Int. J. Heat Mass Transfer. Vol. 24, No. 4, pp. 749–755, 1981
Printed in Great Britain

0017-9310/81/040749-07 $02.00/0
Pergamon Press Ltd

AN APPROXIMATE CALCULATION OF HEAT TRANSFER DURING FLOW OF AN AIR–WATER MIST ALONG A HEATED FLAT PLATE

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(*Received* 19 *March* 1979; *and in revised form* 1 *July* 1980)

Abstract—An approximate calculation of heat transfer during laminar flow of mist air along a flat plate placed in a channel is carried out. The high heat flux is shown to result from the superposition of film evaporation process from the plate surface and convective heat transfer. Convective heat flux appears higher than during a dry air flow due to the enhancement effect of the droplets. Theoretical results show good agreement with experiments.

NOMENCLATURE

- a, thermal diffusivity;
- A, coefficient defined by equation (30);
- B, coefficient defined by equation (30);
- C, coefficient defined by equation (38);
- C_p, specific heat at constant pressure;
- E_c, enhancement factor defined by equation (25);
- E_t, enhancement factor defined by equation (24);
- E_v, enhancement factor defined by equation (26);
- g, Reynolds flux;
- h_d, heat transfer coefficient of dry air;
- h_D, mass transfer coefficient;
- i, enthalpy;
- k, thermal conductivity;
- Le, Lewis number ($Le = Sc/Pr$);
- $\dot{m}$, mass flow rate;
- Nu, Nusselt number;
- Pr, Prandtl number;
- q, heat flux;
- q_b, heat transfer rate of the sinks per unit volume;
- r, latent heat of vaporization;
- Re, Reynolds number;
- Sc, Schmidt number;
- t, temperature;
- u, longitudinal velocity;
- v, perpendicular velocity;
- x, mist quality (moisture content);
- y, distance from wall;
- Y, distance from film.

Greek symbols

- δ_g, boundary layer thickness;
- δ_l, film thickness;
- ρ, density.

Subscripts

- c, conduction and convection;
- co, pure conduction;
- cd, conduction and convection for dry air;
- cr, critical;
- d, deposition;
- f, water film surface;
- g, air;
- l, water film;
- ld, water droplets;
- m, mixture;
- v, vapour, evaporation;
- w, wall;
- ∞, air outside the boundary layer.

Superscripts

- $\bar{\ }$, mean value;
- $\dot{\ }$, per unit time;
- $'$, per unit width;
- $''$, per unit area.

INTRODUCTION

HEAT transfer during a two-phase air–water droplet flow occurs in many technological processes and is distinguished by the heat transfer coefficients much higher than those attainable during a dry air flow.

In the literature one can find a number of papers devoted to this problem. Elperin [1] performed the experiments with tube banks situated in a mist air flow and found the possibility to enhance heat transfer as high as 20 times as compared to that of dry air. This was also confirmed by other authors such as, Acrivos, Ahren and Nagy [2] or Hodgson, Staterbak and Sunderland [3]. A theoretical analysis of heat transfer during a mist air flow around the cylinder was carried out by Goldstein, Yang and Clark [4] and Finlay [5].

Much less attention has been paid to investigations of heat transfer during flow along a flat plate. Bhatti [6, 7] performed the analysis of heat transfer of a laminar mist air flow along a plate at a relatively low concentration of droplets in the air. He assumed that the droplets evaporated entirely or partially when entering the boundary layer and therefore the plate remained dry. The results obtained revealed only a slight enhancement of heat transfer.

Simpson and Brolls [8] considered heat transfer of a turbulent flow along an entirely wetted plate. Their

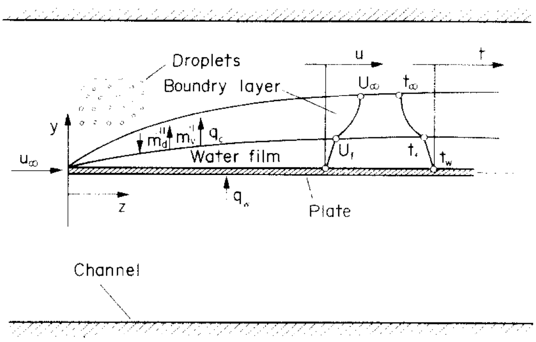

FIG. 1. Analytical model of heat transfer.

model assumed that the droplets crossing the boundary layer destroyed the laminar sublayer contributing to the heat transfer augmentation. These results, supported by experiments, showed a manifold increase of heat transfer rate.

This paper presents an analysis of heat transfer during laminar flow of mist air along a plate placed in a channel. The analysis applies basically to the film covered plate; but under certain conditions remains valid also for a partially wetted plate. The original purpose of this investigation was to learn more about the heat transfer characteristics of cooling towers and evaporative coolers. The analysis is a continuation of the previous papers of the author [9, 10].

SIMPLIFIED THEORETICAL ANALYSIS

The mist air containing small water-droplets flows in the channel over the plate, as shown in Fig. 1. Due to the effect of drag, inertia, buoyancy and gravity forces, together with a certain turbulence of the free stream, the droplets cross the boundary layer and settle on the plate forming a water film. The plate temperature is higher than the air temperature and heat transfer occurs between the plate and the air.

The following simplifying assumptions are taken in the analysis:

1. Water film flow along the plate is laminar.
2. Two-phase flow in the boundary layer of the plate is laminar.
3. The air temperature and the temperature of droplets in the free stream outside the boundary layer are approximately equal.
4. The plate temperature is constant and higher than the air temperature.
5. Reflection of droplets from the film surface does not occur.
6. Concentration of droplets is relatively low, so that the influence of depositing droplets on the boundary layer parameters may be neglected.
7. The mass flux of vapour due to film evaporation does not affect the boundary layer significantly, so that the analogy between heat and mass transfer may be applied to estimate the vapour mass flux.

On the basis of the above assumptions the mass and energy balance equations have been formulated for the water film, the air boundary layer and the free stream in the channel.

Water film

Mass balance of water film on an infinitesimal length dz per unit width of the plate is expressed as

$$\frac{\partial \dot{m}_l'}{\partial z} = \dot{m}_d'' - \dot{m}_v''. \tag{1}$$

With heat conduction along the water film being neglected, its energy balance takes on the form

$$q_w = q_c + \dot{m}_v'' i_v + \frac{\partial}{\partial z}(\dot{m}_l' i_l) - \dot{m}_d'' i_d. \tag{2}$$

Free stream in the channel

Mass balance

$$\frac{\mathrm{d}x}{\mathrm{d}z} = \frac{\dot{m}_v'' - \dot{m}_d''}{\dot{m}_g'}. \tag{3}$$

Energy balance

$$\frac{\mathrm{d}}{\mathrm{d}z}(\dot{m}_g' i_m) = \dot{m}_v'' i_v - \dot{m}_d'' i_d + q_c. \tag{4}$$

Moisture content (in the form of vapour and droplets) of the mist air is

$$x = \frac{\dot{m}_{ld}' + \dot{m}_v'}{\dot{m}_g' + \dot{m}_v' + \dot{m}_{ld}'} \cong \frac{\dot{m}_{ld}'}{\dot{m}_g' + \dot{m}_{ld}'}. \tag{5}$$

From equations (3) and (4) one can obtain the temperature increase of the free stream in the channel [9]

$$\frac{\mathrm{d}t_\infty}{\mathrm{d}z} = \frac{\dot{m}_v''(i_v - i_d) + q_c}{\dot{m}_g' C_{pm}}. \tag{6}$$

Specific heat of mist air is higher than that of dry air and this makes it possible to neglect the temperature increase of the free stream, i.e. $\mathrm{d}t_\infty/\mathrm{d}z = 0$, in many common cases.

Boundary layer of the air

The continuity equation

$$\frac{\partial u_g}{\partial z} + \frac{\partial v_g}{\partial y} = 0. \tag{7}$$

The energy equation

$$v_g \frac{\partial t}{\partial y} = a\frac{\partial^2 t}{\partial y^2} - \frac{q_b}{\rho_g C_{pg}}. \tag{8}$$

In the energy equation two terms have been neglected: the dissipative term because of the low flow velocity and the term $u_g \partial t/\partial z$ because at the film–air interface the relative velocity of air in the longitudinal direction is equal to zero while the transverse velocity reaches a fixed value due to evaporation. Bearing in mind that the transverse temperature gradient is much higher than the longitudinal one, we may neglect the term $u_g \partial t/\partial z$, as compared to $v_g \partial t/\partial y$, in our further analysis. Alternatively, the term $q_b/\rho_g C_{pg}$ appeared in the energy equation expressing heat sinks resulting from the heat transfer between the droplets and air.

The boundary conditions for the boundary layer and the film are formulated as follows (Fig. 1):

$$\left.\begin{aligned}
&\text{at } z = 0;\\
&\delta_l = 0;\quad \delta_g = 0;\quad t = t_\infty;\quad u_g = u_\infty;\quad v_g = 0\\
&\text{and at } z = L\\
&\text{and}\\
&y = 0;\quad t = t_w;\quad u_g = 0\\
&y = \delta_l;\quad t = t_{gf};\quad u_g = u_f;\quad v_g = v_{gf}\\
&y = \delta_l + \delta_g;\quad t = t_\infty;\quad u_g = u_\infty;\quad v_g = v_{g\infty}.
\end{aligned}\right\} \quad (9)$$

The solution of equation (8) requires the velocity profile v_g to be known. This, in turn, may be obtained from the continuity equation (7) by substituting $Y = y - \delta_l$

$$v_g - v_{gf} = -\int_0^Y \frac{\partial u_g}{\partial z}\,\mathrm{d}Y. \quad (10)$$

Having in mind the remarks concerning the convective term in equation (8), one can replace the exact solution of that equation by an approximate one. The simplification depends on substitution of the velocity v_g by mean velocity defined as

$$\bar{v}_g = \frac{1}{\delta_g}\int_0^{\delta_g} v_g\,\mathrm{d}Y. \quad (11)$$

Equation (8) then becomes the linear equation with constant coefficients

$$\frac{\partial^2 t}{\partial Y^2} - \frac{\bar{v}_g}{a}\frac{\partial t}{\partial Y} - \frac{q_b}{k_g} = 0. \quad (12)$$

The solution of equation (12) with the boundary conditions (9) is the temperature profile within the boundary layer

$$t = t_f + \left(t_\infty - t_f + \frac{b}{d}\delta_g\right)\frac{\exp(\mathrm{d}Y) - 1}{\exp(\mathrm{d}\delta_g) - 1} - \frac{b}{\mathrm{d}}Y \quad (13)$$

where

$$\left.\begin{aligned}
d &= \frac{\bar{v}_g}{a}\\
b &= \frac{q_b}{k_g}.
\end{aligned}\right\} \quad (14)$$

The coefficient b should be determined before applying the above temperature profile. When the mist air flows over a plate, the water droplets cross the boundary layer and are deposited onto the plate producing a film. Air temperature in the boundary layer increases from t_∞ to t_f. It follows from the assumptions adopted, that the air is saturated both in the free stream and the boundary layer. Hence, heat and mass transfer occurs between the cooler droplets and the surrounding air, contributing to the increase of the droplet temperature. For the mist air flow over a plate it is reasonable to assume that there is a thermal balance between the droplets and their environment. This is due to the relatively long time period that they remain within the boundary layer, which is connected with only slight forces acting perpendicularly to the plate surface. The heat flux acquired by the droplets during their motion across the boundary layer amounts to

$$\dot{q}_d'' = \dot{m}_d'' C_{pld}(t_f - t_\infty) \quad (15)$$

and therefore

$$q_b = \frac{\dot{q}_d''}{\delta_g} = \frac{\dot{m}_d'' C_{pld}(t_f - t_\infty)}{\delta_g}. \quad (16)$$

The value of the coefficient b is to be calculated from equation (14).

A change in diameter of the droplets during their passage through the boundary layer has been neglected in equation (15).

On the basis of the above set of equations it is possible to determine the wall heat flux assuming adequate velocity and temperature profiles of the film when the mass fluxes $\dot{m}_v''$ and $\dot{m}_d''$ are known. However, this method requires elaborate computations and is less descriptive.

The computations are substantially simplified without a significant decrease in accuracy with an additional assumption that the thermal resistance of the water film is negligibly small when compared with the resistance of the air boundary layer. Simpson and Brolls [8] proved the validity of the assumption. Neglecting the film resistance results in the identity $t_f = t_w$. In this case the energy equation of the film reduces to

$$q_w = q_c + q_v \quad (17)$$

where

$$q_v = \dot{m}_v'' r \quad (18)$$

stands for the heat flux due to film vaporization.

By virtue of the Chilton–Colburn analogy

$$\frac{h_d}{h_D} = \rho_g C_{pg} Le^{2/3} \quad (19)$$

the heat flux q_v can be determined from

$$q_v = h_D(\rho_{vw} - \rho_{v\infty})r = \frac{h_d(\rho_{vw} - \rho_{v\infty})r}{\rho_g C_{pg} Le^{2/3}}. \quad (20)$$

The value of the convective heat flux in the case considered differs from that of the dry air. Having in mind the assumption made that the mass flux $\dot{m}_v''$ is negligible and does not significantly affect the boundary layer, the heat flux q_c is controlled by the temperature distribution within the boundary layer. Thus, from equation (13) with the assumption of $t_f = t_w$ one obtains

$$q_c = -k_g\left(\frac{\partial t}{\partial Y}\right)_{Y=0} = -k_g\left[\frac{[t_\infty - t_w + (b/d)\delta_g]d}{\exp(d\delta_g)-1} - \frac{b}{d}\right]. \quad (21)$$

An essential question in the analysis of heat transfer of mist air flow is the determination of the heat transfer enhancement factor relative to dry air flow.

The convective heat transfer coefficient of dry air flowing along a plate, according to the Newton formula, equals

$$q_{cd} = h_d(t_w - t_\infty). \quad (22)$$

On the other hand, if the mass flow rates of deposition and water vapour are small and do not affect the velocity profile within the boundary layer, the heat flux q_{cd} may be determined from equation (13) at $b = 0$

$$q_{cd} = -k_g\left(\frac{\partial t}{\partial Y}\right)_{Y=0} = -\frac{k_g d(t_\infty - t_w)}{\exp(d\delta_g)-1}. \quad (23)$$

Dividing the both sides of equation (17) by the heat flux q_{cd} one obtains, after appropriate rearrangements, the heat transfer enhancement factor of mist air flowing along a plate

$$E_t = E_c + E_v \quad (24)$$

where

$$E_c = 1 + \frac{\dot{m}_d'' C_{pl}}{k_g d}[(\exp(d\delta_g)-1)/d\delta_g - 1] \quad (25)$$

$$E_v = \frac{(\rho_{vw} - \rho_{v\infty})r}{\rho_g C_{pg} Le^{2/3}(t_w - t_\infty)}. \quad (26)$$

The factor E_c expresses the enhancement of convective heat transfer and is proportional to the mass flux of droplet deposition, as follows from equation (25). According to the model accepted, the deposition of droplets is mainly due to the turbulence of the free stream. The research of Simpson and Brolls [11] produced the evidence that the mass flux of droplet deposition during flow along a flat plate may be assessed by virtue of the Reynolds analogy [12]. According to the analogy the Reynolds flux perpendicular to the plate is

$$g = \frac{h_d}{C_{pg}}. \quad (27)$$

Thus, the mass flux of droplets amounts to

$$\dot{m}_d'' = x\frac{h_d}{C_{pg}}. \quad (28)$$

On the basis of the above equations the heat transfer enhancement factor may be expressed by

$$E_t = Ax + B \quad (29)$$

where

$$A = \frac{h_d C_{pl}}{k_g d C_{pg}}\left(\frac{\exp(d\delta_g)-1}{d\delta_g} - 1\right) \quad (30)$$

$$B = 1 + E_v.$$

The analysis of heat transfer of mist air flowing along a plate, as described above, applies also to the partially wetted or even dry plate. The condition is that the plate temperature is to be lower than the dew point temperature.

If this condition is satisfied, the heat transfer process in the boundary layer remains the same as in the case of a plate entirely covered with film.

If the moisture content of the air decreases during its flow along an entirely wetted plate then at a certain moisture content, referred to as critical, the film flow disappears and small droplets form on the plate. The critical value of the moisture content may be determined by equating the mass flow rates

$$\dot{m}_{vcr}'' = \dot{m}_{dcr}''. \quad (31)$$

Mass flux of vapour may be expressed by combining equations (18), (19) and (26) into

$$\dot{m}_{vcr}'' = \frac{E_v \Delta t}{r} h_d \quad (32)$$

and the critical mass flux of the droplets amounts to

$$\dot{m}_{dcr}'' = x_{cr}\frac{h_d}{C_{pg}}. \quad (33)$$

Hence

$$x_{cr} = \frac{E_v \Delta t}{r} C_{pg}. \quad (34)$$

For $x < x_{cr}$, the total amount of the droplets deposited should evaporate, and the relationship similar to equation (17) may therefore be written as

$$q_w = q_c + \dot{m}_d'' r. \quad (35)$$

Taking into account equations (28) and (34), the above relationship may be rewritten in the form

$$q_w = q_c + \frac{x}{x_{cr}} h_d E_v \Delta t. \quad (36)$$

On the basis of this equation one obtains the formula which describes heat transfer in terms of the enhancement factor

$$E_t = Cx + 1 \quad (37)$$

where

$$C = A + \frac{E_v}{x_{cr}} = A + \frac{r}{C_{pg}\Delta t}. \quad (38)$$

The heat transfer enhancement factor is expressed by a linear relationship, similar to the case of a plate entirely covered by a water film. A comparison of the coefficients of proportions A and C suggests that the increase of the factor E_t due to increasing x is

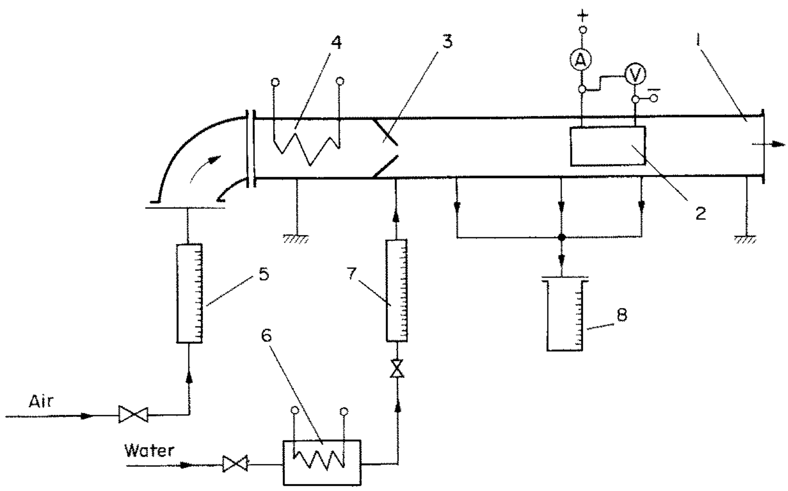

FIG. 2. Layout of the experimental apparatus: 1 channel; 2 test plate; 3 atomizer; 4 air heater; 5 rotameter; 6 water heater; 7 rotameter; 8 measurement tank of deposition water.

substantially higher for a partially wetted plate.

For the sake of making the analysis more comprehensive the contribution of conduction and convection to heat transfer from the water film to the air is considered. If the convective term is omitted in the energy equation (8), this equation takes on the form

$$\frac{\partial^2 t}{\partial Y^2} - b = 0. \tag{39}$$

Solution of the above equation yields the temperature profile

$$t = t_w + \left[\frac{(t_\infty - t_w) - b\delta_g^2}{\delta_g}\right] Y + bY^2. \tag{40}$$

In the same way as above one can determine the heat flux q_c and the heat transfer enhancement factor E_{co}

$$E_{co} = 1 + \frac{\dot{m}'' C_{pl}\delta_g k_g}{k_g} = 1 + x\frac{h_d C_{pl}\delta_g}{C_{pg}k_g}. \tag{41}$$

Calculations performed in [9] have shown that the factor E_{co} is slightly smaller than the factor E_c. For a very high moisture content, $x = 0.1$, the factor E_{co} is scarcely 15% less than E_c and this difference diminishes with the decrease of x. It suggests that conduction prevails in heat transfer from the film to the air.

COMPARISON WITH EXPERIMENTS

In searching the available literature the author has failed to find any experimental results concerning heat transfer during laminar flow of mist air along a plate. Hence, the analytical results presented will be compared with the experiments performed on the test apparatus described by Trela and Kowalski [10]* the layout of which is shown in Fig. 2. An electrically-heated stainless steel plate of dimensions 72 mm × 166 mm × 1 mm was placed in a wind tunnel made from glass tube. Mist air was produced by an ejector atomizer. The measured quantities were: plate temperature, air temperature, electric power of heating, mass flow rate of air, mass flow rate of water, mass flow rate of water deposition and humidity of atmospheric air.

The heat transfer enhancement factor E_t was calculated by determining the heat transfer rate and, in consequence, the heat transfer coefficients at the same flow velocities for both dry and mist air. The value of E_t was the ratio of these coefficients.

The experiments were performed for a horizontal as well as vertical plate. Two control parameters, t_w and Δt, were kept constant at the following values $t_w \simeq 20$, 30, 40, 50°C and $\Delta t \simeq 10$°C.

Most of the experiments were performed within the range of relatively high moisture content of the air. The measurements gave the evidence that the vertical plate was entirely wetted above the critical moisture content. This situation was not achieved with the horizontal plate the lower surface of which was only partially wetted.

After the heat transfer experiment had been completed, the measurements of the droplet diameter in mist air were carried out with the same apparatus [13]. The experiments revealed that within the temperature range of 10–40°C the droplet diameter range was 10–60 μm.

The experimentally determined enhancement factor E_t is shown in Figs. 3 and 4 together with the predicted curves obtained from equations (29) and (37). The heat transfer coefficient h_d was calculated from the correlation

* The experiments were carried out by Kowalski.

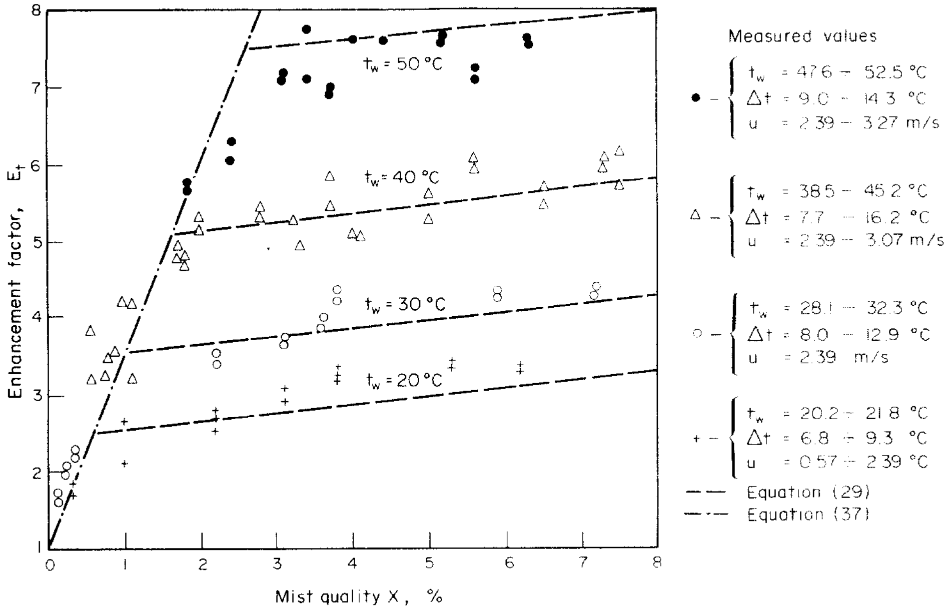

FIG. 3. Predicted and experimental values of the enhancement factor E_t vs mist quality for the vertical plate.

$$Nu_z = 0.332 Re_z^{1/2} Pr^{1/3}. \tag{42}$$

The velocity profile within the boundary layer was assumed to be

$$\frac{u}{u_\infty} = 2\left(\frac{Y}{\delta_g}\right) - 2\left(\frac{Y}{\delta_g}\right)^3 + \left(\frac{Y}{\delta_g}\right)^4 \tag{43}$$

and, in consequence, the boundary layer thickness is

$$\delta_g = \frac{5.83z}{\sqrt{Re_z}}. \tag{44}$$

Perpendicular velocity v_{gf} was calculated from the relation

$$\dot{m}_v'' = \rho_{vw} v_{gf}. \tag{45}$$

The calculations showed that factors A and C in equations (29) and (37) only depend on two para-

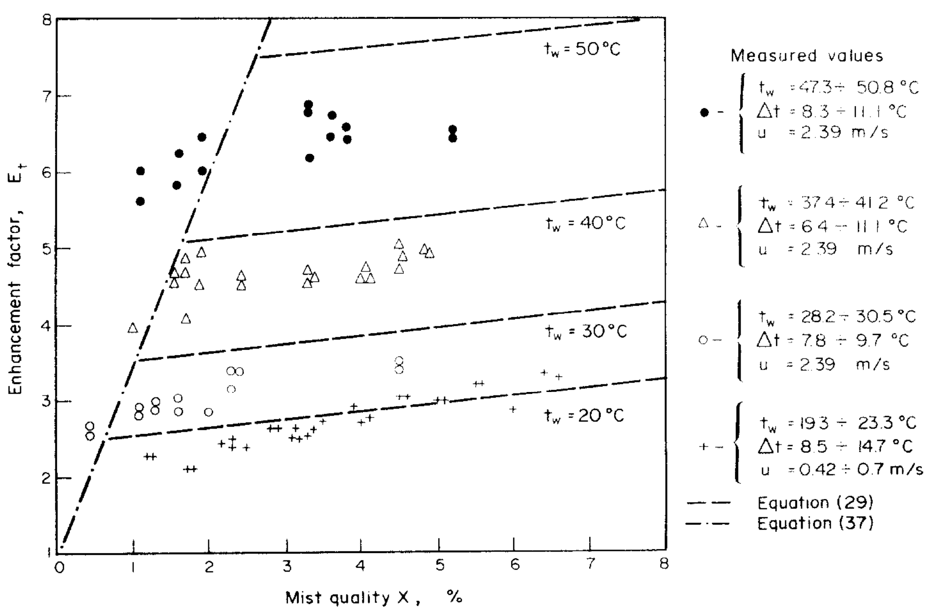

FIG. 4. Predicted and experimental values of the enhancement factor E_t vs mist quality for the horizontal plate.

meters, t_w and Δt. Within the considered range of these parameters, the factors A and C are

$$A \cong 10, \quad C \cong 250.$$

The results presented in Figs. 3 and 4 show good agreement between the predicted and experimental values of the factor E_t, particularly for a vertical plate. The factor E_t of a horizontal plate is about 15% lower than that of a vertical plate due to the different mechanism of deposition.

CONCLUSIONS

A model of heat transfer during laminar flow of mist air along a plate has been presented in this paper. The model holds within the assumption adopted. It also shows that the assumption of negligible effect of the mass flow of deposition on the velocity profile in the boundary layer appears to be important.

Experimental investigations proved the validity of the model at lower wall and air temperatures. The rate of evaporation increases very rapidly at higher temperatures thus affecting the heat transfer process. Therefore at higher temperature levels one may expect larger differences between the calculated and real values of the heat transfer coefficient.

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CALCUL APPROCHE DU TRANSFERT THERMIQUE LORS DE L'ECOULEMENT D'UN BROUILLARD D'EAU DANS L'AIR, LE LONG D'UNE PLAQUE PLANE

Résumé—On calcule de façon approchée le transfert thermique lors de l'écoulement laminaire d'un brouillard le long d'une plaque plane placée dans un canal. Le flux de chaleur élevé résulte de la superposition de l'évaporation du film à la paroi de la plaque et du transfert convectif. Le flux thermique convectif parait plus grand que pour l'écoulement d'air sec à cause de l'effet d'accroissement par les gouttelettes. Des résultats théoriques montrent un bon accord avec les expériences.

NÄHERUNGSWEISE BERECHNUNG DES WÄRMEÜBERGANGS IN DER STRÖMUNG EINES LUFT/WASSER–NEBELS ENTLANG EINER BEHEIZTEN EBENEN PLATTE

Zusammenfassung—Für die laminare Strömung nebelhaltiger Luft entlang einer ebenen Platte in einem Kanal wird der Wärmeübergang näherungsweise berechnet. Es zeigt sich, daß der große Wärmestrom auf die Überlagerung von Filmverdampfung an der Plattenoberfläche und konvektivem Wärmeübergang zurückzuführen ist. Durch die verstärkende Wirkung der Tröpfchen ergibt sich ein größerer Wärmestrom als bei einer Strömung mit trockener Luft. Theoretische und experimentelle Ergebnisse zeigen gute Übereinstimung.

ПРИБЛИЖЕННЫЙ РАСЧЕТ ТЕПЛОПЕРЕНОСА ПРИ ТЕЧЕНИИ ВОДНО-ВОЗДУШНОЙ СМЕСИ ВДОЛЬ НАГРЕВАЕМОЙ ПЛОСКОЙ ПЛАСТИНЫ

Аннотация — Выполнен приближенный расчет теплопереноса при ламинарном течении водно-воздушной смеси вдоль плоской пластины, помещенной в канал. Показано, что высокие значения теплового потока обусловлены процессами пленочного испарения на поверхности пластины и конвективным теплопереносом. Из-за интенсифицирующего воздействия капель величина конвективного теплового потока оказывается выше, чем при течении сухого воздуха. Показано, что теоретические результаты хорошо согласуются с экспериментальными данными.