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FORCED CONVECTIVE HEAT TRANSFER TO SUPERCRITICAL WATER FLOWING IN TUBES

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Abstract—Experimental investigations were made of heat transfer to supercritical water flowing in a horizontal tube and vertical tubes. A comprehensive set of data was obtained for pressures from 226 to 294 bar, bulk temperatures from 230 to 540°C, heat fluxes from 116 to 930 kW/m^2 and mass velocities from 310 to 1830 kg/m^{2}s. Because the physical properties of supercritical fluids change rapidly with temperature in the pseudocritical region, the heat transfer coefficients show unusual behavior depending upon the heat flux. At low or modetate heat fluxes relatively to the flow rate, a satisfactory correlation was obtained, which predicts reasonably well the enhanced heat transfer coefficients near the pseudocritical point. The several characteristics of the deterioration in heat transfer which occurs at high heat fluxes were clarified, and the limit heat flux for the occurrence of the deterioration was determined in connection with the flow rate.

NOMENCLATURE

E, Eckert number defined by equation (1);
G, mass velocity;
Nu, Nusselt number $= \alpha d/\lambda$;
P, pressure;
Pr, Prandtl number $= \mu c_p/\lambda$;
Re, Reynolds number $= Gd/\mu$;
c_p, specific heat of fluid at constant pressure;
d, inside diameter of tube;
h, enthalpy;
q, wall heat flux;
t, temperature;
v, specific volume of fluid;
α, heat transfer coefficient;
λ, thermal conductivity of fluid;
μ, viscosity of fluid.

Subscripts

b, bulk conditions;
cal, calculated value;
exp, experimental value;
m, pseudocritical conditions;
w, wall conditions.

1. INTRODUCTION

THE DEVELOPMENT of supercritical pressure boilers has required more accurate predictions of water and steam heat transfer coefficients over a wide range of operating conditions. Furthermore, in connection with developing nuclear reactors, and cooling rocket motors and gas turbine blades, the increasing knowledge of the heat transfer to fluids near their critical pressure has been required. By these demands of industry a number of investigations have been performed in this field of research in these fifteen years: for example, experimental investigations on forced convective heat transfer to supercritical water by Dickinson and Welch [1], Domin [2], Swenson *et al.* [3] and Shitsman [4], and in addition those on carbon dioxide by several researchers [5–8].

The results of these investigations show that the heat transfer coefficient is remarkably in-

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creased and has a maximum in the neighborhood of the pseudocritical temperature at which the specific heat attains a maximum value. As the heat flux is increased, the peak of the heat transfer coefficient is decreased, and eventually a deterioration in heat transfer coefficient occurs near the pseudocritical point. This has been investigated by Shitsman [9] and others [10–16]. Such unusual behavior of the heat transfer coefficient in the pseudocritical region is attributed to the large variation in physical properties of fluid across the section of a tube, because small changes in temperature do produce large changes in fluid properties in the pseudocritical region, as shown in Fig. 1. Although several models, including ones of pseudo-nucleate-boiling [2, 17] and of pseudo-film-boiling [16, 18] have been proposed, none of them have fully succeeded in explaining such phenomena of heat transfer at supercritical pressures as mentioned above. Also several correlations of heat transfer data have been proposed for supercritical fluids, but they seem to be unsatisfactory, as discussed later. Hence more experimental data which are reliable and systematic may be needed. For example, the data obtained with the same test section in various flow directions may be useful. Little attention has been paid to the effect of flow direction especially at relatively low heat fluxes, and in some papers this condition is not specified.

The authors have been investigating the heat transfer in the critical region since 1960 and have obtained a great many experimental data over a wide range of various parameters. In this paper the results on forced convective heat transfer to water in tubes are reported.

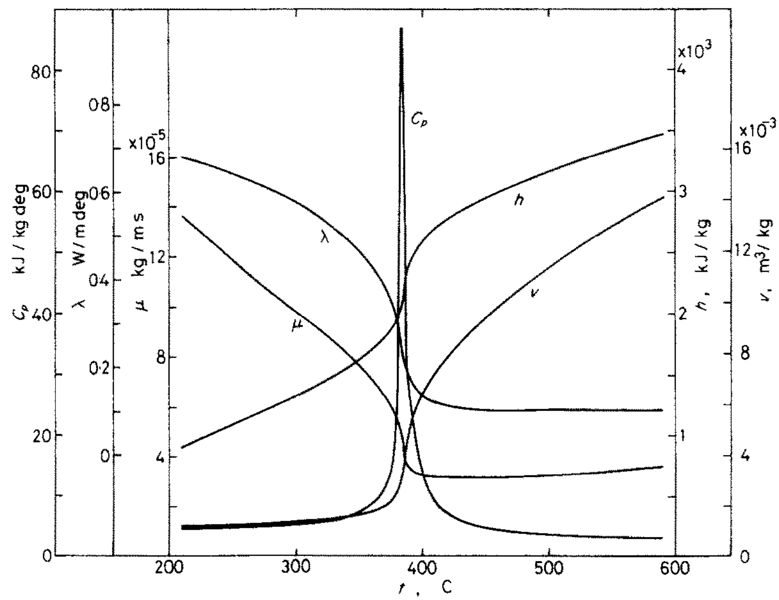


FIG. 1. Physical properties of water at a pressure of 245 bar.

2. EXPERIMENTAL APPARATUS

The experimental apparatus is composed of such a water test loop of once-through monotube as shown schematically in Fig. 2. Its maximum test pressure is 300 bar and the maximum circulating flow rate is 300 kg/h. Water pressurized with a circulating pump, 13, is heated

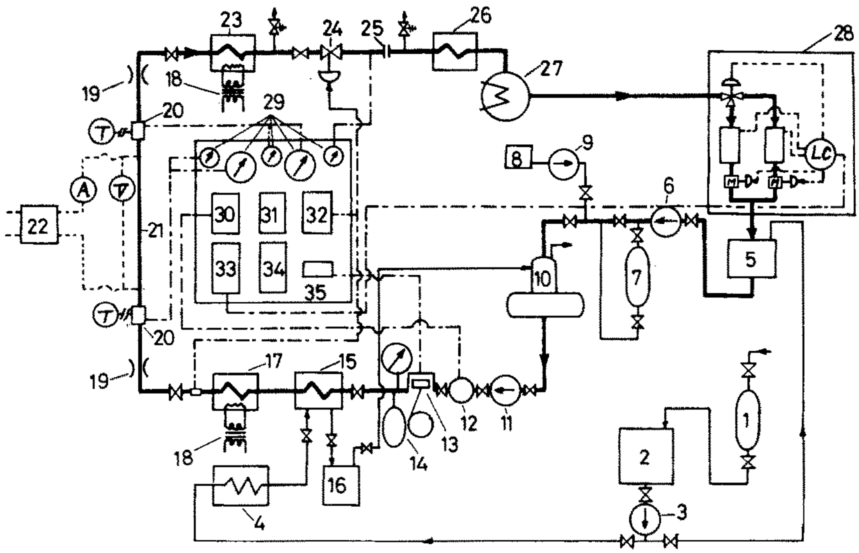


FIG. 2. Schematic diagram of experimental apparatus.

1. Water purifier
2. Purified water tank
3. Supply pump
4. Steam generator
5. Drain tank
6. Pressure pump
7. Polisher
8. Hydrazine tank
9. Injector pump
10. Deaerator
11. Booster pump
12. Flow meter
13. Circulating pump
14. Accumulator
15. Primary heater
16. Flush tank
17. Secondary heater
18. Auto-transformer
19. Venturi tube
20. Mixing chamber
21. Test section
22. Mechanical rectifier
23. Rear heater
24. Throttle valve
25. Electrically insulated flange
26. Cooler
27. Condenser
28. Flow measuring device
29. Pressure gauges
30. Flow recorder
31. Temperature recorder
32. Pressure recorder-controller
33. Flow recorder
34. Temperature recorder
35. Flow regulator

successively by the primary, 15, and the secondary heaters, 17, up to the required temperature and then led to the test section, 21. The water, leaving the test section, is heated up to about 600°C by the rear heater, 23, and then its pressure is reduced down to the atmospheric pressure through a throttle valve, 24. The pressure-reduced steam is sent back to the tank, 5, through a cooler, 26, a condenser, 27, and a flow measuring device, 28. The water from the tank is sent to the circulating pump after water treatment, 8, 9, and deaeration, 10.

The secondary and the rear heaters are electrical furnaces in which spiral water tubes are held. Their load regulation is carried out by three auto-transformers, 18. For heating source of the primary heater, the saturated steam from a 100 bar steam generator, 4, is used, and a part of the condensed water is used as the heating source for the deaerator. The circulating pump is a type of triple connected plunger, whose amount of discharge can be regulated continuously to the one-third of the maximum by speed control. For the reduction of the pressure pulsation, there is installed an accumulator, 14, at the discharge outlet. The throttle valve is a $\frac{1}{4}$ in. globe valve and it functions to maintain the pressure at the test section inlet at a setting value automatically.

The test sections are AISI type 316 stainless steel tubes of 7·5 mm i.d. and 10 mm i.d. with heated lengths of 1500 mm and 2000 mm, respectively. Figure 3 shows the 10 mm i.d. test section. The test section is heated by the direct current of low voltage which is led through

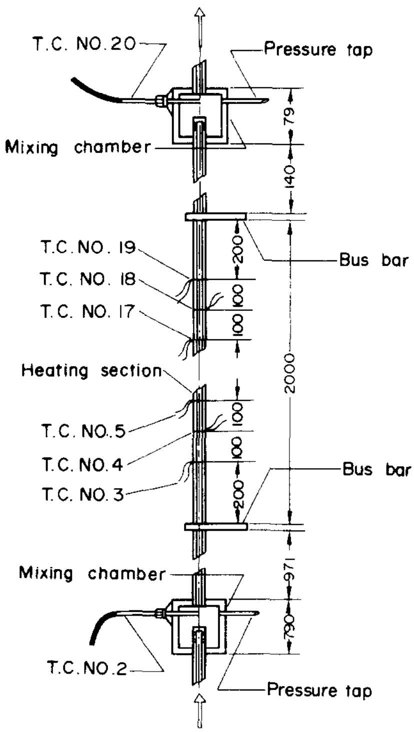


FIG. 3. Test section.

80 kW mechanical rectifier, 22 in Fig. 2. Mixing chambers are arranged before and after the test section, where temperatures and pressures of the fluid at the inlet and the outlet are measured. To measure outside surface temperatures of the test section seventeen chromel–alumel thermocouples for the 10 mm i.d. tube or eighteen for the 7·5 mm i.d. tube are arc-welded to its surface at regular intervals longitudinally on diametrically opposite sides of the tube by turns: for the horizontal tube on its top and on its bottom alternatively.

In order to prevent the electric power supplied to the test section from leaking out, the flange at 25 in Fig. 2 is electrically insulated so that the part between this flange and the outlet end of the test section and the auxiliary circuits connected to this part are isolated from the ground.

The feed water is purified to 3×10^6 Ωcm by a mixed-bed deionizer and a polisher. The pH of the feed water is maintained within 8·5–9·2 by regulating the pouring amount of hydrazine with an injection pump.

3. EXPERIMENTAL METHOD

The effect of each independent variable investigated was determined by varying it over the required range and measuring its effect on the heat transfer coefficient. While this was being done, the other independent variables affecting the heat transfer rate were kept constant. The independent variables investigated and their range were:

Pressure: 226, 245 and 294 bar (230, 250 and 300 ata)

Bulk fluid temperature: 230–540°C

Heat flux: 116–930 kW/m^2 (1×10^5 to 8×10^5 kcal/m^{2}h)

Mass velocity: 310–1830 kg/m^{2}s

Flow direction: horizontal, vertically upward and downward.

The fluid temperature at the inlet of the test section was properly changed according to the heat flux and the flow rate so that continuous tube wall temperature curve might be obtained. For constant pressure, mass velocity and inlet fluid temperature, several heat fluxes were adequately taken within a range where the outside tube wall temperature did not exceed 700°C.

The pressures and the temperatures at the inlet and the outlet of the test section were measured with Bourdon-type gauge and with chromel–alumel sheath thermocouples inserted in the mixing chambers, respectively. The bulk fluid temperature at any axial point in the test section was determined according to the fluid enthalpy at that point which was calculated from the known inlet or outlet enthalpy (whichever was further displaced from the pseudocritical point), the electrical heat input to that point and the flow rate. The inside surface temperatures were estimated from the outside temperature readings by applying the theory of heat conduction [19], after the true thermal electromotive forces were obtained by subtracting the effect of the direct heating current. All the thermocouple outputs were recorded on a chart recorder type of precision potentiometer.

The flow rate was measured at the exit of the condenser by using a flow meter which consists of a three-side electromagnetic valve and two containers, and the time-averaged value was calculated cumulatively.

The heat flux was calculated from the electric power to the test section measured by a 0·2 class direct current voltmeter and an ammeter. Heat losses from the test section were calculated to be negligibly small under the present experimental conditions, which was also confirmed by comparing the indicated electrical input with the heat absorbed by the water.

It was estimated that all temperature readings were accurate within 1 deg C and the power input and flow rate within 1 per cent. Due to the uncertainty on the thermal conductivity of the tube material and heat loss, the accuracy of about 2 deg C could be assumed for the inside wall temperature.

The physical properties of supercritical pressure water were obtained from references [20–22].

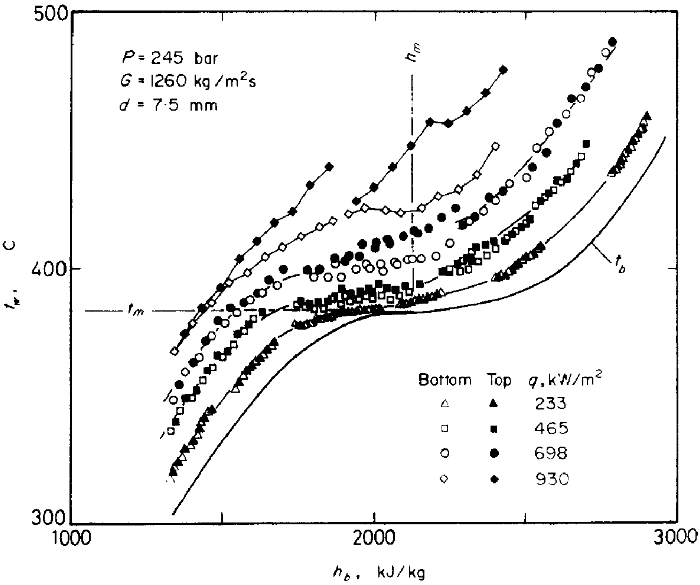
4. RESULTS AND DISCUSSIONS

4.1 *Heat transfer at low heat fluxes*

The heat transfer to supercritical water flowing in a tube was found to be roughly classified into two different regimes according to the heat flux relative to the flow rate. In this section the experimental results that have been obtained at low or moderate heat flux are described.

The measured inside wall temperatures are shown against bulk enthalpy at a pressure of 245 bar, a mass velocity of 1260 kg/m^2s and various heat fluxes of 233, 465, 698 and 930 kW/m^2 for the horizontal flow and the vertically upward flow in Figs. 4 (a) and (b), respectively. For the horizontal flow, at heat fluxes higher than 465 kW/m^2 there exist the differences between temperatures in respect of the top of the tube and the bottom in the pseudocritical region, and the differences are more remarkable at higher heat flux, while there exist no differences in the wall temperatures around the circumference for the vertical flow.

Figures 5 (a) and (b) show the examples of heat transfer coefficients plotted against bulk temperature and inside wall temperature, respectively, for the horizontal flow of such a high mass velocity as 1830 kg/m^2s at a pressure of 245 bar and four different heat fluxes. When the bulk temperature becomes closer up to the pseudocritical temperature (t_m = 382·9°C at 245 bar), the heat transfer coefficient increases abruptly, and it takes a maximum at a bulk temperature which is slightly below the pseudocritical temperature. The wall temperature which corresponds to the maximum in heat transfer coefficient is higher than the pseudocritical temperature, and the difference between the wall temperature and the pseudocritical temperature is remarkably increased with increase of heat flux. The heat transfer coefficient depends strongly upon the heat flux, especially in the pseudocritical region, and its maximum value increases with decrease of heat flux. It seems, however, that there is a limit in this tendency. It is difficult to determine the limit, because the measurement is not sufficiently accurate due to the small temperature difference between the wall and the bulk fluid corresponding to the small heat flux. For example, however, at the condition of a pressure of 245 bar and a mass velocity of 1830 kg/m^2s the heat transfer coefficient appeared not to exceed the value of 80 kW/m^2deg even if the heat flux is less than 233 kW/m^2. For reference, the maximum heat transfer coefficient predicted

a


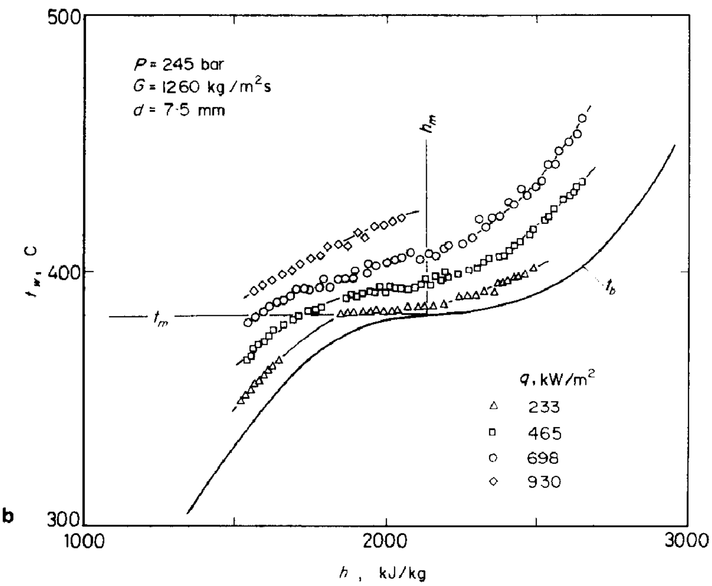
b


FIG. 4. Wall temperature vs. bulk enthalpy
(a) for horizontal flow,
(b) for vertically upward flow.

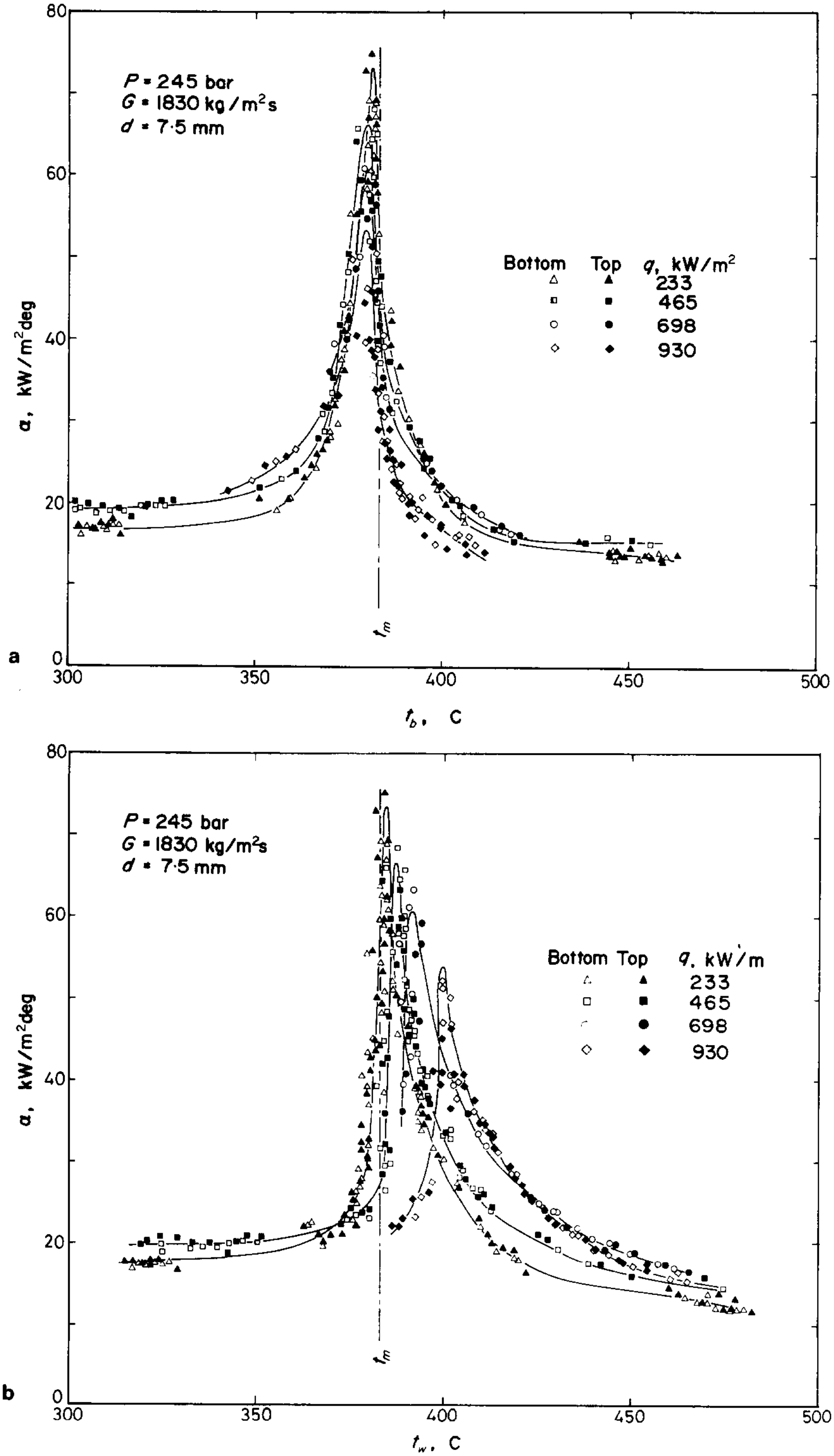


FIG. 5. Heat transfer coefficient for horizontal flow
(a) in relation to bulk temperature.
(b) in relation to wall temperature.

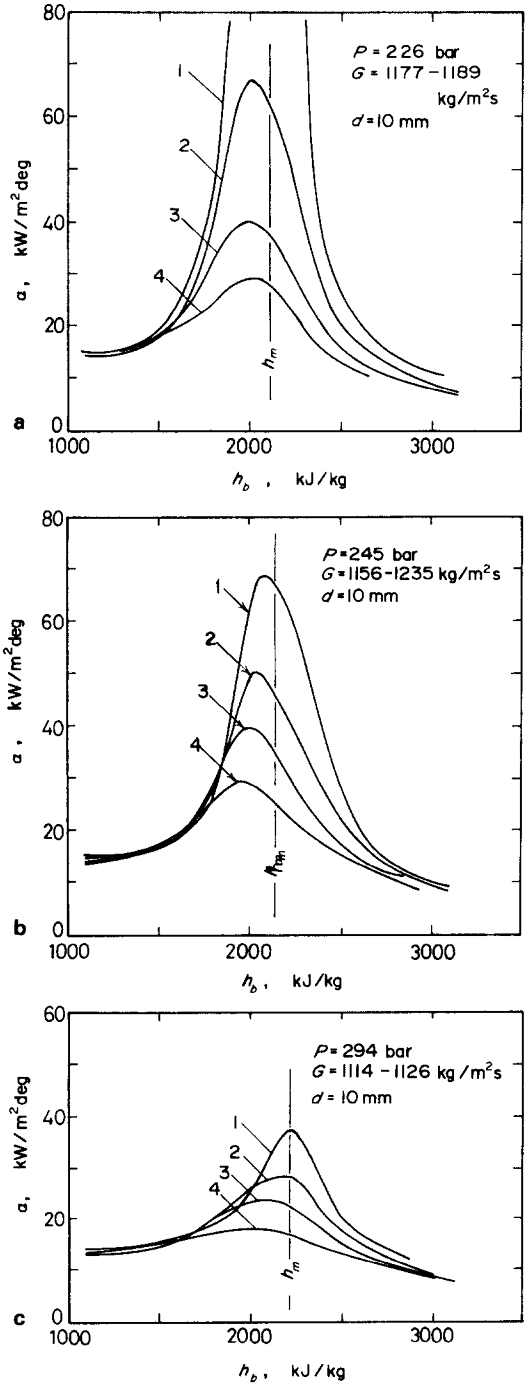


FIG. 6. Heat transfer coefficient vs. bulk enthalpy for vertically upward flow.
1. q = 233; 2. 465; 3. 698; 4. 930 kW/m^2.

by the equation for constant or moderately variable property flow [23], which corresponds to the case of nearly zero heat flux, is 60 kW/m^2 deg at the above-mentioned condition.

Figures 6 (a)–(c) show the heat transfer coefficients calculated from the mean curves of the measured wall temperature at 226, 245 and 294 bar, respectively, for the vertically upward flow in the 10 mm i.d. tube. The heat transfer coefficient shows a similar tendency to that for the horizontal flow: the heat transfer coefficient takes a maximum at a bulk enthalpy slightly less than the pseudocritical enthalpy (h_m = 2116, 2135 and 2206 kJ/kg at 226, 245 and 294 bar, respectively), and its maximum value decreases with increase of heat flux. It is also evident from these figures that the heat transfer coefficient in the pseudocritical region becomes higher as the pressure approaches down to the critical pressure. This tendency of the heat transfer coefficient in relation to the pressure and temperature is similar to that of the isobaric specific heat.

Figures 7 (a) and (b) show the heat transfer coefficients measured with the same test section (7·5 mm i.d. tube) and at the same conditions except the flow directions: horizontal, vertically upward and downward. While there is no appreciable difference in heat transfer coefficients for the flows in three different directions at sufficiently low heat flux (Fig. 7a), at higher heat flux the heat transfer coefficient in the pseudocritical region is highest and lowest at the bottom of the tube and at the top, respectively, for the horizontal flow, and the coefficient for the vertical flow is intermediate between these values. Generally these differences become more remarkable at higher heat fluxes and lower flow rates. It is not clear from the results of this experiment but these differences in heat transfer coefficients may be attributed to a nature of two-phase-like flow or effect of buoyancy.

4.2 *Correlation of heat transfer coefficients*

Most extensive data were obtained for the vertically upward flow in the 10 mm i.d. tube over a wide range of pressures, bulk tempera-

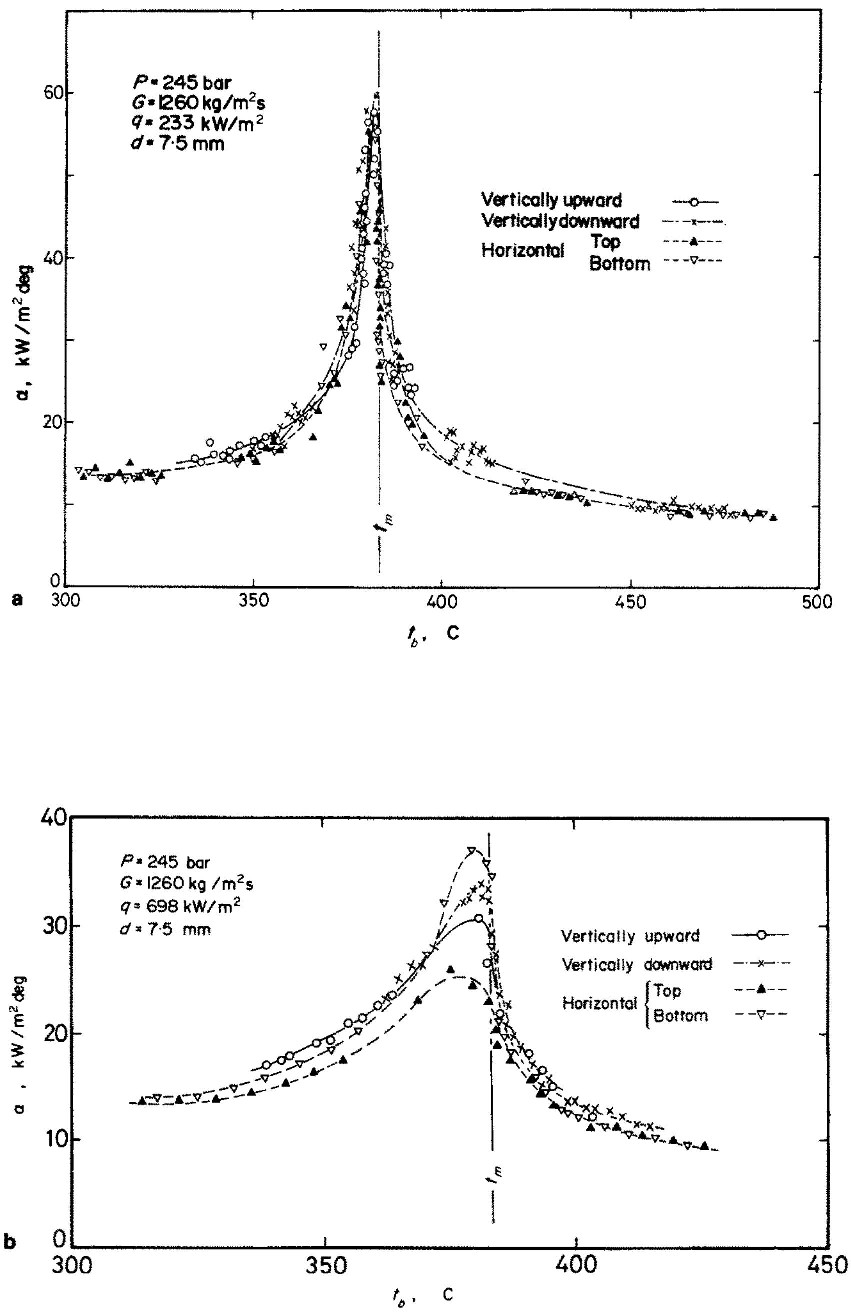


FIG. 7. Comparison of heat transfer coefficients for flows in various directions
(a) at low heat flux, (b) at high heat flux.

tures, mass velocities and heat fluxes. Therefore, heat transfer correlations for supercritical water were investigated, using mainly these data, from which were excluded those obtained in the region of the deteriorated heat transfer which occurs at high heat fluxes as described in the next section.

Figure 8 shows an example of experimental data plotted in terms of the Nusselt number, Reynolds number and Prandtl number which are conventional dimensionless groups for the turbulent heat transfer in a tube. The data are classified into three different domains according to the value of the Eckert number defined by

$$E = \frac{t_m - t_b}{t_w - t_b}. \tag{1}$$

This dimensionless number is one of measures with which the state of the supercritical fluid over a cross section of tube can be designated. In Fig. 8 the slope of the straight line ① is 0·8 and that of the line ② is 0·4. If the Eckert number is greater than unity, that is, the fluid is assumed to be liquid over a cross section, or if the variation of the physical properties is rather small at $E < 0$ where the fluid is assumed to be vapor over a cross section, the data can be correlated well with the three dimensionless groups: Nu_b, Re_b and Pr_b. In the pseudocritical region where the physical properties vary abruptly, however, the heat transfer coefficient does not increase so much as the Prandtl number does, and the conventional correlations do not hold.

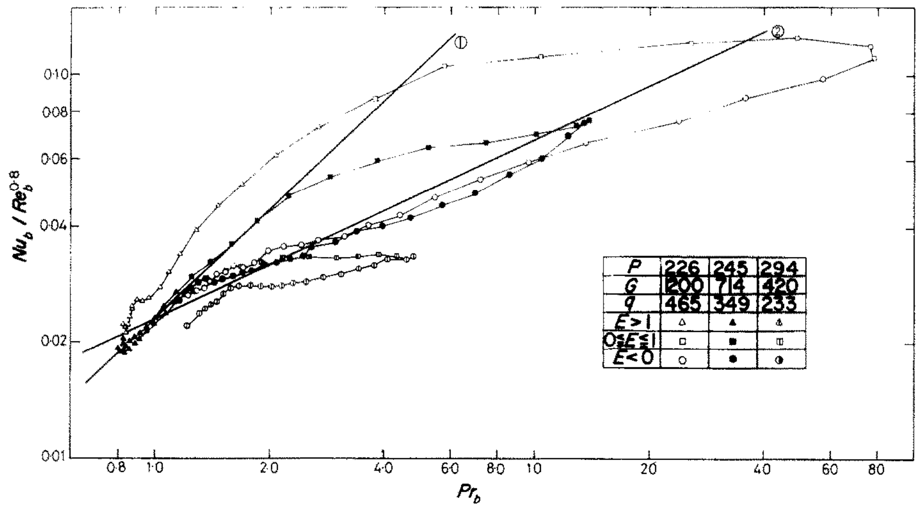


FIG. 8. Data plotted in terms of the conventional dimensionless groups.

From such a tendency of the heat transfer coefficient as mentioned above, Styrikowitsch *et al.* [24] proposed to use the Prandtl number, Pr_{min}, which is the lesser of the Prandtl numbers evaluated at the bulk temperature and at the wall temperature in correlating the data for supercritical water. Figure 9 shows $Nu_b/Re_b^{0.8}$ against Pr_{min} for the same data that were used in Fig. 8. The correlation is not satisfactory yet when the Prandtl number is large.

On the other hand, an attempt was made to take the effect of the variation of the fluid properties into account by evaluating the properties at a specific reference temperature. Eckert suggested on the basis of the analytical results obtained by Deissler [25] that the properties included in the Nusselt and Reynolds numbers might be evaluated at a reference temperature, t_x, which was t_w, t_m or t_b for $E > 1$, $0 \leqq E \leqq 1$ or $E < 0$, respectively, while the Prandtl number was evaluated at the wall temperature. In Fig. 10 the same data that were used in Figs. 8 and 9 are plotted in terms

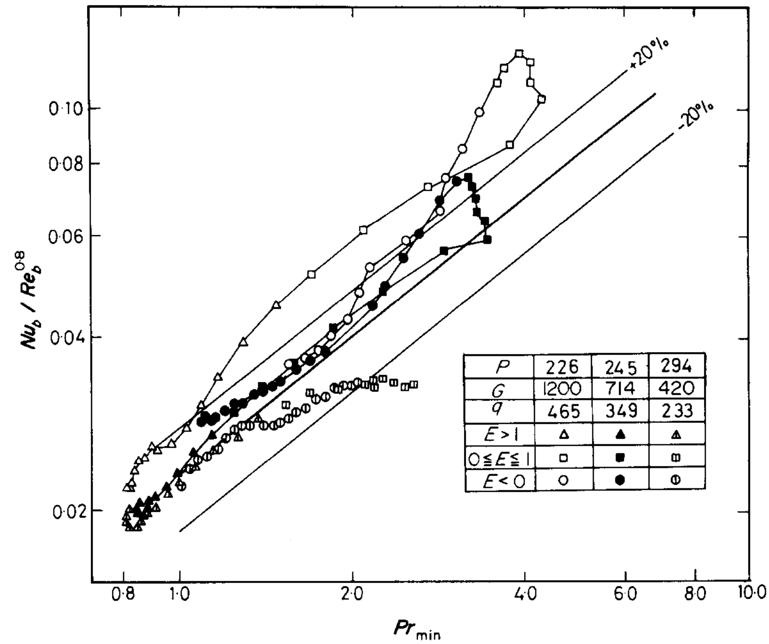


FIG. 9. Correlation of data by the method of Styrikowitsch *et al.*

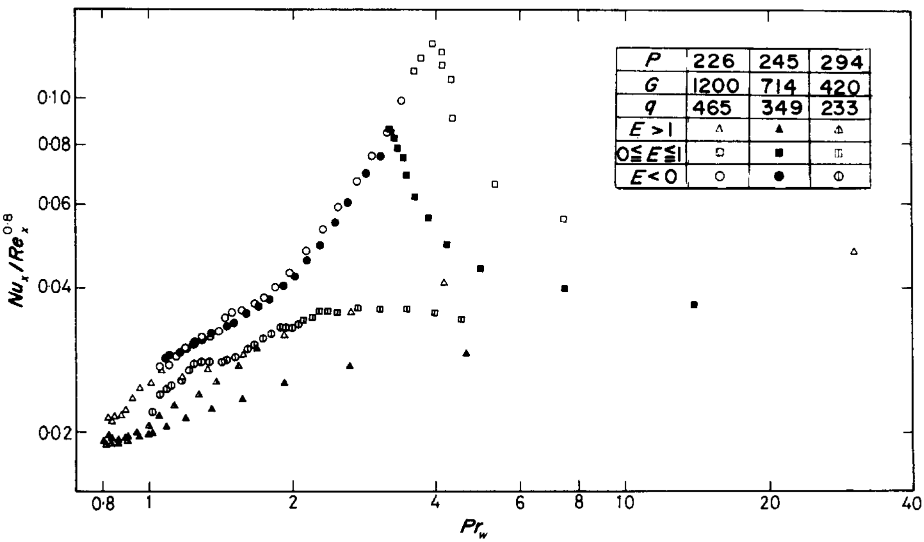


FIG. 10. Correlation of data based on reference temperature suggested by Eckert.

of $Nu_x/Re_x^{0\cdot8}$, which are evaluated at t_x, and Pr_w. The data are very scattered and it is apparent that this method is not useful for the correlation of the data on heat transfer to supercritical water.

It is considered that such tendencies of the heat transfer coefficients as described above require other factors besides the conventional dimensionless groups to be included in the correlation. After great labor, a ratio of specific heat, $\bar{c}_p/c_{pb}$, was found to be most effective in correlating the data, where $\bar{c}_p$ is an integrated average specific heat as defined by

$$\bar{c}_p = \frac{h_w - h_b}{t_w - t_b}. \tag{2}$$

In addition to this, other property ratios, such as density, viscosity and thermal conductivity ratios, were tried to account for the physical property variations, but they hardly improved the fit of the data, though they make the calculation rather tedious.

Figure 11 shows all the experimental values of $Nu_b/kRe_b^m Pr_b^n$ plotted as a function of the ratio $\bar{c}_p/c_{pb}$. By many trials on the various values of the exponents of the Reynolds number and the Prandtl number, it was found that $m = 0\cdot85$ and $n = 0\cdot8$ as shown in Fig. 11 gave the best correlation and then the constant k is 0·0135. In Fig. 11 three regimes are clearly distinguished: $E > 1$ where the fluid is considered to be liquid, $E < 0$ where considered to be vapor and $0 \leq E \leq 1$ where considered to be vapor near the wall and liquid in the core, and different lines fit the data in three specified regimes respectively. If examined in detail, however, data are somewhat scattered systematically depending upon the pressure. This may be considered to result from the fact that the specific heat depends remarkably upon the pressure near the pseudocritical point. Therefore, the correlation was improved by taking account of the Prandtl number, Pr_m, at the pseudocritical point and finally the following correlation was obtained:

$$Nu_b = 0\cdot0135\, Re_b^{0\cdot85}\, Pr_b^{0\cdot8}\, F_c \tag{3}$$

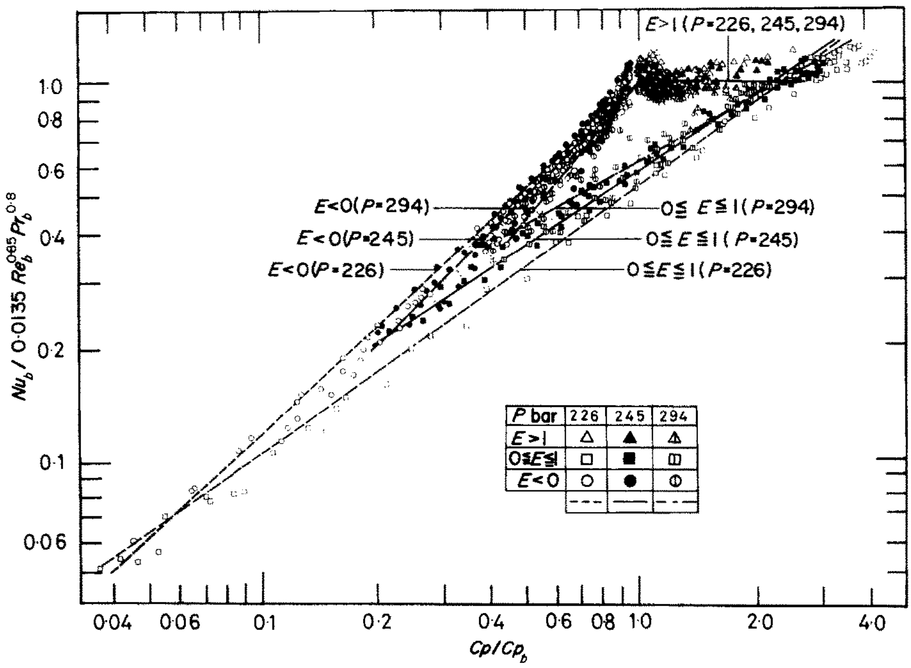


FIG. 11. Dimensionless correlation of heat transfer coefficients for supercritical water in tubes.

where

$$F_c = 1{\cdot}0 \text{ for } E > 1 \quad (4)$$

$$F_c = 0{\cdot}67\, Pr_m^{-0{\cdot}05}\, (\bar{c}_p/c_{pb})^{n_1} \text{ for } 0 \leqq E \leqq 1 \quad (5)$$

$$F_c = (\bar{c}_p/c_{pb})^{n_2} \text{ for } E < 0 \quad (6)$$

$$n_1 = -\,0{\cdot}77\,(1 + 1/Pr_m) + 1{\cdot}49 \quad (7)$$

$$n_2 = 1{\cdot}44\,(1 + 1/Pr_m) - 0{\cdot}53. \quad (8)$$

The values of n_1, n_2 and $0{\cdot}67\ Pr_m^{-0{\cdot}05}$ at several pressures are shown in Table 1. The straight lines shown in Fig. 11 correspond to the above equations at pressures of 226, 245 and 294 bar, respectively.

Table 1. The values of n_1, n_2 *and* ($0{\cdot}67\ Pr_m^{-0{\cdot}05}$) *in equations (5) and (6) at several pressures*

P (bar)	n_1	n_2	$0{\cdot}67\ Pr_m^{-0{\cdot}05}$
226	0·71	0·93	0·54
245	0·66	1·01	0·59
294	0·56	1·21	0·62

Figure 12 shows the comparison of the experimental Nusselt numbers with those calculated by equation (3). The data that show rather considerable discrepancies are several of those corresponding to the region near the point of $E = 0$ or $E = 1$, but almost all data points lie within $\pm$ 20 per cent of the predicted values.

Styrikowitsch *et al.* [24] and Swenson *et al.* [3] proposed the correlations for forced convection to supercritical water, and Petukhov *et al.* [26] and Krasnoshchekov and Protopopov [27] proposed the correlations for carbon dioxide. Figure 13 shows a comparison of the predictions of these correlations and the present

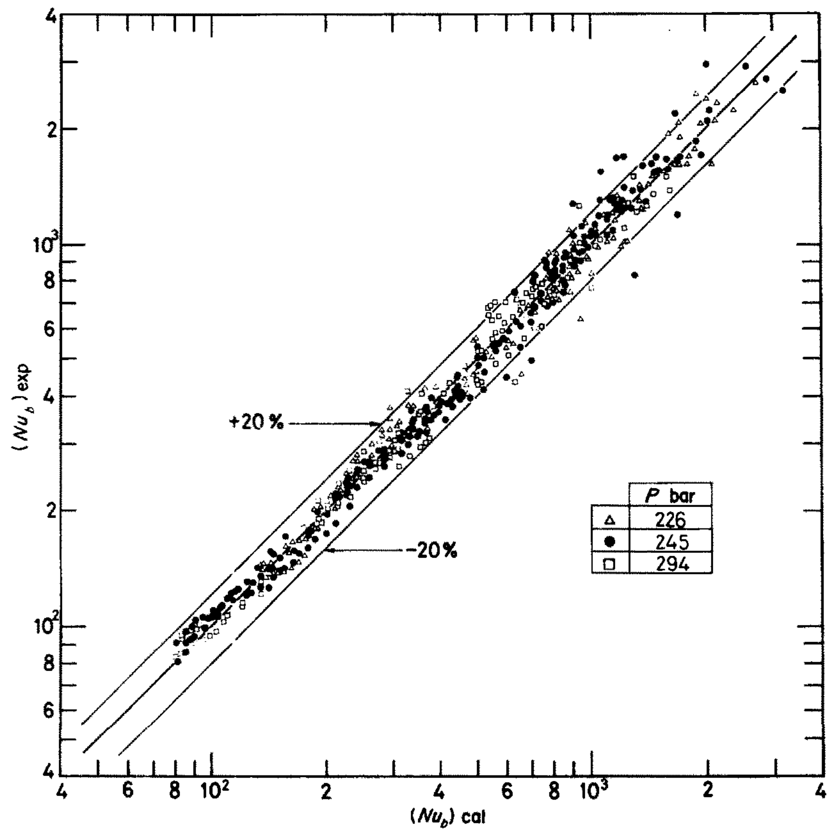


FIG. 12. Experimental data compared with equation (3).

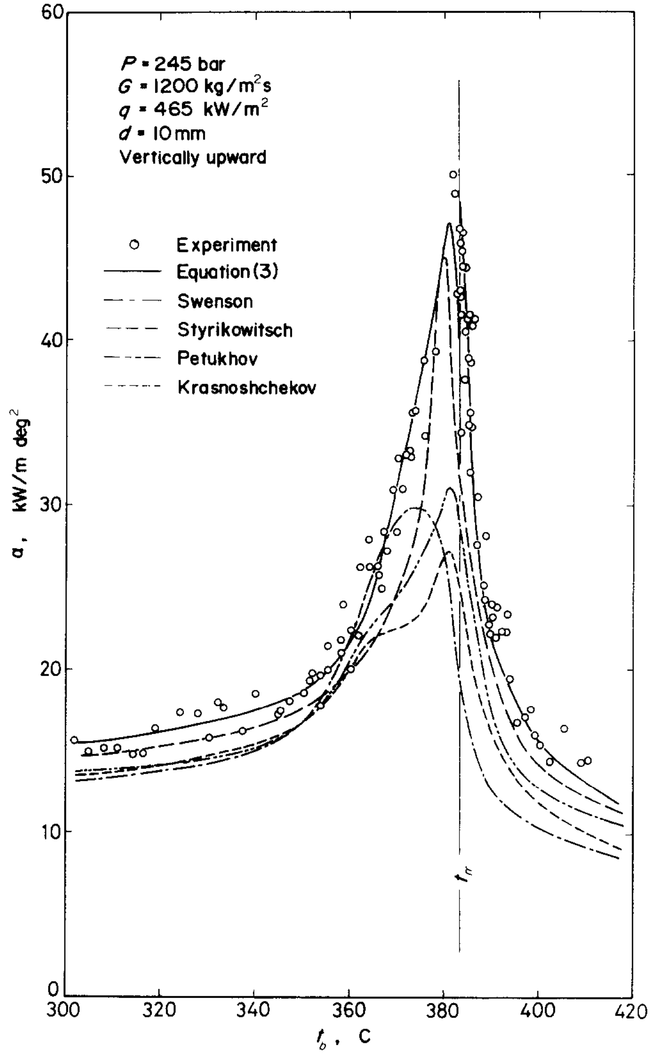


FIG. 13. Comparison of the predictions of various correlations with experimental data.

correlation, equation (3) with the experimental data for the vertically upward flow: the heat transfer coefficient is shown as a function of the bulk temperature on one of the typical conditions of $P = 245$ bar, $q = 465$ kW/m^2 and $G = 1200$ kg/m^{2}s.

The correlation of Styrikowitsch fits the data rather well, but it predicts the heat transfer coefficient smaller than the measured value near the pseudocritical point and the bulk temperature at which the predicted coefficient have a maximum shifts somewhat to lower temperature. The correlation of Swenson generally predicts the heat transfer coefficient considerably smaller than the measured value. The heat transfer coefficients predicted by the correlation of Petukhov and that of Krasnoshchekov, which were derived on the carbon dioxide data, are also considerably smaller and shows the similar trend to each other.

The value of the heat transfer coefficient calculated by equation (3) is discontinuous at the point $E = 1$ from the nature of the equation. This is a weak point of the present correlation. The "jump" of the value at this point is, however, so sufficiently within an acceptable accuracy that it does not matter in practical application. In order to correct this weak point for the value of the heat transfer coefficient to be continuous, if needed, a modification may be suggested: if the calculated value of the heat transfer coefficient, α, in the region of $E < 1$ ($t_b > t_m$) is larger than that, α_m, at $E = 1$ ($t_b = t_m$), then α may be taken to be α_m.

4.3 *Heat transfer at high heat fluxes*

At high heat fluxes the heat transfer in the pseudocritical region assumes a quite different aspect from that at low heat fluxes relatively to the flow rate. Figure 14 shows such an example: the measured wall temperatures are plotted against bulk enthalpy for a heat flux of 465 kW/m^2 and a mass velocity of 403 kg/m^{2}s upwards at a pressure of 245 bar. In this figure a dot-dash-line shows the calculated wall temperature by equation (3). The data points marked with squares correspond to the wall temperature for such flow as the pressure pulsates with about a half second period and one bar amplitude due to the circulating pump, and the points marked with circles correspond to the wall temperature for the flow where the pressure pulsation is removed by the accumulator.

The characteristics of the heat transfer at high heat fluxes which are seen from this example and which have been clarified from the results of the present experiment in wide ranges of parameters are as follows:

(i) The wall temperature is remarkably higher than what is calculated by equation (3) which holds at low heat fluxes relatively to the flow rate, and hence the deterioration occurs in heat transfer.

(ii) The heat transfer coefficient is considerably different according to the flow conditions whether the pressure pulsation caused by the circulating pump is present or not. For the flow with the pressure pulsation, increases in heat transfer coefficient and resulting decreases in

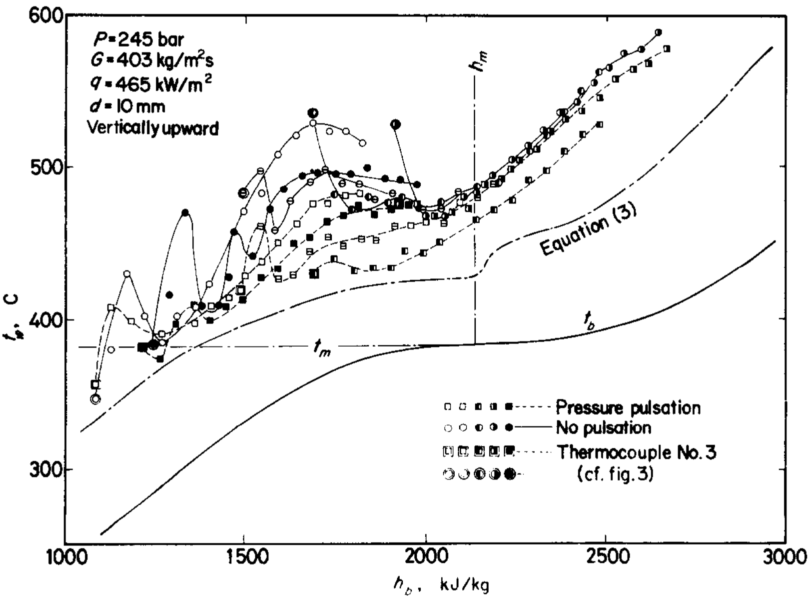


FIG. 14. Wall temperature vs. bulk enthalpy in the deteriorated heat transfer region.

the wall temperature are observed.

(iii) The measured temperature shows remarkable irreproducibility according to the fluid temperature at the inlet of the test section.

(iv) Although it is not very apparent where the deteriorated heat transfer begins, it seems to begin at the lower bulk enthalpy as the heat flux is higher or the mass flow rate is lower.

(v) Generally in the neighborhood of 1700 kJ/kg of bulk enthalpy which is somewhat lower than the pseudocritical enthalpy, the rise of the wall temperature is most remarkable and it covers a considerable length of the tube.

(vi) The sharp rise of the wall temperature as mentioned above is no more observed at the bulk enthalpy above the pseudocritical enthalpy.

(vii) The deterioration in heat transfer is observed over the pressure range of this experiment (226–294 bar).

Recently the deterioration in heat transfer to supercritical fluids has become of intense interest, but a clear explanation has not been made in regard to the mechanism of this phenomenon. Although the occurrence of a pseudo-film-boiling is presumed from the wall temperature rise similar to that on boiling crisis at subcritical pressures, this cannot be definitely concluded from the results of this experiment, and further close investigation will be necessary.

Figure 15 shows the relation between the limit heat flux above which the deterioration occurs in heat transfer (whatever the fluid temperature required) and mass velocity for the vertically upward flow in the 10 mm i.d. tube. The transition to the region of the deteriorated heat transfer which is caused by increasing the heat flux at a fixed flow rate is not abrupt but gradual. Hence in Fig. 15 data are plotted which could be clearly distinguished. Although it seems that the limit heat flux is somewhat increased with increase of pressure, this is not clear and the limit heat flux, q_c, in kW/m^2 may be related to the mass velocity, G, in kg/m^{2}s by the expression as

$$q_c = 0{\cdot}20\, G^{1{\cdot}2} \qquad (9)$$

for the vertically upward flow of water in a 10 mm i.d. tube at pressures of 226–294 bar.

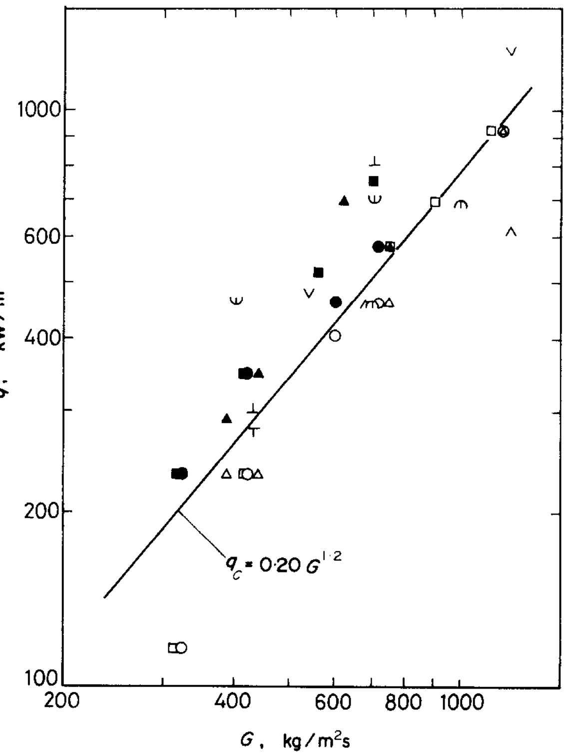


FIG. 15. Limit heat flux vs. mass velocity for deterioration in heat transfer.

	P bar	Deterioration	No deterioration
The authors	226	▲	△
	245	●	○
	294	■	□
Shitsman	233, 253	⊥	T
Vikrev	226–294	ω	ᴖ
Ackerman	228, 310	V	Λ

In Fig. 15 also plotted are the data which are quated from papers by Shitsman (pressures of 233 and 253 bar, an 8 mm i.d. tube) [9], Vikrev and Lokshin (pressures of 226–294 bar, a horizontal 10 mm i.d. tube) [10] and Ackerman (pressures of 228 and 310 bar, a 9·4 mm i.d. tube) [16]. The present result is compatible with these data.

5. CONCLUSIONS

1. At low heat fluxes relatively to the flow rate, the heat transfer coefficient has a maximum at a bulk temperature slightly less than the pseudocritical temperature. The maximum gets progressively less as heat flux or pressure is increased.

2. At sufficiently low heat fluxes, the heat transfer coefficients for horizontal flow are uniform around the tube periphery and are equal to those for vertical flow. At some higher heat fluxes, the heat transfer coefficient at the bottom of the horizontal tube is higher and the coefficient at the top is lower than those for vertical flow.

3. Heat transfer coefficients at low or moderate heat fluxes for supercritical water can be correlated by equation (3), in which the physical property variation across the tube section is taken account of in terms of a ratio of specific heat.

4. At high heat fluxes relatively to the flow rate, a deterioration in heat transfer occurs in the pseudocritical region. Several characteristics of this phenomenon have been made clear.

5. The limit heat flux above which heat transfer is remarkably deteriorated is related to the mass velocity by equation (9) for the vertically upward flow of water in a 10 mm i.d. tube at pressures of 226–294 bar.

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TRANSFERT THERMIQUE PAR CONVECTION FORCEE DANS LES TUBES POUR UN ECOULEMENT D'EAU SUPERCRITIQUE

Résumé— Des recherches expérimentales ont été faites sur le transfert thermique pour un écoulement d'eau supercritique dans un tube horizontal et des tubes verticaux. Un large ensemble de résultats est obtenu pour des pressions variant de 226 à 294 bar, des températures de mélange variant de 230 à 540° C, des flux thermiques de 116 à 930 kW/m^2 et des vitesses massiques de 310 à 1 830 kg/m^{2}s. Parce que les propriétés physiques de fluides supercritiques varient rapidement avec la température dans la région pseudo-critique, les coefficients de transfert thermique montrent un comportement inhabituel dépendant du flux thermique. A des flux thermiques bas ou modérés relativement au débit, on a obtenu une relation satisfaisante qui prédit raisonnablement bien les coefficients de transfert thermique près du point pseudo-critique. Les différentes caractéristiques de la détérioration du transfert thermique qui se produit pour des flux thermiques élevés sont clarifiées, et le flux thermique limite correspondant à l'apparition de cette détérioration est déterminé en relation avec le débit.

ZWANGSKONVEKTIVE WÄRMEÜBERTRAGUNG AN ÜBERKRITISCHES, DURCH ROHRE FLIESSENDES WASSER

Zusammenfassung— Die Wärmeübertragung an überkritisches, durch horizontale und vertikale Rohre fliessendes Wasser wurde experimentell untersucht. Umfassende Ergebnisse wurden erhalten für: Drücke von 226 bis 294 bar, mittlere Temperaturen von 230 bis 540°C, Wärmestromdichten von 116 bis 930 KW/m^2 und Durchsätze von 310 bis 1830 kg/m^{2}s. Da die physikalischen Eigenschaften überkritischer Fluide sich mit der Temperatur im pseudokritischen Bereich rasch ändern, zeigen die Wärmeübertragungskoeffizienten—abhängig von der Wärmestromdichte ungewöhnliches Verhalten. Bei relativ zum Mengenstrom niedrigen Wärmestromdichten wurde eine befriedigende Korrelation erhalten, die recht gut die hier besonders betrachteten Wärmeübertragungskoeffizienten nahe dem pseudokritischen Punkt beschreibt. Die verschiedenen Charakteristiken der Verschlechterung im Wärmeübergang, die bei hohen Wärmestromdichten auftreten, wurden geklärt und die Grenz-Wärmestromdichte für das Auftreten der Verschlechterung wurde in Verbindung mit dem Mengenstrom bestimmt.

ПЕРЕНОС ТЕПЛА ПРИ СВЕРХКРИТИЧЕСКИХ ПАРАМЕТРАХ В ТРУБАХ ПРИ ВЫНУЖДЕННОЙ КОНВЕКЦИИ

Перенос тепла при сверхкритических параметрах в трубах при вынужденной конвекции

Аннотация—Приведено экспериментальное исследование переноса тепла при сверхкритических параметрах в горизонтальной и вертикальной трубах. Получено большое количество данных для давления от 226 до 294 бар, температуры от 230 до 540°C, тепловых потоков от 116 до 930 квт/м^2 и массовой скорости от 310 до 1830 кг/м^2 сек.

Поскольку физические свойства при сверхкритических параметрах быстро изменяются с изменением температуры в псевдокритической области, изменение коэффициентов теплообмена в зависимости от тепловых потоков несколько аномально. Для малых и средних тепловых потоков получено удовлетворительное соотношение, которое позволяет довольно правильно рассчитать коэффициент теплообмена возле псевдокритической точки. Получено несколько характеристик ухудшенного теплообмена при больших тепловых нагрузках, причем устанавливаем связь предельных тепловых нагрузок с величиной расхода для случая ухудшенного теплообмена.