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SHORTER COMMUNICATIONS

DROPWISE CONDENSATION—THE EFFECT OF SURFACE INCLINATION

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1. INTRODUCTION

MOST earlier experimental studies of heat transfer by dropwise condensation have been carried out with steam at pressures near atmospheric and using vertical condensing surfaces. The primary objective has generally been to measure the relation between the steam-to-surface temperature difference and the heat flux. Until relatively recently, agreement between the results of different workers has been poor. During the past few years, good agreement has been found between the results of certain experimenters [1–7]. The heat-transfer coefficients found in these works are generally much higher than those reported earlier. It has been suggested [2, 3, 7] that the main reason for earlier discrepancies has been the presence, in the vapour, of non-condensing gases. These are difficult to eliminate and even very small amounts greatly reduce the extremely high vapour-side heat-transfer coefficients found with dropwise condensation.

Other factors which have received attention are: material of condensing plate [8], surface finish [4, 8], promoter used [2, 4, 5], vapour used [9, 23], vapour pressure [1, 9–11], surface inclination [1, 13] and non-condensing gas content [1, 3, 12]. Unfortunately, owing to uncertainties arising mainly from the diversity of heat-transfer results for the case of steam at near-atmospheric pressures with vertical, smooth condensing surfaces, there is, at present, little generally accepted quantitative information. However, the following have been established: (1) surface finish and promoter used have small but measurable effect on the heat-transfer coefficient; (2) for surface inclinations sufficiently far from the vertical, the coefficient decreases with increasing departure from the vertical in either direction; and (3) the presence of non-condensing gases greatly reduces the coefficient.

Tests concerning the effect of vapour pressure [9–11], conducted at sub-atmospheric pressures, have indicated that the heat-transfer coefficient decreases as the pressure is lowered. This behaviour is to be expected on theoretical grounds [14–17] for condensation processes wherein the temperature drop in the vapour, arising from the non-equilibrium situation near the interface with the liquid, is a significant fraction of the vapour-to-wall temperature difference (as is thought to be the case for dropwise condensation and condensation of liquid metals). However, Wenzel [1], using pressures above atmospheric, observed the opposite trend. Moreover, this same author has proposed a theory [18] in which he disregards the interphase non-equilibrium effect (this may be valid for higher pressures and moderate heat fluxes) and, by relating the number of drop nucleating sites to the pressure, predicts a decrease in heat-transfer coefficient with increasing pressure. A possible alternative explanation for these observations could be that the effectiveness of the promoter (oleic acid) decreased with increasing temperature.

In contrast with the above-mentioned uncertainties regarding heat-transfer observations, distinct progress towards understanding the mechanism of dropwise condensation has recently been made by Umur and Griffith [19] and Westwater *et al.* [9, 20, 21]. These authors have shown that nucleation plays an important role in the mechanism of dropwise condensation and that, apart from a possible adsorbed mono-molecular layer, surface condensate films are not present. This has also recently been confirmed, for the case of mercury, by Ivanovskii, Subbotin and Milovanov [23].

A complete description of the phenomenon requires a knowledge of the heat-transfer rate for a drop of given size, together with the distribution of drop sizes, under prescribed conditions. The former involves the effect of non-equilibrium at the vapour-liquid interface (particularly at low pressures and high condensation rates), the effect of surface curvature in the case of the very small drops, in addition to heat-transfer in the drop itself, for which thermocapillary convection may play a significant role. The drop size distribution is governed by the density of nucleation sites, the growth rate and coalescence of drops and the effect of the sweeping action of falling drops.

Most of the above aspects of the problem have received and are receiving the attention of research workers. The present investigation was undertaken in order to provide reliable data relating to the last of the above-mentioned factors. Since it seems unlikely that surface inclination should have any significant effect on the heat-transfer and growth mechanism for adherent drops of given sizes, its importance lies mainly in governing the size at which a drop begins to slide and the velocity of decent. These in

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turn affect the overall size distribution and hence the mean heat-transfer rate for the surface.

2. APPARATUS

The apparatus described earlier [5–7] was modified to permit inclination of the test plate. Copper bellows, soldered at either end to brass plates, were introduced between the boiler and steam chamber so that the latter was able to rotate through an angle of 180° (see Fig. 1). Condensate drain tubes were fitted to prevent flooding of the condensing plate and window. The angle of inclination was measured by a protractor fitted to the steam chamber in the plane of rotation. The angle was indicated by a fine wire pinned at one end to the centre of the protractor and carrying a weight at the other.

In order to enhance the precision of the surface temperature measurements at the lower steam-to-surface temperature differences, three condensing plates of different thickness were used. In addition to the two plates (0·25 and 2·00 in thick) used earlier [7] for operation at the highest and lowest heat fluxes, a third plate of similar design and of thickness 0·5 in. was used for operation at intermediate fluxes. Thermocouples were located in all of the plates in the manner described earlier [6, 7], so that the heat flux could be found from the temperature gradient, and the surface temperature by extrapolation.

3. PROCEDURE

It was found earlier [6, 7] that, in the absence of forced convection near to the condensing surface, such traces of non-condensing gases as remain in steam after prolonged boiling, give rise to an additional temperature drop in the vapour of similar magnitude to that across the condensate for dropwise condensation. The technique used formerly [6, 7] to eliminate this error was also used in the present work.

The concentration of non-condensing gases in the steam was first minimized by boiling vigorously while "blowing off" steam to atmosphere, through a vent of 13 mm internal diameter in the steam chamber cover plate, for at least

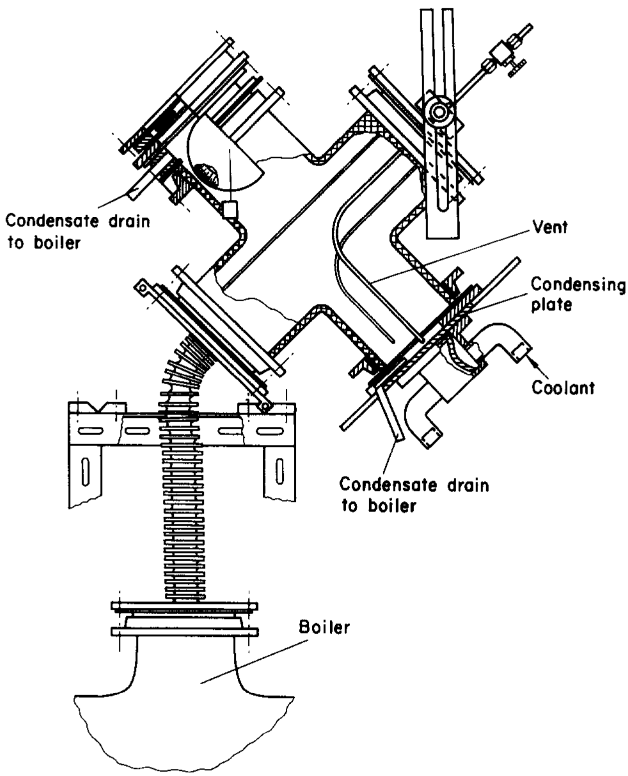

FIG. 1. The steam chamber.

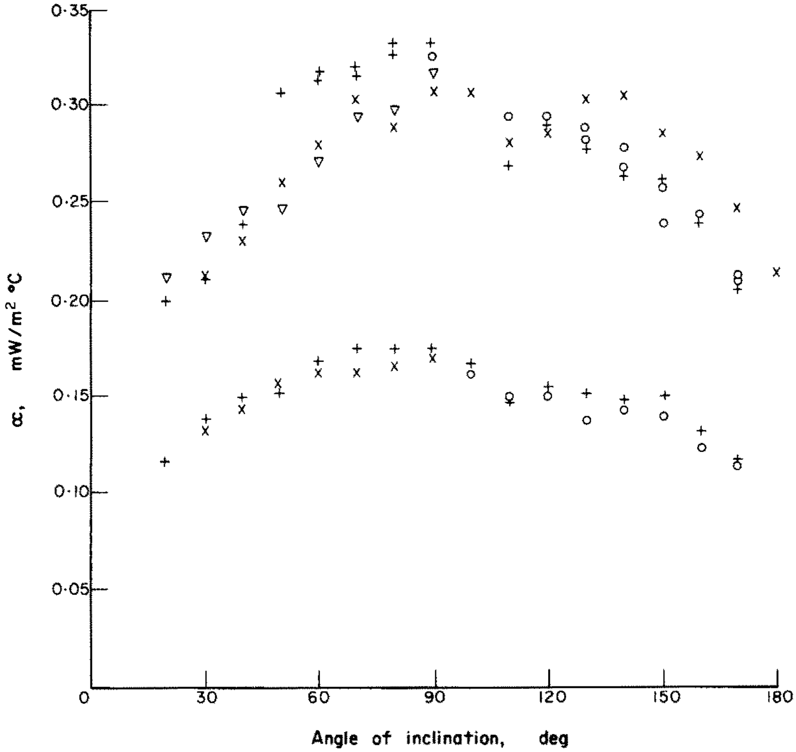

FIG. 2. Variation of the steam-side heat-transfer coefficient with surface inclination.
Sequence of observations: ● 90° → 180°
+ 180° → 20°
× 20° → 180°
△ 90° → 20°

15 min. Venting was thereafter continued using a vent of 4·8 mm internal diameter, situated so that its axis passed through the junctions of the thermocouples in the plate and the open end was 6·4 mm from the condensing surface. The pressure in the steam chamber was maintained at about 24 inH_2O above atmospheric, when the rate of removal of steam through the vent was above 0·65 g/s. Under these conditions the venting rate should be adequate to prevent error due to accumulation of non-condensing gases at the condensing surface, without being so large as to introduce error through disturbance of the condensate [6, 7]. It should be mentioned, however, that the above vent position and venting rate were established for the case of the vertical plate and for heat fluxes less than 1·5 MW/m^2. It is therefore possible that, for inclinations sufficiently far from the vertical, the precision of the present results might have been affected. However, since, as will be seen, the general pattern of the results did not depend on heat flux (which governs the rate at which non-condensing gas is brought to the surface) it is thought unlikely that errors attributable to venting were significant, for heat fluxes less than 1·5 MW/m^2 at least.

The promoter used in the present work was dioctadecyl disulphide. The method of cleaning and promoting of the plates was the same as that described earlier [5, 6]. The condensing surface was always cleaned and re-promoted following an overnight period of shut-down. For newly promoted surfaces, a preliminary condensing interval of at least 3 h was allowed, before any measurements were made, to obtain steady operating conditions [6, 7].

For given condensing plate and a fixed coolant flow rate, heat flux and steam-to-surface temperature difference were observed for a range of angles of inclination from 5° to the horizontal (surface facing upward) to 180° (drops hanging). Measurements were generally made at intervals of 10°. The same procedure was carried out for various coolant flow rates and using all three condensing plates in turn.

4. RESULTS

A typical selection of the results obtained is shown in Figs. 2 and 3. The angles of inclination on these graphs are measured from the horizontal, surface-facing-upward position. A lower limit to the angle of inclination which could be used was set by the problem of surface drainage. Figures 2 and 3 show results for three different coolant flow rates. However, since the varying vapour-side resistance repre-

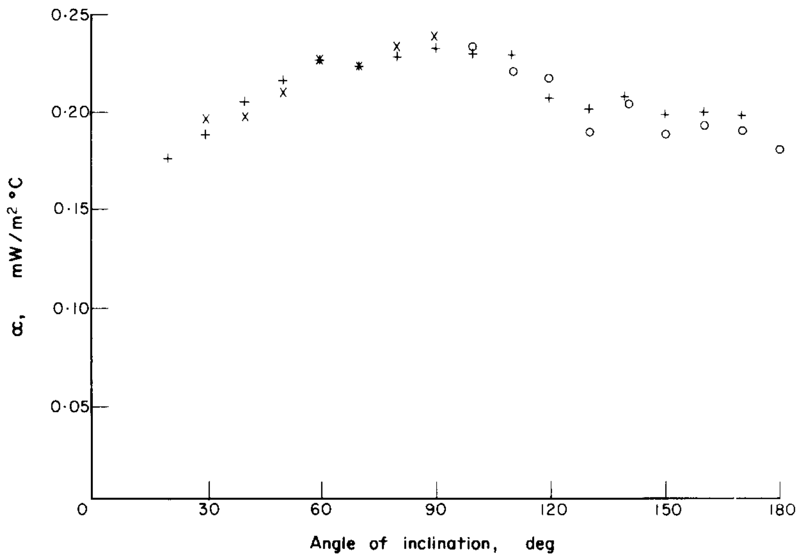

FIG. 3. Variation of steam-side heat transfer coefficient with surface inclination.
Sequence of observations: ● 90° → 180°
\+ 180° → 20°
× 20° → 90°

sents only a small fraction of the overall vapour-to-coolant resistance, the heat flux is essentially independent of inclination for given coolant inlet temperature and flow rate.

Despite some scatter, generally greater at the higher heat fluxes, the figures clearly indicate that the heat-transfer coefficient passes through two maxima, one occurring near the vertical position and the other at around 140°. On some occasions the second maximum was barely distinguishable but on others, when the scatter was small, was very marked as shown by Fig. 4. So far as could be judged the inclinations, at which the second maximum and the intervening minimum occurred, did not depend on the heat flux. In Fig. 5 all of the results are shown on the same graph in which the ratio of the steam-side coefficient to its value for the vertical position is plotted against inclination. Though the scatter is considerable, the two maxima are clearly distinguishable. It was not possible on this graph, to detect systematic variation with heat flux and the tests with different heat fluxes are thus not distinguished.

For angles of inclination sufficiently far from the vertical in either direction, visual observation indicated that the drops grew to a larger size before sliding and the sliding drops moved more slowly with increasing departure from the vertical. Figure 6 shows photographs of the condensing surface for different angles of inclination and for two heat fluxes. The limits of inclination for which photographs could be obtained were set by flooding of the plate and window.

5. DISCUSSION

Earlier investigations of the effect of surface inclination on heat transfer during dropwise condensation have reported that the heat-transfer coefficient was essentially constant [1] or varied only very slightly [12, 13] with inclination, for angles less than 60° to the vertical in either direction. The present results are in broad agreement with this (see Fig. 5).

The above-mentioned previous investigations do not, however, report the double-maximum behaviour found in the present work. It is to be noted that this arises from relatively small changes in the steam-to-surface temperature difference, ΔT, with variation in inclination, θ. For instance, the following data, in the range of interest, relates to the runs shown in Fig. 4.

θ	90°	110°	130°	140°	150°
ΔT/degC	3·3	3·4	3·8	3·7	3·7
ΔT/degC	5·5	6·0	5·6	5·7	5·9

It is possible that the precision of the earlier measurements may not have been adequate to detect these minor variations.

In the case of Hampson and Ozisik [12, 13], measurements were made over a range of heat flux for each of five different angles. From these observations it would not have been possible to obtain detailed information on the behaviour of the steam-side coefficient with inclination for a given heat flux. Wenzel [1] reports results for one heat flux only (about 0·2 MW/m²). It appears that the intervals of inclination used in this work were too large to detect the "double-maximum" effect. Moreover, the present

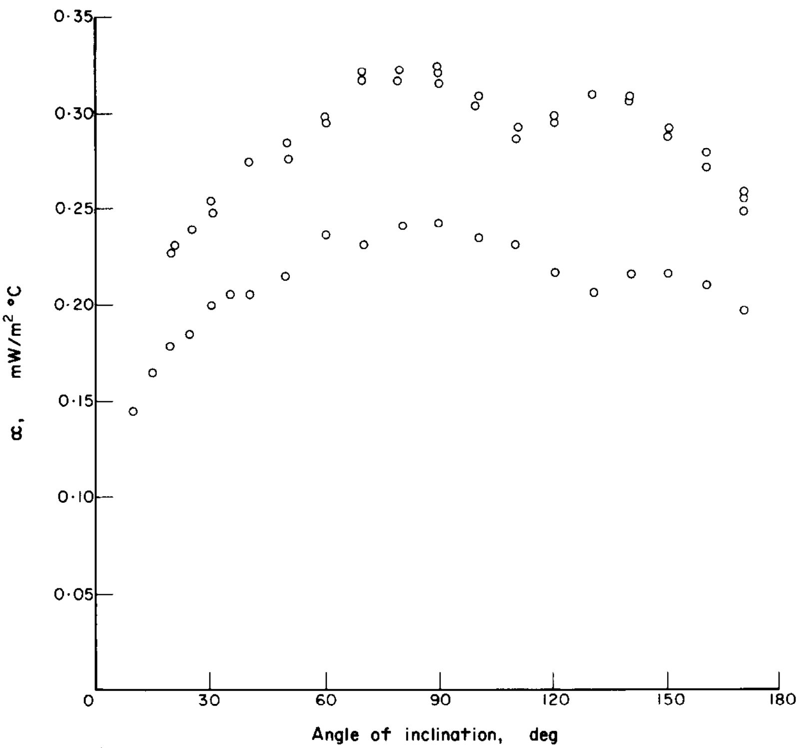

FIG. 4. Variation of steam-side heat-transfer coefficient with surface inclination.

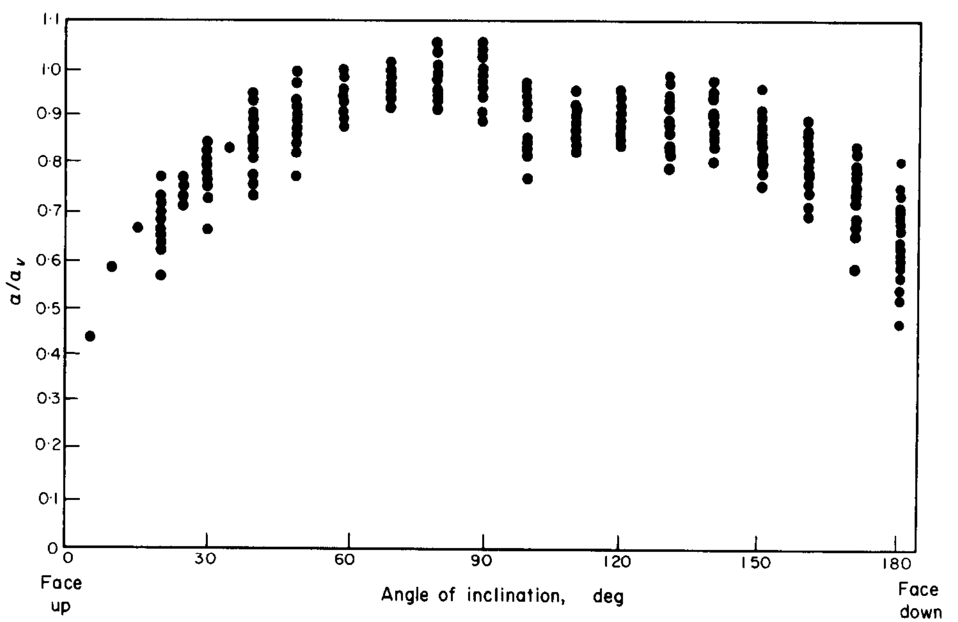

FIG. 5. Variation of the steam-side coefficient, relative to the value for a vertical surface at the same heat flux, with surface inclination.

work suggests this effect would have been slight at the relatively low heat flux.

The somewhat surprising aspect of the present and former observations is the relatively small variation of the heat-transfer coefficient with surface inclination, despite the strong dependence, on inclination, of the size to which drops grow before sliding and the sliding speed. For instance, from Fig. 5 it may be seen that for a horizontal downward facing surface, when the sweeping effect (of falling drops) is zero and the base diameter of a drop of maximum size is several times larger than for a vertical surface, the heat-transfer coefficient was found, on the average, to be as much as about 0·7 times the value for the vertical position. Similarly, for upward-facing, near-horizontal positions, where the size of the largest drops is much greater than for the vertical position (see Fig. 6) and the sliding speed much lower, the heat-transfer coefficient is not less than about half the value for the vertical surface.

The above facts indicate that the average drop size distribution generated by growth, coalescence and sweeping mechanisms, together with the growth rate–size relation for individual drops, is such that the average heat-transfer for the whole plate is only weakly dependent on maximum drop size for the range of the latter occurring in the present tests. Moreover, since the heat transfer through the larger drops is almost certainly negligible in comparison with the high heat transfer rate for the whole surface, it would appear that the mechanism of coalescence is such that the area available, on the average, to the smaller, more effective drops, does not depend strongly on the maximum size which drops may attain before being removed either by falling or by being swept by other drops. A similar deduction may be made from the observation [5] that the local heat-transfer coefficient did not depend on the vertical location of the measuring point in the range 1 in. to 4 in. from the top of the measuring surface, whereas the frequency with which a region is swept is height dependent [22] in this range.

Though the "double-maximum" effect may be of little practical importance, since the changes in the heat-transfer coefficient are small, it is of interest to note that this would suggest that there are two surface inclinations (roughly 90° and 140°) for which the average drop size distribution is somewhat more favorable to heat transfer than neighboring angles. The present results could thus assist in theoretical investigations of the mechanism by which the average distribution is achieved. The immediate implication would appear to be that the maximum drop size and/or the sliding speed have minimum and maximum values respectively, for the angles at which the heat-transfer coefficient has maxima.

From the practical view-point, we may infer from Fig. 5 that the average steam-side coefficient for a horizontal tube might be about 0·8 of the value found for the vertical surface. Thus, when other resistances (wall and cooling side) are taken into account, the horizontal tube may be essentially as effective as a vertical plane surface.

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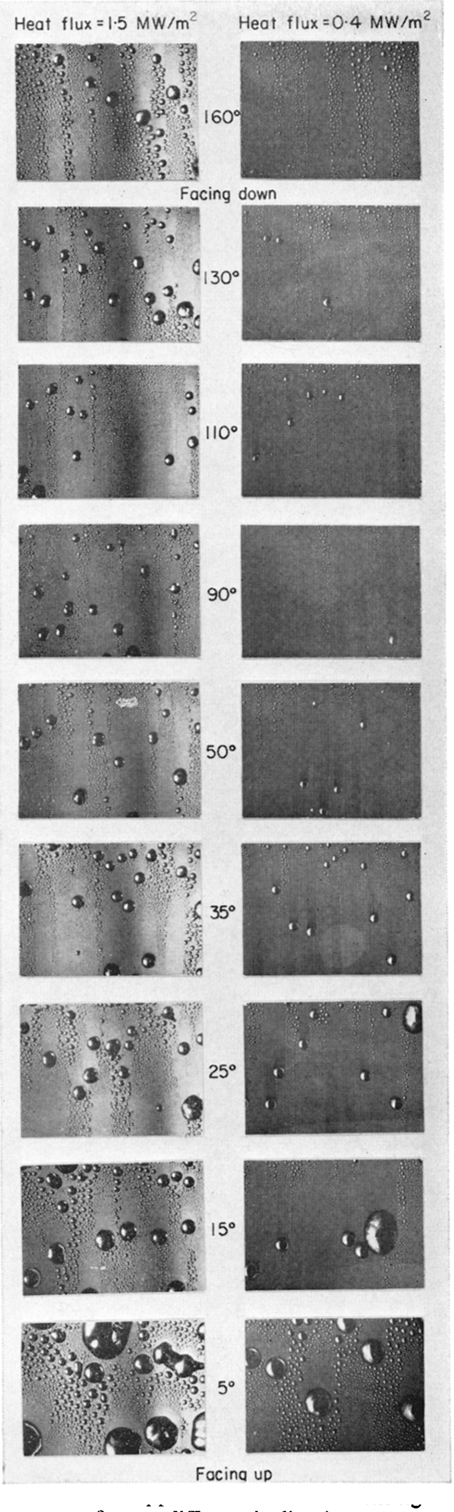

surface at different inclinations

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HEAT TRANSFER AND NATURAL CONVECTION PATTERNS ON A HORIZONTAL CIRCULAR PLATE

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NOMENCLATURE

A, area of heating surface;
Gr, Grashof number;
h, heat transfer coefficient;
Nu, Nusselt number;
p_{pc}, pseudo-critical pressure;
Pr, Prandtl number;
Ra, Rayleigh number;
t, temperature;
ϕ, heat flow rate.

Subscripts
b, bulk fluid;
c, cooling plate;
h, heating plate.

INTRODUCTION

HUSAR and Sparrow [1] reported on flow patterns adjacent to various planforms. Their experiments were performed in water, and flow patterns are shown for a Rayleigh-number range from $2 \cdot 10^6$ to $5{\cdot}5 \cdot 10^8$. In this report flow patterns can be presented for Rayleigh-numbers from 10^9 to 10^{13}. In order to obtain these high Rayleigh-numbers, the experiments were carried out with carbon-dioxide in the supercritical region close to the critical point (31·04°C; 73·84 bar). Three different pressures were applied: 75·84 bar, 89·63 bar and 103·4 bar. Heat transfer coefficients were determined and comparison with Nusselt-type correlations was attempted.

EXPERIMENTAL SET-UP

The experiments were performed in a pressure vessel which is described in [2]. The test arrangement after adjusting this vessel consisted of a circular borosilicate plate, the heating plate, 3·2 mm thick and 50 mm dia. with an electrically conducting surface. The non-conducting side was bonded to a 6·3 mm thick pyrex-glass disc of the same diameter, the so-called measuring-plate, with a diametric groove 0·8 mm deep on each side. Into these grooves thermocouples were cemented with their junctions in the center of the disc. In order to reduce downward heat-losses, another pyrex disc was glued to the measuring plate.

The resulting glass cylinder was set on the lower glass window of the pressure vessel, which had been turned by 90° compared to its former use [2]. It was kept in place by a micarta tube around it and the window, thus also reducing heat losses to the sides. The tube ended 3 mm below the heating surface, so that there were no constraining walls around it. Cooling was provided from tap water running through copper coils wound around the vessel.

Besides the two thermocouples in the glass cylinder, which indicated the temperature-distribution in the center of the disc, another two thermocouples in the vessel gave the bulk temperature.

The pressure vessel was mounted over an optical bench. The light coming from a carbon arc lamp, after passing through a lens system, was directed through the vessel in such a way that the Schlieren-images on a screen gave the view normal to the heating plate. For better contrast a color Schlieren arrangement was used.

RESULTS

A local heat transfer coefficient was calculated from

$$h = \phi/A(t_h - t_b)$$

with the heat flow rate ϕ, the area of the disc A and the temperature difference between the center of the heating surface and the bulk fluid $t_h - t_b$. Figure 1 gives a plot of this heat transfer coefficient vs. the surface temperature at various pressures. As a second parameter, the bulk temperature t_B has to be given for each curve.

Comparison was made between experimental data and Nusselt-type correlations of the form

$$Nu = \text{const.}\ (Gr \cdot Pr)^{\frac{1}{3}}.$$

There was no agreement among the data for the three different pressures. It may be assumed that an extra term in the correlation will consider the pecularities in the vicinity of the critical point [3]. For the lack of sufficient data, especially at various bulk temperatures such a correlation was not performed here.

The flow patterns show distinct differences for various

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