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Effect of Physical and Geometrical Parameters on Nuselt Number

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Abstract

The results of this study show that the Nusselt number decreases with the increase of the dimensionless thermal conductivity and the Prandtl, Froude numbers. The increased Reynolds, Jacob numbers and the dimensionless thickness of porous layer leads to an increase the Nusselt number.

Keywords

Condensation, Forced Convection, Vertical Wall, Nusselt Number

1. Introduction

Numerous studies have been carried out using: numerical experimentation (numerical simulation) and/or practical (laboratory experiment), and theoretical developments.

Pranab Kumar Mondal and al. [1] study in their paper deals with the analytical investigation for the limiting value of the Nusselt number, including the effect of viscous dissipation on heat transfer for a laminar shear driven flow between two infinite parallel plates, where the bottom plate is fixed and the top plate is moving in an axial direction at a constant speed. The study concentrates on hydro-dynamically fully developed flow of a Newtonian fluid of constant properties without considering the axial conduction in the fluid. To investigate the effect of viscous dissipation on heat transfer by defining the limiting Nusselt number, plates are kept at constant unequal temperatures. Expressions in its forms for the limiting Nusselt numbers as a function of the Brinkman number and asymmetry parameter are evaluated. Focus is given to the viscous dissipative effect due to the shear produced by the movable top plate over and above the viscous dissipation due to the internal fluid friction. The interactive effects of the Brinkman number and the degree of asymmetry on the limiting Nusselt number are investigated analytically. Specific to the cases considered for this study, the appearance of the point of singularities due to the variation of the Nusselt number with the Brinkman number is observed, and discussion has been made considering the energy balance.

Tong Bou Chang and al. [2] have studied an analytical investigation realized into the problem of steady film wise condensation flow over the outside surface of a horizontal tube embedded in a porous medium with suction at the tube surface. As in classical film condensation problems, an assumption is made that the condensate and vapor layers meet at a common boundary rather than being separated by an intermediary two-phase zone. Furthermore, it is assumed that the condensate film has constant properties and conforms to Darcy's law within the porous medium. By introducing an effective suction function to represent the effect of the wall suction on the thickness of the liquid film, both the local condensate film thickness and the local Nusselt number are derived using a simple numerical shooting method. The analytical results indicate that the mean Nusselt number depends on the Darcy number, the Jacob number, the Rayleigh number and the suction parameter. Furthermore, it is found that the local Nusselt number has a maximum value at the upper surface of the horizontal tube and reduces toward zero at the lower surface as a result of the finite thickness of the condensate layer.

Ndiaye M. and al. [3] present a numerical investigation of a laminar forced thin film condensation of a saturated vapor along a vertical porous layer. Our study is limited in the liquid medium. To obtain a numerical solution, these problems must be discretized by transforming the differential equations and algebraic equations linearized by a discretization method before solved by direct or iterative methods. We analyze the influence of Prandtl and Froude numbers transfers in the liquid phase.

Louahlia H. and al. [4] have developed forced convection condensation of pure vapour flowing between two horizontal parallel plates with the bottom plate cooled is analysed numerically. The coupled boundary layer equations for the two phases are solved using an implicite finite difference procedure. The pressure gradient, shear stress, inertia and enthalpy convection terms, and turbulence in the two phases are retained in this analysis. The influence of the temperature difference and inclination of the plate on the condensate film thikness are also reported. The results of the present calculations for the condensation of R12, R152a and R113 are represented by a non-dimensional equation.

Chaynane R. and al. [5] have analyzed their side effect on the inclination of the condensation in a forced convection laminar pure saturated steam on a porous plate film. The Darcy-Brinkman model is used to describe the flow in the porous medium, while the classical equations of the boundary layer have been exploited in the pure liquid neglecting the inertia terms and convection enthalpy. The problem was solved by analytical and numerical means. Results are mainly presented as the dimensionless thickness of the liquid film, profiles of speed, temperature and heat transfer coefficients represented by the Nusselt number. The results obtained were compared with those of experimental Renken and al. The effects of various influencing parameters such as the angle, the effective viscosity, Reynolds number, the dimensionless thickness of the porous substrate and the dimensionless thermal conductivity on the dimensionless flow and heat transfer are analyzed.

Asbik M. and al. [6] presented a problem of forced convection condensation in a thin porous layer is considered. The flow in the porous region is described by the Darcy-Brinkman-Forchheimer model (DBF) while classical boundary layer equations without inertia and enthalpie terms are used in the pure condensate region. In order to resolve this problem, an analytical method is proposed. Then, analytical solutions for the flow velocity, temperature distributions and for the local Nusselt number are obtained. The results are essentially presented in the form of the velocity and temperature profiles within the porous layer, the dimensionless film thickness and the heat transfer represented by the local Nusselt number. The comparison of the (DBF) model and the Darcy-Brinkman (DB) one is carried out. The effects of the effective viscosity (Reynolds number Re_K), permeability (Darcy number Da) and dimensionless thickness of porous coating H^* on the flow and the heat transfer enhancement are also documented.

Kuznetsov A. V. [7] present analytical study of fluid flow during forced convection in a composite channel partly filled with a Brinkman-Forchheimer porous medium, Flow, Turbulence and Combustion. In his paper, the problem of fully developed forced convection in a parallel-plate channel partly filled with a homogeneous porous material is considered. The porous material is attached to the walls of the channel, while the center of the channel is occupied by clear fluid. The flow in the porous material is described by a nonlinear Brinkman-Forchheimer-extended Darcy equation. Utilizing the boundary-layer approach, analytical solutions for the flow velocity, the temperature distribution, as well as for the Nusselt number are obtained. Dependence of the Nusselt number on several parameters of the problem is extensively investigated.

Char M. I. and al. [8] have studied the conjugate film condensation and natural convection along the vertical plate between a saturated vapor porous medium and a fluid-saturated porous medium. The solution takes into consideration the effect of heat conduction along the plate. The governing equations along with their corresponding boundary conditions for film condensation and natural convection are first cast into a dimensionless form by a no similar transformation, and the resulting equations are then solved by the cubic spline collocation method. The primary parameters studied include the thermal resistance ratio of film to plate A, the thermal resistance ratio of natural convection to film B, and the Jacob number Ja of the sub cooling degree in the film. The effects of these dimensionless parameters on the plate temperature distribution and the local heat transfer rate on both sides of the plate are discussed in detail. In addition, the interesting engineering results regarding the overall heat transfer rate from the film condensation side to the natural convection side are also illustrated.

Xuehu Ma and al [9] extend their work to the case of film condensation on a vertical porous coated plate is investigated numerically for a porous/fluid composite system based on the dispersion effect in the porous coating. The mathematical model improves the conventional matching conditions by considering the stress force at the porous coating interface. The numerical results show the effects of the porous coating thickness, the effective thermal conductivity and the permeability on condensate film thickness and local Nusselt number. The predicted average Nusselt number has similar tendencies to experimental results reported in literature, such that using a thin porous coating would be a viable alternative for enhancing the condensation heat transfer.

Merouan and al. [10] studied a numerical study of laminar film condensation by forced convection of steam-air mixtures in a vertical tube is presented. The internal face of the tube wall is coated with a thin porous layer. A set of complete boundary layer equations governing the conservation of momentum, heat and mass is used to describe the transfers in the liquid film and the mixture. The flow field in the porous medium is described by the Darcy-Brinkman-Forchheimer model. These three phases are related with the continuity of velocity, shear stress, temperature, heat and mass flux at the interfaces. The dimensionless transfer equations are discretized using an implicit finite difference scheme. The liquid film thickness is determined by solving the liquid mass balance equation. Results were obtained for a saturated steam-air mixture. Profiles of velocity, temperature in the three media and vapor mass fraction in the mixture are presented. The effects of main properties of the porous layer such as thickness and Darcy number are highlighted. Additionally, the influence of the inlet Reynolds number and inlet vapor mass fraction of the mixture on the evolution of heat flux and condensate flow rate is also investigated.

Mohammed Sammouda and al. [11] studied in their paper the phenomena of double-diffusive convection in a cylindrical enclosure filled with a porous medium saturated with a Newtonian fluid. The porosity of the porous media is variable from the walls to the bulk and follows an exponential law. The enclosure is heated from below and two mass concentrations C0, C1 are applied respectively to the surfaces (bottom, top) of the cylinder. The vertical walls are rigid, impermeable and adiabatic. An extended law of Brinkman-Forchheimer (EBFD) describes the fluid flow occurring in the porous layers by using the Boussinesq approximation. The mass concentration follows the basic Fick law, and the model chosen to describe the heat transfer is based on the approximation of one temperature. The heat and mass flow are controlled by the dimensionless numbers such as Rayleigh, Lewis, Darcy, Prandtl number that appear by dimensionless of the system of equations. The finites differences method is used to solve numerically the problem. The study focused on the effect of the dimensionless numbers on the concentrations and on the rate of heat transfers profiles in the overall Nusselt number, while considering the porosity uniform or non-uniform. The numerical code developed can be used for various industrial processes involving the phenomenon of natural convection.

We propose in the present work to study numerically the forced convection along a vertical plane wall covered with a homogeneous porous material saturated by a pure liquid by adopting approximations full Darcy-Brinkman model for the porous medium and those of thermal and hydrodynamic boundary layers for the pure liquid.

2. Mathematical Formulation

2.1. Physical Model and Assumptions

This article is devoted to the mathematical modeling that is to say; first, we will give a physical model and simplifying then mathematically formulate the problem and then transform the equations assumptions.

We consider the phenomenon of thin film condensation on a vertical plate, covered with a thick porous material H, permeability K and porosity ε (**Figure 1**). The vertical flat plate of length L is positioned in a flow of saturated steam and pure, longitudinal velocity U_0. The vapor condenses on the wall of the plate

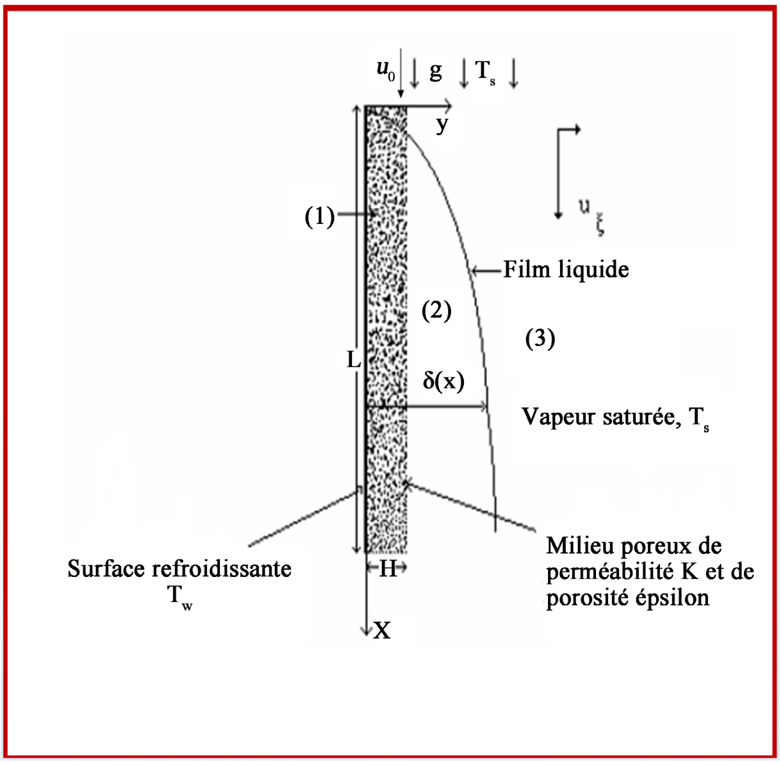

Figure 1. Geometry of the physical model and coordinate system.

maintained at lower than the saturation vapor temperature T_s. The film of condensate flows under the influence of gravity and viscous drag forces.

Our field of study includes three (3) areas.

The area (1) is the porous medium saturated.

The zone (2) corresponds to the liquid film.é

The region (3) is on the saturated vapor.

Let (x, y) and (u, v) respectively be the Cartesian coordinates and the components of the velocity in the porous medium and of the liquid in the frame associated with the model.

The flow is laminar and two-dimensional and the regime is permanent, the porous matrix is homogeneous and isotropic and is in local equilibrium with the condensate that is in the form of a thin film, the thermo-physical properties of fluids and those of the porous matrix are assumed to be constant. Forces are negligible viscous dissipation, saturating fluid, the porous medium is Newtonian and incompressible, the dynamic and kinematic viscosity, the actual porous material is equal to those of the film of the condensate, the liquid-vapor interface is in thermodynamic equilibrium and the shear stress is assumed to be negligible. The variation of the transverse pressure is not taken into account.

The Darcy-Brinkman model is used to describe the flow in the porous medium, while the classical equations of the boundary layer have been used to describe the transfer in the pure liquid medium, the coupling conditions at the interface porous medium/pure liquid, are essentially expressed by the continuity of the shear stresses and heat flux densities.

We will give the equations of motion, heat in the two media (porous and liquid), the heat balance and mass flow without dimensions are expressed respectively by the following equations:

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

$$u_p^* \frac{\partial u_p^*}{\partial X} + \frac{v_p^*}{H^*}\frac{\partial u_p^*}{\partial \eta} = -\frac{\varepsilon^2}{Re_K} u_p^* + \frac{\varepsilon^2}{Re_K}\left(\frac{1}{H^{*2}}\frac{\partial^2 u_p^*}{\partial \eta^2}\right) + \frac{\varepsilon^2}{Fr_K} \tag{1}$$

$$u_p^* \frac{\partial \theta_p}{\partial X} + \frac{v_p^*}{H^*}\frac{\partial \theta_P}{\partial \eta} = \frac{1}{Pr Re_K H^{*2}}\frac{\partial^2 \theta_P}{\partial \eta^2} \tag{2}$$

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

$$u_l^*\left[\frac{\partial u_l^*}{\partial X} - \frac{\eta - 1}{\delta^* - H^*}\frac{\mathrm{d}\delta^*}{\mathrm{d}X}\frac{\partial u_l^*}{\partial \eta}\right] + \frac{v_l^*}{\delta^* - H^*}\frac{\partial u_l^*}{\partial \eta} = \frac{1}{v^* Re_K\left(\delta^* - H^*\right)^2}\frac{\partial^2 u_l^*}{\partial \eta^2} + \left(1 - \frac{\rho_v}{\rho_l}\right)\frac{\left(\delta^* - H^*\right)^2}{Fr_K} \tag{3}$$

$$u_l^*\left[\frac{\partial \theta_l}{\partial X} - \frac{\eta - 1}{\delta^* - H^*}\frac{\mathrm{d}\delta^*}{\mathrm{d}X}\frac{\partial \theta_l}{\partial \eta}\right] + \frac{v^*{}_l}{\delta^* - H^*}\frac{\partial \theta_l}{\partial \eta} = \frac{1}{Re_K Pr\left(\delta^* - H^*\right)^2}\frac{\partial^2 \theta_l}{\partial \eta^2} \tag{4}$$

The heat balance

$$\frac{Ja}{(Pe)_{eff}}\frac{1}{H^*}\frac{\partial\theta_p}{\partial\eta}\bigg|_{\eta=0}=\frac{\mathrm{d}}{\mathrm{d}x^*}\left[H^*\int_0^1\left\{1+Ja\left(1-\theta_p\right)\right\}u_p^*\mathrm{d}\eta\right]+\frac{\mathrm{d}}{\mathrm{d}x^*}\left[H^*\int_0^1\left\{1+Ja\left(1-\theta_p\right)\right\}u_p^*\mathrm{d}\eta\right]+\frac{\mathrm{d}}{\mathrm{d}x^*}\left[\int_1^2\left(\delta^*-H^*\right)\left\{1+Ja\left(1-\theta_l\right)\right\}u_l^*\mathrm{d}\eta\right] \quad (5)$$

The mass flow rate

$$H^*\int_0^1 u_p^*\mathrm{d}\eta+\left(\delta^*-H^*\right)\int_1^2 u_l^*\mathrm{d}\eta=\frac{\rho_v}{\rho_l}\delta^* \quad (6)$$

2.2. Solution Procedure

The expressions of the partial derivatives involved in the equations are treated deducted from the Taylor expansion.

Since the wall and the boundary of the liquid film, the flow is imposed by the boundary conditions. The equations are discretized transfer by an implicit finite difference method. The mesh of the digital domain is considered uniform in the transverse and longitudinal directions. The terms of advection and diffusion are discretized respectively with a rear and centered upwind scheme. The coupled algebraic equations are solved numerically obtained through an iterative relaxation method line by line Gauss-Seidel.

Thus, the discretization in the field of study of the equations of energy and movement leads to the following algebraic equations:

$$c\cdot Int^*\left(i,j\right)=an\cdot Int^*\left(i-1,j\right)+am\cdot Int^*\left(i,j-1\right)+ap\cdot Int^*\left(i,j+1\right)+coef0 \quad (7)$$

$$2\le i\le im \quad \text{et} \quad 2\le j\le jm-1$$

with

$$c=\frac{coefx}{\Delta x}+\frac{coefe}{\Delta\eta}+2\frac{coefe2}{\Delta\eta^2} \quad (8)$$

$$an=\frac{coefx}{\Delta x} \quad (9)$$

$$am=\frac{coefe}{\Delta\eta}+\frac{coefe2}{\Delta\eta^2} \quad (10)$$

$$ap=\frac{coef2}{\Delta\eta^2} \quad (11)$$

Discretization of the equation of heat balance:

$$\delta^*\left(i\right)=\delta^*\left(i-1\right)+R_\delta \quad (12)$$

Coefficients of the discretization of the equation of heat balance:

$$R_\delta = Ja \cdot \Delta x \frac{H^*\left(flux1 - flux10\right)}{\left(1+Ja\right)R_\rho} + Ja \cdot \Delta x \frac{+\left(\delta^* - H^*\right)\left(flux2 - flux20\right) + \frac{1}{Pe * H^* * \Delta\eta}\left(\theta\left(i,j\right) - \theta\left(i,j-1\right)\right)}{\left(1+Ja\right)R_\rho} \quad (13)$$

$$R_\rho = \frac{\rho_v}{\rho_l} \quad (14)$$

3. Results and Discussion

The results of our simulations are presented in the figures below.

3.1. Results

- Influence of dimensionless thermal conductivity on the Nusselt number.

The calculations were performed for the following values: $H^* = 99 \times 10^{-4}$, $Fr_K = 10^{-4}$, $Re_K = 45$, $Ja = 10^{-3}$, $Pr = 2$.

- Influence of Reynolds number on the Nusselt number.

The calculations were performed for the following values: $H^* = 99 \times 10^{-4}$, $Fr_K = 10^{-4}$, $Ja = 10^{-3}$, $Pr = 2$, $\lambda^* = 0.1$.

- Influence of Prandtl number on the Nusselt number.

The calculations were performed for the following values: $H^* = 99\times\ 10^{-4,}$ $Fr_K = 10^{-4}$, $Ja = 10^{-3}$, $Re_K = 45$, $\lambda^* = 0.1$.

- Influence of thickness of the porous layer on the Nusselt number.

The calculations were performed for the following values: $Fr_K = 99.10^{-4}$, $Ja = 10^{-3}$, $Pr = 2$, $Re_K = 45$, $\lambda^* = 0.1$.

- Influence of Froude number on the Nusselt number.

The calculations were performed for the following values: $H^* = 99.10^{-4}$, $Pr = 2$, $Ja = 10^{-3}$, $Re_K = 45$, $\lambda^* = 0.1$.

- Influence of Jacob number on the Nusselt number.

The calculations were performed for the following values: $H^* = 99.10^{-4}$, $Fr_K = 10^{-4}$, $Pr = 2$, $Re_K = 45$, $\lambda^* = 0.1$.

3.2. Discussion

The Nusselt number (Nu) is a dimensionless number used in heat transfer operations. It represents the ratio between the total heat transfer by conduction and transfer: the case of a transfer between two perfect solids. The presence of convection (eg due to the displacement of a fluid), the heat transfer will be made primarily by fluid displacement and will result to reach the Nusselt number to $+\infty$.

Figure 2, Figure 3 and Figure 4 show the changes of the dimensionless thermal conductivity and the Prandtl, Froude numbers in function of the abscissa *x*. It (Nusselt number) decreases with the increase of the Prandtl, Froude numbers and dimensionless thermal conductivity causing the exchange of heat decreases at the interface between the porous medium and the liquid Figure 2, Figure 3 and Figure 4. The Nusselt is the ratio between the total heat transfer and transfer by conduction. In our case (Figure 2, Figure 3 and Figure 4) conduction predominates over total transfer which explains the decrease in the Nusselt number as a function of the abscissa *x*.

The increased Reynolds, Jacob numbers and the dimensionless thickness of porous layer increases the Nusselt number. The Reynolds, Jacob numbers and the dimensionless thickness of the porous layer to promote heat transfer Figure 5, Figure 6 and Figure 7. The total heat transfer is dominant conduction which causes an increase in the Nusselt number as a function of the abscissa *x*.

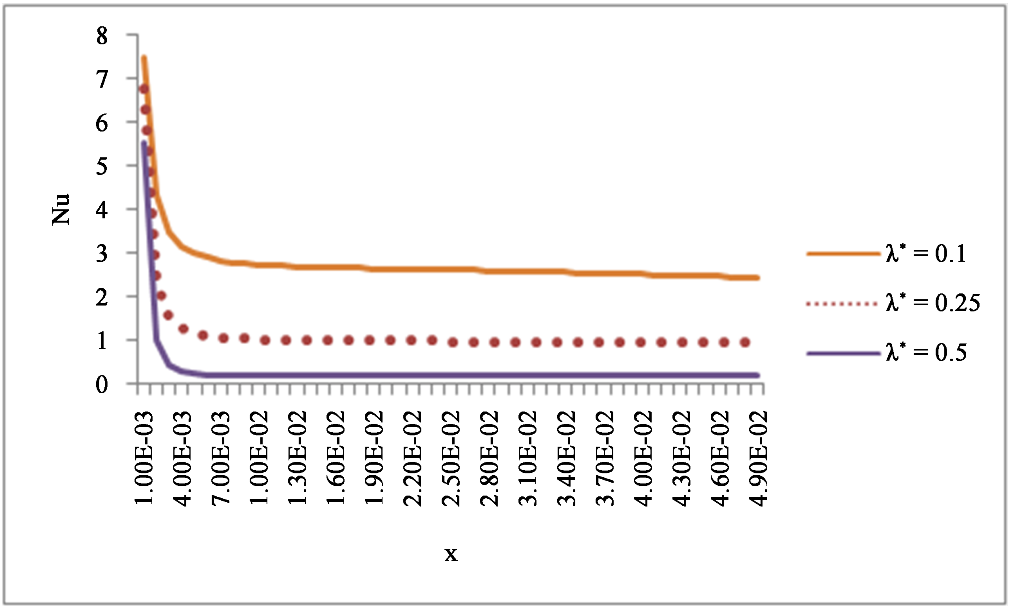

Figure 2. Variation of Nusselt number as a function of the abscissa *x* for different values of dimensionless thermal conductivity.

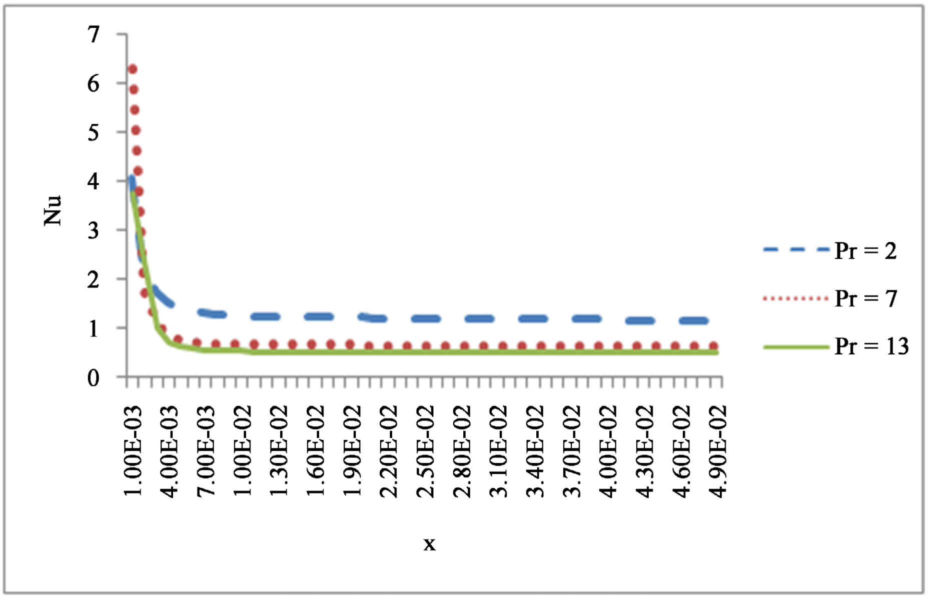

Figure 3. Variation of Nusselt number as a function of the abscissa *x* for different values of *Pr*.

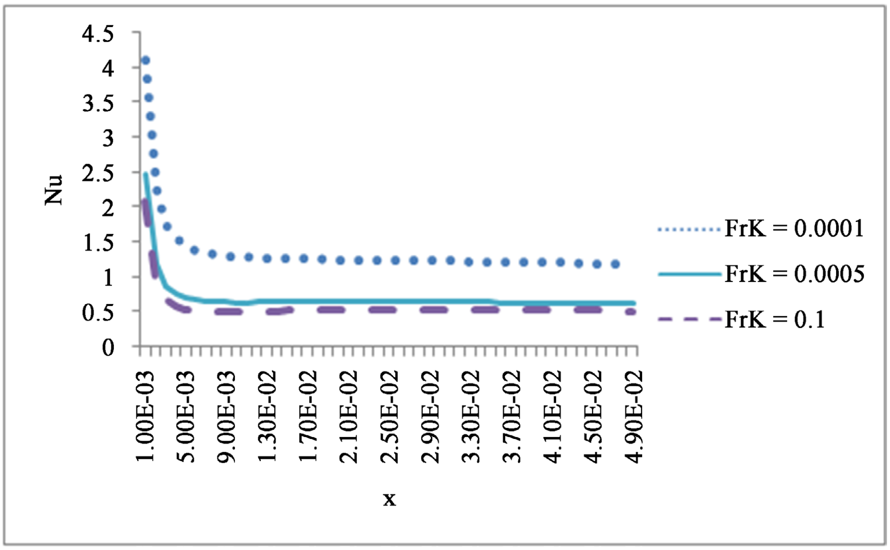

Figure 4. Variation of Nusselt number as a function of the abscissa x for different values of Fr_K.

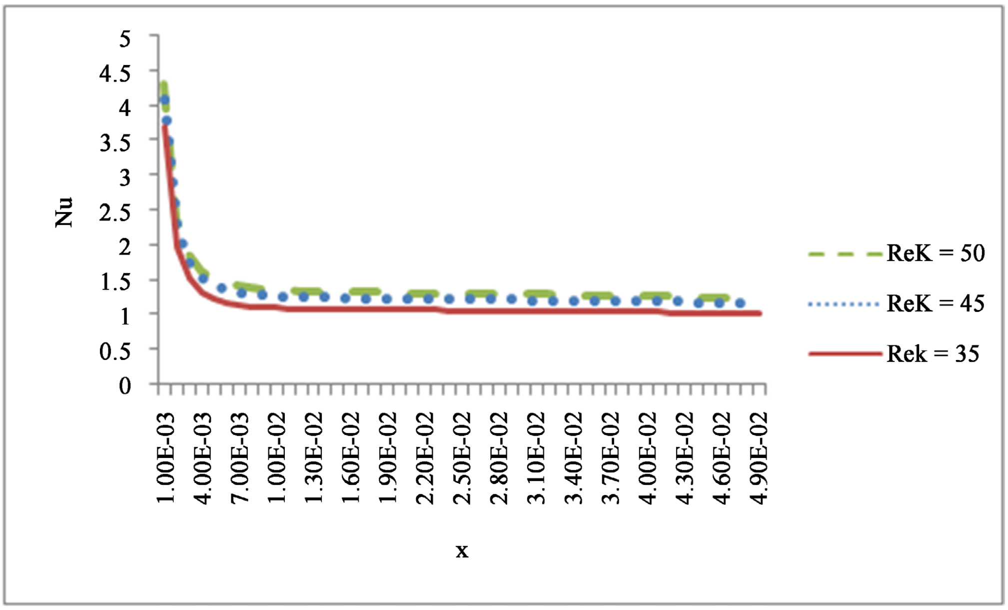

Figure 5. Variation of Nusselt number as a function of the abscissa x for different values of Re_K.

4. Conclusions

We have in this work mathematically and numerically modeled the heat and momentum transfers in a porous medium and a liquid phase surmounted by its saturated vapor. Based on simplifying assumptions and the geometry of the domain, the transfer equations are projected onto the coordinate axes. In order to generalize the problem and to weight the different terms of the equations, we have adimensionalized them, thus generating groupings that compare the different effects. As the physical domain has a curvilinear boundary, the use of a homotopic transformation allowed us to reduce the study domain to a rectangular domain in which the top of the boundary layer is located by a constant coordinate

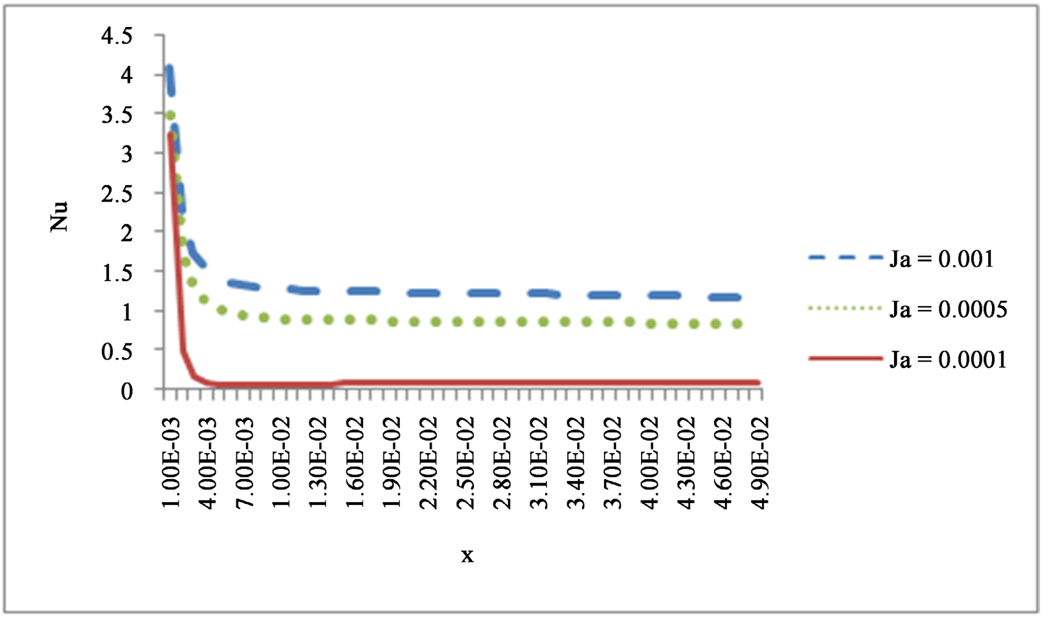

Figure 6. Variation of Nusselt number as a function of the abscissa *x* for different values of *Ja.*

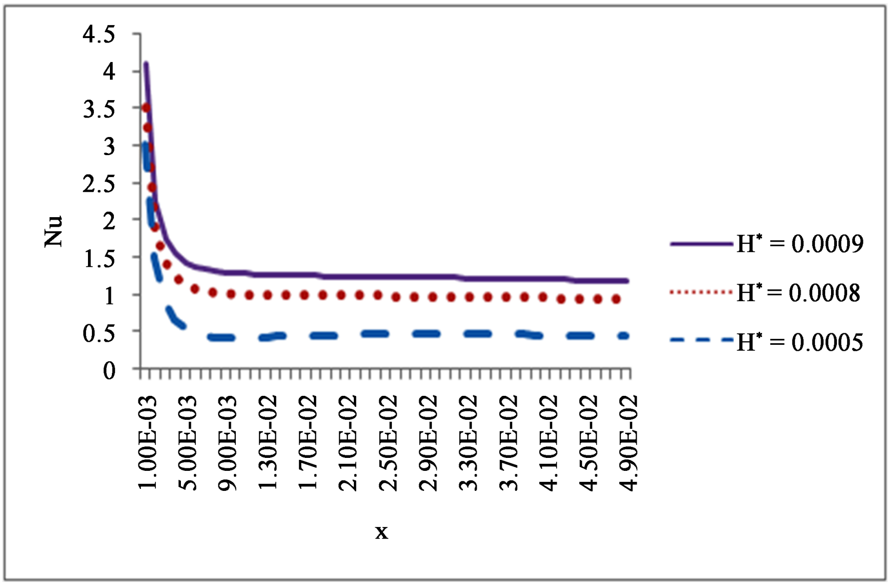

Figure 7. Variation of the Nusselt number depending on the abscissa *x* for different values of H^*.

line. To solve the system of nonlinear and coupled equations between them, an implicit finite difference scheme is used for its relative ease of implementation. The algebraic equations thus obtained are solved using the double-scan method combined with an iterative line-by-line relaxation scheme.

After validating our computer code by comparing our results with those from the literature and testing the space steps, we analyzed the influences of certain parameters on the hydrodynamic and thermal fields as well as on the parietal quantities. We have analyzed the influence of the problem parameters on the Nusselt number in the two media (porous and liquid). The Nusselt number also decreases with increasing Prandtl and Froude numbers and dimensionless thermal conductivity, resulting in weak heat exchange at the interface between the two media. The increase in the Reynolds and Jacob numbers and the thickness of the dimensionless porous layer causes an increase in the Nusselt number, which explains an increase in heat exchange at the interface between the porous medium and the liquid. The logical continuation of this work consists in validating the mathematical and numerical models by comparing the results from our simulations with those found by experience. However, this study deserves to be continued and improved by lifting, for example, certain simplifying hypotheses, by changing geometry, by imposing more physical thermal boundary conditions of the Robin type, for example, by analyzing the effect of inclination. The influences of the other parameters that we have set must also be analysed.

Finally, it is important to study this problem in a turbulent regime because in a descending flow, we can consider that the regime is practically turbulent.

The results obtained will allow us to better quantify their effects and their importance in thin-film flows.

Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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Nomenclature

Greek Symbols

α : Thermal diffusivity, $m^2 \cdot s^{-1}$

δ : Thickness of the condensate, m

ε : Porosity

θ : Temperature dimensionless

λ : Thermal conductivity, $W \cdot m^{-1} \cdot K^{-1}$

μ : Dynamic viscosity, $kg \cdot m^{-1} \cdot s^{-1}$

ν : Kinematic viscosity, $m^2 \cdot s^{-1}$

ρ : Density kg/m^3

Latin letters:

c_p: Specific heat, $J \cdot kg^{-1} \cdot K^{-1}$

Fr_K: Froude number based on $\sqrt{K}$

g: Acceleration of gravity, $m \cdot s^{-2}$

H: Thickness of the porous layer, m

h_{fg}: Heat of evaporation, $J \cdot kg^{-1}$

Ja: Jacob number

K: Hydraulic conductivity or permeability, m^2

L: Length of the plate, m

Pe: Peclet number

Pr: Prandtl number

Re_K: Reynolds number based on $\sqrt{K}$

T: Temperature, K

U_0: Velocity of free fluid (steam), m/s

u: Velocity along x, m/s

u_r: Velocity reference, m/s

v: Velocity along y, m/s

x, y: Cartesian coordinates along x and y, m

Indices Exhibitor

eff: Efficiency value

i: Porous substrate interface/pure liquid

l: Liquid

p: Porous

s: Saturation

v: Steam

w: Wall

*: Dimensionless quantity