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Performance study of an evaporator tube working under high heat fluxes

P.K. Sarma [a,*], V. Srinivas [a], K.V. Sharma [b], V. Dharma Rao [c], T. Subramanyam [c], A.E. Bergles [d]

[a] GITAM, Rishikonda, Visakhapatnam 530 045, India
[b] JNTU, Hyderabad, India
[c] College of Engineering, Andhra University, Visakhapatnam 530 002, India
[d] Rensselear Polytechnic Institute, Troy, New York, USA

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Abstract

The performance of an evaporator tube operating under high fluxes with water is studied analytically for possible thermal conditions in the pre- and post-burnout regions. A correlation is proposed for predicting the critical heat flux under slow burnout conditions making use of the concept proposed by Mozharov that the dry-out conditions in the tube arise due to tearing of the liquid film on the periphery due to shearing action of the lighter phase flowing in the core. The correlation is found to reasonably satisfy experimental data in the Russian literature.

Besides a computational procedure is employed to describe the nature of variation of both heat transfer coefficient and thermal potential ($T_W - T_S$) all along the length of the evaporator tube.

Keywords: Slow burnout; Critical heat flux; Dry-out dryness fraction; Nucleate boiling; Mist flow regime

1. Introduction

The steam generating tube under high heat flux conditions must operate within safe working limits of temperature so that overheating of the wall of the tube and subsequent failure of it due to thermal fatigue or melting can be avoided. With the advent of modern power plants the generating capacities at high pressures and high degrees of superheat of steam are selectively considered as the operating parameters to meet the load requirements. In this regard the feed water quality must be within the strict permissible limits of concentration to avoid scaling of the salts. If salt deposition occurs progressively with the operating time thermal failure of the tube is inevitable leading to shutdown to avert catastrophe.

It is established by Sarma [1] and in subsequent articles by Styrikovich et al. [2–6] that the favored location of scaling of salts is where the critical dryness fraction is such that the heat transfer conditions get impaired drastically due to the absence of liquid film on the tube wall. Absence of the liquid film on the inner periphery of tube wall will lead to sudden rise in wall temperature seriously affecting the heat transfer from the wall to two-phase flow in the core. This is termed the critical heat flux (CHF) condition or crisis. There after the Leidenfrost thermal conditions of the wall make it non-wetting and the mechanism of heat transfer is mostly to the convecting steam phase. The liquid droplet laden stream derives thermal energy either due to radiation from the tube wall or by relative convection between the two-phases. The distinguishing feature of the two-phase flow

* Corresponding author. Tel.: +91 891 279 8899; fax: +91 891 2790 32.
E-mail addresses: sarmapk@yahoo.com (P.K. Sarma), vsvas1973@yahoo.com (V. Srinivas), kvsharmajntu@yahoo.com (K.V. Sharma), vdrao@yahoo.com (V.D. Rao), tsmanyam@gmail.com (T. Subramanyam), abergles@aol.com (A.E. Bergles).

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Nomenclature

A	cross sectional area of thee tube, m^2
A_1, A_2, A_3	constants
D	diameter of the tube, m
f	wall friction coefficient
G	mass velocity, kg/m^2 s
h	heat transfer coefficient, W/m^2 °C
k	thermal conductivity, W/m °C
L	length of the tube, m
$\dot{m}$	discharge rate, kg/s
P	system pressure, bar or N/m^2
q	wall heat flux, W/m^2
T_B	bulk temperature, °C
T_W	wall temperature, °C
T_S	saturation temperature, °C
V	vapor velocity, m/s
V^*	critical velocity, m/s
X	dryness fraction
Nu	Nusselt number, $[hD/k]$
Re	Reynolds number, $[GD/\mu]$
Pr	Prandtl number, $[\nu/\alpha]$
$\frac{dp}{dl}$	pressure drop across length

Greek symbols

α	thermal diffusivity, m^2/s
δ	liquid film thickness,
ρ	density kg/m^3
μ	absolute viscosity, Pa s
ν	kinematic viscosity, m^2/s
σ	surface tension, N/m

Subscripts

con	convection
l	liquid phase
v	gas/vapor phase
tpf	two-phase mixture
cr	critical value
W	at wall
S	saturation
T	transition

on the downstream side of the CHF location is that non-wetting conditions of the wall to the impinging droplets on to the tube wall render it into an inefficient evaporation characterized by the film boiling process. Thus, in a once-through system of steam generation the heat transfer conditions all along the length of the tube are of varying nature leading to non-monotonic variation of the wall temperature.

The CHF may be either "fast" or "slow" – often referred to as "fast burnout" and "slow burnout". There is literature available in abundance on fast burnout [9–11] and slow burnout [14–23]. The fast burnout mostly results in physical destruction of the tube that occurs when the coolant is under subcooled conditions. These investigations are primarily devoted to fix the CHF in nuclear reactors operating under high heat fluxes. The slow burnout, which is specific to steam generating tubes, is not as deleterious as in the case of fast burnout provided the concentrations of impurities in feed water in the form of salts is maintained strictly within norms.

In the literature there are reviews related to convective heat transfer related to single-phase fluid, subcooled and nucleate boiling and mist flow heat transfer in the liquid deficit region of the wall of the tube.

The purpose of the present investigation is two-fold, firstly to establish a correlation based on the extension of the concept proposed by Mozharov [7] and secondly to predict the performance of the steam generating tube in terms of thermal characteristics viz., the wall temperature variation all along the length and the variation of heat transfer coefficients for the total range $0 < X < 1$ considering various flow regimes. Such analysis is generally treated as an essential feature in computations related to once-through evaporation systems.

2. Model of Mozharov

Mozharov in his experimental study [7] of air and water under adiabatic conditions concluded that there exits a critical velocity of the air under given system conditions at which the liquid will be totally in the core in the form of droplets and the liquid film on the inner periphery will be torn leading to dry conditions of the wall. The criterion employed by him is essentially a balance of surface tension forces and inertial forces. Specifically a correlation is arrived at in terms of dimensionless parameters as follows:

$$\left(\frac{G_v^2 D}{\sigma \rho_v}\right)^{1/2} = 115\left(\frac{X}{1-X}\right)^{1/4} \tag{1}$$

where G_v, critical mass velocity of the vapor $[GX]$; D, diameter of the tube; σ, surface tension; ρ_v, density of the lighter phase.

Levitan and Lantsman [12,13] proposed subsequently for diabatic conditions other empirical correlation. For the second type of burnout for steam–water the critical dryness fraction is considered as a function of the system pressure and mass velocity of the two-phase flow as follows and the heat flux does not appear in the correlation:

$$X_{cr} = \left[0.39 + 1.57\left(\frac{P}{98}\right) - 2.04\left(\frac{P}{98}\right)^2 + 0.68\left(\frac{P}{98}\right)^3\right]\left(\frac{G}{1000}\right)^{-0.5} \tag{2}$$

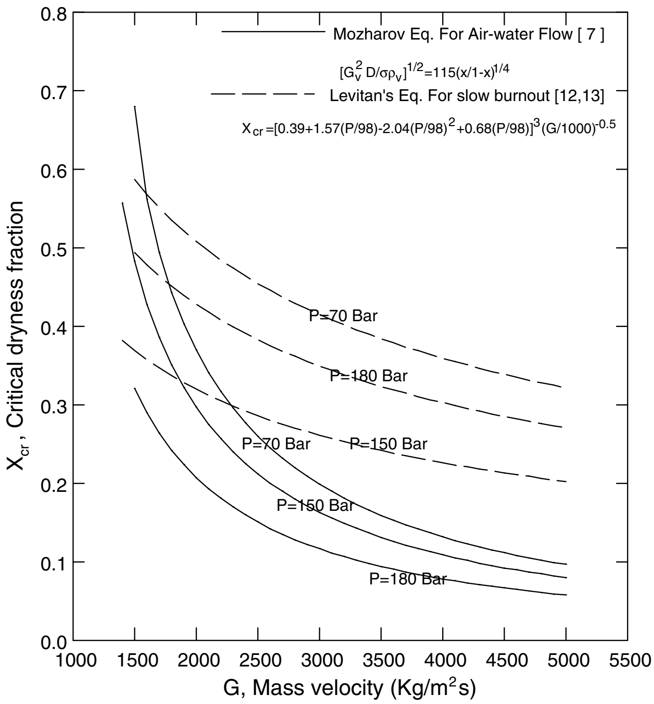

Fig. 1. Comparison of Mozharov's results with predictions of Levitan et al. [12,13].

For critical heat flux of the first kind in an 8-mm tube the critical heat flux is given by the following relationship:

$$q_{cr} = 10^6 e^{(-1.5X)}\left[10.3 - 7.8\left(\frac{P}{98}\right) + 1.6\left(\frac{P}{98}\right)^2\right] \times \left(\frac{G}{1000}\right)^{1.2\{[0.25(P-98)]-X\}} \tag{3}$$

The suggested ranges of applicability of Eqs. (1) and (2) are $29.4 < P < 196$ bar and $750 < G < 5000$ kg/m^2 s. However, a correction factor is suggested to include the diameter effects for data related to tubes other than 8 mm. Because of similarities in dependence of the parameters chosen by them, the equations of Mozharov [7] and that of Levitan's [12,13] are compared in Fig. 1. Except for the pressure dependence, the trends are found to be the same. Thus, to include further data covering a wider range, i.e., $8 < D < 10$ mm, $500 < G < 4500$ kg/m^2 s and $100 < P < 200$ bar from the available literature the experimental data of Peskov et al. [17] are chosen. An expanded model of Mozharov is adapted.

3. Correlation for slow burnout

In the present investigation the following system of criteria is assumed to describe slow burnout.

$$\frac{q_{cr}\delta}{\mu_l h_{fg}} = F\left[\frac{V}{V^*}, \frac{\rho_l}{\rho_v}, X, (1-X), \frac{P}{P_{cr}}\right] \tag{4}$$

where V, is the superficial vapor velocity, GX/ρ_v; V^*, is Mozharov's critical velocity as derived from Eq. (1); δ, is the liquid film thickness just before on set of DNB.

The equation of Bergles and Yadigaroglu [8] is considered as the minimum liquid film thickness on the inner periphery just before the onset of crisis and it is given by the relationship as follows:

$$\delta = \frac{10\mu_l}{\rho_l}\sqrt{\frac{4\rho_l}{\left(\frac{dp}{dl}\right)_{tpf} D}} \tag{5}$$

The data of Russian investigators [17] are chosen for establishing the exact dependence between the parameters of Eq. (4). A regression analysis resulted in a dimensionless correlation as follows:

$$\frac{q_{cr}\delta}{\mu_l h_{fg}} = 2.74\left(\frac{\rho_l}{\rho_v}\right)^{-1.7} X^{0.116}(1-X)^{2.65}\left(\frac{V^*}{V}\right)^{0.35}\left(\frac{P}{P_{cr}}\right)^{-2.95} \tag{6}$$

In Fig. 2 the data are shown plotted along with the correlation, i.e., Eq. (6). The critical heat flux can be correlated with an accuracy of ±19% for the range $100 < P < 200$ bar; $500 < G < 4500$ kg/m^2 s. Subsequently, to establish the validity of the present correlation, in Fig. 3 some of the equations often referred to in the literature are picked up and shown plotted. The present equation agrees very well with that of Katto and Ohno [19] and that of Bowring [14]. On the same plot the correlations such as Biasi et al. [15] and Bertoletti [16] are shown. These predictions substantially differ from the present correlation. In Fig. 4, the present analysis is shown in a different coordinate system with the predictions of Levitan et al. [12,13]. Especially the predictions from the present study are in very close

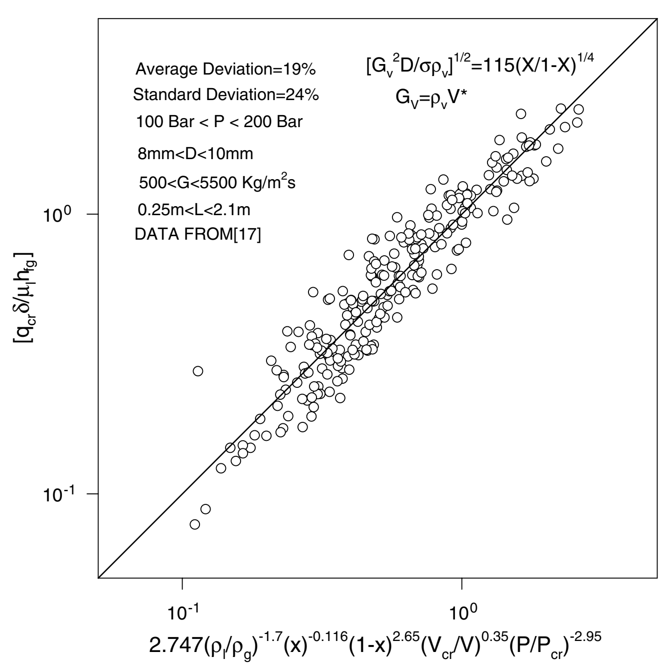

Fig. 2. Validation of critical heat flux equation.

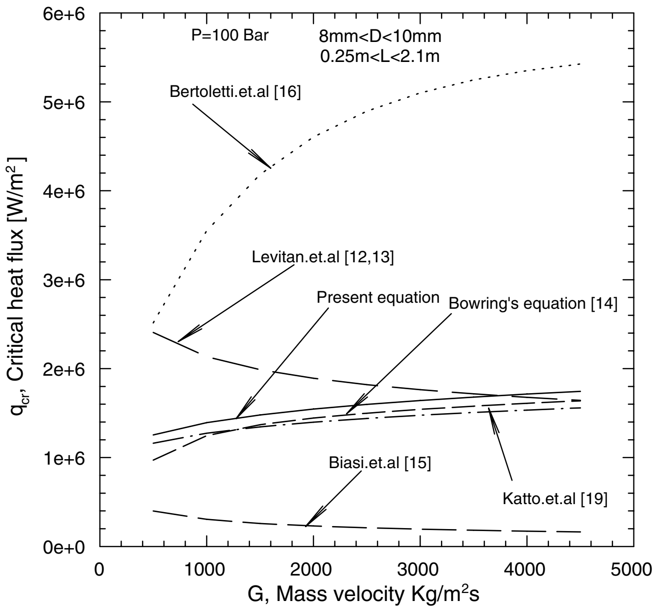

Fig. 3. Comparison of present equation with referred equations.

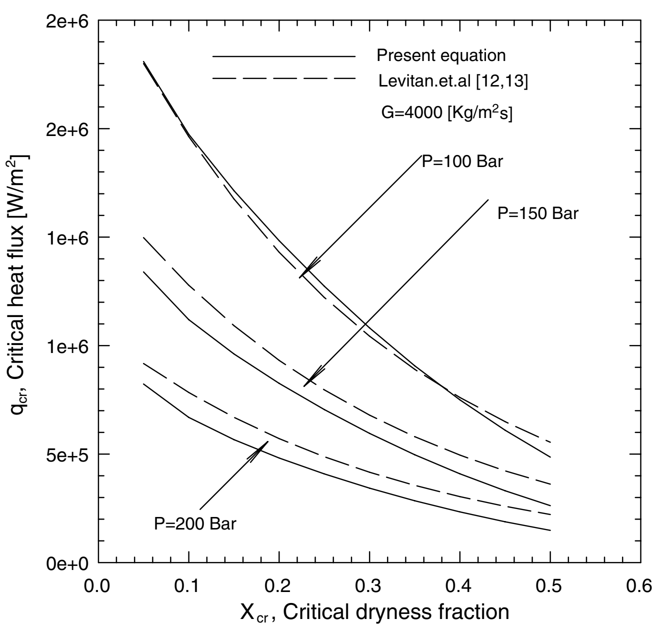

Fig. 4. Variation of critical heat flux with critical dryness fraction.

agreement with those of Levitan et al. [12,13] at $P = 100$ bar. At higher pressures Levitan's equation predicts slightly higher magnitudes. This correlation is used to estimate the performance of once-through evaporator tube from $X = 0$ to 1.

It is noted that at higher operating pressures, forced convection is opted in preference to natural circulation since buoyant forces substantially decrease due to the fact that $\frac{d[\rho_l - \rho_v]}{dp} \to 0$.

This implies that two-phase hydrodynamic conditions are closely interlinked to the thermal conditions as well.

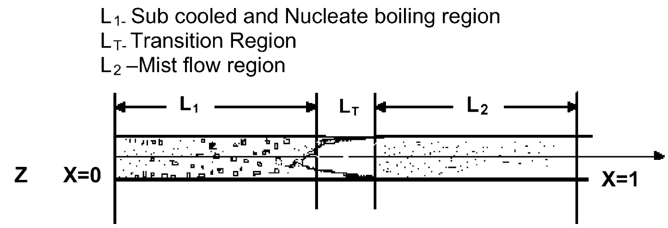

Fig. 5. Configuration of the evaporator tube.

In heat transfer studies two types of thermal conditions viz., constant wall temperature and constant heat flux conditions are generally analyzed. But in high pressure systems employing different modes of firing systems one would expect neither constant wall temperature nor constant heat flux conditions. It can be a hybrid systems leading to variable heat flux or constant wall temperature.

Hence, a typical case of dryness fraction variation in a tube is considered for further study and variation of both heat flux and wall temperature along the length of the tube. The evaporator tube of total length L is divided into three sections between $0 \leqslant X \leqslant 1$. as shown in Fig. 5 with limited transition zone L_T in between.

The region of subcooled and nucleate boiling is studied in detail by many investigators and the process of boiling is found to be hardly dependent on the mass velocity of flow. For low quality forced convective boiling often referred correlations are due to Jens and Lottes [9], Weatherhead [10] and Thom et al. [11]. In the present study the equation of Thom et al. [11] is considered.

The equation of Thom is as follows:

$$[T_W - T_S] = \frac{40}{e^{\left[\frac{P}{18522}\right]}}\left[\frac{q}{3.17 \times 10^6}\right]^{0.5} \tag{7}$$

Though it is established that the nucleate boiling transforms to annular film evaporation with boiling being absent in the annular liquid film as per the observations of Bennett et al. [18] the transition from the regime of net vaporization due to boiling to annular evaporation is not qualitatively established in terms of criteria so far. Hence, in the present study the correlation of Thom is assumed to be the valid relationship up to the transition point. As an approximation the transition zone is assumed to lie in the range $0.95q_{cr} < q < q_{cr}$. After transition, the regime is assumed to transit to the liquid deficit region with liquid in the droplet configuration in the core. Thus, the following computational procedure is employed in establishing the thermal characteristics of the tube.

4. Numerical procedure

1. Prescribe system conditions such as, G mass velocity, D diameter of the tube, P system pressure. The inlet condition of the feed water at entry, i.e., $Z = 0$ is assumed as $X = 0$ at a bulk temperature, $T_B = T_S$.

2. Iterate on the value of $(T_W - T_S)$ such that $q_{con} = q_{Thom}$, where $q_{con} = 0.023Re^{0.8}Pr^{\frac{1}{3}}(T_W - T_S)$ and fix $(T_W - T_S)$ at $X = 0$.

   $$q_{Thom} = 3.17 \times 10^6 \left\{0.025(T_W - T_S)e^{\left[\frac{P}{18522}\right]}\right\}^2$$

   Calculate $X_{dry\text{-}out}$ from Levitan and Lantsman's equation [13]

   $$X_{dry\text{-}out} = \left[0.39 + 1.57\left(\frac{P}{98}\right) - 2.04\left(\frac{P}{98}\right)^2 + 0.68\left(\frac{P}{98}\right)^3\right] \times \left(\frac{G}{1000}\right)^{-0.5}$$

3. Calculate the slow burnout heat flux q_{cr} from the present study, i.e., refer Eq. (6) from the present study.
4. Consider finite number of nodes say, $J = 25$ on the upstream with respect to burnout point, i.e., at $X = X_{dry\text{-}out}$. Choose a step size in X as

   $$\Delta X = (X_{dry\text{-}out} - X_{in})/J$$

   where $X_{in} = 0$ as per the initial assumption.
5. Estimate the corresponding heat fluxes at these nodes since $q_{con} \leqslant q \leqslant q_{cr}$ for the corresponding range $0 \leqslant X \leqslant X_{dry\text{-}out}$ since the heat flux is supposed to vary between the limits q_{con} and q_{cr}

   $$q_{X+\Delta X} - q_X = \frac{0.95q_{cr} - q_{con}}{J}$$

   Such a procedure implies a linear variation of heat flux along the length of the tube.
6. Make use of Thom's equation of step 2 to compute $\Delta T = (T_W - T_S)$ for specific values of q the wall heat flux at corresponding nodes, i.e., $1 \leqslant J \leqslant 25$.
7. Fix the length of the evaporator tube for the range $0 < X < X_{cr}$ on the assumption that the principle of conservation of energy holds well in the tube till burnout occurs, i.e., at least till $q < 0.95q_{cr}$

   $$q\pi D\,dZ = \dot{m}h_{fg}\,dX$$

   The equation in finite difference form with the variable node I can be written as follows:

   $$\Delta Z = Z(I+1) - Z(I) = \frac{\dot{m}h_{fg}\Delta X}{0.5[q(I+1) - q(I)]\pi D} \tag{8}$$

   Thus $Z(J) = L_1$ is the primary length of the evaporator tube for $0 < X < X_{cr}$.
8. For $Z > L_1$ the mist flow regime commences as per the model. The rise in wall temperature at the transition can be estimated from the relationship.

   $$\Delta T_{cr} = (T_W - T_S)_{cr} = \frac{q_{cr}}{h_{Roshenow}[\text{at } X = X_{cr}]} \tag{9}$$

   where $h_{Roshenow}$ [24–27] can be calculated from the equation

   $$\frac{h_{Roshenow}D}{k_v} = 0.023\left[\frac{GD}{\mu_v}\left(X + \frac{\rho_v}{\rho_l}(1 - X)\right)Pr_v^{0.4}\right]$$

9. The change in the wall heat flux after the crisis on the down streamside of the dry-out point can be calculated based on a model as follows.

The wall heat flux is prescribed by a polynomial of second degree with unknown coefficients.

$$\frac{q}{q_{cr}} = A_1 + A_2\left[\frac{X}{X_{cr}}\right] + A_3\left[\frac{X}{X_{cr}}\right]^2 \tag{10}$$

The constants A_1, A_2, A_3 can be evaluated with the help of the boundary conditions viz. At $X = X_{cr}$, $q = q_{cr}$

$$\text{At } X = X_{cr}, \quad q = q_{cr}$$
$$\text{At } X = 1, \quad \frac{\partial q}{\partial X} = 0 \tag{11}$$

The second boundary condition is from the physical reasoning that in the post-burnout region that the wall is under non-wetting conditions and the liquid droplet concentration gradually decreases as $X \to 1$ and hence the thermal transport due to evaporation decreases till two-phase mist flow transits to single-phase forced convection of steam phase.

Application of the boundary conditions will yield a polynomial of the type

$$\frac{q}{q_{cr}} = 1 - 2\frac{A_3}{X_{cr}}\left[1 + \frac{X}{X_{cr}}\right] + A_3\left[1 + \left(\frac{X}{X_{cr}}\right)^2\right] \tag{12}$$

Eq. (12) still contains one unknown A_3, which is yet to be evaluated.

A_3 will be subsequently evaluated from the condition that a reasonable length of the tube, i.e., $Z = L_2$ is required in the post-burnout region such that $X \to 1$. In other words the conservation equation is to be satisfied

$$q\pi D\,dZ = \dot{m}h_{fg}\,dX \tag{13}$$

Substituting the profile equation (12) in Eq. (13) and integrating the same between appropriate limits

$$\int_{(L_1/L)}^{1} d\left[\frac{Z}{L}\right] = \int_{1}^{[1/X_{cr}]} \frac{\dot{m}h_{fg}X_{cr}\,d\left[\frac{X}{X_{cr}}\right]}{\pi D q_{cr}\left\{1 - 2\frac{A_3}{X_{cr}}\left(1 + \frac{X}{X_{cr}}\right) + A_3\left[1 + \left(\frac{X}{X_{cr}}\right)^2\right]\right\}} \tag{14}$$

The value A_3 is obtained to satisfy Eq. (14), i.e., LHS = RHS.

5. Discussion of the results

For different system parameters in Figs. 6–9 the characteristics are shown plotted. Results in Fig. 6 indicate that for a given pressure $P = 100$ bar the critical dryness fraction is found to decrease with the increase in the mass velocity and the nature of variation of dryness fraction in the pre- and post-burnout regions are not the same since

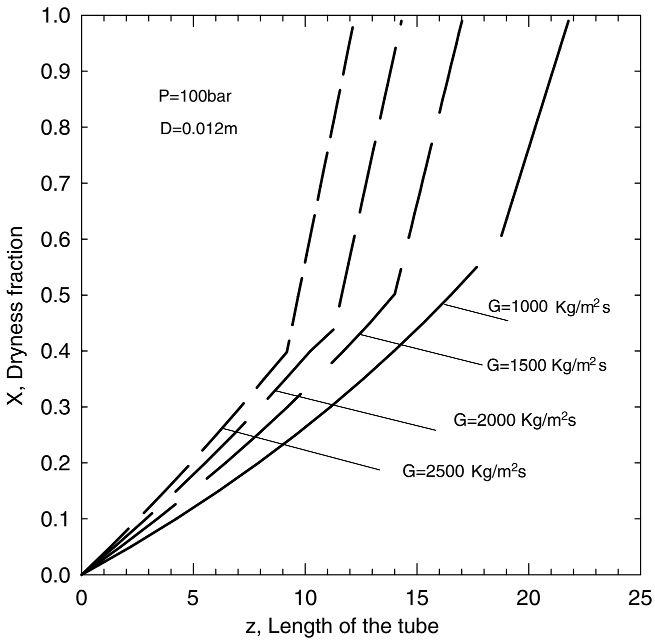

Fig. 6. Typical variation of dryness fraction along length of the evaporator tube.

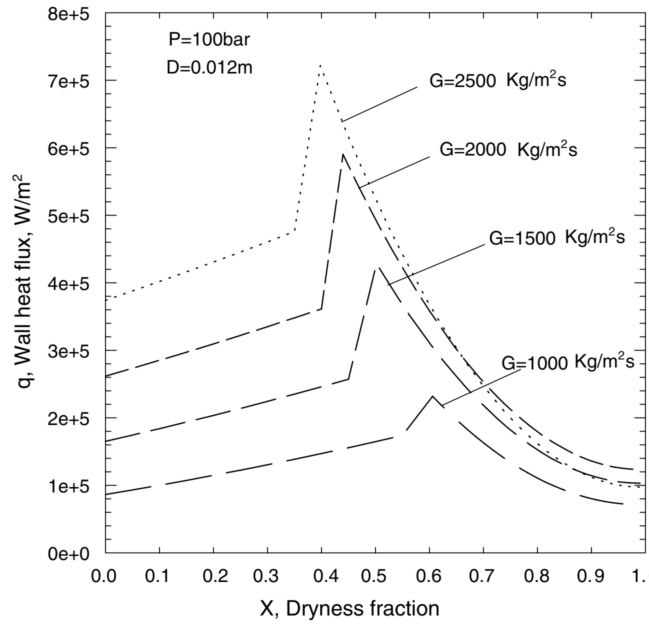

Fig. 8. Variation of wall heat flux with dryness fraction.

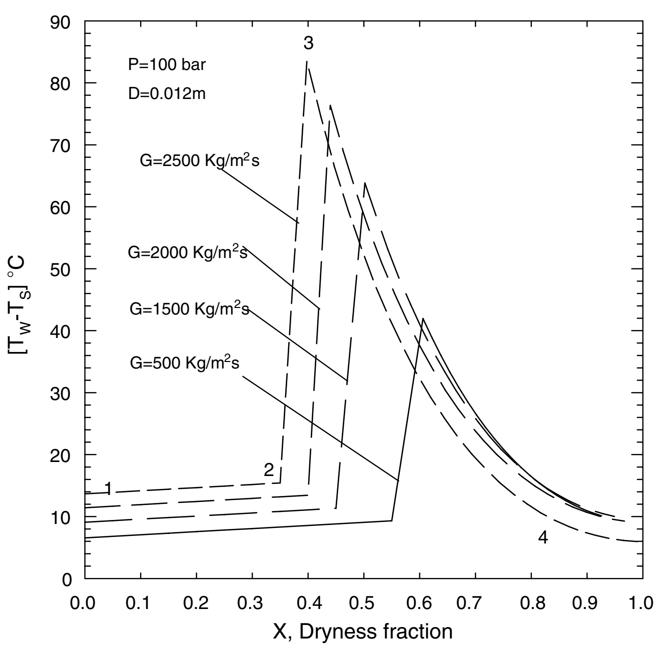

Fig. 7. Variation of wall thermal potential with dryness fraction.

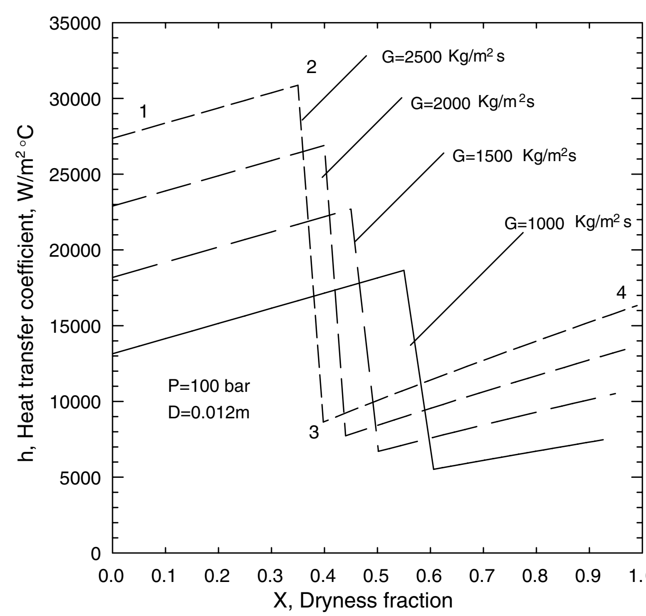

Fig. 9. Variation of local heat transfer coefficient with dryness fraction.

the thermal transport phenomena are of different nature in these regions.

For the corresponding variation of the dryness fraction shown in Fig. 6, the variation of thermal potential $\Delta T = (T_W - T_S)$ is shown plotted in Fig. 7. According to the model envisaged in the analysis, the following demarcation of thermo-hydraulics regimes can be observed. 12-corresponds to subcooled and nucleate boiling regimes: 23-corresponds to transition which includes film evaporation and dry-out conditions and at point 3 the temperature attains maximum value. As per the model it can be seen that the dry-out dryness fraction decreases with the increase in mass velocity of flow. In addition when the dry-out conditions occur, ΔT is found to increase with the increase in mass velocity. In Fig. 8, the wall heat flux is found to increase up to the point of onset of the dry-out conditions. Subsequently in the mist flow regime, the wall heat flux decreases monotonically to a minimum value at $X = 1$. Beyond $X > 1$ the flow is purely convective heat transfer to single-phase steam without droplet contamination. In Fig. 9 the variation of the heat transfer coefficient

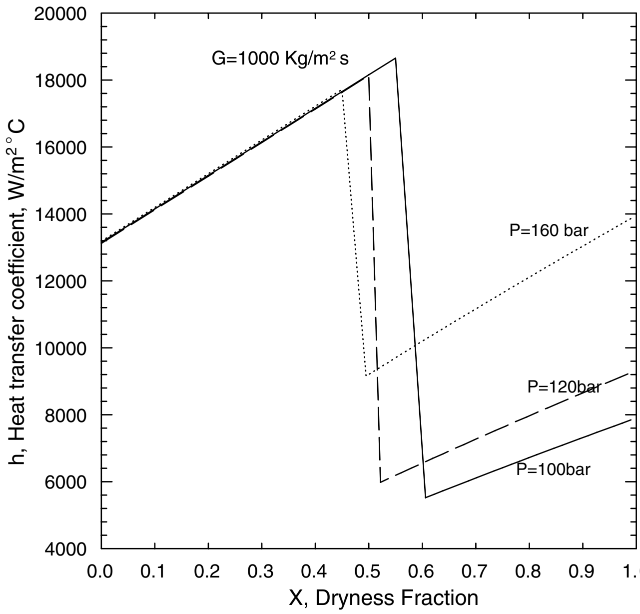

Fig. 10. Effect of pressure on heat transfer coefficient.

with dryness fraction is shown plotted. In the region 12 the heat transfer coefficient gradually increases and reaches maximum at the point 2 and when dryness of the liquid film sets in heat transfer substantially decreases. However, the convective heat transfer in the mist flow region 34 is found to gradually increase. The influence of pressure on heat transfer coefficients is shown plotted in Fig. 10. For the ranges of pressure tested in the analysis the pressure effect does not seem to be profound in the nucleate boiling regime. However the transition to dry-out conditions is influenced by pressure. As the pressure increases the dry-out dryness fraction decreases. In the mist flow regime

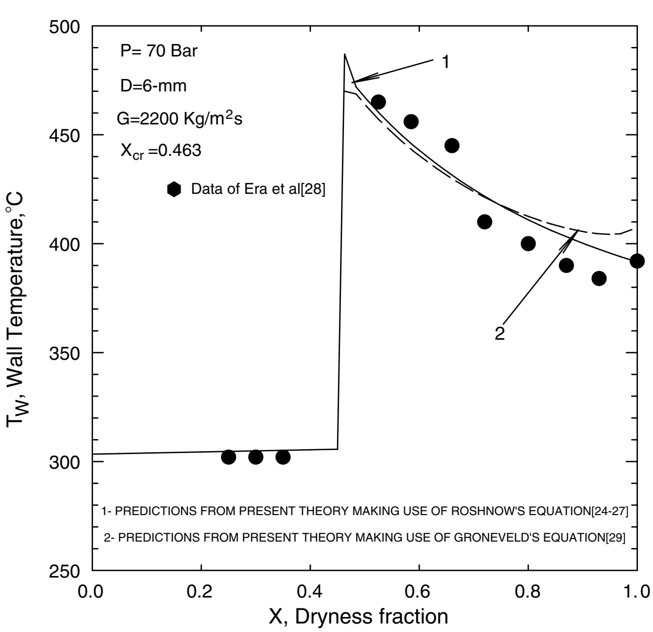

Fig. 11. Comparison of present theory with data of Era et al. [28].

the pressure effect is found to be substantial. To check the validity of the present analysis the computational procedure is repeated for $P = 70$ bar and $G = 2200$ kg/m^2 s the data of Era et al. [28] in Fig. 11. Since the present correlation, i.e., Eq. (6) is only for the given range of data $10 < D < 12$ mm, the diameter correction as suggested by the Russian investigators [30], i.e., $\frac{q_{cr}}{q_{cr \text{ at } (D=8 \text{ mm})}} = (D/8)^{1./2}$ is applied to the value derived from Eq. (6) of the present analysis. It is obvious from Fig. 11 that the present approach has given reasonable agreement with the data of Era et al. [28] in the post-burnout region.

6. Conclusions

Thus, the following conclusions can be arrived at from the study undertaken:

1. The slow burnout wall heat flux can be evaluated from Eq. (6) of the present analysis. This correlation is developed from the concept of Mozharov two-phase flow studies for air–water system without heat addition.
2. A method is outlined to study the performance of steam generating tube under thermal conditions closer to slow burnout. The observed results from the computations indicate trends of rise-fall-rise in the corresponding zones of 12-23-34 respectively (see Figs. 7–9). The computational procedure employed can be used as a first approximation in the design of a steam generating tube.

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