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# Evaporation of Water by Natural Convection in Partially Wetted Heated Vertical Plates: Effect of the Number of the Wetted Zone

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**Abstract:** This research consists of a numerical investigation of coupled heat and mass transfers by natural convection during water evaporation in a vertical channel. The two channel walls were symmetrically heated by a uniform flux density. One wall is partially wetted by an extremely thin water film and the other is dry. The partially humid plate is divided into 2N with equal lengths being alternatively wet and dry zones. The results are reported in terms of local Sherwood number, the inlet velocity and evaporative rate for different wet zone position and for different wet number zones. However, the mass transfer is extremely influenced by the number of the wetted zones and their positions. The evaporative rate is more intense when the wetted zone is situated at the channel exit. Finally, it is observed that the evaporation is intensified by increasing the number of wetted zones.

**Key words:** Natural convection, wetted zone, water evaporation, inlet velocity, water, flux density

## INTRODUCTION

The free convection duct flows with coupled heat and mass transfer in a flowing gas mixture can be significantly affected by the combined buoyancy forces due to the existence of temperature and concentration variations. The understanding of the modification of flow structure in a duct is important in various thermal systems, such as the desalination, solar energy collectors, design of heat exchangers, geothermal energy systems, cooling of the nuclear reactor. Nawayseh *et al.* (1999) studied the performance of desalination systems and insisted on the accurate evaluation of the heat and mass transfer coefficients before the optimization study. The effects of combined buoyancy forces of heat and mass diffusion on laminar free convection heat transfer in vertical and horizontal rectangular ducts were studied extensively (Aung and Worku, 1986a; Dalbert *et al.*, 1981; Prakash and Liu, 1985). The effects of mass diffusion on natural thermal convection flow have been widely investigated for vertical, horizontal and recently inclized flat plates (Gebhart and Pera, 1971; Yan and Lin, 1990; Mammou *et al.*, 1992). Numerically studied the evaporation of water vapor along an inclined heated plate. The influences of the inclined angle, the wall heating flux, the inlet film thickness and the free stream velocity on the momentum, heat and mass transfer in the system are clarified. Mammou *et al.* (1992) presented a numerical study of the laminar heat and mass transfer from an inclined flat plate with a dry zone inserted between two wet zones. They concluded that the inclination angle has a small influence on the local Nusselt and Sherwood numbers. Tsay *et al.* (1990) and Chang *et al.* (1986) have treated the evaporation of liquids by free convection driven by thermal and mass buoyancy forces into air. Yan and Lin (1990) presented a numerical analysis to investigate the effects of the latent heat transfer, in association with the evaporation into air of a finite liquid film on the channel wall, on the free convective heat and mass transfer.

Debbissi *et al.* (2001, 2003) analyzed the evaporation of water by free into humid air and superheated steam. In their research, particular attention is paid to study the effect of the ambient conditions on evaporation rate of water and the inversion temperature of the phenomenon in the condition of free and mixed convection. The effect of ambient conditions, channel width and walls radiation are analysed in this study. Recently, Orfi *et al.* (2004) have studied the evaporation of water by natural convection, of a thin liquid film down on the internal face of one plate of a vertical heated channel.

To our knowledge, the heat and mass transfer by free convection along a partially wetted plate, which is composed, respectively by an alternation of humid and

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dry zones is not studied. The main objective of this research is to study the evaporation of water into natural and mixed convection flow of humid air in a partially wetted channel. A particular attention will be addressed to the effect of the number N of wetted zones on the evaporation rate.

## MATERIALS AND METHODS

This study presents, a numerical analysis of heat and mass transfer during water evaporation by free convection in a finite vertical channel. The studied channel is made up of two parallel plates symmetrically heated by uniform density fluxes. The left plate is made of a 2N equals zones alternately wet and dry. The second plate (y = d) is dry. This geometric configuration can represent a drying system. The role of the dry and heated zone is to generate an upward natural flow in the channel, through the thermal buoyancy forces. The heating imposed flux can be assured by the solar radiation.

At the channel entrances, the moist air flows upwards with the ambient conditions of temperature $T_0$, Pressure $p_0$ and mass concentration $c_0$. The geometry of the problem under consideration (for N = 1) is shown in Fig. 1a, b. The left plate is divided into two regions with equal lengths (H/2) being alternatively wet and dry zones. Two configurations were considered in this study, in the first case (wet exit), the wetted zone is at the channel exit and the part of the plate is dry. In the second case (dry exit) the configuration is reversed.

In order to set the partial differential system equations describing momentum, heat and mass transfers and some simplifying assumptions are taken into consideration. The boundary layer approximations are generally used. The moist air in the channel is considered as an ideal gas with variable thermo-physical properties. The viscous dissipation and the pressure work are negligible. For wet zones, the moist air is assumed to be at thermodynamic equilibrium so that the wall temperature and water concentration can be related through the saturated vapor pressure (Debbissi *et al.*, 2001, 2003). One can note that these humid zones can be modeled by considering an extremely thin liquid film. Thus, transport in the liquid film can be replaced by approximate boundary conditions for gas flow (Debbissi *et al.*, 2001, 2003). Some other classic assumptions are used such as steady state flow, the negligible Dufour and Soret effects and radiative transfer.

From the above assumptions, the bi-dimensional flow of a gas mixture is described in the (x, y) coordinate system by the continuity equation and the balances for momentum, heat and species concentration (Kays and Crawford, 1980; Schlichting, 1979):

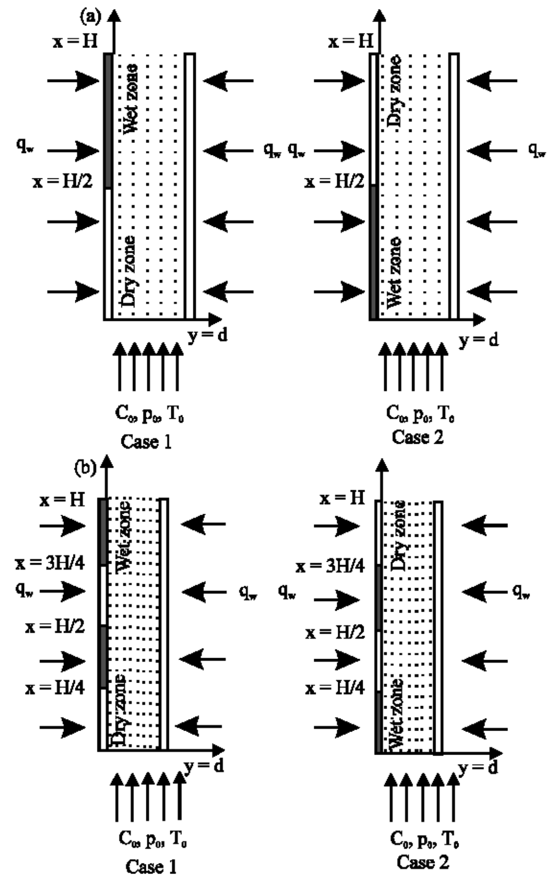


Fig. 1: Schematic diagram of the physical system, a): N = 1 and b): N = 2

$$\frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} = 0 \tag{1}$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho}\frac{dP}{dx} + \beta g(T - T_0) + \beta^* g(C - C_0) + (1/\rho)\frac{\partial}{\partial y}\left(\mu\frac{\partial u}{\partial y}\right) \tag{2}$$

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \frac{1}{\rho C_p}\left[\frac{\partial}{\partial y}\left(\lambda\frac{\partial T}{\partial y}\right) + \rho D(C_{pv} - C_{pa})\frac{\partial T}{\partial y}\frac{\partial C}{\partial y}\right] \tag{3}$$

$$u\frac{\partial C}{\partial x} + v\frac{\partial C}{\partial y} = \frac{1}{\rho}\frac{\partial}{\partial y}\left(\rho\frac{\partial C}{\partial y}\right) \tag{4}$$

where, $(\beta g(T\text{-}T_0) + \beta^* g(C\text{-}C_0))$ represents the momentum transfer caused by the combined buoyancy forces. The thermo-physical properties of gas mixture are considered as variable with temperature and composition.

In this study, the overall mass balance described by the following equation should be satisfied at every axial location:

$$\int_0^d \rho u(x,y)dy = d\rho_0 u_0 + \int_0^x \rho v(x,0)dx \qquad (5)$$

The boundary conditions for the problem are as follows:

$$\begin{gathered} *\text{At } x = 0: \\ -T = T_0,\ C = C_0,\ u = u_0,\ p = 1/2\rho_0 U_0^2 \\ *\text{At } x = H:\ P = 0 \\ *\text{At } y = 0 \\ -u = 0 \end{gathered} \qquad (6)$$

The transverse gas velocity is deduced by assuming that the air-water interface is semipermeable:

$$v(x,0) = \varepsilon\left(\frac{-D}{1-C(x,0)}\left.\frac{\partial C}{\partial y}\right|_{y=0}\right) \qquad (7a)$$

The value of ε is zero for the case of dry zone and unity for the case of wetted zone.

The energy balance at the interface (y = 0) is evaluated by:

$$-\lambda\frac{\partial T}{\partial y} - \varepsilon\left(\frac{\rho L_v D}{1-C(x,0)}\left.\frac{\partial C}{\partial y}\right|_{y=0}\right) = q_w \qquad (7b)$$

It is clear that the imposed heat flux $q_w$ is the sum of a sensible ($q_s$) and a latent ($q_l$) component.

According to Dalton's law and by assuming the interface to be at thermodynamic equilibrium and the air vapor mixture is an ideal gas mixture, the concentration of vapor can be evaluated by:

$$C(x,0) = \frac{M_v/M_a}{p/p_{vs} + M_v/M_a - 1} \qquad (7c)$$

$p_{vs}$ is the equilibrium pressure of vapor given by the Eq. 7d (Vachon, 1989):

$$\log_{10} p_{vs} = 28{,}59051 - 8.2\log T + 2{,}4804.10^{-3}\,T - 3142.32/T$$

$$\begin{gathered} *\ y = d \\ -u = 0,\ v = 0,\ \lambda\left.\frac{\partial T}{\partial y}\right|_{y=d} = q_w \end{gathered} \qquad (7d)$$

The impermeability of the dry plate (y = d) to the water vapor can be described by:

$$\frac{\partial C}{\partial y} = 0 \qquad (7e)$$

In order to describe the mass and energy transfers between the channel walls and moist air, the following dimensionless coefficients are used (Shah and London, 1978):

The local Nusselt number is defined as:

$$Nu_x = \frac{h_x 2d}{\lambda} = -\frac{2d[(\partial T/\partial y)_{y=0}]_x}{T(x,0) - T_m} \qquad (8a)$$

where, $h_x$ is the local heat transfer coefficient. $T_m$ is the fluid bulk temperature at a cross section:

$$T_m = \int_0^d \rho u.T.dy \Big/ \int_0^d \rho u.dy \qquad (8b)$$

The mean Nusselt number is:

$$Nu_m = \frac{1}{x}\int_0^x Nu_x dx \qquad (8c)$$

The local Sherwood number is defined as:

$$Sh_x = -\frac{2d\left[(\partial C/\partial y)_{y=0}\right]_x}{C(x,0) - C_m} \qquad (9a)$$

$C_m$ is the fluid bulk concentration at a cross section:

$$C_m = \int_0^d \rho u.C.dy \Big/ \int_0^d \rho u.dy \qquad (9b)$$

The mean Sherwood number is:

$$Sh_m = \frac{1}{x}\int_0^x Sh_x dx \qquad (9c)$$

The local evaporated mass flux is given by:

$$\dot{m} = \frac{1}{H}\int_0^x \rho v(x,0)dx \qquad (10)$$

**Solution method:** The system of Eq. 1-5 is solved numerically using a finite difference method. The flow area is divided into a regular mesh placed in axial and transverse direction. A fully implicit marching scheme, where the axial convection terms were approximated by the upstream difference and the transverse convection and diffusion terms by the central difference is employed to transform the governing equation into finite difference equations. The resolution of the obtained algebraic equations was marched in a downstream direction since flow under consideration is a boundary-layer type. The discrete equations are resolved line by line from the inlet to the outlet of the channel.

For a given thermal and mass boundary conditions, the resolution procedure is described as follows:

- Guess the inlet velocity $u_0$
- For the given axial location i, guess the wetted wall temperature T* and solve the finite difference form of species equation
- Solve the finite difference form of energy equation and compare the new value T of wetted temperature to T* by testing if

$$\left|\frac{T(i,1)-T^*(i,1)}{T(i,1)}\right| < 10^{-6}$$

If this criteria is not satisfied, return to Eq. 2 and modify the wetted wall temperature by using the bisection method

- Guess a pressure P* at the i axial location and solve the momentum and continuity finite difference equations. Then, verify the satisfaction of the overall conservation of mass expressed by the following criteria

$$\left|\int_0^d \rho u(x,y)dy - (d.\rho_0 u_0 + \int_0^x \rho v(x,0)dx)\right| / (d\rho_0 u_0) < 10^{-6}$$

- If this condition is not satisfied, return to step 4 and modify the pressure value P* and repeat the steps (2-5)
- Test if the exit dynamic pressure is zero, else return to step 1 and modify the inlet velocity by using bisection method

To ensure that results were grid independent, the solution was obtained for different grid sizes for typical case program test. Table 1 shows that the differences in the evaporative rate obtained using 71-71 and 101-101 grids are always <1%. A (71-71) grid is retained for the present computations.

Table 1: Comparison of the total evaporative rate of water at the exit (case when the left plate (y = 0) is entirely wetted) for various grid arrangement ($T_0$ = 298.15 K; $q_w$ = 500 W $m^{-2}$; $p_0$ = 1; d/H = 0.015; $C_0$ = 0.005)

| I×J grid point | m (H) | Shx (H) |
|---|---|---|
| 71×51 | $4.0635\times10^{-4}$ | 5.1516 |
| 71×71 | $4.0656\times10^{-4}$ | 5.1518 |
| 101×71 | $4.0571\times10^{-4}$ | 5.1471 |
| 101×101 | $4.0585\times10^{-4}$ | 5.1491 |

## RESULTS AND DISCUSSION

To validate the numerical scheme adopted in the present study, different limiting cases for laminar mixed and free convection have been considered. The results for the case of mixed convective heat and mass transfers inside a channel were obtained. The plates of the channel are maintained isothermal.

The first plate (y = 0) is wetted by an extremely thin water film and the second one is dry and kept at the ambient temperature $T_0$. The procedure has been tested by comparing the present results for the mass transfer coefficient to those of Shah and London (1978).

Figure 2a shows a good agreement between the result and those obtained by Shah and London (1978). Furthermore, the numerical code has been tested successfully by comparing the present results for mean Nusselt number ($Nu_m$) (Fig. 2b) at the isothermal and dry wall to the analytical solution obtained by Shah and London (1978).

Finally, results of evaporation by natural, forced and mixed convection in a heated channel were validated

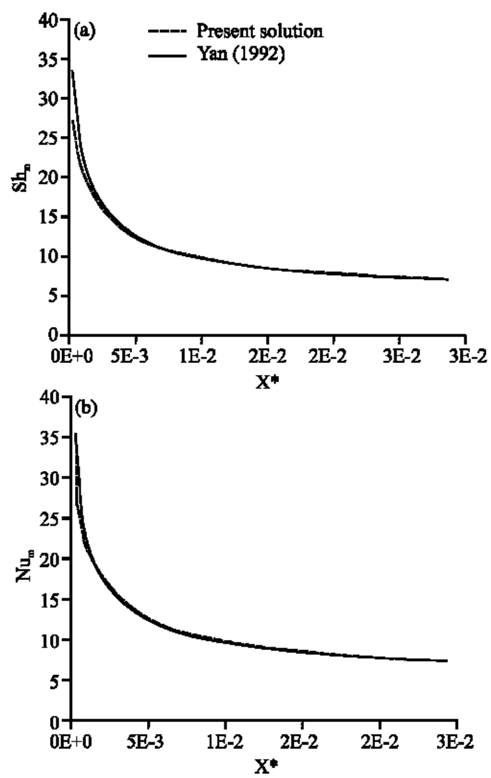


Fig. 2: Axial evolution of a): Mass transfer abd b): Heat transfer coefficient

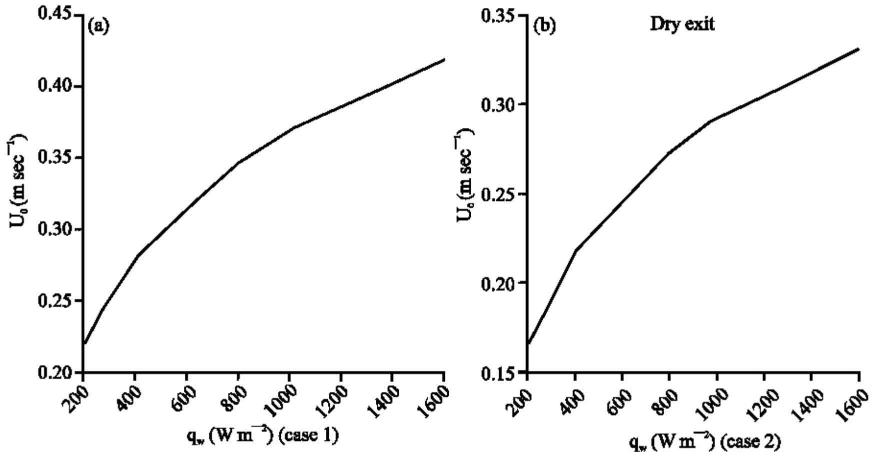


Fig. 3: Effect of the heat flux $q_w$ on the inlet velocity for a): Wet exit, N = 1 ($T_0$ = 298.15 and b): Dry exit K; $p_0$ = 1; d/H = 0.015; $C_0$ = 0.005)

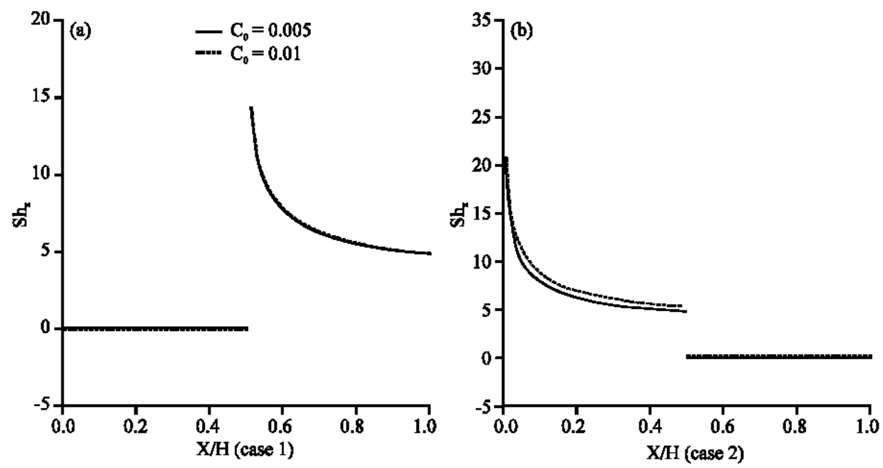


Fig. 4: Effect of the inlet concentration on the local Sherwood number for a): Wet exit and b): Dry exit N = 1 ($T_0$ = 298.15K; $q_w$ = 250 W m$^{-2}$; $p_0$ = 1; d/H = 0.015)

informer paper Dalbert *et al*. (1981, 2001), Prakash and Liu (1985). Through, these program tests, the present numerical code is considered to be suitable for the present investigation.

All the results of this study have been cared out for a channel placed into a upward flow of humid air with the ambient conditions: $C_0$ = 0.005; $T_0$ = 298.15K; the imposed walls heat flux density $q_w$ = 250 W m$^{-2}$, the geometrical ratio is d/H = 0.015. The left plate (x = 0) is divided along the channel into 2N equally wet and dry zones.

The second plate is dry and subjected to a uniform same heat flux density $q_w$. This study includes two configuration, for the first case, the dry zone is located in the first half of the plate (x = 0) and the wetted zone is located at the channel exit. For the second (case 2), situation is inverted.

In this study, attention was paid to the evaporation of water by natural convection driven by the simultaneous presence of combined buoyancy effects of heat and mass diffusion. It is clear in Fig. 3, when the imposed heat flux density $q_w$ increases, the inlet velocity increases.

The effect of the ambiance conditions and of the imposed heat flux on the mass transfer is shown in Fig. 4-6.

Figure 4 shows that the Sherwood number $Sh_x$ is practically not influenced by the inlet concentration and especially, in the case 1.

Figure 5 shows the effect of the inlet temperature on the Sherwood number $Sh_x$. It is clearly seen, when $T_0$ increases, the Sherwood number $Sh_x$ along the wet zone decreases.

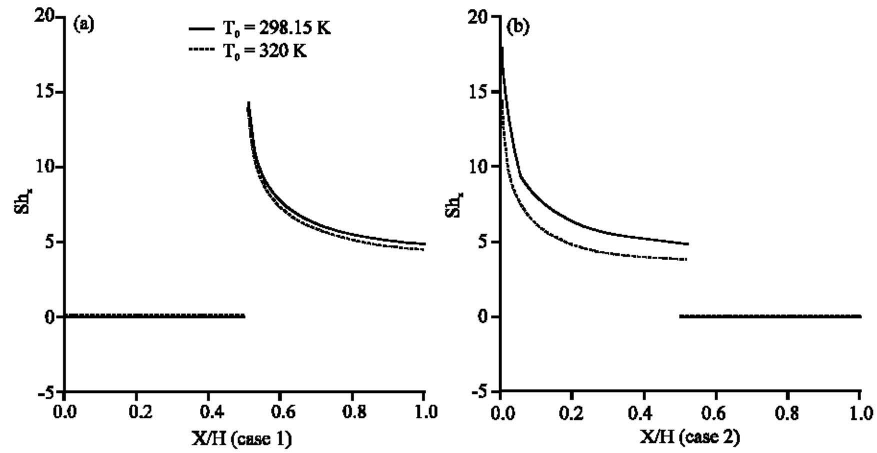


Fig. 5: Effect of the inlet temperature on the local Sherwood number for a): Wet exit and b): Dry exitN = 1 ($C_0$ = 0.005; $q_w$ = 250 W $m^{-2}$; $p_0$ = 1; d/H = 0.015)

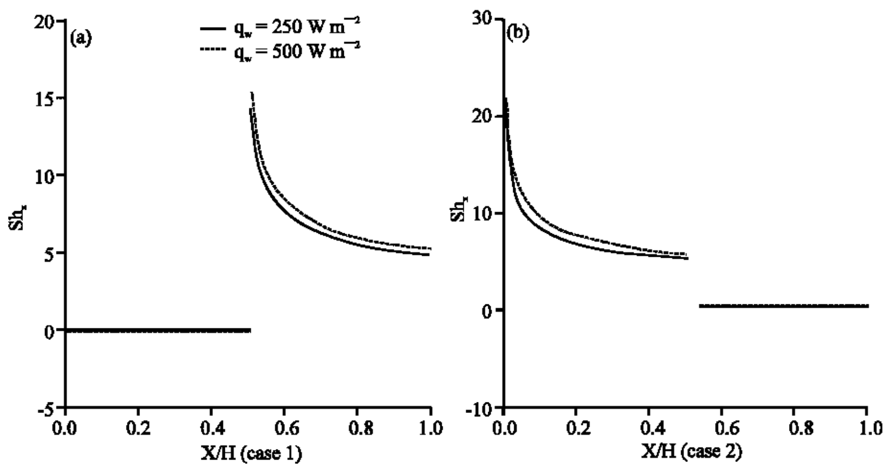


Fig. 6: Effect of the heat flux on the local sherwood number for a): Wet exit and b): Dry exit N = 1 ($C_0$ = 0.005; $T_0$ = 298.15K; $p_0$ = 1; d/H = 0.015)

As shown in Fig. 6 and along the wetted zone, the Sherwood number $Sh_x$ is influenced by the heat flux density applied to the wall. Thus, it is noted that this number increases when the heat flux density applied to the wall increases. This can be allotted to the fact that one increases the imposed heat flux, the evaporation is increasing. Figure 4-6 show that along the dry zone, for which there is practically no mass transfer, the Sherwood number $Sh_x$ is essentially zero, because there is no evaporation process in this zone.

As shown in Fig. 7, a increasing in the heat flux density increases the interfacial mass flux. It is clearly observed in Fig. 8, that the interfacial mass rate increases with an decrease in ambient pressure.

Figure 9 presents the interfacial mass flux at the channel exit $\dot{m}(x=H)$ as the function of inlet temperature $T_0$ for various ambient humidity. It is clearly observed that the interfacial mass rate increases with an increase in inlet temperature. As shown in Fig. 10 that the interfacial mass flux at the channel exit $\dot{m}(x=H)$ increases with an decrease in inlet concentration.

For different configurations, the local evaporative rate for the case of two wetted zones position is plotted also in Fig. 11.

Figure 11 shows that the evaporative rate in the first case (wet exit) is more important than that of the second case (dry exit). This result can be justified by the higher temperature of the fluid arriving to the entry of the second humid zone for the first case.

To provide further perspective about the role of the number of humid zones, for the same wetted length, Fig. 12 displays the result of local evaporative rate for different numbers of the wetted zones (N = 1-4). In all these studied cases, the channel inlet is occupied by a

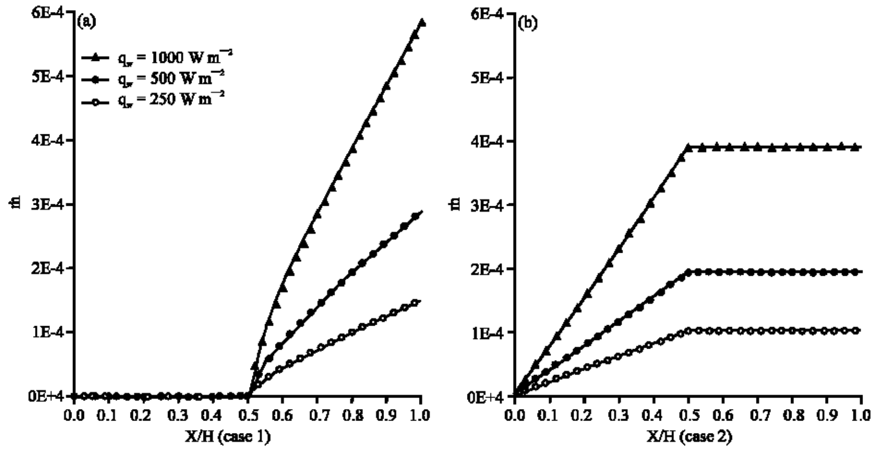


Fig. 7: Effect of the heat flux on the local evaporating rate for a): Wet exit and b): Dry exit N = 1 ($C_0$ = 0.005; $T_0$ = 298.15 K; $p_0$ = 1; d/H = 0.015)

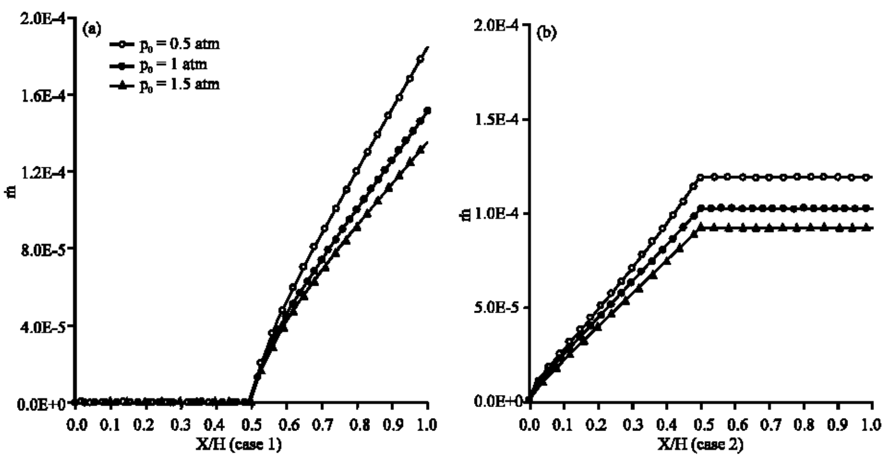


Fig. 8: Effect of the ambient pressure on the local evaporating rate for a): Wet exit and b): Dry exit, N = 1 ($C_0$ = 0.005, $T_0$ = 298. 15K, $q_w$ = 250 W m$^{-2}$, d/H = 0.015)

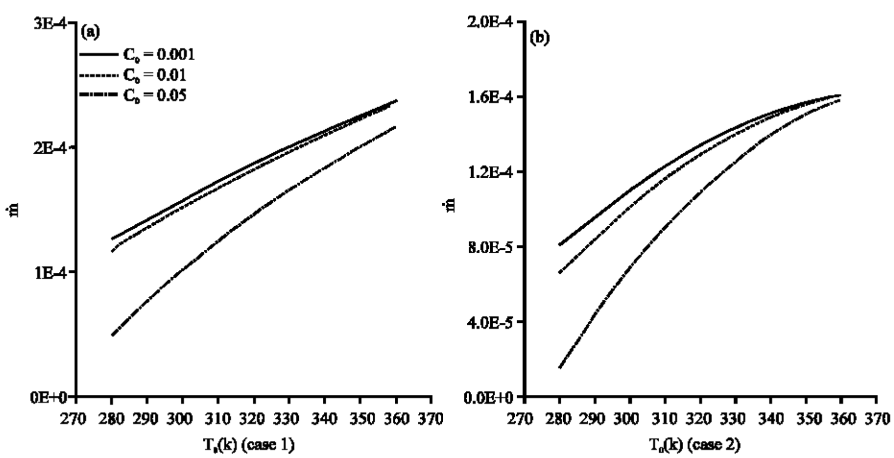


Fig. 9: Effect of the ambient temperature on the local evaporating rate for a): Wet exit and b): Dry exit, N = 1 ($P_0$ = 1atm, $q_w$ = 250 W m$^{-2}$, d/H = 0.015)

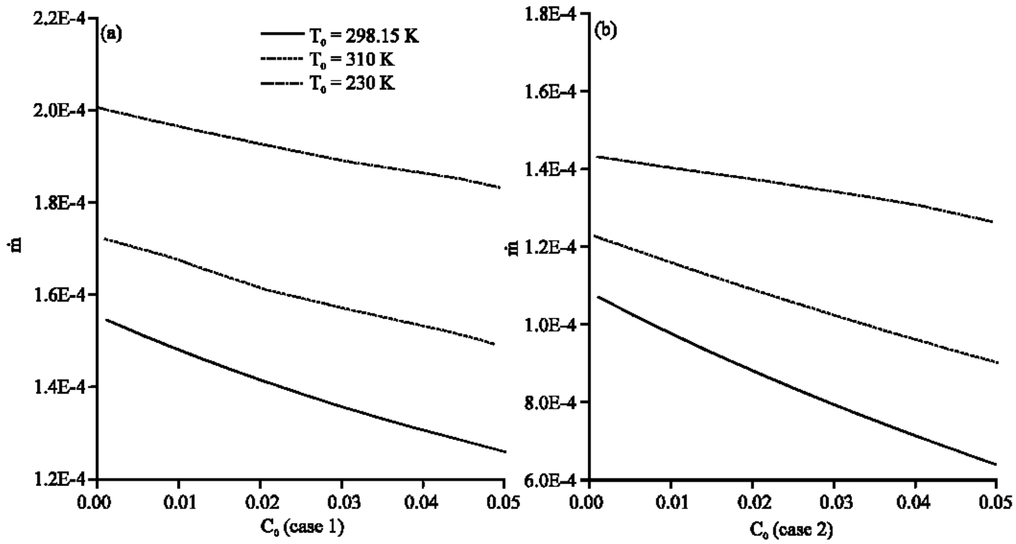


Fig. 10: Effect of the ambient concentration on the local evaporating rate for a): Wet exit and b): Dry exit, N = 1 ($P_0$ = 1 atm, $q_w$ = 250 W m$^{-2}$, d/H = 0.015)

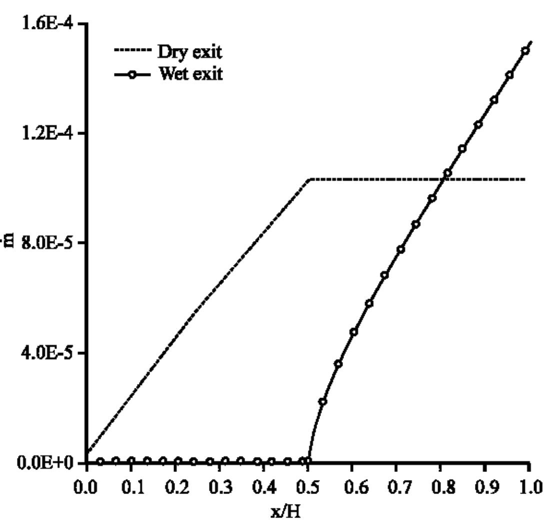


Fig. 11: Effect of the wetted zone position on the local evaporative rate for N = 1 ($C_0$ = 0.005; $T_0$ = 298.15K; $q_w$ = 250 W m$^{-2}$; d/H = 0.015)

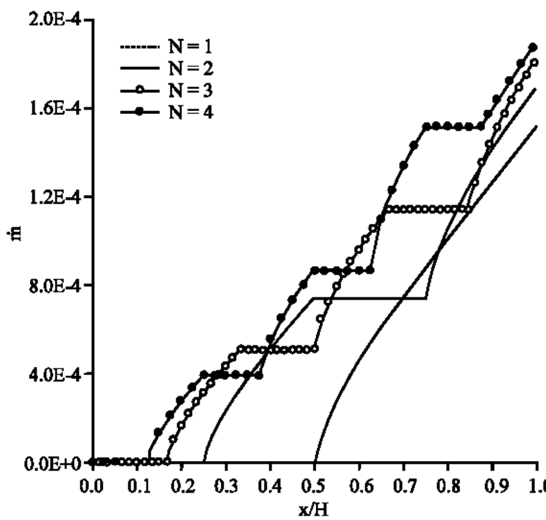


Fig. 12: Effect of the number N of the humid zones on the local evaporative rate ($C_0$ = 0.005; $T_0$ = 298.15 K; $q_w$ = 250 W m$^{-2}$; d/H = 0.015)

dry zone. This choice is approved by the previous results. One notices that at the channel exit, the evaporation is intensified by increasing the number N of the humid zones. This result can be justified by the fact that the fluid temperature reaching every humid zone increases when, one increases the number of these zones. Figure 12 also shows that by passing from N = 1-4, the relative gap of the evaporative rate value can exceed 46%.

## CONCLUSION

The evaporation by free convection in a partially wetted channel has been numerically studied for an air-water system. The studied channel is made up of two parallel plates. The partially wetted plate is heated by uniform density flux, while the second is dry and isothermal. The effect of the humid zones position and the number of wetted zones N on the characteristics of the heat and mass transfers has been analyzed.

The major results are briefly summarized as follows:

- The evaporation is more intense when, the wetted zone is situated at the channel exit (case 1)
- The effect of the number of wetted zones N and their positions on the local evaporative rate across the interface is studied for the same humid surface; it is shown that the evaporative rate depends largely on the Number (N) of humid zones. Results show that the evaporation is intensified by increasing the Number (N) of the humid zones. The rising of the humid zone Number (N) increases the total evaporative rate, which can exceed 46% in relative value

## REFERENCES


Aung, W. and G. Worku, 1986. Theory of fully developed combined convection including flow reversal. AJHT, 108: 485-488.

Chang, C.J., T.F. Lin and W.M. Yan, 1986. Natural convection flows in a vertical open tube resulting from combined buoyancy effects of thermal and mass diffusion. IJHMT, 29 (10): 1543-1552.

Dalbert, A.M., F. Penot and J.L. Peube, 1981. Convection naturelle laminaire dans un canal vertical chauffe à flux constant. IJHMT, 24 (9): 1463-1473.

Debbissi, C., J. Orfi and S. Ben Nassrallah, 2001. Evaporation of water by free convection in a vertical channel including effects of wall radiative properties. IJHMT, 44: 811-826.

Debbissi, C., J. Orfi and S. Ben Nassrallah, 2003. Evaporation of water by free and mixed convection into humid air and superheated steam. IJHMT, 46: 4703-4715.

Gebhart, B. and L. Pera, 1971. The nature of vertical natural convection flows resulting from combined buoyancy effects of thermal and mass diffusion. IJHMT, 14: 2025-2050.

Kays, M.W. and M.E. Crawford, 1980. Convective Heat and Mass Transfer. 2nd Edn. McGraw-Hill Book Company, New York.

Mammou, M., M. Daguenet and G. Le Palec, 1992. Numerical study of heat and mass transfer from an inclined flat plate with wet and dry zones. IJHMT, 35 (9): 2277-2287.

Nawayseh, M.M. Farid, A.Z. Omar and A. Sabrin, 1999. Solar desalination based on humidification process II: Computer Simulation. ECM, 40: 1441-1461.

Orfi, J., C. Debbissi and S. Ben Nasrallah, 2004. Air humidification by free convection in a vertical channel. Desalination, 168: 161-168.

Prakash, C. and Y.D. Liu, 1985. Buoyancy induced flow in a vertical internally finned circular duct. AJHT, 107: 118-123.

Schlichting, H., 1979. Boundry Layer Theory. McGraw-Hill Book Company, New York.

Shah, R.K. and A.L. London, 1978. Laminar Flow Forced Convection in Duct. New York: Academic Press (book).

Tsay, Y.L., T.F. Lin and W.M. Yan, 1990. Cooling of falling liquid film through interfacial heat and mass transfer. IJMF, 16 (5): 853-865.

Vachon, M., 1987. Etude de l' evaporation en convection naturelle. These de doctorat, Universite de Poitiers.

Yan, W.M. and T.F. Lin, 1990. Combined heat and mass transfer in natural convection between vertical parallel plates with film evaporation. IJHMT, 33 (3): 529-541.