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FORCED CONVECTION FILM CONDENSATION ON A HORIZONTAL TUBE WITH AND WITHOUT NON-CONDENSING GASES

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Abstract—Accurate and repeatable heat-transfer data have been obtained for filmwise condensation from pure vapours (steam and R113) and vapour–gas mixtures (steam–air, steam–hydrogen, R113–air and R113–hydrogen) flowing vertically downwards over a single horizontal tube. Surface temperatures were obtained from thermocouples embedded in the tube wall. The heat flux was obtained from coolant measurements which were checked against values obtained by collecting the condensate. The vapour mass flow rate was obtained from the electrical power input to the boiler. The approximate ranges of the variables used were: pressure (4–124 kPa), heat flux (12–455 kW m^{-2}), vapour velocity (0.3–26 m s^{-1}), gas mass fraction (0.02–32%). For pure vapours at low to moderate velocities, the mean vapour-side heat-transfer coefficients were in satisfactory agreement with earlier measurements and with theory. At the higher velocities obtained with steam, the coefficients are somewhat smaller than those predicted by theory incorporating the assumption of uniform wall temperature. For all four vapour–gas mixtures a simple boundary-layer theory-based approximate equation gives results in excellent agreement with the measured mass-transfer resistance. The present results for steam–air mixtures are in good agreement with earlier data.

NOMENCLATURE

$c_{P\mathrm{g}}$	isobaric specific heat capacity of gas
$c_{P\mathrm{L}}$	isobaric specific heat capacity of condensate
D	diffusion coefficient
d	tube diameter
F	dimensionless number, $gdh_{\mathrm{fg}}\mu_{\mathrm{L}}/(k_{\mathrm{L}}u_{\infty}^{2}\Delta T)$
G	dimensionless number, $k_{\mathrm{L}}\Delta T[\rho_{\mathrm{L}}/(\rho_{\mathrm{v}}\mu_{\mathrm{v}}\mu_{\mathrm{L}})]^{1/2}/h_{\mathrm{fg}}$
g	gravitational acceleration
h_{fg}	specific enthalpy of evaporation
k_{L}	thermal conductivity of condensate
M	molar mass
M_{g}	molar mass of gas
M_{v}	molar mass of vapour
m	mass flux of vapour at condensing surface
m_{g}	mass flow rate of gas
m_{v}	mass flow rate of vapour
m_1	condensation rate measured at exit from the auxiliary condenser
m_2	condensation rate on the walls of the vertical supply duct above the test condenser tube
Nu	Nusselt number, $qd/(k_{\mathrm{L}}\Delta T)$
P	pressure
P_{s}	saturation pressure
q	heat flux
q_{H}	power input to the boilers
q_1	heat loss defined in equation (2)
q_2	heat loss defined in equation (3)
Re	Reynolds number, $u_{\infty}d\rho/\mu$
$\tilde{Re}$	'two-phase Reynolds number', $u_{\infty}d\rho_{\mathrm{L}}/\mu_{\mathrm{L}}$
Sc	Schmidt number, $\mu/\rho D$
T	temperature
T_{a}	ambient temperature
T_{i}	vapour–liquid interfacial temperature
T_{w}	wall temperature
T_1	temperature of gas at entry to the boiler
T_2	temperature of the condensate return at entry to the boiler
T_{∞}	free-stream temperature
T^*	reference temperature, see equation (12)
u_{∞}	free-stream approach velocity
W	mass fraction of gas
W_{i}	value of W at the vapour–condensate interface
W_{∞}	free-stream value of W
X	dimensionless number defined in equation (11).

Greek symbols

μ	viscosity of mixture
μ_{L}	viscosity of condensate
μ_{v}	viscosity of vapour
ρ	density of mixture
ρ_{L}	density of condensate
ρ_{v}	density of vapour.

1. INTRODUCTION

SIGNIFICANT vapour velocity and the presence of non-condensing gases in a condensing vapour lead to deviations from the well-established Nusselt theory for condensation of essentially stationary and pure vapours [1, 2]. At relatively high vapour velocities, the effect of vapour drag leads to a thinning of the condensate film and a consequent increase in the surface heat-transfer coefficient. In contrast, non-condensing gases in the vapour can greatly impair the heat-transfer performance of a condenser tube. In the presence of such gases, even small vapour velocities, by limiting the build-up of gas near the condensing surface, may significantly increase the heat transfer.

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For pure vapours, many investigations have been undertaken in recent years to study the effect of vapour velocity on heat transfer for condensation on horizontal tubes [3–29]. Despite the fact that considerable progress has been made there remain several areas of uncertainty, even for the most straightforward case of a single tube with vertical vapour downflow. For condensation on horizontal tubes, a limited amount of experimental data exists for steam [4, 9, 13, 15, 16, 18–20, 23, 25], R21 [8, 17] and R113 [28]. For low to moderate vapour velocities the data are in fairly good agreement with theory [5, 9, 23, 24, 29]. At higher velocities there is evidence that theory, incorporating the assumption of uniform wall temperature, significantly overestimates the mean vapour-side heat-transfer coefficient.

Few reliable data exist for condensation from vapour–gas mixtures under well-defined conditions. Such measurements as are available for forced-convection condensation on single horizontal tubes are for steam–air mixtures only. For this case Berman [30] has given a correlation of the data obtained in the U.S.S.R. and Mills *et al.* [31] have made measurements at relatively low vapour velocities. Mills *et al.* [31] showed that a widely-used 'stagnant film' model [32] underestimated the mass-transfer coefficient, for transfer of vapour to the condensate surface, by a factor of almost two. An approximate equation due to Acrivos [33], based on boundary-layer analysis, was adjusted by Mills *et al.* to give a result which closely represented their data.

Using a somewhat different approach, also based on boundary-layer theory for flow with surface suction, Rose [34] obtained a relatively simple equation giving the vapour mass-transfer rate to the condensate surface (i.e. condensation rate) in terms of the bulk and interface conditions. This may be written as

$$md/\rho D = 0.5Re^{1/2} \times \{[1+2.28Sc^{1/3}(W_i - W_\infty)/W_\infty]^{1/2} - 1\}. \quad (1)$$

No experimental condensation data were used in the derivation of equation (1) which was found to be in excellent agreement with the steam–air correlation of Berman [30], the steam–air data of Mills *et al.* [31] and the steam–air results recently reported by Fujii and coworkers [26, 35].

One of the main objectives of the present work was to validate equation (1) by carrying out tests with other vapour–gas combinations and covering wide ranges of composition, vapour velocity and condensation rate. In particular, by selecting appropriate vapours and non-condensing gases, it has been possible to check the Schmidt number dependence indicated in equation (1).

2. APPARATUS AND PROCEDURE

Referring to Fig. 1, vapour was generated in three stainless steel boilers each fitted with two 5 kW electric immersion heaters. Non-condensing gas could be passed, via a flowmeter, to the base of the boilers. The vapour or vapour–gas mixture flowed vertically downwards via a calming section to the test section (diameter 152.4 mm) where the horizontal condenser tube was located. Vapour and condensate from the test condenser tube passed into the auxiliary condenser

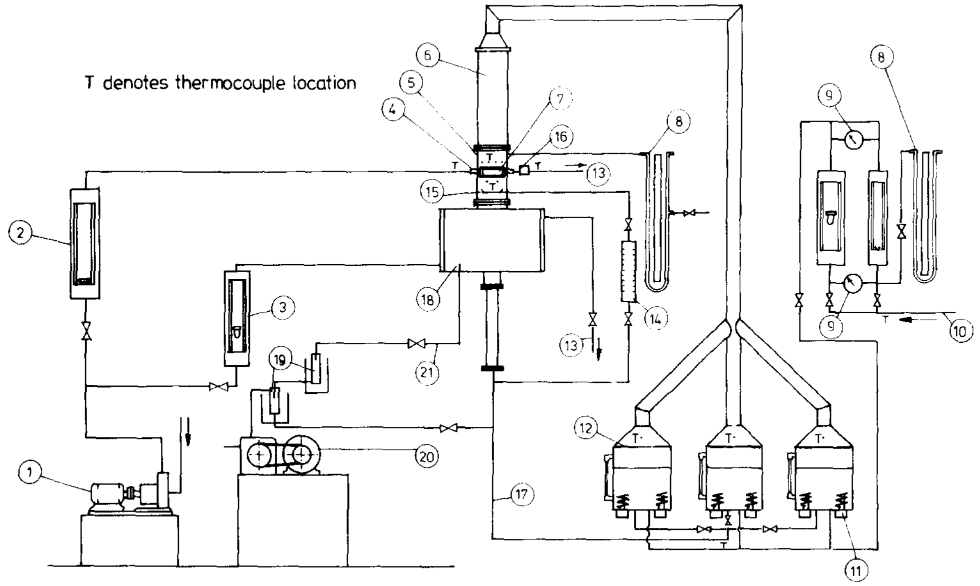

FIG. 1. Apparatus: (1) coolant pump, (2) flowmeter for condenser tube, (3) flowmeter for auxiliary condenser, (4) test condenser tube, (5) test section, (6) calming section, (7) window, (8) mercury manometer, (9) pressure gauge, (10) gas supply, (11) 5 kW immersion heater, (12) boiler, (13) coolant drain, (14) measuring cylinder, (15) condensate collecting tray, (16) mixing section, (17) condensate return line, (18) auxiliary condenser, (19) cold traps, (20) vacuum pump, (21) vent line.

located directly beneath the test section. The auxiliary condenser was vented, via two 'cold traps', by a vacuum pump and the condensate was returned by gravity to the boilers. The boilers, vapour supply duct and test section were thermally well insulated. The condenser tube could be viewed through a heated window. Special efforts were made to ensure that the apparatus was satisfactorily leak tight. The apparatus could readily be pumped down to a pressure of around 2 mmHg and the overnight pressure rise, when isolated, was less than 10 mmHg. The test condenser tube was cleaned prior to each day's tests. Frequent visual observations were made to ensure that film condensation prevailed.

Two copper tubes (I.D. 7.5 mm, O.D. 12.5 mm and I.D. 22.0 mm, O.D. 25.3 mm) were used. Each tube was fitted with four copper–constantan thermocouples which were located in longitudinally machined grooves (1.6 × 1.6 mm) and enclosed by tightly-fitting soldered copper strips. The junctions were orientated at 90° to each other and were located at different axial positions along the tube. The tube was oriented so that the thermocouple junctions were located at angles of 45° and 135° to the vertical on either side. The condenser tube was insulated from the body of the test section by ptfe bushes. The condensing (exposed) length was 109.5 mm.

The mean heat flux to the condenser tube was obtained from the coolant flow rate and the temperature rise. The coolant inlet and exit temperatures were measured by copper–constantan thermocouples, the exit temperature being measured after a well-insulated mixing section. The coolant channel was thermally well insulated at inlet to, and exit from, the test section. In preliminary tests, arrangements were incorporated to obtain a second estimate of the heat flux by collecting condensate over a measured time interval. The values so obtained agreed closely in all cases with those determined from coolant measurements, the largest discrepancy being around 5%.

The local tube surface temperatures were calculated from the observed wall temperatures using the mean heat flux to make a small correction for the depth of the thermocouples below the surface. The mean tube surface temperature was taken as the arithmetic mean of the four local surface temperatures. Data given in ref. [9] suggests that this gives a mean value within about 3% of that found when using eight thermocouples.

The vapour mass flow rate at the test section was obtained from the precisely-measured heater power input to the boilers by application of the steady flow energy balance for the condensing fluid and non-condensing gas streams. A correction for the heat transfer from the apparatus to the environment (i.e. 'thermal losses') was included. The losses were evaluated by preliminary measurements in which condensate was collected and compared with the vapour generation rate determined from the power input to the boilers. To separate the condensate on the walls of the vertical supply duct above the test condenser, due to heat transfer to the environment, from that resulting from condensation of the vapour reaching the test section, a trough was fitted to the test section wall and the condensate collected led out to a measuring cylinder. The thermal losses were obtained as follows:

$$q_1 = q_H - (m_1 + m_2)h_{fg}, \tag{2}$$

$$q_2 = m_2 h_{fg}, \tag{3}$$

where q_1 is the heat-transfer rate to the environment from the boilers, the vertical vapour supply duct above the boilers and the 'horizontal' upper duct which sloped slightly towards the boilers, q_2 is the heat-transfer rate to the environment from the vertical supply duct above the test condenser tube, q_H is the power input to the boilers, m_1 is the condensate flow rate measured at exit from the auxiliary condenser, and m_2 is the condensation rate on the walls of the vertical supply duct above the test condenser tube.

Loss evaluation tests were carried out with steam at atmospheric pressure for a range of boiler input powers. The results are shown in Fig. 2. The slight slope of the q_2 line is thought to be due to 'carry over' of the condensate on the walls of the horizontal supply duct caused by vapour drag which increases with increasing vapour velocity.

In the above tests the vapour and ambient temperatures were also recorded and when calculating vapour velocities in subsequent tests (including those at other (lower) pressures and with other fluids) the losses were evaluated by assuming linear dependence on temperature difference between vapour and environment (the insulation constitutes the major thermal resistance between the vapour and the environment).

It is considered that the thermal losses were certainly determined to better than 10% and since the lowest boiler input power used in subsequent tests was around seven times the total maximum losses, the consequent error in the calculated vapour velocity was less than 1.5% in the worst case.

During the main investigation, all of the condensate was returned continuously to the boilers and in some cases gas was injected into the boilers and removed

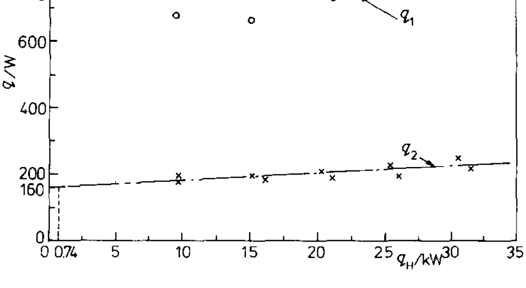

FIG. 2. Thermal losses from apparatus.

from the auxiliary condenser. A steady flow energy balance between the boiler inlets (condensate return and gas) and the test section (vapour, gas, condensate on walls resulting from losses, q_2) gives

$$m_v = \left\{q_H - q_1 - m_g c_{Pg}(T_\infty - T_1) - q_2\left(1 + \frac{c_{PL}(T_\infty - T_2)}{h_{fg}}\right)\right\} \bigg/ \{h_{fg} + c_{PL}(T_\infty - T_2)\}. \quad (4)$$

In equation (4), vapour removed by the vacuum pump is ignored. The consequent error in m_v is negligible.

The mean vapour velocity was calculated from the total mass flow rate (vapour and gas) and the mean density. A mean value over the exposed length of the test condenser tube was obtained from the mass flow rate using a seventh power velocity profile for turbulent flow.

The vapour temperature was measured by four copper–constantan thermocouples located approximately 85 mm above the test condenser tube. The temperatures indicated by the four thermocouples generally agreed to within 0.05 K. A pressure tapping to a mercury manometer, fitted with a vernier scale, was located in the test section at approximately 50 mm above the tube. The pressure was measured to within 0.1 mmHg.

In the pure vapour tests, the pressure and temperature corresponded to saturation values to within the precision of measurements which, in the worst case, amounted to an apparent air mass fraction of around 0.005. In the vapour–gas mixture tests the gas mass fraction was calculated by two methods:

(a) from the mass flow rates of the gas and the vapour

$$W_\infty = m_g/(m_v + m_g), \quad (5)$$

(b) from the pressure and temperature measurements in the test section assuming saturation conditions and the Gibbs–Dalton ideal-gas mixture equation

$$W_\infty = \frac{P_\infty - P_s(T_\infty)}{P_\infty - (1 - M_v/M_g)P_s(T_\infty)}. \quad (6)$$

Excellent agreement was found between the gas mass fraction given by equations (5) and (6), see Fig. 3; the difference being generally less than 0.005. As well as confirming the reliability of the composition measurements, this attests to the accuracy of the vapour flow rate determination and, in turn, that of the vapour velocity.

Special care was taken to ensure high-accuracy thermo-electric measurements (adequate isothermal immersion, lead-wire junctions at reference temperature, heavy draught-free copper switches, precision digital voltmeter).

Full details of the apparatus and calculation methods are given in ref. [27].

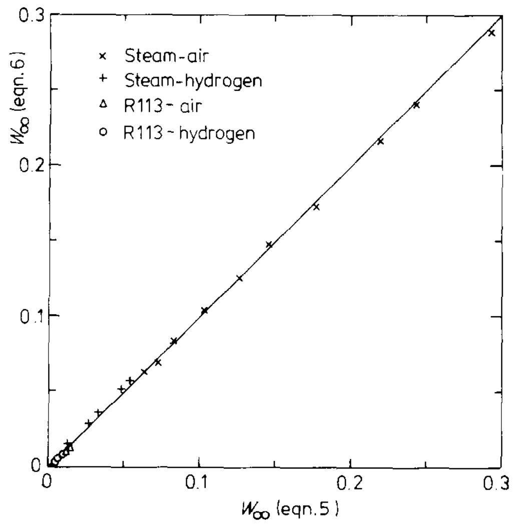

FIG. 3. Comparison of bulk composition determinations.

3. RESULTS AND DISCUSSION

Steam and R113 were used as condensing fluids. Refrigerant-113 was chosen since it is relatively non-toxic and its thermophysical properties are well documented and markedly different from those of steam. Air and hydrogen were used as non-condensing gases. With these vapour–gas combinations, the practically important steam–air case was covered and at the same time, a wide range of variables was permitted, to provide a stringent check on theory [34]. In particular, the range of Schmidt number for the vapour–gas mixtures was from about 0.05 to about 0.5.

3.1. *Pure vapours*

The measurements for pure vapours were made primarily for the purpose of obtaining reliable estimates of the condensate resistance when analysing the vapour–gas data. They are, however, briefly compared with theory and earlier data. As discussed in ref. [29] theoretical solutions of this problem based on the uniform wall heat flux assumption are unsatisfactory. Those based on uniform wall temperature [5, 9, 23, 24, 29] indicate that

$$Nu\,\tilde{Re}^{-1/2} = \psi(F, G), \quad (7)$$

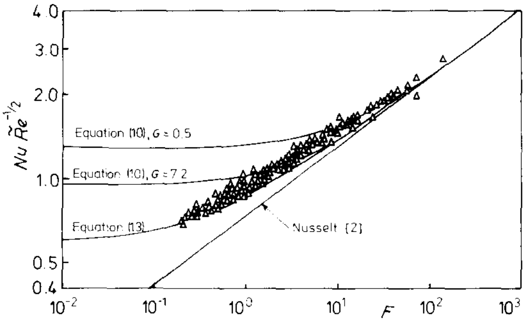

FIG. 4. Comparison of pure steam data with theory. Note: for the data $0.5 < G < 7.2$.

where

$$F = gd\mu_L h_{fg}/(k_L u_\infty^2 \Delta T), \tag{8}$$

$$G = k_L \Delta T(\rho_L/(\rho_v \mu_v \mu_L))^{1/2}/h_{fg}. \tag{9}$$

Effects of vapour boundary-layer separation are relatively unimportant in the calculation of the mean heat-transfer coefficient since most of the heat transfer takes place on the forward part of the tube prior to the separation point. A solution by Fujii *et al.* [9], using an integral treatment of the vapour boundary layer with an assumed velocity profile which avoided the separation problem, led to the result

$$Nu\,\tilde{Re}^{-1/2} = X(1+0.276F/X^4)^{1/4}, \tag{10}$$

where

$$X = 0.9(1+1/G)^{1/3}. \tag{11}$$

Later solutions [23, 24, 29] which included prediction of vapour boundary-layer separation, gave mean Nusselt numbers not greatly different from those given by equation (10).

The present results for steam and R113 are compared with equation (10) in Figs. 4 and 5. When evaluating the experimental results, the condensate properties (except for h_{fg} which was taken at T_i) were determined at the reference temperature

$$T^* = \tfrac{2}{3}T_w + \tfrac{1}{3}T_i, \tag{12}$$

where $T_i = T_\infty$ for pure vapours. The condensation rate was taken as q/h_{fg}. For F greater than about one, the agreement with theory is generally satisfactory. It may be noted that, for both fluids, the data with lower values of G generally had higher values of $Nu\,\tilde{Re}^{-1/2}$ as indicated by theory. At the lower values of F (higher vapour velocities) obtained with steam, the experimentally determined Nusselt numbers are significantly lower than those given by equation (10). A conservative solution by Shekriladze and Gomelauri [5], wherein the asymptotic vapour shear stress (infinite condensation rate) was used, was further conservatively-modified by neglecting all heat transfer beyond the earliest separation point for the vapour boundary layer at 82°. The result was adjusted [36] to blend with the Nusselt theory at low velocity whence the final result is given by

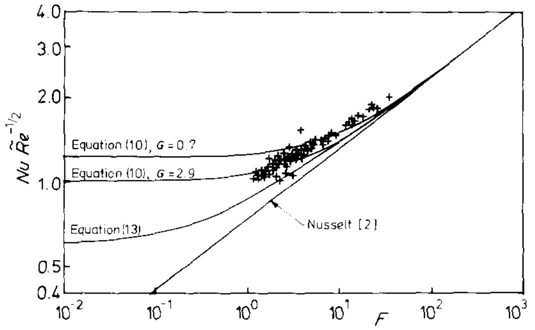

FIG. 5. Comparison of pure R113 data with theory. Note: for the data $0.7 < G < 2.9$.

$$Nu\,\tilde{Re}^{-1/2} = 0.416\{1+(1+9.47F)^{1/2}\}^{1/2}. \tag{13}$$

Equation (13) is also included in Figs. 4 and 5 and, as may be seen, is conservative with respect to virtually all of the data.

In Figs. 6 and 7 the present results are compared with earlier experimental data. As may be seen, the present results are rather less scattered than the earlier data and generally lie near the centre of the overall spread.

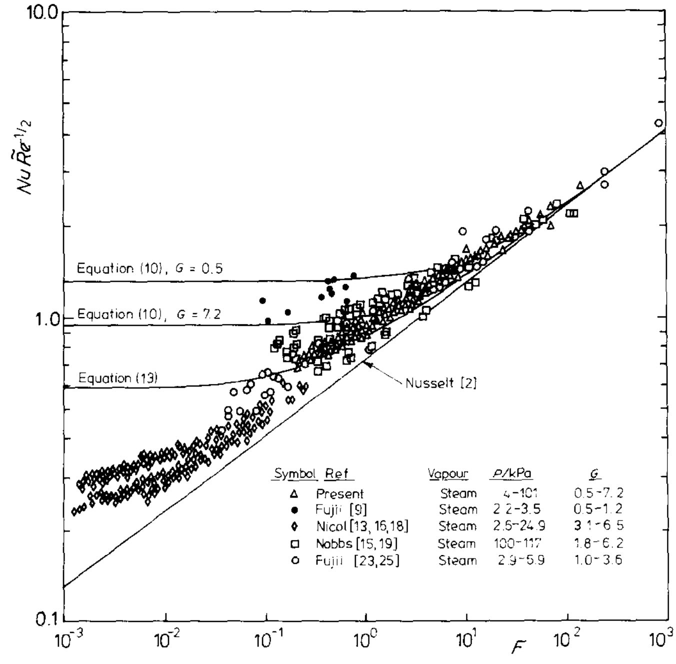

FIG. 6. Comparison of the present and earlier data for steam.

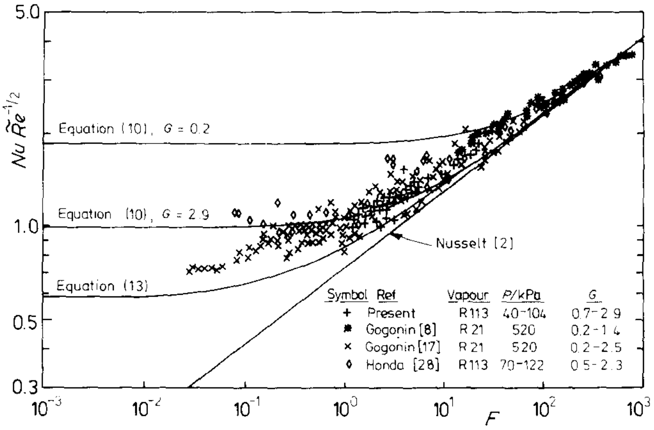

FIG. 7. Comparison of the present and earlier data for refrigerants.

Considering the data as a whole it may be seen that equation (13) is conservative with the exception of the very high velocity data set for steam [13, 16, 18, 20]. With the exception of these data equation (10) is generally satisfactory, though there is some evidence, more particularly in the case of steam, to suggest that this may overestimate the mean Nusselt number when F is less than unity.

In general the above comparisons suggest rather better agreement with theory for refrigerants than for steam. This may be associated with the fact that the uniform wall temperature approximation is better when the condensate resistance dominates than when the coolant side is controlling. In the latter case an overall vapour-to-coolant solution [26, 46] may be satisfactory for predicting the heat-transfer rate but is less suitable for detailed study of the condensate film since predictions for the vapour side are sensitive to values adopted for the coolant-side heat-transfer coefficient.

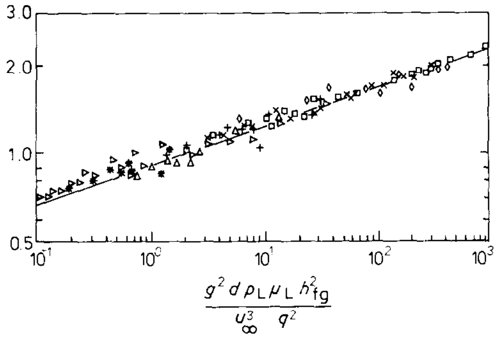

FIG. 8. Comparison of present pure vapour data with equation (14).

	Fluid	d (mm)	P (kPa)
□	Steam	12.5	101
△	Steam	12.5	47
▷	Steam	12.5	5–9
◇	Steam	25.3	101
*	Steam	25.3	4–8
×	R113	12.5	101–105
†	R113	12.5	40–78

3.2. *Vapour–gas mixtures*

In order to evaluate the results for the vapour–gas boundary layer, it is necessary to determine the temperature (and hence the composition) at the vapour–condensate interface, i.e. we require an equation relating the temperature drop across the condensate film to the heat flux. As seen above, equation (10) would not be wholly satisfactory for this purpose. It may be seen from Figs. 4 and 5 that the present data for steam and R113 lie essentially on the same line and are in good agreement with the data for steam of Fujii *et al.* [23, 25] (see Fig. 6). In Fig. 8 the present pure vapour data are compared with a correlation given by Fujii *et al.* [23] (based on steam data only)

$$Nu\,\tilde{Re}^{-1/2} = 0.91(g^2 d\rho_L\mu_L h_{fg}^2/(u_\infty^3 q^2))^{0.13}. \quad (14)$$

It is clear that equation (14) represents the present pure vapour data satisfactorily. The ranges of vapour velocity, heat flux and pressure for the pure vapour observations covered those of the vapour–gas data.

When analysing the vapour–gas data, equation (14) was used with the observed heat flux and mean wall temperature to determine the temperature at the vapour–condensate interface. An iterative procedure was used so as to take the condensate properties at T^* (except h_{fg} which was taken at T_i). The vapour–gas composition at the interface was then found from the equilibrium condition [equation (6) with P_s taken at T_i]. In order to make comparisons with theory, the relevant mean properties of the vapour–gas mixture were evaluated as follows: density was evaluated on the basis of an ideal-gas mixture and viscosity by the method of Wilke [37]; arithmetic means of the values at, and remote from, the interface were adopted. The diffusion coefficient was evaluated at $(T_\infty + T_i)/2$ using the methods given in refs. [38–40]. The vapour mass flux m was taken as q/h_{fg}.

The present results are compared with equation (1) in Figs. 9 and 10. It may be seen that the data are smooth,

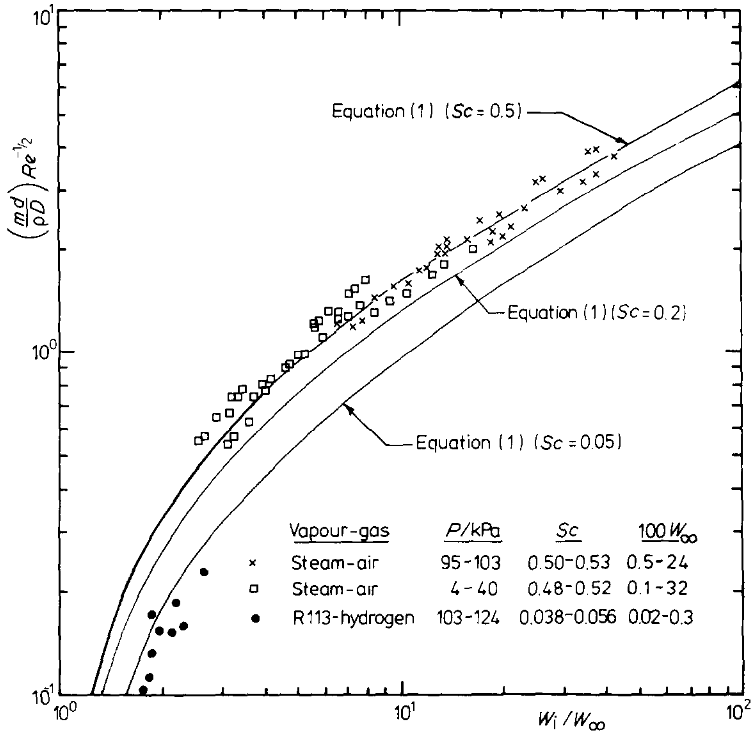

FIG. 9. Comparison of present steam–air and R113–hydrogen data with equation (1).

consistent and in good agreement with the theoretical results. In particular the data exhibit a clear Schmidt number dependence, in line with theory, and the steam–hydrogen results are in close agreement with those for R113–air which have similar values of Schmidt number.

In Fig. 11 the present steam–air data are seen to be in good agreement with earlier results of Mills *et al.* [31]

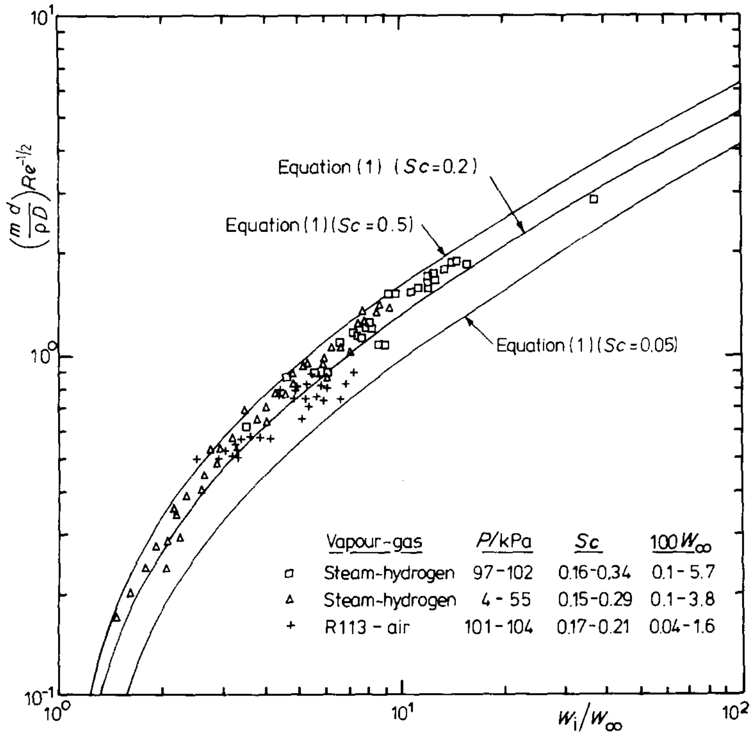

FIG. 10. Comparison of present steam–hydrogen and R113–air data with equation (1).

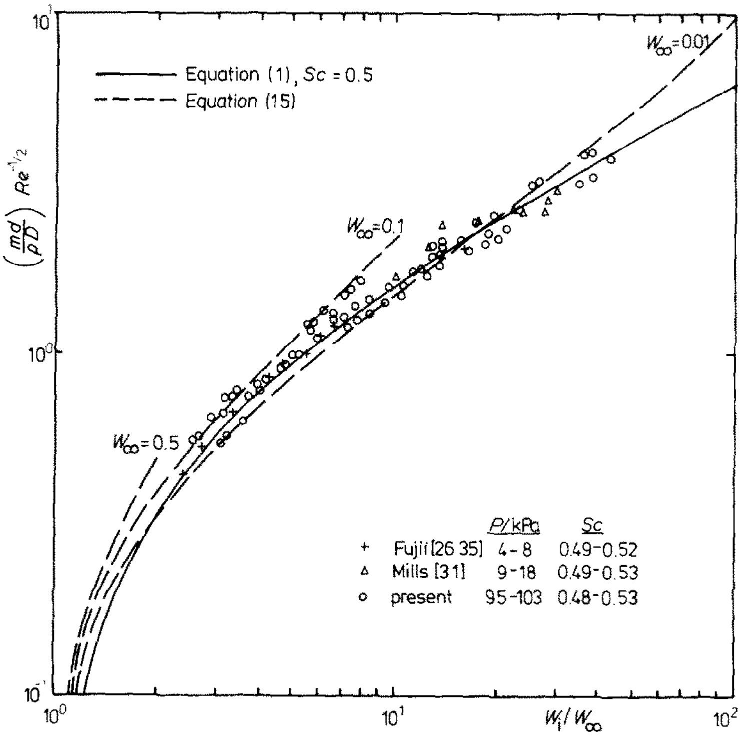

FIG. 11. Comparison of present and earlier steam–air data.

and Fujii [26]. A correlation of Berman [30] of steam–air data has been shown by Rose [34] to be expressible in the form

$$md/\rho D = 0.455 Re^{1/2} \frac{(W_i - W_\infty)^{2/3}(1-0.378W_\infty)^{0.933}}{W_\infty^{0.6}(1-0.378W_i)^{2/3}}. \tag{15}$$

Equation (15) is also shown in Fig. 11 for three values of W_∞ and, as may be seen, agrees quite well with the other data and equation (1).

Calculations were also performed using the method indicated by Mills *et al.* [31] and using the 'film model' [41–43]. In the former case the results were found to be in good agreement with the present data for all vapour–gas combinations. In accordance with the findings of Mills *et al.* [31], mass-transfer rates calculated on the basis of the film model were significantly lower than those observed in the present investigation, the discrepancy increasing with increasing values of W_i/W_∞ (i.e. with increasing condensation rate). More detailed discussions of these comparisons are given in refs. [44, 45].

4. CONCLUDING REMARKS

At moderate velocities the present pure vapour data, for both steam and R113 were in general agreement with theory [equation (10)] incorporating the assumption of uniform wall temperature. At the high velocity end of the steam data, mean Nusselt numbers lower than those indicated by equation (10) were found. Similar results have been found by other workers. These discrepancies may be due to non-uniformity of wall temperature. The present data for pure steam and R113 are correlated well by equation (14).

Equation (1), which gives the gas-phase mass-transfer resistance (i.e. relates the vapour flux at the condensate surface to the interface and remote vapour conditions) is in excellent agreement with the present data for four vapour–gas combinations and for wide ranges of bulk composition, velocity, pressure and heat flux. Equation (1) also agrees well with earlier data for steam–air mixtures and an alternative boundary-layer theory-based calculation method [31, 33]. Calculations based on the 'film model' significantly over-estimated the measured mass-transfer resistance, the discrepancy between calculated and measured results increasing with increasing condensation rate.

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CONDENSATION EN FILM AVEC CONVECTION FORCEE SUR UN TUBE HORIZONTAL, AVEC OU SANS GAZ INCONDENSABLE

Résumé—Des mesures précises et reproductibles sur le transfert thermique sont effectuées pour la condensation en film de vapeurs pures (vapeur d'eau et R113) et de mélanges de gaz–vapeur (air–vapeur d'eau, hydrogène–vapeur d'eau, air–R113 et hydrogène–R113) qui s'écoulent verticalement vers le bas sur un tube unique horizontal. Les températures de surface sont obtenues par des thermocouples noyés dans la paroi du tube. Les flux de chaleur sont déterminés par les mesures qui sont vérifiées avec les valeurs du débit de condensat. Le débit massique de la vapeur est obtenu à partir de la puissance électrique fournie au bouilleur. Les domaines approximatifs des variables sont: pression (4–124 kPa), flux surfacique (12–455 kW m^{-2}), vitesse de la vapeur (0,3–26 m s^{-1}), fraction massique du gaz (0,02–32%). Pour les vapeurs pures à vitesse faible ou modérée, les coefficients moyens de tranfert thermique sont en accord satisfaisants avec des mesures antérieures et avec la théorie. Aux vitesses plus élevées obtenues avec la vapeur d'eau, les coefficients sont plus faibles que ceux prédits par la théorie faite sur l'hypothèse de température pariétale uniforme. Pour tous les quatre mélanges vapeur–gaz, une simple théorie de couche limite avec équation approchée donne des résultats en excellent accord avec la résistance au transfert massique mesurée. Les présents résultats pour les mélanges vapeur d'eau–air sont en bon accord avec des essais antérieurs.

FILMKONDENSATION BEI ERZWUNGENER KONVEKTION AN EINEM HORIZONTALEN ROHR MIT UND OHNE NICHTKONDENSIERBARE GASE

Zusammenfassung—Für die Filmkondensation reiner Dämpfe (Wasser und R113) und Dampf–Gas-Gemische (Wasserdampf–Luft, Wasserdampf–Wasserstoff, R113–Luft und R113–Wasserstoff), die an einem horizontalen Rohr abwärts strömten, wurden genaue und reproduzierbare Wärmeübergangswerte gemessen. Die Oberflächentemperaturen wurden mit Thermoelementen gemessen, die in die Rohrwand eingebettet waren. Der Wärmestrom wurde aus den Temperaturänderungen des Kühlmediums berechnet und mit Werten überprüft, die aus der angefallenen Kondensatmenge ermittelt wurden. Der Dampfmassenstrom wurde aus der elektrischen Leistungsaufnahme des Verdampfers bestimmt. Die ungefähren Parameterbereiche der Untersuchung waren: Druck (4–124 kPa), Wärmestromdichte (12–455 kW m^{-2}), Dampfgeschwindigkeit (0,3–26 m s^{-1}), Gasmassenanteil (0,02–32%). Bei reinen Dämpfen mit niedrigen bis mittleren Geschwindigkeiten ergaben sich dampfseitige mittlere Wärmeübergangskoeffizienten in befriedigender Übereinstimmung mit früheren Messungen und der Theorie. Bei höheren Dampfgeschwindigkeiten wurden mit Wasserdampf etwas kleinere Koeffizienten im Vergleich zur Theorie bei Annahme einheitlicher Wandtemperatur gemessen. Bei allen vier Dampf–Gas-Gemischen ergab eine einfache, auf der Grenzschichttheorie basierende Näherungsgleichung Ergebnisse, die in ausgezeichneter Übereinstimmung mit dem gemessenen Stoffübergangswiderstand standen. Die Ergebnisse aus dieser Arbeit für Wasserdampf–Luft-Gemische stimmen gut mit früheren Meßwerten überein.

ПЛЕНОЧНАЯ КОНДЕНСАЦИЯ НА ГОРИЗОНТАЛЬНОЙ ТРУБЕ В СЛУЧАЕ ВЫНУЖДЕННОЙ КОНВЕКЦИИ ПРИ НАЛИЧИИ НЕКОНДЕНСИРУЮЩИХСЯ ГАЗОВ И БЕЗ НИХ

Аннотация—Получены точные и воспроизводимые данные по теплопереносу при пленочной конденсации чистых газов (пар и фреон 113) и парогазовых смесей (пар–воздух, пар–водород, фреон-113–воздух и фреон-113–водород), стекающих вертикально вниз по горизонтальной трубе. Температура поверхности измерялась термопарами, заделанными в стенку трубы. Величина теплового потока определялась по результатам измерений потока охладителя, и эти данные сопоставлялись с количеством образующегося конденсата. Массовая скорость потока пара определялась по подводимой электрической мощности. Исследования проводились в следующих диапазонах параметров: давления — 4–124 кПа, величины теплового потока — 12–435 кВтм$^{-2}$, скорости пара — 0,3–26 м сек, массовой доли газа — 0,02–32%. В случае чистых паров, когда скорость изменяется от малой величины до умеренной, средние значения коэффициентов теплопереноса со стороны пара удовлетворительно согласовывались с данными ранее проведенных измерений и результатами расчетов. При более высоких скоростях, имеющих место в случае водяного пара, значения коэффициентов оказались несколько ниже рассчитанных с учетом допущения об однородности температуры стенки. Для всех четырех парогазовых смесей простое приближенное уравнение, основанное на теории пограничного слоя, дает результаты, хорошо согласующиеся с измеренными значениями сопротивления массопереносу. Полученные результаты для паровоздушных смесей хорошо согласуются с ранее полученными данными.