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# CONDENSATION OF WET STEAM ON A HORIZONTAL TUBE WITH ADDITIONAL DROPLET DEPOSITION

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**Abstract**—A theory is presented for condensation of downward flowing wet vapour on a horizontal tube. The vapour is assumed to consist of dry saturated vapour and uniformly distributed liquid droplets flowing independently of each other. In addition the droplets are assumed to be so large that they fall vertically on to the tube surface and are unaffected by the vapour flow around the cylinder. The results show that the heat transfer coefficients are extremely dependent on both the droplet mass flux and velocity as well as the steam velocity.

## INTRODUCTION

A particularly difficult problem in the design of condensers is estimating the effect of drainage from the upper tubes on the condensing heat transfer coefficient. It is likely that in the upper tube rows drainage takes place in the form of large discrete drops while lower down the tube bundle the drainage distribution resembles a heavy spray. An associated problem is the estimation of the heat transfer coefficient at inlet to a condenser for a wet, rather than dry saturated, varpour. The liquid content exists as droplets which may vary in size depending on the expansion in the turbine and on whether any large drops from the turbine blades have been entrained. If the droplets are very small, then they can be assumed to flow with the dry vapour and the condensation heat transfer coefficient can be calculated using a modified enthalpy of evaporation term. However if the droplets are large they will flow independently of the vapour and tend to separate on to the upper half of the tubes, while the vapour flows around and condenses on the entire surface. Heat transfer coefficients will therefore be reduced because of the thickening of the film on the upper surface due to droplet deposition which takes place without contributing to the heat transfer through the condensate film.

A theory is developed in this paper for condensation of a vapour flowing vertically downward, with droplet separation, which could be used in cases where the droplet size is such that the droplets flow independently of the dry vapour. This includes condensation of a very wet vapour, or even the case of condensation with inundation from tubes above, provided the droplets weren't so large as to significantly disturb the surface of the film on impact.

## THEORY

The theory introduced here is an extension of that used by Nicol & Wallace (1974, 1976), in their studies on the influence of vapour velocity on condensation. The equations referred to in those papers are modified to take account of the additional mass of liquid, calculated from the mass flux of droplets, $\dot{m}$, being deposited on the upper part of the tube, of inner radius $r_i$, and outer radius $r_0$. Furthermore, if the droplet inundation is heavy or if the droplet velocity, $U_D$, is high, it is likely that a significant amount of momentum could be added to the condensate film and so a term has been included in the force balance to allow for this. As in Nicol & Wallace (1974, 1976), the vapour shear stress, $\tau_v$, unmodified by the effect of mass transfer, is included and so the contribution to the drag on the liquid film from the momentum of the condensing vapour is neglected. This approach has been shown to be valid only at low condensation rates and to underestimate the heat transfer coefficient for high condensation rates. A more recent paper by Fujii *et al.* (1979) demonstrates the use of a shear function taken from the work of

Truckenbrodt (1956) which takes account of suction on the interfacial shear stress. The use of the simpler and easier to handle shear stress function in the present case is justified on the ground that any error in the vapour shear term is considerably reduced when the additional influence of the droplet terms is taken into account.

A force balance on the element of the condensate film shown in figure 1 gives,

$$\tau r_0 \mathrm{d}\phi = g(\rho_L - \rho_V)(\delta - y)\sin\phi r_0 \mathrm{d}\phi + \tau_V r_0 \mathrm{d}\phi + \dot{m} r_0 \mathrm{d}\phi \cos\phi U_D \sin\phi \qquad [1]$$

where $\tau$ is the shear stress at distance $y$ from the tube surface, $\delta$ is the liquid film thickness, $\phi$ is the angle measured from the leading edge of the cylinder, $g$ is the gravitational acceleration, $\rho_L$ is the liquid density and $\rho_V$ is the vapour density. If the film is assumed to flow laminarly despite the bombardment of droplets, then $\tau = \mu_L(\mathrm{d}u/\mathrm{d}y)$, where $\mu_L$ is the liquid viscosity and $\mathrm{d}u/\mathrm{d}y$ the velocity gradient, and the mass flow rate per unit width, $\Gamma$, is given by

$$\Gamma = \rho_L \int_0^\delta u \,\mathrm{d}y$$

$$= \frac{\rho_L g(\rho_L - \rho_V)\sin\phi\delta^3}{3\mu_L} \frac{\rho_L \tau \delta^2}{2\mu_L} + \frac{\rho_L \dot{m}\, U_D \cos\phi \sin\phi\delta^2}{2\mu_L} \qquad [2]$$

from which $\mathrm{d}\Gamma$ as a function of $\mathrm{d}\phi$ and $\mathrm{d}\delta$ can be obtained. The increase in mass flow rate per unit width, $\mathrm{d}\Gamma$, can also be obtained from the heat transferred through the film due to condensation and the additional mass from the capture of droplets, i.e.

$$\mathrm{d}\Gamma = \frac{k_L(T_s - T_{w,\phi}) r_0 \,\mathrm{d}\phi}{\delta h_{fg^*}} + \dot{m} r_0 \,\mathrm{d}\phi \cos\phi. \qquad [3]$$

The first term in this equation is related to the mass of vapour condensed, and $k_L$ is the liquid thermal conductivity, $T_s$ is the vapour temperature, $T_{w,\phi}$ is the local wall temperature, and $h_{fg^*}$ is a modified latent heat term. The second term is the additional mass flow from the droplets

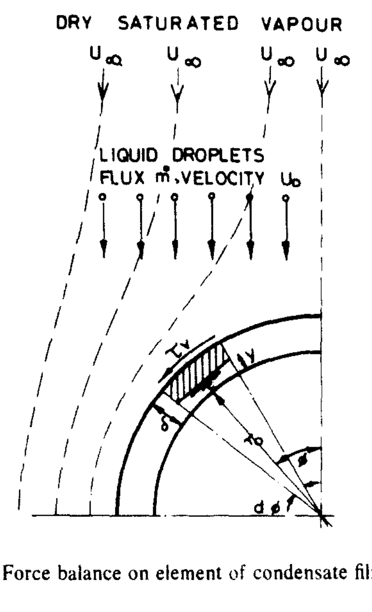


Figure 1. Force balance on element of condensate film.

based on the droplet flux and the projected area. The derivative of the film thickness with respect to angle can therefore be obtained as

$$\frac{d\delta}{d\phi}=\frac{E-\delta^4\cos\phi-\frac{3}{2}\rho_L c\delta^3\frac{d\tau_V}{d\phi}+3\mu_L\dot{m}r_0\cos\phi c\delta-\frac{3}{2}c\rho_L\dot{m}\,U_D\cos 2\phi\delta^3}{3\sin\phi\delta^3+3c\rho\tau_V\delta^2+\frac{3}{2}c\dot{m}\,U_D\sin 2\phi\delta^2} \quad [4]$$

where

$$E=\frac{3k_L(T_s-T_{w,\phi})r_0\mu_L}{gh_{fg^*}\rho_L(\rho_L-\rho_V)} \quad [5]$$

and

$$c=\frac{1}{g\rho_L(\rho_L-\rho_V)}. \quad [6]$$

It has been shown experimentally, Wallace (1975), that with high vapour velocities the tube wall temperature varies considerably and better agreement with experiment has been obtained by Bryce (1977) and Nicol *et al.* (1978), using modified version of [4] to allow for the variation in wall temperature. By introducing the coolant resistance and tube wall resistance the temperature difference in the term, $E$, in [4] becomes

$$T_s-T_{w,\phi}=\frac{T_s-T_c}{1+\frac{k_L r_0}{\delta}\left[\frac{\ln r_0/r_i}{k_w}+\frac{1}{h_c r_i}\right]} \quad [7]$$

where $T_c$ is the coolant temperature, $h_c$ is the coolant heat transfer coefficient and $k_w$ is the tube thermal conductivity. When this temperature difference is used in [4] the solution is referred to as either "variable wall temperature" or "non-isothermal" as distinct from the constant wall temperature solution. Equation [4] for both isothermal and non-isothermal conditions was integrated using a Runge–Kutta–Merson procedure with the starting thickness calculated from the initial condition $(d\delta/d\phi)=0$ at $\phi=0$. The shear force distribution used was the first five terms of the Blasius power series, given in Schlichting (1968).

$$\tau=\frac{\frac{1}{2}\rho U_\infty^2}{\sqrt{\frac{U_\infty r_0}{\nu}}}\left\{6.973\left(\frac{x}{r_0}\right)-2.732\left(\frac{x}{r_0}\right)^3+0.292\left(\frac{x}{r_0}\right)^5-0.018\left(\frac{x}{r_0}\right)^7+0.000043\left(\frac{x}{r_0}\right)^9\right\} \quad [8]$$

where $U_\infty$ is the vapour velocity, $\nu$ is the vapour kinematic viscosity and $x$ is the distance from the leading edge. Although this equation predicts a fixed vapour point unaffected by condensation but approximately in agreement with separation observed for similar suction rates, this defect was not considered to be serious as the main influence on the condensation rate was in the droplet terms.

For purposes of comparison with experiment, all calculations have been made assuming tube dimensions of $r_i=0.00825$ m, $r_0=0.00952$ m and $k_w=101$ W/mK.

## RESULTS AND DISCUSSION

Since most condensation literature is based on constant wall temperature conditions, [4] was first solved to assess the effect of the drainage term only, on the film thickness and heat transfer coefficients. However, for comparison with some of the experimental work, where the heat transfer coefficients were either calculated or deduced using a water side heat transfer coefficient, results were also obtained using the modified version of [4] for variable wall temperature.

Figure 2 shows the variation of film thickness around the tube for vapour velocities 5, 10 and 20 m/s, and drainage mass fluxes of 0 to 5.0 (kg/m$^2$s). The vapour-wall temperature difference ($T_s - T_w$) is 10°C and the vapour pressure $p$ is 0.2 bar. The droplet momentum term has been excluded here so that the effect of the added mass alone could be investigated. The film thickness clearly increases with increased drainage in all three cases, while the effect of increasing vapour velocity or vapour shear is to reduce the film thickness over the upper half of the tube. From the vapour separation point at 108° until 180° the film is assumed to flow under Nusselt conditions, i.e. no shear effect, and for the higher vapour velocities the sudden increase in film thickness because of the reduced film velocity is evident. Figure 3 shows the corresponding trends with the introduction of the momentum term. A droplet velocity of 10 m/s has been selected and the relative influence of droplet momentum and vapour velocity can be discerned. Comparison with the previous figure shows that the droplet momentum acts as an additional shear force term on the film and further reduces the thickness of the film on the upper half of the cylinder—and in fact for the unlikely case of a droplet velocity of 10 m/s and steam velocity 5 m/s the film thickness is reduced to that corresponding to zero drainage.

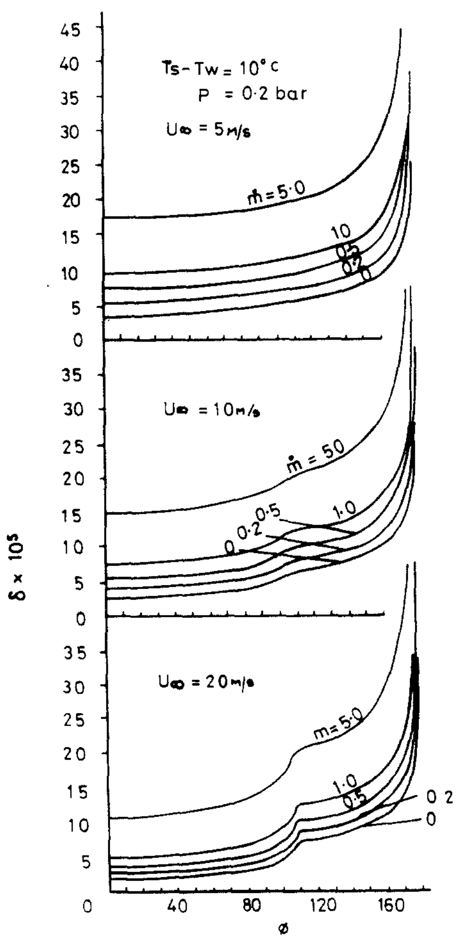


Figure 2. Effect of drainage on film thickness, excluding momentum term.

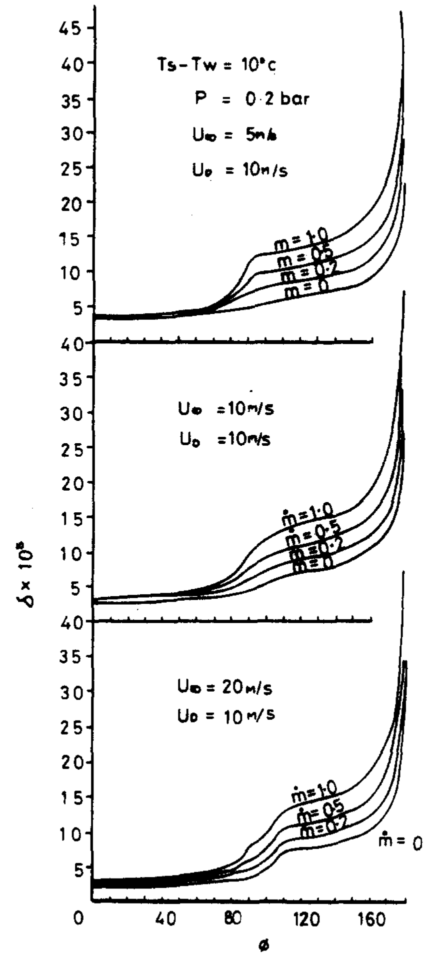


Figure 3. Effect of drainage momentum on film thickness.

However the additional mass flowing with a much higher velocity on the upper half of the tube is apparent in the greater film thickness after the momentum influence ceases at 90°. In the case of the curves for a steam velocity of 20 m/s, the thickening of the film at 90° and also 108° is seen as both the momentum and shear effects cease. From figures 2 and 3 therefore, it can be concluded that while the inundation of droplets on the upper part of the tube increases the thickness, this thickness is substantially reduced if the momentum term is included. However, the film thickness after vapour separation at 180° is considerably greater than for the case of vapour shear alone.

Figure 4 shows the variation in average heat transfer coefficient, $h_{AV}$, with vapour velocity for a range of condensate drainage rates. As expected the average heat transfer coefficient decreases markedly with increased drainage over the complete velocity range, mainly because of the increased thickness of the condensate film on the upper part of the tube. The effect of the momentum term is shown in figure 5 for the case of $\dot{m} = 0.5\ \mathrm{kg/m^2s}$, and the increase in heat transfer coefficient with both vapour velocity and droplet velocity (droplet momentum) is apparent. The same trend could be expected with any other droplet mass flux.

In figure 6, average heat transfer coefficients have been calculated for the case where the droplets travel at the same velocity as the vapour enabling the results to be plotted in terms of constant dryness fraction. As the dryness fraction decreases the heat transfer coefficient also decreases because of the additional liquid content in the wet vapour. This figure can therefore

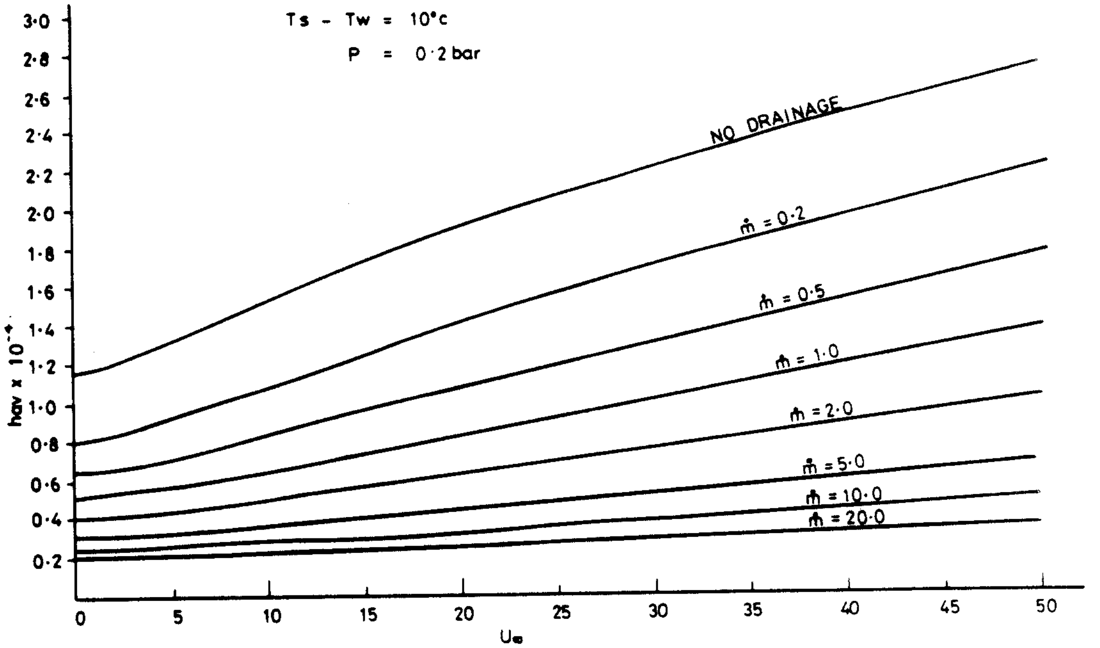


Figure 4. Effect of drainage on average heat transfer coefficient excluding momentum contribution.

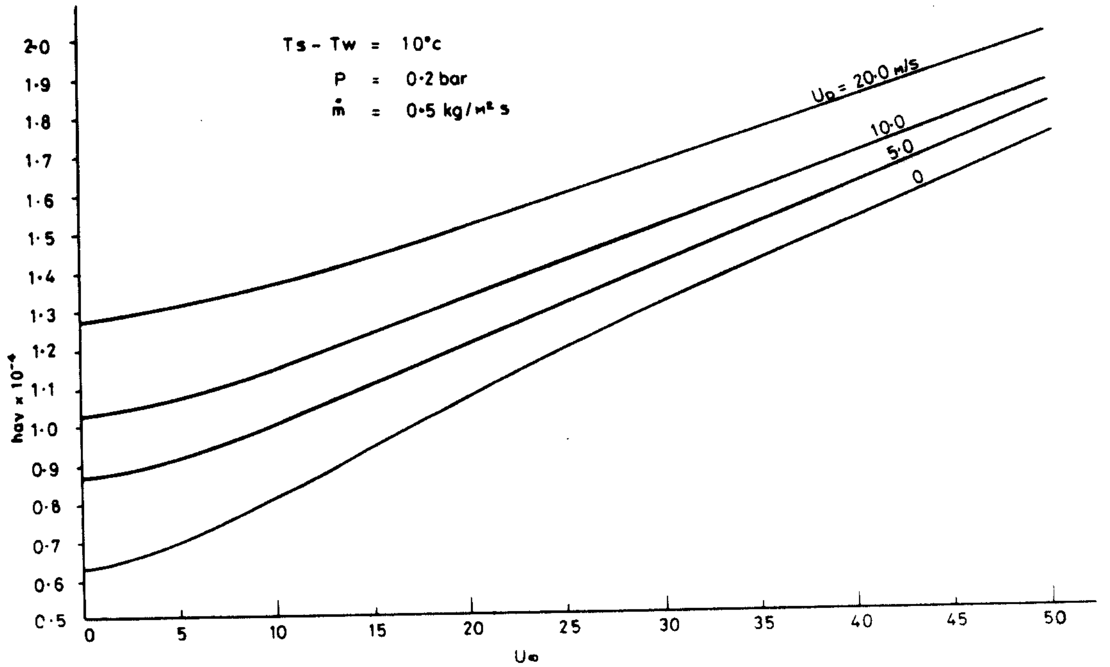


Figure 5. Effect of drainage, including momentum on average heat transfer coefficient.

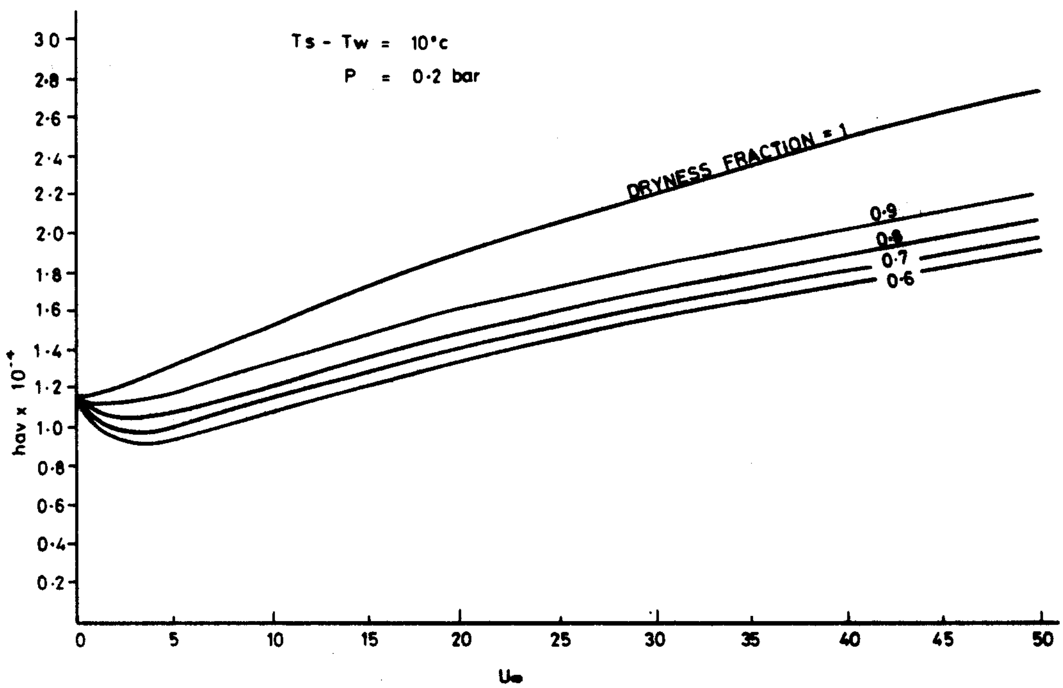


Figure 6. Influence of dryness fraction on the average heat transfer coefficient (droplets travel with the same velocity as the vapour).

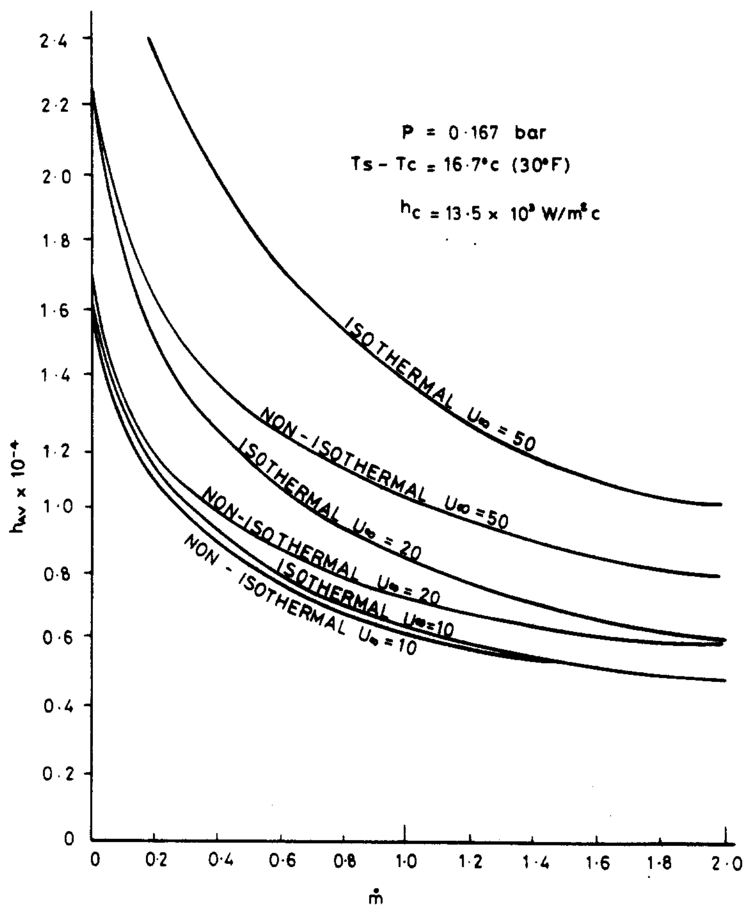


Figure 7. Comparison of isothermal and non-isothermal heat transfer coefficients (without momentum).

be used to determine the heat transfer coefficient for the condensation of a wet vapour, when the droplet size is large enough to cause them to separate on the tube in the manner prescribed by the theory. Because of this restriction to vertical impingement only, a curiosity is observed at zero steam velocity. There, the droplets cease to flow thus eliminating the droplet contribution in [4] and consequently all the curves converge to the Nusselt solution for dry vapour. For the case of very small droplets following the general flow pattern of the vapour around the tube, the heat transfer coefficient would be determined by replacing the $h_{fg*}$ term by the product of the dryness fraction and $h_{fg*}$ and excluding the terms pertaining to the droplets.

The results discussed so far have referred to condensation on an isothermal cylinder, whereas when the vapour velocity is high it has been shown Fujii *et al.* (1972) and Nicol *et al.* (1978) that the wall temperature variation around the cylinder is considerable. In figure 7 results for both isothermal and non-isothermal cases are compared for the same overall temperature difference of 16.7°C (30°F). The isothermal heat transfer coefficients are much greater than those of the non-isothermal case for all velocities, with the difference increasing with vapour velocity because of the reduced local heat flux at the leading edge resulting from the smaller temperature difference between the vapour and the tube wall in this region. A fuller account of non-isothermal condensation is given in Bryce (1977) where local heat fluxes are plotted for different vapour velocities. Figure 8 shows the influence of $(T_s - T_c)$ for three vapour velocities. For the larger drainage rates the effect of $(T_s - T_c)$ is insignificant but at lower drainage rates the trend to higher heat transfer coefficients for smaller values of temperature difference is consistent with the earlier work of Wallace (1975) and Bryce (1976).

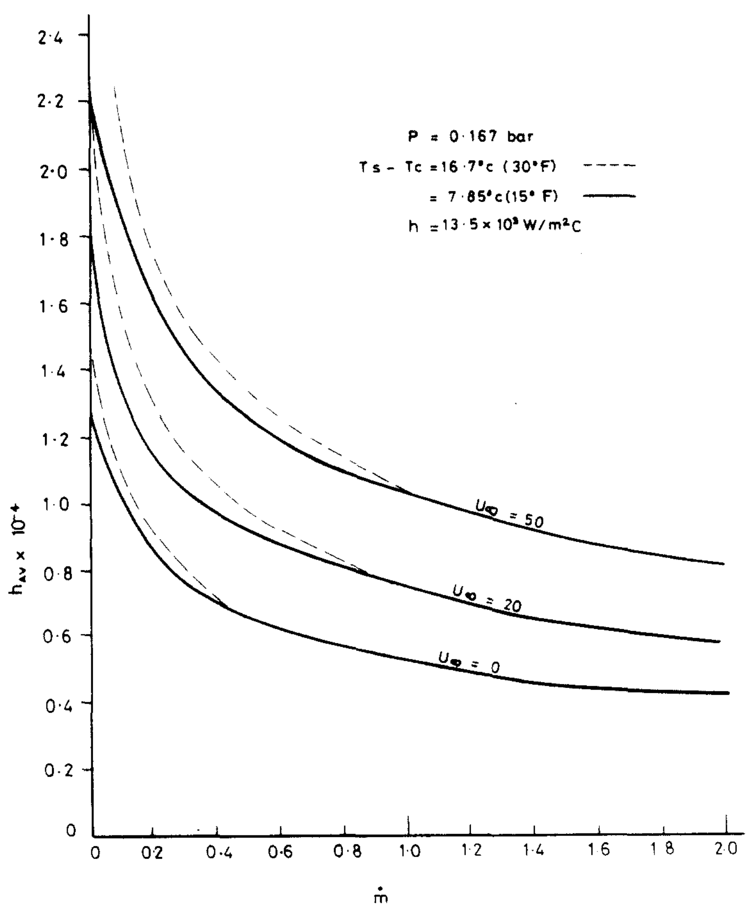


Figure 8. Effect of temperature difference on the average heat transfer coefficient (without momentum).

Finally figure 9 shows a comparison between the present non-isothermal theory and the experimental work of Grant & Osment (1968). This figure is essentially figure 4 from their report with the present theoretical results for two vapour velocities of 10 m/s and 20 m/s superimposed on it. Two possible droplet velocities with associated momentum are also considered for each vapour velocity. From the tables in Grant & Osment (1968) it is difficult to determine the vapour velocity for each individual result but inlet velocities of up to 20 m/s would appear to be implied by the condensation rates quoted. Depending on the influence of the vapour drag on the condensate droplets as they fall from the drainage tube the droplet velocity could lie between almost zero and the velocity of the steam. Thus it can be seen that the present theory for droplet and steam velocities of 10–20 m/s is reasonable consistent with the experimental results. This comparison is not completely justified because the drainage from the upper tubes in the Grant & Osment (1968) experiments is likely to have been in the form of much larger droplets than assumed in the present theory, although the drainage rates are the same. Thus the comparison is mainly of a qualitative nature as is the result predicted by the Nusselt equations for ideal drainage also shown in figure 9. The Nusselt equation is considered to underestimate the heat transfer, because of the idealised flow pattern of the condensate at the leading edge of the tube, giving a thicker film that would exist in practice at that point. The present theory is also likely to underestimate the heat transfer, because it neglects any disturbance or waviness in the film, that might result from the impact of the droplets on the

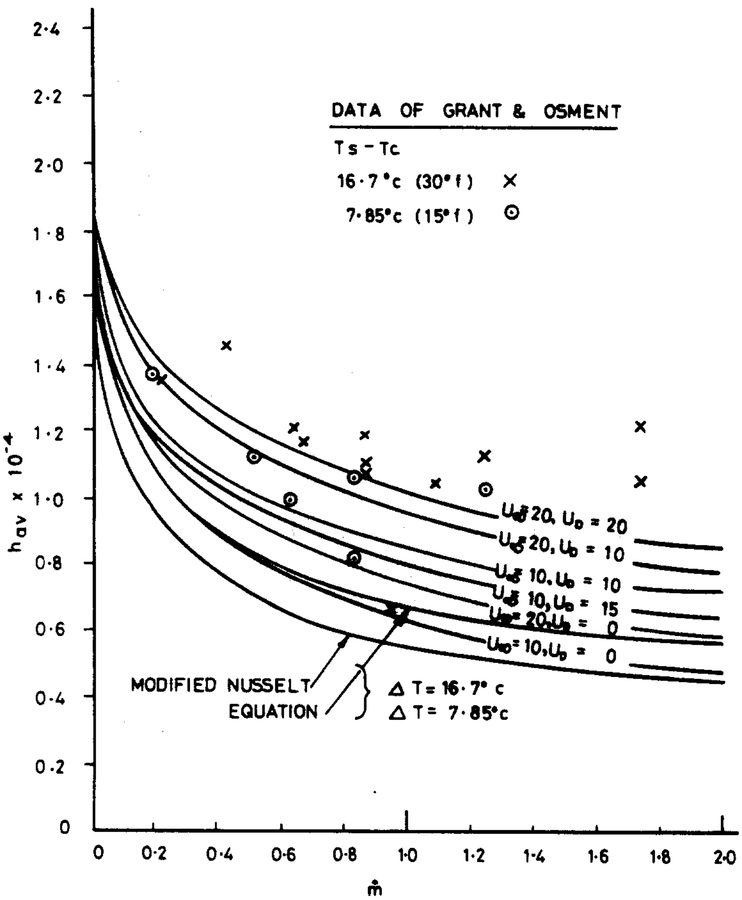


Figure 9. Comparison between theories and the data of Grant and Osment.

surface. While it is possible that small droplets could be absorbed by the film without affecting the flow, it is well known that large drops cause considerable waviness. However, if this limitation in the theory is accepted, it is thought that the present theory takes account more realistically of the kind of drainage that is likely to exist in certain regions of condenser tube bundles, where the condensate is shed as a mist, rather than in a continuous sheet or in very large drops.

## CONCLUSIONS

A theory for the condensation of downward flowing dry saturated steam in the presence of a uniformly distributed mist of liquid droplets has been presented. A range of droplet velocities has been considered so that the theory can be applied to (a) condensation of wet steam of a given dryness, i.e. where the droplets travel with the same velocity as the given vapour, and (b) condensation of steam in the presence of drainage where the droplet velocity is not necessarily that of the vapour.

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