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Effects of Prandtl and Jacob Numbers and Dimensionless Thermal Conductivity on Velocity Profiles in Media (Porous and Liquid)

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Abstract

In this present study, we analyzed the effects of Prandtl and Jacob numbers and dimensionless thermal conductivity on the velocity profiles in media (porous and liquid). The transfers in the porous medium and the liquid film are described respectively by the improved Wooding model and the classical boundary layer equations. The mesh of the digital domain is considered uniform in the transverse and longitudinal directions. The advection and diffusion terms are discretized with a back-centered and centered scheme respectively. The coupled systems of algebraic equations thus obtained are solved numerically using an iterative line-by-line relaxation method of the Gauss-Seidel type. The results show that the parameters relating to the thermal problem (the dimensionless thermal conductivity, the Prandtl (Pr) and Jacob (Ja) numbers) have no influence on the dimensionless speed, although the thermal and hydrodynamic problems are coupled. Via the heat balance equation. The results obtained show that the parameters relating to the thermal problem have no influence on the dimensionless speed, although the thermal and hydrodynamic problems are coupled via the heat balance equation. So, at first approximation with the chosen constants, we can solve the hydrodynamic problem independently of the thermal problem.

Keywords

Condensation, Implicit Finite Difference, Thin Film, Porous Material, Model Wooding, Vertical Wall

1. Introduction

The study of condensation in porous media is a topic of growing interest due to its importance in many technological fields, such as heat exchangers, fuel cells, geothermal energy, energy storage, etc. …It also finds its application in certain technological problems presenting a two-phase zone, called a pasty zone, represented by a saturated porous medium and a binary fluid. Numerous studies have been carried out using: digital experimentation (numerical simulation) and/or practical experimentation (laboratory experiment), as well as theoretical developments. We plan to study the effects of Prandtl and Jacob numbers and dimensionless thermal conductivity on velocity profiles in liquid and porous media.

NDIAYE M. *et al.* [1] proposed a model for the numerical study of the condensation of pure saturated steam of the thin film type in forced convection on a wall covered with porous material. The transfers in the porous medium and the liquid film are described respectively by the Darcy-Brinkman model and the classical equations of the boundary layer. The dimensionless equations are solved by an implicit finite difference method combined with an iterative Gauss-Seidel method. They analyzed the influences of Prandtl and Froude numbers on transfers in the liquid phase. The parameters relating to the thermal problem (Prandtl number) have no influence on the dimensionless speed although the thermal and hydrodynamic problems are coupled via the heat balance equation.

The study showed that heat transfer increases when $Pr \leq 0.2$, however, for values $Pr > 0.2$, there is no change in heat transfer. However, the values of $Fr_K \geq 0.001$ have no effect on the hydrodynamic field, for $Fr_K < 0.001$, the speed increases significantly in the liquid film and the effects of inertia can no longer be ignored (a reduction in the Froude number leads to an increase in the permeability K of the porous medium, thus the substance becomes more and more permeable).

NDIAYE M. *et al.* [2] also proposed a model for the numerical study of the condensation of pure saturated steam of the thin film type in forced convection on a wall covered with porous material and they analyzed the influences of the Reynolds and Jacob numbers, the thickness dimensions of the porous layer and the dimensionless heat transfer conductivity on transfers in the liquid phase. They used the same resolutions as before. Their results showed that the thermal field is not influenced by dynamic problems despite their coupling at the interface. The parameters linked to the thermal problem (thermal conductivity ratio and Jacob number) have no effect on the longitudinal speed. It also appears that the Reynolds number has little influence on the thermal field. The longitudinal speed increases with the thickness of the porous layer. The decrease in the dimensionless thickness of the porous layer and the thermal conductivity ratio and an increase in the Jacob number leads to an increase in temperature.

NDIAYE M. *et al.* [3], using the same numerical model, analyzed the influence of Reynolds numbers and the dimensionless thickness of the porous layer on transfers in the liquid phase and the porous medium as well as the thickness of the liquid film. The thickness of the condensate film is determined by the heat balance equation at the liquid-vapor interface which is solved using an iterative Gauss-Seidel type procedure. Their results showed that the Reynolds number does not influence the thermal field, the temperature increases with the decrease in the thickness of the dimensionless porous layer. The increase in the Reynolds number and the dimensionless thickness of the porous layer leads to an increase in the longitudinal speed and a reduction in the thickness of the liquid film. Increases in Reynolds numbers and dimensionless thickness of the porous layer have adverse effects on condensation (decrease in liquid film thickness). The study also showed that inertia effects can no longer be ignored when the Reynolds number calculated from the permeability coefficient is greater than 7.

PT Ndiaye *et al.* [4] [5] presented the numerical study of the condensation in thin layers in forced convection of a pure saturated vapor in a channel whose walls are covered with a porous material. The generalized Darcy-Brinkman-Forchheimer (DBF) model is used to describe the flow in the porous medium while the classical boundary layer equations have been exploited in the case of a pure liquid. The dimensionless equations are solved by an implicit finite difference method and the Gauss-Seidel iterative method. This study made it possible to examine and highlight the role of parameters such as the Reynolds number and the Prandtl number on the longitudinal speed and on the temperature in the porous medium and in the pure liquid, and the transfer rate heat (local Nusselt number). The increase in Reynolds and Prandtl numbers' results in an increase in longitudinal velocity and temperature. The tangents of the velocity curves at the porous interface on the medium side are smaller than those obtained on the liquid side with low Reynolds and Prandtl numbers. We also note that increasing the Reynolds and Prandtl numbers improves the thermal performance of the interface. The Prandtl number improves heat exchanges at the interface of the porous medium and the liquid film.

Dharma Rao *et al.* [6] developed a model for the study of water vapor condensation in the presence of humid air by forced laminar convection in a vertical tube. The model is developed by considering the classical boundary layer equations but neglecting the convective terms in the liquid phase equations. The temperature at the gas-liquid interface and the condensate thickness are calculated by considering the thermal and mass balances at the interface. The equations are solved numerically by the finite difference method. Calculations were carried out for a wide range of parameters to determine the influence of fluid inlet conditions on transfers. The authors show that the Nusselt and liquid Reynolds numbers increase with temperature, relative humidity and the Reynolds number of the gas mixture at the tube inlet, but decrease as the mixture inlet pressure increases. However, the effects of these parameters on the Nusselt number of the gas phase are less significant.

Maheshwari *et al.* [7] developed a theoretical model to determine the local heat transfer coefficient during vapor condensation in the presence of a non-condensable gas, flowing in a vertical tube for a wide range of the Reynolds number of the mixture. The equations for transfer in the mixture were developed using the analogy relationships between heat and mass transfer. The flow of the liquid film is governed by the Nusselt model modified to take into account the undulatory character 26 of the film at the interface. The results show that the thermal resistance of the layer of non-condensable gases at the mixture-liquid interface is greater than that of the condensate, at low intake Reynolds number of the mixture. When this increases, the thermal resistance of the liquid film becomes preponderant.

Asbik M. *et al.* [8]-[10] were interested in a semi analytical and numerical study of condensation by forced convection laminar film on a vertical plate covered by a porous layer. These authors showed the significant contribution of the porous substrate and the thermal effect of the dispersion on a flat wall causing an increase in the transfer rate.

Chaynane R. *et al.* [11] Also considered by the condensation of a forced convection laminar sublayer of a pure on a saturated porous plate is inclined relative to the vertical steam. 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 in the liquid. 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 *et 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 H * dimensionless thermal conductivity, the heat transfer and flow are illustrated.

K. Renken J. *et al.* [12]-[15] investigated the condensation laminar thin film on a vertical surface with a porous coating. The model simulates the condensation two dimensions within a very thin and very highly permeable porous conductive layer. Model Darcy-Brinkman-Forchheimer is used to describe the flow field in the porous layer while the classical equations of the boundary layer are used in the region of pure condensate. Numerical results, which describe in detail the dependence of the rate of heat transfer of the temperature field and the parameters (e.g., the Reynolds number, and the Prandtl Darcy, the thermal dispersion coefficient and the thickness the porous coating and the rate of thermal conductivity) are calculated using a finite difference scheme.

They also developed experiments on condensation in forced thin, porous coatings convection. This study presents the results of experiments of heat transfer by forced convection and condensation on plates with a thin porous layer. The composite system consists of a very thin porous conductive and permeable material, adhered to a cold condensation surface isotherm moves parallel to the flow of saturated steam. The results of this study provide significant predictions of the heat transfer of condensation that can be used in a number of thermal engineering applications that require improved heat transfer.

2. Mathematical Formulation

2.1. Physical Model

The research carried out in this article generally focuses on the modeling of flows and transfers in porous and liquid media. To carry out a simple formulation of the mathematical model describing the phenomenon of thin film condensation, we consider a saturated porous medium confined on a vertical plate, of thickness H, permeability K and porosity ε (**Figure 1**). This vertical flat plate of length L is placed in a flow of pure and saturated steam, with longitudinal speed U_0. The steam condenses on the wall of the plate maintained at the temperature (T_w) lower than that of saturation (T_s) of the steam. The condensate film flows under the effect of gravity and viscous friction forces. There are three zones: 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. Let (x, y) and (u, v) respectively be the Cartesian coordinates and the components of the speed in the porous medium and of the liquid in the reference frame associated with the model.

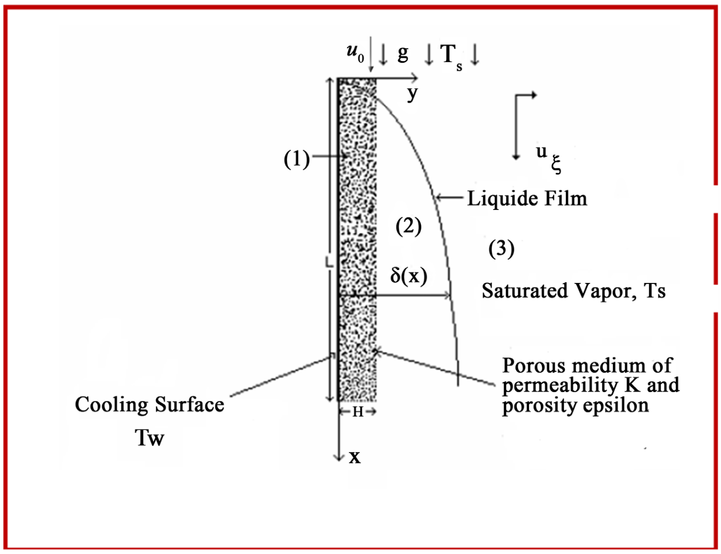

Figure 1. Physical model geometry and coordinate system.

2.2. Assumptions

We accept the following simplifying assumptions:

The porous matrix is isotropic and homogeneous;

The fluid saturating the porous medium is Newtonian and incompressible;

The flow generated is laminar and two-dimensional;

The work, induced by the viscous and pressure forces, is negligible;

The thermo-physical properties of the fluids and those of the porous matrix are assumed to be constant;

The improved Wooding model is used to describe the flow in the porous layer;

The effective dynamic and kinematic viscosities of the porous material are equal to those of the condensate film;

Condensation occurs as a thin film;

The porous matrix is in local equilibrium with the condensate;

The liquid-vapor interface is in thermodynamic equilibrium and the shear stress is assumed to be negligible;

Steam and film are separated by a distinct boundary;

The transverse and longitudinal variation of pressure is not taken into account.

2.3. Nondimensionalization and Transformation Equations

The equations governing transfers in the fields (1) and (2) defined above and the boundary conditions associated with them, were dimensionless using the following variables and parameters (Abik M *et al.* [2003]), we ask:

$$H^* = \frac{H}{\sqrt{K}} \tag{1-a}$$

$$x^* = \frac{x}{\sqrt{K}} \tag{1-b}$$

$$y^* = \frac{y}{\sqrt{K}} \tag{1-c}$$

$$\theta_\xi = \frac{T_\xi - T_w}{T_s - T_w} \tag{2-a}$$

$$u^* = \frac{u_\xi}{u_r} \tag{2-b}$$

$$u_r = \frac{K}{\nu_{eff}} g \tag{2-c}$$

$$\delta^* = \frac{\delta}{\sqrt{K}} \tag{2-d}$$

To pinpoint the liquid-vapor interface and avoid the problems caused by non-uniformity of the mesh in the vicinity of the interface we make the following change of variable:

$$X = x^*;\ \ \eta = coef \cdot \frac{y^*}{H^*} + (1 - coef) \cdot \left\{ 1 + \frac{y^* - H^*}{\delta^* - H^*} \right\} \tag{3}$$

with *coef* = 1 if we are in the porous layer ($0 \le y^* \le H^*$) and *coef* = 0 when we are in the pure liquid ($H^* \le y^* \le \delta^*$).

coef = coefficient

So the plan (x^*, y^*) is transformed into a rectangular domain (X, η).

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

$$\frac{\partial u_p^*}{\partial X} + \frac{1}{H^*}\frac{\partial v_p^*}{\partial \eta} = 0 \tag{4}$$

$$u_p^* \frac{\partial u_p^*}{\partial X} + \frac{v_p^*}{H^*}\frac{\partial u_p^*}{\partial \eta} = -\frac{\varepsilon^2}{R_{e_K}} u_p^* + \frac{\varepsilon^2}{R_{e_K}} \left(\frac{1}{H^{*2}} \frac{\partial^2 u_p^*}{\partial \eta^2} \right) + \frac{\varepsilon^2}{F_{r_K}} \tag{5}$$

$$u_p^* \frac{\partial \theta_p}{\partial X} + \frac{v_p^*}{H^*}\frac{\partial \theta_P}{\partial \eta} = \frac{1}{p_r R_{e_K} H^{*2}} \frac{\partial^2 \theta_P}{\partial \eta^2} \tag{6}$$

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

$$\left(\delta^* - H^*\right)\frac{\partial u_l^*}{\partial X} - (\eta - 1)\frac{\mathrm{d}\delta^*}{\mathrm{d}X}\frac{\partial u_l^*}{\partial \eta} + \frac{\partial v_l^*}{\partial \eta} = 0 \tag{7}$$

$$u_l^*\left[\frac{\partial u_l^*}{\partial X} - \frac{\eta - 1}{\delta^* - H^*}\frac{\mathrm{d}\delta^*}{\mathrm{d}X}\frac{\partial u_l^*}{\partial \eta}\right] + \frac{v_l^*}{\delta^* - H^*}\frac{\partial u_l^*}{\partial \eta} = \frac{1}{R_{e_K}}\left[1 + \frac{\mu^*}{\left(\delta^* - H^*\right)^2}\frac{\delta^2 u_l^*}{\delta \eta^2}\right] \tag{8}$$

$$u_l^*\left[\frac{\partial \theta_l}{\partial X} - \frac{\eta - 1}{\delta^* - H^*}\frac{\mathrm{d}\delta^*}{\mathrm{d}X}\frac{\partial \theta_l}{\partial \eta}\right] + \frac{v_l^*}{\delta^* - H^*}\frac{\partial \theta_l}{\partial \eta} = \frac{1}{R_{e_K} P_r \left(\delta^* - H^*\right)^2}\frac{\partial^2 \theta_l}{\partial \eta^2} \tag{9}$$

Boundary conditions:

in the wall $\eta = 0$

$$u_p^* = v_p^* = 0 \tag{10-a}$$

$$\theta_p = 0 \tag{10-b}$$

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

$$\theta_l = \theta_p \tag{11-a}$$

$$u_l^* = u_p^* \tag{11-b}$$

$$\frac{\mu^*}{H^*}\frac{\partial u_p^*}{\partial \eta} = \frac{1}{\delta^* - H^*}\frac{\partial u_l^*}{\partial \eta} \tag{12-a}$$

$$\frac{1}{H^*}\frac{\partial \theta_p}{\partial \eta} = \frac{\lambda^*}{\delta^* - H^*}\frac{\partial \theta_l}{\partial \eta} \tag{12-b}$$

A liquid interface/steam, $\eta = 2$

$$\theta_l = 1 \tag{13-a}$$

$$\frac{\partial u_l^*}{\partial \eta} = 0 \tag{13-b}$$

The heat and mass balance satisfy the following expressions:

$$\frac{Ja}{(Pe)_{eff}}\frac{1}{H^*}\left.\frac{\partial \theta_p}{\partial \eta}\right|_{\eta=0} = \frac{\mathrm{d}}{\mathrm{d}x^*}\left[H^*\int_0^1 \left\{1 + Ja\left(1 - \theta_p\right)\right\} u_p^* \mathrm{d}\eta\right] + \frac{\mathrm{d}}{\mathrm{d}x^*}\left[\int_1^2 \left(\delta^* - H^*\right)\left\{1 + Ja\left(1 - \theta_l\right)\right\} u_l^* \mathrm{d}\eta\right] \tag{14}$$

with

$$(Pe)_{eff} = \lambda^* Pr R_{e_K} \tag{15}$$

The mass flow rate:

$$H^*\int_0^1 u_p^* \mathrm{d}\eta + \left(\delta^* - H^*\right)\int_1^2 u_l^* \mathrm{d}\eta = \frac{\rho_v}{\rho_l}\delta^* \tag{16}$$

3. Digital Methodology

The transfer equations are discretized by an implicit finite difference method. The mesh of the digital domain is considered uniform in the transverse and longitudinal directions. The advection and diffusion terms are discretized with a back-centered and centered scheme respectively. The coupled systems of algebraic equations thus obtained are solved numerically using an iterative line-by-line relaxation method of the Gauss-Seidel type.

The equations are discretized transfer by an implicit finite difference method. The mesh of the 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.

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

$$\begin{aligned} b.Int^*(i,j) &= ami \cdot Int^*(i-1,j) + amj \cdot Int^*(i,j-1) \\ &\quad + apj \cdot Int^*(i,j+1) + coef\,0 \end{aligned} \tag{17}$$

(i, j) and $(\Delta x, \Delta\eta)$ respectively represent nodes and the step along x and η.

im and jm are the maximum nodes along x and η

$$2 \le i \le im \text{ et } 2 \le j \le jm-1 \tag{18}$$

with

$$b = \frac{coefx}{\Delta x} + \frac{coefe}{\Delta\eta} + 2 * \frac{coef\,2}{\Delta\eta^2} + coefint\ ; \tag{19-a}$$

$$ami = \frac{coefx}{\Delta x}\ ; \tag{19-b}$$

$$amj = \frac{coefe}{\Delta\eta} + \frac{coef\,2}{\Delta\eta^2} \tag{19-c}$$

$$apj = \frac{coef\,2}{\Delta\eta^2} \tag{20-a}$$

$$coef\,2 = \frac{\varepsilon^2}{F_{r_K}} \tag{20-b}$$

$$coef\,1 = \varepsilon^2 / R_{e_K} \tag{20-c}$$

$$Int^*(i,j) = Int^*(i,j-1) + co1 - co2 \tag{21}$$

Discretization of the equation of heat balance:

$$\delta^*(i) = \delta^*(i-1) + R_\delta \tag{22}$$

Coefficients that fall within the terms of the general equation are shown in the appendix.

Coupled algebraic systems thus obtained are solved numerically using an iterative relaxation method line by line Gauss-Seidel.

4. Results and Discussion

Our study makes it possible to examine and highlight the role of parameters such as: Prandtl numbers, Jacob numbers and dimensionless thermal conductivity on the velocity profiles in media (porous and liquid). The results from the numerical simulations relate to:

$\nu^* = 10^{-7}$, $\mu^* = 1$ $\varepsilon = 0.4$ (ν^* dimensionless kinematic viscosity, $\mu^* = 1$ dimensionless dynamic viscosity, ε = porosity). The study of the sensitivity of the mesh led us to choose $im = 51$ et $jm = 101$ (im horizontal sensitivity and jm vertical sensitivity). The convergence criterion is set at 10^{-6}.

To validate our model, we compared the results of Asbik *et al.* [8]-[11] with those from our calculation code in which the inertia terms have been omitted. We find that the correspondence is acceptable.

Figures 2-4 show that the parameters relating to the thermal problem (the dimensionless thermal conductivity, the Prandtl and Jacob numbers) have no influence on the dimensionless speed although the thermal and hydrodynamic problems are coupled via the heat balance equation. So at first approximation with the chosen constants, we can solve the hydrodynamic problem independently of the thermal problem.

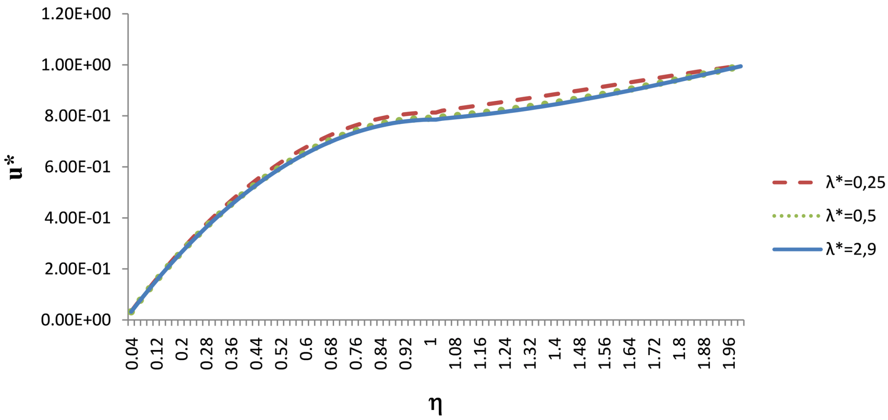

Figure 2. Variation of the longitudinal speed to the ordinate η for different values of λ^* Re_K = 45, H^* = 2.10^{-3}, $Fr_K = 10^{-4}$, $Ja = 10^{-3}$, $Pr = 2$.

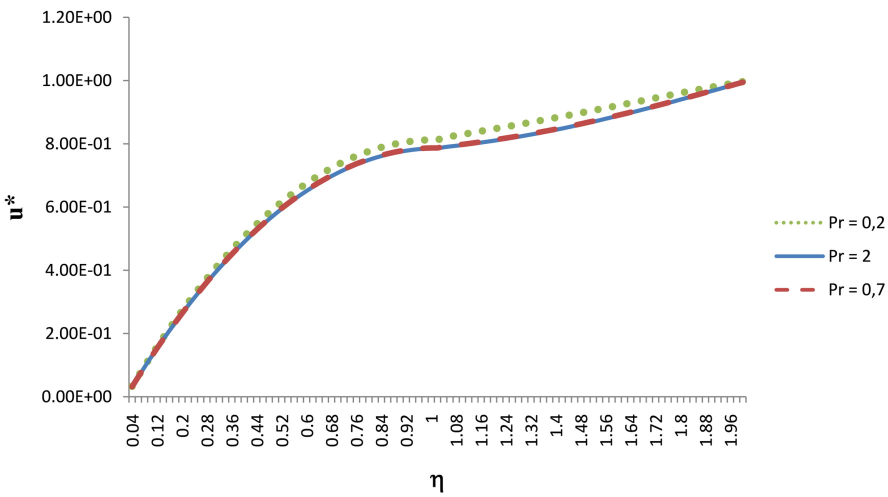

Figure 3. Variation of the longitudinal speed to the ordinate η for different values of Pr $Re_K = 45$, H^* = 2.10^{-3}, $Fr_K = 10^{-4}$, $Ja = 10^{-3}$, $Pr = 2.9$.

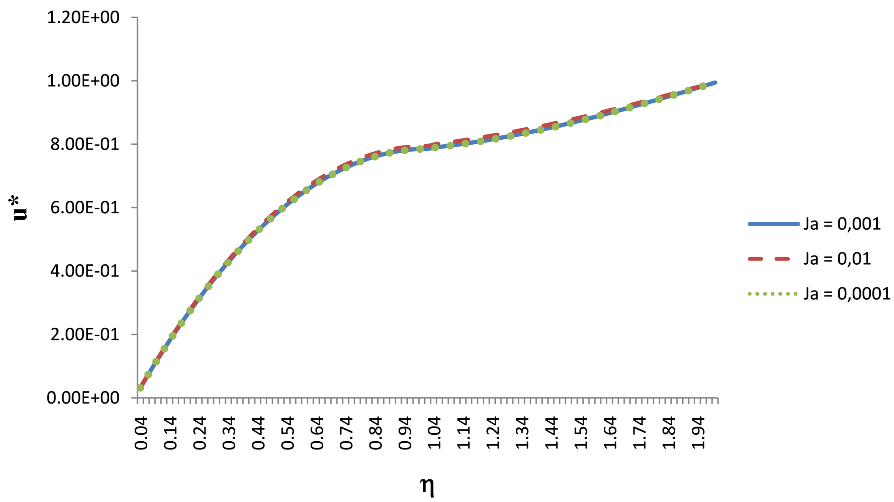

Figure 3. Variation of the longitudinal speed to the ordinate η for different values of *Ja* Re_K = 45, H^* = 2.10^{-3}, $Fr_K = 10^{-4}$, $Ja = 10^{-3}$, $Pr = 2.9$.

5. Conclusion

We studied the forced convection condensation of pure and saturated vapor on a vertical wall covered with porous material. The equations were solved using the back-centered implicit finite difference method for the velocity profiles in the two media (porous and liquid). We analyzed the influence of Prandtl and Jacob numbers and dimensionless thermal conductivity on the velocity profiles in the two media (porous and liquid). The results obtained show that the parameters relating to the thermal problem have no influence on the dimensionless speed although the thermal and hydrodynamic problems are coupled via the heat balance equation. So at first approximation with the chosen constants, we can solve the hydrodynamic problem independently of the thermal problem.

Conflicts of Interest

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

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Nomenclature

Greek symbols:	
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
Indices exhibitor:	
eff	efficiency value
i	porous substrate interface/pure liquid
l	liquid
p	porous
s	saturation
v	steam
w	wall
*	dimensionless quantity
Latin letters:	
c_p	specific heat, $J.\ kg^{-1} \cdot K^{-1}$
Fr_K	Froude number based on $\sqrt{K}$
g	acceleration of gravity, $m \cdot s^{-2}$
H	thickness of the porous layer, m
h_{fa}	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
R_{e_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