Int. J. Heat Mass Transfer. Vol 8, pp. 1117–1133. Pergamon Press 1965. Printed in Great Britain

AN EXPERIMENTAL STUDY OF HEAT TRANSFER BY DROPWISE CONDENSATION

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(*Received* 12 *February* 1965 *and in revised form* 22 *March* 1965)

Abstract—Measurements of heat flux and steam-side temperature difference during dropwise condensation on plane vertical surfaces are reported. These results are considered to have enhanced precision and, in particular their relation to earlier work supports the view that the effects of "non-condensables" have been avoided. Thermocouples, accurately located and spaced through copper plates served to measure the "mean" surface temperature at a known point on the condensing surface (by extrapolation) and the heat flux (from the temperature gradient). Measurements were made at depths of 1 in (25·4 mm), 1·2 in (28·4 mm) and 4 in (101·6 mm) from the top of the condensing surface. The heat flux used ranged from 0·3 to 1·8 MW/m² (i.e. 100000 to 570000 Btu/ft² h or 260000 to 1550000 kcal/m² h). The pressure was approximately 1·04 bar. Four different promoters were used.

The results obtained were very consistent and were reproducible on different days. The steam-side heat-transfer coefficient was found to increase with heat flux over the above range, the maximum coefficient being about 0·3 MW/m² degC (i.e. 53000 Btu/ft² h degF or 260000 kcal/m² h degC). No evidence of dependence on plate height was found. Differences between promoters were clearly established.

NOMENCLATURE

$\dot{Q}''$, heat flux;
$\Delta\theta$, steam-to-surface temperature difference.

Conversion factors

$$\frac{\mathrm{MW}}{\mathrm{m}^2} \simeq 0{\cdot}317 \times 10^6 \frac{\mathrm{Btu}}{\mathrm{ft}^2\,\mathrm{h}} \simeq 0{\cdot}860 \times 10^6 \frac{\mathrm{kcal}}{\mathrm{m}^2\,\mathrm{h}}$$

1. INTRODUCTION

CONSIDERABLE interest in dropwise condensation has been aroused since the discovery by Schmidt *et al.* [1] of this second "ideal" mode of condensation. The report by the above authors in 1930, that the heat-transfer coefficient associated with the dropwise mode was substantially higher than that found in the presence of filmwise condensation, and the potential industrial significance of this, have stimulated this interest.

The commonly seen "mixed" condensation (smudged drops, streaks and patches of continuous film) is so irregular as to defy analysis. On the other hand pure dropwise and pure filmwise condensation each provide an ideal that is so regular as to invite analysis. That there is a well-established theory, due to Nusselt, for the filmwise ideal and no good theory for the dropwise ideal has provided a further stimulus.

In 1934 and 1935 Nagle *et al.* [2, 3, 4] established that clean steam condensing on a chemically clean surface always forms a continuous film and that the presence of substances which render the surface, to some extent, non-wettable is essential for the formation of drops. Much research relating to such substances, their structure, their life (time interval during which they remain effective) and the manner of their breakdown has since been undertaken with a view to producing long-lived or permanent dropwise condensation under industrial conditions. Promoters, as these substances are now called, having seemingly indefinitely long lives on copper or copper-containing surfaces under clean steam conditions, have been synthesized [5, 6]. These promoters have not proved to be long lived under industrial conditions, failure being due either to solid deposits on the promoter, or promoter breakdown caused by impurities in the steam or erosion by water droplets. Such synthetic promoters are, however, entirely satisfactory for experiments under laboratory conditions.

There have been reports of succesful industrial applications of dropwise condensation, using various promoters, by intermittent injection into the steam, notably in connection with evaporators [7, 8] and paper drying [9, 10].

A most important line of research on dropwise condensation is the study of the condensation mechanism with a view to producing a theory of heat transfer. It is with experiments designed to facilitate this that the present paper is concerned.

There have been many attempts to measure the steam-side heat-transfer coefficient during dropwise condensation and, in particular, to determine its dependence on heat flux. The most striking feature of these is the wide diversity between the results of different workers and the considerable scatter in most individual investigations. For instance, it has been reported that the steam-side heat-transfer coefficient:

(1) is substantially independent of heat flux [1, 3, 11]
(2) increases with heat flux [12, 13]
(3) decreases with heat flux [14]
(4) increases at first and subsequently decreases [15, 16, 17].

Furthermore, the reported coefficients vary over the considerable range of approximately 0·02–0·4 MW/m^2 degC (i.e. 15–340 $Mcal/m^2$ h degC; 3000–70 000 Btu/ft^2 h degF). Experiment has thus not only failed hitherto to provide a constructive guide to theory, but has given results of so wide a variance as to be of very limited use even as a check.

The present experimental work was undertaken initially in order to settle discrepancies between earlier results, to attain a precision superior to that of the best of them, and thereby to provide sound data for use in theoretical studies of the mechanism. A brief preliminary report of the main results has been presented by Le Fevre and Rose [18].

2. THE PRESENT WORK

At the outset, it was planned to enhance accuracy by developing an improved technique for accurately measuring the temperature of the condensing surface. It was then thought that this constituted the chief source of error in the work of previous investigators, and was thus responsible for the wide diversity of published results. It was decided that measurement of the temperature on planes spaced through the test plate, and extrapolation to the surface, offered the best prospect of success. The added attraction inherent in this method, was that the heat-flux could also be determined from the temperature gradient. In order to measure "point" temperatures through the plate, the use of thermocouples was clearly indicated. To determine precisely the positions to which the indicated temperatures related, it was necessary that the thermocouples should be as small as possible. Furthermore, to prevent significant conduction along the thermocouple leads, these leads must run in isothermal surfaces, especially in the vicinity of the junction. With these points in mind and also in order accurately to locate the thermocouples, a new technique was adopted.

The test plate was made in halves so that the mating faces were in a vertical plane normal to the steam and coolant faces. Since the plane of contact of the two halves is normal to the isothermal surfaces, imperfections of contact offer no resistance to heat transfer and the steam-side edge of the mating surfaces, being in a vertical plane, should not perturb the descent of the condensate. This is the only geometry satisfying these conditions.

In the mating face of one half, small square grooves were accurately machined parallel to the steam and coolant faces, and fine butt-welded copper–constantan thermocouples were inserted. With this arrangement (see Fig. 1), the thermocouples may be precisely located.

The alternative method of inserting the thermocouples, by drilling holes in the edge of the plate, was rejected since it was considered that adequately small holes could not be drilled to sufficient depths.

It was at first hoped that the joint at the plane of contact might be sealed by carefully finishing the mating surfaces by "lapping". This was possible at room temperature, but under condensing conditions, the differential expansion,

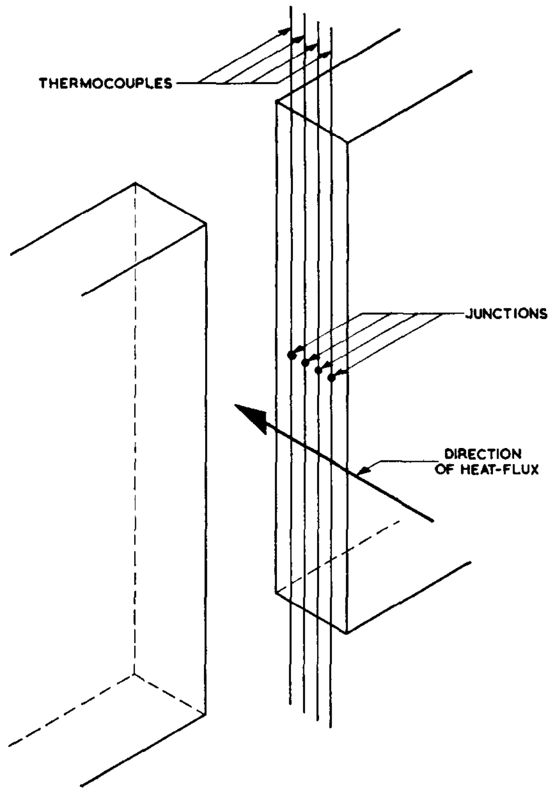

FIG. 1. Location of thermocouples in test plate.

caused by the temperature gradient through the plate, made small leaks unavoidable. To prevent this a thin Neoprene gasket was introduced between the mating faces.

It is apparent that the high thermal resistance of the gasket will cause its steam and coolant edges to attain more nearly the steam and coolant temperatures than the surfaces of the copper plate. Furthermore, it might be thought that this would cause local temperature perturbations in the metal plate adjacent to the gasket, where the thermocouples are situated. This would cause the thermocouples nearer the steam-side to record higher temperatures than those at the corresponding positions in an unperturbed plate, and those near the coolant side to record lower ones. Le Fevre, however, has shown (in a private communication quoted in reference 19) that these effects are unquestionably negligible in the present case. Under the pessimistic assumptions that the steam edge of the gasket had the steam temperature and the coolant edge, the coolant temperature, he showed that if the thermocouples were situated precisely at the edge of even an infinitely thick gasket, the errors in the readings (the thermocouples being situated as they are in the present work) would be negligible. Moreover, the temperature variation along the centre plane of the gasket used, is shown to deviate negligibly from that in the unperturbed plate, at depths corresponding to the thermocouple positions.

Excellent linear temperature gradients were observed in all cases. As a check, the coolant mass flow rate and temperature rise were also measured and a second estimate of the heat flux obtained. Despite the fact that no elaborate precautions were taken regarding mixing, isothermal immersion of thermocouple leads in the coolant, conduction through the brass cooling box and correction to the measured mass flow rate for variation in the density of water and the dimensions of the tank, agreement to within two per cent was obtained with the heat flux more precisely determined from the temperature gradient in the test plate. *The results reported are in all cases based on the latter estimate.* A second plate was later used to investigate the effect of plate height and in this case no provision for coolant measurements was made.

After having established a satisfactory technique for measuring the surface temperature, the first problem encountered was that of the serious temperature fluctuations, which had also been reported by various previous workers [11, 15, 20]. It was considered that fluctuations of the magnitude observed might have been caused, despite precautions to achieve and maintain "gas-free" steam, by an unsteady layer of "non-condensables". This was confirmed by a series of experiments and, as a result of these, a special venting technique was devised to obviate the effects of the "non-condensables". Using this technique, steady readings were observed from all thermocouples, including those nearest to the steam side. Not only were the fluctuations removed but the new steady temperatures were somewhat higher than the old peak temperatures.

It was found impossible, even in several hours, entirely to remove "non-condensable" gases by merely boiling while venting in the ordinary way. The fact that, in the absence of significant steam velocity, even minute traces cause significant lowering and drastic fluctuations of the surface temperature, indicates that the presence of such gases has very probably been the major cause of the wide diversity of published results.

After obtaining steady and reproducible as well as accurately known surface temperatures, the investigation was extended to cover the variation of the steam-to-surface temperature drop over a wide range of heat flux. It was found that the steam-side heat-transfer coefficient *increases* with heat flux over the present range, which extends beyond the highest heat fluxes used in earlier work. The effect of the vertical position of the measuring point was also investigated and, over the limited range used, found to be unimportant. Four promoters were used and it was possible to determine systematic differences between their performances. The results are markedly freer from scatter than earlier reported work and are believed to be freer from systematic error.

The tests were restricted to vertical condensing surfaces and a steam pressure slightly above atmospheric. In order to achieve and maintain clean conditions the apparatus was constructed, so far as possible, in glass.

3. APPARATUS

3.1 *General*

Referring to Figs. 2 and 3, steam was generated

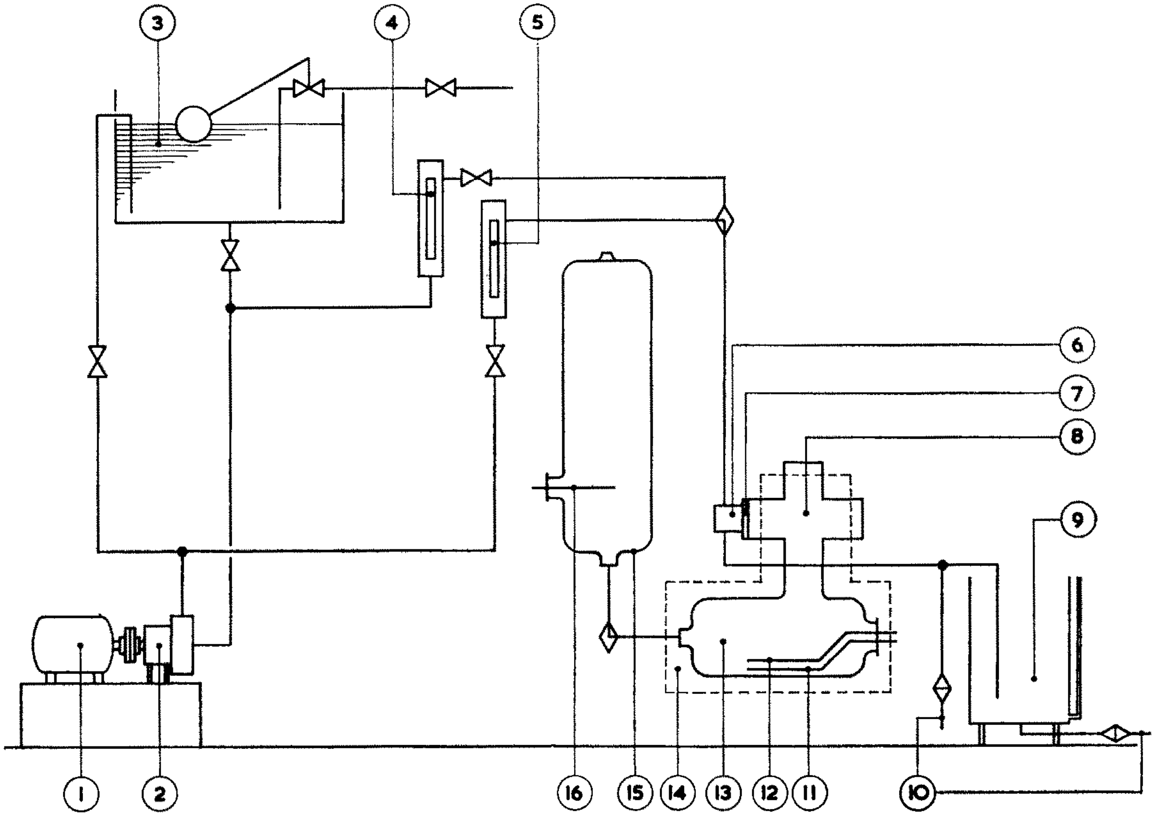

FIG. 2. Layout of apparatus.

(1) Electric motor (10 hp, 3 phase, 2900 rev/min); (2) Centrifugal pump (600 lb/min at 2900 rev/min, against a total head of 160 ft); (3) Supply tank (100 gall); (4) Flowmeter (140 to 1500 lb/h); (5) Flowmeter (15 to 150 lb/min); (6) Cooling box; (7) Test plate; (8) Steam chamber; (9) Measuring tank (100 gall); (10) Drain; (11) Heater (5 kW); (12) Heater (2 kW); (13) Boiler (35 l); (14) Lagging; (15) Supply vessel (50 l); (16) Heater (2 kW).

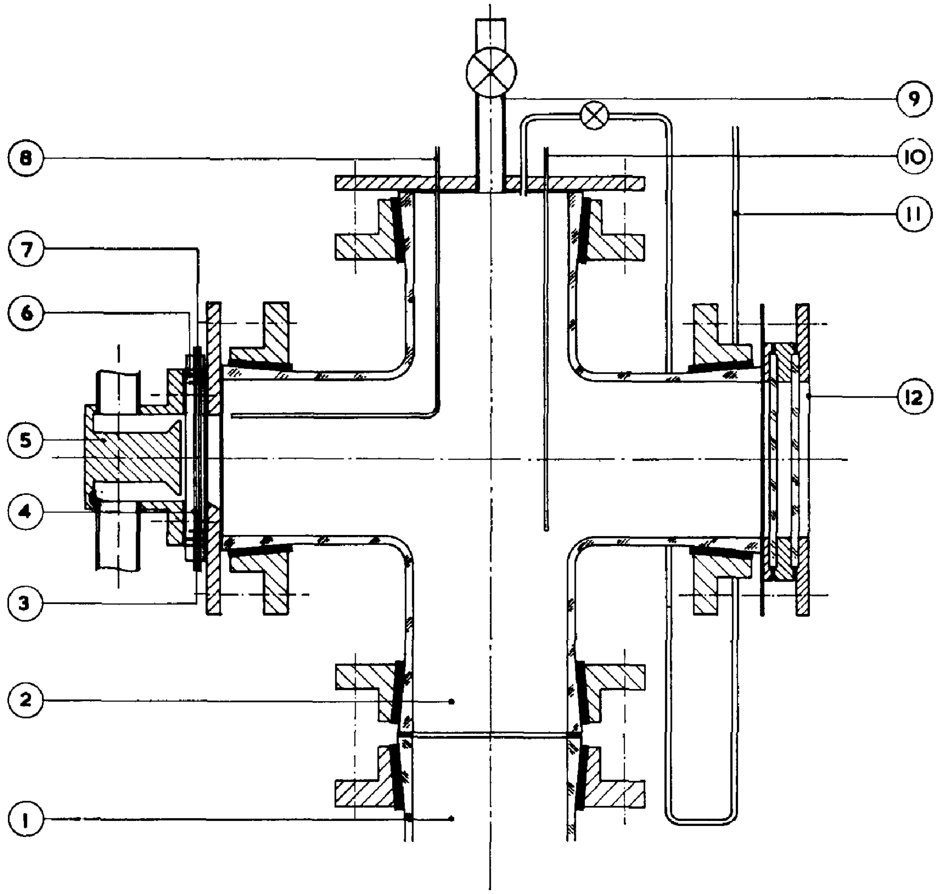

FIG. 3. Steam chamber, test plate and cooling box.
(1) Boiler; (2) Steam chamber; (3) Thermocouples; (4) Test plate (plate 1); (5) Cooling box; (6) Insulating slots; (7) Asbestos lagging; (8) Thermocouple tube; (9) Main vent; (10) Thermocouple tube; (11) Manometer; (12) Double-walled electrically heated window.

from distilled water in the glass boiler and passed into the cruciform glass steam chamber. One of the horizontal limbs of the steam chamber was closed by the test plate and the other by an electrically heated double-glazed window.

The test plate was cooled by mains water flowing vertically downwards. The coolant could either be pumped or allowed to flow by gravity feed from the supply tank. Coolant velocities up to about 30 m/s (100 ft/s) could be obtained. The coolant could be collected in a measuring tank fitted with a transparent level-indicating tube, calibrated using measured masses of water.

3.2 *The test plates*

Two plates of different dimensions were used, each having its own cooling box. The dimensions of the condensing surfaces are given below.

	Horizontal	Vertical
Plate 1	2·750 in	2·438 in
Plate 2	0·875 in	5·000 in

Each plate was of copper, 0·5 in thick, and constructed in halves, the mating faces being in a vertical plane perpendicular to the condensing surface. Four grooves, 0·01 in square, were machined in one mating face of each plate, running parallel to the condensing surface. The distances of the centre lines of the grooves from the condensing surface were:

Plate 1	Plate 2
0·075 in	0·0625 in
0·150 in	0·1250 in
0·225 in	0·1875 in
0·300 in	0·2500 in

A 40 s.w.g. (approx. 0·12 mm) butt-welded copper–constantan thermocouple was inserted in each groove so that the junctions were all in the same horizontal plane at the required vertical position. The plates were assembled with a Neoprene gasket, of thickness about 0·8 mm, between the mating faces. The enlarged sectional plan (Fig. 4) of part of an assembled plate shows the gasket and the method of location and insulation of the thermocouples.

To minimize edgewise heat flux, 0·03 inch slots were machined near the edges in the front and rear faces of the plate. Asbestos strips 0·375 in thick were also screwed to the outer edges of the plates.

The assembly shown in Fig. 3 is that for plate 1. To accommodate the taller plate 2 an extension to the steam chamber was required.

The plate thermocouples were at first butt-welded using a special aligning jig as described by Hickson [21]. It was subsequently found that junctions could more easily be made by hand in a small gas or oil flame.*

Full details of the apparatus are given in [19].

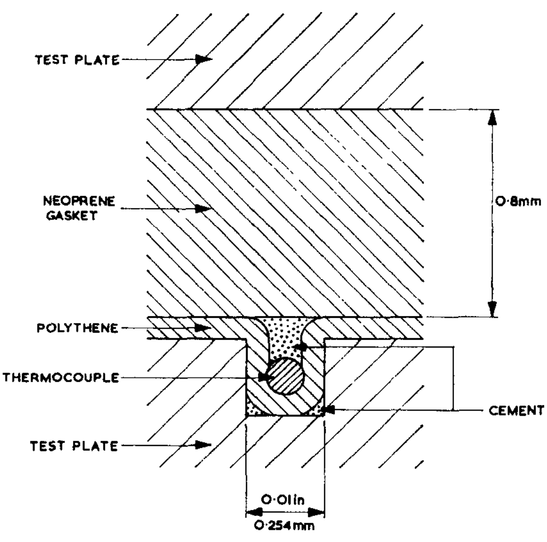

FIG. 4. Scrap sectional plan of test plate showing a thermocouple.

3.3 *Venting arrangements*

In addition to the main vent (see Fig. 3) two other arrangements were used. These consisted of two tubes, each fitted with a valve, which passed into the steam chamber and were situated so that their open ends were close to the condensing surface as shown in Fig. 5. These "close vents" are subsequently referred to as vent 1 (nearer the surface) and vent 2.

4. PREPARATION OF THE CONDENSING SURFACE

The promoters used were:

No. 1: Dioctadecyl disulphide $[C_{18}H_{37}SSC_{18}H_{37}]$†

No. 2: "No. 1 Amine" [chiefly octadecylamine $[C_{18}H_{37}NH_2]$‡

No. 3: Di-S-octadecyl 00-1, 10 decanedixanthate $[C_{18}H_{37}SSCO(CH_2)_{10}OCSSC_{18}H_{37}]$†

No. 4: Dodecanetris (ethanethio) silane $[C_{12}H_{25}Si(SC_2H_5)_3]$†

Hereinafter these promoters will be referred to by the corresponding numbers given above.

Promoters 1, 2 and 3 were powders, 1 and 3 being white and crystalline while 2 was light brown. Promoter 4 was a colourless viscous liquid with a strong nauseating odour. Promoters 1, 3 and 4 were used as a 1 % solution in carbon tetrachloride, and promoter 2 as a 1 % dispersion in hot water.

Preparatory to promoting, the condensing surface of the already assembled plate was

* The authors thank Mr. R. Morris of the National Engineering Laboratory, U.K. for demonstrating this technique.

† Described in [5].

‡ "No. 1 Amine" is a commercial product of Houseman Thomson and Co. Ltd., and is used as a corrosion inhibitor.

All four promoters were supplied by the National Engineering Laboratory, U.K.

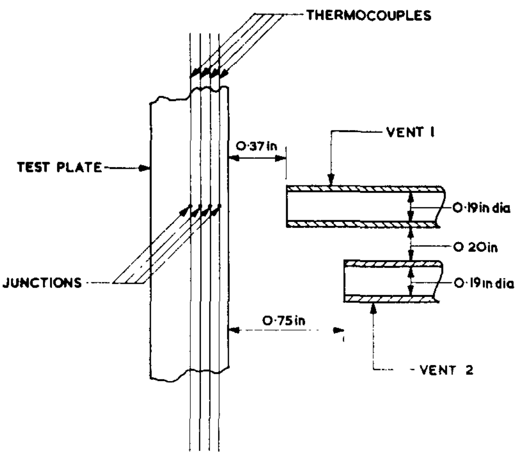

FIG. 5. Arrangement of close vents.

cleaned and given a "mirror" finish with metal polish. It was then more thoroughly cleaned by rubbing vigorously with a succession of clean cloths soaked with acetone, until a piece of cloth remained perfectly clean after being rubbed over the whole surface. The surface was then covered by carbon tetrachloride for a period of 20 min followed by promoter solution for 30 min. It was then quickly mounted on the steam chamber. The coolant was turned on immediately. The mounting procedure took about 3 min.

5. MEASUREMENTS

The surface temperature was found by linear extrapolation from the observed temperatures in the plate, and the heat flux from the temperature gradient in conjunction with the thermal conductivity of the plate. Surface temperature and temperature gradient were determined by linear regression of the temperatures against the distances from the face. However, the linearity of the temperature distributions was such that virtually the same results would have been obtained had the distances been regressed against the temperatures.

The thermal conductivity at 80°C of a specimen of the same material as plate 1 was measured by the National Physical Laboratory (U.K.). The electrical conductivity at room temperature was measured by the authors using a Kelvin Double Bridge, and the thermal conductivity at 80°C calculated therefrom by the method of Smith and Palmer [22]. The two values agreed to better than 0·8 per cent. The electrical method alone was used for plate 2. Corrections for variation in thermal conductivity with temperature were not made since these were smaller than the errors arising from the uncertainties in the adopted values. The excellent observed linear temperature distributions in the plates helped to confirm this.

The temperature of the steam in the steam chamber was observed by two 28 s.w.g. (about 0·38 mm) cotton-covered copper–constantan thermocouples in glass tubes located as shown in Fig. 3. The indications of these never differed by more than 2 μV. The leads from the plate and steam thermocouples were taken through a selector switch (covered by a tightly fitting Perspex box to avoid draughts) to the potentiometer, the calibration of which had been newly checked, measuring to 1 μV. A reversing switch was also incorporated to minimize errors due to spurious e.m.f. The forward and reverse readings never differed by more than 2 μV.

The copper and constantan leads from the measuring junctions were soldered to thicker enamelled copper wires, which led to the selector switch. The copper–copper and copper–constantan soldered junctions so formed, were placed in closely fitting thin-walled glass tubes. The glass tubes (one for each thermocouple) were immersed to a depth of about 11 inches in finely ground, closely packed, melting, distilled-water ice, contained in a large vacuum-walled vessel. These arrangements provided the reference junctions and at the same time obviated any thermal e.m.f. due to the thicker wires. The arrangement is shown in Fig. 6. Care was taken to ensure that the cold junction tubes were not bunched together.

The thermocouples were calibrated against a thermocouple calibrated by the National Physical Laboratory (U.K.). For this purpose, a vacuum-walled vessel fitted with an immersion heater, supplied through a variable transformer, and stirrer was used. The junctions of the standard and of that to be calibrated, were soldered together and placed in a tightly fitting thin-walled glass tube which was immersed to a depth of about 10 inches, in water for calibration up to

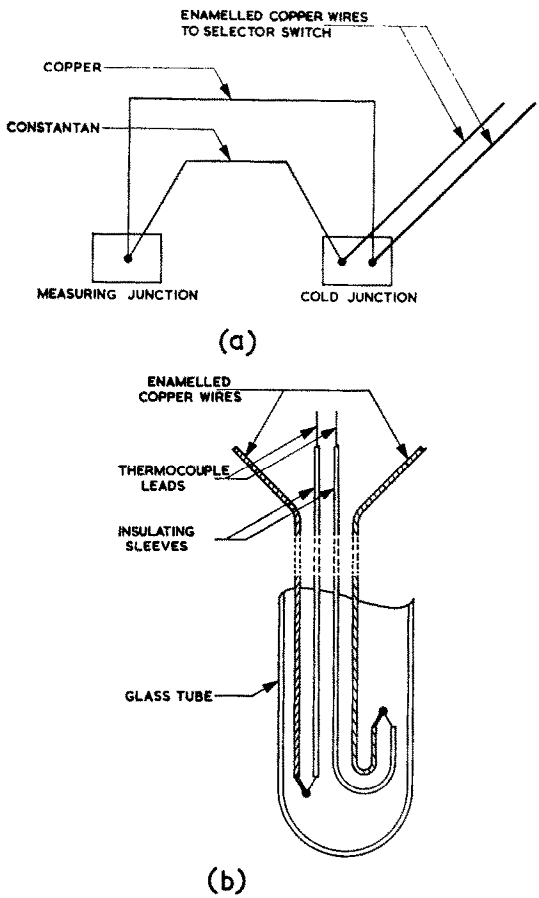

FIG. 6. Thermocouple arrangement.
(a) Connections.
(b) Cold junction tube.

100°C, and in a light machine oil for calibration up to 105°C.

Calibration points were obtained by using a heater current just sufficient to keep the temperature constant at the desired level. The e.m.f. of the standard was first measured in the forward and reverse directions. The thermocouple being calibrated was then read in both directions. The standard was then re-read. If the first and second readings of the standard agreed to within 1 μV the reading was accepted; otherwise the heater current was adjusted and the procedure repeated until such agreement was obtained. The temperature was then adjusted to the new desired level and the procedure repeated. The calibration carried out in this way was found to be reproducible to within 1 μV on different occasions.

As a check on the calibration, a 40 s.w.g. thermocouple (as used subsequently in the plates) and a 28 s.w.g. thermocouple (as used subsequently in the steam) were inserted in the steam chamber and used to measure the steam temperature when the main vent was open and the manometer registered atmospheric pressure. The temperatures measured by the two thermocouples agreed with each other and with the saturation temperature corresponding to atmospheric pressure, to better than 0·04 degC.

6. PRELIMINARY TESTS REGARDING "NON-CONDENSABLE" GASES

6.1 *Attempts to "degas" by boiling*

The boiler was filled and the valve connecting it to the supply vessel (see Fig. 2) was closed. Above the water level in the boiler, the apparatus contained air. With test plate 1 in position and promoted with promoter 1, the coolant was passed at the highest rate which could be achieved without using the pump. The plate thermocouples indicated steady temperatures slightly above that of the coolant.

The heaters in the boiler were switched on and when boiling commenced, steam and air were removed from the main vent. The venting rate was controlled by means of a valve, so that the pressure in the steam chamber was slightly above atmospheric.

At the onset of boiling the temperatures indicated by the plate thermocouples began to rise but with extremely violent fluctuations. The peak-to-peak variations were estimated at around 20 degC. With continued boiling the temperatures in the plate continued to increase and the magnitude of the fluctuations decreased until, after boiling for about 2 h while maintaining a steam-chamber pressure of about 1·03 atm, the fluctuations had diminished to the sort of magnitude shown in Fig. 7. This graph, which relates to the thermocouple nearest the condensing surface, was sketched while very carefully watching the galvanometer spot with the potentiometer set at a point near balance. It is considered that this graph gives a fairly accurate impression both with regard to the maximum and minimum temperatures and the time scale

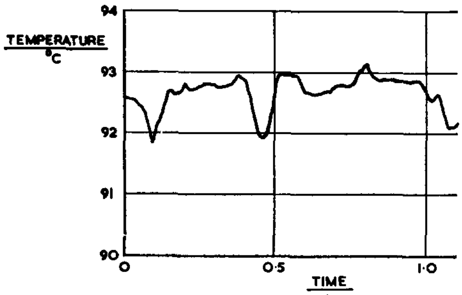

FIG. 7. Indication of plate thermocouple nearest condensing surface after boiling for 2 h. Plate 1; Promoter 1; $\dot{Q}'' \simeq 0{\cdot}9$ MW/m².

of the fluctuating temperature. A further 4 h boiling produced no observable change.

The fact that these fluctuations became less severe, as air was progressively removed by boiling, and yet remained significant after several hours of boiling, suggests that "non-condensable" gases caused the fluctuations, and that a significant concentration of "non-condensables" remained in the chamber even after prolonged boiling,

Before finally abandoning attempts to achieve complete "degassing" by boiling the following experiment was made:

With the promoted plate (promoter 1, plate 1) in position and the main vent open, the valve linking boiler and supply vessel (see Fig. 2) was opened, allowing the boiler and steam chamber to fill with water. When water came out of the main vent it was closed. Then, with the boiler and steam chamber completely filled with water and the supply vessel approximately half-full, the heaters in both boiler and supply vessel were switched fully on. When boiling occurred, the pressure of the steam in the chamber forced water back into the supply vessel, the top of which was open to atmosphere. When the water level fell below the steam chamber, the valve, linking boiler and supply vessel, was closed and the main vent partially opened so as to maintain a steam pressure of about 1·03 atm while venting to atmosphere. This procedure was carried out three times at hourly intervals.

When the test plate was exposed to steam for the third time, the temperature fluctuations were no smaller than had been achieved previously merely by boiling. A further two repetitions of this procedure produced no improvement. It was therefore concluded that the "non-condensable" content of the steam could not be reduced to an insignificant level either by simple boiling or by boiling preceded by flooding.

6.2 *The effect of local venting*

It is apparent that when the steam which is condensed contains "non-condensable" gases, the "gas" concentration in the immediate vicinity of the condensing surface increases to an equilibrium level at which the rate of arrival of gas is equal to the rate of removal by diffusion and natural convection (the "gas-rich" region differing in density from the remoter steam–gas mixture). It was thought that the disturbances, perhaps caused by falling drops, of such a "blanket of non-condensables" might be the cause of the temperature fluctuations in the plate. If this were the case then the prevention of such a build up of "gases" should remove these fluctuations. This hypothesis was tested as follows:

(a) The "close vents" (see section 3.3) were fitted at this stage. With these two vents closed most of the "non-condensables" were removed as before by boiling while venting a considerable proportion of the steam from the main vent. When the mean of the fluctuating temperature, indicated by the thermocouple nearest the condensing surface, had reached its maximum value and the fluctuations had, as described earlier, attained reduced amplitude, the main vent was closed and vent 1 was fully opened, the heater supply being adjusted so as to maintain the same steam pressure (approximately 1·03 atm). The indications of all of the plate thermocouples were then steady. Figure 8 (obtained in the same manner as Fig. 7) compares the indications of the thermocouple nearest the steam surface before and after opening vent 1. Vent 1 was then closed and vent 2 opened. After slightly adjusting the heater to restore the steam pressure, and hence temperature, to its former value, it was found that indications of the plate thermocouples were identical to those found when using vent 1. Both close vents were then opened, and for the same steam temperature, the plate temperatures

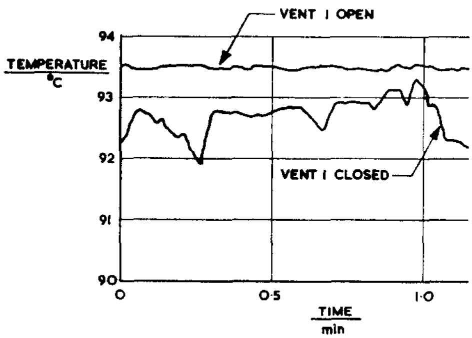

FIG. 8. Indication of plate thermocouple nearest condensing surface showing the effect of close venting. Plate 1; Promoter 1; $\dot{Q}'' \simeq 0{\cdot}9$ MW/m².

were found to be the same as those obtained using either close vent separately. These tests were carried out with a constant coolant-flow rate.

(b) With vent 2 and the main vent closed, a series of tests were carried out at different coolant-flow rates, to determine the minimum venting rates (from vent 1) required to produce the stable condition. A measure of the minimum required venting rate was obtained by counting the number of turns while closing the valve until the onset of fluctuations. It was found that the required venting rate increased with the condensation rate. It was also observed that, provided the steam temperature was held constant (by adjusting the boiler heaters), the plate temperature did not fall until the onset of fluctuation. Repeating the above process using vent 2 and with vent 1 closed, the same was found except that higher minimum venting rates were required throughout and at the highest condensation rates steady temperatures could not be achieved even with vent 2 fully open.

(c) Using a constant coolant flow rate, a quantitative estimate of the effect of venting rate from vent 1 (vent 2 and the main vent being closed) on the steam-to-surface temperature difference was made as follows:

Having first achieved maximum "degassing" by boiling as previously described, and allowed sufficient time for steady conditions to become established, steam and test-plate temperatures were measured for a series of venting rates from fully open to fully closed. For each valve setting the steam temperature was restored to its initial value (corresponding to a pressure of about 1·03 atm) by adjusting the heater power. The heater power was also noted in each case. Since the heat-flux through the plate was measured (from the temperature gradient in the plate) and also the power input to the boiler, the difference between these two quantities for the test with zero venting represents thermal losses from the apparatus. Thus for all venting tests, an estimate of the venting rate could be made by subtracting the power transmitted through the plate plus the power loss from the power to the boiler. The results of this test are shown in Fig. 9.

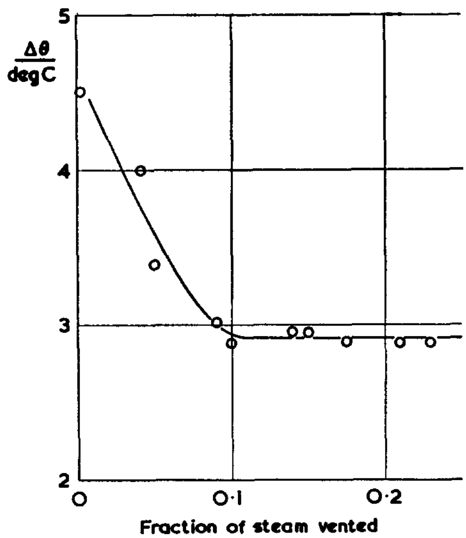

FIG. 9. The effect of close venting rate on steam-to-surface temperature difference. Plate 1; Promoter 1; Vent 1; $\dot{Q}'' \simeq 0{\cdot}9$ MW/m².

It therefore seems apparent that the temperature fluctuations in the test plate are indeed caused by a minute "non-condensable" gas content in the approaching steam which, in the absence of significant non-normal steam velocity or local venting, accumulates at the condensing surface. It has also been seen that neither the venting rate nor the position of the vent has any effect on the steady temperature distribution in the plate provided the venting rate is sufficient. Figure 9 shows that for the particular vent position and heat flux used in this case, consistent and reproducible results will be obtained provided that the venting rate corresponds to the removal of more than one tenth of the approaching steam. If the venting rate is less than this, not only does the steam-to-surface temperature difference increase, but the results are rendered less precise by the onset of fluctuations in the surface temperature. For this reason the curve exhibits scatter for venting rates less than the critical value.

Hence it is to be concluded that close venting provides a method for obtaining reproducible results which are the same as those for "gas-free" steam. Furthermore (as is made particularly clear by the horizontal part of Fig. 9) it has been shown that the moderate steam velocities, caused by close venting, have no effect on the steam-to-surface temperature difference. Indeed in no case did the venting produce any observable change in the appearance of the condensate. Thus the results described can be said to relate to the condensation of "gas-free" steam having no velocity save the normal component necessary to bring the steam to the surface.

6.3. *The effects of "non-condensables" in earlier work*

The tests described above indicate strongly that "non-condensables" are not entirely removed by prolonged boiling. It has been seen also that by preventing the "build-up" in concentration of these gases at the condensing surface, the effect on the surface temperature may be reduced and even eliminated. Thus other tests in which the steam velocity was purposely minimized (in order to avoid effects which it was thought might be due to velocity) will be those most affected by "non-condensables". The extent to which measurements are affected also depends on the geometry of the steam chamber and the fraction of steam condensed. It therefore seems very probable that "non-condensables" have been largely responsible for the wide diversity of published results.

The present apparatus was such that, after minimizing the "non-condensables", their effect, even without close venting, was not too severe since the wide section of the steam chamber assists diffusion of the "gas" away from the condensing surface. Figure 10 compares results for promoter 1 and plate 1, with and without local venting. It is evident that the accuracy and precision of the present measurements are greatly enhanced by local venting. However, it is to be noted that, with the present apparatus, even results obtained without using the close vent represent a marked improvement on most earlier work.

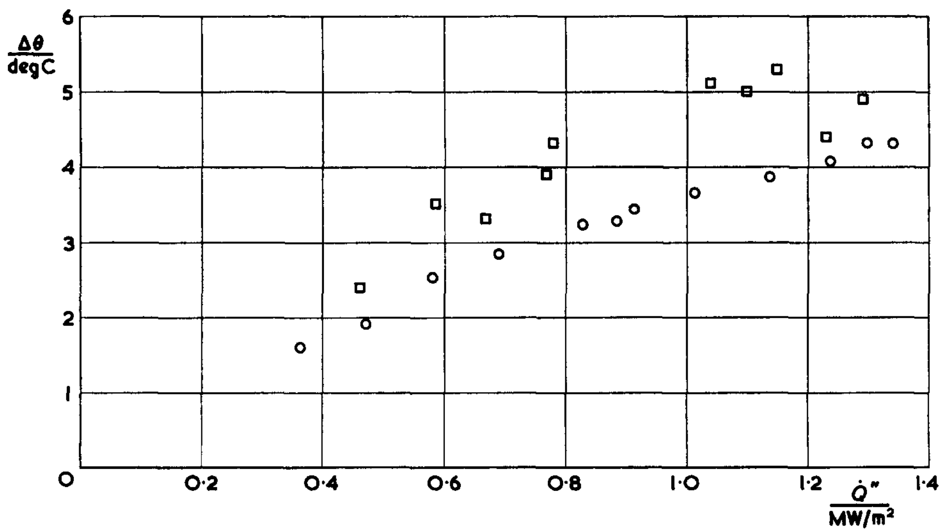

FIG. 10. Comparison of results with and without close venting. Plate 1; Promoter 1; ○ using vent 1, □ no venting.

7. VISUAL OBSERVATIONS

The appearance of a surface promoted with promoter 4 was seen to differ from that of a surface promoted with one of the other promoters. It was noted that in the case of promoter 4 drops grew to a slightly larger size before sliding. Further, it was noticed that during condensation, surfaces promoted with promoters 1, 2 and 3 always retained a bright appearance while in the case of promoter 4 the appearance was always dull. However, it was found that in this last case a single light stroke of the finger, covered by a clean cloth moistened with carbon tetrachloride or acetone, on a plate removed from the steam chamber, revealed a bright polished surface. It would therefore seem that for promoter 4 a layer of more than monomolecular thickness exists during condensation.

A rough estimate of contact angles was obtained using a small tubular condenser. After condensing for about 3 h, the tube was removed and supported horizontally. The drops on the upper edge were photographed after carefully removing those immediately in front and behind with a fine brush. The time between removal of the tube from the steam and photographing was about 2 min. No significant differences between promoters were found, contact angles being estimated at $89^\circ \pm 3^\circ$ in all cases. While it is realized that contact angles so measured are not strictly comparable with those occurring during condensation, when the surface is vertical and in an atmosphere of steam, it was thought that any major differences might be revealed.

It was observed that when the boiler heaters were switched off after a run and the steam chamber allowed to fill with air, a promoted plate would retain the appearance of dropwise condensation for a considerable time provided the coolant flow was continued. This has been observed even after several days during which moisture from the air continued to condense extremely slowly, but in the dropwise mode. When the plate was again subjected to an atmosphere of steam, ideal dropwise condensation was observed. If, however, the coolant was not passed during a non-steaming interval of more than 10–20 h, it was usually found impossible to re-establish ideal dropwise condensation. The breakdown in effectiveness of the surface was usually accompanied by discoloration. This phenomenon has been reported by several earlier workers. Presumably either the plate or the promoter undergoes with substances present in the air, a reaction which is susceptible to inhibition by moisture.

8. HEAT TRANSFER OBSERVATIONS

In all of the tests reported below, vent 1 was operated at a rate sufficient to eliminate temperature fluctuations. In all cases the condensing surface was vertical and a steam pressure of about 1·03 atm was maintained by adjusting the heater power according to the condensation rate.

It was found that in most cases an interval of at least 4 h (after minimizing "non-condensables" by boiling for 2 h) was required for the plate temperatures, and hence the surface temperature, to attain a steady value. The surface temperature increased steadily during this interval (by between 1 and 2 degC at a heat flux of about 0·9 MW/m^2). No further change was detected over the duration of the test. It seems probable that during this preliminary condensing interval "excess" promoter was removed by the condensate. The results presented relate to the steady condition.

The maximum heat flux was limited by the highest obtainable coolant flow rate. A higher maximum heat flux was obtained with the plate 2 assembly. At very low coolant flow rates irregular low-frequency fluctuations of the coolant and plate temperatures occurred. This effect, attributed to natural convection in the coolant, fixed a lower bound for the heat-flux range that could be used.

Runs were made with each promoter in turn:

(a) using plate 1, when the thermocouple junctions lay in a horizontal line 28·4 mm (1·12 in) below the level of the top of the condensing surface.
(b) using plate 2 with the level of the thermocouple junctions 25·4 mm (1 in) below the top of the condensing surface.
(c) using plate 2 with the level of the thermocouple junctions 101·6 mm (4 in) below the top of the condensing surface.

Graphs showing the steam-to-surface temperature difference as a function of the heat flux have been chosen for presenting the results. The traditional plot of heat-transfer coefficient against heat flux has been discarded. In any case, the steam-to-surface temperature difference and the heat flux are the measured quantities. Furthermore, a heat-transfer coefficient vs. heat flux plot can be misleading when the error in the steam-to-surface temperature difference is appreciable, since the heat-transfer coefficient involves the reciprocal of this quantity. When the range of error in the steam-to-surface temperature difference is constant over the heat-flux range, the scatter of the heat-transfer coefficients increases as the heat flux falls. Moreover, a symmetric distribution of temperature-difference errors leads to an asymmetric distribution of heat-transfer coefficients, and the quotient of the mean heat flux by the mean temperature difference fails to coincide either with the unweighted mean of the heat-transfer coefficients or with their median. Figure 11 illustrates this. In (a) the broken lines show the limits of scatter on temperature difference and the full line represents the curve of the mean and median. In (b) the lines from (a) have been transformed to the other basis. It is apparent that from a set of results plotted as in (b) one might easily be misled into believing that the line XX gave the best mean representation of the observations.

In Fig. 12 the results for each promoter are shown separately. It is evident that the scatter on temperature difference is much less than that for earlier experiments. The vertical position of the measuring point, over the range 1–4 in (i.e. about 25 mm–100 mm) from the top of the surface is seen to have no significant effect, i.e. the local heat-transfer coefficient is constant over this range.

For each of the four promoters the results are well represented, over the range considered, by a linear relation. Such a relation was determined in each case by regression of temperature difference against heat flux:

$$\Delta\theta = m\dot{Q}'' + c$$

Promoter No.	$\frac{m}{\text{degC m}^2/\text{MW}}$	$\frac{c}{\text{degC}}$
1	3·0	0·6
2	3·1	0·7
3	3·4	0·8
4	4·7	0·6

The standard deviation of $\Delta\theta$ was found to be just under 0·2 degC.

It is stressed that these linear equations do no more than *represent* the results in the range from 0·4 MW/m² to 1·7 MW/m². The lines are given on Fig. 13 where the results for the different promoters are compared.

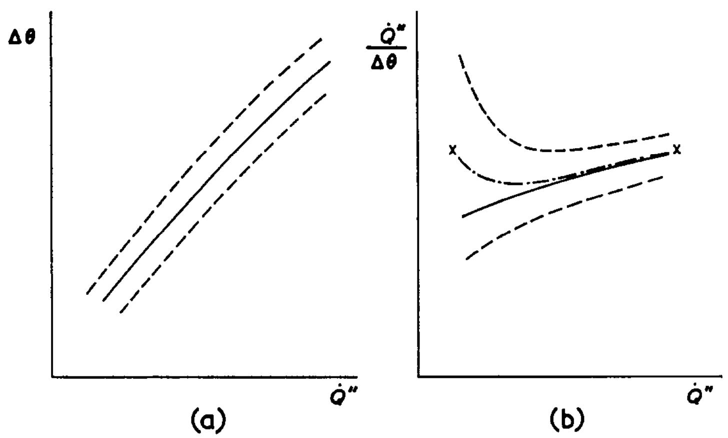

FIG. 11. Distortion of heat-transfer coefficient.
(a) Temperature difference (with uniform error) vs. heat flux.
(b) Corresponding heat-transfer coefficient vs. heat flux, showing magnified error and distorted median.

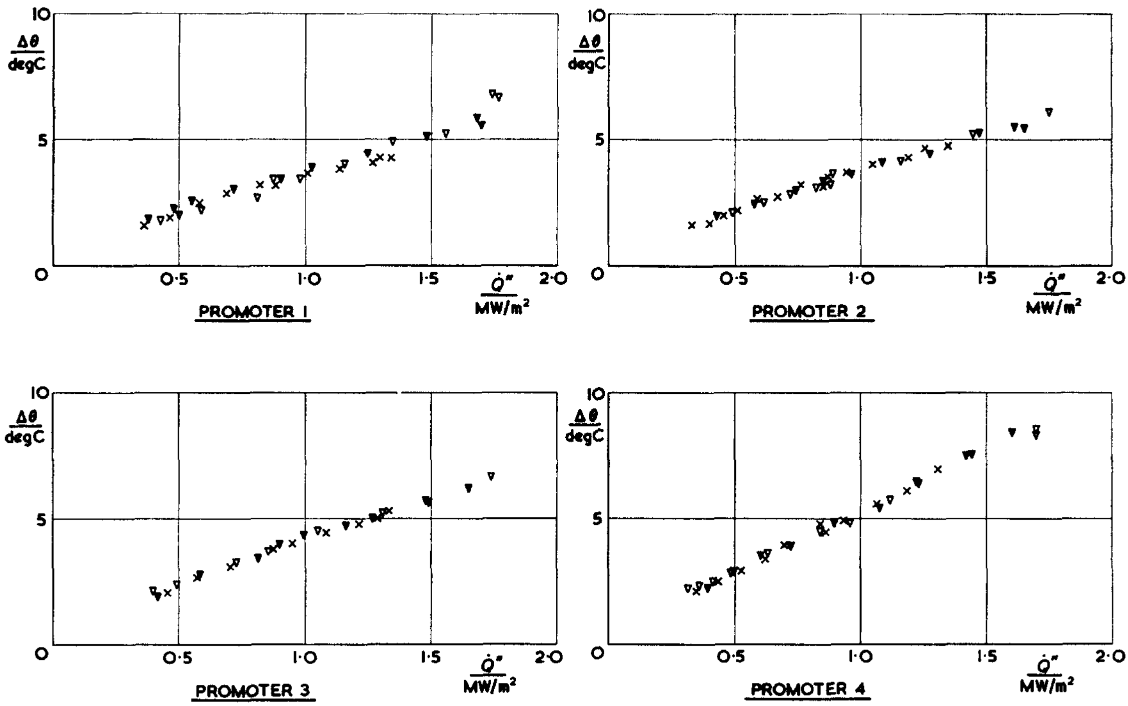

FIG. 12. The effect of the position of the measuring point.
Distance of measuring point from top of surface: × 1·12 in; ▽ 1·00 in; ▼ 4·00 in.

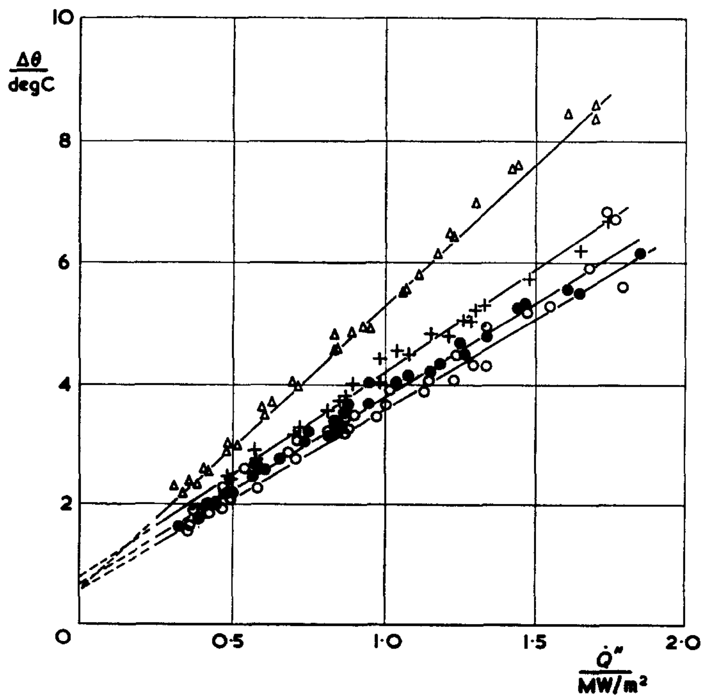

FIG. 13. Comparison of results for the different promoters.
○ Promoter 1;
● Promoter 2;
+ Promoter 3;
△ Promoter 4.

9. DISCUSSION

It is not proposed here to give a detailed survey of earlier results. In most cases the reported steam-side coefficients are lower than the present ones and the scatter of results is greater than in the present work. In Fig. 7 of their recent paper [13] Tanner *et al.* give an indication of the diversity of previously published work. Their graph shows only the smoothed curves and, since the steam-side coefficient is plotted against heat flux, will be subject to distortion as discussed earlier. Of note is the good agreement between the present results, those reported by Wenzel [12] and in particular those of Tanner *et al.* in which the promoter dioctadecyl disulphide (promoter 1 in the present work) was used. Figure 14 shows the above results together with the present ones for promoter 1.

Although both Tanner *et al.* and the present authors have shown that the promoter used influences the results obtained, it is considered that this effect, being relatively small, could not explain the differences between these results and most earlier ones. In view of the present experiments relating to "non-condensables", they would seem to be the major cause of error in earlier work.

That, when the present results are represented by straight lines, the intercepts on the $\Delta\theta$-axis are almost identical (see Fig. 13) suggests that the observed differences between promoters may be described as a simple series resistance having a value dependent on the promoter. This, in conjunction with the observation that in the case of promoter 4 a layer of more than monomolecular thickness was present, suggests that some of the difference between promoter and promoter may be attributable to the resistance offered by promoters layers of different thicknesses and conductivities It is possible to obtain some idea of the required differences in layer thickness by assuming that all promoter layers have a thermal conductivity between that of paraffin wax (0·25 W/m degC) and that of water (0·69 W/m degC). Then, if such layers alone account for all of the difference between the results for promoters 1 and 4, and the heat-transfer were uniform over the surface, a difference in layer thickness of between 0·4 μm and 1·2 μm would explain the observed differences

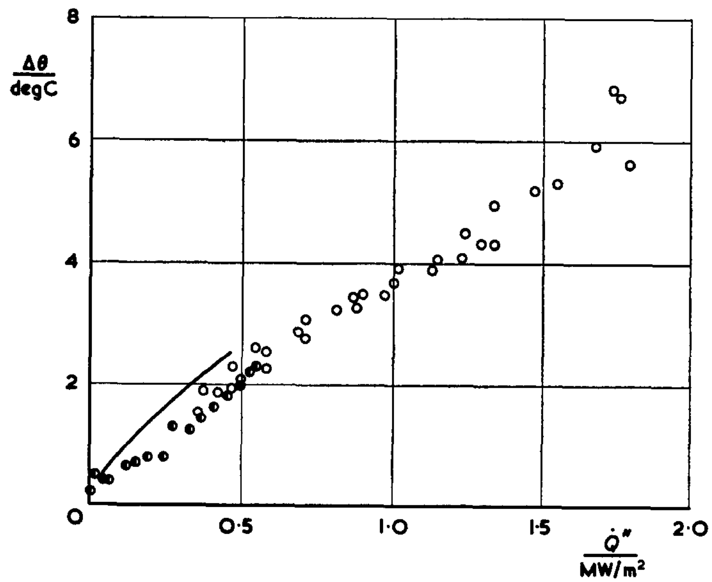

FIG. 14. Comparison of present results for promoter 1 with references 12 and 13.

Author	Promoter
○ Present	Dioctadecyl disulphide
◐ Reference 13	Dioctadecyl disulphide
/ Reference 12	Oleic acid

in results. This crude estimate is of the order of 200–600 times the thickness of a monomolecular layer of these promoters, which is about 0·002 μm.

The fact that the local steam-side coefficient is observed, over the range used, not significantly to depend on height is perhaps a little surprising since the distribution of condensate clearly is height dependent. The frequency with which a region is swept increases and the maximum size of adhering drop decreases with distance down the surface. However this is in agreement with Wenzel's [12] observation that the steam-side coefficient, at a heat flux of 175 Mcal/m^2 h, was independent of surface inclination over a range of about 60° to the vertical in either direction, despite the fact that the maximum size of adhering drops was clearly increased and the transit time of falling drops increased by more than forty per cent.

10. CONCLUSIONS

(1) The present results, see Figs. 12 and 13, together with those of Wenzel [12] and Tanner *et al.* [13], provide the most reliable heat-transfer data for dropwise condensation of gas-free steam at pressures near atmospheric on vertical plane surfaces in the effective absence of lateral steam velocity. The approximate ranges covered are:

	$\dot{Q}''$ (MW/m^2)	Height (mm)	Pressure (bar)
Present	0·3 –1·8	25·4–101·6	1·04
Wenzel	0·07–0·45	70 –280	1·07
Tanner	0·01–0·58	9·2	1·01

The work of other earlier investigators is considered to have been vitiated by the presence of "non-condensable" gases and, in many cases to have suffered from less accurate temperature measurement.

(2) There is no evidence of dependence on plate height for the ranges considered above.

(3) Different promoters give significantly different results.

(4) Variation in the sweeping frequency and maximum size of adhering drops do not, in so far as these vary with height in the ranges considered, effect the steam-side heat-transfer coefficient. Wenzel's tests on tilted surfaces [12] imply that these factors are unimportant over a wider range.

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Résumé—Un compte-rendu de mesures de flux de chaleur et de différences de température du côté de la vapeur pendant la condensation en gouttes sur des surfaces planes verticales est donné. On considère que ces résultats ont une précision améliorée et, en particulier leur relation avec les travaux antérieurs corrobore l'opinion que les effets des gaz non condensables ont été évités. Les thermocouples, situés avec précision et espacés à l'intérieur de plaques de cuivre servaient à mesurer la température de surface "moyenne" en un point connu sur la surface de condensation (par extrapolation) et le flux de chaleur (à partir du gradient de température). On a fait des mesures à des profondeurs de 25,4 mm, de 28,4 mm et de 101,6 mm à partir du sommet de la surface de condensation. Le flux de chaleur employé variait de 0,3 à 1,8 MW/m^2. La pression était approximativement de 1,04 bar. Quatre provocateurs de condensation différents ont été employés.

Les résultats étaient très cohérents et l'on a trouvé qu'ils étaient reproductibles s'ils étaient obtenus à des dates différentes. On a trouvé que le coefficient de transport de chaleur du côté de la vapeur augmente avec le flux de chaleur dans la gamme ci-dessus, le coefficient maximal étant environ égal à 0,3 MW/m^2 degC. On n'a trouvé aucune dépendance en fonction de la hauteur de la plaque. On a clairement établi des différences entre les provocateurs de condensation.

Zusammenfassung—Es wird über Messungen der Wasserstromdichte und der Temperaturdifferenz auf der Dampfseite bei Tropfenkondensation an ebenen, senkrechten Oberflächen berichtet. Diese Ergebnisse haben offensichtlich erhöhte Genauigkeit und insbesondere unterstreicht ihre Beziehung zu früheren Arbeiten den Standpunkt, dass die Einflüsse der nicht kondensierbaren Gase vermieden wurden. Thermoelemente, die genau verlegt und durch Kupferplatten auf gleichen Abstand gebracht werden, dienten zur Messung der "mittleren" Oberflächentemperatur an einer definierten Stelle an der Kondensationsfläche (durch Extrapolation) und zur Bestimmung der Wärmestromdichte (aus dem Temperaturgradienten). Es wurde in Tiefen von 25,4 mm, 28,4 mm und 101,6 mm von der Kondensationsoberfläche aus gemessen. Die dabei aufgetretenen Wärmestromdichten reichten von 30 bis 180 W/cm^2. Der Druck lag angenähert bei 1,04 bar. Es wurden vier verschiedene Promotoren verwendet.

Die erzielten Ergebnisse waren in sich konsistent und liessen sich an verschiedenen Tagen gut reproduzieren. Die dampfseitige Wärmeübergangszahl nahm mit der Wärmestromdichte im oben angeführten Bereich zu; die maximale Wärmeübergangszahl lag bei ungefähr $3 \cdot 10^5$ W/m^2 grd. Ein Nachweis für die Abhängigkeit von der Plattenhöhe wurde nicht gefunden. Die Unterschiede zwischen den Promotoren wurden deutlich festgestellt.

Аннотация—Приводятся результаты измерений теплового потока и температурной разности пар-стенка в процессе капельной конденсации на плоских вертикальных поверхностях.

Предполагается, что эти результаты имеют высокую точность, и сравнение их с более ранней работой подтверждает мнение, что отсутствовало влияние «неконденсируемых» веществ.

Термопары, тщательно заделанные в медные пластины, служили для измерения средней температуры в определенной точке на кондесирующей поверхности (путем экстраполяции) и теплового потока (по температурному градиенту). Измерения выполнены на глубинах: 1 дюйм (25,4 мм), 1,12 дюйма (28,4 мм) и 4 дюйма (101,6 мм), считая от верха конденсирующей поверхности. Тепловой поток изменялся в диапазоне от 0,3 до 1,8 милливатт/$м^2$ (т.е. от 100000 до 570000 БТЕ/кв. фут час или 260000 до 1550000 ккал/$м^2$ час). Давление составляло приблизительно 1,04 бар. Применялись четыре различных активатора.

Результаты показали хорошее согласование и воспроизводимость в различные дни. Было найдено, что коэффициент теплообмена от пара к стенке увеличивается с увеличением теплового потока в указанном выше диапазоне, причем максимальный коэффициент составляет приблизительно 0,3 милливатт/$м^2$ °C (т.е. 53000 БТЕ/кв. фут час °C или 260000 ккал/$м^2$ час °C). Не обнаружено влияния высоты пластины. Была ясно установлена разница между активаторами.