

FILM CONDENSATION OF MERCURY

S. NECMI* and J. W. ROSE

Department of Mechanical Engineering, Queen Mary College, University of London, London, U.K.

(Received 28 October 1975 and in revised form 11 February 1976)

Abstract—Measurements of vapour-to-surface temperature difference and heat flux for film condensation of mercury on a vertical plane square (side 40 mm) nickel-plated copper surface are reported. Thermocouples, accurately located and spaced through the copper condensing block served to measure, by extrapolation, the temperature at the copper-nickel interface and from the temperature gradient, the heat flux. Special care was taken to ensure that the results were not vitiated by the presence in the vapour of non-condensing gases. The results have higher relative precision than other recent heat-transfer measurements for condensation of metals since the present observations were made under conditions (metal used, vapour temperature and condensation rate) for which the vapour-to-surface temperature difference was larger than in the earlier work. The observed vapour-to-surface temperature differences are substantially greater than those given by the Nusselt theory of film condensation. By attributing the excess temperature drop to the vapour-liquid interface, the results are compared with theoretical expressions for interphase matter transfer. As in other recent work, values for the correction factor (or apparent "condensation coefficient") varied from near unity down to about 0.6. The precision of the present results is such as to reveal a dependence of the correction factor on the condensation rate as well as on the vapour pressure. It was found that both could be satisfactorily correlated by a single dimensionless variable.

NOMENCLATURE

A , cross-sectional area of condenser chamber;
 c_p , constant-pressure specific heat-capacity of vapour;
 c_v , constant-volume specific heat-capacity of vapour;
 c_{pf} , constant-pressure specific heat-capacity of condensate;
 g , local gravitational acceleration;
 h_f , specific enthalpy of condensate;
 h_g , specific enthalpy of vapour;
 h_{fg} , $h_g - h_f$;
 k_f , thermal conductivity of condensate;
 Ma , Mach number of vapour flow normal to test condenser surface, $mv_v/\sqrt{(\gamma RT_v)}$;
 m , interphase net mass flux;
 P_s , vapour pressure immediately at vapour-liquid interface;
 P_v , vapour pressure in test condenser, i.e. outside interface zone;
 $P_{sat}(T)$, saturation pressure corresponding to temperature T ;
 Q , heat flux in test condenser block;
 Q_t , heat-transfer rate to test condenser;
 Q_s , heat-transfer rate to secondary condenser;
 R , specific ideal-gas constant;
 T_b , temperature of vapour in boiler;
 T_s , vapour-liquid interface temperature;
 T_v , vapour temperature in test condenser, i.e. outside interface zone;

T_w , surface temperature (vapour-side) of test condenser block;
 u_1 , vapour velocity upstream of test condenser;
 u_2 , vapour velocity downstream of test condenser;
 v_f , specific volume of condensate;
 v_g , specific volume of vapour;
 v_v , specific volume of vapour in test condenser chamber;
 v_{fg} , $v_g - v_f$;
 x , distance from top edge of condensing surface.

Greek symbols

α , $Q/(T_v - T_w)$;
 α_i , $Q/(T_v - T_s)$;
 γ , c_p/c_v ;
 ΔP , $P_{sat}(T_v) - P_{sat}(T_s)$;
 ΔT , $T_v - T_w$;
 δ , local condensate film thickness;
 δQ , estimate of error in observed value of Q ;
 δT_v , estimate of error in observed value of T_v ;
 δT_w , estimate of error in observed value of T_w ;
 $\delta(\Delta T)$, estimate of error in observed value of ΔT ;
 θ_v , Celsius temperature of vapour, i.e. $T_v - 273.15$ K;
 θ_{vN} , nominal vapour temperature, i.e. rounded Celsius temperature close to θ_v ;
 μ_f , viscosity of condensate;
 ξ , defined by equation (5);
 ρ_f , density of condensate;
 ρ_g , density of vapour;
 ρ_{fg} , $\rho_f - \rho_g$;
 σ , condensation coefficient.

*Present address: Whiteshell Nuclear Research Establishment, Atomic Energy of Canada Ltd., Pinawa, Manitoba, Canada.

INTRODUCTION

IN RECENT years, under the stimulus of potential nuclear and space applications, there have been many experimental heat-transfer studies of film condensation of metals. Apart from the fact that these all indicated vapour-to-condensing surface temperature differences in excess of those small (due to the high condensate thermal conductivity) values given by the simple Nusselt theory, agreement between results of different workers in general has been poor.

Most investigators have suggested that the excess temperature drop occurs at the vapour-liquid interface as a result of net transfer of matter from the vapour to the liquid phase. In many cases the results have been used in conjunction with theories of interphase matter transfer (notably due to Schrage [1]) to determine the "condensation coefficient" (the fraction of vapour molecules, incident on the liquid surface, which remain in the liquid phase). The disagreement between the results of different workers referred to above has led to a variety of values of the condensation coefficient ranging from near unity down to about 0.02.

The fact that low values of the condensation coefficient (implying that a large proportion of vapour molecules are reflected at the liquid surface) were not regarded as unlikely, in view of the fact that for solid surfaces as well as for some liquids including mercury, values near unity had been obtained [1], was mainly due to similar low values having been obtained from early unreliable evaporation experiments with water.

A common feature of many experimental studies of film condensation of metals is the fact that the apparent condensation coefficient was found to decrease with increasing vapour pressure. Following the recent work of Wilcox and Rohsenow [2, 3], it seems likely that in many cases the observed pressure dependence may have been due to increased out-gassing of the walls of the apparatus at the higher temperatures, as well as to inaccuracies in the measurement of the smaller vapour-to-surface temperature differences at the higher pressures.

In the light of the available evidence (from experiments on condensation and evaporation at solid surfaces and surfaces of non-metallic liquids, as well as those of liquid metals), the present authors incline to the view that, for clean surfaces, the condensation coefficient is in all cases unity. Comparisons between experiment and theory which yield substantially lower values are thought to involve experimental error or inadequate theory or both.

The main objective of the present work was to measure, for a condensing metal vapour, the relationship between the heat flux and the vapour-to-surface temperature difference, and the dependence of this relationship on the vapour pressure, with sufficient accuracy to enable valid comparison with theories of interphase matter transfer to be made. A condensation experiment using a metal as the condensing fluid is particularly well-suited to this purpose since, for laminar film condensation, the temperature drop in the condensate layer can be calculated with good accuracy

and is generally small in comparison with anticipated values of the temperature drop at the vapour-liquid interface. Of crucial importance is the precision with which the vapour-to-surface temperature difference is measured. Mercury was chosen for the present work since, for convenient experimental conditions, the predicted temperature drop is greater than for other metals.

APPARATUS

Figure 1 shows the all-welded stainless-steel test loop. Vapour passed from the electrically-heated boiler downwards over the vertical plane surface of the copper test condenser block. The secondary condenser, located below the test condenser, could be used when required

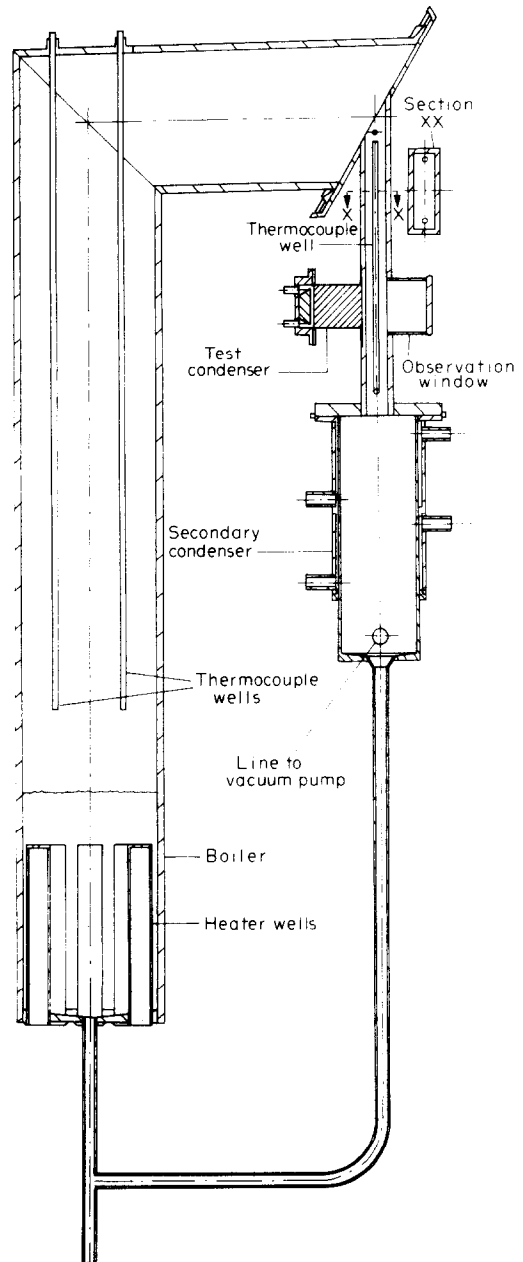


FIG. 1. The test loop (scale 3:20).

to generate cross-flow over the test surface. The condensate from both condensers was returned by gravity to the boiler.

The apparatus could be vented, by a vacuum pump, from a point near the base of the secondary condenser. The vent tube was fitted with a small water jacket and was sloped so that condensed mercury ran back into the boiler. A valve and a liquid nitrogen "cold trap" were located in the venting line between the water-cooled section and the vacuum pump.

Water and air could be used as coolants for both the test and secondary condensers, the flow rate in both cases being measured by "float-type" flowmeters. The whole apparatus was thermally well-insulated. The boiler and test-condenser section were each fitted with two thermocouple wells (wall thickness 0.25 mm) in the form of closed-ended stainless-steel tubes located as shown in Fig. 1. The boiler was fitted with ten 2-kW "cartridge" heaters, each fitting snugly in a vertical closed-ended tube welded to the base. Two of the heaters were supplied via variable transformers. The condensing surface could be viewed through an electrically-heated (to prevent condensation) window.

For maximum precision of location of the thermocouples, it was necessary that the holes in the test condenser block were as small as possible and, to avoid errors due to conduction along the thermocouple leads, these had to run along isotherms in the vicinity of the junction. To obtain adequately small holes of sufficient depth, a development of the split-plate technique used in earlier measurements [4] was adopted. The copper block was first cut into two halves in a plane normal to the condensing surface. Square grooves (side 0.3 mm) were machined in the mating face of one half, parallel to the condensing surface. The block was then re-assembled by "diffusion welding" in a vacuum furnace. The resulting condenser block thus had 0.3 mm square holes passing horizontally from side to side (43 mm) and running parallel with the condensing surface in the central horizontal plane of the block. The block was then machined to its finished dimensions and the distances of the thermocouple holes from the condensing surface measured by travelling microscope with a precision of 0.01 mm. The nominal distances were 14.5, 20, 30 and 40 mm. Finally the block was silver-soldered, at the edges of the condensing face, to a stainless steel flange of thickness 2 mm. The condensing surface was nickel-plated* to a thickness of $0.05 \text{ mm} \pm 0.0025 \text{ mm}$ before being welded, via the steel flange, to the condenser section.

In order to achieve maximum accuracy in the vapour-to-surface temperature difference, the thermocouples used in the vapour and condenser block were made from the same reel of nichrome-constantan wire (0.15 mm dia). Prior to use, the wire was annealed at a temperature exceeding the maximum operating temperature. The junctions of the thermocouples used in the condenser block were butt-welded and when in use

were positioned in the central vertical plane of the block, the nichrome and constantan wires being led out on different sides.

Calibration was carried out against a thermocouple calibrated by the National Physical Laboratory (U.K.). The junction of the standard and that to be calibrated were tightly bound together with fine wire and placed in one of the boiler thermocouple wells. Calibration points were obtained in the range 23–269°C. These were fitted by a polynomial, with which the points, obtained over several days, generally agreed to within about 0.05 K.

The arrangement of the cold-junctions, copper lead wires and the general measuring procedure were the same as described earlier [4]. The thermo-e.m.f. could be measured either by vernier potentiometer or by digital voltmeter.

Special care was taken to ensure that the apparatus was free from leaks. Prior to admitting mercury, "out-gassing" of the inner walls of the test loop was carried out using externally wound heating tapes. Pressures down to 10^{-4} torr could be achieved. The mercury (35 kg) was admitted under vacuum.

Full details of the apparatus are given in [5].

OBSERVATIONS

Preliminary

Throughout the tests described below the apparatus was vented continuously via the "cold trap". On the first day of operation the formerly bright nickel surface became dark grey in colour and the mode of condensation initially was dropwise. After about 4 hours, several small shiny wetted areas appeared. Over the next ten days the apparatus was operated for several hours per day during which time the surface became completely wetted and remained so throughout the investigation. During these preliminary runs it was noticed that the window heater affected the readings of the vapour thermocouples. Subsequently, the window was covered with insulating material which was only removed occasionally, and the heater used, in order to inspect the surface. The window heater was never used while heat-transfer measurements were being made.

Effect of vapour velocity on the vapour temperature and preliminary heat-transfer measurements

When operating under steady conditions, with the two test-section vapour thermocouples fully inserted in their wells and with cooling water passing through both test and secondary condensers, significant differences were found between the readings of the two test-section vapour thermocouples. In some cases the difference between the two temperatures was as much as 8 K, while the two thermocouples measuring the vapour temperature in the boiler agreed closely, their difference generally being less than 0.2 K.

In order to study this effect more closely, tests were carried out in which temperatures indicated by the vapour thermocouples were observed with the junctions at different positions along the wells. The coolant flow rates and the coolant, boiler and test condenser wall temperatures were also observed. During these

*This was in order to protect the copper against attack by mercury and to obtain film condensation.

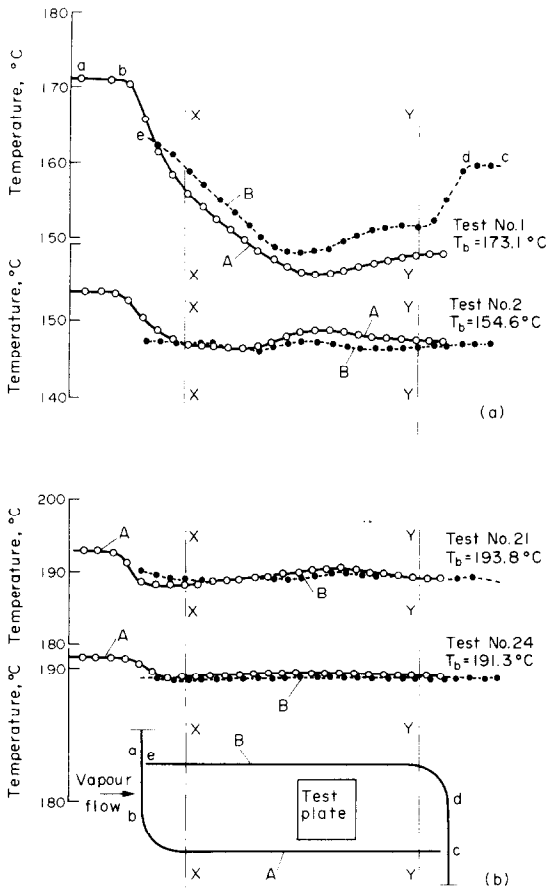


FIG. 2. Temperature distributions along thermocouple wells in the condenser section.

tests the apparatus was vented continuously via the cold trap and vacuum pump.

The variation of the readings of the test-section vapour thermocouples with the position of the junctions in their respective wells, for four of these tests, are shown in Fig. 2. For the two tests shown in Fig. 2(a) the vapour temperature in the vicinity of the condensing surface was about 147°C . The upper curves (test No. 1) are for a case where the secondary condenser was used to give a net cross flow of vapour over the test surface, while the lower curves (test No. 2) are for a case when the secondary condenser was not used.* Figure 2(b) shows the results of a similar pair of tests at a higher vapour temperature where the vapour specific volume and hence vapour velocities were smaller.

The locations of the thermocouple wells in relation to the direction of vapour flow and test condenser surface are shown in the diagram beneath the temperature distributions. The individual test numbers are indicated on Fig. 2 and estimates of the velocities in the vapour chamber, upstream and downstream of the condensing surface, are given in Table 2.

*When water had been used as coolant in the secondary condenser in the previous test, the supply was disconnected and the secondary condenser allowed to boil dry before equilibrium conditions were established and measurements made.

It is apparent from Fig. 2 that the temperature variations observed are related to the vapour velocities. It seems clear that the high temperatures along ab (i.e. where the axis of the well was normal to the vapour flow) for thermocouple A were due to "stagnation" of the vapour along the front edge of the well. It was found that in all cases the temperature along ab was close to the temperature in the boiler. Since the vapour velocity is small in the wider-section boiler, one would expect the boiler temperature to approximate to the stagnation temperature in the condenser section upstream of the test condenser. Similarly "stagnation" would explain the higher temperature along cd for thermocouple B , for those cases where the downstream velocity was significant. The fact that there is no clear evidence of a stagnation temperature increase at the upstream end e , of thermocouple well B is thought to arise from the geometry of the sealed end of the tube shown in Fig. 3. The gas in the cup-shaped end[†] of the

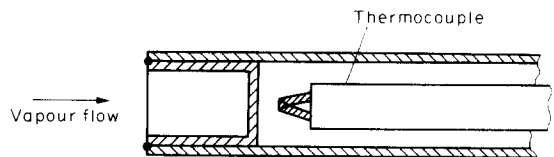


FIG. 3. End of thermocouple well B .

tube tends to insulate the thermocouple junction from the stagnation region. If thermocouple B is not influenced by stagnation at e , then the indication of this thermocouple between e and some position[‡] in plane Y , upstream of d , reflects a streamwise temperature (i.e. "static" temperature) variation in the vapour. This is supported by the fact that the temperature profiles indicated by both thermocouples between planes X and Y , i.e. for the region where neither well is influenced by stagnation on the portions of their length which run normal to the direction of flow of the vapour, are similar.

While it is thought that the vapour thermocouples gave a true estimate of the vapour temperatures near the condensing surface, there remains significant uncertainty at the high vapour velocities, on account of the difference between the two thermocouple readings and the variation of each with position. Figure 2(b) indicates that for moderately low vapour velocities the temperature of the vapour in the condenser section is essentially uniform.

On the basis of the preliminary tests it was decided first to carry out tests with zero§ or moderate (using air as coolant in the secondary condenser) cross flow velocities. The heat-transfer measurements taken during the preliminary tests are included in Table 1.

[†]The tubes were closed in this way to facilitate welding.

[‡]Where the effect of conduction along the wall of the well from the higher-temperature stagnation region was negligible. A simple one-dimensional "slender-fin" calculation indicates that conduction along the wall has negligible effect beyond a distance of about 1 cm.

[§]Save that generated by the vent.

Table 1. Summary of experimental results and estimates of error

Test No.*	θ_{VN}^\dagger (K)	T_c (K)	δT_c (K)	T_w (K)	δT_w (K)	$\frac{Q}{(kW/m^2)}$	$\left(\frac{\delta Q}{Q}\right) \times 100$	$\frac{\Delta T}{(K)}$	$\frac{\delta(\Delta T)}{(K)}$
1	¶147	420.7	2.8	401.0	0.5	281	2.5	19.7	2.9
2	§147	419.3	1.3	399.3	0.4	264	2.0	20.0	1.4
3	160	433.4	0.3	420.9	0.6	243	3.2	12.5	0.6
4	160	433.7	0.1	419.8	0.6	279	2.9	13.9	0.6
5	160	433.1	1.0	414.5	0.5	368	2.0	18.6	1.2
6	160	433.6	0.6	411.4	0.5	432	1.5	22.2	0.8
7	160	432.9	1.7	402.1	0.6	586	1.4	30.8	1.8
8	176	449.4	0.1	438.7	0.7	302	3.2	10.7	0.7
9	176	449.4	0.1	439.0	0.9	317	3.8	10.4	0.9
10	176	449.4	0.1	437.1	0.5	394	1.7	12.3	0.5
11	176	449.0	0.1	433.1	0.5	485	1.3	15.9	0.5
12	176	449.1	0.1	427.0	0.6	635	1.4	22.1	0.6
13	¶187	460.0	2.0	448.9	0.7	395	2.5	11.1	2.1
14	§187	460.1	0.1	449.7	0.7	401	2.4	10.4	0.7
15	190	462.5	0.1	452.4	0.7	379	2.5	10.1	0.7
16	190	462.3	0.1	449.7	0.5	513	1.4	12.6	0.5
17	190	461.8	0.3	446.1	0.5	607	1.1	15.7	0.6
18	‡190	463.1	0.3	453.3	0.5	396	1.9	9.8	0.6
19	‡190	463.1	0.3	450.8	0.6	515	1.6	12.3	0.6
20	‡190	463.4	0.5	448.4	0.6	616	1.4	15.0	0.8
21	190	463.1	0.5	451.6	0.6	490	1.7	11.5	0.8
22	§190	462.0	0.1	449.3	0.5	495	1.5	12.7	0.5
23	¶190	464.1	2.6	453.5	0.8	458	2.4	10.6	2.7
24	§190	463.6	1.4	453.9	0.4	451	1.4	9.7	0.5
25	221	495.0	0.1	484.6	1.0	631	2.2	10.4	1.0
26	221	493.3	0.1	483.3	0.7	666	1.5	10.0	0.7
27	221	493.8	0.1	481.3	0.6	739	1.2	12.5	0.6

*Not chronological order of observations.

†Nominal vapour temperature, i.e. a rounded Celsius temperature close to T_c .

‡Secondary condenser operated with air as coolant.

§Preliminary test—secondary condenser not operated.

¶Preliminary test—secondary condenser operated with water as coolant.

||Preliminary test—secondary condenser operated with air as coolant.

Heat-transfer measurements with zero and moderate crossflow

With the secondary condenser not operated, and the apparatus continuously vented via the cold trap and vacuum pump, measurements were made at vapour temperatures (in the vicinity of the condensing surface) of 160, 176, 190 and 221°C. At each vapour temperature several coolant (water) flow rates were used. Each time the coolant flow rate was changed, the boiler heating rate was adjusted to maintain the desired vapour temperature. For each test the vapour thermocouples in the condenser section were traversed along their wells and temperatures observed at intervals. The condensing plate temperatures, coolant inlet and outlet temperatures, boiler temperature and coolant mass flow rate were also observed.

To check that the results obtained were not influenced by the presence in the vapour of non-condensing gases, tests were carried out while operating the secondary condenser with air as coolant to generate moderate cross flow velocities over the surface of the test condenser. These measurements were made at a vapour temperature of 190°C which had also been used in the preliminary high vapour velocity tests using water as coolant in the secondary condenser.

The final tests to be made were those at the highest

vapour temperature of 221°C. During these tests the nickel surface of the test condenser was noticed to become apparently eroded at various locations. After a further four hours of operation on the following day, irregular depressions, to a depth of one or two millimetres (as subsequently found when the test section was removed) extended over most of the surface. It is thought that the mercury had attacked the copper at the high temperature via an imperfection (possibly at one of the edges) in the nickel plating, the thin nickel layer collapsing as the copper beneath it was dissolved. The investigation was terminated at this point.

RESULTS

The results of all tests, including the preliminary measurements, together with estimates of error, are given in Table 1.

The vapour temperature T_b , recorded in Table 1, is the arithmetic mean of four values; two (one just upstream and one just downstream of the condensing surface) given by each vapour thermocouple.** The

**Significant vapour temperature non-uniformities were found even in tests for which the secondary condenser was not used, in cases where the vapour velocity (generated solely by condensation on the test condenser) in the test section was high.

Table 2. Calculated quantities relating to cross flow and normal flow

Test No.	Fraction* condensed	Cross flow			Flow normal to condensing surface	ξ [equation (5)]	
		u_1 (m/s)		u_2 (m/s)			
		(1)†	(2)‡	*†			
$\left(\frac{Q_t}{Q_t + Q_s}\right)$	$\frac{(Q_t + Q_s)v_g}{h_{fg}A}$	$[2c_p(T_b - T_v)]^\ddagger$	$\frac{Q_s v_g}{h_{fg}A}$	Ma	$\frac{\Delta P}{P_{sat}(T_v)}$		
1	0.31	155	73	107	0.275	0.55	0.65
2	1.00	47	42	0	0.273	0.56	0.64
3	1.00	26	26	0	0.148	0.37	0.52
4	1.00	29	27	0	0.167	0.40	0.55
5	1.00	39	35	0	0.225	0.50	0.59
6	1.00	45	38	0	0.257	0.56	0.60
7	1.00	63	48	0	0.345	0.69	0.66
8	1.00	18	16	0	0.103	0.30	0.45
9	1.00	19	20	0	0.108	0.29	0.49
10	1.00	24	17	0	0.135	0.33	0.54
11	1.00	30	25	0	0.168	0.40	0.55
12	1.00	39	30	0	0.217	0.52	0.56
13	0.21	82	57	65	0.094	0.29	0.43
14	1.00	17	21	0	0.095	0.27	0.47
15	1.00	15	12	0	0.083	0.26	0.42
16	1.00	20	18	0	0.114	0.31	0.49
17	1.00	25	20	0	0.136	0.37	0.49
18	0.66	23	30	8	0.085	0.25	0.45
19	0.71	28	32	8	0.111	0.30	0.49
20	0.75	31	33	8	0.131	0.35	0.50
21	0.73	26	35	7	0.106	0.28	0.50
22	1.00	20	23	0	0.111	0.31	0.47
23	0.15	111	60	94	0.096	0.26	0.48
24	1.00	17	18	0	0.096	0.24	0.53
25	1.00	10	0	0	0.052	0.21	0.33
26	1.00	11	10	0	0.057	0.20	0.40
27	1.00	12	5	0	0.063	0.25	0.34

*Excludes cross flow generated by vent.

†Nominal value assuming saturation conditions at T_v .

‡Estimate based on adiabatic flow between boiler and test section, taking vapour as perfect gas and neglecting velocity in the boiler.

Note: The differences between the two estimates of u_1 , at high velocities result from non-uniform conditions in the vapour, and at the low velocities are due to high relative error in the small values of $T_b - T_v$.

error estimate, δT_v , is half the difference between the highest and lowest of the four values.

The condenser surface temperature, T_w , and the heat flux Q were determined§ with good precision, as may be judged from the typical temperature distributions in the test condenser block shown in Fig. 4. The error estimates, δT_w and δQ , are standard deviations from the "least squares" fits to the temperature distributions in the condenser block.

The error estimate $\delta(\Delta T)$, for the vapour-to-surface temperature difference ΔT (i.e. $T_v - T_w$), given in Table 1, is $(\delta T_v^2 + \delta T_w^2)^{1/2}$.

The heat flux given in Table 1 and used in calculations throughout was that given by the temperature distribution in the test condenser block.

The heat flux was also determined from the coolant measurements. Though these values served only as a

check and no special care was taken to ensure high accuracy, the heat fluxes obtained from the coolant measurements agreed with the more accurate values quoted in Table 1, to within a few per cent.

It may be seen from Fig. 5 that the relationship between Q and ΔT is well determined, as is the dependence of this relationship on the vapour temperature.

The close agreement (see Fig. 6), at a vapour temperature of 190°C, between the results with zero, moderate and high cross flow velocities¶ helps to confirm that significant amounts of non-condensing gases were not present and also indicates that, in the absence of non-condensing gases vapour velocity has little effect on heat-transfer coefficient. These conclusions are also supported by the fact that the preliminary test with high vapour cross flow velocity at a vapour temperature of 147°C is also apparently consistent with the other results.

§The effects of variation with temperature of the thermal conductivity of the copper block, and of the temperature drop across the nickel plating, were barely significant but were included in the determination of Q and T_w .

¶Estimates of vapour velocities in the condenser section upstream and downstream of the test condenser are given in Table 2.

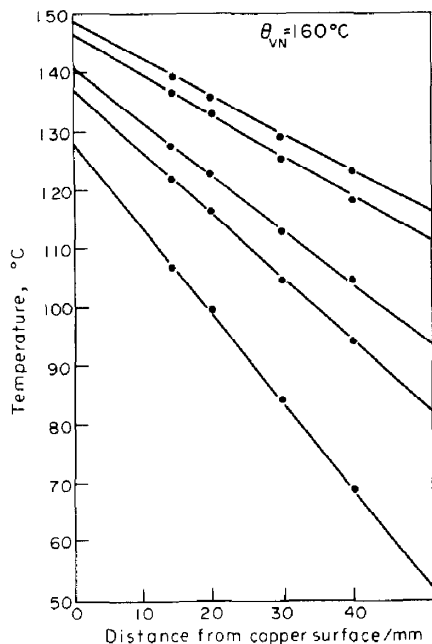


FIG. 4. Specimen temperature distributions in test condenser block.

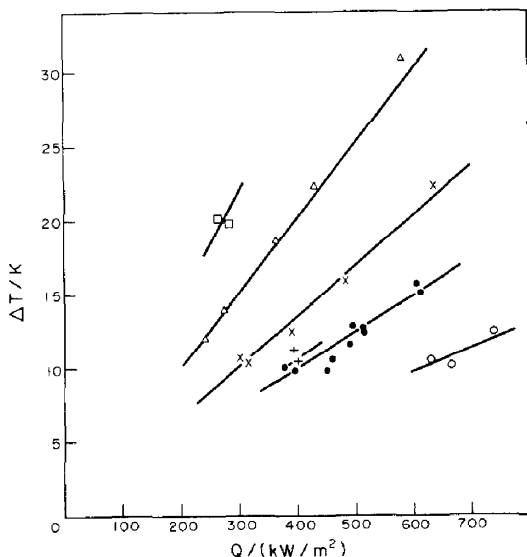


FIG. 5. Observed relationship between vapour-to-surface temperature difference and heat flux at different vapour temperatures: □, 147°C; △, 160°C; ×, 176°C; +, 187°C; ●, 190°C; ○, 221°C. The lines are those given by equation (1).

The results shown in Fig. 5 are adequately represented by straight lines through the origin, indicating that the observed heat-transfer coefficient (vapour-to-wall, i.e. including both the interface and condensate resistances) is, for a given vapour temperature, essentially constant for the experimental ranges of T_v and Q . The results are conveniently summarised by the empirical equation:

$$\alpha / (\text{kW/m}^2 \text{K}) = \exp(a + b\theta_v + c\theta_v^2) \quad (1)$$

where

$$a = -5.20, \quad b = 0.0747 \text{ K}^{-1}, \quad c = -0.000147 \text{ K}^{-2}.$$

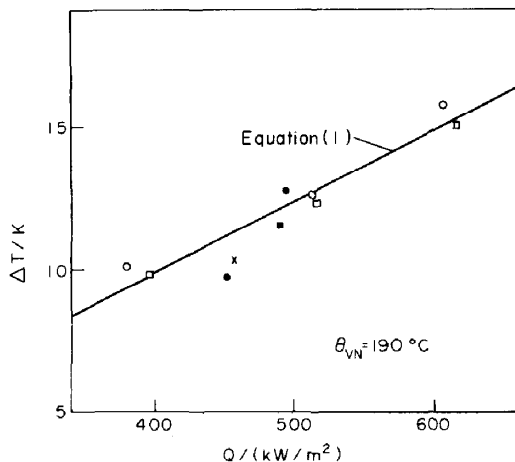


FIG. 6. Results obtained for different cross-flow conditions (see Table 2).

Symbol	Test No.
○	15-17
□	18-20
■	21
●	22, 24
×	23

DISCUSSION

Comparison with theories of interphase matter transfer

Various workers have used kinetic theory arguments to relate the properties of the liquid and vapour phases, in the immediate vicinity of the interface, to the net matter transfer rate between the phases.

Schrage [1] gives two separate analyses, the first of which, when linearized, gives

$$m = \left(\frac{2\sigma}{2-\sigma} \right) \frac{(P_v - P_s)}{\sqrt{(2\pi R T_v)}} \quad (2)$$

A similar result was also found by Kutcherov and Ricenglas [6, 7].

Schrage's second theory, in which the condensation coefficient was set to unity at the outset, relates to monatomic gases only and, when linearized, gives

$$m = \left(\frac{\pi}{8 R T_v} \right)^{\frac{1}{2}} (P_v - P_s) \quad (3)$$

Labuntsov and Muratova [8-11] give

$$m = \left(\frac{2\sigma}{2-0.798\sigma} \right) \frac{(P_v - P_s)}{\sqrt{(2\pi R T_v)}} \quad (4)$$

Each of the above results takes the form

$$m = \xi (P_v - P_s) / \sqrt{(R T_v)} \quad (5)$$

Equation (2) indicates

$$\xi = \left(\frac{2\sigma}{2-\sigma} \right) / \sqrt{(2\pi)} \quad (6)$$

which, with $\sigma = 1$, gives $\xi = 0.798$.

Equation (3) gives $\xi = 0.627$.

Equation (4) indicates

$$\xi = \left(\frac{2\sigma}{2-0.798\sigma} \right) / \sqrt{(2\pi)} \quad (7)$$

which, with $\sigma = 1$, gives $\xi = 0.664$.

The present results were used to determine ζ from equation (5). The Nusselt theory was used to determine T_s , P_s was taken as $P_{sat}(T_s)$ and m was found from the relation

$$Q = mh_{fg} + m^3 v_g^2 / 2. \tag{8}$$

The last term in equation (8) is due to the kinetic energy of the vapour. It was negligible in most cases but, for those tests where the vapour approach velocity was highest, led to reduction in the calculated value of m by about 3%.

The interface temperature cannot strictly be found directly from the wall temperature T_w and the Nusselt theory for the condensate film since, when the temperature drop at the vapour-liquid interface is significant, T_s is not, in general, independent of height. The Nusselt analysis can be simply modified and a more accurate result obtained as indicated in Appendix A. It may be noted, however, that for the present case, the

when used with equation (2) (or equivalent non-linear forms) will still yield values of σ not far from unity. Thus, if a value of $P_t - P_s$, say δP , gives, with equation (2), a value of unity for σ , then values of $P_t - P_s$ in the range $\delta P \pm \delta P$ give values of σ in the range 2/3 to 2. On the other hand, errors in $P_t - P_s$ give rise to proportional changes in the quantity ζ .

The values of ζ (or of σ) shown in Fig. 7 are brought onto a single curve when plotted against the dimensionless quantity $Qv_g/h_{fg}\sqrt{(RT_v)}$, Q and v_g being the dominant variables. Since $Q \approx mh_{fg}$ the dimensionless group $Qv_g/h_{fg}\sqrt{(RT_v)}$ approximates closely to $\gamma^{1/2} Ma$, Ma being the Mach number of the vapour flow normal to the condensing surface. Thus ζ correlates equally well with Ma . The relationship between ζ and Ma is shown in Fig. 8 and can be seen to be satisfactorily represented by the empirical equation:*

$$\zeta = (2/3) - (1/2)\exp(-9Ma). \tag{9}$$

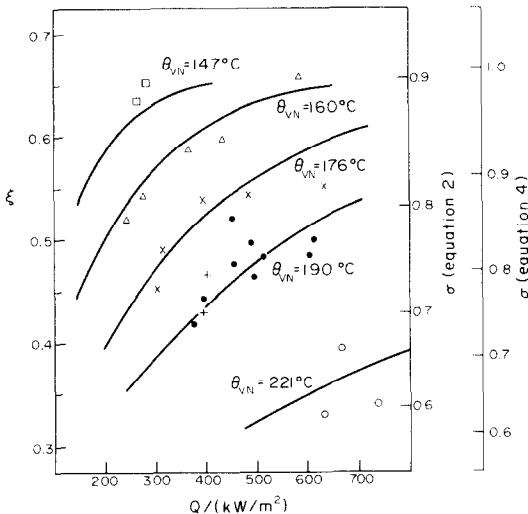


FIG. 7. Variation of ζ , σ (by equation 2) and σ (by equation 4) with heat flux and vapour temperature. The lines are those given by equation (9).

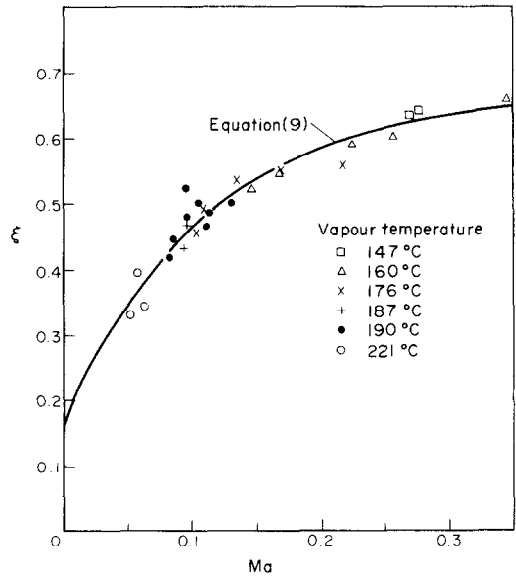


FIG. 8. Correlation of ζ with Ma .

values of T_s found in this way differ negligibly from those obtained directly from the simple Nusselt equation.

In Fig. 7, the values obtained for ζ are plotted against Q . It can be seen that ζ is not constant, as suggested by the foregoing theoretical results, but depends systematically both on the vapour temperature (or pressure) and Q . The highest values obtained are in fair agreement with the theoretical results while the lowest values are about half those suggested by theory.

To facilitate comparison with the measurements of other workers who have calculated "experimental values of the condensation coefficient σ ", subsidiary axes are included in Fig. 7. It should however be noted that values of σ determined from experimental measurements can be misleading since experimental error is disproportionately reflected in the values obtained for σ . If σ is in fact near unity, relatively large error in the experimentally determined small quantity, $P_t - P_s$,

Figure 9 compares the measurements with the relationship between the heat flux and the vapour-to-surface temperature difference given by equation (5) and the Nusselt theory. The theoretical lines are given for (i) $\zeta = 0.798$ (i.e. according to equation (2) with $\sigma = 1$), (ii) $\zeta = 0.627$ (i.e. according to equation (3)), (iii) $\zeta = 0.664$ (i.e. according to equation (4) with $\sigma = 1$) and (iv) using values of ζ given by equation (9).

Comparison with earlier work

Many of the earlier liquid metal condensation measurements suggested that the apparent condensation coefficient decreased with increasing vapour

* The values found (by minimizing the sum of squares of residuals of ζ) for the constants in equation (9) were 0.665, 0.490 and 8.94 respectively. The fit obtained with the rounded values given in equation (9) is not significantly less good than that with the actual least-squares values. The asymptotic value 0.665 is virtually the same as the value of ζ suggested by equation (4). This may be fortuitous.

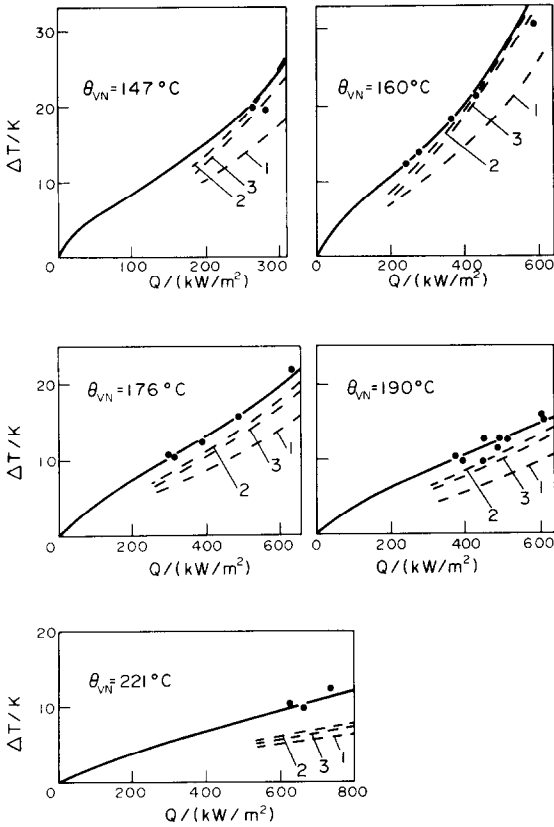


FIG. 9. Comparison of measurements with curves given by equation (5) plus Nusselt theory.
 ——— ζ from equation (9);
 - - - - - { 1. $\zeta = 0.798$ [i.e. using equation (2) with $\sigma = 1$],
 2. $\zeta = 0.627$ [i.e. using equation (3)],
 3. $\zeta = 0.664$ [i.e. using equation (4) with $\sigma = 1$].

pressure. This pressure-dependence, however, was much stronger than that found in the present work, and was most probably due to increased outgassing of the apparatus walls at the higher temperatures [3].

The vapour-to-wall temperature differences found by Misra and Bonilla [13] and Sukhatme and Rohsenow [14] for film condensation of mercury are generally much higher than those obtained in the present work. It is thought that the presence of non-condensing gases, in part arising from "out-gassing" of the walls of the apparatus at the higher temperatures, were the main cause of the higher values of vapour-to-wall temperature difference found in the earlier work. In view of the very high heat fluxes obtained in [13], it is possible that errors in the observed vapour temperature, attributable to vapour velocity effects, might also have played a part.

Ivanovskii *et al.* [12] condensed mercury vapour on a horizontal pool and measured the interface temperature drop directly. The maximum heat flux obtained was 26 kW/m^2 and the maximum vapour temperature used was 136°C , both much smaller than the corresponding minimum values in the present work. However, as a consequence of the high value of the vapour specific volume at the low temperatures, the range of Ma (0.04–0.28) overlaps the present range.

The data of Ivanovskii *et al.* when substituted in equation (5) yields values of ζ scattered in the approximate range 0.4–0.8. No systematic dependence of ζ on either vapour temperature or heat flux could be discerned. The scatter* of the results of Ivanovskii *et al.* stems largely from the small values of the interface temperature difference, encountered at the low heat fluxes used in this work.

Recent condensation measurements for sodium and potassium [2, 15, 16] indicate values of ζ similar to those for mercury [12]. Again the interface temperature differences (both predicted and observed) were significantly smaller than in the present work and the scatter correspondingly greater. No systematic dependence of ζ on either vapour temperature or heat flux is discernible.

CONCLUSION

The present measurements appear to deviate significantly and systematically from current theoretical expressions for interphase matter transfer when applied to condensation from a saturated vapour.

The fact that the deviations are smallest at the highest normal vapour flow Mach numbers and interphase pressure drops, i.e. where departure from equilibrium at the interface is greatest (see Table 2), is unexpected and indicates that the discrepancies are not attributable to linearization of the theoretical results. Values of ζ were determined from non-linearized results [1] and gave virtually the same values as those listed in Table 2.

The possibility, that the observed dependence of ζ both on pressure and heat flux might result from systematic error, is thought to be remote. In this respect the following may be noted:

- (a) An error of about 5 K in the vapour-to-surface temperature difference would be needed to raise the lowest value of ζ to about 0.6.
- (b) The maximum systematic error in T_w resulting from uncertainty in the thermocouple locations, estimated as indicated in [2], was 0.25 K.
- (c) The positions of the thermocouple holes in the block were re-checked when the test section was dismantled.
- (d) The heat flux found from the coolant measurements was in satisfactory agreement with that obtained from the temperature gradient in the block.
- (e) The fact that the calculated value of ζ depends predominantly on a measured temperature difference and a measured temperature gradient, together with the fact that the same thermocouple wire was used both in the vapour and condenser block, renders effects of calibration error minimal. Calculations using the wire manufacturer's nominal calibration yielded virtually the same results as those reported.
- (f) It would be necessary to multiply the calculated temperature drop across the condensate film by

*When the data of Ivanovskii *et al.* are used in equation (2) to determine σ , the values are found to lie in the range 0.7–1.0, i.e. the data are apparently less scattered when judged by values of σ as discussed above.

a factor of about 4 to explain the low values of ξ on this basis.

Finally, it is suggested that the scatter of other recent liquid metal condensation measurements, together with the fact that, in these earlier investigations, tests were not carried out over a range of values of heat flux at fixed vapour temperatures, might have rendered undetectable a dependence of "correction factor" (or apparent "condensation coefficient") on the condensation rate and vapour pressure, similar to that found in the present work.

New measurements are planned in which the accuracy will be further improved and the ranges of pressure and heat flux extended.

REFERENCES

1. R. W. Schrage, *A Theoretical Study of Interphase Mass Transfer*. Columbia University Press, New York (1953).
2. S. J. Wilcox and W. M. Rohsenow, Film condensation of potassium using copper condensing block for precise wall-temperature measurement, *J. Heat Transfer* **92C**, 359 (1970).
3. W. M. Rohsenow, Film condensation of liquid metals, Invited Lecture No. 1, International Seminar—Heat Transfer in Liquid Metals, International centre for heat and mass transfer, Yugoslavia (1971).
4. E. J. Le Fevre and J. W. Rose, An experimental study of heat transfer by dropwise condensation, *Int. J. Heat Mass Transfer* **8**, 1117 (1965).
5. S. Necmi, Ph.D. Thesis, Univ. London (1973).
6. R. Y. Kutcherov and L. E. Ricenglas, On hydrodynamic boundary conditions for evaporation and condensation, *J. Exp. Theor. Phys.* **36**, 125 (1959).
7. R. Y. Kutcherov and L. E. Ricenglas, On the question of measurement of the condensation coefficient, *Rep. Acad. Sci. USSR* **133**(5), 1130 (1960).
8. D. A. Labuntsov, An analysis of evaporation and condensation processes, *Teplofiz. Vysok. Temper.* **5**, 647 (1967).
9. T. H. Muratova and D. A. Labuntsov, Kinetic analysis of vaporization and condensation processes, *Teplofiz. Vysok. Temper.* **7**, 959 (1969).
10. D. A. Labuntsov and T. M. Muratova, Influence of motion on evaporation and condensation, *Teplofiz. Vysok. Temper.* **7**, 1146 (1969).
11. D. A. Labuntsov, Discussion remark, *Proc. Fourth Int. Heat Transfer Conference (Versailles)*, Vol. 10, p. 257. Elsevier, Amsterdam (1970).
12. M. N. Ivanovskii, Yu. V. Milovanov and V. I. Subbotin, Condensation coefficient of mercury vapour, *Atomn. Energ.* **24**(2), 146 (1968).
13. B. Misra and C. Bonilla, Heat transfer in the condensation of metal vapour: mercury and sodium up to atmospheric pressure, *Chem. Engng Prog. Symp. Ser.* **52**(18), 7 (1956).
14. S. P. Sukhatme and W. M. Rohsenow, Film condensation of a liquid metal vapour, *J. Heat Transfer* **88C**, 19 (1966).
15. N. V. Bakulin, M. N. Ivanovskii, V. P. Sorokin and V. I. Subbotin, Intensity of heat transfer in film condensation of pure sodium vapour, *Teplofiz. Vysok. Temper.* **5**(5), 930 (1967).
16. V. I. Subbotin, M. N. Ivanovskii, V. P. Sorokin and B. A. Chulkov, Heat transfer during the condensation of potassium vapour, *Teplofiz. Vysok. Temper.* **2**(4), 616 (1964).

APPENDIX A

Combination of the Interface and Condensate Thermal Resistances

Equation (5), with $T_v = T_{\text{sat}}(P_v)$ and $T_s = T_{\text{sat}}(P_s)$ gives an expression for the local condensate surface temperature in

terms of the local interphase mass flux. The simple Nusselt theory* of laminar film condensation, wherein T_s and T_w are taken to be independent of height x , gives an expression for T_s in terms of the local interphase mass flux and condensate properties:

$$T_s - T_w = \left\{ \frac{4\mu_f h_{fg}^3 x m^4}{\rho_f \rho_{fg} k_f^3 g} \right\}^{1/4}. \quad (\text{A1})$$

Since equation (A1) requires that T_s is independent of x , it is apparent that equations (A1) and (5) cannot hold simultaneously since they do not satisfy the condition of continuity of mass flux at the interface.

Two alternative solutions have been adopted. The Nusselt analysis may be re-worked, replacing the boundary condition— T_s independent of x —by equation (5) and the requirement of continuity of mass flux. Alternatively, the heat flux, rather than T_w may be taken to be independent of x . In this case equation (5) requires that T_s is essentially independent of x and the condensate problem can be treated separately.

It is not clear whether constant T_w or constant Q is the more appropriate in the present case. However, this is not serious since the values obtained for ξ were virtually the same in both cases.

Method (1): Constant interface heat-transfer coefficient

Consider T_w , T_v constant and $T_s = T_s(x)$. For small $P_v - P_s$ and $P_v = P_{\text{sat}}(T_v)$, $P_s = P_{\text{sat}}(T_s)$, we have

$$\frac{P_v - P_s}{T_v - T_s} \approx \left(\frac{dP}{dT} \right)_{\text{sat}} = \frac{1}{T} \frac{h_{fg}}{v_{fg}}. \quad (\text{A2})$$

Thus from equation (5), with $Q \approx mh_{fg}$ the interface heat-transfer coefficient α_i is given by

$$\alpha_i = \frac{Q}{T_v - T_s} = \xi \frac{h_{fg}^2}{v_{fg} T \sqrt{(RT_v)}} \quad (\text{A3})$$

i.e. α_i is essentially independent of height.† Then, as in the Nusselt theory, we have:

$$Q = \frac{gh_{fg}\rho_f\rho_{fg}}{\mu_f} \delta^2 \frac{d\delta}{dx} \quad (\text{A4})$$

and

$$T_s - T_w = \frac{Q}{k_f} \delta. \quad (\text{A5})$$

In this case T_s is a function of x , but

$$T_v - T_s = \frac{Q}{\alpha_i} \quad (\text{A6})$$

where α_i is essentially constant.

Eliminating T_s between equations (A5) and (A6) we have

$$T_v - T_w = Q \left\{ \frac{\delta}{k_f} + \frac{1}{\alpha_i} \right\}. \quad (\text{A7})$$

Eliminating Q between equations (A4) and (A7), integrating and using the condition $\delta = 0$ at $x = 0$ we have:

$$\left(\frac{\mu_f x}{gh_{fg}\rho_f\rho_{fg}} \right) (T_v - T_w) = \frac{\delta^4}{4k_f} + \frac{\delta^3}{3\alpha_i}. \quad (\text{A8})$$

Equations (A7) and (A8) may then be solved simultaneously, using the experimental values of Q , T_v , T_w and x and

*The maximum values of $c_{p,f}(T_s - T_w)/h_{fg}$ and the condensate film Reynolds number (at the bottom edge of the plate) were around 0.0015 and 450 respectively. Each of these values is within the range necessary to the validity of the Nusselt theory.

†The fact that ξ is found to depend on Q does not invalidate this statement since variation in ξ with height, associated with the height-dependence of Q , is negligible.

iterating so as to adopt appropriate mean values* of the properties, to determine α , and δ , and hence, from equation (A5) or (A6), the local value of T_s . Values of ξ are then obtained from equations (5) and (8).

*For those properties arising from equation (A2) arithmetic means of the values at T_w and T_s were used. For those properties arising from equations (A4) and (A5) the following values were used:

$$\begin{aligned} h_{fg} &= h_{fg}(T_s) \\ \rho_f &= \{\rho_f(T_w) + \rho_f(T_s)\}/2 \\ \rho_{fg} &= \{\rho_{fg}(T_w) + \rho_{fg}(T_s)\}/2 \\ k_f &= \{k_f(T_w) + k_f(T_s)\}/2 \\ \mu_f &= 3/\{2/\mu_f(T_w) + 1/\mu_f(T_s)\}. \end{aligned}$$

Method (2): Constant heat flux

Consider Q independent of x . Since $Q \approx mh_{fg}$ equation (5) implies that T_s is essentially independent of x . Equation (A4) may be integrated and the result used to eliminate δ from equation (A5) giving:

$$(T_s - T_w) = \left\{ \frac{3Q^4 x \mu_f}{g k_f^3 h_{fg} \rho_f \rho_{fg}} \right\}^{1/4}. \quad (\text{A9})$$

The experimentally determined local value of T_w may be in equation (A9) to obtain T_s and hence from equations (5) and (8) the value of ξ . The properties were evaluated as in method 1.

CONDENSATION EN FILM DU MERCURE

Résumé—On fait état des résultats de mesures des différences de température entre vapeur et surface ainsi que du flux thermique dans la condensation en film du mercure sur une surface plane verticale carrée (40 mm de côté) en cuivre plaqué nickel. Les thermocouples, implantés et espacés avec précision dans le bloc de cuivre ont servi à mesurer, par extrapolation, la température à l'interface cuivre-nickel et le flux thermique à partir du gradient de température. On a pris un soin particulier à vérifier que les résultats n'étaient pas altérés par la présence d'incondensables dans la vapeur. Les résultats ont une meilleure précision relative que les autres mesures récentes de transfert de chaleur par condensation de métaux, car les présentes observations ont été effectuées sous des conditions (métal utilisé, température de vapeur et taux de condensation) de différence plus élevée de température surface-vapeur. Les différences de température entre surface et vapeur observées sont nettement supérieures à celles données par la théorie de Nusselt de la condensation en film. Attribuant à l'interface liquide-vapeur la chute supplémentaire de température, les résultats ont été comparés aux expressions théoriques du transfert de matière entre phases. De même que dans d'autres travaux récents, les valeurs du facteur de correction (ou "coefficient de condensation apparent") ont varié depuis environ l'unité jusqu'à 0,6. La précision des présents résultats est suffisante pour révéler une influence sur le facteur de correction du taux de condensation et de la pression de vapeur. Tous deux ont pu être représentés de façon satisfaisante à l'aide d'une seule variable adimensionnelle.

FILMKONDENSATION VON QUECKSILBER

Zusammenfassung—Es wird über Messungen der Temperaturdifferenz zwischen Dampf und Oberfläche und der Wärmestromdichte bei der Filmkondensation von Quecksilber an einer quadratischen (40 × 40 mm), vertikalen, nickelbeschichteten Kupferplatte berichtet. Die Temperatur der Kupfer-Nickel-Grenzfläche wird durch Extrapolation der mit genauestens platzierten und über die Kupferplatte verteilten Thermoelemente gemessenen Werte bestimmt. Die Wärmestromdichte wird aus dem Temperaturgradienten ermittelt. Besondere Aufmerksamkeit wurde darauf verwendet, daß die Ergebnisse nicht durch die Anwesenheit nicht-kondensierbarer Gase im Quecksilberdampf beeinträchtigt werden. Die Ergebnisse weisen eine höhere relative Genauigkeit als früher ausgeführte Messungen der Kondensation von Metallen auf. Dies liegt an den Versuchsbedingungen (verwendetes Metall, Dampftemperatur und Kondensationsrate), bei denen die Temperaturdifferenz zwischen Dampf und Oberfläche größere Werte besitzt als bei früheren Untersuchungen. Die beobachteten Differenzen zwischen Dampf- und Oberflächentemperatur waren wesentlich höher als die aus der Nusselt-Theorie berechneten Werte. Durch Zuordnung des Exzesstemperaturabfalles in die Dampf-Flüssigkeit-Phasengrenzfläche wurden die Ergebnisse mit theoretischen Beziehungen für den Stoffaustausch an Phasengrenzflächen verglichen. Wie bei anderen Arbeiten schwanken die Werte für den Korrekturfaktor (oder scheinbarer "Kondensationskoeffizient") von nahezu 1 bis herab zu 0,6. Die Genauigkeit der vorliegenden Ergebnisse ermöglicht es, eine Abhängigkeit des Korrekturfaktors von der Kondensationsrate und vom Dampfdruck festzustellen. Es wurde festgestellt, daß beide Abhängigkeiten in befriedigender Weise mit Hilfe einer einzigen dimensionslosen Variablen erfaßt werden können.

ПЛЕНОЧНАЯ КОНДЕНСАЦИЯ РТУТИ

Аннотация—Приводятся результаты измерения разности температур между паром и стенкой, а также измерение теплового потока при пленочной конденсации ртути на вертикальной плоской квадратной медной пластине (со стороны 40 мм), покрытой никелем. Термопары, тщательно заделанные на определенном расстоянии в медном конденсационном блоке, использовались для измерения методом экстраполяции температуры на границе раздела медь-никель, а также теплового потока по температурному градиенту. Особое внимание уделялось тому, чтобы на точности результатов не сказались влияние имеющихся в паре неконденсирующихся газов. Полученные результаты являются более точными по сравнению

с данными по теплообмену при конденсации металлов, т. к. настоящие исследования проводились при более высокой разности температур между паром и поверхностью. В рассматриваемом случае разность температур была значительно больше той, которая дается теорией Нуссельта для пленочной конденсации. Если отнести это завышение за счет поверхности раздела пар-жидкость, то результаты можно сравнить с теоретическими выражениями для межфазного массообмена. Как и в другой ранней работе значения поправочного коэффициента (или кажущегося «коэффициента конденсации») изменяются примерно от 1 до 0,6. Благодаря точности полученных результатов установлена зависимость поправочного коэффициента от скорости конденсации, а также от давления пара. Найдено, что обе величины можно удовлетворительно обобщить одной безразмерной переменной.