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Effects of latent heat storage on heat transfer in a forced flow in a porous layer

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Abstract

The thermal energy storage and heat transfer between a forced fluid of a pure fluid phase and a porous medium are theoretically studied. A unified enthalpy model is developed to investigate the transient temperature field in the pure fluid phase and in the porous medium. Possible transition from saturated to non-saturated porous medium is taking into consideration. Numerical results provide the parametric information concerning effects of thermal energy stored in porous medium, by latent or sensible heat storage, on the temperature field in the forced fluid flow.

Keywords: Heat transfer; Forced convection; Heat storage; Porous medium; Phase-change

1. Introduction

Heat transfer from a flowing fluid is often enhanced by allowing the fluid flow to come into contact with a porous medium, whereby heat is stored in the porous medium by sensible or latent heat storage [1,2]. The use of porous media is an effective means heat removal and it is motivated by several engineering applications; including the air-conditioning and refrigeration industry, high effective cooling of electronic devices with a porous heat sink, heat exchangers in nuclear industry reactors, and many others.

Recently, Hollmuller [3] has studied the heat diffusion in a cylindrical air/soil heat-exchanger where the air-flow is submitted to a harmonic temperature signal at input. Different geometrical configurations have been studied to characterize the amplitude-dampening and phase-shifting of the input periodic signal.

There has been a current interest of the study of the hydrodynamic as well as the heat transfer behavior of flows through systems partly filled with a porous material. The analysis of fluid flow and heat transfer interfacial conditions between a porous medium and fluid layer is contained in Ref. [4]. The thermal development of forced convection in such system is recently described by Nield and Kuznetsov [5]. The probe of heat transfer in a porous heat sink is well described by Zhang [6].

In the present work, the transient heat transfer between a hot or cold flowing fluid and a parallel porous layer is investigated numerically. The aim of this work is to predict the effects of liquid–vapor phase-change in the porous medium on heat storage and the exit temperature of the flowing fluid variations for different fluid flow characteristics. The case of a time-variation in the inlet temperature was also treated.

One difficult aspect in the modeling of transport mechanism within the porous media with possible phase-change at the transient regime is that the problem suffers from a singularity if the temperature is the primary derived solution from the energy equation. This difficulty can be circumvented by using an enthalpy formulation [7,8].

2. Problem formulation

The schematic diagram considered in this paper is shown in Fig. 1. A horizontal porous layer subjected to a heat exchange in the upper face ($0 < x < L$, $y = d_1$) with a forced fully developed fluid flow (fluid 1) $d_1 < y < d$, entering the system at

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Nomenclature

c	specific heat capacity	$\mathrm{J\,kg^{-1}\,K^{-1}}$
D	capillary diffusion coefficient	$\mathrm{m^2\,s^{-1}}$
h	enthalpy	$\mathrm{J\,kg^{-1}}$
h_{fg}	vaporization latent heat	$\mathrm{J\,kg^{-1}}$
h_w	heat transfer coefficient at the wall	$\mathrm{W\,m^{-2}\,K^{-1}}$
H	volumetric enthalpy	$\mathrm{J\,m^{-3}}$
k_r	relative permeability	
k	thermal conductivity	$\mathrm{W\,m^{-1}\,K^{-1}}$
K	intrinsic permeability	$\mathrm{m^2}$
L	porous cavity length	m
Nu	Nusselt number	
d_1	porous layer width	m
d	system width	m
Q_{net}	thermal energy exchanged with the porous medium	$\mathrm{W\,m^{-1}}$
Re	Reynolds number	
s	liquid saturation	
t	time	s
T_0	initial temperature of the porous layer	K
T_{in}	inlet temperature	K
$T_{m,ex}$	mean temperature value at the outlet face	K
T_{sat}	boiling temperature	K
T_w	mean temperature value at the wall	K
u	velocity	$\mathrm{m\,s^{-1}}$
u_{in}	mean velocity value at the inlet face	$\mathrm{m\,s^{-1}}$
x, y	longitudinal and transversal coordinates	m

Greek symbols

ε	porosity of the porous medium	
γ_h	advection correction factor	
μ	dynamic viscosity	$\mathrm{kg\,m^{-1}\,s^{-1}}$
ν	kinetic viscosity	$\mathrm{m^2\,s^{-1}}$
ρ	density	$\mathrm{kg\,m^{-3}}$
σ	surface tension	$\mathrm{N\,m^{-1}}$

Subscripts

l	liquid phase
$f1$	fluid 1
s	solid phase
v	vapour phase
sat	saturated state
0	initial
eff	effective

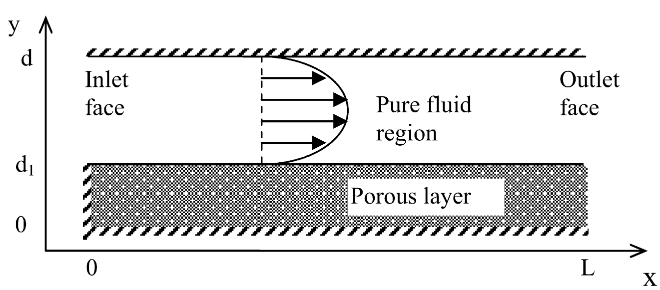

Fig. 1. Schematic of the problem showing coordinates and dimensions of the porous layer and the pure fluid region.

the constant and uniform temperature T_{in}. The top surface of the domain is adiabatic and impervious. The porous layer is initially saturated with a liquid phase (fluid 2) at the temperature T_0 (which may be lower or higher to T_{in}) and is bounded by three impervious faces: ($x = 0$, $0 < y < d_1$), ($0 < x < L$, $y = 0$) and ($0 < x < L$, $y = d_1$). However, the face ($x = L$, $0 < y < d_1$) is permeable while the right boundary is connected to an adjacent liquid tank (fluid 2).

In this work, we focus on the heat transfer and phase-change effects in the porous medium on the temperature profiles in the flowing fluid 1. The forced fluid flow is simplified by the Hagen–Poiseuille profile.

2.1. Velocity field

The fluid 1 is assumed to be Newtonian with uniform properties. The corresponding flow is assumed to be laminar incompressible unidirectional and hydrodynamically developed. Under these assumptions, the analytical solution for the fluid flow problem between impermeable boundaries in the pure fluid regime is given by:

$$u(y) = 6u_{in}\frac{(y-d_1)(d-y)}{(d-d_1)^2}, \quad d_1 \leqslant y \leqslant d \tag{1}$$

where u_{in} is the mean value of the velocity.

Analogous profile has been used by Nield et al. [5]. The steadiness in the fluid flow is due to the negligible change in the pressure gradient which drives the flow.

In the region $0 < y < d_1$, the pressure gradient is insignificant in the porous layer, and so:

$$u(y) = 0, \quad 0 \leqslant y \leqslant d_1 \tag{2}$$

2.2. Energy equation

In the pure fluid region, the heat transfer is governed by the typical single-phase energy equation. The heat transfer in the porous medium will be treated under the assumption of local thermal equilibrium. Local volume averaged forms of governing equations are used.

Various numerical methods have been developed to solve energy equation with phase-change in porous media. A part of models involves the calculation of nodal temperature separately for all phases with appropriate interface conditions. Due to the moving boundaries between single-phase (solid + liquid) and two-phase (solid + liquid + vapor) domains, the solution involves remeshing of the computational domain and real numerical difficulties arise.

A significant amount of the recent literature uses an enthalpy method which involves averaged enthalpy as primary variable in the energy equation. The success of the enthalpy formulation is devoted to the use of a fixed grid in numerical calculations.

In the present study, a single energy balance equation in the entire domain, based on the enthalpy formulation [7], is used. This equation is given as follow:

$$\Omega\frac{\partial H}{\partial t}+\vec{\nabla}.(\gamma_h\vec{u}\,H)=\vec{\nabla}.(\Gamma_h\vec{\nabla}H) \tag{3}$$

For the definition of the variable H and coefficients γ_h, Ω and Γ_h, we consider two cases:

2.2.1. Case 1: pure fluid region

$$H=\rho_{f1}c_{f1}T,\quad \gamma_h=1,\quad \Omega=1\quad\text{and}\quad \Gamma_h=\frac{k_{f1}}{\rho_{f1}c_{f1}} \tag{4}$$

The temperature can be deduced by:

$$T=\frac{H}{\rho_{f1}c_{f1}} \tag{5}$$

2.2.2. Case 2: porous layer

$$H=\rho(h-2h_{v\mathrm{sat}}) \tag{6}$$

with $\rho h=\rho_l s h_l+\rho_v(1-s)h_v$ and $\rho=\rho_l s+\rho_v(1-s)$.

$$\begin{aligned}
\Omega&=\varepsilon+\rho_s c_s(1-\varepsilon)\frac{\mathrm{d}T}{\mathrm{d}H}\\
\gamma_h&=\frac{[s\rho_l+\rho_v(1-s)][h_{v\mathrm{sat}}(1+\lambda_l)-h_{l\mathrm{sat}}\lambda_l]}{(2h_{v\mathrm{sat}}-h_{l\mathrm{sat}})s\rho_l+\rho_v(1-s)h_{v\mathrm{sat}}}\\
\Gamma_h&=\frac{\rho_l h_{\mathrm{fg}}}{\rho_l h_{\mathrm{fg}}+(\rho_l-\rho_v)h_{v\mathrm{sat}}}D+k_{\mathrm{eff}}\frac{\mathrm{d}T}{\mathrm{d}H}
\end{aligned} \tag{7}$$

with $D=\frac{\sqrt{\varepsilon K}\sigma}{\mu_l}\frac{k_{rl}k_{rv}}{\frac{\nu_v}{\nu_l}k_{rl}+k_{rv}}(-J'(s))$, $J(s)=1.417(1-s)-2.120(1-s)^2+1.263(1-s)^3$

$$\lambda_l=\frac{\nu}{\nu_l}k_{rl},\quad \lambda_v=1-\lambda_l,\quad \nu=\frac{\mu}{\rho},\quad \nu_l=\frac{\mu_l}{\rho_l}$$

$$\mu=\frac{[\rho_l s+\rho_v(1-s)]}{k_{rl}/\nu_l+k_{rv}/\nu_v}\quad\text{and}\quad k_{rl}=s,k_{rv}=1-s$$

s is the liquid saturation denoting the ratio of liquid volume and the void space volume in a representative elementary volume. ρ_l and ρ_v are densities of liquid and vapor phases saturating the porous media. $J(s)$ is the Leverett function.

h_l and h_v are the enthalpies of liquid and vapor phases, they are related to temperature by the relations:

$$\begin{aligned}
&h_s=c_sT+h_s^0,\quad h_l=c_lT,\\
&h_v=c_vT+\big[(c_l-c_v)T_{\mathrm{sat}}+h_{\mathrm{fg}}\big]
\end{aligned} \tag{8}$$

T_{sat} is the temperature of phase change, h_{fg} is the latent heat of vaporization and $h_{v\mathrm{sat}}=(h_v)_{T=T_{\mathrm{sat}}}.h_s^0$ is a reference enthalpy and $h_l(T=0)=0$. c_s, c_l and c_v are the specific heats of solid, liquid and vapor phases.

The temperature and liquid saturation can be deduced from the volumetric enthalpy H by:

$$\begin{cases}
T=\frac{H+2\rho_l h_{v\mathrm{sat}}}{\rho_l c_l},\quad s=1,\\
\quad H\leqslant-\rho_l(2h_{v\mathrm{sat}}-h_{l\mathrm{sat}})\\
T=T_{\mathrm{sat}},\quad s=-\frac{H+\rho_v h_{v\mathrm{sat}}}{\rho_l h_{\mathrm{fg}}+(\rho_l-\rho_v)h_{v\mathrm{sat}}},\\
\quad -\rho_l(2h_{v\mathrm{sat}}-h_{l\mathrm{sat}})<H\leqslant-\rho_v h_{v\mathrm{sat}}\\
T=T_{\mathrm{sat}}+\frac{H+\rho_v h_{v\mathrm{sat}}}{\rho_v c_v},\quad s=0,\quad -\rho_v h_{v\mathrm{sat}}<H
\end{cases} \tag{9}$$

Initial and boundary conditions are expressed as follows:

- The faces ($y=0$ and $y=d$), are impervious and adiabatic:

$$\frac{\partial H}{\partial y}=0 \tag{10}$$

- At the inlet ($x=0$), the fluid 1 enters the domain with a constant and uniform temperature T_{in}, except the porous face which is impervious and adiabatic:

$$H=\rho_{f1}c_{f1}T_{\mathrm{in}},\quad d_1\leqslant y\leqslant d \tag{11}$$

$$\frac{\partial H}{\partial x}=0\quad\text{for }0\leqslant y\leqslant d_1 \tag{12}$$

- The outlet face ($x=L$), is in thermally developed conditions:

$$\frac{\partial H}{\partial x}=0\quad\text{for }0\leqslant y\leqslant d \tag{13}$$

- Initially, the porous media is at uniform temperature T_0 and the pure fluid region is at uniform temperature T_{in}:

$$H=\rho_{f1}c_{f1}T_{\mathrm{in}}\quad\text{for }d_1\leqslant y\leqslant d,\ 0\leqslant x\leqslant L \tag{14}$$

$$H=\rho_l(c_lT_0-2h_{v\mathrm{sat}})\quad\text{for }0\leqslant y\leqslant d_1,\ 0\leqslant x\leqslant L \tag{15}$$

Boundary condition given by Eq. (13) reflects that heat transfer at the outlet is assumed to reach the thermally developed conditions. This assumption was justified by previous parametric test in which the computational domain was extended another one-third beyond the physical length, and no appreciable variations in the simulation results were observed.

We use the continuity of the heat flux at the wall separating pure fluid region and porous layer, as an interface condition for the heat transfer.

2.3. Theoretical analysis and governing parameters

For given parameters of inlet parameters for forced flow (u_{in} and T_{in}), heat transfer depends on the value of the started porous medium temperature T_0. If T_0 is lower than T_{in}, the forced flow will loss heat at the interface region of the porous medium. The exiting-mean temperature of the fluid, defined by:

$$T_{m,\mathrm{ex}}=\frac{1}{\rho_{f1}u_{\mathrm{in}}(d-d_1)}\int\limits_{d_1\leqslant y\leqslant d}\rho_{f1}uT_{x=L}\,\mathrm{d}y \tag{16}$$

will be less then T_{in}. The net thermal energy transferred to the porous medium can be calculated by:

$$Q_{\text{net}} = Q_{\text{inlet}} - Q_{\text{outlet}} = \int_{\text{inlet face}} \rho_{f1} c_{f1} u T \, dy - \int_{\text{exit face}} \rho_{f1} c_{f1} u T \, dy \tag{17}$$

The convective heat transfer coefficient at the wall separating the porous layer and the pure fluid region is obtained by:

$$h_w = \frac{Q_{\text{net}}}{L(T_w - T_{\text{in}})} \tag{18}$$

where T_w is the mean temperature of the wall:

$$T_w = \frac{1}{L} \int_{0 \leqslant x \leqslant L} T_{y=d_1} \, dx \tag{19}$$

An overall Nusselt number can be defined by:

$$Nu = \frac{h_w (d - d_1)}{k_{f1}} \tag{20}$$

Temperature at the outlet depends on values of u_{in} and T_{in}. Toward a best comparison with results in the literature it is more convenient to give the magnitude of u_{in} with the Reynolds number defined by:

$$Re = \frac{\rho_{f1} u_{\text{in}} (d - d_1)}{\mu_{f1}} \tag{21}$$

Eq. (3) is solved numerically by the finite volumes method [9]. Integral equations will be converted to algebraic form which is solved numerically by the iterative line-by-line method. The rectangular domain ($d \times L$) is divided by uniform and fixed grid 51×61 nodes. Convergence is considered to be reached when the relative error on the values of H and p are less than 10^{-5} between two consecutive iterations. $\Delta t = 0.1$ s is chosen for all numerical resolutions.

Numerical calculations in the porous region have been validated and used in previous studies [10–12]. In order to verify the validity of the code in the pure fluid region, we consider the case of forced air-flow parallel to an isotherm wall. It is well known from the literature that for air experiment in laminar flow, the wall-averaged Nusselt number has a $Re^{1/2}$ dependence. In Fig. 2 we have plotted results, given by the present code, for Nu as a function of Re. Fitted results show a Re^n dependence with $n = 0.507$.

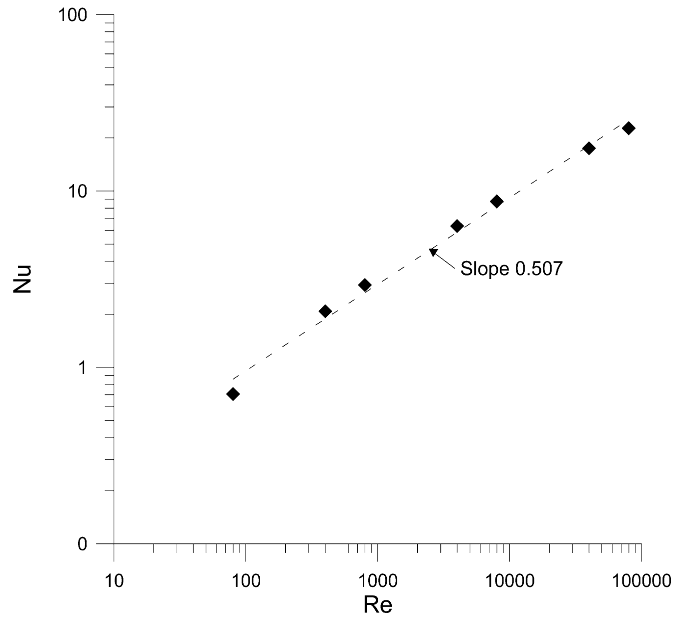

Fig. 2. Averaged Nusselt number for forced air-flow parallel to an isotherm wall.

3. Results and discussions

Heat transfer calculations between forced air flow (fluid 1) and porous medium were conducted with and without phase-change. The porous medium is initially saturated by a fluid (R141b) that evaporates at $T_{\text{sat}} = 32\,^\circ\text{C}$ (corresponding to the pressure 1 bar). Thermophysical and geometrical properties of the porous medium and fluids used are shown in Table 1.

The Reynolds number was varied from 400 to 1000 under conditions of two different values of initial temperature of the porous medium T_0 ($T_0 \ll T_{\text{sat}}$ and $T_0 \leqslant T_{\text{sat}}$). To show the efficiency of the system to store thermal energy, the inlet temperature of forced flow differs from T_0: $T_0 - \Delta T < T_{\text{in}} < T_0 + \Delta T$ ($\Delta T = 10\,^\circ\text{C}$).

In each case, we calculate numerically temperature profiles for the porous medium and the flowing fluid. Resulting profiles serve to evaluate the mean exiting temperature $T_{m,\text{ex}}$ at the outlet face and the corresponding convective heat transfer rate.

Table 1
Thermophysical property data for the porous medium (a), R141b fluid (b) and air (c)

L–d–d_1 (cm)	k_{eff} ($\text{W m}^{-1}\,\text{K}^{-1}$)	Absolute permeability (m^2)	c_s ($\text{J kg}^{-1}\,\text{K}^{-1}$)	ρ_s (kg m^{-3})	ε
150–5–0,5	0.85	10^{-10}	8.79×10^2	2.645×10^3	0.35

(a)

c_l ($\text{J kg}^{-1}\,\text{K}^{-1}$)	c_v ($\text{J kg}^{-1}\,\text{K}^{-1}$)	ρ_l (kg m^{-3})	ρ_v (kg m^{-3})	μ_l ($\text{kg m}^{-1}\,\text{s}^{-1}$)	μ_v ($\text{kg m}^{-1}\,\text{s}^{-1}$)	h_{fg} (J kg^{-1})
1.16×10^3	0.804×10^3	1227	4.8	4.3×10^{-4}	0.939×10^{-5}	2.23×10^5

(b)

c_{f1} ($\text{J kg}^{-1}\,\text{K}^{-1}$)	k_{f1} ($\text{W m}^{-1}\,\text{K}^{-1}$)	ρ_{f1} (kg m^{-3})
1.00×10^3	0.026	1.2

(c)

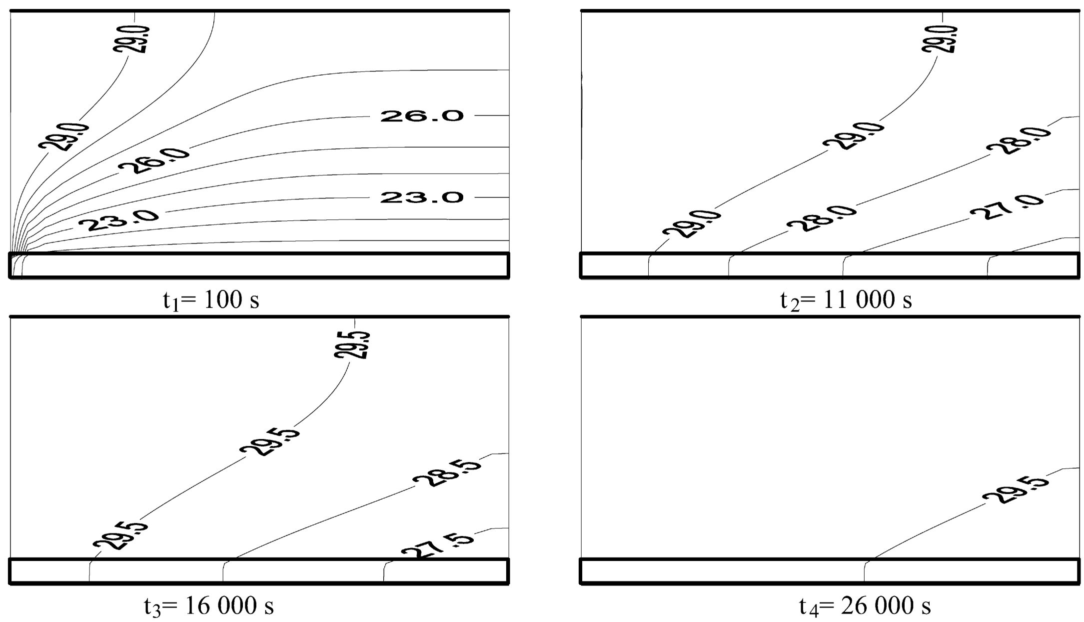

Fig. 3. Time-space evolution of the thermal field: $T_{\text{in}} = 30\,^\circ\text{C}$, $T_0 = 20\,^\circ\text{C}$, $Re = 400$.

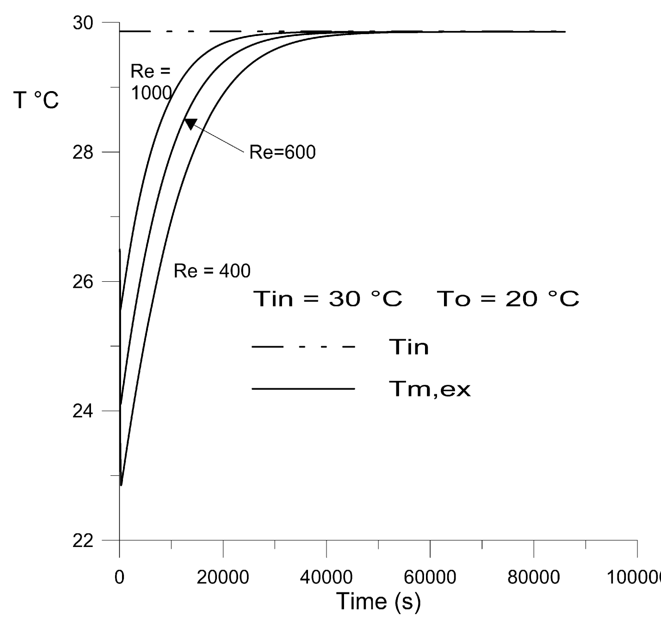

Fig. 4. Transient profiles of the mean exiting temperature, for different values of *Re*: $T_{\text{in}} = 30\,^\circ\text{C}$, $T_0 = 20\,^\circ\text{C}$.

Exiting thermal conditions will be depicted from transient mean temperature profile at the exit face.

In all calculations performed in this study, the maximum variation of the liquid saturation detected is $\Delta s_{\max} = 0.05$ which corresponds to a pressure variation $\Delta p_{\max} = 0.06$ bar. Thus, corresponding phase-change temperature will be less than 2 °C. Boiling temperature is then considered constant and has a fixed value 32 °C.

3.1. The case of fixed inlet temperature $T_{\text{in}} = T_0 + \Delta T$ ($\Delta T = 10\,^\circ C$)

3.1.1. Heat transfer without phase-change

Calculated temperature profiles for $T_0 = 20\,^\circ\text{C}$ and $T_{\text{in}} = 30\,^\circ\text{C}$ are shown in Fig. 3 with $Re = 400$. It is seen that temperature profiles for fluid and porous medium are significantly different. This difference can be attributed to significant different values of thermal conductivities and specific heat capacities for two regions. In the fluid region, the heat transfer mechanism is forced convection in the longitudinal direction and heat transfer by conduction with the porous medium at the separating horizontal wall ($y = d_1$ and $0 < x < L$). As a consequence, the temperature is found to be decreased from the inlet to reach a low value at the exit. The vertical thermal gradient increases at first, then decreases and will have zero value if the entire porous medium is heated at the inlet temperature. At this point, no heat transfer is depicted at the wall and exit temperature equals inlet temperature. In Fig. 4, we have represented transient variations of the exit mean temperature. The net thermal energy transferred to the porous layer increases with time, reaches a maximum and decreases to have zero value (Fig. 5). As a consequence, the outlet temperature falls at first to a lower value $T_{m,\text{ex},\min}$ and then increases to attain the inlet temperature value at the steady state (Fig. 4).

From Fig. 5, it can be seen that heat transfer between fluid flow and porous medium is more efficient at high *Re* number. The steady state is more rapidly attained as *Re* is higher. As could be expected, any increase in the value of *Re* yields an increase in the minimum value of exit temperature (Fig. 4). In fact, with high forced flow intensity, the fluid temperature is highly influenced by forced convection and then, exit temperature equals rapidly the inlet temperature.

3.1.2. Heat transfer with phase-change $T_0 = 30\,^\circ C$

In the previous section, calculations were performed without phase-change. The temperature drop in the fluid phase is

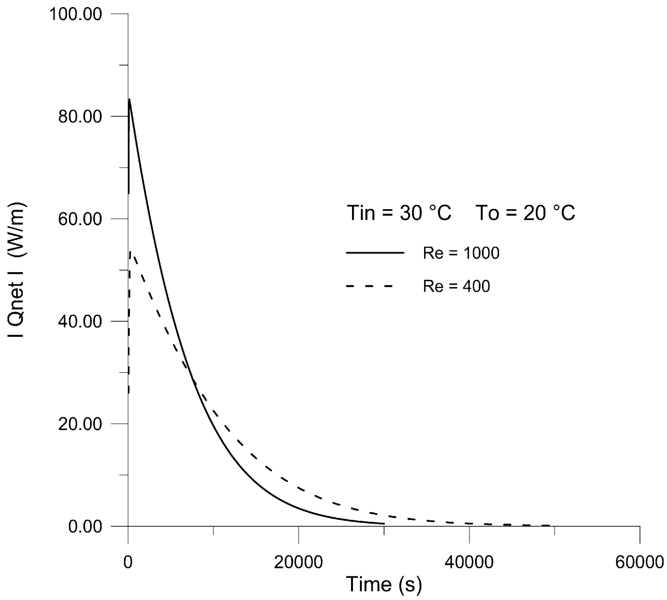

Fig. 5. Transient profiles of the heat exchanged by air-flow with the porous medium, for different values of *Re*: $T_{\mathrm{in}} = 30\,^{\circ}\mathrm{C}$, $T_0 = 20\,^{\circ}\mathrm{C}$.

caused by the sensible heat storage in the porous medium saturated with a single fluid phase. In this section, the initial temperature of the porous medium T_0 approaches the boiling temperature T_{sat} (32 °C) of the fluid saturating the porous medium.

Due to heat transfer with hot forced fluid flow, the temperature in the porous medium increases and reaches the boiling point T_{sat}. As a consequence, liquid–vapour phase-change takes place in the porous medium. An isothermal two-phase zone ($T = T_{\mathrm{sat}}$) will appear in the porous region (Fig. 6). Thermal energy is then stored by latent heat and the forced flow exchanges heat with an isothermal wall $T_w = T_{\mathrm{sat}}$. Heat transfer and outlet

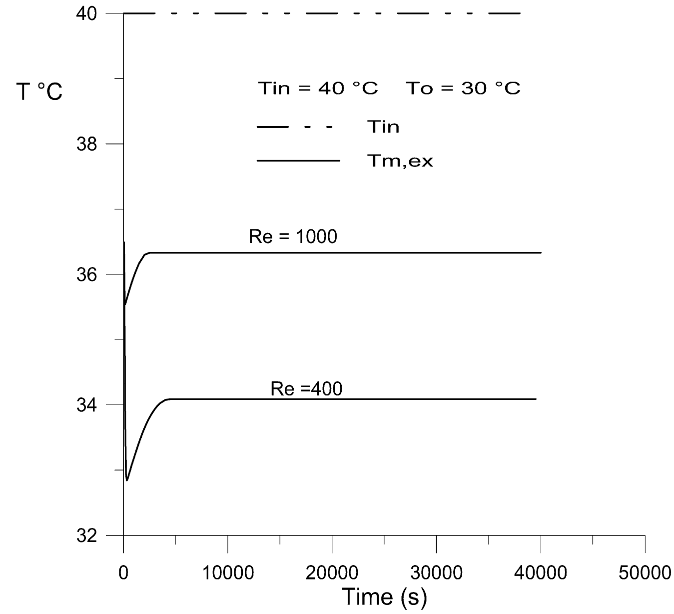

Fig. 7. Transient profiles of the mean exiting temperature, for different values of *Re*: $T_{\mathrm{in}} = 40\,^{\circ}\mathrm{C}$, $T_0 = 30\,^{\circ}\mathrm{C}$.

temperature variations are then less important that the case with sensible heat storage (Figs. 7 and 8). Finally, the outlet temperature will be stabilized at a fixed value if the porous medium is occupied by the two-phase zone and a good uniformity in the wall temperature is achieved. The final value of the outlet temperature decreases with the *Re* number and it is rapidly reached for high values of *Re*.

3.2. The case of square pulsating of inlet temperature

Preceding results show the importance of the use of porous layer to store thermal energy. Now, we will detect the possibility

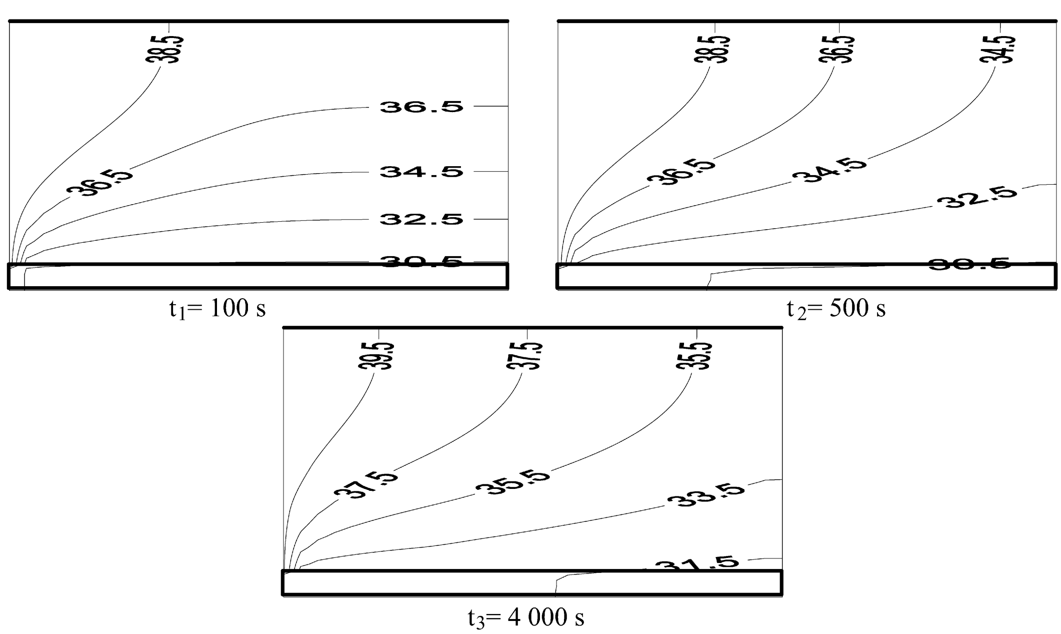

Fig. 6. Time-space evolution of the thermal field: $T_{\mathrm{in}} = 40\,^{\circ}\mathrm{C}$, $T_0 = 30\,^{\circ}\mathrm{C}$, $Re = 400$.

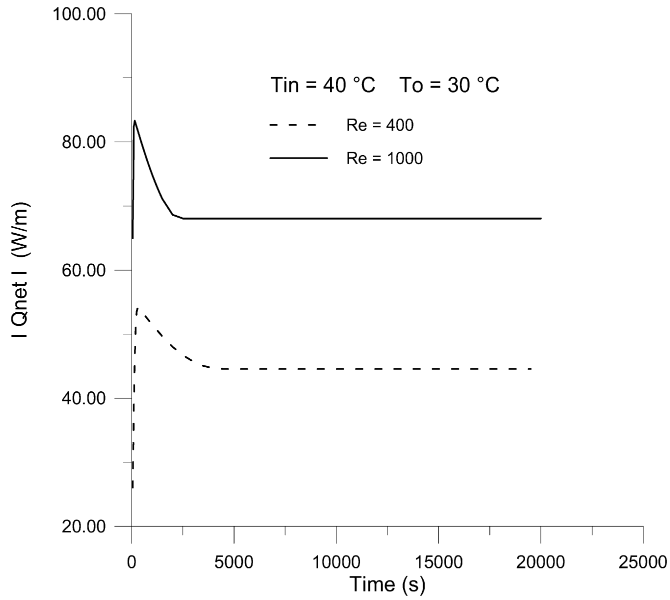

Fig. 8. Transient profiles of the heat exchanged by air-flow with the porous medium, for different values of *Re*: $T_{\rm in} = 40\,^\circ\mathrm{C}$, $T_0 = 30\,^\circ\mathrm{C}$.

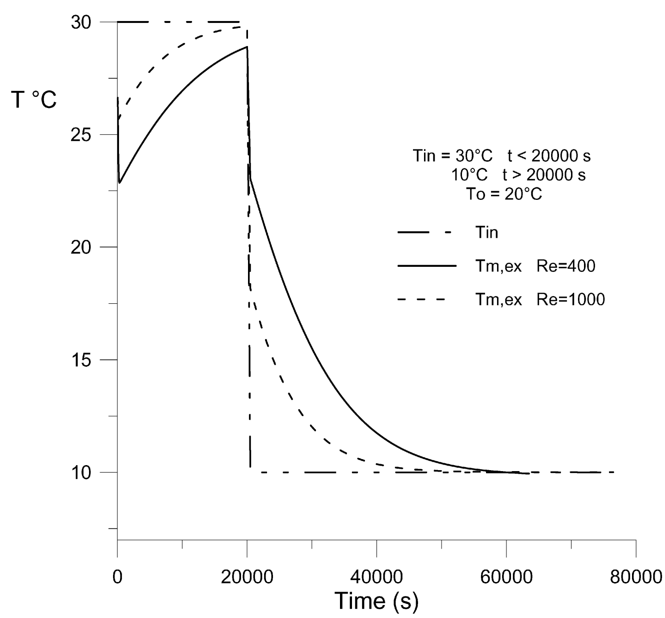

Fig. 9. Transient profiles of the mean exiting temperature, for different values of *Re*: $T_{\rm in} = 30\,^\circ\mathrm{C}$ ($t < 20\,000$ s), $T_{\rm in} = 10\,^\circ\mathrm{C}$ ($t > 20\,000$ s) and $T_0 = 20\,^\circ\mathrm{C}$.

to use the present system for storing thermal energy during a charging period and recovered it during a discharging period.

We consider at present, a forced fluid flow where the inlet temperature pulsates in time. During a first phase, the inlet temperature is kept constant at $T_{\rm in} = T_0 + \Delta T$ ($\Delta T = 10\,^\circ\mathrm{C}$). At $t = 20\,000$ s, $T_{\rm in}$ is fixed at $T_0 - \Delta T$ ($\Delta T = 10\,^\circ\mathrm{C}$). Fig. 9 shows the time variation of the exiting temperature $T_{m,\rm ex}$ in the case "without phase-change" ($T_0 = 20\,^\circ\mathrm{C}$) for different values of *Re* number. As can be seen, the exiting temperature pulsates with time with phase-shifting the inlet temperature variation. This phenomena (phase-shifting) is related to the thermal energy storage in the porous medium and is less and less important as *Re* is higher.

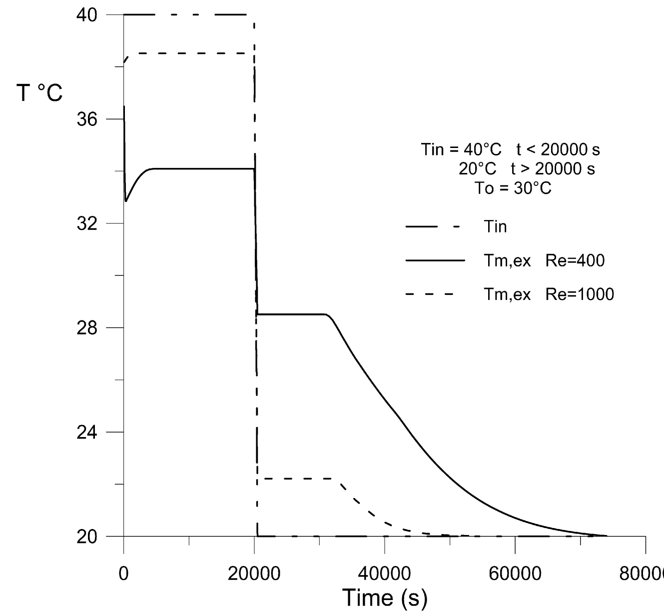

Fig. 10. Transient profiles of the mean exiting temperature, for different values of *Re*: $T_{\rm in} = 40\,^\circ\mathrm{C}$ ($t < 20\,000$ s), $T_{\rm in} = 20\,^\circ\mathrm{C}$ ($t > 20\,000$ s) and $T_0 = 30\,^\circ\mathrm{C}$.

The case of heat storage with phase-change has also been studied where the value of T_0 is fixed at $30\,^\circ\mathrm{C}$ and $T_{\rm in}$ pulsates between $T_0 + \Delta T$ and $T_0 - \Delta T$ ($\Delta T = 10\,^\circ\mathrm{C}$). Resulting temperature profiles at the outlet are presented in Fig. 10. A comparison of Figs. 9 and 10 indicates that outlet temperature will have very weak pulsating around the boiling temperature. However, the pulsating amplitude will be more and more important as *Re* increases.

As mentioned earlier, this difference is due to the large ratio of latent heat to sensible heat and the isothermal behaviour of two-phase change. Thermal energy storage by latent heat allows system to store heat during the charging period. This heat will be recovered in the discharging period by heating the fluid flow. From Fig. 10 we can see that the exit temperature could be kept constant at a large time depending on the Reynolds number value.

3.3. The case of sinusoidal time-variations of inlet temperature

We consider now the problem of a forced fluid flow submitted to periodical variations of inlet temperature where $T_{\rm in}$ is modelized by a sinusoidal time-variation around the temperature T_0:

$T_{\rm in} = T_0 + 10\cos(wt)$. In Fig. 11, we have represented a time-variation of the mean exiting temperature for $Re = 400$ and $Re = 1000$. In this case, $T_0 = 20\,^\circ\mathrm{C}$ and $T_{\rm in}$ oscillates between $30\,^\circ\mathrm{C}$ and $10\,^\circ\mathrm{C}$. So, no phase-change will take place in the porous medium. We can see (Fig. 11) that, due to a sensible heat storage in the porous layer, important amplitude-dampening and phase-shifting appear at low *Re*. However, at high values of *Re*, time-variations at the outlet face approaches

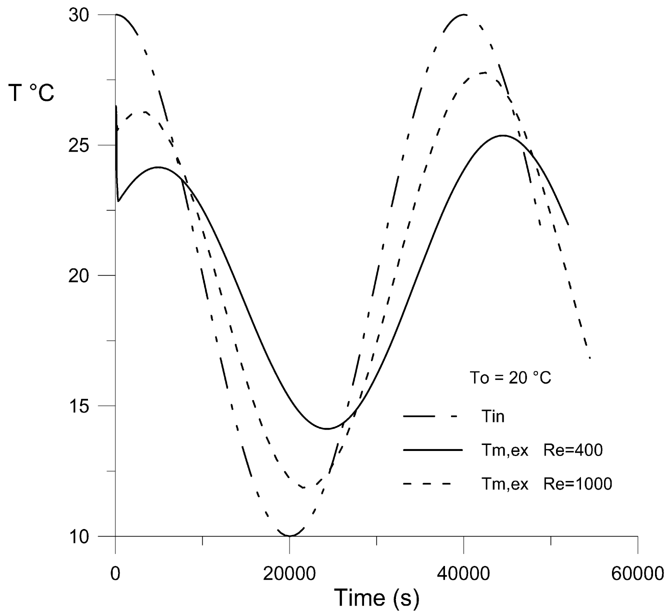

Fig. 11. Transient profiles of the mean exiting temperature, for different values of *Re*: $T_0 = 20\,^{\circ}\mathrm{C}$.

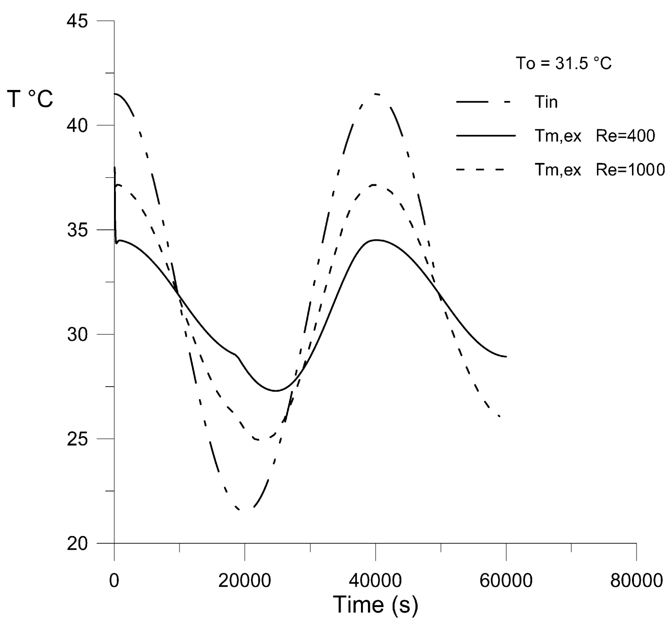

Fig. 12. Transient profiles of the mean exiting temperature, for different values of *Re*: $T_0 = 31.5\,^{\circ}\mathrm{C}$.

that for T_{in}. As discussed above, heat sensible storage in porous layer has slight effects on $T_{m,\mathrm{ex}}$ at high *Re*.

Phase-shifting and amplitude- dampening are also studied in the case "with phase-change". Fig. 12 shows results were $T_0 = 31.5\,^{\circ}\mathrm{C}$ ($T_{\mathrm{sat}} = 0.5\,^{\circ}\mathrm{C}$) and T_{in} oscillates between 41.5 and 21.5 °C. This figure further shows important decrease of the amplitude-oscillations which is very strong for $Re = 400$ and $T_{m,\mathrm{ex}}$ approaches T_{sat}. This phenomenon is a consequence of liquid–vapor phase-change which takes place in the porous layer. In contrary, in this case the phase-shift is less important as compared with the case "without phase-change" (Fig. 11). In fact, in the latter case, sensible heat storage allows the temperature of the porous layer, depending of the *Re* value, to vary between $T_{\mathrm{in,max}}$ and $T_{\mathrm{in,min}}$. Thus, if the temperature of the wall T_w is at a high value (heating regime) and T_{in} changes to a cold regime, T_w will decrease in a large period. However, with a liquid–vapour phase-change, T_w is fixed at T_{sat} and then the phase-shift would be largely smaller.

4. Conclusions

In this article, we developed a theoretical model, based on the enthalpy formulation of the energy equation, to describe heat storage effects in a porous layer on temperature field of a forced fluid (air) flow. A possible transition from saturated porous medium (liquid + solid phases) to a non-saturated porous medium (liquid + vapour + solid phases), is taken into account. This model has been used to predict the effect of thermal energy storage on the average temperature inside the pure fluid forced flow. Concluding remarks of this study can be summarized as follows:

(i) By impregnation of a porous layer into a forced fluid flow, thermal energy may be stored and then serves to cooling hot fluid flow or preheating cold fluid flow.
(ii) Thermal energy storage is more efficient at low forced flow (low values of *Re* number).
(iii) A large heat storage capacity of the porous medium is depicted if the energy is stored by latent heat. Existing temperature of the forced flow is then more stabilized around the phase-change temperature.
(iv) Possible phase-shifting and amplitude-dampening of the inlet forced flow temperature oscillations are depicted. These phenomena depend on stored form of heat (latent or sensible) and forced flow intensity (*Re* number).

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