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TURBULENT FILM CONDENSATION OF PURE VAPORS FLOWING NORMAL TO A HORIZONTAL CONDENSER TUBE - CONSTANT HEAT FLUX AT THE TUBE WALL

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ABSTRACT

A mathematical model is developed for the study of external turbulent film condensation of pure vapors flowing downward and normal to the axis of the condenser tube with constant heat flux condition maintained at the tube wall. The magnitude of interfacial shear is estimated for a given external flow condition of the vapor with the help of Colburn's analogy. The average condensation heat transfer coefficients for different system conditions are evaluated. The present theory is compared with the available experimental and theoretical data in the literature and is found to be satisfactory.

***Keywords*:** Film condensation - Turbulent flow of vapor - Condenser tube – Interfacial shear

INTRODUCTION

The phenomenon of condensation of vapors flowing around a condenser tube is of practical significance in high pressure thermal power plants and chemical process industries. Nusselt (1916) who is the pioneering researcher in film condensation, assumed zero shear at the liquid-vapor interface. The external flow of vapor creates interfacial shear which may enhance the condensation heat transfer coefficient depending on the direction of flow of vapor. Some important investigations on external flow of vapor over the condenser tube are by several researchers (Shekariladge and Gomeluri, 1966; Fujji et al., 1979; Lee et al., 1984; Memory et al., 1993; and Michael et al., 1989). These investigators are primarily concerned with the estimation of condensation heat transfer coefficients for isothermal condition of the tube wall. Fujji et al.(1972) investigated experimentally with the peripheral distribution of wall heat flux and proposed an equation valid for the limiting cases of large and small on-coming vapor velocity. In a later study, Rose (1984) considered the effect of the pressure gradient on forced convection laminar film condensation on a horizontal tube and proposed an equation for the mean heat transfer coefficient. However, in all these investigations the condensate film is assumed to be in laminar regime. Honda et al. (1986) studied experimentally downward film condensation of R113 vapor for wide ranges of vapor velocity, condensate temperature difference and the design equations were proposed. Experimental data of Michael et al. (1989) on condensation of steam with high flow velocities could not be correlated satisfactorily by any of the aforementioned theories. It is further suggested that one of the possible reasons for the deviation might be due the fact that condensate film can be under turbulent regime for certain ranges of the system parameters.

For turbulent film condensation on a horizontal cylinder, Sarma et al. (1998) used the eddy diffusivity of Kato's model in the condensate film with an assumption that shear stress at the vapor liquid interface is of the same order as one would expect for single phase flow. An empirical equation was proposed for an average heat transfer coefficient which is in good agreement with experimental data of Honda et al. (1986) and Lin and Yang (2004) treated the elliptical condenser tube with variable wall temperature. The non-uniform temperature has appreciable effect on local film heat transfer coefficient but its influence on mean heat transfer coefficient is minimal. Yang and Lin (2005) considered the non-isothermal condition for the tuber wall and inter facial eddy diffusivity. It is observed that inclusion of eddy diffusivity effect decreases the mean heat transfer coefficient with increase in wall temperature variation amplitude. Hu et al. (2006) studied turbulent film condensation on an isothermal inclined elliptical tube in terms of the local condensate film thickness and the heat transfer characteristics. Hu et al. (2007) also studied the influence of different eddy diffusivity models and proposed a new model for local film shear which produces higher mean heat transfer coefficients. However, the above-mentioned studies focus on the isothermal and non-uniform wall temperature case. Hence, the present study is aimed at formulating the turbulent film condensation process with diabatic condition of constant heat flux attributed to the tube wall. In addition, the influence of the external flow of the vapor on the condensate film is included in assessing the performance of the condenser tube. Furthermore, the study included a comparison between the present results and the results obtained from previously published theoretical and experimental data.

PHYSICAL MODEL

Pure vapors approach a horizontal tube along the direction of gravity vector. The wall of the condenser tube is maintained at constant heat flux. It is assumed that the condensate film flow is under turbulent regime in the region away from the upper stagnation point. Influence of boundary layer separation is not considered.

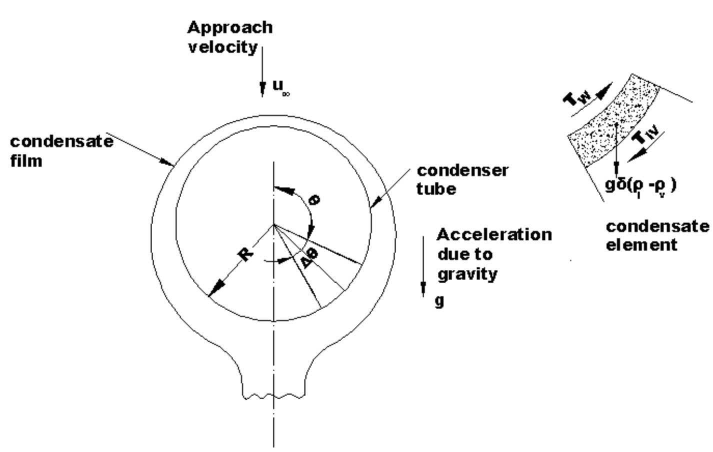

Figure 1. Physical configuration-flow model

Analysis

The shear stress in the condensate film can be expressed as

$$\tau_w = \mu\left(1+\frac{\varepsilon_m}{v_1}\right)\frac{du}{dy} \tag{1}$$

For the physical configuration and coordinate system shown in Figure 1, the force balance applied to an element of the condensate is as follows:

$$\tau_w = \tau_{iv} + g\delta\mu(\rho_l - \rho_v)\sin\theta \tag{2}$$

where τ_w and τ_{iv} are wall and interfacial shear stress.

The case of $\tau_{iv} = 0$ happens to be Nusselt's analysis. In Eq. (2), the influence of inertial force is neglected. In addition, considering that, the turbulent or eddy conduction across the condensate layer is more significant than convective component, hence the equation of conservation of energy in the film is expressed as follows.

$$\frac{d}{dy}\left[\left(1+\frac{\varepsilon_m}{v_1}\mathrm{Pr}\right)\frac{dT}{dt}\right] = 0 \tag{3}$$

The boundary conditions employed for solving Eq. (3) are as follows:

$$\begin{aligned} y = 0;&\quad T = T_w \quad (\theta) \quad \textit{unknown} \\ y = \delta;&\quad T = T_s \end{aligned} \tag{4}$$

In addition, as per the assumption employed in the analysis:

$$y = 0; \quad k_l \left.\frac{dT}{dy}\right|_{y=0} = q_w \quad \text{(a constant)} \tag{5}$$

The phase transformation occurring at the vapor liquid interface is given by the following relationship:

$$\frac{d}{d\theta}\int_0^\delta \rho_l u\,dy = \frac{q_w R}{h_f g} \tag{6}$$

Equation (6) implies that the total heat conducted across the condensate film is equal to the value that would arise due to phase transformation occurring at the vapor liquid interface as result of condensation of vapors around the tube. The expression for convective heat transfer around the cylinder from Holman (1997) is as follows

$$Nu = C\,\mathrm{Re}_v^n\,\mathrm{Pr}^{\frac{1}{3}} \tag{7}$$

The values of C and n are chosen as C = 0.0266 and n = 0.805 from Holman (1997), which is valid for range 40,000 < Re_v < 400,000

Further, it is assumed that the principle of Colburns analogy, modification of Reynold's analogy holds good i.e.,

$$\frac{f}{2} = St\,\mathrm{Pr}^{\frac{2}{3}} \tag{8}$$

From Eqs. (7) and (8) the average wall friction coefficient f can be obtained as follows

$$f = 2C\,\mathrm{Re}_v^{n-1} \tag{9}$$

The local friction f_θ is further defined by the expression

$$f_\theta = C[\mathrm{Re}_v]\sin\theta \tag{10}$$

The average friction coefficient can also be written as

$$f = \frac{1}{\pi}\int_0^\pi f_\theta d\theta \tag{11}$$

Equations (9), (10) and (11) yield the following relationship

$$f_\theta = \pi C\,\mathrm{Re}_v^{n-1}\sin\theta \tag{12}$$

The local shear stress is defined by the relationship

$$\tau_{iv} = \frac{1}{2} f_\theta \rho_v u_T^2 \tag{13}$$

where $u_T = 2u_¥ \ sin\theta$

From Eqs. (12). and (13), the expression for local interfacial shear stress can be obtained as

$$\tau_{iv} = 2\pi C r_v u_¥^2 \ Re_v^{n-1} \ sin^3\theta \tag{14}$$

Thus, it is revealing that the τ_{iv} induced on the condensate film both at $\theta = 0$ $and\,\pi$ is negligible. It is maximum at $\theta = \pi/2$.
The wall shear stress can be expressed as

$$\tau_w = \rho_l u^{*2} \tag{15}$$

Further, the boundary layer separation is ignored in the present analysis. It is close to the assumption employed by Shekriladze and Gomelauri, (1966).
With the aid of Eqs. (14) and (15), Equations (2), (3) and (6) are expressed in

dimensionless form as follows:

$$R^{*3} = R^{*} \varphi \, Fr^{(n+1)/2} \sin^{3}\theta + \delta^{+} \sin\theta \tag{16}$$

$$\frac{dT^{+}}{dy^{+}} = -\left[\frac{1}{R^{*} Gr^{\frac{1}{3}} \left(1 + \frac{\varepsilon_m}{v_1} \Pr\right)}\right] \tag{17}$$

$$\frac{d}{d\theta} \int_0^{\delta^{+}(\theta)} u^{+} dy^{+} = Q \, \mathrm{Re}_l^{\frac{1}{2}} \tag{18}$$

where $\varphi = \left\{ 2^n \pi C \left[\frac{\rho_v}{\rho_l}\right] Gr^{\frac{3n-1}{6}} \left[\frac{v_v}{v_l}\right]^{1-n} \right\}$

From Eqs. (1) and (15), the expression for velocity profile can be obtained as follows

$$\frac{du^{+}}{dy^{+}} = \frac{1}{1 + \frac{\varepsilon_m}{v_1}} \tag{19}$$

The normalized boundary conditions for solving Eqs. (17) and (18) are

$$\left.\begin{array}{ll} y^{+} = 0; & T^{+} = T_w^{+} \\ y^{+} = \delta^{+}; & T^{+} = 0 \\ y^{+} = 0; & u^{+} = 0 \end{array}\right| \tag{20}$$

where $T_w^{+} = \frac{k_l \left(T_s - T_w\right)}{q_w R}$ (21)

According to Sarma et al. (1998), the assumption of universal velocity distribution can be used in evaluation of mass flow rate of condensate at any angular location.

The magnitude of interfacial shear on the vapor-liquid interface is generally assumed to be in terms of a proportionality constant, a factor of multiplication of τ_w. In this particular case the constant of proportionality is tentatively chosen as unity. So, it is assumed that τ_{iv} is of the same order as that of τ_w and Eq. (19) can be considered as valid approximation. In the present study, Kato's expression for eddy diffusivity is used to solve Eqs. (17) and (19). Kato's expression is as follows.

$$\frac{\varepsilon_m}{v_1} = 0.4 y^{+} \left[1 = \exp\left(-0.0017 y^{+^2}\right)\right] \tag{22}$$

The inadequacy in Eq. (19) is that the influence interfacial shear is not properly accommodated. However the Eq. (18) requires integrated values of velocity function and hence it is assumed that the error introduced would be marginal.

The local values of wall temperatures can be estimated with the aid of the following expression:

$$S = \left[Q \mathrm{Re}_l^{\frac{1}{2}} T_w^+ \right] \tag{23}$$

Certain observations can be made from Eq. (16). At $\theta \to 0$ and π, since $R^* \to 0$ it can be seen that local value of temperature tends to infinity as per the formulation. However, to avoid such a situation and to obtain physically meaningful results at the upper and lower stagnation points i.e., at $\theta \to 0$ and π condensation phenomenon is considered taking into account the special conditions that would prevail at these locations. At $\theta \to 0$, the effect of external flow field of vapor on the condensation process can be ignored and is identical to Nusselt's pioneering analysis. According to Nusselt's analysis under constant heat flux conditions the sub-cooling parameter is given by the relationship as follows:

$$\frac{c_p \Delta T}{h_{fg} \Pr} = \frac{1}{3} \left[\frac{3 q_w R}{\mu_l h_{fg}} \right]^{\frac{4}{3}} \left[\frac{\theta}{Gr \sin\theta} \right]^{\frac{1}{3}} \tag{24}$$

It can be seen from the Eq. (24) that $\theta \to 0$, the sub-cooling parameter $\frac{c_p \Delta T}{h_{fg} \Pr}$ tends to a finite value. In addition, $\theta \to \pi$, $\frac{c_p \Delta T}{h_{fg} \Pr}$ will attain infinity.

However, from the physical consideration for cylindrical geometry, the condenser tube with finite wall thickness the boundary condition $\frac{ds}{d\theta}$ at $\theta = \pi$ holds good. This boundary condition will lead the wall temperature to a finite value. In the present problem, the sub-cooling parameter can be assessed with imposition of the condition that laminar film condensation theory holds good at upper stagnation point i.e., $\theta = 0$. However, at the lower stagnation point i.e., $\theta = \pi$, the condition of symmetry viz., $\frac{ds}{d\theta}$ is assumed to hold good such that the sub-cooling parameter tends to a finite value.

By definition the local condensation heat transfer coefficient can be written as follows:

$$q_w = k_l \left. \frac{\partial T}{\partial y} \right|_{y=0} = h\left(T_s - T_w\right) \tag{25}$$

Equation (25) in dimensionless form can be expressed by the relationship:

$$\frac{Nu}{\mathrm{Re}_l^{\frac{1}{2}}} = \frac{Q}{S} \tag{26}$$

Further, the mean Nusselt number and wall temperature are obtained from the following expression

$$\frac{\overline{Nu}}{\mathrm{Re}_l^{\frac{1}{2}}} = \frac{1}{\pi}\int_0^{\pi} Nu d\theta$$
$$\overline{S} = \frac{1}{\pi}\int_0^{\pi} S d\theta \tag{27}$$

In writing down Eq. (19), there is an implied assumption that τ_{iv} is of the same order as τ_w. It is presumed that such an assumption will not lead to substantial error in the estimation of the integral appearing in Eq. (18). In other words universal velocity distribution is being used in the estimation of discharge rate of the condensate at any angular location. In fact, it can be seen that at very high velocities of the external flow of vapor such an assumption, i.e. $\tau_{iv} \to \tau_w$ is valid and hence Eq. (19) can be considered as a valid approximation.

Thus, it is evident that the local condensation heat transfer coefficients are dependent on the following system parameters.

$$\frac{Nu}{\mathrm{Re}_l^{\frac{1}{2}}} = F[\theta, \varphi, Fr, Gr, Q] \tag{28}$$

Solution Procedure

1. The system pressure, the approach velocity u_{∞}, the diameter of the condenser tube and the thermal load are considered as input parameters. Thus *Fr*, *Q*, *Gr* can be computed.
2. To begin the calculation, the dimensionless film thickness δ^+ and the velocity u^+ in the condensate film are assumed to be zero at $\theta = 0$ i.e., node $m = 1$. The temperature is evaluated from Eq. (17) iteratively with an initial guess for T_w^+ so that the boundary condition $T^+ = T_w^+$ at $y^+ = 0$ and $T^+ = 0$ at $y^+ = \delta^+$ are satisfied for a prescribed accuracy of 10^{-5}.
3. The velocity profile across the condensate film and angular positions θ and R^* are obtained by solving Eqs. (16), (17) and (18).

By proceeding to the next node $m = m+1$ i.e., at $m = 2$ the condensate film thickness $d^+(m+1) = d^+(m) + dd^+$, where dd^+ is assumed be 0.003. Equation (18) is written in finite difference form to evaluate the angular position θ.

$$d\theta = \left[\frac{I(\delta^+ + d\delta^+) - I(\delta^+)}{Q\mathrm{Re}^{\frac{1}{2}}}\right] \qquad \text{where} \qquad I(\delta^+) = \int_0^{\delta^+(\theta)} u^+ dy^+$$

The value δ^+ is incremented by of dd^+ and the step 2 and step 3 are repeated to determine the angular position, R^* and T_w^+. Equation (26) is used to solve the local Nusselt number. The procedure outlined in step 3 is repeated till θ advances to π. Equation (27) is further utilized to determine average values of heat transfer coefficient and wall temperatures.

RESULTS AND DISCUSSION

A series of numerical calculations were performed on the model (Figure 1), to investigate the behavior of turbulent film condensation of pure vapors over a horizontal cylinder with constant heat flux at the tube wall.

Characteristics of Flow Hydrodynamics

The variation of the condensate film thickness δ^+ around the periphery is shown plotted in Figure 2 for different values of Q. It is evident that condensate film thickness increases monotonically with increase in Q.

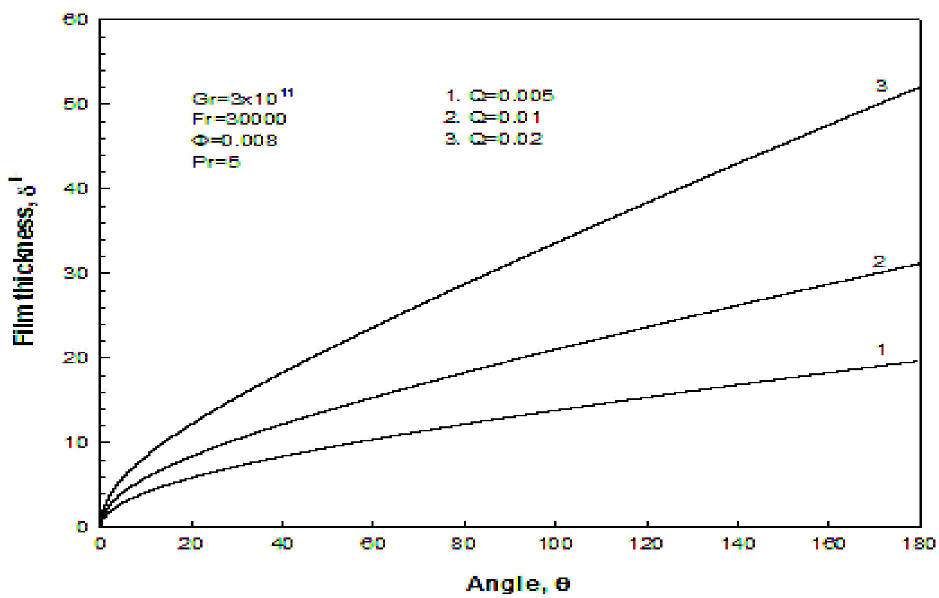

Figure 2. Variation of condensae film thickness around the periphery

Figure 3 reveals the variation of shear along the tube wall for various values of interfacial shear parameter φ. The parameter φ consists of density and viscosity ratios of vapor-to- liquid. Thus parameter φ signifies the effect of system pressure. It can be observed from Figure 3 that the wall shear increases with an increase in system pressure.

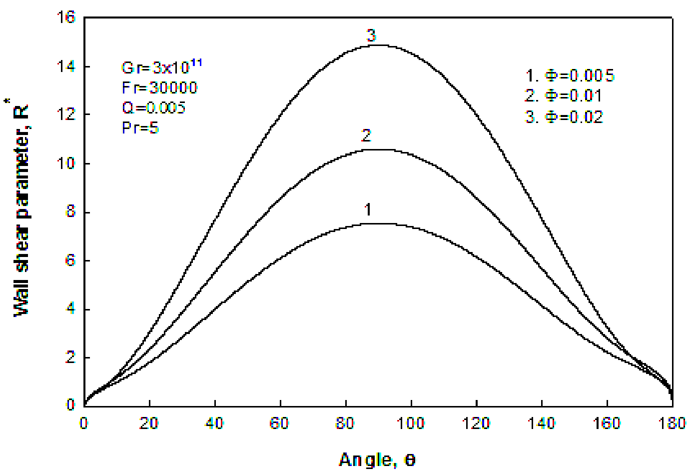

Figure 3. Variation of shear at the tube wall around the periphery- Effect of interfacial shear

Heat Transfer Characteristics

Effect of Sub-cooling Parameter (S)

The constant heat flux condition at the tube wall makes the surface highly non-isothermal. Figure 4 demonstrates variation of sub-cooling parameter S for various values of thermal load parameter Q. It can be inferred from Figure 4 that the wall temperature T_w increases up to $\theta = \pi/2$ and decreases subsequently. It can be noted that the wall temperature T_w decreases as the surface heat flux or cooling rate is increased.

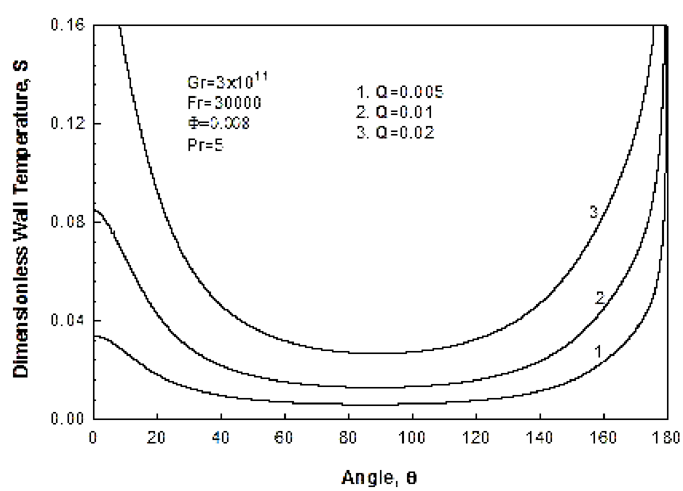

Figure 4. Wall temperature variation around the periphery

Influence of Thermal Load on Local Heat Transfer Coefficient

Equation (26) is shown plotted in Figure 5 for various values of Q. It is evident that the rise and decay characteristics of local Nusselt number indicates that the condensate heat transfer coefficients attain a maximum value approximately at $\theta = 88°$. Further, as $\theta \rightarrow \pi$ the condensate heat transfer coefficient decreases considerably.

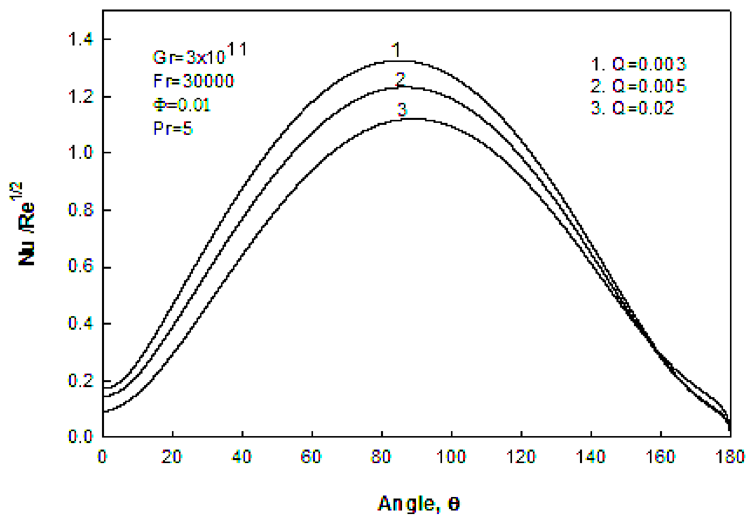

Figure 5. Variation of local Nusselt around the periphery-effect of Q

Effect of External Vapor Velocity

The effect of vapor Froude number on condensation Nusselt number is shown in Figure 6. It can be observed that the Nusselt number increases with an increase in vapor velocity u_{∞}. Further, there is a decrease in Nusselt number with an increase Q, which is due to an increase in condensate film thickness.

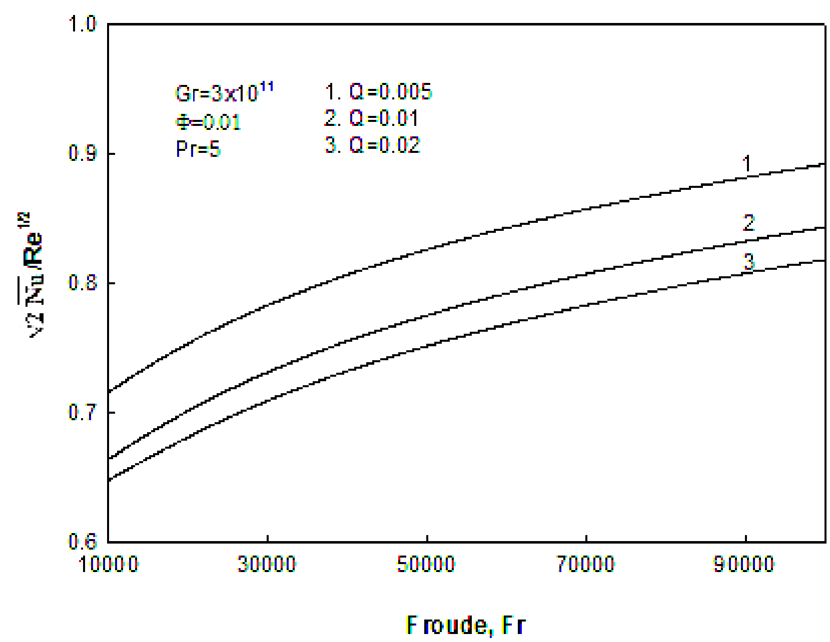

Figure 6. Effect of external flow on the mean condensate heat transfer coefficients

Effect of Thermal Load, Q

An interesting phenomenon is observed in Figure 7, in which the average Nusselt number $\left(\overline{Nu}/Re^{1/2}\right)$ is shown plotted as a function of dimensionless heat flux Q for different Froude numbers. At higher Froude numbers, as the heat flux increases, Nusselt number is found to decrease initially and later assumes an asymptotic value. In other words the average condensation heat transfer coefficient assumes an asymptotic value as the wall heat flux (q_w) is increased. However at lower Froude numbers, such as Fr = 10,000, the condensation heat transfer coefficient is found to increase at higher wall heat fluxes which is due to increased condensation Reynold's numbers.

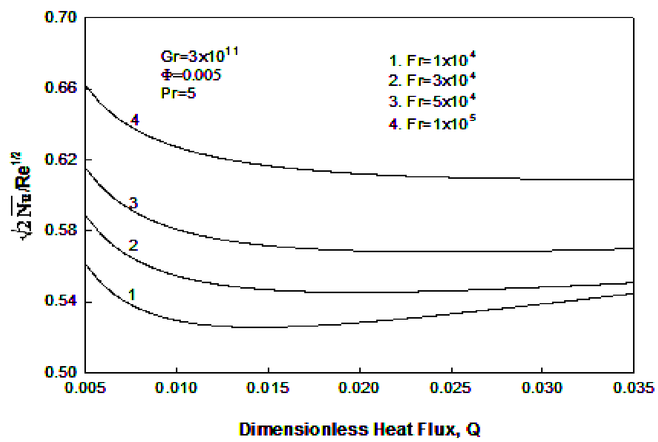

Figure 7. Effect of dimensionless heat flux on mean condensate Nusselt-Influence of Froude

The variation of the Nusselt number with Q is shown in Figure 8 for different values of shear parameter φ. It is found that, the average Nusselt number increases with an increase in shear parameter at any given value of Q.

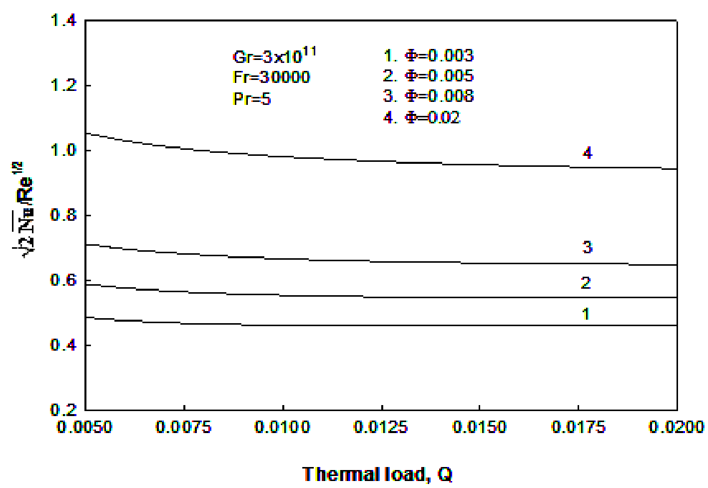

Figure 8. Effect of dimensionless heat flux on mean condensate Nusselt-Influence of interfacial shear

Effect of Shear Parameter, φ

The effect of Prandtl number on Nusselt number is shown in Figure 9 for different values of *Fr* and *φ*. It is found that the Nusselt number increases with an increase in Prandtl number of the condensate. The condensation heat transfer coefficient also increases with an increase in either Froude number or interfacial shear.

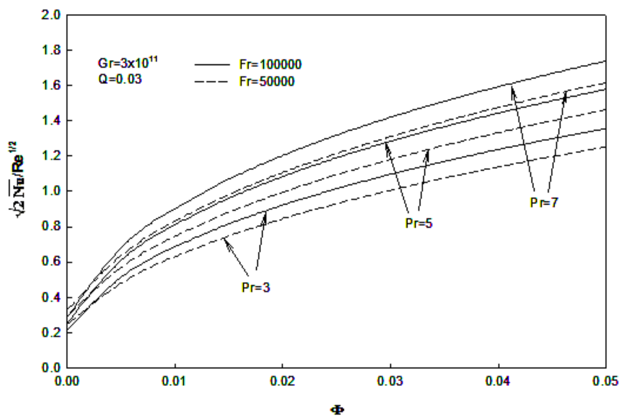

Figure 9. Effect of shear parameter on mean condensate Nusselt

Effect of Inverse Vapor Velocity, F

The analysis is validated with the experimental data of Michael et al. (1989) and is shown plotted in Figure 10. The shear parameter φ comprises of physical properties of the condensate and condenser tube diameter. The ranges of *φ* corresponding to experimental values will be in the range $0.003 \leq \varphi \leq 0.006$ (i.e., system pressure between 0.006 and 0.04 bar) and these magnitudes are computed for the range of the experimental data employed for comparison of present theory. The present theoretical analysis agrees satisfactorily with data for steam. Further, the analytical results of Shekriladze and Gomelauri (1966), Rose (1984), Honda et al. (1986), Sarma et al. (1998) are also shown plotted for comparison in Figure 10.

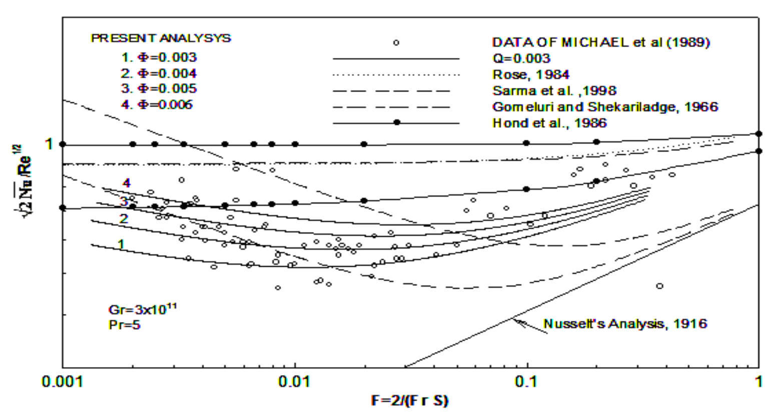

Figure 10. Comparison of present theory with experimental data of Michael el al. (1989) and Rose (1984)

GENERALIZED CORRELATION

The theoretical results of the present analysis for wide ranges of different system parameters are subjected to non linear regression and the following equation for the average Nusselt number is proposed.

$$\frac{\overline{Nu_t}}{Re^{1/2}} = 0.8732 \frac{\varphi^{0.3124}}{(F\overline{S})^{0.03376}\ Q^{0.1306}} \tag{29}$$

Equation (29) is valid for the following ranges of parameters: $0.001 < \varphi < 0.06$; $0.001 < Q < 0.02$. Further Eq. (29) fits the theoretical results with a standard deviation of 5%.

The lowest possible limiting value of Nusselt number is obtained when the vapor is stagnant and condensate film is in the laminar flow. The Nusselt's equation for condensation Nusselt number for the case of laminar film condensation over a horizontal cylinder can be expressed in the following form conveniently.

$$\frac{\overline{Nu}_{lam}}{Re^{1/2}} = 0.4103 \frac{(F\overline{S})^{1/3}}{Q^{1/3}} \tag{30}$$

Thus from Eqs. (29) and (30) one can predict the mean condensate heat transfer coefficients for different ranges of vapor velocity from the following relationship:

$$\overline{Nu}^3 = \left[\overline{Nu}_{lam}^3 + \overline{Nu}_t^3\right] \tag{31}$$

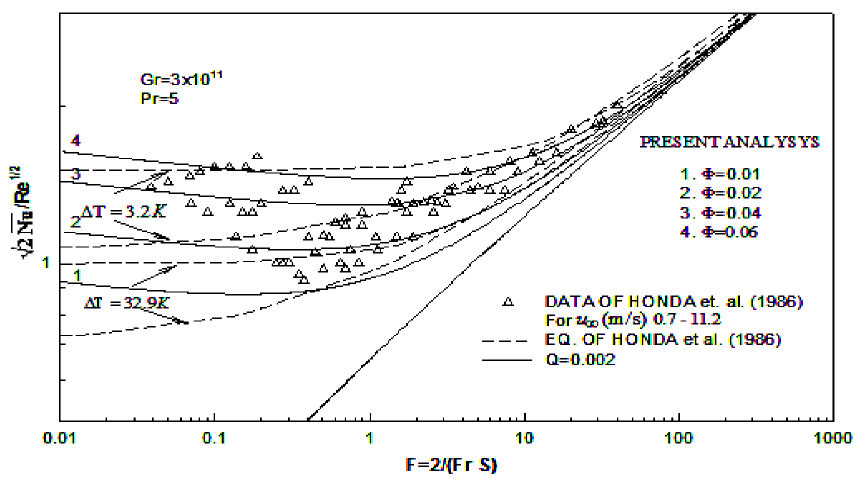

Figure 11. Comparison of present analysis with data of Honda et al. (1986)

Equation (31) of the present analysis is shown plotted together with the experimental data for R-113 in Figure 11. It can be seen that the explicit form of Eq. (31) satisfactorily correlates the experimental data with reasonable accuracy. Hence, Eq. (31) can be used for the design purposes. The comparison of theoretical results for different values of φ with the experimental data of Michael et al. (1989) and Honda et al. (1986)

shown in Figures 10 and 11 respectively indicate clearly the effect of interfacial shear on condensation of flowing vapor over a horizontal cylinder. Hence, the interfacial shear has been shown in this study to be a strong function of system pressure.

CONCLUSIONS

1. The comparison of present analytical results with the available experimental and theoretical data in literature confirms that the present theory is satisfactory for predicting the condensation characteristics.
2. The thermal loading of the condenser tube has an optimal value and increase in flow rate beyond a certain value has no practical significance. It merely increases the pumping power.
3. The increase in approach velocity enhances the mean hear transfer coefficients.
4. The system pressure has a profound influence on condensation heat transfer coefficients, which is brought out in the present study through the interfacial shear parameter φ.
5. A generalized correlation Eq. (29) for average condensate heat transfer coefficient is obtained for high vapor velocities with a standard deviation of 5%**.**
6. Equation (31) is found to be in good agreement with condensation of vapors of R-113 of Honda et al. [5]. Therefore, Eqs. (29), (30) in the form of Eq. (31) can be used for design of condensers for wide range of system parameters.

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Nomenclature

C_p	specific heat capacity of the condensate
D	diameter of the horizontal tube
F	dimensionless inverse vapor velocity, $F = \left(2 / Fr\,\overline{S}\right)$
f	average friction coefficient
f_θ	local friction coefficient
Fr	Froude number, u_∞^2 / gR
Gr	Grashof number, $\left(\dfrac{gR^3}{v_1^2}\right)\left(\dfrac{\rho_l - \rho_v}{\rho_l}\right)$
g	acceleration due to gravity
h	condensation heat transfer coefficient
h_{fg}	latent heat
k	thermal conductivity
Nu	Nusselt number, hR/k_l
$\overline{Nu}$	mean Nusselt number
Pr	Prandtl number
q_w	wall heat flux
Q	dimensionless heat flux, $\left(\dfrac{q_w R}{\mu_l h_{hg}} \mathrm{Re}_l^{-\frac{1}{2}}\right)$
R	outer radius of the tube
R^+	shear radius, $\left(Ru^* / v_l\right)$

R^*	wall shear parameter, $\left(R^+ / Gr^{1/3}\right)$
Re_l	liquid Reynolds number, $\left(u_\infty R / \nu_l\right)$
Re_v	vapor Reynolds number, $\left(u_\infty D / \nu_l\right)$
S	sub-cooling parameter, $\left[c_p\left(T_s - T_w\right)/ h_{fg} \Pr\right]$
$\bar{S}$	average dimensionless wall temperature
St	Stanton number
T	temperature
T^+	dimensionless temperature, $T^+ = k_l\left(T_s - T_w\right)/ q_w R$
u	velocity along the wall
u_∞	approach velocity of steam
u_T	tangential velocity of steam
u^*	shear velocity
u^+	dimensionless velocity, $\left(u/u^*\right)$
y	distance normal to the wall
y^+	dimensionless distance, $\left(yu^* / \nu_l\right)$

Greek symbols

δ	condensate film thickness
δ^+	dimensionless film thickness, $\left(\delta u^* / \nu_l\right)$
ΔT	temperature difference, $\left(T_s - T_w\right)$
ε_m	momentum eddy diffusivity
q	angle measured from the upper stagnation point
μ	absolute viscosity
ν	kinematic viscosity
ρ	density
τ	shear stress
φ	shear parameter

Subscripts

iv	vapor-liquid interface
l	liquid
lam	laminar
s	saturation
t	turbulent
v	vapor
w	wall.