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LATENT HEAT TRANSPORT IN FORCED BOILING FLOW

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Abstract—This paper presents theoretical models describing the mechanisms of heat transfer for nucleate forced boiling and for boiling in the fluid core, i.e. without nucleation at the heat exchange surface. The models result from experimental studies of the hysteresis effect of heat transfer carried out by other workers as well as by the author. In particular, the author's experiment provided data to support the hypothesis that the increase in the heat transfer coefficient from the wall to the flowing fluid is essentially due to vapor bubbles acting as thermal sinks within the thermal boundary sublayer. The above conclusion does not agree with the view that bubbles formed within the thin liquid film for bubble or slug flow regimes affect the forced convection mechanism, which is in the main the only cause of heat transport intensification.

NOMENCLATURE

- c, specific heat;
- D, inner diameter of pipe;
- f, friction coefficient;
- k, fraction of the heat flux absorbed by the bubbles within thermal sublayer;
- r, radius coordinate;
- T, temperature;
- q, heat flux;
- w, velocity;
- z, coordinate in the direction of the flow;
- x, quality, i.e. dynamic dryness fraction, $\dot{m}''/\dot{m}$.

Greek symbols

- α, heat transfer coefficient;
- δ, boundary sublayer thickness;
- ε, turbulent diffusivity ('eddy viscosity');
- ϕ, void fraction;
- ρ, density;
- μ, absolute viscosity;
- λ, thermal conductivity;
- τ, shearing stress, time;
- σ, surface tension;
- ψ, mass flux rate, $w\rho$;
- Ω, total amount of heat flux supplied to the pipe.

Other symbols

- Re, Reynolds number;
- Pr, Prandtl number.

Subscripts

- a, accelerational term;
- bl, boundary sublayer;
- c, core of flow;
- f, frictional;
- h, hydraulic, hydrostatic;
- p, isobaric condition;
- r, radial;
- s, saturation condition, surface;
- T, thermal;
- TP, TPF, two-phase;
- w, condition at the wall.

Superscripts

- $'$, liquid condition;
- $''$, vapor condition.

1. INTRODUCTION

ALTHOUGH the mechanisms of heat transfer from a heated wall to a flowing fluid are not completely understood, it is well known that the processes within boundary layers control heat transfer in forced-convection boiling. The question is whether the bubbles generated on the surface are able to make the flow more turbulent, as does the roughness of the wall. In this case both the momentum exchange rate and the energy or heat exchange rate increase simultaneously near the wall. In other words, the frictional pressure drop and the heat transfer coefficient are supposed to change simultaneously, in accordance with the analogy between heat and momentum transfer [5]. In the light of this, the question arises as to the magnitude of the contribution made by nucleation at a heated wall to the increase in skin friction and heat transfer. A survey of the literature [1–3] leads to the conclusion that most previous works aimed at predicting either the superheat necessary to sustain nucleate boiling or the influence of surface conditions (i.e. surface roughness and trapped gas) on the incipient boiling superheats. There exists no study which makes use of the hysteresis effect as an explanation for the role of small bubbles located within the boundary layer. Whether they act as heat sinks or cause a more turbulent flow, they yield an increase in both skin friction and heat transfer coefficient. The objective of this work is to create two models which would be capable of answering this question. Both experimental and theoretical studies have been undertaken.

2. EXPERIMENTAL

2.1. *Apparatus*

An experiment has been carried out to obtain information on the influence of nucleate bubbles located at the heated wall on skin friction and heat transfer. The forced convection loop used in this study is shown in Fig. 1. The test sections, preheater, and the main loop are built of stainless steel tubing 16 mm O.D. and 13 mm I.D. The loop is designed to operate with Freon 21 at a maximum pressure of 0.6 MPa. Two identical test sections 0.5 m long are used, one being vertical and the other horizontal. They are equipped with transparent tubes at the ends. The temperature difference between the wall and the fluid is measured with the aid of three pairs of iron–constantan thermocouples. The temperature difference between the outlet and inlet of the test sections and the absolute temperature of Freon 21 are measured with identical thermocouples. The system pressures are measured with the aid of a Bourdon gauges. The pressure drops are measured with the aid of strain gauge transducers. The heat flux is measured as a product of voltage and current, i.e. indirectly by the electrical power input to the test sections and to the superheater. A pump is used to circulate the fluid in the loop through a control valve connected to a flow meter. From the flow meter, the liquid is passed through the preheater consisting of a coiled, vertical pipe. The power to the preheater and the test section can be regulated from zero to maximum, as required. In this way it is possible to obtain two-phase flow at the exit of the preheater. The two-phase stream is then passed through a 2.5 m long adiabatic channel to the vertical test section, and next to the horizontal one, where it is heated further. The two-phase mixture is subsequently condensed and subcooled before entering the pump for recirculation.

2.2. *Procedure*

In a typical run, bubble flow is developed before the inlet to the test sections. Steady conditions are obtained by holding constant the inlet temperature, flow rate, heat flux supplied to preheater and static pressure, while increasing in small steps the heat flux received by either the vertical or the horizontal test section. Sufficient time is allowed between each step to ensure that all temperatures reach steady-state conditions. In all runs a gradual increase is observed in the temperature differences between the heated wall and fluid in the test sections until a change of heat transfer mechanism takes place. This change occurs with a sudden drop in the temperature difference, or, strictly speaking, in the wall temperature. At the same time the pressure and fluid temperature drops measured between the two ends of the test section remain practically unchanged, except during transient conditions. After the point of sudden change has been reached, a gradual increase continues in the temperature difference as the heat flux increases, but at a different rate. Next the heat flux is reduced gradually in small increments. The hysteresis effect becomes apparent (Fig. 2), after the completion of a cycle of increasing followed by decreasing heat flux. All runs are carried out for the same two-phase flow pattern, namely bubble flow. The complete cycle of heat transfer is

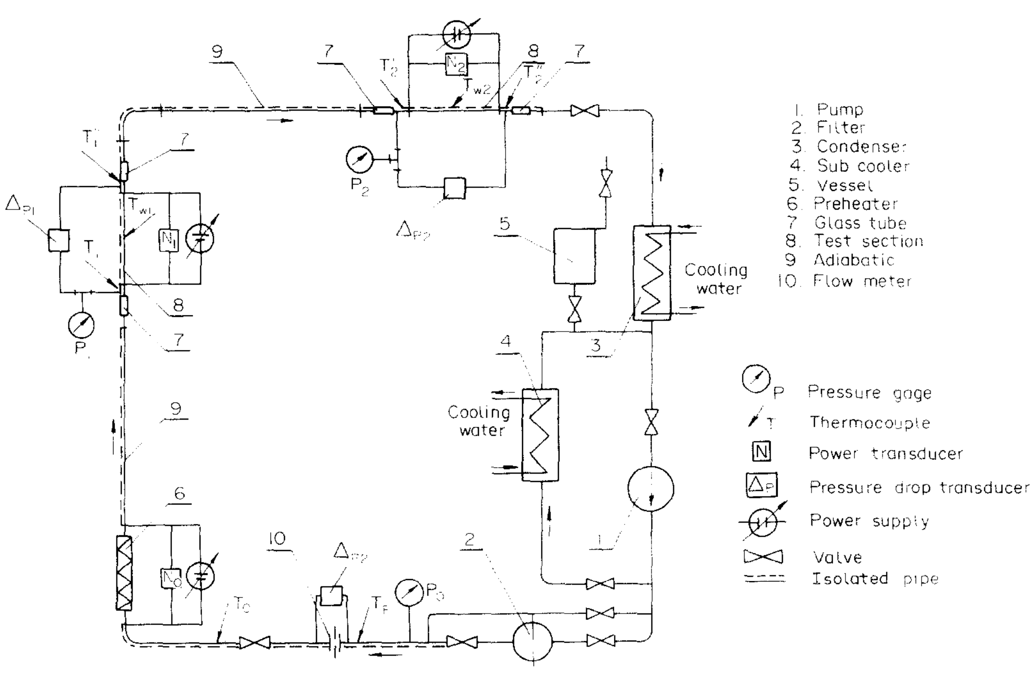

FIG. 1. Schematic diagram of heat transfer loop with Freon 21.

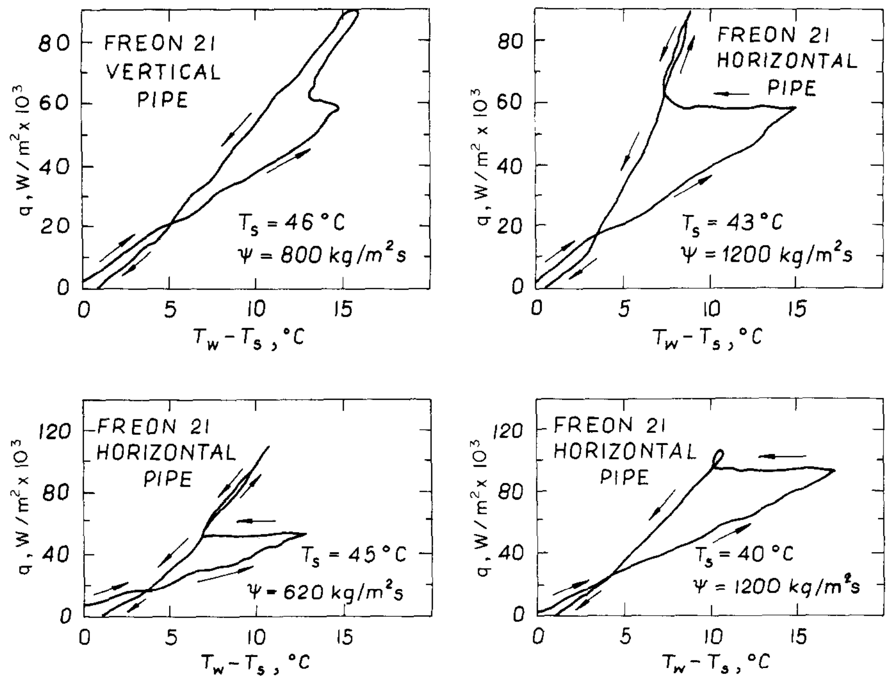

FIG. 2. Experimental curves of the hysteresis effect [4].

shown schematically in Fig. 3. This shows the two different kinds of boiling. Boiling according to line ADB is characterized by the fact that vaporization is confined to the core of the flow and at a distance from the boundary sublayer. In this case the total amount of heat transferred from the wall to the fluid is transferred by means of forced convection. We shall use the term forced-convection boiling, or for brevity f.c.b., for this distinct regime of boiling. The branch of the cycle curve in Fig. 3, denoted by ECD describes fully-developed boiling with bubbles generated on the heated surface and growing within a sublayer. This regime of boiling is called nucleate boiling, or n.b. for brevity. The phenomenon that corresponds to a sudden drop in temperature is referred to as the "zero" boiling crisis.

3. PHYSICAL MODELS

One of the most important observations which results from our experiments is that the pressure drops

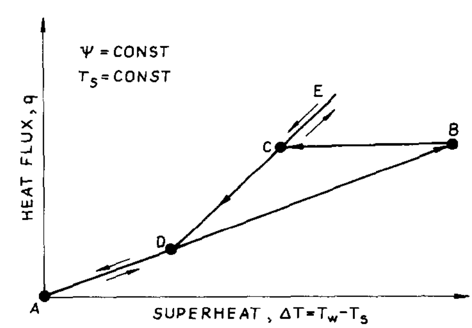

FIG. 3. The cycle of heat transfer.

measured along the test sections are almost the same regardless of whether n.b. or f.c.b. occurs before or after the "zero" boiling crisis. On the other hand, the mass flow rate, inlet temperatures, and inlet quality, as well as the void fraction, remain constant. It seems justified to say that the distribution of the void fraction along the test sections for the two different kinds of boiling under the above conditions is the same. Hence we conclude that the bubbles generated on the wall do not affect the boundary sublayer. In other words, they do not act as a stirring device which increases the intensity of turbulence within the boundary layer next to the heating surface. Thus there is only one reason for the dramatic decrease in the heating surface temperature, namely the major role played by the transport of latent heat in nucleate heat transfer. This seems that bubbles located within the boundary layer might have to be treated as heat sinks. This observation coincides with the conclusion reached by Bankoff [6] on theoretical grounds.

On the basis of the experimental data, an attempt was made to describe each kind of boiling separately. The aim of these descriptions is to explain the "zero" boiling crisis by means of simplified theoretical models. Two models have been devised, one each for f.c.b. and n.b. We will discuss under what conditions they can explain the jump in heated wall temperature, assuming that the jump is caused by a change from f.c.b. to n.b.

Our assumptions that define the simplified physical model of forced convection boiling, f.c.b. are:

(1) The boundary sublayer is filled with liquid for which the thermal conductivity, λ', is constant.

(2) Temperature and velocity profiles are subject to

change only within the boundary sublayer. Owing to turbulence, the distributions of each intensive quantity, such as temperature and velocity, are flat outside the sublayer (Fig. 4).

(3) The liquid velocity varies linearly from zero at the wall to the value of

$$w_1 = \frac{w\rho(1-x)}{\rho'(1-\phi)} \tag{1}$$

within the boundary layer (Fig. 4).

(4) The sublayer thickness is governed by the following equations [7]:

$$\tau_w = \mu'\left(\frac{w_1}{\delta_h}\right) = \frac{f}{2}\rho' w_1^2, \tag{2}$$

$$\delta_{4f} = 0.316\, Re^{-0.25}, \tag{3}$$

$$Re_{TP} = \frac{w\rho(1-x)D}{(1-\phi)\mu'}. \tag{4}$$

(5) The time-averaged temperature within the core of the flowing fluid is equal to the saturation temperature.

(6) The change in temperature measured along the flow is neglected relative to its change in the radial direction, i.e.

$$\frac{\partial T}{\partial z} = 0. \tag{5}$$

(7) The radial component of velocity vanishes within boundary sublayer,

$$w_r = 0. \tag{6}$$

(8) The heat transfer process under consideration is steady-state,

$$\frac{\partial T}{\partial \tau} = 0. \tag{7}$$

The Fourier equation in cylindrical coordinates is

$$\rho' C_p\left(\frac{\partial T}{\partial \tau} + w_z\frac{\partial T}{\partial z} + w_r\frac{\partial T}{\partial r}\right) = \frac{1}{r}\frac{\partial}{\partial r}\left(r\lambda_r'\frac{\partial T}{\partial r}\right) + \frac{\partial}{\partial z}\left(\lambda_z'\frac{\partial T}{\partial z}\right). \tag{8}$$

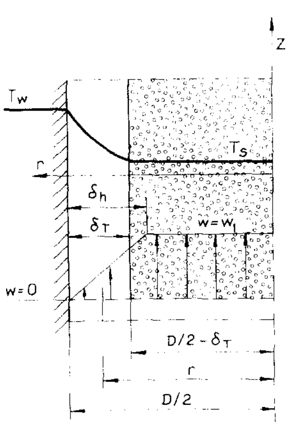

FIG. 4. The physical model of forced convection boiling.

With the above assumptions it reduced to the simple form

$$\frac{1}{r}\frac{d}{dr}\left(r\lambda'\frac{dT}{dr}\right) = 0. \tag{9}$$

The boundary conditions (Fig. 4), are

$$T = T_s \quad \text{for} \quad r = D/2 - \delta_T;\ \delta_T = f(T_w, Pr)$$

$$q_0 = \lambda'\frac{dT}{dr} \quad \text{for} \quad r = D/2. \tag{10}$$

The phenomenon described by equation (9) with boundary conditions (10) is associated with the field of heat conduction. The complication arises from the fact that the boundary conditions change because of the thermal boundary sublayer thickness, δ_T, associated with δ_h which depends, in turn, on the velocity w, the void fraction ϕ, and the dryness fraction, x, as is clear from assumption 4. The solution of the equations (9) and (10) is the function

$$T = T_s - \frac{Dq_0}{2\lambda'}\ln\left[(D/2-\delta_T)/r\right] \tag{11}$$

which is valid in the domain

$$D/2 - \delta_T \leqslant r \leqslant D/2. \tag{12}$$

It may be seen from equation (11) that the wall temperature

$$T_w = T_s - \frac{Dq_0}{2\lambda'}\ln\left(1 - \frac{2\delta_T}{D}\right) \tag{13}$$

strongly depends on the thermal boundary sublayer thickness, δ_T, and the problem arises as to how to evaluate δ_T. It is easy to show that the following relation may be written by using equations (2) and (4):

$$\frac{\delta_h}{D} = 25.32\, Re^{-0.25} \tag{14}$$

Introducing the ratio of the hydraulic and thermal sublayers, taken to be a function of the Prandtl number, in the form

$$\frac{\delta_h}{\delta_T} = Pr^{1/3}; \quad \text{where} \quad Pr = \frac{\mu' C_p'}{\lambda'}, \tag{15}$$

and combining equations (15) and (14), we finally obtain

$$\delta_T = 25.32\, D Re_{TP}^{-0.75} Pr^{-0.33}. \tag{16}$$

From equations (13) and (16) we can show that the temperature difference between the wall and the core of the flow is given by

$$\Delta T = T_w - T_s = \frac{Dq_0}{2\lambda'}\ln\left(\frac{1}{1 - 50.64\, Re^{-0.75} Pr^{-0.33}}\right). \tag{17}$$

The theoretical results contained in equation (17) have been compared with experimental data. Fairly good agreement has been observed. In fact, for more than 85% of the 360 experimental points, the relative deviations are less than 8%.

Let us now turn to nucleate boiling. The assumptions for the theoretical model of nucleate boiling, n.b., are as follows:

(1) Vapor bubbles are spread out in the core of the flow and within the boundary layer close to the wall. They are treated as void spaces playing, however, the role of heat sinks. The rest of the space is filled with liquid of constant conductivity λ'.

(2) The influence of vapor bubbles as stirring devices within the boundary layer is negligible.

(3) The boundary layer thickness is the same for n.b. and for f.c.b. as long as the mass flux, quality, and inlet temperature are held constant, and may be evaluated from equation (16).

(4) Temperature and velocity vary only with the sublayers (Fig. 5).

(5) The time-averaged temperature in the core of the fluid is equal to the saturation temperature.

(6) The change in temperature measured along the flow is negligible relative to its change in the radial direction,

$$\frac{\partial T}{\partial z} = 0. \tag{18}$$

(7) The radial component of velocity is assumed to be equal to zero within the sublayer,

$$w_r = 0. \tag{19}$$

(8) The heat transfer process under consideration is steady-state,

$$\frac{\partial T}{\partial \tau} = 0. \tag{20}$$

(9) The kth part of the total quantity of heat transferred from the wall to the core of the flow is used up for the creation or for the growth of vapor bubbles, and is treated as latent heat transport within the boundary sublayer.

(10) The location of heat sinks is uniform in space.

(11) The void fraction within the sublayer is constant, i.e., independent of the radial coordinate, and equal to the space-averaged void fraction in the same cross-section.

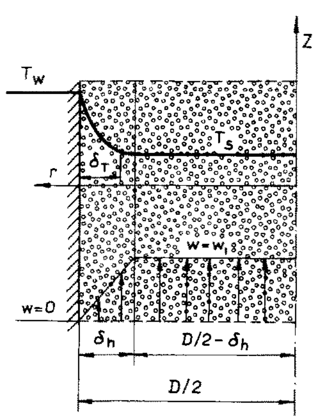

FIG. 5. The physical model of nucleate boiling.

The Fourier equation describing heat transfer within the sublayer may be written as

$$\rho' C_p'\left(\frac{\partial T}{\partial \tau} + w_z\frac{\partial T}{\partial z} + w_r\frac{\partial T}{\partial r}\right)$$
$$= \frac{1}{r}\frac{\partial}{\partial r}\left\{[r(1-\phi)\lambda']\frac{\partial T}{\partial r}\right\} + \left[(1-\phi)\lambda'\frac{\partial T}{\partial z}\right] - k\Omega. \tag{21}$$

The preceding assumptions allow us to simplify this to the form of the heat conduction equation with internal heat sinks:

$$\frac{1}{r}\frac{\mathrm{d}}{\mathrm{d}r}\left[(1-\phi)r\lambda'\frac{\mathrm{d}T}{\mathrm{d}r}\right] = k\Omega, \tag{22}$$

which should be supplemented by the following boundary conditions

$$\left.\begin{aligned} &T = T_s \quad \text{for} \quad r = D/2-\delta_T,\\ &\text{and}\\ &q_0 = (1-\phi)\lambda'\frac{\mathrm{d}T}{\mathrm{d}r} \quad \text{for} \quad r = D/2. \end{aligned}\right\} \tag{23}$$

We assume that the general solution of equation (22) takes the form

$$T = C_1 r^2 + C_2 \ln r + C_3. \tag{24}$$

Substitution of equations (24) into (22) with equation (23) yields

$$C_1 = \frac{k\Omega}{4\lambda'(1-\phi)}, \tag{25}$$

$$C_2 = \frac{q_0 D}{2(1-\phi)\lambda'} - \frac{D^2 k\Omega}{8\lambda'}, \tag{26}$$

and

$$C_3 = T_s - C_1(D/2-\delta_T)^2 - C_2 \ln(D/2-\delta_T). \tag{27}$$

Additional information is needed to describe the quantity Ω in terms of the well-defined heat flux q_0 as an initial condition. This is given by the expression resulting from the definition of Ω

$$\Omega = \frac{q_0}{D[1-(1-2\delta_T/D)^2]}, \tag{28}$$

where Ω, from the physical point of view, describes the heat sinks uniformly distributed within the volume occupied by the sublayer.

At this stage of the investigation we are not in a position to say more about the void fraction, ϕ, and the factor k, beyond what has been said in the assumptions. This is to the effect that ϕ is equal to the average for the section under consideration and that the factor k, which describes the fraction of the heat flux absorbed by the bubbles in the sublayer, must have a value between 0 and 1. It is likely to exceed 0.5. In fact it is possible to estimate ϕ and k further on the basis of assumptions regarding the amount of active nuclei on heated surface,

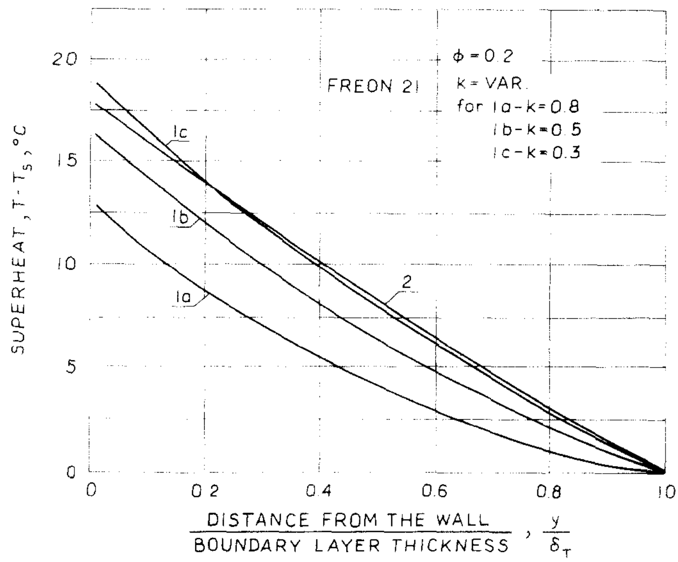

FIG. 6. Theoretical temperature distribution in the boundary layer for nucleate boiling (curves 1a, 1b, 1c) and for forced convection boiling (curve 2). $\psi = 1200$ kg m^{-2} s^{-1}, $T_s = 40$°C, $q = 95$ kW m^{-2}, $D = 0.013$ m.

the time the bubbles spend within the sublayer or their movement, and the heat transfer coefficient between vapor bubbles and surrounding liquid. This kind of calculation does not reduce the number of uncertain quantities; indeed it introduces more unsolved problems requiring experimental investigations. In the light of what has been said we decided to ignore these additional problems associated with k and ϕ and to treat both ϕ and k as parameters.

4. DISCUSSION

The temperature distribution within the sublayer is given f.c.b and n.b. by equations (9) and (24), respectively. They are depicted in Figs. 6 and 7. All curves were drawn for Freon 21 which flows through the 13 mm I.D. pipe with velocities corresponding to those which existed during our own experiments in order to compare them with theoretical results. It is of interest to note that is the case of n.b. for a given void fraction, ϕ, the theoretical wall temperature decreases while k increases. For constant k, we obtain an increase in the wall temperature as the void fraction, ϕ, increases within the sublayer. Each run of n.b. is compared on the same graph with a run of f.c.b. arranged at the same conditions (velocity and temperature). Thus we are able to see how much of a wall temperature jump is predicted theoretically when f.c.b. changes to n.b. The general conclusion is that by suitable fitting we can obtain theoretical results which are in agreement with experiment.

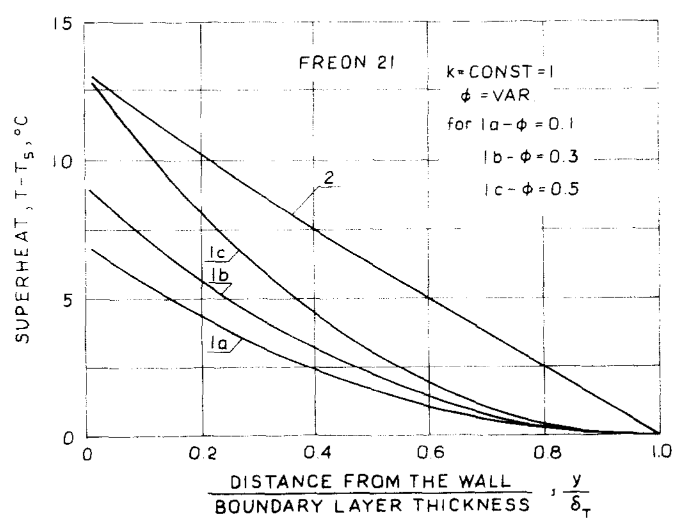

FIG. 7. Theoretical temperature distribution in the boundary layer for nucleate boiling (curves 1a, 1b, 1c) and for forced convection boiling (curve 2). $\psi = 620$ kg m^{-2} s^{-1}, $T_s = 45$°C, $q = 55$ kW m^{-2}, $D = 0.013$ m.

5. CONCLUSIONS

(1) The wall temperature drop, during the zero boiling crisis, may be explained by the fact that f.c.b. changes to n.b.

(2) We have proved that the stirring effect of bubbles generated on the heated surface may be neglected.

(3) The only reason that can explain the decrease in wall temperature and the subsequent increase in heat transfer coefficient after the zero boiling crisis is the effect of the latent heat transport within the boundary sublayer.

(4) By means of suitable fitting of the quantities k and ϕ it is possible to secure agreement between theory and experiment. It is noteworthy that their values fall between 0.5 and 1.0 for k, and 0.2–0.6 for ϕ. The experiment leads us to the reasonable assertion that the ranges of both k and ϕ correspond to bubble flow.

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SUR LE TRANSPORT DE CHALEUR LATENTE DANS L'EBULLITION EN ECOULEMENT FORCE

Résumé—On présente des modèles théoriques qui décrivent les mécanismes de transfert thermique pour l'ébullition nucléée en écoulement forcé et pour l'ébullition dans le coeur du fluide, par exemple, sans nucléation à la surface d'échange thermique. Les modèles résultent d'études expérimentales sur l'effet hysteresis du transfert thermique dégagé par Murphy et Bergles [2] et Abdelmessih *et al.* [3] aussi bien que par l'auteur [4]. En particulier, les expériences de l'auteur ont fourni le support de l'hypothèse selon laquelle l'accroissement du coefficient de transfert thermique à la paroi est essentiellement dû aux bulles de vapeur agissant comme des puits de chaleur dans la souscouche limite thermique. Cette conclusion ne s'accorde pas avec l'idée que les bulles formées dans le film mince de liquide pour les régimes de bulles ou de bouchons affectent le mécanisme de convection forcée qui est généralement la principale cause de l'intensification du transport thermique.

ZUM TRANSPORT LATENTER WÄRME IN EINER ERZWUNGENEN SIEDENDEN STRÖMUNG

Zusammenfassung—Dieser Aufsatz stellt theoretische Modelle vor, die die Wärmeübergangs-Mechanismen beim Blasensieden in einer erzwungenen Strömung und beim Sieden in einem Flüssigkeitskern ohne Keimbildung an der wärmeübertragenden Oberfläche beschreiben. Die Modelle sind von experimentellen Untersuchungen des Hysterese-Effektes beim Wärmeübergang hergeleitet, die von Murphy und Bergles [2], von Abdelmessih u.a. [3] und vom Autor [4] durchgeführt worden sind. Besonders der Versuch des Autors hat Daten geliefert, die die Hypothese stützen, derzufolge die Zunahme des Wärmeübergangskoeffizienten von der Wand an das strömende Fluid vor allem darauf beruht, daß Dampfblasen innerhalb der thermischen Unterschicht als Wärmesenken wirken. Die obige Folgerung stimmt nicht mit der Ansicht überein, daß Blasen, die im Bereich der Blasen- oder Pfropfenströmung innerhalb des dünnen Flüssigkeitsfilms gebildet werden, den Mechanismus der erzwungenen Konvektion beeinflussen, der im allgemeinen der einzige Grund für die Intensivierung des Wärmetransports ist.

О ПЕРЕНОСЕ СКРЫТОЙ ТЕПЛОТЫ ПРИ ВЫНУЖДЕННОМ ТЕЧЕНИИ КИПЯЩЕЙ ЖИДКОСТИ

Аннотация—Представлены теоретические модели, описывающие механизмы теплопереноса при пузырьковом кипении в потоке жидкости и при кипении в ее ядре, т.е. без нуклеации на поверхности теплообмена. Модели построены на основе экспериментальных исследований гистерезиса теплопереноса, проведенных Мэрфи и Берглсом [2] и Абдельмесси и др. [3], а также автором [4]. В частности, полученные экспериментальные данные подтвердили гипотезу о том, что рост коэффициента теплопереноса от стенки к жидкости обусловлен главным образом пузырьками пара, играющими роль стоков тепла в тепловом пограничном подслое. Этот вывод противоречит утверждению о том, что пузырьки, образующиеся внутри тонкой жидкой пленки при пузырьковом или ползучем режиме течения, оказывают влияние на механизм вынужденной конвекции, благодаря которой происходит в основном интенсификация переноса тепла.