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THE EFFECT OF HEAT TRANSFER PARAMETERS UPON THE BRINE HEATERS DESIGN IN A DESALINATION PLANTS

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ABSTRACT

A new trend to a brine heater design with a sprayed saline water on horizontal tubes, where condensation takes place inside the tube bundle while the cooling evaporating film flows over the outside of the tubes. For the condensation inside and film evaporation outside the tube, the analysis of motion and energy equations with each boundary conditions and the laminar theory were undertaken. The resulting system of first order ordinary differential equations were solved numerically by computer program using Runge-Kutta-type method for a wide range of parameters. It was found from the numerical results that the heat transfer coefficients for the condensation and evaporation were inversly proportional to the film thickness around the tube. The average overall heat transfer coefficients increases by decreasing the saline water Reynolds number and the tube radius.

INTRODUCTION

Since desalination is recently popular topic, several workers are conducting research for improvements of the performance of the desalination components. A sprayed film heat exchangers has a high heat transfer coefficients, then used in heating and cooling of liquids as well as in condensation and evaporation process. The film condensation inside a horizontal tube studied by several investigators[1-5], classified the flow regimes of the two phase flow of condensation inside tubes as stratified flow for $Re_v < 35000$ and annular flow for $Re_v > 35000$. Rufer and Kezios [8]indicate that the condensation mechanism in the stratified regime is affected by the viscous, inertial and body forces while in the annular flow regime the condensation process is determined by the dynamics of the condensate and their interaction as shown by the previous investigators[9-11]. The film evaporation outside a horizontal tube investigated by Fletcher and others [6] reported an experimental data for evaporation from a film flowing over a horizontal tube electrically heated from within. The present study investigate the heat transfer parameters associated with the interacting condensation and evaporation phenomena while simultaneously solving for the wall temperature variation in the tangential and axial directions.

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a- The flow of the two films is laminar

6 b- The second derivatives of the velocities are negligible as compared with the corresponding normal derivatives

c- The tangential and normal pressure drop in the two thin films are negligible

d- Surface tension effects are negligible

e- The two phase flow of the condensate inside the tube moves in a stratified flow, the shear stress on the free vapor-liquid interfaces of the two films are negligible

From the continuity equation, the liquid mass flow rate m across the liquid film at the interfaces(y_t+y_v) and $-(y_t+y_c)$ per unit length of the tube is related to a change in y through the velocity distribution expression as;

$$m_c = \int_{-y_t}^{-(y_t+y_c)} \rho\, u_c \, dy \qquad , \qquad m_v = \int_{y_t}^{y_t+y_v} \rho\, u_v \, dy \tag{1}$$

The mass flux $\dot{m} = dm/dx$,then;

$$\dot{m}_c = \int_{-y_t}^{-(y_t+y_c)} \rho\, u_c \, dy \qquad , \qquad \dot{m} = \frac{d}{dx}\int_{y_t}^{y_t+y_v} \rho\, u_v \, dy \tag{2}$$

The momentum equation is given by making a force balance between the shear stress and the gravitational forces in the tangential x-direction as;

$$\frac{d^2u}{dy^2} = \frac{\rho g}{\mu} \sin 0 \tag{3}$$

Where $u = u_v$ or u_c

For the condensate film, the energy released as latent heat for the condensate $= \dot{m}_c L_c$, the heat capacity of the condensate film$=\dot{m}_c C(t_s-t)$ and the heat transferred by conduction from the condensate to the tube per unit area between x and $(x+dx) = k \left.\frac{dt}{dy}\right|_{y=-y_t}$, then the energy equation for the condensate film is;

$$-\dot{m}_c L_c - \dot{m}_c\, C\,(t_s - t) = -k \left.\frac{dt}{dy}\right|_{y=-y_t} \tag{4}$$

By substitution for $\dot{m}$ from Eq.(2) in Eq.(4), then,

$$\frac{d}{dx}\int_{-y_t}^{-(y_t+y_c)} u_c \left[C(t-t_s) - L_c \right] dy = -\,k \left.\frac{dt}{dy}\right|_{y=-y_t} \tag{5}$$

As similar for the evaporating film;

$$\frac{d}{dx}\int_{y_t}^{(y_t+y_v)} u_v \left[C(t - t_v) - L_v \right] dy = -k \left.\frac{dt}{dy}\right|_{y=y_t} \tag{6}$$

With reference to the tube radius r and a reference velocity u^*, define the following nondimensional variables, Where $\bar{U}$, M and δ relate to either the condensation or the evaporation side, also the dimensionless mass flux is given by;

$$\dot{M}_c = \frac{dM_c}{dX} = \delta_c^2 \; B \sin X \frac{d\delta_c}{dX} + \frac{1}{3} \delta_c^3 \; B \cos X$$

$$\dot{M}_v = -\frac{dM_v}{dX} = \delta_v^2 \; B \sin X \frac{d\delta_v}{dX} + \frac{1}{3} \delta_v^3 \; B \cos X \qquad (15)$$

THE TEMPERATURE PROFILES

As the thickness of the two liquid films is very small, then assume the temperature profiles in the two films are linear as for the tube wall. The dimensionless temperature distribution in the evaporating film T_v can be found as;

$$\frac{dT}{dY} = - \frac{(T_{wv} - T_v)}{\delta_v} \qquad (16)$$

And, $T = 0$ at $Y = \delta_t + \delta_v$ and $T = T_{wv}$ at $Y = \delta_t$ (17)

By integrating Eq.(16) with the boundary conditions Eq.(17), then;

$$T_v = - \frac{T_{wv}}{\delta_v} Y + T_{wv}(1 + \frac{\delta_t}{\delta_v}) \qquad (18)$$

As similar the dimensionless temperature profiles in the tube wall T_t and in the condensate film T_c are given by;

$$T_t = - \frac{(T_{wc} - T_{wv})}{2\delta_t} Y + \frac{T_{wc} + T_{wv}}{2} \qquad (19)$$

$$T_c = - \frac{(1 - T_{wc})}{\delta_c} + T_{wc} \; (1 + \frac{\delta_t}{\delta_c}) - \frac{\delta_t}{\delta_c} \qquad (20)$$

T_v and T_c are a functions of the dimensionless temperature at the two sides of the tube wall T_{wv} and T_{wc} which are getting by maching the heat fluxes at $Y = \pm \delta_t$ planes as follows;

$$- k_t \frac{dT_v}{dY}\Big|_{Y=\delta_t} = - k \frac{dT_t}{dY}\Big|_{Y=\delta_t}$$

$$- k_t \frac{dT_c}{dY}\Big|_{Y=-\delta_t} = - k \frac{dT_t}{dY}\Big|_{Y=-\delta_t} \qquad (21)$$

After differentiations of Eq. (21) and rearranging yields;

$$T_{wc} = \frac{2\delta_t + K\delta_v}{2\delta_t + K(\delta_c + \delta_v)} \qquad (22)$$

$$T_{wv} = \frac{K\delta_v}{2\delta_t + K(\delta_c + \delta_v)} \qquad (23)$$

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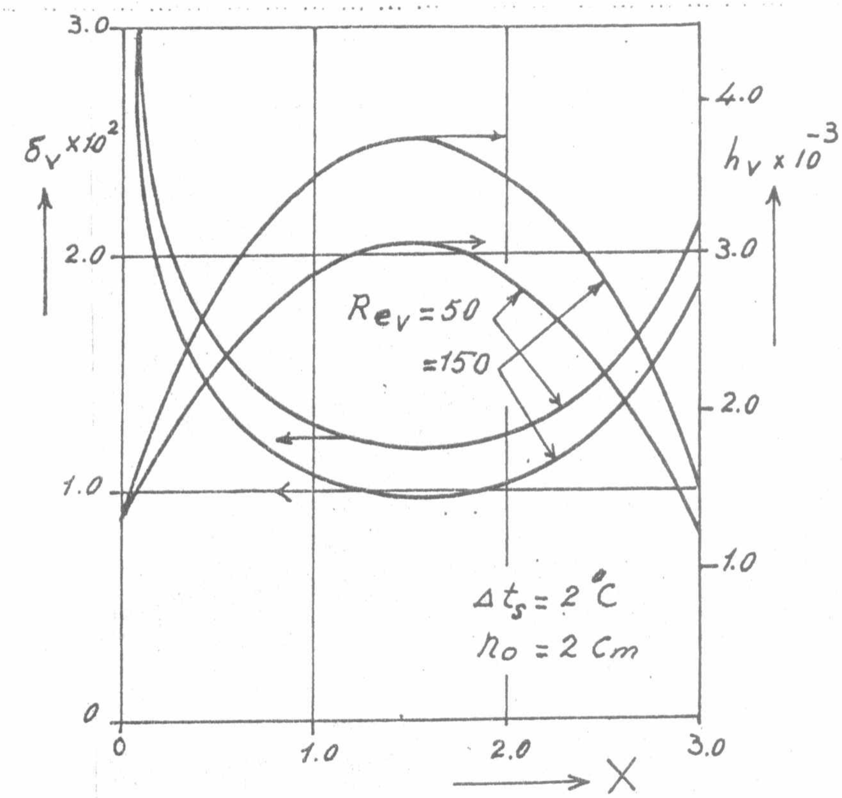

Fig.3. The effect saline water Reynolds number Re_v upon the dimensionless film thickness δ_v and local heat transfer coefficient h_v in the evaporating side around the tube.

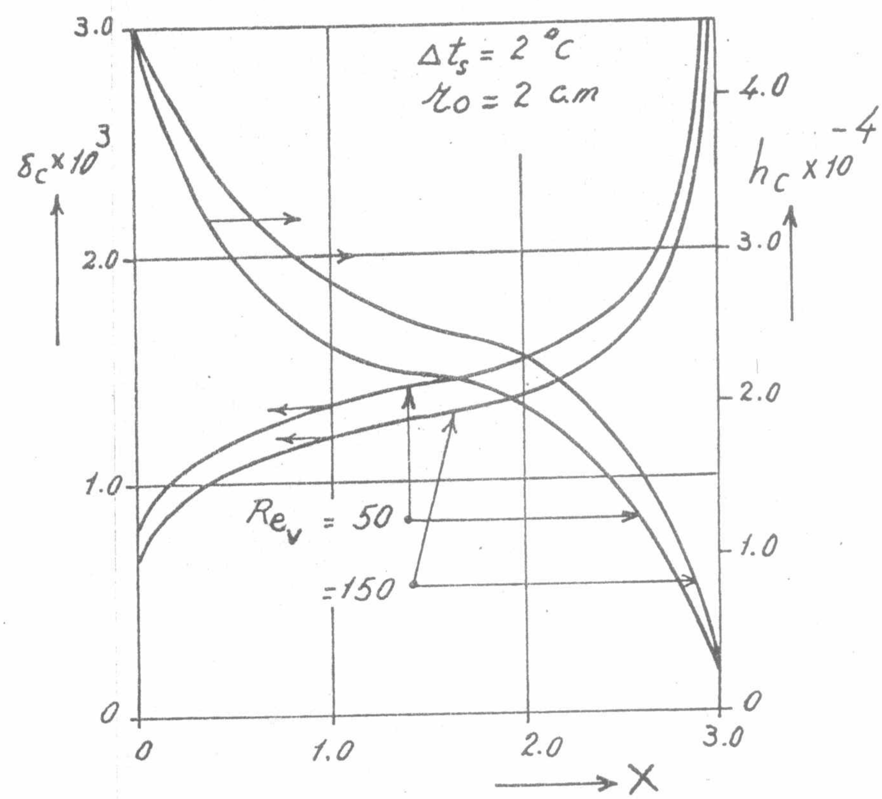

Fig.4. The effect of saline water Reynolds number Re_v upon the dimensionless film thickness δ_c and local heat transfer coefficient h_c in the condensate side around the tube

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tube thickness equals to 0.15 cm and the difference between the saturation etemperature of steam t_s and the saturation temperature of the evaporating film t_v equals to $(2 - 10)^oC$. A numerical results were obtained for the condensate film thickness, evaporating film thickness, condensation and evaporation heat transfer coefficients, local overall heat transfer coefficients around the tube and the average overall heat transfer coefficients at different values of saline water Reynolds number, outside tube radius and temperature driving force $(t_s - t_v)$. Fig. 3, shows the effect of Reynolds number Re_v upon the dimensionless evaporating film thickness δ_v and the local heat transfer coefficient $h_v (kcal/hr.m^2.^oC)$ around the tube. It indicates that for $X = \pi/2$, δ_v tends to a minimum value while h_v tends to a maximum value. The dimensionless condensate film thickness δ_c and the local heat transfer coefficient $h_c (kcal/hr.m^2.^oC)$ around the tube for various values of Reynolds number Re_v can be presented as in Fig.4. It is clear that near the top of the tube, δ_c starts to increase while h_c starts to decrease around the tube. Fig.5, shows the effect of Reynolds number Re_v and the temperature driving force $(t_s - t_v)$ with constant outside tube radius r_o upon the average overall heat transfer coefficient $\bar{H}_i (kcal/hr.m^2 {}^oC)$. It indicates that $\bar{H}_i$ decreases by increasing Re_v and $(t_s - t_v)$. The average overall heat transfer coefficient $\bar{H}_i$ $(kcal/hr.m^2.^oC)$ for various values of Reynolds number Re_v and the outside tube radius r_o can be presented as on Fig.6. It is clear that $\bar{H}_i$ decreases by increasing Re_v and r_o.

CONCLUSIONS

This study indicates that the heat transfer coefficients for the evaporation and condensation were inversly proportional to the film thickness around the tube. The average overall heat transfer coefficients increases by decreasing the saline water Reynolds number and the temperature driving force for a constant tube radius and also increases by decreasing the tube radius for a constant temperature driving force and saline water Reynolds number.

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T_{wc}-[$T_{wc} = (t_{wc}-t_v)/(t_s-t_v)$] - temperature of the tube wall surface at the condensation side, dimensionless
T_{wv}-[$T_{wv}=(t_{wv}-t_v)/(t_s-t_v)$] - temperature of the tube wall surface at the evaporation side, dimensionless
t_s-saturation temperature of steam at the condensate free surface,^oC
t_v-saturation temperature of vapor corresponding to p_v, ^oC
T_{wc},t_{wv}-temperature of the tube wall surfaces on the condensation and the evaporation sides,^oC
$\bar{U}_c$,$\bar{U}_v$-Y-averaged liquid velocities across condensate and the evaporating film, dimensionless
u_c,u_v- liquid velocities across the condensate and evaporating film,m/hr
X-(X=x/r)- tangential coordinate on the tube, dimensionless
Y-(Y=y/r)- normal coordinate, dimensionless
y_c,y_v-condensate and evaporating film thickness,m
y_t-half tube wall thickness,m
z-axial coordinate, final condensation tube length,m

APPENDIX

The constants E_i and C_i(i=1,.....16) in Eqs. (24), (25) are given as follows;

$E_1=K\delta_v$, $C_1=K\delta_c$

$E_2=1+\delta_t/\delta_v$, $C_2=1+\delta_t/\delta_c$

$E_3=\delta_t/\delta_v^2$, $C_3=\delta_t/\delta_c^2$

$E_4=2\delta_t+K\delta_v$, $C_4=2\delta_t+K(\delta_t+\delta_c)$

$E_5=E_1C_5/E_4+2K\delta_t/E_4C_4$ $C_5=KC_1/C_4^2$

$E_6=-KE_1/C_4^2$, $C_6=-KE_4/C_4^2$

$E_7=E_1E_2/C_4-E_v$, $C_7=C_1C_2/C_4+E_c$

$E_8=E_2E_5-E_1E_3/C_4$, $C_8=C_2C_5$

$E_9=E_2E_6$, $C_9=C_2C_6-C_1C_3/C_4$

$E_{10}=\delta_v^4(8E_8-5E_5)+\delta_v^3(24E_7-8E_5\delta_t)$, $C_{10}=\delta_c^4(8C_2+5)+8\delta_t\delta_t^3$

$E_{11}=(5/4)\delta_v^3+(4/3)\delta_t\delta_v^2$, $C_{11}=JC_4C_5C_{10}/12C_1$

$E_{12}=JC_4C_{10}/12E_1-E_{11}$, $C_{12}=\delta_c^4(8C_9+5C_6)+\delta_c^3(24C_7+8C_6\delta_t)$

$E_{13}=\delta_v^4(8E_2-5)-8\delta_v^3\delta_t$, $C_{13}=(5/4)\delta_c^3+(4/3)\delta_t\delta_c^3$

$E_{14}=\delta_v^4(8E_2-5)-8\delta_v^3\delta_t$, $C_{14}=JC_4C_{12}/12C_1-C_{13}$

$E_{15}=\delta_v^4(8C_4E_7/E_1-5)-8\delta_t\delta_v^3$, $C_{15}=\delta_c^4(8C_4C_7/C_1-5)-8\delta_t\delta_c^3$

$E_{16}=(E_{15}/12)\frac{dJ}{dX}$, $C_{16}=(C_{15}/12)\frac{dJ}{dX}$