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FORCED CONVECTION HEAT TRANSFER TO FLUID NEAR CRITICAL POINT FLOWING IN CIRCULAR TUBE

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Abstract—Experiments were performed on forced convection heat transfer under considerably high heat fluxes to supercritical carbon dioxide flowing in a circular tube, and the phenomena of deterioration of heat-transfer coefficient near the pseudocritical temperature were closely investigated. The wall temperature profiles obtained in the authors' experiments and also in those of previous investigators can be explained fairly well by the theory assuming "normal mode turbulent convection". Furthermore, the wall temperature distributions observed in flow boiling heat transfer to a slightly subcritical fluid can also be explained by the present theory. It is also proved that surface roughness has a particular effect on forced convection heat transfer to supercritical fluid.

NOMENCLATURE

c_p, specific heat;
d, tube diameter;
G, flow rate;
h, heat-transfer coefficient;
i, enthalpy;
L, length of test section;
Pr, Prandtl number;
p, pressure;
q, heat flux;
t, temperature;
u, velocity;
u^+, dimensionless velocity;
u^*, shearing stress velocity;
x, steam quality;
y, distance from wall;
y^+, dimensionless distance.

Greek symbols
ε, eddy diffusivity;
λ, thermal conductivity;
μ, viscosity;
ν, kinematic viscosity;
τ, shearing stress;
ρ, density.

Subscripts
B, bulk;
c, critical;
m, momentum;
pc, pseudocritical;
t, thermal;
W, wall.

1. INTRODUCTION

THIS paper presents a study on forced convection heat transfer under considerably high heat flux to a fluid under slightly higher than critical pressure flowing in a tube of circular cross section at a fairly large flow rate. Flow boiling heat transfer to a fluid at slightly subcritical pressure is also discussed.

Recently, forced convection heat transfer to supercritical fluids has attracted increasing interest in relation to the developments of supercritical steam plants and the use of cryogenic fluids in rocket propellant systems. As a matter of course, supercritical fluid shows no distinction between gas and liquid phases. However, as shown in Fig. 1 (the physical properties of carbon dioxide are at 80 kg/cm^2,

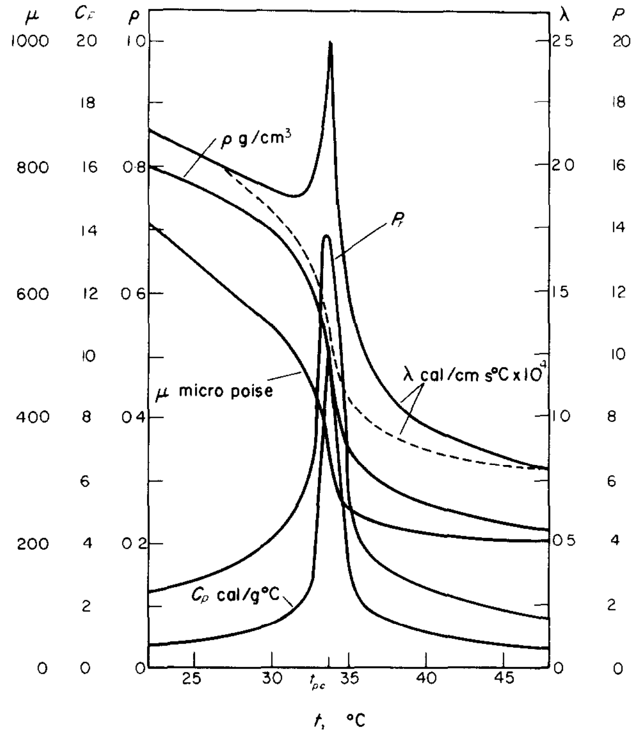

FIG. 1. Physical properties of carbon dioxide at 80 kg/cm².

and the critical constants of carbon dioxide are $p_c = 75{\cdot}3$ kg/cm², $t_c = 31{\cdot}04$°C and $\rho_c = 0{\cdot}468$ g/cm³), the physical properties of supercritical fluid vary extremely around a certain temperature, called the pseudocritical temperature. These properties are apparently divided into a gas-like phase and a liquid-like phase. The main purpose of this paper is to analyse the mechanism of forced convection heat transfer inside a circular tube, taking into consideration the extreme variation of properties of the fluid.

In the previous paper [1], the authors studied forced convection heat transfer to supercritical fluid mainly in cases where the heat flux is relatively small and the difference between the wall temperature and the bulk temperature is somewhat less than 10°C. Under such conditions, special increase of the heat-transfer coefficient near the pseudocritical temperature has been observed by several investigators [2–4]. In the previous paper [1], the authors performed detailed experiments on forced convection heat transfer of supercritical carbon dioxide, both in the case of heating and in that of cooling. It was concluded that forced convection heat transfer of supercritical fluid can be explained, without significant alteration, as an extension of the usual type of heat transfer of fluid with constant physical properties to the case of fluid with varying physical properties.

Experiments on forced convection heat transfer in the supercritical region have also been performed on oxygen [5] and on hydrogen [6] in connection with cryogenic use in rocket propulsion systems. In those experiments, the heat fluxes were considerably high and the temperature differences from the wall to the bulk reached as much as a few hundreds of degrees. In contrast to the case of low heat fluxes, it was observed in those experiments that the heat-transfer coefficients decreased extremely in the neighbourhood of the pseudocritical temperature.

Two kinds of theoretical approaches have been proposed to explain these peculiar phenomena both in the case of low heat fluxes and also in that of high heat fluxes. One approach has postulated the existence of a "boiling-like" phenomenon [7, 8]. The other is based on the assumption that the heat transfer mechanism is primarily turbulent forced convection [9–12].

The previous investigation of the authors into the problem of relatively low heat flux [1] properly belongs to the latter approach, since it assumed "normal mode convection". In this paper the authors hope to explain the phenomenon observed in the instance of very high heat flux also by assuming normal mode turbulent convection

2. THEORETICAL CONSIDERATIONS

In the previous paper [1] the authors explained the physical meaning of increase of the heat-transfer coefficient observed in the case of relatively low heat fluxes, in connection with increase of specific heat in the neighbourhood of the pseudocritical temperature. Namely, an increase of specific heat causes an increase of turbulent thermal conductivity $\rho c_p \varepsilon_t$. When the pseudocritical temperature is located in the buffer zone of the turbulent boundary layer, the temperature drop across the boundary layer is most effectively diminished and the heat-transfer coefficient takes its maximum value.

The phenomenon of decrease of the heat-transfer coefficient observed in the case of very high heat fluxes, which is in contrast to that of low heat fluxes, may also be qualitatively explained by assuming normal mode turbulent convection as follows. In accordance with the increase of heat flux which results in an increase of temperature gradient, the proportion of thickness occupied by the pseudocritical temperature zone (in which the specific heat is very great) in the boundary layer decreases. In consequence the effect of increase of specific heat mentioned above loses its meaning and the flow divides into virtually two phases; that is, a gas-like fluid layer in the neighbourhood of the wall surface and a liquid-like fluid flow near the center of the tube. Under such conditions, when the fluid is heated up from an entrance bulk temperature which is considerably lower than the pseudocritical temperature, the thin layer of gas-like fluid appears on the wall surface in the first place. Then the temperature drop across this gas-like fluid layer increases greatly and the heat transfer coefficient decreases rapidly in the early stage. However, in accordance with the increase of bulk temperature, an increase in the mean velocity of flow occurs and it causes the heat-transfer coefficient to rise. Especially when the fluid temperature near the tube center (which virtually determines the mean velocity) exceeds the pseudocritical temperature, the mean velocity increases greatly and results in a rapid increase of the heat-transfer coefficient.

Based on these qualitative considerations, the authors proceed to evaluate the heat-transfer coefficients by assuming the same theory as discussed in the previous paper [1]. This theory was originated by K. Goldmann [10], and may be outlined as follows. It is assumed that the turbulent mixing process at any point—that is, the growth and damping of turbulent eddies—is a function of the fluid properties at that point. Then it is assumed that the universal turbulent-velocity profile of equation (1), which has been verified for isothermal flow, may be used to describe non-isothermal flow fields with variable properties, provided the velocity and distance parameters are defined by equations (2)–(4).

$$u^+ = \psi(y^+) \tag{1}$$

$$du^+ = du/u^*, \quad u^+ = \int_0^u \frac{du}{u^*} \tag{2}$$

$$dy^+ = \frac{u^*}{\nu} dy, \quad y^+ = \int_0^y \frac{u^*}{\nu} dy \tag{3}$$

$$u^* = \sqrt{(\tau_W/\rho)}. \tag{4}$$

These assumptions can be translated into an assumption in terms of eddy diffusivity; that is, if y^+ in the case of varying physical properties is defined as equation (3), the distribution of eddy diffusivity

$$\varepsilon_m/\nu = \text{func}\,(y^+) \tag{5}$$

in the case of constant properties can be applied as is to the case of varying fluid properties. Either the extended velocity profile in equations (1)–(4) or the extended distribution of eddy diffusivity in equations (3) and (5) is coupled with the energy equation. These equations are solved numerically for the given conditions at the wall surface and then the velocity and temperature profiles in the tube are determined.

On turbulent heat transfer to supercritical carbon dioxide at 80 kg/cm^2 flowing in a circular tube of 10 mm i.d., numerical solutions were calculated by the HITAC 5020E computer at the Computation Center of the University of Tokyo. The details of the calculation procedure are given in the previous paper [1]. In the calculation, the following expression of eddy diffusivity is adopted, according to H. Kato [13]:

$$\varepsilon_m/\nu = 0{\cdot}37\, y^+ [1 - \exp\{-0{\cdot}002\,(y^+)^2\}]. \tag{6}$$

Other diffusivity forms yield almost the same results. Figure 2 is a plot of the calculated wall temperature t_W vs. bulk enthalpy i_B for different flow rates G for constant heat flux $q = 2{\cdot}4 \times 10^5$ kcal/m^{2}h. In this figure, the peculiar behaviour of the wall temperature observed by several investigators is actually calculated.

The temperature distributions in the turbulent boundary layer are also investigated. Figure 3 illustrates the location of the pseudocritical temperature zone in the turbulent boundary layer. The pseudocritical temperature of

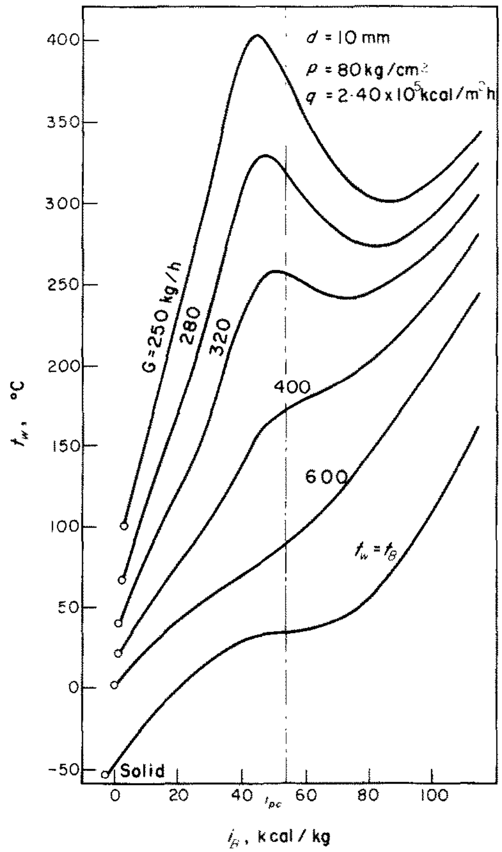

FIG. 2. Calculated wall temperature profiles for forced convection to supercritical carbon dioxide.

carbon dioxide at 80 kg/cm^2 is $t_{pc} = 33{\cdot}75$°C and the physical properties change considerably in the region between $t = \underline{t_{pc}} = 32{\cdot}6$°C and $t = \overline{t_{pc}} = 35{\cdot}2$°C. In Fig. 3, the positions y^+ where the temperature takes these three values are plotted for the case of $G = 280$ kg/h. Figure 3 supports the preceding qualitative discussions: that is, the heat-transfer coefficient takes a minimum value when the pseudocritical temperature point is located near the bottom of the turbulent core region, and under this condition the effect of decrease in the heat-transfer coefficient due to the increase of thickness of the gas-like fluid layer balances with the effect of increase in the heat-transfer coefficient due to the increase of the mean velocity

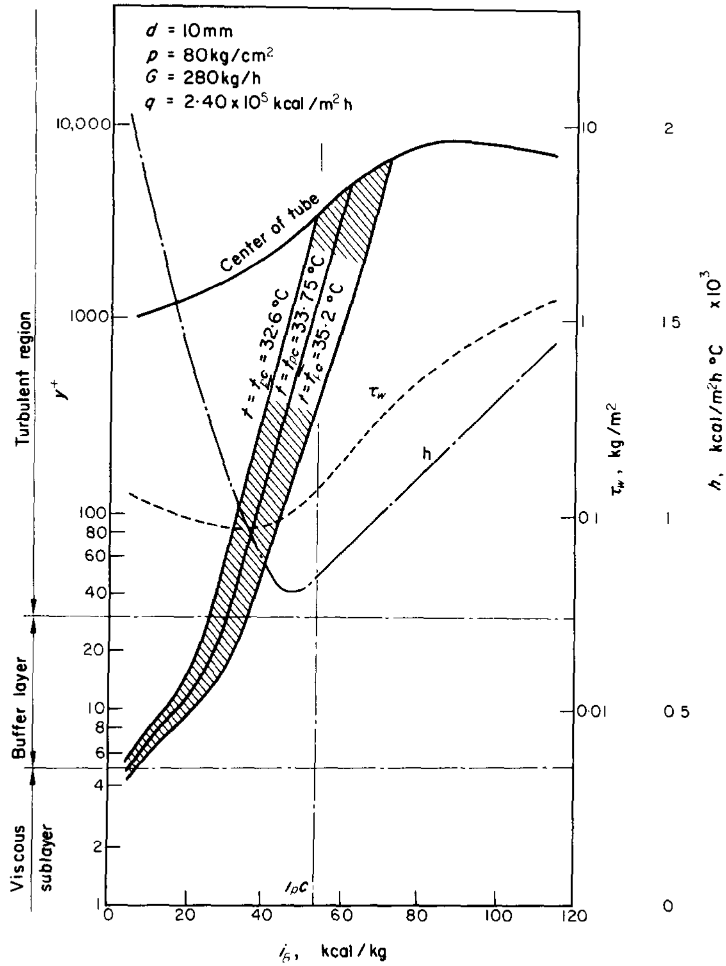

FIG. 3. Location of pseudocritical temperature zone in boundary layer.

of flow. In Fig. 3, the shear stress τ_W at the wall surface is also plotted. It is observed that τ_W decreases to some extent in the region where the heat-transfer coefficient decreases.

3. EXPERIMENTAL APPROACH AND DISCUSSION

3.1. *Experimental apparatus and procedure*

Experiments were performed on forced convection heat transfer under considerably high heat fluxes to supercritical carbon dioxide at 80 kg/cm^2 flowing upwards in a vertically placed circular tube of 6 mm i.d. A smooth tube of 0·2 μ surface roughness and a rough tube of 14 μ surface roughness were used as test sections and the effect of surface roughness on heat transfer was investigated.

A schematic view of the experimental apparatus and details of the test section are shown in Figs. 4 and 5.

The carbon dioxide used in the experiments is the same as that used for welding and its purity is above 99·9 per cent.

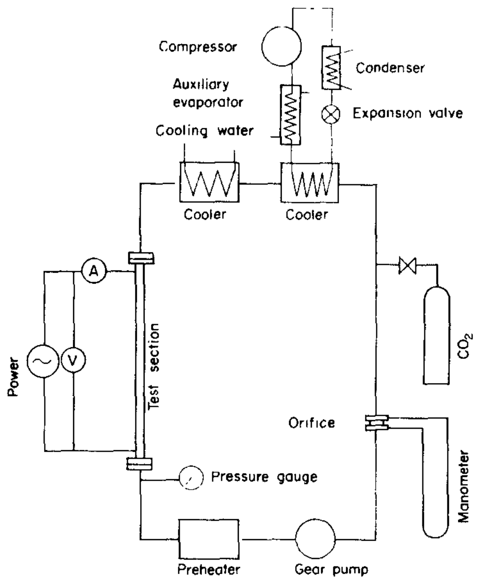

FIG. 4. Schematic drawing of experimental loop.

The pressure in the test loop was measured by a precision pressure gauge.

The carbon dioxide in the test loop was circulated by a gear pump, and the flow rates were measured by an orifice and a water–CO_2 manometer.

The temperatures of the carbon dioxide were measured by the thermocouples t_{in} and t_{out} which were located at the inlet and the outlet of the test section respectively, as shown in Fig. 5. The fluid temperatures along the test section were interpolated from the readings of these thermocouples.

The test section of stainless steel was 6 mm i.d., 8 mm o.d. and 1000 mm long. It was heated by alternating current passing directly through it. Wall temperatures along the test section were measured at 9 points, $t_1, t_2, \ldots, t_9$, by thermocouples attached to the outer surface of the tube 100 mm apart from each other as shown in Fig. 5. The inside wall temperatures were determined from the measured outside wall temperatures by assuming uniform heat generation in the tube.

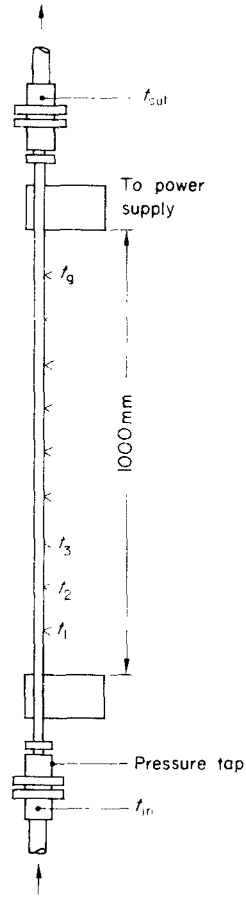

FIG. 5. Test section.

3.2. *Results and discussion for smooth tube*

Temperature profiles were measured for different inlet enthalpies for various flow rates under a constant value of heat flux. For the smooth test section, two series of experiments were performed; that is, one series for $q = 5{\cdot}5 \times 10^5$ kcal/m^{2}h and one for $q = 4{\cdot}2 \times 10^5$ kcal/m^{2}h. The experimental results are plotted in Figs. 6 and 7 in the form of wall temperature t_W vs. bulk enthalpy i_B. A set of 9 points linked together, corresponding to $t_1, t_2, \ldots, t_9$ as shown in Fig. 5, represents the temperature profile measured in each run.

Every temperature profile increases sharply after the inlet point and shows a maximum value, and then turns to decrease. The bulk enthalpy corresponding to the maximum wall temperature point shifts to a higher value as the inlet enthalpy increases. It seems that the heat transfer conditions do not become fully developed in the inlet half of the test section. So in Figs. 6 and 7 a distinction is made between the temperature profile of the inlet half and that of the outlet half by representing these profiles with a fine line and with a heavy line respectively. In the case of forced convection heat transfer to supercritical fluid, the physical properties of the bulk state change considerably along the test section. Therefore, a fully developed state does not exist as a one-dimensional state in radial direction such as develops in case of constant physical properties. However, the term "fully developed" may also be used in case of heat transfer to supercritical fluid, to express the temperature profile which is obtained when the fluid is heated from an inlet bulk temperature sufficiently lower than the pseudocritical temperature by using a very long test section. As shown in Figs. 6 and 7, the entrance region reaches as much as 100 d or so. Similar phenomena were observed in the previous experiments of the authors in case of relatively low heat fluxes [1], and also are reported in the other literature [4, 6]. Furthermore, besides these pure entrance effects, buoyancy effects due to the large density variation take place, and especially at small flow rates in a vertical test section peculiar wall temperature distributions such as hot spots or wavy profiles occur [14]. In the experiments of the authors, the Reynolds number is about $1{\cdot}5 \times 10^5$ in the case: $q = 5{\cdot}5 \times 10^5$ kcal/m^{2}h and $G = 205$ kg/h (to be exact, it varies from about 5×10^4 to about $2{\cdot}5 \times 10^5$ when i_B varies from 40 kcal/kg to 100 kcal/kg), and it is about $1{\cdot}1 \times 10^5$ in the case: $q = 4{\cdot}2 \times 10^5$ kcal/m^{2}h and $G = 155$ kg/h. In the experiments plotted in Fig. 7, the buoyancy seems to have some effect on the results.

The theoretical values of wall temperature were computed for the same conditions as the

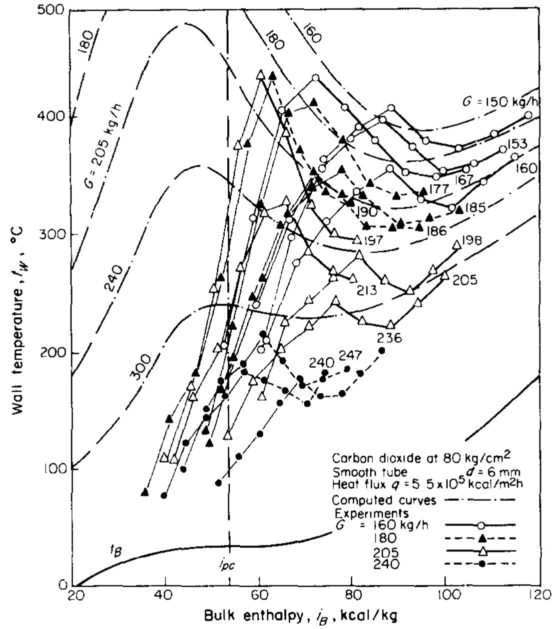

FIG. 6. Experimental results for forced convection to supercritical carbon dioxide flowing in smooth tube ($q = 5{\cdot}5 \times 10^5$ kcal/m^{2}h).

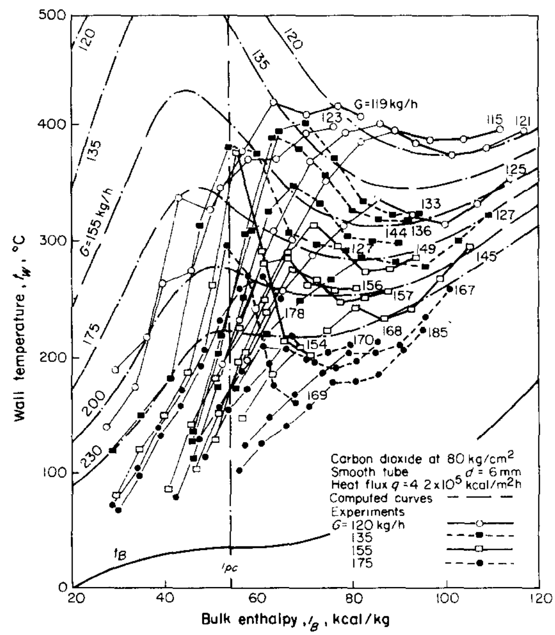

FIG. 7. Experimental results for forced convection to supercritical carbon dioxide flowing in smooth tube ($q = 4{\cdot}2 \times 10^5$ kcal/m^{2}h).

experiments and are shown by chain lines in Figs. 6 and 7. On the whole the experimental results give somewhat larger values of the heat-transfer coefficient than the theoretical results do, and the recovering tendency of the heat-transfer coefficient after its deterioration appears stronger in the experimental results.

Several authors [15] have suggested that eddy diffusivity may be amplified in the presence of a large density gradient. In the previous paper [1], the authors discussed this suggestion negatively. In such a case, if the eddy diffusivity is augmented by 20 per cent, the values of flow rate parameter corresponding to each calculated line in Figs. 6 and 7 are reduced by about 20 per cent, and then the agreement between the experiments and the theory may be improved to a fair extent. However, the authors consider that the time has not yet come to discuss minutely this kind of modification of eddy diffusivity. It is worthy of attention that the present theory, which is merely a first approximation, can explain the experimental results fairly satisfactorily.

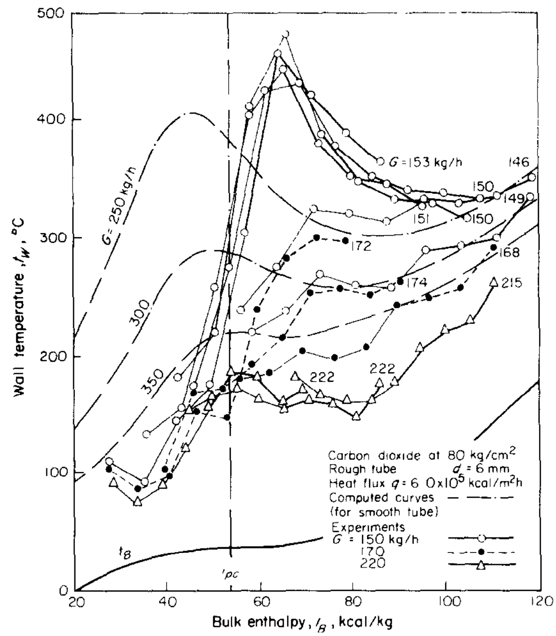

FIG. 8. Experimental results for forced convection to supercritical carbon dioxide flowing in rough tube ($q = 6{\cdot}0 \times 10^5$ kcal/m^{2}h).

3.3. *Results and discussion for rough tube*

For the rough test section, measurements were performed under the heat flux $q = 6{\cdot}0 \times 10^5$ kcal/m^{2}h. Results are shown in Fig. 8.

Nevertheless the heat flux in Fig. 8 is larger than that in Fig. 6, and the wall temperatures in Fig. 8 are lower than those in Fig. 6 for equal values of flow rates. This fact may be attributed to the usual effect of roughness in improving the heat-transfer coefficient.

Considering Fig. 8 in detail, the heat-transfer coefficients are maintained relatively high until the bulk enthalpy i_B gets to 50 kcal/kg. It seems that the deterioration of heat transfer begins at a higher value of bulk enthalpy and occurs more rapidly, compared to the smooth test section. As for the entrance region, equal length of the entrance effect is observed when the inlet enthalpy is higher than 50 kcal/kg. However, when the inlet enthalpy is lower than 50 kcal/kg, the wall temperature profiles trace almost the same curve in spite of different inlet enthalpies, and the entrance effect seems to run short.

As explained in the preceding section on Fig. 3, the heat transfer coefficient for a smooth tube deteriorates in the region: $i_B < 50$ kcal/kg, and in this region the gas-like fluid layer in the neighbourhood of the wall surface is very thin. For instance, according to the theoretical calculation for a smooth tube, the point where $t = \overline{t_{pc}} = 35{\cdot}2$°C ($\overline{t_{pc}}$ is the upper limit temperature of the pseudocritical temperature zone) is located at $y = 40\ \mu$ (corresponding dimensionless parameter is $y^+ = 45$) for $i_B = 35$ kcal/kg under the conditions of Fig. 8: $q = 6{\cdot}0 \times 10^5$ kcal/m^{2}h and $G = 250$ kg/h. On the other hand, the heat-transfer coefficient for a smooth tube recovers in the region: $i_B > 50$ kcal/kg, and in this region the gas-like fluid layer is considerably thicker. The point where $t = \overline{t_{pc}}$ is located at $y = 0{\cdot}6$ mm ($y^+ = 1300$) for $i_B = 60$ kcal/kg under the same conditions as before. In this case, it may well be expected that surface roughness will have a larger effect in the bulk enthalpy region: $i_B < 50$ kcal/kg where the gas-like fluid layer is very thin, than in the region: $i_B > 50$ kcal/kg. Furthermore, by assuming that the entrance effect runs short in a region where surface roughness is predominant, the above-mentioned feature of the experimental results can be explained satisfactorily.

3.4. *Experiments of previous investigators*

R. C. Hendricks *et al.* [6] performed experiments on forced convection heat transfer to supercritical hydrogen (the critical constants of hydrogen are $p_c = 13{\cdot}20$ kg/cm^2, $t_c = 32{\cdot}98$°K and $\rho_c = 0{\cdot}0308$ g/cm^3) flowing upwards in a vertical circular tube. In Fig. 9 the selected data under the conditions of $p = 30{\cdot}94$ kg/cm^2, $d = 8{\cdot}50$ mm, $q = 1{\cdot}45 \times 10^6$ kcal/m^{2}h and different flow rates are compared with the values computed according to the preceding theory. Each measured temperature profile in Fig. 9 shows a rapid increase after the inlet point, and this part is thought to be an entrance region, similar to the experimental results of the authors. Comparing the measured wall temperature with the theoretical value in the fully developed region, the heat-transfer coefficients obtained by the experiments are about two times as high as the theoretical results, and the difference are rather large.

K. R. Schmidt [16] performed experiments on forced convection heat transfer to water (the critical constants of water are $p_c = 225{\cdot}7$ kg/cm^2, $t_c = 374{\cdot}2$°C and $\rho_c = 0{\cdot}32$ g/cm^3) flowing in a horizontal circular tube mainly under subcritical pressures and partly under supercritical pressures. In Fig. 10, the selected data under the conditions of $p = 230$ kg/cm^2, $d = 5$ mm, $G = 49{\cdot}5$ kg/h, and different heat fluxes are compared with the values computed according to the preceding theory. In this case as well, the heat-transfer coefficients obtained by the experiments are somewhat higher than the theoretical results. Especially, the recovering tendency of the heat-transfer coefficient after its deterioration is stronger in the experimental results than in those calculated

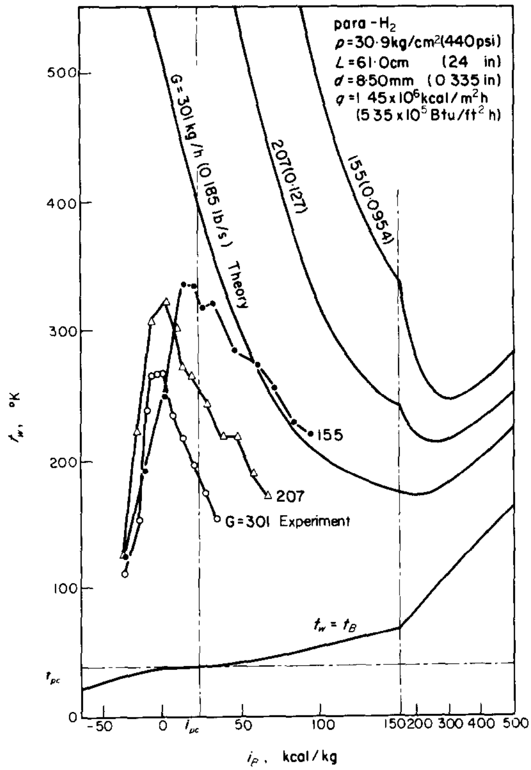

FIG. 9. Comparison between computed and experimental wall temperature profiles for forced convection to supercritical hydrogen.

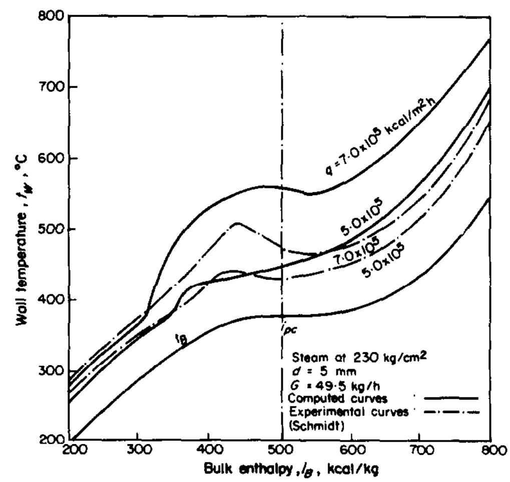

FIG. 10. Comparison between computed and experimental wall temperature profiles for forced convection to supercritical water.

by the theory. This fact is quite similar to the experimental results of the authors.

4. EXTENSION TO FLOW BOILING AND DISCUSSION OF BOILING-LIKE PHENOMENA

From among the above-mentioned experiments of Schmidt [16], the results obtained for flow boiling heat transfer under the subcritical pressure of $p = 210$ kg/cm^2 are shown by chain lines in Fig. 11. In this figure, it is worthy of attention that the wall temperature profile for

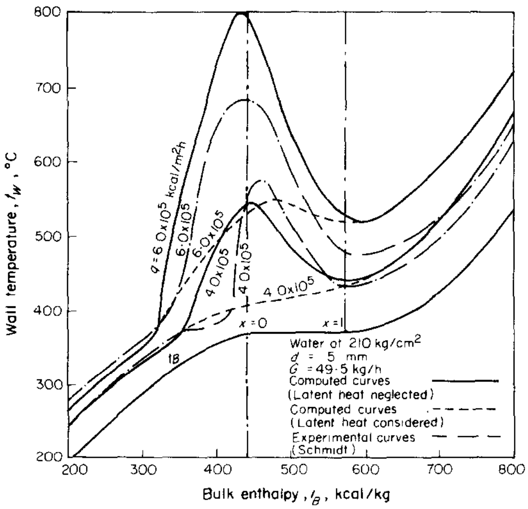

FIG. 11. Comparison between computed and experimental wall temperature profiles for flow boiling of subcritical water.

$q = 6{\cdot}0 \times 10^5$ kcal/m^{2}h is quite analogous to the temperature profiles in the case of forced convection heat transfer to supercritical fluid. In this case, film boiling heat transfer is presumed to take place. Here, although the preceding theory is in principle applicable to fluids with physical properties which vary continuously with temperature, the authors try to apply this theory to subcritical fluids whose properties change discontinuously at saturation temperatures. In this connection, the preceding theory assumes implicitly the continuity of ε_m/v for the boundary condition of turbulence between the gas phase and the liquid phase. However, in due consideration of the mixing motion of turbulent eddies, whose size is estimated to be approximately Prandtl's mixing length, there can be no clearly defined boundary between gas phase and liquid phase, but the fluid properties averaged over time at every point may change somewhat gradually in the radial direction. Then, the preceding theory may be applicable also to the case of subcritical fluid, standing on the primary view of continuously varying properties. (These views call for reconsideration of the preceding computation procedure, in which the property values corresponding to the average temperature at each point are adopted as the local properties of fluid at that point. Nevertheless, such a computation procedure may be admitted if it is considered as only a first approximation.)

In the case of subcritical fluid, the effect of surface tension has to be taken into account. However, we are now discussing flow boiling at slightly subcritical pressure. Here the effect of surface tension may be neglected when making a first approximation.

Latent heat raises another question. In connection with the above suggestion that gas and liquid phases mix together over a certain thickness in the boundary layer, latent heat may make some contribution to heat transfer, operating as if the specific heat has a particular peak near the saturation temperature. Thus, the wall temperatures are theoretically calculated for the conditions of Schmidt's experiments in the following two ways: first, the effect of latent heat is neglected; and second, the latent heat is distributed in a triangular profile extending over 1·5°C. (Judging from the results this value of 1·5°C is thought to be somewhat too large.) In Fig. 11 the results of both calculations are shown by solid lines and by broken lines respectively. Contrary to expectations, the results calculated by neglecting the effect of latent heat give better agreement with the experimental results, and in fact the agreement is quite remarkable.

The experimental result for $q = 4{\cdot}0 \times 10^5$ kcal/m^{2}h, giving a very high heat-transfer coefficient in the slightly subcooled region, does not agree with the theoretical result. In this region nucleate boiling heat transfer takes place, and the present theory, which assumes heat transfer by film boiling, cannot be applied. Thus, nucleate boiling is considered to be a peculiar heat transfer state distinguished from "normal mode turbulent convection" which is represented by a smooth profile of theoretical wall temperature starting from the usual turbulent convection state of cold water, passing through a film boiling state, and ending at the usual turbulent convection state of super-heated steam.

Beginning with the observation by K. Goldmann [7], many discussions have been published on "boiling-like phenomena". According to the above considerations, if such a phenomenon does exist in forced convection, it must be, so to speak, a "nucleate-boiling-like phenomenon". And the authors assume that this nucleate-boiling-like phenomenon will scarcely occur in forced convection heat transfer to supercritical fluid, judging from the tendencies of nucleate boiling observed in forced convection of subcritical fluid.

5. CONCLUSIONS

The experiments were performed on forced convection heat transfer under considerably high heat fluxes to supercritical carbon dioxide flowing in a circular tube, and the phenomena of deterioration of the heat transfer coefficient near the pseudocritical temperature were closely investigated. The wall temperature profiles obtained by the authors experiments and also by previous investigators agreed fairly well with the theory. Along with the conclusions of the previous paper [1], which were mainly concerned with the case of low heat fluxes, it may be concluded that forced convection heat transfer to supercritical fluid is not a new type of heat-transfer problem, but that it can be explained as an extension of the usual turbulent convection heat transfer of fluid with constant physical properties to the case of fluid with varying physical properties, without any significant alterations in the theory. Furthermore, the wall temperature distributions observed in flow boiling heat transfer to a slightly subcritical fluid can also be explained by the present theory. It is proved further that surface roughness has a particular effect on forced convection heat transfer to supercritical fluid.

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CONVECTION THERMIQUE FORCÉE DANS UN TUBE CIRCULAIRE POUR UN FLUIDE PRÈS DU POINT CRITIQUE

Résumé—On rapporte des expériences sur la convection thermique forcée—les flux de chaleur étant considérablement grands—pour du gaz carbonique supercritique dans un tube à section circulaire, et on étudie soigneusement les phénomènes de détérioration du coefficient de transfert thermique aux approches de la température pseudo-critique. A l'aide de la théorie supposant "une convection turbulente de mode normal", il est possible d'expliquer assez bien les profils de température de paroi obtenus par les auteurs lors de leurs expériences, de même que par des chercheurs antécédents. En outre, cette théorie nous aide à expliquer les distributions de température sur la paroi observées lors du transfert de chaleur pour un écoulement avec ébullition de fluide faiblement sous-critique. On prouve aussi que la rugosité de la surface a un effet particulier sur le transfert de chaleur au fluide supercritique en convection forcée.

WÄRMEÜBERTRAGUNG DURCH ERZWUNGENE KONVEKTION IN KREISFÖRMIGEN ROHREN AN FLÜSSIGKEITEN IN DER NÄHE DES KRITISCHEN PUNKTES

Zusammenfassung—Bei verhältnismässig hohen Wärmeströmen wurden Experimente über Wärmeübertragung durch erzwungene Konvektion in kreisförmigen Rohren an überkritisches Kohlendioxyd durchgeführt und die Verschlechterung des Wärmeübergangskoeffizienten in der Nähe der pseudokritischen Temperatur genau untersucht. Die Wandtemperaturprofile, die sich aus den Versuchen der Autoren und aus den Untersuchungen früherer Forscher ergaben, können sehr gut durch die Theorie der gewöhnlichen turbulenten Konvektion erklärt werden. Darüber hinaus können die Wandtemperaturverteilungen, die bei der Wärmeübertragung von einer siedenden Strömung an ein leicht überkritisches Medium beobachtet werden, ebenfalls mit der vorliegenden Theorie erklärt werden. Ausserdem wird bewiesen, dass die Oberflächenrauhigkeit eine besondere Wirkung auf die Wärmeübertragung durch erzwungene Konvektion an überkritische Medien ausübt.

ПЕРЕНОС ТЕПЛА ПРИ ВЫНУЖДЕННОЙ КОНВЕКЦИИ В ПОТОК ЖИДКОСТИ К КРУГОЙ ТРУБЕ ПРИ ТЕМПЕРАТУРЕ, БЛИЗКОЙ К КРИТИЧЕСКОЙ

Аннотация—Проведены эксперименты по переносу тепла при вынужденной конвекции в потоке двуокиси углерода в круглой трубе при сверх критической температуре и больших тепловых нагрузках; тщательно исследовались явления ухудшения коэффициентов теплообмена в области близкой к псевдокритической. Кривые распределения температуры стенки, полученные автором, а также ранее другими исследователями, могут быть хорошо объяснены теорией, допускающей «нормальную турбулентную конвекцию». Кроме того, кривые распределения температуры стенки, наблюдаемые при переносе тепла от кипящего потока в жидкость при температуре несколько более низкой, чем докритическая, могут быть также объяснены этой теорией. Доказано также, что шероховатость поверхности имеет определенное влияние на перенос тепла при вынужденной конвекции в жидкость при сверхкритической температуре.