面的な支配をうけ，かつ重力の加速度 g を含まぬ性格のものである．

3·3 図1（c）の沸騰系　この系に対し液体ヘリウムの限界熱流束の実験がおこなわれている[9]．まず図4で，尾形・佐藤[5]の長方形流路の実験データ（○印）は，$l=25\sim400$ mm，$s=1\sim5$ mm，加熱面幅が 10 mm の条件下のものである．これに対し尾形らは，

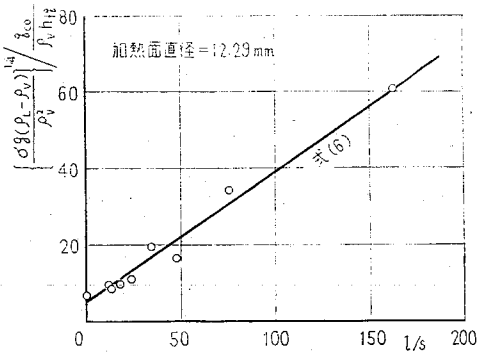

図3　図1（b）の沸騰系の限界熱流束

図4の細い実直線で整理しているが，ここでは $l/s=$ 250～400 付近の データも含めて太い実直線で近似的に整理することとし，それは次式であらわされる．

$$\frac{q_{co}}{\rho_V h_{fg}}\Big/\sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}}=0.12\frac{1}{1+0.012\,l/s} \quad \cdots\cdots(7)$$

なお図4の破線は，Bailey[10] の長方形流路（$l=30$～200 mm，$s=0.8$ mm，加熱面幅 10 mm）の実験結果である．$0<l/s<75$ の範囲を別にすると式（7）にほぼ近い結果を与えている．

一方，図4で Lohongre ら[11]（その論文の図1）のデータのうち，長方形流路のデータ（● 印）は加熱面幅が 40 mm のものである．また二重管流路のデータ（▲ 印）は，内管加熱，外管非加熱，$l=55$ mm のものである．ここで s は次式で定められている．

$s=$（流路断面積）/（加熱面のみの周辺長） ……（8）

しかし内管直径 22 mm に対し，内外管の間の半径すきまが 3 mm 以下であり，近似的に流路厚さ s の長方形流路とみても大きな誤差は生じない．そして図4で，● 印と ▲ 印のデータは鎖線で整理され，それは次式で表される．

$$\frac{q_{co}}{\rho_V h_{fg}}\Big/\sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}}=0.13\frac{1}{1+0.011\,l/s} \quad \cdots\cdots(9)$$

式（7）と式（9）の間のわずかな相違は，加熱面幅（尾形ら[5]および Bailey[10]の 10 mm に対し Lohongre ら[11]の 40 mm ないしそれ以上）の相違に起因するのかも知れない．

なお図4で，Lohongre らの円管流路のデータ（× 印）は，$l=25$～400 mm，管内径 1～5 mm の条件下のもので，s は式（8）で定められている．この時，式（7），（9）の直線の近傍にデータが現れるのが見られる．なお図4の細い実曲線は Lohongre らの与えた実験式を示すものである．

3・4 図1（d）の沸騰系 この系に対し楠田・井村[12]は大気圧下の水，エチルアルコール，ノルマルヘキサン，四塩化炭素を用い，外管加熱，内管非加熱の二重管流路（外管直径 d_0，内管直径 d_i），および円管流路（管直径 d_0）について実験をおこない，かつ半理論的な解析に基づいて無次元整理式を導いている．

ところで上記の実験データ（ここでは飽和沸騰に限るため，サブクール度が 2.5°C 以上のものは除外する）に対し図5のような整理も不可能ではない．ここに s は式（8）によっている．図5をみるに，最右端の水の実験データ（管長の長く半径すきまが非常に小さい条件下のもの）の偏り，また $l=1\,000$ mm と $l=$ 200～300 mm の両データ群の間の多少の分離傾向，およびデータの全体的ばらつきなどの問題があるが，図の実直線でだいたいの整理が可能であり，それは次式であらわされる．

$$\frac{q_{co}}{\rho_V h_{fg}}\Big/\sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}}=0.10\frac{1}{1+0.081\dfrac{l}{s}\left\{\dfrac{\sigma}{g(\rho_L-\rho_V)s^2}\right\}^{0.15}} \quad \cdots(10)$$

上式右辺に含まれる $\{\sigma/g(\rho_L-\rho_V)s^2\}^{0.15}$ は，上記の実験範囲で 0.63～1.27 の値（平均値はほぼ1に等しい）をとり，影響はそれほど大きくはない．この無次元項が，図1（d）の系に 現れるのは，管上端開口部における気液の挙動（向流を生ず）などに関係しているかも知れない．

なお Frea[13]は，大気圧の飽和水，円管流路（$l=$ 88 mm，$d_0=2.5$ mm）について，重力の加速度 g を地上の値の 1～100 倍の範囲で変えた実験をおこなっている．その論文の図3によれば，限界熱流束は $g^{0.4}$

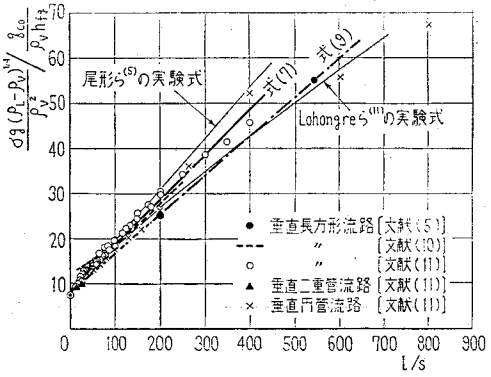

図 4 図1（c）の沸騰系の限界熱流束

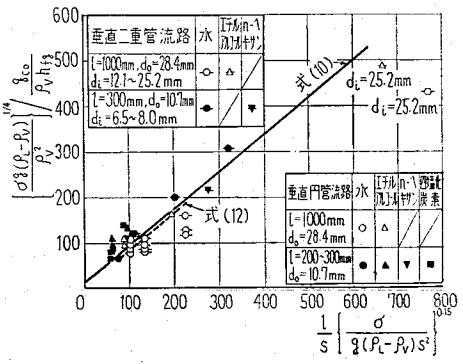

図 5 図1（d）の沸騰系の限界熱流束

にほぼ比例して変化する．そして限界熱流束をフラディングの解析〔実験値は文献(14)〕と関連づけている図12において，上記の重力範囲に属するデータは，$N_{RG}'=1.68\,\phi^{0.8}$ で近似的に表すことができ，これを書き直すと次式のごとくなる．

$$\frac{q_{co}}{\rho_V h_{fg}}\Big/\sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}} =0.42\frac{d_0}{l}\left\{\frac{g(\rho_L-\rho_V)d_0^2}{\sigma}\right\}^{0.15} \left(\frac{\mu_V}{d_0\sigma\rho_V}\right)^{0.1}\left(\frac{\nu_V}{\nu_L}\right)^{0.133} \cdots\cdots(11)$$

ここで，μ_V：蒸気の粘性係数，ν_V：蒸気の動粘性係数，ν_L：液体の動粘性係数である．式(11)右辺の終わりの2個の無次元項は，べき指数の値が小さく影響は大きくない．いま，この2項にFreaの実験条件(大気圧の飽和水，水蒸気の物性値，および $d_0=2.5$ mm)を入れ，かつ式(11)右辺に $d_0=4s$ の関係〔式(8)による〕を入れると式(11)は次のごとくなる．

$$\frac{q_{co}}{\rho_V h_{fg}}\Big/\sqrt[4]{\frac{\sigma g(\rho_L-\rho_V)}{\rho_V^2}} =1.21\frac{s}{l}\left\{\frac{g(\rho_L-\rho_V)s^2}{\sigma}\right\}^{0.15} \cdots\cdots(12)$$

図5の破線は式(12)と，その実験範囲をあらわすものであるが，式(10)と非常に近い結果になっていることは興味深い．

4. 結　　論

3章に掲げた図2～4は，それぞれ1種類の液体，大気圧下だけの結果であり，加熱面の寸法や幅などの実験範囲も限られている．図5については，3·4節に記したような問題点がある．さらには液体粘性がどの程度の影響を持ち得るものか，検討の余地もある．従って式(5)～(7)，(9)，(10)はあくまで暫定的なもので，正確な整理式の決定は今後の研究にまつべきである．

しかし式(5)～(7)，(9)の右辺の値が，$s\to\infty$ の極限において通常のプール沸騰の整理式〔式(2)〕の右辺の値 C〔文献(15)参照〕にだいたい近いこと，あるいは図5で4種類の流体が一応整理されることなど考えると，制限流路内の自然流動沸騰の q_{co} に対し，本報に記した形の無次元整理方式が，少なくとも近似方式として，かなり有効であることが言えそうであり，今後広く検討されることを期待したい．なお2章に記したごとく，本報では l が非常に小(従って l/s も小)なる領域の取扱いは含めていないことを念のため付記する．

本研究に際し，Cryogenics 関係の貴重な資料データを教えていただいた尾形久直氏(日立製作所機械研究所)，文献(8)を教えていただいた 世古口言彦教授(九州大学)，また貴重な実験データを提供していただいた 楠田久男教授(佐賀大学)，井村英昭助教授(熊本大学)に深い謝意を表する．

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B302　垂直二重管流路に於ける自然流動沸騰の限界熱流束

機正　甲藤好郎（東大工）　　機学　黒坂俊雄（東大工院）

制限流路内での自然流動沸騰の限界熱流束は、限界熱流束の基礎的研究として重要であると同時に、超電導装置の設計などにも関係している。しかし、一部（水平二円板間[1]など）を除き実験データは不足しており、研究は不十分である[2]。特に垂直流路に関しては、液体H_e以外の液体での実験は少なく、十分な無次元整理式を得ていない。そこで本研究では、垂直二重管流路に於ける自然流動沸騰の限界熱流束について、数種の液体を用い大気圧近傍で実験を行ったが、以下、その結果得られた3つの領域及び限界熱流束の無次元整理式について報告する。

§2　実験装置および方法

図1，2に示すように、テストブロック周囲に埋め込まれたカートリッヂヒータによって、間接的に電気加熱されるφ10mmの外管加熱（加熱面長ℓ=60mm, 200mm、リーマ仕上面）で、流路内にベークライト非加熱スペーサ（φ6, 7, 8, 9mm）を設ける（スペーサを使用しない実験も行っている）。液体（蒸留水，エチルアルコール，フレオン113）は、テストブロック上方の容器から十分太い管路を通り自然流動により、ほぼ飽和（$\Delta T_{sub} \cong 2°C$）の状態で下方から流入し、加熱面で沸騰し、蒸気柱の様な状態で上方から出る。雰囲気圧力Pは、大気圧が大半であるが、3ataまで変化させた実験も行っている。

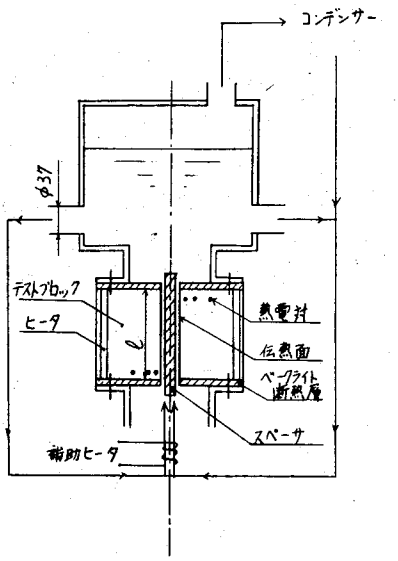

図1　実験装置全体図

実験は、テストブロック温度の定常を待って、ヒータ入力をステップ状に上げる事をくり返し、ブロック温度が定常とならず上昇し続ける事で、限界熱流束に達したと判断した。又、テストブロック材質として、銅及び真ちゅうを用いているが、加熱面長さ方向に熱流束はほぼ一様（実際にはブロック上下で10%程度の差）であり、熱流束の値は、ヒータ入力から検定した熱損失を引いて、熱流束一様として計算した（ブロックの熱効率は大半が85~95%、最低の場合で70%である）。

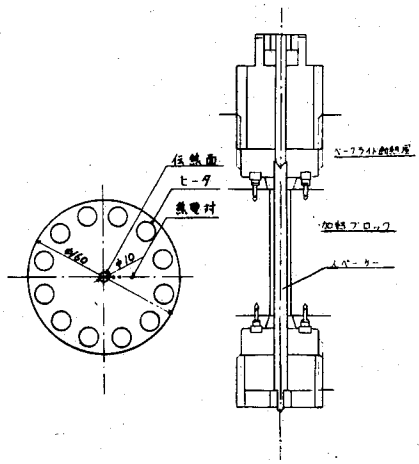

図2　テストブロック及びスペーサ支持図

§3　実験結果と無次元整理

流路の加熱相当すきまΔを次式の形で定義する（図3参照）。

$$\Delta = \text{流路断面積} / (\text{加熱面のみの周辺長}) \qquad (1)$$

そこで、本現象に関係する物理量を選び、次元解析を行うと次式を得る。

$$\frac{q_{co}}{\rho_v h_{fg}} \Big/ \left\{\frac{\sigma g(\rho_\ell-\rho_v)}{\rho_v^2}\right\}^{\frac{1}{4}} = f\left\{\frac{\ell}{\Delta}, \frac{\Delta^2 g(\rho_\ell-\rho_v)}{\sigma}, \frac{\Delta^2 g(\rho_\ell-\rho_v)}{\sigma}, \frac{\rho_v}{\rho_\ell}\right\} \qquad (2)$$

q_{co}；限界熱流束　ρ；密度　h_{fg}；蒸発潜熱　σ；表面張力

ただし、従来の研究から粘性項は無視できると仮定している。又、Δ，ℓに関する3個の無次元数を2個に減らす事は可能である。

図4は大気圧下の実験結果を、$(q_{co}/\rho_v h_{fg}) / \{\sigma g(\rho_\ell-\rho_v)/\rho_v^2\}^{\frac{1}{4}}$の逆数

（同一条件での結果の平均値）と、l/λで整理したグラフである。$l/\lambda \sim 0$の時、q_{c0}はプール沸騰の時の値つまり$\{\sigma g(\rho_l-\rho_v)/\rho_v^2\}^{\frac{1}{4}}/(q_{c0}\rho_v h_{fg}) \simeq 6$である筈だから、$\lambda$が小さくなると急激に$q_{c0}$が小さくなっている。又、変化の仕方は液体，$l$によって異なっているが、図中の実線（原点を通る異なる直線、$q_{c0}l/\rho_v h_{fg}\lambda \simeq$出口蒸気速度が一定に対応）に良く乗る部分が存在している。そこで今、プール沸騰に近く直線からはずれている領域Ⅰ、直線上に乗っている領域Ⅱ、λが小さく直線からはずれている領域Ⅲ、の3つの部分に仮に分けて考える。

この直線上の領域Ⅱでは、直線の傾きと、$l^2g(\rho_l-\rho_v)/\sigma$が関係すると考えられるので、この領域にあると考えられる点の$l^2g(\rho_l-\rho_v)/\sigma$と$(q_{c0}l/\rho_v h_{fg}\lambda)/\{\sigma g(\rho_l-\rho_v)/\rho_v^2\}^{\frac{1}{4}}$との関係を図5に示す。これから、領域Ⅱは次式で良くまとまる事がわかる。

$$\frac{q_{c0}}{\rho_v h_{fg}}\frac{l}{\lambda}\Big/\left\{\frac{\sigma g(\rho_l-\rho_v)}{\rho_v^2}\right\}^{\frac{1}{4}} = 0.658\left\{\frac{l^2g(\rho_l-\rho_v)}{\sigma}\right\}^{0.24} \qquad (3)$$

次に、領域Ⅱ以外の領域Ⅰ，領域Ⅲでは、$\lambda^2g(\rho_l-\rho_v)/\sigma$が関係していると考えられるので、(3)式の定数が仮に$\lambda^2g(\rho_l-\rho_v)/\sigma$の関数であると考え、図6を示す。これから、$\lambda$が小さく$\lambda^2g(\rho_l-\rho_v)/\sigma < 0.14$では、次式で良くまとまる領域Ⅲが存在する事がわかる。

$$\frac{q_{c0}}{\rho_v h_{fg}}\frac{l}{\lambda}\Big/\left\{\frac{\sigma g(\rho_l-\rho_v)}{\rho_v^2}\right\}^{\frac{1}{4}} = 1.17\left\{\frac{l^2g(\rho_l-\rho_v)}{\sigma}\right\}^{0.24}\left\{\frac{\lambda^2g(\rho_l-\rho_v)}{\sigma}\right\}^{0.29} \qquad (4)$$

しかし、通常のプール沸騰に近い領域Ⅰは、図6には、はっきりした傾向は現われず、又、本実験でのq_{c0}の範囲も狭く、整理式を出すに至っていない。

§4 考察

簡略化の為、$q_{c0}l/\rho_v h_{fg}\lambda = v$（出口での蒸気速度を代表する。），$\sqrt{l^2g(\rho_l-\rho_v)/\sigma} = l'$，$\sqrt{\lambda^2g(\rho_l-\rho_v)/\sigma} = \lambda$　とおく。

さて、(3)式を見ると、σが両辺で殆ど消えるので、領域Ⅱはσの影響が極めて小さく、強制流動沸騰系でドライアウトと通称される領域に対応すると考えられる。今、指数0.24を仮に$\frac{1}{4}$として(3)式を変形すると　$\frac{1}{2}\rho_v v^2 = 0.217\, lg(\rho_l-\rho_v)$　の形の興味深い関係式を得る。そこで、$v_0 = \sqrt{lg(\rho_l-\rho_v)/\rho_v}$で定義すると、(3)式は次の形の式になる。

$$v/v_0 = 0.658\left(\frac{\sigma}{\rho_v v_0^2 l}\right)^{0.01} \qquad (5)$$

又、上式と、水平二円板間自然流動沸騰の場合の限界熱流束の整理式[17]とは、σの影響の程度と式の形で大きく異なり、領域Ⅱを見る限り、流路が垂直か水平かでは現象が本質的に異なると考えられる。

次に、(4)式を書き直すと、領域Ⅲの式として次式を得る。

$$v/v_0 = 1.17\left(\frac{\sigma}{\rho_v v_0^2 l}\right)^{0.01}(\lambda)^{0.58} \qquad (6)$$

上式では、(5)式の形の上に$(\lambda)^{0.58}$が加わっており、この領域では領

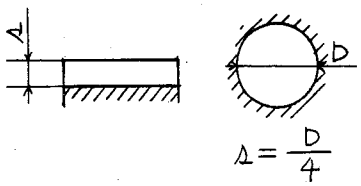

図3　加熱相当すきまλ

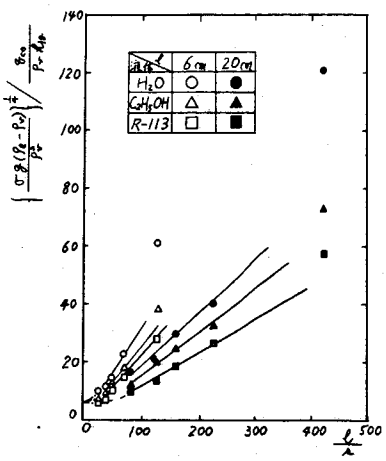

図4　実験結果

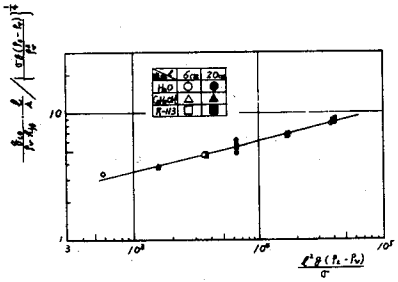

図5　領域Ⅱ

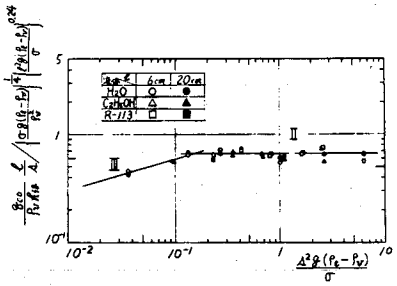

図6　領域Ⅱ，Ⅲ

域Ⅱと異なり、λ'の形でσが大きく現象に関係していると考えられる。その点で、下端を閉じた熱サイフォン系での限界熱流束の暫定的整理式と共通する所があり、出口での気液の挙動が関係しているのかも知れないと思われる。

次に、各領域の境界は、領域Ⅱ、Ⅲに関しては(3)(4)式の交点から$\lambda'=0.37$となるが、プール沸騰に近い領域Ⅰの境界は本実験では不明確である。そこで、非常に暫定的ではあるが、図4から判断して、$\{\sigma g(\rho_\ell-\rho_v)/\rho_v^2\}^{\frac{1}{4}}/(g_{co}/\rho_v R_{fg})<8.5$では領域Ⅰであるとし、式(3)との交点から、近似的に$\lambda'=0.18\sqrt{\ell'}$を境界として得る。ただし、あくまで目安程度の暫定的な境界である。以下、境界をまとめると、

$$\left.\begin{array}{ll} \text{領域Ⅰ} & \lambda'>0.18\sqrt{\ell'} \\ \text{領域Ⅱ} & 0.37<\lambda'<0.18\sqrt{\ell'} \\ \text{領域Ⅲ} & \lambda'<0.37 \end{array}\right\} \quad (7)$$

さて、従来の液体Heの実験[3][4]は、$g(\rho_\ell-\rho_v)/\sigma$の値が他の液体と大きく異なる為（表1参照）、$\lambda,\ell$は同程度でも、$\lambda',\ell'$で考えると本実験と実験範囲はかなり異なっている。今(7)式の領域分けがそのまま適用できるとすると、液体Heのデータは領域Ⅰが中心で、領域Ⅱが多少存在する事になるが、領域Ⅱと思われるデータも一般に(5)式に合致しない。それが、境界分けの(7)式の問題であるのか、(5)式自体に不十分な点があるのか（例えば管径は本実験では変化させていない）は、今後の検討を要する所であろう[5]。

液体	$\sqrt{\sigma/g(\rho_\ell-\rho_v)}$
H_2O	2.50 mm
C_2H_5OH	1.54
R-113	1.00
液体 He	0.307
液体 N_2	1.09
液体 H_2	1.81

表1 各種液体の$\sqrt{\sigma/g(\rho_\ell-\rho_v)}$

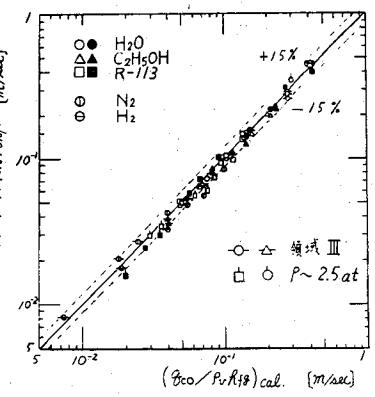

図7 領域Ⅱ,Ⅲのg_{co}に関する計算値と実験値

なお、その他に液体N_2, H_2のデータが若干存在するが、それらのデータ（管径12mm, ℓ=80, 150mm）は、(7)式で領域を分けると、式(5)(6)で良く整理される。又、本実験で圧力を変化させたデータ(3ataまで)も同様に良く整理される。最後に、これら液体Heを除いたデータで、領域Ⅱ,Ⅲにあると考えられる物の$g_{co}/\rho_v R_{fg}$の計算値と実験値を比較したのが図7である。大半が±15%で良く整理されている。

§5 結論

(i)垂直二重管流路（φ10mm外管加熱）に於ける自然流動沸騰の限界熱流束を調べた結果、次の3つの領域がが存在すると考えられる。

領域Ⅰ ；通常のプール沸騰に近い領域。今回の実験では整理式を出すに至らず、今後の一つの課題であろう。

領域Ⅱ ：限界熱流束にσが殆ど影響しないドライアウトの領域(5)式。この領域では、水平流路と現象が本質的に異なると考えられる。

領域Ⅲ ：σの影響が恐らく出口の気液の挙動に関して現われるであろう、λの小さな領域(6)式。

(ii)各領域の境界を(7)式で暫定的に与えたが、今後の検討を必要とする。

(iii)本研究では管径dの変化を行っていない。今後の研究に待つ所である。

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B210　一様加熱垂直円管流路における自然流動沸騰の限界熱流束

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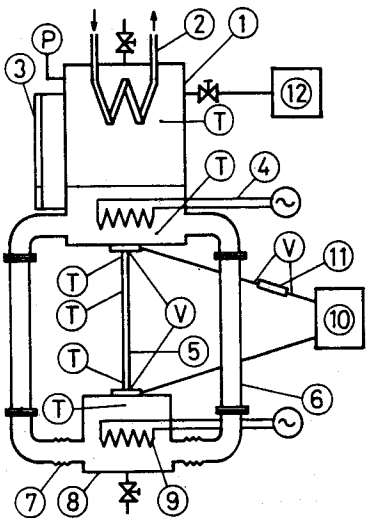
§1 序

制限流路内での自然流動沸騰の限界熱流束は，限界熱流束の基礎的研究として重要であるとともに，超電導装置の設計，原子炉の安全性にも関係している。しかし，一部の沸騰系（水平二円板間[1]など）以外は，通常の単純なプール沸騰系，強制流動沸騰系に比較して著しく実験データが少なく，十分な無次元整理式も得られていない[2]。特に垂直流路に関しては，大気圧下の液体ヘリウム以外の実験は少なく，雰囲気圧力を変化させた実験は，ほとんどないのが実状である。そこで，本研究では，一様加熱垂直円管流路を用い，水及びR12について，雰囲気圧力を変化させ，その限界熱流束の特性を研究した。

§2 実験装置と方法

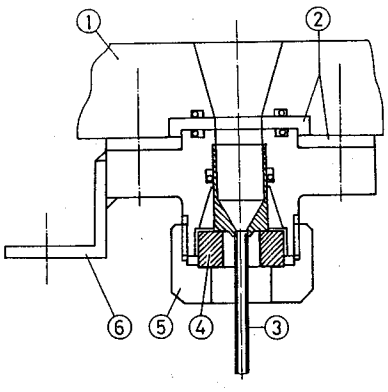
実験装置の概略を図1に示す。液体は自然流動により，ほぼ飽和の状態で下方から流入し，直流直接通電加熱された加熱管内で沸騰する。蒸気はクーラで凝縮し，液体は連結管（内径35.5mm）を通り，下タンクに供給される。下タンクは加熱管長さの変化に応じ，連結管を取りかえることにより，自由に上下方向に移動でき，加熱管入口，出口形状はすべての実験で不変である。なお 加熱管の熱膨張の逃げ，及び取付けの簡便さのために，連結管下部にフレキシブルチューブを設けた。液位は，加熱管出口より25～30cm上方に設定され，系内の雰囲気は，クーラと補助ヒータを併用して調節することにより，所定圧力下の飽和状態に設定される。実験は熱流束をステップ状に上げて加熱管の温度（加熱管外壁面にスポット溶接されたφ0.1mm C-A熱電対により測定）を観察する手法をとって行った。この際に，加熱管温度が定常にならず急激に上昇する点をもって限界熱流束点とした。（すべてのデータは，出口より5～10mm上流側の熱電対が上昇したものである。）加熱管取付部を図2に示す。各加熱管の両端は φ24mmの円筒部が溶接され，標準化されている。電流は押シネジ，加熱管押シを通して供給されており，加熱管押シは2つ割りで，各加熱管に共通である。

本実験の範囲は，水（1ata～9ata），R12（7.6ata～32.9ata），加熱管径 $d=4,\ 6,\ 10$(mm)，加熱管長さ $l=100,\ 200,\ 400,\ 800$(mm)である。また，加熱管には肉厚1mmのステンレスパイプを使用した。



3 液面計	4 補助ヒータ
5 加熱管	6 連結管
7 フレキシブルチューブ	8 下タンク
9 補助ヒータ	10 直流電源装置
11 シャント	12 真空ポンプ
T 熱電対	P 圧力計
V 電圧計	

図1　実験装置概略図



1 タンク	2 テフロン
3 加熱管	4 加熱管押シ
5 押シネジ	6 電極

図2　加熱管取付部

§3 結果

3－1 残留空気の影響

水の実験で当初，下タンク液部内の残留空気のため，熱流束の増加，流動速度の上昇に伴ない空気泡が加熱管に流入し，加熱管入口部の外壁温度の急上昇が観察された。これは本来の限界熱流束とは異なり，これを防止するため系を真空

ポンプで10 mmHg程度に減圧した後水を注入した。但し、その後も加熱管入口部の外壁温度は常に計測したが、温度急上昇は見られなかった。なお、R12の注入の際には、減圧後R12ガスでパージし、再度10 mmHgに減圧後液を注入し、空気の残留を防止した。

3－2　沸騰に伴なう不安定現象　　水の実験において、加熱管長 l が長く、加熱管径 d が太い系で、沸騰に伴なう不安定現象が観察された。即ち、$l=400$, 800 mm × $d=6$, 10 mm の4本の加熱管では密度波振動（振動数 1～15 Hz）が生じ、実験データが得られなかった。また、$d=4$ mm × $l=400$, 800 mm の2本の加熱管については、大気圧近傍の実験では振動が生じたが、2 ata 以上の実験では生じなかった。これは、入口抵抗が大きいほど、また、圧力が高いほど振動が起こりにくいという既知の事実と一致している。密度比の大きいR12では、このような振動は全く観察されなかった。なお、振動により、データが得られなかった場合でも、本実験の実験式（付記参照）で示される値よりも高い熱流束まで熱入力を上昇させた。

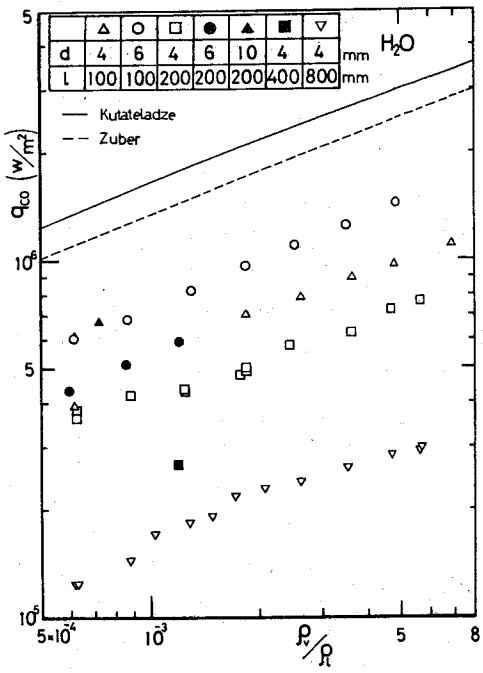


図3　生データ（水）

3－3　実験結果　　実験結果の一部を図3，図4に示す。横軸 ρ_v/ρ_l は圧力の増加に従って単調に増大する。図中，実線及び破線は，それぞれ単純なプール沸騰系において，Kutateladze及びZuberによる無次元整理式から与えられるものである。l/d が大きくなるに従い，この2本の曲線から離れて小さくなっていく。しかし，圧力増加による限界熱流束 q_{co} の変化は，単純なプール沸騰系の場合とかなり似た傾向を示している。（R12において，$d=4$, 6 mm のデータも同様の傾向にある。）図5，図6は，縦軸に上記Kutateladze及びZuberの整理式に表わされた無次元数 $\{\sigma g$

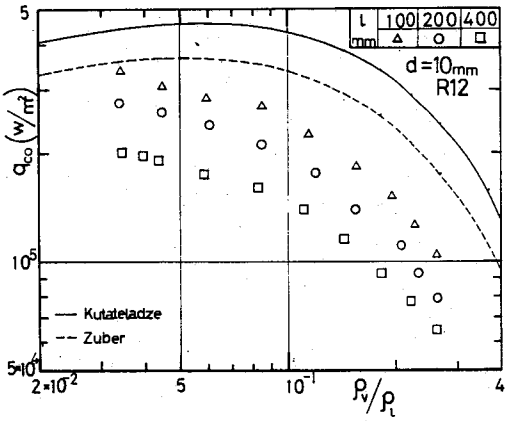


図4　生データ（R12）

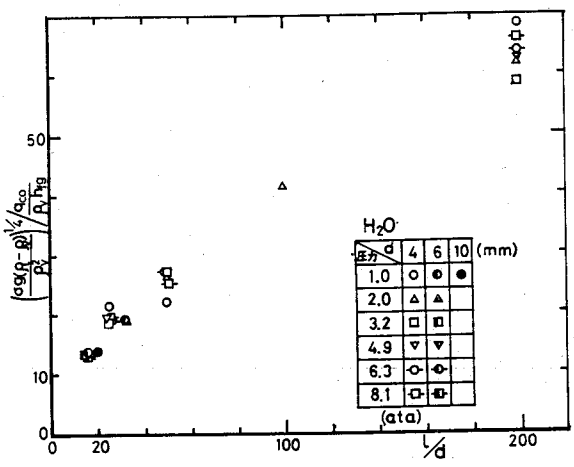


図5　水の限界熱流束

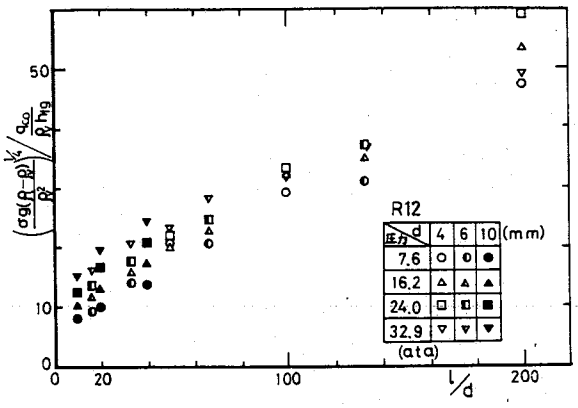


図6　R12の限界熱流束

$\{\sigma g(\rho_\ell-\rho_v)/\rho_v^2\}^{1/4}/(q_{co}/\rho_v h_{fg})$ をとり，l/d の影響をみたものである。両図の傾向は互いに一致する。なお，圧力の影響による変化は，必ずしも規則的ではない。

§4 考察

図7に，大気圧下のヘリウムのデータについて，図5，図6と同様の整理をしたグラフを示す。円管流路以外に，長方形流路，二重管流路でも行なわれているので，次式で定義される等価加熱直径 d_{he} を用いた。

$$d_{he} = 4\times\frac{\text{流路面積}}{\text{加熱面のみの周辺長}}$$

（円管流路では $d_{he}=d$ である）

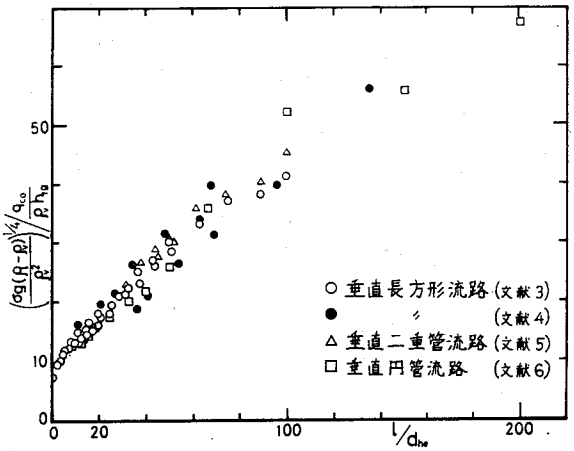


図7 Heの限界熱流束

図5，図6及び図7は，いずれもプール沸騰として，定性的かつかなりのバラツキはあるが，定量的にも一致している。また，ヘリウムの粘度は，本実験の水，R12の値とはかなり異なるが，この比較から粘度の影響は，1次的には小さいと考えられる。流体が，円管などの垂直流路を流れているので，強制流動沸騰的な様相を呈するようにも思えるが，上記の事実から，自然流動沸騰限界熱流束は，強制流動沸騰系とは性格が異なり，本質的に，プール沸騰の限界熱流束特有の性質をもつと考えられる。なお，$l/d\fallingdotseq 0$ 近傍で $\{\sigma g(\rho_\ell-\rho_v)/\rho_v^2\}^{1/4}/(q_{co}/\rho_v h_{fg})$ は，ほぼ一定値になるべきように思える（垂直加熱面であるから，$d\to\infty$ $(l/d\to 0)$ で l の影響が考えられるが，ここでは触れない）が，R12で $l/d=10$ の実験においても，かなりのバラツキを見せる。その原因は不明である。

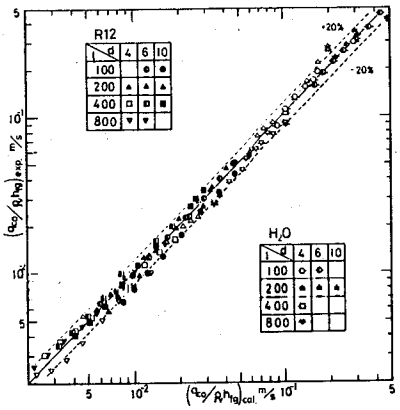


図8 実験値と計算値の比較

このような自然流動沸騰限界熱流束を考える上で，直接的な発生機構の問題以外に，次のような自然流動沸騰特有の影響因子を考慮する必要があろう。即ち，加熱管上下の形状（筒状の非加熱部が上部にあれば，いわゆる煙突効果を生じ，q_{co} を増大させ，下部にあれば入口抵抗となり，q_{co} を減少させる。），降流部分の抵抗の影響，及び加熱管出口での逆流の可能性などである。これらの因子の影響については，今後の研究に待つ所である。

〔付記〕 チャンネルフローの観点から，本実験結果を次のような無次元式でまとめることもできる。

$$\frac{q_{co}}{\rho_v h_{fg}\bar{v}} = C/\left(\frac{l}{d}\right) \qquad \text{ここに } C=0.185\ (\text{水}),\quad C=0.220\left(\frac{\rho_v}{\rho_\ell}\right)^{-0.186}\left(\frac{d^2g(\rho_\ell-\rho_v)}{\sigma}\right)^{-0.248}\quad (\text{R12})$$

但し，$\bar{v}=\sqrt{lg(\rho_\ell-\rho_v)/\rho_v}$，$q_{co}$；限界熱流束，$h_{fg}$；蒸発潜熱，$\sigma$；表面張力，$\rho$；密度

上記の実験式は図8に示すとおり，多くのデータについて±20%程度で合っている。

§5 結論

一様加熱垂直円管流路において，水，R12について自然流動沸騰の限界熱流束を雰囲気圧力を変えて求めた結果，自然流動沸騰限界熱流束は，プール沸騰系としての特性を持ち，ヘリウムのデータとも，その傾向は一致した。

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強制流動沸騰

外部流

536.423.1::532.526

寄　書

液流中に平行においた一様加熱平板上の強制流動沸騰の限界熱流束*

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Yoshiro KATTO, Chiyoharu KURATA

1. 序

強制流動沸騰限界熱流束というと，一般には流路内の内部流の場合を指すのが普通で，研究もこれに集中しているといって過言ではなく，いわゆる外部流の場合の限界熱流束（CHF）の研究は非常に少ない．しかし外部流に属している基本的な沸騰系の CHF は，技術的な応用目的に加え，強制流動沸騰の CHF 機構の考察にも関連して重要と思われる．

さて加熱面上を液膜が流れる所の最も簡単な沸騰系に関しては，甲藤ら[1][2]は一様加熱円板面の中心に小さな円形噴流（飽和液およびサブクール液）がある系に対し CHF の実験とデータの無次元整理の研究をおこない，他方，Lienhard ら[3][4]は上記のデータを基に理論的な解析をしている．また甲藤・石井[5]は一様加熱平板上を飽和平面噴流が流れる系に対し CHF の実験とデータの無次元整理をおこなっている．

一方，加熱面が十分に大きな液体中に沈められた形のものに関しては，Andrew-Mohan Rao[6]は水中噴流に帯状加熱面を直交しておいた場合の CHF の実験をしている．また例えば McKee-Bell[7] などは流れに直交する円柱加熱面上の CHF の実験をおこなったが，その後 Lienhard-Eichhorn[8] は飽和液の流れに直交する円柱面上の CHF について多くの実験をおこない，かつ理論解析に基づくデータの無次元整理をおこなった．

ところで，加熱面が液中に沈められた系のなかには，もっと簡単かつ基本的な形として，一様加熱平板が液流に平行におかれる場合の CHF の問題がある．そこで本報では，この沸騰系に関し大気圧下の飽和水，飽和 R-113 を用い実験をおこない，データを無次元整理した．なお内管が一様に加熱される二重管流路の場合，l/d_{he}（ここに，l：管長，d_{he}：加熱相当直径[9]）が非常に小さい時は本報で扱う系の状況に近くなる〔文献（9）の 4 章（I）項参照〕．そこでこの見地から，この状況下の二重管の既存の実験データ（アセトン，トルエン，モノイソプロピルビフェニル，水）についての検討もおこなった．

2. 実験装置

図 1 は，実験装置の主要部である試験容器を示す．銅ブロックは，左方のみぞに 11 枚の板状電気ヒータを入れて加熱し，右端の長方形加熱面（奥行き 10 mm，流れの方向の長さ l=10，15，20 mm の 3 種）上で沸騰が生ずる．銅ブロックの軸心上にセットした 3 個の熱電対によって，加熱面をよぎる熱流束 q，および CHF の発生（温度の急上昇を伴う）の検出をおこなった．試験容器（図 1 には省略してあるが補助加熱器を内蔵）内には，ほぼ一定量の大気圧飽和液がせき（堰）によって貯えられ，そしてうず巻ポンプ，流量計，予備加熱器を経て供給される飽和液が長方形ノズル（奥行き 15 mm，出口高さ s=5，10 mm の 2 種類）から流出，加熱面に沿う強制液流を作る．ポンプ容量からの制限で，上記のようなノズル流を用いたが，後述のごとく s=5，10 mm による CHF の相違はほとんどみられ

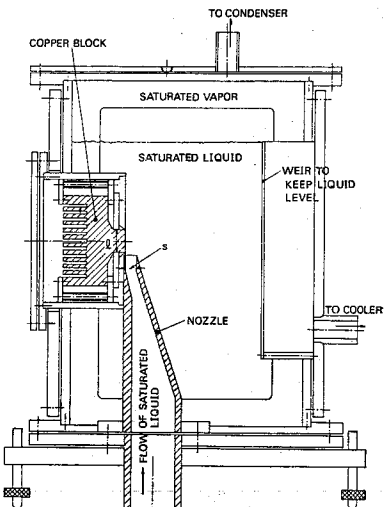

図 1　試験容器

* 原稿受付　昭和 55 年 3 月 5 日．
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ず，従って近似的に加熱面は一様流中にあるとみてよいであろうと考えている．

3. 実験結果の整理

実験は水，R-113 共にノズル出口液体速度 u=1.5～10 m/s の範囲でおこなわれた．瞬間写真観察によると，加熱面近傍の二相流の流動様式は気ほう流で，なお高熱流束になると加熱面近傍で蒸発合体を生ずるようになっている．そして CHF の全測定データを q_{c0}/GH_{fg} 対 $\sigma\rho_l/G^2l$ の形で示すと図2が得られる．ここに，q_{c0}：飽和液の流れに対する CHF，G：ノズル出口からの液体の質量速度，H_{fg}：蒸発潜熱，σ：表面張力，ρ_l：液体密度，l：流れの方向の加熱面長さである．

さて図2において，水と R-113 のデータはそれぞれほぼ同じこう配の直線に沿う形で現れるが，s=5 mm と s=10 mm の場合を区別してみると，s=5 mm の場合のデータの方が s=10 mm のそれよりこう配がわずかに小さく現れているようにもみえる．しかし，その相違自体非常に小さく（仮にあったとしても），またその影響を論ずるに十分なデータもないから，ここでは s による相違はないとみなして扱う．

ところで図2の状況は，以前の研究[1][5]で出会った状況とまったく同じであり，気液密度比 ρ_v/ρ_l によって水と R-113 の両データを合一させることが考えられる．すなわち q_{c0}/GH_{fg} の代わりに $(q_{c0}/GH_{fg})/(\rho_v/\rho_l)^{0.559}$ を使うならば，図2で分離していた水と R-113 の両データが集合し図3の結果を与えることになる．そして図3に示す直線から次の無次元整理式が与えられる．

$$\frac{q_{c0}}{GH_{fg}}=0.186\left(\frac{\rho_v}{\rho_l}\right)^{0.559}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.264} \quad \cdots\cdots(1)^{*1}$$

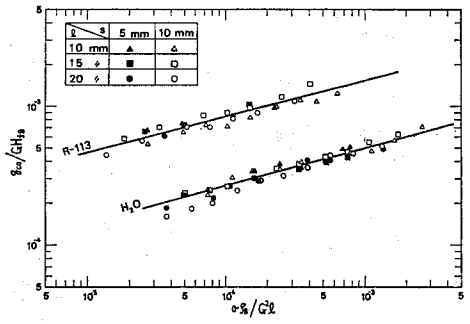

図 2 実験データ

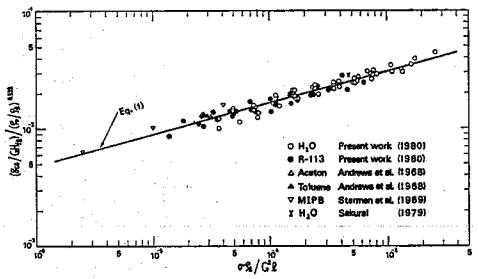

図 3 実験データの無次元表示

4. 二重管の CHF との比較

l/d_{he} の小さい条件下の内管一様加熱の二重管に関し，Andrews ら[10]は大気圧のアセトンおよびトルエンに対し，また Sterman ら[11]は 2.0 bar 下のモノイソプロピルビフェニルに対し，それぞれ実験をおこない，限界熱流束と入口サブクール，エンタルピ ΔH_i の関係を与えている．従って $\Delta H_i \to 0$ に外そうすることにより飽和流の場合の限界熱流束 q_{c0} が容易に求められる．また桜井[12]は，1.43 bar 下の飽和水に対し CHF を求めている．これらの内管加熱二重管の実験条件における ρ_v/ρ_l の値を本報の実験条件下の ρ_v/ρ_l の値と共に表1に示してある．なお表1には二重管の l/d_{he} の値をも示してある．

さて図3には，すでに二重管のデータも図示されているが，それらは式（1）の予測値よりわずかに高目に出る傾向がみられるものの，本の実験データのばらつきの範囲内にあることがわかるであろう．

5. 結　論

飽和液体の一様流に平行におかれた平板加熱面上の限界熱流束 q_{c0} に対し無次元整理式（1）を導いた．本式との比較をおこなった l/d_{he} の小さい内管加熱二重管のデータをも含めて考えると，表1および図3に示す実験範囲は ρ_v/ρ_l=0.006 24～0.010 3，$\sigma\rho_l/G^2l$ =2×10⁻⁶～3×10⁻³ である．

なお，平板加熱面上を飽和液体が非常に薄い膜状で

表 1 限界熱流束のデータに対する ρ_v/ρ_l の値

文　献	流　体	圧　力	$\rho_v/\rho_l\times10^2$	l/d_{he}
Andrews ら[10]	アセトン	1 atm	0.291	1.15
〃	トルエン	1 atm	0.385	1.15
Sterman ら[11]	MIPB*	2.0 bar	1.03	7.05
桜　井[12]	水	1.43 bar	0.086 8	0.058 2
本　報	水	1 atm	0.062 4	—
本　報	R-113	1 atm	0.490	—

* モノイソプロピルビフェニル

*1 本式の定係数 0.186，および指数 0.559，0.264 は図2の実験データから最小二乗法で求めた値である．

流れる場合の限界熱流束に対し，甲藤・石井[5]の与えた無次元整理式を記すと

$$\frac{q_{c0}}{GH_{fg}}=0.0164\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}$$

これを，本報の式（1）と比べると，$\sigma\rho_l/G^2l$ にかかる指数の相違は大きくないが，ρ_v/ρ_l にかかる指数の相違は相当に大きい．換言すれば膜状流で加熱面が覆われる時は ρ_v/ρ_l の影響は非常に弱く，一様流に加熱面が触れる時は ρ_v/ρ_l の影響が大きい．そしてこの事は以下の事柄を考えると自然であるようにも思われる．すなわち，膜状流の場合は加熱面から発生蒸気が周囲空間へ容易に逃げ去り得るのに対し，一様流の場合は蒸気が加熱面近傍に拘束されやすい．従って ρ_v/ρ_l の影響が CHF に対し大きく作用するのかも知れない．ただし上記はすべて推測にすぎず，その当否は今後の研究を待たねばならない．

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536.423.1 : 532.5

バーンアウト機構の研究*

(高バーンアウト熱流束の沸騰系について)

甲藤好郎**, 国広昌嗣***

1. 序　論

さきに甲藤, 横谷ら[1]~[4]は, プール沸騰におけるバーンアウト(および遷移沸騰)に関する実験的研究をおこない, この場合のバーンアウトは一般に次の機構で生ずることを報告した. すなわち, プール沸騰バーンアウトは, 加熱面に接して核沸騰を生じている液層の消耗(蒸気発生により液体を消耗)と, 液層上の蒸気塊の周期的挙動(液層からの蒸気を受けて生長を続けた後に切断浮上する)に関連した現象であり, 核沸騰液層の消耗時間と蒸気塊切断周期とがひとしくなる点でバーンアウトが生ずる. なぜなら, 加熱面への液体補給は, 蒸気塊切断に伴い自動的におこなわれるのみで, 蒸気塊が加熱面上に停留中は液体補給がさえぎられるからである. そして上記のバーンアウト点を越えれば, 蒸気塊の切断浮上がはじまる以前に核沸騰液層が消耗し, したがって蒸気塊切断周期内に加熱面乾燥期間が含まれるようになるので, 熱流束(実験では時間的平均値が測定される)を減じて遷移沸騰にはいる.

上記によれば結局, 通常のプール沸騰バーンアウトは, 蒸気塊の周期的挙動に関連する加熱面上での液体の補給, 消耗の間のバランスから来る限界問題である. そして加熱面上に存在している核沸騰自体にバーンアウト要因が内在するわけではない. そうである以上, 蒸気塊の挙動や, 加熱面への液体補給を人工的に制御することによって, もっと高いバーンアウト熱流束(つまり核沸騰域の拡大)を実現することが可能であり, 場合によっては, 少なくとも液体過熱の許容限度までバーンアウト現象を除去できる可能性すら存在する. しかも同時に, そうした特殊沸騰系の研究は, 核沸騰やバーンアウト機構の全般的な解明に役だつだけでなく, 従来とかく独立に取扱われがちであったプール沸騰バーンアウトと強制対流沸騰バーンアウトの間に橋をかける一助ともなりうるであろう.

本報は, 上記のような見地からおこなった実験的研究であって, 二つの部分からなっている. 第1は, 蒸気塊の挙動の制御効果に関するもので, 機械的な方法によって蒸気塊を破壊し, 加熱面上での蒸気塊停留時間の短縮だけを企図する場合の研究である. 第2は, 加熱面への液体補給の制御効果に関するもので, 小噴流によって加熱面に強制的な液体補給をおこなう沸騰系についての研究である. ただし, 上記の二つの場合とも, できうる限りプール沸騰における核沸騰状態はそのままに保存し, 目的とする制御だけを加える形で, その効果をみる方法をとっている. さもないと, めいりょうな分析ができないからである.

2. 蒸気塊の挙動の制御効果

2·1 実験装置　図1に図式的に実験装置を示す. ここで沸騰容器①の中央におかれた銅ブロック②(上半部が円すい形, 下半部が円柱形)の上端面が直径10 mm の円形の加熱面になる. 加熱面の仕上は, 紙やすり0番で研摩後, アセトンおよび純水で洗浄したものである(一定の実験を経過するごとに仕上げ直す). 銅ブロックの下半部には電気加熱器③(11枚の板状加熱器を銅ブロックみぞにそう入したもの)を内蔵し, 加熱面の熱流束が 2×10^7 kcal/m²h まで可能な容量を持っている. また銅ブロックの上半部の軸心上には4個の銅-コンスタンタン熱電対④があり, 温度分布の測定から加熱面温度 T_w および熱流束 q が求められる. ここで熱流束の計測にあたって必要な銅ブロックの熱伝導率の検定については後記する.

実験は大気圧の純水の飽和沸騰についておこなうが, 補助加熱器⑤によって沸騰容器内の飽和状態が確保されるようになっている. 沸騰による発生蒸気は冷却器⑥で凝縮させて沸騰容器にもどし, 容器内の水位を一定に保持する. なお, 前記の熱流束 q の計測のため, 次のような検定実験をおこなっている. すなわち高熱流束の核沸騰状態において加熱面から発生する蒸気を収集し, その蒸発潜熱から求められる伝熱量, お

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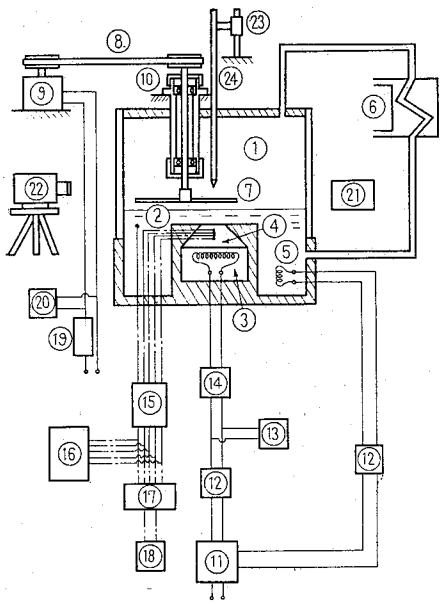
よび銅ブロック内の温度分布から，銅の熱伝導率を決定した．発生蒸気の蒸発潜熱から求められる伝熱量を，銅ブロック電気加熱器入力と比較してみると，両者は非常に近く，入力のうち外部への熱損失はわずか5％以内であった．したがって本報における熱流束の計測誤差は，たかだか数パーセント以内とみてよい状況である．

さて，図1の⑦は機械的に蒸気塊を破壊する目的の回転体であって，直径 2 mm，長さ約 50 mm の金属棒を十文字状に組んだものである．したがって回転体の1回転あたり4本の金属棒が次々に加熱面上を通過していくことになるが，この回転体はベルト⑧を介して直流モータ⑨で駆動され，回転数はしゅう動抵抗器⑲で調整される．回転体の軸は，2個のラジアル玉軸受を持つ軸受箱⑩で支持され，軸受箱を上下に移動させることによって，回転体と加熱面の距離を調整することができる．なお軸受箱の回転軸すきまには，蒸気の漏入防止のためにOリングによるシールがほどこされているものの，軸受箱から沸騰容器へ潤滑油がはいる恐れを完全になくすため，玉軸受に対し潤滑油はいっさい供給せずに運転した．



① 沸騰容器　② 銅ブロック　③ 電気加熱器
④ 熱電対　⑤ 補助加熱器　⑥ 冷却器
⑦ 回転体　⑧ ベルト　⑨ 直流モータ
⑩ 軸受箱　⑪ 定電圧装置　⑫ スライダック
⑬ 交流電圧計　⑭ 電力計　⑮ アイスボックス
⑯ 自動温度記録計　⑰ 切換スイッチ
⑱ mV 計　⑲ しゅう動抵抗器　⑳ 直流電圧計
㉑ 照明装置　㉒ カメラ　㉓ トラバース装置
㉔ 水位計測針

図 1　実験装置（蒸気塊の挙動制御）

2・2 実験の特性　実験は，加熱面から自由水面までの水位 $H=5$ mm および 3 mm の低水位，かつ図1に示すごとく自由水面の上方へ回転体を置く状態でおこなう．これは回転体に対して，水を強制かくはんする作用は与えず，単に蒸気塊の純粋な破壊だけをおこなわせるためである．すなわち，低水位の核沸騰においては，薄い液膜に包まれた形の蒸気塊が周期的に加熱面上に高くもりあがってくる〔文献（2）の図8の写真参照〕．この液膜に包まれた蒸気は，液膜の表面張力などのため一般に大気圧よりやや高い圧力を持っている．そして，もりあがってくる液膜頂部が回転金属棒に触れると，そこで破れ，蒸気が上方へ噴出しつつ蒸気塊は崩壊をはじめる．この液膜破断に伴う蒸気噴出は，高速カメラによる観察において見られる．

なお上記の目的で低水位沸騰を採用したが，これに関連して次のことがらを付記しておくことはむだでないと思われる．すなわち，甲藤・横谷ら[2][3]によれば，直径 10 mm の水平円形加熱面上の大気圧の水の飽和沸騰の場合，高熱流束核沸騰域（$q \fallingdotseq 5\times10^5$ kcal/m²h 以上）の沸騰曲線については，水位 $H=2$ mm 以上で，水位による相違が存在していない．これは，加熱面上の比較的薄い核沸騰液層内で沸騰がおこなわれるからである．またバーンアウトの機構自体についても，水位による質的な相違は存在しない．念のため本報の実験装置でも，これらの点について慎重な追試をおこなったが，上とまったく同じ結果を得た．したがって低水位による実験でも，現在の目的に対し，それほど特殊な条件になるものではないと考えられる．

また図1の実験装置において，軸受箱⑩は沸騰容器①から一応切り離して独立に固定するようにしている．しかし蒸気の漏れ防止のため，軸受箱と容器壁の間にシールをほどこしてあり，それを通して不可避的に多少の振動が沸騰容器に伝わる．この問題に対し，予備実験の結果によれば，本報の実験範囲の回転数に対し，回転体回転時（ここでは蒸気塊から離して回転させる）の沸騰曲線，バーンアウト熱流束ともに，回転体静止時のものとまったく変わらなかった．したがって上記の程度の振動自体は，実験結果になんら影響を与えないと考えてよい状況である．

2・3 実験結果　加熱面から自由水面までの水位 H，回転体と加熱面の間隔 s（回転体は常に水面の上方にあるので $H<s$）を固定した状況下において，バーンアウト熱流束 q_c が回転体の回転数 n によって変化するようすを図2に示す．ただし同図の縦軸には，同じ加熱面，同じ水位において $n=0$（つまり自由沸騰状態）のときのバーンアウト熱流束の測定値 q_{c0} を基準

にした比 q_c/q_{c0} をとり，また横軸には，回転金属棒が加熱面上に到来する周期 t_r（すなわち回転の1周期の1/4）をとってある．

図2は，回転体の回転によって，バーンアウト熱流束が自由沸騰状態の場合より明らかに増加することを示している．このバーンアウト熱流束の増加に伴い，もちろん核沸騰領域の沸騰曲線は高熱流束のほうへ延長されるわけであるが，図3（H=5 mm，s=8 mm の場合の沸騰曲線を例示）にもみるごとく，一般に自由沸騰状態における沸騰曲線が，そのまま高熱流束のほうへ延長される形の特性をもつものである．

2・4 考察 図2によれば，大まかな一般傾向として，（1）回転体と加熱面の間隔 s が小さい場合，（2）回転棒の到来周期 t_r が短い場合，（3）水位 H が高い場合に，バーンアウト熱流束が高くなる．そしてこれは一応，次のごとく解釈することができる．すなわち，s や t_r が小なるほど，蒸気塊の持続周期は短くなり，それだけ加熱面への液体補給ひん度が増す．したがって核沸騰液層の消耗速度を大きくとれるからバーンアウト熱流束が高くなる．また H が大なるほど，加熱面への液体補給時の液体侵入力が強くなるなどの関係で有利である．

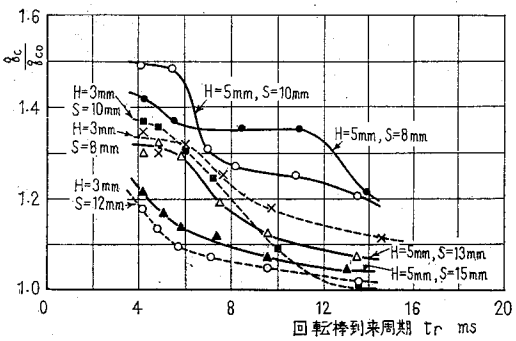
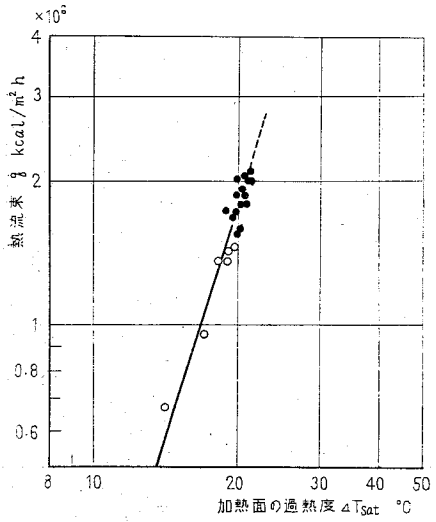


図 2 バーンアウト熱流束 q_c の変化

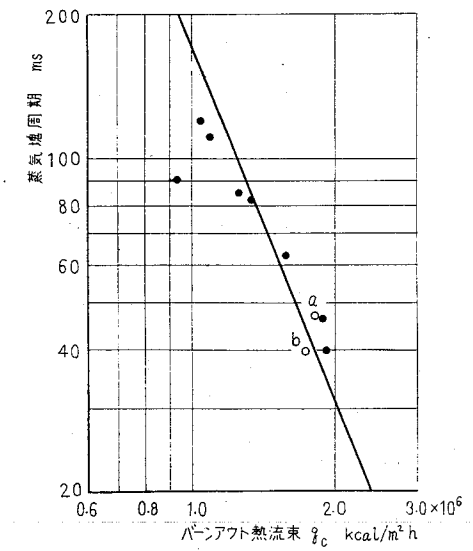


（○印：自由沸騰状態，●印：回転体による核沸騰領域の拡大）

図 3 核沸騰域の沸騰曲線

上記の解釈は，水位 H の影響を別問題とすれば，バーンアウト機構に関し，通常のプール沸騰の場合と本質的な相違を持たぬものである．そして実際，本報の実験は，機械的方法によって蒸気塊の持続周期の短縮をはかっているだけのものであるから，上記の解釈だけで十分のごとくみえる．事実，図4は蒸気塊周期とバーンアウト熱流束の関係で，図中の● 印は通常のいろいろなタイプのプール沸騰について得たデータ[2][3]であるに対し，○印は本報の実験で得たデータを示すものである．データ全体が，一つの統一的な傾向のなかにはいっている．しかしながら，図2が示すごとく，現在の機械的方法ではバーンアウト熱流束 q_c の増加がわずか 1.5 倍程度までしか達しえず，頭打ちになるという重要な問題がある．したがって上記の解釈だけでは欠けるものがあり，もう少し具体的現象をみてみる必要がある．

この問題に関連して，まず容易にわかる単純なことがらがある．すなわち加熱面上への回転棒の到来周期 t_r をいかに短くしても，回転棒と加熱面の間隔 s（水位 H より小にできない）が蒸気塊の持続周期を規制してしまう．なぜなら，少なくとも回転棒の高さまでもりあがってくる間，蒸気塊の破壊はけっしておこなわれないからである．もちろん，この場合，s の値，ま



実験a：H=5 mm，s=12 mm，t_r=7.15 ms
実験b：H=3 mm，s=10 mm，t_r=6.82 ms

図 4 蒸気塊周期とバーンアウト熱流束の関係

た同時にHの値を小ならしめることが考えられるが，すでに述べたように加熱面への液体侵入力の弱化の問題を伴う．

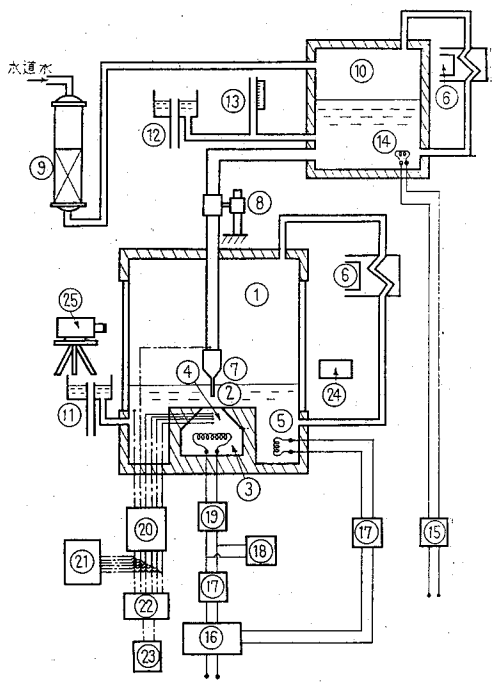
次に，回転棒の到来周期 t_r と，動的な流体挙動とに関連して，本質的に重要なことがらが存在する．すなわち，いま1個の回転棒が蒸気塊の頂部にふれて，そこの液膜を破ったとしても，そこで直ちに蒸気塊が消滅するわけでない．それ以後，2·2 節に述べたような蒸気の噴出をはじめ，さまざまの状態変化に対応する液体の運動を伴いながら時間が経過していくのである．そして，この流体挙動は自然のままに放置されているから，もはや回転棒の回転とは無縁である．したがって，回転棒の到来周期 t_r が小なるとき，あとから次々に回転棒が加熱面上に到来，そこを通過していっても，まだ依然として蒸気塊のほうは，その崩壊過程が独自の速度で経過中という状況におかれる．そして，この状況は高速カメラによる観察でも明白にみられた．また図4において，○印の実験a，b の蒸気塊周期が 40～50 ms 程度であるに対し，回転棒の到来周期 t_r は 7 ms 程度である事実も，主として上記のことがらに関連していることである．結局のところ，回転棒の到来周期 t_r を減少させても，それに対する流体挙動の対応追随は鈍感であり，かつ最終的には追随限界をもっていて，しかもそれがバーンアウト限界を支配する．そして，回転棒でバーンアウト熱流束の増加をはかる場合，その増加の度合に対し，水位 H，つまり加熱面への液体侵入力が影響するという事実も，上記のことがらに関係して意味を持つことになる．

要するに本節の実験は，バーンアウト熱流束を大きく上昇させるという目的に対し適当なものでない．しかし，次の諸件を示している点で意味がある．第1は，蒸気塊は加熱面への液体補給をさまたげるというネガチブな役割を持つものであり，事実，蒸気塊を破壊することによって現実にバーンアウト熱流束が増加する．第2は，蒸気塊は単独に挙動するものでなく，他のあらゆる流体挙動と関連を持つ．したがってバーンアウト熱流束の大幅な上昇のためには，蒸気塊だけの側からの制御は役に立たない．そして上記の目的に対し，とるべき道は，端的に加熱面上へ液体を強制的に補給することである．

3. 加熱面への液体補給の制御効果

3·1 実験装置　前章と同じく，実験は大気圧の純水の飽和沸騰についておこなう．そして使用した実験装置を図5に示すが，前章の装置と特に異なる点は，回転体の代わりに噴流装置を持つことである．すなわち，加熱面の中心上方にノズル⑦を置く．これは，内径 0.7，1.165，1.60 mm の3種類の細いステンレス管を使用し，トラバース装置⑧によって，加熱面とノズル出口の間の高さが変えられるようになっている．そして，純水製造装置⑨，沸騰タンク⑩を経て供給される飽和水の噴流がノズル出口に形成される．沸騰タンクからノズルにいたる流路はグラスウールで保温してあり，噴流の出口温度は最悪の場合でも 97°C 以上であった．したがって，現在の実験状況に対して，少なくとも近似的に飽和水とみなして十分であろう．なお，沸騰容器①内の水位を一定に保つため，オーバフロー装置⑪による排水をおこなう．沸騰容器内の飽和状態は，補助加熱器⑤によって確保されるが，特に加熱面上の自由水面なしの場合（噴流水だけを加熱面に与える場合）の実験においても，この補助加熱器で蒸気を発生させ，沸騰容器内全体を飽和温度に保つ．

沸騰タンク⑩もまたオーバフロー装置⑫を持ち，タンク内の水位が一定に保たれる．そしてノズルからの噴流は，沸騰タンク，ノズル出口間のヘッド差によって



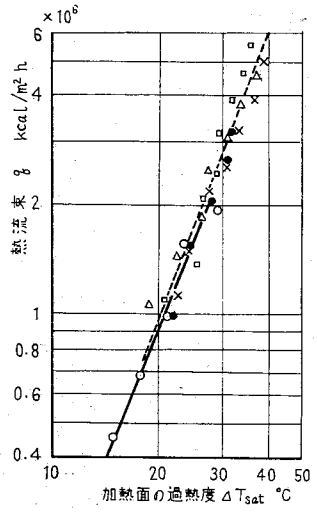
① 沸騰容器　② 銅ブロック　③ 電気加熱器
④ 熱電対　⑤ 補助加熱器　⑥ 冷却器
⑦ ノズル　⑧ トラバース装置　⑨ 純水製造装置
⑩ 沸騰タンク　⑪ オーバフロー装置
⑫ オーバフロー装置　⑬ 水位計
⑭ 電気加熱器　⑮ スライダック　⑯ 定電圧装置
⑰ スライダック　⑱ 交流電圧計　⑲ 電力計
⑳ アイスボックス　㉑ 自動温度記録計　㉒ 切換スイッチ
㉓ mV 計　㉔ 照明装置　㉕ カメラ

図 5　実験装置（液体補給制御）

形成させる．また噴流の流量測定は次の方法による．すなわち沸騰タンクに純水の供給を一時停止し，オーバフロー装置からの排水がない状態を確認後，水位計⑬で沸騰タンク内の水位の降下速度を測定する．そして沸騰タンク内寸法は前もって検定してあるから，流量が算出される．この場合，沸騰タンク内におこる水位降下は，沸騰タンクからノズル出口までの全ヘッド差に比し小さい範囲にあるようにする．したがって，上記の水位降下に基づく噴流流量の変化は非常に小さく無視できる．

3・2 沸騰曲線その他の特性 実験は，加熱面から自由水面までの水位H，ノズル出口と加熱面の間隔h，噴流の流量G，および噴流のノズル出口平均流速u_eを変えておこなう．

上記の実験において，加熱面は強制的な液体の流れにさらされることになる．しかし，ここで扱うような高熱流束の核沸騰では，加熱面上の核沸騰液層内に生ずる非常に激しい沸騰運動自体が支配的である（2・2節に記した，水位$H=2$ mm 以上で沸騰曲線に水位が影響しないという事実も，これに起因する）．したがって本実験の自然流下噴流程度では，強制対流効果はほとんど加わらないと予想される．そして，その確認のため，前もって各種の条件下の実験において沸騰曲線を求め，それらの比較をおこなったが，図6はその代表例である．ここで ○ 印のデータが噴流なしの普通の自由沸騰状態，その他のデータは噴流がある場合の沸騰状態で，特に ● 印のデータは，加熱面上に自由水面が存在せず，噴流水だけが加熱面に与えられるものである．要するに，噴流の有無が沸騰曲線に対しほとんど有意の差を与えないとみてさしつかえないのである．



○：自由沸騰状態（$H=5$ mm）
△：$H=5$ mm，$h=3$ mm，$G=58.0$ g/min，$u_e=2.44$ m/s
□：$H=5$ mm，$h=3$ mm，$G=169$ g/min，$u_e=2.64$ m/s
×：$H=5$ mm，$h=3$ mm，$G=246$ g/min，$u_e=2.04$ m/s
●：$H=0$ mm，$h=3$ mm，$G=62.4$ g/min，$u_e=2.63$ m/s

図 6 核沸騰域の沸騰曲線

なお図6によれば，普通の自由沸騰状態のバーンアウト熱流束を越えた高い熱流束 q（図 6 では $q=6\times10^6$ kcal/m²h にまで達する）の核沸騰域における沸騰曲線が，自由沸騰状態の沸騰曲線をそのまま延長した形になっている．これは図3にもみられたことがらであるが，図6の場合は，ずっと高熱流束までいたっている．そして，このような高熱流束まで，核沸騰の機構自体に特に大きな変化は生じないとみなしてよいように思われる．

なお，加熱面中心に小噴流を加えた沸騰における気液挙動の概況を記しておく．まず特殊な場合として，加熱面上に自由水面が存在せず，噴流水だけが加熱面に与えられる場合（以下にはこれを $H=0$ mm と記すことにする）には，蒸気塊を生じない．しかし，水位 $H>0$ mm では，自由沸騰の場合と同じく，加熱面上に発生する蒸気が集合して，周期的に蒸気塊が発生する．また大まかにいって，その形状や周期なども，自由沸騰の場合と同様な性格のものである．

3・3 実験結果 加熱面から自由水面までの水位

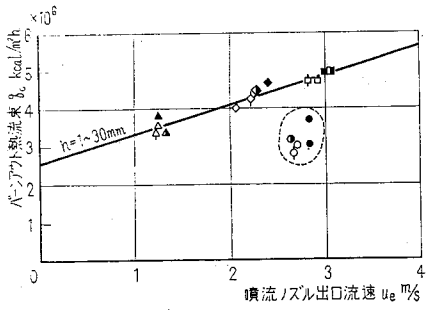


図 7 バーンアウト熱流束（$H=0$ mm の場合）

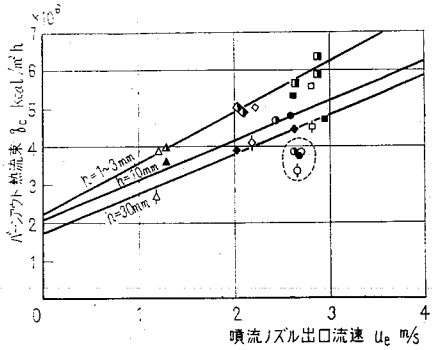


図 8 バーンアウト熱流束（$H=5$ mm の場合）

$H=0$ mm, 5 mm, 30 mm のそれぞれの場合に対し，バーンアウト熱流束 q_e を測定した結果を図7～9（横軸には噴流のノズル出口流速 u_e をとる）に示す．各図のデータ記号は，ノズル出口と加熱面の間隔 h，および噴流の流量 G によるもので，表1を参照されたい．

図7は水位 $H=0$ mm のもので，ノズルからの噴流は，途中に障害がまったくないから，そのまま加熱面中心に到達している．したがって，ノズル出口と加熱面の間隔 h には関係なく，データが1本の直線で整理される．また，これはバーンアウト限界が噴流速度 u_e に支配されていることをも意味している．なお，図7で上記の整理に従わぬデータ（破線で囲むもの）が存在しているが，これらはすべて噴流流量 $G=60$ g/min のものである．すなわち，加熱面への供給流量が少ないため，通常のバーンアウト熱流束まで到達できぬというだけの特殊な状態である．

図8は水位 $H=5$ mm のもので，周期的な蒸気塊の発生などを伴うものである．したがって，ノズル出口と加熱面の間隔 $h=1$～3 mm 程度では，ノズル出口が，ほとんど常に蒸気塊内におかれ噴流に対する障害が少ないが，h が大なるに従い蒸気塊を包む液膜などと噴流との干渉が生ずる．そのため h の値によってデータが分散し，h が大きくなるとバーンアウト熱流束が低くなる．なお図8の場合も，噴流流量 $G=60$ g/min のものは特殊状態で，通常の傾向からはずれている．

表1　図7～9のデータ記号の実験条件

	G≒60 g/min	G≒150 g/min	G≒180 g/min	G≒270 g/min
h= 1 mm	○	△	□	◇
h= 3 mm	◑	▲	◨	◆
h=10 mm	●	▲	■	◆
h=30 mm	◇	◬	◇	◇

（注）バーンアウト熱流束に対し，噴流流量 G は一次的な支配因子でないので，データごとの詳しい流量値を示す繁雑さを避け，上表の4種類の流量値の前後のものをまとめて1個の記号であらわした．

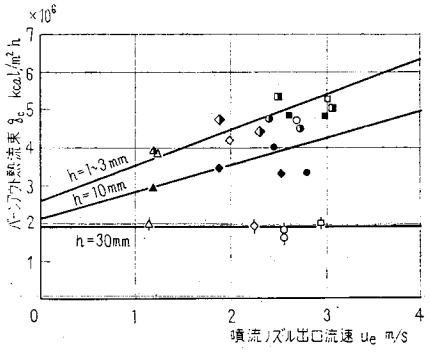


図9　バーンアウト熱流束（$H=30$ mm の場合）

図9は水位 $H=30$ mm のもので，加熱面上にかなり厚い液体層がある．そのため，特にノズル出口と加熱面の間隔 $h=30$ mm の場合には，噴流が途中の大きな障害のため加熱面へは達しえず，したがって自由沸騰状態のバーンアウト熱流束（現在の実験装置では約 2×10^6 kcal/m²h）のままになるものと考えられる．

上記の図7～9におけるデータ整理結果を比較観察しやすくするため，まとめたものが図10である．そして次の一般傾向を指摘することができる．すなわち，噴流速度が大なるほどバーンアウト熱流束 q_e が上昇する．そして $H=0$ mm の場合を除き，ノズル出口が加熱面に近いほど q_e が大となる（噴流に対する障害が少ない）．なお水位 H については，$H\geqq 5$ mm の範囲において低い水位のほうが有利な傾向をもっている．

3·4　バーンアウトの機構　さて図10において，次のことがみられる．すなわちすべてのデータ線が，ノズル出口流速 $u_e=0$ の点で，ほぼ同程度のバーンアウト熱流束の値になる傾向を示している．そして，これは水位 $H=0$ mm の場合を除いて，かなり妥当なことであるといえる．なぜなら，$u_e=0$ は噴流のない自由沸騰状態にほかならず，しかもこの場合，水位 H によるバーンアウト熱流束の相違はそれほど著しいものではない（時には実験ごとのバーンアウト熱流束のばらつきに隠れることさえあるくらいである）からである．

そして上記の見地から考えるとき，図10について，とりあえず次のような解釈を下すことができる．すなわち噴流を加えた場合，自由沸騰状態のバーンアウト熱流束が基礎となり，その上に噴流による液体補給効果が加わっていくものである．そしてこれは，少なくとも $u_e=0$ の近傍で完全に成り立つものとしてよいであろう．しかしながら，われわれは u_e がもっと大きくなった場合，上記の解釈をかえねばならぬ論理的要

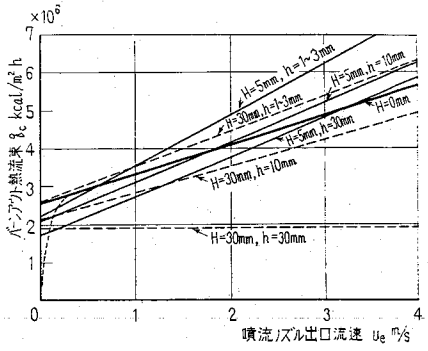


図10　バーンアウト熱流束（図7～9の結果の総括）

請に直面する．なぜなら，上記の解釈は，水位 $H=0$ mm の場合を除外して得られたものだからである．

ところで図10において水位 $H=0$ mm の場合には，ノズル出口流速 $u_e=0$ に近づくに従い，正しくは同図の点線で示すような傾向をとるべきものであろう．なぜなら水位 $H=0$ mm のときは自由水が存在せず，したがって自由沸騰状態というものが存在しないからである．そして，この状況からいって，現在次のような推論をすることが許されるであろう．すなわち図 10 で水位 $H=0$ mm の場合のデータ線は，自由沸騰状態（つまり蒸気塊の作用など）がなんら関与せず，したがって噴流独特の新しいバーンアウト機構によるものとせざるを得ない．そして，水位 $H=5$ mm，30 mm のデータ線（ただし $H=30$ mm，$h=30$ mm は噴流が無効な状態なので除く）が，上記の水位 $H=0$ mm のデータ線の近傍に現われることからみて，これらの場合もノズル出口流速 u_e がある大きさ以上になった後は，やはり噴流独特のバーンアウト機構に主として支配されるものとみてよいであろう．もちろん，水位 $H=0$ mm の場合とは異なり自由水が存在しているので，加熱面周辺からの液体流入，噴流に対する障害，蒸気塊の作用などがからむ．また，それなればこそ図 10 にみるごとく，データが分散する．しかし噴流によってこそ図 10 のようなバーンアウト熱流束の増加が生じている点からみても，上記のように考えることは当面の大筋として妥当と考えられる．

上に記した新しいバーンアウト機構に関連し，核沸騰を生じている加熱面上の流体挙動を観察した結果の一例を図 11 に示す．これは自由水面が存在せず噴流だけを加熱面に加えるところの $H=0$ mm の場合，ノズル出口流速 $u_e=3.04$ m/s のときの状況である．ここで，まず図11（a）のごとく，熱流束が高くない間は加熱面中央に，少なくとも肉眼的には発ぼうのない放射水流があり，周辺近傍に発ぼうを生ずる．しかし熱流束が上昇するに従い，図11（b）のごとく発ぼう領域が加熱面全面にひろがり，肉眼でみると加熱面の全面が白色にみえるようになるのである．この場合，図11（b）にもみられるごとく，核沸騰液層をつき破って上方へ脱出する蒸気の運動に伴い，水滴が多少，飛び散るようになる．なおこの水滴の飛び散りが始まるのは，本報の実験全般を通じ，熱流束がおよそ 1.5×10^6 kcal/m²h（このとき加熱面上からのみかけ蒸気吹出速度は約 1.3 m/s）あたりからであった．3·2 節に述べたごとく，本報の実験の核沸騰は，加熱面上の平均的な意味で，ほとんど強制対流効果を受けていないことに関係しているのかもしれない．

さて，さらに熱流束を上昇させると，図11（c）にみるごとく，噴流が加熱面に衝突する付近からすでに，大部分の水が水滴となって激しく飛び散るようになる．熱流束が高く，したがって加熱面上で発生，吹きあげる蒸気量が多くなるためである．そして，このために加熱面に実際に補給される水量が減じていき，やがて熱流束に対してつりあうだけの水量が加熱面に到達しえなくなる（この状況は一般に加熱面周辺部から始まりやすい）．この状況になると，加熱面上で白色にみえていた部分，すなわち核沸騰液層部分が消失し，代わりに乾燥加熱面の銅の地色が観察されるようになってバーンアウトにはいる．



（a）　（b）　（c）

$H=0$ mm，$h=3$ mm，$G=195$ g/min，$u_e=3.04$ m/s の場合
（a）$q=1.04\times10^6$ kcal/m²h，（b）$q=1.87\times10^6$ kcal/m²h
（c）$q=4.02\times10^6$ kcal/m²h

図 11　熱流束の上昇に伴う加熱面上の流体挙動の変化



（a）　（b）

（a）$H=5$ mm，$h=10$ mm，$G=185$ g/min，$u_e=2.88$ m/s，$q=2.60\times10^6$ kcal/m²h
（b）$H=3$ mm，$h=3$ mm，$G=64.2$ g/min，$u_e=2.71$ m/s，$q=4.16\times10^6$ kcal/m²h

図 12　加熱面上の流体挙動の一状況

なお図12は，水位 $H=5$ mm および $H=3$ mm の場合の一状況を示すものであるが，加熱面上に発生する蒸気が液膜に包まれて蒸気塊を形成することを除けば，熱流束による加熱面上の流体状況変化は基本的に前記 $H=0$ mm の場合とほぼ同じである．なお，これよりずっと高水位の場合（もちろん噴流ノズル出口は加熱面近くに置く）は，加熱面上の状況が光学的に観察しにくくなるので断言はできないが，本質的な相違が生ずることはないと予想してよいようである．

以上，要するに，噴流で液体を加熱面上に補給する場合のバーンアウト機構は，加熱面上の流体挙動の観察から，次のごときものと考えられる．すなわち熱流束に比例して発生，吹きあげる蒸気による液体のはね

とばしが，加熱面への実際補給液量を減じ，熱流束に対してつりあうだけの液量（熱流束に比例して増加）にひとしくなるとき，バーンアウトを生ずる．

一方，図 10 のごとく，バーンアウト熱流束が噴流速度 u_s の関数として整理される事実（噴流の速度でなく流量の関数としての整理は成功しなかったことを付記しておく）も，上記のバーンアウト機構に合致するものである．すなわち，蒸気吹きあげによる加熱面への液体到達阻止の状況下においては，液体噴流速度が高いほど，蒸気の吹きあげに抗して加熱面に液体が侵入しうる力が強くなり，バーンアウト熱流束が上昇するものと解釈ができる．

3・5 付記 噴霧液滴を加熱面に衝突させるミストクーリングは，少なくとも強制的に液体を加熱面に供給するという点で，本報の噴流と外見的に似ている．しかし下記の点で本質的に異なる条件を持ち，両者を混同することは不可である．すなわちミストクーリングのごとく，50 m/s 程度の非常に高速の液滴が加熱面に衝突する状態では，その動圧作用のため核沸騰が抑制されやすいであろう．それに加熱面近傍の流体挙動も，外来液滴の衝突運動による強制支配を受けるから，むしろ強制対流的な性格が強く，いわゆる沸騰からは相当に逸脱した特殊状況にあるものである．なお大気圧下の水滴によるミストクーリングに関する戸田[5]の実験結果から，液滴が飽和温度で加熱面へ飛来する場合のバーンアウト熱流束 q_c のデータが 4 個だけ得られる．これは液滴の平均直径 87.7～146μ，液滴速度 42.3～72.4 m/s の範囲である．このデータ（わずかに 4 個で，やや乏しすぎるが）から強引に推測を試みてみると．バーンアウト熱流束 $q_c=6\times10^6$ kcal/m^{2}h の実現には液滴速度を 100～150 m/s もの高速にする必要があるようである．これに対し本報の噴流ではわずか 3 m/s 以下の速度で実現されている．

4. 結　　論

プール沸騰における加熱面上の核沸騰状態を，できる限りそのままに保存する状況下に，加熱面上の蒸気塊を機械的方法で破壊する場合，および小噴流により加熱面へ液体を補給する場合のそれぞれについて，バーンアウト現象の実験的研究をおこなった．そして下記の結論を得る．

（1） 加熱面上の核沸騰液層の上に構成される蒸気塊を機械的に破壊する場合，通常のプール沸騰と同じ機構でバーンアウトを発生するが，加熱面への液体補給の障害が減ずるために，バーンアウト熱流束がたしかに上昇する．しかし蒸気塊の崩壊は瞬間的に生ずるものでなく，関連する流体挙動を自然のままに放置する限り，それによる限界に支配され，バーンアウト熱流束はそれほど上昇しない．

（2） 加熱面の近くに置いたノズルからの噴流により，加熱面上に液体補給をする場合には，当然，蒸気塊による液体供給阻止作用などは問題にならなくなる．そして，この場合のバーンアウトは，次の新しい機構で生ずる．すなわち，加熱面上の核沸騰で発生する蒸気の吹きあげによる液体飛散のため，加熱面への液体到達量が減じ，ついに熱流束につりあうだけの液量にまで到達してバーンアウトを発生する．

（3） 前記（2）項における蒸気吹きあげによる液体阻止作用は，液体噴流速度をあげれば克服できるものであり，噴流速度の上昇とともにバーンアウト熱流束は着実な上昇を示す．本実験では自由流下式の噴流のため，最高噴流速度が約 3 m/s にとどまるが，その状態で 6.37×10^6 kcal/m^{2}h（大気圧の水の飽和沸騰で強制対流効果は含まず）のバーンアウト熱流束が記録されている．

（4） 前記（2），（3）項において，噴流 1 個あたりの加熱面の大きさ，および噴流流量が関係を持つことはいうまでもない．同一の噴流速度であっても，加熱面が大きくなればバーンアウトに対し加熱面周辺付近の条件はきびしくなり，また噴流流量がある値以下に減じた場合も同様である．

（5） 本報の実験では，通常のプール沸騰のバーンアウト点より，かなり高熱流束の核沸騰域が実現されている．そしてその沸騰曲線は，通常のプール沸騰の沸騰曲線をそのまま延長した形のものである．これからみて，上記のような高熱流束状態でも，依然として核沸騰が存在し，しかも通常の核沸騰と同性格のものと考えられる．

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討　　論

〔質問〕　一色尚次（東京工業大学）

（1）回転棒で蒸気ほう（ドーム）を破壊することにより沸騰面に補給される液体はどこから与えられるのか．単に気ほう膜構成分だけでは不足ではないか．

（2）噴流を垂直に加えるさい，各所に「蒸気の吹上げによる液体飛散のため加熱面への液体到達量が減ずる」という表現が見かけられるが，これをたとえば噴流の動圧と，伝熱面過熱度に対応する蒸気圧のつりあいのようなもので定量的に説明することはできないか．

〔回答〕　（1）このような低水位の核沸騰実験の場合，加熱面は液体の中に置かれており（図1参照：ここで加熱面から自由水面までの水位 H），蒸気塊の破壊時に，液体が加熱面の周囲から加熱面上へと流れこむものである．文献（2）の図8の写真は，蒸気塊の形成を示すことに重点をおいたため，蒸気塊の破壊時の状況が示されていないが，文献（2）の図19の写真（遷移沸騰のものだが）と本質は同じと申しあげれば，前記の状況をご理解いただけると思う．そして，こうした状況なればこそ，本報2·4節において，「H が大なるほど，加熱面への液体補給時の液体侵入力が強くなる」などの記述をしているわけである．なお蒸気塊の上半部を包んでいる薄い液膜（質問者のいわれる「気ほう膜構成分」とは，これのことと思うが）については，あまり微量すぎ，かつ飛散もあり，したがって加熱面への液体補給を考えるとき無視して大過ないものと著者らは思っている．

（2）本実験で速度 $u_e=3$ m/s の水噴流が持つ動圧は約 0.4 kg/cm² であり，これに等しい飽和蒸気圧の上昇に対応する飽和温度上昇は約 10°C くらいになるものである．しかし本実験で噴流断面積は加熱面面積の約 2.6% 以下にすぎないという状況である．したがって，噴流が衝突する加熱面中心付近を除くと，加熱面上の沸騰に対し圧力，温度条件の変化はほとんど無いと考えてよいと思う．事実，図6において $u_e=0$ m/s のデータ（○印）が与える沸騰曲線上に，$u_e=2.04\sim2.63$ m/s のデータもほぼ乗っていることから考えて，本実験は大気圧下の飽和沸騰として大過ないであろうと思う．

この意味において，著者らとしては，「伝熱面過熱度に対応する蒸気圧」を考えることに対し，消極的であるといわざるを得ない．そして，その代わりというわけではないが著者らは本論文を記すにあたり，高熱流束核沸騰の加熱面から吹出してくる蒸気（熱流束 q，蒸発熱 L，蒸気密度 ρ_V のとき，加熱面から発生蒸気は平均速度 $q/L\rho_V$ で吹出す）の持つ動圧，および噴流の動圧の両者を漠然と思い浮かべていたことを記しておきたい．ただし，加熱面上の気液の流れは，単純に衝突するといった形のものでもなく，したがって本文にも動圧などという言葉の使用をひかえている．少なくとも著者らとしては，もっと力学的に気液の流動状況を詳細にはあくしないと，解析的な説明に進むのは無理なような考えを抱いている．

536.423.1 : 532.5

衝突噴流沸騰系のバーンアウト機構の研究*

甲藤好郎**, 門出政則***

1. 序

飽和プール沸騰（低水位沸騰まで含む）のバーンアウト機構に関する実験的研究[1][2]によれば，高熱流束核沸騰域では普通，加熱面に接する比較的に薄い「核沸騰液層」と，その液層から蒸気の供給を受けて液層上で生長と離脱をくりかえす「蒸気塊」の存在が認められる．次いで甲藤・国広[3]は，水平円形加熱面（銅，直径 10 mm）上の大気圧飽和水の核沸騰に対し，（i）「蒸気塊」の人工的破壊による「核沸騰液層」への液体の補給周期の短縮，および（ii）1個の小さな水噴流（噴流の断面積は加熱面面積の 2.6% 以下）による「核沸騰液層」への液体の人工的補給が，それぞれバーンアウトにおよぼす効果の研究をおこなった．ここで特に（ii）の場合，蒸気塊の液体補給阻止作用は，もはや重要な問題ではなくなり，熱流束の増加に伴い加熱面からの蒸気吹きあげによる液体の飛散が生じ，それによる液体補給制限作用がバーンアウト発生に関係するという考えが提示されている．なお彼らの実験では，自然流下式噴流の使用のため，噴流速の変化がわずか 1 m/s～3 m/s の範囲にとどまっているが，その間にバーンアウト熱流束が 3×10^6 kcal/m²h から 5×10^6 kcal/m²h まで上昇（水位零の場合）しており，噴流速度の上昇に伴うバーンアウト熱流束の増加傾向が示されている．

一方，Kopchikov ら[4]は，水平円形加熱面（黄銅，直径 11 mm）上の核沸騰液層に対し，その上方に置いた「噴霧」ノズルから，加熱面上の蒸発量とちょうどバランスするだけの液体を噴霧形式で供給するという特殊条件下の飽和核沸騰の実験をおこなっている．そして，この場合のバーンアウトは一般に，加熱面温度が液層の過熱限界温度に達したときに生ずるという結果が与えられている．これは，本報の研究よりむしろミストクーリングに近い状況であるが，加熱面上の核沸騰液層の存在，および加熱面に沿う流動（オーバフロー）のない状況下のバーンアウトについて記していることに留意すべきであろう．

戸田ら[5]は，かなり大きな水平正方形加熱面（銅，30 mm×30 mm）上に，その一方からほぼ水平に近い角度で液膜流（飽和液の場合を含む）を流入させる状況下に，核沸騰の実験をもおこなっているが，加熱面が「鏡面仕上」であって，核沸騰に対しては相当特殊な条件である．また，この実験では，液膜速度が低い場合でも高熱流束までバーンアウトの発生がみられないと記述される一方，高熱流束になると液膜の乾きが加熱面全面に広がりドライアウトが生ずると記されていて，状況が理解しがたい．ただ，高熱流束域では発生蒸気のため，液膜は流入部近くから上方にはね上げられて流れるようになり，それでも加熱面上（全面か否かは不明）には核沸騰液膜があるとの観察の記述がある．なお彼らは，この液膜のはね上げ現象を，上記のドライアウトに関連する重要現象とみなしている．

さて本報は，高熱流束加熱面上の核沸騰液層への液体の強制補給に関連し，甲藤・国広[3]が可能性を示した衝突噴流沸騰系について，流速 60 m/s におよぶ高速噴流を用い，強制対流の影響のない状況下に非常な高熱流束まで核沸騰域の拡大を試みた実験的研究を報告するものである．核沸騰熱伝達，気液の流動状況の観測などを経て，バーンアウト発生の機構を検討する．なお本実験の準備に際し，熱流束の上昇限度が未知であったため，便宜上，交流による直接通電加熱面を用いた．しかし以下，本報の必要箇所に説明してあるように，本研究の主目的に対し交流加熱は重大な影響を与えていない．

主要記号

G : 飛散液滴流量
G_0 : 噴流流量
q : 熱流束（時間平均値）
q_{BO} : バーンアウト熱流束
q_i : 瞬間熱流束（交流加熱への対応値）
T_{sat} : 飽和温度
T_w : 加熱面表面温度
u_e : 噴流速度

* 昭和48年5月31日　第10回日本伝熱シンポジウムにおいて講演，原稿受付　昭和48年8月27日．
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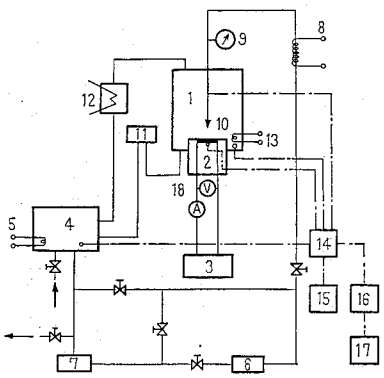
2. 実験装置とその特性

2·1 実験装置　図1は実験装置全体のシステムを示す．ここで沸騰容器*1*内の加熱面*2*において大気圧飽和水の核沸騰の実験をおこなう．この場合，沸騰容器内に水をたくわえれば，上記の加熱面について通常のプール沸騰の実験が可能である．しかし本研究では普通，容器内に加熱面が水をかぶる形での貯水はせず，噴流ノズル*10*から流出する噴流を用い，加熱面へ給水をおこなう系について実験をおこなった．この噴流は，加熱タンク*4*（9 kW 電気加熱器内蔵）から出る96～97°C の水を，高，低圧うず流ポンプ*6*，*7*で加圧（最高 20 kg/cm²），そして水温調整加熱器*8*により97°C に調整後，噴流ノズル*10*から流出させるものである．配管はすべて保温されている．噴流水温は3°C のサブクールを持つが，本報の実験では近似的に飽和温度とみなして差し支えないと考えられる（後述の4·3節参照）．なお図1で，沸騰容器*1*から流出する水は，オーバフロー容器*11*を経て加熱タンク*4*へもどり，また沸騰容器*1*で発生する蒸気は，凝縮器*12*で復水して加熱タンク*4*へもどる．

さて図2は，沸騰容器を中心とする実験装置主要部を示す．ここで加熱器*2*の内部の2個の銅ブロック導体*10*の間が，1枚の Cr 14%ステンレス鋼はく（厚さ80 μ）で結合され，このステンレス鋼はくの水平部分*3*（正方形，8 mm×8 mm）が交流通電により加熱面として働く．ここで銅ブロック導体とステンレス鋼はくとの接合部については，そこでの接触電気抵抗を極力減ずるため，接合面積を広く，かつ全面をはんだ付けとした．その結果，全電気入力の 95% 以上が上記の加熱面*3*での発熱に消費されるようになり，これは電気抵抗測定から確認されている．加熱面としてのステンレス鋼はく表面は市販の状態のままを用い（アセトンでぬぐう），また加熱面*3*の裏側は断熱のためベークライトに接着した（図2参照）．

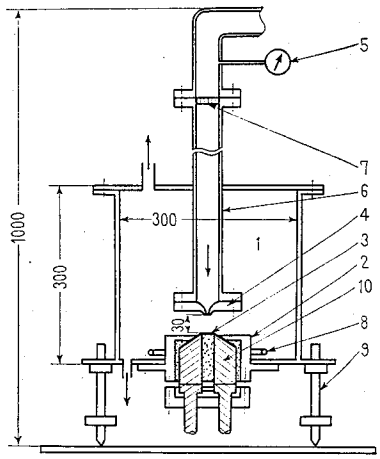
図2の垂直送水管*6*の先端にある噴流ノズル*4*（内径2 mm）は特殊設計で流速 60 m/s（レイノルズ数4 040）以上まで層流を保ち，その層流噴流を加熱面*3*の中心に垂直に置き，それによって加熱面を放射状の水膜流で覆わせる．ここで垂直噴流部分の断面積は加熱面面積のわずかに 4.9% である．噴流ノズル*4*から流出する液体流量 G_0 は，圧力計*5*によって計測される．これに関し，30°～80°C の温度範囲の水で検定した結果によれば，温度による水の密度変化の補正を考慮して誤差は2～3%以内にとどまる．以下の実験では，温度 97°C の水に対する換算検定曲線によって噴流流量 G_0 が求められている．

加熱面からの熱流束 q については，実験装置への電気入力をそのまま発熱量として求めた．ステンレス鋼はくの電気抵抗について前述の事情にあり，かつ核沸騰の熱伝達は良好のため，上記の熱流束 q に含まれる誤差は数%程度以内と考えてよいであろう．なお本報では交流加熱のため，以下熱流速 q とは実効値から求まる時間平均熱流束を意味するものとする．念のため，熱流速 5×10^5～1×10^6 kcal/m²h 程度の高熱流束



1：沸騰容器　*10*：噴流ノズル
2：加熱面　*11*：オーバフロー装置
3：低電圧大電流装置　*12*：凝縮器
4：加熱タンク　*13*：補助加熱器
5：加熱器　*14*：アイスボックス
6：高圧ポンプ　*15*：mV 計
7：低圧ポンプ　*16*：ポテンショメータ
8：水温調整加熱器　*17*：ガルバノメータ
9：圧力計　*18*：交流電流計・電圧計

図 1　実験装置



1：沸騰容器　*6*：送水管
2：加熱器　*7*：フィルタ
3：ステンレスはく加熱面　*8*：補助加熱器
4：噴流ノズル　*9*：高さ調整装置
5：圧力計　*10*：銅ブロック導体

図 2　実験装置主要部

プール沸騰（加熱面上に核沸騰液層が存在し，加熱面からの伝熱量が加熱面近傍で潜熱にほとんど吸収されることが知られている）において，加熱面から発生する蒸気を容器に捕集し，その蒸発潜熱から求めた伝熱量と電気入力との比較を試みたところ，誤差は5％以内であった．

加熱面の表面温度 T_w は，ステンレス鋼はく（加熱部）の裏面にスポット溶接したアルメル・クロメル熱電対（直径50μ）で測定される温度 T_0 から，ステンレス鋼はく（内部で一様発熱）の表，裏面間の温度差を差引くことによって求めた．すなわち

$$T_w = T_0 - \frac{1}{2}\frac{q}{\lambda}\delta \quad \cdots\cdots(1)$$

ここに q：加熱面表面からの熱流束，λ：Cr 14％ステンレス鋼の熱伝達率，δ：ステンレス鋼はくの厚さである．温度 T_0 は，熱電対起電力をポテンショメータおよびガルバノメータで測定（図1参照）するところの時間平均温度であり，従って T_w もまた時間平均温度である．なお，式(1)右辺第1項に対する第2項のオーダに関し，本報の実験範囲で T_0 は160°C前後の値であるのに対し，$q\delta/2\lambda$ は16°C前後の値であることを付記する．

2・2 衝突噴流の動圧の影響　本実験では，加熱面中心付近のいわゆる衝突域で，高速衝突噴流の動圧による圧力上昇とそれに伴う飽和温度上昇（なお噴流水温は97°C一定に保たれるからサブクール度が大になる）の問題を生ずる．

しかし，たとえばSaadら[6]の解析的研究によれば，平板に垂直に円形噴流が衝突する場合，平板に接する流体圧力（ゲージ）は，噴流半径の1.5倍の位置ですでに，よどみ点圧力（ゲージ）の約20％に低下し，さらに噴流半径の2倍の位置では約2.5％に低下，この位置では本報の大部分の実験で動圧の影響は無視できる程度になるものである．この結果から，本実験（噴流半径1mm，加熱面寸法8mm×8mm）では，少なくとも加熱面面積の約80％以上は大気圧の液流に覆われると考えてよいであろう．そして本報では，この大気圧の液流部分の核沸騰とバーンアウトの研究を目的としている．

なお，ここで次の事実を付記しておく要がある．すなわち加熱面の中心，および加熱面の中心と周辺の中間，の両位置で加熱面温度を同時計測，比較した結果によれば，両温度の相違（中心温度のほうが高い）は，それがあっても常にわずか3°C以内であった．この原因は必ずしも単純でないと思われるが，しかし噴流半径がわずか1mmのため，周囲の激しい高熱流束核沸騰の影響が，よどみ点近くまで波及するようなことがあり，それが影響しているかもしれない．そして加熱面温度については，現実として上記の状況にあるので，以下では簡単のため加熱面中心温度をもって加熱面温度 T_w とする．

2・3 発達核沸騰領域　本報の実験装置で噴流速度 u_e を一定に保ち，加熱面温度 T_w を上昇させると，図3に例示する熱伝達特性（加熱面過熱度 $T_w - T_{sat}$ と熱流束 q の関係）が常に得られた．図3において，加熱面温度が低く，熱流束曲線のこう配が比較的にゆるやかな部分では，少なくとも肉眼的には加熱面上に発ぼうを見ることができなかった．しかし熱流束曲線のこう配の性格などからみて，強制対流のなかに弱い核沸騰が混在（なお水は3°Cのサブクールを持つ）するような状況ではないかと考えられる*．他方，図3で加熱面過熱度が高く，熱流束曲線のこう配が急な部分（ここで×印はバーンアウト点）では，加熱面全面にわたり高熱流束の激しい核沸騰が生じ，加熱面から水滴が飛散するのが見られる．以下，本報では，この発達核沸騰領域だけを取扱う．

3. 核沸騰曲線と核沸騰液層

3・1 核沸騰曲線　本報の核沸騰熱伝達の実験では，バーンアウトごとに加熱面のステンレス鋼はくが焼損切断し，実験ごとに新しいステンレス鋼はくが用いられるのであるが，図4において $q > 2\times10^6$ kcal/m²h の高熱流束の領域に示す実験結果は，図3に例示した発達核沸騰領域の核沸騰曲線を集めた結果である（各曲線で×印はバーンアウト点）．一方，図4で

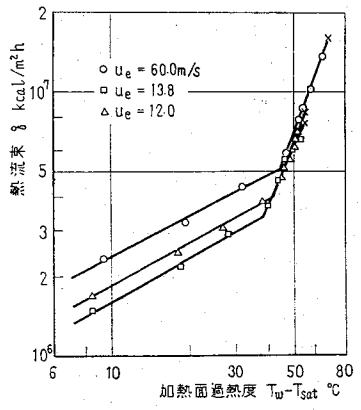


図3 熱伝達特性

* 加熱面に供給される水が，顕熱だけで熱吸収をすると仮定するとき，たとえば熱流束 $q = 4\times10^6$ kcal/m²h の場合，加熱面から流出する水の混合平均温度上昇は，噴流速度 $u_e = 60$ m/s のとき 0.38°C，$u_e = 12$ m/s のときでも 1.9°C である．

$q<2\times10^6$ kcal/m²h の領域に示す実験結果は，本報の実験装置で特に通常のプール沸騰状態（無噴流）について求めた核沸騰曲線を同様に集めたものである．

実験の都合上，図4の両グループの沸騰曲線は，必ずしも同一加熱面に対して求められたものではない．そして核沸騰実験にありがちなばらつきを示している．しかし，それらは少なくとも近似的に図4の細い平行2直線の間におさまり，これから見て，噴流沸騰系の高熱流束の核沸騰曲線（熱流束 $q=2\times10^7$ kcal/m²h におよぶ）は，統計的に通常のプール沸騰曲線（高い熱流束域）の単調な延長上に現れると考えてよいように思われる．そして，これは定性的に甲藤・国広[8]が得たと同じ結果であるだけでなく，図4に示す破直線は甲藤・国広の噴流沸騰系（噴流速度 $u_e=2.0$～2.7 m/s，かつプール沸騰の混在も含む）の核沸騰曲線の代表例を引用したものであって，本報のデータとの定量的一致をも示していることは興味深い．甲藤・国広の場合は定常加熱面（銅），本報の場合は交流加熱面（ステンレス鋼）であるが，加熱方式の差異は，少なくとも沸騰曲線に対し大きな影響を持っていないようである．そして，これについては，(i) 核沸騰では図4にも見るごとく，比較的わずかな加熱面温度変化で熱流束が大きく変わり得ること，(ii) 厚さ80 μとはいえステンレス鋼はくは相当の熱容量効果を持つこと，などの理由によって加熱面温度変化が強く抑制される一方，(iii) 熱流束変化に対し核沸騰は準定常的な形で追随しやすい性格を持つからであろうと考えられる．

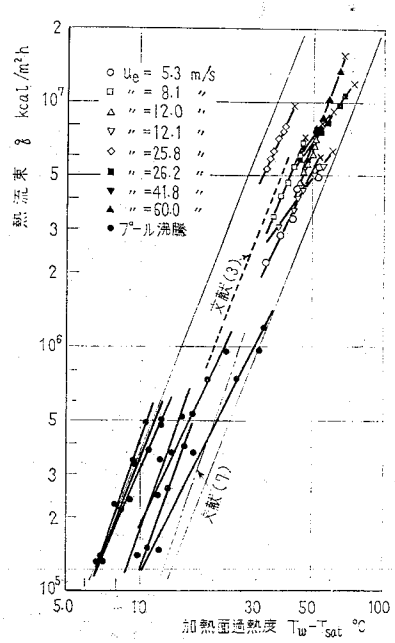


図 4 沸騰曲線（×印：バーンアウト点）

なお図4の鎖線は，正常のプール核沸騰熱伝達に関する西川・山縣[7]の整理式を参考までに示すものである．

3・2 核沸騰液層 噴流沸騰系では，加熱面上の薄い液層流内に核沸騰を生ずるわけである．そして，その伝熱特性について図4に得られた結果は，高熱流束のプール沸騰におけると同じ核沸騰液層が，現在も依然，支配的役割を持つことを示すものと解してよいであろう．なお以前，甲藤・横谷[2]は，水平円形加熱面（銅，直径 10 mm）上の大気圧飽和水のプール核沸騰の実験において，$q>5\times10^5$ kcal/m²h の高熱流束域（加熱面上に核沸騰液層が構成される）では，同一加熱面上の自由水面の水位を 100 mm から 2 mm まで低下させても，沸騰曲線に変化を生じないことを報告している．なお，この沸騰曲線は，やはり図4の平行2直線の間におさまるものであることを付記する．

4. 噴流沸騰系における気液の流動状況

4・1 高速写真による観察 本報における核沸騰の状況を高速写真で撮影した結果について，噴流速度 $u_e=9.4$ m/s および 5.2 m/s の二つの場合を図5に例示する．現在，加熱面は交流（50 Hz であるが加熱変動は 100 Hz）で加熱されており，従って平均熱流束が q の場合，交流加熱に対応するみかけの瞬間熱流束 q_i は $0\sim2q$ の振幅（実際の熱流束振幅は加熱面の熱容量の影響でもっと小さくなるが）を持つ．そして図5には，$q_i\fallingdotseq0$, q, $2q$ の各瞬間の状況が示されている．

$q_i\fallingdotseq0$ の状況では，加熱面からの液滴飛散が少なく，正方形の加熱面がよく見えているが，q_i の増加に伴い液滴飛散が増大し，加熱面が隠れて見えなくなる．なお図5によれば，熱流束 q がほぼ同じとき，噴流速度 u_e が大なるほうが，液滴飛散を多少抑制している傾向がある．

4・2 熱流束による飛散液滴流量の変化 加熱面から飛散する液滴流量（時間平均値）を定量的に測定するため，図6に示す飛散液滴捕集装置を，図2の沸騰容器 *1* のなかにとりつけた．この場合，飛散せずに加熱面上を流れ，加熱面周辺からオーバフローする液体は，図6の細い矢印に沿って流出する．また加熱面からの飛散液滴は，かなり遠くまで飛び，加熱面近くに落下するものは少ない．

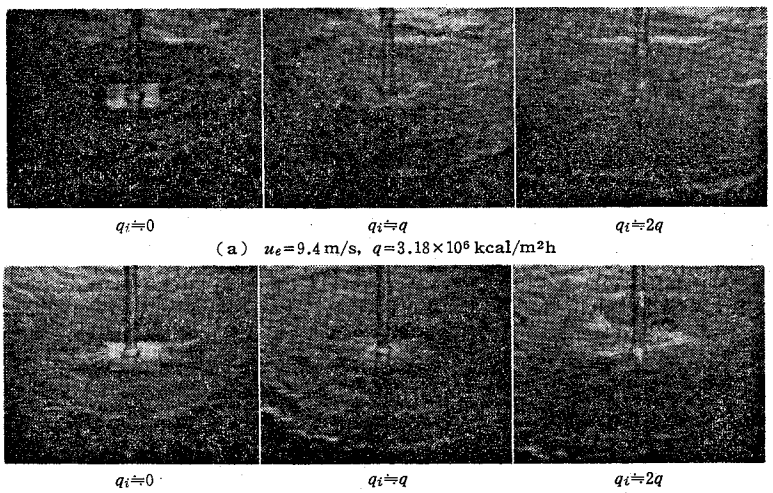
さて噴流ノズルから流出する噴流流量 G_0 を一定に保った状況下に，熱流束 q を変えたとき，加熱面から飛散する液滴流量 G の変化を測定，その代表例を図7

に示す.

図7にみるように，飛散液滴流量Gは，はじめ熱流束 q の上昇に伴って増加（近似的に $G \propto q^2$ の関係）する．しかし $G/G_0 \fallingdotseq 0.4 \sim 0.5$ になると，以後，Gの増加はなくなり，qに独立に G/G_0 がほぼ一定に保たれるという興味ある特性が存在する．そして，この状況下に，熱流束 q が増加してバーンアウト（図7で×印がバーンアウト点）が発生している．

なお，ここで次のことがらを付記しておく．すなわち，このバーンアウト熱流束の状態でさえ，その熱流束で蒸発消耗させられる液量は，上記の G_0 や G の量に比べれば無視できるくらい小さなものである*．また，いかなる瞬間でも，加熱面上を流れ加熱面周辺からオーバフローする水膜流の存在が確認されている．

4・3 流動モデル　本報で扱っている核沸騰系に対し，3・2 節，4・1 節，4・2 節の観測結果を総合して，とりあえず下記のような流動モデルを想定することができよう．



（a） $u_e=9.4$ m/s, $q=3.18\times10^6$ kcal/m²h

（b） $u_e=5.2$ m/s, $q=3.01\times10^6$ kcal/m²h

図 5　沸騰の瞬間状況

すなわち，図8（a）は噴流（流量 G_0，流速 u_e）が加熱面にあたっている場合であるが，ここで加熱面上には「核沸騰液層」が存在し，これが核沸騰を支配する．そして，この核沸騰液層を覆い流れる「上層自由液流」があり，この液流は下方の核沸騰液層から噴出する蒸気噴流の吹きとばし作用をうけて，無数の液滴からなる「飛散液滴流」を発生する．噴流流量 G_0 から飛散液滴流 G を差引い

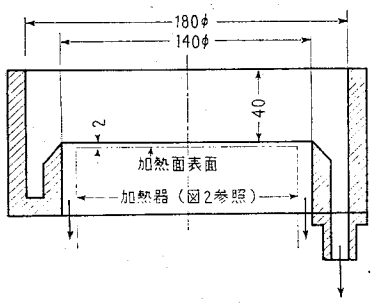


図 6　飛散液滴補集装置

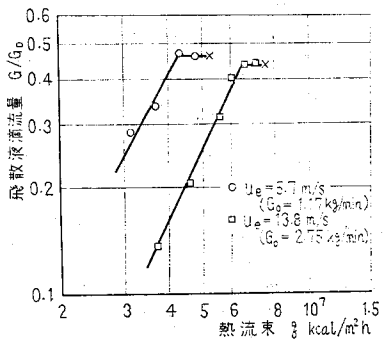
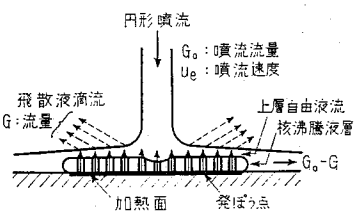
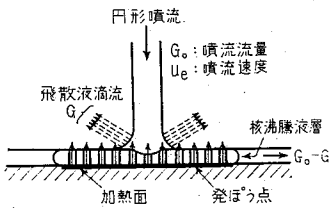


図 7　飛散液滴流量と熱流束



（a） 流れの状況



（b） 流れの極限状況

図 8　流動モデル

* たとえば熱流束 $q=5\times10^6$ kcal/m²h のとき，本報の加熱面上での蒸発消耗液量は 0.01kg/min と計算される．これに対し図7で $u_e=5.7$ m/s のとき $G_0=1.17$ kg/min，あるいは $u_e=13.8$m/s のとき $G_0=2.75$ kg/min の大きさである．

た残り G_0-G は，核沸騰液層および上層自由液流の部分を流れ，加熱面周辺からオーバフローとして流出する．なお前節の終わりに述べたように，沸騰蒸発により消耗する液量は非常に小さいから，簡単のためには無視〔図8(a)では無視〕することができる．また図8(a)のごとく，加熱面上に核沸騰液層が構成される高熱流束沸騰では，この薄い液層内での激しい沸騰流動と，潜熱による熱吸収が支配的であり，従って強制対流および顕熱による熱吸収の影響（本報の噴流の持つ 3°C のサブクールの影響を含む）は無視できると考えられる．

4·4 図7の特性に関する考察 図7に示したごとく，飛散液滴流量 G は，熱流束 q が低い領域で近似的に $G\propto q^2$ の関係をもっている．これについては，下記のような状況が関係しているものと考えられる．すなわち本報の場合，交流加熱のため液滴飛散に関し図5の状況にあるのであるが，ここで，$q_i\fallingdotseq 0$，つまり液滴飛散がほとんど中絶する期間が加熱1周期に占める比率（熱流束 q の上昇とともに減ずるはず）の変化が，G の変化に多少の影響を持つはずである．と同時に，図8(a)の流動モデルに関連し次の力学的状況が考えられる．すなわち飛散液滴流を実際に観察した結果によると，噴流流量 G_0＝一定の状況下に，飛散流出角（加熱面に対する）は熱流束に独立にほぼ一定という大まかな傾向を持っている．そして，これは飛散液滴流が持つ上向き運動量はほぼ G に比例することを意味することになるが，この運動量が，下層の核沸騰液層から上向きに吹き出す蒸気流の平均運動量（q^2 に比例）に対応するとすれば，$G\propto q^2$ に近い関係が成立することになる．

さて図7によれば，熱流束 q がもっと上昇すると，やがて飛散液滴流量 G＝一定の状況に入る．そして，この状況に対しては，図8(a)の流動モデルの極限流動状況として出現する図8(b)の状態を対応させることができるように思われる．すなわち核沸騰液層からの蒸気噴出の激化のため，ここでは上層自由液流の大部分がすでに衝突噴流周辺部付近で飛散される状態になっており，その結果，核沸騰液層はいわば裸の形で外部に露出するようになっている．そしてこの状況下で熱流束 q が上昇する場合核沸騰液層内の核沸騰がさらに激化（図4にみるごとく単調な沸騰曲線に沿って熱伝達の変化）するのみで，飛散液滴流量 G のほうはほとんど変化を生じ得ないということになる．

なお，核沸騰液層内を流れる液体の流量は一般に相当小さいと考えるのが常識であろう．従って図8(b)の極限流動状況では，$G_0-G\fallingdotseq 0$ すなわち $G/G_0\fallingdotseq 1$ が成立すべきものと考えられるのに対し，本報の実験では図7に示したごとく $G/G_0\fallingdotseq 0.4\sim 0.5$ となっている．しかし本報の実験は交流加熱のため，加熱1周期のなかに液滴飛散中絶期間を含んでいる（4·1節参照）．そして，それを考えれば，本報の実験範囲内で $G/G_0\fallingdotseq 0.4\sim 0.5$ の結果は決して不思議なものではないといえよう．なお本件に関連し，著者ら[8]はすでに，本報と同程度の寸法の円形加熱面に対し「定常加熱」の実験をおこない，その場合 $G/G_0\fallingdotseq 0.9$ の結果になることを確かめており，ここに特に付記しておきたい．

5. バーンアウト

5·1 バーンアウト熱流束 図9は，本報の実験で計測されたバーンアウト熱流束 q_{BO} を噴流速度 u_e（5 m/s～60 m/s の範囲）の関数として示すものである．u_e の広い範囲にわたり，q_{BO} と u_e との間に存在する規則的な関係に注意すべきであろう．ここで図9の斜直線は，上記の実験データに関し，最小二乗法で定めたものである．

なお図9には，本報と同程度の寸法の定常加熱面で求めた甲藤・国広[3]の実験データ（u_e＝1 m/s～3 m/s の範囲）をも掲げてある．その q_{BO} の値は，本報よりわずかに高めの傾向であるが，実質的には本報の結果に一致するとみて差し支えないであろう．また4·4節の終わりに付記した著者らの定常加熱の実験で，u_e＝6 m/s～30 m/s の範囲の q_{BO} が求められているが，それは図9の斜直線に一致するものであったことを付記しておく．

図9の斜直線は，数式に表現すれば

$$q_{BO}=2.9\times 10^6 u_e^{0.39} \quad \cdots\cdots(2)$$

ここに q_{BO} kcal/m²h，u_e m/s である．なお式(2)に関連して次のことがらを付記する．すなわち図9で甲藤・国広のデータは，噴流ノズル直径が 0.7，1.165，1.60 mm の3種類のものを含み，q_{BO} に対し，噴流ノズル直径（あるいは噴流流量）は影響しないことを示している．

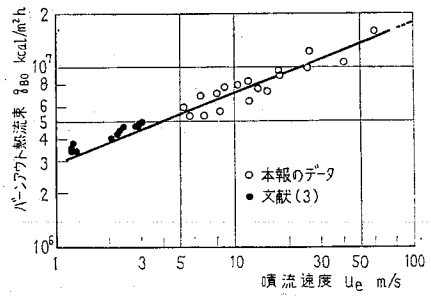


図 9 噴流速度とバーンアウト熱流束

前に図4の核沸騰曲線について，交流加熱面と定常加熱面の差異が少ないという結果であったが，ここにバーンアウト熱流束についても同様な結果を生じたわけである．そしてこれはやはり，加熱面の熱容量*などの慣性効果によるものと想像されるが，なおバーンアウトは多くの場合，ある時間内の平均的状況に基づいて生起する性格を持つことに留意する要があるかもしれない．

5・2 ステンレス鋼はくの切断状況 本報の実験ではバーンアウトによりステンレス鋼はくが切断するが，通常のプール沸騰（無噴流）の状況下では，常に加熱面の中心付近を通る切断（電流方向に垂直）を生じたのに対し，衝突噴流沸騰系の場合は図10に示すごとく，加熱面の端部近くに切断を生じた．なお図10によれば，噴流速度 u_e の上昇に伴い，切断位置がわずかながら加熱面端部のほうへ移動する傾向がみられる．

5・3 バーンアウトの発生機構 さきに甲藤・国広[3]は，衝突噴流沸騰系に関し，液滴飛散のため加熱面への補給液量の実質的減少がおこり，熱流束（蒸発消耗液量）とバランスするだけの液量が加熱面に到達し得なくなる結果としてバーンアウトが発生すると述べている．

しかしながら，本報の4章で明らかにされた気液の流動状況からみて，現在の沸騰系のバーンアウト発生機構を，液体の消耗と補給の間の単純なバランスの問題と考えることは，もはや明らかに不可能である．なぜなら，飛散せずに加熱面上を流れ，加熱面周辺からオーバフローするところの液量 G_0-G に比べ，加熱面上で蒸発消耗する液量は，少なくともオーダが1けた低いものであるからである（4・2節脚注参照）．

すなわち現在の場合，加熱面周辺から相当量の液体がオーバフローする状況下にバーンアウトが発生している．そして加熱面上には，多数の発ぽう点から上方に噴出する吹出し蒸気流が存在するが，ここで核沸騰液層からは液体飛散はあまり生じないと考えてよいように思われる〔図8（b）参照〕．このような液体流れと蒸気の流れのからみ合いのもとに，しかもバーンアウト熱流束（加熱面からの吹出し蒸気量に比例）と，噴流速度（大まかには加熱面上の液流速度に比例）との間に図9のような密接な関係が存在していることになる．従って，蒸気の吹出しを伴う加熱面に沿って流れる液流の一種のはく離の問題が，バーンアウト発生と関係している可能性をここに記しておくことは必ずしもむだでないであろう．ただし現段階では，あくまで推測である．

5・4 付記 吹出しを伴う多孔質壁に沿う乱流境界層の場合，吹出し質量速度の増加に伴い，壁面から境界層がはく離する条件は，普通，近似的に次の形で与えられる．

$$\frac{\rho_w v_w}{\rho_\infty u_\infty}\frac{1}{f_0}=\text{const.} \quad \cdots\cdots(3)$$

ここに ρ_w：吹出し流体の密度，v_w：はく離限界における吹出し流速，ρ_∞：主流密度，u_∞：主流速度，f_0：吹出しのない状態（$v_w\to 0$ の極限）での壁面摩擦係数である．

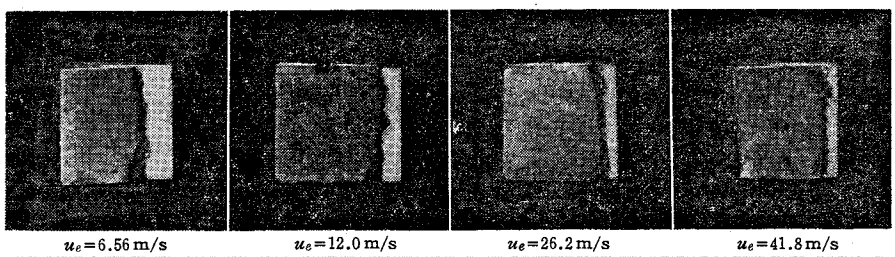
さて，やや強引であるが，噴流沸騰系のバーンアウトがはく離の問題（5・3節に記した推測）であると仮想し，式(3)の応用を試みる．すなわち熱流束 q_{BO} で蒸気を発生しているバーンアウトの場合は，はく離限界の $\rho_w v_w$ に対し，次式が書ける．

$$\rho_w v_w = q_{BO}/h_{fg}\ （ここに\ h_{fg}：蒸発潜熱）$$

また摩擦係数 f_0 については，現在，単相流の状態を考えても無意味に思われるので，とりあえずバブル流（ボイド率零の極限）のそれを使用することとする．ここで Malnes[9] の実験（円管でおこなわれているが，乱流状況なので平板にも近似的に使用できよう）によれば次の関係がある．

$$f_0 \propto u_\infty^{-0.64}\ （ここに\ u_\infty：液体流速）$$

上記の $\rho_w v_w$ および f_0 を式(3)へ代入すれば，次の



u_e=6.56 m/s　u_e=12.0 m/s　u_e=26.2 m/s　u_e=41.8 m/s

図 10 バーンアウトによるステンレス鋼はくの切断（電流は水平方向）

* 厚さ 80 μ のステンレス鋼はく（熱流束 $q=10^7$ kcal/m²h に対応する内部発熱を続けているとする）が温度 160°C から融点 1520°C まで温度上昇するに要する時間を計算してみると，外界と完全な断熱条件下でさえ 34 ms である．これに対し 50 Hz 交流の加熱1周期は 10 ms である．

関係を得る.

$$q_{BO} \propto u_{\infty}^{0.36} \quad \cdots\cdots(4)$$

以上の解析はずいぶん大まかであり，従って式 (4) をそのまま式 (2) と比較するには問題があるかもしれない．しかし両式が，かなり近い関係になっていることは興味あることである．

6. 結　　論

交流加熱の正方形ステンレス鋼はく (8 mm×8 mm) の中心に小さな高速水噴流（直径 2 mm，噴流速度 5～60 m/s）を垂直に置き，加熱面の大部分を大気圧の飽和水膜流で覆わせるという状況下に核沸騰の実験をおこなった．その主要な結果を要約すると以下のごとくである．

（1） 2×10^7 kcal/m^{2}h に至る高熱流束核沸騰域の拡大が実現され，かつその核沸騰曲線は，通常のプール沸騰（高熱流束域の）の核沸騰曲線の単調な延長上に生じた．

（2） 噴流速度一定の状況下に，熱流束 q を変えるとき，核沸騰状況下の加熱面上の水膜流から飛散する液滴流量は，最初 q とともに増加するが，やがて上限に達し，それ以後は一定に保たれるという特性が見出された．そして，その後バーンアウトに到達するものであった．

（3） 噴流沸騰系の核沸騰，および気液の流動状況などに関する諸特性は，「核沸騰液層」，「上層自由液流」，「飛散液滴流」からなる流動モデルを考えることにより，かなりよく説明ができる．

（4） バーンアウト熱流束 q_{BO} と噴流速度 u_e との間には，u_e の広い範囲（1～60 m/s）にわたって非常に規則的な関係：$q_{BO} \propto u_e^{0.39}$ があることが見出された．なお q_{BO} に対し，噴流ノズル径，あるいは噴流流量は影響しない．

（5） 噴流沸騰系におけるバーンアウトは，加熱面からの核沸騰液層（加熱面からの蒸気吹出し，および加熱面に沿う液体の流れを含む）のはく離の問題と関係を持つ可能性がある．これは現段階では推測であるが，上記の可能性に基づく一つの簡単な解析を付記した．

本報の実験装置は，東京大学工学部機械系三学科・機械工学実験所で製作されたものので，ここに記して謝意を表する．

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討　　論

〔質問〕　**一色尚次**（東京工業大学）

巧妙な実験と簡単明確なモデルを提示された努力に敬意を表す．

（1） 図 8 のモデルは大へんわかりやすいが，このモデルは深い液層の下（つまり深い容器内の場合）では飛散水滴の挙動が異なるのでそのままでは成立しないと考えてよいか．

（2） 核沸騰液層なる考えは大へんおもしろいが，その層内は高速水が水平に走っていて，極限の場合最外縁でまずバーンアウトが開始されると考えてよいか．

（3） 貴論文によれば「核沸騰液層内を流れる液体の流量は一般に相当小さいと考えるのが常識である」311 ページ左欄下 3～2 行めといっておられるが，これはさらに詳細にはどのようなモデルを考えておられるか．また後のはく離説と矛盾しないか．

（4） 312 ページ左欄 5～7 行めに述べられている「バーンアウトはある時間内の平均的状況に基づいて生起する性格をもつ」という表現はむしろ，「ある微少かつ有限な時間内の平均的情況の偏差によって統計的に生起する」と書かれるほうが正確ではないか．

（5） 上の有限な時間をバーンアウトの時定数とすると，この時定数は q の増大とともに小さくなるか．

（6） 5·4 節の「はく離による相似」のモデル化は極めてユニークであり，バーンアウトの基本的説（満員説，不安定説など）の新たなる一つとして提起さるべき価値のある大きな業績と思う．しかし単相流吹出しの際にくらべて

（a） 底面において気ほうが占める面積率が大きく，しかも気ほうがねているので，底面で流れにすでにスリップがあると考えられるのではないか．

（b） 吹出しユニットが気ほう塊であり分子にくら

べ大きい.

（c） $u=0$ のときとの連結がうまく行かない.

などの点でそのままの相似には難点があると思うがいかがか.

〔回答〕 （1） 図8のモデルは，深い液層の下などでは成立しないものである.

（2） 噴流沸騰系の場合，加熱面に沿い核沸騰液層内を液体が流れていると考えている．しかし，その流速は相当に低く，かつ噴流衝突点から遠い位置ほど低速になっていると想像している．バーンアウトは最外縁の近傍で開始するように思うが，発生領域に多少の幅を持つ可能性もあろう.

（3） 核沸騰液層は，固体壁に接しているものであり，厚さが薄く，しかも液層を垂直に貫通する多数の蒸気噴流がある．従って液体は層内を相当な低速で流れると考えるのが常識とした次第である．なお5·4節の式(4)の u_∞ は，核沸騰液層内の流速をばく然と考えているものである（噴流衝突速度 u_0 が増大すると u_∞ も大になる）．本報のはく離説は，まだ仮説的なものであると同時に，単相流吹出しの場合のはく離とは相当に異なる流体状況をも含んでいるものゆえ，矛盾の当否については回答を保留させていただきたいと思う.

（4） この表現は，著者ら自身，あいまいな態度で推測を付記したものである．従って，もっと正確な表現をとるとなると，ためらいを感じる．そして，そうした点で本報に記した短い記述 312 ページ左欄 5～8 行めは，実質的には無内容なものであったかもしれないと恥じる次第である.

（5） 本報の扱う噴流沸騰系においては，バーンアウトの時定数なる概念を考えたことはないし，本報でも触れていないとお答えしたい.

（6） 質問者は，充満説，不安定説，はく離説いずれにせよ，どんな流動状況下のバーンアウトにも普遍的に適用できる性格のものと考えておられるように感ぜられる．しかし，これに反し著者らは，プール沸騰と噴流沸騰系のバーンアウト機構を別個のものとする立場に立っているのであり，従って，その立場からはご質問の（c）項の問題は考える必要のないものである．また本報のはく離説では，すでに前記（3）にも述べたごとく，単相流吹出しの場合と，そのままの相似を認めようとしているわけではない．従ってご質問の（a），（b）項などは，難点になるというものではなく，むしろ，（a），（b）項なども含んだ状況下に生ずるところのはく離を考えてみたいというのが著者らの，いつわらざる気持なのである.

536.423.1 : 532.525

衝突噴流沸騰系のバーンアウトの研究*

(第1報，気液の挙動の特性)

門出政則**，甲藤好郎***

1. 序

従来，バーンアウトの理論的な研究のうち，加熱面上に存在する気液の流動状況が比較的簡単なプール沸騰バーンアウト（重力が支配的役割を果たしている）については，その実験結果に関する無次元整理式[1]が見いだされており，かつその物理的説明[2][3]もおこなわれている．しかし，強制流動沸騰系のバーンアウト（重力が影響しないはず）については，特に実用上重要であるにもかかわらず，まだ特定の沸騰形態に対するバーンアウト熱流束の経験式[4]や二，三の理論的解析[5]（現象に即したものとは言い難く，かつ論理の飛躍や任意的仮定を含む）が報告されているような状況にあり，前記のプール沸騰の研究成果に比較して程遠い状況におかれている．

そこで本研究では，衝突噴流沸騰系の場合，(i)加熱面上の気液二相の流動状況の観察が容易であること，(ii)衝突噴流によって実現される液膜流は，平板流などに比較してその境界条件が比較的簡単であること，(iii)高熱流束域の核沸騰を支配する核沸騰液層に強制的に液体の補給をおこなう場合，バーンアウト熱流束が著しく上昇されること[6][7]などの理由から，強制流動沸騰系に属するもののなかから，特に単純な基本形式の一つである衝突噴流沸騰系を選び，その核沸騰熱伝達の特性および加熱面上の流動状況，バーンアウトの発生状況やバーンアウト発生後の加熱面の汚損状況などの詳細な実験観察を経て，バーンアウト熱流束を支配する物理量を明確にし，これを次元解析から得られる無次元数と組合わせることによって，衝突噴流沸騰系で生じるバーンアウト熱流束の無次元整理を試みるものである．

第1報は，まずこの沸騰系の熱伝達特性および加熱面上の気液二相流の詳細な観察をおこない，その特性およびバーンアウトについて検討する．

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主要記号

D：加熱面径（$r_0=D/2$）
d：ノズル径
G：飛散液滴流量
G_0：噴流流量
q：平均熱流束
q_{BO}：バーンアウト熱流束
S：加熱面とガラス平板のすきま幅
T_{sat}：飽和温度
T_W：加熱面温度
ΔT_{sat}：加熱面過熱度　$=T_W-T_{sat}$
u_s：噴流速度

2. 実験装置と実験方法

2·1 実験装置　図1は実験装置全体のシステムを示す．本装置は大別して，温水循環系統，加熱系統，衝突噴流沸騰が生じている実験装置主要部(図2参照)

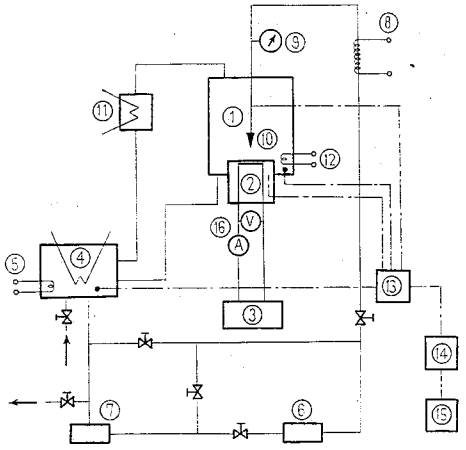
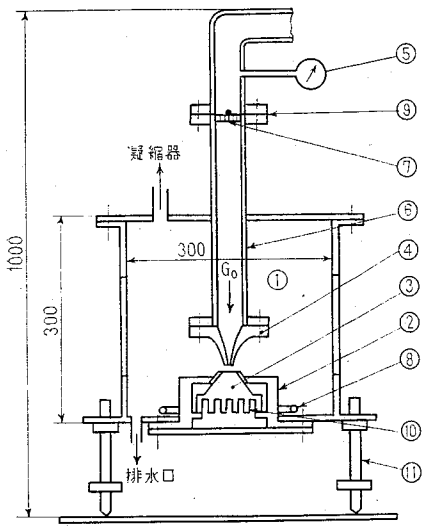


図1　実験装置概略図

の3主要部から成っており，これらに付随して噴流速度測定・噴流温度測定・加熱面温度測定の各測定装置がある．本研究は，沸騰容器①内の加熱面において，水及び F113 の大気圧飽和核沸騰の実験を行うが，加熱面への液体の供給は，噴流ノズル ⑩ から流出する噴流を用いて行う．この噴流は，加熱タンク④（9 kW 電気加熱器，温度自動調節器付き加熱器，熱交換器内蔵）から供給されるサブクール度 3°C～4°C の液体を高・低圧か(渦)流ポンプ ⑥，⑦ で加圧（最高 20 kg/cm²）し，そして水温調節加熱器 ⑧ によってサブクール度 3°C に調整された後，噴流ノズル ⑩ から流出されることによって得られるものである．配管はすべて保温されている．噴流液温は 3°C サブクールを持つが，本報の実験では近似的に飽和温度とみなして差し支えないことを報告[7]している．

図2は上向き加熱面の場合の沸騰容器を中心とする実験装置主要部であるが，重力の影響を調べられるように加熱面の向きが上向きおよび下向きについて実験ができるようになっている．ここで沸騰容器①の中央に置かれた銅ブロック③（上半部が円すい形，下半部が円柱形）の上端面が円形の加熱面（D=10 mm, 15 mm, 20 mm の3種類使用）になる．加熱面の仕上げは，紙やすり0番で研摩後，アセトンで洗浄したものである．なお，加熱面はバーンアウト発生ごとに仕上げ直した．銅ブロックの下半部には，電気加熱器 ⑩（11 枚の板状加熱器を銅ブロックみぞにそう入したもの）を内蔵し，加熱面の熱流束が 2×10^7 kcal/m²h まで可能な容量（本実験では，前記の容量で十分であることを確かめている[7]）を持っている．また銅ブロックの上半部の中心軸上には3個のアルメル-クロメル熱電対（素線径 0.1 mm）があり，温度分布の測定から加熱面温度 T_W および平均熱流束 q が求められる．ところで，熱流束 q の計測のために必要な銅ブロックの熱伝導率を決めるために次のような検定実験をおこなっている．すなわち，高熱流束のプール核沸騰状態において加熱面から発生する蒸気を収集し，その蒸発潜熱から求められる伝熱量，および銅ブロック内の温度分布から銅ブロックの熱伝導率を決定した．発生蒸気の蒸発潜熱から求められる伝熱量を，銅ブロック電気加熱器入力と比較してみると，入力のうち外部への熱損失はたかだか 10% 程度であった．したがって，本実験における熱流束の計測誤差は，たかだか 10% であると考えてよい状況にある．



① 沸騰容器　⑤ 圧力計　⑨ 熱電対（φ0.3 CA）
② 加熱器　⑥ 送水管　⑩ 板状電熱器
③ 銅ブロック　⑦ フィルタ　⑪ 高さ調整装置
④ 噴流ノズル　⑧ 補助加熱器

図 2　実験装置主要部

図2の垂直送水管⑥の先端にある噴流ノズル④（内径 d=2 mm, 2.5 mm 使用）は，流速 60 m/s 以上まで層流を保ち，その層流噴流を加熱面の中心に置き，それによって加熱面を飽和（本報では飽和の場合のみ取扱うが，サブクールのある場合についても実験可能である）の放射状の液膜流（液膜流の厚さは 0.2 mm程度）で覆わせる．ここで垂直噴流の占める断面積は，加熱面面積の 4% 以下である．噴流ノズル ④ から流出する液体流量 G_0 は，圧力計 ⑤ によって計測される．これに関し，温度範囲 30°C～80°C の水と 20°C～25°C の F113 について検定した結果によれば，温度による液体の密度変化の補正を考慮してもその誤差は 2～3% 以内にとどまる．以下の実験においては，サブクール温度 3°C の液体に対する換算検定曲線によって噴流流量 G_0 が求められている．噴流速度 u_0 は，上記の噴流流量 G_0 を噴流ノズル断面で割った平均速度（u_0=1 m/s～30 m/s の範囲）である．

2·2　衝突噴流の動圧の影響　本実験では，加熱面の中心付近のいわゆる衝突域で高速衝突噴流の動圧による圧力上昇と，それに伴う飽和温度上昇の問題を生じるが，前報[7]に記したごとくほとんど無視できることを確かめている．そして本報では，この大気圧下の液流部における核沸騰と，バーンアウトの研究を目的としている．

3.　核沸騰曲線と核沸騰液層

3·1　核沸騰曲線

3·1·1　上向き加熱面の場合　図3は水，図4は

F113 に対し本実験で得られた衝突噴流沸騰系の沸騰曲線の代表的なものである．図中の実線は，水および F113 に対し本実験で得られた沸騰曲線の平均的な直線である．また，図3の破線は，甲藤・国広[6]の核沸騰曲線（噴流速度 u_e=1 m/s～2.7 m/s，かつプール沸騰の混在を含む）の代表例である．更に，本実験で得られる沸騰曲線は，通常のプール沸騰曲線のほぼ延長上にあることを付記[7]しておく．

この図から，沸騰曲線は噴流速度 u_e，噴流流量 G_0，加熱面径 D の影響をほとんど受けず，核沸騰実験にありがちなばらつきの範囲内で一致していることがわかる．またこの沸騰曲線は，一般に噴流速度 u_e の増加および加熱面径の減少に伴って，高熱流束域へ延長されることがわかる．

なお図 3，4 中の一点鎖線は，正常のプール核沸騰熱伝達に関する西川・山懸[8]の整理式を高熱流束まで延長して参考までに示すものである．

3·1·2 下向き加熱面の場合　図5は水，図6は F113 に対し本実験で得られた沸騰曲線の代表的なものである．図 5，6 の実線は，水および F113 の上向き加熱面の場合に対する沸騰曲線である．

この図に示される結果から，下向き加熱面に対する沸騰曲線は，上向き加熱面の場合と同様に噴流速度 u_e，加熱面径 D などの影響をほとんど受けないこと，および両者の沸騰曲線は核沸騰実験にありがちなばらつきの範囲内で一致し，加熱面の向きに支配されないことがわかる．

なお図 5，6 の破線は，Copeland[9]，Ruch[10] が下向き加熱面の中心に，それぞれ水および F113 の噴流を衝突させた場合に得た沸騰曲線を参考までに掲げたものである．

3·1·3 限定空間の場合　噴流ノズル④の先端に固定されたガラス平板を，加熱面上すきま幅 S=0.3 mm あるいは 0.5 mm の位置まで接近（高さ調整装置 ⑪

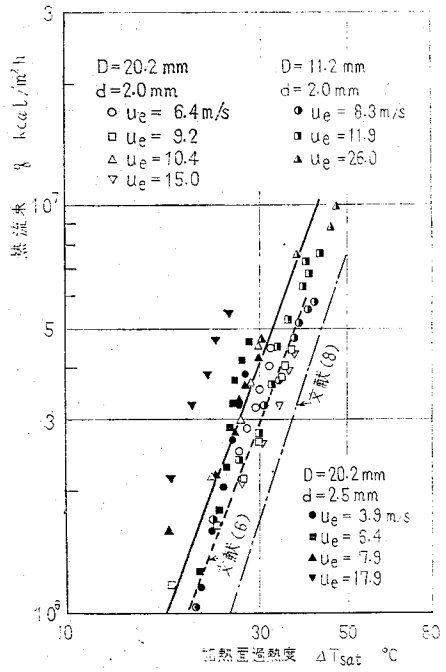


図 3　沸騰曲線（上向き加熱面，液体は水）

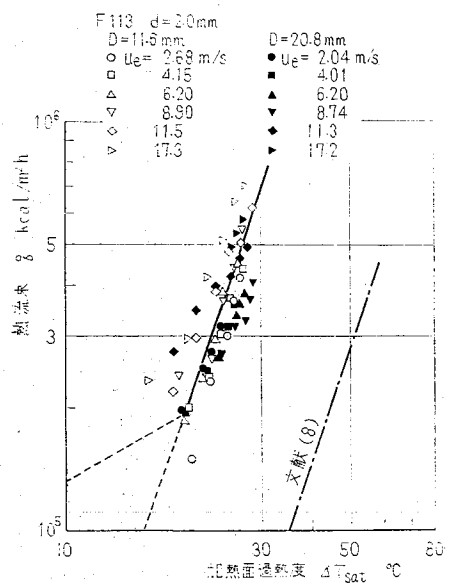


図 4　沸騰曲線（上向き加熱面，液体は F113）

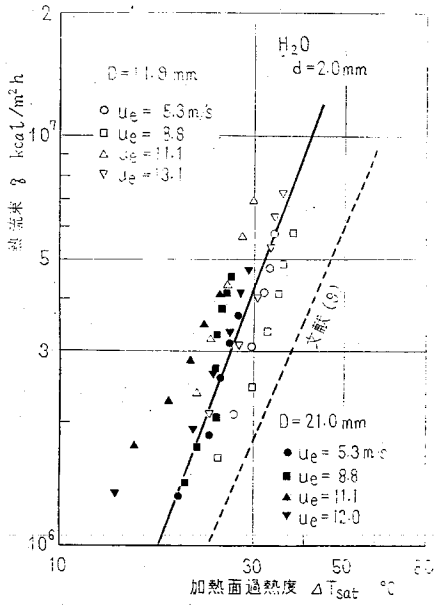


図 5　沸騰曲線（下向き加熱面，液体は水）

を使用）させ，加熱面から発生する蒸気が，ガラス平板に沿って流れる飛散液滴流と加熱面上を流れる液膜流にはさまれた，かなり狭い空間を高速で流れているような状況下の沸騰で，これを限定空間沸騰と呼ぶ．そしてこの沸騰系の実験は，主に加熱面上の流動状況の詳細な光学的観察のために行ったものである．

そこでまず，この沸騰が通常のガラス平板のない場合と比較し，その相違点について検討すると，図7の結果から限定空間の実験で得られる沸騰曲線は，ガラス平板のない場合と同様に噴流速度 u_e，噴流流量 G_0，加熱面径 D の影響を受けないこと，および通常の沸騰曲線と基本的に一致していることがわかる．この結果，加熱面上にガラス平板を取付けても，核沸騰自体の本質には何ら変化が見られないように思われる．

3·2 核沸騰液層 衝突噴流沸騰系の核沸騰は，加熱面上の薄い液膜流内で生じる高熱流束の核沸騰であるが，この核沸騰も通常の高熱流束のプール沸騰において，支配的役割を果たしている核沸騰液層と同じ液層が加熱面を覆う液膜流の中に存在し，そしてこの液層がこの沸騰系の核沸騰を支配しているものと考えられる．この核沸騰液層は，3·1 節の実験結果から，この液層上部に存在する液流の流動状況（噴流速度 u_e，噴流流量 G_0，加熱面径 D，加熱面の向きなど）の変化に支配されない核沸騰固有の特性をもち，そして高熱流束の核沸騰においては，この液層内で自律的とも言える非常に激しい核沸騰流動を生じているものと推察される．

4. 衝突噴流沸騰系における気液の流動状況

4·1 沸騰曲線と流動状況の関係

4·1·1 上向き加熱面の場合 ガラス平板を通して観察される加熱面上の流動状況と，沸騰曲線の関係を図8に例示する．この図8は，水の場合の流動状況であるが，熱流束の増加に伴って以下に記すごとく変化することを示す．すなわち，熱流束が比較的低い範囲（噴流速度，加熱面径によって多少変動するが，約 $q \fallingdotseq 2\times10^6$ kcal/m²h 以下の熱流束域）では図8（a）に示されるごとく加熱面上に十分発達した核沸騰は生じておらず，加熱面からの液滴の飛散は観察されない．更に熱流束を増加し，2×10^6 kcal/m²h 以上の領域になると図8（b）に示されるごとく加熱面周辺部に白いあわ立ち（この白くあわ立って見える部分が衝突噴流沸騰系における核沸騰液層で，本実験の核沸騰はこの液層に支配されている）が観察され始める．また，この核沸騰液層を覆うように流れている液流（この液流を上層自由液流と定義する，図 14 参照）の一部から発生する飛散液滴が，ガラス平板に沿って外部に流出しているのが観察される．しかし，まだ核沸騰の十分発達していない部分が加熱面の大部分を占めている．更に加熱面温度を少し上げ熱流束を上昇させると，図8（c）に示されるごとく加熱面の大部分は，白く見える核沸騰液層に覆われ，非常に激しい核沸騰を生じているのが観察されると同時に，この核沸騰に起因して生じる飛散液滴も加熱面の大部分から発生しており，その流量（飛散液滴流量と熱流束の関係は，4·2 節参照）も増加するのが観察される．バーンアウト発生直前の熱流束では，図8（d）に示されるように

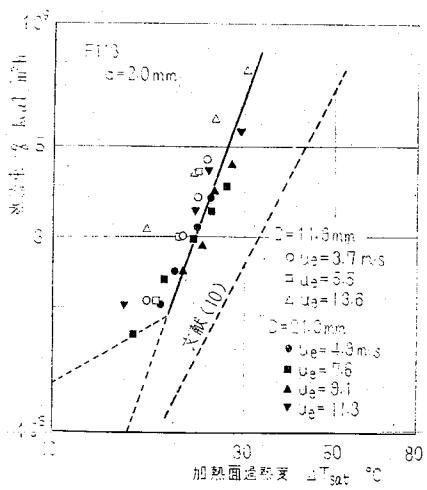


図 6 沸騰曲線（下向き加熱面，液体は F113）

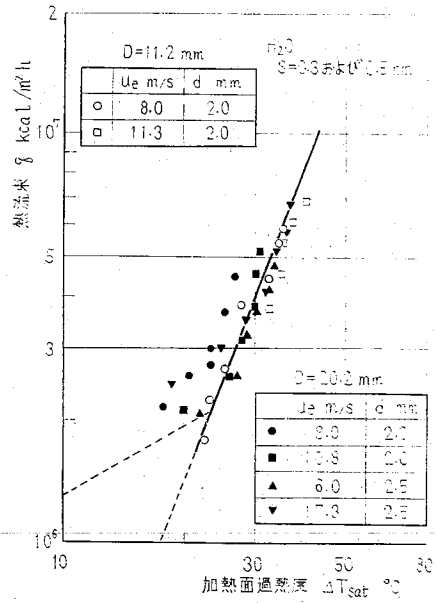
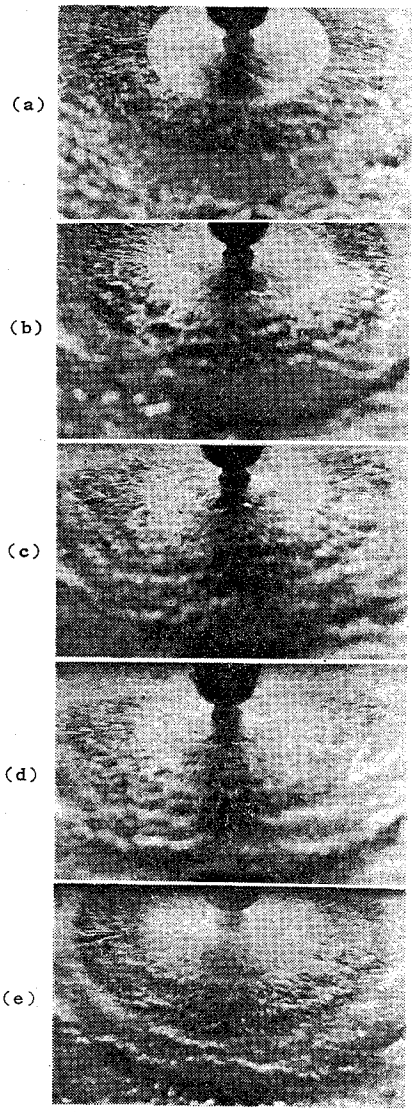


図 7 沸騰曲線（限定空間の場合）

加熱面全面は，非常に激しい核沸騰状態となる，そして上層自由液流は加熱面の中心付近で大部分飛散され，加熱面周辺部に到達する流量はわずかであることが観察される．図8(e)は，バーンアウト発生後の状況であるが，加熱面全面が蒸気で覆われていることを示す．続いて，F113 の場合の流動状況を観察すると，熱流束の増加に伴う核沸騰液層の発達状況や飛散液滴の発生状況は，水の場合とほとんど同一状況変化をすることが確かめられることを付記しておく．

次に，ガラス平板を加熱面上から取り除き，加熱面周辺から流出する液膜流を肉眼観察すると，この液膜流は，加熱面上で激しい核沸騰を生じている場合，加熱器上のある位置で跳水を生じているのが観察される．そこで，この跳水位置に注目し，この位置と熱流束の関係を実測した結果を図9（縦軸：液膜の厚さ h，横軸：加熱面中心からの距離 r）に例示する．この図9から，跳水位置は熱流束の増加に伴って加熱面周辺部に漸近することがわかる．そして，このことは加熱面周辺から流出する液膜流の流速が，だんだん減少することを示すものと考えられる．また一般に，加熱面径が同じ場合，噴流速度を大きくすればする程，同一跳水位置を得るためには，より高い熱流束が必要なことも確かめられている．ところで液膜流の減速は，熱流束の増加に伴う飛散液滴の増加，および加熱面から吹き上げる蒸気流の及ぼす抵抗の増加によって，液膜流が最初持っている運動量を一部持ち去



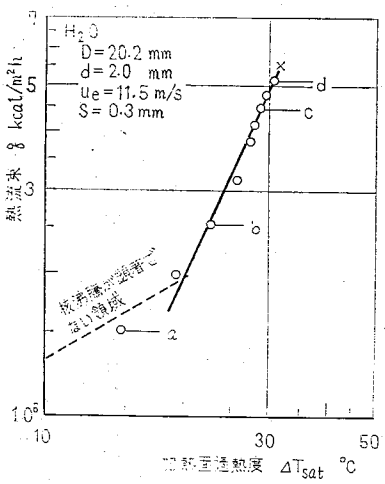


図 8 沸騰曲線と流動状況の関係

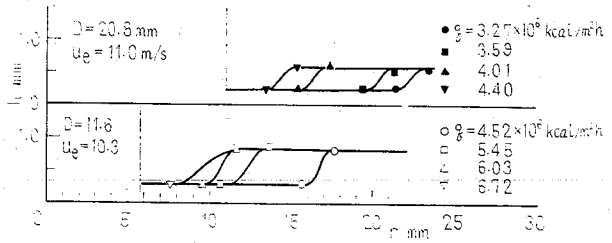
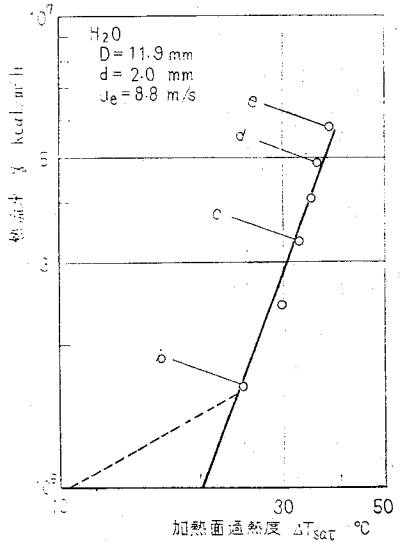


図 9 跳水位置と熱流束の関係

られることによるものと考えられる．

なおこの跳水位置は，噴流速度，噴流流量，加熱面径に関係なく，バーンアウト発生まで常に加熱器上に存在していることが観察されることから，加熱面を通過する液膜流がバーンアウト発生まで常に存在していることが確認されると同時に，更に重要な事実，「加熱面上には，バーンアウト発生まで核沸騰液層が存在し，激しい核沸騰を生じていること」がわかる．



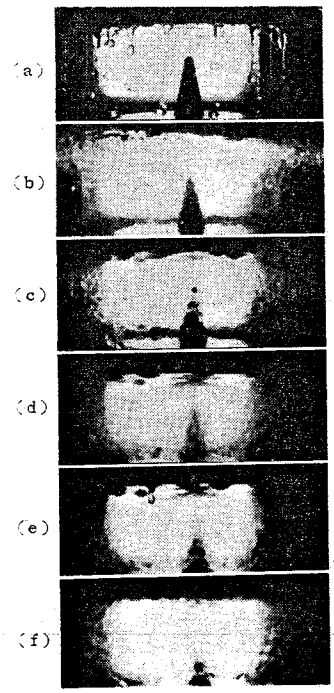


図 10　沸騰曲線と流動状況（下向き加熱面）

4・1・2　下向き加熱面の場合　加熱面を下向きにした場合，飛散液滴流による視野の妨げがなくなり，加熱面周辺部の流動状況の光学的観察（上向き加熱面の場合，肉眼観察のみ）が容易になることを利用して，加熱面周辺の気液の流動状況の詳細な観察を行い，その結果を沸騰曲線に対応させて図 10 に例示（F113 の場合もほぼ同様な結果が得られる）する．図 10(a) は，熱流束 $q=0$ すなわち噴流が加熱面に衝突したときの状況である．また図 10(f) は，バーンアウト発生後の状況であるが，噴流が加熱面に衝突した後，加熱面を覆う液膜流とはならず，あらゆる方向に激しく飛散される不規則な液滴〔定常な核沸騰が生じている場合，図 10(d)，(e) に示されるような非常に規則正しい飛散液滴を形成している〕となることを示している．そして，この不規則な液滴に視野を妨害され，加熱面上の詳細な状況観察は不可能である．

次に，加熱面周辺から流出する液膜流を観察すると，図 10 からこの液膜流は，熱流束の増加に伴う激しい沸騰流動や，飛散液滴によってその運動量の一部を持ち去られ，加熱器上のある位置で失速し液滴状となることが観察される．そして，この液滴は半径方向に流れる蒸気流の作用で加熱器周辺まで移動し，そこで壁面から離れて落下している．ところでこの液滴状となる位置は，熱流束の増加に伴って加熱面に漸近するが，この液滴はバーンアウト発生まで常に観察され，バーンアウト発生まで加熱面上に十分な液体が存在していることがわかる．

なおこの液滴状となる位置は，高熱流束下で激しい飛散液滴が観察されるような状況下では，噴流速度や加熱面径にほとんど影響されないこと，およびこの現象は上向き加熱面の場合に観察される跳水現象と同一原因によるものと考えられる．

4・2　熱流束による飛散液滴流の変化　加熱面から

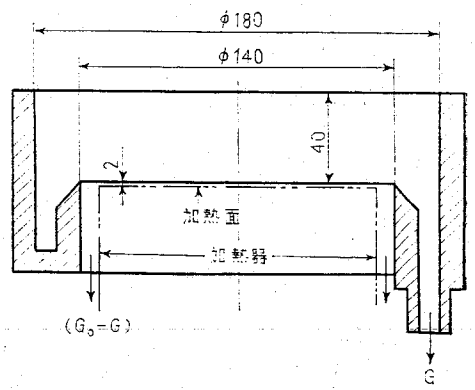


図 11　飛散液滴捕集装置

飛散する液滴流量 G を定量的に測定するため，図 11 に示される飛散液滴捕集装置を沸騰容器内に取付けた．ところで，飛散されずに加熱面周辺から流出する液膜流は，図 11 の細い矢印に沿って流出する．また飛散液滴はかなり遠くまで飛散し，加熱器上に落下するものはほとんどないことが肉眼観察によって確かめられている．

飛散液滴流量の測定は，噴流流量 G_0 を一定に保った状況下に熱流束 q を増加したときに飛散される液滴流を，飛散液滴捕集装置で捕集することによって行った．更に，加熱面周辺から流出する液膜流 (G_0-G) も同時に実測し，両者の捕集流量の和と噴流流量を比較した結果，両者の値は 5% 以内の誤差で一致していた．

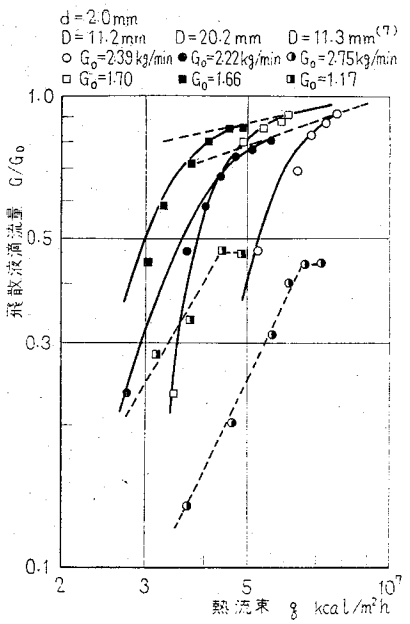


図 12　飛散液滴流量と熱流束

図 12(図中の破線は，交流加熱面で計測された値[7])は，上記方法によって計測された飛散液滴流量 G と熱流束 q との関係であるが，この図から，熱流束によって二つの特徴的な傾向に分類されることがわかる．すなわち，飛散液滴流量は，はじめ熱流束の増加に伴って急激に増加するが，$G/G_0=0.8\sim0.9$ に対応する熱流束を越えると，熱流束の増加に対する飛散液滴流量の増加割合は急激に弱まり，別の傾向（図 12 の一点鎖線で示される傾向）を示すという興味ある特性が存在する．そしてこの傾向は，同一噴流流量について一つの包絡線を形成しそうなことが予想される．バーンアウトは，この包絡線上に達した状況下に熱流束を増加すると発生することがわかった．

なお，加熱面から流出する液膜流の流量は，加熱面上での蒸発消耗液量に比べて相当大きな流量であることを付記しておく．

次に交流加熱面の結果（図 12 中の破線[7]）と本報の定常加熱面の結果を比較すると，その傾向は一見非常に異なっているように思われるが，交流加熱面の場合，加熱 1 周期中に飛散液滴の発生は，ほぼ図 13

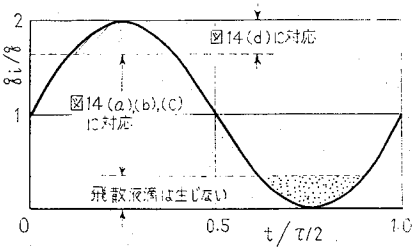
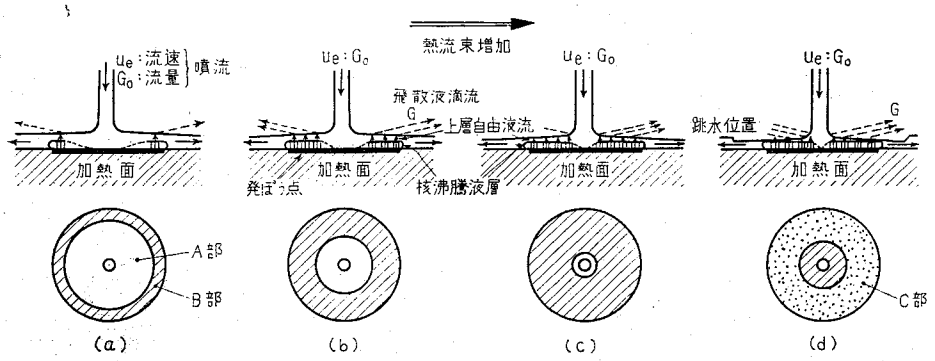


図 13　熱流束と交流電源の関係



A部：核沸騰液層がまだ十分発達せず，飛散液滴が肉眼観察できない部分．
B部：核沸騰液層内の激しい核沸騰で吹出す蒸気流によって，上層自由液流が飛散され飛散液滴流を形成している部分．
C部：上層自由液流はほとんど飛散され，核沸騰液層が加熱面上に露出している．飛散液滴流は，この部分ではほとんど観察されない．

図 14　流動モデル

(q_t：交流による瞬間熱流束，q：平均熱流束，τ：交流周期）に示される状況にあり，1周期中少なくとも2,3 割程度常に飛散の生じていない期間が存在すること，および熱流束がある値以上になると，液滴の発生が頭打ちになることを考慮すると，両者の結果の相違は決して不思議なものでないと言える．

4・3 流動モデル 本研究で取扱っている衝突噴流沸騰系の流動状況は，3章に記した沸騰曲線の特性，核沸騰液層の役割，および4・1節，4・2節に記した加熱面上の詳細な観察結果を総合するに，熱流束の増加に伴って図 14 に示される流動モデルで与えられるような変化をするものと予想される．

図 14 は，噴流（流量 G_0，流速 u_e）が加熱面にあたっている場合であるが，ここで加熱面上には核沸騰液層が存在し，これが核沸騰を支配する．そして，この核沸騰液層を覆うように流れる上層自由液流があり，この液流は下方の核沸騰液層から噴出する蒸気流の吹き飛ばし作用を受けて，無数の液滴からなる飛散液滴流を発生する．噴流流量 G_0 から飛散液滴流量 G を差し引いた流量 (G_0-G) が，核沸騰液層および上層自由液流の部分を流れ，加熱面周辺から流出している．

5. バーンアウトについて

5・1 バーンアウトの発生状況 本沸騰系で生じるバーンアウトは，水の場合，加熱面の大きさ，噴流流量および噴流速度などに関係なく，流動モデル図 14（d）に示される流動状況下に，他方 F113 の場合，噴流流量や噴流速度には関係ないが，加熱面の大きさによってかなり異なり，加熱面径 $D=11.6$ mm では図 14（c），加熱面径 $D=20.8$ mm では図 14（b）の流動状況下に熱流束を増加することによって発生するのが観察された．しかしバーンアウトの発生状況は，両液体ともほとんど同じで，加熱面周辺の一部が，白く見えるその瞬間に加熱面周辺部で液体の加熱面への接触が断たれることによって生じるように思われる．

そして液体の接触が断たれた後，水の場合加熱面全面（衝突域を除く）がただちに乾燥（膜沸騰の状態）するが，F113 の場合図 14 に示されるB部とC部でまず乾燥が起こり，時間の経過につれて乾燥部分が加熱面全面へと拡大して行く状況が観察される．

5・2 バーンアウト発生後の加熱面の汚損状態 バーンアウト発生後の加熱面の汚損（加熱面はバーンアウトごとに研摩）を観察すると，膜沸騰の生じた部分がファウリングによって汚れているのがわかる．

いまこの汚損の状態を水の場合と F113 の場合について比較すると，図 15 に例示されるように顕著な相違が認められる．すなわち F113 の場合，加熱面の汚損は加熱面の大きさによってかなり異なり，$D=11.6$ mm の場合衝突域を除く全面が汚損〔図 15（a）参照〕されるが，他方 $D=20.8$ mm の場合バーンアウト発生後，ただちに加熱を止めると加熱面中心部分に相当大きな汚損されない部分〔図 15（b）参照〕がまだ残っているが，バーンアウト発生後も加熱の状態をそのまま維持すると，衝突域を除く全面が汚損〔図 15（c）参照〕されているのが観察される．そして，この F113 における加熱面の汚損状況は，前記のバーンアウト発生後の液体と加熱面との接触不可能領域の進行状況と密接に関連していると考えられる．

一方，水の場合バーンアウト発生後ただちに加熱を止めても，加熱面径 $D=11.6$ mm および $D=20.2$ mm 共に，図 15（d）（$D=20.2$ mm の場合を例示）

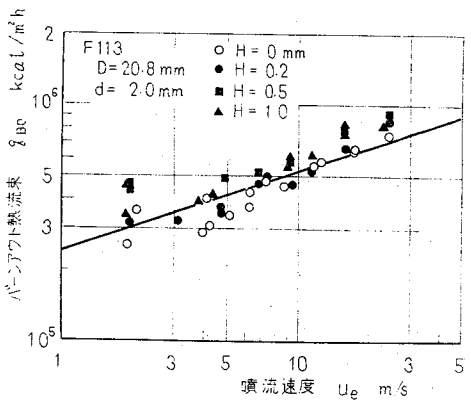
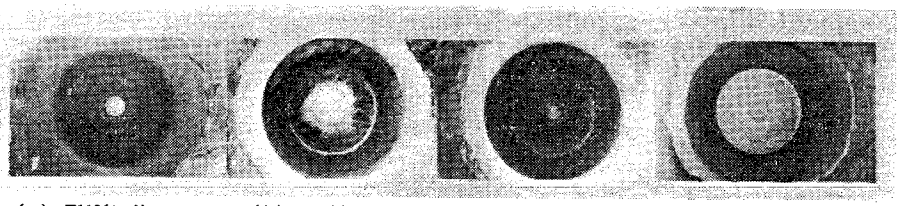


図 16 加熱面周囲の状況変化の影響



（a） F113($t\fallingdotseq 0$) $D=11.6$ mm　（b） F113($t\fallingdotseq 0$) $D=20.8$ mm　（c） F113($t\to\infty$) $D=20.8$ mm　（d） H_2O($t\fallingdotseq 0$) $D=20.8$ mm

図 15 バーンアウトによる加熱面の汚損

に示されるごとく衝突域を除く全面が汚損される．

5·3　加熱面周辺状況のバーンアウトへの影響

図 16 は，加熱面周囲に高さ $H=0.2$ mm, 0.5 mm, 1.0 mm のリング（内径 22 mm）を取付け，加熱面から流出する液膜流を人工的にせき止めた場合，バーンアウト熱流束にどのような影響を及ぼすかを示すものである．

図 16 に，リング高さ $H=0.5$ mm，1.0 mm の場合，リングの影響を受けバーンアウト熱流束が上昇することを示すが，この上昇は，上流において飛散された液滴（F113 の場合，飛散角度が小さく水平に近い状況で飛散される）がリング側壁に衝突することによって，液体が加熱面周辺から補給されるという効果によるものと推察される．そして上記結果は，バーンアウトが加熱面周辺の流れとなんらかの関連をもっていることを示しているようにも思われる．

なお水の場合，飛散液滴がリング高さ $H=1.0$ mm 程度ではリングに衝突することなく飛び越えているため，バーンアウト熱流束へのリングの影響は，ほとんど認められないことを付記しておく．

5·4　バーンアントについて　5·1，5·2 節の観測結果および 3 章，4 章を総合するに，本沸騰系のバーンアウトは，「加熱面を覆うように流れている液膜流と加熱面から発生する蒸気流との間の流体力学的要因によって，加熱面周辺部で液体が加熱面に接触できなくなる」ことによって発生するものと考えられる．

6. 結　　論

円形加熱面（定常）の中心に小さな高速噴流を置き，加熱面の大部分を大気圧下の飽和液膜流で覆わせる形の系で実現される核沸騰の実験をおこない，この沸騰系の沸騰特性および加熱面上の気液の挙動について詳細な検討をおこなった．

その結果

（1）　衝突噴流沸騰系の沸騰曲線は，噴流速度，噴流流量，加熱面径，加熱面の向きおよびガラス平板を，加熱面上 $S=0.3$ mm の位置まで接近させることによる発生蒸気の流れの変化などに支配されることなくすべて一致し，かつ通常のプール沸騰（高熱流束域の）の沸騰曲線の延長上にある．

（2）　衝突噴流沸騰系の沸騰は，通常の高熱流束のプール沸騰において，支配的役割を果たしている核沸騰液層に支配される現象で，かつこの液層内での自律的な現象と言える．

（3）　噴流速度一定の状況下に，熱流束 q を変えるとき，加熱面上の液膜流から飛散する液滴流量は，最初 q とともに急激に増加するが，やがてその増加傾向は急速に鈍り異なる傾向を示すという特性が見いだされた．そして後者の傾向は，同一噴流流量について一つの包絡線を形成しそうなことが予想される．

（4）　衝突噴流沸騰系の核沸騰および気液の流動状況などに関する諸特性は，「核沸騰液層」，「上層自由液流」，「飛散液滴流」からなる流動モデルによってかなりよく説明できる．

（5）　衝突噴流沸騰系のバーンアウトは，加熱面周辺部で液体が加熱面に接触不可能になることによって生じると予想される．

終わりに本研究をまとめるに当たってご教示いただいた東京大学　内田秀雄教授・植田辰洋教授・平田賢教授・秋山　守教授に深じんなる謝意を表する．また，本実験を行うに当たり協力された，当時東京大学卒論生であった野本秀雄君・村中正信君・長谷川　宏君・鈴木隆光君に感謝する．

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衝突噴流沸騰系のバーンアウトの研究*

(第2報，バーンアウト熱流束の無次元整理)

門出政則**，甲藤好郎***

1. 序

第1報では，衝突噴流沸騰系の熱伝達特性を明らかにすると同時に，この沸騰系における加熱面上の流動状況，バーンアウト発生状況やバーンアウト発生後の加熱面の汚損状況などの詳細な実験観察をおこない，これらの結果を基に衝突噴流沸騰系の基本的性質を明らかにした．

本報では第1報に引き続き，この沸騰系で生じるバーンアウト熱流束を支配する物理量を明確にし，これを次元解析から導出される無次元数と組合わせることによって，衝突噴流沸騰系のバーンアウト熱流束の無次元整理式の導出を試みる．更に，上記結果のサブクール条件下の衝突噴流沸騰系への適用を試みる．

なお本論文の実験値は，すべて第1報の実験装置によって計測されたものである．

主要記号

A：面積
C：定数
c_p：定圧比熱
D：加熱面径あるいは加熱面の対角線の長さ
d：ノズル径
G：重量流量
g：重力加速度
L：蒸発潜熱
q：熱流束（時間平均）
R：気液界面の曲率
S：加熱面とガラス平板のすきま幅
T：温度
u：液体の速度（$\boldsymbol{u}$：ベクトル量）
v：気体の速度（$\boldsymbol{v}$：ベクトル量）
α：べき指数
β：べき指数
λ：熱伝導率
μ：粘性係数
ρ：密度
σ：表面張力
τ：時間あるいはせん断応力
ϕ_1：サブクールの影響を示す関数

添字
BO：バーンアウト
0：基準量
l：液体
v：気体
sat：飽和状態
sub：サブクール状態
W：壁面

2. バーンアウト熱流束に及ぼす諸因子の検討

図1～3は，それぞれ上向き加熱面（図1中の記号○は，ステンレス鋼はくに交流を直接通電したものを加熱面に使用したとき計測されたバーンアウト熱流束[1]で，このバーンアウト熱流束は定常加熱面で得ら

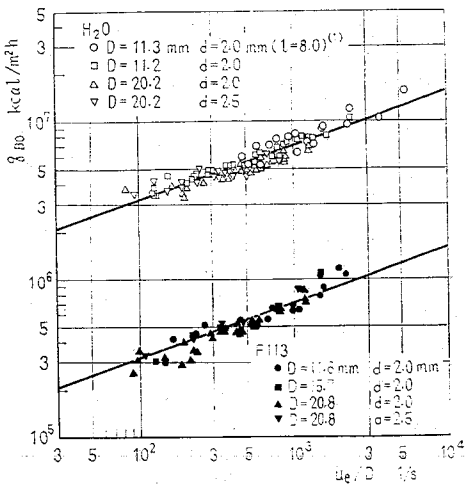

図1 バーンアウト熱流束の u_e/D による整理（上向き加熱面）

* 昭和49年5月29日 第11回日本伝熱シンポジウム，昭和50年5月13日 第12回日本伝熱シンポジウム，昭和50年10月11日 第53期全国大会講演会および昭和50年11月20日 第861回講演会において講演，原稿受付 昭和51年4月19日．

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れるバーンアウト熱流束と差異のないことを報告[1]している）の場合，下向き加熱面の場合，限定空間（ガラス平板を加熱面上すきま幅 $S=0.3$ mm, 0.5 mm の位置に取付け）の場合について，第1報の実験装置で計測された飽和条件下の衝突噴流沸騰系のバーンアウト熱流束 q_{BO} を，噴流速度（$u_e=1$ m/s～60 m/s の範囲）と加熱面径（$D=10$ mm～20 mm の範囲）を組合わせることによって得られる関数形（u_e/D）（$u_e/D=5\times10$～3×10^3 の範囲）で整理したものである．なお，これらの図中に示される実直線は，

$$q_{BO}=C\left(\frac{u_e}{D}\right)^{1/3} \quad \cdots\cdots(1)$$

で与えられる関係で，その定数は物性値によって異なり，水の場合 $C_{H_2O}=6.91\times10^5$，F 113 の場合 $C_{F113}=6.84\times10^4$ である．

これらの図から，バーンアウト熱流束は，加熱面の向きや加熱面から発生する蒸気流の変化（液膜流の厚さ 0.15 mm，蒸気流の厚さ 0.15 mm の場合，簡単な二相流の計算によれば，蒸気流によって加速される液膜流の速度は，加熱面出口での蒸気速度が 100 m/s のときでもたかだか 1 m/s 程度である）に支配されず，式(1)に示される特徴的な関数形で広い範囲にわたって良く整理されることがわかる．

ここで，この沸騰系の核沸騰を支配している核沸騰液層に注目し，この液層内の流れを支配する因子（例えば，噴流速度 u_e，加熱面から垂直に噴き出す平均蒸気速度 $q/\rho_v L$，表面張力 σ，重力 g，粘性 μ など）に起因する力の関係を考えると，上向き加熱面と下向き加熱面とでは，重力の作用のみ異なっている．したがって上記の結果は，この沸騰系で生じるバーンアウトが重力に支配されない現象であることを示すものと考えられる．

また，加熱面にガラス平板を接近（$S=0.3$ mm）しても，バーンアウト熱流束がその影響を受けないという結果は，この沸騰系のバーンアウトが加熱面上に薄く広がっている核沸騰液層[1]（高熱流束域の核沸騰を支配する液層）に支配される現象であることを示しているように思われる．

物性値を変化した場合でも，おのおのの液体に対し定数のみが異なるだけで，式(1)の重要な関係がそのまま成り立つことが図 1, 2 からわかる．

また，図 1, 3 から，バーンアウト熱流束は，同一噴流速度（u_e=一定）における噴流流量 G_0 の増加（噴流径 $d=2.0$ mm と $d=2.5$ mm の場合を比較）による影響をほとんど受けないことがわかる．そしてこの結果は，加熱面上の気液の流動状況と密接な関連があり，下記の事情によるものと考えられる．すなわち，バーンアウト近くでは核沸騰液層上を流れる上層自由液流（第1報，図 14 参照）の大部分が，加熱面の中心に近い部分で加熱面から飛散され，加熱面周辺部まで到達する上層自由液流は，わずかの流量しか存在しないという状況にあることを考慮すると，噴流速度一定の条件下における噴流流量の増加は，そのまま飛散液滴流となる上層自由液流の増加となり，核沸騰液層内の流れに何ら寄与しないという状況にある．

3. バーンアウト熱流束の無次元整理

3·1 次元解析に基づくバーンアウト熱流束の一般的検討とその応用

3·1·1 一般的検討　「加熱面上に液体が十分存在し，かつ流動しているような状況下に生じるバーンアウトは，加熱面上の液体の流れと加熱面から発生する

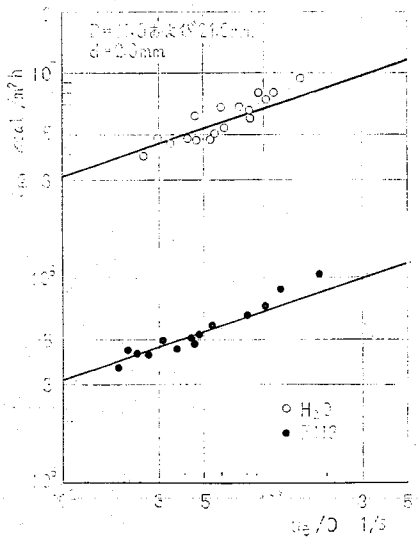

図 2　バーンアウト熱流束の u_e/D による整理（下向き加熱面）

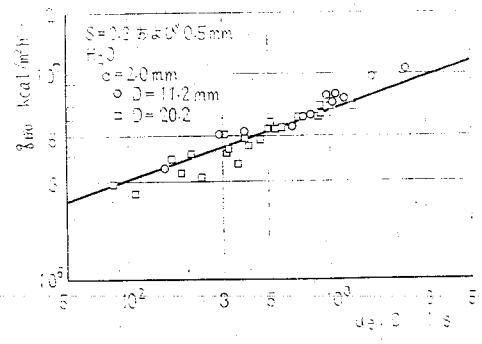

図 3　バーンアウト熱流束の u_e/D による整理（限定空間の場合）

蒸気流との間の流体力学的な力のバランスから生じる限界問題であろう」という基本的仮定に立ち，かつ問題の一般的取扱いのため加熱面上の気液二相の具体的な挙動には触れず，ただ液流と蒸気流とが分離して流れている状況下の非定常・分離二相流を考え，この二相流を支配する一般方程式（流速は，音速に比べて十分小さく，かつ非圧縮性であるとする）並びに境界・界面条件を求めると以下のように与えられる．

[基礎式]　連続の式

液相：$\mathrm{div}\,(\rho_l \cdot \boldsymbol{u})=0$ ……(2)

気相：$\mathrm{div}\,(\rho_v \cdot \boldsymbol{v})=0$ ……(3)

運動方程式

液相：$\rho_l\left[\frac{\partial \boldsymbol{u}}{\partial t}+(\boldsymbol{u},\mathrm{grad})\cdot\boldsymbol{u}\right]$

$=-\mathrm{grad}\,P_l+\mu_l\cdot\nabla^2\boldsymbol{u}+\rho_l\cdot\boldsymbol{g}$ ……(4)

気相：$\rho_v\left[\frac{\partial \boldsymbol{v}}{\partial t}+(\boldsymbol{v},\mathrm{grad})\cdot\boldsymbol{v}\right]$

$=-\mathrm{grad}\,P_v+\mu_v\cdot\nabla^2\boldsymbol{v}+\rho_v\boldsymbol{g}$ ……(5)

エネルギ式

液相：$\rho_l c_{pl}\left[\frac{\partial T_l}{\partial t}+(\boldsymbol{u},\mathrm{grad}\,T_l)\right]$

$=\lambda_l\cdot\nabla^2 T_l+\mu_l\Phi_l$ ……(6)

気相：$\rho_v c_{pv}\left[\frac{\partial T_v}{\partial t}+(\boldsymbol{v},\mathrm{grad}\,T_v)\right]$

$=\lambda_v\cdot\nabla^2 T_v+\mu_v\Phi_v$ ……(7)

ただし，Φ_l，Φ_v は散逸エネルギ

[境界・界面条件]　加熱面について

$\boldsymbol{u}=\boldsymbol{v}=0$ ……(8)

$T_l=T_v=T_W$ ……(9)

$(\pi/4)D^2\cdot q=\lambda_l\,(\mathrm{grad}\,T_l)_n\cdot A_l$

$+\lambda_v\,(\mathrm{grad}\,T_v)_n\cdot A_v$ ……(10)

気液界面について〔気液界面は，$z=g(x,y)$ で与えられるものとする〕

$P_l=P_v+\sigma\left(\frac{1}{R_1}+\frac{1}{R_2}\right)$ ……(11)

$\boldsymbol{u}=\boldsymbol{v}$ ……(12)

$(\tau_l)_{i,j}=(\tau_v)_{i,j}$ ……(13)

$w=\rho_l(\boldsymbol{u},\boldsymbol{n})=\rho_v(\boldsymbol{v},\boldsymbol{n})$ ……(14)

$L\cdot w=-\lambda_l(\mathrm{grad}\,T_l,\boldsymbol{n})+\lambda_v\,(\mathrm{grad}\,T_v,\boldsymbol{n})$ ……(15)

ただし，$\boldsymbol{n}$：界面に垂直な単位ベクトル

$\boldsymbol{n}=\frac{1}{\sqrt{g_x^2+g_y^2+1}}(-g_x,-g_y,1)$

ここで，この気液二相を支配する代表的な物理量；液体の速度：u，気体の速度：v，加熱面の長さ：D，バーンアウト熱流束：q_{BO}，液体のサブクール度：ΔT_{sub}，ある種の不確定な非定常現象の代表時間：τ，粘性：μ_l，μ_v，密度：ρ_l，ρ_v，熱伝導率：λ_l，λ_v，実験系の定常圧力：P を基準に，流れの基礎式 (2)～(7) および境界・界面を決める式 (8)～(15) を無次元化し，その中から独立な無次元数のみを選び出すと，次に示される 14 個の独立な無次元数のグループが導出される．

$$\left(\frac{P}{\rho_l u^2},\ \frac{u\tau}{D},\ \frac{q_{BO}/\rho_v L}{u},\ \frac{\rho_l}{\rho_v},\ \frac{\sigma}{\rho_l u^2 D},\ \frac{\rho_l u D}{\mu_l},\ \frac{q_{BO}/\rho_v L}{v},\ \frac{\mu_l}{\mu_v},\ \frac{gD}{u^2},\ \frac{g(\rho_l-\rho_v)D}{\rho_l u^2},\ \frac{c_{pl}\Delta T_{sub}}{L},\ \frac{\lambda_l}{c_{pl}\rho_l u D},\ \frac{\lambda_v}{\lambda_l},\ \frac{c_{pv}}{c_{pl}}\right)$$

この独立な無次元数によって決定される無次元関係式は，バーンアウトがどのような無次元数に関連する可能性があるかを示す抽象的な関係を与えるだけのもので，厳密には実際に取扱う問題に即した実験を行い，そして具体的に上記の独立な無次元数と，バーンアウトがどのようなからみ合いになっているかを検討することが必要である．

いま飽和沸騰の場合について上記結果を検討すると，液相および気相は共にほぼ飽和の状況下におかれているものとみなされることから，エネルギ式 (6)，(7) は常に満足される自明な方程式となり，この式 (6)，(7) から導出される 4 個の独立な無次元数

$$\left(\frac{c_{pl}\Delta T_{sub}}{L},\ \frac{\lambda_l}{c_{pl}\rho_l u D},\ \frac{\lambda_l}{\lambda_v},\ \frac{c_{pl}}{c_{pv}}\right)$$

は，飽和の沸騰の場合に生じるバーンアウトと独立なものとなる．

したがって飽和沸騰の条件下に生じるバーンアウト熱流束 q_{BO} は，10 個の独立な無次元数を用いて次式に示される無次元関係式

$$\frac{q_{BO}/\rho_v L}{u}=f\left(\frac{P}{\rho_l u^2},\ \frac{u\tau}{D},\ \frac{\rho_l}{\rho_v},\ \frac{\sigma}{\rho_l u^2 D},\ \frac{\rho_l u D}{\mu_l},\ \frac{q_{BO}/\rho_v L}{v},\ \frac{\mu_l}{\mu_v},\ \frac{gD}{u^2},\ \frac{g(\rho_l-\rho_v)D}{\rho_l u^2}\right) \quad \cdots\cdots(16)$$

で与えられる．

3・1・2　本解析の妥当性の検討　詳細な実験的あるいは理論的研究のなされている通常の飽和プール沸騰バーンアウトに式 (16) を適用し，本解析の妥当性を検討してみる．すなわち通常の飽和プール沸騰バーンアウトは，一般に準定常現象（非定常時間 τ が重要な支配因子でない）かつ液体の速度 u は無視できる程微小であること，更に加熱面の大きさ，粘性に支配されないことから，式 (16) において，バーンアウトに

関連しない物理量（τ, u, D, μ_l, μ_v）を消去し，そして気流の速度 v は，飽和の場合バーンアウト発生時では $v \equiv q_{BO}/\rho_v L$ という恒等的な関係が成り立つものと考えられる．

この結果，飽和プール沸騰バーンアウト熱流束は次式で与えられる．

$$\frac{q_{BO}/\rho_v L}{\sqrt[4]{\sigma g(\rho_l-\rho_v)/\rho_v^2}} = f\left(\frac{\rho_l}{\rho_v},\ \frac{\rho_l-\rho_v}{\rho_l},\ \frac{P}{\sqrt{\sigma g(\rho_l-\rho_v)}}\right) \quad \cdots\cdots(17)$$

式（17）の右辺の具体的な関数形は，飽和プール沸騰の実験を行うことによって決定される訳だが，式（17）を用いて従来の飽和プール沸騰バーンアウト熱流束[2]を整理すると，Kutateladze[2] らのプール沸騰バーンアウト熱流束の無次元整理式が導出される．

3・1・3 衝突噴流沸騰系への適用　次に飽和条件下の衝突噴流沸騰系で生じるバーンアウトについて考える．本実験は，大気圧の条件下に発生するバーンアウトのみ取扱っているため，ここでは圧力 P については一応無視して考える．また，大気圧の条件下では，一般に液体と気体の密度の間に $\rho_l \gg \rho_v$ なる関係が成り立ち，$(\rho_l-\rho_v)/\rho_l = 1-(\rho_v/\rho_l) \fallingdotseq 1$ とみなせる結果，現段階では重力を含む二つの独立な無次元数 $[gD/u^2,\ g(\rho_l-\rho_v)D/\rho_l u^2]$ のいずれか一方の無次元数について検討すれば十分である．更に本実験で取扱っているバーンアウト熱流束 q_{BO} は，時間平均についてであるため，非定常時間 τ についても一応無視して考えられそうである．

本実験の場合，液体の速度 u は噴流速度 u_e と密接に関連する量であるから，噴流速度 u_e を液体の代表速度 $u=u_e$ とする．他方気体の速度 v（加熱面から発生する蒸気は凝縮することなくそのまま蒸気流となる）については，バーンアウト発生時の換算蒸気速度 $q_{BO}/\rho_v L$ と密接に関連していることから，この速度を気体の代表速度 $v=q_{BO}/\rho_v L$ とする．

この結果，衝突噴流沸騰系のバーンアウト熱流束は，次に示される無次元関係式

$$\frac{q_{BO}/\rho_v L}{u_e} = f\left(\frac{\rho_l}{\rho_v},\ \frac{\mu_v}{\mu_l},\ \frac{\sigma}{\rho_l u_e^2 D},\ \frac{\rho_l u_e D}{\mu_l},\ \frac{g(\rho_l-\rho_v)D}{\rho_l u_e^2}\right) \quad \cdots\cdots(18)$$

で与えられる．

式（18）において，本実験によって得られた経験式（1）が指数関数で与えられていることを考慮すると，式（18）の右辺は式（19）に示される指数関数で与えられるものと予想される．

$$\frac{q_{BO}/\rho_v L}{u_e} = C_1\left(\frac{\rho_l}{\rho_v}\right)^{\alpha_1}\left(\frac{\mu_v}{\mu_l}\right)^{\alpha_2}\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{\alpha_3} \times\left(\frac{\rho_l u_e D}{\mu_l}\right)^{\alpha_4}\left(\frac{g(\rho_l-\rho_v)D}{\rho_l u_e^2}\right)^{\alpha_5} \quad \cdots\cdots(19)$$

（$\alpha_1, \alpha_2, \alpha_3, \alpha_4, \alpha_5$ は，独立なべき指数）

更に，式（19）に式（1）の関係を適用すると式（19）の独立なべき指数が1個減少し，次式

$$\left(\frac{q_{BO}/\rho_v L}{u_e}\right)\Big/\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3} = C_1\left(\frac{\rho_l}{\rho_v}\right)^{\alpha_1} \times\left(\frac{\mu_v}{\mu_l}\right)^{\alpha_2}\left\{\left(\frac{\rho_l u_e D}{u_l}\right)\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{2/3}\right\}^{\alpha_4} \times\left\{\frac{g(\rho_l-\rho_v)D}{\rho_l u_e^2}\Big/\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3}\right\}^{\alpha_5} \quad \cdots\cdots(20)$$

に示されるごとく変形される．

参考までに，経験式（1）と式（20）〔左辺に注目すると，左辺$\sim q_{BO}/(u_e/D)^{1/3}$〕を比較すると，式（20）の右辺において（$u_e/D$）なる項を含む無次元数のべき指数 α_4，α_5 の間に

$$\alpha_4 + 4\alpha_5 = 0 \quad \cdots\cdots(21)$$

なる関係が必要なことが予想されるが，更に本実験で生じるバーンアウトは，重力に支配されないという本実験の結果（2章参照）を考え合わせると

$$\alpha_4 = \alpha_5 = 0 \quad \cdots\cdots(22)$$

なる関係が成り立つことが予想される．またもし式（22）が成り立つならば，液体の粘性 μ_l がバーンアウトを支配しないことになり，液体の粘性に比べて無視できる程小さい蒸気の粘性 μ_v が，バーンアウトに重要な役割を果たすことはほとんど予想されないように思われる．

この結果，飽和の条件下に生じる衝突噴流沸騰系のバーンアウト熱流束 q_{BO} は，

$$\left(\frac{q_{BO}/\rho_v L}{u_e}\right) = C_2\left(\frac{\rho_l}{\rho_v}\right)^{\alpha_1}\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3} \quad \cdots\cdots(23)$$

（C_2 は，定数である）

で整理されることが最終的に予想される．

式（23）から，密度比（ρ_l/ρ_v）のべき指数 α_1 は，最小限2種類の物性値の異なる液体について実験を行うことによって決定されることがわかる．また粘性や重力が，前記のごとく本実験のバーンアウトを支配する物理量でないかどうかは，具体的に実験を行うことによって得られる結果を基に確かめる必要がある．

3・2 バーンアウト熱流束の支配因子の検討　図4～7は，上向き加熱面および下向き加熱面の実験で計測された水および F 113 のバーンアウト熱流束 q_{BO} を，式（20）で与えられる関数形を用いて整理したものである．なお，図4～7において，密度比のべき指数 α_1 を $\alpha_1 = 0.72$ とすると水，F 113 の両液体に対

する実験結果が一致することがわかったので $\alpha_1=0.72$ とした．また気体の粘性は，バーンアウトに関しあまり重要でない（3·1·3 項参照）ことが予想されるので，ここでは一応無視した．

図 4～7 は，本実験で計測されるバーンアウト熱流束 q_{BO} を $(q_{BO}/\rho_v L)/u_e$ の形に無次元化して表す場合，この無次元数は，密度にかかわる無次元数 ρ_l/ρ_v，表面張力にかかわる無次元数 $(\sigma/\rho_l u_e^2 D)$ と非常に密接に関連しており，他の重力や粘性にかかわる無次元数 $[g(\rho_l-\rho_v)D/\rho_l u_e^2]$，$(\rho_l u_e D/\mu_l)$ と独立な関係にあることを示している．この結果，重力および粘性が衝突噴流沸騰系のバーンアウトを支配する物理量でないことが明らかである．また，本実験の沸騰系において実現される加熱面上の激しい核沸騰流動からみて，上記の結果は妥当なように思われる．

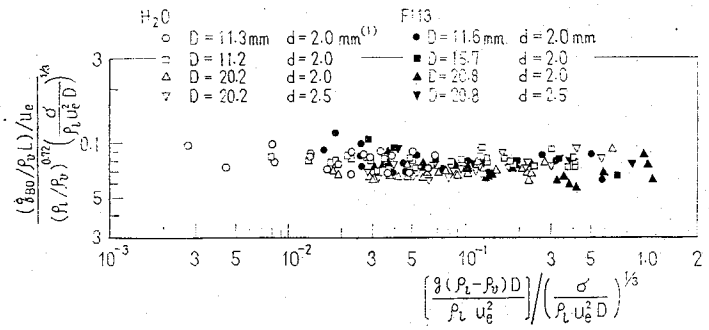

図 4　重力の影響について（上向き加熱面）

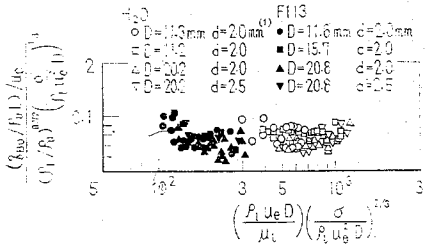

図 5　粘性の影響について（上向き加熱面の場合）

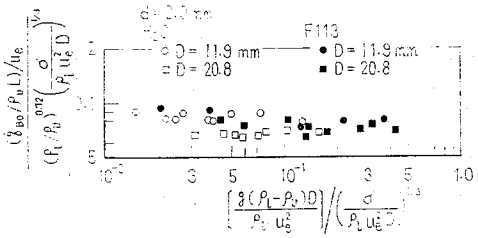

図 6　重力の影響について（下向き加熱面）

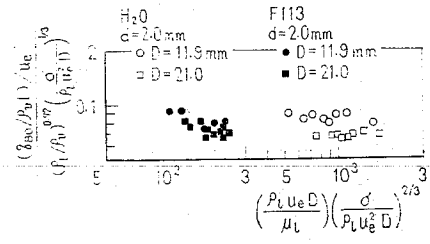

図 7　粘性の影響について（下向き加熱面）

3·3　バーンアウト熱流束の無次元整理式の導出　3·2 節に記した具体的な検討結果から，バーンアウト熱流束は最終的には式（20）から式（24）に示される関数形で与えられる．

$$\frac{q_{BO}/\rho_v L}{u_e}=7.45\times10^{-2}\left(\frac{\rho_l}{\rho_v}\right)^{0.72}\times\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/8} \quad \cdots\cdots\cdots\cdots(24)$$

この式（24）を用いて，本実験の沸騰系で生じる水および F 113 のバーンアウト熱流束 q_{BO} を再整理した結果を図 8（上向き加熱面），図 9（下向き加熱面），図 10（限定空間）に示す．

これらの図から，式（24）は本実験で計測されたバーンアウト熱流束を広い範囲にわたって良く整理することがわかる．

ところで，式（24）の密度比のべき指数の決定に関し，3·1 節に記したごとく 2 種類の異なる物性値の液体について実験すれば十分であるが，水，F 113 と異なる他の物性値についての実験結果があればより信頼性が増すことを付記しておく．

4.　サブクール条件下のバーンアウト熱流束への適用

4·1　衝突噴流沸騰系の沸騰におけるサブクールの特性

4·1·1　沸騰曲線　　図 11，12 は，それぞれ水と F 113 に対しサブクール条件下（第 1 報と同一実験装置で，飽和噴流の代わりにサブクール度 $\Delta T_{sub}=10°C\sim30°C$ の噴流を用いておこなったものである）に計測された加熱面温度と熱流束の代表的な関係を示すものである．なお図 11，12 中の実線は，それぞれ水および F 113 の飽和の条件下に得られた沸騰曲線（第 1 報の沸騰曲線と同一もの）である．

図 11，12 に示されるごとく，この沸騰曲線は水および F 113 共に十分高熱流束域では噴流速度 u_e，加熱面径 D および液体のサブクール度 ΔT_{sub} にほと

んど関係なく，核沸騰固有のばらつきの範囲内で一致していること，および核沸騰の十分発達した高熱流束域では飽和の沸騰曲線と一致することがわかる．そしてこの結果は，次の事情によるものと考えられる．すなわち，サブクールの条件下において核沸騰の十分発達した高熱流束域（飽和の場合，第1報に記したごとく加熱面上の液膜流の中に白っぽいあわ立ちができ始め飛散液滴が観察されたが，サブクールの場合も加熱面上の流動状況は飽和の場合とほとんど同一状況変化を呈することが観察される）では，加熱面に直接接する核沸騰部分は，飽和の場合と同一状況下におかれており，非常に激しい自律的な核沸騰を生じていることに起因するものである．

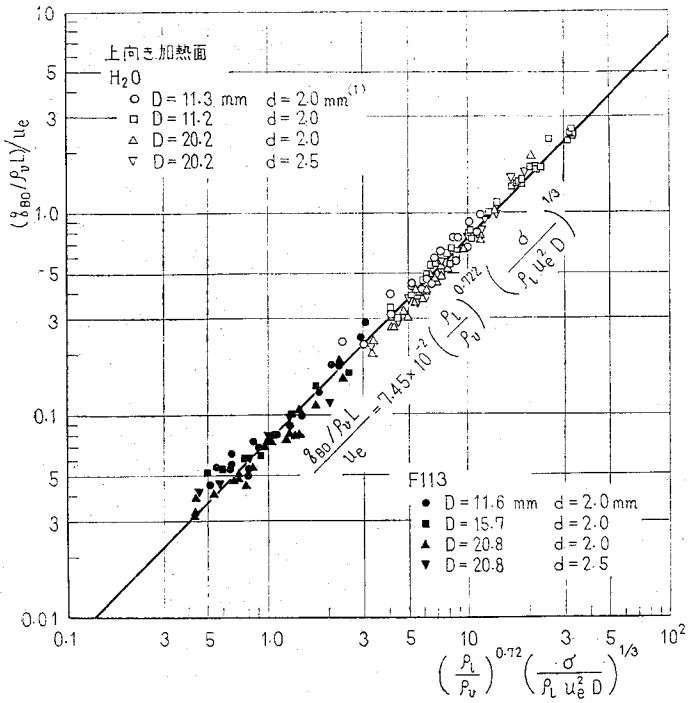

図 8　バーンアウト熱流束の無次元整理（上向き加熱面）

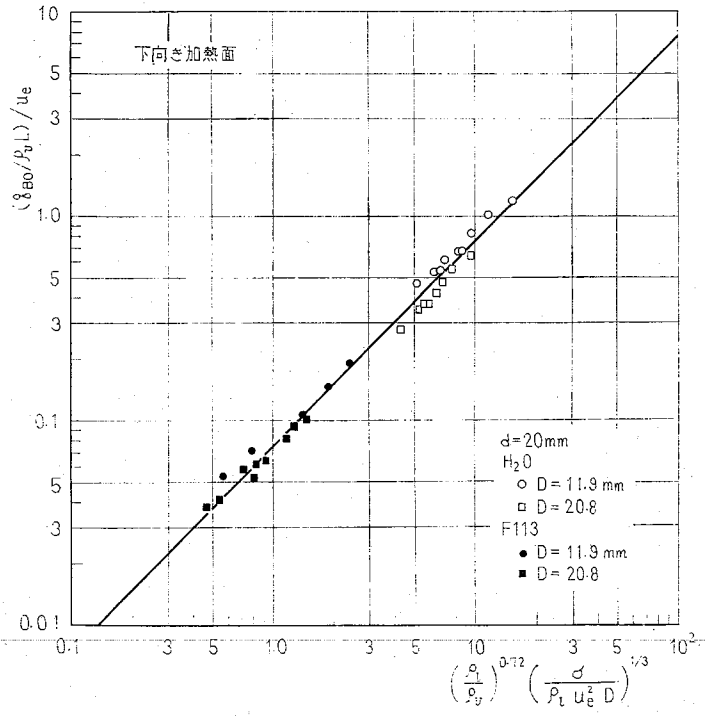

図 9　バーンアウト熱流束の無次元整理（下向き加熱面）

4・1・2　バーンアウト熱流束　図13は，サブクールの条件下に本実験で計測されたバーンアウト熱流束 q_{BO} を (u_e/D)（噴流速度 u_e，加熱面径 D）なる関数形で整理したもので，サブクールがバーンアウト熱流束に及ぼす効果を示すものである．図13中の実線は，式（1）で与えられる直線である．

図13から，サブクール度の増加に伴うバーンアウト熱流束の上昇が見られるが，これは加熱面から発生する蒸気が加熱面を覆うように流れているサブクールされた液膜流のために凝縮され，そしてサブクール度の増加が核沸騰液層への液体の供給を促進する方向に作用することに起因するものと考えられる．

しかし同一サブクール度に注目すると，水およびF 113のバーンアウト熱流束 q_{BO} は，噴流速度 u_e と加熱面径 D に関して飽和の場合とほとんど同一変化をすることが図13から明らかである．すなわちサブクールのある場合も，噴流速度 u_e と加熱面径 D に関して飽和の場合と

同一関数形

$$q_{BO}=C_3\left(\frac{u_e}{D}\right)^{1/3} \quad \cdots\cdots(25)$$

という特徴的な性格をもっている．

なお式（25）の定数 C_3 は，物性値とサブクール度によって決まる値である．

4・1・3 バーンアウトについて　前記のごとくバーンアウトに及ぼすサブクールの効果は，主に加熱面から発生する蒸気流の一部凝縮によって核沸騰液層への液体の供給を促進することにあり，バーンアウトの本質は，飽和の場合と同一機構（加熱面を覆う液膜流と加熱面から発生する蒸気流との間の流体力学的な力のバランスから生じる限界問題）であろうと予測される．

4・2 バーンアウト熱流束の無次元整理　問題にしている沸騰系がサブクール条件下の沸騰である場合，液体はサブクールの状況下にあるが，他方気体はほとんど飽和の状況下にあるとみなせ，液体のエネルギ式（6）を考慮することが必要となる．従ってサブクール沸騰系の場合，前記の飽和沸騰系のバーンアウト熱流束を支配する独立な無次元数のほかに，サブクールの影響を示す2個の独立な無次元数（$c_{pl}\Delta T_{sub}/L$, $\lambda_l/c_{pl}\rho_l u_e D$）を加えることが必要である．

したがって，本実験で得られるサブクール条件下のバーンアウト熱流束 q_{BO} は，次式に示される無次元関係式（26）

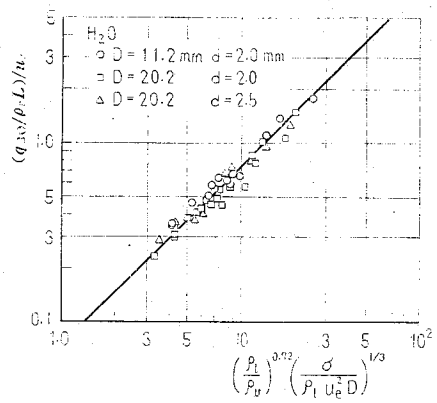

図 10　バーンアウト熱流束の無次元整理（限定空間の場合）

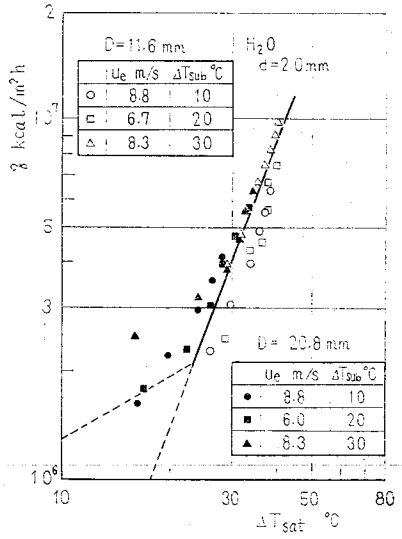

図 11　沸騰曲線

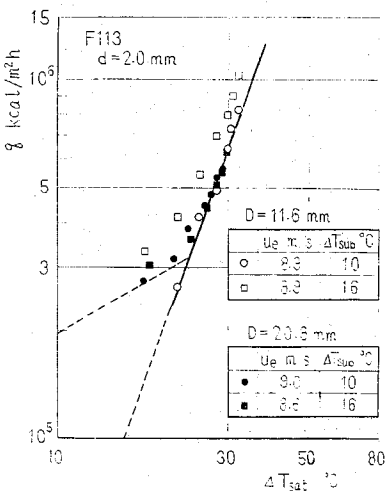

図 12　沸騰曲線

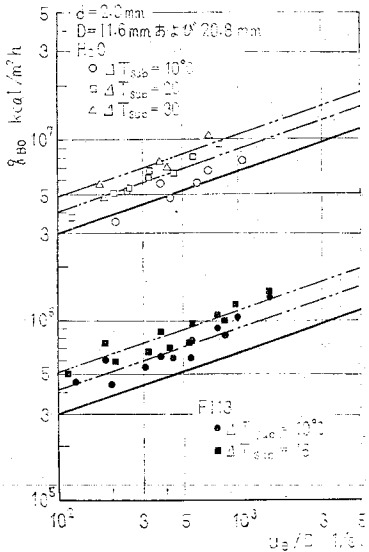

図 13　バーンアウト熱流束の u_e/D による整理

$$\frac{q_{BO}/\rho_v L}{u_e}=f\left(\frac{\rho_l}{\rho_v},\ \frac{\mu_v}{\mu_l},\ \frac{\sigma}{\rho_l u_e^2 D},\ \frac{\rho_l u_e D}{\mu_l},\ \frac{g(\rho_l-\rho_v)D}{\rho_l u_e^2},\ \frac{c_{pl}\Delta T_{sub}}{L},\ \frac{\lambda_l}{c_{pl}\rho_l u_e D}\right) \quad \cdots\cdots(26)$$

で与えられる．

上式（26）にサブクールの実験で得られた重要な関係式（25）（飽和の場合と同一関係式で定数のみ異なる）を適用すると，式（26）は次式

$$\frac{q_{BO}/\rho_v L}{u_e}=7.45\times10^{-2}\left(\frac{\rho_l}{\rho_v}\right)^{0.72}\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3}\times\phi_1\left(\frac{\rho_l}{\rho_v},\ \frac{c_{pl}\Delta T_{sub}}{L},\ \frac{\lambda_l}{c_{pl}\rho_l u_e D}\right) \quad \cdots\cdots(27)$$

に示されるごとく変形される．

次に，サブクールの影響を示す無次元関係式 ϕ_1 について考えると，三つの独立な無次元数のうち（$\lambda_l/c_{pl}\rho_l u_e D$）は，本実験範囲では 10^{-9} のオーダで他の無次元数に比較して無視できる程小さいので，現段階では無視できるものと考えられる．更に，サブクール度 ΔT_{sub} を小さくすると飽和の場合の次元関係式（24）に漸近することから，関数 ϕ_1 は次の性質をもつ必要がある．

$$\Delta T_{sub}\to 0,\quad \phi_1\to 1.0 \quad \cdots\cdots(28)$$

簡単のため，式（28）の性質を考慮して関数 ϕ_1 を指数関数で与えると，関数 ϕ_1 は，

$$\phi_1=1.0+C_4\left(\frac{\rho_l}{\rho_v}\right)^{\beta_1}\left(\frac{c_{pl}\Delta T_{sub}}{L}\right)^{\beta_2} \quad \cdots\cdots(29)$$

で与えられる．

ここで，サブクールがバーンアウトに及ぼす効果が，主に加熱面から生じる蒸気と密接にからんでいるという結果（4·1·2項参照）を考慮すると，サブクール条件下で生じるバーンアウト熱流束は，式（30）で表したほうがその物理的意味が明確になる．

$$\frac{(q_{BO}/\rho_v L)/\phi_1}{u_e}=7.45\times10^{-2}\left(\frac{\rho_l}{\rho_v}\right)^{0.72}\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3} \quad \cdots\cdots(30)$$

式（30）における関数 ϕ_1 の未知数 C_4，β_1，β_2 について，本実験で計測されたサブクール条件下のバーンアウト熱流束を式（29），（30）に基づいて最小二乗法によって決定すると定数 C_4，およびべき指数 β_1，β_2 は，それぞれ

$$C_4=2.7,\quad \beta_1=0.5,\quad \beta_2=2.0 \quad \cdots\cdots(31)$$

で与えられることがわかった．

そして図 14 は，式（29），（30），（31）を用いて本実験で計測されたバーンアウト熱流束を整理した結果を示す．この図 14 から，サブクール条件下のバーンアウト熱流束も式（30）で示される無次元関係式でよく整理されることが明らかである．

式（30）で本実験のバーンアウト熱流束がよく整理されるという上記の結果は，バーンアウト熱流束に及ぼす力学的因子（噴流速度，加熱面径，表面張力など）と熱的因子（サブクール度，比熱など）がバーンアウトに関し互いに独立に作用することを示しているように予想される．

参考までに，本実験のサブクール範囲における関数 ϕ_1 の値を表 1 に掲げておく．

この表 1 から，本実験では水の場合，約 $\Delta T_{sub}=10°C$，F 113 の場合，約 $\Delta T_{sub}=3°C$ 程度までほと

表 1

H_2O		F113	
ΔT_{sub} °C	ϕ_1	ΔT_{sub} °C	ϕ_1
3	1.004	3	1.015
10	1.038	10	1.159
20	1.151	16	1.407
30	1.341	—	—

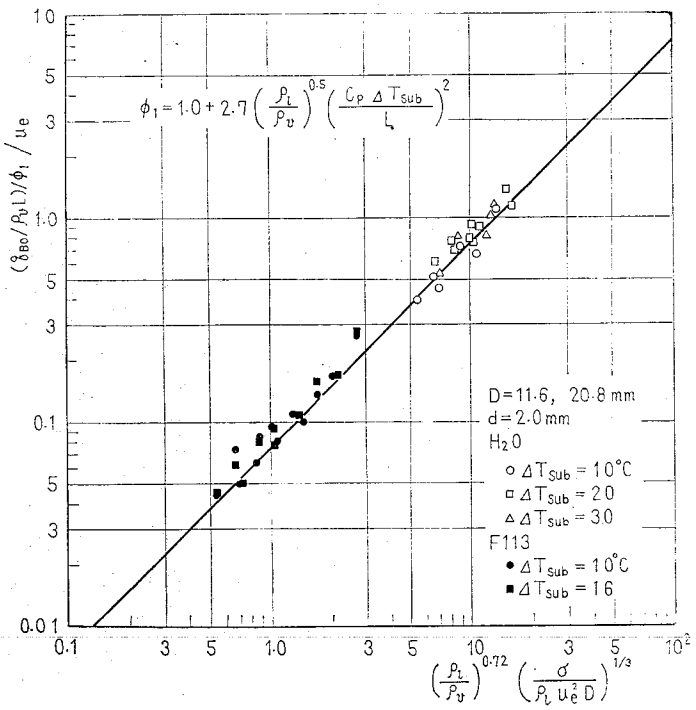

図 14　バーンアウト熱流束の無次元整理（サブクールの場合）

んど飽和状態とみなせることを付記しておく．

5. 結　　論

円形加熱面の中心に小さな高速噴流を置き，加熱面の大部分を大気圧の液膜流で覆わせる形の系において生じるバーンアウト熱流束に注目し，その特性についての検討およびこの検討結果と，次元解析から決定される無次元数とを組合わせることによって，バーンアウト熱流束の無次元整理式の導出を試みた．

その結果

（1） バーンアウト熱流束 q_{BO} と噴流速度 u_e および加熱面径 D との間には，広い範囲にわたって非常に規則的な関係，

$$q_{BO}=(\text{定数})\left(\frac{u_e}{D}\right)^{1/3}$$

があることが見いだされた．

なお，上記の関係式は物性値やサブクール度に関係なく成り立ち，この式の定数は物性値とサブクール度によってのみ決まる値である．また q_{BO} に対し加熱面の向き，噴流流量，加熱面上のガラス平板の有無は影響を及ぼさない．

（2） バーンアウトを支配する力学的因子と熱的因子とは，バーンアウトに対し互いに独立に作用し，飽和の場合バーンアウトは力学的因子にのみ支配される．

（3） 重力と粘性は，バーンアウトを支配する物理量でない．

（4） 本実験で計測されたバーンアウト熱流束は，次式に示される無次元関係式，

$$\frac{(q_{BO}/\rho_v L)/\phi_1}{u_e}=7.45\times10^{-2}\left(\frac{\rho_l}{\rho_v}\right)^{0.72}\left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3}$$

$$\phi_1=1.0+2.7\left(\frac{\rho_l}{\rho_v}\right)^{0.5}\left(\frac{c_{pl}\Delta T_{sub}}{L}\right)^2$$

で非常によく整理される．

終わりに本研究をまとめるに当たってご教示いただいた東京大学　内田秀雄教授・植田辰洋教授・平田賢教授・秋山　守教授に深じんなる謝意を表する．また，本実験を行うに当たり協力された，当時卒論生であった野本秀雄君・村中正信君・長谷川　宏君・鈴本隆光君に感謝する．

文　　献

（1） 甲藤・門出，機論，41-341（昭 50-1）306.
（2） Kutateladze, *AEC-tr*-3770, (1952), 99.

討　　論

〔質問〕　**柴山信三**（早稲田大学理工学部）

本論文は著者の実験範囲内ではよくまとめられていて，非常に有益な結果がえられたと思う．念のために次の2点についてご回答いただきたい．

（1） 序文の最後に，本論文の実験値はすべて第1報の実験装置によって計測したとあるが，第1報では 8 mm×8 mm の正方形伝熱面を使用したと書いてある．本論文で D の値が数種類あるのは，どのような寸法の伝熱面を使用したのか．

（2） 3·1·1 項に非定常分離二相流の概念から，基礎式及び条件式 (2)～(15) を記載したのは，単に有効な無次元数を誘導するためのものと解釈してよいか．

〔回答〕　（1） 本論文で使用した加熱面は，上半部が円すい形，下半部が円柱の銅ブロック（円柱部にそう入された板状電気電熱器で定常的に加熱）の上端面の円形部分である．そして，その加熱面の寸法は，D=10 mm，15 mm，20 mm の3種類を基準寸法として用いたが，バーンアウト発生ごとに加熱面を研摩するため，その直径は 0.01 mm 程度ではあるが使用ごとに大きくなる．なお，詳細については第1報(付1)の2章，実験装置と実験方法の箇所に記述しているのでご覧いただきたいと思う．

（2） ご指摘のとおり有効な無次元数を導出するためのものである．

なお，バーンアウトを支配する物理量があらかじめわかっている場合，本論文中の無次元関係式と同一関係式が π 定理から導出される訳であるが，本研究の場合最初から支配物理量が明確にされていないので，全体をはあくする意味で，本論文で行ったような方法で出発し，バーンアウト現象に関係する可能性のある無次元数を細大もらさず誘導し，各々の無次元数について検討するという手順を取った．

（付 1） 門出・甲藤，本論文集 3399 ページ．

536.423.1 : 532.525

平面噴流による強制液体供給，高熱流束沸騰系のバーンアウト*

甲藤好郎**，石井和徳***

1. 序

プール沸騰については，すでに Kutateladze[1]，Rohsenow ら[2]，Zuber ら[3]，その他によって，簡単な基本沸騰系のバーンアウト熱流束の無次元整理式（ないしそれに近いもの）が求められている．

他方，強制対流沸騰においては，原子炉設計の問題などから，流路内強制流動沸騰のバーンアウトの研究が続けられて来ている．しかし現象の複雑さのため，一般的に適用できるような無次元整理式の導出にはほど遠い状況にある．そしてこの際，もっと簡単な強制対流沸騰系のバーンアウトに眼をむけるのもむだではないであろう．最近，Lienhard ら[4]は，速度 u_0 の一様な液体の流れに直交する直径 D_0 の円柱面上のバーンアウト熱流束 $\dot{q}_{BO}$ について研究をおこない，2種類の無次元整理式を提出しているが，共に無次元数 $\dot{q}_{BO}/(\rho_v \Delta h_v u_0)$ がウェバー数 $\rho_v u_0^2 D_0/\sigma$ と密度比 ρ_l/ρ_v の関数として与えられている（記号についてはつぎの記号表を参照）．

また門出ら[5][6]は，上向きおよび下向き加熱面（円形または正方形）の中心に，小さな液体噴流が垂直にあたる沸騰系〔図1(a)〕のバーンアウトの研究をおこなった．それによれば，加熱面から激しい液体の飛散を生ずるが，加熱面上の放射状の流れが加熱面外周をこえて流出する状況下にバーンアウトを生じ，そのバーンアウト熱流束 $\dot{q}_{BO}$ は，加熱面直径（または対角線の長さ）D，噴流速度 u_e のとき次の無次元整理式でよく整理される．

$$\frac{\dot{q}_{BO}}{\rho_v \Delta h_v u_e} = 0.0745 \left(\frac{\rho_l}{\rho_v}\right)^{0.725} \left(\frac{\sigma}{\rho_l u_e^2 D}\right)^{1/3} (1+\varepsilon_{sub}) \quad \cdots\cdots(1)$$

ここに，ε_{sub} は，噴流液体のサブクール度にかかわる因子で，次式で与えられる．

$$\varepsilon_{sub} = 2.7 \left(\frac{\rho_l}{\rho_v}\right)^{0.5} \left\{\frac{c_{pl}(T_s - T_{liq})}{\Delta h_v}\right\}^{2.0} \quad \cdots\cdots(2)$$

しかし上記の沸騰系は，液体が加熱面中心から放射状に流れるという点で特殊な形式である．従って，噴流沸騰系のバーンアウトについて，より一般的な視点を得るには，図1(b)に示すように加熱面上に平行流（一次元流）がある場合のバーンアウトについて情報を得ることが少なくとも必要である．

従って本報では，加熱面（長方形）の前縁近くに薄い長方形ノズルを設け，それから流出する平面噴流によって液体が供給される沸騰系のバーンアウトについて実験をおこなった結果について報告する．

記　号

c_{pl}：液体の定圧比熱
D：円形（または正方形）加熱面の直径（または対角線）
D_0：円柱加熱面の直径
g：重力の加速度
Δh_v：蒸発熱
l：長方形加熱面の流れ方向の長さ
$\dot{q}$：熱流束
S_e：ノズル出口における液体噴流厚さ
T：温度
u_e：ノズル出口における液体噴流速度
u_0：一様な液体流の速度
η：粘性係数
ρ：密度
σ：表面張力
ϕ：加熱面に対する噴流の入射角

添字
BO：バーンアウト
l：飽和液
liq：ノズルからの流出液体
s：飽和
v：飽和蒸気
w：加熱面

2. 実験装置

実験装置の全体の系を図2に示す．加熱タンク④

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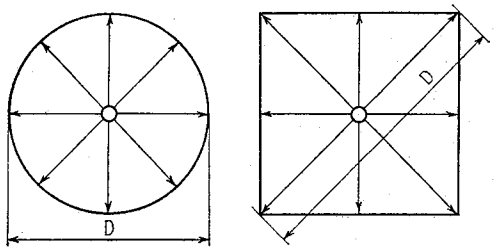
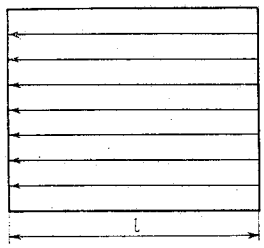
からポンプ⑥を経て長方形ノズル⑨に供給される液体は，加熱タンク用加熱器⑤および温度調整加熱器⑦によって加熱され，飽和温度 T_s より3～5°C だけ低い温度 T_{liq} で流出する．しかし，最大の温度差 $T_s-T_{liq}=5$°C の場合でも，式(2)の ε_{sub} の値は，試験液体（水，フロン 113，トリクロルエタン）に対して0.01～0.04にすぎず，従ってノズル⑨から流出する液体は飽和液とみなしても差し支えないものと推定される．

実験装置の主要部は図 3 に示す．沸騰容器①の内部は大気圧に保たれるようになっている．銅ブロック②の上部のみぞのなかに板状電気加熱器が設置され，同ブロックの下面（長方形で幅 15mm）が下向き加熱面となる．加熱面の流れ方向の長さは，3種類（l=10, 15, 20mm）について実験した．この加熱面と同じ幅の長方形ノズルから流出する平面噴流は，いわゆる衝突領域の影響を避けるため，加熱面前縁より手前約 2mm のところにあてた．加熱面上の平行流

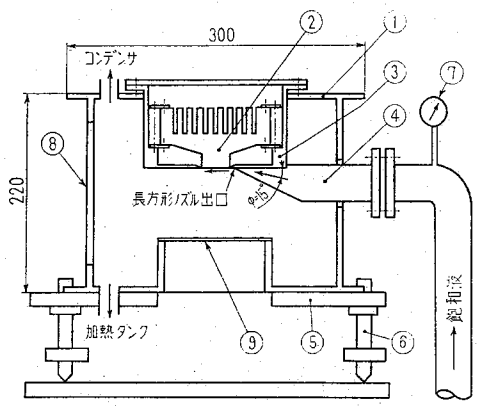


（a） 衝突噴流沸騰系



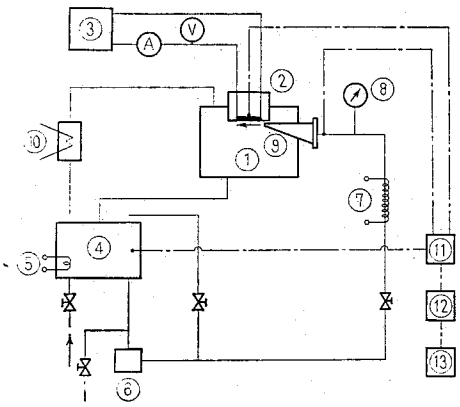
（b） 平面噴流沸騰系

図 1　2種類の沸騰系



① 沸騰容器
② 銅ブロック
③ 加熱カプセル
④ 長方形ノズル
⑤ 沸騰容器支持台
⑥ 高さ調整装置
⑦ 圧力計
⑧ 観察用窓ガラス
⑨ 観察用窓ガラス

図 3　実験装置主要部



① 沸騰容器
② 加熱カプセル
③ 定電圧装置
④ 加熱タンク
⑤ 加熱タンク用加熱器
⑥ ポンプ
⑦ 温度調整加熱器
⑧ 圧力計
⑨ 長方形ノズル
⑩ コンデンサ
⑪ アイスボックス
⑫ 多点温度記録計
⑬ ディジタルマルチメータ

図 2　実験装置全体図

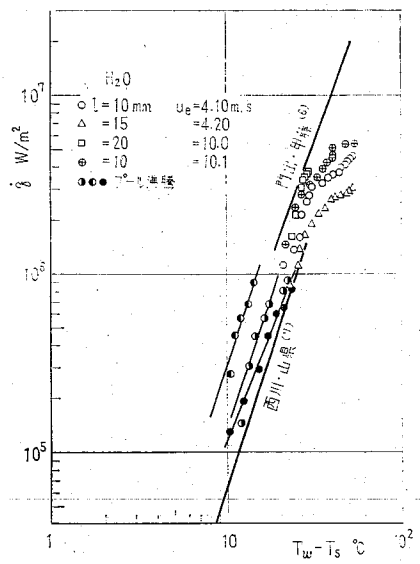


図 4　核沸騰熱伝達（水）

を保証するため，加熱面の両側面に沿い薄板を固定してある．実験は，2種類のノズル出口噴流厚さ（S_e=0.56, 0.77 mm），2種類の噴流入射角（ϕ=15°, 60°），およびノズル出口の噴流速度 u_e=1.5～15 m/s の範囲でおこなった．なお，図3は噴流入射角 ϕ=15°C の標準の場合を示し，本報の実験の大部分はこれによっておこなわれた．

加熱面温度 T_w，および加熱面をよぎる熱流束 $\dot{q}$ は，銅ブロック角柱部の中心軸上にある間隔で配置した3個のクロメル・アルメル熱電対によって測定した．なお，ここで必要な銅ブロックの熱伝導率については，高熱流束 $\dot{q}$=10^5～10^6 W/m² の飽和プール核沸騰の予備実験（図4のプール沸騰のデータ参照）を水についておこない，加熱面上に発生する蒸気が吸収する潜熱から求めてある．

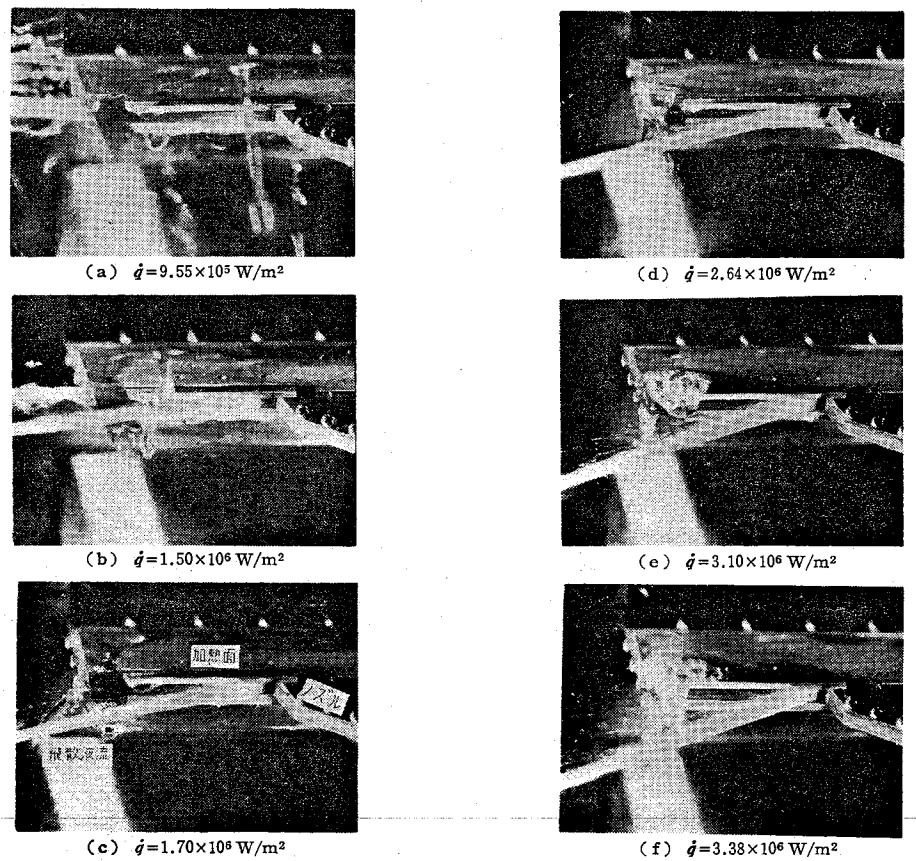
ノズル出口の液体噴流速度 u_e は，前もってノズルの流量係数を求めておけば，図3の圧力計⑦と沸騰容器①内部との間の圧力差から正確に定めることができる．

3. 核沸騰の状況

3·1 核沸騰熱伝達　2章に記した水の飽和プール核沸騰の予備実験で得られた沸騰熱伝達の実験結果（3例）を図4に示し，同時にプール核沸騰熱伝達に関する西川・山県[7]の整理式の指示する結果も示してある．

さらに図4には，本報の平面噴流沸騰系で得られた熱伝達の代表的実験例，および門出ら[6]が衝突噴流沸騰系で得た平均沸騰曲線を示してある．

図4においてプール沸騰および噴流沸騰に見られるところの核沸騰に不可避的なデータのちらばり（同程度のちらばりが以前の研究[5][6]にも現れており，かつバーンアウト熱流束には影響がないことが知られているので，細かい検討は省略した）を別にすると，図4から次の結論をしてもよいであろう．すなわち，噴流速度の実験範囲内で，核沸騰は十分に発達した状態（沸騰が非常に激しく，もはや強制対流の影響を受けない状態）に到達する．そしてその状態の沸騰曲線



（a） $\dot{q}$=9.55×10⁵ W/m²　（d） $\dot{q}$=2.64×10⁶ W/m²

（b） $\dot{q}$=1.50×10⁶ W/m²　（e） $\dot{q}$=3.10×10⁶ W/m²

（c） $\dot{q}$=1.70×10⁶ W/m²　（f） $\dot{q}$=3.38×10⁶ W/m²

水，l=10 mm，u_e=3.82 m/s

図5　流体挙動の変化

は，高熱流束域の通常のプール沸騰曲線の単調な延長上に出現する．なお，この結論はすでに文献 (5), (8) の研究で見いだした事実に一致している．また戸田ら[9]は，本報と同種の沸騰系について伝熱の測定をおこない，上記とは異なる特性を得ているが，彼らの用いた加熱面は鏡面仕上げであり，従ってその結果（核沸騰に対する）はかなり特殊なものと考えるべきであろう．

ところで，図4からさらに，本報の沸騰系の沸騰曲線は，熱流束の上昇とともに，ある程度こう配が変化する傾向が見られる．しかしこの特性は水の沸騰の場合にのみ観察された現象で，フロン113およびトリクロルエタンでは見られなかったことを記しておく．

3・2 熱流束による流体挙動の変化 図5は，長さ 10 mm の加熱面，速度 3.82 m/s の平面噴流の沸騰系で熱流束を上昇させたときの流体の状況変化（水の実験）を示す．状況 (a) では，核沸騰が弱くて液体の飛散がなく，液体は加熱面上を流れている．しかし状況 (b) では，飛散液流が加熱面後縁近くで加熱面からはく離し始めている．以下，熱流束がさらに上昇するに従い，飛散液量は増し，かつ加熱面からはく離する位置が加熱面前縁のほうへ移動すると共に飛散角度が大きくなる．そして状況 (f) は，バーンアウト点に比較的近い状況であるが，ノズルから供給される液体の大部分が加熱面前縁近くで飛散，しかしそれでも加熱面の後縁をこえて流れる残存液流の存在することが，これとほとんど同じ状況に関する文献 (6) の研究（そこでは残存液流の流量計測，および残存液流の存在確認の研究がおこなわれている）から推測できる．

フロン113およびトリクロルエタンでも，多かれ少なかれ上記と似た状況が観察される．しかし，飛散角度は水の場合より小さく，また飛散液流のはく離も水の場合ほど明確でない．これらの液体の場合は，加熱面からの蒸気のみかけ流出速度 $\dot{q}/(\rho_v \Delta h_v)$ が低く，すなわち加熱面から蒸気の噴出が弱いことに留意する要がある．

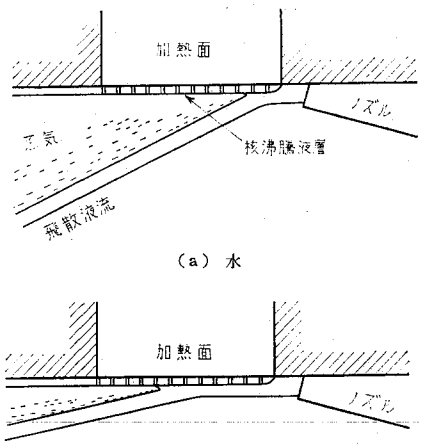


図 6 流れのモデル

4. バーンアウト

4・1 バーンアウト発生時の流体状況 3・1 節で図4について述べた沸騰伝熱の特性，3・2 節に記した写真観察および文献 (6) の研究などをもとに，バーンアウト発生の直前における流れの状況に対し，図6のような流れのモデル（関連する諸現象の統一的説明をなるべく可能にするよう仮想された状況）を考えることができる．ここで水の場合，大部分の液体は加熱面前縁近くで飛散され，加熱面は核沸騰液層（そのなかで激しい核沸騰が保たれている）に覆われ，そこから蒸気が空間に噴出している．他方，フロン（あるいはトリクロルエタン）の場合は，飛散液流のはく離点はかなり不明確，かつ加熱面前縁のほうへ移動しないうちにバーンアウトが発生する．そして，このような状況は，加熱面の長さが大きく，噴流速度が低いとき，特に著しい．

図6 (a) および (b) のいずれの状況からでもバーンアウトが発生した直後，加熱面後縁からドライアウトが始まり，加熱面前縁近くまで拡大，そして加熱面前縁近くで全液体が飛散されるようになる．

上記の状況観察は，門出ら[6]のそれ（衝突噴流沸騰系では加熱面から放射状に液体の飛散がおこるため，流れの状況観察がやや困難）にほぼ一致している．そして門出ら[6]と同様に，図6 (a), (b) 間の状況の相違はバーンアウト発生に無関係と結論できる．なお戸田ら[9]は，液体の飛散現象が，加熱面上の液膜（核沸

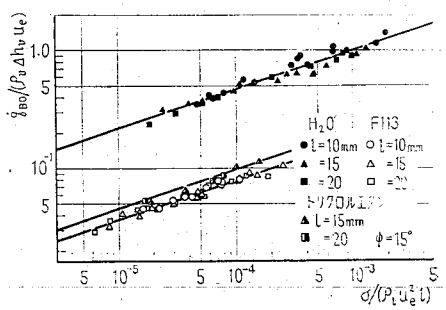


図 7 水，フロン 113，トリクロルエタンに対するバーンアウト熱流束の実験データ

騰液層）のドライアウト現象に関連をもつ重要現象であると結論しているが，液体の飛散現象自体はバーンアウト発生に無関係と考えられる．すなわちバーンアウトは，加熱面上の核沸騰液層内の流体力学的状況だけに起因して発生する現象のように思われる．

4・2 バーンアウト熱流束の実験結果 噴流入射角 $\phi=15°$ の状況下，3種類の液体に対して求めたバーンアウト熱流束 $\dot{q}_{BO}$ の全データを図7に示す．ここでは，$\dot{q}_{BO}$ に関する無次元数 $\dot{q}_{BO}/(\rho_v \Delta h_v u_e)$ を逆ウェバー数 $\sigma/(\rho_l u_e^2 l)$ の関数として示してある．ノズル出口における噴流の厚さ S_e に関連し，$\dot{q}_{BO}$ の実験データはほぼ半々くらいの割合で2群（$S_e=0.56$ mm および 0.77mm の場合）にわかれるのであるが，S_e の実験範囲で S_e は $\dot{q}_{BO}$ に影響しない（なお〔付録〕参照）ので，図7には S_e の区別を示していない．図7において，$\dot{q}_{BO}/(\rho_v \Delta h_v u_e)$ は，どの液体に対しても $\sigma/(\rho_l u_e^2 l)$ の 1/3 乗に比例する変化を示している．

4・3 液体噴流の入射角度の影響 これまで入射角 $\phi=15°$ の場合のデータだけを扱って来たが，入射角がバーンアウト熱流束に影響するか否かは重要な問題である．そこで $\phi=60°$ の場合の水の沸騰の実験を図8の装置でおこない，そのバーンアウト熱流束のデータを図9に示す．図9の斜直線は，図7で水のデータにひいた斜直線と同じものである．図9で実験データは，上記の斜直線よりわずかに偏りを示しているが，図7の水のデータの分散の程度を考えると，図9の結果からバーンアウト熱流束は噴流入射角にほとんど影響されないと結論しても重大な誤りを生ずることはないと思われる．

4・4 バーンアウト熱流束の無次元整理 バーンアウトは流体力学的な状態に関係する現象であるという立場に立って，門出ら[6]は，簡単な沸騰系のバーンアウト熱流束を支配する基本的無次元数を次のごとく与えている．

$$\frac{\dot{q}_{BO}}{\rho_v \Delta h_v u_e} = C\left(\frac{\rho_l}{\rho_v}\right)^{n_1}\left(\frac{\eta_v}{\eta_l}\right)^{n_2}\left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{n_3} \times\left(\frac{\eta_l}{\rho_l u_e l}\right)^{n_4}\left\{\frac{g(\rho_l-\rho_v)l}{\rho_l u_e^2}\right\}^{n_5} \quad \cdots\cdots(3)$$

ここに，u_e は強制流の速度，l は加熱面の代表寸法，$g(\rho_l-\rho_v)$ は液体，蒸気間の比重量差である．式(3)右辺で，第3番めの無次元数（ウェバー数の逆数）は表面張力，第4番めの無次元数（レイノルズ数の逆数）は粘性力，第5番めの無次元数は重力にそれぞれ関係するものである．なお慣性力は，上記の3個の無次元数のすべてに関係している．

プール沸騰の場合，$\dot{q}_{BO}$ に対し強制流の速度，粘性力，加熱面寸法が影響しないとすると，式(3)から直ちに Kutateladze-Zuber の形の無次元整理式が導かれる．

さて本報の強制流動沸騰系の場合，$\dot{q}_{BO}$ に対し重力が影響しないと仮定できる．さらに普通の液体では $\dot{q}_{BO}$ に対し粘性がほとんど影響しないと考えてよいであろう．そうすると式(3)から直ちに次式が得られる．

$$\frac{\dot{q}_{BO}}{\rho_v \Delta h_v u_e} = C\left(\frac{\rho_l}{\rho_v}\right)^{n_1}\left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{n_3} \quad \cdots\cdots(4)$$

そして本式を図7の3本の斜直線と比較すれば，$n_3=1/3$ と決定される．次に式(4)に残る2定数 C，n_1 については，水，フロン 113，トリクロルエタンの実験データを比較して決定することができて，結局，次の無次元整理式が求められる．

$$\frac{\dot{q}_{BO}}{\rho_v \Delta h_v u_e} = 0.0164\left(\frac{\rho_l}{\rho_v}\right)^{0.867}\left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{1/3} \quad \cdots(5)$$

図10は，式(5)を本報の全実験データと比較したものである．

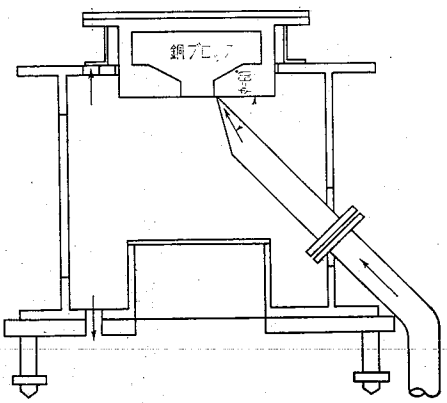


図 8 入射角 $\phi=60°$ に対する実験装置

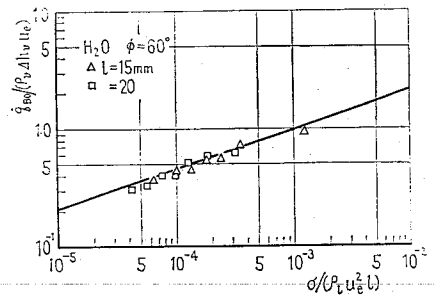


図 9 バーンアウト熱流束に対する噴流入射角の影響

5. 衝突噴流沸騰系との比較

平面噴流沸騰系に対して得られた式(5)は，衝突噴流沸騰系に対して求められた式(1)に形が似ている．特に両式でウェバー数にかかる指数が同一なのは驚くべきことのように思われる．

上記の事柄は，バーンアウト発生機構に関し，両沸騰系に強い相似性の存在を示唆しているようにみえる．ところで，式(1)，(5)を定量的に比較するためには，式(1)の中に含まれる D を $2l$ で置きかえるべきである．なぜなら，l は噴流の流入位置から測った加熱面上の最大流路長（図1参照）とみなされるからである．そして式(1)は次のように書き直される．

$$\frac{\dot{q}_{BO}}{\rho_v \Delta h_v u_e} = 0.059\,1\left(\frac{\rho_l}{\rho_v}\right)^{0.725}\left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{1/3} \cdots (6)$$

図11は，式(6)を本報の平面噴流沸騰系のバーンアウト熱流束の実験データと比較した結果である．もちろん図10のように一致は良くない．しかし同時に，式(6)は正確な整理式から遠くかけはなれているというものでもないことがわかる．結局，近似的に言えば，式(1)，(5)は形式的のみならず定量的にも近い関係にある．

6. 結　　論

（1） 長方形加熱面の前縁近傍に設けた薄い平面噴流によって，飽和液体（水，フロン113，トリクロルエタン）を供給する沸騰系に対しバーンアウト現象の研究をおこなった．

（2） 十分に発達した核沸騰（核沸騰が強く沸騰熱伝達がもはや強制対流の影響を受けなくなった核沸騰）の状況下にバーンアウトが発生する．

（3） 十分に高熱流束の発達核沸騰では，平面噴流から供給された液体の一部は，加熱面を覆う核沸騰液層から吹出す蒸気によって飛散させられる．しかし核沸騰液層自身は，バーンアウトが発生するまで，ずっと加熱面上に存在する．

（4） 平面噴流については，その流入速度だけがバーンアウト熱流束に重要な影響を持つ．噴流の厚さ（〔付録〕参照），および入射角ともにバーンアウト熱流束に影響しない．

（5） 長方形加熱面については，その長さだけがバーンアウト熱流束に重要な影響を持つ．しかしバーンアウト熱流束は，液体の粘性に影響されないことに留意すべきである．

（6） 本報の沸騰系において得られたバーンアウト熱流束の実験データは，次の無次元整理式でよく整理できる．

$$\frac{\dot{q}_{BO}}{\rho_v \Delta h_v u_e} = 0.016\,4\left(\frac{\rho_l}{\rho_v}\right)^{0.867}\left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{1/3}$$

（7） バーンアウトに関する限り，平面噴流沸騰系と衝突円形噴流沸騰系との間には親近性があるようである．

本報の実験装置は，東京大学工学部機械系三学科，機械工学実験所で製作されたものである．また本研究に対し熱田正房，笠石公平の両君の協力を得た．これらをここに記して謝意を表する．

〔付　録〕　本報では，液体噴流の流量が，加熱面に加えた全熱負荷に比べて欠乏するだけで生ずるところの簡単な形式のバーンアウトを除外している．この現象は，文献(8)の実験（同文献の図7，8において破線で囲んだデータ）でも観察されている．本報のバーンアウト熱流束 $\dot{q}_{BO}$ より低い熱流束において，この種の流量欠乏バーンアウトが発生しないための条件は容易に下記のごとく求められる．

$$\rho_l u_e S_e \Delta h_v > \dot{q}_{BO} l$$

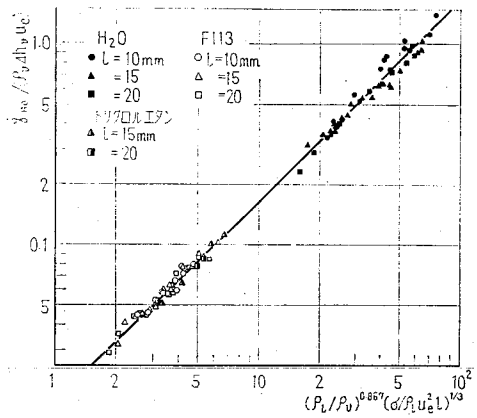


図 10　バーンアウト熱流束，無次元整理

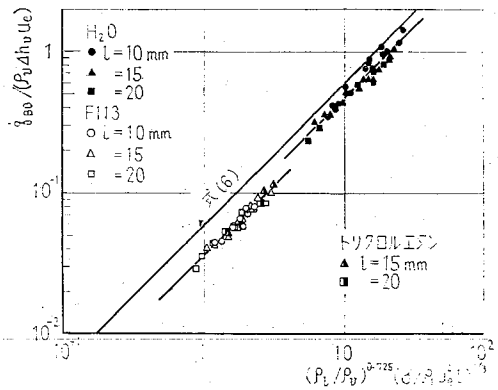


図 11　式(6)と本報の実験データとの比較

すなわち

$$\frac{S_e}{l} > \frac{\rho_v}{\rho_l} \frac{q_{BO}}{\rho_v \Delta h_v u_e} \quad \cdots\cdots(7)$$

ここで，式(5)の左辺を式(7)の右辺へ代入すれば，直ちに

$$\frac{S_e}{l} > 0.016\,4 \left(\frac{\rho_v}{\rho_l}\right)^{0.133} \left(\frac{\sigma}{\rho_l u_e^2 l}\right)^{1/3} \quad \cdots\cdots(8)$$

ところで本報では，$S_e \geqq 0.56$mm，$u_e > 1.5$m/s，20mm$\geqq l \geqq$10mmの範囲で実験がおこなわれている．ゆえに，不等式(8)の左辺の値は0.028より大であり，他方式(8)の右辺の値は，水の場合0.000 86より小，フロン113の場合0.000 62より小，トリクロルエタンの場合0.000 69より小である．従って条件式(8)は本報の実験で十分に満足されている．すなわち本実験では，流量欠乏バーンアウト発生の可能性はまったく存在していないといってよいであろう．

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Upper Limit of CHF in the Forced Convection Boiling on a Heated Disk with a Small Impinging Jet

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Critical heat flux (CHF) in the forced convection boiling on an open, heated disk being supplied with saturated liquids through a small high-speed jet impinging on the center of disk is studied experimentally, employing Freon-12 at comparatively high pressures from 6.0 to 27.9 bars as well as water and Freon-113 at atmospheric pressure. Generalized correlations of CHF are obtained for two characteristic regimes: V-regime where CHF is variable and I-regime where CHF is invariable for the change of jet velocity. Then, the boundaries of each regime are discussed clarifying the aspects for the lower limit of jet velocity capable of generating I-regime as well as the upper limit of CHF.

1. まえがき

沸騰伝熱の上限を区切る限界熱流束 (CHF) 現象は，沸騰水型原子炉，超電導装置など重要な設備の設計に大きな影響を持つ。ただ現象の複雑さのゆえに，従来，簡単な形状の加熱面上のプール沸騰の場合だけを除き，その一般性格はほとんど把握できなかった。しかし最近，ようやく強制流動を含む複雑な沸騰系の CHF に対し解明の手が及ぶに到り，本研究は，そうした研究と密接な関連を持つ基礎研究の一つである。

本研究で扱う系は，直径 D の加熱円板面の中心に，飽和液の小さな噴流（その直径 d に対し $d^2/D^2 \ll 1$）を垂直におき，加熱面上を放射状の液膜流が覆う状況下の沸騰である。この系に対し筆者らの従来の研究[1),2)]により

$$q_{co}/(\rho_v H_{fg} u) \propto (\sigma/\rho_l u^2 D)^{1/3} \qquad (1)$$

の関係が見出されている。ここに q_{co}: 限界熱流束，ρ_v: 蒸気密度，H_{fg}: 蒸発潜熱，u: 噴流速度，σ: 表面張力，ρ_l: 液体密度，D: 加熱面直径。しかし (1) 式によれば，噴流速度の増加と共に，q_{co} は限りなく上昇し得ることとなる。本研究は q_{co} の上昇限度の探究を主目的とする。

2. 実験方法

Fig. 1 に示す圧力容器の上蓋中心にはめこまれた銅ブロックの下方の円柱部分の端面が下向き加熱面 (D=0.01 m) として作用するもので，加熱は銅ブロックの上方円錐部に挿入した 11 枚の板状電気ヒータでおこなわれる。一方，Fig. 1 の圧力容器の中心軸上方のノズル (d=0.002 m) から液体噴流は上向きに流出，前述の下向き加熱面の中心に衝突した後，加熱面上に放射状の液膜流を構成する。沸騰によって発生した蒸気は，容器上部

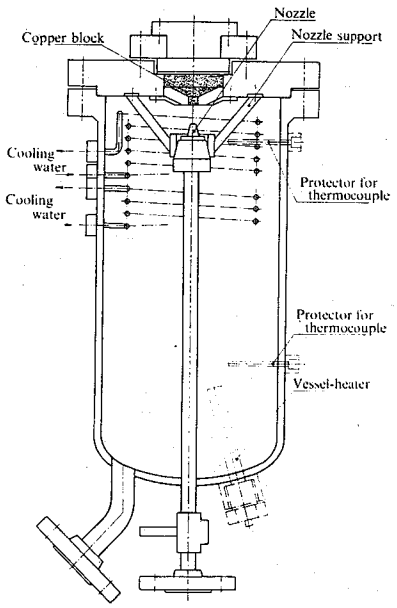

Fig. 1. Pressure vessel.

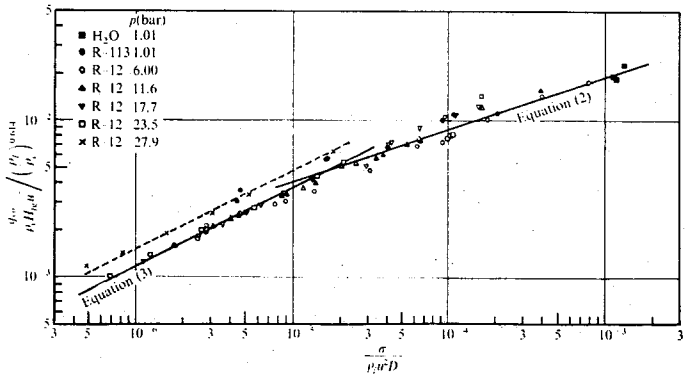

Fig. 2. Comparison of correlation equations (2) and (3) with experimental data.

の冷却コイルで凝縮され，残りの液体と共にポンプに戻される。

流体には，比較的高圧 6.0～27.9 bar の状況下の R-12 を使用した。実験装置の容量から，噴流速度は $u<20$ m/s に押さえられるが，圧力の上昇と共に表面張力 σ が低下するため，$\sigma/(\rho_l u^2 D)$ の値が減少し，噴流速度 u を増加させるのと同じ効果を発揮する。なお流体の密度比 ρ_l/ρ_v の実験範囲を拡大するため，大気圧下の水と R-113 の実験をも加えた。

3. 実験結果の整理と特性領域

Fig. 2 は本研究の実験データの整理結果である。同図の右半分にある実直線は次式の無次元整理式を与える。

$$\frac{q_{co}}{\rho_v H_{fg} u}=0.188\left(\frac{\rho_l}{\rho_v}\right)^{0.614}\left(\frac{\sigma}{\rho_l u^2 D}\right)^{1/3} \qquad (2)$$

これは式 (1) の関係に属し，q_{co} は $u^{1/3}$ に比例して変化する (variable) ので V 領域と呼ぶことにする。

一方，Fig. 2 の左半分にある実直線は次式で表わされる。

$$\frac{q_{co}}{\rho_v H_{fg} u}=1.188\left(\frac{\rho_l}{\rho_v}\right)^{0.614}\left(\frac{\sigma}{\rho_l u^2 D}\right)^{1/2} \qquad (3)$$

ここでは，u が変化しても q_{co} が変化せず (invariable)，従って I 領域と呼ぶことにする。

さて前記の V 領域より，さらに $\sigma/(\rho_l u^2 D)$ の値が大きい領域では，供給液体の全量が蒸発，加熱面がドライアウト (従って D 領域と呼ぶ) するという単純な機構の CHF が存在し，これに対し次の関係が容易に成立する。

$$\frac{q_{co}}{\rho_v H_{fg} u}=\frac{\rho_l}{\rho_v}\left(\frac{d}{D}\right)^2 \qquad (4)$$

この D 領域が出現し得る $\sigma/(\rho_l u^2 D)$ [式 (2), (4) から q_{co} を消去して下限値を知る] の範囲は，本報の $\sigma/(\rho_l u^2 D)$ の実験範囲 (Fig. 2 の横軸参照) よりはるかに大きい。

他方，Fig. 2 に示す破直線は，最高圧力 27.9 bar の R-12 の実験データに関するものである。これが式 (3) から偏倚するのは，前述の D, V, I, 領域以外の第 4 の領域が高圧力下に出現することを意味する。従って式 (3) の適用範囲は

$$\rho_l/\rho_v>6.0 \qquad (5)$$

式 (5) を仮に水の場合にあてはめると，圧力 $p<152$ bar (臨界圧 p_c を用いると $p/p_c<0.69$) の範囲で式 (3) が適用できることになる。すなわち適用範囲は相当の高圧まで及ぶ。

4. CHF の上限とそこに到達する噴流速度

式 (3) の q_{co} は CHF の上限であるゆえ，特に $q_{co,\max}$ と記すことができ，かつ式 (3) の両辺から u が消去できて

$$\frac{q_{co,\max}}{\rho_v H_{fg}\sqrt{\sigma/(\rho_l D)}}=1.18\left(\frac{\rho_l}{\rho_v}\right)^{0.614} \qquad (6)$$

また，この $q_{co,\max}$ を出現し得る噴流速度 u の下限 (それを $u_{\min}$ と記す) は，式 (2), (3) から q_{co} を消去して

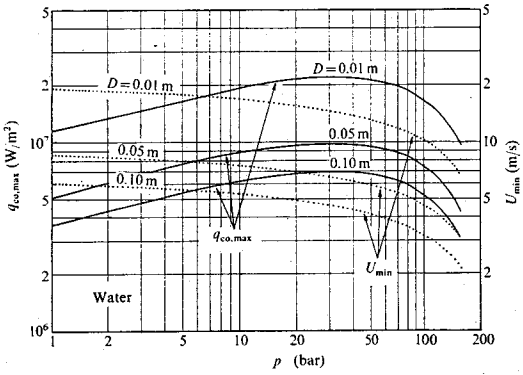

Fig. 3. Upper limit of q_{co} and corresponding lower limit of u for water boiling on heated disks.

$$\frac{\sigma}{\rho_l u^2{}_{\min} D}=1.64\times 10^{-5} \qquad (7)$$

Fig. 3 は，直径 $D=0.01$, 0.05, 0.10 m の加熱面の場合に対し，水の沸騰について式 (6), (7) から求めた $q_{co,\max}$ と $u_{\min}$ とを示すものである。比較的低い噴流速度ですでに CHF が上限に達してしまうことがみられ，CHF の向上をはかる技術の開発は容易でないことを示している。

5. 流路内の強制流動沸騰 CHF との関連

技術上，特に重要な円管，二重管，あるいは燃料集合体などの流路内の複雑な強制流動沸騰の CHF と，本報のごとき単純形式沸騰系の CHF の間の安易な比較は避けるべきである。しかし円管，二重管の CHF の実験データを筆者の一人が分析した結果[3)-4)] と比較してみると，4 個の特性領域の存在をはじめ，各領域における流速と CHF の変化傾向など，相当な近似性を持つことがわかり，従って，そのような意味からもまた，本報のごとき基本沸騰系の研究の重要性が存在している。

6. おわりに

本研究で導かれた式 (2)～(7) はすべて無次元整理式である。従来，強制流動沸騰系の CHF に関し無次元整理式の導出が困難で，従って CHF 発生機構についてもほとんど不明に近い状況であったが，考え方としては CHF に対する表面張力の影響は無視される一方，流体の粘性の影響を重視する議論が大勢を占めていたといっても過言ではない。しかし本報の結果は，こうした従来の与え方に相当の修正を必要とすること，また強制流動沸騰系の CHF にも相当の規則性が存在することを示している。

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M57 噴流沸騰系における限界熱流束

指導教官　甲藤教授
小林公正

§1 序

沸騰現象における限界熱流束は，工業上極めて重大な問題であり，特にその発生機構の解明は重要な事柄である．そこで本実験は特に次の観点により強制流動沸騰系の中の一形式である高圧下の衝突噴流沸騰系について研究した．i)衝突噴流によって形成される液膜流は，その境界条件が比較的簡単であること，ii)高熱流束域の核沸騰を支配する核沸騰液層へ強制的に液体の補給を行なうことにより，限界熱流束が著しく上昇する可能性があること．ところでこの系に関しては，門出・甲藤，甲藤・清水の研究があり，さらに前者のデータを用いたLienhard等の解析がある．

§2 実験装置および方法

実験装置の全体図を図1，主要部を図2，加熱面近傍を図3に示す．使用した液体は主にR-12である．

実験は，まずタンクヒータと凝縮器によりタンク内の雰囲気を所定圧力の飽和状態に設定する．

ついで，噴流流体をバイパスヒータとノズルヒータによりほぼタンク内の飽和温度($\Delta T_{sub}=0.2$K)まで加熱し，これを差圧変換器で目的とする噴流流速Uに設定して，加熱面下の円形ノズルから円形加熱面の中心に噴出させる．一方，加熱ブロックにはヒータ入力を加え，ブロック内温度の定常性を確認した上でこの温度測定を行なう．

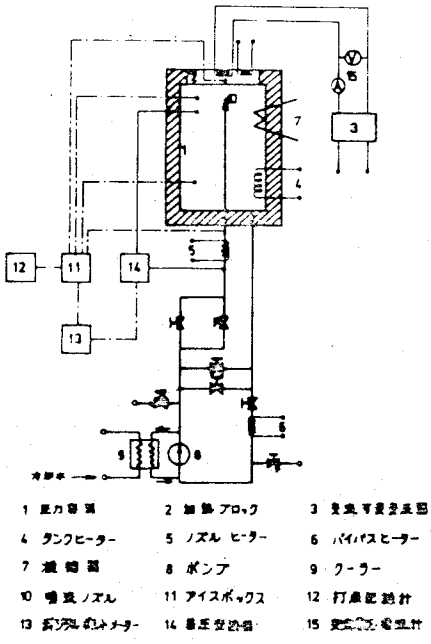


図1　実験装置全体図

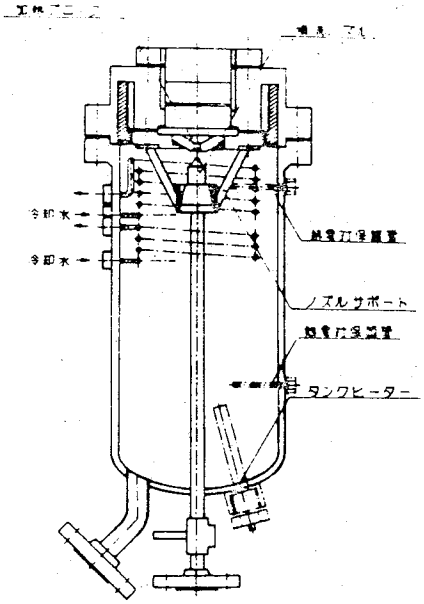
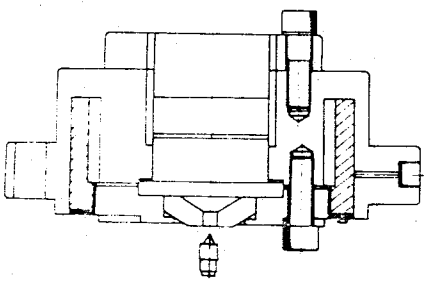


図2　実験装置主要部



加熱面直径　$D=10$mm
ノズル内径　$d=2.048$mm
ノズル・加熱面距離　10mm

図3　加熱面近傍

その後ヒータ入力をステップ状に増加させ，上記の方法により，測定を行ない，以後これを限界熱流束q_{co}を得るまでくりかえす．

§3 実験結果および考察

本実験で得られたR－12の代表的な沸騰曲線を図4に示す．

そして，限界熱流束q_{co}と噴流流速Uの関係を図5に示す．図5において限界熱流束q_{co}は噴流流速Uの増加に伴い多少の増加を示している．また圧力が増加した場合，限界熱流束q_{co}が高く出ているが，これは清水の結果と逆であり，この点に関しては更に検討が必要である．

次に図5の結果を無次元数$\{(q_{co}/\rho_v h_{fg})/U\}$と$\{\sigma/(\rho_l U^2 D)\}$で整理した関係を図6に示す．ここで，$\rho$：密度，$h_{fg}$：蒸発潜熱，$\sigma$：表面張力である．これに適合した無次元整理式は次のようになる．

$$(q_{co}/\rho_v h_{fg})/U = 4.99\{\sigma/(\rho_l U^2 D)\}^{0.528} \quad (1)$$

ところで，このような沸騰系においては清水は2つの領域に別れる事を見い出し，領域Iに対して式(2)，領域IIに対して式(3)を得ている．

$$(q_{co}/\rho_v h_{fg})/U = 0.188(\rho_l/\rho_v)^{0.614}\{\sigma/(\rho_l U^2 D)\}^{1/3} \quad (2)$$

$$(q_{co}/\rho_v h_{fg})/U = 1.18(\rho_l/\rho_v)^{0.614}\{\sigma/(\rho_l U^2 D)\}^{1/2} \quad (3)$$

そして，領域I，IIの境界は式(4)としている．

$$\sigma/(\rho_l U^2 D) = 1.64\times10^{-5} \quad (4)$$

本実験では領域IIの式(3)に近い結果が得られた．

また，Lienhard等は式(5)，(6)，(7)を得ている．

$$(q_{co}/\rho_v h_{fg})/U = f(\rho_l/\rho_v)(D/d)^3\{\sigma/(\rho_l U^2 D)\}^{A}/(D/d) \quad (5)$$

$$A(\rho_l/\rho_v) = 0.4376 + 0.1027\ln(\rho_l/\rho_v) - 0.0174\{\ln(\rho_l/\rho_v)\}^2 + 0.00426\{\ln(\rho_l/\rho_v)\}^3 \quad (6)$$

$$f(\rho_l/\rho_v) = 0.744 + 0.0084(\rho_l/\rho_v) \quad (7)$$

式(6)では，圧力P＝9.61 barの場合A＝0.426となり，かなり本実験とは異なる傾向を示す．

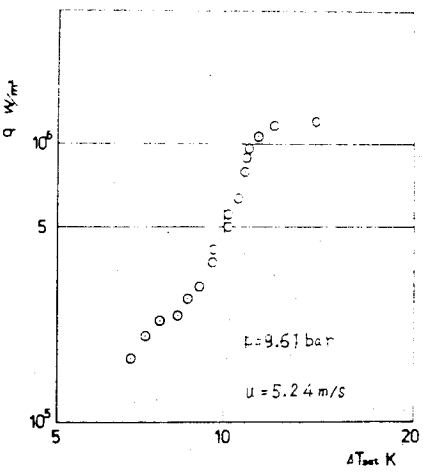


図4 R－12沸騰曲線

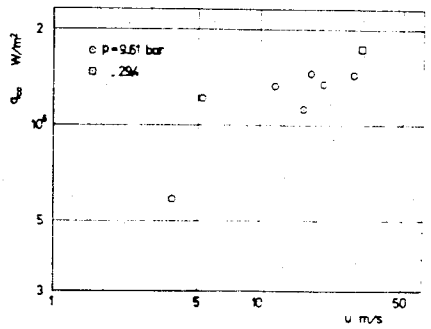


図5 q_{co}とUの関係

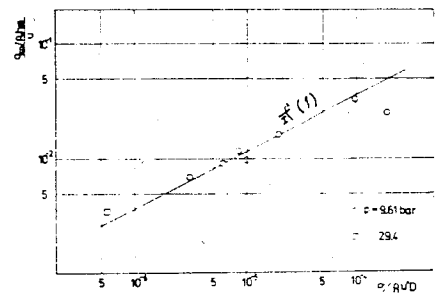


図6 $\{(q_{co}/\rho_v h_{fg})/U\}$と$\{\sigma/(\rho_l U^2 D)\}$の関係

§4 結論

円形加熱面と円形ノズルによる衝突噴流沸騰系においてR－12を用いて実験を行ない，限界熱流束q_{co}に関して式(1)の関係式を得た．

C113 平面噴流で冷却される一様加熱平板の沸騰CHFに対する速度(ウェーバ数)の影響

機正 甲藤好郎(東大工)　　機学 ※原村嘉彦 (東大工院)

1. 序　飽和・強制流動沸騰系で外部流の場合、限界熱流束 q_{co} と液体流速 u の関係を

$$q_{co} = k u^m \qquad (1)$$

と記すとき，速度指数 m に対し既存の実験データは次のような結果を示している．(A)一様流中に加熱面を沈める場合：流れに直交する円柱では $m=0.5$ (Vlietら)，$m \fallingdotseq 1/3$ (Lienhardら)，$m=0.48$ (Yilmazら)；また流れに平行な平板では，$m=0.5$ (Kutateladzeら)，$m=0.472$ (甲藤・倉田)で，これらは $u=0.3\sim10$ m/s の範囲で測定されている．(B)加熱面上の液膜流の場合：放射状液膜流では，甲藤・門出・清水らが，$u=0.16\sim60$ m/s で $m=1/3\sim0.39$ ないし、場合により $m=0$，勝田・黒瀬は $u=1\sim4$ m/s の実験で $m=0.5$；また平行液膜流では甲藤・石井[1]が $u=1.5\sim15$ m/s の実験で $m=1/3$ を与えている．以上を見るに，(A)では大部分が $m \fallingdotseq 0.47\sim0.5$ であるのに対し，(B)では大部分が $m=1/3$ の状況にあるので確認の必要がある．本報では，一様熱流束条件の正確な実現のため直流通電加熱ステンレス鋼板を用い，大気圧下の R-113 の平行液膜流に対し，$u=1.8\sim65$ m/s の広い流速範囲の実験を行ない，式(1)の m の値を検討した．この場合，鋼板の長さ ℓ が短いと，銅電極と接合する端部熱伝導損が影響するので，$\ell=20$, 40 mm を主に使用．なお参考のため $\ell=10$ mm の測定も行なった．また鋼板の厚さは，厚いと端部熱伝導損を生じ，薄すぎるとCHFへの厚さの影響が顕著になるので 0.5 mm の厚さのステンレス鋼板を使用した．

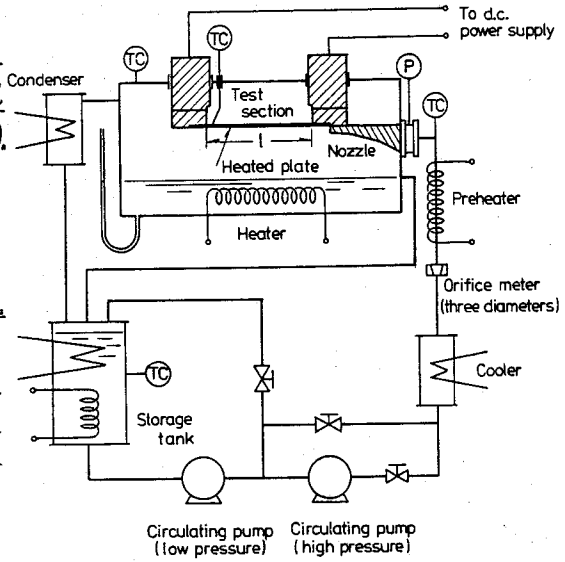


図1　実験装置

2. 実験装置　図1に装置を示す．貯タンク内のR-113は飽和温度近くに保持，高速流の実験のため低高圧2個の循環ポンプを使用，オリフィス(大中小3個)で流量測定後，予熱器で飽和温度(±0.5 K)に調整．長方形ノズル(幅16 mm，高さ $s=0.4\sim1.5$ mm)から試験加熱面(下向き，幅12 mm)に液体を供給，加熱面の両側に流れのガイド(幅16 mm)を設けて平行液膜流を保持した．CHFの検出は，ステンレス鋼板の裏面，出口端近くにスポット溶接したC-A熱電対(ϕ0.1 mm)の指示による．ステンレス鋼板に沿い長さ ℓ の間で電圧が正確に直線降下(一様加熱)することを検定．ただ両端接合部付近に僅かな遷移部分(鋼板の厚さ0.5 mmと同程度の長さ)があり，従って $\ell=10$ mm の加熱面では一様加熱性にかなりの誤差があり得るが，$\ell=20\sim40$ mm では数%以下の誤差とみてよいであろう。なお，長方形ノズルの出口高さ s は，本実験範囲内でCHFに対し影響はないものと認められた．

3. 実験結果と検討　図2に実験結果を示す．加熱面長さ ℓ 一定のとき，液体流入速度 u の広い範囲にわたり式(1)の成立がみられる．(高速領域 $u>30$ m/s でややデータのばらつきが増大しているが)．図2に示す m の値は，最小自乗法で求めたもので，$\ell=20\sim40$ mm では実質上 m の値は等しく，$\ell=10$ mm ではそれより約13%低い値を示している．また図3は，同じデータを無次元数 $q_{co}/\rho_v u H_{fg}$ 対 $\sigma/\rho_l u^2 \ell$ (ここに ρ_v：蒸気密度，H_{fg}：蒸発潜熱，σ：表面張力，ρ_l：液体密度)の形で示すもので，ℓ 一定のデータに対し次の関係を持つことは図2の結果から当然である．

$$q_{co}/\rho_v u H_{fg} \propto (\sigma/\rho_l u^2 l)^n, \quad n=(1-m)/2 \qquad (2)$$

l の大きさでデータ群が分離することが見られるが、$l=20, 40$ mm のデータ群はかなり近い。前に甲藤・石井[1]が銅ブロック加熱面（$l=10\sim20$mm）上の水, R-113 トリクロルエタン（大気圧）の実験から求めた整理式：

$$q_{co}/\rho_v u H_{fg} = 0.0164(\rho_l/\rho_v)^{0.867}(\sigma/\rho_l u^2 l)^{1/3} \qquad (3)$$

を，その R-113 の実験範囲で示すと図3の破線のようになり，本報の実験 $l=20\sim40$ mm の結果の範囲に入っている。

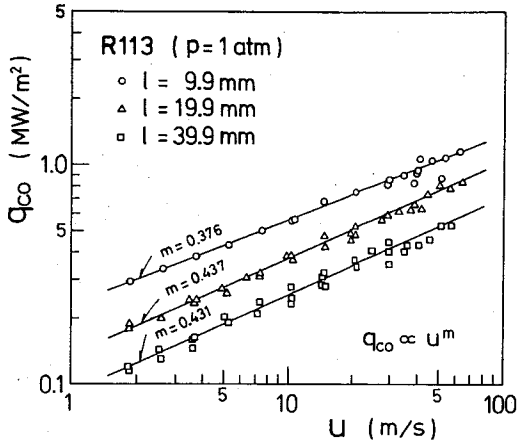


図2　実験結果（R-113, 1atm.）

なお本報の実験装置で参考のため大気圧の水に対し，$l=20$ mm，$u=3.70$, 5.44 m/s の時の2実験データを求めている。いま密度比 ρ_l/ρ_v（R-113 は 204.2 に対し水は 1603）に関し，式(3)右辺の $(\rho_l/\rho_v)^{0.867}$ で補正した水のデータを図3に ▲ 印で示す。これらは R-113 の $l=20$mm のデータ群と一致する位置にある。

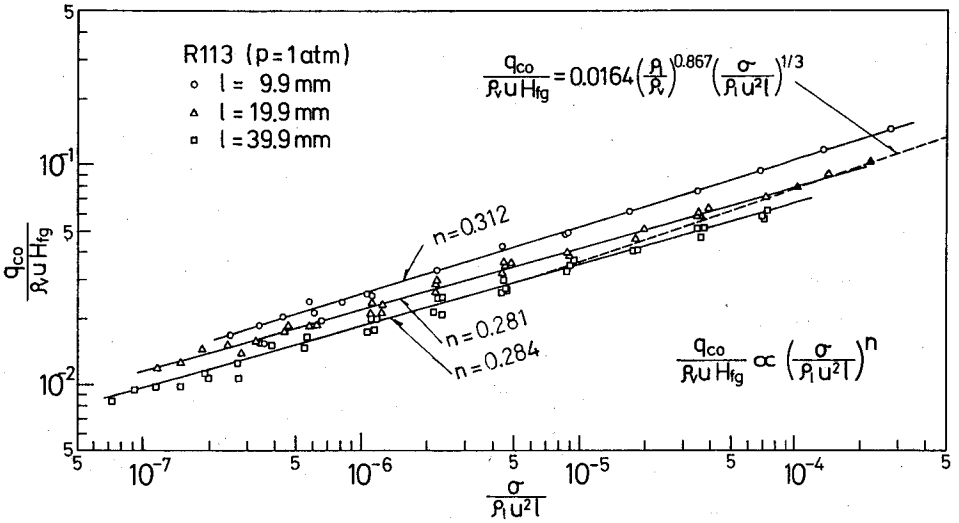


図3　実験結果の無次元表示

さて図3で $l=20$mm のデータにひいた直線は，上記の式(3)とほぼ近い（甲藤・石井の水のデータ範囲上限 $\sigma/\rho_l u^2 l < 2\times10^{-3}$ まで考慮）関係を持ち，また $\sigma/\rho_l u^2 l > 10^{-7}$ の範囲まで本報データと近似的に合うので，これをもって式(3)を置きかえるとすると，暫定的に次式が得られる。

$$q_{co}/\rho_v u H_{fg} = 0.0106(\rho_l/\rho_v)^{0.867}(\sigma/\rho_l u^2 l)^{0.281} \qquad (4)$$

4. 流路内の強制流動沸騰（H領域）のCHF　著者の一人[2~4]は最近，一様加熱の流路内・強制流動沸騰のCHFデータの無次元整理を試み，H領域の基礎限界熱流束 q_{co} に対し次の諸式を与えた。

円管：　$q_{co}/GH_{fg} = 0.10(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{1/3}/(1+0.0031\, l/d)$　(5)

内管加熱二重管：　$q_{co}/GH_{fg} = 0.12(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{1/3}/(1+0.0081\, l/d_{he})$　(6)

長方形管：　$q_{co}/GH_{fg} = 0.15(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{1/3}/(1+0.0077\, l/d_{he})$　(7)

ここにGは質量速度（流路入口の飽和液流入速度 u に対し $G=u\rho_l$），d_{he} は加熱相当直径である。

ところで，著者の一人[5]によれば，上記のH領域のCHFは，流路の出口端で環状噴霧流の流動様式の状況下で生じているものである。この意味で，本報で扱っている平行液膜流と多少の類似性（本報では液滴のデポジションなし）がある。実は式(3)を $G=u\rho_l$ で書きかえると直ちに次式となる。

$$q_{co}/GH_{fg} = 0.0164(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{1/3} \qquad (8)$$

この式(8)を式(5)～(7)と比較すると，ρ_v/ρ_l，$\sigma\rho_l/G^2 l$ の各指数が共通の値になっている。しかし本報の研究の結果，液膜流に対し，式(3)よりむしろ式(4)の方が妥当とされる結果を得た。従って流路内

の強制流動沸騰（H領域）のCHFについて、式(5)～(7)の逆ウェーバ数指数の値を再検討しておく必要がある．

さて図4，5，6は，それぞれ円管，内管加熱二重管，長方形管のH領域のCHFの既存データ（甲藤[2~4]で用いたデータ）を図示し，ウェーバ数による変化をみたものである．これらのデータの ρ_v/ρ_ℓ，ℓ/d（または ℓ/d_{he}）の変化範囲は下記のとおりである．

	$(\rho_v/\rho_\ell)\times 100$	
（円管）		ℓ/d
H_2O	0.0623 ~ 34.7	20~500
H_2	6.80 ~ 7.10	24~ 28
He	15.1 ~ 18.3	50~ 51
R-12	2.41 ~ 9.38	33~337
R-21	2.40 ~ 4.85	151~318
R-22	2.10 ~ 8.36	150
R-114	4.86 ~ 8.46	65~150
（二重管）	$(\rho_v/\rho_\ell)\times 100$	ℓ/d_{he}
H_2O	2.12 ~ 16.0	18~433
R-12	3.60 ~ 8.49	85~310
R-114	4.81	90~180
（長方形管）	$(\rho_v/\rho_\ell)\times 100$	ℓ/d_{he}
H_2O	2.61~ 13.6	26~500

図4～6の結果によれば，式(5)~(7)右辺の逆ウェーバ数 $\sigma\rho_\ell/G^2\ell$ の指数を1/3より小さくする必要はないように見える．なお図4，5でフレオン類のデータが式(5)，(6)の予測より多少低く現われているが，これは文献[2,3]ですでに指摘されている特性である．

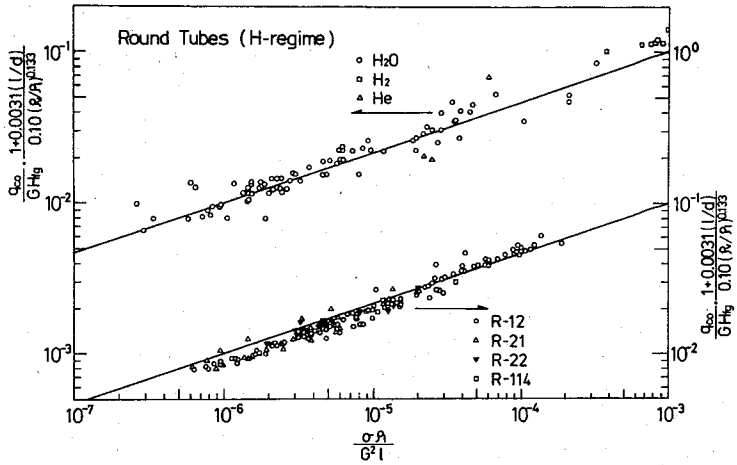


図4　円管内の強制流動沸騰のCHF（H領域）

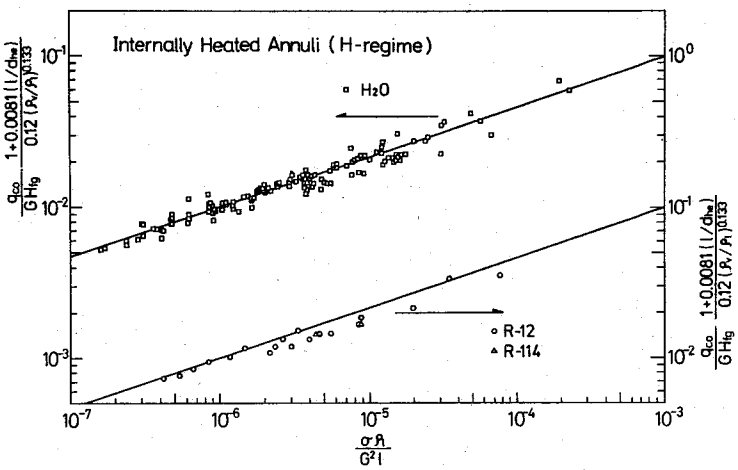


図5．内管加熱二重管内の強制流動沸騰のCHF（H領域）

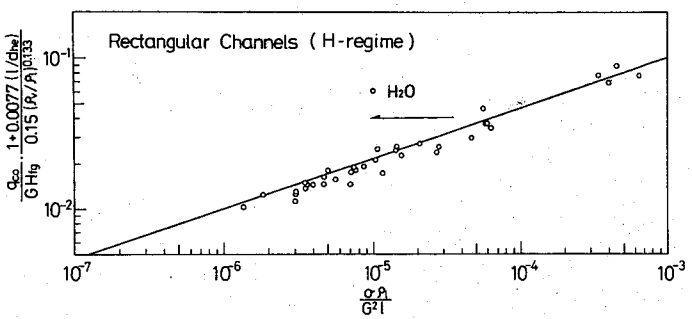


図6　長方形流路内の強制流動沸騰のCHF（H領域）

5. むすび　一様加熱される長方形加熱面の飽和液・平行液膜流について，流入速度 $u=1.8\sim65$ m/s に対しCHFの精密な実験を行ない，加熱面長さ ℓ によってデータが多少分離すること，および式(1)の速度指数が1/3より確かに大きい（1/3と1/2のほぼ中間の値）ことを見出した．また，これに関連して一様加熱流路内の飽和強制流動沸騰のH領域のCHFデータの再検討を行ない，現状では逆ウェーバ数の指数nが $n=1/3$ のままでよいことを示した．

文献　(1) 甲藤・石井：機論44（昭53），2817．(2) 甲藤：機論46（昭55），1721．(3) 甲藤：機講論 No.800-13（昭55），70　(4) 甲藤：機講論 No.818-1（昭56）131．　(5) 甲藤：機論47（昭56）220

強制流動沸騰

内部流

536.423.1 :: 532.542

一様加熱垂直円管内の強制流動沸騰の限界熱流束の無次元整理*

甲　藤　好　郎**

1. 序

一様加熱垂直円管内の強制流動沸騰の限界熱流束（以下 CHF と略記）は，原子炉の設計その他，エネルギ問題に関連して工学上，重要な基本的問題の一つであり，多くの研究がおこなわれている[1]~[4]．本報では加熱管入口で流体がサブクール液（極限として飽和液を含む）状態で蒸気を含まず，かつ安定流の場合を扱う．これに対し Macbeth ら[5]は，水の CHF の実験データ（圧力範囲 1~211 kg/cm²）を広くしゅう集，数個の代表的圧力に対し各圧力ごとに実験整理式を作成した．その後，Becker ら[6]は，圧力範囲 142~204 kg/cm² の水の実験をおこない，上とは別種の実験整理式を作成した．他方，実験の容易なフロンによって設計資料を得る目的から，フロンによる CHF の実験，および水の場合への変換尺度の研究が Barnett ら[7]，Stevens ら[8]，Dix[25] などによっておこなわれている．また CHF に関する解析的研究も Hewitt ら[9]などによっておこなわれて来ている．しかし上記の諸研究にかかわらず，任意の流体に適用できる CHF の無次元整理式はまだ導かれていない．

さて最近，門出，石井，著者ら[10][11]により噴流沸騰系の CHF の実験がおこなわれ，その無次元整理式が求められた．そこで，この結果と方向性次元解析を一つの手がかりとして，各種流体（水，フロン R 12, R 144, R 21，液体窒素，液体パラ水素，液体ヘリウム I）の CHF の実験データの分析，および無次元整理式の導出を試みてみた．本報はその結果について報告するものである．

記　号

d ：加熱管内径
G ：質量速度
h_{fg} ：蒸発潜熱
Δh_i ：入口サブクールエンタルピ
K ：Δh_i の影響に関係する係数，式 (19)
l ：加熱管長
p ：圧力
q_c ：限界熱流束（q_{c_0} は $\Delta h_i=0$ の場合の値）
u ：流入速度
ρ_L ：液体密度
ρ_V ：蒸気密度
σ ：表面張力
μ ：粘性係数
χ_{ex} ：出口クオリティ

2. データ分析の手がかりに関する考察

飽和液の沸騰の場合を考え，その限界熱流束 q_{c_0} が気液二相の運動の流体力学的条件から決まるという場合を想定してみると，これに関係する物理量は普通，流体代表速度 u，加熱面代表寸法 l のほかに h_{fg}, ρ_L, ρ_V, σ, μ_L, μ_V，および重力の作用 $g(\rho_L-\rho_V)$ である[10]．ただし通常，q_{c_0} に対し粘性は大きな作用を持たないから，μ_L, μ_V を無視し，残りの量について次元解析をおこなうと容易に

$$q_{c_0}/(\rho_V h_{fg} u)=f\{\rho_L/\rho_V,\ \sigma/(\rho_L u^2 l),\ g(\rho_L-\rho_V)l/(\rho_L u^2)\} \quad \cdots\cdots(1)$$

さてプール沸騰の場合は，式 (1) に含まれる各項から u が消える．従って次の関係が得られる．

$$q_{c_0}/(\rho_V h_{fg})=\sqrt[4]{\sigma g(\rho_L-\rho_V)/\rho_V{}^2}\times f\{\rho_L/\rho_V,\ g(\rho_L-\rho_V)l^2/\sigma\} \quad \cdots\cdots(2)$$

加熱面寸法 l が非常に小さい場合を除き，一般に右辺 $f\{\}$ 内の第2項はほとんど影響せず，式 (2) は Kutateladze[13]，Zuber ら[14] の式の形となる．他方，強制流動沸騰の場合は式 (1) から g が消えて直ちに

$$q_{c_0}/(\rho_V h_{fg} u)=f\{\rho_L/\rho_V,\ \sigma/(\rho_L u^2 l)\} \quad \cdots\cdots(3)$$

これに対し，加熱面に平行に一次元噴流（流入速度 u）が流入する場合の実験[11]では次式が得られている．

$$q_{c_0}/(\rho_V h_{fg} u)=0.016\,4(\rho_L/\rho_V)^{0.867}\sqrt[3]{\sigma/(\rho_L u^2 l)} \quad \cdots\cdots(4)$$

さて，ここで式 (2), (3) に対し方向性次元解析[12]の立場からの考察を加える．図1に示す気液二相界面

* 昭和53年4月3日　第55期通常総会講演会において論文講演として講演，原稿受付　昭和52年10月3日．
** 正員，東京大学工学部（東京都文京区本郷 7-3-1）．

(重力に垂直な場合) の微小波動において，表面張力 σ および重力の加速度 g は流体の圧力を介して波動に関与し，そして図1のごとく座標 x, y (紙面に垂直に z) を定めるとき，上記の波動に関する理論[15]から直ちに σ および g は次元を持つことが知られる．

$$[\sigma]=ML_y^2/(T^2L_xL_z),\ [g]=L_y/T^2 \quad \cdots\cdots(5)$$

ここに M, T は質量，時間の次元，L_x, L_y, L_z はそれぞれ x, y, z 方向の長さの次元を示す．次に，図1の二相界面に対し，重力方向 (図1の破線の g) が平行な場合を考える．この時，g は波動に関係せず，従って通常の g の次元を考えるべきであって $[g]=L_y/T^2$ となる．これは形式上，式(5)の第2式と同じである．つまり界面が重力に対し垂直，平行どちらでも式(5)が成立する．

図2 (a) および (b) は，水平上向き，および垂直加熱面上のプール沸騰を示す．両者共に，加熱面から垂直に吹き出す蒸気に対して考えられる二相界面に式(5)が適用され，また他の物理量の次元は，H を熱量の次元として直ちに，$[q_{c_0}]=H/(TL_xL_z)$，$[h_{fg}]=H/M$，$[\rho]=M/(L_xL_yL_z)$．ゆえに

$$[q_{c_0}/(\rho_V h_{fg})]=L_y/T,\ [\sigma g(\rho_L-\rho_V)/\rho_V^2]=(L_y/T)^4,\ [\rho_L/\rho_V]=1 \quad \cdots\cdots(6)$$

従って式 (2) (l の影響が無視できる場合) は両沸騰系に対し，真に無次元であるということができる．

次に図2(c)は，長さ l の加熱面に平行に速度 u で流体が流入する場合の強制流動沸騰を示す．加熱面に対し垂直な蒸気吹出しで生ずる二相界面について，σ は式(5)第1式の次元，また $[q_{c_0}]=H/(TL_xL_z)$，$[h_{fg}]=H/M$，$[\rho]=M/(L_xL_yL_z)$，$[u]=L_x/T$，$[l]=L_x$ の各次元を考慮すると

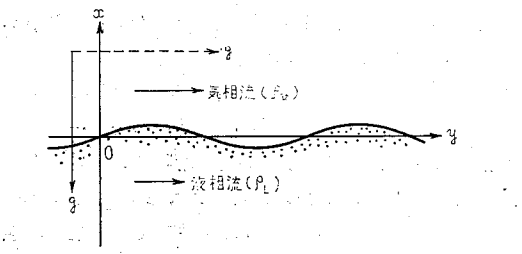


図1　二相界面の微小波動

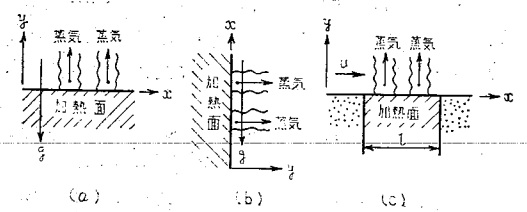


図2　3種の基本沸騰系

$$[q_{c_0}/(\rho_V h_{fg} u)]=L_y/L_x,\ [\rho_L/\rho_V]=1,\ [\sigma/(\rho_L u^2 l)]=(L_y/L_x)^3 \quad \cdots\cdots(7)$$

従って式(3)が真に無次元形式であるためには，$q_{c_0}/(\rho_V h_{fg} u)=\sqrt[3]{\sigma/(\rho_L u^2 l)}\, f(\rho_L/\rho_V)$ の形をとるべく，これは実験で得られた式(4)の形にほかならない．そして以上，図2(a)～(c)について得られた結果は，CHF の解析に方向性次元解析がある程度有効であることを示すものといえるであろう．

さて，内径 d，加熱長 l の一様加熱円管に液体が速度 u で流入している場合，$d\to\infty$ の極限を考えると，そのときの状況は図2(c)の状況にほかならず，従って CHF に対し式(4)に近い整理式が成立する可能性がある．一方，$d\to 0$ の極限を考えると，図3に示すような状況で CHF が生ずるようになろう．この時の二相界面は加熱面に平行であり，これに方向性次元解析を適用すると

$$[q_{c_0}/(\rho_V h_{fg} u)]=L_y/L_x,\ [\sigma/(\rho_L u^2 l)]=1 \quad \cdots\cdots(8)$$

従って，この場合の真に無次元な形式の整理式としては次式が想定される．

$$4q_{c_0}l/(\rho_L h_{fg} u d)=\text{const.}\,(\rho_L/\rho_V)^a,\ \{\sigma/(\rho_L u^2 l)\}^b \quad \cdots\cdots(9)$$

式(9)で左辺は管出口のクオリティ χ_{ex} に等しい量である．そして，d に関する前記の両極限で，式(4)と(9)の形にそれぞれなるような簡単な形の式を考えてみるのに，次式はその一つになり得る．

$$q_{c_0}/(Gh_{fg})=\text{const.}\,(\rho_V/\rho_L)^{0.133}\{\sigma\rho_L/(G^2 l)\}^{1/3}/[1+\text{const.}\,(\rho_L/\rho_V)^m\{\sigma\rho_L/(G^2 l)\}^n l/d] \quad \cdots\cdots(10)$$

ここで G は，$G=\rho_L u$ (本報では管入口で液体状態を扱っている)，すなわち質量速度である．まだ式(10)の当否はわからないが，一つの手がかりとして試みる価値はあろう．

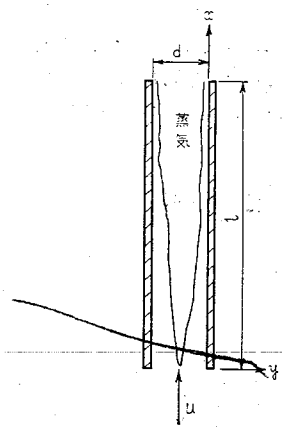


図3　加熱円管 ($d\to 0$ の極限)

3. q_{c_0} に関するデータ分析と無次元整理

本章では式 (10) を手がかりに，$(\rho_V/\rho_L)^{0.133}\{\sigma\rho_L/(G^2l)\}^{1/3}/\{q_{c_0}/(Gh_{fg})\}$ を，ρ_L/ρ_V，$\sigma\rho_L/(G^2l)$，l/d の関数とみなして実験データの分析を試みる．ここで q_{c_0} は次の方法で求める．すなわち質量速度 G を一定にして，入口サブクールエンタルピ Δh_i を変える時，限界熱流束 q_c は，多くの場合，図4のaのごとく Δh_i に比例して変化するが，l/d が小さい場合などは図4のbのような変化をすることが知られている[3]．しかしいずれの場合でも，十分な個数の実験データがあれば，$\Delta h_i \to 0$ の外そうにより q_{c_0} の値を正確に求めることができる．そして，水に関し文献(5)（その表 1～10），文献(6)（その図12のサブクール 10°C のデータ），文献(16)，文献(17)（その表Ⅳ），文献(18)（その図3），文献(19)～(22)，フロン 12 に関し文献(23),(24)，フロン114に関し文献(25)（その付録データ表で CHF×10^{-4} は CHF×10^{-5} の誤り），フロン21に関し文献(26)，液体窒素，液体水素（パラ水素）に関し文献(27)，（本文献で Max，CHF と称しているもの）のデータから q_{c_0} を求めた．

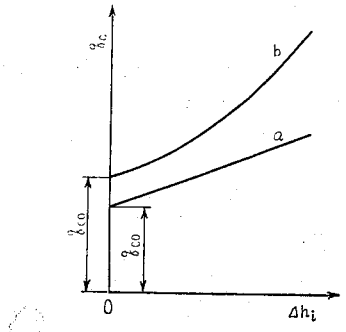


図 4　q_c と Δh_i の関係

3・1　H 領域　図5は $\sigma\rho_L/(G^2l)=10^{-7}$～$10^{-3}$ の領域で見いだされた結果（水，フロン）の代表例で，l/d＝一定のデータに対し縦軸の値がほぼ一定に保たれるという特色を持つ．これは式 (10) の導出で想定した流体力学的状況に近い状況にあると推定される．これに属する実験データは次の無次元整理式でまとめられる．

$$q_{c_0}/(Gh_{fg})=0.10(\rho_V/\rho_L)^{0.133}\times\{\sigma\rho_L/(G^2l)\}^{1/3}/(1+0.0031\,l/d) \quad \cdots\cdots\cdots\cdots(11)$$

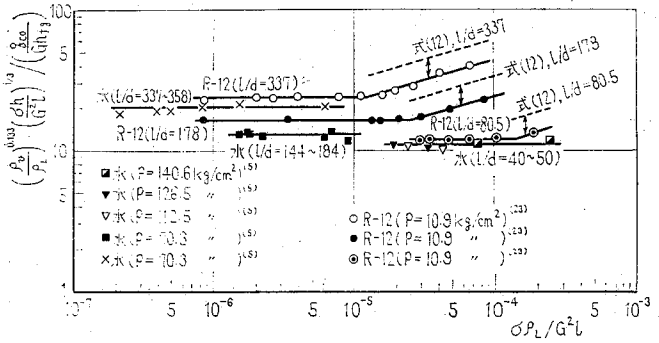


図 5　q_{c_0} の変化特性（H 領域）

図6は，式 (11) を本領域の q_{c_0} の全データと比較したものである．本領域は Macbeth ら[5]のいう高速域とだいたいの対応を持つので，以下 H 領域と呼ぶ．式 (11) によれば q_{c_0} は質量速度 G の 1/3 乗に比例して変化する．サブクールの影響については後に5章で記すが，H 領域では q_c は Δh_i に比例した変化（図4のa）を示す．

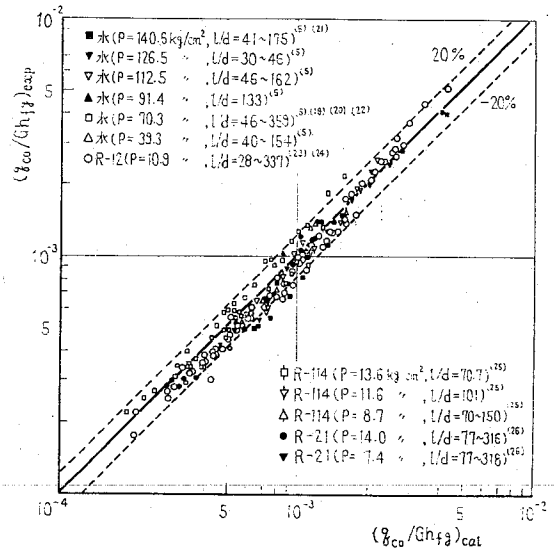


図 6　式 (11) と実験データ（H 領域）の比較

3・2　L 領域　図5より相対的に $\sigma\rho_L/(G^2l)$ の値が大きい領域で見いだされた結果（水，フロン，液体窒素，液体水素）の代表例を図7（および図5の一部）に示す．これに属する実験データは，フロン系を除き次の無次元整理式でまとめることができる．

$$q_{c_0}/(Gh_{fg})=0.25\{\sigma\rho_L/(G^2l)\}^{0.048}/(l/d) \quad \cdots\cdots\cdots\cdots(12)$$

フロン系については，図5に例示するごとく，q_{c_0} の実験結果が式 (12) の値（図

5に破線で示す）より 1.36 倍くらい高い．従って式 (12) 右辺の定数 0.25 の代わりに 0.34 を用いればよい．この原因はまだ不明だが，加熱面上でのフロンの熱分解などに関係するかも知れない．図8は，式 (12)（フロン系では定数0.34とする）を本領域の q_{c_0} の全データと比較したものである．本領域は Macbeth ら[5]のいう低速域に対応するので，以下L領域と呼ぶ．

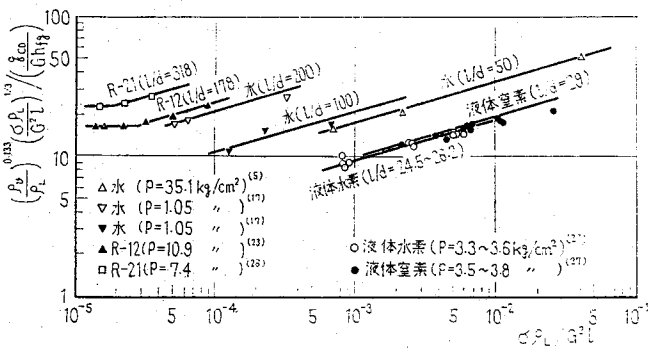


図 7 q_{c_0} の変化特性（L 領域）

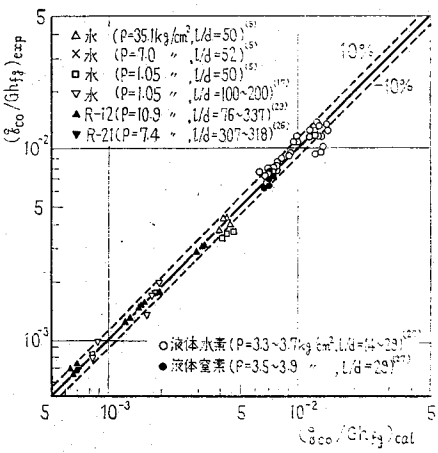


図 8 式 (12) と実験データ（L 領域）の比較

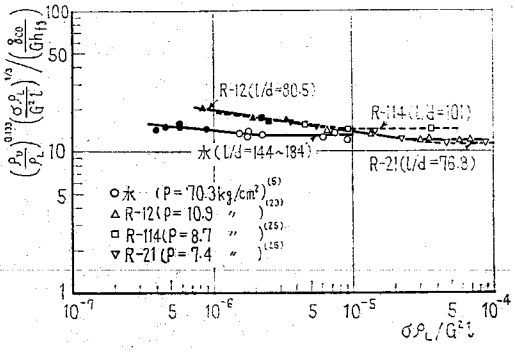


図 9 q_{c_0} の変化特性（N 領域）

L領域の CHF は，主に管内の液体の蒸発欠乏に起因して発生するものである．単純に液体の完全蒸発が CHF の原因とすると直ちに $q_{c_0}/(Gh_{fg})=0.25/(l/d)$ の関係が導かれるが，これに $\sigma\rho_L/(G^2l)$ の弱い影響の加わったものが式 (12) にほかならず，これは式 (10) で想定した流体力学的不安定状況とは全く無縁の状況である．式 (12) によれば，q_{c_0} は質量速度 G にほぼ比例して変化する．サブクールの影響は，q_c-Δh_i の直線関係を持つ．

3・3 N 領域　H領域の状態から，l/d=一定下に $\sigma\rho_L/(G^2l)$ の値が減少すると，図9に示すごとく縦軸の値が増加し，しかも q_c-Δh_i が非直線性（図4のb）の関係を示す領域に入る．図9で黒塗りの記号はこの両性格を持つ実験データを示し，これらの q_{c_0} は次式でまとめられる．

$$q_{c_0}/(Gh_{fg})=0.098(\rho_V/\rho_L)^{0.133}\times\{\sigma\rho_L/(G^2l)\}^{0.433}(l/d)^{0.27}/(1+0.003\,1\,l/d) \quad \cdots\cdots\cdots(13)$$

図 10 は，式 (13) を本領域の q_{c_0} の全データと比較したものである．他の領域に比し誤差がやや大きい．本領域は q_c-Δh_i の非直線性を特色とするので N 領域と呼ぶことにする．式 (13) によれば，q_{c_0} は質量速度 G が増してもほとんど増加せず，ほぼ一定に近い値をとる．なお図9に示す右下がりの線上でH領域に属するデータ（白抜きの記号）の q_{c_0} は元来，式 (13) でまとめるべきである．しかし偏差はあまり大きくないので，繁雑さをさけるため本報では式 (11) で扱うこととしており，図6でも上記のデータが式 (11) と比較されている．

3・4 HP 領域　H領域より $\sigma\rho_L/(G^2l)$ の値が小で

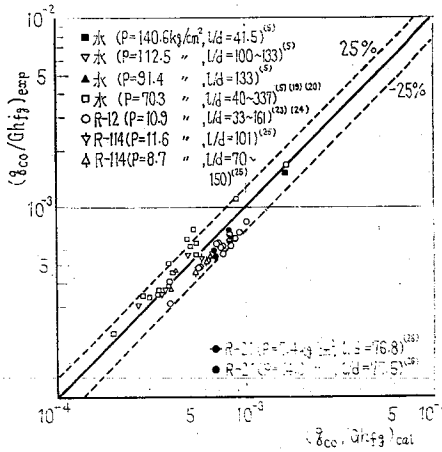


図 10 式 (13) と実験データ（N 領域）の比較

ありながら，q_c-Δh_i の直線性が保たれ，かつ図 11 に示す特性を持つ領域がある．この領域では水の実験データしか得られていないが，それらは次の無次元整理式でよくまとめられる．

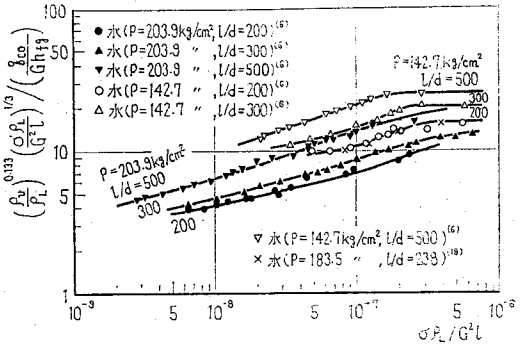


図 11　q_{c_0} の変化特性（HP 領域）

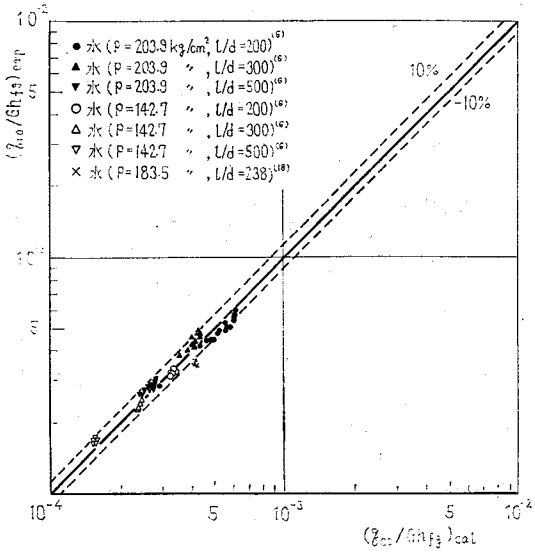


図 12　式 (14) と実験データ（HP 領域）の比較

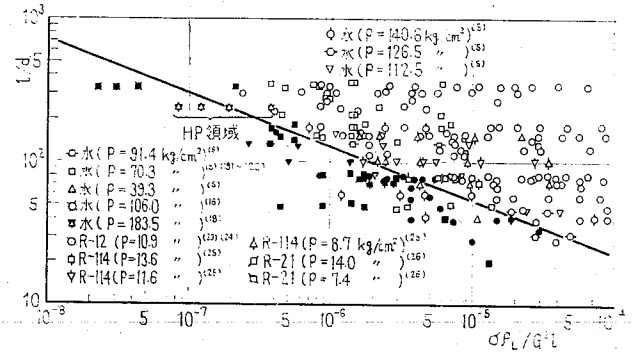


図 13　H 領域（白抜きのデータ）と N 領域（黒塗りのデータ）の境界

$$\frac{q_{c_0}}{Gh_{fg}}=8.20\frac{(\rho_V/\rho_L)^{0.65}\{\sigma\rho_L/(G^2l)\}^{0.453}}{1+107\{\sigma\rho_L/(G^2l)\}^{0.54}l/d} \quad \cdots\cdots\cdots\cdots(14)$$

図 12 は，式 (14) を本領域の q_{c_0} の全データと比較したものである．本領域は Becker ら[6] の言うごとく，また 4 章でより明らかになるごとく高圧で生じやすいから，以下 HP 領域と呼ぶ．なお後述 (5·3 節) するごとく，Macbeth ら[5] のしゅう集データの中にすでに本領域のデータが含まれている．

4.　各特性領域の境界

3 章に得た整理式 (11)～(14) の有効な利用のためには，4 個の領域間の境界を明らかにしておく要がある．まず L 領域と H 領域の間では，図 5 にみられるごとく，q_{c_0} の値が連結する．従って，これら 2 領域の境界線は近似的に，式 (11)，(12) から q_{c_0} 消去して定めることができ，次式で与えられる．

$$l/d=1/[0.4(\rho_V/\rho_L)^{0.133}\{\sigma\rho_L/(G^2l)\}^{0.29}-0.0031] \quad \cdots\cdots\cdots\cdots(15)$$

ただし，フロン系の場合は，式 (15) 右辺分母の定数 0.4 の代わりに 0.29 となる．

次に H 領域と N 領域の境界線に関し，仮に式 (11)，(13) から q_{c_0} を消去してみると次の結果を得る．

$$l/d=1.077/\{\sigma\rho_L/(G^2l)\}^{0.37} \quad \cdots\cdots\cdots\cdots(16)$$

しかし，これは図 9 からわかるように，H 領域と N 領域の境界そのものではない．そこで，この両領域に属する実験データに関し，q_c-Δh_i の関係の直線性，非直線性から H 領域と N 領域に分類，図示した結果が図 13 である．そして同図の斜めの境界線から，H, N 両領域が次のごとく与えられる．

$$l/d=0.77/\{\sigma\rho_L/(G^2l)\}^{0.37} \quad \cdots\cdots\cdots\cdots(17)$$

最後に H 領域と HP 領域の間については，実験データ不足のため両領域間の遷移域の広がりなど必ずしも明確でない．しかし図 11 で $p=142.7$ kg/cm² の 3 組の実験データでは，H 領域と HP 領域で q_{c_0} が連結している．そこで一応，式 (11)，(14) から q_{c_0} を消去する方法で次式を得る．

$$\frac{l}{d}=\frac{82(\rho_V/\rho_L)^{0.517}-\{\sigma\rho_L/(G^2l)\}^{-0.12}}{\left[\begin{array}{l}107\{\sigma\rho_L/(G^2l)\}^{0.42}\\ -0.254(\rho_V/\rho_L)^{0.517}\end{array}\right]} \quad \cdots\cdots\cdots\cdots(18)$$

さて以上に求めた式 (15)，(17)，(18) を図示した結果が図 14 である．フロン系の場合は，式 (15) の定数の変更に伴い H-L 境界線が図 14 の鎖線のように右方へ多少移動する．図 14 から次の特徴が観察さ

れる．ρ_V/ρ_L の値が大になる（すなわち高圧になる）に従い，L 領域が $\sigma\rho_L/(G^2l)$ の小さいほうへ広がる一方，HP 領域が $\sigma\rho_L/(G^2l)$ の大きいほうへ広がる．その結果，H領域がせばめられ，$\rho_V/\rho_L=0.325$（この時の圧力は水を例にとれば 200 kg/cm² で臨界圧力 225 kg/cm² に非常に近い）で H 領域は非常に狭い．他方，H領域とN領域の境界は ρ_V/ρ_L に関係しない．

なお図 14 において，N 領域と HP 領域の間の遷移は存在しないもののようである．すなわち H⇄N および H⇄HP の遷移だけが生ずる（図 13 の HP 領域のデータ参照）．

〔付記〕 液体ヘリウムⅠのデータ分析　尾形ら[28]の液体ヘリウムⅠの実験データ（$p=1.1\sim2.0$ kg/cm², $l/d=25.7$ および 51.4）を図 14 で判定すると，大部分が N 領域，一部が H 領域にあり，かつ各領域の q_{c_0} のデータがそれぞれ図 10 および 6 の実験データのちらばりの範囲内に入る．すなわち本報の整理式は液体ヘリウムⅠにも適用可能のようである．

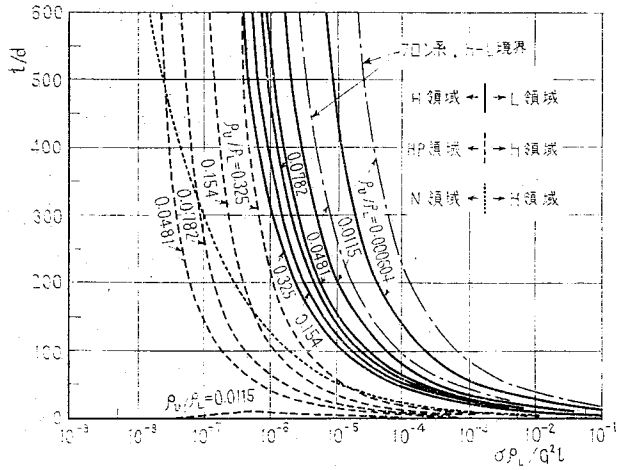


図 14　境界領域の境界線図

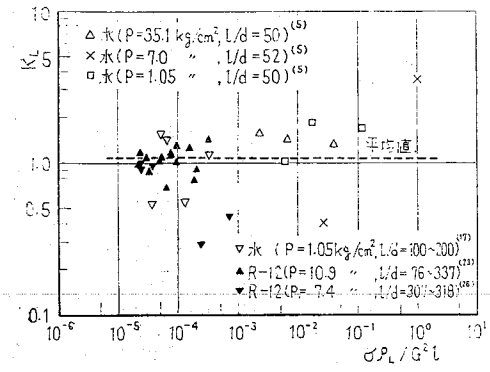


図 15　K_L の実験データ

5. サブクールの影響に関するデータ分析と無次元整理

管入口で液体がサブクールされている時，図 4 の a のごとく q_c-Δh_i の直線性が成立する場合は

$$q_c=q_{c_0}(1+K\Delta h_i/h_{fg}) \quad \cdots\cdots(19)$$

と書くことができる．ここに K は Δh_i に独立な係数であり，図 4 の直線 a のこう配から求められる．

5·1 L 領域における K：K_L　図 15 は，L 領域に属する水，フロンの実験データ（液体窒素[27]，水素[27]については Δh_i がわずかで K_L が求められず）から求めた K_L を示す．この領域では，3·2 節に述べた性格から $K_L\fallingdotseq1$ を予想できる．図 15 のデータの平均をとれば

$$K_L=1.16 \quad \cdots\cdots(20)$$

ただし図 15 にみるごとく，K_L の値のばらつきは小さくない．

5·2 H 領域における K：K_H　図 16 は H 領域に属する全実験データ（水，フロン）から求めた K_H（0.5～6.0 程度の値をとる）を整理したもので，同図の実験は次式であらわされる．

$$\left.\begin{aligned}&\sigma\rho_L/(G^2l)<3\times10^{-6}\text{ に対し}\\&\quad K_H=1.8\{l/(130d)\}^{-5(\rho_V/\rho_L)}\\&3\times10^{-6}<\sigma\rho_L/(G^2l)\text{ に対し}\\&\quad K_H=0.075\{l/(130d)\}^{-5(\rho_V/\rho_L)}\\&\qquad\times\{\sigma\rho_L/(G^2l)\}^{-1/4}\end{aligned}\right\} \quad \cdots\cdots(21)$$

上記の整理に対し実験データにある程度のばらつきがある．なお図 16 で $\sigma\rho_L/(G^2l)$ の小さな領域で破線の近傍に少数のデータが現れる事実があるが，その原因については将来の研究を待つ要がある．

5·3 HP 領域における K：K_{HP}　HP 領域のデータのうち，5 章はじめに述べた方法で直接 K_{HP} の値が求められるのは Chojnowski ら[18]のものだけである．そこで Macbeth ら[5]のしゅう集データ（その表 11～14 のうち $l/d=365$ のデータ*），および Becker ら[6]のデータ（その図 12 のサブクール度が 100°C のもの）が与える q_c と Δh_i の組合せに対し，式 (19) を適用，ここで q_{c_0} については式 (14) の計算値を使用，間接的に K_{HP} の値を求め，これらも分析に利用した．図 17 は，こうして求めた K_{HP}（1.0～3.0 程度の値をとる）を整理したもので，同図

* 図 14 により判定すると，表 11～14 のうち $l/d=11.7$, 50.0, 80.0 のデータは N 領域に属する．

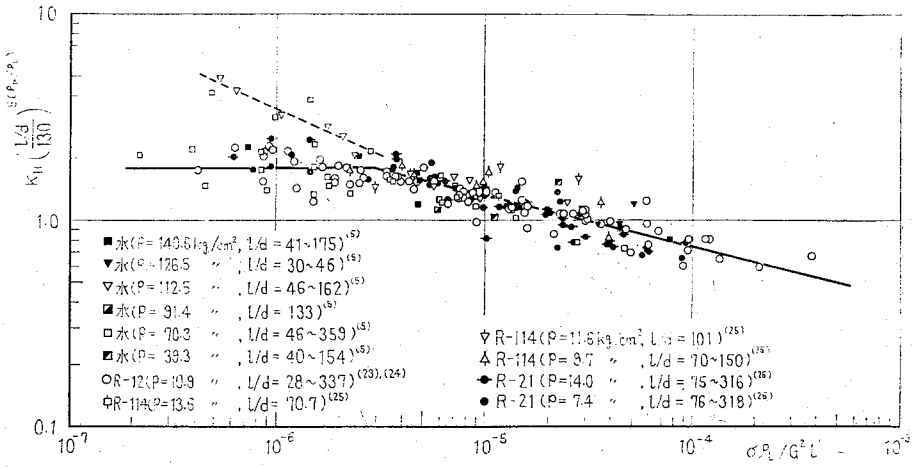


図 16 K_H の実験データ

の実線は次式であらわされる．

$\sigma\rho_L/(G^2l)<4\times10^{-8}$ に対し

$$K_{HP}=0.664(\rho_V/\rho_L)^{-0.6}$$

$4\times10^{-8}<\sigma\rho_L/(G^2l)$ に対し

$$K_{HP}=3.08(\rho_V/\rho_L)^{-0.6}\{\sigma\rho_L/(G^2l)\}^{0.09} \quad \cdots\cdots(22)$$

5·4 N領域におけるサブクールの影響 これについては q_c-Δh_i の直線性がなく，かつ実験データも不十分で，また信頼性ある無次元整理式を導くことができない．

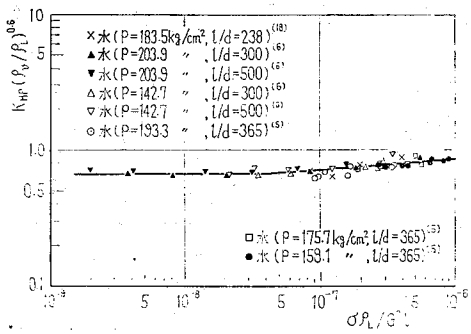


図 17 K_{HP} の実験データ

6. 限界熱流束 q_c と出口クオリティ χ_{ex} の関係

一様加熱の場合，エネルギのつりあいから q_c, Δh_i, χ_{ex} の間に次の関係が成立する．

$$q_c/(Gh_{fg})=[\chi_{ex}+(\Delta h_i/h_{fg})]/(4l/d) \quad \cdots\cdots(23)$$

従って，式 (19), (23) から $\Delta h_i/h_{fg}$ を消去し，かつ q_{c_0} に式 (11), (12), (14) を与えて次の諸関係を得る．

L領域：

$$\frac{q_c}{Gh_{fg}}=0.25\left(\frac{\sigma\rho_L}{G^2l}\right)^{0.043}\frac{K_L\chi_{ex}-1}{\{K_L(\sigma\rho_L/G^2l)^{0.043}-1\}l/d}$$

（フロン系では右辺定数を0.34に代える）…(24)

H領域：

$$\frac{q_c}{Gh_{fg}}=0.10\left(\frac{\rho_V}{\rho_L}\right)^{0.133}\left(\frac{\sigma\rho_L}{G^2l}\right)^{1/3}\frac{1-K_H\chi_{ex}}{1+\left\{0.003\,1-0.40K_H\left(\frac{\rho_V}{\rho_L}\right)^{0.133}\left(\frac{\sigma\rho_L}{G^2l}\right)^{1/3}\right\}\frac{l}{d}} \quad \cdots\cdots(25)$$

HP 領域：

$$\frac{q_c}{Gh_{fg}}=8.20\left(\frac{\rho_V}{\rho_L}\right)^{0.65}\left(\frac{\sigma\rho_L}{G^2l}\right)^{0.453}\frac{1-K_{HP}\chi_{ex}}{1+\left\{107\left(\frac{\sigma\rho_L}{G^2l}\right)^{0.54}-32.8K_{HP}\left(\frac{\rho_V}{\rho_L}\right)^{0.65}\left(\frac{\sigma\rho_L}{G^2l}\right)^{0.453}\right\}\frac{l}{d}} \quad \cdots\cdots(26)$$

ここに，K_L, K_H, K_{HP} は式 (20)～(22) に与えられる．なお式 (24)～(26) の使用に際しては，必ず式 (23) を考慮し，与えられた Δh_i の変化範囲に対応する χ_{ex} の変化範囲に対してのみ適用すべきである．

7. 結　　論

一様加熱垂直管内の安定な強制流動沸騰の限界熱流束に関し，水，フロン(R-12, R-144, R-21)，液体窒素，液体水素（パラ），液体ヘリウムIの実験データを分析した．そして4個の領域 (L, H, N, HP) を区別し，各領域における飽和液の限界熱流束の特性を明らかにした上で無次元整理式〔式 (11)～(14)〕を導き，各領域間の境界

について図 14 を求め，次いでサブクールの影響に関し無次元整理式〔式 (20)～(22)〕を求めた．以上の結果，管内強制流動沸騰の限界熱流束の挙動には，非常な規則性があることが判明し，本現象に関連ある技術問題に対し見通しがよくなったといえるであろう．

本報で利用した実験データを発表された著者らに心からの敬意を，また本研究は文部省から科学研究費補助を受けたもので謝意を表する．なお本研究の展開のための必要事項などについて，楠田久男教授（佐賀大学），尾形久直氏（日立製作所機械研究所）から重要な援助を，また文献の入手などについて，数土幸夫氏（日本原子力研究所），大浪美雪氏，横谷定雄氏（東京大学）から援助を受けたので，ここに記して謝意を表する．

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討　論

〔質問〕　西川兼康（九州大学工学部）

従来混とんとしていた強制流動沸騰における限界熱流束についてその規則性を明らかにし，一般性の高い整理式を求められたことに対し，敬意を表する．

(1) 式 (5) は

$$c^2=\frac{g}{m}\frac{\rho_L-\rho_V}{\rho_L+\rho_V}+\frac{\sigma m}{\rho_L+\rho_V}$$

c：乱れの伝ぱ速度，$2\pi/m$：乱れの波長，から導いたものと思われるが，例えば g の次元がそれが作用している方向ではない方向の L_y を含んでもよいものであろうか．逆に $[g]=[L_x/T^2]$, $[m]=[1/L_y]$ とすれば，$[c^2]=[L_xL_y/T^2]$ となり，$[L_y^2/T^2]$ とはならない．それでも物理的に正しいのであろうか．

(2) 管内流動において $d\to\infty$ と $d\to 0$ の両極限における状況を考え，式 (10) を導き，それを手掛かりとして四つの領域別に CHF の整理式を求めておられるが，CHF が生じるときの流体力学的状況は各領域でどのようになっているとみればよいのであろうか．H領域とL領域については図2と3を用いた説明で一応理解できるが，N 領域と HP 領域については特に明記されていないようである．この二つの領域の機構に対する著者の考え方があれば，本文に記されているこの領域の限界熱流束の特性とも関連してご教示願いたい．

(3) 四つの領域の境界を示す境界線図（図 14）は，提案された整理式の適用範囲および各特性領域の広がりを知る上で重要であるが，HP 領域⇄H領域と N 領域⇄H 領域の二つの境界線が交差する場合が生じており，領域が重複しているようにも受け取られる．このような場合，いずれの領域の特性が優先すると考えて整理式を適用すべきか伺いたい．

(4) 6章における式 (24)～(26) の使用上の注意で，「与えられた Δh_i の変化範囲」とは具体的にどのような範囲を指しているのか．

(5) この論文で検討の対象とされていない流体の CHF を提案された整理式で推測することの可否について，著者のご見解を伺いたい．

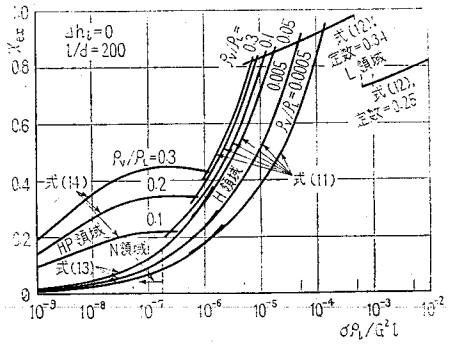
〔回答〕　(1) 現在の現象に対し，σ および g は，その発生する圧力変化を介して，はじめて現象に関与している．ところで流体内の任意の点の圧力は，

あらゆる方向に同じ強さで作用する．従って σ, g の効果は，種々の方向におよぶ．このため，その効果を考えた後の σ, g は方向に関し，もととは異なる次元を持つ形となるのであって，このような例は他にも多く例をあげることができる．なお式(5)は，質問者の記された c^2 の式だけでなく，それを導く理論過程でも当然，成立している．

（2） 式(10)に合致するのはH領域の式(11)だけで，他の3領域は式(10)から偏よるものとして導いたというのが本報の筋である．事実，式(12)～(14)は，手がかりの式(10)に対し異質の形をとっている．なお4領域のCHFの発生機構は今後の研究を待つべきものと思う．ただ漸定的という条件下に著者の推量を言わしていただければ次のごとくである．すなわちCHF発生時の管出口付近の状況は，L領域でミスト流（液滴の壁面へのデポジット速度と，気流へのエントレインメント速度がCHFに関係），H領域では環状流（ないしフロス，スラグ流で，界面の流体力学的条件がCHFに関係），N領域では気ほう流（いわゆる核沸騰から膜沸騰への遷移でCHF発生），またHP領域についてはまだ見当がつかない．なお参考までに本報の整理式による出口クオリティの状況を付図1（これは著者の次報に掲載予定のもの）を示す．

（3） 本文に記した「H⇄N およびH⇄HPの遷移だけが生ずる」の原則で処理する．すなわち図14で $\sigma\rho_L/G^2l$ の大なる領域（H領域）から $\sigma\rho_L/G^2l$ を減少させ，先にN領域に入れば以後N領域のみ続き，先にHP領域に入れば以後HP領域のみが続くとすればよい．付図1は，この方式で領域を定めている．

（4） 他の条件一定に，Δh_i だけを変える時，式(24)～(26)により，q_c と χ_{ex} は線形関係を持つが，現実の Δh_i の変化範囲に対応する χ_{ex} の変化範囲は比較的狭く，従って $\chi_{ex}=0$～1 のような広域変化範囲を機械的に考えるのは非現実というのが趣旨である．この注意を記した理由は，本報の式(24)～(26)が，従来のlocal conditions hypothesis（本報はこの考えを否定する立場にある）に基づく q_c-χ_{ex} の関係と同一視されることを防ぐためであった．本報の立場からいうと，例えば数個の異なる管長に対し，式(24)～(26)で与えられる q_c と χ_{ex} の関係（数個の）を近似的に一つに連ねたものがlocal conditions hypothesis の q_c-χ_{ex} の関係に相当する．



付図1 出口クオリティ χ_{ex}（$\Delta h_i=0$, $l/d=200$ の場合）

（5） 本研究を完成した時点では，著者自身，ご質問の問題に対し少からぬ不安があった．しかし，その後の研究（次報で発表予定）により，実験との比較を広げた結果，現在ではかなり自信を持つに至っている．なお一般論としては，流体の種類により，その粘性が影響を持つ可能性が十分にあり，この時はレイノルズ数 Gd/μ_L および粘性比 μ_L/μ_V が影響因子として加わるはずである．従って，このような流体に本報の整理式の適用は不可であろう．また液体金属の場合などは，L領域には使えても，H領域やN領域には熱伝導に関する配慮を必要とするといった事態が生ずるかも知れない．なお本報の式(12)や図14で，フロン系が特別扱いになっているが，その後の研究（次報で発表予定）で，流体の種類には関係なく，式(12)右辺の定数は $\sigma\rho_L/G^2l<5\times10^{-4}$ で0.34，$\sigma\rho_L/G^2l>5\times10^{-4}$ で0.25とするのが適当のごとき結果が得られているので一応付記する（付図1参照）．

〔質問〕 赤川浩爾（神戸大学工学部）

限界熱流束を広範囲の条件に対して統一的に整理方式でまとめられたことについて敬意を表する．

（1） 図14において，あるいは式(17)，(18)によって領域を判定する場合に，HPであるのかNであるのかはどのようにして定めればよいか．

（2） L領域は3868ページ右欄1行めに明記されているように，「主に液体欠乏に起因して発生するCHF」の領域であって物理的によくわかる．一方，ここで分類されているH, HP，およびN領域のCHFはどのような物理的現象であるのかご教示ねがいたい．

〔回答〕 （1） 西川氏への回答(3)を参照願いたい．

（2） 西川氏への回答(2)を参照願いたい．

〔質問〕 鳥飼欣一（日本原子力研究所）

（1） クオリティの q_{c_0} に対する影響は，どのように考慮されているのか．著者の無次元整理の過程をみると，直接はクオリティに触れておられないようにみえる．代表長さ l などでクオリティと関連する気液二

相流動状態をも代表されているのか．また，2章の初めに飽和液とあるが，これはクオリティ0の場合で，関係式を展開されているのか．なお6章で，クオリティについて述べておられるが，これはそれまでの式を整理したものと考えてよいか．

（2） 2章の初めに，加熱面代表寸法 l のみで式(11)を求めておられるが，後で内径 d が入ってくる．最初から l，d 共に q_{c_0} に関係する物理量とは考えられないか．

（3） 3866 ページ左欄 4 行めに「波動に関する理論から…」とあるが，どのような条件の気液境界面をここでは考えておられるか．このあとの論理の展開の基になっているかと思われるので質問する．

（4） 式(10)を式(4)，(9)から直接導出はされないようにみえるが，式(10)はどこから導かれたものか．一つの仮定なのであろうか．

〔**回答**〕 （1） 従来，強制流動沸騰系の CHF に対し，local conditions hypothesis あるいはこれに類する考えが非常に優勢で，q_c はクオリティに強く関係すると考えるものが多いといってよいであろう．また表面張力 σ についても，強制流動系の場合は軽視されてきたように思われるだけでなく，仮に σ を含む無次元量を考えるにしても，管径 d に基づく $\sigma\rho_L/G^2d$ の形のものであり(付1)，本報で全面的に用いている管長 l に基づく $\sigma\rho_L/G^2l$ の使用は，上記の立場からいえば気違いざたと考えられるものではないかと思う．そして実は，これらの諸点に反逆〔その出発点は文献(10),(11)の研究〕したのが本研究である．もし本研究に，いささかの成功があるとすれば，その反逆によるものであり，従って従来の考え方の多くは改める要があるというのが卒直にいって著者の主張である．また6章でクオリティについて述べたものは，6章以前の式を再整理したものにすぎず，6章を省いても本報の本質には触れない．

なお本報は，図4で $\Delta h_i \geqq 0$（加熱管入口にサブクール液か飽和液が流入）の範囲を扱っている．この範囲は，加熱管入口の流体の状況が Δh_i の値だけで一義的に確定できる．これに対し，$\Delta h_i < 0$（加熱管入口に気液二相流が流入）の場合は，Δh_i の値だけではフローパターンが確定できぬ問題あり，やや複雑になり得るが，この $\Delta h_i < 0$ の場合については著者の次報で触れる予定である．

（2） 最初から l，d の両者を含めて解析しても何ら差し支えない．ただ，そうすると記述が長くなるので，本文のようにしただけである．

（3） Lamb の書物[15] に出ているところの気液二相界面の「微小波動」に関する周知の諸理論で扱っている条件である．著者の書物(付2)にも一部記してある．

（4） $d \to \infty$ および $d \to 0$ の両極限で，それぞれ式(4)と(9)の形になるという条件を満足する式（無数にあり得る）のなかで，恐らく最も簡単な形のものとして式(10)を考えたにすぎない．従って，この段階では式(10)に現実的価値はなく，式(10)に価値がでるのは実験との比較から有用性が認められてからである．

（付1） たとえば Ahmad, S.Y., *Int. J. Heat & Mass Transf.*, **16**-3 (1974-3), 641.

（付2） 甲藤，伝熱概論，(昭 39)，287，養賢堂．

B301　圧力降下を伴う管内強制流動サブクール沸騰の限界熱流束

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機正　横谷定雄（東大工）

§1　序論

強制流動沸騰現象は　工業上熱交換，熱除去等に多く用いられており　その限界をなす限界熱流束の定量的な把握は実用的な意義が大きい。このうち最も多く用いられている管内沸騰系の限界熱流束に関して　最近甲藤[1),(2)]は多くの実験データを収集し　これらに関して方向性次元解析を背景とした無次元整理を行い，限界熱流束に対する4つの特性領域に区分した上でかなり一般に適用出来る限界熱流束の無次元整理式を提唱した。このうちのN-領域に着目して本研究を行った。N－領域は，相対的に質量流量Gが大，加熱管の径長比 L/d が小の場合に実現される領域で，限界熱流束と入口サブクール度との間の関係が非直線性を示す等の特徴を持っている。本研究については，一般の管内沸騰の研究に比較して　大気圧近くの圧力範囲（出口圧力 P_e = 1～1.5 (bar)）に於る径長比 L/d の小さい加熱管を使用した実験を行ったことが特色として掲げられる。これに依り，現象的には　加熱管入口-出口間の圧力降下が大きくなり，またこれに伴い入口に於るサブクール度も大きくなった。この複雑な現象の整理に当っては　圧力降下を考慮した沸騰長さの概念を導入して本現象を飽和沸騰に帰着させる方法をとり，限界熱流束の無次元整理式を導出した。

§2　実験装置及び実験方法

実験装置はFig.1に示すように密閉系内を流体が循環する構成となっている。流体はタンク内で温度が設定されポンプによって系内を循環する。流量は主流路及びバイパス流路に設けられたバルブで調節され　オリフィス前後の差圧から測定される。主流路内の流体はFig.2に示す加熱管部分に導かれここで沸騰現象を起す。加熱管はステンレス製であり内径 d = 3,4,5 (mm) のものを使用し，長さ l は約70～240 (mm) の範囲にある。加熱方法は低電圧大電流電源使用の直接通電加熱である。実験は，流量を設定し熱流束を階段状に上げて加熱管の温度（出口より 10～40 mm 上流側の外壁に取付けた数個の熱電対で測定）を観察する手法をとって行った。この際に，加熱管温度が定常にならず急激に上昇する点をもってバーンアウト現象発生とみなし限界熱流束データとして採用した。他に加熱管の入口及び出口の圧力についても測定した。

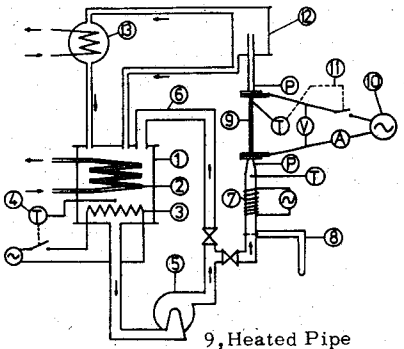

Fig.1　実験装置概略図

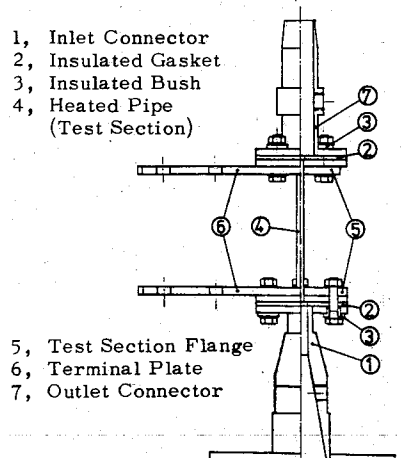

Fig.2　実験装置主要部

本実験の範囲は，使用流体はR-113のみ，加熱管3種類（内径 d = 5.79 mm，長さ L = 240mm，径長比 L/d = 41.5 及び d = 2.96 mm，l = 89.6 mm，L/d = 30.3 及び d = 3.99 mm，l = 88.1 mm，L/d = 22.1），質量流量 G = 4.3×10^6～4.9×10^7 [kg/m²hr]（入口流速 0.8～9 [m/s]），出口圧力 P_e = 1～1.5 [bar]，入口-出口間圧力降下 0.3～2.7 [bar]，（入口圧力 P_i = 1.8～4 [bar]），入口サブクール温度 ΔT_{sub} = 10～60 [°K]，気液密度比 ρ_v/ρ_l = 0.006～0.014 である。

§3 実験結果及び整理

3-1 実験結果　実験結果の例として Fig.3 に入口サブクール温度 ΔT_{sub} 一定のもとでの入口流速 u_i と限界熱流束 q_c との関係を示す。しかし，このグラフでは各点での入口，出口圧力はまちまちの値をとっており 限界熱流束の特性の定量的な把握には更に入口，出口圧力に関する整理が必要となる。

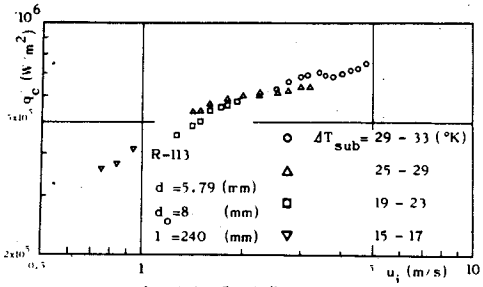

Fig.3　実験結果の例

3-2 整理の方針　本実験結果の整理に当っては，前出甲藤の限界熱流束の無次元整理式を手掛りとして無次元整理を行うこととする。甲藤は飽和液流入時の N－領域の限界熱流束について次式 (1) に示す無次元整理式を提唱している。

$$(q_c/G h_{fg})=0.098(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2 l)^{0.433}(l/d)^{0.27}/(1+0.0031(l/d)) \quad \text{----------(1)}$$

h_{fg}；蒸発潜熱，ρ_v；蒸気密度，ρ_l；液体密度，σ；表面張力，l；加熱管長さ，d；加熱管内径

本現象はこれに加え，入口－出口間の圧力降下及び入口サブクール度の影響が加わるため，関与する無次元数は簡略化して考えても式 (1) に表われた4個に2個加わり，計6個となる。本実験の範囲のみからこれら6個の無次元数の間の関数関係を定めることは困難である。そこで，沸騰長さの概念を導入して，本現象を飽和沸騰に帰着させ 整理の簡単化を計った。

3-3 沸騰長さ　加熱管内に於て局所バルククオリティ $x=0$ となる点より下流の部分の長さを沸騰長さ l_b とする。この場合一般の沸騰長さと異なり 加熱管内に流れ方向の圧力変化を考え，これに依る飽和温度の変化を考慮して 次式 (2) により近似的に x を求めた。

$$x(x)=(\pi d x q-(\pi/4)d^2 G(h_f(T_{sat}(p(x)))-h_f(T_i)))/(\pi/4)d^2 G h_{fg} \quad \text{----------(2)}$$

x；流れ方向座標，q；熱流束，$h_f(T)$；温度 T での液体エンタルピ，$T_{sat}(P)$；圧力 P での飽和温度，$p(x)$；圧力分布を与える関数 である。式 (2) から $x=0$ となる点を求めれば l_b が近似的に定まる。沸騰長さの求め方を Fig.4 に示す。尚，図中の液体温度 T_L は，加熱管から流体へ伝わる熱が総て液体温度の上昇のみに用いられるとした場合の液体温度であり $T_L = T_{sat}(p(x))$ となる点 x に於て局所バルククオリティ $x=0$ となる。

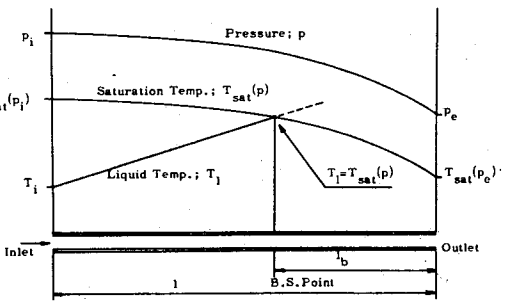

Fig.4　沸騰長さの求め方

3-4 圧力分布　沸騰長さを求めるためには，加熱管内の流れ方向の圧力分布関数 $P(x)$ を定める必要がある。ここでは，圧力分布を正確に求めることが主目的ではないので，沸騰長さを求め易いような圧力分布に関してかなりの仮定を設け，簡単化した圧力分布関数を定めた。先ず，加熱管入口から直ちにサブクール沸騰を開始してボイド率が0から増加すると考え，またボイド率 f_g は式 (3-1) に示す様にクオリティ x の一次式で表わされると仮定した。また，ボイド率と局所圧力降下 $\frac{dp}{dx}$ については 式 (3-2) の関係が知られている。TP；二相流，L；液単相流，γ；定数 $\gamma=-2$ 程度，であり，本現象ではボイド率が小さい（Bubbly Flow と考えられる）ので 式 (3-2) は式 (3-3) のように近似される。以上から式 (3-4) に示す如く，局所圧力降下が，クオリティ x の一次式で表わされると考える。式 (2) を式 (3-4) を代入すると $P(x)$ に関する微分方程式が得られ近似的にこれを解くと式 (4) に示す形の圧力分布が得られる。式 (4) の係数 A は積分定数，B，C は式 (3-4) の係数 a，b を含む係数，K は熱流束と物性値から定まる定数である。式 (3-4) の係数

$$f_g=a_1 x+b_1 \quad \text{----------(3-1)}$$

$$\left(\frac{dp}{dx}\right)_{TP}=\left(\frac{dp}{dx}\right)_L(1-f_g)^{\gamma} \quad \text{--(3-2)}$$

$$\frac{dp}{dx}=a_2 f_g+b_2 \quad \text{----------(3-3)}$$

$$\frac{dp}{dx}=a x+b \quad \text{----------(3-4)}$$

$$p(x)=A\exp(Bx)+Kx+C \quad \text{-----(4)}$$

$$p(0)=p_i,\ p(1)=p_e,\ \left.\frac{dp}{dx}\right|_{x=0}=-0.3164\,Re^{-1/4}\rho_l u_i^2/2d \ \text{(Blasius)}$$

a, b は具体的に定められないので、係数 A,B,C は式(4)に示す3個の境界条件から定め、これを用いて沸騰長さをボめた。

3-5 実験結果の無次元整理　圧力降下及び入口サブクール度の影響を沸騰長さ l_b に代表させ 本現象を加熱管長さが l_b で、飽和液流入の沸騰現象に帰着させて整理を試みる。これに依り、本現象に関与する無次元数は、加熱管長さ l を沸騰長さ l_b に置換えた式(1)の4個の無次元数となる。つまり、無次元数 (q_c/Gh_{fg}) が、他の3個の無次元数 (ρ_v/ρ_l), (l_b/d), $(\sigma\rho_l/G^2l_b)$ の関数で与えられると考え実験結果の整理を行う。ここで、各無次元数に含まれる物性値については バルククオリティ $x=0$ となる点に於る飽和温度のものを用いた。 実験を行った主な範囲については Fig.5 に示す。グラフ中の実線は甲藤が与えた H-, N-領域の境界であり、これより殆どの実験点が、N-領域に属すると考えられる。加熱管長さに対する沸騰長さの比率 l_b/l は 0.01 ～ 0.75 の範囲にあるが、l_b/l く 1/2 と比較的小さい l_b/l の値を持つ実験結果が多い。無次元整理式の導出に当っては、無次元数 (ρ_v/ρ_l) の実験範囲が 0.006 ～ 0.014 と狭いため (ρ_v/ρ_l) の指数については、式(1)のもの 0.133 を用いた。Fig. 6 に密度比の影響を除いた限界熱流束の無次元数を無次元数 $(\sigma\rho_l/G^2l_b)$ でプロットしたグラフを示す。ここで (l_b/d) の値の近いデータ同士に直線性が見られることから (l_b/d) の指数を決定した。尚、(l_b/d) の項の分母についても、本実験の範囲からは定められないので 式(1)のものを用いた。以上で定めた項を用いて、Fig. 7 に示すように $(\sigma\rho_l/G^2l_b)$ の指数を全データから決定して式(5)に示す限界熱流束の無次元整理式を得た。

$$(q_c/Gh_{fg})=0.103(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l_b)^{0.415}(l_b/d)^{0.30}$$
$$/(1+0.0031(l_b/d)) \quad \text{----------(5)}$$

Fig. 8 に実験結果と式(5)による計算値 (exp. 及び cal. で示す) との比較を示すが、両者はほぼ ±10% の誤差の範囲内で良く一致している。また式(5)の係数及び本整理で定めた2つの指数は式(1)のものとかなり近い値である。これより、本現象がかなり良い精度で飽和沸騰に帰着されたことが判り、また 以上の整理方法の有効性が確かめられた。

§4 結論

圧力降下を伴い入口サブクール度の大きい管内強制流動沸騰系を 圧力降下を考慮した沸騰長さを用いて飽和沸騰系に帰着して整理を行うと、その限界熱流束は、飽和沸騰系の場合と同様な無次元整理式で良く整理されることが判った。

参考文献 (1)甲藤　機論 44-385 ('78-11) 3867
(2)甲藤　機講論 第56回全国大会 ('78-10)

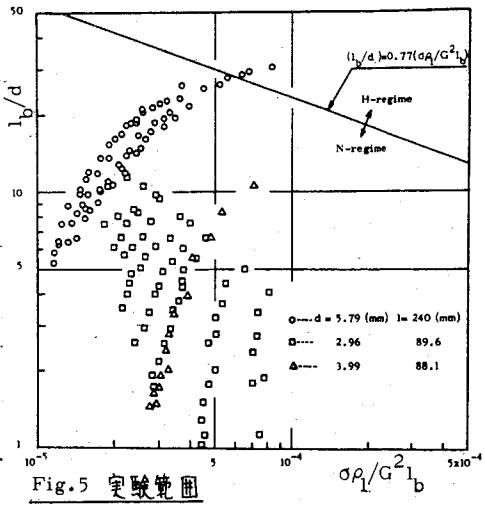

Fig.5 実験範囲

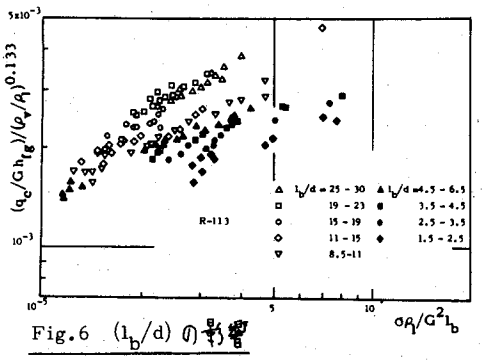

Fig.6 (l_b/d) の影響

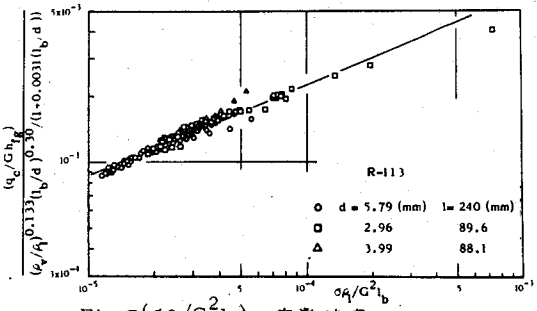

Fig.7 $(\sigma\rho_l/G^2l_b)$ の指数決定

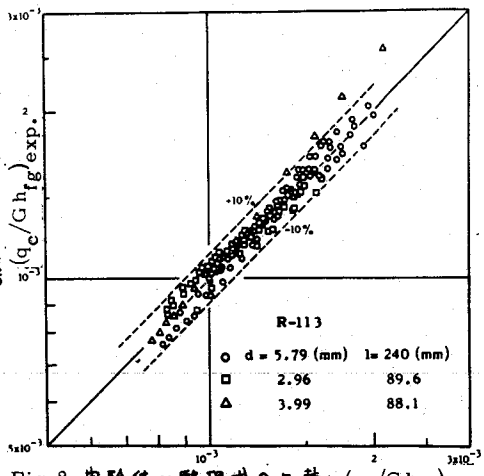

Fig.8 実験値と整理式の比較 $(q_c/Gh_{fg})_{cal.}$

536.423.1 :: 532.542

一様加熱垂直円管内の強制流動沸騰の限界熱流束の特性について*

甲　藤　好　郎**

1. 序

一様加熱垂直円管内の強制流動沸騰の限界熱流束(CHF)について，種々の特色ある研究がおこなわれて来ている．比較的最近では，ヨーロッパの8研究所の間で，水について同じ条件下のCHFの実験データの再現性を調べる実験がおこなわれ，特に信頼性の高いデータ（ミラノ再現性試験データ[1]）が得られている．一方，Beckerらは，非常に直径の小さい管[2]，あるいは非常に長い管[3]について水のCHFの実験をおこなっている．またCumoら[4]は，フロン12を用い，上昇流と下降流のそれぞれの場合のCHFの測定を比較し，Groeneveld[5]はフロン12で入口サブクールが正から負にわたる領域に関しCHFの変化，および円管内のCHFの生ずる位置を調べ，Campolunghiら[6]は水のCHFの特性が圧力90～110 barの区間で急変する結果を示している．他方，Alad'yevら[7]はカリウムのCHFの測定をおこない，Pokhvalovら[8]はベンゼンのCHFについて報告している．またCHFのモデリングの研究は，Ahmad[9]その他によって大きく進展している．しかし，上記の諸研究の間の相互関係は必ずしも明白でない．

さて最近著者[10]は，方向性次元解析などを一つの手がかりとして，一様加熱円管内の強制流動沸騰のCHFに対する無次元整理式を導くことができた．そして本報は，この無次元整理式を用い，前述の諸研究のデータを分析，この整理式の適用性や問題点を調べると同時に，CHFの特性を統一的な立場から明らかにすることを試みた結果について報告するものである．

記　号

d ：加熱管内径
F_G ：質量速度変換係数
G ：質量速度
H_{fg} ：蒸発潜熱
ΔH_i ：入口サブクール・エンタルピ
K ：ΔH_i の影響に関係する係数
l ：加熱管長
p ：絶対圧力
q_c ：限界熱流束
q_{c0} ：$\Delta H_i=0$ の場合の q_c
ρ_l ：液体密度
ρ_v ：蒸気密度
μ_l ：液体の粘性係数
μ_v ：蒸気の粘性係数
σ ：表面張力
$\chi_{ex,0}$ ：$\Delta H_i=0$ の場合の管出口クオリティ

2. 無次元整理式

前報[10]で著者が導いた無次元整理式のうち，本報で必要とするものを以下に再録する．まず管入口に飽和液が流入（すなわち $\Delta H_i=0$）の場合の限界熱流束 q_{c0} は次の諸式で与えられる（記号は表参照）．

L領域：

$$\frac{q_{c0}}{GH_{fg}}=C\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d} \quad \cdots\cdots(1)$$

（C は定数で $C=0.25$ または $C=0.34$）

H領域：

$$\frac{q_{c0}}{GH_{fg}}=0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\frac{1}{1+0.003\,1\,l/d} \quad \cdots\cdots(2)$$

H領域およびN領域：

$$\frac{q_{c0}}{GH_{fg}}=0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}\frac{(l/d)^{0.27}}{1+0.003\,1\,l/d} \quad \cdots\cdots(3)$$

HP領域：

$$\frac{q_{c0}}{GH_{fg}}=8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.453}\times\frac{1}{1+107(\sigma\rho_l/G^2l)^{0.54}l/d} \quad \cdots\cdots(4)$$

なお式（3）に関し，H領域とN領域の境界は次式で定められる．

$$\frac{\sigma\rho_l}{G^2l}=\left(\frac{0.77}{l/d}\right)^{2.70} \quad \cdots\cdots(5)$$

次に入口サブクール ΔH_i が正である場合，その限界熱流束は，各領域により図1の $\Delta H_i>0$ の部分に示すごとき特性の変化を示し，L，H，HP領域につ

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いては次式で q_c が求められる．

$$q_c=q_{c0}(1+K\Delta H_i/H_{fg}) \quad \cdots\cdots(6)$$

ここで K は，各領域に対し次のごとく与えられる．

L 領域の K：

$$K_L=1 \quad \cdots\cdots(7)$$

H 領域の K：

$$\left.\begin{aligned} &\frac{\sigma\rho_l}{G^2l}<3\times10^{-6} \text{ で} \\ &K_H=1.8\left(\frac{130}{l/d}\right)^{5\rho_v/\rho_l} \\ &3\times10^{-6}<\frac{\sigma\rho_l}{G^2l} \text{ で} \\ &K_H=0.075\left(\frac{130}{l/d}\right)^{5\rho_v/\rho_l}\Big/\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/4} \end{aligned}\right\} \quad \cdots\cdots(8)$$

HP 領域の K：

$$\left.\begin{aligned} &\frac{\sigma\rho_l}{G^2l}<4\times10^{-8} \text{ で} \\ &K_{HP}=0.664\left(\frac{\rho_v}{\rho_l}\right)^{-0.6} \\ &4\times10^{-8}<\frac{\sigma\rho_l}{G^2l} \text{ で} \\ &K_{HP}=3.08\left(\frac{\rho_v}{\rho_l}\right)^{-0.6}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.09} \end{aligned}\right\} \quad \cdots\cdots(9)$$

なお $\Delta H_i=0$ の場合，限界熱流束 q_{c0} と管出口クオリティ $\chi_{ex,0}$ の間にはヒートバランスから次の関係がある．

$$\chi_{ex,0}=4(q_{c0}/GH_{fg})l/d \quad \cdots\cdots(10)$$

式 (10) 右辺で q_{c0}/GH_{fg} は式 (1)～(4) で与えられ，そして図 2 はその結果（参考として $l/d=200$ の場合）を示すものである．図 2 の特性から CHF 発生時の管出口近くの流れの状況を次のごとく推定できるように思われる．すなわち，L 領域ではミスト流（液滴の壁面へのディポジット率と気流へのエントレインメント率が CHF に関係），H 領域では環状流(なおフロス，スラグ流も含み，界面の流体力学的条件が CHF に関係)，N 領域では気ほう流（いわゆる核沸騰から膜沸騰への遷移で CHF 発生）である．もちろん，これらについては，さらに慎重な研究を必要とするが，液滴のディポジット率と CHF の関係に関する Bennett ら[11]，Whalley ら[12]の研究，あるいは β 線および X 線により管出口付近のボイド率を測定した Nevstrueva[13] の研究などが参考になると思われる．なお HP 領域の CHF の管出口近くの状況については，著者にはまだ不明である．

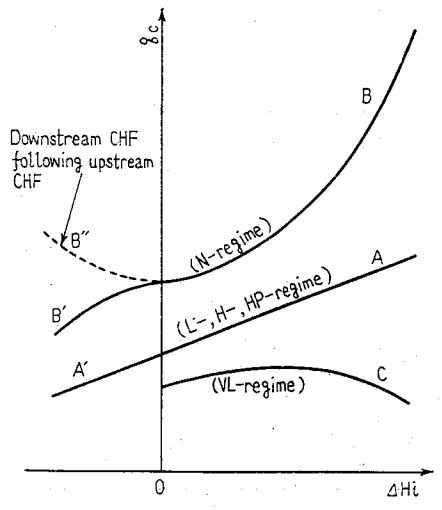

図 1　q_c と ΔH_i の関係 (VL 領域は $d/l<50$, かつ入口流速が 0.2～0.3 m/s 以下の特殊領域)

3. L　領　域

3·1　ミラノ再現性試験データなどとの比較　図 3 は L 領域に関し，水についてのミラノ再現性試験デー

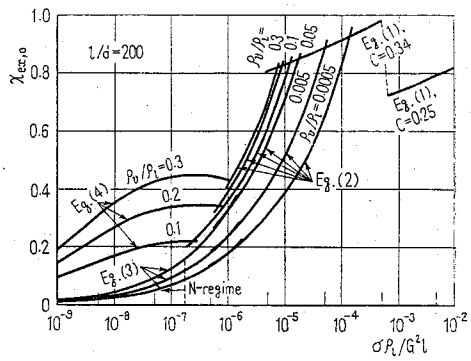

図 2　出口クオリティ $\chi_{ex,0}$ ($\Delta H_i=0$, $l/d=200$ の場合)

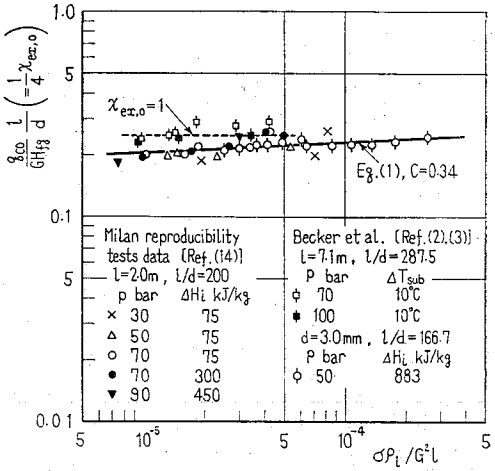

図 3　L 領域における水のデータと式 (1) の比較

タ（ここでは Becker ら[14]が各種の経験整理式との比較に用いたデータを利用），および Becker らの非常に小さい管径（d=3.0 mm）のデータ[2]ならびに非常に長い管（l=7.1 m）のデータ[3]を式（1）と比較した結果である．ただし，ここに q_{c0} は上記のデータ（q_c, ΔH_i）から式（6），（7）によって求めたものである．図3をみるに，非常に長い管では $\chi_{ex,0}=1$ の状況下に CHF を生ずる傾向（図3の破線）が見られるものの，他のデータはすべて式（1）に合致しているとみられる．ただ式（1）の定数 C の値に関し，前報[10]では，フロン類に対して C=0.34，他の流体に対し C=0.25 と定めたのに対し図3の実験データはすべて水であるにかかわらず C=0.34 が対応して前報の結論と矛盾している．

そこで Thompson ら[15]（その表 1-10）の水，Lewis ら[16]の液体水素，窒素，Stevens ら[17]のフロン 12，および Barnett ら[18]のフロン 12，21 のデータから，式（6），（7）によって L 領域の q_{c0} を求め，それらを図4に示した．図3,4の結果によれば，流体の種類によるのではなく，$\sigma\rho_l/G^2l=5\times10^{-4}$ 付近を境に C の値が変わるもののように見える．そして前報[10]の場合には，図1の A 形の実験結果の直線性から q_{c0} を求めたので利用できるデータの数が限られ，そのため $\sigma\rho_l/G^2l<5\times10^{-4}$ の領域にフロンのデータだけが存在，フロンに対し C=0.34 という結果になったものである．

以上により，L 領域の CHF には，一般に図4に示す2種類（C=0.25 および C=0.34）の領域があり，なお管が非常に長いなど特殊な場合は，図3の破線（$\chi_{ex,0}=1$）の場合があって，計3種類の形態が存在し得るといってよいようである．ただし Lowdermilk ら[19]（その表 III）の大気圧の水に関する膨大なデータ（d=1.3～4.8 mm，l/d=50～250）は，図4の $\sigma\rho_l/G^2l$ の全領域にわたり C=0.25 の直線に合致する結果を示す（図4の破線参照）．従って，L 領域の CHF の特性をもっと明確に決定するためには，今後の研究がなお必要と思われる．

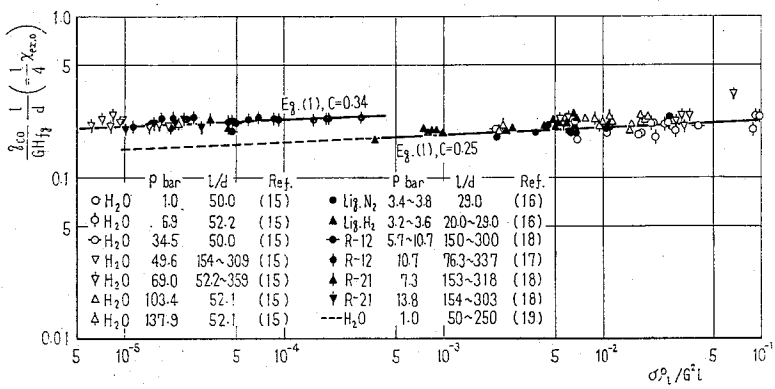

図 4　各種の実験データによる式 (1) の定数 C の検討の値

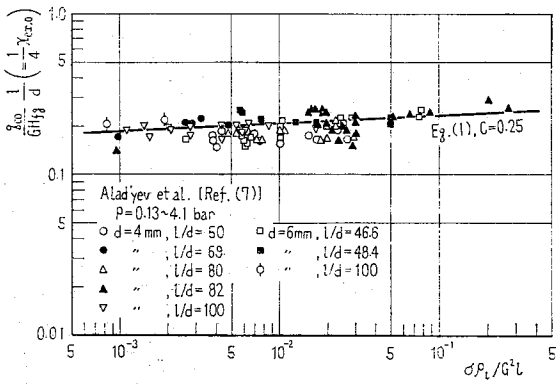

図 5　L 領域におけるカリウムのデータと式 (1) の比較

3・2　カリウムのデータとの比較　図5は，Alad'yev ら[7]のカリウム（圧力 p=0.13～4.1 bar の範囲）の q_c のデータから，式（6），（7）によって求めた q_{c0} を式（1）と比較した結果である．ここでカリウムの表面張力については Kiriyanenko[20] の精密な測定値を用いた．なお l/d=30 の場合のデータは，入口流速が 0.3 m/s 以下の条件のものであるので図5から省いてある（図1の図説参照）．ともあれ実験範囲が $\sigma\rho_l/G^2l>10^{-3}$ にあり，そして C=0.25 の式（1）に合致するとみて差し支えなく（多少，実験データが低めの傾向にあるが），3・1 節の結果に一致しているといえよう．なお Alad'yev らのカリウムのデータは L 領域のものだけである．従ってカリウムの CHF が，L 領域以外の領域で著者の整理式〔式（2），（3）など〕と合うか否かは不明であることを付記する．

3・3　上昇流と下降流の比較　図6は，フロン 12 を用いた Cumo ら[4]の実験データから，図1の直線 A を与える実験結果を用い，ΔH_i=0 における q_c の値（すなわち q_{c0}）を求め，それを式（1）と比較したものである．実験範囲は $\sigma\rho_l/G^2l$

$<2\times10^{-4}$ にあり，そしてデータは $C=0.34$ の式（1）にほぼ合致している．ただし，上昇流の q_{c0} は下降流の q_{c0} に比べ 11％程度高い値を示している．なお，この実験は，質量速度 $G=120\sim325$ kg/m²s (L 領域において)，$l/d=256.4$ の条件下のものである．

4. H 領 域

4・1 ミラノ再現性試験データとの比較 H領域に関し，水についてのミラノ再現性試験データ (Becker ら[14] の使用したもの) から式（6），（8）を通して q_{c0} を求め，それを式（2），（3）と比較したものが図7である．図7で垂直破線は，式（5）による H, N 領域の境界を示す．圧力が3倍も異なるデータを含むにかかわらず，実験データと式（2），（3）との合致はかなりよいといってよいであろう．なお必要があれば式（2），（3）を修正して，図7のデータ群にさらによく合うようにすることは容易であることを付記する．

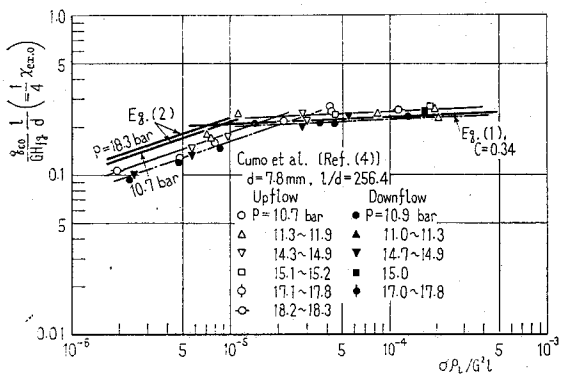

図 6 L および H 領域におけるフロン 12 の上昇流，下降流

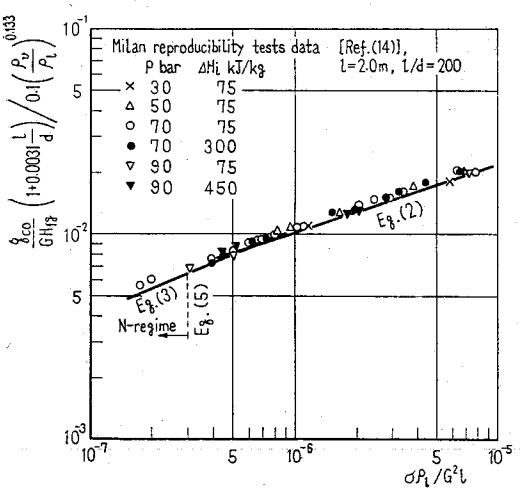

図 7 H 領域における水のデータと式 (2), (3) との比較

4・2 水に関する $d=3$ mm のデータ，$l=7.1$ m のデータとの比較 図8は，Becker ら[2][3] の水に関する $d=3$ mm，および $l=7.1$ m の H 領域のデータから式（6），（8）によって求めた q_{c0} を，式（2），（3）と比較した結果である．$d=3$ mm のデータは式（2）とよく合致している．しかし管長が 7.1 m と長い場合のデータは，精密にいうと式（2）より多少高めに現れているほか，圧力 $p=70$, 100 bar による分離を示し，かつ q_{c0}/GH_{fg} は $(\sigma\rho_l/G^2l)^{1/3}$ ではなく $(\sigma\rho_l/G^2l)^{0.29}$ に比例して変化している．これらの点に関する詳細については将来の研究が必要である．

4・3 上昇流と下降流の比較 図6には，H 領域について Cumo ら[4] の実験データ（図1の直線Aのデータから q_{c0} を求める）と式（2）との比較が示されている．実験は $G=503\sim1\,043$ kg/m²s, $l/d=256.4$ における結果であるが，上昇流の q_{c0} は下降流の q_{c0} の約 11% 高い値を示している．なお，これらの q_{c0} の実験データは全体として式（2）の計算値より低く現れているが，前報[10] の図6，11（各種流体の q_{c0} の実験値と計算値の統計的比較を示す）にもみるごとく，式（2），（3）はフロン 12, 21, 144 の実験値に対し，多少高い計算値を与える傾向を持っているようである．この問題については，あとで8章で論ずる．

5. N 領 域

5・1 フロン 12 のデータとの比較 Groeneveld[5] は，フロン12について実験をおこない，ΔH_i の正負にわたる領域について図1の形（A-A′, B-B′, B-B″）の q_c-ΔH_i の関係を報告して

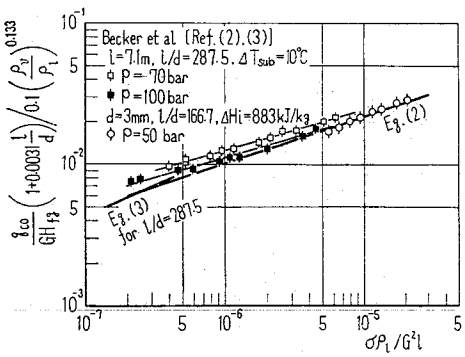

図 8 H 領域における水のデータと式 (2), (3) との比較

いる．従って，その結果から $\Delta H_i=0$ における q_c（すなわち q_{c0}）が容易に求められ，それを式（2），（3）と比較したものが図9である．実験値と計算値との合致はよく，さらに次のことが知られる．すなわち，図9で垂直破線は式（5）による H，N 領域の境界であるが，Groeneveld らの実験データを，H，N 領域に分類したもの（図1の A 形の特性を示すものは H 領域に属し，○印であらわし，他方図1のB形の特性を示すものは N 領域に属し●印で区別）もそれに合致して現れている．

5・2 ベンゼンのデータとの比較 Pokhvalov ら[6]は，$\Delta H_i=0$（飽和液流入）の条件下のベンゼン（圧力 $p=1.53\sim37.6$ bar）のデータを与えている．ただし彼らの実験は，$l/d=16$ と非常に小さい上に，質量流量 G が非常に低い範囲を含んでいる．このような特殊領域（図1の図説参照）では，強制流動沸騰よりむしろ自然流動沸騰が支配する状況が出現し，Pokhvalov ら自身も，G が低い場合の CHF はプール沸騰の CHF についての Kutateladze-Zuber 式の示す特性に近いということを述べている．ところで図10は，彼らのデータを式（1）～（3）と比較〔式（10）で $\chi_{ex,0}$ に変換〕したものである．図10によれば，G が小（すなわち $\sigma\rho_l/G^2l$ が大）の領域は前述のごとく自然流動沸騰が支配的であるため，実験データと整理式はまったく相違するが，G が増加（すなわち $\sigma\rho_l/G^2l$ が減少）するにつれて，$p=1.53\sim16.1$ bar の範囲の実験データが式（3）と合致するようになる．ただし，$p=28.7\sim37.6$ bar（$\rho_v/\rho_l=0.14\sim0.23$）の高圧下の実験データについては，式（3）より多少高めの $\chi_{ex,0}$ の値を示している．

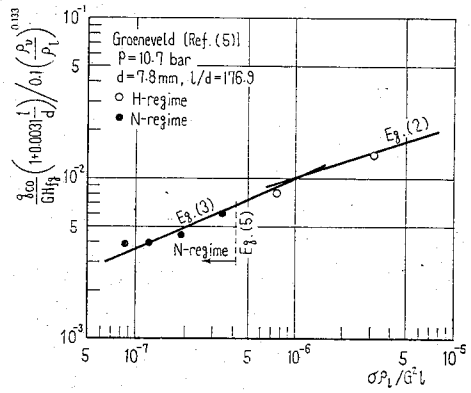

図9 H，L 領域におけるフロン12のデータと式(2)，(3)との比較

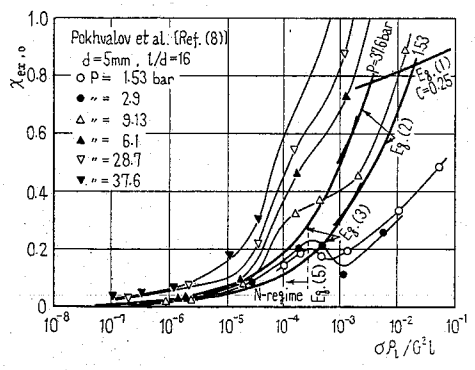

図10 ベンゼンのデータと整理式の比較

6. $\Delta H_i<0$ の状況下における q_c-ΔH_i の関係

管入口に流入する流体が負のサブクール度（$\Delta H_i<0$）を有する場合は，気液二相の流入のため，厳密にいうと，ΔH_i の値の指定だけでは管入口の流れの状況が一義的に決定できないという問題が存在する．しかし，その問題についてここでは触れない．さて図6の実験データを与える Cumo ら[4]の実験結果をみると，L および H 領域では $\Delta H_i<0$ の範囲で，図1の A′ のごとく直線性が成立するとしてよいようである（多少データに不規則性が現れることもあるが）．また，図9の実験データを与える Groeneveld[5]の実験結果をみると，$\Delta H_i<0$ の範囲における q_c-ΔH_i の関係は，H 領域では図1の A′，また N 領域に入ると最初は図1の B′ の特性（しかし CHF は依然管出口で生ずる）を示すが，さらに $\sigma\rho_l/G^2l$ が減少すると図1の B″ の特性（そして CHF は管出口より上流にまず発生）を示すとみてよいようである．ただし上記の $\Delta H_i<0$ の範囲の特性の一般性を確認するためには，もっと多くの実験結果を集積する要がある．

7. H 領域と HP 領域の間の遷移

Campolunghi ら[6]は，非常に長い一様加熱垂直管（$l=11.25$ m，$d=12$ mm）の前流側に同じ長さのコイル状の予熱管を接続した系において水の CHF の実験をしている．彼らの報告からは，垂直管入口に流入する水の ΔH_i の値が不明であるが，主として垂直管内で沸騰を生ずるむねが記されているので，ここでは近似的に長さ $l=11.25$ m の垂直管の入口で $\Delta H_i=0$ の状況にあるものと仮定する．そして，この条件下に式（2），（4）〔および式（10）〕から求めた $\chi_{ex,0}$ の計算値を Campolunghi らの実験結果〔文献（21）の図5から採録〕と比較したものが図11である．

前述のごとく実験条件に不明確さがあるが，図11をみるに，HP 領域で実験と式（4）の計算が比較的近くに現れる反面，H 領域では実験が式（2）の計算

より相当に高めである．この点は，l=11.25 m という管長を考えると，図8に示した l=7.1 m の場合の特性と同傾向といえるかも知れない．なお図 11 をみると，H 領域から HP 領域に移る間の遷移について，大まかな傾向としては式（2），（4）は実験と合っているといえるものの，H 領域の $\chi_{ex,0}$ の実験データが p の上昇と共にかなり急な低下を示す傾向とはやや相違している．従って，この付近の CHF の正確な変化状況について今後の研究が必要であるように思われる．

8. CHF のモデリング

8·1 無次元量 $\sigma\rho_l/G^2l$ の支配下の F_G いま式（1）～（9）の無次元整理式（これらは完全なものではない）から離れ，ただし，一様加熱垂直円管内の CHF が5個の無次元量 q_c/GH_{fg}, $\sigma\rho_l/G^2l$, ρ_v/ρ_l, l/d, $\Delta H_i/H_{fg}$ に支配される現象であるとすると，次のことが成立する．すなわち相異なる流体を用いる2個の実験系 W, F の間において，いま

$$\left.\begin{aligned}&(\rho_v/\rho_l)_W=(\rho_v/\rho_l)_F\\&(l/d)_W=(l/d)_F\\&(\Delta H_i/H_{fg})_W=(\Delta H_i/H_{fg})_F\end{aligned}\right\}\cdots\cdots\cdots\cdots\cdots\cdots(11)$$

の条件を満足させたうえで，両実験系の質量速度 $(G)_W$, $(G)_F$ を

$$(\sigma\rho_l/G^2l)_W=(\sigma\rho_l/G^2l)_F\quad\cdots\cdots\cdots\cdots\cdots\cdots(12)$$

を満足するように選んで実験すると次式が成立し，F 系の q_c の値から W 系の q_c の値を知ることができる．

$$(q_c/GH_{fg})_W=(q_c/GH_{fg})_F\quad\cdots\cdots\cdots\cdots\cdots\cdots(13)$$

いま簡単のため $(d)_W=(d)_F$ の場合を考えると，式（11），（12）の条件下に式（13）は次のごとく書き直せる．

$$\left.\begin{aligned}&\frac{(q_c)_W}{(q_c)_F}=F_G\frac{(H_{fg})_W}{(H_{fg})_F}\\&\text{ここに}\\&F_G=\frac{(G)_W}{(G)_F}=\sqrt{\frac{(\sigma\rho_l)_W}{(\sigma\rho_l)_F}}\end{aligned}\right\}\cdots\cdots\cdots\cdots\cdots\cdots(14)$$

ここに F_G は質量速度変換係数と呼ばれ，通常は W に水，F に他のモデル流体をとるものをいう．

8·2 考察 質量速度変換係数 F_G に関し，Dix[22] はフロン 114, 12, 22, 21 の CHF の実験データ（おそらく H および N 領域と推定される）を水の CHF の実験データと比較検討，経験的に求めた F_G の値が $\sqrt{(\sigma\rho_l)_W/(\sigma\rho_l)_F}$ に比例することを示している．この経験的結果は，式（14）の指示に一致する関係である．

一方，表1は ρ_v/ρ_l=0.05 の場合に対し，実験結果（H および N 領域のものと推定される）の分析から，R12 に対し Stevens ら[23], R22 に対し Staub[24], R113 に対し Coffield[25], CO_2 に対し Hauptmann ら[26]が求めた F_G の経験値を，式（14）の与える F_G の値と比較したものである．なお表1には，Ahmad[9] が次元解析を基に導いた F_G の値をも示すが，これは次式で与えられるものである．

$$F_G=\left\{\frac{(\gamma/\rho_l\mu_l)_F}{(\gamma/\rho_l\mu_l)_W}\right\}^{1/3}\left\{\frac{(\mu_l/\mu_v)_F}{(\mu_l/\mu_v)_W}\right\}^{1/8}\quad\cdots\cdots(15)$$

ここに $\gamma=|\partial(\rho_l/\rho_v)/\partial p|_{\text{saturation}}$ である．表1によると，フロンの場合，式（14）の F_G の値は，F_G の経験値より小さいことが見られる．これは 4·3 節に述べた事実，すなわちフロンの q_{c0} の実験値は式（2），（3）の q_{c0} の計算値より低いということに一致する事柄でもある．そして Ahmad の与えた式（15）による F_G の方が，F_G の経験値に近い．元来，Ahmad の F_G は，本報で使用した無次元量 $\sigma\rho_l/G^2l$ の代わりに次の無次元量 ψ_{CHF} を用いる時に導かれる F_G であるとみなすことができる．

$$\frac{1}{\psi_{CHF}}\equiv\left(\frac{\sigma\rho_l}{G^2d}\right)^{2/3}\left(\frac{Gd}{\mu_l}\right)^{1/3}\left(\frac{\mu_l}{\mu_v}\right)^{1/5}\quad\cdots\cdots(16)$$

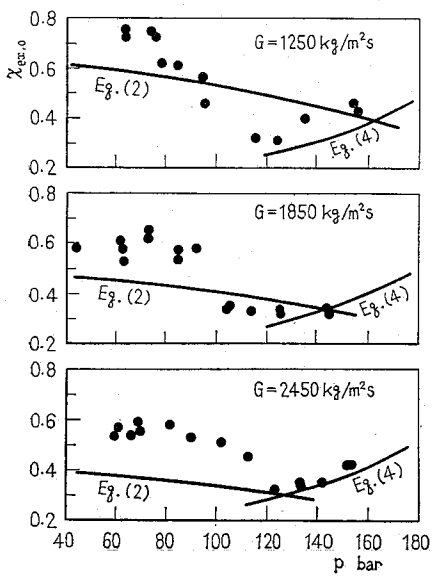

図 11 l=11.25 m, l/d=937.5 の垂直管による水の CHF の実験（Campolunghi ら[6][21]）

表 1 F_G の値の比較（ρ_v/ρ_l=0.05 の場合）

		R12	R22	R113	CO_2
F_G	経験値	1.40[23]	1.40[24]	1.46[25]	1.13[26]
	式 (14)	1.28	1.34	1.21	1.19
	Ahmad[9]	1.38	1.38	1.34	1.2

この事実は，少なくともHおよびN領域における問題に関し，流体の粘性がCHFの発生に対し二次的な影響を持ち得ることを意味しているといえる．すなわち，より一般性のある整理式やモデリングのためには，無次元量 $\sigma\rho_l/G^2l$ のほかに，レイノルズ数 Gd/μ_l および粘性比 μ_l/μ_v を考慮する要があるであろう（前報[10]の2章では μ_l, μ_v を無視して次元解析をすることをことわってある）．しかし，特に精密さが強く要求されない場合は，無次元量 $\sigma\rho_l/G^2l$ だけによる取扱いの方が簡便，かつ広い見通しを与えるのに有効であろう．少なくとも本報の式（1）～（9）は，かかる性格のものと解すべきであろう．

なお強制流動沸騰のCHFに関する従来の研究では，管径 d による無次元量 $\sigma\rho_l/G^2d$ はときどき考慮されているが，管長 l による無次元量 $\sigma\rho_l/G^2l$ を考慮する研究は見当たらないように思われる．式（1）～(9) の有効性は前報[10]および本報で明らかと考えられるが，これは無次元量 $\sigma\rho_l/G^2l$ の重要性を示すものということができるであろう．

8. 結　論

（1） 無次元整理式（1）～（9）の適用性はかなり広く，実験結果の統一的な比較検討のための媒体としても有効な役割を持ち得ると考えられる．

（2） ミラノ再現性試験データは，L領域で式(1)，H領域で式（2），(3）にかなりよく合う．一方，非常に長い管の場合，CHFの実験データは式（1）～（3）から多少偏る傾向があるようである．

（3） L領域で式（1）の定数 C は，$\sigma\rho_l/G^2l=5\times10^{-4}$ 付近を境にして $C=0.34$ から $C=0.25$ に変わることが多いようである．なお管長が特に長い管では $\chi_{ex,0}=1$ の条件でCHFが発生することがあり得る．

（4） 少なくともHおよびN領域では，一般的な整理式ないしモデリングの構成を目的とする場合，無次元量 $\sigma\rho_l/G^2l$ に加え，レイノルズ数 Gd/μ_l，粘性比 μ_l/μ_v などの二次的影響を考慮する要があるようである．

（5） H領域からHP領域へ遷移する領域のCHFの変化の詳細については，将来の研究を要する．

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討　論

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限界熱流束に関する精密にして斬新なるご検討結果発表に対し深く敬意を表す．

（1） 式（1）～（4）で（$\sigma\rho_l/G^2l$）なる一種のウェーバ数の逆数を基本無次元数とされておられ，実験結

果の整理もみごとにされているが，ウェーバ数としては物理的モデルとしてより長さの短いdをとる方が考えやすく，lを取る事は考え難いように思われる．そのため$(\sigma\rho_l/G^2d)$を基準とし，lの影響をすべて$f(l/d)$の形で分けてしまう方が物理的意味の説明には都合はよくないか．

（2） 内部流動の変化にてN領域とH領域を分離しておられるが，観察によるこのことの確かさはいかほどか．

〔回答〕 （1） ウェーバ数の物理的モデルには，長さの短いdをとる方が考えやすく，lをとることは考え難い，従って$\sigma\rho_l/G^2l$を使うより$(\sigma\rho_l/G^2d)(d/l)$を使う方が都合がよいのではないかとのご意見だが，そういわれるからには，それを支持するような，なんらかの物理的イメージを質問者は持っておられるはずだと思う．しかしそれにしても，そのイメージが現在の現象に本当に合ったものであることが明確にされぬ限り，どんな議論も無意味であろう．そもそも，本報で使う無次元量$\sigma\rho_l/G^2l$の源泉は，長方形加熱面上に液体を流す沸騰系に関する著者ら(付1)の実験結果（流れの方向の加熱面の長さをlとするときCHFが$\sigma\rho_l/G^2l$に支配される）にある．この実験事実は長さlの上に成立し，それ以外の長さは見当たらぬという状況下のものである．

（2） N領域とH領域は，流れの変化から分離したのではなく，図1に示すq_c-ΔH_iの実験特性の相違（H領域は直線関係，N領域は非直線関係）から分離，そして前報(10)の図13のように多数の実験データについて検討した結果である．これに対し流れの変化の方は，あくまで推定にすぎない．

〔質問〕 **成合英樹**（船舶技術研究所）

これまでまとまりの悪かった限界熱流束の実験データの整理に，次元解析により無次元量を定めみごとに整理されたことに敬意を表す．

ところで，限界熱流束は基本的には流路中の局所的な条件で定まると考えられる．それに対し，本解析での重要なパラメータ$\sigma\rho_l/(G^2l)$には，管入口からの長さlという量が含まれている．この管長lは，均一加熱の円管のヒートバランスから，出口のバーンアウト乾き度に対応するものと考えてよいのか．その観点からすると，無次元量$\sigma\rho_l/(G^2l)$と$\sigma\rho_l/(G^2d)$の相違は，出口乾き度をimplicitに含むかどうかということになる．

次に，二相流における局所値という観点から速度uのかわりに，液相速度u_l，気相速度u_vを用いて次元解析を行うと，本論文中の無次元量は乾き度をexplicitに含み，また気液スリップ比（あるいは密度，乾き度，ボイド率の比の関数）などの無次元量がでてくる．このような局所値での整理はまとまりが悪くなるのか．局所値と結びつけられれば，均一加熱でない場合などに対する適用も可能と考えられる．

〔回答〕 著者は，質問者のお考えに反し恐縮だが，L，H，HPなどの領域により異なるものの（次々報(付2)参照），一般論をいうと限界熱流束は基本的には局所条件では定まらないという考えに立っている．

もちろん管長lは，一様加熱の場合，ヒートバランスから出口クオリティχ_{ex}との間に常に次の関係を持つ．

$$\pi dlq_c=\frac{\pi}{4}d^2G(\Delta H_i+\chi_{ex}H_{fg}) \quad \cdots\cdots\cdots\cdots\cdots\cdots(\mathrm{i})$$

しかしlocal conditions hypothesisの立場に立ち，χ_{ex}がCHF発生の主要支配因子と考える形のCHF整理方式〔一般に管長lは式（i）の関係で問題に入ってくるだけ〕は，従来あまりうまく行っていないのが現実（それゆえ無次元整理式も導かれない）ではないであろうか．周知の事実といってもよいであろう．

一方，本報で使う無次元量$\sigma\rho_l/G^2l$が出て来た源泉は，前報(10)にみるごとく，著者ら(付1)の長方形加熱面上に液体を流す沸騰系のCHFの実験結果（流れの方向の加熱面の長さをlとするときCHFが$\sigma\rho_l/G^2l$に支配される）に存している．この実験事実について，まだ完全な物理的解釈は与えられていないにしても，著者の前報(10)の水力不安定解釈（方向性次元解析による），あるいはLienhardら(付3)による理論的説明などが試みられている．そして，そういうことも関連してくる問題なので，それを抜きにして$\sigma\rho_l/G^2l$に勝手な解釈ないし変更を加えるようなことは軽々にはできないと思う．

それにしても，はっきり言えることは，とにかく$\sigma\rho_l/G^2l$を使うと，CHFに関するいろいろのことが，統一的にとてもよく整理できるということである．それは事実である．

〔質問〕 **井上 晃**（東京工業大学原子炉工学研究所）

種々の流体と流動条件について，適用性の広い限界熱流束（CHF）の整理式が得られ，しかも二相流の流動様式に対応していくつかの領域に分けられているが，複雑と考えられていた管内流のCHFが，比較的

（付1） 甲藤・石井，機論，**44**-384（昭53-8），2817.

（付2） 甲藤，機講論，No. 790-6（昭54-4），63.

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簡単な整理式で統一的にまとめられ得たことに興味深いものを感じている．これらの無次元整理式の意味と以下に述べる特殊な条件下への適用についてご意見をうかがいたい．

（1）（前報で詳しく説明されているのかも知れないが，前報を拝見する機会がなかったので，質問する）．

一連の無次元整理式の最も興味深い点は，本文中でも強調されているようにウェーバ数（$\equiv\sigma\rho_l/G^2l$）に加熱部長さ l を用いた点にあると考えられるが，この無次元量の CHF または二相流動に関した物理的意味は，どう考えるべきか．

（2） H 領域と N 領域は，それぞれ環状流と気ほう流に対応していると説明されている．流動様式は系圧力に応じて変化して行き，流動条件だけでは一意的に定まらないと考えられるが，領域の境界を示した式（5）に，系圧力の情報（P または ρ_v）またはその帰結としての q_{c0} が入って来ないのはどうしてか．流動様式の変化はそれほど厳密には CHF が対応しないということか．

（3） 入口で二相流となっているような特殊な条件（$\Delta H_i<0$）では，図2のように $\chi_{ex,0}$ と $\sigma\rho_l/G^2l$ が一意的に決まらず ΔH_i（または χ_i）によっても変化すると考えられるが，このような条件では整理式はどう考えればよいのか．

（4） 本論文で得られた整理式は，単円管の場合ということだが，環状流路中の円管加熱とか原子炉炉心などの Rod bundle などの CHF に適用する場合，単に熱または水力等価直径などの補正でよいのか．

〔回答〕 （1） 流路内の流れの CHF について，一般に $\sigma\rho_l/G^2l$ の物理的意味はなにかということになると，卒直にいって，著者の現在の研究水準をいささか越えたご質問になろうかと思う．ただ無次元量 $\sigma\rho_l/G^2l$ の出て来た源泉は，前報[10]にみるごとく長方形加熱面上に液体を流した沸騰系に関する著者ら(付1)の実験結果（流れの方向の加熱面の長さ l に対し CHF が $\sigma\rho_l/G^2l$ に支配される）にある．

（2） H 領域と N 領域の分離は，環状流と気ほう流（これらはあくまでも推定）に対応させたものではなく，図1に示す q_c-ΔH_i の特性の相違（H 領域は直線関係，N 領域は非直線関係）から分離したもので，前報[10]の図 13 のごとく多数の実験データの検討から式（5）が得られている．なお，ここで N 領域にまで入っている実験データの圧力範囲は，水の場合，70.3～112.5 kg/cm² abs にわたる状況にある．ただし N 領域（そこでは $\sigma\rho_l/G^2l$ が小）は実現が比較的困難なので，データも手薄な傾向にあり，従って本当に正確なことは将来を待った方がよいかも知れない．

（3） $\Delta H_i<0$ の場合も，6章に記した条件下に，L, H, HP 領域では図1のごとく q_c-ΔH_i の直線関係が成立する．従って与えられた管長 l を用い，q_{c0} に関し式（1）～（4），K に関し式（7）～（9），そして式（6）〔ここで $\Delta H_i<0$ となるのみ〕によって普通のとおり q_c が求められる．なお著者の次々報(付2)で扱っているので，それを参照いただきたく思うが，特に L 領域では CHF の発生に上流の影響がほとんどなく，従って $\Delta H_i<0$ の場合（いわゆる mixed inlet condition），ΔH_i（または χ_i）が零となる点まで仮想的に管を延長し，その管の長さに対し式（1）で求められる q_{c0} が，前記の方法で求めた q_c に一致することが証明される．

（4） 直接的な回答をまず申し上げると，一般には，そのような補正だけではだめで整理式自体が少し変わるし，また上記の補正にしても，CHF に対し水力等価直径の使用（従来の整理式ではよく用いられている）は原理的におかしく，一様熱流束条件下では加熱等価直径のみを使うべきである．なお著者の次報(付4)でこの二重管の CHF の無次元整理を扱っているので，それを参照していただきたい．

〔質問〕 **鳥飼欣一**（日本原子力研究所）

ミラノ再現試験データが昭和 45 年に示されているが，我が国においても「燃料集合体のバーンアウト機構に関する試験研究」が機械学会で実施(付5)され，得られた結果は，かなりのデータのばらつきがサブクーリングがちょうど0になるあたりであった．これとの比較はどうか．

ただし，上記実験は，同じ寸法の試験部を各所の実験装置に取付けて実施したもので，上向き円管内丸棒発熱の試験部であった．

〔回答〕 （1） 円管内丸棒加熱の条件は，二重管の内管のみ加熱の場合に相当する．そして垂直二重管の場合の CHF の管理については，著者の次報(付4)で扱っているので，それを参照していただきたく思うが，その結果によると二重管の内管加熱の場合は特に，単一円管の場合とかなり性格が異なり，従って残念ながら本報の単一円管の整理式では処理できない．

（2） しかし，ご指摘いただいたデータは貴重なものである．なお著者の次報(付4)にみられるように，

（付4） 甲藤，機講論，No. 780-18（昭 53-11），37.
（付5） 燃料集合体のバーンアウト機構に関する試験研究分科会報告，（昭 43），日本機械学会．

従来発表されている二重管のデータ数は，特にL領域，N領域に関し，非常に少ないので，もし上記のデータが，この領域のデータを多く含んでいると，さらに貴重なものになる．従って今後，機会をみて分析してみたく思っている．

〔質問〕　赤川浩爾（神戸大学工学部）

広範囲の流動状況に対してCHFが統一的に整理されていることに敬意を表す．

L領域の整理方法について次のことに関するご見解をお聞きしたい．各実験系列の q_c の実験値から式(7)の $K_L=1$ として q_{c0} を求めて図4の結果がえられている．これによると本文にあるように「$\sigma\rho_l/G^2l=5\times10^4$ 付近を境に C の値が変わるもののように見える」．しかし一方H，HP領域の K は式（8），（9）のように2領域に分けて表示されている．そこでL領域についても K_L の値を $\sigma\rho_l/G^2l$ の2領域に分けて表示すると，式（1）の C は一定にならないだろうか．

〔回答〕　図4は，ご指摘のごとく

$$q_{c0}=q_c/(1+K_L\Delta H_i/H_{fg}) \quad \cdots\cdots\cdots\cdots\cdots\cdots(ii)$$

で $K_L=1$ とおき，q_c，ΔH_i の実験データから求めた q_{c0} を図示している．しかし，$\Delta H_i/H_{fg}\fallingdotseq0$ の条件にあるデータでは，K_L の値の大小は q_{c0} の結果にほとんど影響を与えない．また特に前報[10]の場合は，実験結果から直接求めた q_{c0}（これには K_L は関係せず）を分析して $C=0.34$ と $C=0.25$ の区別があったわけである．従ってL領域の場合は，まず C の二つの値が存在，それに対応して K_L の値が二つ存在（ただし両者とも1に近い）というのが筋と思う．実は著者の次々報(付2)において，この入口サブクールの影響因子 K の理論解析を試みており，それを参照いただきたく思うが，その結果によると K_L は次式のごとく導かれる．

$$K_L=\frac{1}{4C}\frac{1}{(\sigma\rho_l/G^2l)^{0.043}} \quad \cdots\cdots\cdots\cdots\cdots\cdots(iii)$$

ここに，$C=0.34$ 及び 0.25．そして式（iii）の理論結果は，ばらつきは別にして K_L の実験データ（前報[10]の図15に図示）とかなりよい一致をみせるように思われる．

536.423.1

一様加熱垂直円管内の強制流動沸騰限界熱流束の研究*

（ドライアウト，入口サブクール，沸騰長さなどに関連する事項について）

甲　藤　好　郎**

1. 序

一様加熱垂直円管内の強制流動沸騰の限界熱流束(CHF)に関して，さきに著者は無次元整理式を導き[1]，さらにいくつかの重要なデータとの比較により，その整理式がCHFの各種実験結果の統一的な比較の物指しになり得ることを示した[2]．

さて円管内の強制流動沸騰のCHFの機構に関し，環状流ないし環状噴霧流のCHFと気液挙動の関係が次第に明らかにされつつあるようである〔例えば文献(3)，(4)参照〕．そして現在のところ，気液の挙動が最も明確にされているといえるのは，高クオリティ領域でのドライアウト（およびドライアウト後の熱伝達）についてであろう．従って，ドライアウトの研究[5]~[9]にかかわるCHFの実験データについて，その特性を検討しておくことが必要である．

他方，限界熱流束 q_c に対する入口サブクールエンタルピ ΔH_i の影響に関し，著者の無次元整理式では，次式（記号の説明については記号表参照）の形で与え

$$q_c=q_{c0}(1+K\Delta H_i/H_{fg}) \quad \cdots\cdots(1)$$

ここに入口サブクールの影響因子 K に関し，実験データから求めた K の値を純経験的に整理した式を与えている[1]．しかし，この整理法について，それがなんらかの論理性を持つものか否か，またいわゆる沸騰長さ（boiling length）による整理法とどんな関係を持つか，といった面に関し検討しておくことが必要である．

本報では，上記の諸事項（以下にみられるようにCHFの発生に対する上流履歴の影響の問題が共通のベースを構成している）について研究した結果を報告する．

記　号

d：加熱管内径
G：質量速度
H_{fg}：蒸発潜熱
ΔH_i：入口サブクールエンタルピ
K：入口サブクール影響因子
l：加熱管長
l_b：沸騰長さ
p：絶対圧力（図2～6），8～11 bar
q_c：限界熱流束
q_{c0}：$\Delta H_i=0$ の場合の q_c
z：管軸方向座標
α：ボイド率
ρ_l：液体密度
ρ_v：蒸気密度
σ：表面張力
$\chi_{d.o.}$：ドライアウト発生点のクオリティ
χ_{ex}：管出口クオリティ
χ_{in}：管入口クオリティ

2. ドライアウトの実験データの分析

2·1 R-113の実験　植田ら[5][6]は，予備円管内で作ったR-113(圧力 $p=3.0$ bar)のミスト流を，一様加熱垂直円管（直径 $d=0.01$ m）の下端からクオリティ

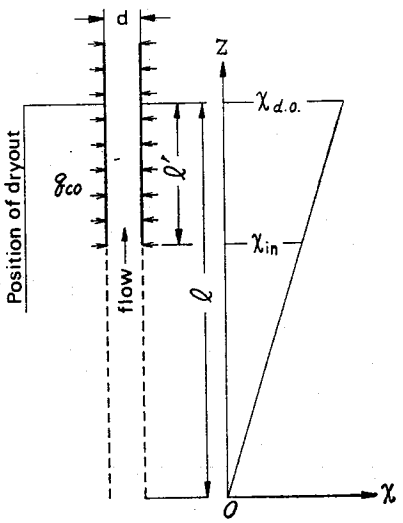


図 1　垂直円管

* 昭和54年4月5日　第56期通常総会講演会において論文講演として講演，原稿受付　昭和53年10月5日．
** 正員，東京大学工学部（〒113 東京都文京区本郷 7-3-1）．

χ_{in}，質量速度 G で流入させる状況（いわゆる mixed inlet condition）下の実験で，壁温分布の測定をおこなっているが，それからドライアウト点（壁温が急上昇する位置）を知ることができる．いまもし，このような高クオリティ下の CHF は，ドライアウト点の比較的近くの流れの状況だけに支配されて発生するものと仮定すると，図1の破線のごとく一様加熱円管（熱流束は不変で q_{c0} とする）を延長し，その下端から飽和液（$\chi=0$）が流入する場合を考えても，上と同じドライアウト状況が保たれることになる．そしてこの時，図1の関係から次の2式が成立する．

$$\pi d l' q_{c0}=\frac{\pi}{4}d^2G(\chi_{d.o.}-\chi_{in})H_{fg} \quad \cdots\cdots\cdots(2)$$

$$\chi_{in}/\chi_{d.o.}=(l-l')/l \quad \cdots\cdots\cdots(3)$$

式（2），（3）から l'，χ_{in} が消去され，直ちに次式が得られる．

$$l/d=(\chi_{d.o.}/4)(GH_{fg}/q_{c0}) \quad \cdots\cdots\cdots(4)$$

式（4）により，$\chi_{d.o.}$，G，q_{c0} の実験データから，管長 l（飽和液の流入する円管入口から CHF 位置までの距離）を定めることができる．そして図2は，このようにして求めた l を用い，かつ〔付録1〕に示す式（20）によって植田ら[5][6]のドライアウトの実験条件を L 領域と H 領域に分類したものである．すなわち△印が L 領域の実験，▲印が H 領域の実験，また△印で境界線近くの H 領域実験を示している．

なお最近，植田[7]は長さ $l_0=2.45$ m，直径 $d=0.01$ m の一様加熱垂直円管の下端から R-113（圧力 $p=3.2$ bar）のサブクール液が流入する条件下に，管出口に生ずるドライアウトの実験をおこなっている．この時も，前と同様な考えで管長 l（現在，入口クオリティ $\chi_{in}<0$ の条件に対し，$\chi=0$ の位置から管出口までの距離，すなわち沸騰長さに相当）を考え，この l について次式が得られる．

$$l/d=(l_0/d)+(\chi_{in}/4)(GH_{fg}/q_{c0}) \quad \cdots\cdots\cdots(5)$$

ここに，$\chi_{in}<0$

図2には，植田[7]の実験条件も図示してあるが，これらに関し l は式（5）によっている．

2・2 液体窒素の実験　Hynek ら[8]は，液体窒素（圧力 $p=1.38$ bar）を一様加熱垂直円管（直径 $d=0.0102$ m）内を質量速度 G で流した場合，管途中に生ずるドライアウト点における $\chi_{d.o.}$ を与えている〔文献（8）の図 12, 19, 20〕．この場合，式（4）により管長 l が求められ，これを用いると図3にみる

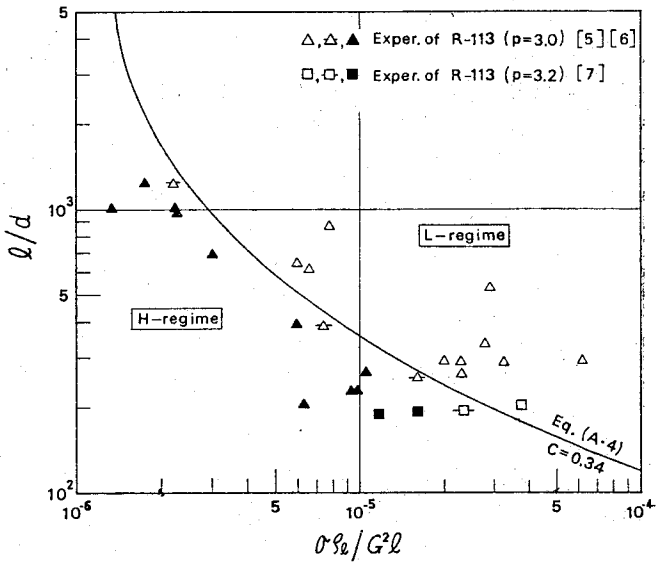
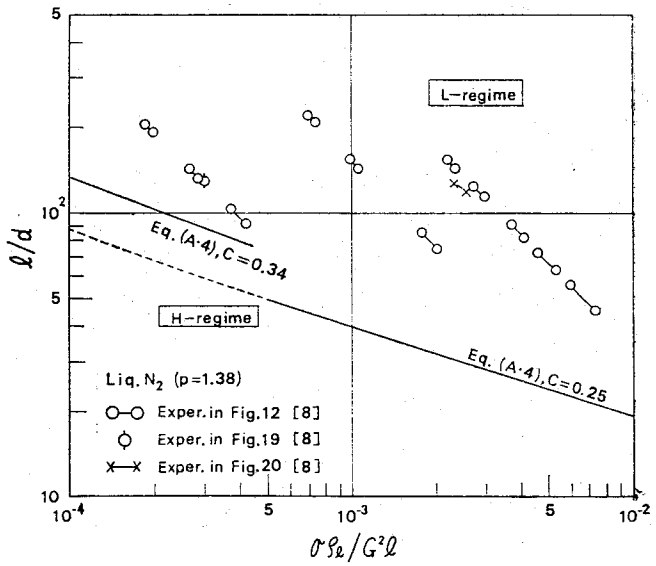


図 2　R-113 のドライアウトの実験条件



文献（8）の図 19, 20 でそう入テープを持つ実験は除く．また図 20 で $G=2\times10^4$ lb/ft²h の実験は，$l/d<29$，入口流速≒0.03 m/s で VL 領域〔文献（2）の図1説明参照〕に属する特殊条件なので除く

図 3　液体窒素のドライアウトの実験条件

ごとく，全実験が L 領域にあることがわかる．

2·3 水の実験　Bennett ら[9]は，圧力 p=69.0 bar のサブクール水を一様加熱垂直円管（直径 d=0.012 6 m）内に質量速度 G で流入させる場合について，ドライアウト後の熱伝達の研究をおこない，これに関連して管途中に生ずるドライアウト位置を沸騰長さ（χ=0 の位置からドライアウト点までの距離）の形で示している〔文献（9）の図 14〕．そして，この沸騰長さは，2·1 節，2·2 節で用いた管長 l と同内容ゆえ，とりあえず，この沸騰長さを管長 l として実験条件を分類してみた結果が図 4 である．

2·4 L 領域の CHF の実験データ　図 2～4 において L 領域に属するとみなされる実験（および L，H 境界線近くの実験）で求められた CHF のデータを，〔付録 1〕に示す L 領域 CHF 整理式（17）と比較した結果が図 5 である．実験データに多少のばらつきはあるが，式（17）との一致はかなり良いといってよいであろう．特に植田ら[5][6]のデータは，l/d に関し l/d=200～1 250 という広い範囲（図 2 参照）にわたって式（17）との一致を示している点で興味がある．なお図 5 の縦軸の値から明らかなごとく，大部分のデータに対し，ドライアウト点のクオリティ $\chi_{d.o.}$ は，ほぼ $0.7<\chi_{d.o.}<1.0$ の高クオリティの範囲にある．また図 5 に得られた結果，および図 1 のような特殊条件（mixed inlet condition）のデータも含むことから考えて，一般に L 領域の CHF というものは，いわゆるドライアウトと呼ばれている現象に一致すること，および CHF 発生点の近傍の流れの状況に主として支配（上流の影響はあまり受けない）されて発生する性格のものであるといってよいであろう．

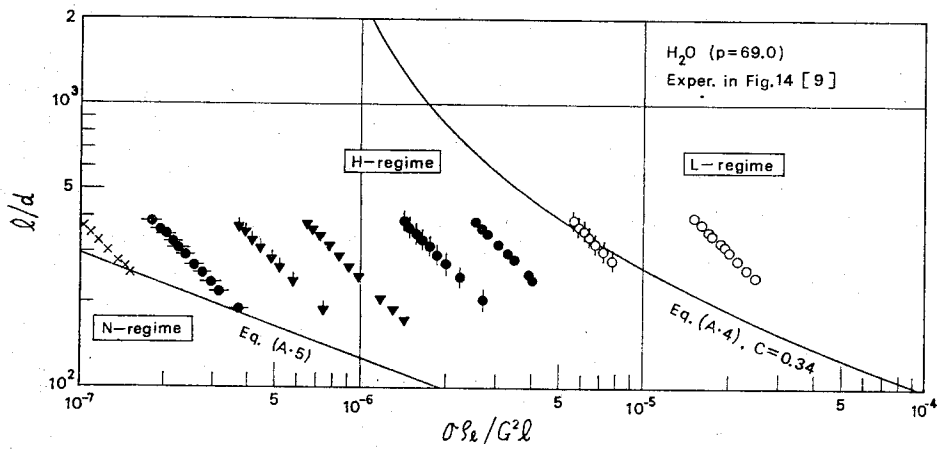


図 4　水の CHF の実験条件

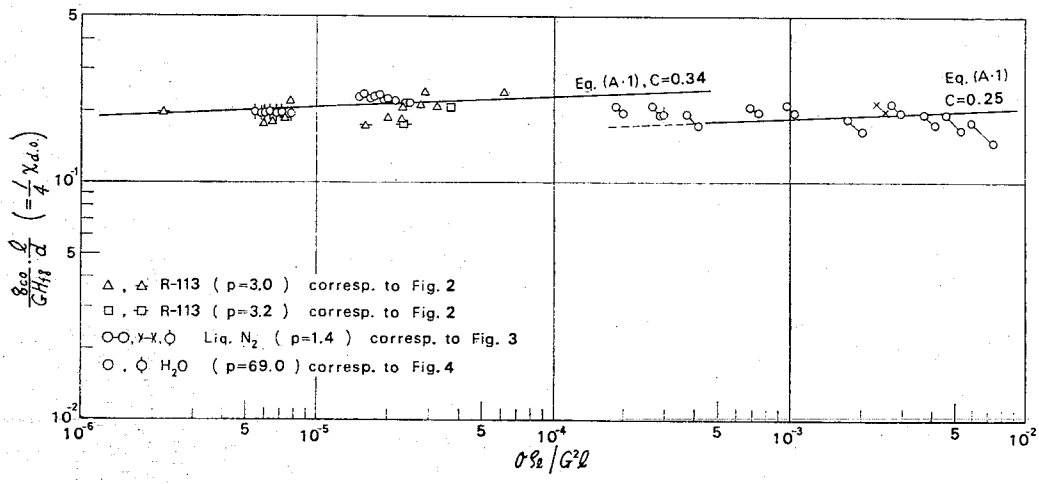


図 5　L 領域の CHF の実験データと式（17）の比較

2·5 H 領域の CHF の実験データ　図2～4において H 領域に属するとみなされる実験（および L，H 境界線近くの実験）で求められた CHF の実験データを，〔付録1〕に示す H 領域の CHF 整理式 (18) と比較した結果が図6である．ここでも実験データと式 (18) との一致はかなり良いといってよいが，なお特に注目すべき点が二つある．

第一に，植田ら[5][6]の特殊入口条件下のデータ（▲印）を観察すると，他のデータに比し非常にばらつきが大きく，かつ統計的にみて整理式 (18) からの偏りも大きい．これは，H 領域の CHF は，L 領域の CHF とは違って，上流からの履歴の影響（記憶効果）を受けるものであることを示しているように思われる．

また第二に注目すべき点は，Bennett ら[9]の水に関するデータである．これは2·3節に述べたように，沸騰長さを l にとって整理されているが，式 (18) との一致が $\sigma\rho_l/G^2l$ の相当に小さな値（4×10^{-7} くらい）にまで良好である．しかし一般的に言えば，沸騰長さの起点とされる $\chi=0$ の位置（加熱管の途中にある）での流れの状況は，純粋の飽和液が流入している管入口条件〔整理式(18)はこの条件に対し成立〕とは相違している．従って上述の図6にみられるところの良好な一致は，Bennett ら[9]の実験が比較的に小さな入口サブクール度（$\Delta H_i/H_{fg}=0.036\sim0.115$）でおこなわれていることに起因すると考えられる〔後述の式 (14) 参照〕．なお図6において $\sigma\rho_l/G^2l<4\times10^{-7}$ の範囲の実験データは式 (18) から偏るが，これは沸騰長さの適用の困難な条件になること，および N 領域に近づいていること（図4参照）などによると考えられる．ただし，このような領域は本報の主題外である．

3. L 領域と H 領域の特性の比較

図2～4にみたように，著者の整理式に従って分類すると，いわゆるドライアウトの実験は，L，H 両領域にわたっておこなわれている．しかしドライアウト点ないし環状流領域の CHF の従来の解析的研究〔文献（5），（9），(10) など〕では，上記のような CHF 特性の変化に対応する特性領域の区別はしていない．そして時には低クオリティの N 領域まで含む広い範囲にわたり，壁面上の液膜流量が零になる点（液膜からの液体の蒸発，および液滴のエントレインメントと，主流から液膜への液滴のデポジットのつりあいで生ずるとする）が CHF 点という単一機構モデルの想定下に CHF の推定計算がおこなわれている．なお Hewitt[11]は，水（圧力 $p=69.0$ bar）を円管内に流した時の液膜流量を測定しており，文献 (11) の図6の結果から得られる CHF 点の液膜流量の状況は表1

表 1　Hewitt[11] の CHF 液膜流量の測定結果

水（$p=69.0$ bar），$l=3.66$ m，$d=0.012\,6$ m

	G kg/m²s	$\sigma\rho_l/G^2l$	l/d	CHF 点での液膜流量
(1)	1.35×10^3	1.90×10^{-6}	290	～0
(2)	2.03×10^3	8.42×10^{-7}	290	～0
(3)	2.71×10^3	4.88×10^{-7}	290	小さくはあるが有限値

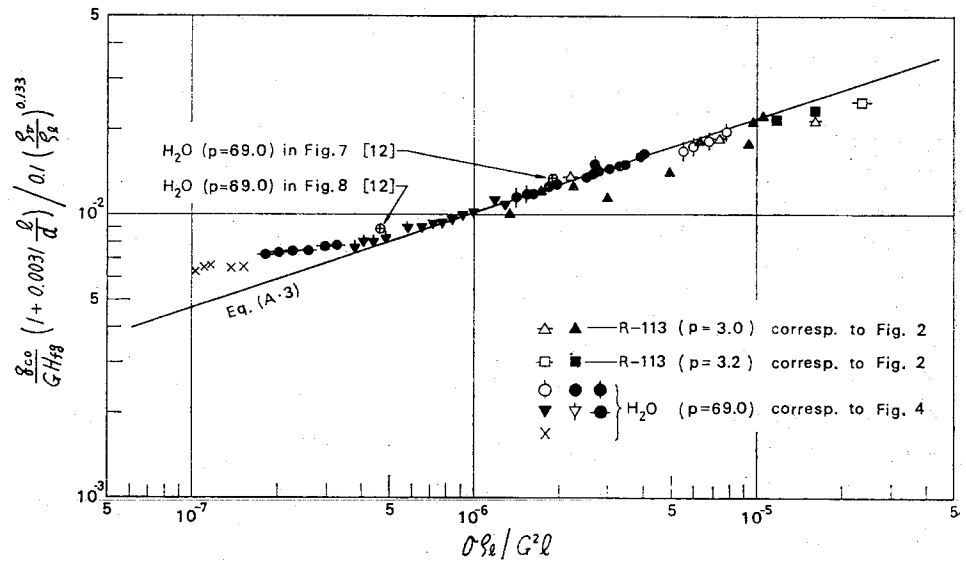


図 6　H 領域の CHF の実験データと式 (18) の比較

のごとくである．表1の3実験（1），（2），（3）は，それぞれの $\sigma\rho_l/G^2l$ および l/d の値から図4により所属領域の区別ができ，それによれば図4の右から数えて第4～6番めの実験グループに含まれる実験条件である．すなわち3実験とも H 領域にあり，そして表1によれば実験（1），（2）では CHF 点で液膜流量はほぼ零という結果にある．なお実験（3）では液膜の破断が CHF 点で生ずるとも解し得るかも知れないが判然とはしない．

他方，Bennett ら[12]は，管壁への液滴のデポジットが CHF を支配する条件を，実際の CHF 条件と比較した結果，暫定的に次の4領域の存在が考えられるとしている*1．

領域Ⅰ：高クオリティで熱流束が非常に低く，液滴のデポジットが CHF を支配する（ただし Bennett らは，mixed inlet condition でないと，本領域は実現しにくいとしている）．

領域Ⅱ：この領域では，加熱管長の変化に対し，CHF 発生の加熱入力はほぼ一定である．

領域Ⅲ：この領域では，限界熱流束の増加と共に出口クオリティ χ_{ex} が急低下する．

領域Ⅳ：この領域の CHF は，古典的な膜沸騰の概念に対応する．

さて上記の Bennett らの領域Ⅰは，その性格から著者の L 領域（前報[1]の3·2節，および前報[2]の2章参照）に属するとみてよい．また領域Ⅱも L 領域に属する．なぜなら，加熱管への入力 $Q=\pi dlq_{c0}$ を式（17）に代入すると次のごとく書きかえられる．

$$Q=C(\sigma\rho_l/G^2l)^{0.043}\pi d^2GH_{fg} \quad \cdots\cdots(6)$$

ここで $\sigma\rho_l/G^2l$ の指数 0.043 はきわめて小さいから，Q は管長 l にほとんど独立に一定となり，Bennett らの領域Ⅱの性格に一致する．なお，式（17）の定数 C に関連し，領域Ⅰと $C=0.25$，領域Ⅱと $C=0.34$ の対応が考えられるかも知れない．次に，Bennett らの領域Ⅲは，その性格から明らかに著者の H 領域に対応する．すなわち，H 領域では，式（18）により限界熱流束は $G^{1/3}$ に比例して増加する一方，出口クオリティ χ_{ex} は $\sigma\rho_l/G^2l$ の減少（G の増加）と共に急激に減少〔前報[2]の図2における式（2）の曲線参照〕する．最後に領域Ⅳは，その性格から N 領域（前報[2]の2章参照）に対応する．

以上，要するに Bennett ら[12]の考えた4領域と，著者の L，H，N 領域の間には相当密接な対応性がある．しかし，ここで少なくとも領域Ⅰ，Ⅱ，Ⅲ，あるいは領域 L，H の区別の生ずる原因が，単に液滴のデポジットの影響の大小といったようなことだけで説明できるものか否か．そうしたことに結論を与え得るほどには，環状流領域の CHF の解析的研究の精度はまだ高くないように思われる．なお〔付録2〕に示すような実験結果もあるので，L 領域と H 領域の CHF の機構の同異については，さらに研究が必要である．

4. 入口サブクール影響因子 K の特性

前報[1]に述べたごとく，従来の CHF の実験結果を検討すると，L，H，HP の各領域では，限界熱流束 q_c と入口サブクールエンタルピ ΔH_i の間に直線関係がある．そして式（1）右辺の比例定数 K の値が純経験的に与えられ，その K の実験値を，無次元量 ρ_g/ρ_l，l/d，$\sigma\rho_l/G^2l$ の関数として整理した結果が前報[1]の図 15～17 である．しかし，この純経験的な整理法に対し，従来から ΔH_i の影響を沸騰長さの考えで処理しようとする考えがある．従って以下には，両

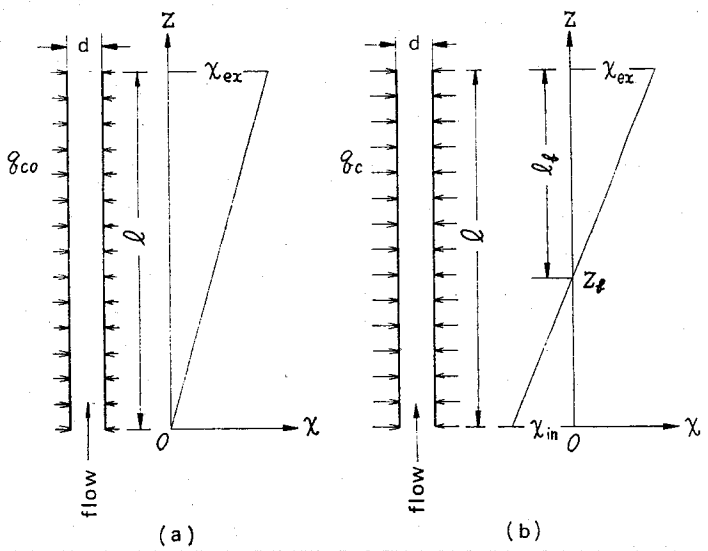


図7　円管内のクオリティ変化

*1 Bennett ら[12]の論文の図7，8のデータ（水，$p=69.0$ bar）のうち，一様加熱円管の場合，$\Delta H_i\to 0$ の限界熱流束 q_{c0} を外そうで求めた結果を図6に示してあるが，式（18）にほぼ一致しているのがみられる．また4領域については文献（13）も参照．

整理法の関係について分析，またそれによって K の特性の検討をおこなう．

4·1 L領域　図7(a)は，飽和液が流入する場合を示し，L 領域では，この円管の限界熱流束 q_{c0} に対し式(17)が成立する．一方，図7(b)は，他の条件は前と同じで，ただ入口クオリティ $\chi_{in}(=-\Delta H_i/H_{fg})$ のサブクール液が流入する場合を示し，この時の限界熱流束 q_c に対し，図7(b)に示す沸騰長さ l_b を用いた式(17)が適用できると仮定すると次式が成立する．

$$\frac{q_c}{GH_{fg}}=C\left(\frac{\sigma\rho_l}{G^2l_b}\right)^{0.043}\frac{1}{l_b/d} \quad \cdots\cdots(7)$$

そして式(7)，式(17)から次の関係を得る．

$$q_c(l_b/l)^{1.043}=q_{c0} \quad \cdots\cdots(8)$$

ところで図7(b)に示す沸騰長さ l_b に対し，ヒートバランスから次式が成立する．

$$\left.\begin{aligned}&\pi d(l-l_b)q_c=\frac{\pi}{4}d^2G\Delta H_i\\&\therefore\quad l_b/l=1-(d/4l)(G\Delta H_i/q_c)\end{aligned}\right\}\cdots\cdots(9)$$

いま，$0<l_b$ の範囲[*2]の l_b に対し，式(7)が成立するものと仮定（この仮定は，2·4 節に述べたところの，L 領域の CHF は上流の影響をあまり受けないという実験事実に対応）する．なお，$0<l_b$ の条件は式(9)から次のごとく書くことができる．

$$(d/4l)(G\Delta H_i/q_c)<1$$

これに式(1)を代入して書き直すと

$$\frac{\Delta H_i/H_{fg}}{1+K\Delta H_i/H_{fg}}<4\frac{q_{c0}}{GH_{fg}}\frac{l}{d} \quad \cdots\cdots(10)$$

さて式(8)，(9)から

$$q_c[1-(d/4l)(G\Delta H_i/q_c)]^{1.043}=q_{c0}$$

ここで左辺の［ ］内を式(10)の条件を考慮して展開，第2項までとり，かつその第2項に含まれる G に式(17)左辺の G を代入すると直ちに

$$q_c=q_{c0}\left\{1+\frac{1.043}{4C(\sigma\rho_l/G^2l)^{0.043}}\frac{\Delta H_i}{H_{fg}}\right\}$$

これを式(1)と比較すれば，L 領域の K（それを K_L と書く）は直ちに次のごとく定まる．

$$K_L=\frac{1.043}{4C(\sigma\rho_l/G^2l)^{0.043}} \quad \cdots\cdots(11)$$

ここに C は式(17)に付記した値をとる．そして図8は，前報[1]の図15に示した K_L の実験データと，式(11)と比較したものである．実験データの数が少なく，かつばらつきがあるが，それでも式(11)とほぼ合うとみなせるようである．

4·2 H 領域　4·1 節と同様に，図7(a)の q_{c0} に対し式(18)が成立，図7(b)の q_c に対し次式が成立する．

$$\frac{q_c}{GH_{fg}}=0.10\left(\frac{\rho_g}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l_b}\right)^{1/3}\times\frac{1}{1+0.0031l_b/d} \quad \cdots\cdots(12)$$

そして式(12)，(18)から次の関係を得る．

$$q_c(l_b/l)^{1/3}(1+0.0031l_b/d)/(1+0.0031l/d)=q_{c0} \quad \cdots\cdots(13)$$

本式に含まれる沸騰長さ l_b に対し式(9)が成立するが，H 領域の CHF には，2·5 節に述べたように，上流の影響が無視できない性格がある．しかし少なくとも $l_b\fallingdotseq l$ の条件，すなわち式(9)から

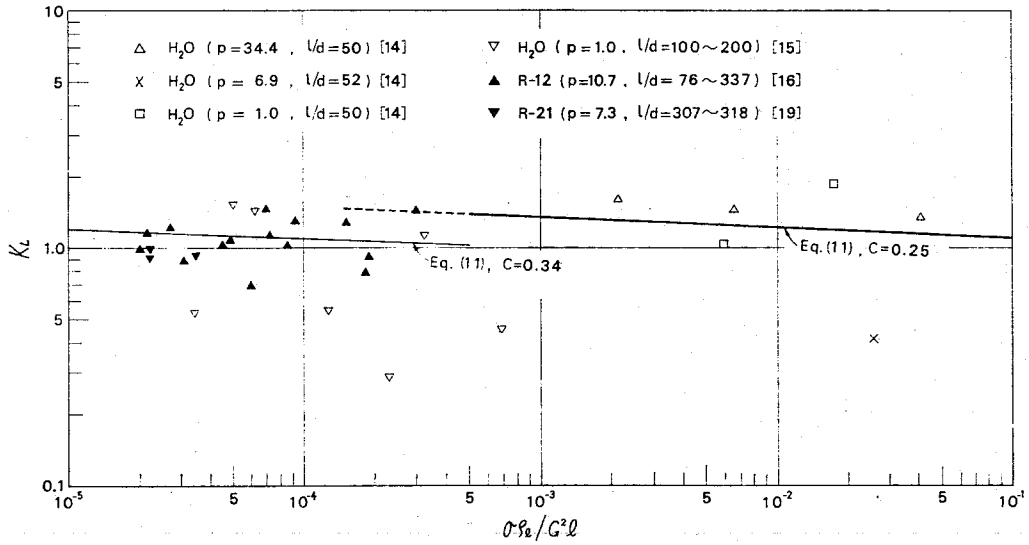


図 8　K_L の実験値と式(11)の比較

*2　図1のように $\chi_{in}>0$，すなわち $\Delta H_i<0$ の場合［この時，式(9)から $l<l_b$］も含めて考える．この場合も，L，H，HP 領域で式(1)が成立することは前報[2]の6章参照．

$$\left.\begin{aligned}&\frac{1}{4}\frac{d}{l}\frac{G|\Delta H_i|}{q_c}\ll 1\\ \therefore\quad &\frac{|\Delta H_i|/H_{fg}}{1+K|\Delta H_i|/H_{fg}}\ll 4\frac{q_{c0}}{GH_{fg}}\frac{l}{d}\end{aligned}\right\}\cdots\cdots(14)$$

の条件下には近似的に式（12）が成立すると仮定してよいであろう．そこで，式（9）の l_b を式（13）左辺の l_b に代入し，式（14）の条件を考慮して展開，第2項までとり，かつ G に式（18）左辺の G を代入すると次式を得ることができる．

$$q_c=q_{c0}\left\{1+\frac{5}{6}\frac{0.0124+d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{1/3}}\frac{\Delta H_i}{H_{fg}}\right\}$$

これを式（1）と比較すれば，H 領域の K（それを K_H と書く）は直ちに次のごとく定まる．

$$K_H=\frac{5}{6}\frac{0.0124+d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{1/3}}\cdots\cdots\cdots\cdots(15)$$

図9は，前報[1]の図 16 にした K_H の実験データと式（15）とを比較したものである．実験データは式（15）とかなりよく一致すると結論してよいであろう．そして，これは q_{c0} に関する式（18）の整理式の妥当性の間接的証明になると同時に，また前報[1]図 16 の整理方法（K_H の実験データを ρ_v/ρ_l, l/d, $\sigma\rho_l/G^2l$ の関数として整理）に相当な妥当性があったことを示

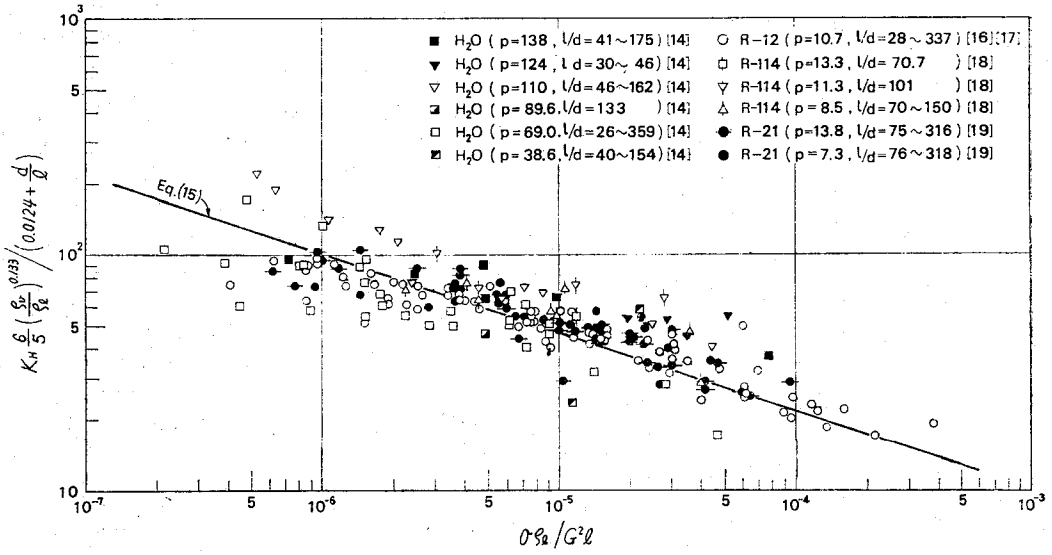


図 9　K_H の実験値と式（15）の比較

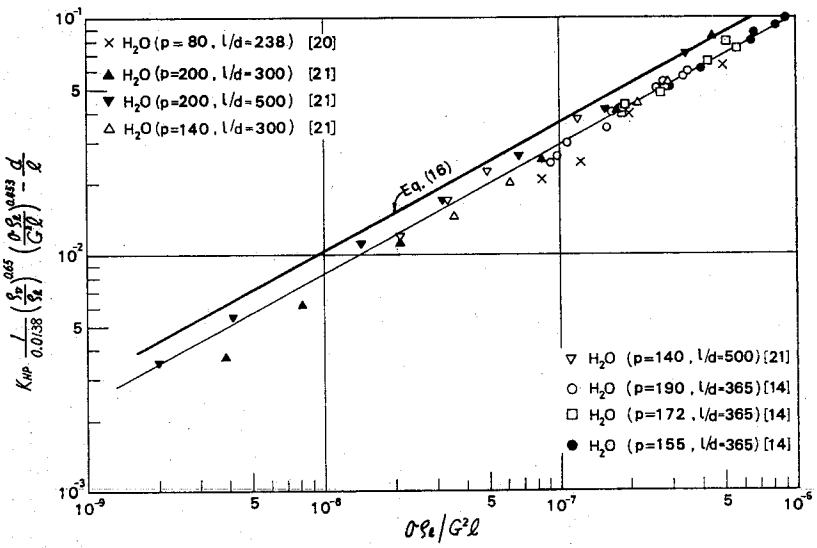


図 10　K_{HP} の実験値と式（16）の比較

するものと解してよいであろう．

4·3 HP 領域　4·2 節（H 領域）とまったく同様にして，式（19）を用い，かつ式（14）の条件下に，HP 領域の K（それを K_{HP} と書く）が次のごとく導かれる．

$$K_{HP}=0.0138\frac{215(\sigma\rho_l/G^2l)^{0.54}+d/l}{(\rho_v/\rho_l)^{0.65}(\sigma\rho_l/G^2l)^{0.453}} \quad \cdots(16)$$

なお，もし式（16）の右辺分子から d/l を取り去ると，

$$K_{HP}=2.97(\sigma\rho_l/G^2l)^{0.087}/(\rho_v/\rho_l)^{0.65}$$

となり，これは前報[1]の K_{HP} の経験整理式の一つ

$$K_{HP}=3.08(\sigma\rho_l/G^2l)^{0.09}/(\rho_v/\rho_l)^{0.6}$$

と非常に近い形である．

ともあれ，図 10 は，前報[1]の図 17 に示した K_{HP} の実験データと式（16）を比較したもので，実験データは式（16）よりわずかに低めに現れる（その理由は現在，不明）ものの，式（16）にかなり近いことを示している．従って前報[1]図 17 の整理方法に妥当性を認めてよいであろう．

4·4 沸騰長さの適用可能範囲　4·1～4·3 節の解析は，L，H，HP の各領域に対し，$\Delta H_i=0$ の場合の限界熱流束 q_{c0} について式（17）～（19）の整理式，また $\Delta H_i\neq 0$ の場合の限界熱流束 q_c について式（1）が成立するという二つの経験的事実に基づいておこなわれている．そして，この二つの経験的事実に抵触しないという条件から沸騰長さ l_b の適用可能範囲を定めることができるのであって，上記の解析結果によれば，L 領域の場合は式（10）のごとく l_b の適用可能範囲が広いが，H および HP 領域では式（14）のごとく適用範囲がきわめて狭いという結論が導かれる．

5. 結　　論

（1）いわゆるドライアウトの研究で得られている CHF の実験データも，著者の CHF 無次元整理式（17），（18）とよく一致する．なお，L 領域の CHF は上流の履歴の影響をあまり受けず，他方 H 領域の CHF はその影響が無視できぬことが分析の結果として観察された．

（2）式（1）右辺に含まれる入口サブクール影響因子 K について，L，H，HP の各領域の無次元整理式を基礎に解析的結果〔式（11），（15），（16）〕を導き，それらが K の実験データにほぼ一致することを示した．これは q_{c0} に関する著者の無次元整理式（17）～（19）の妥当性の間接証明にもなる．

（3）入口サブクールのある状況下の CHF の整理に関し，沸騰長さによる方法の適用可能範囲は，L 領域に対し式（10），H，HP 領域に対し式（14）で与えられる．誤差の定量的計算は容易である．

本研究は，昭和 53 年度科学研究費補助金〔特定研究（1）〕によっておこなわれた．また本報で使用した文献および実験データについて植田辰洋教授（東京大学）から重要な援助を受けた．以上に対し，ここに記して深い謝意を表する．

〔**付録 1**〕　円管の CHF の無次元整理式[1][2]のうち本報で必要なものを記す．まず $\Delta H_i=0$ の CHF に対し

L 領域：

$$\left.\begin{aligned}&\frac{q_{c0}}{GH_{fg}}=C\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d}\\&\left(\frac{\sigma\rho_l}{G^2l}<5\times10^{-4}\text{ で }C=0.34\right)\\&\left(5\times10^{-4}<\frac{\sigma\rho_l}{G^2l}\text{ で }C=0.25\right)\end{aligned}\right\}\cdots\cdots\cdots\cdots(17)$$

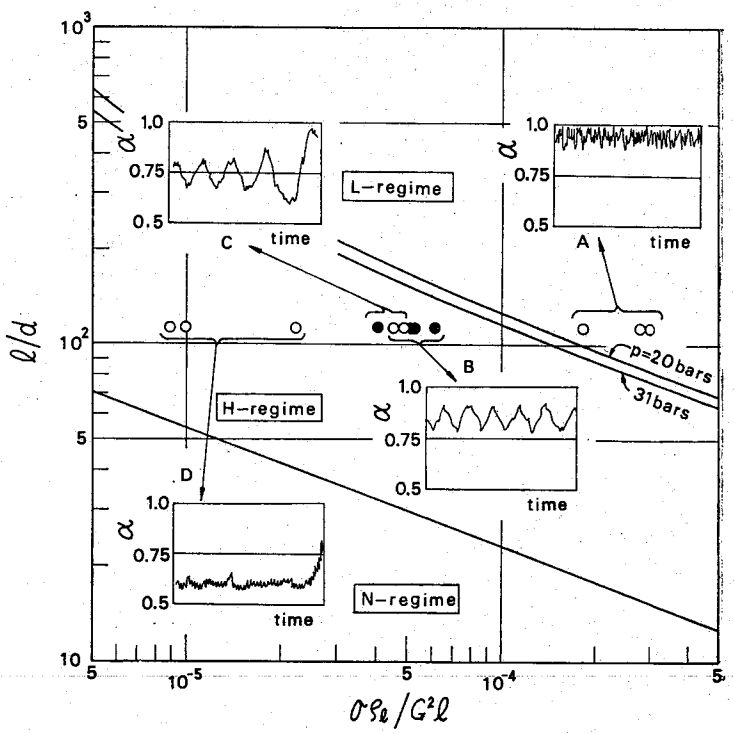


図 11　CHF 直前におけるボイド率 α の変動

H 領域：

$$\frac{q_{c0}}{GH_{fg}}=0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\times\frac{1}{1+0.003\,1\,l/d} \quad \cdots\cdots(18)$$

HP 領域：

$$\frac{q_{c0}}{GH_{fg}}=8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.453}\times\frac{1}{1+107\{\sigma\rho_l/(G^2l)\}^{0.54}l/d} \quad \cdots\cdots(19)$$

次に，L 領域と H 領域の境界，H 領域と N 領域の境界に対し

L-H 境界：

$$\frac{l}{d}\frac{1}{(0.10/C)(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{0.29}-0.003\,1} \quad \cdots\cdots(20)$$

H-N 境界：

$$\frac{l}{d}=\frac{0.77}{(\sigma\rho_l/G^2l)^{0.37}} \quad \cdots\cdots(21)$$

〔付録 2〕　Nevstrueva[22] は，圧力 20～31 bar の水の円管内強制流動沸騰に対し，放射性物質からのベータ線（および X 線）により CHF 発生直前のボイド率の時間変動を測定している．図 11 は，その結果を整理して示すもので，○および●印が実験条件である．そしてボイド率の変動については，L 領域では A（不規則振動），H 領域では B（一定振幅の規則振動），C（振幅増加の規則振動），D（無振動）のごとき結果が報告されている．

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討　論

〔質問〕　**一色尚次**（東京工業大学）

（1） サブクールのある時も何らかの方法で別に l_b がわかれば，q_{c0} の式の l に l_b を入れればそのまま使用できるという結論と考えてよいか．

（2） 最近は l もしくは l_b による q_{CHF} の整理式が多いが，物理的な説明は局所表示の方がしやすいと考えられる．ご意見はいかがか．

（3） 式 (17)～(19) の右辺のウェーバ数の指数（べき数）は l の影響の度合を示すものと考えてよいか．しからば L 領域より H, HP 領域となるほど l（管長履歴）の影響が強いと考えられるが，物理的にどのように説明できるか．

〔回答〕　（1） 入口サブクール ΔH_i がある時の限界熱流束 q_c を求めるには，式（1）を使い，ここに K の値は L 領域で式 (11)，H 領域で式 (15)，HP 領域で式 (16)〔ただし式 (16) よりは前報[1] の経験式 (22) の方が優れる〕を用いるというのが著者の方法の原理である．なお，もし ΔH_i の値が L 領域では式 (10)，H, HP 領域では式 (14) の条件内にある時は，q_{c0} の式の l に l_b を入れれば q_c の値が近似的に求められるというのが本報の趣旨である．そして前者の方法〔式（1）と K〕による q_c に対して，後者の方法〔q_{c0} の式と l_b〕による q_c の近似値が持つ相対誤差（これは ΔH_i の大きさで変化する）については，本報にその計算式を示さなかったが，しかし容易に定量的計算が可能であるということを述べたのが本報の結論（3）である．

（2） 限界熱流束の機構自体が実は明確とはいえな

い現在，この問題についての一般論は不可能にみえる．そこで質問者の論文(付1)(付2)を参照することにすると，質問者は q_c に関し結局のところ次の関係を考えておられる（本報の記号使用）．

$$q_c=f(G, d, \rho_e, \rho_v, \mu_e, \mu_v, \sigma, H_{fg}, \chi_{ex}) \quad \cdots\cdots(\mathrm{i})$$

式（i）右辺 f（ ）内の物理量のうち，管出口の局所値は χ_{ex} だけで，他の量はすべて管入口から出口まで一定とみられる．従って質問者が「局所表示」と言われるのは，q_c-χ_{ex} 関係のことと言える．なお一様加熱円管では次の単純なエネルギつりあい式が常に成立する．

$$\chi_{ex}=\frac{q_c}{GH_{fg}}\frac{4l}{d}-\frac{\Delta H_i}{H_{fg}} \quad \cdots\cdots(\mathrm{ii})$$

従って質問者は，管長 l や入口サブクール ΔH_i は式（ii）を通し χ_{ex} に関係するだけと考えておられることになる．しかし l や ΔH_i が式（ii）以外に，さらに式（i）の右辺 f（ ）内に入ってはいけないという証明が一般的になされてはいない現状において，局所表示の方が物理的な説明がしやすいということを主張しては論理の飛躍にならないだろうか．

（3） 式 (17)～(19) には逆ウェーバ数 $\sigma\rho_l/G^2l$ 以外に l/d の形でも l が含まれている．しかも l/d の影響については，$4(q_{c0}/GH_{fg})(l/d)=\chi_{ex}$ の関係があるということ以外は不明である．従って，このような状況下に，あまり性急な物理的解釈を求めることはつつしんだ方がよいのではないかと考えられる．ただし本報の結論（1）でL領域の CHF について述べた事柄に関連しつつ，次の事柄には多少の意味があるかも知れない．すなわち式 (17) と上記の関係とから L 領域では

$$\chi_{ex}=4C(\sigma\rho_e/G^2l)^{0.043} \quad \cdots\cdots(\mathrm{iii})$$

そして，ごく大まかに言えば式（iii）右辺の値は1に近く，従ってこの意味において $\chi_{ex}\fallingdotseq 1$ が L 領域の CHF 発生条件であるとみることができる．

（付1） 堀・一色，機講論，No. 780-18（昭 53-11），45.
（付2） 堀・一色，機講論，No. 790-6（昭 54-4），71.

536.423.1 :: 532.542

一様加熱垂直円管(飽和液流入)内の限界熱流束の全般的特性の観察*

甲　藤　好　郎**
Yoshiro KATTO

1. 序

一様加熱円管内の限界熱流束(CHF)に関し，最近Shah[1]および著者[2]~[4]によって実験データの無次元整理が試みられた．しかしShahの整理では，無次元パラメータ $Y \leqq 10^5$ の2領域でまったく異なる形式を用いている(なお Y に重力の加速度 g を含む点も問題)．また従来の著者の研究でもCHFの検討が個々の特性領域に分けておこなわれ，ゆえに全般的状況の鳥かん(瞰)ができず不便である．そこで本報では，精度は多少犠牲にしてもCHFの全般的状況の観察を目的に，14種類の純液体に関し47編にのぼる文献から集めた多数の実験データを，著者の整理法をベースにして統一的な形で図示することを工夫した．なお著者の整理では，入口サブクール・エンタルピが ΔH_i の時の限界熱流束 q_c に対し次の形に書かれる(記号は後述の記号表参照)．

$$\left.\begin{aligned} &q_c = q_{co}(1+K\Delta H_i/H_{fg}) \\ &\text{ここに} \\ &q_{co}/GH_{fg} = f(\rho_v/\rho_l, \sigma\rho_l/G^2 l, l/d) \end{aligned}\right\} \cdots\cdots\cdots(1)$$

ところで前報[4]によれば，式(1)の第2式の関数形 $f(\ )$ が定まるとき，多くの場合，式(1)第1式の K の値を解析的に定めることができる．すなわち原理的にいって，q_{co} (すなわち $\Delta H_i=0$ の時の q_c) の整理が最も重要と考えられるので，従って本報では q_{co} の整理についてのみ取扱うものである．

記　号

C：定数 [式(4)]
d：加熱管内径
G：質量速度
H_{fg}：蒸発潜熱
ΔH_i：入口サブクール・エンタルピ
K：ΔH_i の影響係数 [式(1)]
l：加熱管長
q_c：限界熱流束
q_{co}：$\Delta H_i=0$ の時の q_c
ΔT_i：入口サブクール温度
ρ_l：液体密度
ρ_v：蒸気密度
σ：表面張力
χ：クオリティ (χ_{ex}：加熱管出口クオリティ，χ_{in}：加熱管入口クオリティ)．

2. CHF実験データからの q_{co} の集積

2·1 q_{co} の求め方　表1に示した諸文献[5]~[52]から，大部分の q_{co} のデータが下記の方法(i)，(ii)で求められた．また実験条件範囲を広げるために集めた一部のデータは，方法(iii)~(v)によった．

(i) G=一定の条件下，ΔH_i の変化に従う q_c の変化が与えられているデータからは，$\Delta H_i \to 0$ への外そう(時には内そう)により q_{co} が求められる(前報[2]の図4，前報[3]の図1参照)．ΔH_i の代わりに，ΔT_i ないし χ_{in} が用いられている場合も同原理で扱える．なお特殊な場合として，実質上 ΔH_i が零に等しいとみなし得る条件下におこなわれた実験データからは直ちに q_{co} を知ることができる．(ii)限界熱流束 q_c が，出口クオリティ χ_{ex} の関数として与えられているデータの場合は，一様加熱円管に対し一般的に成立する周知のエネルギ・バランス式：

$$4(q_c/GH_{fg})(l/d) - \Delta H_i/H_{fg} = \chi_{ex} \cdots\cdots\cdots(2)$$

を使えば容易に q_c と ΔH_i の関係を知ることができて，その上で(i)の方法が使える．ただしこの場合，加熱管入口で気液二相が混在流入する所の，いわゆる混合流入条件($\Delta H_i<0$)の度合いの強い実験データが与えられている場合があるが，そのようなデータだけから q_{co} を求めることは避けた．

(iii)文献[18][24][35]は，サブクール入口条件($\Delta H_i \geqq 0$)の実験で壁温上昇位置を与えている．この場合，クオリティ $\chi=0$ の位置を計算で求め，その位置から壁温急上昇位置までの距離を l，その時の熱流束を q_{co} とした．(iv)文献[47][48][51]は，混合流入条件($\Delta H_i<0$)で壁温急上昇位置を与える．この時，(iii)と同じ方法で求められる l は仮想的な長さになるが，前報[4]の検討により，L領域(後述3·2節参照)のデー

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* 昭和54年11月22日　第905回講演会において論文講演として講演，原稿受付　昭和54年6月7日．
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タに関する限り，そのlを用いて特に問題ないことが示されている．（v）文献[13][45]は，G, ΔH_i, q_c の一つの組合せごとに実験結果を与える．そこで式（1）第1式によって，前報[4]の与えるKの値を用い q_{co} の値を推定した．この場合，$\Delta H_i/H_{fg}$ の値が大きくなければ，q_{co} の推定に大きな誤差は生じない．

2・2 実験データに関するメモ 文献[7]~[10]のデータは，Collier の書物[59]に引用されているデータである．文献[11]のデータは，Handbook of Heat Transfer の中に Rohsenow[60] によって引用されているデータである．文献[15] に関しては厚肉管のデータを採用した．文献[21]では，l=157 cm のデータは精度が非

表 1 集めた q_{co} データの一覧

文献	流体	l cm	d cm	l/d	$(\rho_v/\rho_l)\times10^2$	$\sigma\rho_l/G^2l$	データ数
Lowdermilk et al.[5]	Water	12.9~48.6	0.129~0.243	100~200	0.062 3	3.42×10^{-5}~6.82×10^{-4}	7
Aladyev et al.[6]	Water	16.0	0.800	20	7.82	3.19×10^{-4}	1
Lee-Obertelli[7]	Water	43.2~366	1.08	40~338	4.81	7.79×10^{-7}~6.60×10^{-6}	5
Matzner[8]	Water	198	0.558~3.75	52~359	4.81	1.45×10^{-6}~1.89×10^{-6}	7
Weatherhead[9]	Water	45.7	0.772	59	13.5	1.26×10^{-6}~1.01×10^{-5}	3
Barnett[10]	Water	178	1.27~3.81	47~140	4.81	3.01×10^{-6}	2
Bergles[11]	Water	1.19~11.4	0.118~0.458	5.0~ 25	1.24	1.36×10^{-5}~5.22×10^{-4}	12
Thompson-Macbeth[12]	Water	15.2~365	0.304~3.74	26~358	0.0623~13.5	2.14×10^{-7}~9.30×10^{-1}	81
Thompson-Macbeth[13]	Water	8.51~312	0.170~1.00	50~792	0.0623~13.5	7.11×10^{-6}~1.00×10^{0}	139
Matzner et al.[14]	Water	244~488	1.02	240~480	4.84	2.90×10^{-8}~2.43×10^{-6}	15
Hewitt et al.[15]	Water	121	0.929	131	0.0736~0.122	4.72×10^{-4}~3.16×10^{-3}	3
Lee[16]	Water	63.5~152	1.40~2.82	42~108	6.51~11.2	4.58×10^{-7}~4.67×10^{-5}	38
Bennett et al.[17]	Water	144	1.25	290	4.81	4.74×10^{-7}~1.89×10^{-6}	2
Bennett et al.[18]	Water	342~416	1.26	271~330	4.81	1.16×10^{-7}~2.38×10^{-5}	36
Babarin et al.[19]	Water	96.0~180	1.20	80~150	0.173	1.33×10^{-4}~1.99×10^{-2}	26
Hewitt[20]	Water	580	0.617	940	4.81	1.33×10^{-7}~2.98×10^{-7}	2
Sterman-Nekrasov[21]	Water	37.0~50.0	0.800	46~62	1.78	1.40×10^{-5}~6.73×10^{-5}	4
Ornatskii et al.[22]	Water	8.70~44.0	0.290~0.800	30~150	1.79~7.82	7.80×10^{-4}~1.66×10^{-3}	8
Midler-Shitsman[23]	Water	321	1.00	321	8.05	7.96×10^{-6}	1
Roko et al.[24]	Water	360	1.25	288	2.53	5.12×10^{-5}	1
Doroschuk et al.[25][26]	Water	150	0.800	188	3.19~27.1	1.50×10^{-7}~1.14×10^{-5}	27
Peskov et al.[27]	Water	165	0.800	206	36.1	2.73×10^{-8}	1
Becker et al.[28]	Water	200~500	1.00	200~500	14.0~34.7	2.78×10^{-9}~6.47×10^{-7}	83
Chojnowski-Wilson[29]	Water	762	3.20	238	24.3	8.43×10^{-8}~3.90×10^{-7}	4
Watson et al.[30]	Water	549	3.78	145	27.1	4.68×10^{-8}~1.17×10^{-6}	6
Noel[31]	Ammonia*	8.00	0.589	14	5.26	8.20×10^{-7}~3.13×10^{-6}	2
Sterman et al.[32]	Benzene	10.0	1.00	10	0.675	1.55×10^{-6}~1.40×10^{-5}	3
Pokhvalov et al.[33]	Benzene	8.00	0.500	16	0.978~14.3	1.08×10^{-7}~2.65×10^{-5}	10
Tolubinskiy-Matorin[34]	Benzene	6.00	0.400	15	2.37	1.10×10^{-5}	1
Tolubinskiy-Matorin[34]	Ethanol	6.00	0.400	15	1.49	1.17×10^{-5}	1
Giarratano et al.[35]	He**	1.09~7.64	0.213	5.1~36	15.1~21.1	2.72×10^{-7}~1.10×10^{-5}	9
Keilin et al.[36]	He**	10.0	0.200	50	17.0	6.04×10^{-5}	1
Ogata-Sato[37]	He**	2.80~5.60	0.109	26~51	15.1~41.3	2.90×10^{-6}~5.00×10^{-5}	10
Lewis et al.[38]	H_2+	28.0~40.6	1.40	20~29	6.34~7.10	3.72×10^{-4}~7.74×10^{-3}	21
Sterman et al.[32]	MIPB++	10.0	1.00	10	1.03	1.23×10^{-6}~1.10×10^{-5}	3
Lewis et al.[38]	N_2	40.6	1.40	29	1.78~1.97	2.16×10^{-3}~2.61×10^{-2}	11
Papell[39]	N_2	30.5	1.28	24	3.00~6.52	1.39×10^{-6}~1.93×10^{-6}	3
Stevens et al.[40]	R-12	24.1~357	0.525~1.61	28~337	4.82	2.37×10^{-7}~3.80×10^{-4}	102
Stevens et al.[41]	R-12	32.4~259	0.965	33~268	4.82	9.52×10^{-7}~6.10×10^{-5}	11
Barnett-Wood[42]	R-12	43.9×241	0.770~0.965	57~313	2.41~4.81	7.91×10^{-7}~1.00×10^{-4}	31
Groeneveld[43]	R-12	138	0.780	177	4.98	8.47×10^{-8}~3.14×10^{-6}	6
Cumo et al.[44]	R-12	200	0.780	256	4.97~9.88	1.96×10^{-6}~1.97×10^{-4}	17
Barnett-Wood[42]	R-21	52.8~331	0.671~1.61	76~318	2.40~4.85	6.17×10^{-7}~9.37×10^{-5}	100
Staub[45]	R-22	152	1.02	150	2.10~4.82	1.99×10^{-6}~3.71×10^{-5}	32
Coffield et al.[46]	R-113	76.2	1.02	75	4.89~13.9	1.15×10^{-7}~1.84×10^{-6}	4
Ueda et al.[47][48]	R-113	254~882	1.00	254~882	1.47	2.20×10^{-6}~6.17×10^{-5}	9
Dix[49]	R-114	91.4~150	0.998~1.41	65~150	4.85~8.46	1.62×10^{-6}~3.97×10^{-5}	19
Hoffman-Krakoviak[50]	K	37.1~183	0.825~2.21	45~83	0.0820	2.67×10^{-3}~5.11×10^{-3}	6
Bond-Converse[51]	K	87.1~612	1.07~1.94	45~569	0.526~0.953	4.42×10^{-5}~2.89×10^{-3}	16
Alad'yev et al.[52]	K	12.0~40.0	0.400~0.600	30~100	0.0270~0.336	1.88×10^{-3}~2.87×10^{-2}	4
Total							946

* Anhydrous ammonia. ** Helium I. + Para-hydrogen. ++ Monoisopropylbiphenyl.

常に悪いので省略した．文献[27]では G＝一定の条件を満たすデータがほとんどなく，q_{co} が定め難く，p＝200 atm の時の1データのみを利用した．文献[35]では簡単のため $\Delta H_i=0$ のデータのみを利用した．

前報[3]の図1に VL 領域として示したごとく，l/d が小さく，かつ入口流速が低い条件下では，CHF に自然対流の影響が強く入り易い．この面から，文献[33]の $\sigma\rho_l/G^2l>3\times10^{-5}$ の範囲のデータ（前報[3]の図 10 参照），および文献[38]の水素のデータの一部は省略した．また同様な理由で，文献[39]では最高流速（3.5 m/s）の上向き流のデータ（下向き流の q_{co} との差は 7.2% 以内にとどまる）だけを用いた．文献[53]のデータは上記の VL 領域に属すると判断されるので採用せず，また文献[54]~[56]のデータは管長 l が不明なので利用できなかった．

表 2　図1で流体区別に用いた記号

文献番号	流　体	記号	文献番号	流　体	記号
(5)~(24)	Water	○	(32)	MIPB	■
(25), (26)	Water	◐	(33), (39)	N_2	■
(27)	water	◪	(40)~(44)	R-12	△
(28)~(30)	Water	●	(42)	R-21	▽
(31)	Ammonia	⊙	(45)	R-22	□
(32)~(34)	Benzene	⊖	(46)~(48)	R-113	▲
(34)	Ethanol	⊕	(49)	R-114	▼
(35)~(37)	He	＋	(50)~(52)	K	×
(38)	H_2	◆			

3.　q_{co} データの統一的図示

3·1　データの図示　前章で集めた q_{co} の全データについて，その液体の種類，実験条件範囲，およびデータ数を表1（データ数の総計は946個）に示す．なおここで特にモノイソプロピルビフェニル（MIPB）の物性値に限り，文献[57][58]の方法で推定した値を用いている．さて，これらの q_{co} のデータが式（1）第2式の形で整理できるものと仮定する．そして図1のごとく，l/d に関し13種類の大きさの図面（q_{co}/GH_{fg} 対 $\sigma\rho_l/G^2l$ の関係を示す）を用意し，表1の全データをプロットした（ρ_v/ρ_l の影響の処理については後述の3·3節参照）．なお図1では，表2の記号を用いて流体の種類を区別している．

3·2　著者の整理式の図示　図1には，前報[2][3]の結果に基づき，下記の諸式により，領域(L, H, N, HP)，整理曲線ⓐ~ⓔ，およびH領域とN領域の境界ⓕをも示してある．

L領域
$$\begin{cases}\text{ⓐ}: q_{co}/GH_{fg}=0.25/(l/d) & (\text{これは } \chi_{ex}=1 \text{ に相当}) \quad\cdots\cdots(3)\\ \text{ⓑ}: q_{co}/GH_{fg}=C(\sigma\rho_l/G^2l)^{0.043}/(l/d) & (C=0.25 \text{ または } 0.34) \quad\cdots\cdots(4)\end{cases}$$

HおよびN領域
$$\begin{cases}\text{ⓒ}: \dfrac{q_{co}}{GH_{fg}}=0.10\left(\dfrac{\rho_v}{\rho_l}\right)^{0.133}\left(\dfrac{\sigma\rho_l}{G^2l}\right)^{1/3}\dfrac{1}{1+0.0031(l/d)}, & \left(\text{ⓒでは } \dfrac{\rho_v}{\rho_l}=0.048\right)\quad\cdots\cdots(5)\\ \text{ⓓ}: \dfrac{q_{co}}{GH_{fg}}=0.098\left(\dfrac{\rho_v}{\rho_l}\right)^{0.133}\left(\dfrac{\sigma\rho_l}{G^2l}\right)^{0.433}\dfrac{(l/d)^{0.27}}{1+0.0031(l/d)}, & \left(\text{ⓓでは } \dfrac{\rho_v}{\rho_l}=0.048\right)\quad\cdots\cdots(6)\end{cases}$$

HP 領域
$$\text{ⓔ}: \frac{q_{co}}{GH_{fg}}=8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.453}\frac{1}{1+107(\sigma\rho_l/G^2l)^{0.54}(l/d)} \quad\cdots\cdots(7)$$

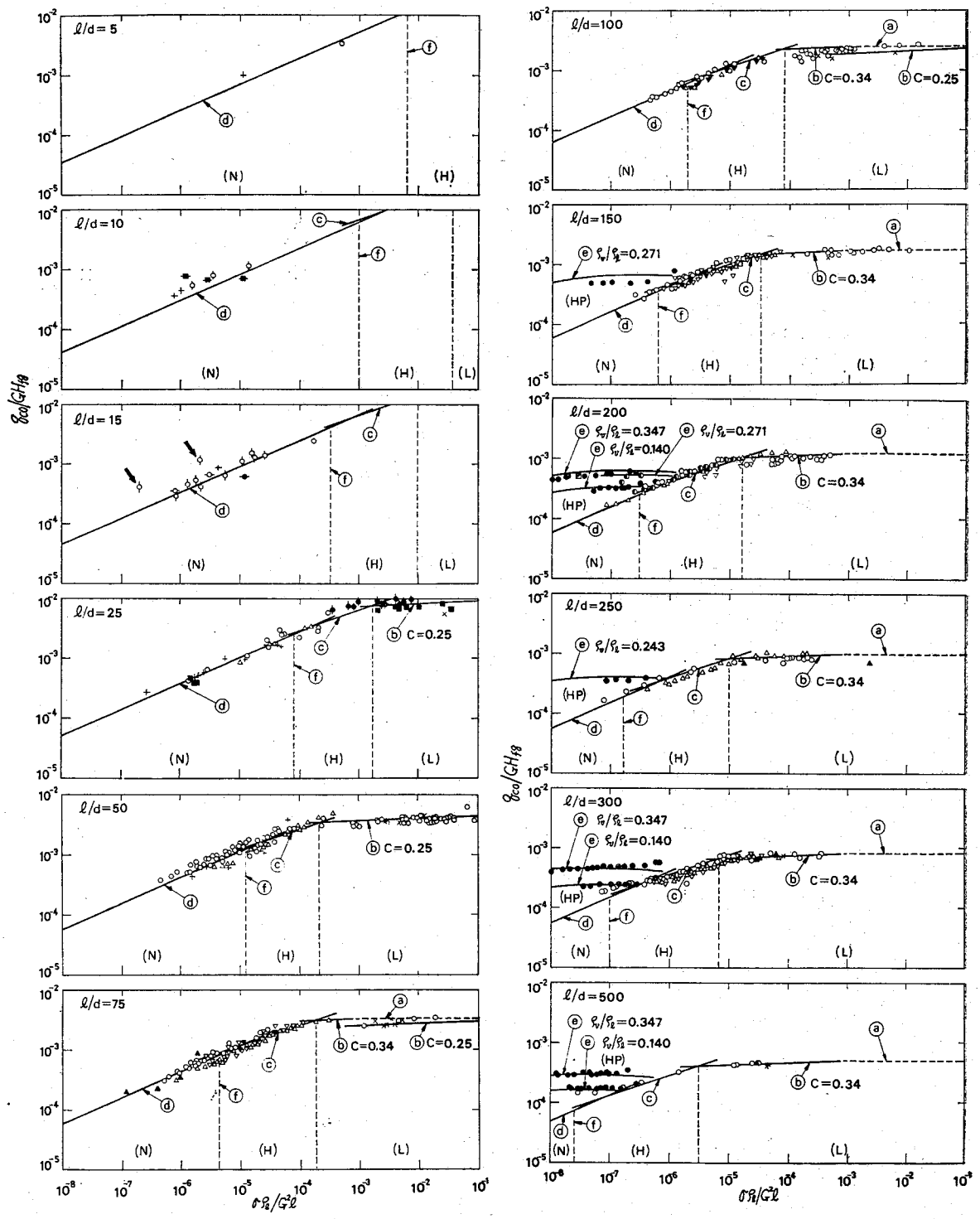
HおよびN領域の境界—ⓕ：
$$\sigma\rho_l/G^2l=[0.77/(l/d)]^{2.70} \quad\cdots\cdots(8)$$

さて図1で，L領域のⓐ，ⓑは，式（3），（4）にみるごとく ρ_v/ρ_l に関係しない．なおⓐとⓑの交点の位置は，式（3），（4）から q_{co} を消去し，そして図1の l/d=75~800 では C=0.34 であるゆえ，次のごとく定まる．
$$\sigma\rho_l/G^2l=7.84\times10^{-4} \quad\cdots\cdots(9)$$

次に，図1でHおよびN領域に示すⓒ，ⓓは式（5），（6）で ρ_v/ρ_l=0.048（これは圧力 68.5 bar の水の密度比）とおいたものを示す．一方，図1で HP 領域のⓔは，式（7）で4種類の ρ_v/ρ_l の値（0.140, 0.243, 0.271, 0.347）にとった結果を示している．

3·3　ρ_v/ρ_l の影響について　前節のごとくして図1上に定めた L，H+N，HP の各領域に対し，全データを分類し，その実験条件を各領域について示した結果が表3である．ただし，H+N 領域にあって ρ_v/ρ_l の値が特に高いか，特に低い特殊な少数のデータのみは表3から除き表4に示してある．

さて表3で，まずL領域についてみると，そのデータの範囲内で ρ_v/ρ_l の値が 500 倍（=13.5÷0.027 0）も



(L)：L領域，(H)：H領域，(N)：N領域，(HP)：HP領域，ⓐ：式（3），ⓑ：式（4），ⓒ：式（5）で $\rho_v/\rho_l=0.048$，ⓓ：式（6）で $\rho_v/\rho_l=0.048$，ⓔ式（7），ⓕ：式（8）．

図 1　q_{co} の実験データの無次元整理（記号に関し表2参照）（その1）

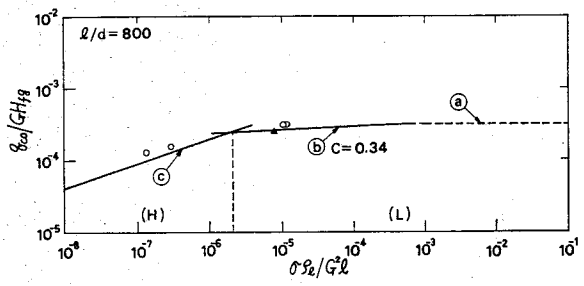


図 1　q_{co} の実験データの無次元整理（記号に関し表2参照）（その2）

変化している．しかし，これらのデータを図1上にプロットした結果では，特に ρ_v/ρ_l による相違はないとみなしてよいようである．そしてこれはL領域の整理式（3），（4）の特性と一致している．

次に表3で，HおよびN領域についてみると，そのデータの ρ_v/ρ_l の実験範囲は，$\rho_v/\rho_l \fallingdotseq 0.048$ の前後に多少変化している．しかしこれらのデータは，図1でほぼⓒ，ⓓの近くに現れており，このことは，整理式（5），（6）で ρ_v/ρ_l の影響が $(\rho_v/\rho_l)^{0.133}$ でしか効かないという事柄に一致すると考えてよいであろう．なおこれらのデータの図示に際し，$(\rho_v/\rho_l)^{0.133}$ による補正をしなかったのは，L領域とH領域の接続部付近のデータの取扱いが，L領域に関し不自然になる恐れからである．ただし表4に示す少数の特殊データについては，H+N 領域のみに属し，かつ ρ_v/ρ_l の実験値が $\rho_v/\rho_l=0.048$ に比べかなり大きく異なっているので，式（5），（6）を参考にして $[0.048/(\rho_v/\rho_l)]^{0.133}$ の補正を加えた q_{co} の値を図1にプロットしている．

最後に表3で，HP 領域についてみると，そのデータは図1にみるごとく ρ_v/ρ_l による影響を強く受けることがわかる．そして図1には，それぞれの実験における ρ_v/ρ_l の値に対応する式（7）を図示してある．

4.　図1の結果の検討

前章で求めた図1の結果に関し，その検討から下記のような事柄が見出される．

（1）　データ源[5]~[52]から求めた各種流体の多数の q_{co} データが，l/d の値の広い範囲にわたり，図1のごとく相当な規則性を示すことは興味あることである（ただし図1，$l/d=15$ で矢印を示す2個のベンゼンのデータが強い偏差を示しているが，前報[3]の図 10 ですでに指摘したことで，原因不明）．そしてこの結果は，一様加熱円管内の強制流動沸騰の CHF に関し無次元数 $\sigma\rho_l/G^2l$（加熱管長 l を含む）が持つ重要性を示すものとみてよいであろう．

（2）　図1の結果からみて，著者の整理式（3）~（7）は，実験データの全般的な傾向をかなり良くあらわしていると結論してよいであろう．

（3）　前報[3]では，式（4）の C の値を暫定的に $5\times10^{-4}<\sigma\rho_l/G^2l$ で $C=0.25$，$\sigma\rho_l/G^2l<5\times10^{-4}$ で $C=0.34$ とした．そしてこれは図1でも，おおまかには成立している．しかし図1の結果によれば，$l/d<50$ で $C=0.25$，$150<l/d$ で $C=0.34$，そして $50<l/d<150$ で C の値が 0.25~0.34 の遷移がおこるとみるのがよいように思われる．

（4）　$\sigma\rho_l/G^2l$ の値が減少した時，N領域と HP 領域のどちらの CHF が生ずるかの判定問題に対し，真

表 3　図1の各領域のデータ（表4のデータを除く）の l/d，ρ_v/ρ_l の実験範囲

公称 l/d	HP-領域		H および N-領域		L-領域		文献番号*
	l/d	$\rho_v/\rho_l\times10^2$	l/d	$\rho_v/\rho_l\times10^2$	l/d	$\rho_v/\rho_l\times10^2$	
10	—	—	10.0	1.03	—	—	32
15	—	—	13.6~16.0	0.979~6.42	—	—	31, 33, 34
25	—	—	20.0—33.4	1.79~7.82	20.0~30.0	0.105~7.10	6, 12, 22, 38, 39, 40, 41, 52
50	—	—	39.2~59.2	1.79~13.5	40.2~60.1	0.0623~13.5	7, 8, 9, 10, 12, 13, 16, 21, 22, 35, 40, 42, 50, 51, 52
75	—	—	67.0~83.5	2.41~11.2	62.5~83.0	0.0820~4.82	7, 8, 12, 16, 19, 21, 40, 41, 42, 46, 49, 50, 52
100	—	—	90.2~112	1.79~11.2	86.8~105	0.0270~2.07	5, 12, 13, 16, 19, 22, 40, 42, 49, 51, 52
150	145	27.1	123~161	1.79~9.28	123~159	0.736~4.85	7, 8, 10, 12, 13, 15, 19, 22, **30**, 40, 41, 42, 45, 49, 51
200	188~206	14.0~36.1	175~214	2.41~13.1	178~224	0.0623~4.82	5, 8, 12, 13, 25, **25**, 26, **26**, **27**, **28**, 40, 42, 43, 51
250	238	24.3	229~268	4.84~9.88	249~268	0.404~9.81	12, 13, 14, 29, **29**, 41, 44, 47, 48
300	300	14.0~34.7	271~360	2.40~8.05	286~337	0.293~4.86	7, 8, 12, 13, 14, 17, 18, 23, 24, **28**, 40, 42, 47, 48, 51
500	500	14.0~34.7	480	4.84	528~569	0.614~2.28	13, 14, **28**, 47, 48, 51
800	—	—	940	4.81	792~882	1.47~2.21	13, 20 47, 48

*　ゴジック番号は，HP-領域のデータを与える文献を示す．

表 4　HおよびN-領域で ρ_v/ρ_l の値が特に高い，または特に低いデータ

公称 l/d	HおよびN-領域		データ数	文献番号
	l/d	$\rho_v/\rho_l\times10^2$		
5	5.0	0.124	2	(11)
	5.14	15.1	1	(35)
10	7.86～10.9	21.1	2	(35)
	10.0	0.675	3	(32)
15	15.0	0.124	1	(11)
	12.5	0.151	1	(35)
25	25.0	0.124	9	(11)
	21.8～31.2	15.1～21.1	4	(35)
	25.7	15.1～41.3	5	(37)
50	35.5	15.1	1	(35)
	50.0	17.0	1	(36)
	51.4	15.1～41.3	5	(37)
200	188	15.4～20.1	7	(25)(26)
Total			42	

表 5　HP-領域の CHF 出現のための ρ_v/ρ_l の下限値

l/d	5	10	15	25	50	75	100
$\rho_v/\rho_l\times10^2$	129*	106*	93.8	78.8	59.0	47.8	40.0

150	200	250	300	500	800
29.8	23.2	18.7	15.4	8.31	4.27

* $\rho_v/\rho_l>1.0$ は HP-領域を欠くことを意味する．

に確定的なことはいまだいえない．しかしいま，HP 領域の㊄線は，図1のH領域の©線にのみ接続して発生すると仮定すると，HP 領域の CHF 発生のための ρ_v/ρ_l の下限値は次のごとく定められる．すなわち，©線と㊄線の交点［式（5）と式（7）から q_{co}/GH_{fg} を消去して定められる］が，©線と㊃線の交点［式（5）と式（6）から q_{co}/GH_{fg} を消去して定められる］に一致する時が ρ_v/ρ_l の下限である．ゆえに次式を得る．

$$\frac{\rho_v}{\rho_l}=\left\{\frac{116(l/d)^{-0.134}+0.976(l/d)^{0.324}}{0.254(l/d)+82}\right\}^{1.93} \quad \cdots\cdots\cdots\cdots(10)$$

そして表5は，式（10）から求めた ρ_v/ρ_l の下限値をの関数として示すが，これは図1における HP 領域の発生状況とおおまかに一致しているといえる．すなわち，表4に見るごとく，l/d=25 および 50 のデータは，0.413 にまでおよぶ大きな値の ρ_v/ρ_l を持っているが，図1の l/d=25, 50 では HP 領域は出現していない．これは表5の指示（l/d=25, 50 で ρ_v/ρ_l の下限値は0.413より大）に一致する．他方，表3には，図1の l/d=150～500 に HP 領域としてあらわれたデータの ρ_v/ρ_l の値が示されているが，これらの ρ_v/ρ_l の値は表5の ρ_v/ρ_l の下限値より大体において大きな値になっている．

（5）　図1の l/d=50～300 の図面で，HおよびN領域では，統計的にフレオン系の q_{co} データは水のそれより多少低くあらわれている．これはすでに前報[3]で，Ahmad[61] の流体のモデリングの結果との比較において指摘（領域は H, N と推定）したことと一致している．また図1の $l/d\geqq300$ でH領域において実験データは©線よりこう配が少しゆるい傾向を持っているように見える．

（6）　カリウムの実験データはL領域のみに存在している．L領域以外の領域での液体金属の CHF の挙動については今後の研究にまつべきであろう．また HP 領域の実験データは水だけである．従って HP 領域の無次元整理を確実にするため，水以外の流体の実験が必要であろう．

5. 結　　論

47 編にのぼる多数の文献から，14 種類の流体の q_{co} に関し計 946 個のデータを集め，それらの統一的な図式表示をおこない，一様加熱円管内の強制流動沸騰の CHF 現象に相当な規則性があることを示した．なお同時に，これらのデータの示す諸特性が，著者の整理式（3）～（7）によってかなりよくはあくし得ること，またその整理式の構成上，前報[2][3]ではまだ不明であった二三の事柄［式（4）の定数 C の値，および HP 領域出現のための ρ_v/ρ_l 下限値］についての検討結果を示した．

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討　　論

〔質問〕　**青木成文**

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（1） 貴論文の目的に照らして「精度を多少犠牲にしても」ということは，CHF 整理の上で，どの程度までの精度の悪化を考慮されているのか，定量的数値があれば伺いたい．

（2） 2·1 節（iii）で原論文の q_c から必要な q_{co} を求める際，l を決定した後で式（1）第1式を用い，ΔH_i の q_c への影響を除去して q_{co} を求められたのであろうか．

（3） H+N 領域の ρ_v/ρ_l についての考察の中で，ρ_v/ρ_l 値の特殊なデータをあえて分離して図1のように整理されたが，このデータ分離にあたりその物理的意味または判別条件についてご教示願いたい．

〔回答〕　（1） H，N領域では，著者の整理式（5），（6）からも推察されるように，q_{co}/GH_{fg} の実験値を図1の形に図示する時，ρ_v/ρ_l の弱い影響が残る可能性がある．しかし恐らく $(\rho_v/\rho_l)^{0.133}$ の程度の影響にすぎないから，ある範囲内の ρ_v/ρ_l の相違については，いろいろその相違を考慮せず図1のように図示したことが精度を犠牲にしたことの一つである．具体的には表3のH+N領域の実験値がそれに当たり，従って図1でⓒ，ⓓの直線を規準に考える時，表3のH+N領域の ρ_v/ρ_l の実験範囲に対し $[(\rho_v/\rho_l)/0.048]^{0.133}=1.14\sim0.80$ 程度の分散がⓒ，ⓓの上下に生じていることになる．なおL領域，HP 領域の図示にあたって

は上記の問題は存在しないようになっている．また，もう一つ精度を犠牲にした事柄は，いうまでもなく図1が l/d のとびとびの値に分割して実験値を図示していることである．これについては，表3，4において l/d の公称値と実験値の間の差を見て頂き，式(3)～(7)で l/d の相違がどの程度の影響を持つかを検討して頂ければ分散の度合いがわかるはずである．それほど大きな分散にはなっていないとおもう．

(2) ここでは熱流束に関し何らの操作はせず，実験で求められた限界熱流束 q_c の値が，そのまま q_{co} と見なし得る（少なくとも近似的に）状況になっているわけである．

(3) 「あえて分離」というようにおおげさなことではなく，単に上記(1)項の問題に関連した事柄にすぎない．すなわち表4に掲げた実験値は ρ_v/ρ_l の値が0.048に比べて，かなり大きかったり小さかったりしている．(1)項と同じ計算をすると

$$[(\rho_v/\rho_l)/0.048]^{0.133}=1.33\sim0.62$$

程度の分散となり，従って ρ_v/ρ_l の相違を考えないで図示すると少し分散が過大になる．そこで本文に記したように $[0.048/(\rho_v/\rho_l)]^{0.133}$ を乗じ補正した q_{co} を図1に図示しただけのことである．なお，それならば表3，4のH，N領域の全データに $(\rho_v/\rho_l)^{0.133}$ の補正をして図示すればとのご意見が出るかも知れないが，もともと $(\rho_v/\rho_l)^{0.133}$ は著者の整理式のものにすぎず，そうした人為的操作をなるべく加えず自然のままのデータから傾向を鳥かんしようとしたのが本報の主旨である．従って表4の少数のデータのみに少しの補正をするに止めることにしたわけである．

〔質問〕 一色尚次（東京工業大学）

CHF に関しさらに精密な整理をされたことに敬意を表す．

(1) LからH領域へ移行するのは物理的にどのような状況で生ずるのかのご意見を伺いたい．

(2) 上記の移行点は水とフロン等有機流体との間で若干の相違はないか．

(3) HP 領域は再びL領域の（a）（b）曲線に平行になってくるようであるが，両者に何らかの物理的共通性は考えられるか．

〔回答〕 (1) 著者の研究は CHF の経験的整理の段階にあり，ゆえにこの問題に対し科学的な回答は不可能である．ただ単なる推測として申しあげることを許して頂けるなら，LからH領域へ移行に際し多分フローパターンは環状流のまま変わらないであろう．しかし管壁上の薄い液膜の厚さ δ が $\delta=0$ になって CHF を生ずる（いわゆるドライアウト）のがL領域，これに対し有限の厚さの液膜（$\delta>0$）が破断して CHF を生ずるのがH領域というようなことがあるかも知れない．

(2) 本文4章(5)項に述べたようなことは存在する．それ以上の詳細は現段階では不明である．

(3) 式(4)，(7)，あるいは図1にみるように，L 領域では ρ_v/ρ_l が影響しないのに，HP 領域では ρ_v/ρ_l の影響が強いといった事情もあるので，この面からみれば両領域は互いに異質だとする見方も成立しそうである．

〔質問〕 西川兼康・伊藤猛宏（九州大学工学部）

第4章の(4)において，与えられた l/d と $\sigma\rho_l/G^2l$ の値に対してNも HP も可能であると考えられる場合，ρ_v/ρ_l の値が式(10)の値より大きいことは HP が出現する十分条件と考えるのか，あるいは必要条件と考えるのであろうか．また ρ_v/ρ_l の上限は原理的に存在しないと考えてよいか．

〔回答〕 第4章(4)において著者が暫定的に仮定した判定法では，ρ_v/ρ_l の値が式(10)の値より大であれば $\sigma\rho_l/G^2l$ の減少と共に HP 領域が出現（ⓒ線とⓓ線の交点より上方のⓒ線の途中からⓔ線が分枝できる），ρ_v/ρ_l の値が式(10)の値より小であれば $\sigma\rho_l/G_l^2$ の減少と共にN領域に入る（ⓒ線とⓓ線の交点から下方のⓓ線に沿って変化する）ことになっているわけである．そして上記において，ρ_v/ρ_l の値が式(10)の値より大である時に HP 領域が出現するということを事項A，またⓒ線とⓓ線の交点より上方のⓒ線と交点を持つⓔ線を HP 領域と考えることを事項Bとする時，論理的に言えば事項A，Bは単に互いに等しい関係にある．従ってまた，このような時は十分条件とか必要条件という言い方はしないわけである．

次に ρ_v/ρ_l の上限が原理的に存在するか，あるいは存在しないかは，現状では不明という他ない．本報の結果は絶経験的なものであって原理的なものではないからである．つまり現存する実験データの範囲内で第5番めの特性領域が現れていないので，第5領域を考えない取扱いをしている次第であり，これは熱力学第1，第2法則などの経験法則（例として大げさであるが）の置かれる状況と似ていると申せるかも知れない．ただ，しかし次の問題はある．すなわち文献(2)の図14からもすでに予測されるように ρ_v/ρ_l が非常に大になると，著者の整理式による限り，HP 領域とL領域が重なり合う部分を生じ始める．この辺でどのようなことになるのかは今後の研究を待つ他ない．それにしても，表3の HP 領域に現れている実験条件で ρ_v/ρ_l

の最高値0.361は，水の場合でいえば圧力 206 kg/cm² に相当する．水の臨界圧は約 225 kg/cm² ですから，圧力の面だけから言えば未知の領域はそれほど広くはないと言えるであろう．

〔質問〕　西川兼康・伊藤猛宏

質問の前半については，質問の真意が通じていないようですから，再度質問する．

NもHPも可能であるような $\sigma\rho_l/G^2l$ の領域でHPが生ずることを事項Aとし，ρ_v/ρ_l が式（10）の値より大きいことを事項Bとしよう．このBはAに対して十分条件であるか，必要条件があるか，という意味である．

〔回答〕　ご質問の趣旨は，本報第4章（4）で著者が暫定的に仮定した判定法に範囲を限るとしてよいと思う．そしてNもHPも可能であるような l/d, $\sigma\rho_l/G^2l$ の領域にあることを前提とする．この時，HPが生ずることを事項A，ρ_v/ρ_l が式（10）の値より大きいことを事項Bとする．なお必要，十分条件については数学で周知の定義（文献(付1)など参照）によるものとする．

さて本報では，AならばBが成立（実験データからみれば近似的な話であるが，いまはそのことは問題外）することを見出した形になっている．従ってBはAの必要条件である．他方，著者の判定法，ならびに上記の前提条件下には，B以外の問題は存在し得ないので，BならばAが成立，すなわちAはBの必要条件になる．ゆえに以上を総合すれば，BはAに対し必要かつ十分な条件である．なお，このように互いに関連するA，B以外に他の事項が存在しないことが明らかな時は，必要とか十分とかを区別する要はなく，通常は面倒な言い方をしないというのが著者の前回の回答（舌足らずだったことを申し訳なくおもう）の実は趣旨であった．

ところで，もう少し問題を広げて，上記の前提条件内にある「HPが可能」ということまで考えるならば，一応そのための条件をいくつか考えることが必要になる．第一にはHP領域の ρ_v/ρ_l の上限問題がある．しかし本報の性格（前回の回答も参照）からいっても，図1にみられるようなHPの経験的出現範囲を考えれば十分で，しかも現在の問題はNとHPの分離の問題であるから ρ_v/ρ_l の上限値まで考える要は必ずしもなく，従って本問題は常識的に対象外としてよいであろう．また第二の問題としては，図1のH領域に有限長さの©線が存在することが必要（さもないとⓔ線の起点がなくなる）になる．しかし，この問題も現実に図1の範囲内で十分満たされているから対象外としてよいであろう．さらに第三には，ρ_v/ρ_l の上昇と共に図1でⓔ，©両線の交点が上方へ必ず移動するか否かの問題がある．しかし式（5）と式（7）に対する ρ_v/ρ_l の影響度の大小などを考えると，この問題も満足されることが明らかで，考慮の対象外として差し支えない．従って上記のような諸問題は特に云々する要はなく，またその意味でBはAに対し必要かつ十分な条件ということが依然として成立するといえるであろう．

要するにBは，HPの発生判定のため必要として与えた条件であり，かつ当面の目的にはそれで十分である．

〔質問〕　老固潔一
（川崎重工業会社技術研究所）

垂直円管内の限界熱流束についてご提案された無次元表示式の適用性が広範囲の実験値により検証されており，設計への適用に対しても非常に有益なものになると考える．ついては設計への適用という観点から以下の点についてお教えいただければ幸いである．

（1）　表示式の精度について

貴論文の前報〔参考文献(2)，174ページ〕ではその時点までに検証された実験値の精度が図示されているが，今回の論文で明確にされた点（定数Cのとりかた）も考慮し，今回追加された多数の実験値を含めての精度をお教え下さい（出来れば各領域について）．

（2）　不均一加熱への適用について

標題で示されているように均一加熱が貴論文の対象ですが，不均一加熱の限界熱流束の算定にご提案された表示式を拡張または適用することに関してご意見をおうかがいしたい．

〔回答〕　（1）　本報で扱った実験値を，文献(2)でやったと同じように著者の整理式と機械的に比較してみることは難しいことではない．しかし本文第4章（5）に述べたような事柄も含めて，せっかく図1を示してあるので，まずこれをご利用頂けないか．以下のように見て頂ければ，精度の大きさをほぼはあくして頂けると思う．

図1の表示上，実験データに生ずる分散の原因として l/d および $\rho_v\rho_l$ の影響がある．まず l/d の影響の大きさについては，青木氏への回答（1）を参照して頂きたい．また ρ_v/ρ_l については各領域に対し次のような状況にある．

（i）　L領域：整理式（3），（4）には ρ_v/ρ_l の影響が含まれないので，図1のⓐ，ⓑ線と実験データの

（付1）　平凡社，世界大百科事典，**14** (1959), 160; **24** (1959), 180.

比較から直ちに精度（前記の l/d の影響を含むが）がわかる．

（ii） HP 領域：図1では ρ_v/ρ_l の値が共通な実験データとⓒ線を図示してあるので，これも直ちに精度を見ることができる．なお，ここで $l/d=150$ の図に示す Watson ら[30]のデータは整理式（7）からの偏りが大である．その理由は明白ではないが，加熱管内径 $d=3.78$cm が他の実験に比べ少し大きいものであることを付記しておく．

（iii） H，N領域：青木氏への回答（1）に記すごとく，表3の実験データについて

$$[(\rho_v/\rho_l)/0.048]^{0.133}=1.14\sim0.80$$

程度の分散がある．しかし，すでにご承知のように，ヨーロッパの8研究所で実験比較されたミラノ再現性試験データのように，特に信頼性のあるデータでも，偏りの大きなものを除いた正常データのみの比較においてすら +19～−14% の最大偏差が存在してる（文献(付1)参照）．従って，それを考えると，少し荒っぽい話になるが，図1のⓒ，ⓓ線と実験データをそのまま比較し，それをもって整理式の精度をご判断頂いても大きな誤りはないように思う．ただしこの際，設計への適用という観点からは，本文第4章（5）および文献(4)の結論（4）に記した事柄にご留意をお願いする．そして，ひとつひとつ特定の流体ごとに図1の整理結果を見て頂いた方が設計の場合にはよいように思う．

（2） この問題に関心は持っているが，著者自身にはこれを検討する時間的余裕が得られない状況にあり，誰か検討して頂けないだろうかと思ったりする．ただ原理的な面からだけ言えば，均一加熱と不均一加熱とは違うものであるから，lacal conditions hypothesis のようなものが確実に成立でもしない限り，一般の不均一加熱に対し本整理式を拡張することは難かしさがあるような気もする．少なくとも，さらに近似的な性格を加えることになることだけは確かであろう．

（付1） Marinelli, V., *Nucl. Technol*, 34-7 (1977-7), 135.

536.423.1::532.542

一様加熱垂直円管内の強制流動沸騰の限界熱流束*

(高圧領域の研究と CHF 特性領域線図)

甲　藤　好　郎**
Yoshiro KATTO

1. 序

著者は最近[1]，一様加熱垂直円管内の強制流動沸騰の限界熱流束 (CHF) について，その実験データの整理式を提出した．すなわち，入口サブクールエンタルピが ΔH_i の場合の限界熱流束 q_c に対し

$$q_c=q_{c0}\{1+K(\Delta H_i/H_{fg})\} \quad \cdots\cdots(1)$$

ここに，q_{c0} は $\Delta H_i=0$ の時の q_c であり，また H_{fg} は蒸発潜熱で，従って係数 K は無次元である．そしてL領域(いわゆるドライアウトの近傍領域)，N領域(いわゆる DNB の近傍領域)，H 領域 (L, N の中間領域)，および HP 領域 (高圧下に N 領域の代わりに出現する領域) の4個の特性領域を区別し，まず q_{c0} に関して，各領域ごとに次の形式の無次元整理式 (使用記号については後述の記号表参照) を導いた．

$$q_{c0}/GH_{fg}=f(\rho_v/\rho_l, \sigma\rho_l/G^2l, l/d) \quad \cdots\cdots(2)$$

この整理式の特徴は，加熱管長 l を含む逆ウェーバ数 $\sigma\rho_l/G^2l$ を使用している点にある．そしてその後の研究[2]~[4]において，著者は上記の q_{c0} の整理式を既存の多くの実験データとさらに比較し，その結果，L 領域内の q_{c0} を整理式の適用方法に関し少しの修正をほどこす必要性を見いだした以外は，初報[1]に与えた無次元整理式のままで実験データの特性を一般的にかなり良く示し得ることを知った．

他方，$\Delta H_i>0$ の時，q_c-ΔH_i の直線関係が多くの場合，成立することが知られている (例えば Collier[5] 参照) が，この場合，式 (1) 右辺の係数 K の値を上記直線関係のこう配から容易に求めることができる．そして著者の初報[1]では，N 領域を除く残りの 3 領域 (L, H, HP) において q_c-ΔH_i 直線関係が成立することを示し，かつそれら 3 領域の K の実験データに対してそれぞれ経験的な無次元整理式を導いた．しかし，その後の研究[3]において，沸騰長さの概念を使えば，q_{c0} に関する整理式 (2) から解析的に K の整理式が導けること，ただし HP 領域に限り，この K の整理式と実験データの間にわずかながら規則的な偏差が存在することを見いだした．

さて上述したように，式 (2) を基礎にして解析的に求めた K の整理式が，HP 領域において K の実験結果に対して規則的な偏差を示すことは緊急に解決を要する問題といえるであろう．また同時に HP 領域については著者の前報[4]が与える次の問題がある．すなわち，前報[4]で著者は HP 領域発生の境界条件式を導いたが，それは実験結果とだいたいの傾向の一致を示すものの，なお定量的に満足できない点があった．そこで本報では，これら HP 領域に関し残された問題を検討解決し，またその結果に基づき，これまであいまいさを残していた CHF 特性領域線図を定めることを目的におこなった研究について報告する．

記　号

C : 定数〔式 (4)〕
d : 加熱管内径
G : 質量速度
H_{fg} : 蒸発潜熱
ΔH_i : 入口サブクールエンタルピ
K : ΔH_i の影響係数〔式 (1)〕
l : 加熱管長
p : 圧力　bar
q_c : 限界熱流束
q_{c0} : $\Delta H_i=0$ の時の q_c
ρ_l : 液体密度
ρ_v : 蒸気密度
σ : 表面張力

2. 前報までに得られた整理式の総括

これまでの著者の研究[1]~[4]の結果を考慮すると，CHF 整理式は以下のように要約される．

2・1 q_{c0} の整理式　整理式の全体傾向を示すため，まず図1に $l/d=50, 200$ の場合の例を掲げ，それぞれに対し $\rho_v/\rho_l=0.017\,87$ および 0.3216 (水の場

* 昭和 55 年 3 月 18 日　九州支部第 33 期総会講演会において論文講演として講演，原稿受付　昭和 54 年 10 月 22 日．
** 正員，東京大学工学部 (〒113 東京都文京区本郷 7-3-1)．

合には圧力 p=29.5 bar, 196 bar に対応する）に関する整理式を掲げておく．そして図1に図示されている諸式は下記のものである．

L 領域：

$$q_{c0}/GH_{fg}=0.25/(l/d) \quad (l/d>150 \text{ に対して}) \cdots\cdots(3)$$

$$q_{c0}/GH_{fg}=C(\sigma p_l/G^2l)^{0.043}/(l/d)$$

ここに，

$l/d<50$ の時 $C=0.25$

$l/d>150$ の時 $C=0.34$

$l/d=50\sim150$ で C は遷移 $\cdots\cdots(4)$

H 領域：

$$\frac{q_{c0}}{GH_{fg}}=0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\frac{1}{1+0.003\,1l/d} \cdots\cdots(5)$$

$$\frac{q_{c0}}{GH_{fg}}=0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}\times\frac{(l/d)^{0.27}}{1+0.003\,1l/d} \cdots\cdots(6)$$

N 領域：

$$\frac{q_{c0}}{GH_{fg}}=0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}\times\frac{(l/d)^{0.27}}{1+0.003\,1l/d} \cdots\cdots(6)$$

本式は H 領域の式 (6) と同じである（図1の[illegible]N, H 領域参照）．ただし H 領域と N 領域の境界は次式（これは初報[1]で q_c-ΔH_i の直線関係の成立する領域と，q_c-ΔH_i の非直線関係の成立する領域の境界として決定されたもの）で定められる．

$$\sigma\rho_l/G^2l=\{0.77/(l/d)\}^{2.70}$$

または

$$l/d=0.77/(\sigma\rho_l/G^2l)^{0.37} \cdots\cdots(7)$$

HP 領域：

$$\frac{q_{c0}}{GH_{fg}}=8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.453}+\frac{1}{1+107(\sigma\rho_l/G^2l)^{0.54}l/d} \cdots\cdots(8)$$

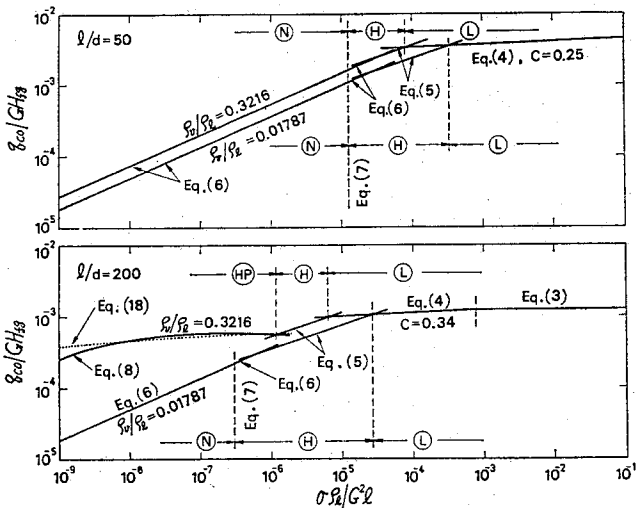

図 1 q_{c0} の無次元整理結果の一例

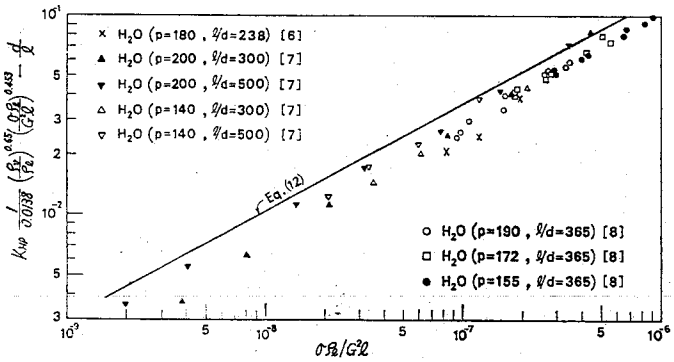

図 2 K_{HP} に関する式 (12) と実験データの比較

2・2 K の整理式 著者の前報[3]の方法に従い，沸騰長さの概念を用いることによって，q_{c0} の無次元整理式〔式 (4)～(6), (8)〕から解析的に求められる K の整理式は次のごとくである．

L 領域：式 (4) から

$$K_L=\frac{1.043}{4C(\sigma\rho_l/G^2l)^{0.043}} \cdots\cdots(9)$$

H 領域：式 (5) から

$$K_H=\frac{5}{6}\frac{0.0124+d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{1/3}} \cdots\cdots(10)$$

式 (6) から

$$K_H=0.416\frac{(0.0221+d/l)(d/l)^{0.27}}{(\rho_v/\rho_l)^{0.113}(\sigma\rho_l/G^2l)^{0.433}} \cdots\cdots(11)$$

HP 領域：式 (8) から

$$K_{HP}=0.0138\times\frac{215(\sigma\rho_l/G^2l)^{0.54}+d/l}{(\rho_v/\rho_l)^{0.65}(\sigma\rho_l/G^2l)^{0.453}} \cdots\cdots(12)$$

3. HP 領域における問題点の詳細

3・1 第一の問題点 図2は，HP 領域における ΔH_i の影響係数 K_{HP} に関して，2・2 節に掲げた解析的な K_{HP} の式 (12) を，初報[1]で集めた K_{HP} の実験値（Chojnowski ら[6]，Becker ら[7]の求めたもの，および Thompson ら[8]のデータ集積表 11～

14 に掲げられた l/d=365 のデータから得た K_{HP}) と比較したものである．そして本図から明らかなように，式 (12) と実験値の間には規則的な偏差が見られる．しかし，式 (12) の導出の基礎となる q_{c0} の無次元整理式 (8) は，初報[1]の図 12 に見られるように，q_{c0} の実験データとかなりよい一致を示すものである．そこで前報[3]の段階では，図 2 にみられる偏差の原因について不明と考えざるを得なかった．

3・2 第二の問題点 L 領域と H 領域の境界については，前報[3]で示したように，式 (4)，(5) から q_{c0}/GH_{fg} を消去して次式が得られる．

$$\frac{l}{d}=\frac{1}{(0.10/C)(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{0.29}-0.0031} \quad \cdots\cdots(13)$$

また H 領域の中で，式 (5)，(6) の各適用範囲の境界については，初報[1]でおこなったように両式から q_{c0}/GH_{fg} を消去して次式が与えられる．

$$\frac{l}{d}=\frac{1.077}{(\sigma\rho_l/G^2l)^{0.37}} \quad \cdots\cdots(14)$$

そして H 領域と N 領域の境界は式 (7) で与えられ，また H 領域と HP 領域の境界については，初報[1]でおこなったように式 (5)，(8) から q_{c0}/GH_{fg} を消去して次式が定められる．

$$\frac{l}{d}=\frac{82(\rho_v/\rho_l)^{0.517}-(\sigma\rho_l/G^2l)^{-0.12}}{107(\sigma\rho_l/G^2l)^{0.42}-0.254(\rho_v/\rho_l)^{0.517}} \quad \cdots\cdots(15)$$

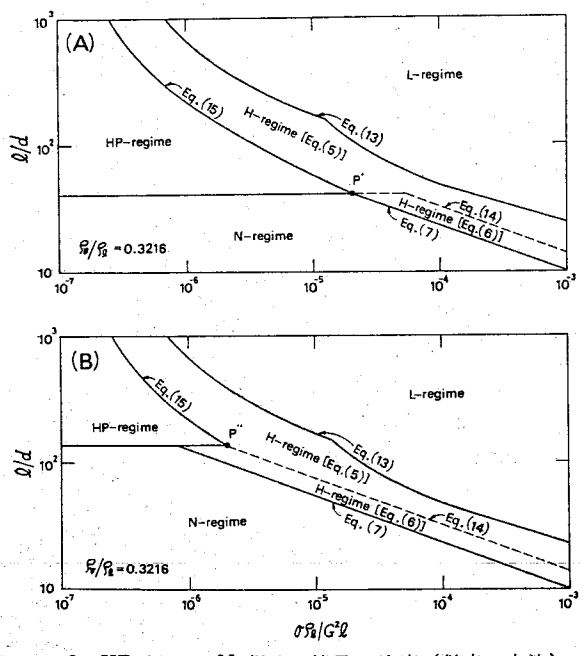

図 3 HP 領域と N 領域の境界の決定（従来の方法）

そして図 3 は，ρ_v/ρ_l=0.3216 の場合を例にとり，式 (13)～(15) を用いて作成した CHF 特性領域線図を示すものである．ここに式 (13) に含まれる C の値については，式 (4) に従い，なお l/d=50～150 の間では 0.25 から 0.34 まで l/d と共に直線的に変化するとした．

さて，まず図 3 (A) では，式 (15) と式 (7) の交点 P′ を通る水平線によって HP 領域と N 領域が分割されるとしているが，これは初報[1]で採用した考えにひとしい．そして，この点 P′ における ρ_v/ρ_l と l/d の関係は，式 (15)，(7) から $\sigma\rho_l/G^2l$ を消去して次式で与えられる．

$$\frac{\rho_v}{\rho_l}=\left\{\frac{79.5(l/d)^{-0.134}+1.08(l/d)^{0.324}}{0.254(l/d)+82}\right\}^{1.93} \quad \cdots\cdots(16)$$

本式は，与えられた l/d の条件に対し，HP 領域の発生可能な ρ_v/ρ_l の下限値を与えることになる．

ところで表 1 は，CHF に関する既存の実験データの中から，HP 領域の CHF の発生をみていると判断されるものを選び，その実験条件（管内径 d，管長径比 l/d，密度比 ρ_v/ρ_l，および HP 領域の CHF データの $\sigma\rho_l/G^2l$ の範囲）を示したものである．なお表 1 の中で Campolunghi ら[12]のデータでは管長 l の正確な値がわからず，従って表 1 の l/d の値は近似値である．そして図 4 は，表 1 に掲げた HP 領域発生に関する ρ_v/ρ_l と l/d の関係を図示するが，これらに対し式 (16) による HP 領域発生の境界条件は妥当な性格を示しているといえる．しかしながら，同じく図 4 に示すヘリウム I の CHF の 2 実験データ（尾形ら[13][14]による）は，前報[4]で検討した結果によると共に N 領域に属するもので HP 領域にはないと結論される．従って，このヘリウム I の実験データに対し式 (16) は妥当性を欠くと言わざるを得ない．なお上記のヘリウムのデータは共に ρ_v/ρ_l=0.413 で，かつそれぞれ次の条件下のものである．

l/d=25.7, $\sigma\rho_l/G^2l=5.70\times10^{-6}$

l/d=51.4, $\sigma\rho_l/G^2l=2.90\times10^{-6}$

さて，そこで次に図 3 (B) では，式 (14) を利用する点で物理的にやや不自然さを持つが，仮に式 (15)，(14) の交点 P″ を考えることとし，この点 P″ を通る水平線によって HP 領域と N 領域が分割されると仮定する．これは前報[4]で採用した方法である．そして，この点 P″ における ρ_v/ρ_l と l/d の関係は，式 (15)，(14)

から $\sigma\rho_l/G^2l$ を消去して次式を得る．

$$\frac{\rho_v}{\rho_l}=\left\{\frac{116(l/d)^{-0.134}+0.976(l/d)^{0.324}}{0.254(l/d)+82}\right\}^{1.93} \quad \cdots\cdots(17)$$

そして図4をみるに，式（17）の与える HP 領域発生の境界条件は，ヘリウム I に対しては問題がなくなり，大まかな傾向としては，かなり妥当のように見える．しかし詳細をみれば，HP 領域の CHF 発生の実験データに対し定量的に合致しない部分があり，問題を残しているといわざるを得ないであろう．

4. HP 領域の CHF 実験データの再検討

4·1 q_{c0} の無次元整理式　初報[1]で求めた HP 領域の q_{c0} の無次元整理式（8）の精度は，既存の HP 領域の CHF のデータに関し必ずしも悪くないことを3·1節に記した．しかしながら，図2に示される解析，実験の偏差の問題は，論理的にみて，q_{c0} の整理式（8）に原因があると考えるのが最も自然である．そこで念のため，HP 領域の q_{c0} の実験データの慎重な再整理を試みてみたところ，その結果，式（8）の代わりに次式を導くことができた．

$$\frac{q_{c0}}{GH_{fg}}=0.038\,4\left(\frac{\rho_v}{\rho_l}\right)^{0.60}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.173}\times\frac{1}{1+0.280(\sigma\rho_l/G^2l)^{0.233}l/d} \quad \cdots\cdots(18)$$

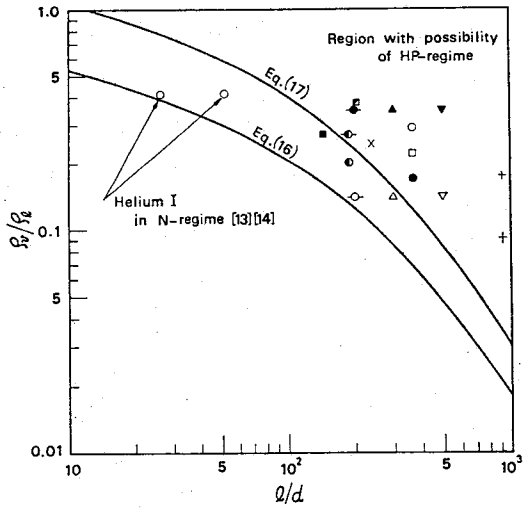

図 4　HP 領域の発生限界の検討（HP 領域の実験データは表1に対応）

図1の $l/d=200$ において，$\rho_v/\rho_l=0.321\,6$ のときの式（8），（15）の q_{c0}/GH_{fg} の比較が示されているが，両者の間にそれほど大きな相違は存在しないようにみえる．しかし，例えば初報[1]で整理に用いた q_{c0} の実験値について，式（8），（18）の RMS 誤差を考えてみると，式（8）が 8.0% であるに対して，式（18）は 5.58% であって，式（18）の方が精度は確かにすぐれているといえる．なお図5は，参考までに，表1に掲げた文献〔ただし文献（12）を除く〕の実験データによる $(q_{c0}/GH_{fg})_{Exp}$ の値を，式（18）による計算値 $(q_{c0}/GH_{fg})_{Cal}$ と比較したものである．ここで大部分の実験データは，式（18）の計算値の前後の値をとるが，Chojnowski ら[6]のデータ（×印），および Watson ら[10]のデータ（■印）は式（18）の計算値より約 10% 低い方に偏って現れている．この2実験は，表1にみるように加熱管内径 d が他のものに比べやや大きな条件になっている．しかし，これが直ちに上述の偏りの原因になるか否かは不明である．

表 1　HP 領域の CHF の発生を示す実験の条件

出　典	実験範囲					データ数	記　号
	流　体	d mm	l/d	ρ_v/ρ_l	$\sigma\rho_l/G^2l$		
Thompson-Macbeth[8]	水	1.9	365	0.170	2.69×10^{-7}—9.15×10^{-7}	9	●
Thompson-Macbeth[8]	水	1.9	365	0.219	1.86×10^{-7}—5.67×10^{-7}	9	□
Thompson-Macbeth[8]	水	1.9	365	0.289	9.17×10^{-8}—3.49×10^{-7}	9	○
Peskov ほか[9]	水	8.0	206	0.361	2.73×10^{-8}	1	◩
Becker ほか[7]	水	10.0	200	0.140	4.83×10^{-8}—2.07×10^{-7}	8	-○-
Becker ほか[7]	水	10.0	200	0.347	6.49×10^{-9}—2.40×10^{-7}	14	-●-
Becker ほか[7]	水	10.0	300	0.140	3.40×10^{-8}—2.03×10^{-7}	7	△
Becker ほか[7]	水	10.0	300	0.347	6.41×10^{-9}—2.61×10^{-7}	14	▲
Becker ほか[7]	水	10.0	500	0.140	2.13×10^{-8}—1.28×10^{-7}	10	▽
Becker ほか[7]	水	10.0	500	0.347	2.78×10^{-9}—1.32×10^{-7}	20	▼
Chojnowski-Wilson[6]	水	32.0	238	0.243	8.43×10^{-8}—1.97×10^{-7}	3	×
Watson-Lee[10]	水	37.8	145	0.271	4.67×10^{-8}—4.21×10^{-7}	5	■
Doroschuk ほか[11]	水	8.0	188	0.201	1.50×10^{-7}	1	◐
Doroschuk ほか[11]	水	8.0	188	0.271	1.76×10^{-7}—3.14×10^{-7}	2	-◐-
Campolunghi ほか[12]	水	12.0	937*	0.090 2—0.176	——	—	+

* 近似値．

4・2 K_{HP} の無次元整理式　前報[3]の方法に従い，沸騰長さの概念を用いて式（18）から解析的に K_{HP} の値を求めると，式（12）の代わりに次式が得られる．

$$K_{HP}=1.12\frac{1.52(\sigma\rho_l/G^2l)^{0.233}+d/l}{(\rho_v/\rho_l)^{0.60}(\sigma\rho_l/G^2l)^{0.173}} \quad \cdots\cdots(19)$$

そして図6は，上記の式（19）を，図2に掲げたのとまったく同じ K_{HP} の実験データと比較したものであるが，図2に比べて K_{HP} の解析値と実験値の関係が非常に改善されていることがわかる．また従って，この図6の結果は，式（18）の整理式の精度がかなり十分であることを示すものと解してよいであろう．

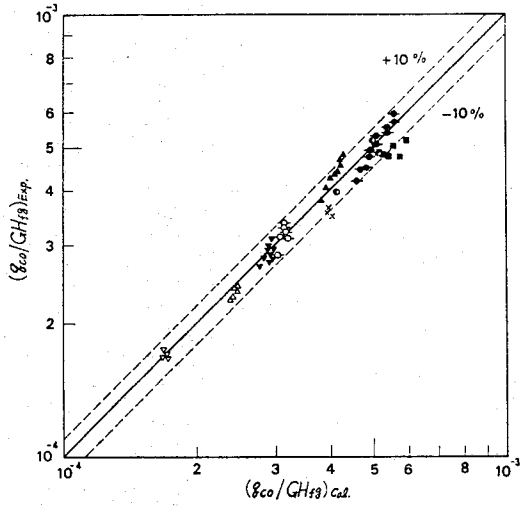

図 5　HP 領域の q_{c0} の無次元整理式（18）と実験データの比較

4・3 HP 領域の発生限界　H 領域と HP 領域の境界に関し，式（5），（18）から q_{c0}/GH_{fg} を消去して

$$\frac{l}{d}=\frac{0.384(\rho_v/\rho_l)^{0.467}-(\sigma\rho_l/G^2l)^{0.160}}{0.280(\sigma\rho_l/G^2l)^{0.393}-0.001\,19(\rho_v/\rho_l)^{0.467}} \quad \cdots\cdots(20)$$

そして図3と同じく，$\rho_v/\rho_l=0.3216$ の場合を例にとって，式（20）を用いて作製した CHF 特性領域線図が図7である．図7では，式（20）と式（7）の交点 P を通る水平線によって HP 領域と N 領域が分割されるとしている．この点 P における ρ_v/ρ_l と l/d の関係は，式（20），（7）から $\sigma\rho_l/G^2l$ を消去して

$$\frac{\rho_v}{\rho_l}=\left\{\frac{89.3(l/d)^{-0.432}}{+21.2(l/d)^{-0.062}}\right\}^{2.14}_{0.119(l/d)+38.4} \quad \cdots\cdots(21)$$

そして図8は，図4に掲げたのと同じ実験データと式（21）を比較したもので，これらの実験データに関する限り，式（21）は HP 領域発生の境界条件として妥当なものであることがわかる．

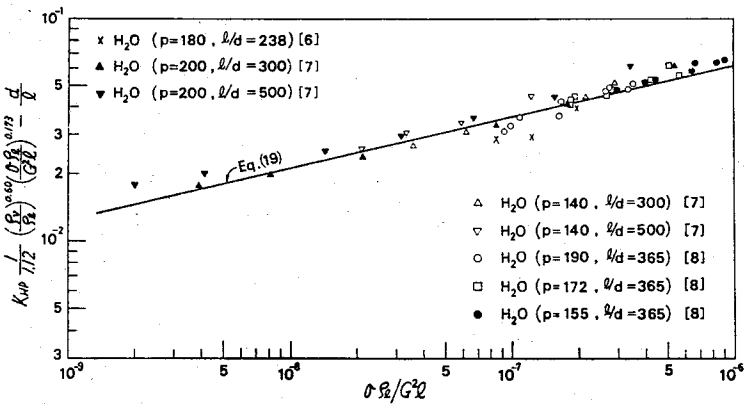

図 6　K_{HP} に関する式（19）と実験データの比較

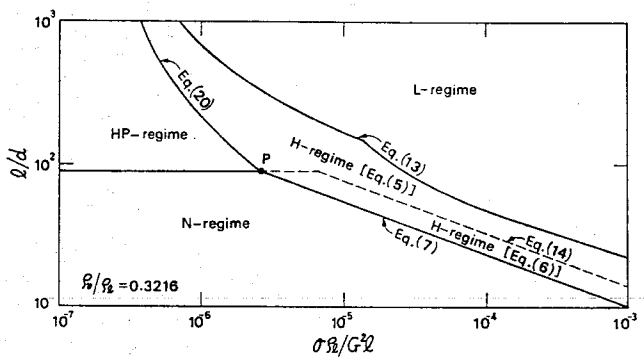

図 7　HP 領域と N 領域の境界の決定（本報の方法）

なお図8において，HP 領域発生可能域に画いてある $\sigma\rho_l/G^2l$＝一定の曲線群は式（20）で求めたもので，その位置（ρ_v/ρ_l と l/d の組合せで定まる）において HP 領域が発生するために必要な $\sigma\rho_l/G^2l$ の限界値を示す（記入してある $\sigma\rho_l/G^2l$ の値より小さな $\sigma\rho_l/G^2l$ の状態のとき HP 領域が生ずる）．そして表1に掲げた HP 領域の実験データについ

て，その $\sigma\rho_l/G^2l$ の範囲を調べてみると，図8上の対応する位置での $\sigma\rho_l/G^2l$ の限界値よりだいたい低い値になっている．ただし Thompson-Macbeth[8] のデータのなかで，$\rho_v/\rho_l=0.170$ のデータ（図8に●印で示される）だけは，図8に示す $\sigma\rho_l/G^2l$ の限界値よりやや高めにある．そして，この $\rho_v/\rho_l=0.170$ のデータは調べてみると，q_{c0} について式（18）の計算値に対し実験値の偏差が多少大きく，従って HP 領域と H 領域の間の遷移の状況に近いデータであると言えそうである．

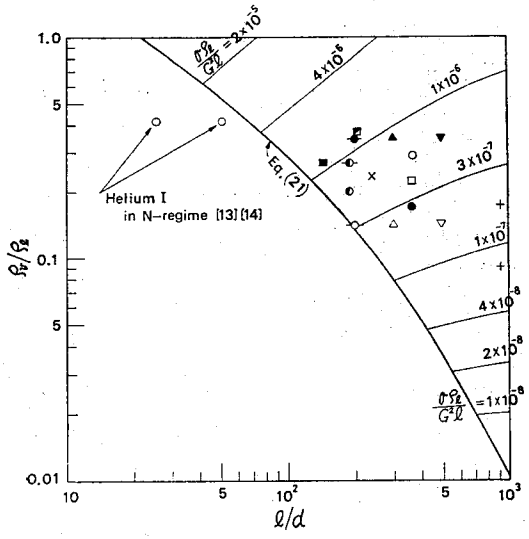

図 8　HP 領域の発生限界の式（21）と，HP 領域の実験データとの比較

ともあれ図8によれば，ρ_v/ρ_l が高い（すなわち系の圧力が高い）ほど，l/d の小さい状況まで HP 領域発生の可能性が広がることが知られる．また l/d が一定の時は，ρ_v/ρ_l が高いほど，$\sigma\rho_l/G^2l$ の大きな値で HP 領域が出現し得る．

5. CHF 特性領域線図

図8に示す HP 領域発生の実験の個数は必ずしも十分に多数というわけにはいかない．また，その実験データも，現在のところ，水の実験に限られている．従って，HP 領域発生限界に関する 4·3 節の結論についても，信頼性を高めるために今後さらに実験的な研究と検討を必要とするであろう．

しかし，少なくとも現時点では，CHF 特性領域の境界の決定について図7の方式は，かなり確実性があると考えられる．そこでいま，この方式に従い，$\rho_v/\rho_l=0.017\,87, 0.048\,47, 0.135\,1, 0.321\,6$（これは水の場合，圧力 $p=29.5$ bar, 69 bar, 137 bar, 196 bar にそれぞれ相当する）の各場合に対し CHF 特性領域線図を作成してみた結果が図9である．これは，著者の初報[1]の図 14 に与えた CHF 特性領域線図に対し，それ以後の研究結果を加え改訂，完成した結果であるということができる．

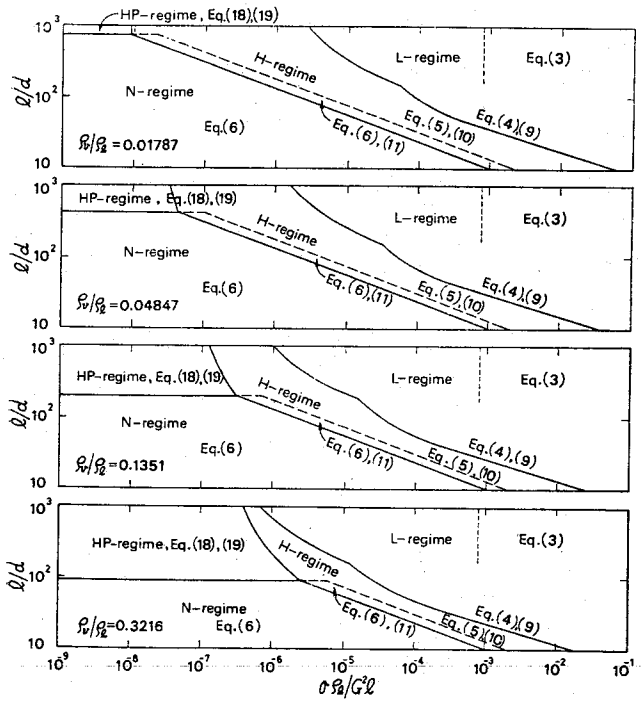

図 9　CHF 特性領域線図（各特性領域間の境界線の決定は図7による）

なお図9の各特性領域内には，その領域で適用される q_{c0} の無次元整理式，および ΔH_i の影響係数 K の無次元整理式の式番号を記入してある（ただ N 領域では q_c-ΔH_i 直線関係が成立しないので K の整理式が得られていない）．また図9各図の右上方部の垂直破線は，図1の下図（$l/d=200$）の L 領域内にみられるような式（3），（4）（ここでは $l/d>150$ なので $C=0.34$ を適用）の交点に関連し，その位置 $\sigma\rho_l/G^2l=7.84\times10^{-4}$ の位置を示すものである．

6. 結　　論

（1）　一様加熱垂直円管内の強制流動

沸騰の限界熱流束の無次元整理式に関し，著者のこれまでの研究[1]~[4]の結果を要約し式 (3)～(12) に示した．

（2） HP 領域の q_{c0} の実験データの再整理を試み，またそれに基づいて二，三の重要特性を検討した結果，著者の従来の研究結果のうち，q_{c0} に関する式 (8) を式 (18) で，K_{HP} に関する式 (12) を式 (19) で，H，HP 両領域の境界に関する式 (15) を式 (20) で，また HP 領域の発生限界に関する式 (16)，(17) は式 (21) で，それぞれ置きかえるべきことを見いだした．なお，これらの諸式と実験データとの比較を図 5，6，8 に示した．

（3） 著者の初報[1]以来，前報[4]に到るまで，CHF 特性領域線図については必ずしも完全にめいりょうとはいえない面があった．しかし，本研究の結果，図 9 のような特性領域線図を提出することができた．

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討　論

〔質問〕　西川兼康・伊藤猛宏
（九州大学工学部）

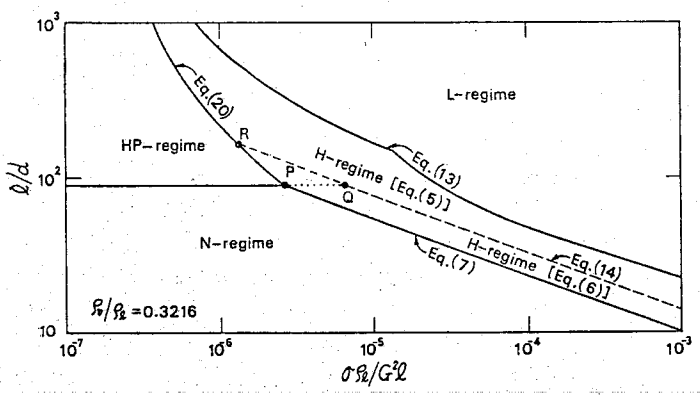
（1） HP 領域の限界熱流束が式（6）に接続して実現される可能性についてはいかがお考えか．

（2） 限界熱流束の形式（領域）の判定法を，数式でまとめて表現することはできないか．特定の条件において限界熱流束を推定することを念頭においた質問である．

（3） 限界熱流束が 4 個の形式（領域）に分類され，さらに一つの領域において整理式が 2 個になったり，一つの領域内で整理式の係数が変化している．限界熱流束の発生機構を考えるフィロソフィとして，整理式の数がこのように多くなることは合理的であろうか．

〔回答〕　（1） HP 領域が式（6）に接続して生ずる可能性を考える場合は，図 7 が付図 1 のようになる*1．ところで H 領域内にある式 (6) の適用領域は，N 領域への遷移的な領域として見いだされており（前報[1]の図 9 参照），この意味から付図 1 で PQR の領域の存在は不自然にみえる．そこで本報では図 7 の形式を考えているわけである．

（2） できるが，相当に複雑になり実用的でないように思う．そして質問者の言われる目的のため図 7（各形式の境界をきめる式を示す）



付図 1

*1 精密に言うと付図 1 で RP の間は，式 (20) の代わりに，式 (6)，(8) から q_{c0}/GH_{fg} を消去した式となるが，あまり大きな影響はなく，かつ現在の問題に本質的なかかわりを持たないのでそのままにする．

を掲げ，かつ ρ_v/ρ_l のだいたいの実用範囲内での状況変化を図9に示したのである．図9でだいたいの見当をつけ，次いで図7に示す式で細かい検討が可能である．なお図9に示す式（3）の適用範囲は，まだ多少不分明な点があるが，前報[4]の図1の結果から，だいたい $l/d>150$，$\sigma\rho_l/G^2l>7.84\times10^{-4}$ の範囲と考えてよいと思う．

（3） 著者の初報[1]以来の経緯もからむ問題だが，現時点では次のように考えればよい問題である．すなわち整理の根本はもちろん整理式(3)～(7),(18)であり，これに対しL，H，N，HPはもう少し大まかな区分概念である．そして限界熱流束の発生機構（現時点での知識水準で考える）の面からは，上記の大まかな区分概念がむしろ大いに有効と思われる．なぜなら図1にみるように，

L 領域：図の右側部に CHF 線はほぼ水平．

H，N 領域：図の中央部に右上がりの斜線（しかしN領域は q_c-ΔH_i の非直線関係で明確に区別）．

HP 領域：図の左側部に曲線の形をとる．

そして旧来の常識との対比も，例えばL領域をドライアウト，N領域をDNB，H領域はその中間と考えるようなことが可能であり，なお著者の整理式によれば特にL領域（ドライアウト）がもう少し微細区分ができるという状況なのではないであろうか．

なお著者は，初報[1]以来5編に及ぶ報告を通し，初報で与えた整理式の数と形式をほとんどそのまま維持して来たことを特に記しておきたく思う．違った点といえば，ただL領域に関し，式(3)〔これは式(4)にたいへん近く，かつごく特殊な状況のみ〕が加わったことと，式(4)の C の値が0.25と0.34の2個であったのを，式（4）のように多少細かく規定されたことだけである．

〔質問〕 **老固潔一**（川崎重工業会社技術研究所）

HP領域の q_{c0} の整理式として従来の式(8)を式(18)に修正された事により K_{HP} の値及びHP領域の発生限界について比較された実験データに，定量的に一致した式を導出された事は一連のご研究の発展だと思われ，興味深く読ませていただいた．HP領域の発生限界について以下のような素ぼくな疑問を感じたので著者のご意見をいただきたい．

従来のHP領域の q_{c0} の式〔式(8)〕と今回修正された式(18)は図1で見る限りその差は極めて小さく，特にHP領域とH領域の交点近傍では両者はほとんど一致しているように見られる．一方HP領域発生限界に関する図4中の式(16)と今回検討された図8中の式(21)の違いは，HP領域の q_{c0} の整理式として，式（8）の代わりに式(18)を用いられた事によるだけであるにもかかわらず，式(16)と(21)は定量的にかなり大きな差があるように見られる．これは図7又は3に示されているようにHP領域発生条件をかなり傾向の似ている二つの曲線〔式(20)と(7)又は式(15)と(7)〕の交点として導出されているため，式(20)と(15)の〔すなわち式(18)と(8)の〕わずかな相違がHP領域発生限界を（すなわち図7のP点を）大きく移動させたためだと考えられる．すなわち図8はHP領域の q_{c0} の整理式に対しきわめてセンシティブであると言える．他方，式(18)は実験式であるため，新たな実験データの出現などによっても若干の修正が生じる可能性を持つ性質のものであり，それにより図8も定量的にかなり大きく変化する可能性があると思われる．その意味から図8〔又は式(20)及び(21)〕がどの程度の確かさを持つものかお教え願いたい．

〔回答〕 （1） 式（8）と(18)の間の小さな修正（例えば前報[1]の図12と本報の図5を比較）が，いろいろの点に相当の影響をおよぼす事実には著者も気付き，HP領域の q_{c0} の整理式には微妙な性質があるように感じていた所である．そしてその事柄は，q_{c0} の整理式から解析的に導いた K_{HP} の計算式を K_{HP} の実験値と比較した時の図2，6間の相違にすでに出現しているように思う．

（2） それにしても，式(18)から解析的（その考え方には相当の理論的妥当性があるように思う）に求めた K_{HP} の計算式(19)が，K_{HP} の実験値と比較して図6の程度のよい一致を示したことは，今後新たな実験データの出現で式(18)が改変されるとしても，その改変の度合いはそれほど大にはならないことを予想させる．

（3） さてさらにご質問の内容に入るが，ご指摘のようなことが確かにあろうかと思う．ただ以下のような事情を考える価値もあろう．すなわち，

（i） 図1で $l/d=200$ の時，式(8)と式(18)の差が極めて小である問題： これは図1が例としては非常に不適当だったことが反省される．なぜなら図4〔ここで式(17)は小細工的なので省き，式(16)の方を考えていただきたい〕と図8を比較すると，両図の境界線が $l/d=200$ で偶然一致するのである（従って図1で差が極小になるはずである）．

（ii） かなり傾向の似ている2曲線の交点を考える問題： 確かに図3（A）ではその傾向が著しいといえる．しかし改訂された図7では，式(20)と式(7)

の曲線のこう配に相当の差が存在するようになっている．さらに図9によれば，ρ_v/ρ_l が小になるほど上記のこう配差は増大している（なお $\rho_v/\rho_l=0.3216$ は現状では通常の実験圧力の上限付近になると考えてよいと思う）．

（4） 以上の次第で，HP 領域の q_{c0} の整理式の微妙な性格は感じつつも，著者としては，図8はかなりよい線に達している（換言すれば図3の段階のようなあいまいさはない）と考えている．なお本文中にも書いたように HP 領域の CHF の既存のデータは水に限られている．そこで著者の実験室では R 12 による実験を企図し，現在，$l/d=200$ で $\rho_v/\rho_l=0.214$ の状況下のデータしか得ていないが，図8の指示に一致し，HP 領域が出現していることを付記したく思う．

〔質問〕 **一色尚次**（東京工業大学）

（1） 各式の精度については HP 領域（図5）と同じく ±10% と考えてよいか．

（2） 長さ l の影響については，質問者も今までは影響がないとする立場で考えて来たが，サブクールの場合必ずしも完全には消去し切れないものがあることに気が付いている．著者の論文ではむしろ l/d や l に関するウェーバ数を表に出しておられ，この方法もまた整理上は極めて有効であることを示しておられる．その点は質問者も十分認めるものであるが，将来さらにミクロかつ物理的なモデル説明を与えるさいにかなり難点を与えると感じている．その点につき，質問者は，このような二相流沸騰時には，サブクールがある程度強い領域では，CHF は必ず，何らかの「遷移点脈動」を伴わざるを得ず，その脈動の安定，不安定に対しては流路内流体通過時間が影響するので，その流体通過時間をなかだちとして長さ l の影響が効いてくるのではないかと考えているが，このような考え方に対するご意見などを伺いたい．

〔回答〕 （1） まず q_{c0} の整理式については，前報[4]の図1を参考にしていただきたく思う〔なお同論文への老固氏の討論への回答（1）も参照〕．また K に関する式は，上記の q_{c0} の整理式から解析的に定まるが，実験値との比較は前報[3]の図8，9，および本報の図6を見ていただきたい．ただ K については，式（1）にみるように $K\Delta H_i/H_{fg}$ の1に対する大きさも大切になる（例えば $\Delta H_i/H_{fg}$ が小さい時は，K の誤差はあまり影響しない）．

（2） l に関しては，著者の整理式では $\sigma\rho_l/G^2l$ および l/d の両方の形で入ることが特色である．そこで，この両方にわけて意見を記す．

（i） $\sigma\rho_l/G^2l$ については次の事実がある．すなわち，加熱面上を薄い液膜状で飽和液体が流れる単純な沸騰系で，流れの方向の加熱面の長さを l として，$\sigma\rho_l/G^2l$ を使って CHF のデータがよく整理できる〔文献（付1）～（付3）参照〕．これに対し Lienhard ら(付4)(付5)の解析的説明も試みられているが，必ずしも納得できない点が多々ある．すなわち現在の知識の水準では $\sigma\rho_l/G^2l$ の形で上記の沸騰系になぜ l が効くのか説明しにくく，質問者の言などを借りれば難点があるわけだが，むしろそれに合う説明を見いだす必要があるのではないか（ただしもちろん，上記の $\sigma\rho_l/G^2l$ が円管内の場合とどんな関連を持つかの問題でもあるが）．

（ii） l/d の影響に関しては，著者の整理式に次の特色がある．すなわち，L 領域（ドライアウト形式の CHF）の式（3），（4）は l/d の影響を直接受ける形である．次に H 領域で主要な式（5）では，$1/(1+0.003\,1l/d)$ の形で効くので l/d の影響は前よりずっと小さくなる．さらに N 領域（DNB 形式の CHF）に入ると，その整理式（6）に含まれる $(l/d)^{0.27}/(1+0.003\,1l/d)$ の値は，$l/d=10$～$1\,000$ の範囲内で変化がさらに小さくなる．そしてこのような l/d の仕方は，質問者のご意見（ここで遷移的脈動とはドライアウト的な状況で壁温が脈動することなどと解釈）にかなり合致する傾向にあるとも考えられるがいかがなものであろうか．

〔質問〕 **井村英昭**（熊本大学工学部）

HP 領域と N 領域の境界の決め方において，図3（A）および（B）の二つの方法を考えておられるが〔最終的には図7，8で（A）の方法が採用されている〕，141ページ右欄下7，6行めの「図3（B）では，式（14）を利用する点で物理的にやや不自然さを持つが」とは具体的にどういう意味か．また，この文章から，逆に図3（A）の方法が物理的に合理的とも解釈される．この点についてご教示願いたい．

〔回答〕 図1で式（5），（6）の両直線を見ていただくとわかるように，両者は大まかに言うと，ほとんど同じものといえるような状況である．従って両直線の交点〔式（14）で与えられる〕は元来，あまり明確な物理的状況の区別のできる点ではないようにみえる．これに対し N 領域では q_c-ΔH_i の非直線関係と

（付1） 門出・甲藤，機論，**43**-373（昭52），3399，3408.
（付2） 甲藤・石井，機論，**44**-384（昭53），2817.
（付3） Katto, Y. and Shimizu, M., *Trans. ASME, J. Heat Transf.*, **101**-2 (1979), 265.
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（付5） Lienhard, J. H, and Hasan, M. Z., *Trans. ASME, J. Heat Transf.*, **101**-2 (1979), 383.

いう著しい特性を持ち，これによってH領域から明確に区別される．従って，その境界を示す式(7)の方が，物理的に式(14)よりはるかに重要と考えてよいのではないか．そしてこのような意味で，式(14)を基礎にした図3(B)〔単にデータに合わせる目的で式(14)を使用してみたという形〕が不自然で，式(7)を基礎にした図3(A)の方が合理的と考えた次第である．

〔質問〕　**鳥飼欣一**（日本原子力研究所）

（1）入口サブクールの広範囲にわたっての q_c については，図9には示されてないと思われるが，この場合の領域について，図9に類するものも考えられるかどうか．考えられるならば，どのような図に（図9と比較して）なるか．

（2）なお，サブクールのある場合とない場合の流動状態は観察上も異なる場合が多く，特に流動が振動的になる場合もあるがどう思われるか．式(1)の K も，扱い方に難しい点はないか．

〔回答〕　（1）図9の各領域には，そこで入口サブクール（$\Delta H_i>0$）がある時に使う計算式(9)～(11)，(19)も書きこんである．なお入口サブクールが極端に大になった時にどうなるかは研究を要する問題だが，従来の通常の実験における ΔH_i の変化範囲は，ある大きさの中に限られており，その程度の範囲内では，一般に図9の方法（つまり $\Delta H_i=0$ のときと同じ特性領域にあるとする）で実用上十分のようである．

（2）サブクールのない場合に二つの状況があり得る．一つは $\Delta H_i<0$（二相混合入口条件），他は $\Delta H_i=0$（飽和入口条件）である．さて著者の整理式は元来，$\Delta H_i>0$（その極限として $\Delta H_i=0$ を含む）を対象とするので，$\Delta H_i<0$ の場合は除外することにする．次に $\Delta H_i=0$（ないし $\Delta H_i\fallingdotseq 0$）の場合だが，著者の調査した多くの実験（通常，系内の可圧縮部は絞りにより動的接続を絶つなど振動防止の配慮をしている）の中には，$\Delta H_i=0$ の近傍の実験をしたものもかなりあり，しかし特に振動は問題になっていないようである．また式(1)の K について扱い方に困難を感じたこともなかった．

536.423.1 :: 532.542

寄　書

一様加熱垂直円管における強制流動沸騰の限界熱流束と流動様式との関係について*

甲　藤　好　郎**
Yoshiro KATTO

1. 序

最近著者は，一様加熱垂直円管内の強制流動沸騰（入口サブクールないし入口飽和）の限界熱流束（CFH）に関する実験データを分析し，4個の特性領域（L, H, N, HPと名付ける）に分けてCHF無次元整理式を提出[1]~[3]，なお多くの既存実験データの図式表示により上記の整理式がCHFの一般特性をかなりよくあらわし得ることを示した[4]．ここでL領域は，他の条件は一定として質量速度Gが小さい時，N領域はGが大きい時に生じ，そしてH領域は上記のL，Nの中間に生ずる領域である．また，HP領域は，通常，圧力が非常に高い時，N領域の代わりに出現する領域である．なお上記の4領域のうち，N領域では，限界熱流束q_cと入口サブクール・エンタルピΔH_iとの間に直線関係が成立しないという特徴があり，この特性によってH領域から区別される（著者の前報[1]参照）．

さて一様加熱円管の強制流動沸騰の場合，CHFの発生は最初，管出口付近からおこる[5]．従って，前述の各特性領域のCHF発生時，加熱管出口付近における流動様式（フローパターン）を調べてみることは興味あることであり，なおCHFの発生機構の研究にも寄与するものがあると考えられる．そこで本報では，一様加熱垂直管出口の流動様式を実測しているBerglesら[6]，およびBennettら[7]の研究結果を利用してCHF特性領域と流動様式の関連を調べてみる．ただし上記の2研究[6][7]ではHP領域の状況を含んでいないので，以下にはL，H，Nの3領域のみを扱う．

2. 一様加熱円管出口の流動様式

Berglesら[6]は水を用い，圧力p=34.5 bar（この場合，飽和温度241°Cに対し入口液温は121, 163, 204°Cの3種類）および69 bar（この場合，飽和温度283°Cに対し入口液温は149, 204, 260°Cの3種類），管内径d=0.010 2 m，管長径比l/d=60, 150, 240の条件下で質量速度Gおよび熱流束qを変えて流動様式を調べている．ここで流動様式の検出は，電気抵抗プローブを管出口におき，プローブと管壁間のコンダクタンスの時間変動を記録することによっている．そしてその結果，気ほう（bubbly），スラグ（slug），フロス（froth），および環状噴霧（spray annular）の4種類の流動形式が区別され，流動様式線図（質量速度Gと出口クオリティχ_{ex}間の関係として流動様式の境界を図示）が与えられた．

ところで一様加熱円管では，エネルギのつりあいから次式が一般的に成立する．

$$\frac{q}{GH_{fg}}=\frac{1}{4}\frac{d}{l}\left(\chi_{ex}+\frac{\Delta H_i}{H_{fg}}\right) \quad \cdots\cdots(1)$$

ここに，H_{fg}は蒸発潜熱である．従って，上記の流動様式線図〔文献(6)の図6～8〕上の任意の点におけるq/GH_{fg}の値が上式によって容易に計算され，また同じ点における$\sigma\rho_l/G^2l$（ここにσは表面張力，ρ_lは液体密度）の値の計算も容易である．すなわちBerglesらの与えた流動様式線図を$(q/GH_{fg})-(\sigma\rho_l/G^2l)$線図上に移すことができ，図1，2はその結果である．図1はp=34.5 bar，図2はp=69 barの場合，ただし簡単のため入口液温は図1で204°C，図2で260°Cの場合だけを示す．なお図1，2の実験は入口サブクール状態であるので，管出口でχ_{ex}=0になる時の熱流束の計算値を点線で示してある．また図1の最上図に示す3個の○印はBerglesらのCHF実測値である．

一方，Bennettら[7]は同じく水を用い，圧力p=34.5および69 bar（共に入口サブクール温度は11～17°C），管内径d=0.012 6 m，管長l=3.66 mの条件で実験している．ただ流動様式の検出には，管出口直後の流れの瞬間写真および高速映画（なお補助的に加熱管内の流れのX線写真を使用）を用いている．そしてG-χ_{ex}形式の流動様式線図〔文献(7)で圧力34.5 barの図5·7，圧力69 barの図5·8〕を与えているが，簡単のため本報では圧力69 barの場合の流動様式線図を(q/GH_{fg})-$(\sigma\rho_l/G^2l)$線図に移した結果

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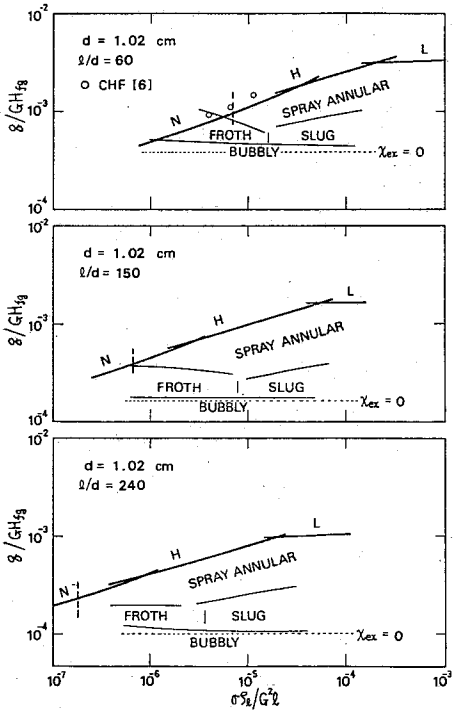
を図3に示す．なお同図上の鎖曲線は，同じ装置で Bennett らの求めた CHF 実測結果を示している．さて図3の結果を与える実験条件は，図2の最下図のそれに近い．図3の場合の入口サブクール度は図2の場合の約半分のため，$\chi_{ex}=0$ の位置が図2より下方にずれているが重大な問題ではない．また文献 (6), (7) によれば，図2の領域 froth と図3の領域 wispy annular とは同じものを指していると解してよく*1，従って図2と図3は定性的にほぼ似た結果であるといってよい．図2，3で特性領域の境界位置に大きな不一致がみられるのは恐らく，検出方法の相違による所が大であろう〔文献 (6) 参照〕．

3. 限界熱流束の特性領域

入口サブクール・エンタルピ ΔH_i の時の限界熱流束 q_c を次のように書く．

$$q_c=q_{c0}\{1+K(\Delta H_i/H_{fg})\} \quad \cdots\cdots(2)$$

この時，右辺の q_{c0} および K は以下のように与えられる．



水，p=34.5 bar. 飽和温度：241°C，
入口水温：204°C ($\Delta H_i/H_{fg}=0.0975$)

図 1 管出口の流動様式 (Bergles ら[6]) と限界熱流束の予測

*1 我が国では，froth は churn に等しい，あるいは froth と wispy annular は別ものとすることがある〔文献 (8), (9) など参照〕が，本文での用語法はそれとは違うので注意のこと．

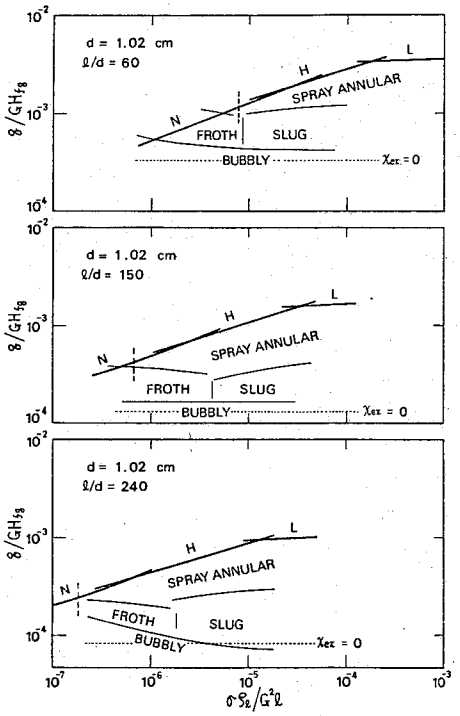
まず q_{c0} については，著者の前報[4] から

L領域：

$$\frac{q_{c0}}{GH_{fg}}=C\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d} \quad \cdots\cdots(3)$$

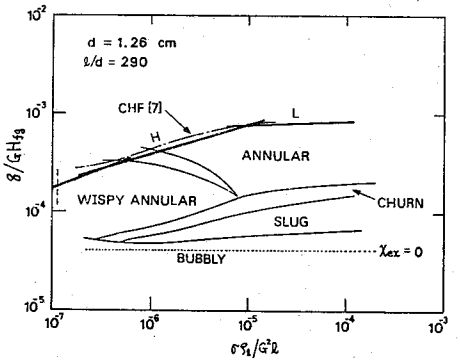
C=0.25 (l/d<50)，C=0.34 (l/d>150)，

l/d=50～15 では C は 0.25～0.34 間を直線的に変化．



水，p=69 bar. 飽和温度：283°C，
入口水温：260°C ($\Delta H_i/H_{fg}=0.0799$)

図 2 管出口の流動様式 (Bergles ら[6]) と限界熱流束の予測



水，p=69 bar. 飽和温度：283°C，
入口水温（平均）：269°C ($\Delta H_i/H_{fg}=0.0476$)

図 3 管出口の流動様式 (Bennett ら[7]) と限界熱流束の予測

HおよびN領域：

$$\frac{q_{c0}}{GH_{fg}}=0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\times\frac{1}{1+0.003\,1l/d}\quad\cdots\cdots(4)$$

および

$$\frac{q_{c0}}{GH_{fg}}=0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}+\frac{(l/d)^{0.27}}{1+0.003\,1l/d}\quad\cdots\cdots(5)$$

ここに，ρ_v は蒸気密度，なお H，N 領域の境界は次式で与えられる．

$$\sigma\rho_l/G^2l=\{0.77/(l/d)\}^{2.70}\quad\cdots\cdots(6)$$

次に式（2）右辺のKは，著者の前報[3]に示したごとく，q_c-ΔH_i の直線関係および沸騰長さの概念を利用して式（3）～（5）から理論的に次のように求められる．

L領域：　式（3）に対応して

$$K=\frac{1.043}{4C(\sigma\rho_l/G^2l)^{0.043}}\quad\cdots\cdots(7)$$

H領域：　式（4）に対応して

$$K=\frac{5}{6}\frac{0.012\,4+d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{1/3}}\quad\cdots\cdots(8)$$

および式（5）に対応して

$$K=0.416\frac{(0.022\,1+d/l)(d/l)^{0.27}}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{0.433}}\quad\cdots\cdots(9)$$

なおN領域では q_c-ΔH_i の直線関係が成立しないのでKは求められていないが，H領域に近いN領域の範囲で，かつ ΔH_i の値が小さい場合に限れば，式（9）を近似的に使ってよいと考えられるので，本報ではこの方法によってN領域のq_cを計算する．そして図1～3の太い線は，このようにして式（2）～（9）で計算したCHFを示すもので，これらのCHF線上に，L，H，Nの記号を付し特性領域を区別してあるが，H領域とN領域を区分する垂直破線は式（6）によるものである．

4. 討　　論

図1～3の結果から以下のような事柄を指摘することができるであろう．

（i）　図1の最上図の Bergles らの3個のCHFの実験データは，著者の整理式によるCHF予測線の近くにあらわれている．また図3の Bennett らの CHF 実測線（鎖線）についても同様なことがいえる．

（ii）　図1，2に示す6枚の図面すべてを通じて，L，H領域の CHF に対し管出口の流動様式は環状噴霧流の状況にあり，またN領域に対してはフロス流（ないし気ほう流）の状況にあるとみてよいようである．また図3についても，大まかには上に記したと同様な状況にあるといってよいであろう．

(iii)　上記（ii）項の結果によれば，L，H両領域を通して管出口の流動形式は同一（環状流）とみなしてよい．従って CHF の特性がL領域とH領域とで変化する原因は，流動形式のような大まかなものでは判断ができず，もっと微細な流動挙動（壁面上の液膜が，L領域では乾燥することにより，またH領域では有限の厚さの液膜が何らかの原因で破断することにより CHF が発生するなど）によって生ずるものと考えられる．

(iv)　上記（ii）項の結果からみて，N領域は，いわゆる DNB (departure from nucleate boiling) 形式の CHF に近い状況にあると考えられる．なおN領域のみに現れる q_c-ΔH_i の非直線関係は，管出口でフロス流（ないし気ほう流）という流動形式に関係しているかも知れない．

（v）　図1～3の結果から見る限り，管出口がスラグ流（ないしチャーン流）になる状況下に CHF が生ずることはないと考えてよいようである．

(vi)　加熱円管内の二相流については，HP 領域の CHF が出現するような状況下に新しいタイプの流動形式がみられるかも知れない．しかし，これについては将来の研究が必要である．

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536.423.1 :: 532.542

寄　書

一様加熱垂直円管内の強制流動沸騰限界熱流束の「熱流束-クオリティ」形式の整理について*

甲　藤　好　郎**
Yoshiro KATTO

1. 序

限界熱流束-出口クオリティの形のデータ整理法は簡便なので，現在でも限界熱流束（CHF）の主要な整理形式の一つをなしている．ところで最近，著者[1]~[5]は一様加熱円管内のCHFの無次元整理式を求め，既存の実験データの一般特性をよく表し得る結果を得た．そこで本報では，これによって前記の整理形式の特性を検討してみた結果を報告する．

2. 著者のCHF整理式

一様加熱円管に対し次の式が一般に成り立つ．

$$\chi_{ex}=4\frac{q_c}{GH_{fg}}\frac{l}{d}-\frac{\Delta H_i}{H_{fg}} \quad \cdots\cdots(1)$$

ここに，χ_{ex}：管出口クオリティ，q_c：限界熱流束，G：質量速度，H_{fg}：蒸発潜熱，l：管長，d：管内径，ΔH_i：入口サブクール・エンタルピである．そしていま上式右辺の q_c を次のように書いてみよう．

$$q_c=q_{c0}\{1+K(\Delta H_i/H_{fg})\} \quad \cdots\cdots(2)$$

ここでまず右辺の q_{c0} は，著者の前報[5]から，L，H，N，HPの4領域にわけて下記の無次元整理式によって定まる．

L領域：

$$\frac{q_{c0}}{GH_{fg}}=C\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d} \quad \cdots\cdots(3)$$

ここに，C は定数で，$C=0.25$ $(l/d<50)$，$C=0.34$ $(150<l/d)$，また $50<l/d<150$ では 0.25 から 0.34 まで直線的に変化するとする．なお σ：表面張力，ρ_l：液体密度である．

HおよびN領域：

$$\frac{q_{c0}}{GH_{fg}}=0.01\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}+\frac{1}{1+0.003\,1l/d} \quad \cdots\cdots(4)$$

および

$$\frac{q_{c0}}{GH_{fg}}=0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}+\frac{(l/d)^{0.27}}{1+0.003\,1l/d} \quad \cdots\cdots(5)$$

ここに，ρ_v は蒸気密度である．

HP領域：

$$\frac{q_{c0}}{GH_{fg}}=0.038\,4\left(\frac{\rho_v}{\rho_l}\right)^{0.60}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.173}+\frac{1}{1+0.280(\sigma\rho_l/G^2l)^{0.233}l/d} \quad \cdots\cdots(6)$$

他方，式（2）右辺のKは著者の前報[3][5]に示したごとく，q_{c0} の整理式（3）~（6）から解析的に下記のごとく求められる．

L領域：　式（3）に対応して

$$K=\frac{1.043}{4C(\sigma\rho_l/G^2l)^{0.043}} \quad \cdots\cdots(7)$$

H領域：　式（4）に対応して

$$K=\frac{5}{6}\frac{0.012\,4+d/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{1/3}} \quad \cdots\cdots(8)$$

また式（5）に対応して

$$K=0.416\frac{(0.022\,1+d/l)(d/l)^{0.27}}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{0.433}} \quad \cdots\cdots(9)$$

HP領域：　式（6）に対応して

$$K=1.12\frac{1.52(\sigma\rho_l/G^2l)^{0.233}+d/l}{(\rho_v/\rho_l)^{0.60}(\sigma\rho_l/G^2l)^{0.173}} \quad \cdots\cdots(10)$$

また図1は著者の前報[5]で定めた特性領域線図で，ここには3種類の密度比 ρ_v/ρ_l の値に対し，L，H，N，HP各領域の範囲を示し，その領域で用いる q_{c0}，K の計算式を書き込んである．なおN領域では，式（5）に対応するKの計算式は求められない[1]．しかしH領域に近いN領域で，かつ ΔH_i が大でなければ，式（9）が近似的に適用できる．この意味で本報では式（9）をN領域でも使うことにする．

3. q_c-χ_{ex} の計算結果

圧力p，質量速度G，管径dが一定の条件下，種々の管長lごとに，入口サブクール・エンタルピ ΔH_i を変える時の q_c-χ_{ex} の関係は次のごとくして計算さ

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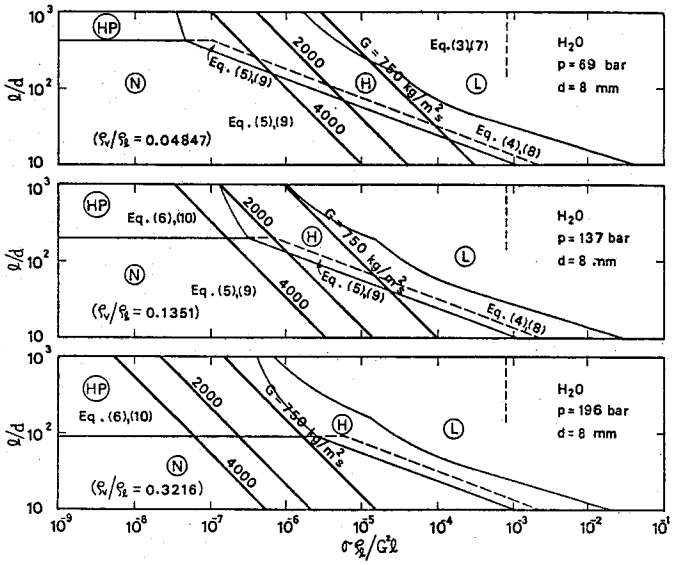
れる．いま水，d=8 mm，G=750，2 000，4 000 kg/m²s の場合を例にとると，まず図1の各図に示す圧力 p のもとで l を変化させる時の斜直線が描かれる．次に図1でこれら斜直線の通る特性領域の整理式によって，与えられた l（従って l/d）の値ごとに q_{c0} と K の値が計算できる．ゆえに式（1），（2）によって l/d の値ごとに，ΔH_i をパラメータとして q_c-χ_{ex} の関係が定まる．

さて本報では，上記の計算にあたり，通常の実験範囲などを参考に，ΔH_i=0～400 kJ/kg の範囲で q_c-χ_{ex} の関係を計算した．これは l/d の値ごとに，q_c-χ_{ex} 線図の上に一つの短い線分を描く．そして図2～4は，こうして求めた q_c-χ_{ex} の関係（10～793 の間の l/d のとびとびの値に対し）を示す．なお図2～4でN領域の結果は破線，H領域は細線，L領域は太線，HP 領域のものは鎖線で区別してある．



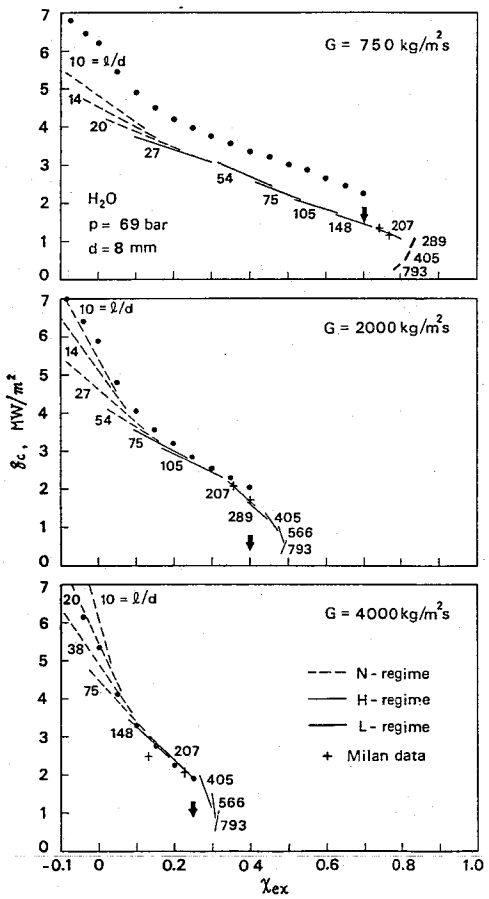
（ρ_v/ρ_l=0.048 47，0.135 1，0.321 6 の各場合）

図 1　CHF 特性領域線図

4. 討　　論

（I）　著者の整理式に基づき，前章のようにして求めた q_c-χ_{ex} の関係（図2～4）から次の特徴がみられる．まずN領域（ここでは l/d が小）では q_c-χ_{ex} の線分が分散し，かつ G が大きいほど分散が大きい．次にH領域では，q_c-χ_{ex} 関係は近似的に一つの曲線を形成する形となり，かつ l/d が大になると曲線が下向きに曲がるようになる．これらの特徴はすべて，Stevens ら[6]によって実験的（R-12 による）に見いだされた特徴に一致する．なおL領域は，本報の計算範囲では，圧力低く，G が小の時，しかも l/d が非常に大きい時にわずかに現れるにすぎない．また図4にみるように，HP 領域では q_c-χ_{ex} 関係は比較的簡単な形状の曲線を近似的に形成する形で出現する．

（II）　図2～4にプロットした●点は，ソ連科学アカデミー[7]が実験データから決定した水の CHF の標準値表（d=8 mm に対してのみ与えられる）の値であり，また↓印はその近傍の χ_{ex} 位置で q_c-χ_{ex} の関係に屈折点が生ずると説明されているものである．さて上記の CHF 標準値を計算結果と比べると，まず図2で G=750 kg/m²s の場合の偏差が大きい．しかし図2に＋印で示すミラノ再現性試験データ〔文献（2）参照〕なども含めて判断すると，ソ連の標準値に少し



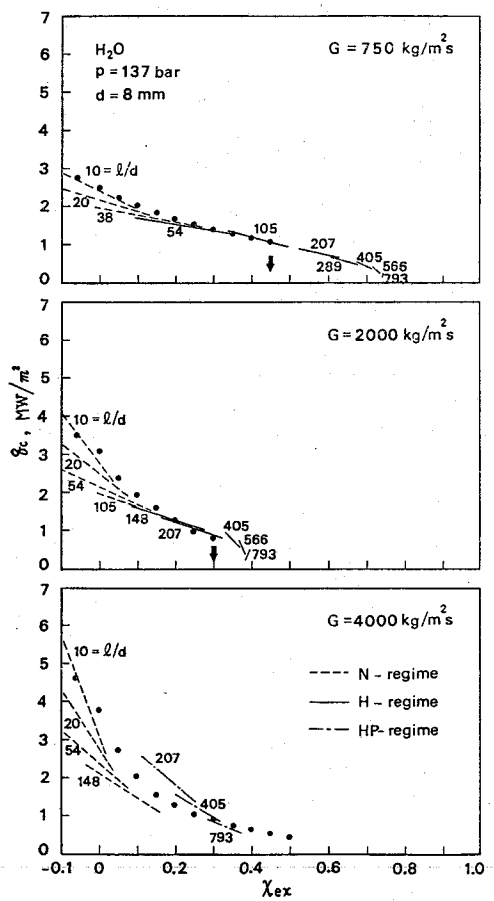
（水，p=69 bar，d=0.008 m）

図 2　q_c-χ_{ex} の関係

問題があるように思える．次に図3の G=750 kg/m²s の場合，↓印の位置が，CHF 計算曲線の下向き曲がり位置から相当にずれているが，この原因はわからない．なお図3で G=4 000 kg/m²s の場合，計算結果ではN，HP 領域間が不連続になるが，ソ連の標準値は連続的に定められている．しかし以上のような事を除くと，著者の無次元整理式を基にした q_c-χ_{ex} 関係は，ソ連の標準値の結果とかなりよく一致するといってよい．

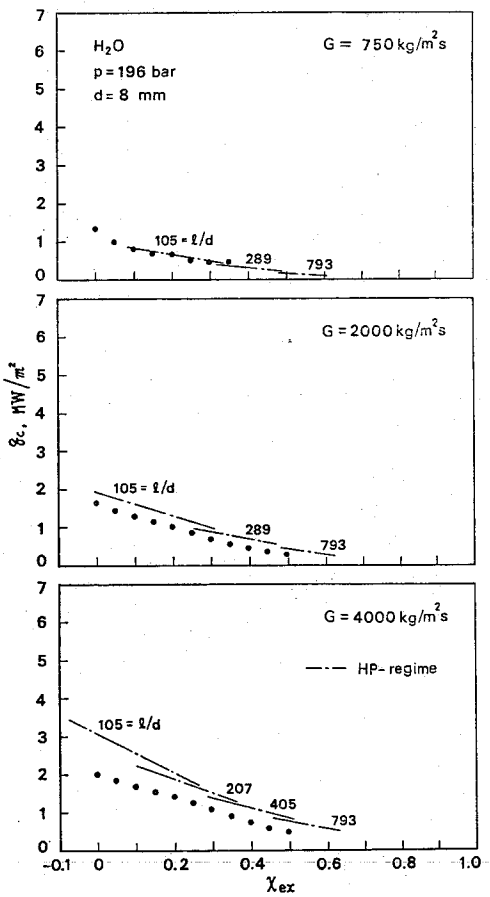
（Ⅲ） しかし図2，3にある↓印について次の問題がある．ソ連の CHF 標準値表では，↓印位置より小さい χ_{ex} 範囲の q_c（図で●点で表されている）は DNB による限界熱流束であり，それが↓印付近でドライアウトによる限界熱流束に変化すると考えている．しかし本報の計算結果では，q_c-χ_{ex} 関係曲線が↓印の近くで下向きに曲がるもののH領域を保持して特性領域の変化がみられない（図2の最上図の場合を除く）．しかも著者の整理でH領域は，L領域（ドライアウトに相当）とN領域（DNB に相当）の中間にある領域と考えられている．従って上記の矛盾について今後の研究を必要とするが，次の事柄を指摘しておかねばならない．すなわち，入口サブクール条件下の実験では，実は l/d>400 のH領域の実験データはほとんど無く〔文献（4）参照〕，図2，3の↓印付近における CHF の実験（特にソ連における）の大部分は気液二相の混合入口条件でおこなわれており，ゆえに Hewitt[8] が指摘するように，その結果の一般性に疑問があるとせざるを得ない．

（Ⅳ） 著者の整理式によると，図2～4にみるごとく，HP 領域を除けば常に q_c-χ_{ex} 曲線は χ_{ex} の増加と共に下向きに曲がり，そして χ_{ex} の最大限界値（それ以上 χ_{ex} が上昇しない）を持つ．ところで CHF に関する従来の常識では，p，G，d 一定条件下に l を増加すると χ_{ex} が増加してドライアウト状態になると考えられ，q_c-χ_{ex} 曲線は最終的に χ_{ex}=1，q_c=0 点に向かうとされていたといえる（二，三の論文や教科



（水，p=137 bar，d=0.008 m）

図3 q_c—χ_{ex} の関係



（水，p=196 bar，d=0.008 m）

図4 q_c—χ_{ex} の関係

書にはそれを図示している). しかし, この常識は必ずしも自明なものでなく, 上記の計算結果からみると, かなり疑わしく思われる. なおこの点に関し, 入口サブクール条件下の実験では, 実用になる長さの管の場合, $\chi_{ex}=1$ の近傍領域に入るのは困難とされて来た事実[9][10] にも注目する要がある.

5. 結　　論

著者の CHF 無次元整理式を基に q_c-χ_{ex} を計算し, 実測特性と比較することにより, 従来 q_c-χ_{ex} 形式のデータ整理で常識的に考えられていた事柄に疑問点もあることなどを示した. なお本報の結果から考えて, q_c-χ_{ex} 特性から限界熱流束の機構を推測することには危険性があると思われる.

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〔604〕 一様加熱垂直円管内の強制流動沸騰の限界熱流束：R-12によるHP領域の実験的研究

論文前刷

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1．序

一様加熱垂直円管内の強制流動沸騰に関し，Peskovら[1]，Beckerら[2]，およびCampolunghiら[3]の高圧下の諸実験（実験範囲は表1参照）において，高圧下の水の限界熱流束（CHF）の特性が指摘され，またPeskov[1]，Lee[4]，Becker[3]のCHF経験整理式がそれぞれ導かれた．一方，著者の一人，甲藤[5~7]は最近，一様加熱垂直円管のCHFの無次元整理の研究をおこない，上記の特殊領域をHP領域と名付けて区別し，その発生条件について図1の結果を得た．すなわち図1（ここに ρ_v：蒸気密度，ρ_l：液体密度，l：加熱管長，d：加熱管内径）の曲線の右側の各場所で無次元量 ρ_l/G^2l（ここに σ：表面張力，G：管内の質量速度）の値が図示の値より小さい状況下にHP領域のCHFが生ずる．そして甲藤は，蒸気発生器などに関連しておこなわれた最近のChojnowski-Wilson[8]，Watsonら[9]，Doroshukら[10]またはLevitan-Lantzman[11]の水のCHFの実験データないしその一部がHP領域に属することを示し，なお古くThompson-Macbeth[12]（その表11~14）の收集したデータの中にもHP領域に属するものがすでに含まれていることを示した．以上の諸実験は表1に示すごとく，Peskovら，Beckerらなどと同じく，l/d が比較的大きく，ρ_v/ρ_l が高い状態でおこなわれている．

一方，Watersら[13]，Matznerら[14]，Merilo[15]，およびMerilo-Ahmad[16]の垂直円管内の水またはR-12の実験は，図1のHP領域発生可能範囲の境界線付近から隔たった範囲にわたる条件下（表1参照）の

表1．引用した垂直円管内のCHF実験の主要条件

Main Regime	Author	Fluid	l(cm)	d(cm)	l/d	ρ_v/ρ_l	G(kg/m^{2}s)	Remark
HP	Peskov et al.1969	Water	89.0-210	0.80	111-263	0.144-0.366	1110-5000	Slow and small temp. rise CHF*
	Becker et al.1972	Water	200-500	1.0	200-500	0.140-0.347	1000-7000	
	Campolunghi et al.1974	Water	∿1100	1.2	∿937	0.0902-0.176	1100-2600	
	Chojnowski-Wilson 1974	Water	762	3.2	238	0.243	682-1462	
	Watson et al.1974	Water	549	3.78	145	0.271	407-2034	Upstream CHF
	Doroschuk et al.1975	Watwe	150	0.80	188	0.201-0.271	1500-3000	
	Levitan-Lantzman 1975	Water	150	0.80	188	0.201-0.271	1500-3000	
	Thompson-Macbeth 1964	Water	69.6	0.191	365	0.171-0.285	1940-3780	Data in Tables 11-14
Near HP	Waters et al.1964	Water	366	1.12	327	0.0493-0.0853	6690-9320	Upstream CHF for G = 6690-9320 kg/m^{2}s
	Matzner et al.1965	Water	244-488	1.02	240-480	0.0484	1340-9480	Upstream CHF for G = 6900-9300 kg/m^{2}s
	Merilo 1977	R-12	244-488	1.26	194-387	0.0487	950-6500	Upstream CHF for G = 5500-6500 kg/m^{2}s
	Merilo-Ahmad 1979	R-12	103-305	0.53	194-575	0.0487-0.0768	1600-8100	Upstream CHF for G = 8100 kg/m^{2}s
H and N	Groeneveld 1972, 1974	R-12	138	0.78	177	0.0498	1330-8100	Upstream CHF** and slow CHF
HP	Present study	R-12	100	0.30-0.50	200-333	0.109-0.306	1100-9000	

* in case of high pressure and small ΔH_i.　** for mixed inlet conditions G > 4050 kg/m^{2}s.

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ものであるが，質量速度Gの非常に高い値まで実験され，表1右端の注に示すGの範囲に対し，上流CHF（upstream CHF），すなわち加熱管出口端より上流位置におけるCHF状態開始が観測されている．他方，Groeneveld[17-19]は，R-12の実験をおこない，$G > 4050\ kg/m^2s$ で上流CHFの発生がみられることを報告しているが，彼の実験条件（表1参照）はHP領域のものでなく，かつ二相混合入口条件（mixed inlet condition）のときだけに観測される点で上記の諸研究とは相違している．またGroeneveld[17][19]は，Gが高いとき，熱流束の上昇と共に壁温が徐々に上昇する低速CHF（slow CHF）がみられることを述べているが，前述のPeskovら[1]は高圧下，入口サブクール度が低いときCHF温度上昇が小さくなることを報告している．

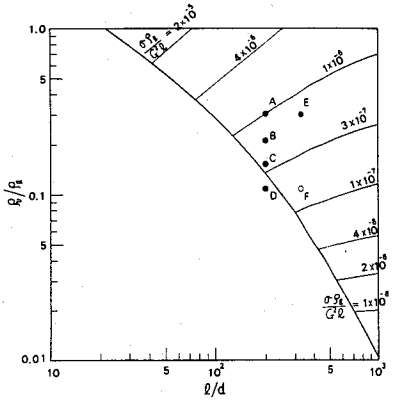

図1．HP領域の発生条件，および本報の実験条件

さて本研究では，上述以外には研究がまだ必ずしも十分でなく，かつ水の実験結果しか報告されていないHP領域のCHFに対し，サブクール入口条件下のR-12の精密な実験をおこない，HP領域のCHFに対する因源の無次元整理式（水のデータからポめられた）および境界条件の妥当性の検討をおこなう．また同時に，質量速度Gの高い領域に対し，上流CHFや低速CHFなどの特殊状況の検討をおこなうもので，表1最下段に示す実験範囲の結果を報告する．

2．実験装置

実験装置の概要を図2に示す．循環ポンプを出るR-12サブクール液のうち，その一部が流量計を通り，電気式予熱器（容量6KVA）で入口液温を調整後，垂直試験部に流入，その上端から流出する気液二相流は圧力調整器（直径200mm，高さ650mm容器）に入る．一方，循環ポンプを出るR-12サブクール液の残りは，冷却器（向流型二重管式熱交換器で水冷却）を経たのち，その大部分はフィルタを通って循環ポンプに環流し，残りは圧力調整器にスプレー状で流入し，試験部からの蒸気を冷却凝縮するのに使われる．この圧力調整器は，水冷凝縮器および電気加熱器を内蔵しており，容器内の飽和R-12液の温度調整によって系の圧力を設定する．圧力調整器から流出する液体は，冷却器からのサブクール液と合流し，フィルタを経て循環ポンプに戻る．

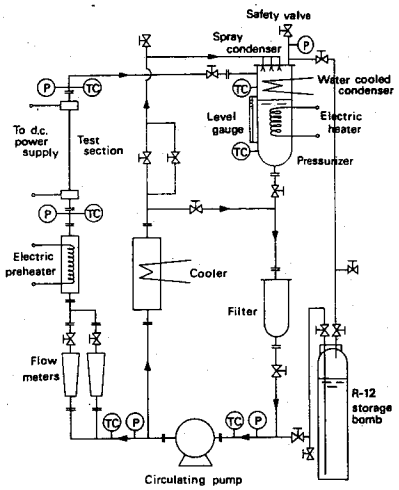

図2．実験装置（P：圧力計，TC：熱電対）

試験円管は内径 $d = 5mm$ または $3mm$（共に肉厚1mm），加熱部長さ $\ell = 1000mm$ のステンレス管で，直流電源装置から供給される直流の直接通電によって加熱される．そして図3に示すように，加熱部外壁上に11個のクロメル・アルメル熱電対（直径0.1mm）がとりつけてある．CHF検出器は，加熱管出口端付近の熱電対TC1の温度が飽和温度より約60°C以上上昇すると作動して自動的に加熱電源を切るようにセットされているが，TC1～11の各熱電対の指示温度も計測される（上流CHFや低速CHFの検出のため）．

3．実験結果

3.1 内径5mm管の場合　内径 $d = 5mm$，長さ $\ell = 1000mm$ の加熱管に対し，図1のA，B，C，Dの各点の圧力条件（$p = 34.3, 29.4, 24.5, 19.5\ bar$）において実験をおこなった．図4はその結果を示すもので，圧力pと質量速度Gが一定の条件下に，入口サブクール・エンタルピ ΔH_i による限界熱

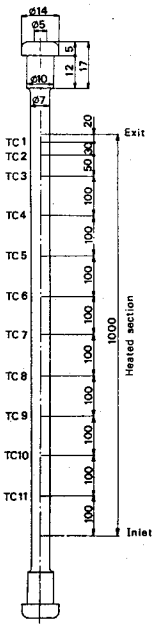

図3. 試験円管
($d=5mm$ の場合)
と熱電対の位置.
長さ単位：mm.

流束 q_c の変化を示す.

図4で○印データは，加熱管出口付近の熱電対TC1で検出される通常形式のCHFである．一方，●印データは上流CHFである．繁雑さを避けるため $p=34.3$ bar および 19.6 bar の一部の●印データだけに，CHF状態発生が検出される上流位置の熱電対番号（図3参照）を記入してあるが，Gが大きいほど上流CHFを示す ΔHi の範囲が広がり，かつ $\Delta Hi=0$ に近い状況下ほどCHF状態開始位置が加熱管入口の方へ移行する様子がみられる．上流CHFは，図4の破曲線の左側領域に含まれているので，この領域を一時的に「上流CHF領域」と名付ける．また図4の破曲線の右側領域では直線 $q_c-\Delta Hi$ 関係が成立するので，これを「直線 $q_c-\Delta Hi$ 領域」と名付ける．

図4で●印の上流CHFでは，CHF発生時の壁温上昇値が通常の上昇値（CHF検出器の設定温度上昇60°C以上）より小さくなることが観測された．さらに図4の上流CHF領域内の○印のCHF，および直線 $q_c-\Delta Hi$ 領域内の $G=1100\sim3300$ kg/m²s のCHFで $\Delta Hi=0$ の近傍のものでも，CHF発生時の壁温上昇値が通常状況より小さくなる傾向が観察された．以上の状況は，Peskovら[1]の記している状況に類似ではないかと推測される．

図5は，図4のCHF発生直前において測定された試験円管前後の圧力降下 $-\Delta p$（入口圧力 p に対する比率）を参考に示すものである．

<u>3.2 内径3mm管の場合</u>　内径 $d=3$ mm，長さ1000mmの加熱管に対しては，図1のE点の圧力条件（$p=34.3$ bar）において求めた実験結果を図6に示す．図1のF点（$p=19.6$ bar）で

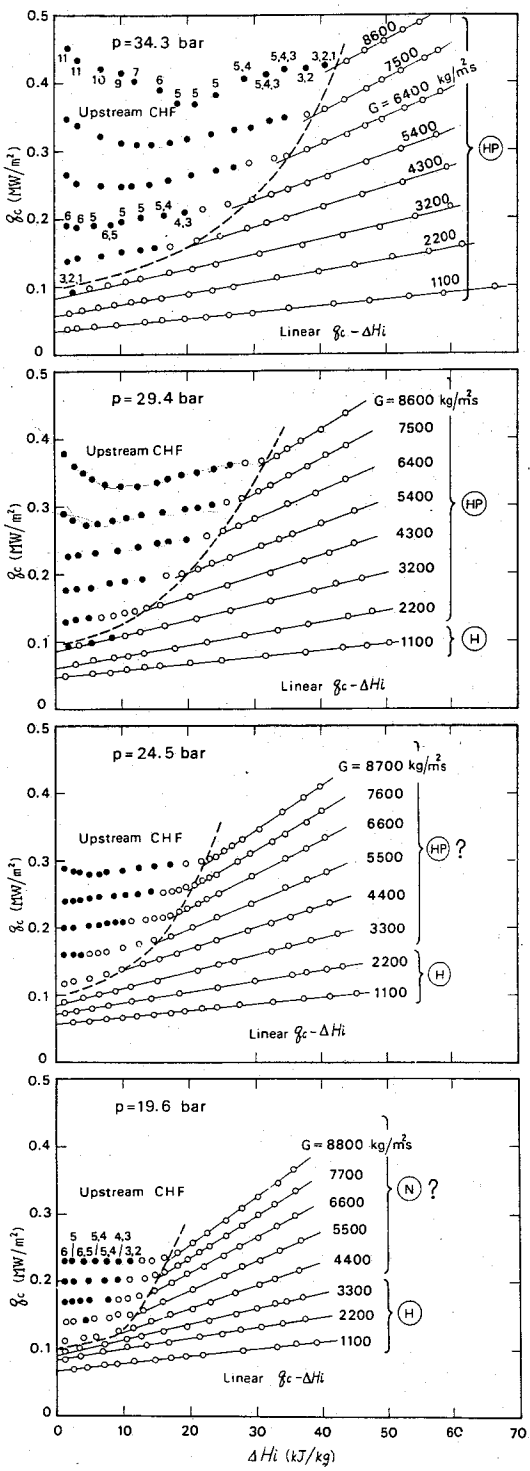

図4. 1.2 の限界熱流束 q_c の実験結果.
$d=5mm$，$l=1000mm$.

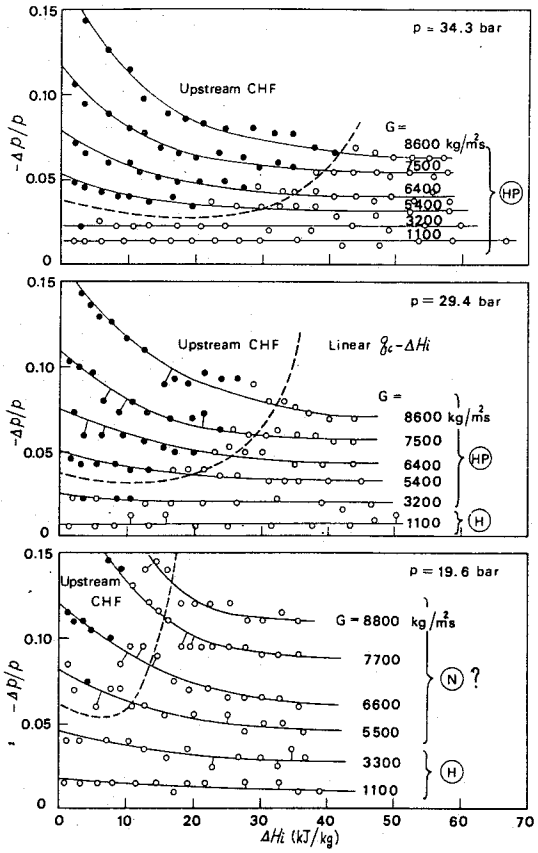

図5. R-12 のCHF発生前の加熱管前後の圧力降下 $-\Delta p$ の実験結果. $d = 5mm$, $\ell = 1000mm$.

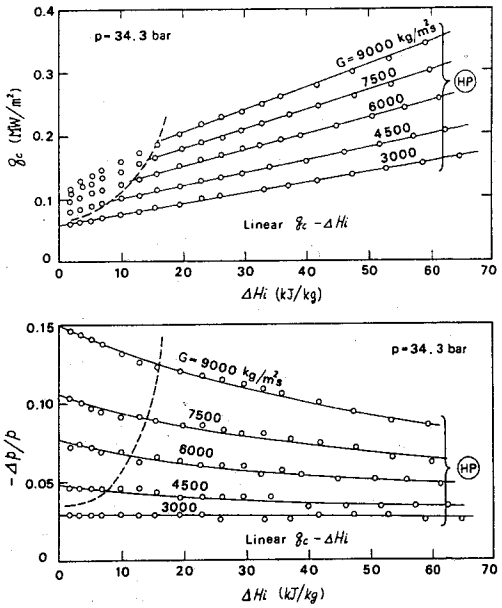

図6. R-12 の限界熱流束 g_c および加熱管前後の圧力降下 $-\Delta p$ の実験結果. $d = 3mm$, $\ell = 1000mm$.

も実験をおこない，図6とほぼ同性格の結果を得ているが，試験円管前後の圧力降下 $-\Delta p$ が非常に大きいので，ここにはデータの呈示を省略する．図6にみるごとく，$d = 3mm$ の加熱管の場合，データはすべて○印であり，測定範囲内で上流CHFは出現しない．ただし破曲線の左側に $g_c - \Delta H_i$ の関係がやや特異な領域がある．また図6の大部分のデータ（$\Delta H_i \fallingdotseq 0 \sim 50$ kJ/kgの範囲内）で，CHF発生時の壁温上昇値が通常の上昇値より小さくなることが観察された．しかし，これは管内径3mmに比べ比較的大きな肉厚1mmという状況に関係するかも知れない．

4. 分析と検討

4.1 準備 著者の一人[7]が定めた無次元整理式のうち，本章で必要なものを以下に記す．入口サブクール・エンタルピ ΔH_i のときの限界熱流束 g_c に対し

$$g_c = g_{co}\{1 + K(\Delta H_i / H_{fg})\} \qquad (1)$$

ここに H_{fg} は蒸発潜熱である．まず式(1)右辺の g_{co} に対して

H および N領域：

$$\frac{g_{co}}{G H_{fg}} = 0.10\left(\frac{\rho_v}{\rho_\ell}\right)^{0.133}\left(\frac{\sigma\rho_\ell}{G^2\ell}\right)^{1/3}\frac{1}{1+0.0031\,\ell/d} \qquad (2)$$

$$\frac{g_{co}}{G H_{fg}} = 0.098\left(\frac{\rho_v}{\rho_\ell}\right)^{0.133}\left(\frac{\sigma\rho_\ell}{G^2\ell}\right)^{0.433}\frac{(\ell/d)^{0.27}}{1+0.0031\,\ell/d} \qquad (3)$$

HP領域：

$$\frac{g_{co}}{G H_{fg}} = 0.0384\left(\frac{\rho_v}{\rho_\ell}\right)^{0.60}\left(\frac{\sigma\rho_\ell}{G^2\ell}\right)^{0.173}\frac{1}{1+0.250\,(\sigma\rho_\ell/G^2\ell)^{0.233}\,\ell/d} \qquad (4)$$

次に式(1)右辺の K については

式(2)の g_{co} に対応し：
$$K = \frac{5}{6}\,\frac{0.0124 + d/\ell}{(\rho_v/\rho_\ell)^{0.133}\,(\sigma\rho_\ell/G^2\ell)^{1/3}} \qquad (5)$$

式(3)の g_{co} に対応し：
$$K = 0.416\,\frac{(0.0221 + d/\ell)\,(d/\ell)^{0.27}}{(\rho_v/\rho_\ell)^{0.133}\,(\sigma\rho_\ell/G^2\ell)^{0.433}} \qquad (6)$$

式(4)の g_{co} に対応し：
$$K = 1.12\,\frac{1.52\,(\sigma\rho_\ell/G^2\ell)^{0.233} + d/\ell}{(\rho_v/\rho_\ell)^{0.60}\,(\sigma\rho_\ell/G^2\ell)^{0.173}} \qquad (7)$$

なお式(3)に関連し，H領域とN領域の境界は次式で与えられる．
$$\sigma\rho_\ell/G^2\ell = \{0.77/(\ell/d)\}^{2.70} \qquad (8)$$

4.2 実験データの特性領域区分　図4，6のCHF実験データに付した記号 (HP), (H), (N) はそれぞれHP領域，H領域，N領域を意味し，これらは図1および式(8)の判定条件で定めたものである．ただし図4の $p = 24.5$ bar, 19.6 bar の実験は，図1のC，Dよりそれぞれ対応し，HP領域可能範囲の境界線（この付近は厳密には遷移域になる）の近くにある．従って，質量速度 G が大になったとき，HP領域，N領域と必ずしも判定し難い条件下にあるので，(HP) および (N) に疑問符？を付してある．

図4でH領域の g_c は，ΔH_i の測定全域で直線 g_c-ΔH_i 関係を示し，N？領域の g_c は，ΔH_i の測定全域に対しては直線 g_c-ΔH_i 関係が成立しない．これらは著者の一人[5]がH，N各領域に付与した特性に合致している．他方，図4，6でHP領域の g_c は，$G < 3800$ kg/m²s では ΔH_i に対し直線関係を持つが，$G > 3800$ kg/m²s の時は，図4，6の破曲線の右側領域では非直線関係を示す．さきに著者の一人[5]は，Chojnowski-Wilson[9] の実験データ（これは $G < 1500$ kg/m²s），Becker[2] のCHF経験公式を参考に，HP領域に対し直線 g_c-ΔH_i 関係という特性を付与した．しかし図4，6の前述の結果によれば，HP領域でも G が非常に大きくなると，ΔH_i の小さい領域で g_c-ΔH_i の直線関係はくずれるものであることがわかる．

4.3 上流CHF　図7は，Watersら[13]が直径 11.2 mm，長さ 3660 mm の加熱管で圧力 $p = 104$ bar の水に対して求めたCHFの実験データ［文献[13]の表1 (a), (b), (c), (e) の全データ］を示し，白抜きデータが管出口端で発生する通常のCHF，黒ぬりデータが上流CHFである．このWatersらのデータは，Collierがその有名な著書[20]に引用し，超高質量速度（ultra high mass velocity）に基づく当則効果と説明しているものである．

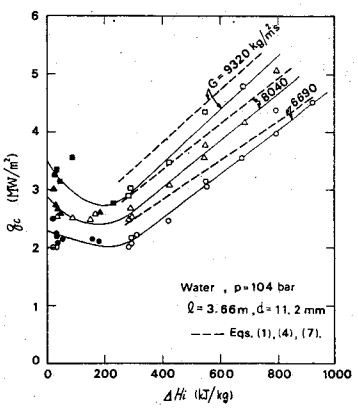

図7．水の限界熱流束 g_c の実験結果（Watersら[13]）．$\rho_v/\rho_\ell = 0.0853$，$\ell/d = 327$．

さて図7の実験条件（$\rho_v/\rho_\ell = 0.0853$，$\ell/d = 327$）は図1の判定条件によればHP領域* に入るが，HP領域可能範囲の境界線の近くにあり，その意味で図4，$p = 24.5$ bar の実験条件と性格が共通である．そして両方の実験結果が，g_c-ΔH_i の関係，および上流CHFの発生状況において互に非常に似た性格を持つことが見られる．これは図4の結果がR-12のみの特性でなく水でも実現することを示す．なお表1にみるごとく，Matznerら[14]，Meliro[15]，Meliro-Ahmad[16] の実験も，ℓ/d，ρ_v/ρ_ℓ，G に関しWatersら[13]のものと同程度であり，従ってその上流CHFの発生状況は本報の場合に似ていると考えてよいであろう．一方，Groeneveld[17],[18] の観察した上流CHFは，1章に述べたごとく実験条件がHP領域になく，かつ二相混合入口条件（$\Delta H_i < 0$）にだけ観測されるものであり，従って現状ではまだ上記の上流CHFと直接の関係をつけることは出来ない．

＊ 文献[5]の図13では，古い判定条件のためN領域に入れられている．

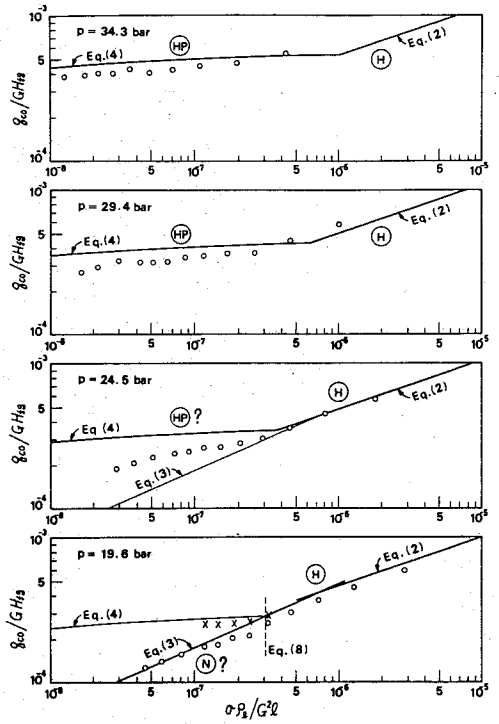

図8. q_{co}の実測値と予想値の比較
(R-12, d=5mm, ℓ=1000mm).

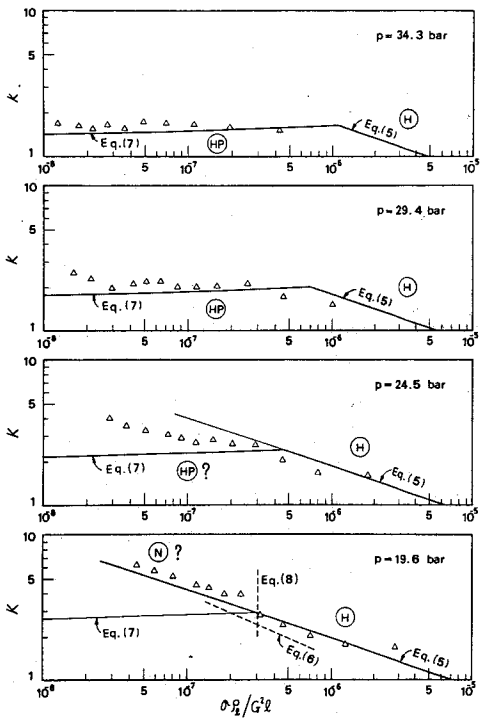

図9. K の実測値と予想値の比較
(R-12, d= 5mm, ℓ= 1000mm).

<u>4.4 直線 q_c-ΔH_i 領域のCHF</u>　図4, 6の直線 q_c-ΔH_i 領域内の G=一定の各直線から，式(1)右辺の q_{co} および K の実験値が求められ，それらを式(2)～(4)による q_{co} の予想値，式(5)～(7)による K の予想値（ここに物性値は管入口状態の値を用いた）と比較すると図8～10の結果が得られる．

まず図8で○印は，図4から上記の方法で求めた q_{co} の実験値*で，q_{co}/GH_{fg} 対 $\sigma\rho_e/G^2\ell$ の関係として示すものである．一方，図8の太線は図1の判定条件で定めた特性領域の q_{co} の予想値を示す．ここで p=34.3, 29.4 bar の2条件（図1のA, B点）では実験値と予想値がかなり良く合っている．一方，p=24.5 bar（図1のC点）では，$\sigma\rho_e/G^2\ell < 3\times10^{-7}$ の範囲で実験値は，式(4)と式(3)の予想の中間に現われ，HP領域からの遷移特性を示す．さらに p=19.6 bar（図1のD点）では，$\sigma\rho_e/G^2\ell < 3\times10^{-7}$ の範囲で実験値は式(3)の予想値と合い，HP領域からの離脱を示している．ただし，このことは上記 p=19.6 barの実験値がN領域のものということを意味するものではないことを付記する．なぜなら，図4の p=19.6 bar の図面で破曲線の左側のデータから

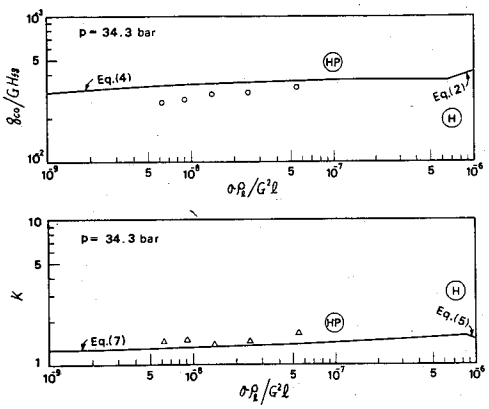

図10. q_{co} と K の実測値と予想値の比較（R-12, d=3mm, ℓ=1000mm）

* 繁雑さを避けるため，図4には図示してないが，G=1600, 2700, 3800（または3900），4800（または4900）kg/m²s の時のCHFのデータもあり，図8, 9にはそれらから求めた q_{co}, K も図示してある．

$\Delta H_i \to 0$ の外挿で求めた g_{co} は，図8の $p=19.6$ bar の図面で×印の位置に現われ，式(3)の予想値（N領域では×印がこれに一致する筈）から外れるからである．

次に図9の△印は，図4から求めたKの実験値を $\sigma\rho_\ell/G^2\ell$ の関数として示すもので，式(5)，(7)の予想値との関係において，図8の場合とまったく同じ特性を示している．なお図9で $p=19.6$ bar の場合，$\sigma\rho_\ell/G^2\ell < 7\times10^{-7}$ の範囲でKの実験値は，式(6)より式(5)の予想値に近いことが注目される．つまり図8の $p=19.6$ bar で $\sigma\rho_\ell/G^2\ell < 7\times10^{-7}$ の範囲の g_{co} は式(3)に合い，従ってそれに対応するKは式(6)で予想される筈のものが，ここでは成立しないことになる．しかし上記と同じ状況がすでに著者の一人の研究[21]（その3.1節を見よ）でも見出されており，従って式(6)を除去し，代りに式(5)を使う方がよいようである．

図10は，図6（図1のE点に対応）の各直線から求めた g_{co} とKの実験値に関し，式(2)，(4)，(5)，(7)の予想値との比較を示す．ここに実験値と予想値はかなりよく合い，その状況は図8，9の $p=34.3$，29.4 barの場合と同様である*．

最後に，図7に示す4本の破線は，与えられた実験条件に対し，HP領域の整理式(4)，(7)および式(1)を用いた時の g_c の予想値である．実験値と多少の偏差（15%以内）がみられるが，4.3節に記したごとく図7の実験条件はHP領域可能範囲の境界線近くにあり，従って式(4)，(7)の予測には誤差があり得ることに注意する要もある．

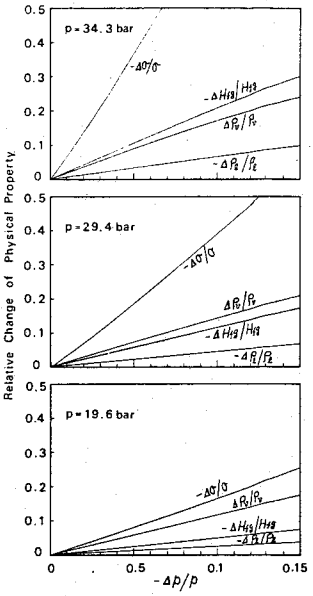

図11．加熱管の入口，出口間の圧力降下 $-\Delta p$ に対応するR-12の物性値の変化．
（ρ_ℓ：液体密度，ρ_v：蒸気密度，H_{fg}：蒸発潜熱，σ：表面張力）

4.5 HP領域における加熱管前後の圧力降下と物性値変化　加熱管前後の圧力降下 $-\Delta p$（入口圧力 p に対する比率）を図5，6に示したが，これに対応するR-12の物性値変化（入口の値に対する比率）は図11のごとくである．さて式(4)，(7)によれば，HP領域のCHFに影響する物性値は，H_{fg}，ρ_v/ρ_ℓ，$\sigma\rho_\ell$ であるが，圧力による ρ_ℓ の変化は比較的小さい（図11参照）ので無視することにすれば，H_{fg}，ρ_v，σ が主として影響することになる．さて図11によればHP領域（$p=34.3$，29.4 bar）では σ の変化が非常に大きい．しかし図8～10にみられるごとく，HP領域における g_{co} やKの値は，$\sigma\rho_\ell/G^2\ell$ による変化，すなわち σ による変化が非常に小さいから，その影響は近似的に無視してよいであろう．一方，4.2節に記したところの $G<3800$ kg/m²s の状況（ここではHP領域で ΔH_i の全測定域に対し直線 g_c-ΔH_i 関係が成立する）だけを対象にすることにすれば，HP領域で圧力変化は $-\Delta p/p < 0.03\sim0.04$，従って物性値 H_{fg}，ρ_v の変化比率は約2.5%以下とみることが出来る．そしてHP領域CHFに対するBecker，Lee，甲藤の各整理式（すべて直線 g_c-ΔH_i 関係を想定している）の本来の適用領域は上記の対象範囲にあるというべきかも知れない．

4.6 CHF発生時の小さい壁温上昇　CHF発生時の壁温上昇が通常の上昇値より小さい場合の発生状況は3.1節および3.2節に記した．なお本報で観察された状況は，低速CHF（slow CHF）と呼ぶほど特異なものではなかったことを付記する．

5. 結論

一様加熱垂直円管内の強制流動沸騰のCHFに関し，著者の一人[7]はL，H，N，HPの4特性領域の区分をしたが，特にHP領域については水以外の実験データが見当らず，また実験数も限られている．本報

* 図8，10ともに，HP領域の g_{co} の実験値が式(4)の予想値よりやや低く出る傾向がみられる．これがR-12特有の性格かどうかについては今後の研究が必要である．

では，このHP領域に対して，R-12を用い，かつ質量速度Gの大きな範囲まで実験をおこなって下記の事柄を明らかにした．

(i) HP領域（およびそれに近い領域）では，ほぼ質量速度 $G < 3800\ kg/m^2s$（本報の場合の数値）の範囲では，$\Delta H_i > 0$ の全域で直線 $q_c - \Delta H_i$ 関係が得られる．しかし $G > 3800\ kg/m^2s$ では，ΔH_i が零からある値までにわたる範囲（Gが大なるほど広くなる）内で当則的な $q_c - \Delta H_i$ 関係を示し，それより ΔH_i の値が大きい範囲で直線 $q_c - \Delta H_i$ 関係を示す．

(ii) 前記の $G < 3800\ kg/m^2s$，$G > 3800\ kg/m^2s$ の両範囲を通し，直線 $q_c - \Delta H_i$ 関係を持つ領域のCHFは，すべて加熱管出口端で開始するところの通常形式のCHFである．そして本報で求めたR-12の実験データは，HP領域において，甲藤[7]のCHF無次元整理式（水のデータから求めたもの）の予想値とかなりよく一致し，またHP領域可能範囲の境界線付近の状況ではHP領域からの遷移特性を明らかに示す．

(iii) 前項(ii)に記した結果（水，R-12のデータが甲藤の無次元整理式に合う）から考えて，HP領域の直線 $q_c - \Delta H_i$ 関係を持つCHFに対し，表面張力は僅かな影響しか持たぬと推測される．一方，このCHFに影響すると考えられる物性値（H_{fg}，ρ_v など）については，もしG<3800 kg/m^2s の範囲に限ると，加熱管前後の圧力降下に伴う物性値変化率を約25％（本報の場合の数値）以内におさえることを得る．

(iv) 前記の(i)項に記したところの $G > 3800\ kg/m^2s$ における当則的な $q_c - \Delta H_i$ 関係の領域のCHFは，内径5mmの加熱管の場合，上流CHFであった．しかし内径3mmの加熱管になると，当則的な $q_c - \Delta H_i$ 関係の領域も狭く，かつ上流CHFは発生しなかった．なお上流CHFにおいては，CHF発生時の壁温上昇値が通常の場合の上昇値より低くなる傾向が観察された．

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〔710〕一様加熱流路で「加熱長さ」/「加熱相当直径」比の小さい場合の限界熱流束

論文前刷

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1，序

最近，著者の一人は，サブクール入口条件下の一様加熱垂直流路の限界熱流束（CHF）に対し既存実験データを分析し，円管[1,2]，二重管[3,4]，矩形管[5]の各場合に対しそれぞれ無次元整理式を導くと共に，データとの比較を示した．そこでは，CHFに対し4個の特性領域（L，H，NおよびHP）が区分され，大まかに言って，L領域のCHFは質量速度が比較的小さい場合に生じ，いわゆるドライアウト形式に近いもの；N領域のCHFは質量速度が比較的大きい場合で，いわゆるDNB形式に近いもの；H領域はLとNの中間領域；またHP領域は特に高圧時にN領域の代りに発生する領域である．なお上記L，H，Nの3領域のうち，L，H領域では限界熱流束 q_c と入口サブクール・エンタルピ ΔH_i 間に直線関係が成立するが，N領域では非直線関係になるという特色を有している．本報では以下，上記のような一様加熱流路限界熱流束のことを，簡単のため「流路CHF」と呼ぶことにする．

一方，甲藤・倉田[6]は，飽和液の一様流中の流れに平行な平板加熱面上の沸騰の実験をおこない，CHFの実験データに対し無次元整理式を導いた．以下，この形式の沸騰のCHFのことを「平板CHF」と呼ぶことにする．なお，上記論文[6]には，内管加熱二重管で，加熱流路長さ l と加熱相当直径 d_{he} の比 l/d_{he} が非常に小さい場合，そのCHFのデータが上記の「平板CHF」の整理式に一致することが記され，また著者の一人の別の研究[5]においては，矩形管のCHFのデータにも，上と同様の性格が見られることが示されている．

表1．l/d_{he} がほぼ10以下の流路のCHFに対する実験の条件

	Source	Fluid	l (cm)	d_o or e (cm)	d_i or b (cm)	d (cm)	l/d_{he} or l/d	$\frac{\rho_v}{\rho_l} \times 10^2$	Type* of CHF	Remark
Internally heated annulus	Sakurai[7]	Water	7.00	3.80	0.120	---	0.0582	0.0868	∿ Plate(1)	
	Andrews et al.[8]	Acetone	7.62	2.09	0.604	---	1.15	0.291	∿ Plate(1)	
	Andrews et al.[8]	Toluene	7.62	2.09	0.604	---	1.15	0.385	∿ Plate(1)	
	JSME[9]	Water	30.0	1.90	1.00	---	11.5	4.05	∿ Channel(2)	
	Tolubinskiy et al.[10]	Water	22.0-33.0	1.30	0.500	---	7.64-11.5	10.3	∿ Channel(2)	
	Tolubinskiy et al.[11]	Water	10.0	1.30	0.900	---	10.2	16.0	∿ Channel(2)	
	Tolubinskiy et al.[12]	Water	10.0	1.30	1.00	---	14.5	16.0	∿ Channel(2)	
Round tube	Bergles[13]	Water	1.19	---	---	0.238	5.00	0.124	∿ Channel(3)	
	Sterman et al.[14]	Benzene	10.0	---	---	1.00	10.0	0.675	∿ Channel(3)	
	Giarratano et al.[15]	Helium-I	1.09	---	---	0.213	5.14	15.1	∿ Channel(3)	
	Giarratano et al.[15]	Helium-I	1.67-2.33	---	---	0.213	7.86-10.9	21.1	∿ Channel(3)	
Rectangular, one-side heating	Goto[16]	Water	1.00-1.50	0.300-0.500	1.80	---	0.417-0.694	0.0624	∿ Plate(4)	Horizontal, Top heated
	Yücel-Kakaç[17]	Water	12.7	0.635	0.952	---	3.07	0.0624	∿ Plate(4)	Vertical upflow
	Yücel-Kakaç[17]	Water	12.7	0.635	0.652	---	4.47	0.0624	∿ Plate(4)	Horizontal, Bottom heated
	Isshiki et al.[18]	Water	30.0	1.10	1.70	---	6.02	0.0624	∿ Channel(4)	Horizontal, Bottom heated

＊ (1) 文献[6]の図3参照．(2) 文献[4]の図3参照．(3) 文献[2]の図1参照．(4) 文献[5]の図5参照．

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ところで前述の諸研究[2,4,5,6]で分析した実験データ（ただし物性値の信頼性の面からモノイソプロピルビフェニルのデータは省略する）のうち，特に ℓ/d_{he} の値がほぼ10より小さいものの実験条件を示すと表1のごとくなり，ここで「平板CHF」の状況に近いものには～Plate，「流路CHF」の状況に近いものには～Channel の記号を付してある．いま条件を一定にするため，垂直流路の場合のみを考える（すなわち表1で水平流路の実験を除く）ことにすれば，表1から $\ell/d_{he}<3$ で「平板CHF」，$\ell/d_{he}>5$ で「流路CHF」を生ずるという結果になる．

しかし，CHFは液体からの蒸気発生を伴なう現象であり，そこに生ずる二相流の流動様式とも関連を有するから，「流路CHF」と「平板CHF」の間の遷移が ℓ/d_{he} の大きさだけに支配されると即断はできない．本報ではこの問題に対しておこなった研究の結果について報告する．

記号　ℓ：矩形断面の巾（≒加熱側壁の巾），d：加熱円管の内径，d_{he}：加熱相当直径＝（4×流路断面積）/（加熱周辺長），d_i：二重管の内径（加熱内管の外径），d_o：二重管の外径（非加熱外管の内径），e：矩形断面の高さ，G：質量流量，H_{fg}：蒸発潜熱，ΔH_i：入口サブクール・エンタルピ，l：加熱長さ，p：絶対圧力，q_c：限界熱流束，q_{co}：基礎限界熱流束（$\Delta H_i=0$ の時の q_c），$[q_{co}]_d$：流路CHFの q_{co} の予測値，$[q_{co}]_p$：平板CHFの q_{co} の予測値，$u_{i,min}$：最小入口速度，ρ_l：液体密度，ρ_v：蒸気密度，σ：表面張力，x_{ex}：出口クオリティ（CHF発生時）．

2．実験装置

図1に実験装置を概念的に示す．試験液体R-113（～大気圧）の貯蔵タンクは，水冷クーラおよび電気加熱器を内蔵しており，温度調整器によってタンク内液温を設定値に保つ．循環ポンプを出る液体の大部分は，バイパスIを経て貯蔵タンクに還流し，残りが試験部の方へむかうが，試験部への流量の微調整は，バイパスIIの弁，およびオリフィス後の弁の調整によっておこなう．オリフィスは大小2個から成り，流量の大小によって，どちらか一方を使用する．試験部を出た液体は，気液分離タンクに入り，ここで分離された液体は貯蔵タンクに戻り，また蒸気は水冷凝縮器で凝縮後，貯蔵タンクに戻る．

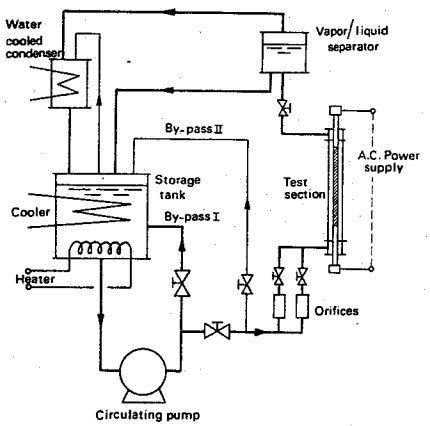

図1．実験装置

試験部の二重管は，耐熱ガラス製の外管，およびステンレス鋼製の内管（肉厚0.5 mm）から成り，内管はその上下端を同じ外径の銅棒に銀ロウ付で接続してあり，直接通電（50サイクル低電圧交流）により加熱される．図2は，二重管の助走部への流入流路付近の構造の詳細を示し，なお二重管の出口付近の構造もほぼこれに等しいものである．加熱される内管の端部から銅棒への熱伝導損失は，内管の肉厚が0.5 mmと薄いせいで，CHFの実験結果に重大な誤差を与えるほど大きくないことを確かめている．CHF発生は，加熱内管の後流端近くの外壁上にスポット溶接した熱電対の温度急上昇によって検出し，このときCHF防止装置により自動的に電気入力の遮断をおこなうようになっている．

3．実験結果と分析

図3は，圧力 1.17 bar（$\rho_v/\rho_l=0.00563$），および加熱部長さ $\ell=150$ mm，内管外径 $d_i=10.0$ mm，外管内径 $d_o=16.0$ mm（$\ell/d_{he}=9.60$）の場合の実験結果で，質量速度 G 一定の条件下に入口サブクール・エンタルピ ΔH_i による限界熱流束 q_c の変化を示す．これらのデータから $\Delta H_i\to 0$ の外挿により求めた基礎限界熱流束 q_{co}（$\Delta H_i=0$ の時の q_c）実測値を手がかりに分析をはじめる．

さて「流路CHF」の q_{co} に対しては，内管加熱二重管の場合，著者の一人[3,4]により下記の無次元型

理式が与えられている．すなわち，H領域およびN領域に対し

$$\frac{q_{co}}{GH_{fg}} = 0.12\left(\frac{\rho_v}{\rho_\ell}\right)^{0.133}\left(\frac{\sigma\rho_\ell}{G^2\ell}\right)^{1/3}\frac{1}{1+0.0081\,\ell/d_{he}} \qquad (1)$$

$$\frac{q_{co}}{GH_{fg}} = 0.22\left(\frac{\rho_v}{\rho_\ell}\right)^{0.133}\left(\frac{\sigma\rho_\ell}{G^2\ell}\right)^{0.433}\frac{(\ell/d_{he})^{0.171}}{1+0.0081\,\ell/d_{he}} \qquad (2)$$

式(2)はH，N領域にまたがるが，2領域の境界は次式で与えられる．

$$\sigma\rho_\ell/G^2\ell = \{0.0206/(\ell/d_{he})\}^{1.71} \qquad (3)$$

また「平板CHF」のq_{co}に対しては，甲藤・倉田[6]により次の無次元整理式が与えられている．

$$\frac{q_{co}}{GH_{fg}} = 0.186\left(\frac{\rho_v}{\rho_\ell}\right)^{0.559}\left(\frac{\sigma\rho_\ell}{G^2\ell}\right)^{0.264} \qquad (4)$$

そして前に求めたq_{co}の実測値を，式(1)～(4)によるq_{co}の予測値と比較した結果が図4である．白丸で示すq_{co}の実測値は，「流路CHF」および「平板CHF」の予測値のほぼ中間に現われる．もし仮に，これらの実測点が，式(1)，(2)による「流路CHF」の予測線上にあったとすると，図3においてG＝260～2200 kg/m²sの条件下の実験値はH領域にあり，従ってその属性であるところの直線$q_c-\Delta H_i$関係を示すべきである．しかし現実には，図3において，G＝260～2200 kg/m²sのデータも非直線$q_c-\Delta H_i$関係を示しており，従って純粋の「流路CHF」からは性格が偏っていることを明示している．

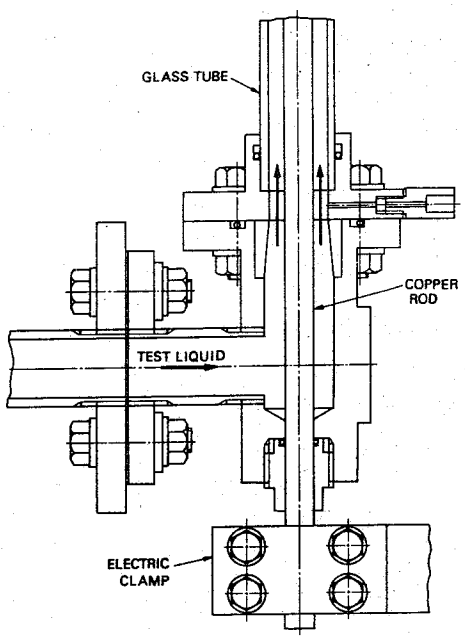

図2．二重管流入部付近の構造詳細

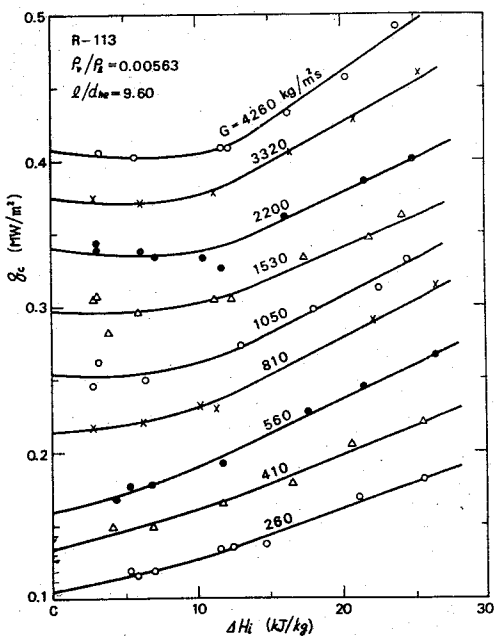

図3．CHF実験結果．R-113，ℓ＝150 mm，d_i＝10.0 mm，d_o＝16.0 mm．

上記の事柄を，流動様式の面から検討すると以下のごとくである．すなわち図4の実測点のうち，A，B，C，Dの記号を付した実験の状況下の二重管内の流れの状況写真（$\Delta H_i \fallingdotseq 0$でCHF発生直前の状況）を図5に示すが，ここで加熱部出口端付近の流動様式は，気泡流（ただし流路断面の寸法に比べれば未だ小さい気泡合体を伴なう）の状況に近いように思われる．前に著者の一人[19]が「流路CHF」の一つ，すなわち垂直円管内のCHFに対し，CHFの特性領域と，管出口の流動様式との関係を調べた結果によれば，H領域は環状流に伴っている．しかし図5の流動様式は，これと性格がかなり異なるものであり，従って「平板CHF」の場合の流れの性格をあわせ持つ状況にあるといえるであろう．

次に図6は，p＝1.36 bar（ρ_v/ρ_ℓ＝0.00654），およびℓ＝100 mm，d_i＝10.0 mm，d_o＝20.1 mm（ℓ/d_{he}＝3.29）の場合の実験結果を前と同様の形で示すものである．ただしこの場合は，図3のよう

なΔHiによるgcの変化は求めておらず，ΔHi≒0 のgcの測定（それを図6に示す）だけをしているので，gcoの値は図6の○印のデータ点より多少低くなるべきであろう．それを考慮しつつ図6の結果を図4と比較すると，実験点は「平板CHF」の方へ少し近づいているとみてよいであろう．なお図6でA，B，C，Dの記号を付した実験点の状況下の流れの状況写真（ΔHi≒0 でCHF発生直前）を図7に示す．これによれば，「平板CHF」に近いA点では，特に気泡流に近く，また蒸気は加熱面近傍に，外壁面近くに液体が流れる傾向が観察されるように思われる．

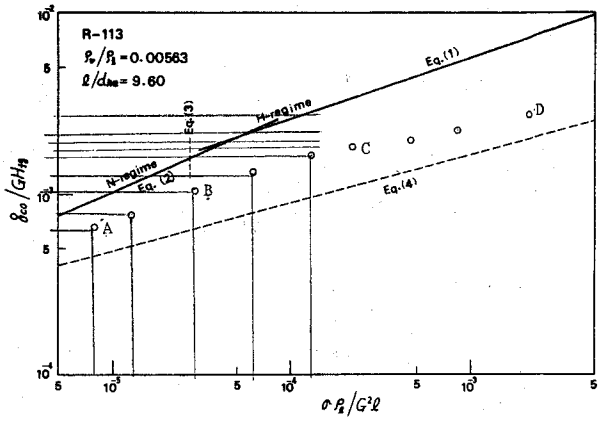

図4．gco実測値と，「流路CHF」，「平板CHF」予測値の比較

4．流路CHFと平板CHFの発生に関するルール

前章に記した内管加熱二重管の実験結果によれば，l/dhe = 9.60 の場合（図4）でも，「流路CHF」と「平板CHF」の中間状況を示していて，これは1章で表1から導いた結果（l/dhe < 3で「平板CHF」，l/dhe > 5で「流路CHF」を生ずる）に明らかに合致しない．そこで，表1に示した実験のうち，二重管および円管のものについて，図4，6と同じく「流路CHF」および「平板CHF」の予測値との比較を試みると，図8，9の結果を得る．ここに図8は内管加熱二重管，図9は円管の場合である．図9の場合，「流路CHF」の無次元整理式は，二重管の場合の式(1)～(3)に対応するものとして次の3式が与えられる[1]．

$$\frac{g_{co}}{GH_{fg}} = 0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3} \times \frac{1}{1+0.0031\,l/d} \qquad (5)$$

$$\frac{g_{co}}{GH_{fg}} = 0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3} \times \frac{(l/d)^{0.27}}{1+0.0031\,l/d} \qquad (6)$$

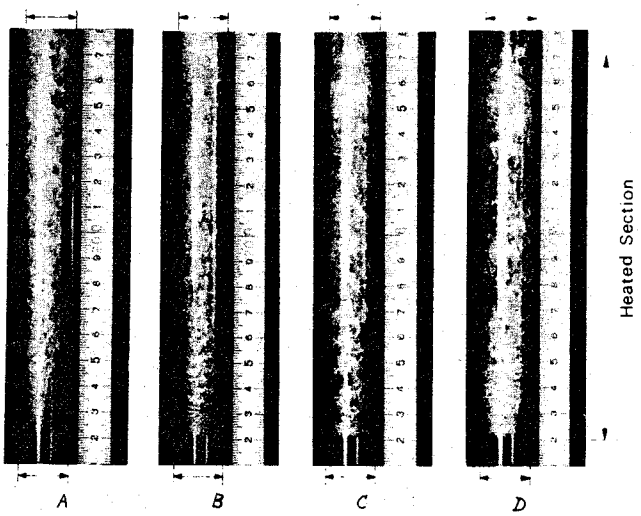

図5．二重管内の流動様式（A，B，C，Dは図4の実験点に対応）．ガラス管外径（印）20.0mm，内径16.0mm，加熱内管径10.0mm．

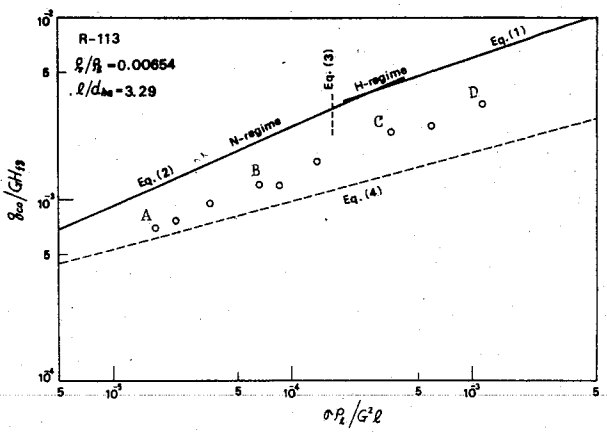

図6．gco実測値と，「流路CHF」，「平板CHF」予測値の比較

$$\frac{\Delta P_e}{G^2 l} = \left(\frac{0.77}{l/d}\right)^{2.70} \quad (7)$$

さて図8の場合，上方の2図では実線（流路CHFの予測値）が破線（平板CHFの予測値）より高く，中央図では両者がほぼ等しく，下方の2図では逆に破線が実線より高い．そして黒矢で示す実測点は，上方の2図（ここでは $l/d_{he} \leqq 1$）では破線の近くに現われるが，他の3図では実線の近くに現われるとみなすことが出来る．

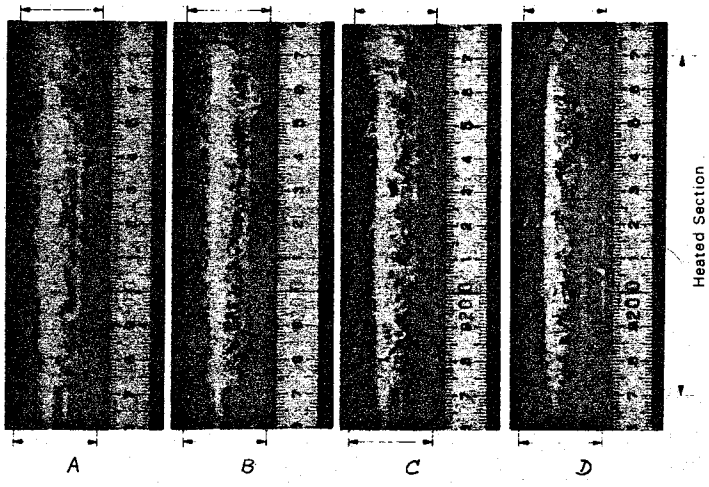

図7. 二重管内の流動様式（A, B, C, Dは図6の実験点に対応）．ガラス管外径(公称) 25.0mm，内径 20.1mm，加熱内管径 10.0mm．

他方，図9の場合，最上段図では実線が破線より高く，第二図では実線，破線がほぼ等しく，下方の2図では破線が実線より高い．そして実測点は，図9のすべての図面を通して，実線の近くに現われているとみてよいであろう．

さて，「流路CHF」，「平板CHF」の発生に関して，図4，6，8，9の諸結果のすべてに適用するルール（少くとも暫定的な）を考えてみると次のごとくなる．すなわち，式(1)，(2)，(5)，(6)による「流路CHF」の g_{co} の予測値を $[g_{co}]_C$，式(4)による「平板CHF」の g_{co} の予測値を $[g_{co}]_P$ と記すことにすれば

(i) 圧力（すなわち P_v/P_l）が高くて $[g_{co}]_P \geqq [g_{co}]_C$ となる場合は，常に「流路CHF」が生ずる．

(ii) 圧力（すなわち P_v/P_l）が低くて $[g_{co}]_P < [g_{co}]_C$ となる場合は，l/d_{he} が大きいと「流路CHF」が生ずるが，l/d_{he} が減少するに従い「流路CHF」から離れて「平板CHF」の状況に近づく．「流路CHF」から離れる時の l/d_{he} の値はまだ確定していないが，ほぼ1～10の程度の値で，流路断面の形状で異なるもののごとくである．

なお，上記のルール(i)，(ii)を導くのに基礎とした図4～9のCHF実験データは，ほぼ $\Delta P_e/G^2 l < 10^{-3}$ の範囲にあり，質量速度 G が比較的大きい状況下のものである．また図4～9の縦軸 g_{co}/GH_{fg} の値に対し，出口クオリティ χ_{ex} は

$$\chi_{ex} = (g_{co}/GH_{fg}) \times (4l/d_{he})$$

で与えられる故，l/d_{he} の値が小さい時 χ_{ex} の値は零に近い．同時に，上記の実験データに対応する「流路CHF」も，主としてN領域［式(2)または式(6)］からH領域にかかるという状況下にあり，従ってこの「流路CHF」の流動様式（文献

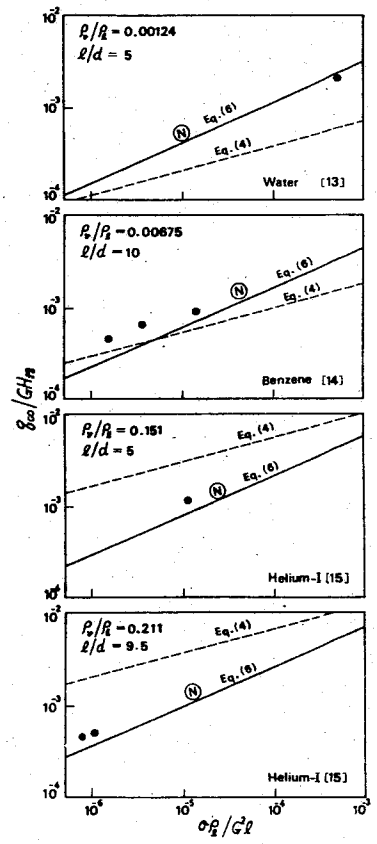

図9. g_{co} 実測値と，「流路CHF」，「平板CHF」予測値の比較．

（図8は次頁にあり）．

[19]参照）は「平板CHF」の流動様式に比較的関連を持つものといえる．すなわち，前記のルール(i)，(ii)は，「流路CHF」と「平板CHF」が相互に関連を持ち易い状況下に成立しているということが出来る．

5．質量速度が小さい場合

次に質量速度Gが小さく（たとえば $\sigma\rho_l/G^2l > 10^{-2}$ ），対応する「流路CHF」がL領域ないしそれに近い状況になる場合について検討しておく必要がある．

前に著者の一人は，垂直円管のCHFデータの無次元整理に関する最初の報告[20]において，VL領域（very low mass velocity regime）という特殊状況について記した．すなわち質量速度Gが非常に小さく，かつ管長 l が相当に小さいと，基礎限界出流束 q_{co} が，下記の通常のL領域の q_{co} の無次元整理式[1]から相当な偏りを示すという状況である．

$$\left.\begin{array}{l} q_{co}/GH_{fg} = C\,(\sigma\rho_l/G^2l)^{0.043}/(l/d) \\ \text{ここに } l/d < 50 \text{ では } C = 0.25;\ l/d > 150 \\ \text{では } C = 0.34\ ;\ \text{また } 50 < l/d < 150 \text{ では} \\ C = 0.25 + 0.09\{(l/d) - 50\}/100 \end{array}\right\} \quad (8)$$

そして，その具体例としてLewisら[21]の液体水素，Hewittら[22]の水の実験データを示した（実験条件については表2参照）．

しかし上記のLewisらの液体水素のデータについては，図9と同様な形でCHFの予測値と比較を試みたところ図10の結果が得られた．すなわち図10によれば，Lewisらのデータは，l/d の減少と共にL領域［式(8)］の予測値から確かに偏倚するが，しかしH領域［式(5)］の予測値にほぼ一致するものであって，前述のVL領域と称する特殊状況とは無関係なことが判明した（4章の終りに記した「流路CHF」，「平板CHF」発生ルールも成立している）．著者の最初の報告[20]では，まだCHF研究の初期段階でCHF特性領域の区分について不明確さが残っていたため，この誤りを生じたものと考えられる．

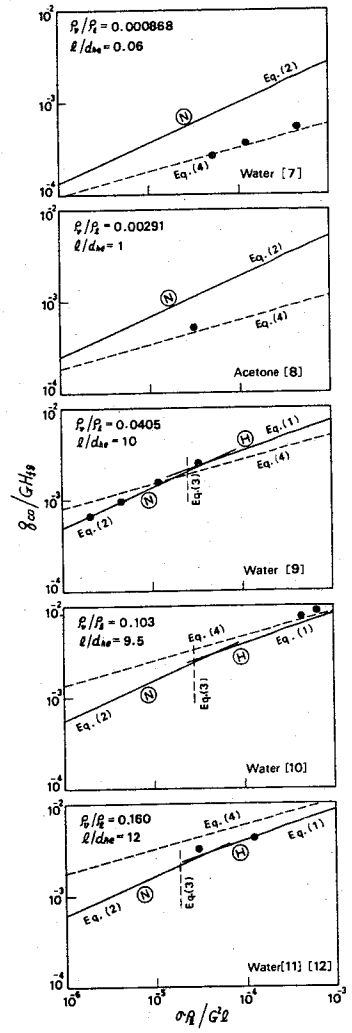

図8．q_{co} 実測値と，「流路CHF」「平板CHF」予測値の比較．

一方，前述のHewittらの水のデータについては，図11の上段の図が与えられる．これによれば，$l/d \geqq 49$ のデータは比較的よく式(8)の予測値に合うが，$l/d = 25$ および33のデータは明らかに式(8)の予測値と大きく相違する．そのうえ，Hewittらの論文[22]の図2・7によれば，$l/d \leqq 66$ では q_c と ΔH_i の間に変則的な関係が示されている．従って，これらからみて，図11で l/d が小さいとき，CHFは明らかに特殊状況（VL領域）に入ると結論してよい．

次に図11の中央の図は，Barnardら[23]のR-113に対するデータで，ここでは $l/d \geqq 17$ のデータは予測値とほぼよく合うが，$l/d = 8.5$ のデータが式(5)の予測値と大きく相違している．また図11の下段の図は，Pokhvalovら[24]のベンゼンのCHFデータで，$l/d = 16$ の条件下，図11の上段，中央の図のデータと定性的に同様な特性を示している．そしてPokhvalovらは，質量速度Gが低いとき，プール沸騰CHFに対するKutateladze－Zuberの整理式に近い特性を示すと述べており，上記のベンゼンのデータが図11下段図の特性を示す主原因は浮力の影響と考えられる．またこれから類推して，図11の上段，中央の図にみ

られる予測値からの偏差も同じ原因によると考えてよいであろう．なお浮力の影響は，ρ_v/ρ_l の値が小さい場合ほど大きくなると考えられるが，図11の水のデータ（$\rho_v/\rho_l = 0.000909$）では $l/d = 25$～33で予測値からの偏差を示すのに対し，図11のR-113のデータ（$\rho_v/\rho_l = 0.00511$）では $l/d = 8.5$ で偏差を示し，一方，図10同一図の液体水素のデータ（$\rho_v/\rho_l = 0.0685$）では $l/d = 8.4$ でもまだ偏差を示していない事実は，これに対応すると考えてよいであろう．図11でBarnardらのR-113のデータ（$l/d = 17$）とPokhvalovらのベンゼンのデータ（$l/d = 16$）は，ρ_v/ρ_l の値がほぼ同じであり，かつ最小流入速度 $u_{i,min}$ の値も表2にみるごとく同程度であるにかかわらず，ベンゼンの方が式(5)からの偏差をみせる．しかしベンゼンの実験では低流量のとき，ある種の不安定指示が圧力計に現われたことが観察されており，恐らく実験装置の循環ループの構造上の相違によって前述の差が生じたものと推察される．なおPappel[25]は液体ちっ素のCHFの実験（実験条件は表2参照）をおこない，ρ_v/ρ_l が小さいほど浮力の影響が大きい結果を示している．

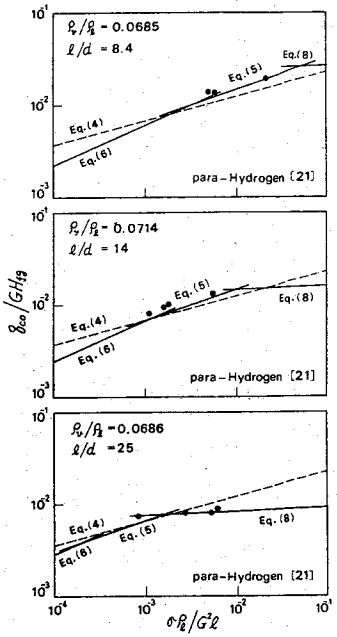

図10. q_{co} 実測値と，「流路CHF」，「平板CHF」予測値の比較

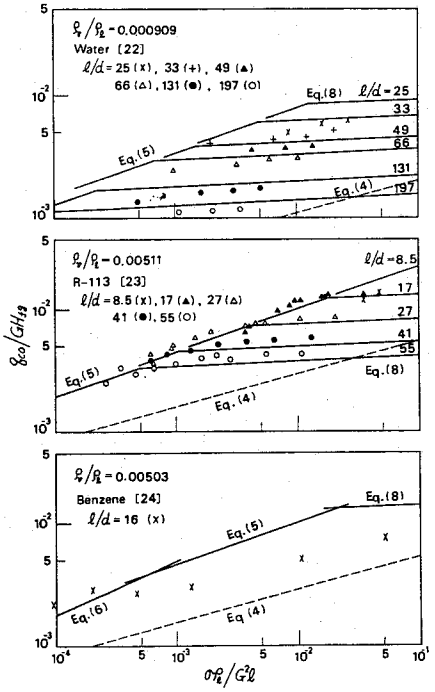

図11. q_{co} 実測値と，「流路CHF」，「平板CHF」予測値の比較．

さて，質量速度が低いときは以上のような状況を生ずるので，4章の終りに記したルール(ii)に対し下記の補足ルール(ii·s)を追加しておくのがよいと考えられる．

(ii·s) 圧力（すなわち ρ_v/ρ_l）が低くて $[q_{co}]_p < [q_{co}]_c$ となる場合，質量速度Gが小さくて $[q_{co}]_c$ がL領域ないしH領域に多少かかる程度のものである場合，そのCHFは l/d_{he} が減少すると「流路CHF」から，浮力の影響をうける特殊領域に入る．

表2．比較的に質量速度の低いCHFの実験の条件

Source	Fluid	l (cm)	d (cm)	l/d	$\frac{\rho_v}{\rho_l} \times 10^2$	$u_{i,min}$ (m/s)	Remark
Lewis et al.[21]	H_2	11.2-40.6	1.40	8.00-29.0	6.35-7.25	0.081	l: boiling length
Hewitt et al.[22]	Water	22.9-183	0.930	24.6-197	0.0909	0.097	
Barnard et al.[23]	R-113	14.2-98.2	1.74	8.25-57.0	0.511	0.032	l: boiling length
Pokhvalov et al.[24]	Benzene	8.00	0.500	16.0	0.503-23.6	0.075	
Pappel[25]	N_2	30.5	1.28	23.8	1.79-6.53	0.23	

なお「平板CHF」に関する整理式(4)は，現在までのところ $\sigma\rho_e/G^2\ell < 3\times10^{-3}$ の範囲でのみ検証されているもので（文献(6)参照），従って $\sigma\rho_e/G^2\ell > 10^{-2}$ 程度の範囲（重直上昇流に平行な重直平板加熱面を考える）においては，「平板CHF」自体も，なんらかの浮力の影響をうけるものに変る可能性がある．

6．結　論

本報では，一様加熱重直流路内の強制流動沸騰に対し，ℓ/d_{he} の値が非常に小さい場合のCHFについて，「平板CHF」との関連を念頭におきつつ研究をおこなった．質量速度Gが比較的に大きい場合については，本報の内管加熱二重管の実験結果，ならびに既存データの検討により，「流路CHF」から「平板CHF」への変移条件は ℓ/d_{he} の大きさだけでなく，ρ_v/ρ_l や $\sigma\rho_e/G^2\ell$ の大きさにも関係することを見出し，両形式のCHFの発生に対し4章に記すルール(i)，(ii)を導いた．質量速度Gが比較的に小さい場合については，既存データの分析から，浮力の影響が関与することを示し，そして5章に記す補正ルール(ii-s)を導いた．

なお本報ではHP領域には触れていないが，以前の研究(1)によれば，ℓ/d_{he} が非常に小さい時，HP領域のCHFは小じないようである．従って ℓ/d_{he} の小さい範囲を対象とする本報では対象外にしてよいと考えられる．

本研究に対し文部省から特定研究（昭和54年度 No.411002）科研費補助金の援助をうけ，また実験に対し浦木泰男，名取頌二の両君の援助をうけたので，これらに対し謝意を表する．

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一様加熱垂直二重管内の強制流動沸騰の限界熱流束の無次元整理*

甲　藤　好　郎**

1. 序

原子炉の燃料集合体内の燃料棒のまわりの流れは，二重管内の流れに似た面があり，そのようなことなどのため，二重管内の強制流動沸騰の限界熱流束(CHF)の研究がおこなわれて来ている．本報では，一様に加熱される垂直の同心二重管で，サブクール液（その極限として飽和液）が安定（振動のない）に入口から流入する場合を扱う．これに関し，内管加熱（外管非加熱）の場合の水の CHF に対し，Barnett[1] は8データ源から圧力 69.0 bar 下の 724 個の実験データを集録，さらに Barnett[2] は新しい3データ源から圧力 69.0 bar 下の 106 個のデータを追加集録した．Janssen ら[3] は圧力 41.4～100.0 bar の水の CHF の実験をおこない（そのうち圧力 69.0 bar のデータは文献（1）に集録されている），他方 Little[4] の研究には一様加熱二重管の圧力 51.7～69.0 bar の水の CHF の実験が含まれている．水以外では，内管加熱（外管非加熱）の場合のフロン 12 の CHF の実験があり，すなわち Stevens ら[5] の研究には圧力 8.1～10.7 bar 下の実験が含まれ，また Ahmad ら[6] は圧力 10.7～16.4 bar 下の 418 個の実験データを求めている．他方，内管加熱，外管加熱，および両管同時加熱を含む条件下の実験が水についておこなわれ，このうち Becker ら[7]（圧力 8.3～36.8 bar），Jensen ら[8]（圧力 69.0 bar）は一方の管壁の加熱が他方の管壁の CHF に影響があることを示すに対し，Ornatskiy ら[9]（圧力 172～216 bar），Tolubinskiy ら[10]（圧力 150～200 bar）はその影響がないという結果を得ている．なお Ornatskiy ら[11]の別の研究には，外管加熱（内管非加熱），圧力 98.1～186.5 bar の水のデータが含まれている．

実験データの整理に関していえば，内管加熱（外管非加熱）の水の CHF に対し，Barnett[1]，Janssen ら[3]，Tong ら[12]，Bertoletti ら[13]，Becker[14]，Hewitt ら[15]が経験的整理をおこなっている．また Ahmad ら[6][16]は，CHF のモデリングの研究に関連し，一種の無次元整理の方法を示している．なお内，外管加熱共用のものとして，Tolubinskiy ら[17] は水に対する経験整理式，Ornatskiy[18] は一種の無次元整理式を与えているが，前者には自然流動沸騰時の CHF，また後者には重力の加速度 g が含まれる形式である．従って現状では，まだ真の意味での CHF の無次元整理がおこなわれていない状況にある．ところで，さきに著者[19][20] は単一円管内の強制流動沸騰の CHF の無次元整理式を提出，かつその適用性の広さを示した．そこで本報は，それと同様な分析手法を二重管の場合に拡張し，CHF の無次元整理式を求めた結果について報告する．

記　号

d ：単一円管内径
d_{he} ：加熱相当直径
d_i ：内管外径
d_0 ：外管内径
G ：質量速度
H_{fg} ：蒸発潜熱
ΔH_i ：入口サブクール・エンタルピ
K ：ΔH_i の影響に関係する係数
l ：加熱管長
p ：絶対圧力
q_C ：限界熱流束
q_{CO} ：$\Delta H_i=0$ の場合の q_C
α ：外管加熱割合（全加熱負荷に対する比率）
ρ_l ：液体密度
ρ_v ：蒸気密度
σ ：表面張力
χ_{ex} ：出口クオリティ

2. 加熱相当直径

任意の流路断面に対し，加熱相当直径 d_{he} は一般に次式で定義される．

$$d_{he}=4\times(\text{流路断面積})/(\text{加熱辺長}) \quad \cdots\cdots(1)$$

そして一様加熱の場合，エネルギのつりあいから次の関係（記号については記号表参照）が容易に導かれる．

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$$q_C/(GH_{fg})=[\chi_{ex}+(\Delta H_i/H_{fg})]/(4l/d_{he}) \quad \cdots\cdots(2)$$

さて二重管（内管外径 d_i，外管内径 d_0）に関し，式（1）の定義から次式が得られる．

内管加熱（外管非加熱）の場合：

$$l/d_{he}=(l/d_i)/[(d_0/d_i)^2-1] \quad \cdots\cdots(3)$$

外管加熱（内管非加熱）の場合：

$$l/d_{he}=(l/d_i)(d_0/d_i)/[(d_0/d_i)^2-1] \quad \cdots\cdots(4)$$

図1（a）は式（3），図1（b）は式（4）の関係を図示し，なお文献（1）～(11) の実験条件をも示してある．図1に明らかなように，d_0/d_i が1に近い時は d_0/d_i の値のわずかな変化によって l/d_{he} の値が大きく変化するという特性がある．なお図1（a），（b）の原点（$l/d_i=0$，$d_0/d_i=1$）は長方形管に対応する．

3. 内管加熱（外管非加熱）の場合の CHF

3·1 q_C と ΔH_i の関係　文献（1）～（6）の実験データについて q_C と ΔH_i の関係を調べてみると，質量速度 G が小なる間は図2の A の形の直線関係があるが，G が十分に大になると図2の B の形が出現し，これは単一円管の場合[19]と同じ傾向である．ただし二重管の場合は，G の大なる領域で時として図2のbの形の非直線関係が出現する場合もあるようである．ともあれ図2の q_C·ΔH_i の関係から，$\Delta H_i \to 0$ の外そうにより $\Delta H_i=0$ における q_C の値 q_{co} が求められる．ただ図1（a）の原点付近の条件下の実験（● 印で示す）のデータは，他のデータに比べわずかではあるが，q_{co} の値に原因不明の偏りを示すので以下の分析から除いた（元来 $d_0/d_i \fallingdotseq 1$ および $l/d_i \fallingdotseq 0$ の条件は特異な状況を出現しやすい）．

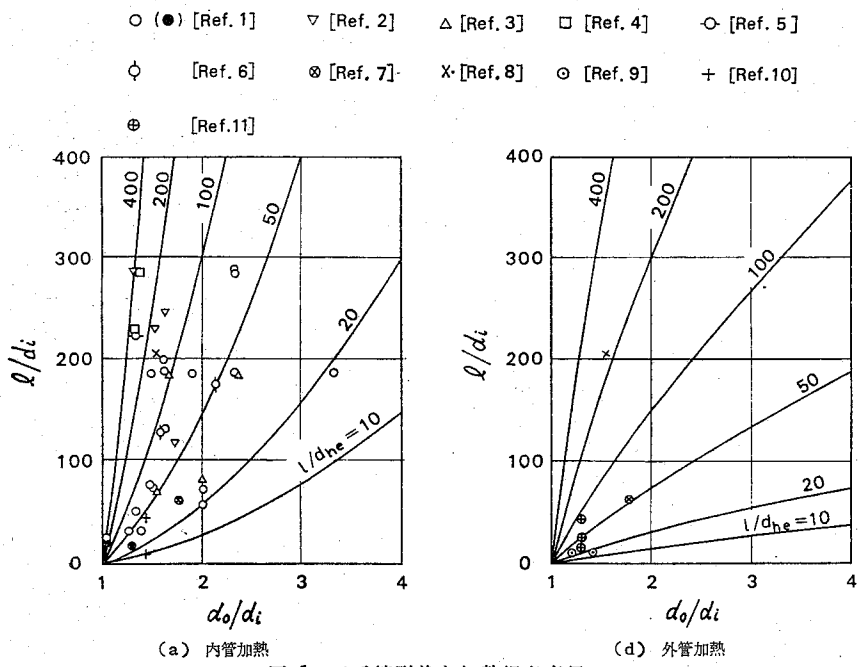

図 1　二重管形状と加熱相当直径

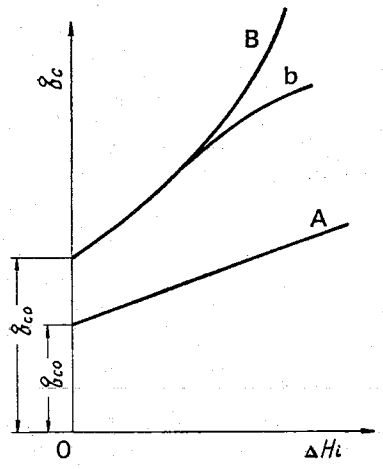

図 2　q_C と ΔH_i の関係

3·2 q_{co} の分析の手がかり　3·1 節に求めた q_{co} の分析に際し，単一円管で用いた手法[19]を用いると次のごとくなる．すなわち二重管で $d_0 \to \infty$（d_i=一定）の極限を考えると次の形の整理式が成立する可能性がある．

$$q_{co}/(GH_{fg})=\text{const.}\,(\rho_v/\rho_l)^{0.133}\{\sigma\rho_l/(G^2l)\}^{1/3} \quad \cdots\cdots(5)$$

一方，$d_0 \to d_i$（d_i=一定）の極限を考えると，次元の面から次の形の整理式が成立する可能性がある．

$$\{q_{co}/(GH_{fg})\}(4l/d^*)=\text{const.}\,(\rho_v/\rho_l)^a \times\{\sigma\rho_l/(G^2l)\}^b \quad \cdots\cdots(6)$$

ここに d^* は直径で，この d^* にどの直径をとるかが

問題である．もし加熱相当直径 d_{he}〔現在，式（3）による〕をとれば，$\Delta H_i=0$ および式（2）から明らかなごとく，式（6）左辺は出口クオリティ χ_{ex} に等しい．そしてこれは，$d_0\to d_i$ の極限状態の CHF の発生条件として物理的意味があるように見える．なお CHF のデータ整理に関する従来の諸研究〔文献（1），（3），(12)～(18)〕の多くは，水力相当直径を d_{he} と併用，ないし単独に用いている．しかし，もともと水力相当直径は粘性流体の流路内流れの圧力降下の問題に用いて有効な量である．従って著者の前報[19][20]に見るごとく，CHF に対し粘性の影響は二次的なものであるならば，CHF に対し水力直径はそれほど重要な量でないと考えるのが妥当であろう．そこで，いま式（6）の d^* に d_{he}〔式（3）による〕を用いることに決めよう．この時，d_0 に関する上記の両極限で，式（5），（6）の形にそれぞれなるような式として，次式はその候補になり得る．

$$q_{co}/(GH_{fg})=\text{const.}\,(\rho_v/\rho_l)^{0.133}\times\{\sigma\rho_l/(G^2l)\}^{1/3}/[1+\text{const.}\,(\rho_l/\rho_v)^m\times\{\sigma\rho_l/(G^2l)\}^n l/d_{he}] \quad \cdots\cdots(7)$$

現在，式（3）から $d_0\to\infty$（d_i=一定）の時 $l/d_{he}\to 0$，また $d_0\to d_i$（d_i=一定）の時 $l/d_{he}\to\infty$ となるゆえ，この両極限で式（7）は確かに式（5），（6）に一致する．

さて q_{co} の実験データに関し，式（7）の見地から，$\{\sigma\rho_l/(G^2l)\}^{1/3}/\{q_{co}/(GH_{fg})\}$ と $\sigma\rho_l/(G^2l)$ の関係を調べてみるとき，図3はその代表例を示すもので，単一円管の場合と同様な特性を持っている．すなわち，縦軸の値がほぼ一定に保たれる H 領域を中心に，$\sigma\rho_l/(G^2l)$ が大になると縦軸の値が増加する L 領域，および $\sigma\rho_l/(G^2l)$ が小になると縦軸の値が増加し，かつ q_C-ΔH_i に非直線関係が出現（図3で▲のデータはそれを示す）する N 領域の存在が見られる*1．

3・3 H 領域の q_{co} の分析　図3に見られるような H 領域の特性に従う q_{co} の全データを，式（7）を参考に整理してみた結果が図4である．データは，多少のばらつきを示すものの，ほぼ ±10 %の範囲内で整理され，これは次式を与える．

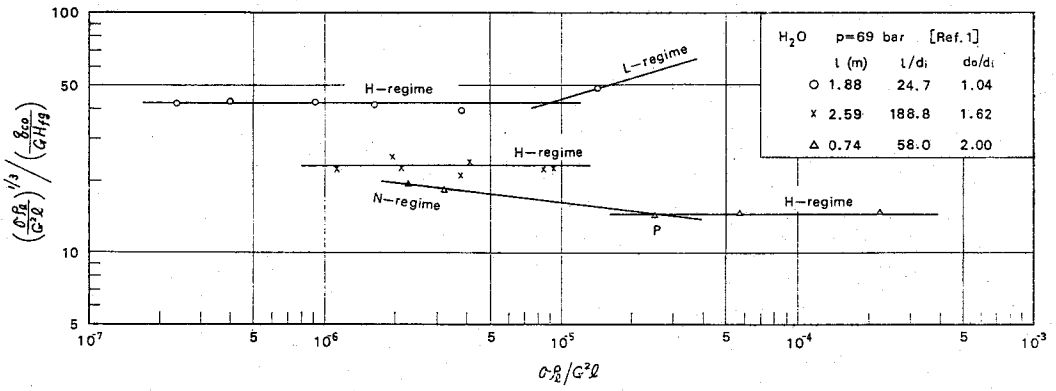

図 3　q_{co} の変化特性（圧力一定の条件下）

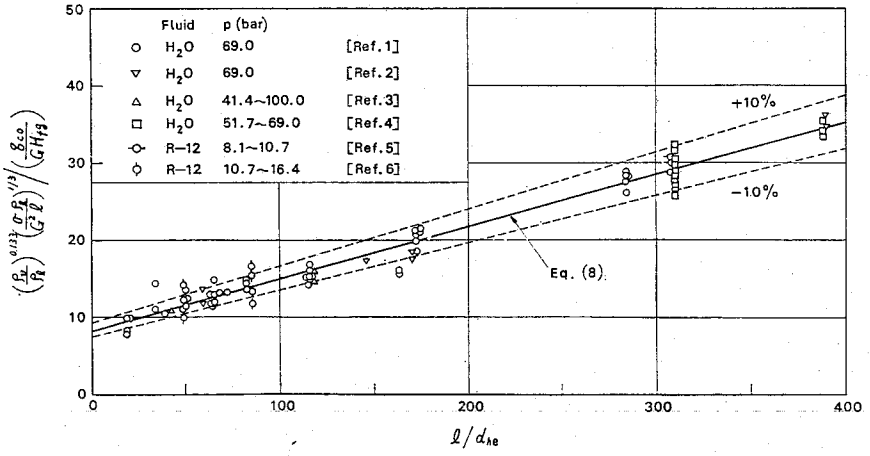

図 4　H 領域の q_{co} の無次元整理

*1 以下3～5章にあらわれる L, H, N, HP の各領域の名称の由来については文献 (19) 参照．

$$\frac{q_{co}}{GH_{fg}}=0.12\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\frac{1}{1+0.0081l/d_{he}} \quad \cdots\cdots(8)$$

単一円管の場合の H 領域の整理式を付録の式 (23) に示すが，右辺分母の定係数の値の相違が特に注目される．

3・4 L 領域の q_{co} の分析　文献 (1)～(6) の実験データには L 領域のデータがわずかしか含まれていない．しかし一応それらにつき，単一円管の場合の L 領域の整理式〔付録の式 (22)〕を参考に，かつ l/d_{he} を用いて整理した結果が図 5 であり，次式を与える．

$$\frac{q_{co}}{GH_{fg}}=0.25\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d_{he}} \quad \cdots\cdots(9)$$

データが少ないので暫定的とすべきであるが，式(9)は単一円管の場合の式 (22) と同形式であり，これは L 領域の CHF が流路内の液体が欠乏に近い状況で生ずることを意味しているといえるであろう．

3・5 N 領域の q_{co} の分析　厳密にいえば N 領域は，$q_C\cdot\Delta H_i$ の非直線関係が成立する領域であるが，図 3 の P 点から左の斜線にみるごとく，q_{co} の変化特性が同じである「前 N 領域」も存在する．従って，これら両領域の q_{co} の実験データを含めて考える．文献 (1)～(6) にはこれに属するデータが多くないが，単一円管の場合の式 (23)，(24) の形の関係を参考にしつつ式 (8) を考慮して，上記のデータを整理してみた結果が図 6 である．データのばらつきが大（単一円管の場合も N 領域ではその傾向がある），かつデータ数が少ないのでやや精度がおちるが，図 6 から次式が得られる．

$$\frac{q_{co}}{GH_{fg}}=0.22\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}\times\frac{(l/d_{he})^{0.171}}{1+0.0081l/d_{he}} \quad \cdots\cdots(10)$$

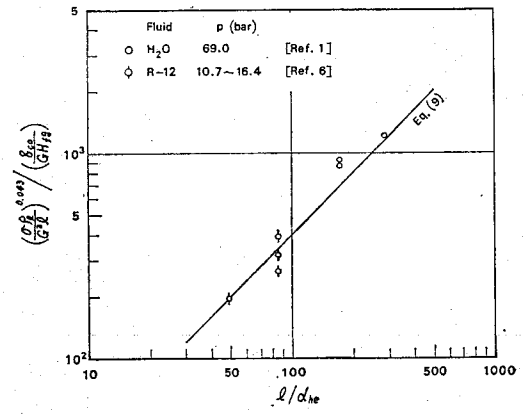

図 5　L 領域の q_{co} の無次元整理

単一円管の場合の整理式 (24) と比べ，右辺分子の l/d_{he} の指数の値の相違が注目される．

3・6 HP 領域について　文献 (1)～(6) には，内管加熱（外管非加熱）の HP 領域のデータが含まれていない．従って，ここに HP 領域の CHF の整理式を与えることができないが，この領域の CHF の存在の可能性は，外管加熱の場合の状況（後述の 4・3 節参照）などから十分にあると考えられる．

3・7 各特性領域の境界　単一円管の場合と同様な方法で，L，H，N 領域の境界を定める．まず L 領域と H 領域の境界は，式 (8)，(9) から q_{co} を消去して求められ，次式で与えられる．

$$\frac{l}{d_{he}}=\frac{1}{0.48(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{0.29}-0.0081} \quad \cdots\cdots(11)$$

図 7 に，$\rho_v/\rho_l=0.0484$ および 0.0849 の場合に対し，式 (11) の与える境界線を示してある．なお図 7 に示す実験データは，本章で分析に用いた全データであり，上記の境界線の近傍から右方のデータが L 領域に属する．次に H 領域と N 領域の境界に関連し，式 (8)，(10) から q_{co} を消去すると次式が得られる．

$$\frac{l}{d_{he}}=0.0288\frac{1}{\{\sigma\rho_l/(G^2l)\}^{0.584}} \quad \cdots\cdots(12)$$

これは図 7 に破線で示される．しかし，これは図 3 で P 点の位置を示すもので，H 領域と N 領域の境界ではない．そして図 7 に示す実験データのうち，黒ぬりのデータは $q_C\cdot\Delta H_i$ の非直線性（図 2 の B または b の形）を示すものである．このグループのデータの個数が必ずしも十分でないため，多少の不確定性を残しているが，図 7 の実斜線をもって H 領域と N 領

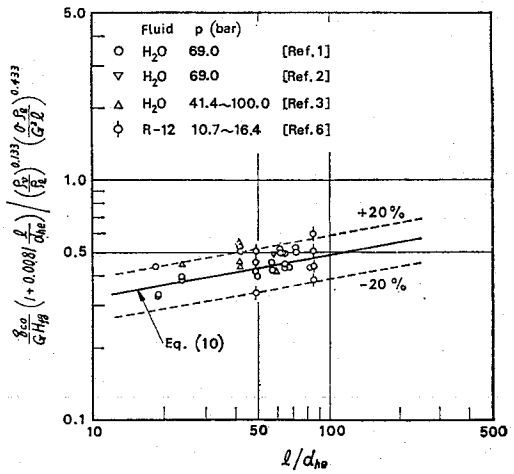

図 6　N 領域の q_{co} の無次元整理

域の境界と定めると，これは次式を与える．

$$\frac{l}{d_{he}}=0.0206\frac{1}{\{\sigma\rho_l/(G^2l)\}^{0.584}} \quad \cdots\cdots\cdots\cdots(13)$$

式（13）は単一円管の場合の式（26）に対応するものである．

3·8 CHF に対する入口サブクールの影響　二重管流路入口で液体がサブクールされている時，図2のA のごとく q_C-ΔH_i に直線関係が成立する L 領域，H 領域では次のごとく書くことができる．

$$q_C=q_{CO}(1+K\Delta H_i/H_{fg}) \quad \cdots\cdots\cdots\cdots(14)$$

ここに K は ΔH_i に独立な係数である．

さて表1は，L 領域の実験データから，その K（これを K_L と記す）の値を求めた結果を示す．K_L の値

表 1　K_L の実験データ

流　体	p bar	l/d_{he}	$\sigma\rho_l/(G^2l)$	K_L
水	69.0	284.2	1.44×10^{-5}	0.74
〃	69.0	173.5	1.48×10^{-5}	1.55
〃	69.0	172.5	1.42×10^{-5}	1.08
フロン12	10.7	85.5	2.91×10^{-4}	1.34
〃	10.7	85.5	7.66×10^{-5}	1.52
〃	16.4	85.5	1.19×10^{-4}	0.91
〃	10.7	49.4	2.80×10^{-4}	1.11
平　均				1.18

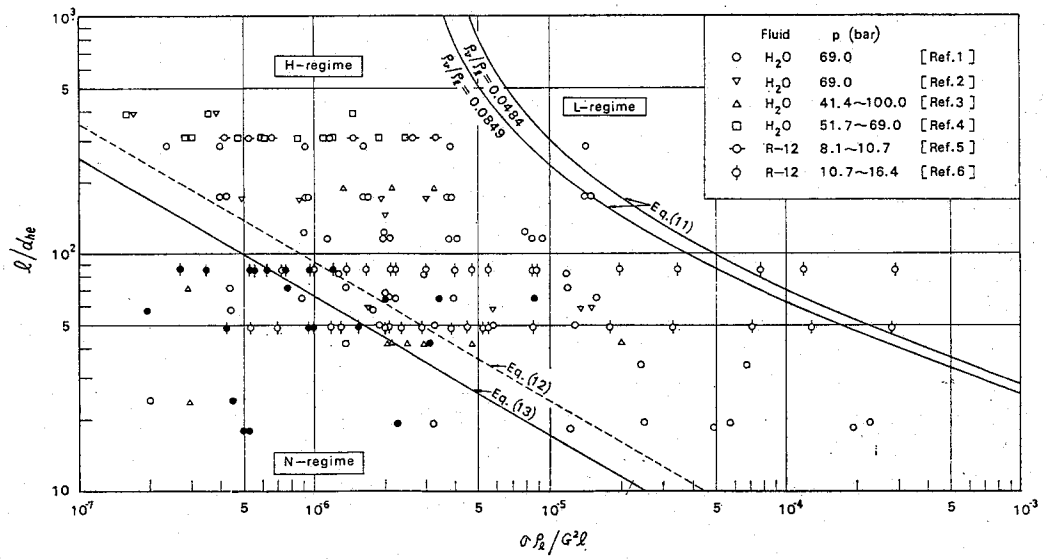

図 7　特性領域の境界線図

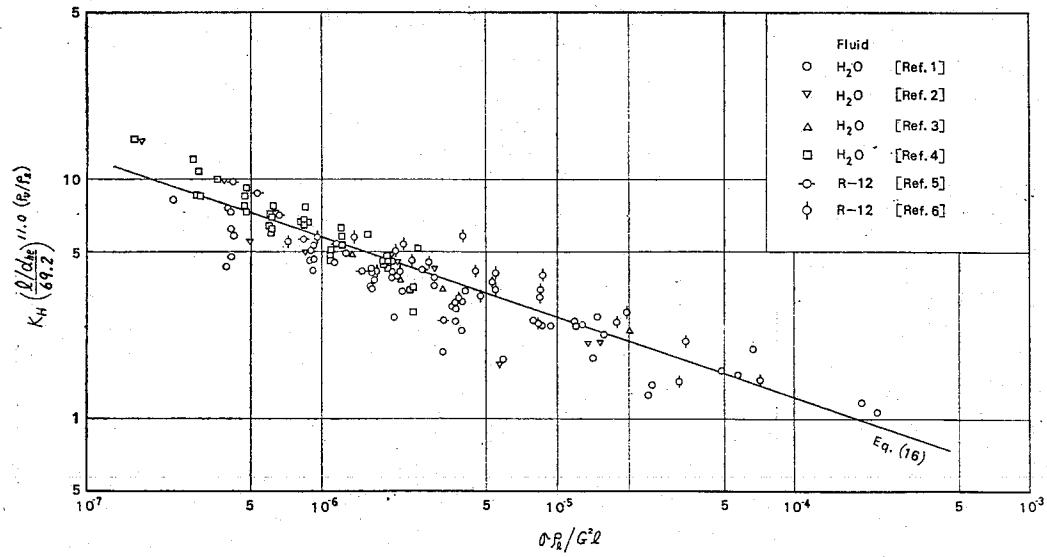

図 8　K_H の実験データの整理

にばらつきがあるが平均値は $K_L=1.18$（これは単一円管の場合の平均値 $K_L=1.16$ に近い）であり，近似的に次のごとく置いて可であろう．

$$K_L=1.0 \quad \cdots\cdots(15)$$

次に図8は，H 領域に属する実験データから求めた K（これを K_H と記す）の値を，単一円管の場合〔付録の式 (28) 参照〕に似た形に整理したもので，同図の直線は次式であらわされる．

$$K_H=0.057\left(\frac{69.2}{l/d_{he}}\right)^{11.0(\rho_v/\rho_l)}\frac{1}{\{\sigma\rho_l/(G^2l)\}^{1/3}} \quad \cdots\cdots(16)$$

単一円管の場合は，$\sigma\rho_l/(G^2l)=3\times10^{-6}$ の前後で K_H の特性が変化したが，図8の場合は K_H の特性にそのような変化が見られない．

4. 外管加熱（内管非加熱）の場合の CHF

外管加熱（内管非加熱）の場合の CHF については，既存の実験データが少なく，従って3章に対応する分析ができない．しかし幸いなことに，単一円管の場合の整理式〔付録の式 (22)～(29)〕において，l/d を式（4）の l/d_{he} で置きかえさえすれば，外管加熱（内管非加熱）の二重管の CHF の整理式になるようであり，この事について以下に例証を示す．

4·1 H 領域　図9は，内外管双方を同時に加熱した二重管において CHF 発生時の流路出口クオリティ χ_{ex} の実験結果（Jensen ら[8]による）を示す．図9の横軸の α は

$$\alpha=L_{tube}/(L_{rod}+L_{tube}) \quad \cdots\cdots(17)$$

ここに L_{tube} は外管の加熱負荷，L_{rod} は内管の加熱負荷である．従って図9の $\alpha=1$ の実験データは外管加熱（内管非加熱）の場合のものであり，その χ_{ex} のデータから，式（2）〔ここに l/d_{he} は式（4）で与える〕によって直ちに $q_C/(GH_{fg})$ の実験値が求められ，それを表2に示す．一方，この実験に関し，領域の判定〔文献 (19) の図 14 によるが，表2に見るごとく $\sigma\rho_l/(G^2l)<5\times10^{-4}$ の範囲にあるので文献 (20) に従い式 (22) で $C=0.34$ とする〕をおこなうと H 領域にあることがわかる．そこで式 (23)〔l/d の代わりに式（4）の l/d_{he}〕から $q_{CO}/(GH_{fg})$ を計算，式 (28) および式 (14) を経て $q_C/(GH_{fg})$ の計算値を求めた結果，表2の最下段のごとくなる．表2において，$q_C/(GH_{fg})$ の実験値と計算値はよく一致している（ただし $G=800$—$1\,400$ kg/m²s で両者がよく合いすぎているのは，むしろ偶然である）ということができる．

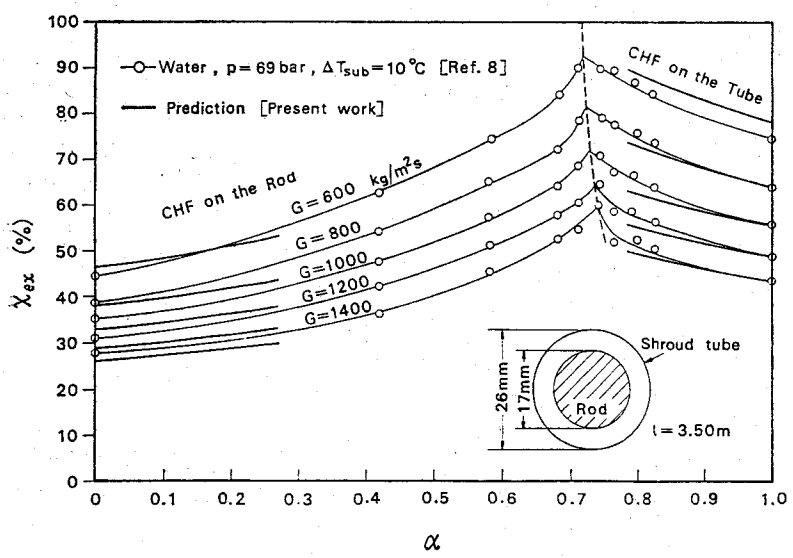

図 9　内外管同時加熱の二重管の CHF

4·2 N 領域　図10の○,●,△は Ornatskiy ら[11]による外管加熱（内管非加熱）の場合の実験データで，4·1 節と同じ方法により N 領域（ないし前 N 領域）にあることがわかる．なお本実験は大部分，入口サブクール ΔH_i が零に比して大きな条件下におこなわれている．一方，図10

表 2　外管加熱（内管非加熱）の CHF-H 領域

G kg/m²s		600	800	1 000	1 200	1 400
χ_{ex}		0.746	0.640	0.560	0.488	0.437
$\sigma\rho_l/(G^2l)$		1.01×10^{-5}	5.67×10^{-6}	3.63×10^{-6}	2.52×10^{-6}	1.85×10^{-6}
$\frac{q_C}{GH_{fg}}$	実験値	8.30×10^{-4}	7.17×10^{-4}	6.32×10^{-4}	5.56×10^{-4}	5.01×10^{-4}
	計算値	8.68×10^{-4}	7.21×10^{-4}	6.25×10^{-4}	5.55×10^{-4}	5.00×10^{-4}

に記入してある × 印は，$\Delta H_i=0$ の場合に対し式(24)〔l/d の代わりに式（4）の l/d_{he}〕および式（2）から求めた q_C，χ_{ex} の計算結果を示す．そして図10に見るごとく，上記の実験データと計算値をスムースな曲線で結び得る事実は，外管加熱（内管非加熱）のN領域に対する式（24）の適用性を示すものと判断して可であろう．

4·3 HP領域　図11に示す実験データは，Ornatskiy ら[9]が圧力 216 bar の水（臨界点に非常に近い）について得たデータ（これらは図2のAの直線関係を示し，それから求めた q_{co} を図11に図示）によるものである．さて前と同じ方法〔式（4）の l/d_{he} を用い文献（19）の図14による〕で領域の判定をおこなうと，上記の実験は十分にHP領域に入っていることがわかる．なお前記の q_C-ΔH_i の直線関係も，このHP領域にあることに矛盾しない．そして式（25）による $q_{co}/(GH_{fg})$ の計算値と比較してみると図11にみるごとくだいたいよい一致をみせる．なお念のため図11には，式（23）によるH領域，式（24）によるN領域の計算値も掲げてあるが，実験データから遠く離れている．

〔付記〕　上記の Ornatskiy らの実験について次の事柄を記しておく要がある．すなわち，彼らは二重管の流路幅 $(d_0-d_i)/2$ が実験範囲で CHF に影響しないと記している．しかし式（25）の計算値の方は，図11の実曲線と破曲線の間の相違を生じている．また彼らの実験データから求めた K_{HP} の値は，式（29）から求められる K_{HP} の計算値に比し大きさがかなり相違する．このような問題が生ずる原因の解明は今後の研究にまたねばならない．

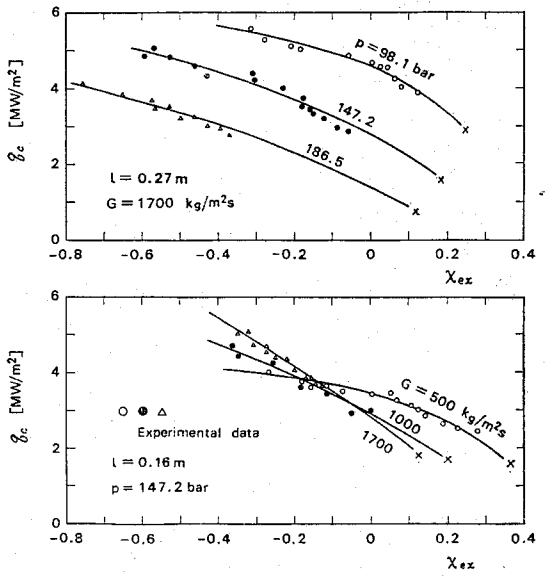

図 10　外管加熱，内管非加熱の二重管（d_0=13 mm, d_i=10 mm）の水の CHF

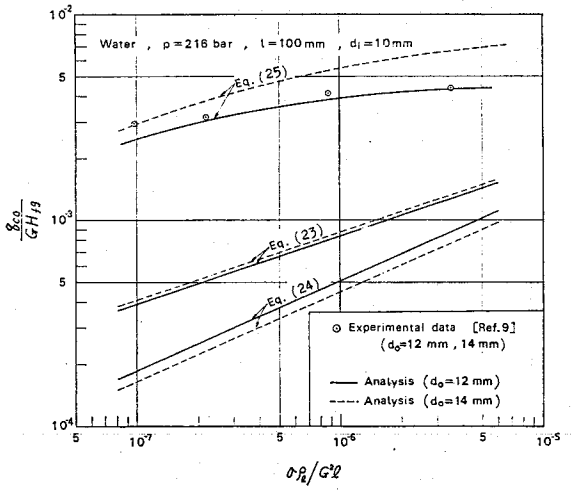

図 11　外管加熱，内管非加熱の二重管の水の CHF

5.　内外管同時加熱の場合の CHF

5·1 LおよびH領域の近似的取扱い　式(17)の α の割合で内外管が同時に加熱されている二重管において，いま図12に破線で示すごとく直径 d_m の仮想壁で流路を二分割し，各部分流路の断面積が，それに接する壁面の加熱負荷に比例するようにすれば

$$(d_0^2-d_m^2)/(d_m^2-d_i^2)=P_{tube}/P_{rod} \quad \cdots(18)$$

式(17)，(18)から

$$(d_m/d_i)^2=\alpha+(1-\alpha)(d_0/d_i)^2 \quad \cdots\cdots(19)$$

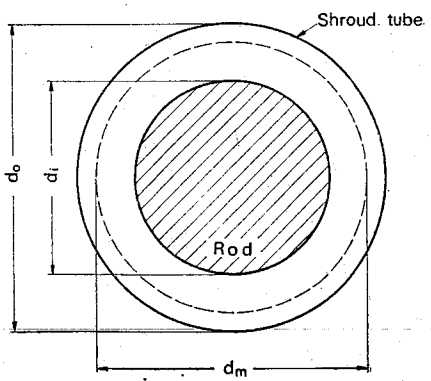

図 12　二重管流路の分割

さて，もし流路の全断面上で，質量速度および軸方向エンタルピ変化が一様とみなし得る状態にあったとすると，上記の仮想壁を通して物質およびエネルギの移動がないと考えても実質的に同じ状態が得られる．そこで，図 12 の内側流路を内管加熱（外管非加熱）の二重管とみなすと，その加熱相当直径 d_{he} は式 (1) の定義から，$d_{he}=(d_m^2-d_i^2)/d_i$ である．また同様に外側流路を外管加熱（内管非加熱）の二重管とみなすと，その加熱相当直径 d_{he} は式 (1) から $d_{he}=(d_0^2-d_m^2)/d_0$ である．従って，これらの式と式 (19) から d_m を消去すると

内側流路に対し

$$\frac{l}{d_{he}}=\frac{1}{1-\alpha}\frac{l}{d_i}\frac{1}{(d_0/d_i)^2-1} \quad \cdots\cdots\cdots\cdots(20)$$

外側流路に対し

$$\frac{l}{d_{he}}=\frac{1}{\alpha}\frac{l}{d_i}\frac{d_0/d_i}{(d_0/d_i)^2-1} \quad \cdots\cdots\cdots\cdots(21)$$

式 (20) は $\alpha=0$ のとき式 (3) に一致し，式 (21) は $\alpha=1$ のとき式 (4) に一致する．

従って，まず $\alpha=0$ に近い状況下では，式 (20) の l/d_{he} を用い，3章の無次元整理式によって CHF が近似的に計算できる可能性がある．図9に示す実験の場合について調べてみると，$\alpha=0$ の近傍で H 領域にあることが判定され，従って式 (8)，(14)，(16)，(20) および式 (2) によって χ_{ex} を計算した結果，図9の $\alpha=0\sim0.27$ の範囲に示す太い曲線が得られる．次に，$\alpha=1$ に近い状況下では，式 (21) の l/d_{he} を用い，〔付録〕の無次元整理式によって CHF が近似的に計算できる可能性がある．図9の実験の場合，$\alpha=1$ の近傍で H 領域にあることが判定され，従って式 (23)，(28) および式 (14)，(20)，(2) によって χ_{ex} を計算した結果，図9の $\alpha=0.8\sim1.0$ の範囲に示す太い曲線が得られる．なお図9は $\Delta T_{sub}=10°C$（入口サブクール温度）の場合であるが，Jensen ら[8]の $\Delta T_{sub}=100°C$ の場合についても上と同様な結果が得られる．従って上記の近似的計算は，非常に簡単な方法であるにかかわらず，$\alpha=0$ および 1 の近傍で有効であると結論してよいであろう．

なお Becker ら[7]の実験データには，入口サブクール ΔH_i（または質量速度 G）の記載がないため，定量的比較ができない．しかし彼らのデータは図9と同傾向を示すもののごとく考えられ，そして彼らの実験範囲は L 領域と H 領域にわたっている．従って上記の CHF の近似的計算法はおそらく L 領域に対しても適用可能であろう．

5・2 HP 領域の状況　図 11 に示す実験結果を求めた Ornatskiy ら[9]は，圧力 172～216 bar 下に $\alpha=0.6\sim0.7$ の状況下の実験をおこない，内管の加熱は外管の CHF に影響がないという結論を与えている．また Tolubinskiy ら[10]は，圧力 150～200 bar で $\alpha=0\sim0.5$ の状況下の実験をおこない，外管の加熱は内管の CHF に影響せずといっている．この Tolubinskiy らの与える実験データのうち，圧力 150 bar のものは H 領域に属するが，圧力 200 bar のものは図 10 の方法で検討するとき H 領域でも N 領域でもないようで，従って HP 領域にあると推測される．上記の2報告を考えるとき，少なくとも HP 領域では内外管の相互干渉が消滅する可能性があるように思われる．

6. 結　　論

（1） 内管加熱（外管非加熱）の二重管の CHF に対し，既存の実験データを分析，$\Delta H_i=0$ の状況下の CHF の無次元整理式〔式 (8)～(10)〕，特性領域の境界線図（図7），および入口サブクールの影響に関する無次元整理式〔式 (15)，(16)〕を求めた．ここで管の直径としては式 (3) の与える加熱相当直径 d_{he} だけが用いられる．なお HP 領域については実験データの不足で整理式を導くに到っていない．

（2） 外管加熱（内管非加熱）の場合は，式 (4) の与える加熱相当直径 d_{he} を用いれば，単一円管の場合の無次元整理式〔付録の式 (22)～(29)〕がそのまま使える．

（3） 内外管同時加熱の場合については，外管加熱割合 α の値が $\alpha=0$ および $\alpha=1$ の近傍の場合に適用できる近似計算法（L 領域，H 領域で適用可能）を示し，また HP 領域では内外管の相互干渉が消える可能性があることを記した．

本報で利用した実験データを発表された著者らに心からの敬意を，また本研究は文部省から科学研究費（特定研究：エネルギーの有効利用の工学的研究）の援助をうけたもので謝意を表する．なお図面の作成について特にお世話になった大鳴　登君に感謝する．

〔付　録〕　単一円管の CHF の無次元整理式[19][20]のうち本報で必要なものを記しておく．まず $\Delta H_i=0$ に対し

L 領域：

$$\left.\begin{aligned}&\frac{q_{CO}}{GH_{fg}}=C\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d}\\&\left(\frac{\sigma\rho_l}{G^2l}<5\times10^{-4}\text{ で }C=0.34\right)\\&\left(5\times10^{-4}<\frac{\sigma\rho_l}{G^2l}\text{ で }C=0.25\right)\end{aligned}\right\}\cdots\cdots\cdots(22)$$

H 領域：

$$\frac{q_{co}}{GH_{fg}}=0.10\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\frac{1}{1+0.003\,1l/d} \quad \cdots\cdots(23)$$

H および N 領域：

$$\frac{q_{co}}{GH_{fg}}=0.098\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}\frac{(l/d)^{0.27}}{1+0.003\,1l/d} \quad \cdots\cdots(24)$$

HP 領域：

$$\frac{q_{co}}{GH_{fg}}=8.20\left(\frac{\rho_v}{\rho_l}\right)^{0.65}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.453}\frac{1}{1+107\{\sigma\rho_l/(G^2l)\}^{0.54}l/d} \quad \cdots\cdots(25)$$

なお式 (24) に関し，H 領域と N 領域の境界は次式で定められる．

$$\frac{l}{d}=0.77\frac{1}{\{\sigma\rho_l/(G^2l)\}^{0.37}} \quad \cdots\cdots(26)$$

次に入口サブクールエンタルピ $\Delta H_i>0$ の場合に対し，式 (14) の K の値は次のごとくである．

L 領域：

$$K_L=1 \quad \cdots\cdots(27)$$

H 領域：

$$\left.\begin{aligned}&\sigma\rho_l/(G^2l)<3\times10^{-6}\ \text{で}\quad K_H=1.8\{130/(l/d)\}^{5.0(\rho_v/\rho_l)}\\&3\times10^{-6}<\sigma\rho_l/(G^2l)\ \text{で}\quad K_H=0.075\{130/(l/d)\}^{5.0(\rho_v/\rho_l)}\{\sigma\rho_l/(G^2l)\}^{-1/4}\end{aligned}\right\}\quad\cdots\cdots(28)$$

HP 領域：

$$\left.\begin{aligned}&\sigma\rho_l/(G^2l)<4\times10^{-8}\ \text{で}\quad K_{HP}=0.664(\rho_v/\rho_l)^{-0.6}\\&4\times10^{-8}<\sigma\rho_l/(G^2l)\ \text{で}\quad K_{HP}=3.08(\rho_v/\rho_l)^{-0.6}\{\sigma\rho_l/(G^2l)\}^{0.09}\end{aligned}\right\}\quad\cdots\cdots(29)$$

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(19) 甲藤, 機論, **44**-387 (昭 53-11), 3865.
(20) 甲藤, 機論, **45**-394, B (昭 54-6), 859.

討　論

〔質問〕　青木成文・井上　晃
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整理式中に重要なパラメータとして加熱長さ l が入って来ている意味は，

（i） 整理式が加熱部入口のパラメータで整理している（たとえば ΔH_i や q_{co} の式で $\Delta H_i=0$ とおいていること）．

（ii） CHF 発生点で同一クオリティでも二相流の流動状態が加熱長さによって差違している．

という理由以外に l が式中に入って来る理由があるのか．いいかえれば，CHF 発生点での二相流の流動がもっとくわしくわかれば CHF は局所パラメータのみで整理できるものと考えていいのかどうか．

〔回答〕　$\sigma\rho_l/G^2l$ を使う動機となったのは，長方形加熱面の一端から，速度 u の液体噴流を流しこんだ系に関し，その CHF が $\sigma/\rho_l u^2 l$（ここに l は流れの方向の加熱面長さ）で整理できるという経験的事実(付1)である．そして，この非常に簡単な沸騰系においてすら，なぜ CHF が流入速度 u と流出端まで

(付1) 甲藤・石井, 機論, **44**-384 (昭 53-8), 2817.

の距離 l の関数になるのか不明な状態であり，実際，加熱面流出端の流れの局所状態が u, l だけで指定されるはずもなく，不思議にもみえるわけである．

〔質問〕 **中西重康**（大阪大学工学部）

（1） L 領域はドライアウト，他の領域は DNB に対応すると考えてよいか．また，$\Delta H_i=0$ でドライアウトを生じる二重管は ΔH_i を大にしても DNB を生じることはないとしてよいか．

（2） 本論文で提出された相関を入口クオリティが正の範囲に対して使用しようとする場合，式 (14) を延長する方法によってよいか．

（3） 二重管のすきまが極端に狭い場合にも，l/d_{he} が大きければ，提案された相関は有効とお考えか．

〔回答〕 （1） 特性領域と CHF 発生機構との関係はまだ必ずしも明白ではない．しかし $\Delta H_i=0$ の場合，少なくとも L 領域は，いわゆるドライアウトの状況に対応すると考えてよいようである．円管の場合であるが，著者の次報(付2) でも，従来ドライアウトの名の下に測定されたデータが L 領域の整理式に一致することが示される．他方，いわゆる DNB の状況に対応するのは，出口クオリティの大きさから判断して N 領域であろうと推測している．また H, HP 領域については，現在のところ，こういう割り切った考えはしない方がよいように思っている．

次に $\Delta H_i>0$ の問題については，次の 2 点を回答申しあげたい．（i）厳密に言うと，本報では，式 (14) の K の値が ΔH_i に独立な一定値をとるという経験的事実だけを利用しているだけで，CHF の機構は考慮外である．（ii）円管についてであるが，次報(付2) で $\Delta H_i>0$ の場合の L 領域について，ある解析を試みている．それによると少なくとも

$$\frac{\Delta H_i/H_{fg}}{1+K_L\Delta H_i/H_{fg}}<4\frac{q_{co}}{GH_{fg}}\frac{l}{d}$$

を満足する ΔH_i までは L 領域の CHF が起こるといえる．ただし，これ以上の ΔH_i に対しどうなるかは不明である．

（2） 入口クオリティが正，すなわち $\Delta H_i<0$ の場合（いわゆる mixed inlet condition），二重管については検討の機会を持っていないが，円管の場合(付3) は，利用可能の少数のデータに関する限り，L, H, HP 領域で q_C-ΔH_i 直線関係（図 2 の A）が $\Delta H_i<0$ の領域まで延長可能の結果を示している．そしてこの場合はもちろん，式 (14) でただ $\Delta H_i<0$ となるだけのことになる．ただ入口クオリティが非常に大きい場合は，前報(付3) の 6 章にすでに記したように，管入口の流れの状況が ΔH_i の値の指定だけでなく，二相流のフローパターンにも強く支配されることになるから注意が必要になると思う．

（3） 有効と考えていない．二重管のすきまが極限に狭い，すなわち d_0/d_i が 1 に近いときは，2 章の終わりに記したように，実際問題として d_0/d_i の値にかかわるわずかな製作誤差でも l/d_{he} の値を大幅に変化させてしまう．従って誤差の点からまず実用にはならぬと思う．また特に $d_0/d_i=1$ の場合は，実現可能なのは $l/d_i=0$ ($d_i=\infty$) の場合，つまり長方形状流路である．しかしこの状況では，図 1 にみるごとく l/d_{he} が不定となり，従って本報の整理式は使えない．

〔質問〕 **鳥飼欣一**（日本原子力研究所）

（1） 原子炉の燃料集合体の燃料棒のまわりの流れの大部分はむしろ円管内流れに近く，集合体の周辺部で二重管的要素が加わると考えられる．従って，貴論文の最初の話とやや趣きを異にするかと思うがいかがか．

（2） 安定（振動のない）に入口から流入する場合を扱われるとある．しかし，採用される各種の実験値がそうであるとの証拠はどの程度あるのか．

この種の実験では，一般に時間的に振動がなく流量一定に流すことはかなり困難であるからである．

（3） l は加熱管長とあるが，バーンアウト点は必ずしも出口部分ではないので，実験値の扱いはどのようにされたのか．

（4） 3·1 節 q_C と ΔH_i の関係の節で「$\Delta H_i\to 0$ の外そうにより $\Delta H_i=0$ における q_{co} の値が求められる」とあるが，単純に外そうにより求められるであろうか．サブクーリングが 0 ($\Delta H_i=0$) のあたりが，最も問題が多いよう（各種の実験結果からして）であるのでお伺いする．

（5） 式 (6) の d^* に d_{he}〔式 (3)〕を用いることは，式 (6) は式 (3) より連想されたもののように思われる．そうすると，式 (2) の [ ] の内，主としてクオリティ χ_{eq} は式 (6) の const.$(\rho_g/\rho_l)^a\{\ \}^b$ に相当するということになるが，そのような関係なのか．

〔回答〕 （1） 次の 2 点をお答えしたく思う，（i）本報の最初の記述は，著者自身の意見ではなく，従来のいくつかの研究の動機として記したものである（例えば Marinelli(付4) の 144 ページ参照）．（ii）著者は燃料集合体の CHF の分析まで手が及んでいないので白紙の立場である．しかしそれでも質問者のお考えが必ずしも一般的になっているとはいえないと思

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(付3) 甲藤，機講論，No. 780-16 (昭 53-10)，87.
(付4) Marinelli, V., *Nucl. Technol.*, **34**-7 (1977-7), 135.

う．集合体のコーナの流路状況は，「内管のみ加熱」の二重管流路のそれに似ている（例えばJanssen ら[3]の8ページ，Marinelli(付4) 145 ページ）ことはもちろんである．しかし他方，例えば Barnett(付5)(付6)が考えた集合体内燃料棒と二重管の相関方法（mixed flow correlation）は質問者のお考えとは違うベースの上にのっている．また最近の subchannel model による CHF 解析法においては，燃料棒を中心におく subchannel 方式もまた使われ排除されてはいない〔文献（付7）参照〕．なお本報にもみられるように，凸面加熱面と凹面加熱面は CHF に対し，性格を異にする問題も忘れてはならぬであろう．

（2） CHF 試験部と，ループの他の部分の間のダイナミックカップリングの問題については，1958 年の Lowdermilk ら(付8)，1961 年の Aladyev ら(付9)の論文など，古くから知られている．そして上記の Lowdermilk らによれば，大気圧の水の CHF の実験に際し，試験部入口に設けた弁において 3.5～14.0 kg/cm² 以上もの圧力降下（試験条件で異なる）を与えると初めて流れが安定となり，一定の CHF が得られるようになることが示されている．従って，その後の多くの実験では，この点に十分注意しておこなわれるのが普通で，例えば本報で引用した最も古い文献(3)でも，その 45，46 ページに安定性に関する明確な記述がされている．

なお質問者の抱かれる不安の原因は，内管加熱の二重管の CHF に関する研究成果報告書（1967 年）(付10)による所が大きいのではなかろうかと思ったりする（間違っていたらお許し願いたい）．この報告書では大気圧，および 60 kg/cm² の水の試験がされ，そのうち大気圧のデータはまるで散乱して規則性がないに等しいともいえるほどで，これに比べると 60 kg/cm² のデータの方はまだ規則的な傾向を持っている．そして，この報告書には，前述の Lowdermilk らのような安定化に関する記述はまったく見られないので，特に大気圧の試験の場合，試験部の流れを安定化する抵抗が不足しやすく，流れが不安定になっている可能性が考えられる．ともあれ著者の見た限りでは，上記の報告書(付10)の大気圧のデータのような例は非常に特異で，他にはどこにも見られなかった．

（3） 熱流束一定の円管の場合だが，まずドライアウト（これは L 領域の CHF に対応）の実験で壁温急上昇位置の測定がおこなわれている(付11)～(付14)．これらについて検討した結果では，管出口で CHF と考えて何の問題も生じない（著者の次報(付2)参照）．次に Groeneveld(付15)は，H および N 領域についての実験から CHF 発生位置を報告しているが，N 領域で特に $\Delta H_i<0$，かつ $\sigma\rho_l/G^2l$ がかなり小さい特殊状況以外はすべて，CHF は管出口に生じている．また Bennett ら(付16)の図7，8に，軸方向に熱流束分布が一定の場合と変化した場合の CHF 位置が報告されているが，熱流束一定のときは常に管出口でバーンアウトしている．さらに，これ以外にも熱流束一定時のバーンアウト発生位置の測定の報告があるが，管出口の付近に生じている．

なお Collier の教科書(付17)には，一様加熱流路の場合，安定流の条件下には一般に，CHF の開始は流路出口におこると考えることができる，不安定な流れの状況下では，管の出口，入口，または中間の任意の位置で過熱がおこると記されているが，著者も多くの CHF のデータを分析した体験から，Collier のように考えてよいと思う（もちろん出口付近で CHF 発生位置の上流側への多少のばらつきはあり得るとして）．なお（2）項に述べた報告書(付10)では，加熱管の焼き切れ位置が測定されているが，（2）項に述べた流れの不安定性の問題があり，また CHF 発生点が必ずしも焼き切れ位置になるとは限らない場合の存在も考える要がある．

（4） 次の2点をお答えしたい．（i）$\Delta H_i\to 0$ の極限での q_{co} を求める件．円管に関する前報(19)，二重管に関する本報のぼう大なデータの分析（$\Delta H_i>0$ のデータから外そう）を通し，すべて単純なもので何の問題もなかった．換言するに，著者の扱った実験データに関する限り，ほとんど例外なく q_c と ΔH_i の間に非常にきれいな関係を示した．（ii）次に，$\Delta H_i=$

（付5） Barnett, P.G., *AEEW-R*, 463 (1966).
（付6） Barnett, P.G., *AEEW-R*, 558 (1968).
（付7） Butterworth, D. and Hewitt, G.F., *Two-Phase Flow and Heat Transfer*, (1977), 316, Oxford Univ. Press.
（付8） Lowdermilk, W.H., ほか2名, *NACA TN*, 4382 (1958).
（付9） Aladyev, I.T., ほか3名, *Int. Devel. in Heat Transfer*, Pt. II, (1961), 237, ASME.
（付10） 日本機械学会，バーンアウト機構に関する試験研究・研究成果報告書，(昭 42).
（付11） Ueda, T., ほか2名, *Proc. Heat Transf., 1978*, Vol. 1, (1978), 423.
（付12） Koizumi, Y., ほか2名, *Int. J. Heat & Mass Transf.*, **22**-5 (1979), 669.
（付13） Hynek, S.J., ほか2名, *MIT Tech. Rep.*, No. 70586-63, (1969).
（付14） Bennett, A.W. ほか3名, *AERE-R*, 5373 (1967).
（付15） Groeneveld, D.C., *Proc. Heat Transf. 1974*, Vol. IV, (1974), 265.
（付16） Bennett, A.W., ほか4名, *Trans. Inst. Chem. Engrs.*, **45** (1967), T-319.
（付17） Collier, J.G., *Convective Boiling and Condensations*, (1972), 238, McGraw-Hill.

0とその近傍の q_C の実験値についても，Groeneveld(付16)の R-12 の測定，尾形（私信による研究成果）の液体ヘリウムの測定などがあるが，これらを見るに $\Delta H_i \geqq 0$ の範囲で q_C-ΔH_i の関係に特異になる結果は何も見られない．従ってここでもまた質問者は，文献（付 10）などを根拠にして $\Delta H_i = 0$ の付近に問題が多いなどと考えておられるのであろうかと思ってみたりする．

（5） 次の3点を回答したい．（i）式（6）は式（3）から連想したものではない．単一円管の場合[19]に記したごとく，方向性次元解析の考察から $d_0 \to d_i$ の極限で真に無次元な形式として導いたのが式（6）である．（ii）式（6）の左辺 $\{q_{CO}/(GH_{fg})\}(4l/d^*)$ は，d^* に d_{he} を使うとき，出口クオリティ χ_{ex} に等しくなる．このことは，すでに単一円管の場合[19]の式（9）について記してある説明とまったく同じである．（iii）これ以上のことは単なる推測になるので論文には記していないが，単一円管の場合の $d \to 0$ の極限の式（9），本報の場合の $d_0 \to d_i$ の極限の式(6)において，左辺の値が出口クオリティ χ_{ex} になるということは，これらの場合に限り，CHF の発生が χ_{ex} に密接な関係をもつ可能性を暗示する．そして，これは $d \to 0$ の円管，$d_0 \to d_i$ の二重管の状況（前報[19]の図3のような）に対し，おかしくないことのようにも思われ，真疑はともかく，興味のあることと思っている．

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内管一様加熱の垂直二重管（飽和液流入）内の限界熱流束の一般特性の観察*

甲　藤　好　郎**
Yoshiro KATTO

1. 序

著者は最近，一様加熱垂直円管内の強制流動沸騰の限界熱流束（CHF）の無次元整理の研究[1]~[3]を行い，その後さらに円管内 CHF の一般特性を鳥かん(瞰)することを目的として，飽和液流入条件下の限界熱流束 q_{c0} のデータを諸文献から集収して図示する研究[4]をおこなった．その一方，著者は二重円管内の CHF の無次元整理についても検討し，外管のみ一様加熱の場合の CHF は上記の単一円管の整理式で扱うことができることを示す一方，内管のみ一様加熱の場合の CHF に適用できる無次元整理式を求めた[5]．そして本報は，この内管一様加熱の二重管内強制流動沸騰の CHF に対し，円管の場合の研究[4]と同様に，飽和液流入条件下の限界熱流束 q_{c0} の実験データをできるだけ集めて図示することを試みたものである．

記　号

d_{he}：加熱相当直径［内管一様加熱の場合，式（3）］
d_i：内管外径
d_0：外管内径
G：質量速度
H_{fg}：蒸発潜熱
ΔH_i：入口サブクールエンタルピ
l：加熱管長
q_c：限界熱流束
q_{c0}：$\Delta H_i=0$ のときの q_c
ΔT_i：入口サブクール温度
ρ_l：液体密度
ρ_v：蒸気密度
σ：表面張力
χ：クオリティ（χ_{ex}：出口クオリティ，χ_{in}：入口クオリティ）

2. CHF 実験データからの q_{c0} の集積

2・1　q_{c0} の求め方　表1に示す諸文献 (6)~(31) から q_{c0}（$\Delta H_i=0$ のときの限界熱流束 q_c）を求めたが，その大部分は下記の方法（i），（ii）で求め，またごく一部は（iii），（iv）の方法で求められている．

（i）　図1，2に例示するごとく，G=一定の条件下に，ΔH_i（または ΔT_i，χ_{in} など）の変化に従う q_c の変化が $\Delta H_i=0$ の近傍まで求められている実験からは，$\Delta H_i=0$（または $\Delta T_i=0$，$\chi_{in}=0$ など）への外そうにより q_{c0} の値が求められる．なお図2の実験データは Knoebel ら[19]の求めたもので，Knoebel らは $\Delta T_i<25$°C の領域を付加ボイド効果があるとして除外し，$\Delta T_i>25$°C の領域の q_c のデータ（図2の結果をも含む軽水，ステンレス加熱管の全データ）に対して次の経験整理式を与えている．

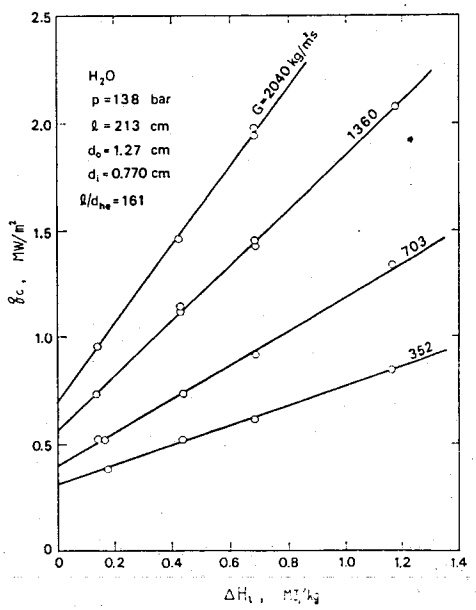


図 1　q_c-ΔH_i の直線関係の例［文献 (25) の表 A の実験データ］

* 昭和55年8月22日　第58期全国大会講演会において論文講演として講演，原稿受付　昭和55年1月14日．
** 正員，東京大学工学部（〒113 東京都文京区本郷 7-3-1）．

$$q_c=4.844\times10^5(1+0.169V)(1+0.12\Delta T_i)\cdots(1)$$

ここに使用単位は q_c W/m², V m/s, ΔT_i °C で，かつ V は液体の流入速度である．そして図 2 の破線は式（1）を示す．しかしながら，著者の CHF の整理の研究（円管[1][2]，二重管[5]）では，図 2 の結果はN領域（q_c-ΔH_i が非直線関係を持つ領域）の特性と解釈され，従って本報でも ΔT_i の領域の区別などはせず図 2 からそのまま単純に $\Delta T_i\to0$ の外そうによって q_{c0} を求めている．

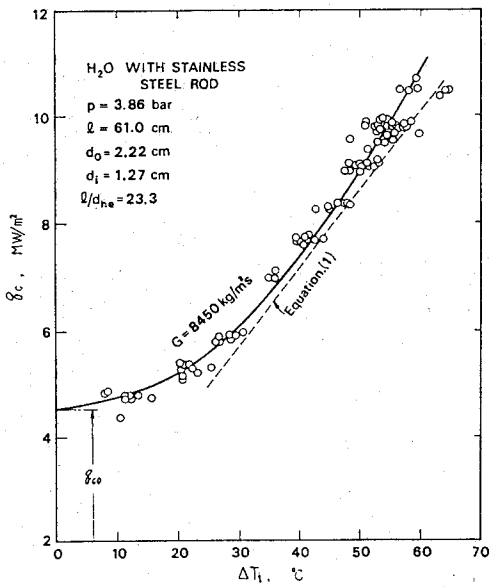


図 2 q_c-ΔH_i の非直線関係の例〔文献（19）の表 B-1 のデータのうち加熱管に欠陥などある場合を除く〕

（ii） 限界熱流束 q_c が出口クオリティ χ_{ex} の関数として与えられている報告の場合は，一様加熱の二重管流路に対し成立する次のエネルギバランス式から容易に q_c と ΔH_i の関係を知ることができて，その上で（i）の方法が使える．

$$4(q_c/GH_{fg})(l/d_{he})-\Delta H_i/H_{fg}=\chi_{ex}\quad\cdots\cdots(2)$$

ここに d_{he} は加熱相当直径と呼ばれるもので，内管のみ一様加熱の場合は次式で与えられる[5]．

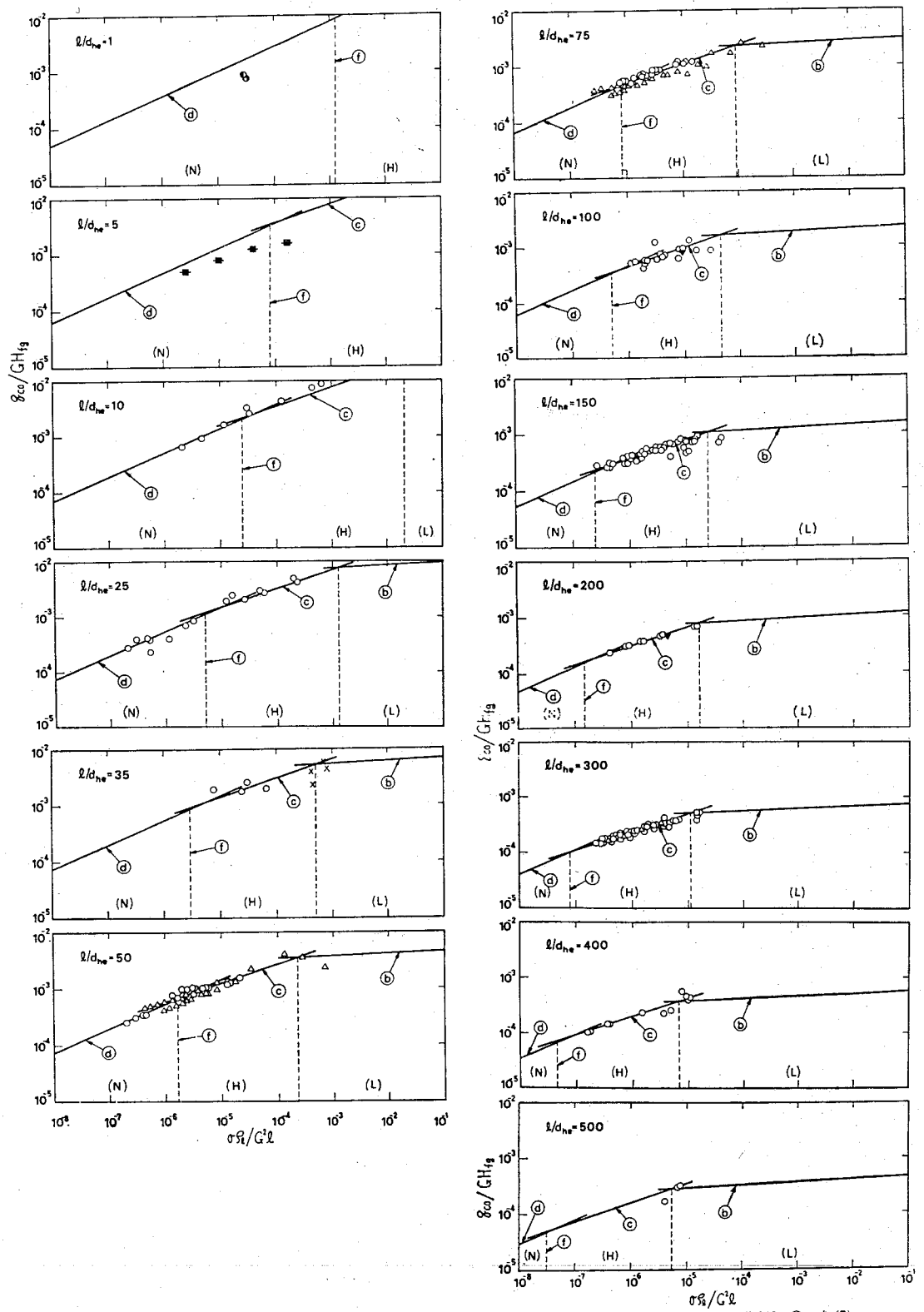
$$l/d_{he}=(l/d_i)/[(d_0/d_i)^2-1]\quad\cdots\cdots(3)$$

（iii） 文献（8），(10)，(37）のデータ表示は，形式上（ii）に属しているが，ただ二重管入口で気液二相混合流入条件（すなわち $\Delta H_i<0$, $\chi_{in}>0$）下の実験に関するデータである．この条件の実験に対し q_{c0} を考える場合は，実験に用いた実際の管長 l' に対して，$l=l'/[1-(\chi_{in}/\chi_{ex})]$ で求められる仮想管長 l が，飽和液流入の場合の管長（実験限界熱流束を q_{c0} として）に相当することになるが，円管の場合の研究[3]によれ

表 1 集めた q_{c0} のデータの一覧表

文献	流体	l cm	d_0 cm	d_i cm	l/d_{he}	ρ_v/ρ_l	$\sigma\rho_l/G^2l$	データ数
G. E.[6]	Water	274	2.22	0.952	64.8	4.84×10^{-2}	$8.93\times10^{-7}-8.04\times10^{-6}$	4
Blackford-Matzner[7]	Water	107	4.43	3.49	50.0	4.84	$1.78\times10^{-6}-1.24\times10^{-5}$	4
Bennett et al.[8]	Water	471～649	1.42	0.952	402～554	2.13	$7.24\times10^{-6}-9.96\times10^{-6}$	5
Janssen-Kervinen[9]	Water	91.4～177	2.22～2.54	0.952～1.27	24.0～42.0	2.61～7.60	$2.95\times10^{-7}-2.00\times10^{-5}$	3
Bennett et al.[10]	Water	459～884	1.58	0.952	271～522	2.13～4.84	$3.85\times10^{-6}-6.61\times10^{-6}$	13
日本機械学会[11]	Water	30.0	1.90	1.00	11.5	4.05	$2.05\times10^{-6}-3.28\times10^{-5}$	4
Barnett[12][13]	Water	73.6～274	1.40～10.1	0.952～9.64	18.4～287	2.61～8.04	$1.97\times10^{-7}-2.25\times10^{-4}$	82
Moeck et al.[14]	Water	188	7.95～10.1	7.63～9.64	172～287	4.84	$4.07\times10^{-7}-1.46\times10^{-5}$	20
Barnett[15]	Water	182～457	2.09～2.67	1.48～1.54	59.0～387	4.84	$1.71\times10^{-7}-1.51\times10^{-5}$	11
Little[16]	Water	365～457	2.09	1.58	310～387	3.39～4.84	$1.60\times10^{-7}-2.46\times10^{-6}$	40
Moeck[17]	Water	131	2.37	1.97	146	2.11～4.84	$5.27\times10^{-6}-9.07\times10^{-6}$	2
Tolubinskiy et al.[18]	Water	22.0	1.30	0.500	7.64～11.5	10.3	$4.50\times10^{-4}-6.75\times10^{-4}$	2
Knoebel et al.[19]	Water	61.0	2.22	1.27	23.3	0.123～0.224	$4.59\times10^{-7}-1.12\times10^{-6}$	3
Tolubinskiy et al.[20]	Water	10.0	1.30	1.00	14.5	16.0	1.23×10^{-4}	1
Tolubinskiy et al.[21]	Water	10.0～100	1.30	0.900	10.2～102	16.0	$3.07\times10^{-6}-3.07\times10^{-5}$	7
Andersen et al.[22]	Water	350	2.72	1.70	132	4.94	$1.55\times10^{-6}-9.74\times10^{-6}$	10
Jensen-Manov[23]	Water	350	2.60	1.70	154	4.84	$1.85\times10^{-6}-1.01\times10^{-5}$	5
Becker-Letzter[24]	Water	300	2.13	1.20	116	1.83～4.94	$1.90\times10^{-6}-3.04\times10^{-5}$	5
Mortimore-Beus[25]	Water	213	1.27	0.770	161	6.16～13.5	$2.61\times10^{-7}-4.33\times10^{-5}$	12
Stevens et al.[26]	R 12	365	2.09	1.58	310	3.60～4.82	$4.22\times10^{-7}-3.31\times10^{-6}$	8
Ahmad-Groeneveld[27]	R 12	182	2.22	1.04～1.39	49.4～85.5	4.82～8.49	$2.67\times10^{-7}-7.07\times10^{-4}$	48
Shiralkar[28]	R 114	183	2.22	1.43	90.3～180	4.81	$4.36\times10^{-6}-8.72\times10^{-6}$	2
Andrews et al.[29]	Acetone	7.62	2.09	0.604	1.15	0.290	3.13×10^{-5}	1
Andrews et al.[29]	Toluene	7.62	2.09	0.604	1.15	0.385	2.90×10^{-5}	1
Sterman et al.[30]	MIPB*	11.0	1.60	1.00	7.05	1.03	$2.50\times10^{-6}-1.60\times10^{-4}$	4
Noyes et al.[31]	Sodium	68.6	1.27	0.635	36.0	0.005 80～0.021 1	$4.22\times10^{-4}-7.64\times10^{-4}$	4
							Total	301

* Monoisopropylbiphenyl



(L)：L 領域，(H)：H 領域，(N)：N 領域，ⓑ：式(4)，ⓒ：式(5)で $\rho_v/\rho_l=0.048$，ⓓ：式(6)で $\rho_v/\rho_l=0.048$，ⓕ：式(7)

図 3　q_{c0} の実験データの無次元整理（記号に関し表 2 参照）

ば，χ_{ex} が1に近い高出口クオリティ下の CHF に限り，上記の仮想的方法で実験データの整理が可能である．そこで本報でもこのことを考慮しつつ，まず文献（8）では，その表Ⅱのデータのうちで $\chi_{ex}>0.75$ のデータ5個（実験 No. 691～695）を用い，また文献（10）では，その表2のデータのうちで $\chi_{ex}>0.75$ のデータ 13 個（実験 No. 2582～2588 および 2599～2604）を使用した．なお文献（37）のデータについては，χ_{ex} の値が 0.75 以上のものがないので使用を差しひかえた．

（iv） 文献（2）～（24）では，ΔT_i のある値（特定の）における q_c のデータが与えられている．そこで著者の前報[5]の式（14），（16）を用いて，上記の q_c の値から q_{c0} の値を推定して求めた．この際，$\Delta T_i=10°C$ の場合のデータを使ったが，これに対し $\Delta H_i/H_{fg}$ の値は大きくないから，q_{c0} の推定値に大きな誤差は生じないと考えられる．

2・2 実験データに関するメモ 文献（11）に報告される実験は，大気圧下のものと圧力 59.9 bar のものとで実験者，実験装置が異なる．そして大気圧下では，恐らく絞り弁による十分な脈動防止効果を加えられない状況下に実験を遂行したものと推定され，データがまったく不規則なので採用せず，圧力 59.9 bar のデータのみを使った．文献（17）の実験では，コイル状リボン加熱器を内管内に内蔵する方式で加熱をしているので，熱流束の一様分布条件，および加熱管長 l の精度に多少誤差が入る可能性を残している．また文献（19）には，重水，アルミニウム加熱管の場合のデータが報告されているが，残念なことに $\Delta T_i=0$ の近傍のデータを欠くため q_{c0} が求められず，従って軽水，ステンレス加熱管のデータのみを使用した．文献（21）では，実験データの掲示図面がきわめて小さく，従ってそれから求めた q_{c0} の値の精度は多少劣る可能性がある．文献（31）のナトリウムのデータは，全般的にばらつきが大で，ただ圧力 0.15 bar，流量 0.056 7 kg/s のデータのみ比較的に規則性を持つので本報ではこれらを用いた．なお文献（38）にも内管一様加熱のデータがあるが，質量速度 G の値が与えられていないので利用できなかった．

表 2 図3で流体の区別に用いた記号

文献番号	流体	記号	文献番号	流体	記号
(6)～(25)	Water	○	(29)	Toluene	◐
(26),(27)	R 12	△	(30)	MIPB	■
(28)	R 114	▼	(31)	Sodium	×
(29)	Acetone	●			

3. q_{c0} データの統一的図示

3・1 データの図示 2章のごとくして集めた q_{c0} の全データに関し，その液体の種類，実験条件範囲，およびデータ数を表1に示す（データの総計301個）．なおここで特にモノイソプロピルビフェニル（MIPB）の物性値に限り，文献（35），（36）の方法で推定した値を用いている．さて以下には，これらの q_{c0} のデータが，無次元量 q_{c0}/GH_{fg}，ρ_v/ρ_l，$\sigma\rho_l/G^2l$，l/d_{he} の間の関係として整理できるものと仮定する．そして l/d_{he} に関し $l/d_{he}=1$～500 にわたる 13 枚の図面を用意し，表1のデータをプロットした結果が図3である（ρ_v/ρ_l の影響の処理については後述の 3・3 節参照）．なお図3では，表2の記号を用いて流体の種類を区別している．

3・2 著者の整理式の図示 図3にはなお，著者の前報[5]の結果から，内管加熱に関する下記の整理式（4）～（9）によって，CHF の特性領域（L, H, N）[*1]，整理曲線 ⓑ～ⓓ，および H 領域と N 領域の境界 ⓕ をも示してある．

L 領域—ⓑ：

$$\frac{q_{c0}}{GH_{fg}}=0.25\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d_{he}} \quad \cdots\cdots(4)$$

H および N 領域—

ⓒ：
$$\frac{q_{c0}}{GH_{fg}}=0.12\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\times\frac{1}{1+0.008\,1\ l/d_{he}} \quad \cdots\cdots(5)$$

（図3のⓒでは $\rho_v/\rho_l=0.048$）

ⓓ：
$$\frac{q_{c0}}{GH_{fg}}=0.22\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}\times\frac{(l/d_{he})^{0.171}}{1+0.008\,1\ l/d_{he}} \quad \cdots\cdots(6)$$

（図3のⓓでは $\rho_v/\rho_l=0.048$）

H および N 領域の境界—ⓕ：

$$\frac{\sigma\rho_l}{G^2l}=\left(\frac{0.020\,6}{l/d_{he}}\right)^{1.71} \quad \cdots\cdots(7)$$

さて以上の諸式のうち，L 領域の式（4）は，ρ_v/ρ_l に関係しないので図3の各図上で1本の直線ⓑで表される．これに対し，H および N 領域の式（5），（6）は，ρ_v/ρ_l の影響を受けるが，図3に示す直線 ⓒ，ⓓ はともに $\rho_v/\rho_l=0.048$（これは圧力 68.5 bar の水の密度比に相当）とおいたものである．

3・3 ρ_v/ρ_l の影響 図3を見ると，集められた q_{c0} の実験データは，ほとんどすべて H および N 領域に分布するといえる．そして，これらの領域では，

*1 特性領域 L，H，N の名称の由来については文献（1）参照．

式（5），（6）からみて ρ_v/ρ_l の影響はわずかに $(\rho_v/\rho_l)^{0.133}$ の程度と考えられる．そこで表1の実験データを，ρ_v/ρ_l の値の大きさによって表 3，4 の二群に分け，表3に示すデータ（$0.010\,3<\rho_v/\rho_l<0.160$ の範囲にある）に関しては，ρ_v/ρ_l の値が 0.048 に比較的に近い範囲にあるので，q_{c0} の生の値をそのまま図3にプロットする一方，表 4 に示す 9 個のデータ（$0.000\,058\,0<\rho_v/\rho_l<0.003\,85$ の範囲にある）では，ρ_v/ρ_l の値の 0.048 から非常にへだたっているので，q_{c0} の生の値に $[0.048/(\rho_v/\rho_l)]^{0.133}$ を乗じて補正した q_{c0} の値を図3にプロットした．

4. 図3の結果の検討と結論

内管一様加熱の二重管内強制流動沸騰の CHF（飽和液流入の条件）について，25 編の文献から 7 種類の純粋流体に対し計 301 個の q_{c0} データを集め，円管の場合[4]と同様な方法で，それらの統一的な図式表示をおこない図3の結果を得た．そして図3の結果を検討して下記のような結論が導かれる．

（I） 文献 (6)～(31) から求めた多くの q_{c0} データは，$l/d_{he}=10\sim500$ の広い範囲にわたって，相当な規則性を示し，かつ著者の無次元整理式（5），（6）とかなりよい一致を示す．しかし l/d_{he} が 1～5 の程度に小さくなると，実験データと整理式との間に偏差が出るようである．もともと l/d_{he} が小さくなると，二重管内の流れというよりは，むしろ一様な流れの中に平行に加熱円柱面を置いた状況になるから，上記の偏差出現は当然の結果のように思われる．

（II） 図3で $l/d_{he}=10\sim500$ の範囲にある実験データの大部分は水のデータであり，それ以外の流体は，R12，R114，およびナトリウムに限られている．もっと，いろいろな流体による実験が望まれる．

（III） 図3に明らかなように，L 領域の実験データは，H 領域との境界付近のものを除くと，ほとんど存在していない．従って，この領域の実験がもっとおこなわれることが望まれると同時に，L 領域に関する著者の無次元整理式（4）は現段階では単なる暫定式と考えるべきである．

（IV） 図3の $l/d_{he}=50$ および 75 の場合に見られるように，統計的に言って，H，N 領域におけるフレオン系の q_{c0} データは，水のそれより多少低めに現れる傾向がある．そして，これは円管の場合（前報[4]参照）と同じ傾向といってよいであろう．

（V） 円管の場合と，外管加熱の二重管の場合に見いだされている HP 領域[1]~[5]に関して，内管一様加熱の場合，図3にプロットし得る実験データがまだ提出されていないようである．

表 3 図3のデータ（表4のデータは除く）の l/d_{he}，ρ_v/ρ_l の実験範囲

公称 l/d_{he}	$\frac{l}{d_{he}}$	$\frac{\rho_v}{\rho_l}$	文献番号
5	7.05	1.03×10^{-2}	(30)
10	7.64～14.5	4.05～16.0	(11),(18),(20),(21)
25	18.4～26.8	4.84～16.0	(9),(12),(21)
35	33.8～40.9	4.84～16.0	(12),(21)
50	42.0～59.0	2.61～8.49	(7),(9),(12),(13),(15),(27)
75	64.6～85.5	4.82～8.49	(7),(12),(27)
100	90.3～116	1.83～16.0	(12),(13),(21),(28)
150	123～172	4.84～13.5	(12),(14),(15),(17),(22),(23),(25)
200	173～180	4.84	(12),(14),(28)
300	283～348	2.13～4.84	(10),(12),(14),(16),(26)
400	316～443	2.13～4.84	(6),(8),(10),(15),(16)
500	509～553	2.13	(8),(10)

表 4 ρ_v/ρ_l の値が特に低いデータ

公称 l/d_{he}	$\frac{l}{d_{he}}$	$\frac{\rho_v}{\rho_l}$	データ数	文献番号
1	1.15	$0.290\sim0.385\times10^{-2}$	2	(29)
25	23.3	0.123～0.224	3	(19)
35	36.0	0.005 80～0.021 1	4	(31)
		Total	9	

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討　　論

〔質問〕　**鳥飼欣一**（日本原子力研究所）

限界熱流束の実験では，サブクーリング（ΔH_i や ΔT_i など）が大きい場合，データのばらつきが少ないようであるが，液が飽和温度，すなわち，$\Delta H_i=0$ 近傍では，データのばらつきの多い実験が多く，また，$\Delta H_i=0$ の近傍の実験データのないものは，著者も $\Delta H_i=0$ のデータからの外そうで取扱っておられることから，$\Delta H_i=0$ の近傍の事象は，本論文の範囲外のことのように思われるが，どうか．

〔回答〕　すでに著者の前報[5]に対し同じ質問者から同趣旨の質問をいただいている．そのとき，当時までに検討したデータに関し $\Delta H_i=0$ 付近に特殊状況は見いだされなかったことを実例も付して回答した．なおそれ以後の研究(円管の場合も含む)で，上記の問題に特別の注意を払ってきた．しかし今回もまた前と同じ回答の線（$\Delta H_i=0$ の近傍の事象を範囲外とする必要はない）のままになるように思う．その理由の二，三を以下に述べる．

（1） 2·1 節（ii）に記す実験データについて円管の場合も含めて次のことが言える．つまり特にソ連のデータでは $\Delta H_i=0$ の前後にわたる実験のものが大部分といってよいが，それらのどの結果をみても $\Delta H_i=0$ の近傍に特殊状況（ばらつきが多いなど）はみられず，$\Delta H_i<0$ と $0<\Delta H_i$ の両領域の間をなめらかに内そうする形で q_{c0} の値を求めることができる．

（2） 西欧側の実験では $0<\Delta H_i$ の条件下のものが多いが，それでも $\Delta H_i\fallingdotseq 0$ の状況の実験をしているものもかなりある．しかし特にばらつきが多いという状況は出ていない．

（3） $0<\Delta H_i$ の条件下の実験報告で，$\Delta H_i=0$ の近傍のデータが少ないことに対し，ばらつきが多いのでデータの報告がないと，もし勝手な推測をしたりしたとすれば不可であろう．その報告の著者がデータの取捨選択に対し，上記の判断をしていたとすれば，そのことを報告に明記してあるはずである．およそ $0<\Delta H_i$ の条件の実験装置では，限界熱流束の特性が原因で $\Delta H_i=0$ の近傍のデータを欠くのではなく，単に $\Delta H_i=0$ の近傍でポンプの運転が困難（キャビテーション発生）になることなどのためである．

（4） $0<\Delta H_i$ の条件下の限界熱流束の整理に対し，沸騰長さの概念が有力であることがよく知られている．この概念は，加熱管の途中でクオリティ $\chi=0$ に到達した位置から以後の状況は，$\Delta H_i=0$ の入口条件下の状況に等しいと考えるものである．従って $\Delta H_i=0$ の近傍の実験にばらつきが多いとすれば，$0<\Delta H_i$ の実験でもすべてばらつきが生じておかしくないことになる（もちろん加熱管途中の $\chi=0$ の位置の流れの状況と，$\Delta H_i=0$ の管入口の流れの状況は厳密には一致しないから速断はつつしむべきであろうが）．

（5） 現時点では以上のようなことを考えるが，あくまで経験的な結論であるから，さらに将来も，この問題に注意を払って行く必要はあると思う（質問者のいわれる状況にはあまりならないと考えるが）．

M49　短い矩形流路（一面加熱）内の限界熱流束

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§1 序

沸騰現象における限界熱流束は，工業上極めて重要な問題であり，とくに，管内における強制流動系については，原子炉，超伝導装置の設計等の観点から，数多くの研究がなされている。加熱面上に流体を吹き付ける，いわゆる噴流系についての実験研究もなされており，門出は，円形加熱面を用いた衝突噴流系，石井は，矩形加熱面を用いた平面噴流系について，限界熱流束の無次元整理式を得ている。

本研究は，石井が使用したものと同一の矩形加熱面を用い，ただし流路の四方が剛体壁で囲まれた場合の，矩形流路内における沸騰実験を行ない，矩形流路内流動系の限界熱流束に関しての考察を試みるものである。

§2 実験装置および方法

実験装置の全体図を図1，主要部を図2，テストセクションを図3に示す。予熱タンクで飽和温度近傍まで加熱された流体（本研究では水を用いている）は，ポンプにより，ループ内を循環させられる。流体は，途中で補助ヒータで加熱され，飽和温度でテストセクションに入る。テストセクションは図に示すような，一面から加熱された矩形流路で，加熱面付近を一様流に近い状態にさせるため，流路直前に絞りを設けてある。また，テストセクション出口は大気に開放されており，テストセクション内の圧力はほぼ大気圧である。流体は，テストセクション内の加熱面上で核沸騰を行なう。加熱面からの熱の供給は，銅ブロック（その頭部角柱部の上面が加熱面となっている）を介しての電気加熱により行なっている。

実験は，流量，熱流束に関して，定常状態で行なう。流量を設定量に調節した後，加熱量をステップ状に増加させて，加熱面温度が急上昇し始めた点で終了する。流量は，テストセクション上流のオリフィス，熱流束は，ブロック頭部に埋め込まれた熱電対により測定する。

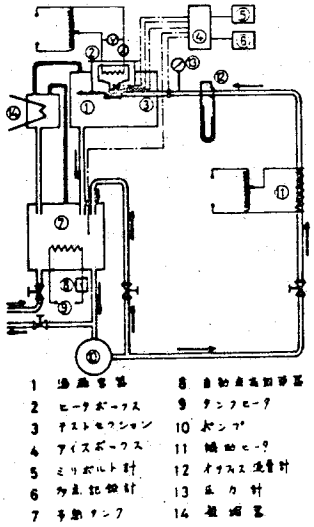

図1　実験装置全体図

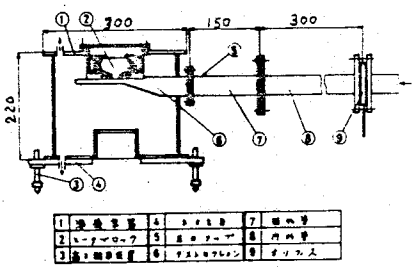

図2　実験装置主要部

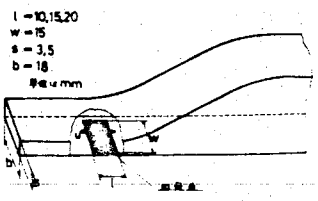

図3　テストセクション

§3 実験結果および考察

流速（体積流量を流路断面積で割って求めたもの）uに対して，限界熱流束 q_{cr} をプロットしたものを図4に示す。噴流系と同様に，uの増加に伴って q_{cr} が増加しているが，流れ方向の加熱面長さ l については，本研究では，lが大きいほど q_{cr}が大きい，という結果が現われており，石井とは全く逆の様相を呈している。流路高さSの q_{cr} への影響は，本研究の範囲内ではほとんど認められない。

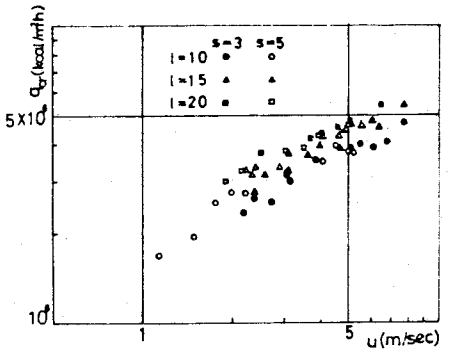

図4　q_{cr}とuの関係

無次元数 $\sigma/\rho_l u^2 l$ に対して無次元数 $q_{cr}/\rho_v u h_{fg}$ をプロットしたものを図5に示す。ここで，ρ_l, ρ_v：液体，気体の密度，h_{fg}：蒸発潜熱，σ：表面張力。最小二乗法により求めた結果，限界熱流束は次の無次元式で整理することができる。

$$\frac{q_{cr}}{\rho_v u h_{fg}} = K\left(\frac{\sigma}{\rho_l u^2 l}\right)^{0.267} \quad (1)$$

$l=10$: $K=5.96$
$l=15$: $K=7.77$
$l=20$: $K=8.92$

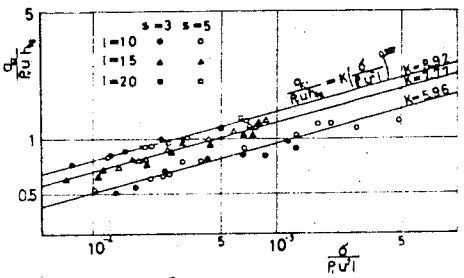

図5　$\frac{q_{cr}}{\rho_v u h_{fg}}$ と $\frac{\sigma}{\rho_l u^2 l}$ の関係

さらに無次元数 l/s_0（s_0は，流路高さ方向の仮定した代表長さで，本研究の範囲内では一定値をとるもの。ここでは便宜上，$s_0=3$ mmとしている。）を用いると，次の無次元式で整理することができる。

$$\frac{q_{cr}}{\rho_v u h_{fg}} = 2.97\left(\frac{l}{s_0}\right)^{0.586}\left(\frac{\sigma}{\rho_l u^2 l}\right)^{0.267} \quad (2)$$

（図6参照）

石井は，平面噴流系について，次式を得ている。

$$\frac{q_{cr}}{\rho_v u h_{fg}} = 1.64\times10^{-2}\left(\frac{\rho_l}{\rho_v}\right)^{0.867}\left(\frac{\sigma}{\rho_l u^2 l}\right)^{\frac{1}{3}} \quad (3)$$

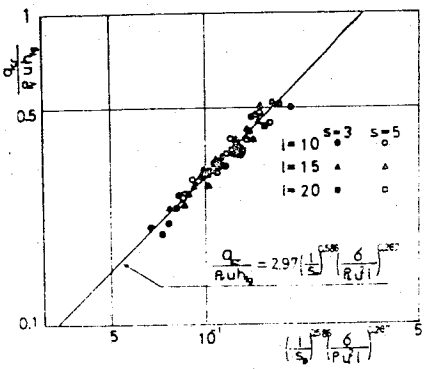

図6　q_{cr}の無次元整理

本研究では，密度比 ρ_l/ρ_v の影響については不明であるが，(2),(3)を比較すると，lの指数に大きな差があり，両者は，機構的に異った現象であると考えられる。平面噴流系では，加熱面上の核沸騰液層なる薄い液膜を通して，蒸気の大部分が，q_{cr}に関与する流れから自由に離散してしまうのに対し，本研究では，ウォリティが流路全域にわたって殆んどゼロであり，発生した蒸気はあくまでも流路内にとどまっている。このようなことが，両者の相違に関係しているものと思われる。

§4 結論

一方の面から加熱された，比較的短い矩形流路内の沸騰実験を，水を用いて行なった結果，限界熱流束に関して，平面噴流系との相違が確認され，実験式(2)を得た。

〔401〕　一様加熱の矩形流路内の強制流動沸騰の限界熱流束の無次元整理

正　甲藤好郎　（東大）

1. 序

一様に加熱された矩形流路内の限界熱流束（CHF）に対して，古くMacbeth[1]は整理式を導いているが，それは水だけを対象とした経験式である．またMacbethw以外の研究では，円管のCHFに対して導いた整理式がそのまま矩形管の場合にも適用できるとしているものが多い．さて著者は最近，一様加熱円管[2~7]および一様加熱二重管[8,9]内のCHFの研究をおこない，広い条件範囲内の実験データの特性をかなり良く予測できる無次元整理方法を見出した．そこで本報では，一様加熱矩形流路内のCHFに対し，上と同様な方式に従って既存の実験データの整理を試み，その方式の有効性を確かめると同時に，矩形流路の場合のCHFの特性を明らかにするための研究をおこなった．ただし矩形流路壁の全周加熱では，出口近傍の流路壁の隅の部分にオーバーヒートをおこすことが知られており，従って本報では図1のように対向2壁面加熱（A型）および1壁面加熱（B型）の場合を取扱う．これらの場合でも隅の部分のオーバーヒートの危険を避けるため，図1のごとく加熱面中wを流路中bより僅かに小さくすることが多い．またA型では，対向する2加熱面からの加熱量は互にひとしいものとする．このとき流路断面の高さに対し図1A型，B型にそれぞれ示した方式でsの数値を定めるならば，加熱相当直径 d_{he} =（4×流路断面積）÷（加熱辺の長さ）の値は，図1のA型，B型共通に次式で与えられる．

$$d_{he} = 2sb/w \qquad (1)$$

本報ではこの方式により，A型，B型をまとめて扱う．なお一様加熱の流路内のCHFの整理にあたり，加熱相当直径が重要な働きをする理由づけについては，二重管に関する論文[8]の3.2節に記してある．

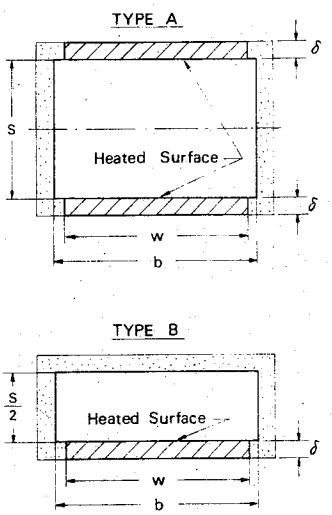

図1. 流路断面の型式

記号　b：流路断面の巾，C：定数［式(3)］，d_{he}：加熱相当直径［式(1)］，G：質量速度，H_{fg}：蒸発潜熱，ΔH_i：入口サブクール・エンタルピ，K：ΔH_iの影響係数［式(6)］，l：加熱板の長さ（流路軸方向），q_c：限界熱流束，q_{co}：基礎限界熱流束（$\Delta H_i = 0$の時のq_c），s：流路断面の高さ（図1参照．A型では実際の高さ自体，B型では実際の高さの2倍にとる），w：加熱板の巾，δ：加熱板の厚さ，ρ_ℓ：液体密度，ρ_v：蒸気密度，σ：表面張力．

2. 限界熱流束

2.1 q_{co}のデータ　図2はTroy[12]の実験結果の与える限界熱流束q_cと入口サブクール・エンタルピΔH_iの関係の一例である．こうした結果から，基礎限界熱流束q_{co}（ΔH_i=0におけるq_c）の値を容易に求めることができる．表1は，こうしてq_{co}を求めたデータ源とその実験条件範囲などを示す．

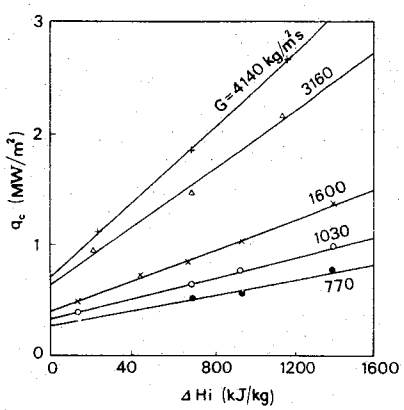

図2. q_c-ΔH_iの直線関係（文献[12]の実験データから）．水，p=138 bar，l=190 cm，s=0.190 cm，b=w=5.71 cm，A型．

表1. 集めた q_{co} および K のデータの一覧表

文献	流体	型	l[cm]	s[cm]	b[cm]	w/b	l/d_{he}	$(\rho_v/\rho_l)\times10^2$	$\sigma\rho_l/G^2l$	データ数	δ[mm]
DeBortori et al.[11]	Water	A	15.2	0.257	2.54	0.880	26.1	3.67-13.6	2.80×10^{-5}-5.61×10^{-3}	10	1.27
DeBortori et al.[11]	Water	A	30.6	0.246	2.54	0.880	54.7	13.6	9.43×10^{-7}-3.11×10^{-4}	8	1.27
DeBortori et al.[11]	Water	A	30.6	0.127	2.54	0.880	106	2.61-13.6	3.06×10^{-6}-2.12×10^{-4}	8	1.27
DeBortori et al.[11]	Water	A	68.6	0.246	2.54	0.880	122	3.67-13.6	3.00×10^{-7}-8.53×10^{-5}	10	1.27
DeBortori et al.[11]	Water	A	68.6	0.150	2.54	0.880	201	3.67-13.6	3.02×10^{-6}-3.46×10^{-4}	7	1.27
Troy[12]	Water	A	190	0.190-0.195	5.71	∿1.0	487-500	6.16-13.6	1.24×10^{-7}-1.99×10^{-5}	9	——
Gambill-Bundy[13]	Water	A	30.5-30.8	0.109-0.125	2.65-2.73	0.930-0.958	114-132	1.96-2.42	7.45×10^{-7}-1.03×10^{-6}	3*	1.52
Tippets[14]	Water	A	94.0	0.635	5.33	1.00	74.0	4.84	3.65×10^{-6}-5.97×10^{-5}	3	0.254
Tippets[14]	Water	A	94.0	1.27	5.33	1.00	37.0	4.84	1.43×10^{-5}-5.55×10^{-5}	2	0.254
Tippets[14]	Water	B	94.0	2.54	5.33	1.00	18.5	4.84	1.41×10^{-5}	1	0.254
Tong et al.[15]	R-113	B	63.5	1.27	6.03	∿1.0	25.0	0.587-0.719	6.97×10^{-6}-1.28×10^{-4}	3	∞ **

* ニッケル加熱板のデータ　　** 非常に薄い加熱板

ただし表1のデータ源のうち，Gambill-Bundy[13]は G, ΔH_i の各一つの組合わせごとに q_c のデータを与えるもので，図2の直接的方式で q_{co} が求められない（その求め方は2·4節[注意]に記載．Tongら[15]の R-113 のデータも上と同形式だが，ΔH_i が零に近いので近似的に $q_c \fallingdotseq q_{co}$ とみなして q_{co} を求めている．またTippetsら[14]の実験では，加熱板の厚さ δ が非常に薄く $\delta=0.254, 0.152$ mm である．従って本報では，加熱板の厚さの影響をなるべく避けるため $\delta=0.254$ mm の方のデータを採用した．なお表1に示す実験のうち，Gambill-Bundy（垂直流路下降流）のものを除き，他はすべて垂直流路上昇流である．またLevy[16], Chernobay[17], その他は全周加熱の実験なので除外している．

2.2 q_{co} の無次元整理　表1のデータ源からの q_{co} の実験データを l/d_{he} の値で分類し，円管[5]，二重管[9]の場合とまったく同じ方式で図示した結果が図3である．表1にみるように，これらのデータは $\rho_v/\rho_l=0.048$ に比較的に近い条件下の実験で得られているので，図3には q_{co} の実測値をそのまま図示している．ただしTongら[15]のデータは，ρ_v/ρ_l の値が 0.048 に比べかなり小さいので，円管や二重管の場合を参考にして，q_{co} の実測値に $\{0.048/(\rho_v/\rho_l)\}^{0.133}$ を乗じ補正した値を図3に図示している．

一方，図3に示す直線ⓐ, ⓑ, ⓒ, ⓓ はそれぞれ下記の各式を示すものである．

$$\text{ⓐ}: \frac{q_{co}}{GH_{fg}} = 0.25\frac{1}{l/d_{he}} \quad \cdots\cdots (2) \qquad \text{ⓑ}: \frac{q_{co}}{GH_{fg}} = C\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.043}\frac{1}{l/d_{he}} \quad \cdots\cdots (3)$$

$$\text{ⓒ}: \frac{q_{co}}{GH_{fg}} = 0.15\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{1/3}\frac{1}{1+0.0077\,l/d_{he}} \quad \cdots\cdots (4)$$

$$\text{ⓓ}: \frac{q_{co}}{GH_{fg}} = 0.26\left(\frac{\rho_v}{\rho_l}\right)^{0.133}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.433}\frac{(l/d_{he})^{0.171}}{1+0.0077\,l/d_{he}} \quad \cdots\cdots (5)$$

ここで式(4), (5)右辺には ρ_v/ρ_l が含まれているが，図3の直線ⓒ, ⓓは式(4), (5)で $\rho_v/\rho_l=0.048$ としたものである．

さて図3によれば，直線ⓐ, ⓑの領域の実験データは非常に少ないことがわかる．従って，これら短形流路のデータから q_{co} の整理式を定めることは不可能である．図3で直線ⓐ, ⓑを与える式(2), (3)は，円管[5]の場合の整理式をそのまま（ただし直径 d の代りに加熱相当直径 d_{he} を使う）一時的に使ったものである．q_{co} の少ない実測値にはほぼ一致し，かつ式(3)の定数 C に関し，円管の場合と同様に l, d_{he} が小

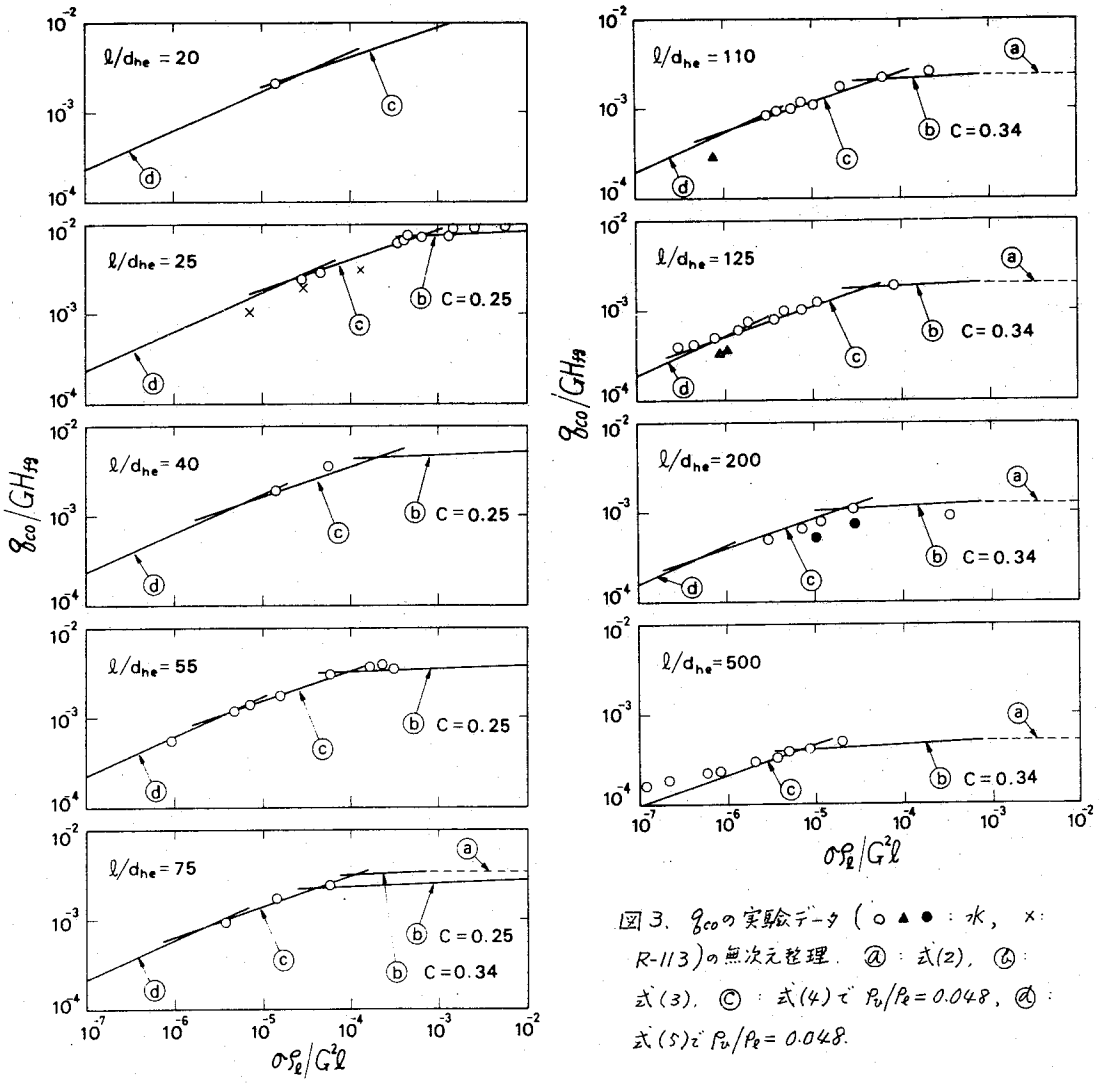

図3. q_{co}の実験データ（○▲●：水，×：R-113）の無次元整理．ⓐ：式(2)，ⓑ：式(3)，ⓒ：式(4)で$P_v/P_\ell=0.048$，ⓓ：式(5)で$P_v/P_\ell=0.048$．

さいとき $C=0.25$，l/d_{he} が大きいとき $C=0.34$ となる傾向があるようにみえる．

次に図3にみるごとく，既存実験データの大部分は直線ⓒの領域に集中している．そして直線ⓒを与える式(4)は，これらの実験データに合うように定めた（文献[8]の図4の方法による）ものである．実験データに関し二三の注意を以下に記す．まず図3の $l/d_{he}=25$ に示す×印はTongら[15]のえるR-113のデータである．円管[5]，二重管[9]の場合もフレオンのデータは水のそれより僅かに低目に出る傾向がみられたが，なおTongらの実験は他の実験（表1の）と異なり，直接通電加熱でなく，水蒸気の注入加熱によるもので一様熱流束条件から少し偏る可能性も含んでいる．また図3の $l/d_{he}=110$，125 に示す▲印は2.1節に述べた Gambill-Bundy[13] のデータ，つまり2.4節L注記]に記す間接的方法で求められたものである．なおこの実験は垂直流路下降流のもので，従ってデータがやや低目に出るのが普通であるが，円管の場合の例（文献[3]の図6）から判断すると図3の▲印のデータは低くすぎるように思われる．図3の $l/d_{he}=200$ に示す2個の●印は，DeBortoliら[11]の統計処理のデータ（表1参照）中に含まれるものであるが，2.4節に述べるごとく，入口サブクール・エンタルピの影響の面でも異常性を示している．

最後に図3で ⓓ の領域の実験データはほとんど無いに等しい．従って直線ⓓを与える式(5)は暫定的なもので，二重管の場合の整理式の形を参考にしつつ図3の $l/d_{he}=20, 55, 125$ などを基に定めたものである．

2.3 CHFの特性領域に関する注意　円管[6,7]の場合，CHFはL，H，N，HPの4領域に区分された．ここにL領域は質量速度が低いとき，N領域は質量速度が高いときに生ずるもので，H領域は前二者の中間領域である．またHP領域は，圧力が高いとき質量速度の大きな領域でN領域の代りに出現する領域である．さて本報の場合，L領域とH領域の境界は，図3で直線ⓑとⓒの交点として容易に求められる．しかしN領域の実験データがほとんど存在しないため，H領域とN領域の境界を与える式が求められない．なお図3の $l/d_{he}=500$ において，図の左端付近のデータは直線ⓒから上方へかなり偏っているのがみられる．これらのデータは気液出度比 $\rho_v/\rho_l=0.136$ とかなり大きく，かつ l/d_{he} の値も大であり，従ってHP領域に入っているものと推察される（円管の場合のHP領域発生判定法[6]によればHP領域に入る条件を十分にみたしている）．

2.4 Kのデータの無次元整理　表1に集めたデータ源の実験では大部分，図2のような $q_c-\Delta H_i$ 直線関係を示す．この直線関係に対し次式が書ける．

$$q_c = q_{co}\left\{1+K\cdot(\Delta H_i/H_{fg})\right\} \tag{6}$$

ここにKは入口サブクール・エンタルピ ΔH_i の影響係数（無次元）で，その値は図2の各直線の勾配から容易に求められる．一方，著者の前報[4]に明らかにしたごとく，もし滞時長さの概念が $\Delta H_i=0$ に近い入口サブクール条件に成立すると仮定すれば，q_{co} に関する整理式(2)〜(5)の各式からそれぞれ対応するKの値を理論的に導くことが出来て次の結果を与える．

式(2)から：　$K=1$　(7)

式(3)から：　$K=\dfrac{1.043}{4C(\sigma\rho_l/G^2l)^{0.043}}$　(8)

式(4)から：　$K=\dfrac{5}{9}\cdot\dfrac{0.0308+d_{he}/l}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{1/3}}$　(9)

式(5)から：　$K=0.252\,\dfrac{(0.0371+d_{he}/l)(d_{he}/l)^{0.171}}{(\rho_v/\rho_l)^{0.133}(\sigma\rho_l/G^2l)^{0.433}}$　(10)

さて2.2節に述べたように，表1に集めたデータ源のCHFデータは大部分，整理式(4)に対応するものである．従って式(4)に対応する実験から求めたKの実測値を，式(9)による予測値と比較してみた結果が図4である．図4によれば，Kの実測値は式(9)に近似的に一致する．しかし図4に示すデータに関する限り，統計的にKの実測値は式(9)よりやや小さい値（約80%程度）をとることがみられる．現在のところ，その原因は不明である．なお図4で2個の●印の実験データは，図3の $l/d_{he}=200$ の●印と同じ実験のものである．

［注意］　図3の $l/d_{he}=110, 125$ に示す▲印のデータは，式(6)，(9)［ただし式(9)のKの80%の値を使用］を用い，q_c の実験データから間接的に求めた q_{co}

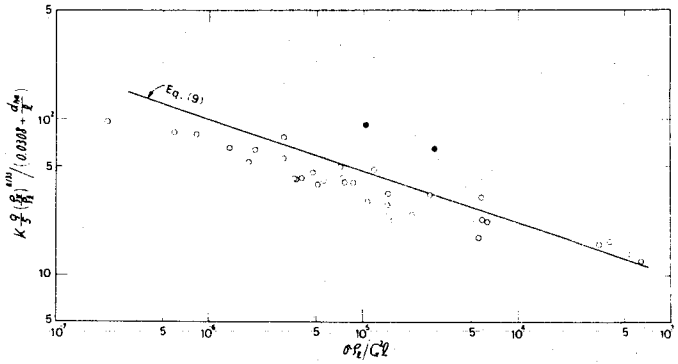

図4．Kに関する式(9)と実験データの比較

表2. l/d_{he} が非常に小さい場合に対して集めた q_{co} のデータ一覧表

文献	流体	型	l[cm]	s[cm]	b[cm]	w/b	l/d_{he}	$(\rho_v/\rho_l)\times10^2$	$\sigma\rho_l/G^2l$	データ数	δ[mm]
一色ら [18]	Water	B	30.0	2.20	1.70	0.882	6.02	0.0624	8.37×10^{-3}-9.31×10^{-2}	5	∞ *
Yücel-Kakaç[19,20]	Water	B	12.7	1.27	0.652	0.896	4.47	0.0624	1.11×10^{-5}-2.79×10^{-4}	4	0.178
Yücel-Kakaç[19,20]	Water	B	12.7	1.27	0.952	0.613	3.07	0.0624	1.11×10^{-5}-2.79×10^{-4}	2	0.178
後藤 [21]	Water	B	1.00	0.600	1.80	0.833	0.694	0.0624	1.01×10^{-4}-1.23×10^{-3}	10	∞ *
後藤 [21]	Water	B	1.50	1.00	1.80	0.833	0.625	0.0624	1.33×10^{-4}-8.24×10^{-4}	9	∞ *
後藤 [21]	Water	B	1.00	1.00	1.80	0.833	0.417	0.0624	2.38×10^{-4}-4.73×10^{-3}	10	∞ *

* 非常に厚い加熱板

の値を示すものである.

3. l/d_{he} が非常に小さい場合の限界熱流束

内管を加熱する二重管のCHFの既存データの無次元整理の研究[9]において, l/d_{he} の値が1～5程度の大きさまで減少すると, 図3の形式の整理からの偏りを生ずることが見出されている. 次いで甲藤・倉田[10]は, 速度の一様な飽和液の流れに平行な平面加熱面上のCHFについて, 水およびR-113の実験をおこない, 実験データに対し次の無次元整理式:

$$\frac{q_{co}}{GH_{fg}} = 0.186\left(\frac{\rho_v}{\rho_l}\right)^{0.559}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.264} \qquad (11)$$

を等くと共に, 式(11)は l/d_{he} のきわめて小さな内管加熱二重管の q_{co} のデータ(アセトン, トルエン, モノイソプロピルビフェニル, 水)にも一致することを示した. 同様な状況が矩形流路の場合にも当然予想されるので以下の検討をおこなつた.

すなわち表2は, 表1と同様なものであるが, ただ l/d_{he} が非常に小さい矩形流路に対し q_{co} を与えるデータ源と, その実験条件範囲を示すものである. これらの実験はすべて図1のB型である. そして一色ら[18]のものは水平流路(下型面加熱), Yücel-Kakaç[19]のものは第1行は水平流路(下型面加熱のもの, および上型面加熱のもの), 第2行は垂直流路上昇流, また後藤[20]のものはすべて水平流路(上型面加熱)の形式の実験である. なおKakaç[20]によれば文献[19]の実験において $l=127mm$, $w=5.84mm$ ($5.50mm$ ではなく)である.

さて図5は, 上記の実験データを図示するもので, 同時に矩形流路の q_{co} の整理式(4), (5), および流れに平行な平面加熱面の q_{co} の整理式(11)を示している. ただし図5の実験データはすべて大気圧の水のものである. 従つて式(4), (5), (11)の図示にあたり $\rho_v/\rho_l = 0.000624$ (大気圧の飽和水蒸気, 飽和水の値)にとつている.

図5によれば以下のようなことが観察される. まず $l/d_{he}=6.0$ では, q_{co} の実測値(一色らのデータ)はほぼ直線ⓒの近傍に現われている. 次に $l/d_{he}=3.0$, 4.5 に示すデータはYücel-Kakaçによるものである. 垂直上昇流, 水平流路で下型面加熱, 上型面加熱によって q_{co} の値に相違が生じ, 彼等はそれを重力の作用によるものと主張している. しかしいま, 上記の相違の問題から離れて $l/d_{he}=3.0$, 4.5 の結果をみるに, q_{co} の実測値は直線ⓒ, ⓐより式(11)に近いことが知られる. ただ, これらのデータは表2にみるごとく, 加熱板の厚さが非常に薄い ($\delta=0.178mm$)実験によるので, 通常より q_{co} が低く出る可能性がないとはいえぬかも知れない. けれども図5で $l/d_{he}=0.5$ のデータ(後藤のもの)は非常に厚い加熱面で得られたものである. また l/d_{he} がこれ位小さくなると, 流路の入口から出口までクオリティが零に近く保持され, 流路のオリエンテイションや加熱位置の相違はCHFに対し大きな影響を与えなくなるのではないかと想像される. 結局, l/d_{he} がきわめて小さな矩形流路では, q_{co} は式

(11) に近い値を与えると結論してよいであろう。なお ℓ/d_{he} がきわめて小さくなると，ℓ/d_{he} は CHF にほとんど影響を持たなくなるが，図5の結果によれば ℓ/w（加熱板の長さと巾との比）が零に近いほど q_{co} が式(11)に接近する傾向を持つている．後藤[21]の実験データに関する限り，表2に掲げた実験条件以外のデータをも含めて，暫定的に次式が得られる．

$$\frac{q_{co}}{GH_{fg}} = 0.186\left(\frac{\rho_v}{\rho_\ell}\right)^{0.559}\left(\frac{\sigma\rho_\ell}{G^2\ell}\right)^{0.264}\left\{1+0.53\left(\frac{\ell}{w}\right)^{1.78}\right\} \qquad (12)$$

本式を求めた実験条件範囲は，$\ell/d_{he}=0.417\sim1.39$，$\ell/w=0.667\sim1.33$，$w/\ell=0.833$ である．

4. 結論

(i) 図1に示すような断面の一様加熱矩形流路内の強制流動沸騰の CHF に対し，既存の実験データを分析し，基礎限界熱流束 q_{co} に対しては図3および整理式(2)～(5)，また入口サブクール・エンタルピの影響係数 K に対しては図4および式(7)～(10)の結果を得た．また ℓ/d_{he} が非常に小さい場合についても検討し，図5の無次元整理，式(12)の結果を得た．

(ii) 矩形流路内の強制流動沸騰の場合，既存実験データの数が少なく，実験条件範囲も限られ，かつほとんど水のデータである（表1，2参照）．従って CHF の無次元整理の整備のためには，さらに多くの実験的研究が必要と思われる．しかし本報の無次元整理は，円管や二重管の場合に似た結果を示している．従ってその観点から考えると，精度があまり必要でない場合には，本報の結果でも，いろいろの条件下の CHF の予測に対し有用と考えられる．

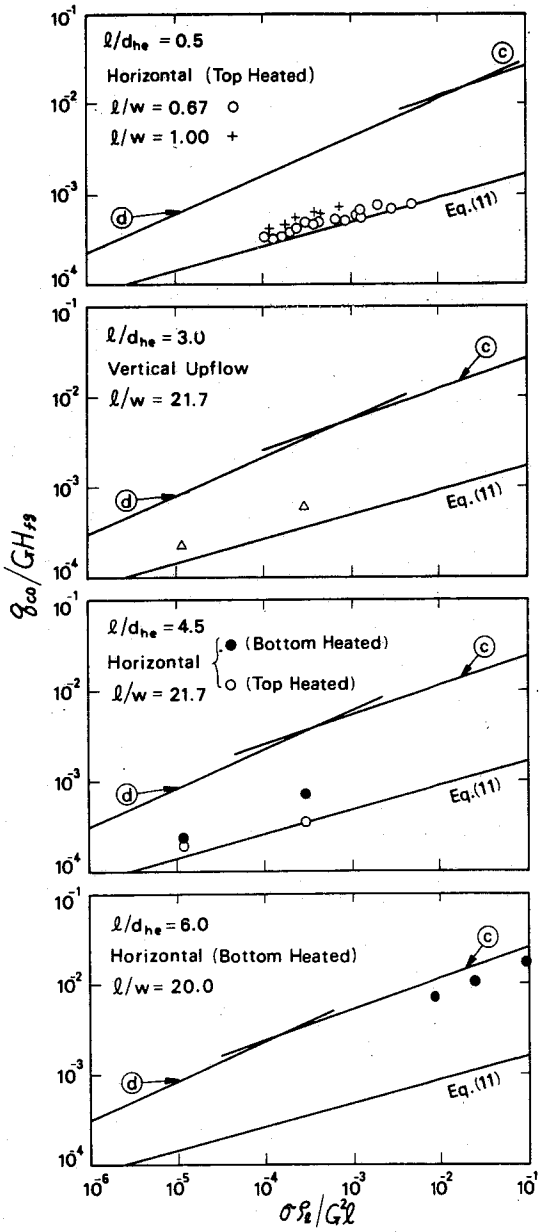

図5. ℓ/d_{he} が非常に小さい時の q_{co} の実験データの無次元整理（すべて大気圧の水）．式(4)，(5)，(11)で $P_v/P_\ell = 0.000624$．

(iii) 内管加熱二重管内の CHF の分析[8,9]では，L領域の q_{co} に対し，その領域の実験データの不足から，式(3)で $C=0.25$ としたものだけを暫定的に考えた．しかし本報の結果をみると，円管[5]の場合と同様に，矩形流路の場合も ℓ/d_{he} の大小によって式(3)の C の値が 0.25～0.34 の間で変化するもののようである．これから類推して二重管の L 領域でも同様な状況にあると考えられる．

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