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CONDENSATION HEAT TRANSFER IN THE PRESENCE OF NONCONDENSABLES

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ABSTRACT

The effect of noncondensables upon the laminar film condensation of vapor-gas flow inside a horizontal tube were investigated. The analysis was undertaken for the continuity, momentum and the thermal energy with its boundary conditions for the condensable film. Also the diffusion equation with its boundary conditions for the gas phase. The resulting partial differential equations were solved numerically by finite difference technique using a computer program for a wide range of parameters. Numerical results were obtained for the heat transfer coefficients with and without noncondensable for a wide range of parameters. It was found from the numerical results that the heat transfer coefficients inside the tube of the horizontal condenser are strongly dependent upon the axial and circumferential position. The noncondensable reduces the heat transfer and these reductions are accentuated at low operating pressures. For small steam velocities, a severe effects of the noncondensables are expected at the back end of the tube.

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INTRODUCTION

Condensation of vapor inside a horizontal tube results in a two phase flow. Bell, et al [1] showed the various two phase flow regimes during condensation. The performance of a condenser is governed to a large extent by the heat transfer on the inside of its tubes and thus depends to a large extent on these two phase flow patterns which exist along the tube. Previous works [2 - 3] investigated the condensation in annular flow regime where vapor shear is the dominant force acting on the condensate. Investigators [4 - 5] performed the analysis for the condensation in stratified flow regime where vapor shear is negligible and gravity the dominant force. Little study, however, apart from Rosson and Myers [6] , has been done in the wavy flow regime where the vapor shear and gravity are both significant forces. In the main of these investigations were concerned with the condensation of pure saturated vapors, that is, it was assumed that noncondensable gases were not present. In the present investigation the effect of noncondensable gases on laminar film condensation on the inside of horizontal circular tube with vapor shear and gravity forces is treated.

THE THEORETICAL MODEL

Consider a mixture of a vapor and a noncondensable gas entering a horizontal pipe of uniform radius R, length l and a constant uniform inside wall temperature T_w . The saturation temperature corresponding to the partial pressure of the vapor in the inlet mixture T ($T > T_w$). Cylinderical coordinate system is chosen with the axis oz lying along the centre of the horizontal pipe and v , u are taken to be the velocity components in the directions of θ, z increasing respectively. While the vapor-gas mixture flowing in the z-direction condenses on the inside surface of the tube. The condensate film flows down in the tangential direction along the periphery of the tube. The tube section between z and z + Δz represents the control volume around which a balance of forces is taken. A coordinate system is shown in Fig.1.

THE GOVERNING EQUATIONS

The formulation of the governing momentum and energy equations for the condensate film can be simplified by introducing the following assumptions :

1. The flow of the liquid film is laminar.
2. The tangential pressure drop in the film is negligible as compared with the axial pressure drop.
3. The second derivatives of the velocities are negligible as compared with the corresponding normal derivatives.
4. Surface tension effects are negligible.
5. The two phase flow of the condensate inside the tube moves in the wavy flow regime, the shear stress on the free vapor liquid interface of the film in the axial direction is considered.

FOR PURE SATURATED VAPOR

The momentum equations in the condensate film can be written as follows :

$$\frac{\partial^2 v}{\partial r^2} = - \frac{\rho g}{\mu} \sin \theta \qquad (1)$$

$$- \frac{\partial p}{\partial z} - \frac{\partial \tau_{rz}}{\partial r} = 0 \qquad (2)$$

With the following boundary conditions :

$$r = R \;, \quad v = 0 \;, \quad u = 0 \qquad (3)$$
$$r = r_i \;, \quad \partial v / \partial r = 0 \;, \quad \tau = \tau_i$$

The integration of Eqs.(1-2) with the boundary conditions Eq.(3) ,yields the velocity profiles, then the average velocity components can be written as :

$$\bar{v} = \frac{\delta^2}{2} \frac{\rho g}{\mu} \sin \theta \qquad (4)$$

$$\bar{u} = \frac{\delta^2}{3} \frac{1}{\mu} \frac{\partial p}{\partial z} + \frac{\tau_i \delta}{2 \mu} \qquad (5)$$

The thermal energy equation is,

$$q = k \frac{\partial T}{\partial r} = k \frac{(T - T_w)}{\delta} = k \frac{\Delta T}{\delta} \qquad (6)$$

And the continuity equation on the film is,

$$\rho \frac{\partial}{\partial z}(\delta \bar{u}) + \frac{\rho}{R} \frac{\partial}{\partial \theta}(\delta \bar{v}) = \frac{q}{L} \qquad (7)$$

By substitution of Eqs.(4-6) into Eq.(7) yields a partial differential equation describing the film thickness distribution over the condensate surface as follows :

$$\frac{\rho}{\mu}\frac{\partial p}{\partial z} \delta^3 \frac{\partial \delta}{\partial z} + \frac{\rho \tau_i \delta^2}{\mu} \frac{\partial \delta}{\partial z} + \frac{\rho^2 g}{R \mu} \delta^3 \sin\theta \frac{\partial \delta}{\partial \theta}$$

$$+ \frac{\rho^2 g}{3 R \mu} \cos\theta \, \delta^4 - \frac{k \, \Delta T}{L} = 0 \qquad (8)$$

The momentum balance in the vapor phase can be written as :

$$\frac{\partial p}{\partial z} = \frac{2 \tau_i}{R} - \frac{\delta q}{L} \frac{2 \bar{U}}{\pi R} \int \frac{d\theta}{\delta} \qquad (9)$$

The simultaneous numerical solution of Eqs.(8-9) using a backward difference technique giving the film thickness δ at any point. Hence the local heat transfer coefficient can be calculated as :

$$h = \frac{k}{\delta} \qquad (10)$$

FOR VAPOR-AIR MIXTURE

The accumulation of noncondensables at the (fluid) condensation surface reduce the saturation pressure of the condensing steam and the corresponding saturation temperature at the surface. As the condensation proceeds along the tube the effect of inerts increases due to the reduction of the saturation temperature brought about by steam disappearance and the increased concentration of inerts. T varies with the axial direction and henceforth denotes the local z dependent temperature of the free surface of the condensate film.

THE DIFFUSION PROBLEM

If W denote the local mass fraction of the noncondensable gas as,

$$W = \rho_g / (\rho_g + \rho_v) \tag{11}$$

The diffusion equation is,

$$U \frac{\partial W}{\partial z} = D \left(\frac{\partial^2 W}{\partial r^2} + \frac{1}{r} \frac{\partial W}{\partial r} \right) \tag{12}$$

With the following initial and boundary conditions,

$$z = 0 \;,\; 0 < r < R \qquad U = U_o(r) \;, \quad W = W_o(r) \tag{13}$$

$$0 < z < 1 \begin{cases} \text{(centre)}\ r=0 \;, & D\dfrac{\partial U}{\partial r} = 0 \;, \quad \dfrac{\partial W}{\partial r} = 0 \\ \text{(interface)}\ r=r_i \simeq R, & U = 0 \;, \quad -D\dfrac{\partial W}{\partial r} = 0 \end{cases} \tag{14}$$

Assume a parabolic concentration distribution of the inerts associated with the steam at the inlet. The condensate is impermeable to the inerts. Neglect the volume occupied by the condensate film within the tube($r = r_i \simeq R$ at the vapor-liquid interface). $W(z , R)$ or $W_i(z)$ at the interface between the vapor-gas mixture and the liquid film can be determined by the numerical solution of Eq.(12) using finite difference technique.

INTERFACE TEMPERATURE

Consider the condensation of a specific vapor and noncondensable gas where in the following quantities are specified :

$$T_o \;,\; W_o \;,\; T_w \text{ and } U_o .$$

First the total pressure p of the system can be evaluated as : Assume the mixture and its components behave like perfect gases such as :

$$\frac{p_v}{p} = \frac{1 - W}{1 - W(1 - M_v/M_g)} \tag{15}$$

Where p_v represents the vapor pressure. If T_o is specified, the mixture is assumed to be at saturation then p_v is available from the tabulated properties of the vapor. With this and with the known value of W_o , the total pressure p follows Eq.(15). The resulting W_i from the numerical solution of the diffusion equation is converted to $p_v)_i$ by applying Eq.(15) , and T_i follows from the tabulated saturation data for the vapor. By substitution of T_i in Eqs.(8-9) instead of T then the film thickness at the presence of noncondensable gas δ' can be determined and also the local heat transfer coefficient h' as $h' = k / \delta'$.

HEAT TRANSFER

Once T_i has been found, then heat transfer rate follows directly. Let q' denote the local heat transferred to the surface per unit area, so that,

$$q' = k \frac{(T_i - T_w)}{\delta'} \qquad (16)$$

It is particularly interesting to compare the heat transfer rate in the presence of noncondensables with that for the case of pure vapor. The comparison is made under the condition that T_o and T_w are the same in the two cases.
For the pure vapor $T = T_o$,

$$q = k \frac{(T_o - T_w)}{\delta} \qquad (17)$$

Then upon ratioing these two equations, follows,

$$\frac{q'}{q} = \left(\frac{\delta}{\delta'}\right)\left(\frac{T_i - T_w}{T_o - T_w}\right) \qquad (18)$$

The ratios of the condensate layer thicknesses and the temperature differences are evaluated as before. It is seen that for prescribed values of T_o , T_w and W_o , the heat transfer ratio q'/q can be calculated for a given mixture of vapor and noncondensable gas. The departure of q'/q from unity is direct measure the effect of noncondensable gas.

All calculations have been made by a computer program. The values of all data necessary for calculations are arbitrary, however the values used in the present calculations for a horizontal condenser characterized by the dimensions of l=200 cm and R=2 cm. The tube was made of copper. The vapor-air mixture was in a dry saturated state. For condensation of steam with air as noncondensable, computations of q'/q were performed for T_o = 355 and 370 °k which correspond approximately to the system pressure p = 0.5 and 0.9 atmosphere. At each T_o , the mass fraction W_o of the air in the vapor-air mixture at the inlet was assigned values of 0.01 , 0.05 and 0.1 . In addition, at each fixed W_o , the temperature difference (T_o - T_w) was varied between 2 °k to 50 °k. The inlet mixture velocity U_o was varied from 15 m/s to 30 m/s.

RESULTS AND DISCUSSIONS

Some of the obtained numerical results are shown in Figs.(2-5). Circumferential variations in the heat transfer coefficients for pure saturated vapor at fixed axial positions and T_o=355 °k are shown in Fig. 2. It was found that for laminar film condensation the film thickness increased around the tube, the resistance to heat transfer would increase and hence the local heat transfer coefficient would decrease. The axial variations in heat transfer coefficients for a pure saturated vapor at constant circumferential positions and T_o = 355 °k are shown in Fig. 3. It is clear that the heat transfer coefficient decreases along the tube due to increase in the film thickness as condensation proceeded along the tube.

The heat transfer results of q'/q were presented in Figs.(4-5). The departure of the curves from unity is a direct measure of the effect of the noncondensable gas. It was found that for any fixed temperature difference and fixed T_o , the heat transfer decreases monotonically as the mass fraction of the noncondensable increases. The presence of the noncondensable is more strongly manifested when the condensation takes place at sub-atmospheric pressures. At the higher T_o and the lower mass fractions, q'/q is rather insensitive to the temperature difference (T_o - T_w). When T_o decreases and W_o increases the heat transfer ratio becomes more sensitive to the temperature difference, decreasing as (T_o - T_w) increases.

The buildup of the noncondensable gas at the interface tends to lower the corresponding partial pressure of the vapor, then lowers the interface temperature T_i at which condensation occurs (the interface is a saturation state). The thermal driving force (T_i - T_w) is lowered, thereby decreasing the heat transfer rate. Moreover, at small steam velocities, where a concentration profile of inerts is built up, some more severe effects of the noncondensables are expected at the back end of the tube.

CONCLUSIONS

From the presented numerical results, it can be concluded that :

1. Heat transfer coefficients inside the tubes of horizontal condensers are strongly dependent upon the axial and the circumferential position.
2. The noncondensable reduces the heat transfer and these reductions are accentuated at low operating pressures and small steam velocities.

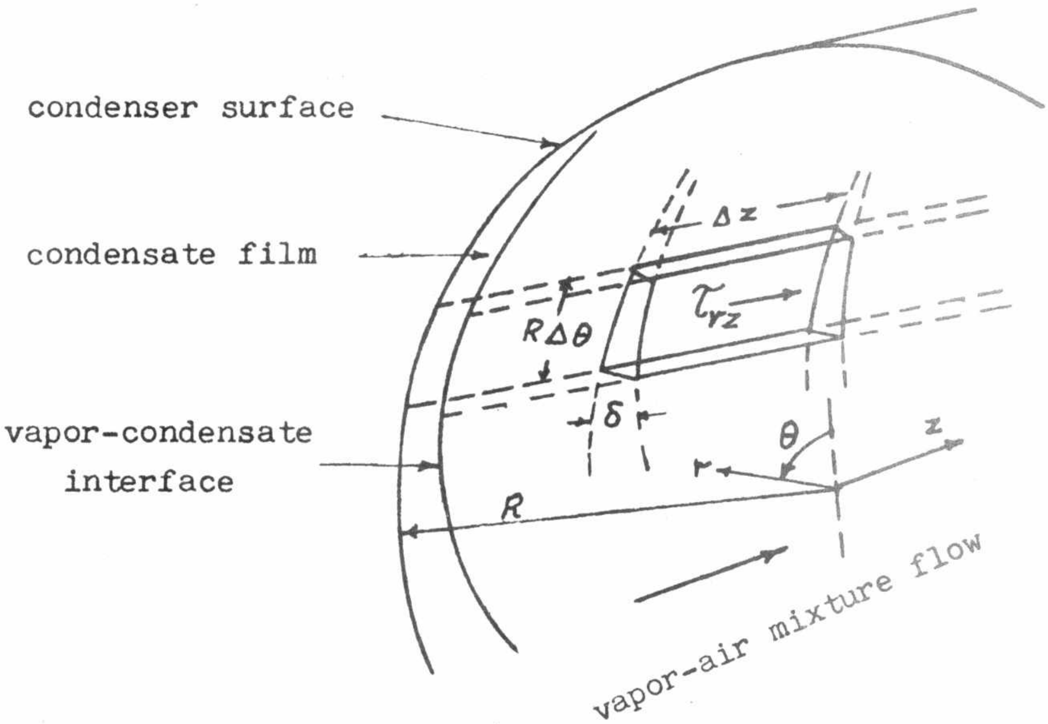

Fig. 1. Physical system and coordinates

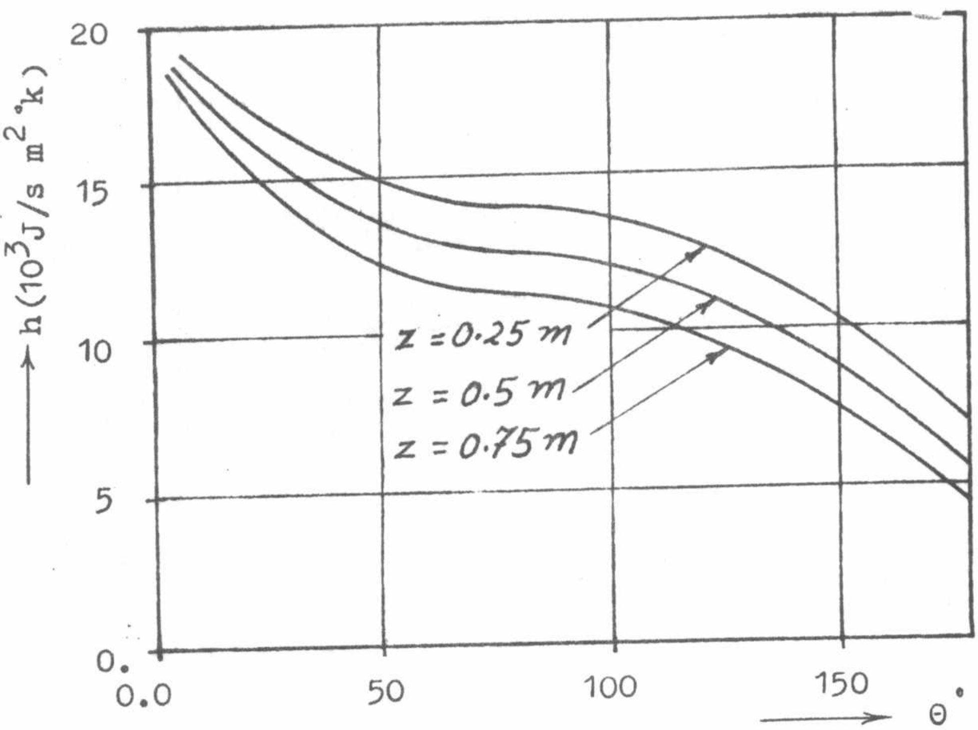

Fig. 2. Local heat transfer coefficient around the tube, T_o = 355 °k

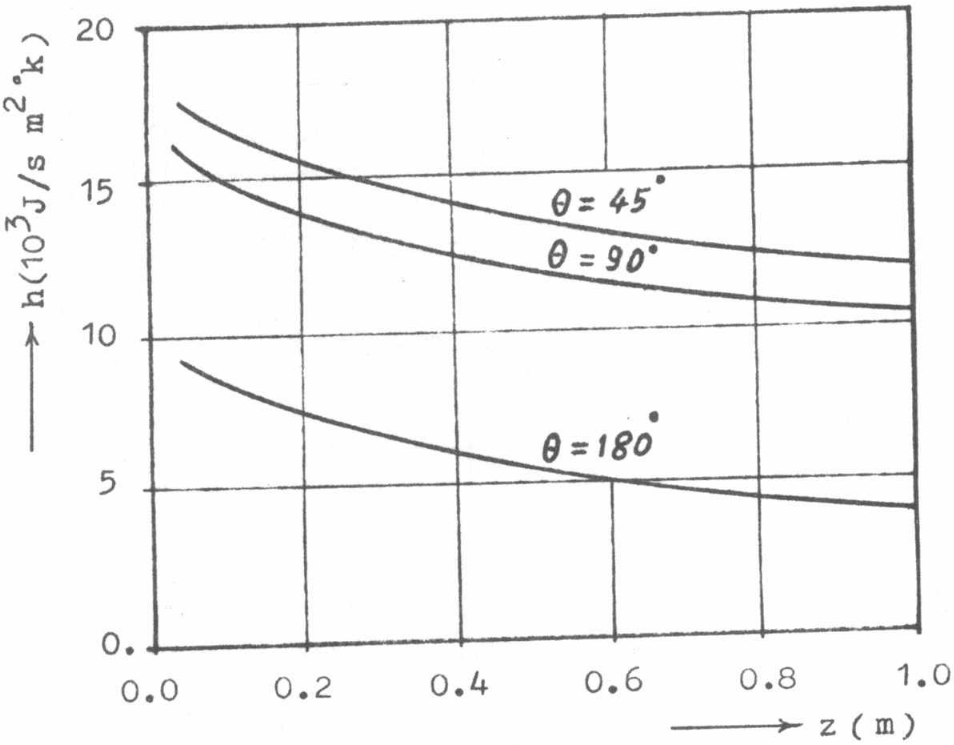

Fig. 3. Local heat transfer coefficient along the tube, T_o = 355 °k

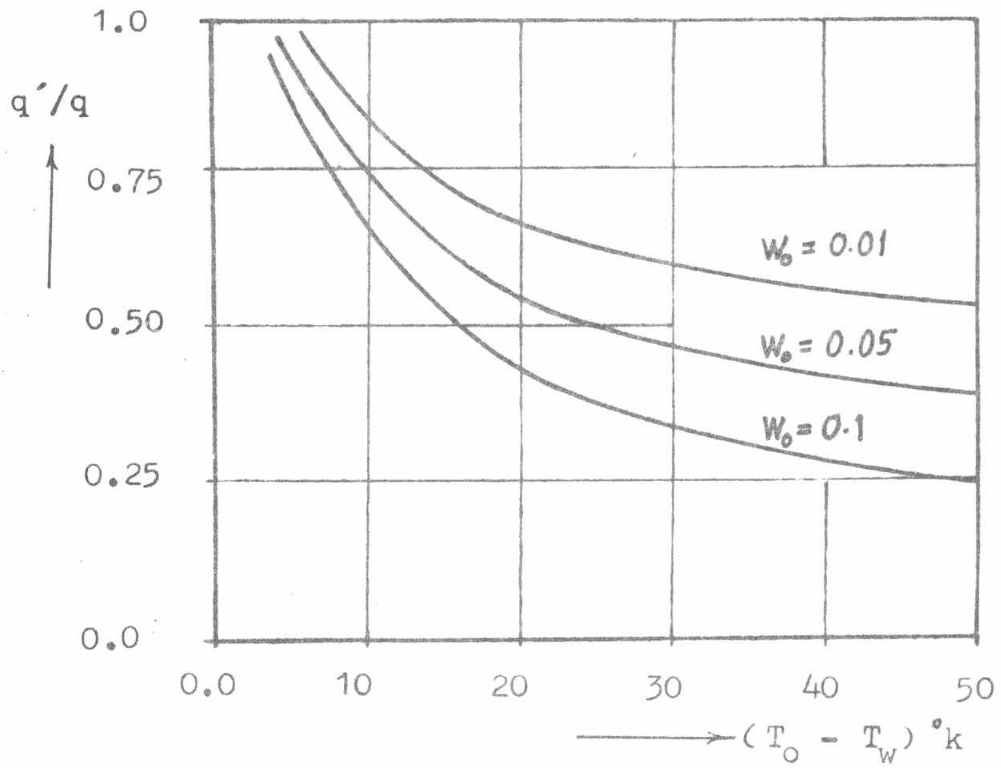

Fig. 4. Condensation heat transfer for steam-air mixture, $T_o = 355$ °k

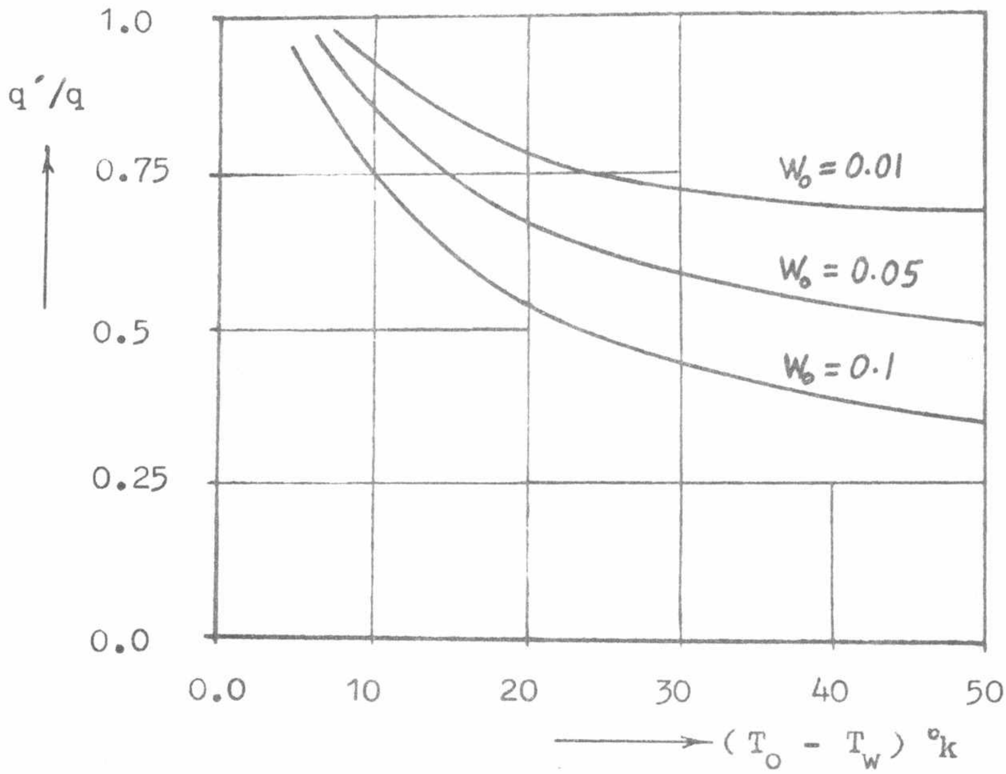

Fig. 5. Condensation heat transfer for steam-air mixture, $T_o = 370$ °k

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NOMENCLATURE

D - binary diffusion coefficient
g - gravitation constant
h - local heat transfer coefficient
k - thermal conductivity of liquid
L - latent heat of condensation
l - tube length
M - molecular weight
p - pressure
q - heat flux
R - tube radius
r - radial coordinate
T - saturation temperature of steam corresponding to the pressure p
U - axial velocity of the vapor phase
u - axial velocity of the condensate
v - tangential velocity of the liquid film
W - local mass fraction of the noncondensable gas
z - axial coordinate of the tube
δ - condensate film thickness
θ - angular coordinate
μ - liquid viscosity
ρ - density
τ - shear stress

Subscripts:

g - gas
i - interface
v - vapor
o - inlet
w - at the wall

Superscripts:

$\bar{}$ - mean value
$'$ - vapor-gas mixture