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Theoretical Study of Boiling Heat Transfer in a Thin Film on Horizontal Tube.

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THEORETICAL STUDY OF BOILING HEAT TRANSFER IN A THIN FILM ON HORIZONTAL TUBE

دراسه نظرية لانتقال الحراره بالغليان في طبقة رقيقة على أنبوبة أفقية

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خلاصه: في هذا البحث تم دراسه ظاهره انتقال الحراره المصاحب للغليان على انبوبه افقيه واشتملت هذه الدراسه بحث دور كل من العوامل المؤثره في عمليه انتقال الحراره - مثل الفيض الحراري ومعدل السريان وقطر الانبوبة على معامل انتقال الحراره . تم حل النموذج الرياضي الواصف لهذا السريان عدديا بطريقه الفروق المحدده وذلك بتصميم وتنفيذ برنامج للحاسب الالى . باستخدام هذا النموذج امكن الحصول على توزيع درجات الحراره خلال الطبقه الملاصقه للجدار الخارجي للانبوبه و من ثم تم حساب معامل انتقال الحراره عند الظروف المؤثره المختلفه . لاختبار صلاحيه هذا النموذج المقترح تم مقارنه النتائج المستخلصه منه مع مثيلاتها من الابحاث السابقه . هذا البحث درس انتقال الحراره لانابيب ذات قطر ١٢ , ١٩ , ٣٨ مم وكان مدى رقم رينولدز من ١٠٠ الى ٥٠٠ ، و مدى فرق درجات الحراره يصل الى ٣٥ درجة مئوية .

BSTRACT

Boiling heat transfer process in a thin film on horizontal tube s ,theoretically, investigated . This subject is important for esign of the horizontal tube evaporator- condenser (HTE) , which s applied in distillation processes. The effect of the operating arameters (heat flux, mass flow rate and tube diameter) is nvestigated . To perform this study , a theoretical model is roposed, and a computer program is developed to solve this model umerically. This program is used to determine local and average oiling heat transfer coefficient for different operating arameters in laminar flow regime. The range of Reynolds number s taken as 100-500 and wall superheat up to 35 $^{\circ}$C. The diameter f tested tube was 12, 19 and 38 mm.

- INTRODUCTION

Heat transfer through falling-film or spray-film evaporation has been widely employed in heat exchange devices in the chemical, refrigeration, petroleum refining, desalination and food industries . Horizontal Tube Evaporator (HTE) is an important thermal desalination device, where boiling takes place in a thin film on horizontal tubes. Many investigators show that. the world dependence on desalination increases greatly in the last twenty years. Sea water desalination seams to be the best solution for the water shortage problem.

Many investigators studied the boiling heat transfer from theoretical and experimental point of view [1-9]. Experimental and theoretical work of H. M. Mostafa [1] for boiling heat transfer in a thin film on horizontal tube heated by a waste steam. W. H. Parken et al [3] studied the same problem using electrically heated tubes . P. K. Tewari [5] studied the nucleate boiling in a thin film on horizontal tube at atmospheric and sub-atmospheric pressures by using distilled and Sodium Chloride solutions.

Heat transfer for saturated falling-film evaporation on a horizontal tube has been analytically and experimentally studied by M. C. Chyu and A. E. Bergles [6] . The effect of film flow rate, liquid feed height and wall superheat are investigated. Two models have been proposed, both models based upon three defined heat transfer regions, the jet impingement region, the thermal developing region and the fully developed region. Both two models assumed heat is conducted across the film and evaporation takes place at the free surface. The influence on heat transfer coefficient is even smaller at low Renynolds number and independent of Reynolds number at high heat flux (208 KW/m^2). Both models and experimental data demonstrate, that heat transfer coefficient is independent on wall superheat.

Theoretical analysis was performed by D. Moalem and S. Sideman [8] to study the overall heat transfer coefficient in a horizontal evaporator- condenser tube for low heat flux in laminar flow regime. Local evaporation heat transfer coefficient around the tube has a maximum value at angle equal to $\Pi/2$ from the top because the film thickness was a minimum at this angle. In laminar flow regime the average overall heat transfer coefficient decrease with increasing Reynolds number or increasing tube radius.

2- GOVERNING EQUATIONS

Fig.(1-a) shows the system of coordinate used to analyze, mathematically, the present problem. According to the present proposed model some assumptions are made. Hydrodynamic as well as, thermal flow field are assumed to be identical along the tube length. The radial velocity is assumed to be very small compare to the tangential component. According to these assumption the energy equation in cylinderical coordinate is simplified to the form:

$$\frac{V}{r}\frac{\partial T}{\partial \phi} = \frac{K}{\rho C_p}\left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r}\right), \qquad (1)$$

with the boundary condition:

$$\text{at } r=R_o \quad T=T_w \; ; \qquad \text{at } r=R_o+\delta \quad T=T_v \; . \qquad (2)$$

In equation (1), V is the average tangential component of the velocity which is determined according to the relation:

$$V = \Gamma / \rho\,\delta_o \qquad (3)$$

Where Γ is the rate of falling water per unit length of the tube per one side ($\Gamma = m^{\cdot}/2L$) and δ_o is the film thickness at the position ϕ=0.0 and is approximated by

$$\delta_o = A_o / (2\ L) \qquad (4)$$

The film thickness δ at general position ϕ is estimated with the aid of the equation:

$$\delta = \delta_o \, EXP(-q'' \, R_o \, \phi \, / \, \Gamma \, h_{fg}) \qquad (5)$$

Equation (5) is derived according to the mass balance of the evaporation process from the free surface of the film .Figure (1-b).

To put flow describing equations (1-5) in dimensionless form, one defines the following dimensionless independent and dependent variables as follows:

$$R = r/R_o \quad : \quad \Phi = \phi \, / \, \phi_o \quad :$$
$$\hat{\delta} = \delta \, / \, R_o \quad : \quad \theta = (T - T_v)/(T_w - T_v) \quad . \qquad (6)$$

With the aid of the foregoing definitions of variables (equations (6)), the dimensionless form of energy equation , can be written as:

$$\frac{1}{R} \frac{\partial\theta}{\partial\Phi} = (4\Pi/Re*Pr)*(\hat{\delta}_o) \; \left[\frac{\partial^2\theta}{\partial R^2} + \frac{1}{R} \frac{\partial \theta}{\partial R} \right] \quad . \qquad (7)$$

With the boundary condition:

$$\text{at} \quad R = 1 \quad : \quad \theta = 1 \quad :$$
$$\text{at} \quad R = 1 + \hat{\delta} \quad : \quad \theta = 0.0 \quad . \qquad (8)$$

Where Re & Pr are Reynolds and Prandtl numbers , which have the following definitions:

$$Re = 4\Gamma/\mu \quad : \quad Pr = \mu C_p/K \quad . \qquad (9)$$

Solving equations (7-9), the temperature distribution through out the flow field can be evaluated and, in turn, one calculates the local heat transfer coefficient by using the local heat flux and wall superheat as follows:

$$h_\phi = q''_{w,\phi} \; / \; (\Delta T)_{sup} \quad ; \qquad (10)$$

The local heat flux at the wall is calculated according to the equation:

$$q''_{w,\phi} = - K * (\Delta T)_{sup} * (\partial\theta/\partial R)_o \; / \; R_o \qquad (11)$$

Where $(\partial\theta/\partial R)_o$ is the gradient of the dimensionless temperature at the tube wall.

The average boiling heat transfer coefficient is calculated through the following relation:

$$h = \frac{1}{\Pi} \int_o^{\pi} h_\phi \, d\phi \qquad (12)$$

3- NUMERICAL PROCEDURE

The dimensionless energy equation and its boundary conditions equations (7-9) are solved .numerically, using finite divided difference method. As shown in Fig. (1-c) R-Φ flow field is covered with a mesh, their nodes are identified by the identifier

(...). Accordingly, at any position i,j , the value of R, Φ can be determined as:

$$R_i = 1-(i-1)S \quad ; \quad \Phi = (j-1)\Delta\Phi \quad ;$$

where S and ΔΦ are the step size in radial and tangential directions respectively . Taking n as the total number of nodes in radial direction , the step size in this direction is defined as:

$$S = \delta / (n-1)$$

The derivatives of the variable θ with respect to R,Φ can approximated by the following finite divided differences as:

$$\begin{aligned} \partial\theta/\partial R &= (\theta_{i+1,j} - \theta_{i-1,j}) / 2S \\ \partial^2\theta/\partial R^2 &= (\theta_{i+1,j} - 2\theta_{i,j} + \theta_{i-1,j}) / S^2 \\ \partial\theta/\partial\Phi &= (\theta_{i,j} - \theta_{i,j-1}) / \Delta\Phi \end{aligned} \qquad (13)$$

Substitution of the approximate derivatives in the dimensionless energy equation leads to the following set of linear algebraic equations:

$$C_i\,\theta_{i-1,j} + D_i\,\theta_{i,j} + F_i\,\theta_{i+1,j} = A_i\,\theta_{i,j-1} \qquad (14)$$

Where the coefficients of the foregoing equations (14) are defined as:

$$i = 1, 2, \ldots, n$$

$$\begin{aligned} C_i &= -1 + S/[2(1+(i-1)S)] \\ D_i &= 2 + S^2/((1+(i-1)S) \cdot \Delta\Phi \cdot (4\Pi/Re \cdot Pr) \cdot \delta_j) \\ F_i &= -1 - S/[2(1+(i-1)S)] \\ A_i &= S^2/((1+(i-1)S) \cdot \Delta\Phi \cdot (4\Pi/Re \cdot Pr) \cdot \delta_j) \end{aligned} \qquad (15)$$

Equations (14) are of the tri-diagonal matrix type . The tri-diagonal matrix equations are solved by the well known back substitution technique. The solution of equations (14) is repeated in iterative manner till a proper value of δ is achieved. A computer program is designed and proposed to predict the temperature profile across film thickness and local heat flux. And thus local and average boiling heat transfer coefficient and Nusselt number can be calculated.
The suitable total number of nodes in both R and Φ directions are found to be 100 .

4- RESULTS AND DISCUSSION

The dimensionless thickness for the evaporating film around the tube circumference is shown in Fig. (2). The film thickness decreases with the angular position specially for smaller values of Reynolds number . The developing of the temperature profile becomes more and more linear with angular position and it is very close to linear distribution for Φ=Π/2 as shown in Fig.(3). This profile is in good agreement with the results obtained by M. C. Chyu and A. E. Bergles [6]. The local heat flux decreases with the angular position as shown in Fig. (4). It is clear that the local heat transfer coefficient and local Nusselt number have the highest value at the top of the tube and then decrease rapidly till they has an asymptotic values (starting from $\theta = \Pi/6$).

The effect of mass flow rate (or Reynolds number) on local Nusselt number is shown in Fig. (5). It is found that , Reynolds number has a little effect on local Nusselt number specially at higher angular position.

The effect of tube diameter on the heat transfer coefficient is studied as shown in Fig. (6). From this figure it is clear that the heat transfer coefficient increases with decreasing tube diameter in laminar flow.

In laminar flow (Re<750) the average Nusselt number increases with increasing Reynolds number , as shown in Fig. (7). The amount of liberated vapor from the water film is significant for low Reynolds number (smaller film thickness) and thus a relative rapid decrease in the film thickness with the angle ϕ is expected . This causes an increase in the local boiling heat transfer coefficient and local Nusselt number , specially at higher degree of superheat .

Fig. (8) shows a comparison between the present results and M. C. Chyu et al (6). A good agreement between the two models proposed by M. C. Chyu et al and present work is found .

A comparison between the average boiling heat transfer coefficient obtained from theoretical results and experimental results obtained by Mostafa, H. M.[1] is shown in Fig. (9). The difference between the experimental and theoretical results is. probably, due to the nonuniform spread of water along the tube circumference. Moreover in experimental work , a fraction of the total area is covered by a relatively thin film where as in other parts the flow is turbulent and or wavy. Also, the non-uniform rain-like drops falling on the tube may enhance the transfer rate by initiating concentric waves. These effects are not accounted in theoretical work. For these reasons the experimental heat transfer coefficient is greater than the theoretical one by about 17% at the same working conditions.

5- CONCLUSION

In this study a model describing the boiling heat transfer process over a horizontal tube at constant wall temperature is proposed. To check the validity of this model, a comparison between the obtained results with that of the previous works proved the validity of this model. It is found, that the boiling heat transfer coefficient increases with decreasing tube diameter. Also, increasing wall superheat, heat flux and Reynolds number cause an increase in boiling heat transfer coefficient.

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NOMENCLATURE

A_o	Distributer orifice area	m^2
D	Diameter	m
h	Average boiling heat transfer coefficient	$W/m^2.{}^oC$
h_ϕ	Local boiling heat transfer coefficient	$W/m^2.{}^oC$
h_{fg}	Latent heat of evaporation	$J/kg.{}^oC$
I	Increment in nodes in r-direction	—
J	Increment in nodes in ϕ-direction	—
K	Thermal conductivity	$W/m.{}^oC$
L	Tube length	m
m	Number of nodes in ϕ-direction	—
$\dot{m}_{cir}$	Circulated mass flow rate	kg/s
n	Number of nodes in r-direction	—
q"	Heat flux.	W/m^2
r	Radial coordinate	m
R_o	Outer tube radius	m
R	Dimensionless tube radius	—
T	Temperature	^oC
T^*	Saturation temperature of waste steam	^oC
ΔT_{sup}	Superheat temperature difference (T_w-T_v)	^oC
U_o	Velocity of free falling film	m/s
u	Radial velocity	m/s
V	Tangential velocity	m/s
Z	Axial coordinate of the test tube	m

Greek symbols:

Γ	Mass flow rate per unit length per one side of the tube	kg/m.s
δ	Film thickness	m
θ	Dimensionless temperature difference$=(T-T_v)/(T_w-T_v)$	—
μ	Dynamic viscosity	$N.s/m^2$
ν	Kinematic viscosity	m^2/s
ρ	Density	kg/m^3
ϕ	Angle of inclination	Rad.
Φ	Angular position	—

Subscripts:

c	Condensate
L	Liquid
o	Orifice, outer, initial
s	Steam
sup	Superheat

v Vapor
w Wall

Dimensionless numbers:

Nu Nusselt Number (hD/K)
Pr Prandtl Number ($C_p.\mu/K$)
Re Reynolds Number ($4\Gamma/\mu$)

REFERENCES

1-H. M. Mostafa (Boiling heat transfer in a thin film on horizontal tubes) Ph.D. Thesis. El-Mansoura University . 1994.

2-S.Zhang and G.Gogos (Film evaporation of a spherical droplet over a hot surface ;fluid mechanics and heat /mass transfer) J.Fluid Mechanics. Vol.222, pp. 543-563., 1991.

3-W.H.Parken. L.S.Fletcher, V.Sernas and J.C.Han (Heat transfer through falling film evaporation and boiling on horizontal tubes) ASME Journal Of Heat Transfer Vol.112. pp. 744-750. Aug. 1990.

4-S.G.Bankoff (Dynamics and stability of thin heated liquid film) ASME Journal Of Heat Transfer Vol.112, pp. 538-546. Aug. 1990.

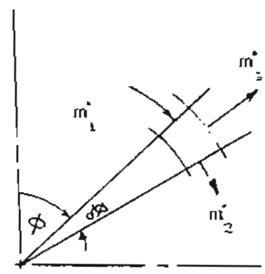
5-P.K.Tewari. P.K.Verma and and M.P.S. Ramini (Nucleate boiling in a thin film on a horizontal tube at atmospheric and sub-atmospheric pressures) Int. J. Heat Mass Transfer. Vol. 32 .No. 4 .pp.723-728 .1989.

6-M.C.Chyu and A.E.Bergles (An analytical and experimental study of falling film evaporation on a horizontal tube) ASME Journal Of Heat Transfer Vol. 109. pp. 983-990 .Nov. 1987.

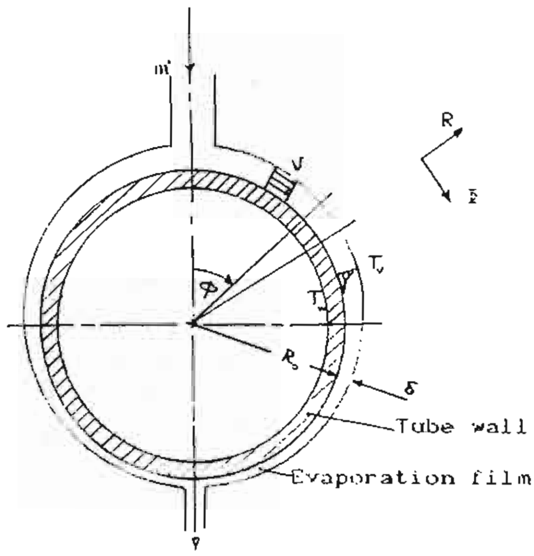
7-K. Edahiro. T. Hamada, M. Arai and Y. Hirao (Research and development of multi-effect horizontal tube film evaporator) Desalination. Vol.22. pp.121 .1977.

8-D.Moalem and S.Sideman (Theoretical analysis of a horizontal condenser-evaporator tube) Int. J. Heat Mass Transfer. Vol. 19 .pp. 259-270 ,1976.

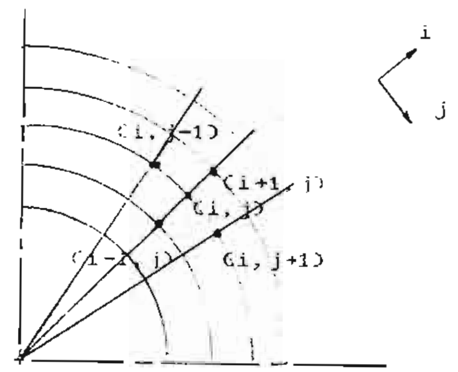
9-S.Sideman. D.Moalem and R.Semiat (Theoretical analysis of horizontal condenser-evaporator conducts of various cross sections) Desalination Vol.17. pp. 167-192, 1975.



(1.a) Mass balance of differential element



(1.b) Test tube co-ordinates system



(1.C) The used mesh in calculating procedure

Fig.(1) Schematic description of the flow field

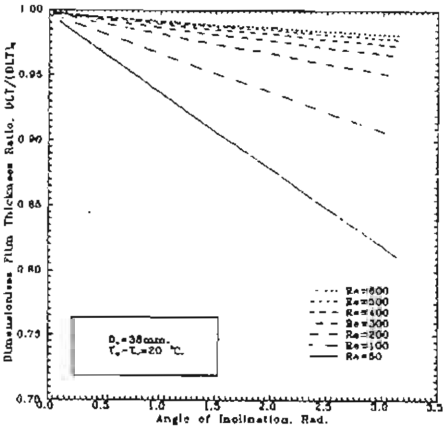


Fig.(2)Variation of the theoretical dimensionless water film thickness around the test tube for laminar flow.

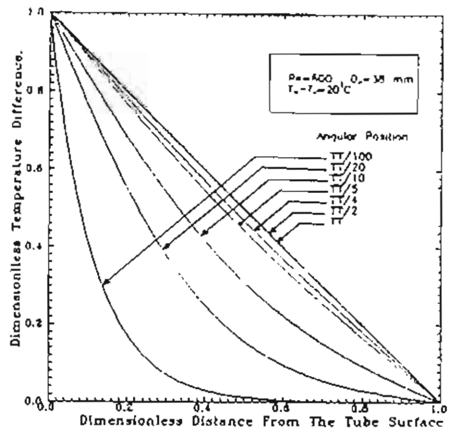


Fig.(3) Theoretical temperature profile across water film at different angular position.

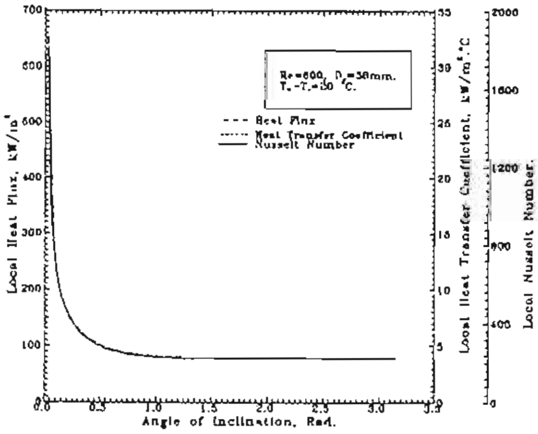


Fig.(4)Distribution of local heat flux ,Nusselt number and heat transfer coefficient versus angular position.

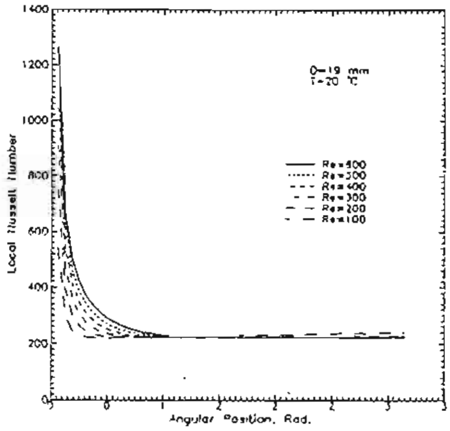


Fig.(5) Distribution of local Nusselt number versus angular position for laminar flow

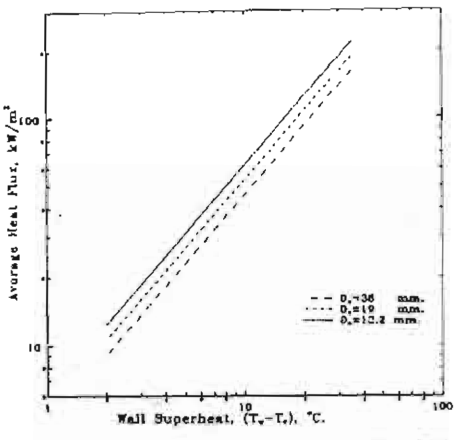


Fig.(6) Boiling curve for laminar flow (Re=500) at various tube diameters

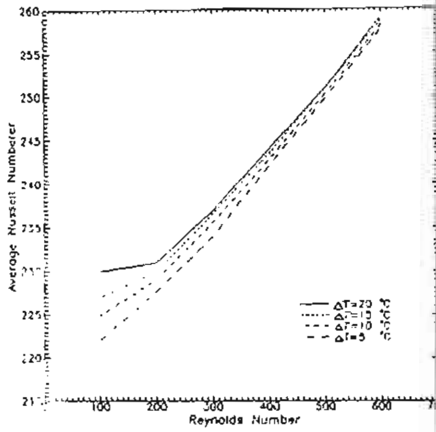


Fig.(7) Effect of wall superheat on the average Nusselt number in laminar flow (D_o=19 mm)

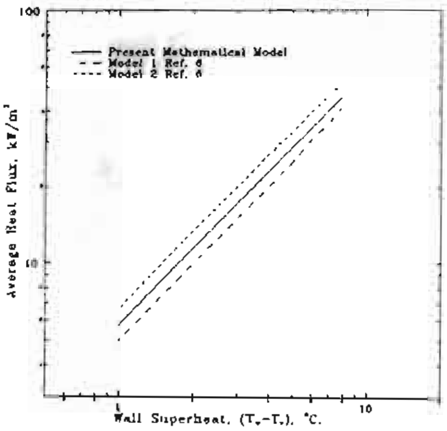


Fig.(8) Copmarison between prsent results with previous data for laminar flow.

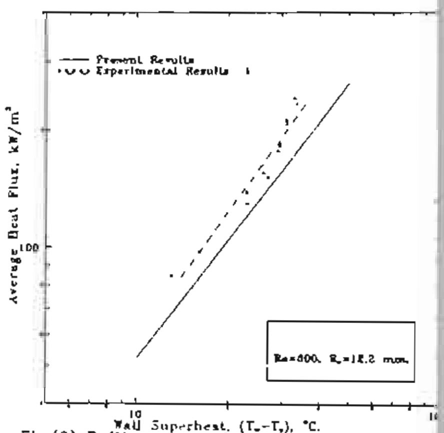


Fig.(9) Boiling curve for present theoretical results and another experimental results in laminar flow