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THEORETICAL STUDY OF HEAT TRANSFER FROM SATURATED VAPOUR CONDENSING OUTSIDE INCLINED TUBES

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Abstract

A theoretical study for condensation of saturated vapour outside inclined tubes is presented. A mathematical model is developed starting from the governing equations of the condensate film around the inclined tube which are mass, momentum and energy equations for two dimensional flow. The resulting equations are solved together to give a partial differential equation relating the condensate film thickness at any location on the surface of the tube with the tube inclination-angle. A solution for the liquid-film thickness and the average heat-transfer coefficient is determined analytically using the method of characteristics with the aid of a computer program.

The presented mathematical model is examined by deducing the previously-known relations for condensation of saturated vapour on vertical tubes, horizontal tubes and inclined flat plates as special cases of the general relation for inclined tubes.

The effects of temperature difference between condensing saturated vapour and tube surface as well as the tube inclination on the average heat-transfer coefficient are plotted and discussed. It is concluded that the average heat-transfer coefficient increases with decreasing the temperature difference and with increasing the tube inclination-angle from the vertical.

1. INTRODUCTION

Condensation occurs when the temperature of a vapour is reduced below its saturation temperature corresponding to its pressure. In engineering practice, the process occurs when the vapour comes in contact with a cooling surface. The condensate vapour forms a liquid film on the cooling surface which resists heat transfer from the vapour to the cooling surface. The condensate film moves downwards on the cooling surface under the action of gravity, and therefore, the condensate film-thickness as well as the average heat-transfer coefficient depend on the orientation of the cooling surface.

Nusselt [1] derived a theoretical relation for laminar-film condensation of saturated vapour on a vertical surface as follows:

$$h_{av} = 0.943 \left(\frac{\rho^2 \, g \, L \, k^3}{\mu \, \Delta t \, H} \right)^{0.25} \tag{1}$$

This equation may be applied for condensation outside vertical tubes and inside vertical tubes of large diameters. Holman [2] considered the buoyancy force in the original analysis of Nusselt, and the following relation is obtained:

$$h_{av} = 0.943 \left\{ \frac{\rho_l (\rho_l - \rho_v) g L k^3}{\mu \Delta t H} \right\}^{0.25} \qquad (2)$$

For condensation of vapour over inclined flat plates, Nusselt [1] showed that:

$$h_{av} = h_{ver} (\cos\beta)^{0.25} \qquad (3)$$

Moreover, Holman [2] stated that Eqn. (3) may be applied, also, for condensation outside inclined tubes. However, McAdams [3] reported that Eqn. (3) could be applied only for inclined flat plates, but this is not true for inclined tubes.

A theoretical study for condensation of pure saturated vapours using inclined tubes was presented by Hassan and Jacob [4]. The theoretical analysis was formulated without differentiating between the two cases of condensation on outside and inside surfaces of the tube. A partial differential equation was derived and solved, numerically, using four approximated assumptions to obtain the average heat-transfer coefficient.

In the present study, a mathematical model for pure saturated vapour condensing outside inclined tube is developed starting from the governing equations of mass, momentum and energy in two dimensional flow. A differential equation similar to that of ref.4 is obtained. However, an analytical solution of this differential equation is developed.

2. THEORETICAL ANALYSIS

The following assumptions are considered to simplify the theoretical study. They are the same assumptions considered by Nusselt [1] for condensation on a vertical surface, but the effect of the buoyancy force is taken into consideration.

a) The flow of condensate film is laminar.
b) The vapour is considered dry and saturated.
c) The vapour is pure and does not contain any non-condensable gases.
d) The vapour velocity is very small, thus the shear stress at vapour-liquid interface can be neglected.
e) The tube surface temperature is uniform.
f) The inertia force of condensate film is negligible.
g) There is no heat transfer by convection between vapour and liquid.
h) Heat is transferred across the condensate film by conduction, where the temperature distribution is linear.
i) Variation in physical properties of condensate film with temperature is neglected.
j) The only force acting on condensate film is the gravity force.
k) The flow of condensate film is continuous and steady.
l) The condensate film thickness is negligible with respect to the tube radius.

2.1. Condensate Motion

A droplet of the condensed vapour tends to move vertically downwards under the action of gravity, but this motion is constrained to be in contact with the tube surface. The expected distribution of condensate-film thickness around the surface of inclined tube is illustrated in Fig. (1). The flow of condensate film is two dimensional in axial and tangential directions and symmetrical about the plane $\phi=0$. The condensate-film thickness is maximum at $\phi=\pi$ and minimum at $\phi=0$, and it increases as the axial distance measured from the top edge of the tube increases.

2.2 Mass Conservation

Figure (2) shows an element of the condensate film on the tube. The element starts at radial distance r measured from the tube surface and ends at the liquid-vapour interface. The mass flow rate entering the element in the axial direction is:

$$\dot{m}_{i,x} = \rho_l \int_0^\delta u_x \left(\frac{D}{2} + r\right) d\phi \, dr \qquad (4)$$

The mass flow rate leaving the element in the axial direction is:

$$\dot{m}_{o,x} = \dot{m}_{i,x} + \frac{\partial}{\partial x} (\dot{m}_{i,x}) \, dx$$

Hence,

$$\dot{m}_{o,x} = \rho_l \int_0^\delta u_x \left(\frac{D}{2} + r\right) d\phi \, dr + \frac{\partial}{\partial x} \rho_l \int_0^\delta u_x \left(\frac{D}{2} + r\right) d\phi \, dr \, dx \qquad (5)$$

The mass flow rate entering the element in the tangential direction is:

$$\dot{m}_{i,\phi} = \rho_l \int_0^\delta u_\phi \, dr \, dx \qquad (6)$$

The mass flow rate leaving the element in the tangential direction is:

$$\dot{m}_{o,\phi} = \dot{m}_{i,\phi} + \frac{\partial}{\partial \phi} (\dot{m}_{i,\phi}) \, d\phi$$

$$= \rho_l \int_0^\delta u_\phi \, dr \, dx + \frac{\partial}{\partial \phi} \rho_l \int_0^\delta u_\phi \, dr \, dx \, d\phi \qquad (7)$$

The net increase of the mass flow rate is, then:

$$d\dot{m} = (\dot{m}_{o,x} - \dot{m}_{i,x}) + (\dot{m}_{o,\phi} - \dot{m}_{i,\phi})$$

$$= \rho_l \frac{\partial}{\partial \phi} \int_0^\delta u_\phi \, dr \, dx \, d\phi + \rho_l \frac{\partial}{\partial x} \int_0^\delta u_x \left(\frac{D}{2} + r\right) dr \, dx \, d\phi \qquad (8)$$

Neglecting the condensate-film thickness with respect to the tube radius, it is found that:

$$d\dot{m} = \rho_l \left(\frac{\partial}{\partial x} \int_0^\delta \frac{D}{2} u_x \, dr \, d\phi \, dx + \frac{\partial}{\partial \phi} \int_0^\delta u_\phi \, dr \, dx \, d\phi \right) \qquad (9)$$

2.3. Momentum Equation

Figure (3) shows an element of the condensate film and the system of forces acting on it. The element is acted upon by the force of gravity downwards, the buoyancy force upwards in the vertical direction, and the viscous force components in the axial and tangential directions. The element is in equilibrium under this system of forces. A force balance in the tangential direction gives:

$$m \, g \sin \beta \, \sin \phi = \mu \frac{\partial u_\phi}{\partial r} \left\{ \left(\frac{D}{2} + r\right) d\phi \right\} dx \qquad (10)$$

A force balance in the axial direction gives:

$$m \, g \cos \beta = \mu \frac{\partial u_x}{\partial r} \left\{ \left(\frac{D}{2} + r\right) d\phi \right\} dx \qquad (11)$$

with $\dot{m}$ = mass of the element − mass of displaced vapour

$$= (\rho_l - \rho_v) \left\{ \left(\frac{D}{2} + r\right) d\phi \right\} (\delta - r) \, dx \qquad (12)$$

By substituting, the value of m in the last two equations, it is found that:

$$\frac{\partial u_x}{\partial r} = \frac{(\rho_l - \rho_v) \, g}{\mu} \cos \beta \, (\delta - r), \qquad (13)$$

and

$$\frac{\partial u_\phi}{\partial r} = \frac{(\rho_l - \rho_v) \, g}{\mu} \sin \beta \, \sin \phi \, (\delta - r) \qquad (14)$$

Integrating the last two equations and using the boundary conditions:

$u_x = 0$ at $r = 0$ and

$u_\phi = 0$ at $r = 0$.

it is found that:

$$u_x = \frac{(\rho_l - \rho_v)\, g}{\mu} \cos\beta \left(\delta r - \frac{r^2}{2} \right), \tag{15}$$

and

$$u_\phi = \frac{(\rho_l - \rho_v)\, g}{\mu} \sin\beta \sin\phi \left(\delta r - \frac{r^2}{2} \right) \tag{16}$$

2.4. Energy Equation

By applying the concept of energy conservation on the study element, it is found that the heat conducted across the condensate film equals the heat rejected from the vapour during condensation. The energy balance for the element gives:

$$q \left(\frac{D}{2}\, d\phi\, dx \right) = d\dot{m}\ L \tag{17}$$

with q = the heat flux by conduction from. the vapour side to the surface of the tube

$= \frac{k}{\delta}\ \Delta t$

Hence.

$$d\dot{m} = \frac{k\ \Delta t}{\delta\ L} \left(\frac{D}{2}\, d\phi \right) dx \tag{18}$$

2.5. Mathematical Model

By substituting from Eqns. (15), (16) and (18) into Eqn. (9), it is found that:

$$\frac{k\,\Delta t}{\delta\, L}\ \frac{D}{2}\, d\phi\ dx = \rho_l \frac{D}{2}\, d\phi\ dx \left\{ \frac{2}{D} \frac{\partial}{\partial\phi} \int_0^\delta \frac{(\rho_l - \rho_v)\, g}{\mu} \sin\beta \sin\phi \left(\delta r - \frac{r^2}{2} \right) dr + \frac{\partial}{\partial x} \int_0^\delta \frac{(\rho_l - \rho_v)\, g}{\mu} \cos\beta \left(\delta r - \frac{r^2}{2} \right) dr \right\} \tag{19}$$

Integrating the last equation:

$$\frac{k\ \mu\ \Delta t}{\rho_l(\rho_l - \rho_v)\, g\, L\, \delta} = \frac{2}{D} \frac{\partial}{\partial\phi} (\sin\beta\ \sin\phi) \frac{\delta^3}{3} + \frac{\partial}{\partial x} \cos\beta\ \frac{\delta^3}{3}$$

Hence,

$$\frac{k\ \mu\ \Delta t}{L\ \rho_l\, (\rho_l - \rho_v)\, g} = \frac{2 \sin\beta}{3\, D}\, \delta\, \frac{\partial}{\partial\phi}(\sin\phi\ \ \delta^3) + \frac{\cos\beta}{3}\, \delta\, \frac{\partial}{\partial x}(\delta^3) \tag{20}$$

Or,

$$\frac{3\, D\, k\ \mu\ \Delta t}{2\, L\ \rho_l\, (\rho_l - \rho_v)\, g\, \sin\beta} = \delta \frac{\partial}{\partial\phi}(\delta^3 \sin\phi) + \frac{D}{2} \cot\beta\ \delta \frac{\partial}{\partial x}(\delta^3) \tag{21}$$

Let $N = \dfrac{3\, D\, k\ \mu\ \Delta t}{2\, L\ \rho_l\, (\rho_l - \rho_v)\, g\, \sin\beta}$, and $Z = \dfrac{2\, x}{D\ \cot\beta}$.

By substituting for the values of N and Z in Eqn. (21), it is found that :

$$N = \delta \frac{\partial}{\partial\phi} (\delta^3 \sin\phi) + \delta \frac{\partial}{\partial Z} (\delta^3)$$

$$= \frac{3}{4} \sin\phi\ \frac{\partial\delta^4}{\partial\phi} + \delta^4 \cos\phi + \frac{3}{4}\ \frac{\partial\delta^4}{\partial Z} \tag{22}$$

Let $Y = \frac{\delta^4}{X}$, and substituting in the last equation, it is found that:

$$\frac{3}{4}\sin\phi \frac{\partial Y}{\partial \phi} + Y \cos\phi + \frac{3}{4}\frac{\partial Y}{\partial Z} = 1$$

Or,

$$\frac{\partial Y}{\partial Z} + \sin\phi \frac{\partial Y}{\partial \phi} = \frac{4}{3}\,(1 - Y\cos\phi) \qquad (23)$$

Equation (23) is a nonlinear, first-order partial differential equation, and its solution gives the condensate-film thickness at any point on the surface of the tube.

The problem now is solving Eqn. (23) under the condition that the film thickness (δ) is equal to zero at the upper edge of the tube for any perimeter angle ϕ, i.e. :

$$Y(Z,\phi) = 0 \quad \text{for } Z = 0 \quad \text{and} \quad 0 \leq \phi < 2\pi$$

This problem can be solved analytically by the method of characteristics, as described by Smith [5]. The solution will be described in the following section.

2.6. Solution of the Mathematical Model

Smith [5] showed that the solution of a partial differential equation in the form: $a\frac{\partial u}{\partial x} + b\frac{\partial u}{\partial y} = c$ can be obtained by investigating the possibility of finding a direction at each point on the integral surface $\{u = f(x,y)\}$ along which the partial differential equation is transformed to an ordinary differential equation. This direction is called the characteristics curve. Smith [5] showed that the solution along this characteristics curve is given by:

$$\frac{dx}{a} = \frac{dy}{b} = \frac{du}{c}$$

and the equation of the characteristics curve is obtained by solving the equation:

$$\frac{dx}{a} = \frac{dy}{b}.$$

According to this method, the solution of the present problem along the characteristics curve is given by:

$$\frac{dZ}{1} = \frac{d\phi}{\sin\phi} = \frac{dY}{4/3\,(1 - Y\cos\phi)} \qquad (24)$$

The equation of the characteristics curve is given by:

$$\frac{dZ}{1} = \frac{d\phi}{\sin\phi}$$

Integrating the last equation gives:

$$e^{-Z}\tan\frac{\phi}{2} = C_1$$

The solution of the partial differential equation along this characteristics curve is given by:

$$\frac{dY}{4/3\,(1 - Y\cos\phi)} = \frac{d\phi}{\sin\phi}$$

Hence,

$$\frac{dY}{d\phi} + \frac{4}{3}\cot\phi\; Y = \frac{4}{3}\operatorname{cosec}\phi \qquad (25)$$

Equation (25) is an ordinary, linear, first-order differential equation. The solution of this equation has the following form as given by Wylie and Barrett [6]:

$$Y(z,\phi) = e^{-\frac{1}{3}\int \cot\phi \, d\phi} \left\{ \int \frac{1}{3} \left(e^{\frac{1}{3}\int \cot\phi \, d\phi} \right) \operatorname{cosec}\phi \, d\phi - C_2 \right\}$$

$$= (\sin\phi)^{-4/3} \left\{ \frac{1}{3} \int (\sin\phi)^{1/3} \, d\phi + C_2 \right\} \qquad (26)$$

Making use of the boundary condition $Y(0,\phi) = 0$, C_2 can be expressed in terms of C_1 as follows:

$$C_2 = -4/3 \int_0^{2 \arctan(C_1)} (\sin(\phi))^{1/3} \, d\phi .$$

Substituting the values of C_1 and C_2 in the solution of the partial differential equation, the analytical solution of equation (23) with its boundary conditions is:

$$Y(Z,\phi) = \frac{4}{3} (\sin\phi)^{-4/3} \left\{ \int_0^{\phi} (\sin\phi)^{1/3} d\phi - \int_0^{2 \arctan(e^{-Z} \tan\frac{\phi}{2})} (\sin\phi)^{1/3} \, d\phi \right\} \qquad (27)$$

Substituting for the various quantities in the last equation by their original ones, the expression of the film thickness results in:

$$\delta(x,\phi) = \left\{ \frac{2 D k \mu \Delta T}{L \rho_l (\rho_l - \rho_v) g \sin\beta} \right\}^{1/4} (\sin\phi)^{-1/3}$$

$$\times \left\{ \int_0^{\phi} (\sin\phi)^{1/3} d\phi - \int_0^{2 \arctan \left\{ \tan\frac{\phi}{2} e^{-(\frac{2x}{D} \tan\beta)} \right\}} (\sin\phi)^{1/3} d\phi \right\}^{1/4} \qquad (28)$$

The local heat-transfer coefficient at any point on the surface of the tube is given by:

$$h(x,\phi) = \frac{k}{\delta(x,\phi)}$$

and the average heat-transfer coefficient is given by:

$$h_{av} = \frac{\int_0^H \int_0^{\pi} h_{(x,\phi)} \left(\frac{D}{2} d\phi \right) dx}{\pi \, (D/2) \; H}$$

$$= \frac{k}{\pi H} \int_0^H \int_0^{\pi} \delta^{-1} (x,\phi) \quad d\phi \quad dx \qquad (29)$$

Equation (29) gives the average heat-transfer coefficient for every length and diameter of the tube and at any inclination of its axis with respect to the vertical at a given temperature difference between the outer surface of the tube and condensing vapour. The integrals in equation (28) cannot be calculated analytically. However, a computer program is developed to calculate these integrals numerically.

2.7. Verification of the Mathematical Model

Equation (28), obtained for vapour condensing outside an inclined tube, can be considered as a general equation for vapour condensing at any surface orientation. In the following sections, Nusselt equations for vapour condensing on vertical tube, horizontal tube and inclined flat plate will be derived from this general equation.

2.7.1 Vertical tube

The condensate film thickness for vapour condensing outside a vertical tube can be obtained by substituting with zero for the angle of inclination in Eqn. (20) as follows:

$$\frac{k\ \Delta t\ \mu}{\rho_l\,(\rho_l - \rho_v)\ g\ L} = \frac{1}{3}\ \delta\ \frac{d}{dx}(\delta^3)$$

Or,

$$\frac{3\ k\ \Delta t\ \mu}{\rho_l\,(\rho_l - \rho_v)\ g\ L} = \delta\ \frac{d\delta^3}{dx}$$

Hence,

$$\frac{3\ k\ \Delta t\ \mu}{\rho_l\,(\rho_l - \rho_v)\ g\ L} = \frac{3}{4}\ \frac{d\delta^4}{dx}$$

Or,

$$\frac{4\ k\ \Delta t\ \mu}{\rho_l\,(\rho_l - \rho_v)\ g\ L} = \frac{d\delta^4}{dx}$$

Integrating this equation gives:

$$\frac{4\ k\ \Delta t\ \mu}{\rho_l\,(\rho_l - \rho_v)\ g\ L}\ x = \delta^4 + C$$

Applying the boundary condition:

$$\delta = 0 \qquad \text{at} \qquad x = 0$$

gives

$$C = 0$$

Hence,

$$\delta = \left\{ \frac{4\ \mu\ k\ \Delta t\ x}{\rho_l\,(\rho_l - \rho_v)\ g\ L} \right\}^{0.25}$$

The local heat-transfer coefficient is:

$$h_{(x)} = \frac{k}{\delta}$$

$$= \left\{ \frac{\rho_l\,(\rho_l - \rho_v)\ g\ L\ k^3}{4\ \mu\ \Delta t\ x} \right\}^{0.25}$$

The average heat-transfer coefficient is:

$$h_{av} = \frac{\int_0^H h(x)\ dx}{H}$$

$$= 0.943\ \left\{ \frac{\rho_l\,(\rho_l - \rho_v)\ g\ k^3\ L}{\mu\ \Delta t\ H} \right\}^{0.25} \qquad (30)$$

The above expressions are the same as those obtained according to Nusselt analysis for steam condensing outside a vertical tube.

2.7.2 Horizontal tube

The codensate-film thickness for vapour condensing on a horizontal tube can be obtained by substituting with 90° for the angle of inclination in Eqn. (28) as follows:

$$\delta = \left\{ \frac{4\ \mu\ k\,(D/2)\ \Delta t}{\rho_l\,(\rho_l - \rho_v)\ g\ L\ (\sin\phi)^{4/3}} \int_0^{\phi} (\sin\phi)^{0.25}\ d\phi \right\}^{0.25} \qquad (31)$$

The local heat-transfer coefficient is:

$$h = \frac{k}{\delta} = \left\{ \frac{k^3 \rho_l (\rho_l - \rho_v) g L (\sin \phi)^{4/3}}{\mu (D/2) \Delta t \int_0^{\phi} (\sin \phi)^{1/3} d\phi} \right\}^{0.25}$$

The average heat-transfer coefficient is given by:

$$h_{av} = \frac{1}{\pi} \int_0^{\pi} h \, d\phi = \frac{1}{\pi} \int_0^{\pi} \left\{ \frac{k^3 \rho_l (\rho_l - \rho_v) g L (\sin \phi)^{4/3}}{4 \mu (D/2) \Delta t \int_0^{\phi} (\sin \phi)^{1/3} d\phi} \right\}^{0.25}$$

Using numerical integration, it is found that:

$$h_{av} = 0.74 \left\{ \frac{k^3 \rho_l (\rho_l - \rho_v) g L}{\mu D \Delta t} \right\}^{0.25} \tag{32}$$

The above expressions are the same as those obtained according to Nusselt analysis for vapour condensing outside a horizontal tube.

2.7.3 Inclined flat plate :

The condensate film-thickness for steam condensing on an inclined flat plate can be obtained by substituting with ∞ for the tube diameter in Eqn. (20) as follows:

$$\frac{k \Delta t \mu}{L \rho_l (\rho_l - \rho_v) g} = \frac{\cos\beta}{3} \delta \frac{d}{dx} (\delta^3) = \frac{\cos\beta}{3} \frac{3}{4} \frac{d\delta^4}{dx}$$

Integrating this equation gives:

$$\left\{ \frac{4 \mu k \Delta t x}{\rho_l (\rho_l - \rho_v) g \cos\beta L} \right\}^{0.25} = \delta$$

The local heat-transfer coefficient is:

$$h = \frac{k}{\delta} = \left\{ \frac{\rho_l (\rho_l - \rho_v) g \cos\beta k^3 L}{4 \mu \Delta t x} \right\}^{0.25}$$

Then, the average heat-transfer coefficient is:

$$h_{av} = 0.943 \left\{ \frac{\rho_l (\rho_l - \rho_v) g \cos\beta k^3 L}{\mu \Delta t H} \right\}^{0.25} \tag{33}$$

It is the same equation obtained according to Nusselt analysis.

3. RESULTS AND DISCUSSIONS

The results of the theoretical study obtained for vapour condensing around inclined tubes at different angles of inclination with the vertical are presented in Table (1). The theoretical results are for a condenser tube having a length of 2 m and a diameter of 0.021 m.

Moreover, the average heat-transfer coefficient is plotted versus the temperature difference for different inclination angles in Fig. (4). As shown in Fig. (4), the average heat-transfer coefficient increases with decreasing the temperature difference and with increasing the inclination angle from the vertical. The reasons of this behavior may be explained as follows : (a) Increasing the temperature difference increases the rate of heat transfer. The rate of condensation increases and, therefore, the thickness of condensate film increases as well. With increasing the thickness of condensate film, the thermal resistance to heat transfer increases. In other words, the heat transfer coefficient decreases. (b) The effect of the tube inclination on condensation process could be attributed to the behavior of condensate film and the path of condensate droplets on tube surface. This path is dependent upon the angle of inclination. As the angle of inclination from the vertical increases, the path of the condensate droplet on the tube surface will be shorter, and the amount of condensate remaining on the tube surface decreases to the extent where all condensate leaves in the form of droplets at various points on the underside of the tube. The number of such points increases as the angle of inclination from the vertical increases. This results in a reduced effective liquid film-thickness which contributes to a higher condensation heat-transfer coefficient.

4. NOMENCLATURE

Unless otherwise stated, the symbols used in the paper have the following meaning:

C	constant of integration	
D	tube outside-diameter	m
$d\dot{m}$	mass of condensate element	kg
g	acceleration due to gravity	m/s^2
H	length of cooling surface	m
h	local heat-transfer coefficient	$W/m^2 \cdot C$
h_{av}	average heat-transfer coefficient	$W/m^2 \cdot C$
k	condensate film thermal conductivity	$W/m \cdot C$
L	latent heat of condensation	J/kg
$\dot{m}$	mass flow rate of condensate film	kg/s
$\dot{m}_{i,x}$	inlet mass flow rate of condensate in x direction	kg/s
$\dot{m}_{i,\phi}$	inlet mass flow rate of condensate in ϕ direction	kg/s
$\dot{m}_{o,x}$	outlet mass flow rate of condensate in x direction	kg/s
$\dot{m}_{o,\phi}$	outlet mass flow rate of condensate in ϕ direction	kg/s
q	rate of heat transfer	W
r	radial distance measured from tube surface	m
u_x	velocity component in x direction	m/s
u_ϕ	velocity component in ϕ direction	m/s
x	distance measured from the tube top edge	m

Greek letters:

β	tube inclination angle from the vertical	radian
δ	condensate-film thickness	m
μ	dynamic viscosity	kg/m.s
ρ	density	kg/m^3
ρ_l	density of liquid	kg/m^3
ρ_v	density of vapour	kg/m^3
ϕ	perimeter angle of the tube	radian
Δt	temperature difference	°C

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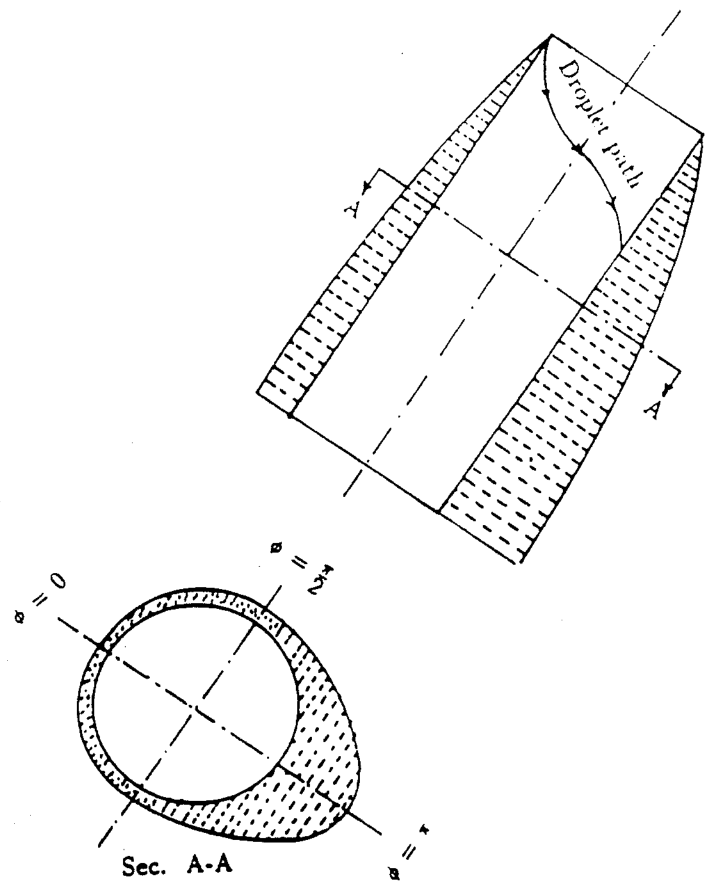

Distribution of condensate-film thickness

Fig. (1) : Condensation on inclined tube.

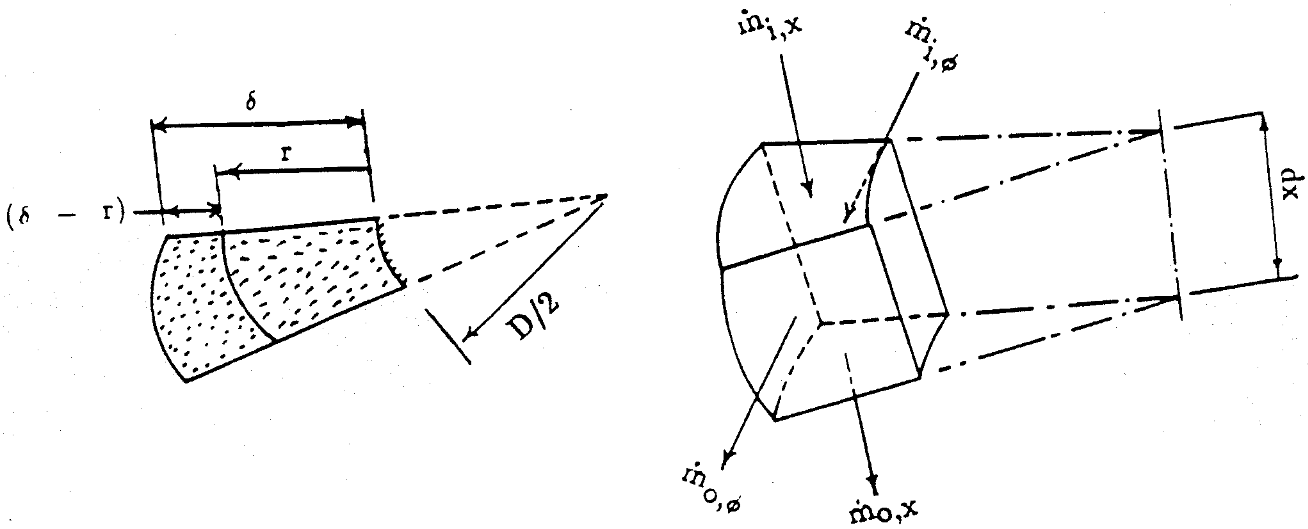

Fig. (2) : Element of the condensate film

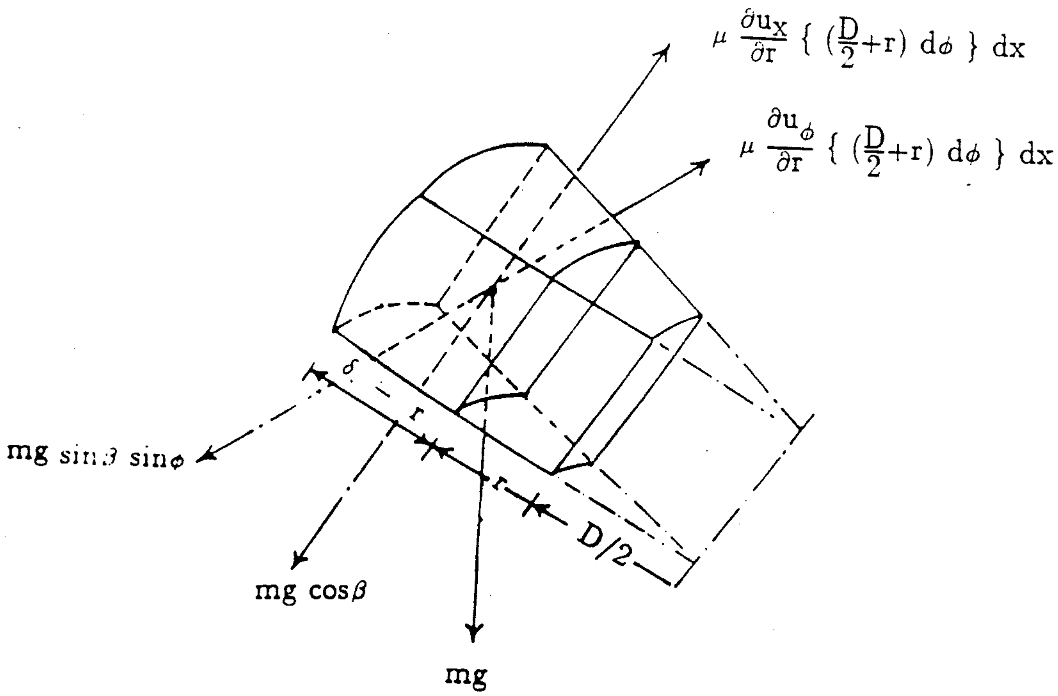

Fig. (3) : Forces balance on the study element

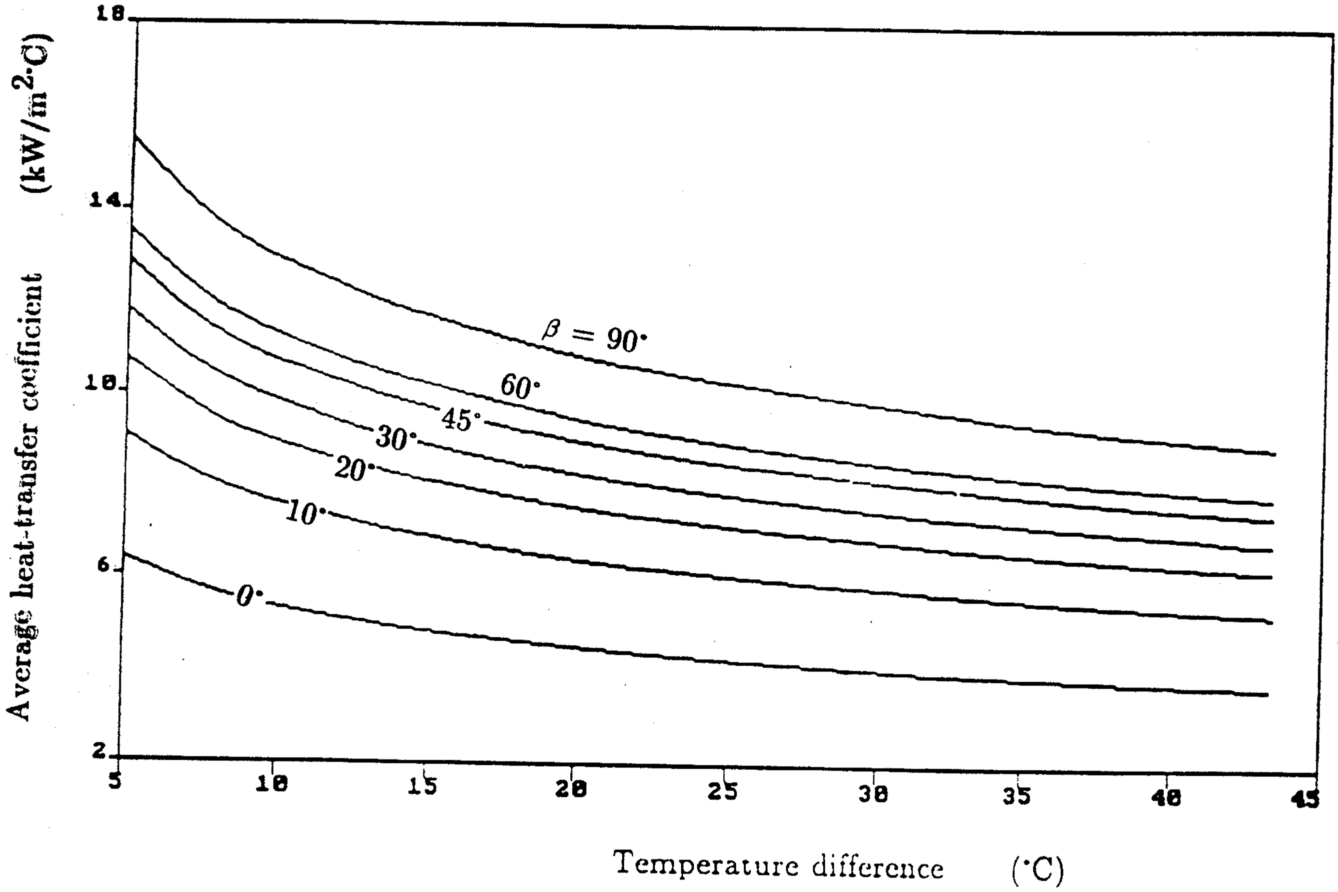

Fig. (4) : Variation of theoretical average heat-transfer coefficient with temperature difference.

Table (1) : Theoretical condensation average heat-transfer coefficient, $W/m^2 \cdot C$:

β / Δt (°C)	0°	10°	20°	30°	45°	60°	90°
4	6700.7	9554.2	11278.3	12395.2	13515.5	14217.9	16293.3
6	6054.8	8633.2	10191.1	11200.3	12212.6	12847.3	14722.6
8	5634.6	8034.1	9483.9	10423.0	11365.1	11955.7	13701.0
10	5328.9	7598.2	8969.3	9857.54	10748.5	11307.0	12957.6
12	5091.5	7259.6	8569.7	9418.3	10269.5	10803.2	12380.2
14	4899.0	6985.2	8245.7	9062.3	9881.3	10394.8	11912.2
16	4738.1	6755.9	7975.0	8764.7	9556.9	10053.5	11521.1
18	4600.7	6559.8	7743.6	8510.4	9279.6	9761.9	11186.8
20	4481.1	6389.3	7542.3	8289.2	9038.4	9508.1	10896.0
22	4375.5	6238.9	7364.7	8094.0	8825.6	9284.2	10639.4
24	4281.4	6104.6	7206.2	7919.8	8635.7	9084.4	10410.5
26	4196.6	5983.7	7063.4	7762.9	8464.6	8904.5	10204.2
28	4119.5	5873.8	6933.8	7620.4	8309.2	8741.0	10016.9
30	4049.1	5773.4	6815.2	7490.1	8167.1	8591.5	9845.7
32	3984.3	5681.0	6706.1	7370.2	8036.4	8454.0	9688.1
34	3924.4	5595.5	6605.3	7259.4	7915.5	8326.9	9542.4
36	3868.7	5516.1	6511.5	7156.4	7803.2	8208.7	9407.0
38	3816.7	5442.1	6424.1	7060.3	7698.4	8098.5	9280.7
40	3768.1	5372.7	6342.3	6970.3	7600.3	7995.3	9162.4
42	3722.4	5307.6	6265.4	6885.8	7508.2	7898.4	9051.3
44	3679.4	5246.2	6192.9	6806.2	7421.4	7807.1	8946.7