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TRANSITION AND FLOW BOILING HEAT TRANSFER INSIDE A HORIZONTAL TUBE

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Abstract—Experiments on transition and flow boiling heat transfer with refrigerant R114 inside a horizontal tube were performed at bubble flow, critical heat flux and in the transition region between bubble flow and film boiling at mass fluxes between 1200 and 4000 kg/m^2 s and in the pressure range between 5 and 15 bar. In comparison with pool boiling bubble flow heat transfer depends essentially on the mass flow rates and on the vapor quality. The critical heat flux depends less on the temperature difference than in pool boiling heat transfer and exhibits a maximal and a minimal value as a function of the pressure. The critical heat flux increases with mass flow rate as already shown by Collier. In the region of transition boiling the heat flux over the difference between wall and saturation temperature approaches a horizontal curve. Therefore in this region an evaporator may always be operated under stable conditions and burn out does not occur.

NOMENCLATURE

f, correction factor;
h, heat-transfer coefficient;
h_{ncb}, heat-transfer coefficient in natural convection boiling;
h_{fc}, heat-transfer coefficient in forced convection flow;
$\dot{m}$, mass flow density;
$\dot{m}''$, mass flow density of vapor;
$\dot{m}'$, mass flow density of liquid;
p, pressure;
p_c, critical pressure;
$\dot{Q}$, heat flux;
$\dot{q}$, heat flux density;
$\dot{q}_{ncb}$, heat flux density at given ΔT in natural convection boiling;
$\dot{q}_{nf}$, heat flux density in forced convection flow;
R_w, resistance to heat transfer of the heating wall;
T, absolute temperature;
T_w, wall temperature;
T_s, saturation temperature;
ΔT, $= T_w - T_s$, temperature difference;
ΔT_f, temperature difference in pool boiling;
ΔT_c, temperature difference in convection flow boiling;
x^*, $= \dot{m}''/(\dot{m}''+\dot{m}')$, vapor quality;
Re, Reynolds number of the liquid defined as

$$Re = \frac{\dot{m}(1-x^*)d}{\nu_1 \cdot \rho_1},\ \dot{m} = \text{total mass}$$

flow rate, x^* = vapor quality, ν_1 = viscosity, ρ_1 = density of the liquid, d = ID of tube;

Fr, Froude number of the liquid defined as

$$Fr = \frac{\dot{m}^2}{\rho_1^2}\frac{(1-x^*)^2}{gd}$$

g = gravity.

1. INTRODUCTION

EVAPORATORS are usually operated well below the maximum heat flux density either in the region of convection boiling or in some cases also in the region of film boiling. There has been little interest in transition boiling heat transfer until recent years due to the difficulties of stable operation in this region. However, it is possible to transfer heat also at maximum heat flux density and in the transition region between nucleate and film boiling without a change of the boiling mechanism. The stability criteria were derived by Stephan [1, 2], then established by Kovalev [3–6] and also by Grassmann and Ziegler [7].

Experiments in the transition region were performed by Stephan [1, 2], Kovalev [3] and also by Nishikawa, Hasegawa and Hondu [8], Kesselring, Rosche and Bankoff [9], Veres and Florschuetz [10], Hesse [11], Happel and Stephan [12] and recently by Canon and Park [13], who again demonstrated that stable boiling can be obtained over the complete transition region.

As shown in earlier papers [1, 2] stable operating points of an evaporator are obtained, if at the intersection point of the characteristic boiling, curve 1 in Fig. 1, and the characteristic of the heating surface, curve 2, the gradient of the boiling curve is greater than the gradient of the heating surface characteristic

$$\frac{1}{R_w} > -\frac{d\dot{Q}}{d(\Delta T)},$$

where R_w is the resistance to heat transfer of the heating wall.

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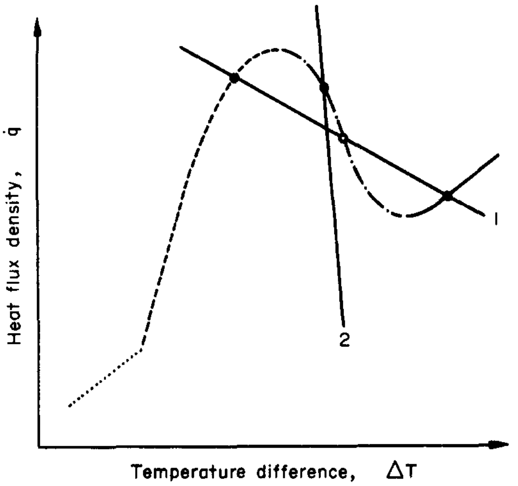


FIG. 1. Operating points.

The above cited experiments refer to pool boiling heat transfer on flat surfaces or outside horizontal tubes. This paper deals with experiments on flow boiling heat transfer inside a horizontal tube. Contrary to the large number of test results for the range of nucleate flow boiling in tubes there exist only few data concerning the maximum heat flux density and no data concerning heat transfer in the transition region of forced convection flow. The present experiments therefore appear to be the first to study transition flow boiling inside a horizontal tube. The experiments were performed with refrigerant R114, flowing inside a horizontal tube at mass flow rates between 1200 and 4000 kg/m^2 s and pressures between 5 and 15 bar. The measurements reported here were done at low vapor qualities $x^* \leqslant 0.1$, in order to eliminate the influence of vapor content on heat transfer. The results therefore may be applied to heat transfer in short tubes. Further experiments taking into account higher vapor qualities are presently being done.

2. EXPERIMENTAL EQUIPMENT

The experimental equipment is shown in Fig. 2. Main parts are the evaporator circuit, through which the evaporating liquid is pumped, and the stabilizing circuit which heats and stabilizes the boiling liquid. The liquid flows from the pump of the evaporator circuit through a mass flow control device. Small

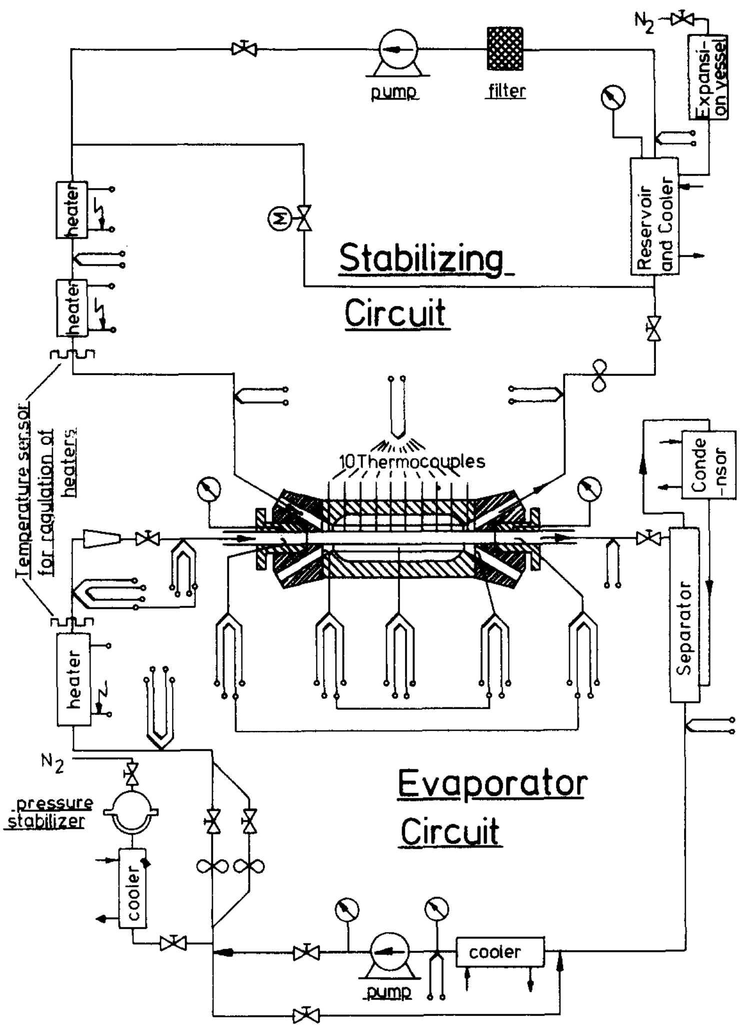


FIG. 2. Experimental equipment.

pressure fluctuations coming from the pump are eliminated by a pressure stabilizer. The volume flow rate is measured by turbine flow meters. Hence the mass flow rate is given by multiplication with the density which itself is determined by temperature measurements with a thermocouple. The fluid in the heater reaches saturation temperature, the pressure inside the heater being above the saturation pressure in the evaporator. After leaving the heater the fluid passes a nozzle where its pressure is reduced to saturation pressure. In all experiments the vapor content at the inlet of the test section was kept zero. Therefore no vapor was allowed to be formed during the expansion in the nozzle, which was inforced by equal temperatures at the inlet and outlet section of the nozzle. The saturation temperature could be controlled by a thermocouple at the entrance of the test section. Boiling occurred inside the horizontal tube of 14 mm diameter of the test section. This tube was heated by the stabilizing liquid in an annular space outside of 3 mm width. Furthermore part of the liquid evaporates due to the expansion that takes place in the test section. The two-phase flow leaving the test section with a maximum vapor quality of $x^* = 0.1$ passes through a temperature and pressure measuring cylinder into the separator, from where the vapor moves into the condenser. It is condensed by cooling with air or with pressurized water. The condensate flows via a cooler to the suction side of the pump. By means of the cooler cavitation inside the pump could be avoided. As stabilizing liquid fully degassed water was chosen. Decisive factors for the choice of water as stabilizing liquid were its high heat capacity and furthermore the high transfer coefficient that can be achieved. Due to the high heat capacity fluctuations in heat flux cause only small fluctuations of temperature. The high heat-transfer coefficients produce a steep evaporator characteristic, curve 2 in Fig. 1, and therefore favorize stable states in boiling. The water flows from the pump through a control device for the volume flow rate into a multi-stage heater and from there into the annular space of the test section. In order to achieve a regular flow in the test section and avoid secondary flow the device before the entrance is subdivided into 10 separate sections which continuously pass over into the annular space. Wall temperatures were measured by means of 10 thermocouples stemmed into the inside wall along an axial line. The temperature difference of the stabilizing liquid was registered by two thermocouples in difference connection (3). Multiplication with the mass flow rate of the water gave then the mean heat flux density transferred in the evaporator. The mass flow rate itself was determined in a device behind the test section consisting of a turbine flow meter and a thermocouple. From this device the stabilizing liquid flew back over a reservoir, serving also as a cooler, across a filter and into the suction side of the pump. The reservoir was connected to an expansion vessel, where the liquid might be pressurized by means of nitrogen in order to avoid evaporation in the stabilizing circuit even at water temperatures above 100°C.

3. EXPERIMENTAL RESULTS

In Fig. 3 the boiling characteristic of R114 at a pressure of 6 bar is compared with the curve measured by Hesse [11] in natural convection boiling outside a horizontal tube of 14 mm in diameter. In the nucleate boiling region both curves are practically parallel. In

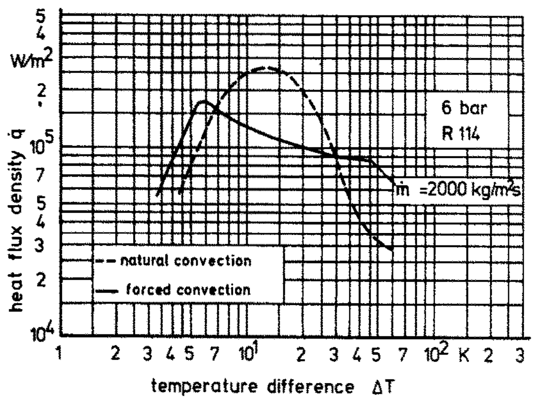


FIG. 3. Characteristic boiling curve.

addition to the parameters for natural convection boiling the mass flow rate and the vapor content become important in forced convection boiling. As the comparison shows at constant heat flux density $\dot{q}$ the temperature difference ΔT in free convection boiling is greater than that in forced convection boiling. The ratio of heat-transfer coefficients h_{fc} in forced convection boiling and h_{ncb} in natural convection boiling is practically constant in the region of nucleate boiling as already shown by Chawla [14] in experiments with different refrigerants. All heat flux densities plotted in the following diagrams are mean values over the length of the test section. In fact the heat flux densities and also the saturation temperature, the pressure, the vapor content, the velocity, the flow pattern and also the wall temperature change along the tube axis. In forced convection boiling, heat-transfer coefficients are therefore, as already known, greater and the maximum heat flux densities smaller than in natural convection boiling. As the experiments show, the characteristic boiling curve in the transition region is flatter than the dashed curve for natural convection boiling. This is caused by the liquid flowing inside the tube thus preventing the formation of large vapor nests.

Figure 4 gives characteristic boiling curves, at a constant mass flow rate of $\dot{m} = 2000\,\mathrm{kg/m^2\,s}$ and for pressures between 5 and 15 bar. In the region of nucleate boiling it is well known that heat flux densities at constant temperature difference increase with increasing pressure. The maximum heat flux density at first increases with pressure up to a maximum value at about 7 bar, then decreases to a minimum value at about 9 bar and increases then again. From the experiments by Cichelli and Bonilla [15], later on confirmed by many other authors, it is known that the maximum heat flux density in natural convection boiling over the reduced pressure p/p_c first increases, reaches a maximum value and then decreases again.

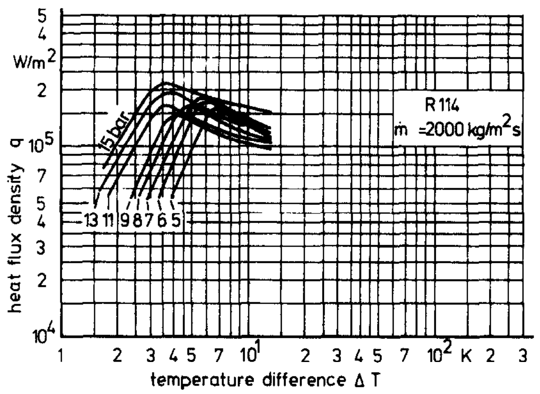
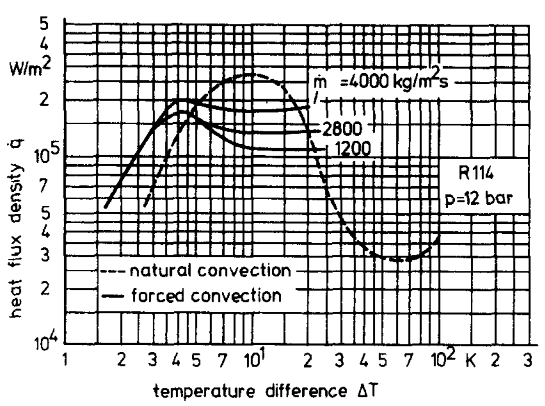


FIG. 4. Characteristic boiling curves for different pressures.

Contrary to this result in forced convection flow the maximum heat flux density experiences a second maximum after having passed through a minimum value. This effect may be explained if one considers that the liquid viscosity decreases when the saturation pressure is raised, hence the liquid boundary layer thickness decreases also. Simultaneously the surface tension decreases considerably from a value of 8.3 dyn/cm at a pressure of 6 bar to 3.8 dyn/cm at 15 bar and to 2.4 dyn/cm at 20 bar. The break-off diameter therefore decreases, the frequency becomes higher, heat transfer also increases and eventually the maximum heat flux density increases again in spite of the higher pressure. Furthermore one observes that also



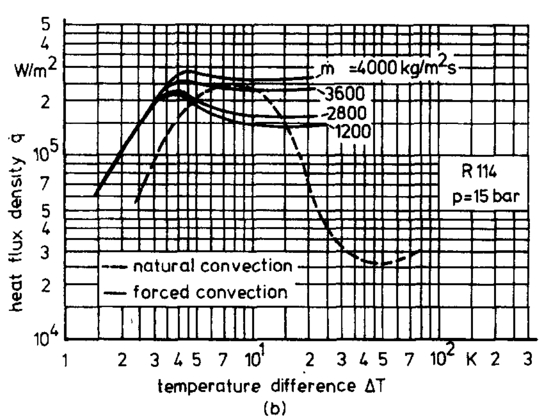


FIG. 5(a, b). Influence of mass flow rate on heat flux density.

the heat flux density in the transition region shows a similar tendency as a function of pressure.

In Fig. 5 the characteristic boiling curve is represented for different mass flow rates and pressures of 12 and 15 bar. As a comparison the dashed line represents the characteristic boiling curve for natural convection boiling. In the region of nucleate boiling there is only a small influence of mass flow rate on heat flux density. According to Chawla's [14] equation for nucleate flow boiling when applied to mass flow rates of 4000 kg/m^2 s one obtains heat-transfer coefficients which are only 13% greater than those for 1200 kg/m^2 s. The maximum heat flux density and the respective temperature difference ΔT increase with the mass flow rate which is in agreement with predictions by Collier [16]. For a pressure of 15 bar the maximum heat flux density exceeds even the values for natural convection flow. This may be explained by the high heat-transfer coefficients due to the thin boundary layer at higher mass flow rates. In the region of transition boiling the suspending curves become more and more horizontal when mass flow rates increase. These experimental findings are physically plausible because the high liquid velocities prevent the vapor to adhere at the wall and thus cause higher heat transfer coefficients.

4. TEMPERATURE DIFFERENCES IN AXIAL DIRECTION

For evaluation of the temperature difference needed for the characteristic boiling curve the arithmetic mean value of the local temperature differences over the test section were determined. However, the local temperature differences along the test section give some qualitative information on the local state of the boiling liquid. Due to the variation of velocity, flow pattern, vapor quality, saturation temperature and pressure in flow direction also the heat flux densities are locally different and cannot be assessed from the present experiments. By comparing the local temperature differences for different heat flux densities one can however draw some conclusions about the local heat flux density.

The characteristic boiling curve of Fig. 6 represents heat flux densities referring to the arithmetic mean value of the temperature differences over the tube length. The curves in Fig. 7 demonstrate how the local temperature differences in the different boiling ranges

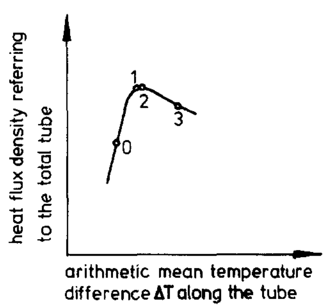


FIG. 6. Characteristic boiling curve.

change along the tube, the curves indicated by the symbols 0* to 3* in Fig. 7 refer to the states 0 to 3 in Fig. 6.

As long as boiling occurs only in the region of nucleate boiling, the local temperature difference remains practically constant in flow direction, curve 0* in Fig. 7. For a mean heat flux density, point 1 in Fig. 6, the local temperature difference is represented by curve 1* in Fig. 7. As a result from the experiments one finds an increase of the temperature difference followed by a slight decay towards the exit of the tube. Quite a

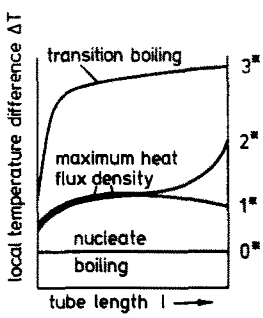


FIG. 7. Local temperature difference as a function of tube length.

different shape of the temperature difference near the tube exit was registered for mean temperature differences, point 2 in Fig. 6, slightly larger than the temperature difference for maximum heat flux density. The corresponding curve 2* in Fig. 7 found in the experiments, at first practically coincides with curve 1*. However, near the exit section of the tube, it shows a remarkable increase in temperature difference. This is evidently a consequence of the higher heat-transfer resistance due to the increase in vapor production towards the end of the tube. From this sharp increase in local temperature differences as soon as the average temperature difference is slightly above that of maximum heat flux density we may therefore draw the conclusion that for all states characterized by point 1 in Fig. 6, also the local maximum heat flux density will be obtained in the exit section of the test tube, as already predicted by Bell [17] and Bartoli [18]. Otherwise one would observe a pronounced increase in temperature differences towards the exit section.

5. EMPIRICAL CORRELATIONS

In order to correlate heat-transfer experiments in the region of nucleate boiling, one usually starts from the equations for natural convection boiling and introduces a correction term taking into account the influence of forced convection. Thus one obtains equations of the form

$$\dot{q} = \dot{q}_{ncb} + \dot{q}_{fc} \quad \text{or} \quad h = h_{ncb}\left(1 + \frac{h_{fc}}{h_{ncb}}\right) = h_{ncb}\cdot f. \tag{1}$$

Equations of the first type were introduced by Rohsenow [19], equations for the correction factor f depending on several dimensionless numbers given by Chawla [20].

The correction factor f by Chawla depends on the Reynolds- and the Froude-number of the liquid

$$f = 29 \cdot Re^{-0.3} Fr^{0.2}. \tag{2}$$

As shown by Danilowa [21] and Gorenflo [22] heat-transfer coefficients in natural convection boiling of refrigerants may be calculated from equations of the form

$$h_{ncb} = K \cdot \dot{q}^{n}(0.14 + 2.2\, p/p_c) \tag{3}$$

with $p/p_c \leqslant 0.5$. From equations (1) and (2) one finds for the heat-transfer coefficient in forced convection boiling the empirical formula

$$h = K'\dot{q}^{n}(0.14 + 2.2\, p/p_c) \cdot Re^{-0.3} Fr^{0.2}, \tag{4}$$

where the constant K' and the exponent n may be determined from experiments. As found from the measurements an average value for n proved to be 0.6, so that only one further experiment is necessary in order to determine the constant K'. As a reference point for K' the value $\dot{q} = 133\,500\,\mathrm{W/m^2}$, $\Delta T = 3.59\,\mathrm{K}$, $p = 9\,\mathrm{bar}$ ($p_c = 32.62\,\mathrm{bar}$), $\dot{m} = 2000\,\mathrm{kg/m^2\,s}$ was used. With this one finds $K' = 774.7$. Figure 8 compares calculated heat-transfer coefficients for $n = 0.6$ and $K' = 774.7$ from equation (4) and experimental results. The agreement is satisfactory.

In cases where no experiments on flow boiling heat transfer exist, heat-transfer coefficients may be calculated starting from one of the well-known equations for heat transfer in pool boiling that yields h_{ncb} at usually normal pressure. By means of equation (3) one obtains then the heat-transfer coefficient h_{ncb} for the desired pressure. From multiplication of h_{ncb} with the correction factor f by Chawla, equation (2), then follows the heat-transfer coefficient in forced convection boiling. This procedure, however, being not based on additional flow boiling experiments is less accurate as the one cited above. When using, e.g. the equation by Stephan [23] for pool boiling heat transfer, together with the Danilowa-correction and the Chawla-factor the error between experimental and calculated values is between 30 and 50% according to the pressure and mass flow density.

An analysis of the maximum heat flux data is very difficult because the influence of pressure in the high mass flow rate region studied in the experiments is evidently more complex than in the low mass flow region. There exist only a few experiments on the influence of system pressure on the maximum heat flux, especially for high mass flow rates. The well-known empirical equations for the maximum heat flux given by Thomson and Macbeth [24] correlate experimental data for water and may not be applied to boiling refrigerant R114. Insofar correlation of the few data obtained in these experiments seems to be useless until sufficient data exist. It should, however, be emphasized that Collier [25] predicted from the Thomson–Macbeth-correlation a possible secondary maximum when maximum heat flux density is plotted over the pressure, as found in the present experiments.

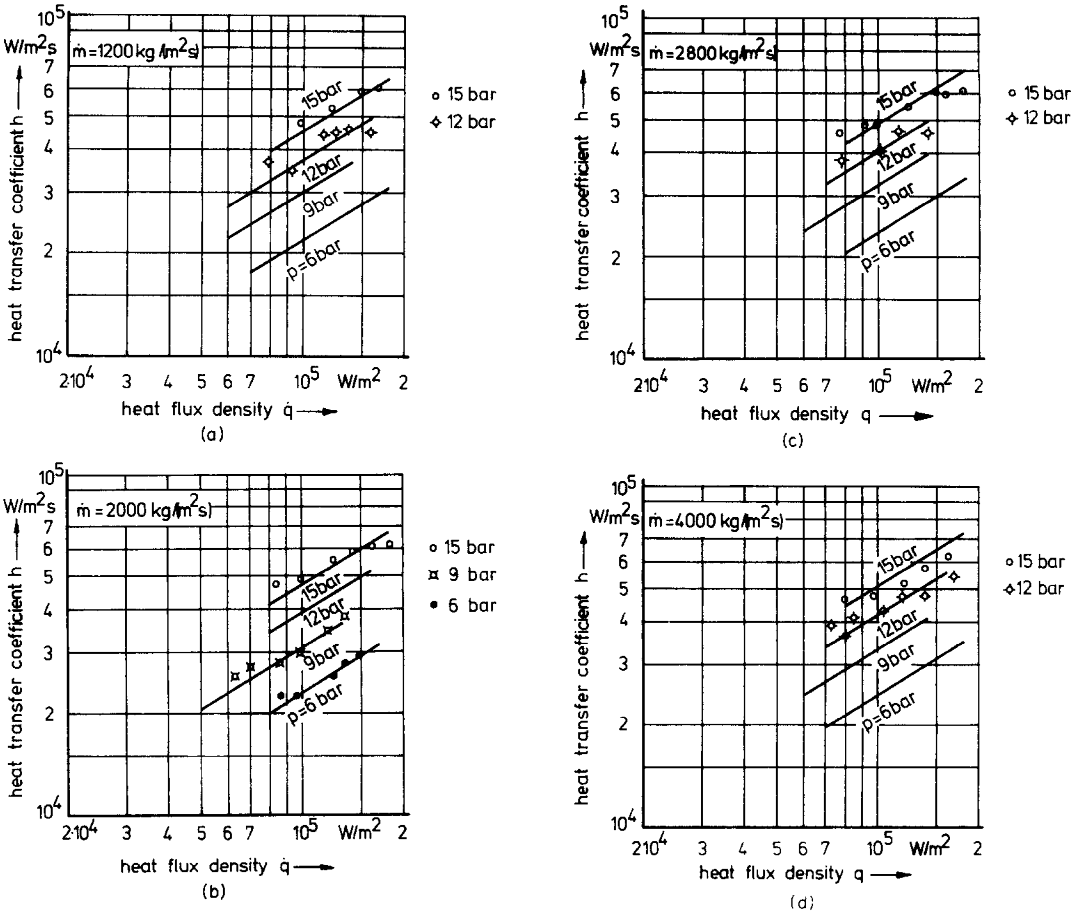


FIG. 8(a–d). Empirical correlations and experimental data.

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TRANSFERT THERMIQUE A LA TRANSITION ET POUR UN ECOULEMENT EN EBULLITION DANS UN TUBE HORIZONTAL

Résumé—On a effectué des expériences sur la transition et le transfert thermique dans un écoulement de fluide en ébullition, avec le réfrigérant R 114, dans un tube horizontal, pour l'écoulement avec bulles, pour le flux critique et dans la région de transition entre l'écoulement avec bulles et l'ébullition en film, à des débits massiques spécifiques entre 1200 et 4000 kg/m^2 s et à des pressions comprises entre 5 et 15 bar. Par rapport à l'ébullition en réservoir, le transfert thermique dans l'écoulement avec bulles dépend essentiellement des débits massiques spécifiques et de la qualité de la vapeur. Le flux critique dépend moins de la différence de température que dans l'ébullition en réservoir et on trouve que la valeur maximale et la valeur minimale sont fonctions de la pression. La densité de flux critique croît avec le débit massique spécifique comme cela a été montré par Collier. Dans la région de l'ébullition de transition, la densité de flux thermique divisée par la différence entre les températures de paroi et de saturation s'approche d'une courbe horizontale.

Par suite, dans cette région, un évaporateur peut toujours opérer dans des conditions stables et la crise ne peut se produire.

WÄRMEÜBERGANG IM WAAGERECHTEN ROHR BEI BLASEN- UND ÜBERGANGSSIEDEN IN ERZWUNGENER STRÖMUNG

Zusammenfassung—Es wurde der Wärmeübergang an siedendes R 114 im Bereich des Blasensiedens, der kritischen Wärmestromdichte und des Übergangssiedens bei erzwungener Strömung in einem horizontalen Rohr gemessen. Die Massenstromdichten wurden zwischen 1200 und 4000 kg/m2 s, die Drücke zwischen 5 und 15 bar geändert. Im Vergleich zum Sieden in freier Strömung hängt der Wärmeübergangskoeffizient maßgeblich von der Massenstromdichte und vom Strömungsdampfgehalt ab. Die kritische Wärmestromdichte hängt von der Temperaturdifferenz schwächer ab als beim Sieden in freier Strömung und durchläuft einen maximalen und einen minimalen Wert als Funktion des Druckes. Wie auch von Collier gezeigt, nimmt die kritische Wärmestromdichte mit der Massenstromdichte zu. Im Bereich des Übergangssiedens nähert sich die Wärmestromdichte als Funktion der Differenz zwischen Wand und Siedetemperatur einer horizontalen Linie. In diesem Bereich kann man daher einen Verdampfer immer stabil betreiben, ohne daß die Gefahr des Durchbrennens besteht.

ПЕРЕНОС ТЕПЛА В ОБЛАСТЯХ ПЕРЕХОДНОГО РЕЖИМА КИПЕНИЯ И КИПЕНИЯ ПРИ ТЕЧЕНИИ В ГОРИЗОНТАЛЬНОЙ ТРУБЕ

Аннотация — Экспериментально исследовался перенос тепла при течении хладагента R-114 в горизонтальной трубе в условиях пузырькового кипения, критического теплового потока и в области перехода от пузырькового кипения к пленочному при массовых потоках от 1200 до 4000 кг/м^2 сек и давлениях от 5 до 15 бар. В отличие от кипения в большом объеме при пузырьковом кипении в условиях течения теплоперенос существенно зависит от массового расхода жидкости и истинного паросодержания. По сравнению с процессом переноса тепла при кипении в большом объеме критический тепловой поток в меньшей степени зависит от разности температур и, в зависимости от давления, достигает максимального или минимального значения. Критический тепловой поток увеличивается с ростом массового расхода жидкости, как уже показано Колльером. В области переходного режима тепловой поток при температурах, превышающих разность между тепмературой стенки и температурой насыщения, стремится к горизонтальной кривой. Следовательно, в этой области испаритель может всегда работать при стационарных условиях.