Influence of thickness of the porous layer on thin film condensation in forced convection in a canal whose walls are covered with a porous material: Determination of lengths of entry

Abstract – *We have examined film condensation in a channel whose walls are covered with porous media numerically. The transfers in the porous medium and the liquid film are respectively described by the Darcy-Brinkman-Forchheimer model and the equations of the hydrodynamic and thermal boundary layers and are solved by the decentered implicit finite difference method and the iterative Gauss-Seidel method.*
After validating, the influence of the thickness of the porous layer on the longitudinal velocity *and the temperature profiles in both media (pure liquid and porous medium), the thickness of the liquid film, the local Nusselt number and the lengths of entry have been studied.*
We note that an increase in the thickness of the porous layer increases the friction and decreases the contact of the fluid with the cold plate and allows a decrease in the longitudinal velocity *and an increase in the temperatures in the porous medium and the pure liquid, a decrease liquid film thickness (disadvantaged condensation) and increases the local Nusselt number and also an increase in the length of entry. The increase in length of entry is quasi-linear. The sensitivity of condensation to a change in thickness of the porous layer is constant.*

Keywords*: Channel with Porous Wall, Condensation Thin Film, Generalized Darcy-Brinkman- Forchheimer Model, Lengths of Entry, Thickness of the porous layer*

1. Introduction

Heat and mass transfers during the condensation of saturated steam in a porous medium are generally encountered in practice such as in many technological fields: heat exchangers, refrigeration, cooling of electronic components, energy storage, air conditioning, desalination, distillation, chemical reactors, drying.
Vapor condensation in a porous medium is the subject of many recent and current theoretical and experimental studies. [1-19].
Shekarriz and Plumb [1] by examining the effect of the presence of porous medium on the condensation of the film on the external wall of a horizontal tube, showed that they contributed to significantly reducing the thickness of the liquid film. So it (the presence of porous medium) improves the heat exchange with the wall. By carrying out numerical and an analytical studies of film condensation on a wall covered with an inclined porous material, Chaynane R. and al. [2] produced “a comparison between the Darcy-Brinkman (DB) model and the Darcy-Brinkman-Forchheimer (DBF) model. They also investigated the effects of

permeability, effective viscosity and dimensionless thickness of the porous coating on flow and heat transfer enhancement". With numerical and analytical studies of a thin film vapor condensation problem in a porous medium, Asbik M and al. [3] compared the Darcy-Brinkman (DB) and Darcy-Brinkman-Forchheimer (DBF) models. They exposed in their results: the temperature and velocity profiles in the porous layer, the thickness of the liquid film, the local Nusselt number. They showed the influences of the dimensionless thickness of the porous layer, the Darcy number and the Reynolds number. Ndiaye M. and al. [4-7] studied the influences of many parameters such as the dimensionless thickness of the porous layer, the Froude, Prandtl, Jacob, Reynolds numbers and the ratio of thermal conductivity on the transfers in the both media (porous and pure liquid) by proposing a numerical model of the condensation of pure saturated vapor in thin film in forced convection on a wall covered with porous material. Patil A.A. and al. [8] submitted "a solar water distillation system with a combination of surface condenser and vacuum pump. They showed that the daily distillate production increases by up to 43% by coupling the surface condenser and the vacuum pump due to the increased rate of evaporation and condensation". R.S. Jha and al. [9] exposed "the numerical simulations based on the analytical modeling of the heat and mass transfer processes involved in the condensation of moisture in the flue gases in a tubular counter-current heat exchanger. They investigated operating parameters such as fluid flow inlet temperatures and flow ratio with the aim of achieving maximum heat and moisture recovery". A. Nasr and S. Al-Ghamdi [10] showed that "the presence of the porous layer enhances the heat and mass transfer performance at the liquid-gas interface during the liquid film evaporation in the free convection by proposing a numerical study of coupled heat and mass transfers during the evaporation of a flowing liquid film". Numerically, Abdelaziz Nasr [11] proved that "the presence of the porous layer during the liquid film condensation improves the performance of heat and mass transfer at the liquid-gas interface". Charef A. and al. [12] studied and showed numerically that R152a-air condensation corresponds to better condensation with higher condensed mass ratio (m_{cd}) and local heat transfer coefficient (h_T) compared to R134a-air under the same conditions. Mosaad M. E-S. and al. [13] presented "a model of a regular process of condensation of a laminar film on a vertical wall with the back face cooled by free convection in a porous medium saturated with fluid. The conjugate solution of the laminar film condensation problem is different from a Nusselt-type solution was concluded". Charef A. and al. [14] presented "a model on liquid film condensation in a thermal desalination process, which is based on the phase change phenomenon. First, they learned that the condensation process and liquid film thickness improved with increasing temperature difference between the inlet and the wall. Decreasing the radius of the tube increased the process of condensation. The radius of the tube and the incondensable gas were finally retained as relevant factors to improve the efficiency of thermal desalination units". Sellami K. and al. [15] found and claimed that for thicker porous medium and high porosity, the evaporative cooler is more powerful with up to 23% improvement by studying numerically, from medium porous laminar airflow the combined exchanges of heat and mass in the direct evaporative cooler process. Ndiaye P.T. and al. [16] presented "a numerical study of thin layer type condensation in forced vapor convection on a channel whose walls are covered with porous material. They studied the influence of Reynolds and Prandtl numbers. The increase in the Reynolds number and the Prandtl number led to an increase in the longitudinal velocity and the temperature, improves the heat exchanges at the interface of the porous medium and the liquid film (local Nusselt number)".

In their article, Ndiaye P.T. and al. [17-18] presented "the numerical study of thin film type condensation in forced convection of pure saturated steam in a channel whose walls are covered with a porous material". By noting that the thickness of the two liquid films vary according to variable (parameters of the physical problem) and can meet, they [17-18] studied this phenomenon (the meeting between the two liquid films) thanks to the length of entry, which will make it possible to determine the sensitivity of the condensation to a variation of a variable (or parameter of the physical problem). They studied the influence of Jacob numbers and ratio of form on flow, transfer and condensation. Ndiaye G. and al. [19] worked on a study of condensation by forced convection of a laminar film of pure and saturated vapor on a porous plate inclined towards the vertical. The influences of the variations of different parameters such as: the inclination, the dimensionless thermal conductivity, the effective viscosity on the transfers were presented.

As Ndiaye P. T. and al. [17, 18], in our case we will study this time the influence of the thickness of the porous medium instead of the Jacob number and a ratio of form on the flow , transfer and condensation (unlike Ndiaye P. T. and al. [17, 18]).

The hydrodynamic and thermal boundary layer equations will be employed in the pure liquid and will be used together with the generalized Darcy-Brinkman-Forchheimer (DBF) equations in the porous medium. To carry out this project well, first, we will proceed to the mathematical formulation with the descriptions of the simplifying hypotheses, the equations of the problem and the boundary conditions. Then announce the numerical resolution method. Finally evaluate the influence of the thickness of the porous medium on the velocity and temperature profiles in both media (porous medium and liquid film or pure liquid), the thickness of the liquid film, the local Nusselt number (the rate of heat transfer) and the lengths of entry (which characterizes condensation).

2. Mathematical formulation:

2.1 Physical model and assumptions :

This work focuses on the modeling of flows and transfers in porous media and pure liquids.

We consider a vertical channel (physical model) of width *2A*, formed by two flat plates of length L, symmetrical, each covered with a porous layer of thickness *H*, permeability *K*, porosity ε (Fig. 1). At the entrance to the channel, a flow of pure vapor arrives at temperature

T_S, speed U_0, assumed to be uniform. The plates (porous walls) are assumed to be isothermal at the temperature Tw lower than the saturation temperature T_s of the pure steam.
During the flow of vapor in the channel, there is then a condensation of the vapor on the porous wall (cold) leading to the formation of a liquid film flowing on the wall.
We thus discover three regions in the channel:
(1) The porous layer or porous medium,
(2) The liquid film or condensate film,
(3) The pure steam
The condensate film flows under the effect of the forces of gravity and viscous friction. (x,y) and (u,v) are respectively the Cartesian coordinates and the velocity components in the porous medium and the liquid in the coordinate system associated with the model.

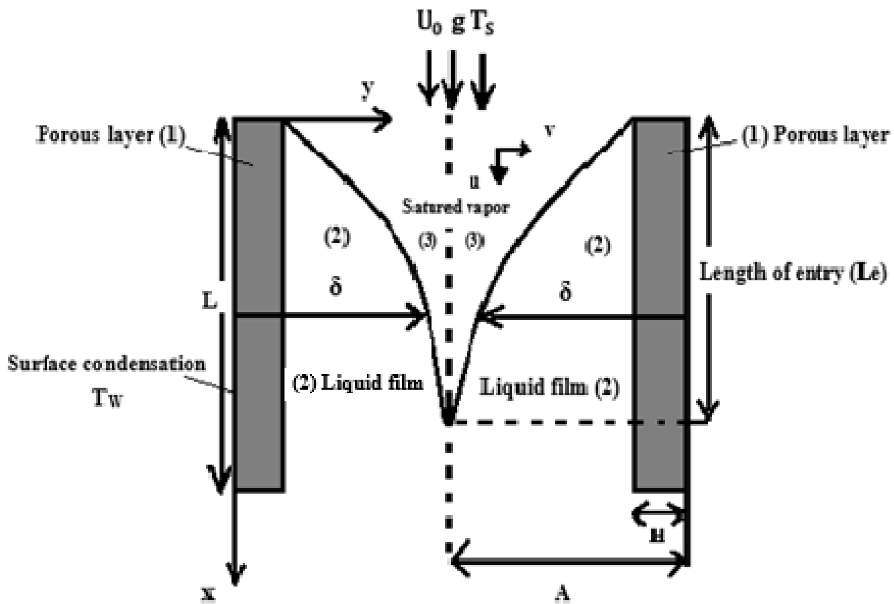

Fig 1 Geometry of the physical model and coordinate system

In order to achieve a coherent mathematical modeling of the physical problem and to reason the difficulties related to the resolution of the governing equations of the phenomenon, we considered the following simplifying hypotheses:
1-The thermal and hydrodynamic approximations of the boundary layers are acceptable.
2- The matrix or porous material is homogeneous and isotropic.
3-The porous medium is saturated with a supposedly incompressible and Newtonian fluid (where generally in the vapor phase (3), the flow and transfer equations are neglected).
4-The flow generated is laminar, permanent and two-dimensional
5-the work induced by the pressure forces and the viscous stresses is neglected.

6-The thermo-physical properties of the fluids and those of the porous matrix are supposed to be constant in the studied temperature range
7-To describe the flow in the porous layer the generalized model of Darcy-Brinkman-Forchheimer (DBF) is used.
8. Condensation appears as a thin film thicker than the porous material.
9-The porous matrix and the condensate are in local equilibrium.
10-The liquid-vapor interface is in thermodynamic equilibrium.
11- The fluid and solid phases of the porous medium are in thermal equilibrium
12-The transverse and longitudinal variation of the pressure is not taken into account in the porous matrix.
13- The transfer of energy by radiation is neglected.
14- The flow considered as axisymmetric because of the geometry of the problem.

2.2 Dimensionless Equations, Determination of local Nusselt number and length of entry

In the domains (1) and (2) already defined, the equations which govern the transfers and the boundary conditions associated with them have been made dimensionless by using the following variables and parameters, we pose:

$$x^* = \frac{x}{L} \quad (1) \; ; \quad y^* = \frac{y}{A} \quad (2) \; ; \quad H^* = \frac{H}{A} \quad (3) \; ; \quad \delta^* = \frac{\delta}{A} \quad (4) \; ; \quad u^* = \frac{u}{U_0} \quad (5)$$

$$v^* = \frac{v}{U_0} \quad (6) \; ; \quad \theta = \frac{T - T_W}{T_S - T_W} \quad (7) \; ; \quad \lambda^* = \frac{\lambda_l}{\lambda_{eff}} \quad (8) \; ; \quad \nu^* = \frac{\nu_{eff}}{\nu_l} \quad (9)$$

By homotopic transformation, we will transform the adimensional physical domain (x*,y*) obtained, presenting a curvilinear interface (liquid/pure vapor interface) into a rectangular domain (***X***,***η***) in which, the boundary and the interfaces (interfaces porous medium/pure liquid and pure liquid/vapor) will be identified by lines of constant coordinates.

We pose:

$$\begin{cases} X = x^* \\ \eta = coef.\dfrac{y^*}{H^*} + (1-coef)\left\{1 + \dfrac{y^* - H^*}{\delta^*(X) - H^*}\right\} \end{cases} \quad (10)$$

With *Coef* it is a coefficient equal to 1 in the porous layer and 0 in the pure liquid.
Thus the frame (x*, y*) is transformed into a rectangular field (***X***, ***η***).
The porous medium/pure liquid and pure liquid/vapor interfaces are identified respectively by the coordinate lines η = 1 and η = 2.

The dimensionless equations in the rectangular domain (***X***, ***η***) in both media are written:

Porous layer: $0 \le \eta \le 1$

The equation of conservation of mass or equation of continuity is:

$$\frac{\partial u_p^*}{\partial X} + \frac{L}{A}\frac{1}{H^*}\frac{\partial v_p^*}{\partial \eta} = 0 \quad (11)$$

The equation of momentum following X-direction is:

$$u_p^*\frac{\partial u_p^*}{\partial X} + \frac{L}{A}\frac{v_p^*}{H^*}\frac{\partial u_p^*}{\partial \eta} = \frac{\varepsilon^2}{Fr} - \varepsilon^2\left(\frac{L}{A}\right)^2\frac{v^*.Da}{\mathrm{Re}}u_p^* + \varepsilon^2\left(\frac{L}{A}\right)^2 v^*\frac{1}{\mathrm{Re}\,H^{*2}}\frac{\partial^2 u_p^*}{\partial \eta^2} - \varepsilon^2 F\sqrt{Da}\,\frac{L}{A}u_p^{*2} \quad (12)$$

The equation of heat is:

$$u_p^*\frac{\partial \theta_p}{\partial X} + \frac{L}{A}\frac{v_p^*}{H^*}\frac{\partial \theta_P}{\partial \eta} = \left(\frac{L}{A}\right)^2\frac{1}{\mathrm{RePr}_{eff}\,H^{*2}}\frac{\partial^2 \theta_P}{\partial \eta^2} \quad (13)$$

Pure liquid: $1 \prec \eta \prec 2$

The equation of conservation of mass or equation of continuity is:

$$\frac{\partial u_l^*}{\partial X} - \frac{(\eta-1)}{\left(\delta^*(X)-H^*\right)}\frac{d\delta^*(X)}{dX}\frac{\partial u_l^*}{\partial \eta} + \frac{L}{A}\frac{1}{\left(\delta^*(X)-H^*\right)}\frac{\partial v_l^*}{\partial \eta} = 0 \quad (14)$$

The equation of momentum following X-direction is:

$$u_l^*\left[\frac{\partial u_l^*}{\partial X} - \frac{\eta-1}{\delta^*(X)-H^*}\frac{d\delta^*(X)}{dX}\frac{\partial u_l^*}{\partial \eta}\right] + \frac{L}{A}\frac{v_l^*}{\delta^*(X)-H^*}\frac{\partial u_l^*}{\partial \eta} = \left(\frac{L}{A}\right)^2\frac{1}{\mathrm{Re}\left(\delta^*(X)-H^*\right)^2}\frac{\partial^2 u_l^*}{\partial \eta^2} + \frac{1}{Fr}\left(1-\frac{\rho_v}{\rho_l}\right) \quad (15)$$

The equation of heat is:

$$u_l^*\left[\frac{\partial \theta_l}{\partial X} - \frac{\eta-1}{\delta^*(X)-H^*}\frac{d\delta^*(X)^*}{dX}\frac{\partial \theta_l}{\partial \eta}\right] + \frac{L}{A}\frac{v^*_l}{\delta^*(X)-H^*}\frac{\partial \theta_l}{\partial \eta} = \left(\frac{L}{A}\right)^2\frac{1}{\mathrm{Re\,Pr}\left(\delta^*(X)-H^*\right)^2}\frac{\partial^2 \theta_l}{\partial \eta^2} \quad (16)$$

To these equations are added the following boundary and dimensionless continuity conditions:

Conditions at the entry: $X = 0$

$$u_p^*(0,\eta) = 1 \quad (17) \quad ; \quad v_p^*(0,\eta) = 0 \quad (18) \quad ; \quad \theta_p(0,\eta) = 1 \quad (19)$$

Boundary conditions at the wall**:** $\eta = 0$

$$u_p^* = v_p^* = 0 \quad (20) \; ; \quad \theta_p = 0 \quad (21) \; ; \quad u_p^* = v_p^* = 0 \quad (22) \; ; \quad \theta_p = 0 \quad (23)$$

At the porous/pure liquid interface**:** $\eta = 1$

$$u_l^* = u_p^* \quad (24) \quad ; \quad \theta_l = \theta_p \quad (25)$$

Continuity of constraints

$$\frac{\mu^*}{H^*}\frac{\partial u_p^*}{\partial \eta} = \frac{1}{\left(\delta^*(X) - H^*\right)}\frac{\partial u_l^*}{\partial \eta} \quad (26)$$

Continuity of thermal flows:

$$\frac{1}{H^*}\frac{\partial \theta_p}{\partial \eta} = \frac{\lambda^*}{\left(\delta^*(X) - H^*\right)}\frac{\partial \theta_l}{\partial \eta} \quad (27)$$

At the liquid / vapor interface $\eta = 2$

$$\theta_l = 1 \quad (28) \quad ; \quad \frac{\partial u_l^*}{\partial \eta} = 0 \quad (29)$$

The dimensionless speed and temperature depending on the thickness of the liquid film, the heat and mass balances can be coupled. The coupled equation of mass flow and heat balance made dimensionless is expressed by the following relationship:

$$\left(\frac{L}{A}\right)^2 \frac{J_a}{(Pe)_{eff}} \frac{1}{H^*} \frac{\partial \theta_p}{\partial \eta} /_{\eta=0} = \frac{d}{dX}\left\{(1+Ja)\delta^*(X)\frac{\rho_v}{\rho_l}\right\} - Ja\frac{d}{dX}\left\{H^*\int_0^1 \theta_p u_p^* d\eta + \left(\delta^*(X) - H^*\right)\int_1^2 \theta_l u_l^* d\eta\right\} (30)$$

With:

The mass flow rate:

$$H^*\int_0^1 u_p^* d\eta + \left(\delta^*(X) - H^*\right)\int_1^2 u_l^* d\eta = \frac{\rho_v}{\rho_l}\delta^*(X) \quad (31)$$

Number of Jacob:

$$Ja = \frac{Cp_l (T_S - T_W)}{h_{fg}} \quad (32)$$

Number of Froude:

$$Fr = \frac{U_0^2}{gL} \quad (33)$$

Number of Reynolds:

$$\mathrm{Re} = \frac{U_0 L}{\nu_l} = \nu^* . \frac{U_0 L}{\nu_{eff}} \quad (34)$$

Number of Darcy:

$$Da = \frac{A^2}{K} \quad (35)$$

Number of Prandtl:

$$\Pr = \frac{\nu_l}{\alpha_l} = \frac{\mu_l C_{p_l}}{\lambda_l} \qquad (36)$$

Number of Peclet:

$$Pe = \mathrm{Re}.\Pr \qquad (37)$$

Number of Peclet modified:

$$Pe_{eff} = \lambda^* Pe = \lambda^* \mathrm{Re}.\Pr \qquad (38)$$

Number of de Prandtl modified:

$$\Pr_{eff} = \lambda^* \Pr \qquad (39)$$

Determination of the Lengths of Entry (Le)

The length of entry already defined by Ndiaye P. T. and al. [16]: 'The length of entry (Le) is defined as the length from the channel inlet to the meeting of the two liquid films. This is the length of the channel from which the boundaries layers conditions are not applicable anymore [16].

$$\delta(x = Le) = A \qquad (41)$$

In dimensionless form and a rectangular domain, it is:

$$\delta^*(X = Le^*) = 1 \qquad (42)$$

The partial differential equations which govern our problem are not only nonlinear, coupled between them, also coupled with their boundary conditions. They generally do not have analytical solutions, except for very simplified cases. The use of numerical methods becomes inevitable for complex cases closer to reality.

3. NUMERICAL METHODOLOGY

Numerical resolutions include two steps: a meshing step and a discretization step. The transfer equations are discretized by an implicit finite difference method.

The transfer equations are discretized by an implicit finite difference method. The advection terms are discretized respectively with a backward off-center scheme and the diffusion terms are discretized respectively with a centered in order to make possible the main diagonals of the most dominant matrices possible (for more stability).

We used a double-scan method combined with an iterative Gauss-Seidel type line-by-line sub-relaxation scheme to solve numerically the systems of coupled algebraic equations thus obtained. We have defined the relaxation coefficients empirically to guarantee the non-growth of calculation errors during the iterative process, which will make our diagrams stable and convergent. For the local Nusselt number (heat transfer rate), a second order progressive finite difference scheme will be used for more precision.

4. RESULTS AND DISCUSION

4.1 Results of method validation:

We validated our model by comparing it to the work of Ndiaye and al. [4-7].
For this we considered the same conditions as him, a low number ($Re = 2$). Because Ndiaye M. (2014) had retained that his model was only valid for values of the Reynolds number lower than 7. As we considered two characteristic lengths L and A to make our model dimensionless, we will consider them equal ($L/A = 1$) because they [4-6] had a single characteristic length and low permeability, therefore a high Darcy number ($Da = 10^{12}$).
For the study of the sensitivity of the mesh we chose $\Delta X = 0.0001$ and $\Delta\eta = 0.02$ and the stopping criterion (of convergence) in the iterative process is fixed at $\varepsilon_{arrêt}=10^{-6}$ to respond to both the speed, the precision and the convergence of the calculations. Lower values of $\varepsilon_{arrêt}$ will unnecessarily increase the volume of calculations without a noticeable and significant improvement in accuracy.
We note a small superiority of the dimensionless longitudinal velocity, particularly in the porous medium of our study (compared to the model of Ndiaye M. and al. [4-7], which is due to the inertial effects that we took into account in the porous medium (unlike Ndiaye M. and al. [4-7]) which are not predominant in our case (because Re = 2, Re is low) (Fig. 2) We can retain a good agreement for the dynamic field (Fig. 2). a very good agreement is noted for the thermal field despite the slight superiority of the values of our dynamic field (Fig. 3). Because advection is negligible with low values of the Reynolds number. The agreement is good, our model is valid.

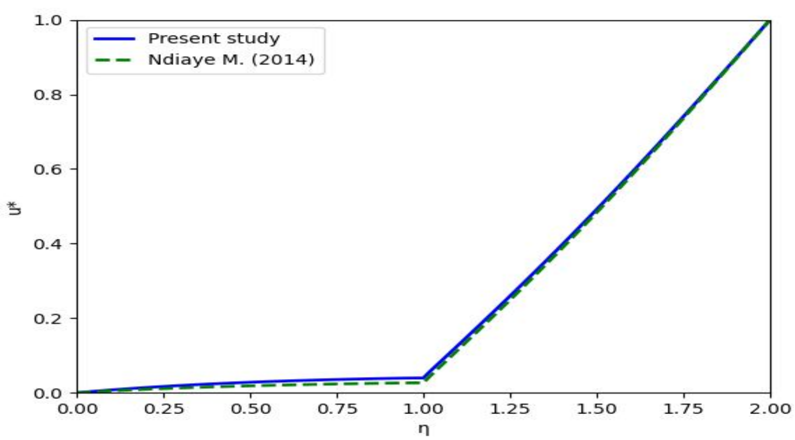

Fig. 2: Comparison of the velocity profile of our study with that of Ndiaye M. and al. [4-7]

$Re=2; Fr=10^{-4}; \lambda^*=2.9; H^*=2.10^{-3}; Ja=10^{-3}; v^*=1; L/A=1; Da=10^{12};$ Ɛ=0.4; F=0.55

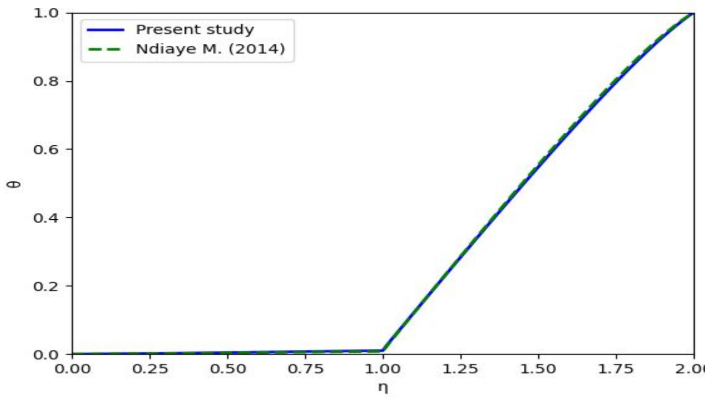

Fig. 3: Comparison of the temperature profile of our study with that of Ndiaye M. and al. [4-7]

$Re=2;\ Fr=10^{-4};\ \lambda^*=2.9;\ H^*=2.10^{-3};\ Ja=10^{-3};\ \nu^*=1;\ L/A=1;\ Da=10^{12};\ ;\ \varepsilon=0.4;\ F=0.55$

4.2- Results and discussions

In this study the influence of the thickness of the porous layer on the behavior longitudinal velocity, the temperature, the thickness of film, the thermal rate of transfer (local Nusselt number) and finally over the length of entry will be shown.

The results from the numerical simulations are related to : $\varepsilon=0.4$; $F=0.55$; $\upsilon^*=1$; $Da=10^9$; $\lambda^*=3$; $Fr=10^{-4}$.

We note that longitudinal velocity are more significant in liquid film than in the porous medium. What is due to viscosities (microscopic and macroscopic) and to the effects of the friction imposed by the porous medium. The longitudinal velocity decreases as the thickness of the porous layer increases (Fig. 4). The presence and increase in the thickness of the porous layer perturbs the flow by its friction.

The further one moves away from the cold plate of the porous medium, the more the temperature increases and tends towards the dimensionless saturation temperature (equal to 1).We also notice an increase in temperature with increasing dimensionless thickness of the porous layer (Fig. 5). Indeed, the increase in the porous coating decreases the contact between the fluid and the (cold) condensation plates, which increases the temperature.

The thickness of the liquid film is an increasing function of the abscissa X, due to the increase in the mass flow of liquid along the channel. It determines the effectiveness of condensation (Fig. 6). It shows an increase in the thickness of the condensate film as the thickness of the porous layer decreases. It can be interpreted by the fact that the decrease in the thickness of the porous layer allows better contact with the cold plate. This promotes condensation and therefore increases the thickness of the liquid film.

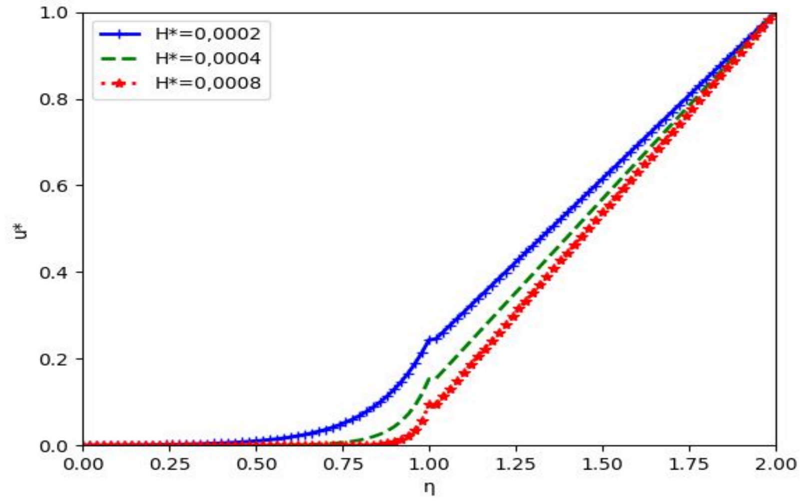

Fig. 4: Variation of the longitudinal velocity as a function of the ordinate η for different values of H* at the position X= 0.05.
Re=200; Pr=5; L/A =25 ; Ja=10^{-8}

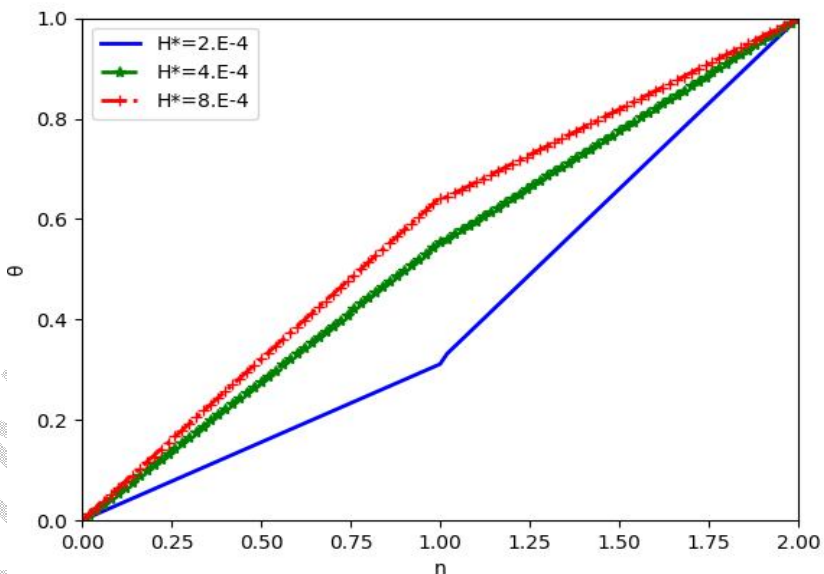

Fig. 5: Variation of the longitudinal temperature as a function of the ordinate η for different values of H* at the position X= 0.05.
Re=200; Pr=5; Ja=10^{-7} ; L/A =75 ; Ja=10^{-7}

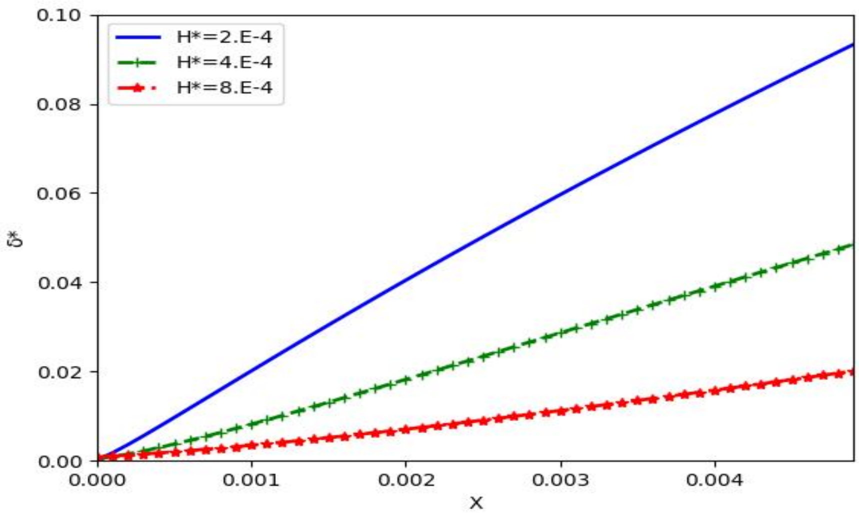

Fig. 6: Variation of the thickness of the liquid film as a function of the abscissa X for different values H*

Re=25; Pr=5; L/A =150 ; Ja=10^{-7}

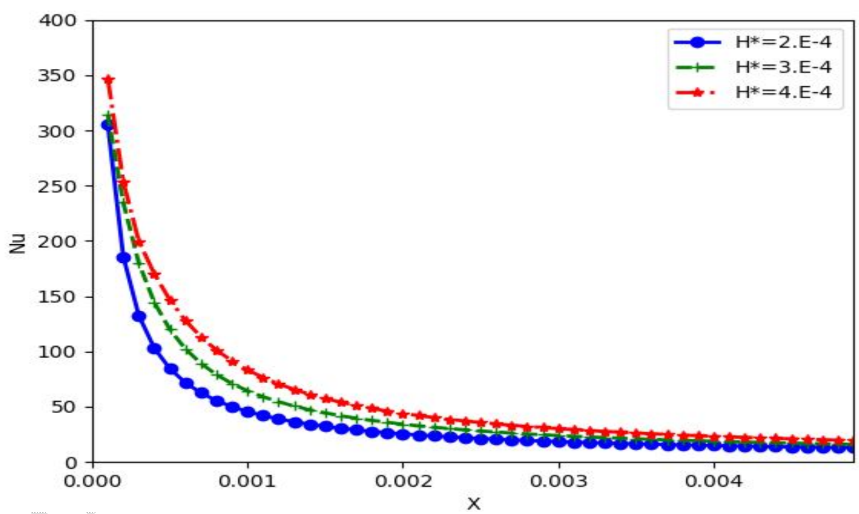

Fig. 7: Variation of the Nusselt number as a function of the abscissa X for different values of H*

Re=25; Pr=2 ; L/A =200 ; Ja=10^{-7}

The local Nusselt number, rate of heat transfer at the porous medium/pure liquid interface, is the ratio of the total heat transfer to the transfer by conduction. When the local Nusselt number is low, much of the heat transfer is by conduction with no flux or very low flux because there is little convection. When the Nusselt number is large, transfer by convection is active. We study the variation of the Nusselt number as a function of the abscissa X. We note the value of the Nusselt number is very large at the entrance of the channel, then decreases rapidly along the porous wall to reach very low asymptotic values. The heat transfer by conduction (weak at the entrance of the channel) gradually increases along the porous wall on the convective heat transfer, which explains a decrease in the Nusselt number as a function of the abscissa X. that the fluid descends along the abscissa X in the channel; its temperature tends towards that of the plates and the temperature gradients become weaker. The local Nusselt number decreases along X (along the channel). An increase in the Nusselt number is noted with the increase in the thickness of the porous layer (Fig. 7).

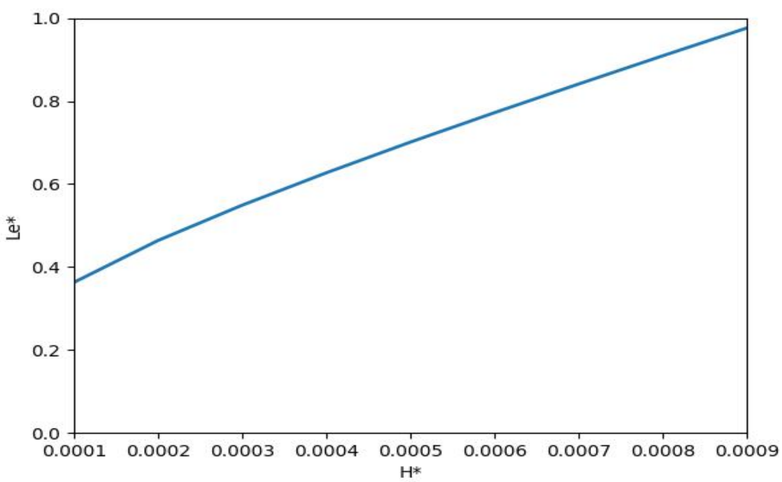

Fig. 8: variation of the dimensionless length of entry ($\boldsymbol{Le^*}$) as a function of H*
Re=50; Pr=3; Ja=10^{-7}; L/A=100

The variation of the dimensionless lengths of entry highlights the ease of variation of the dimensionless liquid film thickness which depends exclusively on condensation. Thus, the variations of the lengths of entry as a function of the thickness of the porous layer (Fig. 8) make it possible to characterize the sensitivity of the condensation to a variation of the thickness of the porous layer. An increase in the thickness of the liquid film (promoted condensation) leads to a decrease in the lengths of entry and respectively its decrease increases the lengths of entry. Hence the variations of the dimensionless lengths of entry (Fig. 8) confirm those of the dimensionless thickness of the liquid film (Fig. 6).

Figure 8 shows an increase in length of entry as a function of dimensionless porous layer thickness. This increase is almost linear unlike Ndiaye P. T. and al. [17-18] (Variations in the dimensionless length of entry according to the Jacob number and ratio of form). Here, the sensitivity of the condensation to a change in thickness of the dimensionless porous layer is constant whatever its value (thickness of the porous layer).

5. Conclusion

We have studied forced convection condensation of the thin film type. Our equations (generalized Darcy-Brinkman-Forchheimer (DBF) equations in the porous medium and those of the hydrodynamic and thermal boundary layers in pure liquid) were solved using the decentered implicit finite difference method. The advection terms are discretized with a backward-centered scheme and the diffusion terms with a centered scheme.

After validating our model in comparison to Ndiaye M. [4-7], the influence of the thickness of the porous layer on the longitudinal velocity, the temperature profiles in both media (pure liquid and porous medium), the thickness of the liquid film , the local Nusselt number (the rate of heat transfer) and the lengths of entry have been studied.

It is found that an increase in the thickness of the porous layer increases the friction and decreases the contact of the fluid with the cold plate and allows a decrease in the speed and an increase in the temperatures in the porous medium and the pure liquid, a decrease liquid film thickness (disadvantaged condensation) and increases the local Nusselt number (heat transfer rate at the porous medium/liquid film interface) and also an increase in the dimensionless length of entry. The increase in length of entry is quasi-linear. The sensitivity of condensation to a change in thickness of the dimensionless porous layer is constant.

Nomenclature

Greeks symbols

α: thermal diffusivity, m2.s-1
δ thickness of condensate, m
ε : Porosity
η : dimensionless coordinate in the transverse direction
θ : temperature dimensionless
λ : thermal conductivity, W.m-1.K-1
μ : viscosité dynamique, kg.m-1.s-1
ν : dynamic viscosity, m2.s-1
ρ : density, kg.m-3

Latines letters:

A: half-width of the channel, m
Cp: specific heat, J.kg-1. K-1
Da: Darcy's number
F: Forchheimer coefficient
Fr: Froude number
g: gravitational acceleration, m.s-2
H: thickness of the porous layer, m
hfg: enthalpy of evaporation, J.kg-1
Ja: Jacob number
K: hydraulic conductivity or permeability, m^2
L: length of the plates of the channel, m
Nu: local Nusselt number
Pe: Peclet number
Pr: Prandtl number
Re: Reynolds number
T: temperature, K
u: velocity along x, m.s-1
U0: velocity of the free fluid (at the entrance of the channel), m.s-1
v: velocity along y, m.s-1
x, y: Cartesian coordinates, m
X: dimensionless coordinate in the longitudinal direction

Subscripts:

eff: effective value
int: porous substrate / pure liquid interface
l: liquid
p: porous

s: saturation	w: wall
v: steam (vapor)	*: dimensionless quantity

References

[1] Shekarriz A. and Plumb O.-A. Enhancement of film condensation using porous fins, *J. Thermo physics and heat Transfer.* 1989:3: 309-314.

[2] Chaynane R., Asbik M., Boushaba H., Zeghmati B., Khmou A. Study of the laminar film condensation of pure and saturated vapor on the porous wall of an inclined plate. *Mechanics & Industries.* 2004:5(4): 381-391.

[3] Asbik M., Chaynane R., Boushaba H., Zeghmati B. and Khmou A. Analytical investigation of forced convection film condensation on a vertical porous-layer coated surface. Heat and Mass Transfer. 2003: 40 (1-2): 143 – 155. DOI 10.1007/s00231-002-0406-8

[4] Momath Ndiaye, Cheikh Mbow, Joseph Sarr, Belkacem Zeghmati, Modou Faye. Numerical Investigation of Laminar Forced Thin Film Condensation of a Saturated Vapor along a Vertical Wall Covered with a Porous Material: Effect of Prandtl and Froude numbers. International Journal on Heat and Mass Transfer Theory and Applications (IREHEAT). 2013:1(6): 339-344

[5] NDIAYE M, MBOW C, SARR J., ZEGHMATI B. Numerical Study of the Thin Film-type Condensation of Saturated Forced into a Vertical Wall Covered with a Porous Material Vapor Convection. *International Journal on Heat and Mass Transfer Theory and Applications* (IREHEAT). 2013:1(6): 330-338.

[6] NDIAYE M, MBOW C, SARR J. Numerical Investigation of Laminar Forced Thin Film Condensation of a Saturated Vapor along a Vertical Wall Covered with a Porous Material. 3rd International Francophone Symposium on Energy and Mechanics, Renewable Energies and Mechanics Applied to Industry. CIFEM 5-6-7 May 2014. French

[7] Momath NDIAYE**.** Numerical Study of the Thin Film-type Condensation of Saturated Forced into a Vertical Wall Covered with a Porous Material Vapor Convection. Ph.D. dissertation, Dpt of Physics, Cheikh Anta Diop University, Dakar, Senegal, 2014.

[8] Atul A. Patil, Atul, Tejas G. Patil, Aniruddha Y. Chaudhari. Performance Evaluation of Passive Solar Water Distillation System with Separate Surface Condenser and Vacuum Pump. International Review of Mechanical Engineering (IREME). 2017: *11(* 7): 467-472.

10.15866/ireme.v11i7.12877

[9] Jha, R., Haribhakta, V., Kolte, A., Shekhadar, S., Tengale, S., Tare, S. Design and Simulation of Condensing Heat Exchanger. International Review of Mechanical Engineering (IREME). 2017:11(2) 473-480. http://dx.doi.org/10.15866/ireme.v11i7.12879

[10] A. Nasr, S. Al-Ghamdi, A, Numerical study of evaporation of falling liquid film on one of two vertical plates covered with a thin porous layer by free convection. International Journal of Thermal Sciences. 2017:112: 335–344. https://doi.org/10.1016/j.ijthermalsci.2016.10.018

[11] Abdelaziz Nasr, Heat and mass transfer for liquid film condensation along a vertical channel covered with a thin porous layer. International Journal of Thermal Sciences. 2018:124: 288-299. http://dx.doi.org/10.1016/j.ijthermalsci.2017.10.016

[12] Charef A, Feddaoui M, Najim M, Meftah H. Comparative study during condensation of R152a and R134a with presence of non-condensable gas inside a vertical tube. Heat and Mass Transfer. 2018:54:1085-1099. https://doi.org/10.1007/s00231-017-2205-2

[13] Mohamed El-Sayed MOSAAD and Rashed AL-AJMI. Film condensation generated by free convection in a porous medium. THERMAL SCIENCE. 2018: 22(6B):2699-2710. https://doi.org/10.2298/TSCI160820316M

[14] Adil Charef, M'barek Feddaoui, Abderrahman Nait Alla, Monssif Najim. Computational Study of Liquid Film Condensation with the Presence of Non-Condensable Gas in a Vertical Tube. Desalination and Water Treatment, Chapter 4, November 5th 2018, pp.55-76. http://dx.doi.org/10.5772/intechopen.76753

[15] Karima Sellami, Nabila Labsi, Monssif Najim, Youb Khaled Benkahla, Numerical Simulations of Heat and Mass Transfer Process of a Direct Evaporative Cooler From a Porous Layer. Journal of Heat Transfer. 2019: 141(7): 071501. *https://doi.org/10.1115/1.4043302*

[16] Ndiaye P. T., Ndiaye M., Mbow C., Ndiaye G. Influence of Reynolds and Prandtl Numbers on Thin Film Condensation in Forced Convection in a Canal Covered with a Porous Material. International Journal on Engineering Applications (IREA). 2020:8(5): 178-187. https://doi.org/10.15866/irea.v8i5.18678

[17] Ndiaye P. T., Ndiaye M., Mbow C., Ndiaye G. Numerical Study of Thin Film Condensation in Forced Convection in a Canal whose Walls Are Covered with a Porous Material: Influence of Jacob Number- Determination of Lengths of entry.

International Journal on Engineering Applications (IREA). 2020:8(4): 125-132. https://doi.org/10.15866/irea.v8i4.18687

[18] Ndiaye P. T., Ndiaye M., Ndiaye G., Mbow C. Numerical Study of Thin Film Condensation in Forced Convection in a Canal Whose Walls Are Covered with a Porous Material: Influence of ratio of form-Determination of Lengths of Entry. Journal of Chemical, Biological and Physical Sciences An International Peer Review E-3 Journal of Sciences Available online at www.jsbsc.org Section C: Physical Sciences, JCBPS; Section C. 2022: 12(3):166-180.
DOI: 10.24214/jcbps.C.12.3.16680

[19] Ndiaye G., Sambou V., Ndiaye, M. and Ndiaye P.T. Numerical Study of Thin Film Condensation in Forced Convection on an Inclined Wall Covered with a Porous Material. Open Journal of Applied Sciences. 2022:12:793-805. https://doi.org/10.4236/ojapps.2022.125053