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Study of the solidification process of a phase change material contained in a heat exchanger with the presence of a porous medium

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Nomenclature

a - permeability, m²; A_m - consecutive number; A_{sf} – interfacial surface area, 1/m; *c* - specific heat, J/(kg K); C_i - inertia coefficient; d_f - ligament diameter, m; d_p - pore diameter, m; *g* - gravity, m/s²; *h* - heat transfer coefficient, W/(m² K); *k* - thermal conductivity, W/(m K); *L* - Latent heat, J/kg; *MP* - metal foam; *P* - pressure, Pa; *PCM* - phase change material; *PPI* - pore number per inch; *Pr* - Prandtl number; *Ra* - Rayleigh number; *S* - source term; *t* - time, s; *T* - temperature, K; *u* - X-velocity, m/s; *v* - Y-velocity, m/s; *V* - velocity, m/s; x, y - Cartesian coordinates; β - liquid volume fraction; γ - expansion factor, 1/K; ε - porosity, μ - dynamic viscosity, kg/(m s); ρ - density, kg/m^3; ω - constant number;
subscripts –
0 - initial; *eff* - effective; *fe* - effective thermal conductivity of fluid; *m* - metal foam; *p* - phase change material, *sf* - between solid and fluid; *se* - effective thermal conductivity of solid; *td* - thermal dispersion; *x* - X direction; *y* - Y direction.

1. Introduction

To eliminate the discrepancy between supply and demand for thermal energy, storage system will be needed. Latent heat storage systems have been widely used due to the high density storage and distribution of an almost constant operating temperature. Most of the PCM used in such storage systems have low thermal conductivity, which is a major drawback. Different techniques are used to improve the heat transfer between the PCM and the heat transfer fluid.

Improvement methods include the use of multiple PCM [1, 2], adding axial or radial fins [3-10], dispersion of highly conductive particles in the PCM [11-19] and use metal foams [20-25].

The use of metal foam is found most effective for improving the heat transfer. Krishnan and al. [20] studied the phase change solid-liquid appearing in a PCM embedded in a metal foam. The natural convection in the melt is considered. Because of the difference in thermal diffusivity between the metal foam and the PCM, thermal equilibrium is not assured. The Enthalpy method is used to reflect the phase change. The influence of Rayleigh numbers, Nusselt and Stefan on the evolution of the melting front, the temperature difference between the solid and the fluid and the melting rate are discussed. For metal foams with Nusselt numbers above 5.9, a mono-temperature model is sufficient for analysis. But for small values of Nusselt, the metal foam and the PCM are sufficiently out thermal equilibrium. For this, a model with two temperatures is necessary. An experimental and analytical study was carried out by Siahpush and al. [21] to evaluate how the copper porous foam (CPF) improves heat transfer in a thermal energy storage system using the PCM. The used metal foam had a porosity of 95% and the PCM is 99% pure eicosane. The PCM and the CFP were contained in a vertical cylinder where the temperature at the radial boundary was kept constant for both the melting and solidification of the PCM to the interior. The CPF has increased the effective thermal conductivity from 0.423 W / mK to 3.06 W / mK. Medrano and al. [22] studied experimentally the heat transfer process during melting and solidification in five small heat exchangers as latent heat storage system. Commercial paraffin RT 35 is used as PCM and water is used as heat transfer fluid. The results show that the Reynolds numbers in the turbulent regime are desirable for the faster phase change process, which reduces the time of phase change in approximately one half. Increasing the difference between the temperature of the inlet water and the phase change temperature of the PCM (15°C to 25°C) causes a considerable decrease of the phase change time between 30% and 60%. An experimental device was designed and constructed by Cui and al. [23] in which the paraffin was used as PCM and copper foam as porous medium. The results indicate that the metal foam increases the effective thermal conductivity during the melting process. Heat transfer rate can be improved by 36%. Consequently, the use of the metal foam can accelerate the melting process and reduce the period of the charge. When the PCM starts melting, natural convection can improve heat transfer performance, which reduces the temperature difference between the wall and the PCM. Taeil Kim and al. [24] performed a study on a latent heat storage system, composed of a tank filled with a PCM and tubes carrying a heat transfer fluid. The effect of the graphite foam was considered in the PCM region. System optimization was performed by varying the thermal conductivity of the combination foam / PCM, the velocity of heat transfer fluid, the diameter and the thickness of the tube wall. The effect of wall thickness of the tubes having the HTF is negligible on the system. The considerable reduction in the number of tubes can be achieved using graphite foam and turbulent flow. The results indicate that the system cost can be reduced considerably by using the combination graphite foam/ PCM as the heat storage medium. A test system was constructed by X.Xiao [25] for measuring the effective thermal conductivity of composite paraffin/metal foam. The effective thermal conductivity is significantly improved by using metal foams compared with that of pure paraffin. For example, the effective conductivities of PCM composites made by copper foams with porosities: 96.95%, 92.31%, 88.89% and pore size of 25 PPI were about 13, 31, 44 times greater than that of pure paraffin, respectively. In addition, if the porosity decreases, the effective thermal conductivity increases. No significant change of the effective conductivity was detected by varying the pore size of the metal foam for the same porosity. The experimental results show good agreement with the theoretical predictions of the literature models.

In our work, a heat exchanger with triple tube containing a PCM is studied. The finite volumes method and SIMPLE algorithm is used for solving governing equations. The effect of porous medium, the number and the position of fins will be discussed.

2. Problem description

The use of phase change materials for thermal energy storage is an effective means if it can improve the heat transfer between the heat transfer fluid and the phase change material to gain more stored energy. The storage system in our work is a heat exchanger with triple concentric tubes. The importance of using three tubes to increase the heat exchange area between the phase change material and the heat transfer fluid. This latter circulates in the two tubes: interior and exterior. The intermediate tube is used for the phase change material. To improve the heat transfer, we used metal foam as a porous medium initially. In the second part of our work, we will add fins to this porous medium.

This heat exchanger provided the energy required for the supply of a solar cooling system.

We have taken the geometric parameters of Abduljalil et al. [26] who performed a numerical study on the PCM solidification in heat exchanger with internal and external fins. Figure 1 shows the physical configurations of all the cases studied. Case A represents a heat exchanger with PCM only [26], case B: addition of metal foam as porous media to PCM zone, case C: addition of four fins to case B, case D: addition of six fins to case B, case E: addition of eight fins to case B and the cases (F, G, H) show the different position of fins.

The radius of the inner tube was 25.4 mm, the radius of the intermediate tube is 75 mm and the radius of the outer tube is 100 mm.

The PCM used in this study is the RT82. The thermo-physical parameters of the PCM and copper are mentioned in Table 1.

3. Mathematical approach

In our work, we adopted the mathematical model that is presented in the study of Liu et al. [27]:

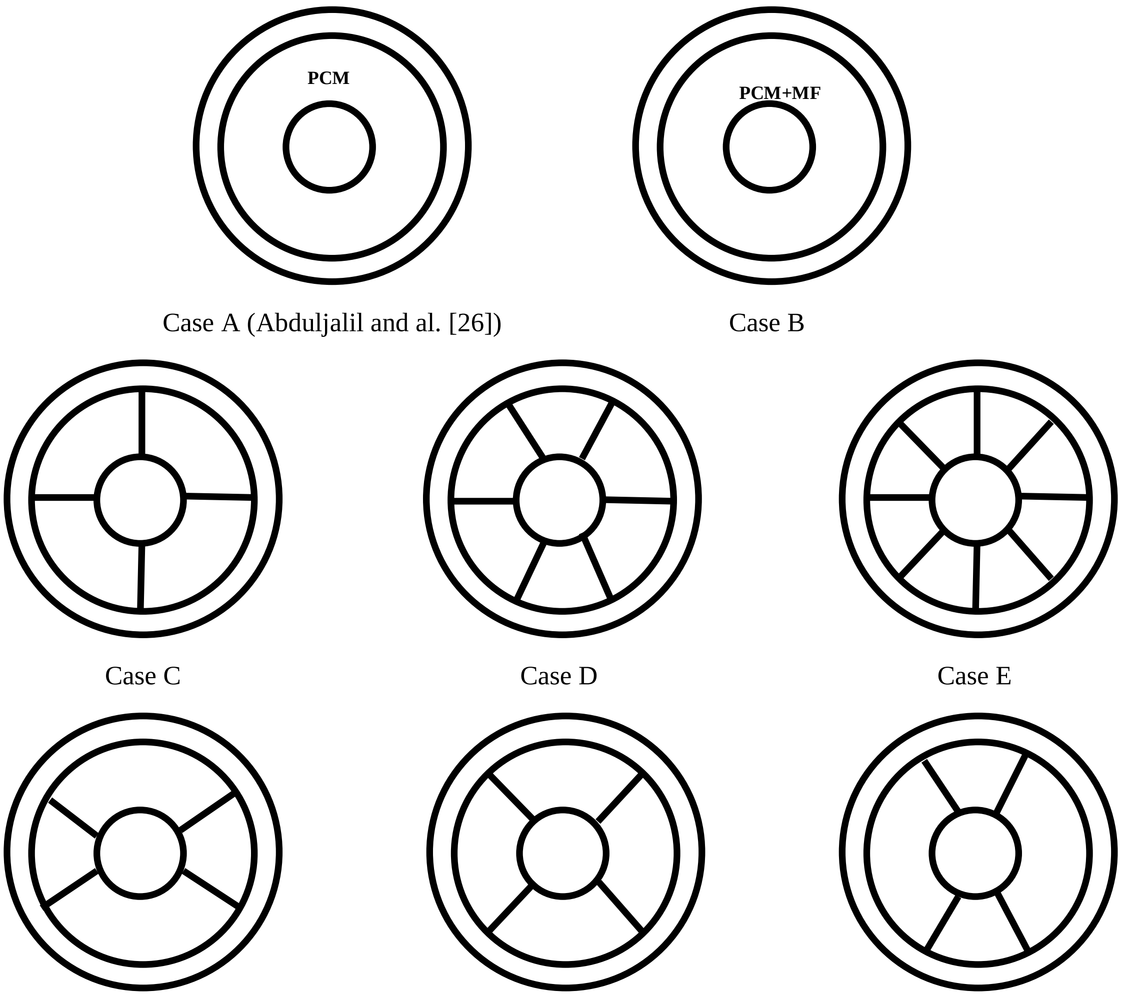

Fig. 1 Physical configurations of all cases

Table 1

Thermo-physical properties of materials

Property	RT 82	Copper
Density of solid, kg/m^3	950	8978
Density of liquid, kg/m^3	770	-
Specific heat, J/kg K	2000	381
Latent heat of fusion, J/kg	176000	-
Melting temperature, K	350.15-358.15	-
Thermal conductivity, W/m K	0.2	387.6
Thermal expansion coefficient, 1/K	0.001	-
Dynamic Viscosity,kg/m s	0.03499	-

+

- The continuity equation

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}=0. \tag{1}$$

- Momentum equations

$$\rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right)-\frac{\partial P}{\partial x}+S_x; \tag{2}$$

$$\rho\left(\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)-\frac{\partial P}{\partial y}+S_y; \tag{3}$$

$$S_x=\frac{(1-\beta)^2}{\left(\beta^3+\omega\right)}A_m u+\frac{\mu}{a}u+\frac{1}{2}C_i\rho u|u|; \tag{4}$$

$$S_y=\frac{(1-\beta)^2}{\left(\beta^3+\omega\right)}A_m v+\frac{\mu}{a}v+\frac{1}{2}C_i\rho v|v|-\rho g\gamma\left(T-T_0\right). \tag{5}$$

- Energy equations

For heat transfer, there are two models as given by Khashan et al. [28]: a single equation model and a model with two equations. There are researches that ignore the transfer of heat between the fluid and the solid phase [29, 30]. This is the thermal equilibrium model. There are other researches that assume that the fluid and porous media do not have the same temperature [31-34].This is the non equilibrium thermal model.

When the difference between the thermal conductivities of the fluid and porous media is important, it is preferable to use the non equilibrium thermal model [35].

In our work, we used metal foams which have thermal conductivities higher compared to the thermal conductivity of PCM, that way we used non equilibrium thermal model.

For the PCM:

$$\varepsilon(\rho c)_f\frac{\partial T_f}{\partial t}+\varepsilon(\rho c)_f\left(u\frac{\partial T_f}{\partial x}+v\frac{\partial T_f}{\partial y}\right)=\left(k_{fe}+k_{td}\right)\times$$
$$\times\left(\frac{\partial^2 T_f}{\partial x^2}+\frac{\partial T^2{}_f}{\partial y^2}\right)+h_{sf}A_{sf}\left(T_f-T_s\right)-\varepsilon\rho_p L\frac{\partial \beta}{\partial t}. \tag{6}$$

For porous media (metal foam):

$$(1-\varepsilon)(\rho c)_s\frac{\partial T_s}{\partial t}=$$
$$=k_{se}\left(\frac{\partial^2 T_s}{\partial x^2}+\frac{\partial T^2{}_s}{\partial y^2}\right)+h_{sf}A_{sf}\left(T_s-T_f\right). \tag{7}$$

- Permeability and inertia coefficient

For porous media that have a very high porosity, the permeability a and the inertia coefficient C_i can be determined as follows [36]:

$$a=0.00073(1-\varepsilon)^{-0.224}d_f^{-1.11}d_p^{0.89}; \tag{8}$$

$$C_i=0.00212(1-\varepsilon)^{-0.132}\left(\frac{d_f}{d_p}\right)^{-1.63}. \tag{9}$$

- Effective thermal conductivity and thermal dispersion

For the effective thermal conductivity, we used the model presented in the work of Boomsma and al. [37]:

$$k_{eff}=\frac{\sqrt{2}}{2\left(R_A+R_B+R_C+R_D\right)}; \tag{10}$$

$$R_A=\frac{4\sigma}{\left(2e^2+\pi\sigma(1-e)\right)k_m+\left(4-2e^2-\pi\sigma(1-e)\right)k_p}; \tag{11}$$

$$R_B=\frac{(e-2\sigma)^2}{(e-2\sigma)e^2k_m+\left(2e-4\sigma-(e-2\sigma)e^2\right)k_p}; \tag{12}$$

$$R_C=\frac{\left(\sqrt{2}-2e\right)^2}{2\pi\sigma^2\left(1-2e\sqrt{2}\right)k_m+2\left(\sqrt{2}-2e-\pi\sigma^2\left(1-2e\sqrt{2}\right)\right)k_p}; \tag{13}$$

$$R_D=\frac{2e}{e^2k_m+\left(4-e^2\right)k_p}; \tag{14}$$

$$\sigma=\sqrt{\frac{\sqrt{2}\left(2-\left(\frac{5}{8}\right)e^3\sqrt{2}-2\varepsilon\right)}{\pi\left(3-4e\sqrt{2}-e\right)}}; \tag{15}$$

$$e=0.339;$$

$$k_{fe}=k_{eff}\rightarrow k_m=0;$$

$$k_{se}=k_{eff}\rightarrow k_p=0.$$

Concerning thermal dispersion, we adopted the model determined by Georgiadis and Catton [38]:

$$k_{td} = \frac{0.36}{1-\varepsilon}\rho_p c_p d_f \left|\vec{V}\right|. \tag{16}$$

- Specific area

Calmidi and Mahajan [36] presented a model that allows us to calculate specific area based on geometric parameters of the metal foam:

$$a_{sf} = \frac{3\pi d_f \left(1-e^{-1-\varepsilon/0.04}\right)}{\left(0.59 d_p\right)^2}. \tag{17}$$

- Heat transfer coefficient interfacial

Zhao and al. [37] have adopted a model for the heat transfer coefficient between the fluid and the porous medium based on the Rayleigh number:

$$h_{sf} = \frac{k_f}{d_f}\left(0.36 + \frac{0.518 Ra^{1/4}}{\left(1+\left(\frac{0.559}{Pr}\right)^{9/16}\right)^{4/9}}\right), \tag{18}$$

where $Ra = g\gamma \Delta T d_f^3 / \left(\alpha_f \upsilon_f\right)$.

- Initial and boundary conditions

At t = 0, the phase change material has a temperature of 363 K.

The inner tube and the intermediate tube are maintained at a constant temperature of 338 K.

The outer tube is insulated.

Due to the symmetry, only half of the system will be studied.

4. Results and discussions

4.1. Numerical model validation

This section is dedicated to validate the numerical approach. The validation is conducted using numerical and experimental results of Tian and Zhao [35], who investigated the effect of the metal foam on improving the heat transfer in a phase change material. Rectangular copper foam is filled with PCM, and then heated by an electrical resistance.

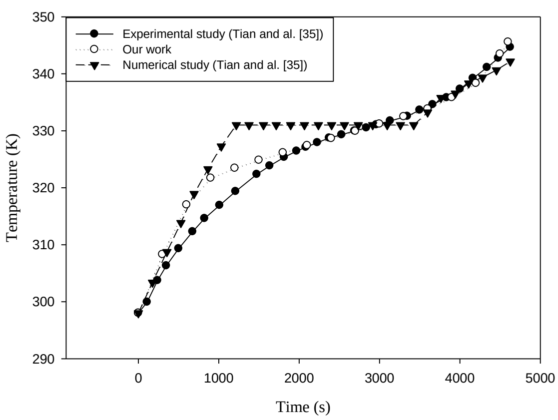

Fig. 2 Evolution of the PCM temperature versus time

Fig. 2 shows the temperature behaviour of the phase change material during the heating process. Indeed, we can see that results given by our numerical approach are much more closed to the experimental work carried out by Tian and Zhao [35], compared to his numerical analysis.

4.2. Choice of the mesh

In the previous section related to the validation study corresponding to rectangular foam, the mesh used is the same adopted by Tian and Zhao [35]: (50 × 200). However, further analyses are conducted in triple coaxial tubes. This requires a meshing optimisation analysis. For choosing an optimal mesh, different tests are performed with different numbers of cells: 12642, 14994, 18522, 21168 and 24696 cells. Fig. 3 shows the variation of the liquid fraction versus time for those numbers of cells. We noticed that the two curves (21468 cells and 24696 cells) overlap, that why we consider the optimal mesh using 21468 cells.

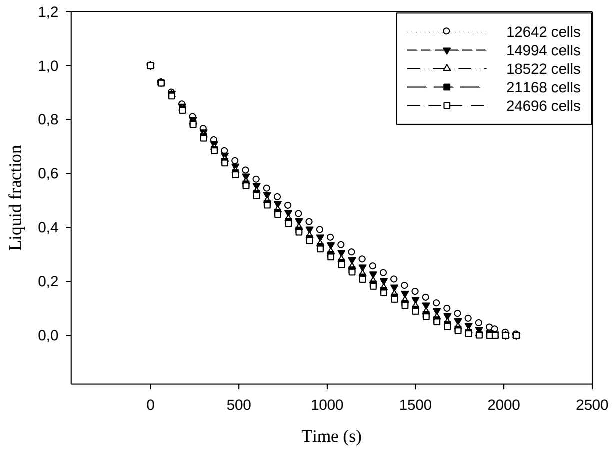

Fig. 3 Grid size dependence of the numerical solution

4.3. Influence of the porous medium on the phase change process

In this section, a comparison between two systems: heat exchanger without porous medium and heat exchanger with porous medium is conducted.

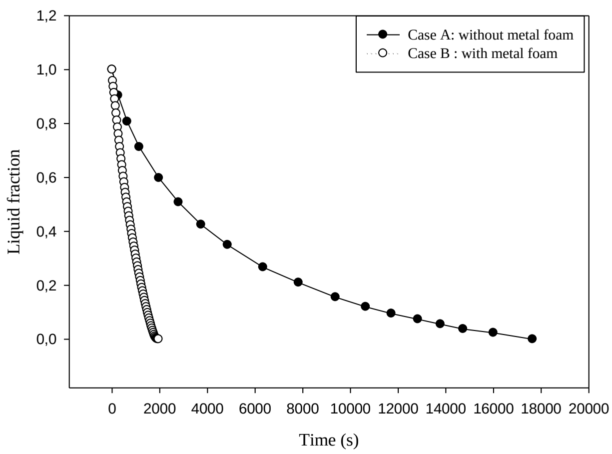

Fig. 4 Liquid fraction behaviour versus time

Fig. 4 shows the behaviour of the liquid fraction versus time for the two cases without and with porous medium. We note that the PCM solidifies more rapidly and reaches the complete solidification at $t = 1950$ seconds when the porous medium is used. For the case without porous medium, the complete solidification is obtained at $t = 17640$ seconds, which means that the in the first case, the solid state is reached 9 times more rapidly than the second case. We can easily conclude that using metal foam increase dramatically the heat exchange.

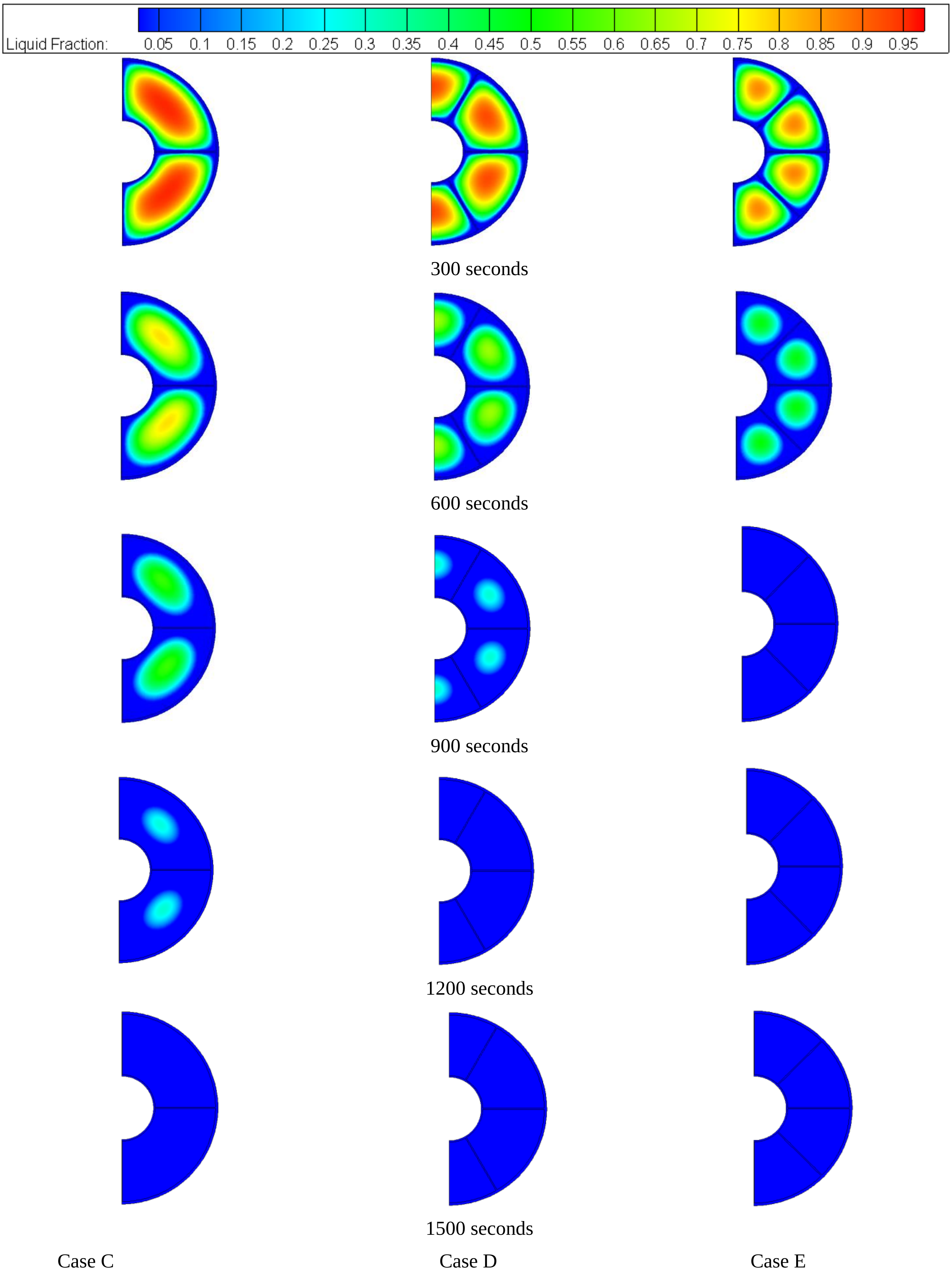

Fig. 5 Liquid fraction contours; case C: four fins; case D: six fins; case E: eight fins

4.4. Effect of fins

In this part, we will add fins to the porous medium to see their effect on the solidification process. The effect of number of fins and the fins position in the presence of the metal foam will be discussed.

Fig. 5 shows the outline of the liquid fraction for three different cases: heat exchanger with four fins (case C), heat exchanger with six fins (case D) and a heat exchanger with eight fins (case E). At $t = 0$, the PCM is in the liquid state and its temperature equal to 363 K. The temperature of the walls is lower than the solidification temperature and is equal to 338 K. At $t = 300$ seconds, a solid film is formed from the cold walls and from the fins. At $t = 900$ seconds, the PCM is completely solid for case E but remains 13% liquid for case C and 4% liquid for case D. At $t = 1200$ seconds, the PCM is completely solid to both D and E, but there is 2% liquid in case C. At $t = 1500$ seconds, the PCM is completely solid for all three cases.

So we can say that the PCM solidifies rapidly if the number of fins increases. This is due to the increase of the exchange surface.

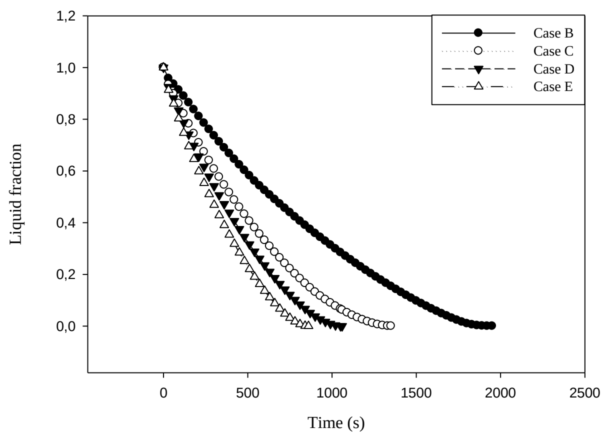

Fig. 6 Liquid fraction behaviour versus time for different numbers of fins

Fig. 6 shows the variation of the liquid fraction over time for four cases: B, C, D and E. It is noted that the solidification takes place rapidly by increasing the number of fins. PCM reaches the complete solidification at $t = 1950$ seconds for the case B (porous media only), at $t = 1350$ seconds for the case D, at $t = 1060$ seconds for the case E, and at $t = 860$ seconds for the case F.

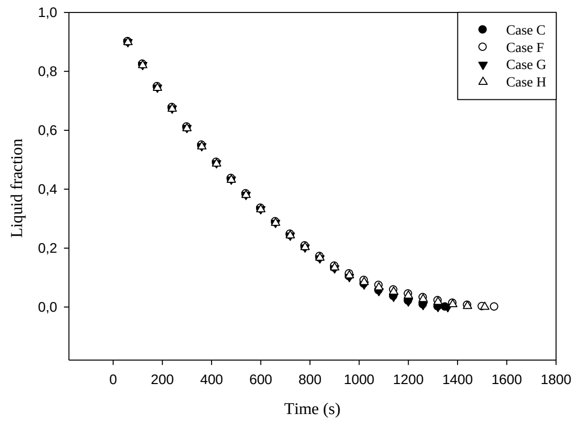

Fig. 7 Liquid fraction behaviour versus time for different positions of fins

Fig. 7 shows the variation of the liquid fraction over time for the four positions of the fins (case C, F, G and H). We see that the four curves are merged until $t = 1060$ seconds. After $t = 1060$ seconds, we see that there is a difference between the four cases. From the figure, it is found that the complete solidification is achieved in a shorter time (1350 seconds) by the case C.

4. Conclusions

To improve heat transfer in a heat exchanger with triple concentric tube, two enhancement techniques are used simultaneously: incorporating the PCM into metal foam and the addition of longitudinal fins. The non-thermal equilibrium model is used because of the great difference between the thermal conductivities of the PCM and the metal foam (copper). The results show that:

- The time of PCM solidification is reduced to 9 times using a porous medium.
- Compared with the case C (four fins), a heat exchanger with six fins leads to a reduction of 21.48% of the solidification time, and a heat exchanger with eight fins leads to a reduction of 36.29% of solidification time.
- The position of the fins has no great influence on the solidification process and the best result is achieved by the configuration C.

If the solidification time decreases, then the rate of energy release will be improved.

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STUDY OF THE SOLIDIFICATION PROCESS OF A PHASE CHANGE MATERIAL CONTAINED IN A HEAT EXCHANGER WITH THE PRESENCE OF A POROUS MEDIUM

S u m m a r y

The present study is conducted in order to evaluate the impact of a porous medium containing a phase change material (PCM) on the heat transfer behaviour in triple tube heat exchanger, using Ansys fluent software. The non-thermal equilibrium model is adopted. Different influences such as introducing a porous medium, the numbers of the fins and the fins position are studied. The time of PCM solidification is reduced to 9 times using a porous medium. Compared with a heat exchanger with a porous medium only, a heat exchanger with a porous medium and with fins results in a reduction of the solidification time. In addition, the free convection is eliminated. If the number of fins increases, the solidification time will be reduced. We found that the position of the fins has no great influence on the solidification process.

Keywords: heat exchanger, PCM, porous medium.

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