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THE EFFECT OF INCLINATION ON THE HEAT-TRANSFER COEFFICIENTS FOR FILM CONDENSATION OF STEAM ON AN INCLINED CYLINDER

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Abstract—A theory has been developed by Nusselt, and by Hassan and Jakob, for the laminar film condensation of saturated vapours. The following work for steam condensing on the outside surface of a $\frac{3}{4}$ in OD × $\frac{5}{8}$ in ID × $12\frac{1}{2}$ in long copper tube has verified the effect of the inclination of the cylinder on the heat-transfer coefficient as predicted by the above theory. The experimentally determined coefficients ranged between 95 and 109 per cent of the values determined from theory, with the greatest deviation occurring with the condensing tube in the vertical position.

NOMENCLATURE

A, area [ft^2]

h, local condensate film heat-transfer coefficient [Btu/h ft^2 degF];

h_m, mean condensate film heat-transfer coefficient on a ring of infinitesimal width on cylinder surface;

h_M, mean condensate film heat-transfer coefficient over entire cylinder as determined from theory;

h_A, actual mean condensate film heat-transfer coefficient over entire cylinder as determined by measurement;

G, mass of vapour condensed [lb/h];

g, gravitational constant;

k, conductivity of condensate [Btu/h ft degF];

L, length of cylinder [ft];

q, heat-transfer rate [Btu/h];

r, cylinder radius [ft];

Re, Reynolds number
- for horizontal tube $Re = 4G/\mu g L$
- for vertical tube $Re = 4G/2\pi\mu g r$
- for inclined tube $Re = 4G/\mu g L \cos \alpha$;

t_M, mean surface temperature of entire tube [°F];

t_s, local surface temperature of tube [°F];

t_V, temperature of condensing vapour [°F];

Δt, $t_V - t_S$;

Δt_M, $t_V - t_M$;

Δt_w, temperature rise of water;

U, heat-transfer coefficient ratio, $(h_M)_{X_L}/(h_M)_{X_L} = \infty$;

w, water flow rate [lb/h];

x, length measured along the tube [ft];

X, reduced length, $x/(r \tan \alpha)$;

X_L, $L/(r \tan \alpha)$;

α, angle of inclination of the cylinder above the horizontal;

γ, specific weight [lb/ft^3];

λ, latent heat of condensation [Btu/lb];

μ, dynamic viscosity [lb h/ft^2];

Φ, angular position from top of cylinder.

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INTRODUCTION

A GREAT deal of work has been done on the heat-transfer coefficients of vapours condensing on tubes. The basic assumptions were first stated by Nusselt [1] and from these assumptions a theory for film condensation was developed, leading to separate relations for the average heat-transfer coefficients on horizontal and vertical tubes.

Hassan and Jakob [3] applied Nusselt's basic ideas and developed a theory for filmwise condensation on inclined tubes. Their analysis enabled them to express the local film thickness at any point on the condensing surface, in terms of certain dimensionless groups. From this information they computed the local film coefficient at any point, and then a mean coefficient for the entire surface of the cylinder at any inclination, α.

To verify their theory they condensed steam inside a thick copper tube $1\frac{1}{2}$ in OD $\times$ $\frac{3}{4}$ in ID $\times$ $8\frac{1}{2}$ in long. They found that their actual coefficients ranged from 28 to 100 per cent higher than their theory predicted. They also found that the discrepancy increased with the inclination of the tube, and with the temperature difference, Δt_M, between the steam and the mean surface temperature of the tube.

Selin [4] has presented data for condensation of pure saturated vapours outside inclined tubes. Three alcohols (methanol, 2-propanol, and 1-butanol) were condensed outside copper tubes 2 m in length, and with three different outside diameters (22 mm, 28 mm, and 42·1 mm). His actual coefficients ranged from 15 per cent below the theoretical for a horizontal tube to 30 per cent above for a vertical tube.

The present investigation* applies the theory of Hassan and Jakob to the condensation of steam on the outside of an inclined cylinder. The agreement obtained has been much closer—the experimental coefficients ranging from 95 to 109 per cent of the theoretical.

ANALYSIS

The basic assumptions proposed by Nusselt are quoted below, essentially in the words of Hassan and Jakob. Their validity is discussed later.

1. The vapour is pure, dry, and saturated.
2. The condensate film flow is laminar.
3. The vapour at the vapour–liquid interface is stagnant, and the shear stress at the interface is negligible.
4. The wall temperature is uniform.
5. The liquid–solid and the liquid–vapour interfaces are smooth.
6. The velocity distribution at any point on the cylinder surface is the same as that in a fully developed isothermal film flowing on a plane tangent to the surface at that point. Under such conditions the effect of acceleration in the film is neglected. Also the effect of the curvature of the surface on the velocity distribution in the film is neglected.
7. The curvature of the surface is large enough that the effect of capillary forces may be neglected.
8. The liquid temperature at the liquid–vapour interface is that of the saturated vapour.
9. The convective heat transfer along the condensate film is neglected. Further the heat entering an element of the film is equal to the latent heat liberated by the condensing vapour.
10. The physical properties of the condensate are constant.

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A further assumption, that the radial temperature distribution through the condensate film is linear, is implied in their development.

Based on these assumptions they developed a relation from which they could compute the local condensate film thickness and the local film coefficient, h, at any point on the surface. Since the mean coefficient for the entire cylinder is a function of a reduced length $X = x/(r \tan \alpha)$, and Φ, the angular position around the cylinder, the averaging was carried out in two stages. First an integrated average value, h_m, was found for an infinitesimal ring around the cylinder. At $X = 0$, $h_m = \infty$, but it decreases rapidly and asymptotically, reaching within 2 per cent of its final value for $X \geqslant 3{\cdot}2$. Thus the cylinder can be considered as composed of two parts, a starting length $L_{St} = 3{\cdot}2r \tan \alpha$, and a portion over which the condensation is approximately equivalent to that on an infinitely long cylinder.

Then h_m was integrated numerically over the reduced length X yielding

$$h_M = \frac{1}{X_L}\int_0^{X_L} h_m(X)\,\mathrm{d}X = 0{\cdot}8045\left[\frac{\gamma^2\ \lambda\ k^3 \cos\alpha}{3\ \mu r\ \Delta t}\right]^{1/4} \quad \text{for } X_L = \infty \qquad (1)$$

At $X = 0$, $h_M = \infty$, but it decreases rapidly and asymptotically, reaching within 2 per cent of its final value at $X_L = L/(r \tan \alpha) = 50$.

It is evident that a short cylinder is more

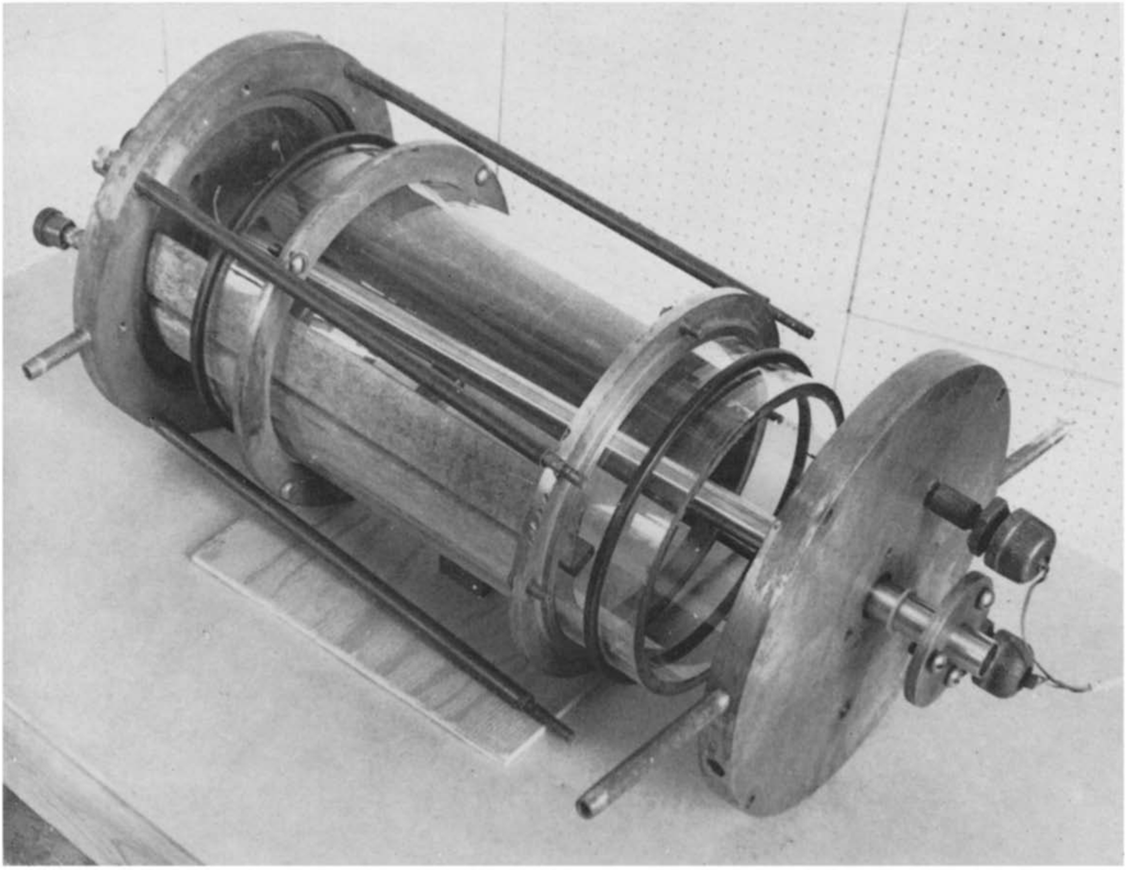

FIG. 2. The steam chamber.

effective as $X_L \to 0$, and a "long" cylinder as $a \to 0$.

Hassan and Jakob tested their theory by condensing pure steam on the inside surface of a copper tube. The fact that their experimental results ranged from 28 to 100 per cent above their predicted values appears to cast doubt upon either their development or their experimental procedure. The present writers decided that the theory was sound but that the choice of the test system was unfair to the theory and its authors, because it violated their third assumption—"The vapour at the vapour–liquid interface is stagnant and the shear stress at the interface is negligible". Accordingly, apparatus was designed and built to provide condensation on the outside of the tube thus avoiding the effects of two phase flow inside it.

APPARATUS

Figure 1 shows the general arrangement of the test equipment. The core of the apparatus was a $\frac{3}{4}$ in OD by $\frac{5}{8}$ in ID copper condensing tube mounted along the axis of a 6 in ID $\times$ $12\frac{1}{2}$ in long Pyrex pipe. End plates, seals and tie rods were added as shown in Fig. 2 to provide an enclosed condensing chamber. Steam was admitted to the chamber through two $\frac{1}{8}$ in pipe manifolds located 180° apart near the glass wall. Each manifold contained a row of small orifices discharging steam radially outwards. This steam was distributed circumferentially by means of sheet metal turning vanes so that its velocity was reduced to a minimum before the steam reached the condensing tube. It is essential that this be done since the condensate film and consequently the surface temperature distribution

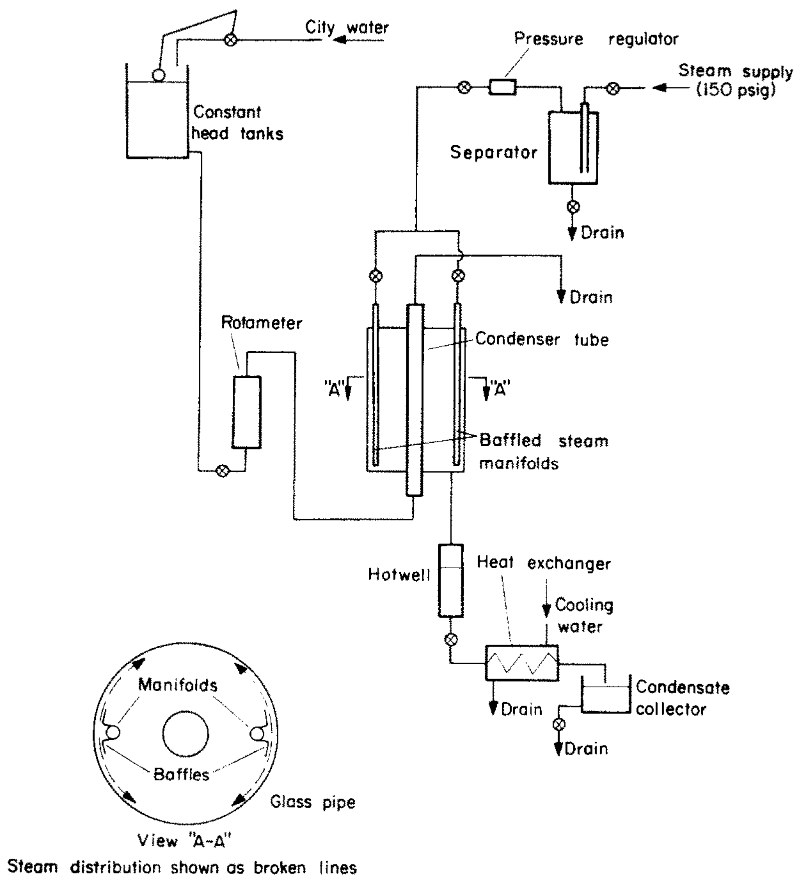

FIG. 1. Schematic arrangement of apparatus.

on the copper tube are sensitive to the method of introducing the steam.

Provision was made for rotating the condensing tube about its axis, and for altering the angle of inclination of the assembly.

The steam supply was obtained from the University power house at 150 psig wet saturated. It passed through a moisture separator and pressure regulator before entering the condensing chamber at pressures ranging from 0 to 10 psig. Condensate from the chamber was drained to a hot well and then to waste. The cold water inside the copper tube was supplied at $52° \pm 3$ degF from a float controlled constant head tank through a Rotameter, and discharged to waste.

INSTRUMENTATION

The most difficult problem was the measurement of the surface temperature of the copper tube with minimum disturbance to the surface conditions and to heat conduction in the metal. Five 30 gauge copper–constantan thermocouples were buried 0·030 in below the tube surface at $2\frac{1}{4}$ in centres along the tube, each being displaced 90° circumferentially from the preceding. Thus five circumferential temperature distributions could be obtained by rotating the tube. The five couples were laid in grooves milled in the tube wall, and the insulated leads were carried $1\frac{3}{16}$ in along the groove before entering the water stream inside the pipe. The hot junctions were covered with 14 gauge copper wire which was then soldered to the tube. The excess solder was filed away leaving a smooth surface comparable to the undisturbed tubing. (Some anxiety had been felt that the soldering might damage the insulation but this proved unwarranted.) Since the hot junction was not in contact with the surface, the thermocouple readings obtained from the recorder had to be corrected. To reduce the effect of the thermocouple insulation on the heat transfer through the tube, the layer of Fibreglas common to both leads was removed up to the point where the leads entered the cooling water inside the tube.

All other measurements of pressure, flow and temperature were made by standard means. Water temperatures were measured by standardized thermometers, and the steam temperature in the condensing chamber was obtained from four copper–constantan couples connected in parallel.

PROCEDURE

Once steady state conditions have been achieved in any test run, the heat flow through the condenser tube wall has to be equal to the heat carried away in the cooling water, i.e.

$$q = h_A \, A \, \Delta t_M = w \, C_p \, \Delta t_w.$$

Since A, w, C_p and Δt_w are either known or readily measurable, h_A, the actual or effective heat-transfer coefficient, can be found once Δt_M has been computed.

Δt_M was found using a two stage averaging technique. First the tube was rotated and surface temperature measurements were recorded every 45° at each of the five thermocouple stations. Then the average circumferential temperature was obtained arithmetically at each station. These average temperatures were then plotted vs distance along the tube, and the overall mean surface temperature, t_M, was obtained from the plot by planimeter. Since the steam temperature is known from thermocouple readings, Δt_M, and therefore h_A, can be found.

It was noted that the local surface temperature decreases as Φ increases—as would be expected from the increasing thickness of the condensate film.

The symmetry which would be expected in a plane normal to the axis of the tube was not exactly attained. The asymmetry increased with Δt_M. Since higher values of Δt_M are accompanied by larger quantities of steam, entering at higher velocities, it appears that the admission and distribution of the steam in the condensing chamber are important in determining the film characteristics, and therefore the surface temperature distribution and the local heat-transfer coefficient.

A total of 63 test runs were made at various angles of inclination and heat-transfer rates. The last 5 were replicates of the runs made at the start of the programme to see if any "aging" effect had occurred over the five week test period. The later runs yielded values of h_A averaging 2·7 per cent higher than the initial runs.

Table 1 summarizes the data obtained from

Table 1. Comparison of actual and theoretical heat-transfer coefficients

Run	Inclination α	Heat-transfer rate q	Mean temperature difference Δt_M	Actual coefficient h_A	Theoretical coefficient h_M	h_A/h_M
2	90	12700	48·9	1280	1178	1·087
4		11980	44·0	1335	1208	1·105
1		10600	40·0	1300	1215	1·071
3		9650	35·6	1330	1260	1·056
55	75	14100	48·2	1435	1425	1·005
51		13000	42·2	1510	1475	1·024
54		10000	32·1	1530	1550	0·987
48	60	14650	45·6	1575	1630	0·967
43		12900	38·7	1630	1695	0·962
44		10950	31·7	1690	1765	0·958
14	45	15170	41·9	1775	1800	0·985
15		14100	38·8	1780	1830	0·972
13		13350	37·9	1750	1850	0·945
10		11530	31·0	1825	1920	0·951
34	30	14900	41·0	1780	1885	0·944
36		12550	31·4	1960	2000	0·980
32		11300	27·5	2010	2055	0·978
28	15	15450	42·5	1780	1905	0·935
27		13350	34·9	1875	2005	0·935
24		11570	27·8	2040	2095	0·974
23	0	16300	42·5	1880	1915	0·982
20		15710	49·9	1880	1925	0·977
19		13570	35·0	1900	1980	0·959
22		11200	27·0	2035	2110	0·965

typical runs. It shows a pronounced increase in h_A as the inclination decreases from the vertical to the horizontal position, and also a lesser increase in h_A as the condensing load q decreases.

The values of h_M show the same trends. They can be obtained from Fig. 9 of the paper by Hassan and Jakob [3] for $L/r = 33$ and $0 < \alpha < 90°$, the conditions of the test programme. (Actually Hassan and Jakob's basic relation was re-integrated numerically to obtain better accuracy than could be had from the curve.)

Figure 3 compares the theoretical and actual heat-transfer coefficients as functions of α. At inclinations less than 75° from the horizontal the experimental curve is lower than the theoretical by a maximum of 5 per cent. For $\alpha > 75°$ the experimental curve is higher, with a maximum deviation of 9 per cent occurring at 90°.

The points on this plot were obtained by arithmetically averaging the values of h_A

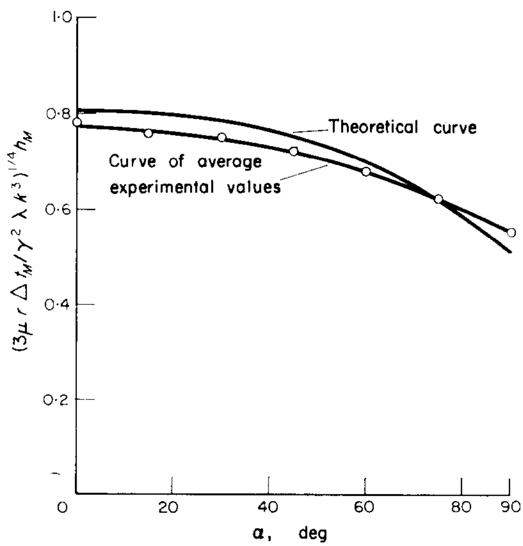

FIG. 3. Variation of the mean heat-transfer coefficient with the inclination of the tube.

obtained for all runs at a given value of α, and various values of condensing load. The maximum deviation from the average was 7·5 per cent. It was obtained at $\alpha = 15°$.

Selin obtained similar curves for the three alcohol vapours—methanol, 2-propanol, and 1-butanol, condensing on the outside surfaces of three inclined tubes with L/r ratios of 95, 143, and 182. At inclinations less than 60° to the horizontal, his experimental curves were lower than the theoretical by a maximum of 15 per cent at $\alpha = 0°$. For $\alpha > 75°$ his experimental curves were higher, with a maximum deviation of 30 per cent occurring at 90°.

In Fig. 4 the present results are replotted in dimensionless numbers. The 95 per cent confidence belt of Selin's results for the three alcohol vapours condensing on a tube with an L/r ratio of 143 is also shown. The solid line represents the equation

$$Y = \left(\frac{\mu^2 g}{k^3 \gamma^2}\right)^{1/3} U^{-4/3} h_M =$$

$$1{\cdot}51 \left(\frac{4G}{\mu g L \cos \alpha}\right)^{-1/3} = 1{\cdot}51\, Re^{-1/3} \qquad (2)$$

Selin obtained equation (2) by evaluating $\lambda/r\,\Delta t = 2\pi L h_M/G$ from $h_M \equiv \lambda G/2\pi r L\,\Delta t$ and substituting into equation (1). He then introduced the heat-transfer ratio

$$U \equiv (h_M)_{X_L}/(h_M)_{X_L=\infty} = 1{\cdot}0]$$

for $X_L > 50$, and obtained

$$h_M^{3/4} = 0{\cdot}8045\, U \left[\frac{2\pi}{3}\; \frac{\gamma^2 k^3 L \cos \alpha}{\mu g}\right]^{1/4}$$

or

$$\left(\frac{\mu^2 g}{k^3 \gamma^2}\right)^{1/3} U^{-4/3} h_M = 1{\cdot}51 \left(\frac{\mu g L \cos \alpha}{4G}\right)^{1/3}$$

which yields equation (2). If U (or $U^{4/3}$), is evaluated as a function of X_L equation (2) can be applied to condensing tubes of any length.

Figure 5 gives the relation between the correction factor U and the reduced length X_L. It can be obtained from the plot of

$$\left[\frac{3 \mu r \Delta t}{\gamma^2 \lambda k^3 \cos \alpha}\right]^{1/4} h_M$$

vs X_L in reference [3], but original data supplied by Hassan in a private letter, were used for better accuracy. The bracketed coefficient of h_M was kept constant, and the ratio

$$U = (h_M) X_L/(h_M)_{X_L=\infty}$$

was found for each value of X_L, using equation (1) to evaluate the denominator.

Figure 4 can now be used to evaluate h_M

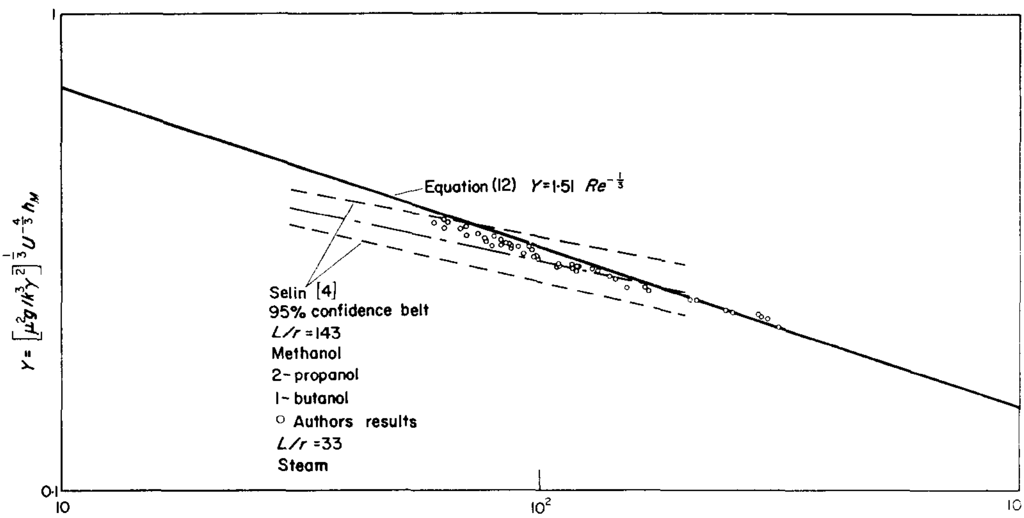

FIG. 4. Variation of the mean heat-transfer coefficient with the Reynolds number.

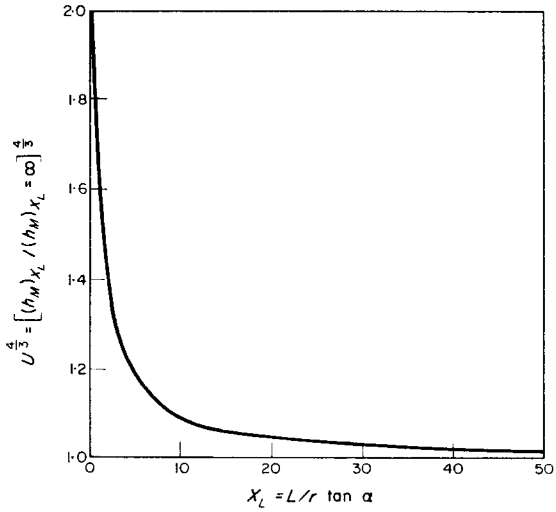

FIG. 5. Variation of the heat-transfer coefficient ratio with X_L.

for any tube at any inclination less than 90°, if the necessary physical properties are known, and the basic assumptions are applicable. For $\alpha = 90°$, $\cos \alpha = 0$, $Re = \infty$, and Fig. 4 no longer applies.

The main variables that affect the value of h are q, α, L/r, and μ. The value of q was varied between say 40000 and 80000 Btu/h ft^2. At lower values of q the temperature difference between the condensing vapour, and the condensing surface, becomes small, and further testing becomes impractical.

The present results and those of Selin cover the entire range of inclinations, L/r ratios from 33 to 182, and four different values of μ. Over this range of variables the agreement with the theory is good.

If a smaller value of L/r had been chosen the starting length would have been a more significant portion of the total, but the error due to heat conduction along the condenser tube to colder regions outside the steam chamber would have been significantly increased.

The velocity of cooling water in the condenser tube was maintained at a low value, about 0·5 ft/s, to limit errors in measuring its temperature rise. Thus the inside temperature of the tube was not uniform, and neither was the outside temperature. However, Hassan [5] has shown that the heat-transfer coefficient for a non-isothermal tube is practically the same as that for the corresponding isothermal tube, at the integrated mean temperature of the non-isothermal tube, so there is no difficulty here.

Throughout the tests at all inclinations of the condensing tube, all condensate flowed to the lower end of the tube before leaving the surface. However, at $\alpha = 0$ the water on the underside of the tube moved horizontally and formed drops at various points. These fell off when the accumulated weight became large enough.

DISCUSSION OF ASSUMPTIONS

The various Nusselt assumptions listed above were carefully considered and in most cases it was found they could introduce only trivial errors. There were three that required special consideration—purity of the condensing steam, smoothness of the liquid–vapour interface, and absence of shear at that interface.

In Fig. 3 the experimental curve falls below the theoretical curve for most of its length. This was originally attributed to the presence of air since Hampson [6] has shown that a small amount of air can cause a large reduction in the steam side coefficient, e.g. 1 per cent of air causes about 30 per cent decrease in the coefficient. However, 0·1 per cent of air causes a decrease of only about 1 per cent. Therefore the precautions taken by the power house to obtain well deaerated feedwater (to forestall corrosion) were investigated, and the data and test procedure were reconsidered. As a result it was concluded the steam used was essentially air free.*

Both Garrett and Selin attribute the low experimental values to the condensate not draining from the tube in the manner assumed by Nusselt. In Nusselt's theory the condensate film is assumed to be very thin around the tube except for a small area (about 4 per cent of the perimeter) where the condensate falls from the tube in a continuous film. In Selin's and the present work it was observed that the condensate thickness built up over a much larger area under the tube. When the weight of condensate

* In a long condensation test 2·4 g of non-condensables were collected from 159 kg of condensate. This run was made well after the end of the research and it can be considered only as order of magnitude verification.

was sufficient to overcome the surface tension forces, the condensate separated from the tube in the form of droplets. As Nusselt's theory assumes that the total thermal resistance to heat transfer is directly proportional to the integrated mean thickness of the condensate film, then the practical heat-transfer coefficient will be less than the theoretical since the integral mean film thickness of the condensate film will be greater than assumed in the theory.

The assumption of a smooth liquid–vapour interface was not met. Widely spaced ripples with long troughs between them were visible in the flowing condensate film. The character of the interface depended upon the angle of inclination of the tube, the position on the tube surface, and the heat-transfer rate. At $\alpha = 0$ the film was smooth. At $\alpha = 15^\circ$ the top surface was smooth, but rippling occurred on the underside. The intensity of this rippling increased with the value of α. At $\alpha = 75^\circ$, ripples appeared on the upper surface. At $\alpha = 90^\circ$, ripples appeared all around the tube except for a few inches at the upper end. At all values of α the intensity of rippling increased with increasing heat-transfer rate. These observations indicate that the degree of rippling is directly proportional to the *Re* of the condensate film, as also observed by Dukler and Bergelin [7] and Grimley [8]. From their work it seems that the flow is laminar* regardless of the ripples.

These ripples tend to increase the rate of heat transfer, and as the rippling becomes more intense, the heat-transfer coefficient may be expected to rise. This mechanism undoubtedly accounts for the fact that the experimental curve of Fig. 3, which is about 5 per cent below the theoretical at low values of α, where rippling does not occur, gradually rises to about 9 per cent above the theoretical as the rippling becomes more intense. The more negative slope of the present results in Fig. 4 suggests that the increase in rippling of the condensate film with *Re* was not as pronounced as that experienced in Selin's work. Hassan and Jakob [3] state that for $Re =$ 785 the heat-transfer coefficient of a rippling film will be 50 per cent larger than that of a smooth film. In the present work the maximum value of *Re* was 313, so the increase in heat transfer was always much less.

The assumption that the vapour–liquid interface is stagnant needs consideration. During preliminary testing the condensing tube was rotated to obtain the temperature distribution around the circumference of the tube. This distribution should be symmetrical about a vertical axis. This symmetry was not attained until considerable attention had been devoted to the distribution of the incoming steam by suitable manifolding and baffling. The asymmetry was attributed to incoming steam wiping the surface of the condensate thus affecting the local thickness of the condensate film, and the local surface temperature of the tube. No test runs were made until this asymmetry had been almost completely removed by reducing the entering velocity of the steam to as low a value as possible, and distributing it over the inside surface of the 6 in Pyrex pipe surrounding the condensing tube.†

CONCLUSIONS

1. The results of this programme, along with those of Selin, have verified the theory developed by Hassan and Jakob for all inclinations, for L/r ratios from 33 to 182, and for steam, methanol, 1-butanol, and 2-propanol vapours.

2. The flow volume and direction of the steam entering the condenser have a substantial effect on the heat transfer. Considerable attention must be paid to the design of the steam distribution if theory is to be verified.

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* This has been questioned by a colleague on the basis that flow may be laminar in the sense that while film thickness may be still predictable from the Nusselt theory, and the flow may be laminar beneath the ripples, the ripples themselves should not be considered laminar.

† It has been stated that a full scale condenser would not be built with the special arrangements used here to obtain uniform steam distribution—the cost would be greater and the heat transfer less. However the purpose here was to verify theory, not to develop condenser design.

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Résumé—Une théorie a été établie par Nusselt, et par Hassan et Jakob, pour la condensation par film laminaire de vapeur saturées. L'étude suivante de la condensation de la vapeur sur la surface intérieure d'un tube de cuivre de 317 mm de long, de 16 mm de diamètre intérieur et de 19 mm de diamètre extérieur, a vérifié l'effet de l'inclinaison du cylindre sur le coefficient de transport de chaleur comme il l'est prédit par la théorie ci-dessus. Les coefficients déterminés expérimentalement se placent entre 95 et 109 pour cent des valeurs déterminées à partir de la théorie, la plus grande déviation se produisant avec le tube de condensation dans la position verticale.

Zusammenfassung—Für laminare Filmkondensation gesättigter Dämpfe wurde eine Theorie von Nusselt und von Hassan und Jakob entwickelt. Für Dampf, der an der äusseren Oberfläche eines Kupferrohres von 19,5 mm Aussendurchmesser, 15,9 mm Innendurchmesser und 317 mm Länge kondensiert, weist diese Arbeit den Einfluss der Zylinderneigung auf die Wärmeübergangszahl nach, wie er von der oben erwähnten Theorie vorhergesagt wurde. Die experimentell bestimmten Übergangszahlen bewegten sich zwischen 95 Prozent und 109 Prozent der theoretisch ermittelten Werte. Die grösste Abweichung ergab sich bei vertikaler Lage des Rohres.