ELSEVIER

Int. J. Therm. Sci. 41 (2002) 295–301

International Journal of Thermal Sciences

www.elsevier.com/locate/ijts

Convective condensation heat transfer in a horizontal condenser tube

P.K. Sarma [a,*], C.V.N. Sastry [b], V.D. Rao [b], Sadik Kakac [c], Hongton Liu [c]

[a] College of Engineering, GITAM, Visakhapatnam 530 045, India
[b] College of Engineering, Andhra University Visakhapatnam 530 003, India
[c] College of Engineering, University of Miami, Miami, FL 33124-0620, USA

Received 17 January 2001; accepted 4 May 2001

Abstract

The purpose of this article is to solve analytically the problem of convective condensation of vapors inside a horizontal condenser tube. Homogeneous model approach is employed in the estimation of shear velocity, which is subsequently, made use of in predicting local convective condensation heat transfer coefficients. The resulting analysis of the present study is compared with some of the available equations in the literature. It is observed that the agreement is reasonably satisfactory validating the assumptions and the theory presented.

Keywords: Convective in; Tube condensation; Heat transfer coefficient; Homogenous model; Friction multiplier

1. Introduction

Investigators considered the two-phase convective condensation process of vapors in tubes in view of its industrial importance variedly as either stratified or annular flow. The possible hydrodynamic flow regimes for given system parameters can be demarcated with the aid of plots proposed by Baker [1] and Taitel and Dukler [2]. At low velocities of vapor, the condensate tends to roll down the wall towards the bottom leading to stratification of heavier component. Kutateladze [3] developed analysis both for stratified and annular flow of the condensate in a horizontal tube. Jester and Kosky [4] considered the problem of convective condensation from the concepts of a homogeneous model. Soliman et al. [5] considered that condensation heat transfer is dependent on the cumulative effect of shear terms associated with momentum, friction and gravity. In their analysis Lockhart and Martinelli [6] correlations were employed. Ananiev et al. [7] proposed a correlation for convective condensation to predict local heat transfer coefficients. Traviss et al. [8] presented a correlation treating local Nusselt as a function of various system parameters as follows for different ranges of Re.

$$Nu_x = F[Re, Pr, x, X_{tt}] \tag{1}$$

Correlation of Shah [9] is regarded as the one satisfying wide range of system pressures, flow conditions and media of varying Prandtl numbers. Carey [10] gave a theoretical analysis in predicting local convective condensation heat transfer coefficients. However, the predictions are on the high side in relation to the values computed from the correlations of other investigators. The review has revealed that many of the analyses are either accomplished through the non-dimensional approach or semi-theoretical considerations with constants being evaluated with the help of the experimental data. The purpose of the investigation is to propose an analytical approach in the estimation of local convective heat transfer coefficients based on certain assumptions and hypothesis.

2. Formulation and analysis

The convective condensation heat transfer inside tubes is generally treated making use of dimensionless analysis resulting in a correlation or as separated flow with the principles of turbulent boundary layer applied to the annular liquid film. The condensation of vapors irrespective of the type of flow regime in the condenser tube can be treated

* Correspondence and reprints.
E-mail addresses: sarmapullela@hotmail.com (P.K. Sarma), vdharmarao@hotmail.com (C.V.N. Sastry), vdharmarao@hotmail.com (V.D. Rao), skakac@miami.edu (S. Kakac), hliu@miami.edu (H. Liu).

1290-0729/02/$ – see front matter
PII: S1290-0729(01)01317-5

Nomenclature

a_1, a_2, a_3, a_4, a_5	coefficients in Eq. (31)	
A	cross sectional area of the tube	m^2
C	constant in Eq. (27)	
C_p	specific heat	$J\cdot kg^{-1}\cdot K^{-1}$
D	diameter of the tube	m
F	see Eq. (33)	
f	friction coefficient	
k	thermal conductivity	$W\cdot m^{-1}\cdot K^{-1}$
g	acceleration due to gravity	$m\cdot s^{-2}$
h	heat transfer coefficient	$W\cdot m^{-2}\cdot K^{-1}$
h_x	local heat transfer coefficient	$W\cdot m^{-2}\cdot K^{-1}$
h_{fg}	latent heat of vaporization	$J\cdot kg^{-1}$
L	length of the condenser tube	m
$\dot{m}$	discharge rate of the condensate	$kg\cdot s^{-1}$
p	exponent in Eq. (16)	
P	pressure	$N\cdot m^{-2}$
P_{CR}	critical pressure	$N\cdot m^{-2}$
Q^+	dimensionless heat flux	
T	temperature	K
T^+	dimensionless temperature $= (T_S - T)/(T_S - T_W)$	
u	velocity	$m\cdot s^{-1}$
u^+	dimensionless velocity $= u/u^*$	
u^*	shear velocity	
x	dryness fraction by weight	
X_{tt}	dimensionless constant in Eq. (27)	
y	distance measured normal to the tube wall	m
y^+	dimensionless distance measured normal to the wall $= yu^*/\nu$	
Z	distance measured from the entry of the tube	m
Nu	Nusselt number $= hD/k$	
Pr	Prandtl number $= \mu C_p/k$	
Re	Reynolds number of the condensate at the exit $= 4\dot{m}/\pi D\mu$	

Greek symbols

ϕ_L	friction multiplier $= \sqrt{\frac{\Delta P_{T.P}}{\Delta P_{L.P}}}$	
ψ	function of X (see Eq. (33))	
τ	shear stress	$N\cdot m^{-2}$
ρ	density	$kg\cdot m^{-3}$
η	dimensionless distance $= Z/L$	
μ	dynamic viscosity	$kg\cdot m^{-1}\cdot s^{-1}$

Subscripts

g	gravity or elevation
L	liquid
m	momentum
V	vapour
W	wall
S	saturation condition
T.P	two-phase
L.P.F	liquid phase friction
T.P.F	two-phase friction
G.P.F	gas phase friction

as a homogeneous model under diabatic conditions with heat removal. In single-phase convective problems related to either heating or cooling of the medium without phase change it is established that modified Reynolds analogy with correction for Prandtl number variation responds favorably and the heat transfer coefficients can be predicted from the momentum transfer analysis. In two-phase convective condensation studies Colburn and Carpenter [11] applied some of these concepts assuming that the wall of the tube is invariably in contact with the liquid medium. Hence, it is suggested that the thermal resistance to heat transfer in the sub layer would play significant role facilitating condensation at the vapor-liquid interface. By definition, the wall heat flux can be written as follows:

$$q_W = h_{T.P}(T_S - T_W) = k_L \left[\frac{\partial T}{\partial y} \bigg|_{Y=0} \right]_{T.P} \tag{2}$$

Alternatively, in non-dimensional form it can be written as follows:

$$\frac{h_{T.P} D}{k_L} = D^+_{T.P} \left[\frac{\partial T^+}{\partial y^+} \bigg|_{y^+=0} \right]_{T.P} \tag{3}$$

where

$$D^+_{T.P} = \frac{D u^*_{T.P}}{\nu_L}, \qquad y^+ = \frac{y u^*_{T.P}}{\nu_L}$$

Similarly, if liquid is flowing with a mass flow rate of $\dot{m}(1 - x)$, the single-phase convective heat transfer coefficient can be expressed as follows.

$$\frac{h_L D}{k_L} = D^+_L \left[\frac{\partial T^+}{\partial y^+} \bigg|_{Y^+=0} \right]_L, \quad \text{where } D^+_L = \frac{D u^*_L}{\nu_L} \tag{4}$$

Hence, from Eqs. (3) and (4)

$$\frac{h_{T.P}}{h_L} = C_3 \frac{D^+_{T.P}}{D^+_L}, \quad \text{where } C_3 = \frac{[\frac{\partial T^+}{\partial y^+}|_{y^+=0}]_{T.P}}{[\frac{\partial T^+}{\partial y^+}|_{y^+=0}]_L} \tag{5}$$

The following definitions are further employed in the formulation

$$\tau_{T.P.F} = \rho_L u^{*2}_{T.P} \quad \text{and} \quad \tau_{L.P.F} = \rho_L u^{*2}_L$$

$$\phi^2_L = \frac{\tau_{T.P.F}}{\tau_{L.P.F}} = \left(\frac{D^+_{T.P}}{D^+_L} \right)^2$$

$$\frac{\Delta P_{T.P.F}}{L} = \frac{4 f_{T.P.F} \rho_{T.P} u^2_{T.P}}{2D} \quad \text{and}$$

$$\frac{\Delta P_{\mathrm{L.P.F}}}{L} = \frac{4 f_{\mathrm{L.P.F}} \rho_{\mathrm{L}} u_{\mathrm{L}}^2}{2D} \tag{6}$$

where $u_{\mathrm{T.P}} = \frac{\dot m}{A\rho_{\mathrm{T.P}}}$ and $u_{\mathrm{L}} = \frac{\dot m(1-x)}{A\rho_{\mathrm{L}}}$.

$$\phi_{\mathrm{L}}^2 = \frac{\Delta P_{\mathrm{T.P.F}}}{\Delta P_{\mathrm{L.P.F}}} \tag{7}$$

where ϕ_{L} is the friction multiplier as defined by Lockhart–Martinelli parameter [6]. Eq. (5) with the help of the definitions shown above can be written as follows

$$\frac{Nu_{\mathrm{T.P}}}{Nu_{\mathrm{L}}} = C_3 \phi_{\mathrm{L}} \tag{8}$$

where $Nu_{\mathrm{L}} = 0.023 Re^{0.8} Pr^{0.4} (1-x)^{0.8}$. Thus, the convective condensation heat transfer is given by the following equation.

$$Nu_{\mathrm{T.P}} = 0.023 C_3 \phi_{\mathrm{L}} Re^{0.8} Pr^{0.4} (1-x)^{0.8} \tag{9}$$

where $Re = \frac{4\dot m}{\pi D \mu_{\mathrm{L}}}$.

Eq. (9) consists of two unknown constants C_3 and ϕ_{L}. These constants will be evaluated for a homogeneous model with certain assumptions. The constant C_3 is dependent on the hydrodynamics of two-phase flow in the core region of the tube. The interfacial shear resistance on the condensate layer depends on the conditions of flow of vapor in the core and consequently it reflects on the gradients of temperature and velocity conditions prevailing at the wall. Besides, in closed systems working on Freon-11, 12, 113, etc., the thermal resistance at the wall of the condenser tube can be influenced by contaminants like traces of lubricants generally employed for efficient functioning of the compressor. The homogeneous model analysis of the present study is tested with an assumed value of $C_3 = 1$. However, in reality the type of flow regime is dependent on the local dryness fraction, flow rates of the two phases and system pressure. The regimes can be prefixed with help of the plots of Baker [1] or Taitel and Dukler [2]. In the present analysis these regimes are ignored treating the flow as a homogenous model. Further, to make use of the present theory the values of ϕ_{L} and $[\rho_{\mathrm{L}}/\rho_{\mathrm{T.P}}]$ implicitly appearing in Eq. (9) must be known. Some relationships applicable to condenser tubes of horizontal orientation are derived.

3. Evaluation of two-phase density, $\rho_{\mathrm{T.P}}$

McAdams [14] defined the two-phase density as follows:

$$\frac{1}{\rho_{\mathrm{T.P}}} = \frac{1-x}{\rho_{\mathrm{L}}} + \frac{x}{\rho_{\mathrm{V}}} \tag{10}$$

However, in the present study a different approach is undertaken. At any section of the condenser tube the sum of the kinetic energies of the two phases must be equal to the total kinetic energy of the two-phase homogeneous two-phase medium possessing a density $\rho_{\mathrm{T.P}}$, i.e.,

$$\left(\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{T.P}}}\right)^2 = \frac{(1-x)^3}{(1-\alpha)^2} + \frac{x^3}{\alpha^2}\left(\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}}\right)^2 \tag{11}$$

where the void fraction $\alpha = A_v/A$.

Zivi [12] proposed the following relationship applying the principle of minimum entropy production.

$$\alpha = \frac{1}{1 + ((1-x)/x)((\rho_{\mathrm{V}})/\rho_{\mathrm{L}})^{2/3}} \tag{12}$$

Baroczy [13] proposed another form of correlation as follows:

$$\alpha = \frac{1}{1 + ((1-x)/x)^{0.74}((\rho_{\mathrm{V}})/\rho_{\mathrm{L}})^{0.65}((\mu_{\mathrm{L}})/\mu_{\mathrm{V}})^{0.13}} \tag{13}$$

Substitution of Eq. (12) in Eq. (11) will yield an expression for two-phase density as follows:

$$\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{T.P}}} = \left[1 + x\left\{\left[\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}}\right]^n - 1\right\}\right]^{1/n} \tag{14}$$

where $n = 2/3$.

Alternatively, $\rho_{\mathrm{T.P}}$ can be expressed in the form

$$\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{T.P}}} = (1-x)^p + x^p \frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}} \tag{15}$$

where the exponent 'p' is unknown and in the present study employing the air–water data of Lockhart Martinelli [6] a correlation is obtained.

$$p = 0.4 - 0.14 \ln(P/P_{\mathrm{cr}}) \tag{16}$$

These relationships, i.e., Eqs. (14), (15), and (16) will be subsequently used.

4. Evaluation of ϕ_{L} for a condenser tube

In a once-through condenser tube the flow of the medium is single phase both at entry and exit of the tube with several regimes occurring in between. This situation allows us to assume a suitable form of two-phase local friction coefficient $f_{\mathrm{T.P.F}}$ satisfying the conditions both at the exit ($x = 0$) and entry ($x = 1$) of the tube. For example, two models are developed as follows.

Model 1 (Linear variation with respect to x).

$$f_{\mathrm{T.P.F}} = (1-x) f_{\mathrm{L.P.F}} + x f_{\mathrm{G.P.F}} \tag{17}$$

Model 2 (Exponential variation with respect to x).

$$f_{\mathrm{T.P.F}} = f_{\mathrm{L.P.F}} \exp(-\lambda x) \quad \text{and} \quad \lambda = \ln(f_{\mathrm{L.P.F}}/f_{\mathrm{G.P.F}}) \tag{18}$$

where for turbulent–turbulent regime of vapor–liquid phases, respectively.

$$f_{\mathrm{L.P.F}} = \frac{0.046}{Re_{\mathrm{L}}^{0.2}}, \qquad f_{\mathrm{G.P.F}} = \frac{0.046}{Re_{\mathrm{V}}^{0.2}}$$

$$Re_{\mathrm{L}} = \frac{4\dot m(1-x)}{\pi D \mu_{\mathrm{L}}}, \qquad Re_{\mathrm{V}} = \frac{4\dot m x}{\pi D \mu_{\mathrm{V}}} \quad \text{and} \tag{19}$$

$$\frac{f_{\mathrm{G.P.F}}}{f_{\mathrm{L.P.F}}} = \left(\frac{\mu_{\mathrm{V}}}{\mu_{\mathrm{L}}}\right)^{0.2}\left(\frac{1-x}{x}\right)^{0.2}$$

Following Lockhart–Martinelli definition of two-phase multiplier, ϕ_{L}, i.e., as given in Eq. (6)

$$\phi_{\mathrm{L}}^2 = \frac{\Delta P_{\mathrm{T.P.F}}}{\Delta P_{\mathrm{L.P.F}}} = \frac{f_{\mathrm{T.P.F}}}{f_{\mathrm{L.P.F}}} \frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{T.P}}} \frac{1}{(1-x)^2} \tag{20}$$

Model 3 (Linear variation of $f_{\mathrm{T.P.F}}$ as per Eq. (17)).

Thus, making use of Eqs. (14), (17) in Eq. (20) the friction multiplier ϕ_{L} can be obtained.

$$\phi_{\mathrm{L}}^2 = \left(\frac{1}{1-x}\right)\left[1 + x\left\{\left(\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}}\right)^{2/3} - 1\right\}\right]^{3/2} \times \left\{1 + \left(\frac{\mu_{\mathrm{V}}}{\mu_{\mathrm{L}}}\right)^{0.2}\left(\frac{x}{1-x}\right)^{0.8}\right\} \tag{21}$$

Model 4 (Exponential variation of $f_{\mathrm{T.P.F}}$ as per Eq. (18)).

$$\phi_{\mathrm{L}}^2 = \frac{\exp[-\lambda x]}{[1-x]^2}\left[1 + x\left\{\left(\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}}\right)^{2/3} - 1\right\}\right]^{3/2} \tag{22}$$

Model 5 (Two-phase density, $\rho_{\mathrm{T.P}}$ as per Eq. (16)).

$$\phi_{\mathrm{L}}^2 = (1-x)^{p-1}\left\{1 + \left(\frac{x}{1-x}\right)^{p} \frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}}\right\} \times \left\{1 + \left(\frac{x}{1-x}\right)^{0.8}\left(\frac{\mu_{\mathrm{V}}}{\mu_{\mathrm{L}}}\right)^{0.2}\right\} \tag{23}$$

where the exponent p can be obtained from Eq. (17).

Model 6. (Employs void fraction relationship α as proposed by Barcozy [13], i.e., Eq. (13) in the exponential variation relationship, i.e., Eq. (18))

where $\lambda = \ln\left(\frac{f_{\mathrm{L.P.F}}}{f_{\mathrm{G.P.F}}}\right)$

$$\phi_{\mathrm{L}}^2 =_1 \frac{\exp[-\lambda x]}{(1-x)^{0.5}}\left[1 + \left(\frac{x}{(1-x)}\right)^{3.0}\left(\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}}\right)^{2.0} \times \left(\frac{1-\alpha}{\alpha}\right)^{2.0}\right]^{0.5} \tag{24}$$

$$\frac{f_{\mathrm{G.P.F}}}{f_{\mathrm{L.P.F}}} = \left(\frac{\mu_{\mathrm{V}}}{\mu_{\mathrm{L}}}\right)^{0.2}\left(\frac{1-x}{x}\right)^{0.2} \tag{25}$$

Where α is given by Eq. (13).

Correlation of Lockhart and Martinelli [6]:

Lockhart and Martinelli [6] proposed the following correlations for ϕ_{L} based on data from a series of studies of air–water data under isothermal conditions for turbulent–turbulent regime of the media

$$\phi_{\mathrm{L}} = \left[1 + \frac{C}{X_{\mathrm{tt}}} + \frac{1}{X_{\mathrm{tt}}^2}\right]^{1/2} \tag{26}$$

where

$$X_{\mathrm{tt}} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_{\mathrm{V}}}{\rho_{\mathrm{L}}}\right)^{0.5}\left(\frac{\mu_{\mathrm{L}}}{\mu_{\mathrm{V}}}\right)^{0.1} \quad \text{and} \quad C = 20 \tag{27}$$

The validity of these equations is to be tested by comparing the convective heat transfer predictions with the results of other investigators for low, medium and high system pressures. The present approach is compared with some of the equations often referred in the literature. Thus, Ananiev et al. [7] gave the following equation

$$\frac{Nu_{\mathrm{T.P}}}{Nu_{\mathrm{L}}} = \left[\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{T.P}}}\right]^{0.5} \tag{28}$$

where

$$\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{T.P}}} = \frac{1-x}{\rho_{\mathrm{L}}} + \frac{x}{\rho_{\mathrm{V}}} \tag{29}$$

$$Nu_{\mathrm{L}} = 0.021 Re^{0.8} Pr^{0.43}$$

Soliman et al. [5] gave the following correlation

$$Nu_{\mathrm{T.P}} = 0.036 Pr^{0.65} Re^{*} \tag{30}$$

where

$$Re^{*} = u_{\mathrm{T.P}}^{*} D/\nu_{\mathrm{L}}$$

and

$$u_{\mathrm{T.P}}^{*} = (\tau_{\mathrm{T.P}}/\rho_{\mathrm{L}})^{0.5}$$

$$\tau_{\mathrm{T.P}} = (\tau_m + \tau_f + \tau_g) \tag{31}$$

where

$$\tau_m = \left(\frac{\dot{m}}{A}\right)^2 \frac{D}{4}\left[-\frac{1}{L}\right]\sum_{n=1}^{5} a_n \left(\frac{\rho_{\mathrm{V}}}{\rho_{\mathrm{L}}}\right)^{n}$$

$$a_1 = (2x - 1 - \beta x)$$

$$a_2 = 2(1-x)$$

$$a_3 = 2(1 - x - \beta + \beta x)$$

$$a_4 = \left[\frac{1}{x} - 3 - 2x\right]$$

$$a_5 = \beta\left[2 - \frac{1}{x} - x\right]$$

and $\beta = 1.25$ and 2.0 (respectively—turbulent and laminar annular liquid film)

$$\tau_g = \frac{D}{4}(1-\alpha) g \sin\theta$$

$$\alpha = \left[1 + \frac{1-x}{x}\left(\frac{\rho_{\mathrm{V}}}{\rho_{\mathrm{L}}}\right)^{2/3}\right]^{-1.0}$$

$$f_{\mathrm{V}} = 0.046\left(\frac{4\dot{m}x}{\pi D \mu_{\mathrm{V}}}\right)^{-0.2} \quad \text{and}$$

$$\tau_f = \frac{D}{4}\phi_{v}^2 \frac{2 f_{\mathrm{V}}}{D\rho_{\mathrm{V}}} x^2, \qquad \phi_{v}^2 = 1 + 2.85 X_{\mathrm{tt}}^2$$

Shah [9] gave the following correlation

$$\frac{Nu_{\mathrm{T.P}}}{Nu_{\mathrm{L}}} = (1-x)^{0.8}\left\{1 + 3.8\left(\frac{x}{1-x}\right)^{0.76}\left(\frac{P_{\mathrm{CR}}}{P}\right)^{0.38}\right\} \tag{32}$$

where $Nu_{\mathrm{L}} = 0.023 Re^{0.8} Pr_{\mathrm{L}}^{0.4}$

Traviss et al. [8] presented the following equation

$$Nu_{\mathrm{T.P}} = 0.15 Pr_{\mathrm{L}} Re^{0.9} \frac{\psi}{F} (1-x)^{0.9} \tag{33}$$

$$\Psi = \frac{1}{X_{\mathrm{tt}}} + \frac{2.85}{X_{\mathrm{tt}}^{0.476}}$$

$$F = 5 Pr_{\mathrm{L}} + 5 \ln\left[1 + Pr_{\mathrm{L}}\left(0.0964 Re_{\mathrm{L}}^{0.585} - 1\right)\right]$$

$$Re_{\mathrm{L}} > 1125$$

Kutateladze [3] proposed the following equation for evaluating mean condensation heat transfer coefficient.

$$Nu_{\mathrm{T.P}} = 0.0387 \left[\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}}\right]^{0.5} \left[\frac{\mu_{\mathrm{V}}}{\mu_{\mathrm{L}}}\right]^{0.1} Re^{0.8} Pr^{0.4} \tag{34}$$

5. Results and discussion

Thus, the present analysis is compared with the equations mentioned and plots for low, medium and high pressures are drawn.in the Figs. 1, 2, 3. It is evident that the hypothesis and assumptions of the present theory favorably predict the local condensation heat transfer coefficients. Obviously, the present results show reasonable agreement with correlations developed by Shah [9] and Traviss et al. [8]. For the sake of clarity in figures the results of models 1, 3 and 5 are only presented since the predications of the other two models are of the same trends and orders. However, it is to be pointed that the correlation equations proposed by the earlier authors agree with the experimental data with an accuracy of ±20–30%. In the context of such variations in the predictions, the present theory can be considered as a satisfactory proposition purporting and validating the assumptions employed in the evaluation of ϕ_{L}. The results are obtained for model 2, i.e., for the case of exponential variation of the two-phase friction coefficient over a wide range of pressures $P = 1$–150 bar for steam–water data. These results are subjected to regression analysis and the following correlation is obtained for 140 points considered in the computations with an average deviation of 8.8% and standard deviation of 11%.

$$\phi_{\mathrm{L}} = 1 + 2.164 \left(\frac{x}{1-x}\right)^{0.85} \left(\frac{P_{\mathrm{cr}}}{P}\right)^{0.57} \tag{35}$$

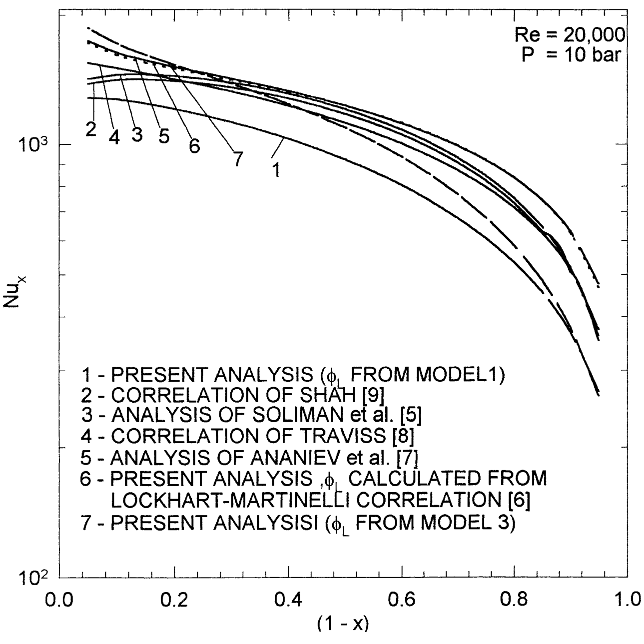

Fig. 1. Variation of local heat transfer coefficient.

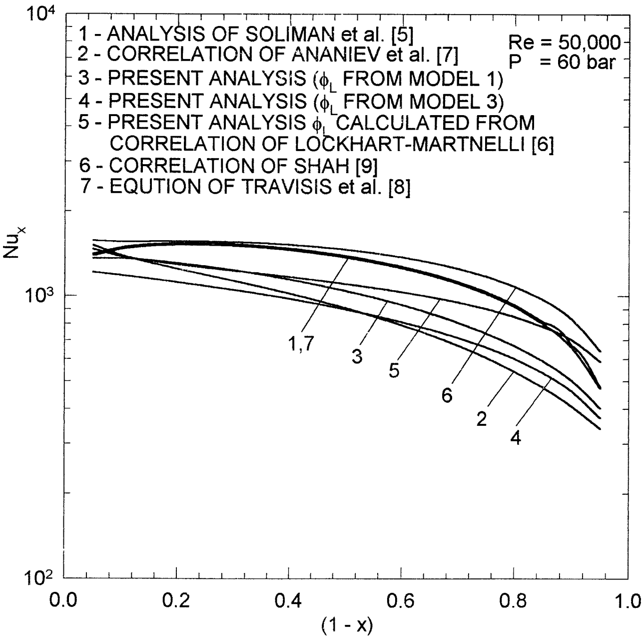

Fig. 2. Variation of local heat transfer coefficients.

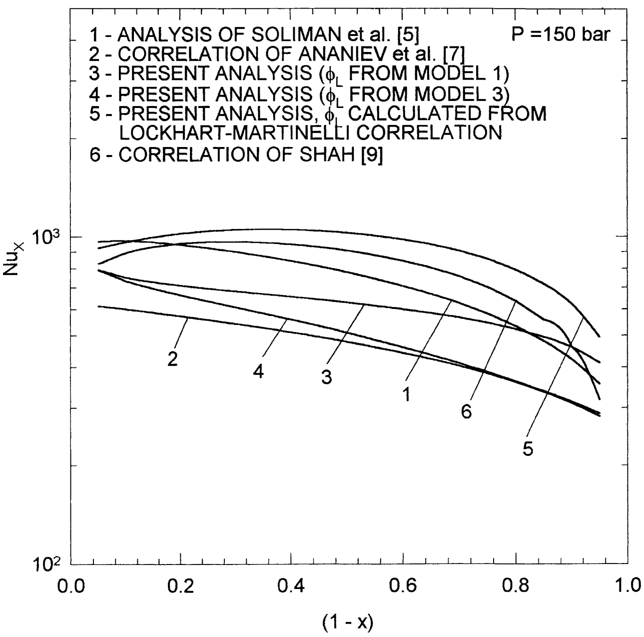

Fig. 3. Variation of local Nusselt number with dryness fraction.

Substituting Eq. (35) in Eq. (9) with $C_3 = 1$, the following expression similar to the correlation of Shah, i.e., Eq. (32) can be obtained.

$$Nu_{\mathrm{T.P}} = 0.023Re^{0.8}(1-x)^{0.8}Pr^{0.4} \times \left[1 + 2.164\left(\frac{x}{1-x}\right)^{0.85}\left(\frac{P_{\mathrm{cr}}}{P}\right)^{0.565}\right] \quad (36)$$

where $Re = \frac{4\dot{m}}{\pi D \mu_{\mathrm{L}}}$

Thus, the difference between the present analysis and the correlation of Shah [9] is only marginal to the extent that the magnitudes of the exponents vary. It can be construed that the present theory offers physical justification to the correlation of Shah.

6. Length of the condenser tube

The average heat transfer coefficients are needed for design. Hence, in Figs. 4 and 5 the results of the present analysis together with those of other investigators are shown. It is evident that the present theory yields the same order of magnitude as the values computed from other analyses. The results of Fig. 4 reveal that as the flow rate increases the average Nusselt number increases indicating that the convective conditions of the two phase media would decrease the thermal resistance at the wall facilitating conditions for phase transformation. Nevertheless from the theory it can be seen that for a given flow rate of the condensate at the exit of the condenser tube the wall must be maintained at a prescribed temperature. For a given system pressure increase in flow rate of the vapor requires substantial decrease in the wall temperature to achieve total condensation, as can be observed from Fig. 5.

In order to estimate the total length of the condenser tube for complete condensation of the vapors, an energy balance for differential element of the tube can be written as follows.

$$h_{\mathrm{X}}(T_{\mathrm{S}} - T_{\mathrm{W}})\pi D\,\mathrm{d}Z = -\dot{m}h_{fg}\,\mathrm{d}x \quad (37)$$

Or in dimensionless form

$$\frac{\mathrm{d}x}{\mathrm{d}(Z/L)} = -4\left(\frac{h_{\mathrm{X}}D}{k_{\mathrm{L}}}\right)\left(\frac{C_{\mathrm{P}}(T_{\mathrm{S}} - T_{\mathrm{W}})}{h_{fg}}\right)\left(\frac{K_{\mathrm{L}}}{\mu_{\mathrm{L}}C_{\mathrm{P}}}\right) \times \left(\frac{1}{Re}\right)\left(\frac{L}{D}\right) \quad (38)$$

Thus, for the case of constant wall temperature integration of the equation between proper

$$\frac{L}{D} = \left(\frac{1}{4\int_0^1 Nu_{\mathrm{X}}\mathrm{d}\eta}\right)\left(\frac{h_{fg}}{C_{\mathrm{P}}(T_{\mathrm{S}} - T_{\mathrm{W}})}\right)Re_{\mathrm{L}}Pr \quad (39)$$

$$\frac{L}{D} = \frac{Re_{\mathrm{L}}}{Q^+}, \quad \text{where } Q^+ = \frac{q_{\mathrm{W}}D}{\mu h_{fg}} \quad \text{and } \eta = \frac{Z}{L} \quad (40)$$

Similarly, for constant heat flux conditions, i.e., $q_{\mathrm{W}} =$ constant.

It can be seen that for constant heat flux conditions the wall temperature will be under non-isothermal conditions and the variation of the dryness fraction is linear along the length of the tube.

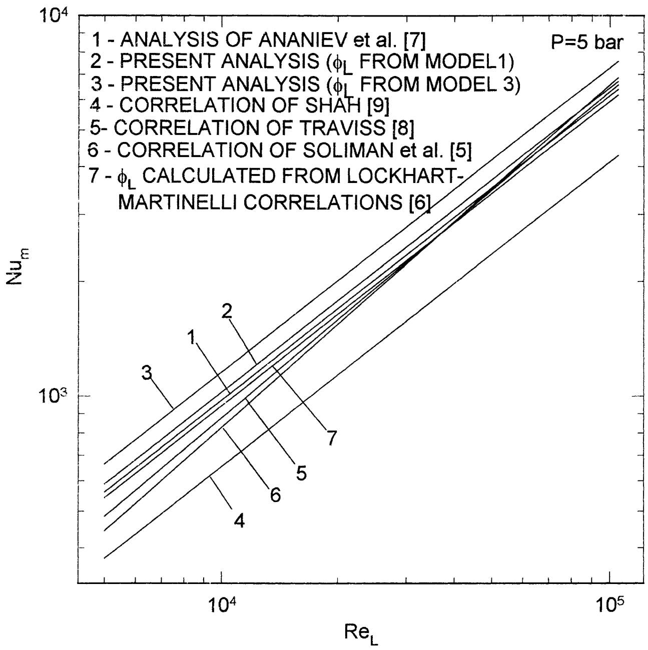

Fig. 4. Variation of maen heat transfer coefficient with exit Reynolds number.

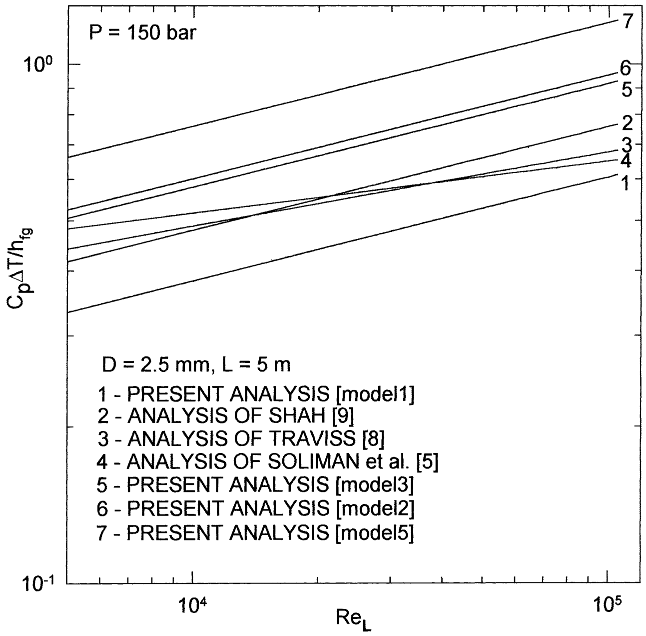

Fig. 5. Variation of ΔT with Re_{L}.

7. Conclusions

(1) The friction multipliers ϕ_{L}, i.e., Eqs. (21)–(24) can be made use of in the estimation of the frictional pressure-drop in condenser tubes. Further, the convective condensation heat transfer coefficients can be evaluated from the

present theory since the present analysis shows reasonable agreement with the equations of other investigators.

(2) The temperature gradients at the wall for single-phase turbulent flow conditions are approximately close to those at the wall for two-phase flow in as much as the wall is always wetted by the condensate and the influence of the hydrodynamic regime in the core can exert marginal influence on the heat transfer rate. Thus, the homogeneous model analysis gave reasonable agreement with the separated flow analysis of Soliman et al. [5] for horizontal flow in the condenser tube. Further, it is an indirect verification of the Carpenter and Colburn [11] proposition that the thermal resistance of the liquid sublayer seems to be the factor controlling the process of condensation.

Acknowledgements

The authors thank D.S.T., Govt. of India, New Delhi and N.S.F, Washington USA for their financial support.

References

[1] O. Baker, Simultaneous flow of oil and gas, Oil Gas J. 53 (1954) 185–195.
[2] Y. Taitel, A.E. Dukler, A model for predicting flow regime transitions in horizontal and near horizontal gas-liquid flow, AICHE J. 22 (1976) 47–55.
[3] S.S. Kutateladze, Aspects of Heat Transfer and Hydraulics of Two-Phase Medium, Energy Publishing House, Moscow, 1961 (in Russian).
[4] H. Jaster, P.G. Kosky, Condensation in a mixed flow regime, Internat. J. Heat Mass Transfer 19 (1976) 95–99.
[5] M. Soliman, J.R. Schuster, P.J. Berenson, A General Correlation for Annular Flow condensation, J. Heat Transfer 90 (1986) 267–276.
[6] R.W. Lockhart, R.C. Martinelli, Proposed correlation of data for isothermal two-phase, two component flow in pipes, Chem. Engrg. Progress 45 (1949) 38–48.
[7] E.P. Ananiev, L.D. Boyko, G.N. Kruzhilin, Heat Transfer in the presence of steam condensation in a horizontal tube, in: 1st Internat. Heat Transfer Conf., Part II, 1961, p. 290.
[8] D.P. Traviss, W.M. Rohsnow, A.B. Baron, Forced convection condensation in tubes: A heat transfer correlation for condenser design, ASHRAE Trans. 79 (1) (1973) 157–165.
[9] M.M. Shah, A General Correlation for Heat Transfer during film condensation inside pipes, Internat. J. Heat Mass Transfer 22 (1989) 547–556.
[10] V.P. Carey, Liquid–Vapor Phase Change Phenomenon, Taylor and Francis, 1992.
[11] F.G. Carpenter, A.P. Colburn, The effect of vapor velocity on Condensation inside tubes, in: Proceedings of the General Discussions of Heat Transfer—The Institute of Mechanical Engineers and the ASME (July 1951), pp. 20–26.
[12] M.S. Zivi, Estimation of steady state steam void fraction by means of the principle of minimum entropy production, J. Heat Transfer 86 (1964) 247–252.
[13] C.J. Baroczy, A systematic Correlation for two-phase pressure drop, AIChE, Reprint 37, presented at the 8th Nat. Heat Transfer Conf., (August 1965), Los Angeles.
[14] W.H. McAdams, Heat Transmission, 3rd edn., McGraw-Hill, New York, 1954.