MILITARY TECHNICAL COLLEGE

CAIRO - EGYPT

1- # HEAT TRANSFER IN TWO PHASE FLOW

2- # OVER A FLAT PLATE

3- M. M. SHAMLOUL*

4- ## ABSTRACT

The theoretical model of heat and mass transfer during laminar two phase (air-water) flow along a flat plate is carried out . The governing continuity and energy equations with its boundary conditions are written for the free stream of two phase flow , the air water droplets mixture boundary layer and the water film over the plate. The equations are given for determining; the mass flux of water evaporation from film surface, the convective heat flux and the mass flux of droplets depositing on the flat plate. The analysis shows high flux resulting from the superposition of the film evaporation process from the plate surface and the heat transfer by convection. The convective heat flux appears higher than during dry air flow due to the enhancement effect of the droplets. Experimental work is carried out. The comparison between the results obtained in the theoretical model and those obtained from the experimental results is presented and it was found an agreement.

5- * Associate Professor, Mechanical Power Engineering Department, Faculty of Engineering, Zagazig University, El-Sharkia, EGYPT.

INTRODUCTION

Heat transfer in two phase flows has a considerable attension in the recent years due to the need to improve heat transfer rate . Previous works [1-2] investigated the heat transfer during the laminar flow of air water mist across a heated circular cylinder. Heat transfer to a mist flow inside a tube was studied by the Authors [3-4] . Simpson and Brolls [5] performed the analysis for droplet deposition on a flat plate from an air water mist in a turbulent flow. All the previous investigators found that for a given wall temperature or a specific heat transfer area, the heat transfer rate for two phase flow is higher than that may be in a single phase flow. Most of these studies are theoretical only. In the present work, theoretical and experimental investigations of heat transfer in a two phase system consisting of air with water droplet suspension flowing over a plate is presented.

THE THEORETICAL ANALYSIS

Consider an air-water two phase mixture flowing over a flat plate of uniform wall temperature t_w. The coordinates y and z are chosen with z along the plate and y perpendicular to it and v , u are taken to be the velocity components in the directions of y and z increasing respectively. While the mixture of air and water droplets flowing in the z direction, water droplets are settled on the plate forming water film. The physical system and coordinates are shown in Fig.1. The formulation of the governing continuity and energy equations for water film, free stream and air boundary layer can be simplified by introducing the following assumptions:

1. The flow of the water film over the plate is laminar.
2. Two phase, air boundary layer over the water film is laminar.
3. The temperatures of air and water droplets in the free stream are approximatively equal.
4. The temperature of the plate is constant and higher than the temperature of the air.
5. The effect of mass stream in the transversal direction; Separation of water droplets as well as water vapor upon the

velocity profile in air boundary layer is neglected.
The continuity and energy equations can be written as follows :

For The water film :

$$\frac{\partial m_l'}{\partial z} = m_d'' - m_v'' \quad (1)$$

$$q_w = q_c + m_v'' i_v + \frac{\partial}{\partial z}(m_l' \, i_l) - m_d'' \, i_d \quad (2)$$

For the free stream :

$$\frac{dx}{dz} = \frac{m_v'' - m_d''}{m_a'} \quad (3)$$

$$\frac{d}{dz}(m_a' \, i_m) = m_v'' \, i_v - m_d'' \, i_d + q_c \quad (4)$$

For the air boundary layer :

$$\frac{\partial u_a}{\partial z} + \frac{\partial v_a}{\partial y} = 0 \quad (5)$$

$$v_a \frac{\partial t}{\partial y} = \alpha \frac{\partial^2 t}{\partial y^2} - \frac{q_b}{\rho_a \, c_a} \quad (6)$$

With the following boundary conditions :

$$\left.\begin{array}{l} \text{For } z = 0 \text{ , } \delta_l = 0 \text{ , } \delta_a = 0 \text{ , } t = t_\infty \text{ , } u = u_\infty \text{ , } v_a = 0 \\ \text{For } z = l \text{ and } y = 0 \text{ , } t = t_w \text{ , } u = 0 \\ \qquad y = \delta_l \text{ , } t = t_f \text{ , } u = u_f \text{ , } v_a = v_f \\ \qquad y = \delta_l + \delta_a \text{ , } t = t_\infty \text{ , } u = u_\infty \text{ , } v_a = v_{a\infty} \end{array}\right\} \quad (7)$$

The energy dissipation is neglected in Eq.(6) but considering the heat vent q_b which depends upon the motion of the water droplets through the air boundary layer. These droplets are penetrating through the boundary layer which increases the temperature gradient in this layer and intensification the heat transfer by convection. The heat received through separation of droplets stream can be determined as ;

$$q''_d = m''_d \; c_l \; (t_f - t_\infty) \tag{8}$$

Hence the heat vent can be written as ;

$$q_b = \frac{q''_d}{\delta_a} = \frac{m''_d \; c_l \; (t_f - t_\infty)}{\delta_a} \tag{9}$$

To solve the system of equations from Eq.(1) to Eq.(6), consider the following two additional foundation. The first is that a small water film resistance occurs, then $t_f = t_w$. The second is that the transversal velocity is substituted by its average value as follows ;

$$\bar{v}_a = \frac{1}{\delta_a} \int_0^{\delta_a} v_a \; dY \tag{10}$$

Where ; $Y = y - \delta_l$

With these accepted establishments, Eq.(2) tends to the form ;

$$q_w = q_c + q_v \tag{11}$$

Where ;

$$q_v = m''_v \; H \tag{12}$$

And represents the heat stream by evaporation of the water film. Chilton-Colburn analogies [7] found that ;

$$\frac{h_{ao}}{h_D} = \rho_a \; c_a \; Le^{2/3} \tag{13}$$

The heat stream q_v can be determined as ;

$$q_v = h_D \; (\rho_{vw} - \rho_{v\infty}) \; H \doteq \frac{h_{ao} \; (\rho_{vw} - \rho_{v\infty}) H}{\rho_a \; c_a \; Le^{2/3}} \tag{14}$$

The convective heat stream q_c depends upon the temperature profile of the air in the boundary layer. By using the above foundation, Eq.(6) can be written as ;

$$\frac{\partial^2 t}{\partial y^2} - A \frac{\partial t}{\partial y} - \frac{B}{A} = 0 \tag{15}$$

With the following boundary conditions ;

$$\text{For } Y = 0 \text{ , } t = t_w \text{ and for } Y = \delta_a \text{ , } t = t_\infty \tag{16}$$

Where ;

$$A = \bar{v}_a / \alpha \quad , \quad B = q_b / k_a \quad \text{and} \quad Y = y - \delta_1 \tag{17}$$

Then Eq.(15) can be solved to get the following temperature distribution as ;

$$t = t_w + \left(t_\infty - t_w + \frac{B}{A}\delta_a \right) \frac{\exp(A\,Y) - 1}{\exp(A\,\delta_a) - 1} - \frac{B}{A} Y \tag{18}$$

Hence the heat stream q_c is written as ;

$$q_c = - k_a \frac{\partial t}{\partial Y}\bigg|_{Y=0} = - k_a \left[\frac{(t_\infty - t_w + (B/A)\,\delta_a)\,A}{\exp(A\,\delta_a)} - \frac{B}{A} \right] \tag{19}$$

It is convenience to transform the energy equations Eq.(11) , Eq.(14) and Eq.(19) to the nondimensional forms. Divide these equations by the heat stream for the flow of dry air (without droplets) over the plate q_{co} where ;

$$q_{co} = h_{co} (t_w - t_\infty) = h_{co} \; \Delta t \tag{20}$$

Hence the energy equation can be written in the form of the intensification of the heat transfer coefficient as ;

$$Q_t = Q_c + Q_v \tag{21}$$

Where ;

$$Q_t = \frac{q_w}{q_{co}} \tag{22}$$

$$Q_c = \frac{q_c}{q_{co}} = 1 + \frac{m_d'' \, c_1}{k_a \; A} \left(\frac{\exp(A\,\delta_a) - 1}{A\,\delta_a} - 1 \right) \tag{23}$$

$$Q_v = \frac{q_v}{q_{co}} = \frac{(\rho_{vw} - \rho_{v\infty})\,H}{\rho_a \; c_a \; Le^{2/3} \; \Delta t} \tag{24}$$

It is clear from Eq.(23) that the heat stream by convection is proportional to the separation stream of droplets.
Simpson and Brolls [5] found that the separation stream of the droplets can be written as ;

$$m_d'' = x \; h_{ao} \; / \; c_a \tag{25}$$

The present analysis concerns with the accident where the separation stream of droplets is greater than the mass vapor stream. If the droplets capacity in air decreases, it must be a limited value as ;

$$m_{vlm}'' = m_{dlm}'' \tag{26}$$

Based upon Eq.(12) , Eq.(13) and Eq.(24) the mass vapor stream can be represented as ;

$$m_{vlm}'' = Q_v \; \Delta t \; h_{ao} \; / \; H \tag{27}$$

But the droplets stream as ;

$$m_{dlm}'' = x_{lm} \; h_{ao} \; / \; c_a \tag{28}$$

Hence the limit of the droplets capacity is ;

$$x_{lm} = Q_v \; \Delta t \; c_a \; / \; H \tag{29}$$

If $x < x_{lm}$, it means that all quantity of the droplets separation are evaporated, then Eq.(11) can be written as ;

$$q_w = q_c + m_d'' \; H \tag{30}$$

The energy equation can be transformed to the nondimensional form as before by dividing both sides of Eq.(30) by q_{co} getting ;

$$Q_t = 1 + x \left[\frac{m_d'' \; c_l}{x \; k_a \; A} \left(\frac{\exp(A\,\delta_a) - 1}{A\,\delta_a} - 1 \right) + \frac{H}{c_a \Delta t} \right] \tag{31}$$

The calculated values of the intensity coefficient of the heat transfer Q_t are based on Eq.(21) and Eq.(31) .

THE EXPERIMENTAL WORK

Schematic layout of the experimental stand is shown in Fig.2. A steel plate of dimensions 80 x 160 x 1 mm is located in a circular channel of 1000 mm length and 250 mm diameter. The plate is connected with arm supported on an angle indicator so that its inclination angle may be adjusted. Air mist is created as the water droplets mixed in the air stream were sprayed through the nozzle. The capacity of the water droplets in the air is measured by the water balance to the air and the separated part in the channel. The flow rates of the water and the air were measured by a calibrated rotameters. But the flow rate of the separated part of the water in the channel was measured by a graduated tank and a stop watch. There were eight locations for the temperature measurements indicated in Fig.2. by a symbol Th and distributed as ; two locations for the air before and after the nozzle, one location for the water before the nozzle and five locations each 4 mm apart for the plate at its mid width as shown in Fig.2. The plate is shown in the vertical position. The temperatures were measured by a calibrated thermocoax thermocouples which are chromel-alumel type, connected directly with digital multimeter (Sinclair DM 350) . The sinclair multimeter have its selector switch for all locations of the thermocouples. The temperatures were measured directly by the pointer readings on the meter which are indicated in °C. All the measurements were taken for the horizontal and the vertical positions of the plate in the cicular channel. The experimental test rig was carried out in Heat Transfer Laboratory at Faculty of Engineering, Zagazig University.

RESULTS AND DISCUSSIONS

The boundary conditions for the theoretical solutions were determined from the experimental measurements. The theoretical and the experimental results of the intensity coefficient of heat transfer Q_t were obtained for a wide range of parameters ($x = 0 - 8$ % , $t_w = 20 - 50$ °C and $u = 0.4 - 3.3$ m/s) . The theoretical results were represented by a dashed lines while the experimental results were represented by a symbols as shown in Figs.(3-4) .

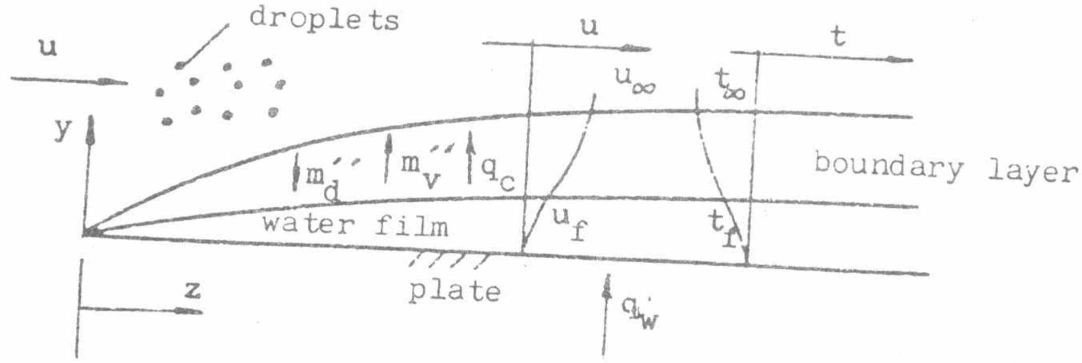

Fig.1. Schematic representation of the physical system.

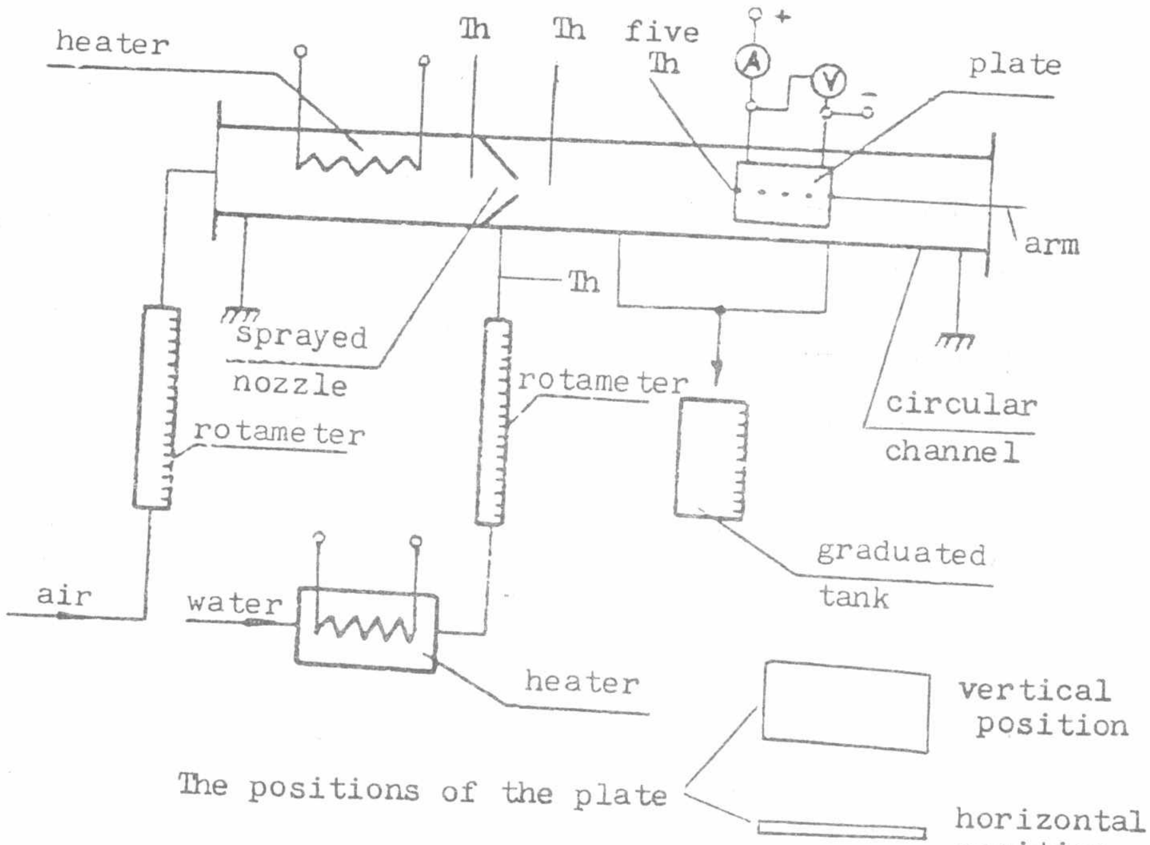

Fig.2. Schematic layout of the experimental rig.

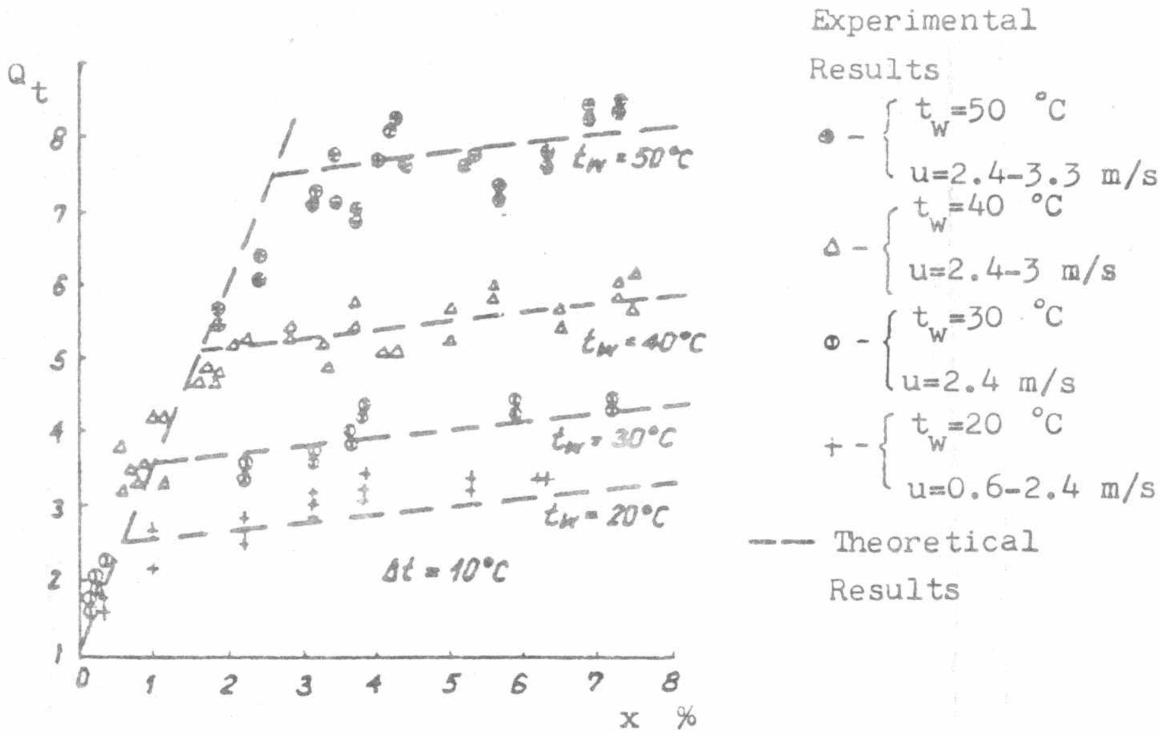

Fig.3. Theoretical and experimental results of the coefficient Q_t for a vertical plate.

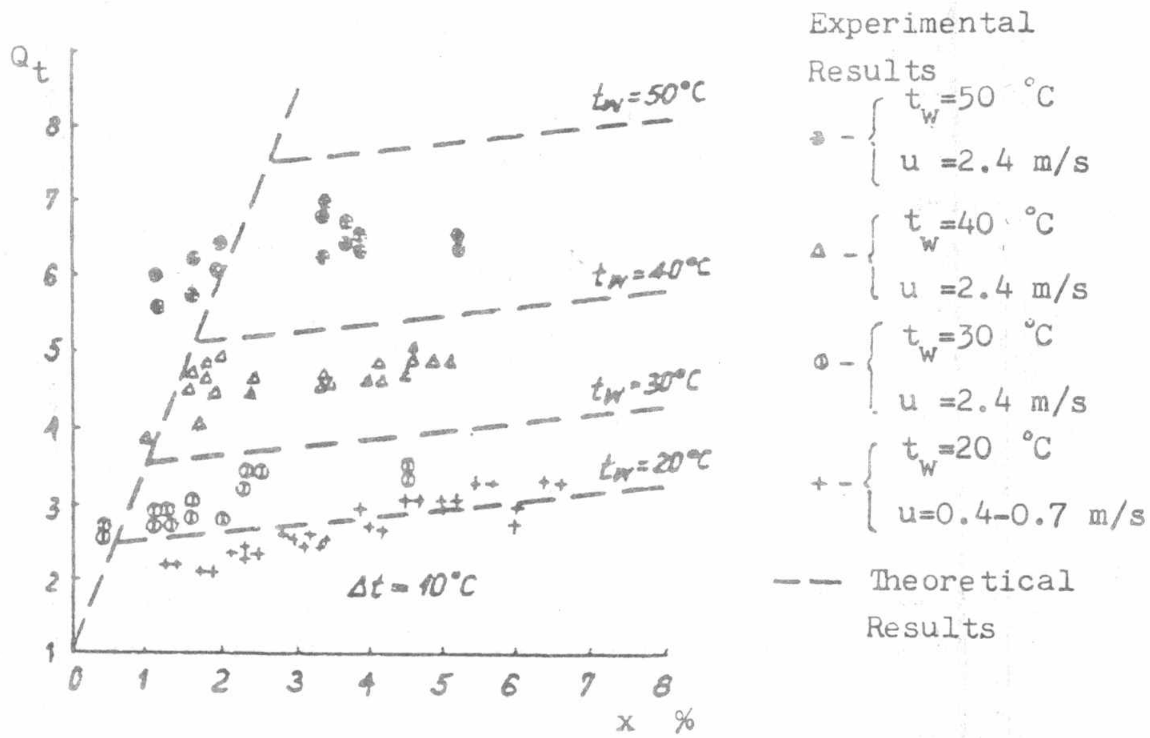

Fig.4. Theoretical and experimental results of the coefficient Q_t for a horizontal plate.

It is clear that the intensity coefficient of heat transfer Q_t was increased by increasing x and t_w for constant value of the temperature difference $\triangle t = 10\ ^oC$. It was found an agreement between the theoretical and the experimental results specially for the plate in a vertical position.

CONCLUSIONS

From the presented theoretical and experimental results, it can be concluded that ;

1. The intensity of heat transfer was increased by increasing the capacity of the droplets in the air stream and the plate temperature for constant temperature difference between the plate and the free stream of the air water mist.
2. The higher growth of the intensity of heat transfer occurs for the case where the capacity of the water droplets in the air is smaller than the limited value.

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NOMENCLATURE

c - specific heat at constant pressure

H - heat of evaporation

h - coefficient of heat or mass transfer

i - enthalpy

k - thermal conductivity

Le - Lewis number

l - length of the plate

m'' - mass flux of the stream

m - mass flow rate

Q - nondimensional heat flux

q - heat flux of the stream

t - temperature

u - longitudinal velocity

v - transversal velocity

x - capacity or contents of the droplets in the air

y - coordinate perpendicular to the plate

z - coordinate along the plate

α - thermal difussivity

ρ - density

δ - boundary layer thickness

Subscripts

a - air

ao - dry air

b - vent

c - convection

D - mass

co - convection without droplets

d - droplets separation

l - water

f - film

lm - limit

m - air mist

t - total

v - evaporation or vapor

w - wall

∞ - free stream