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Numerical Analysis of the Evaporation of Water by Forced Convection into Humid Air in Partially Wetted Vertical Plates

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Abstract: The present study focuses on a numerical investigation of steady conjugated heat and mass transfers by forced convection in an externally heated or insulated channel. One wall is partially wetted by an extremely thin water film, while the other is dry and impermeable. The partially humid plate is divided into 2.n equal regions, which are alternately humid and dry zones. The effect of the number of wetted zones and their positions on the flow, on the heat and mass transfers is analysed. The results are reported in terms of axial distribution of wall temperature, relative heat fluxes and evaporative rate for different wetted zone positions. It is noticed that the change of the wetted zone position has no significant effect on the moist air flow. However, the heat and mass transfers are extremely influenced by the presence of the wetted zones and their positions. As the condition of an insulated channel, the evaporative rate is more intense when the wetted zone is situated at the channel entrance. In case of the condition of a heated wall channel, the situation is generally inversed. It is also shown that there exists a critical value for the density heat flux from which the behaviour of the evaporative rate is reversed. Finally, it is noticed that the evaporation is intensified by increasing the number of humid zones.

Key words: Evaporation, forced convection, heat and mass transfer, thin film, vertical plates

INTRODUCTION

The combined heat and mass transfers over vertical ducts have a wide range of applications in the field of science and technology such as film cooling of electronic equipments, drying processes, air conditioning and desalting. Owing to these widespread applications, several researches have been investigated in these topics.

A vast amount of study is about the evaporation in the case of uniform wall temperature or uniform wall concentration (Wei-Mon and Lin, 1990; Wei-Mon, 1992; Hammou, 2004; Agunoun *et al*., 1994; Kaoua *et al*., 1996). Gebhart and Pera (1971) obtained similar solutions for air and water over a wide range of Schmidt number (Sc). Nelson and Wood (1989) have introduced a numerical investigation of combined heat and mass transfers in a developing natural convection flow between vertical plate with uniform temperature and concentration.

The case of uniform wall heat or mass fluxes has also been studied. A numerical study of finite liquid film evaporation, on laminar convection in a vertical parallel plate channel with uniform heat flux, is studied in (Wei-Mon, 1995; Baumann and Thiele, 1986; Shah and London, 1978; Wei-Mon and Soong, 1993; Debbissi *et al*., 2001). Wei-Mon (1995) investigated numerically the effects of film evaporation in turbulent mixed convection heat and mass transfer in a vertical channel. Baumann and Thiele (1986) studied the heat and mass transfers in 2-component film evaporation in a vertical tube. Wei-Mon and Soong (1995) studied the convective heat and mass transfers along an inclined heated plate with film evaporation. Debbissi *et al*. (2003) considered the evaporation, by free or mixed convection, of a thin liquid film and presented the effects of wall radiative properties. A numerical study of liquid film evaporation on laminar convection heat and mass transfers in a vertical parallel plate channel with adiabatic wall is reported in Baumann and Thiele (1990), Chow and Chung (1983), Wei-Mon and Soong (1995), Splettstober (1975) and Shembharkar and Pai (1986). Salah El-Din (2003) has examined the effect of mass buoyancy forces on the development of laminar mixed convection between vertical parallel plates with uniform wall heat and uniform mass fluxes. The author studied the effect of the buoyancy ratio on heat and mass transfers between plates.

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Most of the previewed studies deal with a uniform temperature, concentration or heat and mass wall fluxes. Less attention is given to discrete or variable heat and mass boundary conditions (Mammou *et al*., 1992; Lee, 1999; Rani, 2003). Mammou *et al.* (1992) presented a numerical study of laminar heat and mass transfers from an inclined flat plate with a dry zone inserted between 2 wet zones. They concluded that the inclination angle has a small influence on the local Nusselt and Sherwood numbers. Lee (1999) presented a numerical analysis to investigate the effects of the heat transfers, in a partially heated vertical parallel plate. In this study, both boundary conditions of uniform wall temperature/uniform wall concentration and uniform heat flux/uniform mass flux are considered. Their analysis showed that the presence of the unheated entry and unheated exit severely affects the heat and mass transfers. They present theoretical correlations for average Nusselt number and Sherwood number. Rani (2003) investigated a numerical study of transient natural convection along a vertical cylinder under combined effects of thermal and mass diffusions with power law in wall temperature and concentration. Schroppel and Thiele (1983) presented a numerical analysis to investigate the impact of wall temperature variation on the film condensation of binary gas-vapour mixtures. Their analysis showed the effect of a linearly varied wall temperature, compared to the isothermal case on the heat transfer to the wall and interface velocities. To our knowledge, the heat and mass transfers along a partially wetted plate, which is composed, respectively of an alternation of humid and dry zones, is not studied. The main objective of this research is to perform a numerical study to investigate the evaporation of a thin liquid film by forced convection in partially wetted vertical plates. Its main concern is to study the influence of the numbers and the positions of wetted zone on the heat and mass transfer coefficients and on the evaporation rate.

MATERIALS AND MEDTHODS

Analysis: This study presents a numerical analysis of heat and mass transfers during water evaporation by forced convection in a finite vertical channel. The studied channel is made up of two parallel plates. The first plate (y = 0) is externally insulated while the second one (y = d) is dry and isothermal. The imposed temperature is maintained at T_w = 100°C for all computations. The left plate is made of a succession of 2.n zones alternately wet and dry. At the channel entrance, the moist air flows upwards with the ambient conditions of temperature T_0, pressure p_0, mass concentration c_0 and velocity u_0. The geometry of the problem under consideration (for n = 1) is

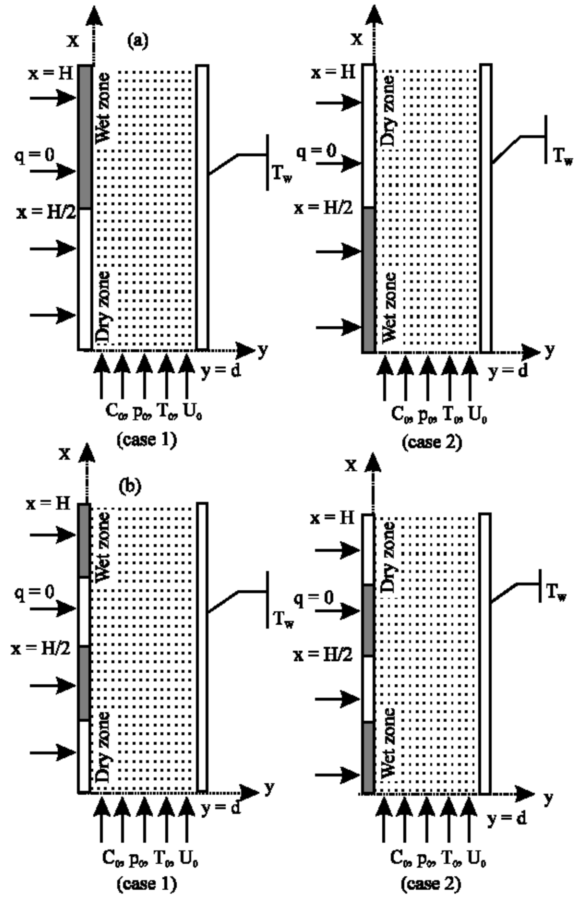

Fig. 1: Schematic diagram of the physical system, a (n = 1): case for 1 wetted zone, b (n = 2): case for 2 wetted zone

shown in Fig. 1a. For these case (n = 1), the left plate is divided into 2 regions with equal lengths (H/2), which are alternatively wet and dry zones. Two configurations were considered in this study, in the first case (dry exit), the wetted zone is at the channel entrance and the second 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, 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 vapour pressure (Wei-Mon and Lin, 1990). One can note that these humid zones can be modelled 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 the gas mixture is described in the (x, y) coordinate system by the continuity equation and the balances for momentum, heat and species concentration:

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

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{dP}{dx}+(1/\rho)\frac{\partial}{\partial y}(\mu\frac{\partial u}{\partial y}) \tag{2}$$

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

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

Thermo-physical properties of gas mixture are considered as variable with temperature and composition. In this study, the overall mass balance described by the Eq. 5 should be satisfied at every axial location:

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

Boundary conditions:

$$\begin{gathered} \text{* At } x=0 \\ -\ u=u_0,\ T=T_0 \text{ and } C=C_0 \\ \text{* At } y=0\text{: } u=0 \end{gathered} \tag{6}$$

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

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

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:

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

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

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

p_{vs} is the equilibrium pressure of vapour given by the Eq. 7d (Debbissi *et al.*, 2003):

$$\begin{aligned} \log_{10} p_{vs} &= 28{,}59051-8.2\log T+2, \\ &\quad 4804.10^{-3}T-3142.32/T \end{aligned} \tag{7d}$$

$$\begin{gathered} \text{* } y=d \\ -\ u=0,\ v=0 \end{gathered}$$

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

$$\frac{\partial C}{\partial y}=0 \tag{7}$$

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

The peripheral local Nusselt number is defined as:

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

where:

h_x = The local heat transfer coefficient

T_m is the fluid bulk temperature at a cross section:

$$T_m=\int_0^d \rho u.T.dy \Big/ \int_0^d \rho u.dy \tag{8b}$$

The local evaporated mass flux is given by:

$$\dot{m}=\frac{1}{H}\int_0^x \rho v(x,0)dx \tag{9a}$$

The water evaporating rate, commonly used in the previous studies (Schwartze and Bröcker, 2000; Chow and Chung, 1983; Wu *et al.*, 1987; Haji and Chow, 1988) is expressed as:

$$R_{ev}=\frac{10^4\dot{m}}{\sqrt{\rho_0 u_0/H}} \tag{9b}$$

For the dry zone: In this region, the governing equations for flow and heat transfer were obtained by adjusting the above equations (boundary conditions). In this case, the left plate was considered to be impermeable.

Solution method: The presented system Eq. 1-5 are solved numerically using a finite difference method. The flow area is divided into a regular mesh placed in axial and transverse direction and a (71, 71) grid is retained for computations. 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 equations into finite difference equations. The resolution of the obtained algebraic equations was marched in 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. The solution procedure is briefly outlined as following:

1. Give the flow, thermal and mass boundary conditions
2. For the given axial location i, guess the wall temperature T* and solve the finite difference form of species equation
3. Solve the finite difference form of energy equation and compare the new value T to T* by testing if:

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

If this criterion is not satisfied, return to (2) and modify the wall temperature by using the bisection method.

4. 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:

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

5. If this condition is not satisfied, return to step 4 and modify the pressure value P* and repeat steps (2-5).
6. For the dry plate, the species equation and the evaporative rate in the overall conservation of mass, were omitted.

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 = 373.15K, T_w = 373.15K, q_w = 0, p_0 = 1, d/H = 0.015)

I×J grid point	$\dot{m}(H)$: C_0 = 0.005	$\dot{m}(H)$: C_0 = 0.01
71×51	5.152×10^{-4}	5.021×10^{-4}
71×71	5.175×10^{-4}	5.043×10^{-4}
101×71	5.167×10^{-4}	5.035×10^{-4}
101×101	5.058×10^{-4}	5.032×10^{-4}

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 71×51 and 101×101 grids are always <1%.

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 of the case of mixed convective heat and mass transfers inside a channel have been treated. The channel wall is maintained isothermal. The first plate 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 latent Nusselt number. Figure 2a shows a satisfactory conformity. Furthermore, the numerical code has been tested successfully by comparing the present results for mean Nusselt number Nu_m (Fig. 2b) at the isothermal wall to the analytical solution found in Shah and London (1978).

Finally, the results of evaporation by natural convection in a heated channel were validated in a former paper (Debbissi *et al*., 2001, 2003). Through these program tests, the present numerical code is considered to be suitable for the study of the present problem.

This research includes 2 sections. The first one concerns the case of an insulated partially humid plate. In the case of n = 1, the effect of the wetted zone position on the development of velocity, temperature and concentration profiles as well as on the characteristics of heat and mass transfer is investigated. For different of dry zones number (n), the results for evaporative rate of water from the partially wetted plate are also presented in this part. The second section concerns the case of partially wetted plate being submitted to a uniform heat flux (q). All the above cases are based on a vertical channel with length of 1 m and width of 0.015 m. Moreover, the dry wall temperature is always dry and kept at T_w = 100°C.

Case of insulated channel wall: In the first section of this study, the humid plate is externally insulated. The results have been carried out for a channel placed into an upward

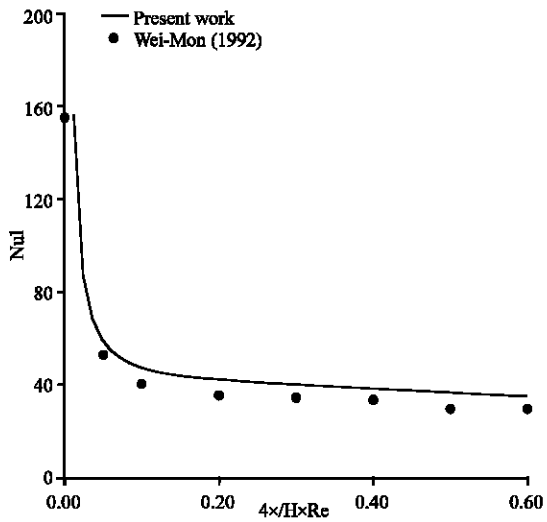

Fig. 2a: Variation of the local Nusselt latent heat number

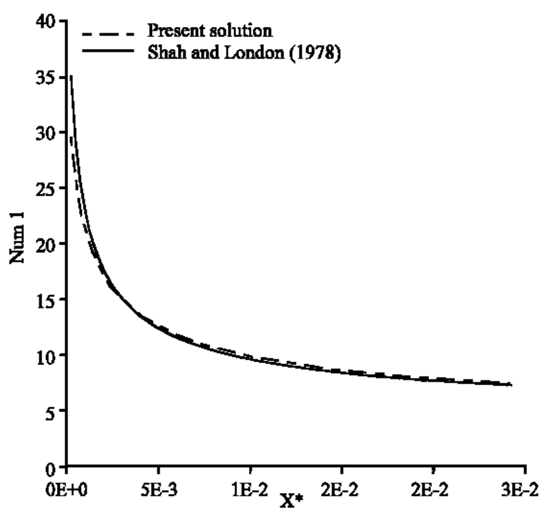

Fig. 2b: Axial evolution of heat coefficient

flow of humid air in the ambient conditions: $u_0 = 1$ m s^{-1}; $C_0 = 0.005$; $T_0 = 100°C$ and the geometrical ratio is d/H = 0.015. The right plate (y = 0) is divided along the channel into 2n equally wet and dry zones. The second plate is dry and isothermal. The imposed temperature is maintained at $T_w = 100°C$. In order to study the effect of the number of wet zones and their position on the flow as well as on the effectiveness of water evaporation along the partially wetted plate, the cases of n = 1, n = 2, n = 3 and n = 4 are examined. The first part of this investigation deals with the case of n =1, it includes 2 configurations. For the first one (case 1), the dry zone is located in the first half of the plate (x = 0) whereas the wet zone is located at the channel exit. For the second one (case 2), the configuration is inverted.

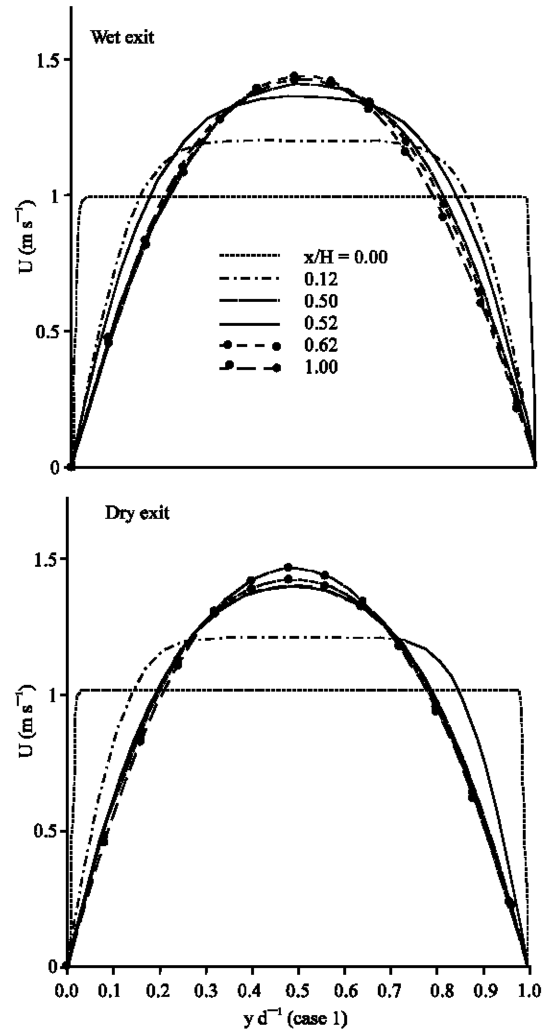

Fig. 3: Effect of the wetted zone position on the x-velocity component profiles for n = 1 (u_0 = 1 m s^{-1}; C_0 = 0.005; T_0 = 373.15 K; T_W = 373.15 K; q = 0; d/H = 0.015)

For several channel sections, Fig. 3-5 present the x-velocity component, the temperature and the concentration profiles. From Fig. 3, one can see that the velocity axial profile develops gradually from a uniform distribution at the inlet to parabolic ones as the flow goes up stream. By comparing the results of the 2 cases, it is obvious that the axial velocity profile keeps increasing at the centreline and is slightly affected by the separation zone between the dry and the wet region. This it can be justified by the difference of the fluid viscosities.

On the other hand, the temperature and the vapour concentration profiles are extremely influenced by the inversion of the wet zone position. As can be seen from Fig. 4 and for the first case, the temperature is constant

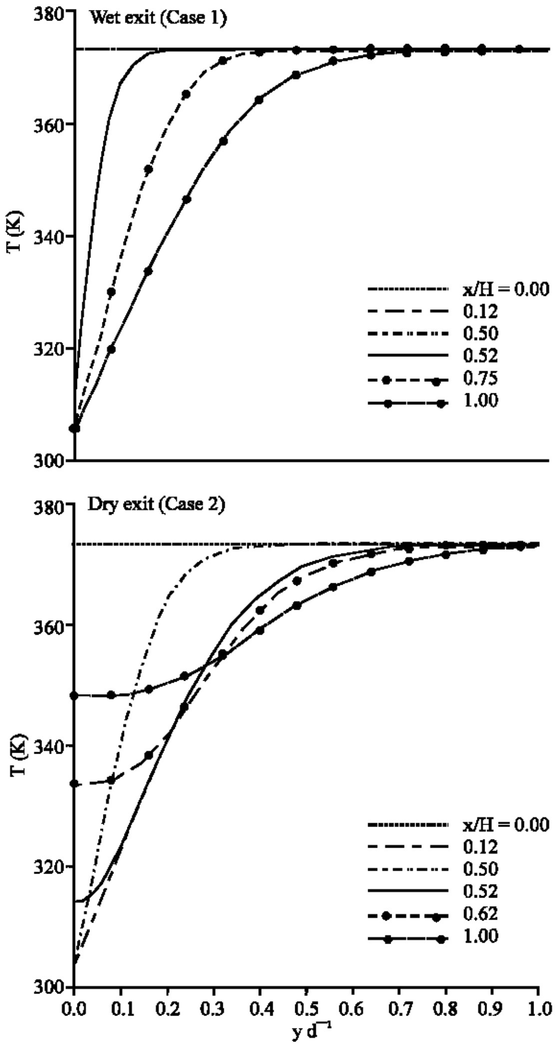

Fig. 4: Effect of the wetted zone position on the temperature profiles for n = 1 (u_0 = 1 m s^{-1}; C_0 = 0.005; T_0 = 373.15 K; T_W = 373.15 K; q = 0; d/H = 0.015)

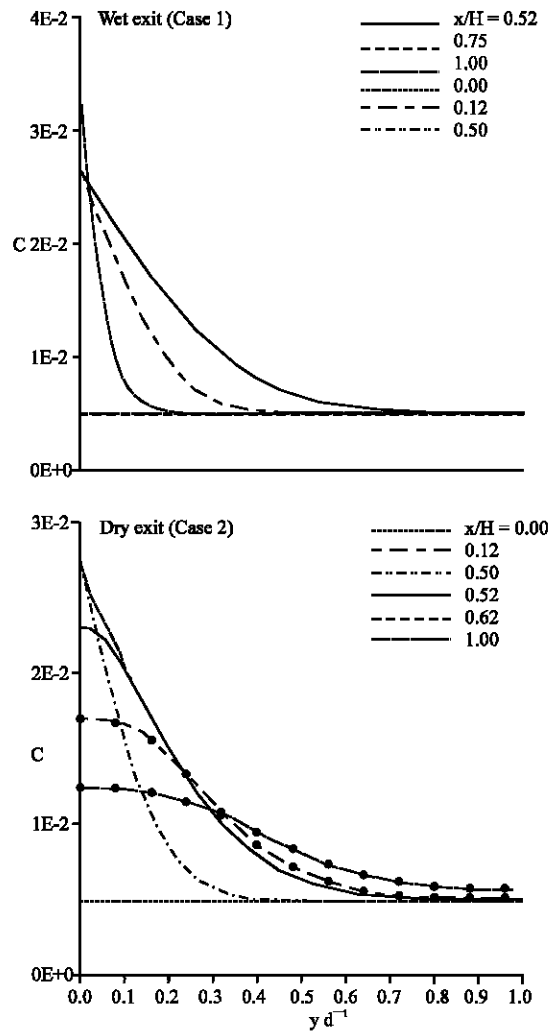

Fig. 5: Effect of the wetted zone position on the vapour concentration profiles for n = 1 (u_0 = 1 m s^{-1}; C_0 = 0.005; T_0 = 373.15K; T_W = 373.15K; q = 0; d/H = 0.015)

along the dry zone but it decreases in the wetted zone. This sudden decrease is justified by the cooling through evaporation. In the second case, the gas flowing near the humid zone is cooled especially at the channel inlet because of the amount of energy needed for water evaporation. Consequently, the temperature profiles decrease at the first half of the channel. Along the dry zone, the Fig. 4 shows that the fluid temperature increases to approach the ambient temperature. Figure 5 shows that the water vapour concentration increases along the wet zone for the first case as well as for the second case.

It is clear in Fig. 6a and for the first case, that the parietal temperature along the dry zone keeps constant since the first plate (y = 0) is externally insulated. However, in the wetted zone (first case), one can notice a rapid decreasing of the interfacial temperature via the cooling by evaporation. For the second case and under the present conditions of adiabatic evaporation, Fig. 6a reveals that the parietal temperature of the humid zone is almost constant along the wetted zone whereas it increases considerably in the dry one.

In line with the above result and according to the relation Eq. 7c and 6b shows that for both cases studied and along the humid zones, the parietal vapour concentration decreases slightly since the parietal temperature in these zones keeps decreasing slightly. But along the dry zone (case 1), the parietal water vapour

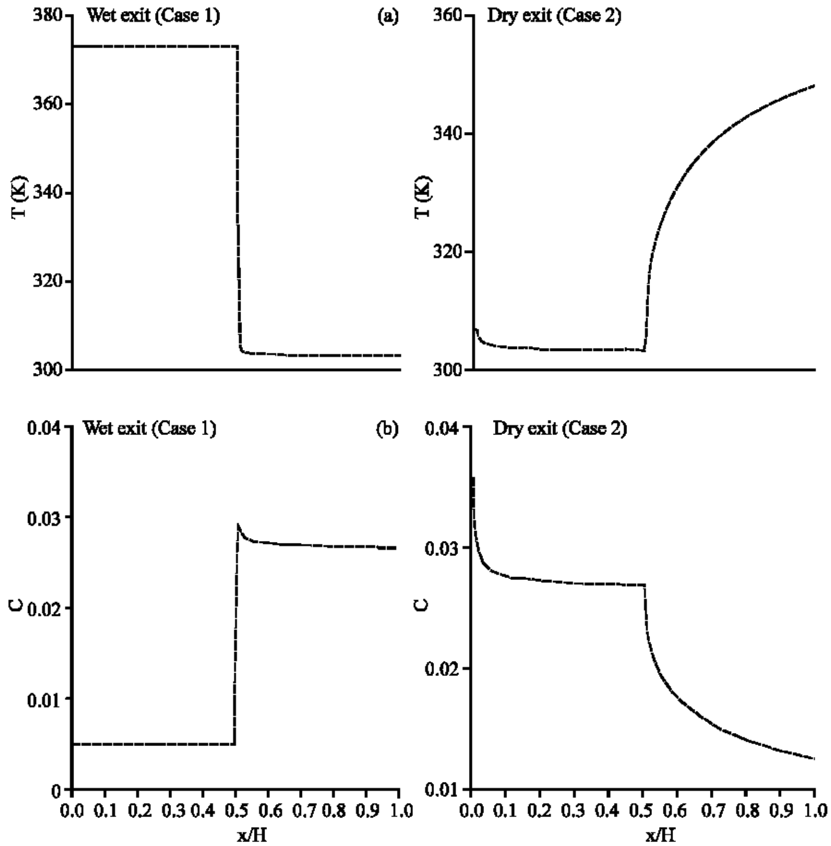

Fig. 6: Effect of the wetted zone position on the interfacial temperature and concentration evolution for n = 1 (u_0 = 1 m s^{-1}; C_0 = 0.005; T_0 = 373.15K; T_W = 373.15K; q = 0; d/H = 0.015)

concentration keeps constant because the water vapour concentration in the channel entrance is uniform. For the second case, the parietal concentration along the dry zone drops because the water vapour concentration at the entrance of this zone (dry zone) is not uniform and there is no evaporation process in the last one.

To investigate the importance of the sensible and the latent heat exchange along the partial wetted plate, Fig. 7 illustrates the heat transfer rates along the interface. Figure 7 shows that for a dry zone, the sensible and the latent heat are null, because the right plate (y = 0) is externally insulated. On the other hand and according to the imposed thermal boundary conditions, the figure shows that the sensible and latent heat fluxes are symmetrical along the humid zone. It can also be noted that q_s increases towards the flow direction and reaches a constant value near the channel exit. According to the relation Eq. 7b, an opposite trend is shown for q_l in order to respect the energy balance at the interface. As predicted from temperature profiles presented in Fig. 4 and 7, the sensible heat transfer at the humid regions is towards the interface. For the latent heat transfer, it can be noted that q_l is always positive because only the case of evaporation is dealt with. By comparing the 2 configurations (case 1 and 2), one can note that the latent heat (needed for evaporation) at humid regions is more effective in the second case. This can be explained by the fact that for this case the axial velocity near the wall is more important.

The effect of the number of wetted zones and their positions on the local evaporative rate across the interface is illustrated in Fig. 8 and 9. For the case n =1, Fig. 8a shows that in accordance with the above result for latent heat transfer, the evaporative rate at the channel exit is more effective for the case when the humid zone is located at the channel entrance. This result can be justified by the fact that the axial velocity in the second case (dry exit) is more important than the one in the first case (wet exit) and this tendency has the effect of activating the evaporation process.

For different configurations, the local evaporative rate for the case of 2 wetted zones (n = 2) is plotted in

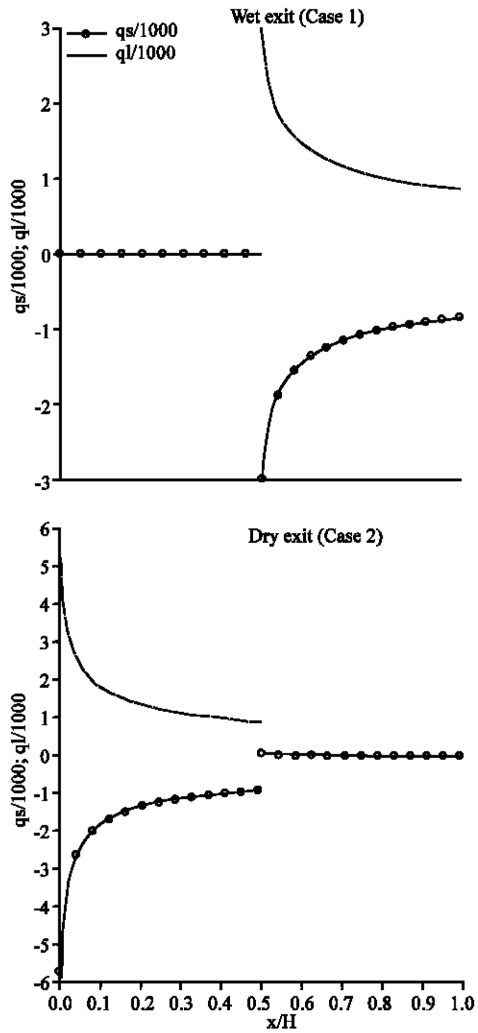

Fig .7: Effect of the wetted zone position on the interfacial relative heat fluxes for n = 1 (u_0 = 1 m s^{-1}; C_0 = 0.005; T_0 = 283.15K; q = 0; T_W = 373.15K; d/H = 0.015)

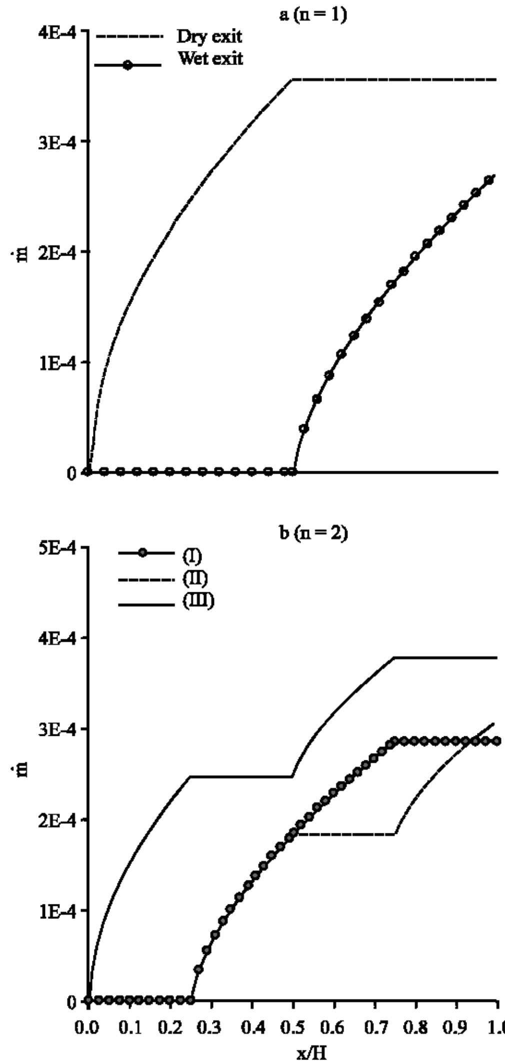

Fig. 8: Effect of the wetted zone position on the local evaporative rate (u_0 = 1 m s^{-1}; C_0 = 0.005; T_0 = 373.15K; T_W = 373.15K; q = 0; d/H = 0.015)

Fig. 8b. First, it is clearly seen that evaporation is more efficient in the case of the humid zone being located at the channel entrance (configuration III: wetted-dry-wetted-dry). This result can be allotted to the effect of convection which is important, particularly near the inlet, because the relative high heat transfer is associated with the development of the flow. The careful observation of curves given in Fig. 8b shows that the evaporative rate in the first configuration (configuration I: dry-wetted-wetted-dry) is slightly more important than that of the second configuration (configuration II: dry-wetted-dry-wetted). This result can be justified by the higher temperature of the fluid reaching the entry of the second humid zone for the first configuration (configuration I).

To provide a further perspective about the role of the humid zones number (n), for the same wetted length, Fig. 9 displays the result of local evaporative rate for different numbers of the wetted zones (n = 1, n = 2, n = 3 and n = 4). In all these studied cases, the channel inlet is occupied by a wetted zone. This choice is approved by the previous results. One notices that at the channel exit, the evaporation is intensified when we increase the number 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 (n) of these zones. The Fig. 9 also shows that by passing from n = 1-4, the relative gap of the evaporative rate value can exceed 12%.

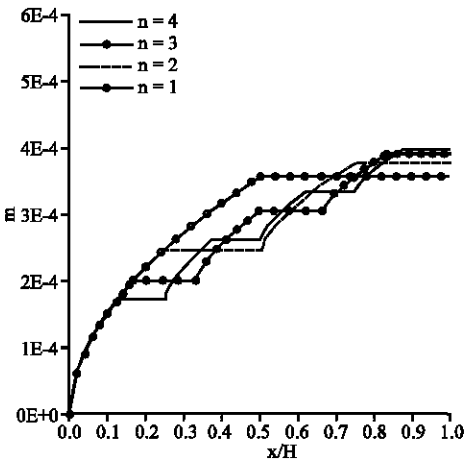

Fig. 9: Effect of the number of wetted zone on the local evaporative rate (u_0 = 1 m s^{-1}; C_0 = 0.005; T_0 = 373.15K; T_W = 373.15K; q = 0; d/H = 0.015)

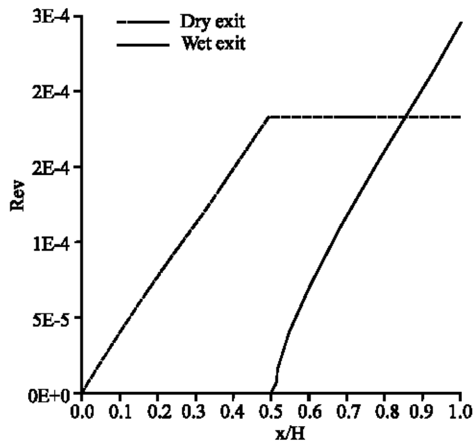

Fig. 10: Effect of the wetted zone position in the case of heated channel wall (u_0 = 1 m s^{-1}; C_0 = 0.005; T_0 = 373.15K; T_W = 373.15K; q = 500W/m^2; d/H = 0.015)

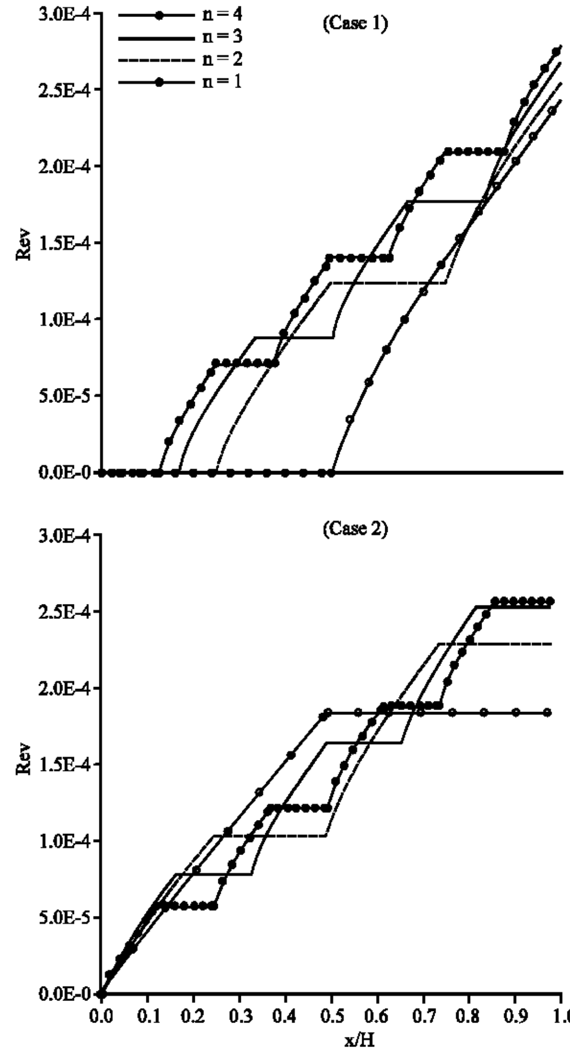

Fig. 11: Effect of the number of wetted zone in the case of heated channel wall (u_0 = 1 m s^{-1}; C_0 = 0.005; T_0 = 373.15K; T_W = 373.15K; q = 500W/m^2; d/H = 0.015)

Case of heated channel wall: In this study attention was paid to the evaporation of water in the case of the partially wetted plate being submitted to a constant heat flux. The ambient temperature considered in this part is T_0 = 20°C. Figure 10 gives the local evaporative rate for the 2 cases n = 1 (dry-wet and wet-dry). It is seen that in contrast with the results obtained for an insulated channel wall, the evaporation is more effective for the case of wet channel exit. This can be attributed to the quantity of heat accumulated along the first dry half of the channel.

For a given wetted length (H/2), the effect of the humid zones number (n) on the local evaporative rate is also studied, for the case of a heated channel wall. Generally, the Fig. 10 shows that an increase of the number n of the humid zones generates an increase of the evaporative rate. In contrast with the above case of insulated channel wall, this increase is more significant in the case where the entry of the channel is dry.

In the following section one is interested to the case of n = 1. In fact, the study of an insulated plate is just a particular case of a plate subject of a constant heat flux (with q = 0). On the other hand, we have shown that for an insulated channel plate, the evaporation is more efficient in the case of the entry of the channel being humid. But for a plate subject of a relatively important heat flux, the evaporation is more intense in the case where the entry is humid. So, one can foresee the existence of a critical value for the density heat flux from which the behaviour of the evaporative rate is reversed. Below this

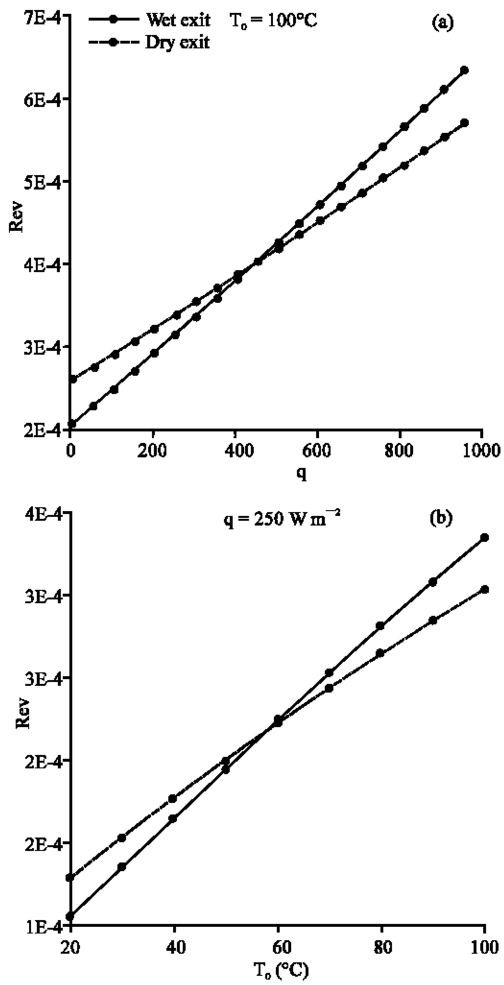

Fig. 12: Effect of thermal conditions on inlet evaporative rate (n = 1) (u_0 = 1 m s^{-1}; C_0 = 0.005; T_W = 373.15K; d/H = 0.015)

critical value of q, evaporation is more efficient in the dry exit case and above this value this tendency is reversed.

At a fixed free stream temperature (T_0 = 100°C) and for the two cases n =1 (dry-wet and wet-dry), the Fig. 11 display the evolutions of the total evaporative rate at the channel exit with the imposed heat flux. As foreseen, the Fig. 11 shows that the evaporation rates for the 2 studied cases converged as the density heat approaches a critical value (q ≈ 450 W m^{-2} for T_0 = 100°C). Below this value, water evaporation is more important for the case (wet-dry). Whereas above this flux density, water evaporates faster in the other case (dry wet).

For the same above reason, Fig. 12 illustrates the effect of the ambient temperature T_0 variation on the total evaporative rate for a given heat flux (q = 250°C). One notices that the increase of the temperature ambient T_0 enhances the evaporation in the case where the channel exit is dry. On the other hand, for a low value of T_0 the evaporation becomes more pronounced when the exit is humid. One remarks also that there exists an intermediate value (T_0) for which the evaporation is independent of the studied configuration.

CONCLUSION

The evaporation by forced convection in a partially-wetted channel has been numerically studied for an air-water system. The studied channel is made up of 2 parallel plates. The partially wetted plate is externally insulated or subject to uniform heat flux. The second plate is dry and isothermal. The effect of the humid zones position on the characteristics of the heat and mass transfers has been analyzed. Furthermore, the effectiveness of the evaporation rate for the same humid surface is studied for different numbers of the humid zones. The major results are briefly summarised in the following:

- For the case of one humid zone, the position of the wetted zone has no significant effect on the fluid flow
- The heat and mass transfers are considerably influenced by the position of the dry zone. Particularly, it is verified that for n = 1 and for an insulated humid wall, the latent heat needed for the evaporation at the humid region is more effective in the case of the first half of the plate being wetted. This trend is inverted for a heated humid wall
- The effect of the number of wetted zones and their positions on the local evaporative rate across the interface is studied for the same humid surface length. It is shown that the evaporative rate depends largely on the number (n) of humid zones. Results show that the evaporation is intensified when we increase the number of the humid zones
- The rising of the humid zone number increases the total evaporative rate which can exceed 12% in relative value
- For the 2 studied cases n = 1, it is shown that there exists a critical value for the density heat flux from which the behaviour of the evaporative rate is reversed. Below this value of q, evaporation is more efficient in the dry exit case and above this value the evaporation is more intense for the wet exit case

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