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Effects of non-condensable gas on laminar film condensation in a vertical tube

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Abstract—This paper develops a theory to reveal the effect of small amounts of non-condensable gas on laminar filmwise condensation of a vapour–gas mixture flowing turbulently in a vertical tube. The reductions in heat transfer due to the non-condensable gas are found to be more significant at low pressures and at low Reynolds numbers of the mixture. Comparisons of the theory with some experimental data reported in the literature are in good agreement.

INTRODUCTION

THE ANNULAR filmwise condensation of vapours inside a vertical tube is an important process in the chemical and power industries, and has been extensively studied [1–8]. The pioneering work of Nusselt has been modified recently to include both the effect of interfacial shear stress on condensate flow and the characteristic of vapour velocity diminishing along the length of the tube [4–8]. All these investigations were concerned with condensation of pure saturated vapours. However, in practical operations of tube condensers, small amounts of non-condensable gas may exist in working vapours due to the leakage of the system or dissolution of working vapours. By the investigations regarding condensation in unconfined spaces, such as on flat plates or outside horizontal tubes [9–12], it has been well established that the existence of non-condensable gas in vapours can greatly reduce condensation heat transfer and deteriorate the performance of condensers. Thus, predicting the effects of non-condensable gas on annular filmwise condensation of vapours in a vertical tube seems to be of important technical and theoretical interest.

For condensation in tubes, vapour flow decelerates and non-condensable gas in vapours is concentrated more and more along condensing surfaces of tubes. These factors precluded the application of the theoretical results obtained by investigating effects of non-condensable gas on condensation in unconfined spaces to the inside tube case. Therefore, a new analysis which should consider these two factors simultaneously is needed to be performed for condensation of a vapour–gas mixture in tubes. Although the condensation of vapours from a vapour–gas mixture in a tube has been studied historically [13–15], the main interest of these studies is not to detect the effects of non-condensable gas on condensation heat transfer, but to predict heat and mass transfer coefficients in the gas phase. Related to the former case, the search of literature has revealed that no work has ever been published to indicate how remarkable reductions in condensation heat transfer in tubes are induced by the presence of a small amount of non-condensable gas, except for a few rough experimental data [16]. Thus the topic, as pointed out by Westwater [17], 'is wide open for study'. The aim of this work is to present a theoretical model to predict the effect of non-condensable gas on condensation heat transfer in a vertical tube with turbulent vapour–gas mixture flow, while in the previous paper [18] laminar mixture flow was dealt with.

At small temperature driving forces and low vapour velocities, laminar film condensation occurs, and Nusselt's assumptions [1] for laminar film condensation in a vertical tube hold. The theory developed here is based on the analogy between the condensation process and the convective transport with boundary suction. By means of the known results for hydrodynamics and convective diffusion of turbulent flow in porous tubes, and the well-known assumptions adopted by Nusselt for filmwise condensation, the theoretical treatments for fluid and vapour diffusion in a gas core as well as heat transfer in a condensate film are combined together, then the effect of non-condensable gas on local characteristics of heat transfer in a vertical tube is numerically predicted.

THE PHYSICAL MODEL

Consider steady, laminar, filmwise condensation of a vapour–gas mixture in a vertical tube with downflow. The mixture entering the tube is saturated and flows with a fully developed profile. The tube surface temperature is kept constant. Figure 1 illustrates such a condensation problem schematically.

According to Nusselt's assumptions, the equation of motion in the condensate layer is given by

$$\mu_l \frac{\partial^2 u_l}{\partial y^2} = \frac{\mathrm{d}p}{\mathrm{d}z} - \rho_l g \tag{1}$$

NOMENCLATURE

D	inner diameter of tube	z	axial distance.
D_H	hydraulic diameter of tube		
f	friction factor	Greek symbols	
g	gravity	δ	condensate film thickness
k	thermal conductivity	μ	viscosity
l	length of tube	ν	kinetic viscosity
L	latent heat	ρ	density
Nu	Nusselt number, hD/k	τ	shear stress
p	pressure	ϕ_F	correction coefficient for shear stress
p_v	vapour partial pressure	ϕ_a	correction coefficient for mass transfer
q	heat flux	ω	see equation (18).
Q	flow rate		
R	radius of tube	Subscripts	
Re	Reynolds number	c	condensate
Sc	Schmidt number	i	interface of liquid–gas
Sh	Sherwood number	l	liquid
T	temperature	m	vapour–gas mixture
u	velocity of liquid film in z-direction	s	saturated
v	velocity normal to surface	v	vapour
W	mass fraction of non-condensable gas	w	wall
x	vapour quality	0	condensation of pure vapour or single phase flow.
y	coordinate, see Fig. 1		

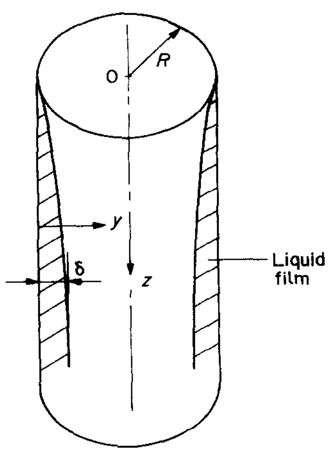

FIG. 1. Filmwise condensation in a vertical tube.

where p is the local pressure. Assuming the no slip condition at the tube wall surface and a velocity gradient of τ_i/μ_l at the gas–liquid interface $y = \delta$, integrating equation (1), yields

$$u_l = \frac{1}{\mu_l}\left(\frac{dp}{dz} - \rho_l g\right)\left(\tfrac{1}{2}y^2 - \delta y\right) + \frac{\tau_i}{\mu_l} y \tag{2}$$

where τ_i is the interface shear stress in the flow direction at $y = \delta$. τ_i is related to the axial pressure gradient by

$$-\frac{dp}{dz} = \frac{4\tau_i}{D_H} + \frac{d(\rho_m u_m^2)}{dz} \tag{3}$$

if the body force on the mixture is neglected ($\rho_m \ll \rho_l$), and the hydraulic diameter, D_H, for the vapour–gas mixture flow is given by

$$D_H = D - 2\delta. \tag{4}$$

Equation (3) can be easily derived from the cross-sectional averaged momentum equation on mixture gas flow, and the second term on the right-hand side represents the pressure recovery from the vapour deceleration as condensation proceeds along the tube wall.

The interfacial shear stress τ_i is calculated based on the local mixture velocity

$$\tau_i = \tfrac{1}{2}\rho_m u_m^2 f \tag{5}$$

where f is the gas phase friction factor at the liquid–gas interface. It should be noted that the friction factor f will be affected by the condensation process at the liquid–gas interface which behaves like the surface suction of mass. As proposed by Mickley [19]

$$f = \frac{\phi_F \exp(\phi_F)}{\exp(\phi_F) - 1} f_0 \tag{6}$$

where

$$\phi_F = \frac{v_w/u_m}{f_0/2} \tag{7}$$

and f_0 is the gas phase friction factor for an impermeable tube wall, and given by

$$f_0 = 0.079 Re^{-1/4}. \tag{8}$$

The liquid flow rate is obtained by integrating the liquid velocity field over the liquid cross-sectional area. Thus

$$Q_l = 2\pi \int_0^{\delta} \rho_l (R-y) u_l \, dy$$

or

$$Q_l = 2\pi \left\{ \frac{\tau_i}{\nu_l} \left(\tfrac{1}{2}R\delta^2 - \tfrac{1}{3}\delta^3\right) + \frac{1}{\nu_l}\left(-\frac{dp}{dz} + \rho_l g\right) \times \left(\tfrac{1}{3}R\delta^3 - \tfrac{5}{24}\delta^4\right) \right\}. \quad (9)$$

The local heat flux is given by Nusselt's solution, i.e.

$$q = \frac{k}{\delta}(T_i - T_w) \quad (10)$$

and the local film condensate rate over the whole wall is obtained by the energy balance at the interface

$$Q_c = 2\pi R q / L = 2\pi R k (T_i - T_w)/L\delta \quad (11)$$

where the sensible heat transferred from the mixture core is neglected.

A mass balance in the axial direction reads

$$\frac{dQ_l}{dz} = Q_c \quad (12)$$

or

$$\frac{\tau_i}{\nu_l}\left(\tfrac{1}{2}R\delta^2 - \tfrac{1}{3}\delta^3\right) + \frac{1}{\nu_l}\left(-\frac{dP}{dz} + \rho_l g\right)\left(\tfrac{1}{3}R\delta^3 - \tfrac{5}{24}\delta^4\right) = \frac{1}{2\pi} Q_l|_z + \frac{Rk(T_i - T_w)}{L\delta}\Delta z. \quad (13)$$

Once T_i is known, equation (13) can be solved with the boundary condition $Q_l = 0$ at $z = 0$ (mixture inlet). The calculation proceeds in a stepwise manner by first evaluating the film thickness δ for $z = \Delta z$ which, in turn, is used to calculate the local liquid rate from equation (9). The local vapour flow rate is then corrected by accounting for the condensation rate. The new local values of Q_v, D_H, dp/dz and τ_i are then evaluated for the preparation for the next calculation step.

Where only condensation of pure vapours occurs $T_i = T_s$. In the presence of non-condensable gas in the vapour core, the interface temperature will be lower than the saturated temperature T_s of the mixture corresponding to the local pressure, due to the resistance of vapour diffusion in the gas phase. So the determination of interfacial temperature T_i relies on the diffusion process of non-condensable gas towards the interface in the mixture core, and the mass transfer coefficient along the interface should be known before the calculation for condensation heat transfer of the vapour–gas mixture. Like the friction factor, the mass transfer coefficient Sh is also influenced by vapour condensation occurring at the liquid–gas interface, and therefore an equation similar to equation (6) is suggested here to account for this effect of condensation mass flux

$$Sh = \frac{\phi_a \exp(\phi_a)}{\exp(\phi_a) - 1} Sh_0 \quad (14)$$

where Sh_0 is the single phase mass convection coefficient in turbulent tube flow without wall suction, and its value is given by the Dittus–Boelter equation [21]

$$Sh_0 = 0.023 Re^{0.8} Sc^{0.4}. \quad (15)$$

The mathematical expression for Sh/Sh_0 in equation (14) corrects the convective mass transfer for the effect of mass transfer to the vapour–liquid interface, where the quantity ϕ_a is defined as

$$\phi_a = Re_w Sc / Sh_0. \quad (16)$$

Such a form of correction was first derived by Ackermann [20] to account for the effect of mass transfer on heat transfer, and recently has been successfully used in the studies of the condensation process of mixture flowing over a flat plate, on a horizontal tube or in a vertical tube with laminar vapour flow [12, 18].

For turbulent vapour–gas flow, a comparison between the empirical correlation (6) and numerical results in ref. [22] for the friction factor is shown in

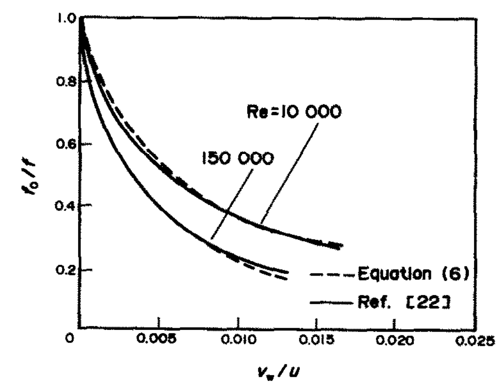

FIG. 2. Comparison of gas phase friction factor, equation (6), with the numerical results of Kinney and Sparrow [22].

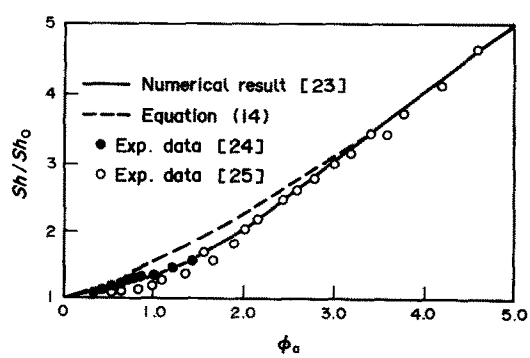

FIG. 3. Comparison of convective mass transfer, equation (14), with the numerical results of Yeroshenko *et al.* [23] and the experimental data of refs. [24, 25].

Fig. 2. Also, equation (14) is compared with numerical as well as experimental results in refs. [23–25] in Fig. 3. The agreement is good enough for the present analysis of the condensation process in a tube with turbulent vapour–gas mixture flow, because the value of ϕ_a is likely larger than 2.5 for the case of condensation in the presence of small amounts of non-condensable gas. The interface is considered as impermeable for a non-condensable gas, and hence

$$Sh = Re_w\, Sc/(1-\omega) \tag{17}$$

where

$$\omega = W_\infty / W_i \tag{18}$$

is the ratio of non-condensable gas content in the core to that at the interface.

Combining equations (14) and (16), one obtains

$$Sh = Sh_0(-\ln\omega)/(1-\omega) \tag{19}$$

$$Re_w = Sh_0(-\ln\omega)/Sc \tag{20}$$

where Re_w is defined as $v_w D/v_m$, and the condensation film rate Q_c can be related with Re_w as

$$Q_c = \pi\mu_m\, Sh_0(-\ln\omega)/Sc. \tag{21}$$

Either W_i or W_∞ is calculated by the Gibbs–Dalton relation for a perfect mixture, i.e.

$$W = \frac{1-p_v/p}{1-p_v/p(1-M_v/M_g)} \tag{22}$$

where p_v represents the vapour partial pressure, and is available from the tabulated properties of the vapour corresponding to local temperature, and p is the total pressure. The molecular weights of vapour and non-condensable gas are M_v and M_g, respectively. Hence, W_i is a function of interface temperature T_i.

In each step Δz of the axial distance, there are only two unknowns, film thickness δ and interface temperature T_i. Using equations (13), (21) and (22), they could be calculated by the above-mentioned solution procedure with the Newton–Raphson iterative method. Once T_i and δ have been found, the local heat flux q is determined by equation (10), and the Nusselt number gives

$$Nu = \frac{hD}{k} = \frac{D(T_i - T_w)}{\delta(T_s - T_w)}. \tag{23}$$

It is the main objective of this paper to compare the heat transfer rate in the presence of noncondensables with that for the case of a pure vapour. The comparison is made under the condition that T_s, T_w and u_{in} are the same in the two cases.

For pure vapour (subscript 0), one has

$$q_0 = \frac{k(T_s - T_w)}{\delta_0} \tag{24}$$

where δ_0 is calculated from equation (13) by substituting T_s for T_i. Then, upon ratioing equation (10) and (24)

$$\frac{q}{q_0} = \frac{\delta_0(T_i - T_w)}{\delta(T_s - T_w)}. \tag{25}$$

RESULTS AND DISCUSSION

The theoretical model established here was first verified by the experiments for condensation of pure vapours in vertical tubes at high Reynolds numbers. The type of data which was found in the literature, as listed in Table 1, is in the form of the average heat transfer coefficient over the entire tube condensing surface. The wall temperature of the tube in ref. [6] is evaluated at $l/2$ while data from both references have been reduced by evaluating liquid properties at $T_f = T_w + 0.3(T_s - T_w)$ and vapour properties at T_s. From Table 1 it can be seen that the prediction error is within 20% for a wide range of experimental parameters.

The theory presented here was applied to the steam–air system, because of its engineering interest. A tube of length 1.5 m and diameter of 40 mm served as a computational example. Computations for the reduction in condensation heat flux due to non-condensable gas, q/q_0, were performed for three values of core saturated temperature T_s varying between 70 and 130°C, which correspond approximately to a range of system pressures from 0.3 to 3 bar. At each T_s, the mass fraction W_∞ of the air in the core ranges from 0 to 0.1. In

Table 1. Comparison of analytical and experimental data for the average Nusselt number at high Reynolds numbers

Ref. No.	Run in Ref.	Fluid	u_{in} (m s^{-1})	l (m)	D (m)	T_s (°C)	ΔT (°C)	Re	$Nu(l)$ exp.	$Nu(l)$ theory	Percentage (%)
2	Fig. 6	H_2O	20	1.21	0.04	100	28	39 500	325	325.8	−0.25
2	Fig. 6	H_2O	40	1.21	0.04	100	28	79 000	475	387.0	18.5
2	Fig. 6	H_2O	20	1.21	0.04	100	11	36 900	369	417.1	13.5
2	Fig. 6	H_2O	40	1.21	0.04	100	11	73 800	491	501.6	−2.2
2	Fig. 6	H_2O	20	1.21	0.04	100	4	36 900	460	545.6	−18.6
2	Fig. 6	H_2O	10	1.21	0.04	100	6	19 885	363	435.4	−19.9
2	Fig. 6	H_2O	10	1.21	0.04	100	12	19 885	340	365.0	−7.4
2	Fig. 6	H_2O	20	1.21	0.04	100	20	38 000	353	356.7	−1.1
6	5	H_2O	20	2.32	0.0158	130	23.5	25 800	112	106.2	5.2
6	13	H_2O	26.5	1.86	0.0158	127	39.5	44 600	154	125.5	18.5

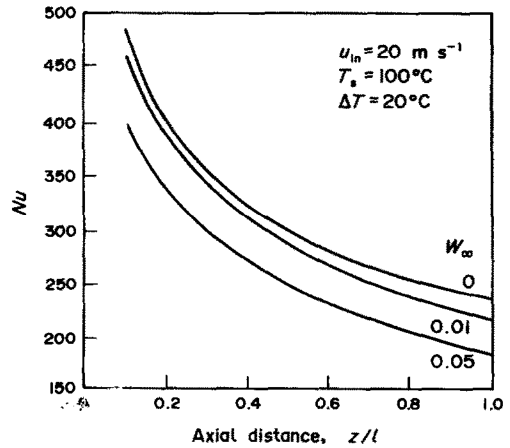

FIG. 4. Local heat transfer coefficient for condensation of steam–air mixture in the vertical tube.

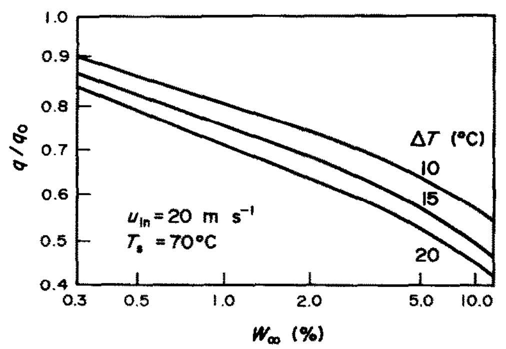

FIG. 5. Effects of non-condensable gas on condensation heat transfer in the vertical tube under sub-atmospheric pressure.

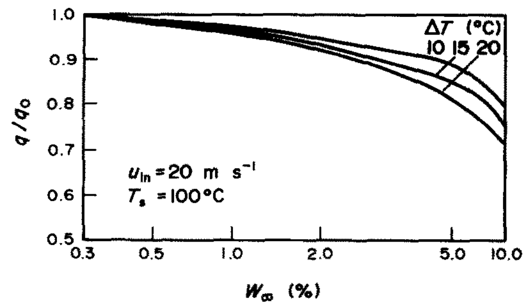

FIG. 6. Effects of non-condensable gas on condensation heat flux in the vertical tube for various temperature differences under atmospheric pressure.

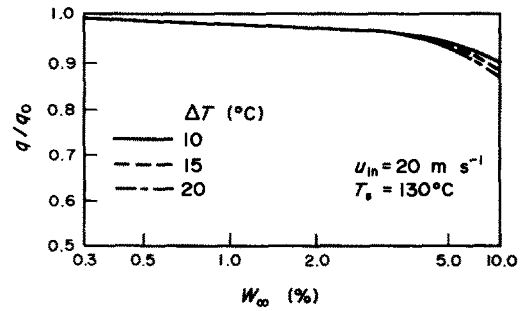

FIG. 7. Effects of non-condensable gas on condensation heat transfer in the vertical tube under sub-atmospheric pressure.

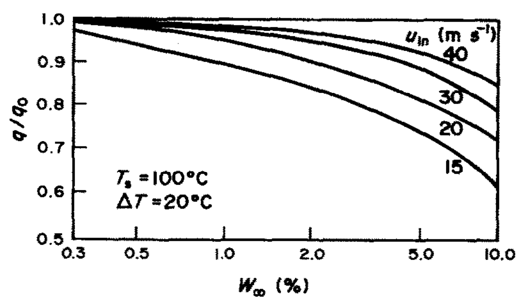

FIG. 8. Effects of non-condensable gas on condensation heat flux for different inlet velocities.

addition, at each fixed W_∞, the temperature difference $(T_s - T_w)$ was assigned values of 10, 15 and 20°C.

A typical distribution of the local heat transfer coefficient is shown in Fig. 4. The curve changes dramatically near the inlet of the tube and flattens gradually as the distance z increases. The presence of non-condensable gas, although it reduces the magnitude of heat transfer coefficients, does not seem to affect the changing trend of the local distribution of Nu.

The average heat transfer reductions due to non-condensable gas are presented in Figs. 5–7, which correspond respectively to T_s values of 70, 100 and 130°C. The departure of the curves from unity is a direct measure of the effects of non-condensable gas. Several general trends are seen in these figures. At any fixed temperature difference and fixed core temperature level, the heat transfer decreases monotonically as the mass fraction of the non-condensable gas increases. The reduction in condensation heat flux is perceptibly accentuated as the core saturated temperature decreases. Thus, the presence of the non-condensable gas is more strongly manifested when condensation takes place at sub-atmospheric pressures. In general, the increase of temperature difference $(T_s - T_w)$ results in the decrease of q/q_0, especially at the lower core saturated temperature level and at higher mass fractions, this effect is more remarkable. When T_s increases and W_∞ decreases, the heat transfer ratio becomes rather insensitive to the temperature difference.

The actual magnitudes of the reduction in heat transfer are as interesting as the afore-mentioned trends. For a trace amount of non-condensable gas, say $W_\infty = 0.005$, the heat transfer is only slightly decreased. Indeed, even for $W_\infty = 0.02$, appreciable effects are encountered only at the lower pressure levels and large temperature differences. With larger values of W_∞, such as 0.10, $q/q_0 \ll 1$, indicating that the effects of non-condensable gas should not be neglected.

Clearly, removal of non-condensable gases from the interface by forced convection will be beneficial. Such a conclusion was reached first by Sparrow *et al.* [10]. In the present system, the augmentation of condensation with non-condensable gas by increasing inlet mixture velocity is displayed in Fig. 8 for four

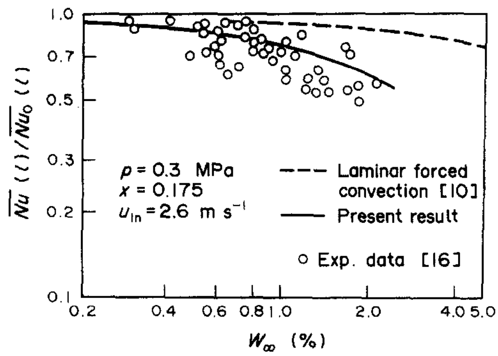

FIG. 9. Comparison of the present results with the experimental data of Borishansky *et al.* for condensation of a steam–nitrogen mixture in a vertical tube.

values of u_{in} ranging from 15 to 40 m s^{-1}. As expected, the forced flow condensation with lower velocities is much more sensitive to a non-condensable gas than with higher velocities, a finding that is intuitively reasonable.

Reference [18] indicated that the effects of non-condensable gas on the condensation heat flux is more appreciable in a tube than in an unconfined space, because in the former case the mass fraction of inert gas in the free stream becomes more and more as condensation proceeds along the condensing surface. In addition, the effects are more strongly manifested in the vertical orientation of the tube than in the horizontal one. As a simultaneous result of these two factors, the relative reduction in heat transfer due to inert gas is greater in the present system than in the case of laminar forced convection condensation along the horizontal plate dealt with by Sparrow *et al.* [10], even though the free stream here is in turbulent flow, as seen in Fig. 9.

A comparison of the theory presented here and experimental data indicating the effect of inert gas is certainly limited by the scarcity of available heat transfer coefficients corresponding to the present system. A meaningful comparison can be achieved by referring to the data obtained by Borishansky *et al.* [16] in a vertical tube with 10 mm i.d. and a length of 3 m for a moderate system pressure, as shown in Fig. 9. The concentration of nitrogen gas entering the condenser tube ranged from 0.26 to 2.5%. The experiments were carried out at $p = 0.3$ MPa, and the outlet vapour quality X varied from 0.05 to 0.5. The experimental data for Nu/Nu_0 (identical to q/q_0) were correlated at the same average velocity of steam in the tube, so were the numerical predictions for Nu/Nu_0 in Fig. 9. The calculated values are in good agreement with the experimental ones. This primarily indicates that the model is well developed for the prediction of effects of non-condensable gas on condensation heat transfer in vertical tubes. A systematic and comprehensive study concerning laminar film condensation of vapour–gas mixture in vertical tubes with turbulent flow of the core awaits further experimental effort.

CONCLUSIONS

An analytical study of filmwise, annular condensation of a saturated vapour–gas mixture in turbulent forced flow through a vertical tube has been conducted in order to reveal the detrimental effects of non-condensable gas on heat transfer.

It was found that the effects of non-condensable gas on condensation is more significant in ducts than in an unconfined space. Indeed, in the former case, the presence of trace amounts ($W_\infty = 0.01$) can lead to q/q_0 values of the order of 0.7–0.95, while larger concentrations such as $W_\infty = 0.10$ can reduce q/q_0 to 0.4. These values of q/q_0 are comparable with those for the case of laminar convection condensation along a horizontal plate, even though the free stream here is in turbulent flow. The reductions in heat transfer due to the non-condensable gas are accentuated at low operating pressures. As the inlet velocity of the mixture decreases, the effects become more and more appreciable.

All the computational results reported here were obtained for the steam–air system. However, such an analysis could be made for turbulent flow of any mixture consisting of a vapour and a non-condensable gas, and can be extended to condensation of a mixture flowing in parallel ducts as well as annuli.

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EFFETS DES GAZ INCONDENSABLES SUR LA CONDENSATION EN FILM LAMINAIRE DANS UN TUBE VERTICAL

Résumé—On développe une théorie pour révéler l'effet de petites quantités de gaz incondensables sur la condensation en film laminaire d'un mélange vapeur–gaz qui s'écoule de façon turbulente dans un tube vertical. La réduction du transfert thermique par le gaz incondensable est plus marquée aux faibles pressions et aux faibles nombres de Reynolds. Des comparaisons de la théorie avec quelques données expérimentales dans la littérature sont en bon accord.

DIE AUSWIRKUNGEN EINES NICHT KONDENSIERBAREN GASES AUF DIE LAMINARE FILMKONDENSATION IN EINEM SENKRECHTEN ROHR

Zusammenfassung—In diesem Artikel wird eine Theorie entwickelt, die den Einfluß kleiner Mengen nicht kondensierbarer Gase auf die laminare Filmkondensation von turbulent strömenden Dampf–Gas-Gemischen in einem senkrechten Rohr aufzeigt. Eine deutliche Verringerung des Wärmetransports durch das nicht kondensierbare Gas zeigt sich bei kleinen Drücken und niedrigen Reynolds-Zahlen des Gemisches. Vergleiche zwischen dieser Theorie und einigen experimentellen Werten aus der Literatur zeigen gute Übereinstimmung.

ВЛИЯНИЕ ПРИСУТСТВИЯ НЕКОНДЕНСИРУЕМОГО ГАЗА НА ЛАМИНАРНУЮ ПЛЕНОЧНУЮ КОНДЕНСАЦИЮ В ВЕРТИКАЛЬНОЙ ТРУБЕ

Аннотация—Предложена теория, объясняющая влияние небольшого количества неконденсируемого газа на ламинарную пленочную конденсацию парогазовой смеси при турбулентном течени в вертикальной трубе. Найдено, что в этом случае интенсивность теплопереноса значительно снижается при низких давлениях и числах Рейнольдса для смеси. Сравнение теории с некоторыми представленными в литературе экспериментальными данными дает хорошее соответствие.