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Numerical Study of Thin Film Condensation in Forced Convection of Saturated Vapor on a Vertical Wall Covered with a Porous Material

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Abstract

The study of forced convection in a porous medium has aroused and still arouses today the interest of many scientists and industrialists. A considerable amount of work has been undertaken following the discovery of the phenomenon. Solving a standard problem of forced convection in porous media comes down to predicting the temperature and velocity fields as well as the intensity of the flow as a function of the various parameters of the problem. A numerical study of the condensation in forced convection of a pure and saturated vapor on a vertical wall covered with a porous material is presented. The transfers in the porous medium and the liquid film are described respectively by the Darcy-Brinkman model and the classical boundary layer equations. The dimensionless equations are solved by an implicit finite difference method and the iterative Gauss-Seidel method. Our study makes it possible to examine and highlight the role of parameters such as: the Froude number and the thickness of the porous layer on the speed and the temperature in the porous medium. Given the objective of our study, the presentation of velocity and temperature profiles is limited in the porous medium. The results show that the Froude number does not influence the thermal field. The temperature increases with an increase in the thickness of the dimensionless porous layer. The decrease in the Froude number leads to an increase in the hydrodynamic field.

Keywords

Condensation, Implicit Finite Differences, Thin Film, Porous Material, Darcy-Brinkman Model, Vertical Wall

1. Introduction

The study of condensation in porous media is a subject of growing interest due to its importance in many technological fields such as heat exchangers, fuel cells, geothermal energy, and energy storage. It also finds its application in certain technological problems presenting a two-phase zone, called pasty zone, represented by a saturated porous medium and a binary fluid. Many studies have been carried out using: numerical experimentation (numerical simulation) and/or practice (laboratory experiment), as well as theoretical developments.

Renken KJ *et al.* [1], present a theoretical investigation of laminar film condensation along a solid impermeable surface coated with a porous material. This material may be a conductive metallic foam used to augment heat transfer, scale deposited over a long period of time on a surface or any relatively thin porous matrix layer. The dependence of the local heat transfer coefficient (represented by the Nusselt number) on the porous matrix parameters is presented and compared to the classical study of Nusselt. Representative results of the velocity and temperature distributions as well as the film thickness are also presented. It is shown that a conductive coating may yield a considerable heat transfer enhancement.

Renken KJ *et al.* [2], study a Film-condensation enhancement by a porous/fluid composite system is investigated numerically. The numerical simulations detail laminar film condensation from an inclined thin porous-layer coated plate that is immersed in pure saturated steam at atmospheric pressure. A mathematical model is formulated, and the numerical results are compared with Nusselt's theory and preliminary experimental data. The dependence of the average heat transfer coefficient (represented by the Nusselt number and the average heat flux) on the surface subcooling, the gravity field, and the thin porous-layer coating characteristics (thickness, effective thermal conductivity model, porous structure, etc.) is documented.

Renken KJ *et al.* [3], present the effect of thermal dispersion during laminar forced convection filmwise condensation within a thin porous/fluid composite system is examined numerically. The model simulates two-dimensional condensation within a very permeable and highly conductive thin porous-layer coated surface. The local volume-averaging technique is utilized to establish the energy equation and to account for the thermal dispersion effect. The Darcy-Brinkman-Forchheimer model is employed to describe the flow field in the porous layer while classical boundary layer equations are used in the pure condensate region. The numerical results, which detail the dependence of the heat transfer rate and temperature field on the governing parameters (e.g., Reynolds number, Rayleigh number, Darcy number, Prandtl number, thermal dispersion coefficient, as well as porous coating thickness and thermal conductivity ratio), are calculated using a finite difference scheme. It is found that due to the better mixing of the thermal dispersion effect, the heat transfer rate is greatly increased and the effect becomes more pronounced as the Reynolds number increases. The results of this study provide valuable fundamental predictions of enhanced film condensation that can be used in a number of practical thermal engineering applications.

Renken KJ *et al.* [4], present in this paper reports the experimental findings of forced convective condensation heat transfer on plates with a thin porous coating. The composite system consists of a relatively thin, highly conductive and permeable porous coating bonded to a cold isothermal condensing surface which is placed parallel to saturated steam flow. The porous coatings ranged in thickness from 0 to 254 μm and are used as a passive technique for heat transfer augmentation. The inclined and isothermal plates were exposed to bulk vapor velocities of 0.9 - 6.5 $m \cdot s^{-1}$ resulting in free-steam Reynolds numbers of 5.0×10^4 - 2.7×10^6. The forced convective heat transfer coefficients show a substantial heat transfer enhancement (250% increase) when compared to noncoated surface under the prescribed conditions. The experimental data exhibits as much as a 600% increase over previously measured free convection condensation results. The effect of subcooling temperature and plate inclination (vertical to horizontal) is also documented. The results of this study provide valuable fundamental predictions of condensation heat transfer that can be used in a number of thermal engineering applications which require heat transfer enhancement or retardation of an impermeable surface.

Asbik M. *et al.* [5] study 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.

Asbik M. *et al.* [6] An analytical solution to the problem of condensation by natural convection over a thin porous substrate attached to a cooled impermeable surface has been conducted to determine the velocity and temperature profiles within the porous layer, the dimensionless thickness film and the local Nusselt number. In the porous region, the Darcy–Brinkman-Forchheimer (DBF) model describes the flow and the thermal dispersion is taken into account in the energy equation. The classical boundary layer equations without inertia and enthalpyterms are used in the condensate region. It is found that due to the thermal dispersion effect, the increasing of heat transfer is significant. The comparison of the DBF model and the Darcy–Brinkman (DB) one is carried out.

Chaynane R. *et al.* [7] for their part, analyzed the effect of inclination on the condensation in forced convection of a laminar film of pure and saturated vapor on a porous plate. The Darcy-Brinkman model is used to describe the flow in the porous medium, while the classical boundary layer equations have been exploited in the pure liquid by neglecting the terms of inertia and enthalpy convection. The problem posed was solved analytically and numerically. The results are essentially presented in the form of the dimensionless thickness of the liquid film, the velocity and temperature profiles and the heat transfer coefficients represented by the Nusselt number. The results obtained were compared with the experimental results of Renken *et al.* The effects of various influential parameters such as: inclination, effective viscosity, Reynolds number, dimensionless thickness of the porous substrate and dimensionless thermal conductivity on flow and heat transfer are analyzed.

However, these authors in their studies have neglected the effects of inertia in the Darcy-Brinkman equations.

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

The objective of this study is to analyze the influence of the Froude number and the thickness of the porous layer on the velocity and temperature profiles.

2. Mathematical Approach

We are interested in the phenomenon of thin film condensation on a vertical plate, covered with a porous material of thickness H, permeability K and porosity ε (**Figure 1**). This flat vertical plate of length L is placed in a flow of pure and saturated steam, of longitudinal velocity U0. The steam condenses on the wall of the plate maintained at a temperature lower than that of saturation Ts of the steam. The condensate film flows under the effect of gravity and the forces of viscous friction.

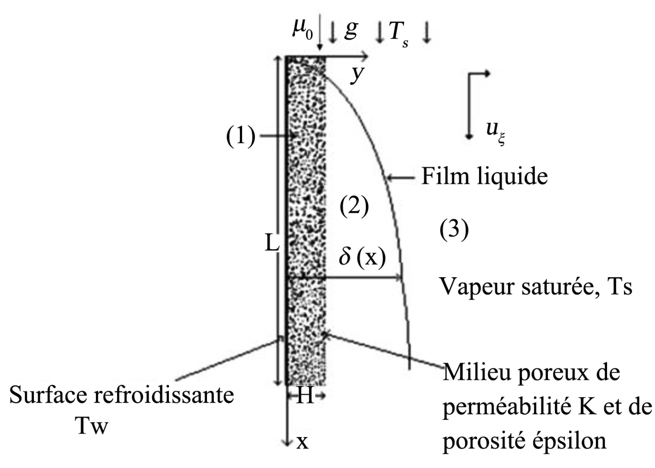

Figure 1. Geometry of the physical model.

Our field of study includes three areas. Zone (1) is the porous medium saturated by the liquid, zone (2) corresponds to the liquid film while zone (3) relates to the saturated vapor see **Figure 1**.

We accept the following simplifying assumptions:

- The generated flow is laminar and two-dimensional and the regime is permanent;
- The porous matrix is isotropic and homogeneous and is in local equilibrium with the condensate which is represented in the form of a thin film;
- The thermo-physical properties of the fluids and those of the porous matrix are assumed to be constant;
- The saturating fluid, the porous medium is Newtonian and incompressible;
- Work induced by viscous and pressure forces is negligible;
- The effective dynamic, kinematic viscosities of the porous material are equal to that of the film of the condensate;
- The liquid-vapor interface is in thermodynamic equilibrium and the shear stress is assumed to be negligible.

The equations which govern the transfers in the domains (1) and (2) defined above as well as the boundary conditions associated with them, have been dimensionless using the following variables and parameters [5],

The Dimensionless values are

$$H^* = \frac{H}{\sqrt{K}}, \quad x^* = \frac{x}{\sqrt{K}} \tag{1.a-b}$$

$$y^* = \frac{y}{\sqrt{K}}, \quad \theta_\xi = \frac{T_\xi - T_w}{T_s - T_w} \tag{2.a-b}$$

$$u^* = \frac{u_\xi}{u_r}, \quad u_r = \frac{K}{\nu_{eff}} g \tag{3.a-b}$$

$$\delta^* = \frac{\delta}{\sqrt{K}}, \quad \lambda^* = \frac{\lambda_l}{\lambda_{eff}}, \quad \mu^* = \frac{\mu_{eff}}{\mu_l} \tag{4.a-b-c}$$

In order to bring the physical domain which presents a curvilinear interface (liquid/vapor interface) into a rectangular domain, we perform the following change of variable:

$$X = x^* \tag{5.a}$$

$$\eta = coef \frac{y^*}{H^*} + (1 - coef)\left\{1 + \frac{y^* - H^*}{\delta^* - H^*}\right\} \tag{5.b}$$

With *coef* a coefficient equal to 1 if we are in the porous layer and 0 in the pure liquid.

Thus the plane (x^*, y^*) is transformed into a rectangular domain (X, η) and the porous medium/liquid and liquid/saturated vapor interfaces are respectively parametrized by the coordinate lines $\eta = 1$ and $\eta = 2$.

In the two media, the dimensionless equations for the conservation of mass, momentum and heat are written:

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

Conservation equation of mass in the porous medium

$$\frac{\partial u_p^*}{\partial X} + \frac{1}{H^*}\frac{\partial V_p^*}{\partial \eta} = 0 \tag{1}$$

Velocity equation in the porous medium

$$u_p^* \frac{\partial u_p^*}{\partial X} + \frac{V_p^*}{H^*}\frac{\partial u_p^*}{\partial \eta} = -\frac{\varepsilon^2}{R_{e_K}} u_p^* + \frac{\varepsilon^2}{R_{e_K}}\left(\frac{1}{H^{*2}}\frac{\partial^2 u_p^*}{\partial \eta^2}\right) + \frac{\varepsilon^2}{F_{r_K}} \tag{2}$$

Heat equation in the porous medium

$$u_p^* \frac{\partial \theta_p}{\partial X} + \frac{V_p^*}{H^*}\frac{\partial \theta_P}{\partial \eta} = \frac{1}{p_r R_{e_K} H^{*2}}\frac{\partial^2 \theta_P}{\partial \eta^2} \tag{3}$$

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

Conservation equation of mass in liquid

$$\left(\delta^* - H^*\right)\frac{\partial u_l^*}{\partial X} - (\eta - 1)\frac{\mathrm{d}\delta^*}{\mathrm{d}X}\frac{\partial u_l^*}{\partial \eta} + \frac{\partial V_l^*}{\partial \eta} = 0 \tag{4}$$

Equation of velocity in liquid

$$\begin{aligned} & u_l^*\left[\frac{\partial u_l^*}{\partial X} - \frac{\eta - 1}{\delta^* - H^*}\frac{\mathrm{d}\delta^*}{\mathrm{d}X}\frac{\partial u_l^*}{\partial \eta}\right] + \frac{V_l^*}{\delta^* - H^*}\frac{\partial u_l^*}{\partial \eta} \\ & = \frac{1}{v^* R_{e_K}\left(\delta^* - H^*\right)^2}\frac{\partial^2 u_l^*}{\partial \eta^2} + \left(1 - \frac{\rho_v}{\rho_l}\right)\frac{\left(\delta^* - H^*\right)^2}{Fr_K} \end{aligned} \tag{5}$$

Heat equation in the liquid

$$u_l^*\left[\frac{\partial \theta_l}{\partial X} - \frac{\eta - 1}{\delta^* - H^*}\frac{\mathrm{d}\delta^*}{\mathrm{d}X}\frac{\partial \theta_l}{\partial \eta}\right] + \frac{V_l^*}{\delta^* - H^*}\frac{\partial \theta_l}{\partial \eta} = \frac{1}{R_{e_K} P_r\left(\delta^* - H^*\right)^2}\frac{\partial^2 \theta_l}{\partial \eta^2} \tag{6}$$

To these equations, we associate the following dimensionless boundary conditions:

At the wall, $\eta = 1$

$$u_p^* = V_p^* = 0, \quad \theta_p = 0 \tag{6.a-b}$$

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

$$u_l^* = u_p^*, \quad \theta_l = \theta_p \tag{7.a-b}$$

$$\frac{1}{H^*}\frac{\partial u_p^*}{\partial \eta} = \frac{1}{\delta^* - H^*}\frac{\partial u_l^*}{\partial \eta} \tag{8.a}$$

$$\frac{1}{H^*}\frac{\partial \theta_p}{\partial \eta} = \frac{\lambda^*}{\delta^* - H^*}\frac{\partial \theta_l}{\partial \eta} \tag{8.b}$$

at the liquid/vapor interface, $\eta = 2$

$$\theta_l = 1, \quad \frac{\partial u_l^*}{\partial \eta} = 0 \tag{9.a-b}$$

Since the dimensionless velocity and temperature depend on the thickness of the liquid film δ^*, the dimensionless heat balance is expressed respectively by the following relationship:

$$\frac{Ja}{Pe_{eff}}\frac{1}{H^*}\frac{\partial\theta_p}{\partial\eta}\bigg|_{\eta=0} = \frac{\mathrm{d}}{\mathrm{d}x^*}\left[H^*\int_0^1\left\{1+Ja\left(1-\theta_p\right)\right\}u_p^*\mathrm{d}\eta\right] + \frac{\mathrm{d}}{\mathrm{d}x^*}\left[H^*\int_0^1\left\{1+Ja\left(1-\theta_p\right)\right\}u_p^*\mathrm{d}\eta\right] + \frac{\mathrm{d}}{\mathrm{d}x^*}\left[\int_1^2\left(\delta^*-H^*\right)\left\{1+Ja\left(1-\theta_l\right)\right\}u_l^*\mathrm{d}\eta\right] \quad (7)$$

With $Pe_{eff}=\lambda^* p_r R_{e_K}$

Mass flow:

$$H^*\int_0^1 u_p^*\mathrm{d}\eta+\left(\delta^*-H^*\right)\int_1^2 u_l^*\mathrm{d}\eta=\frac{\rho_v}{\rho_l}\delta^* \quad (8)$$

Dimensionless numbers are defined by the following relations:

$$P_r=\frac{\mu_l C_{P_l}}{\lambda_l}=\frac{\nu_l}{\alpha_l},\quad F_{r_K}=\frac{u_0^2}{g\sqrt{K}} \quad (10.\text{a-b})$$

$$R_{e_K}=\frac{u_0\sqrt{K}}{\nu_{eff}},\quad Ja=\frac{C_{P_l}\left(T_S-T_w\right)}{h_{fg}} \quad (11.\text{a-b})$$

3. Resolution Methodology

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:

$$c\cdot Int^*\left(i,j\right)=an\cdot Int^*\left(i-1,j\right)+am\cdot Int^*\left(i,j-1\right)+ap\cdot Int^*\left(i,j+1\right)+coef0 \quad (12)$$

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

with

$$c=\frac{coefx}{\Delta x}+\frac{coefe}{\Delta\eta}+2\frac{coefe2}{\Delta\eta^2} \quad (13)$$

$$an=\frac{coefx}{\Delta x} \quad (14)$$

$$am=\frac{coefe}{\Delta\eta}+\frac{coefe2}{\Delta\eta^2} \quad (15)$$

$$ap = \frac{coef\,2}{\Delta\eta^2} \tag{16}$$

Discretization of the equation of heat balance:

$$\delta^*(i) = \delta^*(i-1) + R_\delta \tag{17}$$

Coefficients of the discretization of the equation of heat balance:

$$\begin{aligned} R_\delta &= Ja \cdot \Delta x \frac{H^*(flux1 - flux10)}{(1+Ja)R_\rho} \\ &+ Ja \cdot \Delta x \frac{+(\delta^* - H^*)(flux2 - flux20) + \dfrac{1}{Pe * H^* * \Delta\eta}\left(\theta(i,j) - \theta(i,j-1)\right)}{(1+Ja)R_\rho} \end{aligned} \tag{18}$$

$$R_\rho = \frac{\rho_v}{\rho_l} \tag{19}$$

4. Results and Discussions

We show in this study the influence of parameters such as: the Froude number and the thickness of the porous layer on the velocity and longitudinal temperature profiles in the porous medium. The results from the numerical simulations relate to $\nu^* = 10^{-7}$, $\varepsilon = 0.4$.

The study of the sensitivity of the mesh led us to choose $\Delta\eta = 0.02$ and $\Delta x = 0.004$ constitutes a good compromise between a good precision and an acceptable calculation time. Δx and $\Delta\eta$ are the space steps along x and η. The convergence criterion in the iterative process is fixed at 10^{-6}.

Figure 2 shows that when the Froude number decreases the longitudinal velocity in the porous medium increases. Indeed, a decrease in the Froude number results in an increase in the permeability coefficient of the porous substance, so the medium becomes more and more permeable. For large Froude numbers ($F_{r_K} \geq 0.001$) the shapes of the longitudinal velocity curves are identical to those obtained by Asbik *et al.* [5] [6] and therefore it is legitimate to neglect the effects of inertia.

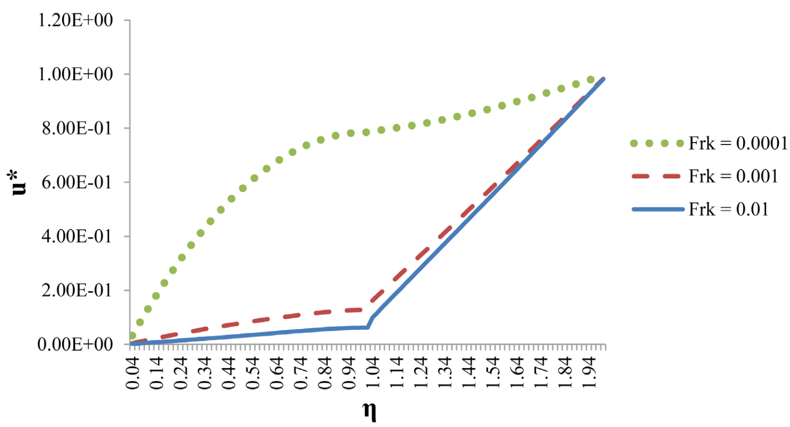

Figure 2. Variation of the longitudinal velocity in the porous medium as a function of the ordinate η for different values of Fr_K $Re_K = 45$, $H^* = 2 \times 10^{-3}$, $Ja = 10^{-3}$, $Pr = 2$, $\lambda^* = 2.9$.

On the other hand when $F_{r_K} \prec 0.001$, the medium being very permeable, the effects of inertia become preponderant and the hypothesis which consisted in neglecting the terms of advection and convection is no longer justified. The values of the longitudinal velocity increase with the increase in the thickness of the porous substance and its profile tends towards that of a flow in the boundary layer. So for low values of the thickness of the dimensionless porous layer the Darcy approximation can be used (**Figure 3**). **Figure 2** and **Figure 3** show that the effects of the thickness of the dimensionless porous layer and the Froude number are opposite.

The analysis of **Figure 4** shows that the thermal field is not influenced for Froude numbers equal to 0.01 and 0.001.

$F_{r_K} = 0.0001$, the dimensionless temperature decreases. The fact that the Froude number practically does not vary over the thermal field proves that the effects of the intrinsic permeability of the porous medium does not have a large effect on the temperature. When the thickness of the porous layer decreases, heat transfers are favored because the resistance to exchanges decreases. However, values of H* less than 0.0001 have no effect on the thermal field (**Figure 5**).

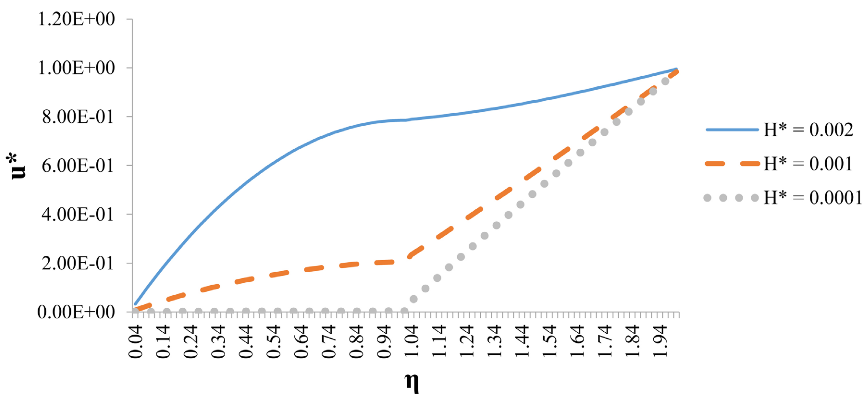

Figure 3. Variation of the longitudinal velocity in the porous medium as a function of the ordinate η for different values of H^* $Re_K = 45$ $Fr_K = 10^{-3}$, $Ja = 10^{-3}$, $Pr = 2$, $\lambda^* = 2.9$.

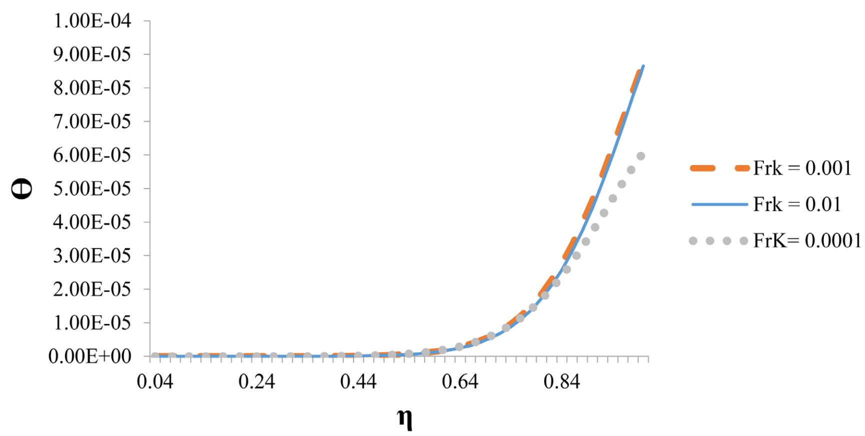

Figure 4. Temperature variation in the porous medium as a function of the ordinate η for different values of Fr_K $Re_K = 45$, $H^* = 2 \times 10^{-3}$, $Ja = 10^{-3}$, $Pr = 2$, $\lambda^* = 2.9$.

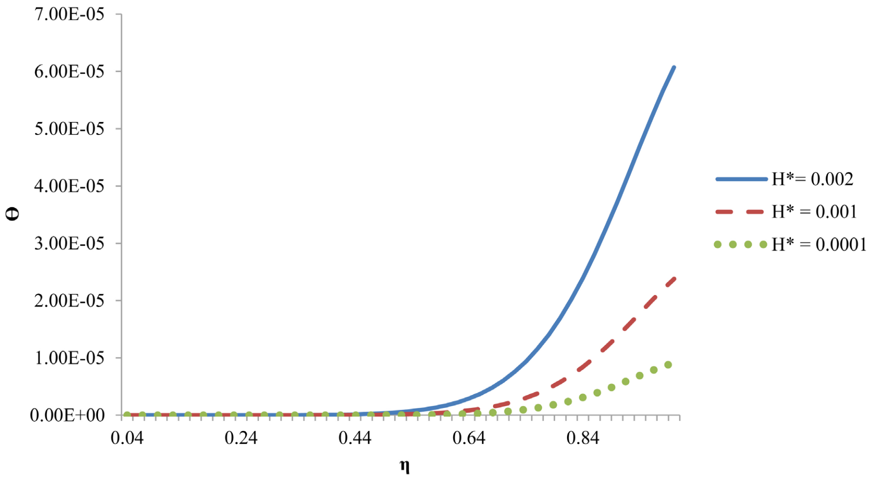

Figure 5. Temperature variation in the porous medium as a function of the ordinate η for different values of H^* $Re_K = 45$, $Fr_K = 10^{-3}$, $Ja = 10^{-3}$, $Pr = 2$, $\lambda^* = 2.9$.

5. Conclusions

We have studied the forced convection condensation of a saturated vapor on a vertical wall covered with porous material. The equations were solved using the back-offset implicit finite difference method.

We also analyzed the influence of the Froude number and the thickness of the dimensionless porous layer on the velocity and temperature profiles in the porous medium. The results show that the Froude number does not influence the thermal field.

The temperature increases with an increase in the thickness of the dimensionless porous layer.

The decrease in the Froude number leads to an increase in the hydrodynamic field. It appears from our study that the Froude number and the dimensionless thickness of the porous layer have opposite effects on the dynamic field. We are in the presence of a fluvial regime since the Froude number is less than 1 ($F_{r_K} \prec 1$). The increase in the thickness of the dimensionless porous layer promotes heat transfer. We compared our results with those of Asbik *et al.* [5] [6] whose terms of inertia are neglected, we note an acceptable correspondence.

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Conflicts of Interest

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

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Nomenclature

α: Thermal diffusivity, $m^2 \cdot s^{-1}$
δ: Thickness of condensate, m
ε: Porosity
η: Dimensionless coordinate in the transverse direction
θ: Temperature dimensionless
λ: Thermal conductivity, $W \cdot m^{-1} \cdot K^{-1}$
μ: Viscosity dynamic, $kg \cdot m^{-1} \cdot s^{-1}$
ν: Viscosité cinématique, $m^2 \cdot s^{-1}$
ρ: Density, $kg \cdot m^{-3}$
Cp: Specific heat, $J \cdot kg^{-1} \cdot K^{-1}$
Da: Darcy's number
F: Forchheimer coefficient
Fr: Froude number
g: Gravitational acceleration, $m \cdot s^{-2}$
H: Thickness of the porous layer, m
h_{fg}: Enthalpy of evaporation, $J \cdot s \cdot kg^{-1}$
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, $m \cdot s^{-1}$
U_0: Velocity of the free fluid, $m \cdot s^{-1}$
v: Velocity along y, $m \cdot s^{-1}$
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 (vapeur)
w: Wall
*: Dimensionless quantity