AN ANALYTICAL MODEL TO PREDICT HEAT TRANSFER COEFFICIENT FOR CONDENSATION INSIDE LONGITUDINALLY FINNED TUBES

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ABSTRACT

The analytical model developed by Kaushik and Azer (1989), to predict the heat transfer coefficient for condensation inside longitudinally finned tubes was modified to include the effect of the gravity force on the liquid film in addition to the surface tension force of the original model. Also the analysis was modified to include a trapezoidal fin profile rather than a rectangular profile. The parametric study, using the present model showed that the gravity force has very small effect while, the number, height, width, angle and material of fin have a significant effect on condensation heat transfer coefficient.

Keywords: Heat transfer, Finned tubes, Gravity force, Surface tension, Fin profile.

INTRODUCTION

Several analytical models were developed to investigate the outside heat transfer enhancement by the use of externally finned tubes during condensation. Non has been attempted for condensation inside internally finned tubes except the analytical model developed by Kaushik and Azer (1989). The following is a brief review of a few references on condensation on externally finned tubes to show the effect of surface tension on the condensation process and the relation between this effect and fin geometry.

Beatty and Katz (1948) were the first to propose a model for the condensation of vapors on outside of finned tubes. Their results were quite reasonable only for low surface tension fluids. Krakhu and Borovkov (1971) developed an analytical model to predict the condensation heat transfer coefficient of steam on a horizontal tube with trapezoidally shaped fins. They showed that surface tension played an important role in enhancing the heat transfer. They stated that their model could be applied only when surface tension have marked effect i.e for high values of Bond number, Bo, ($Bo \geq 10.$).

Edward et al. (1973) developed a model for evaporation or condensation with liquid filled circumferential grooves in which surface tension determines the free surface configuration of meniscus. The effect of groove pitch, half angle, and overhead feed rate were shown to be interconnected and highly significant. On the other hand, a few experimental studies such as the study of Shklover et al. (1981) did not show any improvement in the heat transfer coefficient for finely finned tubes during condensation of steam. There were also extensive studies such those of Bell (1983), and Rudy and Webb (1985) which showed that the surface tension effect on condensate retention caused a decrease in heat transfer enhancement. The investigation of Rudy and Webb (1985) proposed a model to predict the condensate retention on horizontal integral finned tubes. Webb et al. (1985) developed their model, using Rudy and Webb's model (1985) to predict the condensation heat transfer coefficient. Their results agreed fairly well with the experimental data. The model of Honda and Nozu (1987) considered many effects that were not considered by previous investigators. Their model predicted the experimental data fairly well.

From the above review, it can be seen that the surface tension effect plays an important role in enhancing the condensation heat transfer on finned tubes. The value of this effect is dependent mainly on the fin geometry.

Numerous correlation, to predict the condensation heat transfer coefficient inside internally finned tubes have been proposed. Although these correlations have been useful as a design tool, they do not explain the influence of the various geometric parameters of the fins and condensing fluid properties. Also, these correlations are limited to the range of experimental parameters from which they were developed. As mentioned earlier, only Kaushik and Azer (1989) proposed a model which dealt with this problem. They concluded that their model predicted well the experimental data for higher surface tension fluids and over predicted the experimental data for lower surface tension fluids. In their model, they neglected the gravity and buoyancy forces and considered only the surface tension and viscous forces. Also, they approximated the trapezoidal fins profile by a rectangular shape neglecting the fin angle.

In the present model the gravity and buoyancy forces were considered in addition to the surface tension and viscous forces. Also, the actual profile of fins is considered where, the fin angle could play an important part.

ANALYSIS

Physical Model

In this model, it is assumed that the condensing fluid is drained in the valley between any two adjacent fins by the net force produced from the balance of surface tension and gravity forces. The condensate also forms a circular film, between two fins, whose thickness is less than the fin height. Thus, this model can be applied at least in the upstream section of the tube and as long as the thickness of the liquid film is less than the height of the fin. Also, the surface tension force must be larger than the gravity force to ensure the draining of condensate to the valleys between fins i.e when Bond number, Bo is more than 1.0, (Bo>1.).

The following assumptions were made;

1- The fluid properties in the condensate film are constant.
2- The pressure of the vapor in the condenser tube at any location is constant.
3- The temperature of the fin is constant and equal to the tube wall temperature.
4- The temperature distribution in the condensate film on the fin is linear.
5- Momentum changes through the film are neglected i.e., static balance of forces is considered.
6- Referring to Figure (1), surface AB of the fin and surface CD of the tube contribute to the heat transfer.
7- The heat transfer at the fin tip was included in the calculation of the fin efficiency.

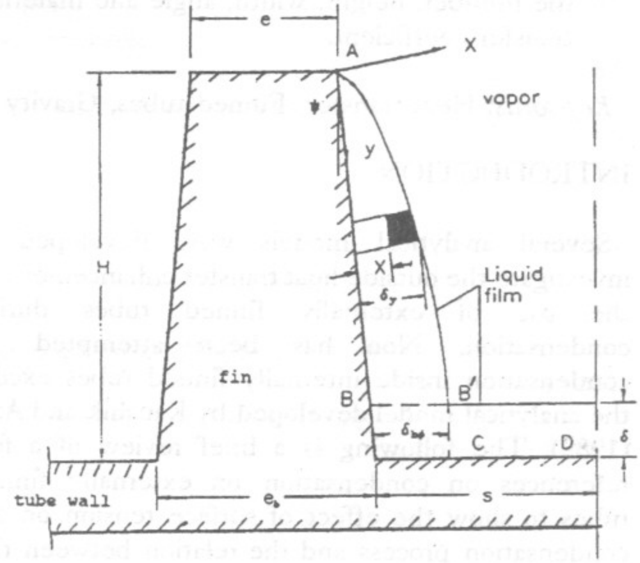

Figure 1. Schematic view of the flow model.

The analysis is carried out in two steps. First, a heat transfer expression is proposed to calculate the heat transfer from the fins surface. Second, the surface between the fins is treated as a smooth tube with an annular liquid film. This was the same approach followed by Kaushik and Azer(1989) in developing their model.

Heat Transfer from the Surface of Fins

Referring to Figure (1) and Figure (2), for the forces balance within a liquid element on a fin at angle ϕ inside the tube, the following equation can be written;

$$\mu_\ell \frac{du}{dX} + \frac{dp}{dy}(\delta_y - X) = (\rho_\ell - \rho_v) g'(\delta_y - X) \qquad (1)$$

where,

$$g' = g\ \cos\phi.\ \cos\psi, \qquad (2)$$

$$\phi = (n-1).\theta \qquad (3)$$

where, n varies from 1 to N, and

$$\theta = 2.\pi/N \qquad (4)$$

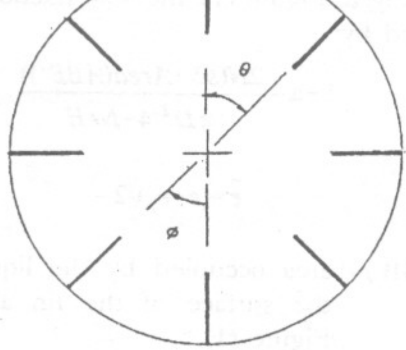

Figure 2. Schematic view for an internally finned tube with 8 fins.

By integration of equation 1, one obtains

$$\mu_\ell . u + \frac{dp}{dy}(\delta_y . X - \frac{X^2}{2}) = \rho_\ell g'(\delta_y . X - \frac{X^2}{2}) \qquad (5)$$

The above equation can be solved for the velocity u as follows;

$$u = -\frac{1}{\mu_\ell}(\frac{dp}{dy} - \rho_\ell g').(\delta_y X - \frac{X^2}{2}) \qquad (6)$$

where

$$\frac{dp}{dy} = \frac{d}{dy}(p_i - p_o) = \frac{\sigma}{dy}(\frac{1}{R_1} + \frac{1}{R_2}) \qquad (7)$$

The vapor pressure was assumed constant inside the condenser tube. Also, the variation in liquid film on fin along the tube can be neglected, thus

$$\frac{dp}{dy} = \frac{dp_i}{dy} = \sigma \frac{d}{dy} \left| \frac{1}{R_1} \right| \qquad (8)$$

Using the relation given by Webb et al. (1982) of the linearity change in $1/R_1$ along the line AB (Figure (1)), one can write;

$$\frac{dp_i}{dy} = \sigma \frac{d}{dy} \left| \frac{1}{R_1} \right| = \frac{\sigma \cos\psi}{(H-\delta)} \left| \frac{1}{R_B} - \frac{1}{R_T} \right| \qquad (9)$$

where

$$\frac{1}{R_B} = -1/s \qquad (10)$$

For R_T, the relation of Krakhu and Borovkov(1971) is considered;

$$\frac{1}{R_T} = \frac{2}{e(1+\tan\psi)} \qquad (11)$$

using equations 8, 9, 10, and 11, one can show;

$$u = \frac{1}{\mu_\ell}(\frac{2\sigma\cos\psi}{(H-\delta)}(\frac{1}{2s} + \frac{1}{e(1+\tan\psi)}) + \rho_\ell g')(\delta_y X - \frac{X^2}{2}) \qquad (12)$$

The mass flow rate of the condensate flowing on the fin, at any distance y from the tip of the fin is given by;

$$\dot{m}_y = \int_0^{\delta_y} \rho_\ell . u . dX \qquad (13)$$

By substituting equation 12 into equation 13, the following equation results;

$$\dot{m}_y = \frac{\rho_\ell}{3\mu_\ell}(\frac{2\sigma\cos\psi}{(H-\delta)}(\frac{1}{2s} + \frac{1}{e(1+\tan\psi)}) + \rho_\ell g')\delta_y^{\ 3} \qquad (14)$$

From the energy balance, it can be shown that;

$$h_{fg} \frac{d\dot{m}_y}{dy} = k_\ell \frac{\Delta T}{\delta_y} \qquad (15)$$

By differentiating equation 14 and substituting in equation 15, one can get;

$$\frac{k_\ell \Delta T}{\frac{\rho_\ell h_{fg}}{\mu_\ell}\left(\frac{2\sigma\cos\psi}{(H-\delta)}\left(\frac{1}{2s}+\frac{1}{e(1+\tan\psi)}\right)+\rho_{\ell v} g'\right)} dy = \delta_y^{\,3}.d\delta_y \quad (16)$$

Then, integrating equation 16 gives;

$$\delta_y = \left[\frac{4k_\ell \Delta T.y}{\frac{\rho_\ell h_{fg}}{\mu_\ell}\left(\frac{2\sigma\cos\psi}{(H-\delta)}\left(\frac{1}{2s}+\frac{1}{e(1+\tan\psi)}\right)+\rho_{\ell v} g'\right)}\right]^{0.25} \quad (17)$$

To account for the subcooling in the liquid film, the modification of Rohsenow (1956) is applied to h_{fg} as follows;

$$\delta_y = \left[\frac{4k_\ell \Delta T.y}{\frac{\rho_\ell \overline{h_{fg}}}{\mu_\ell}\left(\frac{2\sigma\cos\psi}{(H-\delta)}\left(\frac{1}{2s}+\frac{1}{e(1+\tan\psi)}\right)+\rho_{\ell v} g'\right)}\right]^{0.25} \quad (18)$$

where

$$\overline{h_{fg}} = h_{fg} + \frac{3}{8} C_{p\ell} \Delta T \quad (19)$$

Then, δ_y can be written as;

$$\delta_y = (4.C.y)^{0.25} \quad (20)$$

where

$$C = \frac{k_\ell .\Delta T}{\frac{\rho_\ell}{\mu_\ell}\left[h_{fg}+\frac{3}{8}C_{p\ell}.\Delta T\right]\left[\frac{2\sigma\cos\psi}{(H-\delta)}\left(\frac{1}{2s}+\frac{1}{e(1+\tan\psi)}\right)+\rho_{\ell v} g'\right]} \quad (21)$$

The local heat transfer coefficient on the fin, h_{fy} at any distance y from the tip of the fin is obtained by equating the heat conducted by the liquid film and the heat convected, thus

$$h_{fy} = k_\ell / \delta_y \quad (22)$$

By substituting equation 20 into equation 22 and integrating it over the active surface AB of the fin, the average heat transfer coefficient is obtained;

$$h_{f\phi} = \frac{1}{(\overline{H-\delta})}\int_0^{(\overline{H-\delta})} k_\ell (4.C.y)^{-0.25}.dy$$

then,

$$h_{f\phi} = \frac{4k_\ell}{3[4C(\overline{H-\delta})]^{0.25}} \quad (23)$$

where

$$(\overline{H-\delta}) = (H-\delta)/\cos\psi \quad (24)$$

To determine, the liquid film thickness between the fins, δ, the following steps are considered;

Referring to Figure (1), the void fraction, α, can be calculated by:

$$1-\alpha = \frac{2N(s\delta + Area(ABB'))}{\pi D^2/4 - N\bar{e}H} \quad (25)$$

where,

$$\bar{e} = (e+e_b)/2 \quad (26)$$

and

Area(ABB')= area occupied by the liquid film on the surface of the fin as shown in Figure (1).

Then, it can be shown that,

$$Area(ABB') = \int_0^{(\overline{H-\delta})} \delta_y.dy$$

or

$$Area(ABB') = \int_0^{(\overline{H-\delta})} (4Cy)^{0.25}.dy = 1.13137 C^{0.25} (\overline{H-\delta})^{1.25} \quad (27)$$

From Zivi (1964), the void fraction, α is given by;

$$\alpha = \frac{1}{1+((1-x)/x)(\rho_v/\rho_\ell)^{2/3}} \quad (28)$$

Also,

$$s = (\pi D - N e_b)/2N \quad (29)$$

Substituting equations 27, 28, and 29 into equation 25, the following equation results;

$$\frac{((1-x)/x)(\rho_v/\rho_\ell)^{2/3}}{1+((1-x)/x)(\rho_v/\rho_\ell)^{2/3}} = \frac{(\pi D - N e_b)\delta + 2.2627 N C^{0.25}(\overline{H-\delta})^{1.25}}{(\pi D^2/4 - N\bar{e}H)} \quad (30)$$

If the quality, x, the saturation temperature of the condensing fluid, and the pipe wall temperature and if the geometric parameters, N, H, D, e, and e_b are specified, equations 21 and 30 can be solved to determine δ and C.

The heat transfer per unit length of condenser tube for the fin at angle ϕ is given by;

$$q_{f\phi}=2\eta_f h_{f\phi}(\overline{H-\delta})\Delta T \tag{31}$$

To get the heat transfer per unit length of condenser tube due to all fins, the following relation can be applied;

$$q_f=\sum_{n=1}^{N} q_{f\phi}=\sum_{n=1}^{N} 2\eta_f h_{f\phi}(\overline{H-\delta}).\Delta T \tag{32}$$

where η_f is the fin efficiency. From Kraus(1964) η_f can be obtained as follows;

$$\eta_f=\frac{\beta_o}{2\gamma^2 H_c}\left[\frac{K_1(\beta_1).I_1(\beta_o)-I_1(\beta_1).K_1(\beta_o)}{I_o(\beta_o).K_1(\beta_1)+I_1(\beta_1).K_o(\beta_o)}\right] \tag{33}$$

where

$$\beta_o=2\gamma\sqrt{H_c+\frac{e(1-\tan\psi)}{2\tan\psi}}, \tag{34}$$

$$\beta_1=2\gamma\sqrt{\frac{e(1-\tan\psi)}{2\tan\psi}}, \tag{35}$$

$$\gamma=\sqrt{h_{f\phi}/(k_f\sin\psi)}, \tag{36}$$

and

$$H_c=(H-\delta)+e/2 \tag{37}$$

From equation 37, it can be seen that the corrected fin height, H_c, is used to account for the heat lost at the tip of the fin.

Heat Transfer from the Tube Wall Between the Fins

For the inter-fin spacing, Kaushik and Azer (1989) assumed that the liquid forms an annular film, as in smooth tubes. They used the correlation of Soliman et al. (1968) to predict the heat transfer coefficient, h_b. The same procedure was applied in the present model as follows;

$$h_b=0.036\frac{k_\ell\rho_\ell^{0.5}}{\mu_\ell}Pr_\ell^{0.65}F_o^{0.5} \tag{38}$$

where

$$F_o=F_f+F_m \tag{39}$$

$$F_f=0.45\left(\frac{8W_T^2}{\pi^2\rho_v D^4}\right)\left(\frac{4W_\ell}{\pi D\mu_v}\right)^{-0.2}$$
$$\left[x^{1.8}+5.7\left(\frac{\mu_\ell}{\mu_v}\right)^{0.0523}(1-x)^{0.47}\left(x^{1.33}\left(\frac{\rho_v}{\rho_\ell}\right)^{0.261}\right.\right.$$
$$\left.+8.11\left(\frac{\mu_\ell}{\mu_v}\right)^{.105}(1-x)^{0.94}x^{0.86}\left(\frac{\rho_v}{\rho_\ell}\right)^{0.522}\right] \tag{40}$$

$$F_m=0.5\left(D\frac{dx}{dz}\right)\left(\frac{8W_T^2}{\pi^2\rho_v D^4}\right)\left[2(1-x)\left(\frac{\rho_v}{\rho_\ell}\right)^{2/3}+\left(\frac{1}{x}-3+2x\right)\left(\frac{\rho_v}{\rho_\ell}\right)^{4/3}\right.$$
$$\left.+(0.751x-1)\left(\frac{\rho_v}{\rho_\ell}\right)^{1/3}+\left(2.5-1.25\left(x+\frac{1}{x}\right)\right)\left(\frac{\rho_v}{\rho_\ell}\right)^{5/3}+0.5(x-1)\left(\frac{\rho_v}{\rho_\ell}\right)\right] \tag{41}$$

The surface CD shown in Figure (1) was considered only for heat transfer from the inter-fin spacing per unit length of condenser tube. The heat transfer from the inter-fin spacing adjacent to the fin at angle ϕ, $q_{b\phi}$ can be given as follows;

$$q_{b\phi}=h_b(2S-2\delta_{b\phi}).\Delta T \tag{42}$$

where h_b can be calculated from equation 38 and the film thickness of the liquid, $\delta_{b\phi}$ at distance $(H^{-}\delta^{-})$ from the tip can be obtained from equation 20.

Then, the heat transfer q_b for all inter-fin spacings for an internally finned tube can be given by;

$$q_b=\sum_{n=1}^{N} q_{b\phi}=2\sum_{n=1}^{N} h_b(s-\delta_{b\phi}).\Delta T \tag{43}$$

The local condensation heat transfer coefficient from the longitudinally finned tube is;

$$h=\frac{q_f+q_b}{\pi D\Delta T} \tag{44}$$

Finally, to obtain the average heat transfer coefficient over a given length of the tube, the following formula can be used;

$$\frac{1}{h^{-}}=\frac{1}{x_{in}-x_{out}}\int_{x_{out}}^{x_{in}}\frac{dx}{h} \tag{45}$$

COMPARISON WITH EXPERIMENTAL DATA

Kaushik and Azer (1989) used the experimental data of Royal (1975) and Said (1982) for comparison with their theoretical results. Royal reported condensation heat transfer coefficients for steam condensed inside longitudinally finned tubes, while Said reported similar data for R-113. In both investigations the test condenser was constructed from four subsections. In their comparisons, they considered only the subsection nearest to the inlet of the condenser. For each subsection, the inlet and outlet quality of the condensing fluid, its saturation temperature, the wall temperature of the condenser tube, and the mass flow rate of the condensing fluid were given. Also, they started the numerical calculations at inlet quality, x_{in}, equal to 0.999 neglecting any inlet superheat encountered during experimentation. This is due to the fact that their model could only be applied in the condensation region.

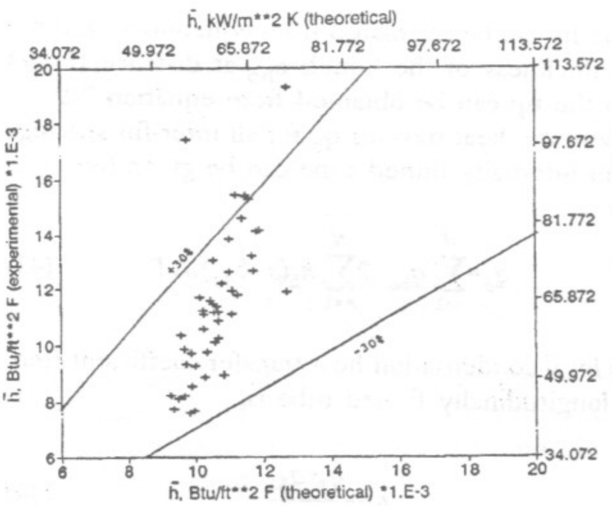

Figure 3. Comparison of experimental average inside heat transfer coefficients of Royal (1975), tube 7, with the predictions of eq. 45.

Figures (3) and (4) show comparisons between the predictions of the heat transfer coefficient by equation 45 and the experimental data of Royal (1975) for steam, and Said (1982) for R-113 respectively. The results show that 91.3% of the data points of Royal and 84% of the data points of Said were predicted within $\pm 30\%$. For Kaushik and Azer's model these values were 87% and 47% respectively, which indicates an improvement in the predictions of the new model. Also, it is important to note that, the majority of the points outside the $\pm 30\%$ region were for data of higher inlet quality where a partial length in the entrance region of the condenser tube could have acted as a desuperheater.

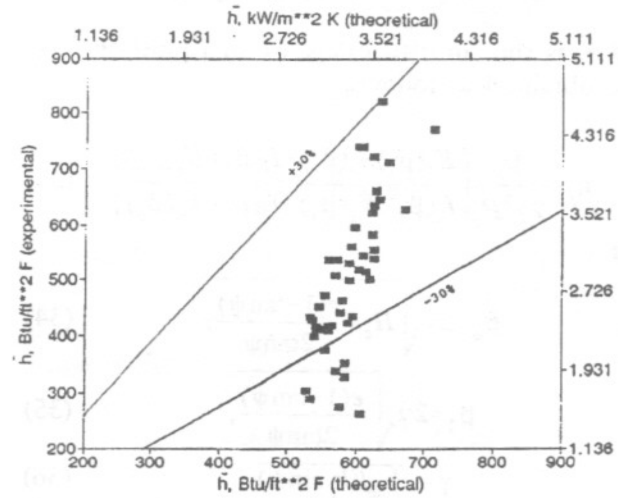

Figure 4. Comparison of experimental average inside heat transfer coefficients of Said (1982), tube 2, with the predictions of eq. 45.

EFFECT OF GRAVITY FORCE

Kaushik and Azer (1989) predicted the effect of the tube parameters for a longitudinally finned tube, on the heat transfer coefficient. The tube had the following characteristics;

Inside diameter, D = 0.0378 ft (0.01153 m)
Number of fins, N = 6
Fin height, H = 0.00535 ft (0.00163 m)
Fin width, e = 0.003 ft (0.0009114 m)
Saturation temperature, T_s = 266°F (130.0 °C)
Temperature difference, ΔT = 12.5 °F (6.944 °C)
Mass flow rate, W_T = 0.06481 Ib/s (0.031673 kg/s)

The condensing fluid was assumed to be steam and the tube material was assumed to be copper.

These were the characteristics of the tested tube by Royal (1975) except for the fin width. They have taken the average width, e^{-}, neglecting the fin angle, ψ, where e^{-} is given in equation 26.

It is to be recalled that in the present model two additional factors were taken into consideration, namely the gravity force and the fin angle, ψ. The following section compares the prediction of the present model and the Kaushik and Azer's model. For the sake of this comparison the same fin parameters used by them were used, with the exception of considering the effect of the gravity force on the prediction. In this case, the fin efficiency, η_f, is given by;

$$\eta_f = \tanh(m.(H-\delta))/(m.(H-\delta)) \tag{46}$$

where

$$m = (2h_{f\psi}/k_{\ell}.e)^{0.5} \tag{47}$$

The results were obtained by the two models for the variation of heat transfer coefficient with the number of fins are shown in Figure (5).

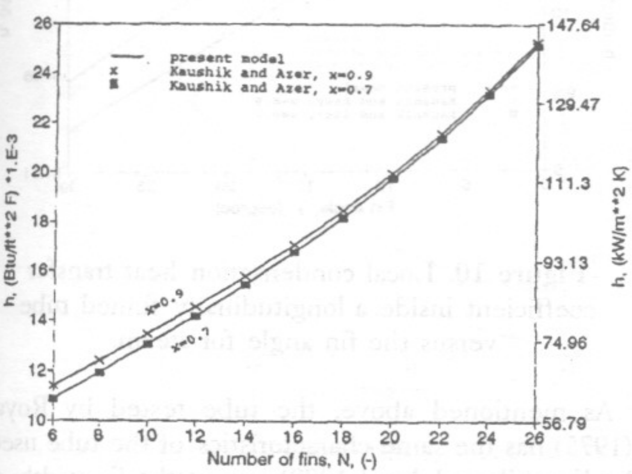

Figure 5. Local condensation heat transfer coefficient inside a longitudinally finned tube versus the number of fins for steam.

Figures (6), (7), and (8) show comparison, between the two models, of the effect of the height, width, and material of fins on the heat transfer coefficient respectively. As can be seen the predictions of the two models were nearly identical. Thus, one can concludes that the introduction of the gravity force has a negligible effect on the prediction of the heat transfer coefficient for condensation of steam inside finned tube i.e the surface tension force is the dominant force.

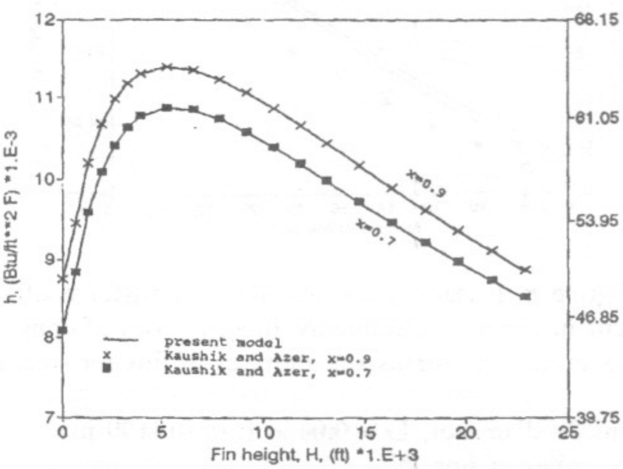

Figure 6. Local condensation heat transfer coefficient inside a longitudinally finned tube versus the fin height for steam.

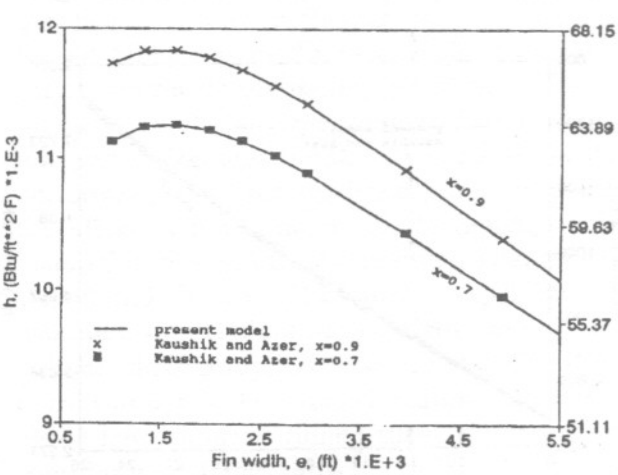

Figure 7. Local condensation heat transfer coefficient inside a longitudinally finned tube versus the fin width for steam.

To study the effect of the gravity force for low surface tension fluid, the analysis was applied to the following tube characteristics. This tube was tested by Said (1982).

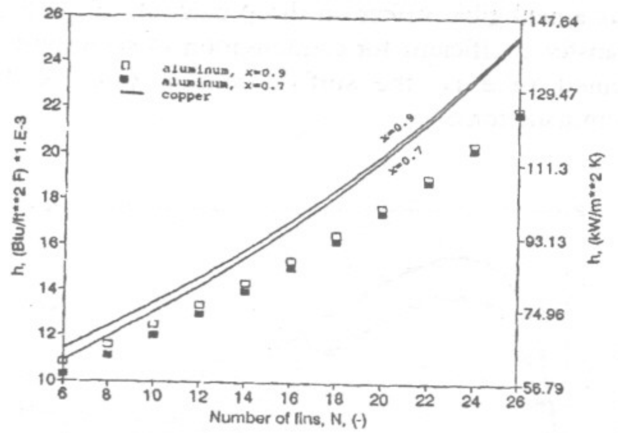

Figure 8. Local condensation heat transfer coefficient inside a longitudinally finned tubes of copper and aluminum versus the number of fins for steam.

Inside diameter, D = 0.0466 ft (0.014199 m)
Number of fins, N = 10
Fin width at tip, e = 0.002789 ft (0.00085 m)
Fin width at base, e_b = 0.004892 ft (0.001491 m)
Saturation temperature, T_s = 149 °F (65.0 °C)
Temperature difference, ΔT = 12.5 °F (6.944 °C)
Mass flow rate, W_T = 0.04189 Ib/s (0.019 kg/s)

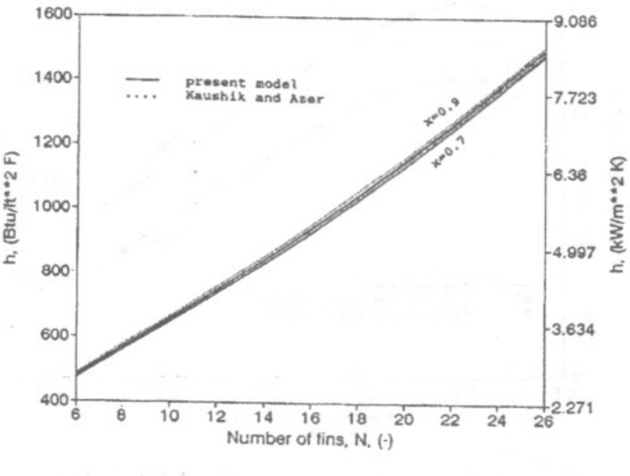

Figure 9. Local condensation heat transfer coefficient inside a longitudinally finned tubes versus the number of fins for R-113, (tube 2, Said (1982)).

The condensing fluid was assumed to be R-113 and the tube material was assumed to be copper. The fin width was taken as constant and equal to e to predict the gravity effect only i.e ψ=0. Figure (9) shows that the gravity effect caused a slight decrease in the heat transfer coefficient by 1 to 2% only for different values of number of fins, N. It appears that the small difference results from the fact that the surface tension and gravity forces act in the same direction in the lower half of the tube while, they act opposite each other in the upper half of the tube.

EFFECT OF FIN ANGLE

To study the effect of the fin angle, ψ, on the heat transfer coefficient, the same tube used by Kaushik and Azer(1989) is considered. Figure (10) shows that as the fin angle, ψ, increases the heat transfer coefficient, h decreases. The value of h obtained from Kaushik and Azer's model is shown as a constant value at ψ=0.

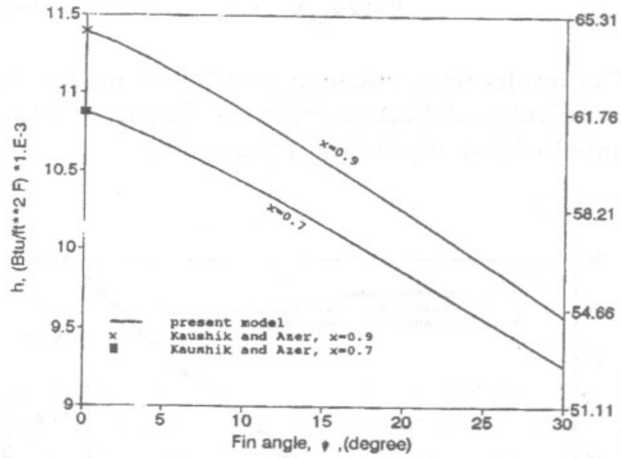

Figure 10. Local condensation heat transfer coefficient inside a longitudinally finned tube versus the fin angle for steam.

As mentioned above, the tube tested by Royal (1975) has the same characteristics of the tube used by Kaushik and Azer(1989) except the fin width, e, where it was equal to 0.0004343 m. For this fin width, Figure (11) shows that the heat transfer coefficient, h increases with the increase of ψ to a maximum value, then it decreases. This phenomenon occurs only for small fin width at the tip. The reason for the increase in the heat transfer coefficient, h, is due to the fact that the increase in h due to the increase of fin efficiency is higher than the decrease in h due to the decrease of surface tension effect.

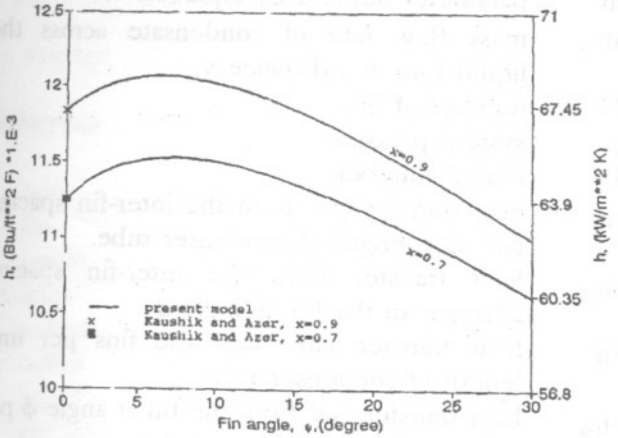

Figure 11. Local condensation heat transfer coefficient inside a longitudinally finned tube versus the fin angle for steam, (tube 7, Royal (1975)).

For the tube tested by Said (1982), the heat transfer coefficient decreased with the increase of the fin angle, ψ, as shown in Figure (12) for R-113.

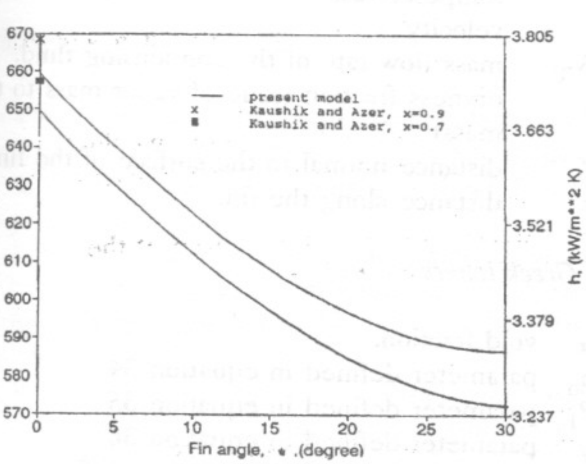

Figure 12. Local condensation heat transfer coefficient inside a longitudinally finned tube versus the fin angle for R-113, (tube 2, Said (1982)).

The effect of the saturation temperature, T_s, and temperature difference, ΔT, is shown in Figure (13). It shows that the heat transfer coefficient, h, decreases with the increase of ψ. The value of the decrease in h seems to be higher for the lower quality of R-113 (x=0.7) at the higher saturation temperature (T_s=80 °C) and lower temperature difference (ΔT=4 °C).

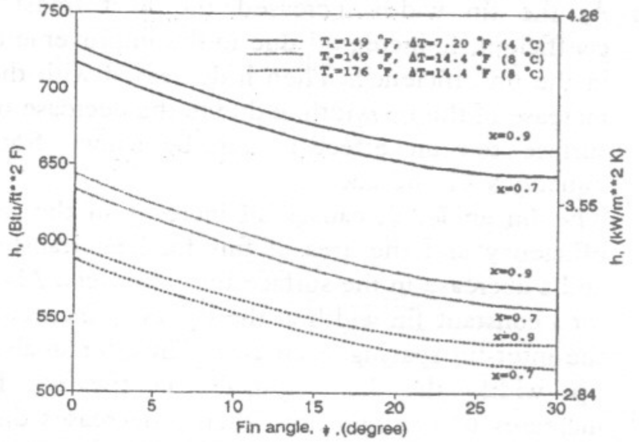

Figure 13. Variation of local condensation heat transfer coefficient with the angle at different saturation temperatures and temperature differences for R-113, (tube 2, Said (1982)).

SUMMARY AND CONCLUSIONS

The model of Kaushik and Azer (1989) to predict the heat transfer coefficient inside longitudinally finned tubes was modified to include the effect of gravity on the condensating liquid film. The model predicted well the experimental data for high and low surface tension fluids. A parametric study was performed, using the modified model to determine the effect of the number, width, height, angle and material of fins on the condensation heat transfer for steam and R-113. The model capabilities were extended to include a trapezoidally fin profile.

From this study, it was concluded that the enhancement in heat transfer during condensation inside longitudinally finned tubes resulted from the contribution of the surface tension effect and the increase of the area of the fins for heat transfer. In addition, the gravity force had a negligible effect on the heat transfer.

The results also showed that:

a- The heat transfer coefficient increased with the increase in the fin height, H, up to a certain height, then it decreased due to the decrease of surface tension effect despite the increase in the area of fins for heat transfer. This can be noted from equation 21 where the value of C decreases with the increase of H, the matter which cause a decrease in $h_{f\phi}$ as shown in equation 23.

b- As the fin width increased the heat transfer coefficient, h, increased due to the improvement in the fin efficiency. Then h decreased with the increase of the fin width, e due to the decrease of surface tension effect as can be shown from equations 21 and 23.

c- The fin angle, ψ, causes an increase in the fin efficiency and the area of fins for heat transfer and a decrease in the surface tension effect. Also, for a constant fin width at the tip, as ψ increases the inter-fin spacing decreases. Thus, for smaller tip width the heat transfer coefficient, h, increases to a certain limit then h decreases due to the decrease of surface tension effect as shown from equations 21 and 23, and due to the decrease in the heat transfer in the inter-fin spacing, as shown in equation 42.

NOMENCLATURE

Bo	Bond number($[2\sigma.(1/(2s)+1/e)]/[\rho_l.g.(H-\delta)]$)
C_p	specific heat.
D	inner diameter of the tube.
e	fin width at the tip.
e_b	fin width at the base.
F_f	parameter defined in equation 40
F_m	parameter defined in equation 41
F_o	parameter defined in equation 39
g	acceleration of gravity.
h	local condensation heat transfer coefficient.
$\bar{h}$	average heat transfer coefficient over a given length of the tube.
h_b	heat transfer coefficient in equation 38
h_{fg}	latent heat of vaporization.
h_{fy}	local heat transfer coefficient at a distance y from the tip of the fin.
$h_{f\phi}$	average heat transfer coefficient for the fin at angle ϕ.
H	fin height.
H_c	corrected fin height, equation 37.
$I_o(\beta)$, $I_1(\beta)$	modified Bessel function of the first order
$K_o(\beta)$, $K_1(\beta)$	modified Bessel function of the second order
k	thermal conductivity.
m	parameter defined by equation 47
$\dot{m}_y$	mass flow rate of condensate across the liquid film at a distance y,
N	number of fins
p	system pressure
Pr	Prandl number
q_b	heat transfer rate from the inter-fin spacing per unit length of condenser tube.
$q_{b\phi}$	heat transfer from the inter-fin spacing adjacent to the fin at angle ϕ.
q_f	heat transfer rate from the fins per unit length of condenser tube.
$q_{f\phi}$	heat transfer rate from the fin at angle ϕ per unit length of condenser tube.
R_B	radius of curvature at the base of the fins, equation 10
R_T	radius of curvature at the tip of the fin, equation 11
R_1,R_2	principal radii of curvature.
s	half of the inter-fin spacing.
T	temperature.
ΔT	difference between the saturation and wall temperatures.
u	velocity
W_T	mass flow rate of the condensing fluid.
x	dryness fraction (ratio of vapor mass to total mass)
X	distance normal to the surface of the fin.
y	distance along the fin.

Greek letters

α	void fraction.
β_o	parameter defined in equation 34
β_1	parameter defined in equation 35
γ	parameter defined in equation 36
δ	liquid film thickness between the fins.
$\delta_{b\phi}$	thickness of the condensate film at the base of the fin at angle ϕ.
δ_y	condensate film thickness on the fin at a distance y.
μ	dynamic viscosity.
θ	angle between two adjacent fins.
η_f	fin efficiency.
σ	surface tension.
ϕ	angle of fin with the direction of gravity, Fig.2.
ψ	fin angle, Fig.1.

Superscripts

-- average

Subscripts

B base
f fin
i inside
in inlet
o outside
out outlet
ℓ liquid
s saturation
T tip
v vapor

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