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Numerical Modeling of the Effect of the Ratio of Thermal Conductivity on the Thin Film Condensation in Forced Convection in a Canal Whose Walls are Covered with a Porous Material

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Authors' contributions

This work was carried out in collaboration among all authors. All authors read and approved the final manuscript.

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ABSTRACT

A numerical modeling of the effect of the ratio of thermal conductivity on the thin film condensation in forced convection in a canal whose walls are covered with a porous material is presented. In this work, the generalized Darcy-Brinkman-Forchheimer (DBF) equations in the porous medium and the hydrodynamic and thermal boundary layer equations in the pure liquid, were used.

Rendered dimensionless and homotopically transformed into a new rectangular basis, we used a finite difference method to discretize them. The advection and the diffusion terms are discretized with respectively a backward-centered scheme and a centered scheme.

After validation, we find that a variation of the longitudinal velocity as a function of the ratio of thermal conductivity only for low values of the Peclet number. When the ratio of thermal conductivity increases, corresponding to an increasingly conductive medium, the longitudinal velocity, the temperature and the Nusselt number increase (even when the Peclet number is high for the thermal field). While the thickness of the liquid film decreases (disadvantaged condensation) and leads to an increase in the length of entry, increase almost linear. The sensitivity of condensation to variations in the ratio of thermal conductivity is constant, whatever its value.

The ratio of thermal conductivity is a very decisive and predictable physical quantity to properly examine the performance of condensation.

Keywords: Channel with porous wall; condensation thin film; generalized darcy-brinkman-forchheimer model; iterative gauss-seidel relaxation method; lengths of entry; ratio of thermal conductivity.

NOMENCLATURE

Greeks Symbols

α : *thermal diffusivity, $m^2.s^{-1}$*
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, $m^2.s^{-1}$*
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, m2*
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, $m.s^{-1}$*
U_0 : *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*
v : *steam (vapor)*
w : *wall*
**:* *dimensionless quantity*

1. INTRODUCTION

"Implemented in many technological fields such as distillation, heat exchangers, energy storage, cooling of electronic components, drying, desalination, distillation …, heat and mass transfer during condensation saturated steam in porous media have received considerable attention in many theoretical and experimental studies" [1-20].

Shekarriz and Plumb [1] have shown that "the presence of porous medium on the condensation of the film on the external wall of a horizontal tube contributes to significantly reduce the thickness of the liquid film and therefore improves the heat exchange with the wall". Chaynane R. and al. [2] investigated "the effects of effective viscosity, permeability, and dimensionless thickness of the porous coating on flow and heat transfer refinement". "With analytical and numerical studies of a vapor thin film condensation problem in a porous medium Asbik M et al. [3] compared the Darcy-Brinkman (DB) model and the Darcy-Brinkman-Forchheimer (DBF) model. They showed in their results the influences of the Reynolds and Darcy numbers and the dimensionless thickness of the porous layer on the velocity and temperature profiles in the porous layer, the liquid film thickness, the local Nusselt number". "By producing a numerical model of the condensation of pure saturated vapor of the thin film type in forced convection on a wall covered with porous material, Ndiaye M. and al. [4-7] exposed the influences of the thermal conductivity ratio, of the Froude, Reynolds, Prandtl, and Jacob numbers, of the dimensionless thickness of the porous layer and on the transfers in the porous medium and in the liquid phase". Patil A. A. and al. [8] showed that "the daily distillate production increases up to 43% by coupling the surface condenser and vacuum pump due to the increase in evaporation and condensation rate by presenting a distillation system of solar water with a combination of surface condenser and vacuum pump". "By proposing a numerical study A. Nasr and S. Al-Ghamdi [10] confirmed that the presence of the porous layer improves the performance of heat and mass transfer at the liquid-gas interface during the evaporation of the film liquid in free convection". Abdelaziz Nasr [11] showed "numerically that the performance of heat and mass transfer at the liquid-gas interface during the condensation of the liquid film is improved by the presence of the porous layer by studying a numerical model of improvement of the liquid heat and mass transfer film condensation by covering a porous layer on one of the vertical channel plates". Charef A. et al. [12] presented "a numerical study of liquid film condensation from vapour-gas mixtures of HFC refrigerants inside a vertical tube and concluded that the condensation of R152a-air corresponds to accumulated condensation m_{cd} and a higher local heat transfer coefficient h_T compared to R134a-air under the same conditions". Mosaad M. E-S. et al. [13] presented "a model of a regular process of condensation of a laminar film on a vertical wall with the rear face cooled by free convection in a porous medium saturated with fluid. They found a difference compared to the Nusselt type solution (absence of the porous medium)". "In a thermal desalination process, which is based on the phase change phenomenon, Charef A. et al. [14] proposed a model on liquid film condensation and showed that increasing the temperature difference between the inlet and the wall improves condensation (and therefore the thickness of the condensate film), the radius of the tube and non-condensable gas are also relevant factors to improve the efficiency of thermal desalination". Sellami K.et al [15] found that "the evaporative cooler is more efficient for high porosity and thick porous media, with up to 23% improvement for high porosity". Ndiaye P.T. et al. [16-18] presented "numerical models of thin film type condensation in forced convection of pure saturated steam in a channel whose walls are covered with a porous material. They studied the influence of the Reynolds, Prandtl and Jacob numbers, the aspect ratio on the longitudinal velocity and the temperature, the heat transfer rate at the interface of the porous medium and the liquid film (Nusselt number local) and on condensation and its sensitivity via liquid film thickness and length of entry". Ndiaye G. and al. [19-20] proposing on "numerical studies of the condensation in forced convection of a laminar film of a pure and saturated vapor on a porous plate inclined towards the vertical, examined the effects of different parameters such as: the inclination, effective viscosity and dimensionless thermal conductivity, Reynolds and Prandtl numbers on flow and heat transfer".

In this work, we want to numerically study the effect of the ratio of thermal conductivity on the heat transfer and on the performance of the condensation. To properly carry out this project, we will first proceed to the mathematical formulation with the descriptions of the simplifying hypotheses, the equations of the problem and the boundary conditions. The hydrodynamic and thermal equations of the boundary layer will be used in the pure liquid and the generalized Darcy-Brinkman-Forchheimer (DBF) equations in the porous medium. Then announce the numerical resolution method and finally evaluate the influence of the ratio of thermal conductivity on the velocity and temperature profiles in porous medium and pure liquid , the thickness of the liquid film, the local Nusselt number (the transfer rate of heat) and lengths of entry (which characterizes condensation).

2. MATHEMATICAL FORMULATION

2.1 Physical Model and Assumptions

We present the modeling of flows and transfers during thin film vapor condensation in porous media. To investigate this resulting condensation phenomenon, we consider the physical model (channel) and coordinate system shown in Fig. 1.

This channel is formed by two flat plates, vertical and parallel, the internal walls of which are covered with a porous material of permeability *K* and porosity *ε*, of thickness *H*. These plates are spaced apart by a distance *2A*, of length *L* and are maintained at a temperature T_w lower than that of a saturated vapor T_S which flows at a uniform speed U0 at the channel inlet.

This shows the condensation of pure vapor on the porous medium and the presence of three zones: (1) the porous medium saturated with the liquid (2) the film of condensate (3) pure saturated vapor. The condensate film flows under the effect of the forces of gravity and viscous friction. The Cartesian coordinates *(x, y)* and the velocity components *(u, v)* in the porous medium and the pure liquid in the frame are associated with the model (channel).

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.

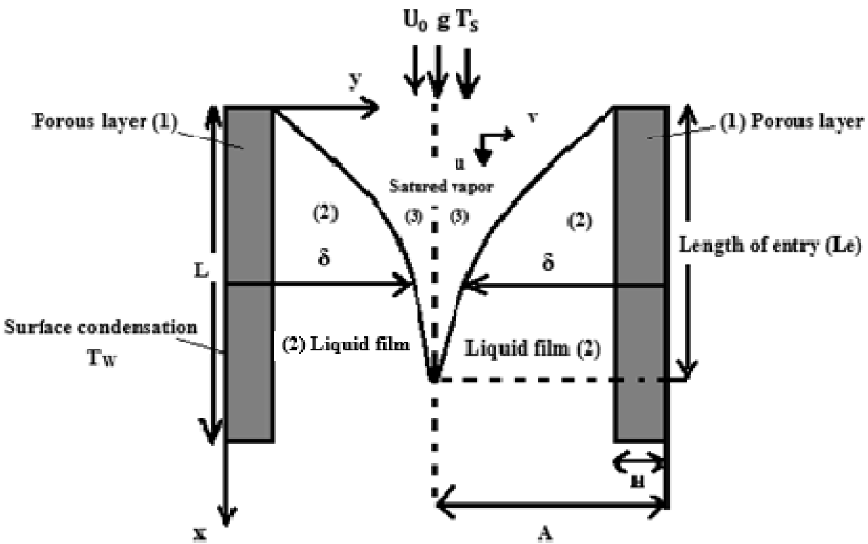

Fig. 1. Geometry of the physical model and coordinate system

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:

$$x^* = \frac{x}{L} \quad (1)$$

$$y^* = \frac{y}{A} \quad (2)$$

$$H^* = \frac{H}{A} \quad (3)$$

$$\delta^* = \frac{\delta}{A} \quad (4)$$

$$u^* = \frac{u}{U_0} \quad (5)$$

$$v^* = \frac{v}{U_0} \quad (6)$$

$$\theta = \frac{T - T_W}{T_S - T_W} \quad (7)$$

$$\lambda^* = \frac{\lambda_l}{\lambda_{eff}} \quad (8)$$

$$\nu^* = \frac{\nu_{eff}}{\nu_l} \quad (9)$$

The dimensionless physical domain (x*,y*) thus obtained presents a curvilinear interface (liquid/pure vapor interface), we will transform it into a rectangular domain (***X***,***η***) in which the boundary and the interfaces (porous medium interfaces /pure liquid and pure liquid/vapour) will be identified by lines of constant coordinates. The following change (homotopic transformation) of variable is made:

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

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 interface is parametrized by the straight line with coordinate *η = 1* and the pure liquid / vapor interface is parametrized by the straight line with coordinate *η = 2*.

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

Porous layer or porous medium: $0 \le \eta \le 1$

The continuity equation or equation of conservation of mass is:

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

The X-momentum equation is:

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

The equation of heat or energy equation is:

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

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

The continuity equation or equation of conservation of mass 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 X-momentum equation 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 or energy equation is:

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

$$=\left(\frac{L}{A}\right)^2\frac{1}{\mathrm{Re}\,\mathrm{Pr}\left(\delta^*(X)-H^*\right)^2}\frac{\partial^2\theta_l}{\partial\eta^2}$$

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

Conditions at the entry: $X=0$

$$u_p^*(0,\eta)=1 \quad (17)$$

$$v_p^*(0,\eta)=0 \quad (18)$$

$$\theta_p(0,\eta)=1 \quad (19)$$

The boundary conditions at the wall are: $\eta=0$

$$u_p^*=v_p^*=0 \quad (20);$$

$$\theta_p=0 \quad (21)$$

$$u_p^*=v_p^*=0 \quad (22)$$

$$\theta_p=0 \quad (23)$$

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

$$u_l^*=u_p^* \quad (24)$$

$$\theta_l=\theta_p \quad (25)$$

The continuity of constraints give:

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

The continuity of thermal flows give:

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

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

$$\theta_l=1 \quad (28)$$

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

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 relation below:

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

With:

The mass flow rate:

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

(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:

$$\mathrm{Re}=\frac{U_0L}{\nu_l}=\nu^*.\frac{U_0L}{\nu_{eff}} \quad (34)$$

Number of Darcy:

$$Da = \frac{A^2}{K} \tag{35}$$

Number of Prandtl:

$$\Pr = \frac{\nu_l}{\alpha_l} = \frac{\mu_l C_{p_l}}{\lambda_l} \tag{36}$$

Number of Peclet:

$$Pe = \text{Re}.\Pr \tag{37}$$

Number of Peclet modified:

$$Pe_{eff} = \lambda^* Pe = \lambda^* \text{Re}.\Pr \tag{38}$$

Number of de Prandtl modified:

$$\Pr_{eff} = \lambda^* \Pr \tag{39}$$

2.3 Determination of Local Nusselt Number

The **local Nusselt number** is the gradient in temperature at the interface porous medium and liquid film.

$$\boldsymbol{Nu} = \frac{1}{\delta^*(X) - \boldsymbol{H}} . \frac{\partial \boldsymbol{\theta}_l}{\partial \boldsymbol{\eta}} \bigg]_{\eta=1} \tag{40}$$

2.4 Determination of the Lengths of Entry (Le)

Already defined by Ndiaye P. T. et al. [16-18], the length of entry (Le) is a length from the channel inlet to the meeting of the two liquid films.

It is the length on the channel beyond which the boundaries layers conditions are not applicable any more

$$\delta(x = Le) = A \tag{41}$$

In dimensionless form and a rectangular domain, it is:

$$\delta^*(X = Le^*) = 1 \tag{42}$$

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. A numerical resolution is essential for not simplified cases and thus for ours.

3. NUMERICAL METHODOLOGY

It comprises two parts: a mesh part and a discretization part. We considered the mesh of the numerical domain uniform in the longitudinal and transverse directions and for the discretization an implicit finite difference method. The advection terms are discretized respectively with a backward off-center scheme. The terms and diffusion are discretized with a centered scheme. This will make possible the main diagonals of the most dominant matrices possible (for more stability). The systems of coupled algebraic equations thus obtained will be solved numerically thanks to a double sweep method combined with an iterative scheme of sub-relaxation line by line of the Gauss-Seidel type, in general faster than several other algorithms (such as the Jacobi's algorithm). Under-relaxation coefficients have been empirically defined to guarantee the non-growth of calculation errors during the iterative process. This will make our patterns or system more stable and convergent. All the results from the numerical simulation were obtained from a FORTRAN code.

4. RESULTS AND DISCUSSION

4.1 Results of Method Validation

The results of our calculation model have been validated in comparison with the work of Ndiaye M. and al. [4-7]. Since his model is only valid for values of the Reynolds number lower than 7, we made the comparison by taking low Reynolds numbers (*Re=2*) and low permeability (high Darcy number, $Da=10^{12}$). We took $\Delta X = 0.0001$ and $\Delta\eta = 0.02$, for the study of the sensitivity of the mesh in order to answer at the same time to the speed, the precision and the convergence of the calculations.

We have a good concordance with Ndiaye M. and al. [4-7], which will allow us to validate our model.

4.2 Results and Discussions

In this study, we show the effect of the ratio of thermal conductivity on the longitudinal velocity, the temperature, the film thickness, the thermal transfer rate at the porous medium/liquid film interface (local Nusselt number) and finally along the length of entry. We remind that the ratio of thermal conductivity is the ratio of the thermal conductivity of the liquid film on the thermal conductivity of the porous material.

The results from the numerical simulations are related to: $\varepsilon = 0.4;\ F = 0.55;\ \upsilon^* = 1;\ Da = 10^9;\ Fr = 10^{-4}$ The curve in Fig. 2 shows that when the ratio of thermal conductivity increases, the longitudinal velocity increases ($\boldsymbol{Re} = 10\ ;\boldsymbol{Pr} = 1$) and for Fig. 3 the variations in the ratio of thermal conductivity are accompanied by a small variation in the longitudinal velocity ($\boldsymbol{Re} = 50\ ;\boldsymbol{Pr} = 7$). In general we have a variation of the longitudinal velocity as a function of the thermal conductivity ratio only for low values of the Peclet number ($Pe = \mathrm{Re}.Pr$). The flow is characterized by the Peclet number which indicates the relative importance of convection and diffusion (thermal conduction), diffusion is strongly dominant when the Peclet number is low (Fig. 2, *Pe=10*). When it is high (Fig. 3, *Pe=350*) convection is strongly dominant and the velocity is not influenced by the variation of the ratio of thermal conductivity.

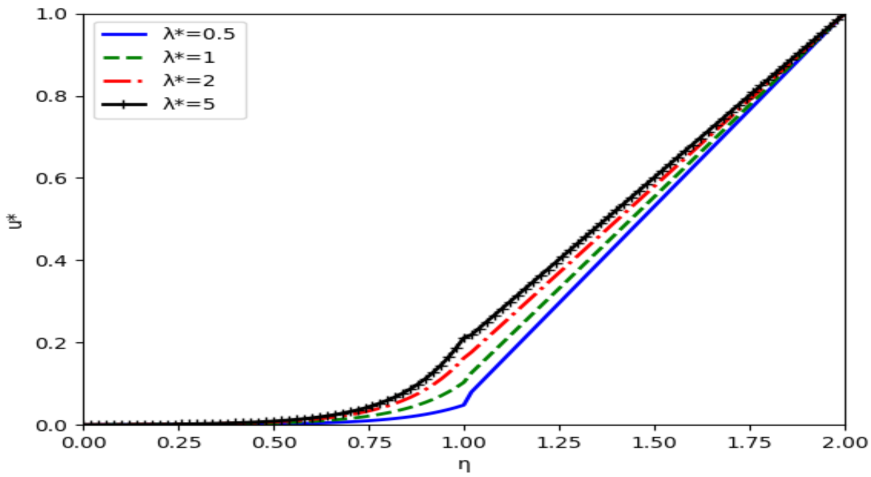

Fig. 2. Variation of the longitudinal velocity as a function of the ordinate η for different values of λ* à la position X=0.05
Re=25 ; Fr=10^{-4} ; Pr=1; H=2.10^{-4}; Ja=10^{-8}; v*=1 ; Da=10^{9}; L/A=10*

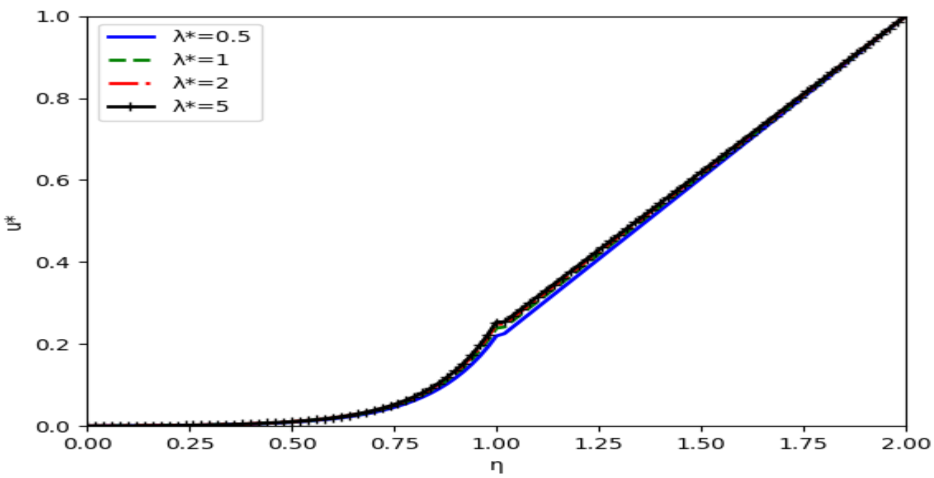

Fig. 3. Variation of the longitudinal velocity as a function of the ordinate η for different values of λ* à la position X=0.05
Re=50 ; Fr=10^{-4} ; Pr=7; H=2.10^{-4}; Ja=10^{-8}; v*=1 ; Da=10^{9}; L/A=10*

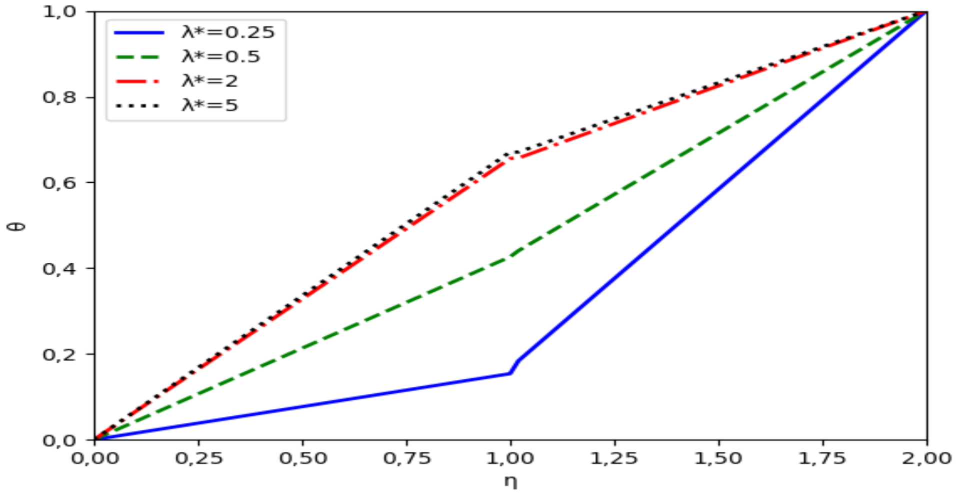

Fig. 4. Variation of the longitudinal temperature as a function of the ordinate η for different values of λ*

Re=100 ; Fr=10^{-4} ; Pr=5; H=2.10^{-4}; Ja=10^{-8}; v*=1 ; Da=10^{9}; L/A=100*

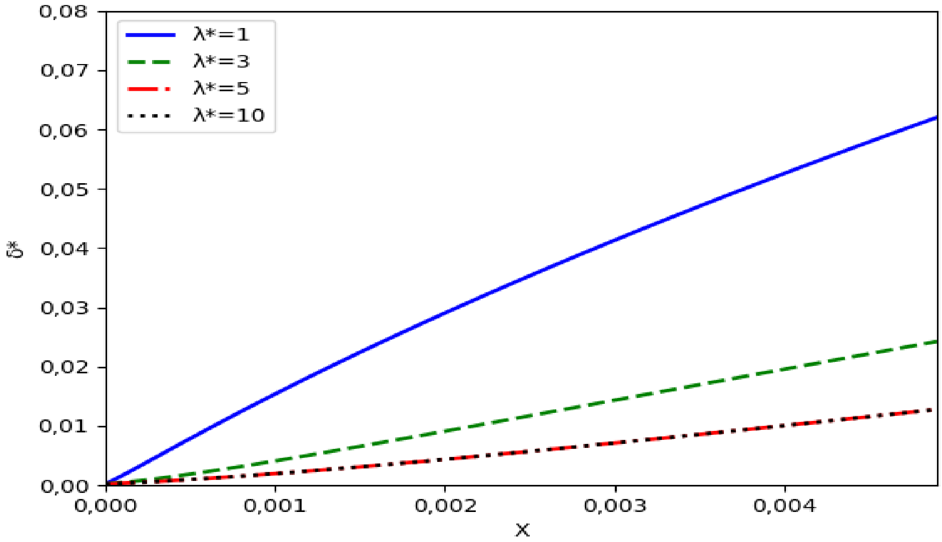

Fig. 5. Variation of the thickness of the liquid film as a function of the abscissa X for different values of ***λ****

Re=100 ; Fr=10^{-4} ; Pr=5; H=2.10^{-4}; Ja=10^{-8}; v*=1 ; Da=10^{9}; L/A=100*

For Fig. 4 when the ratio of thermal conductivity increases, the temperature increases even when the Peclet number is high (*Pe=500*). The thickness of the liquid film being much greater than that of the porous material, hence the conductive transfers in the liquid film are much more dominant, most of the transfers take place in the liquid film. Large values of the thermal conductivity ratio correspond to a highly conductive liquid phase.

With Fig. 5 we found that the dimensionless thickness of the liquid film decreases when the ratio of thermal conductivity increases (Fig. 5). By reducing the thermal conductivity of the liquid film compared to that of the porous material, the transfer and the heat exchange (conductive) are lowered, which allows better contact with the cold plate and condensation is favored.

We study the variation of the Nusselt number, heat transfer rate at the porous medium/pure liquid interface and the ratio between convective heat transfer and conductive transfer as a function of the abscissa X (Fig. 6). At the entrance to the channel, the value of the Nusselt number is very high, then rapidly decreases along the porous wall to reach very low asymptotic values. Conduction gradually increases along the porous wall on convective heat transfer. This justifies the decrease in the Nusselt number as a function of the abscissa X. We note the increase in the ratio of thermal conductivity (increase in the thermal conductivity of the liquid film compared to that of the porous material) leads to an increase in the Nusselt number (Fig. 6), even when the Peclet number is high (*Pe=125*). Their growth is consistent with an increasingly conductive environment. (This increases the heat exchanges at the interface between the porous medium and the pure liquid.)

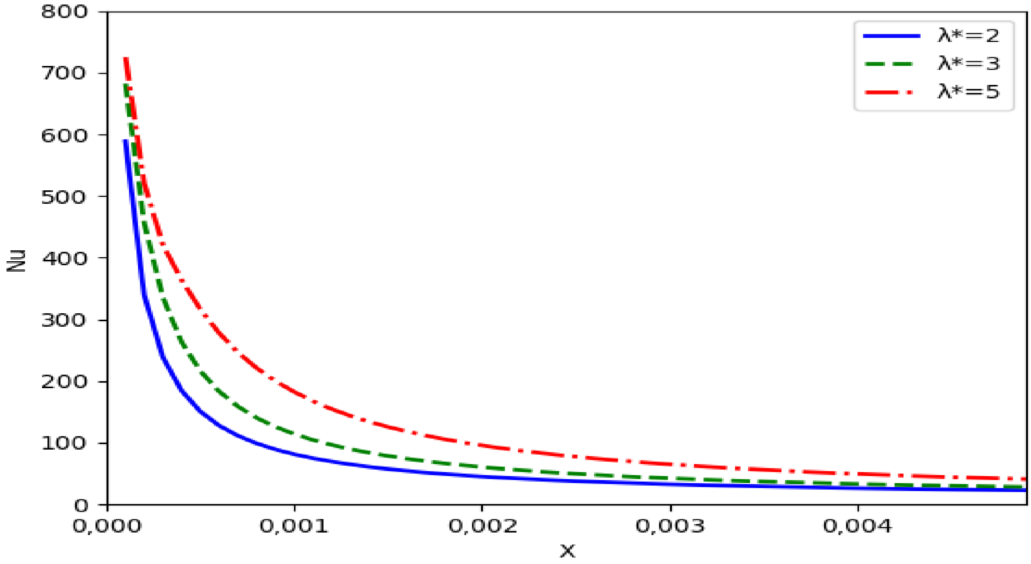

Fig. 6. Variation du nombre de Nusselt en fonction de l'abscisse X pour différentes valeurs de λ*
Re=25 ; Fr=10^{-4} ; Pr=5; H=2.10^{-4}; Ja=10^{-8}; v*=1 ; Da=10^{9}; L/A=100*

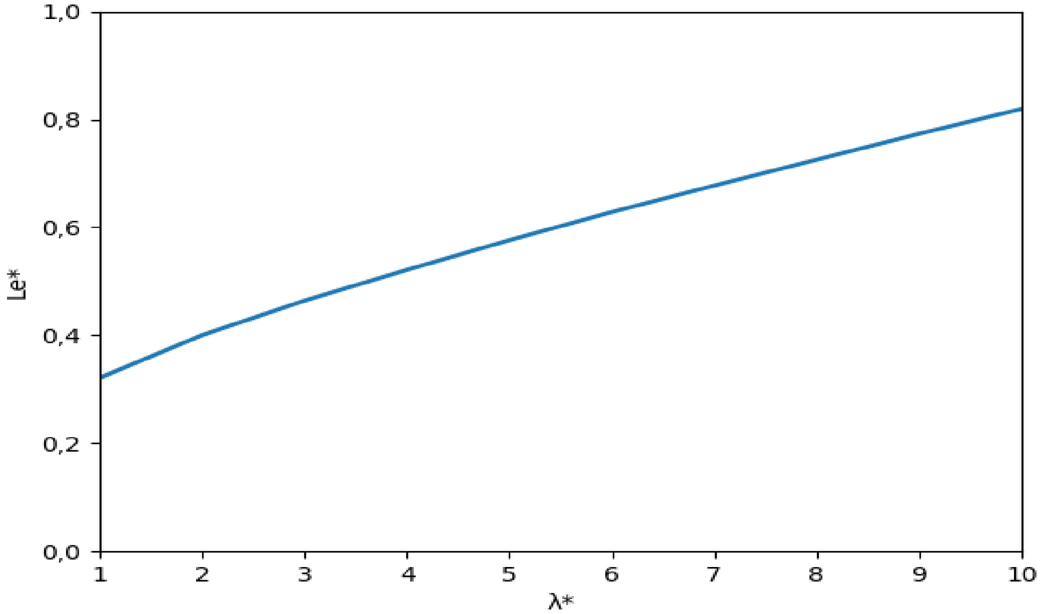

Fig. 7. variation de la longueur d'entrée adimensionnelle (Le^*) en fonction de λ*
Re=50 ; Fr=10^{-4} ; Pr=5; H=2.10^{-4}; Ja=10^{-8}; v*=1 ; Da=10^{9}; L/A=100*

The Fig. 7 is the variation of the dimensionless length of entry as a function of the ratio of thermal conductivity (increase in the thermal conductivity of the liquid film compared to that of the porous material). It provides information on the sensitivity of the condensation to the variation of the ratio of thermal conductivity. The variations of the dimensionless length of entry (Fig. 7) confirm those of the dimensionless thickness of the liquid film (Fig. 5).

A reduction thickness of liquid film produces an increase length of entry. From where the variations in the dimensionless length of entry (Fig. 7) confirm those the dimensionless thickness of liquid film (Fig. 5), i.e. the increase in the ratio of thermal conductivity decrease the thickness of liquid film and increase the length of entry. This increase of length of entry is almost linear. The sensitivity of condensation to variations in the ratio of thermal conductivity is constant, whatever its value. The ratio of thermal conductivity is a very decisive physical parameter to properly examine the performance of condensation.

5. CONCLUSION

We have proposed a numerical modeling of the effect of the ratio of thermal conductivity on the thin film condensation in forced convection in a canal whose walls are covered with a porous material.

Using the generalized Darcy-Brinkman-Forchheimer (DBF) equations in the porous medium and the hydrodynamic and thermal boundary layer equations in the pure liquid, we used a finite difference method to discretize the latter rendered dimensionless, homotopically transformed into a new rectangular base. The advection and the diffusion terms are discretized with respectively a backward-centered scheme and a centered scheme. Our results, compared to those of Ndiaye M. and al. [4-7] have been validated.

We analyzed the influence of the ratio of thermal conductivity on the longitudinal velocity, the temperature, the film thickness, the thermal transfer rate at the porous medium/liquid film interface (local Nusselt number) and finally on the length of entry. We have a variation of the longitudinal velocity as a function of the ratio of thermal conductivity only for low values of the Peclet number. When the ratio of thermal conductivity increases, corresponding to an increasingly conductive medium, the longitudinal velocity, the temperature and the Nusselt number increase (even when the Peclet number is high for the temperature and the Nusselt number, so the thermal field). While the thickness of the liquid film decreases (disadvantaged condensation) and leads to an increase in the length of entry. This increase in the length of entry as a function of the ratio of thermal conductivity is almost linear. The sensitivity of condensation to variations in the ratio of thermal conductivity is constant, whatever its value. The ratio of thermal conductivity is a very decisive and predictable physical quantity to properly examine the performance of condensation.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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