Numerical heat transfer analysis of the packed bed latent heat storage system based on an effective packed bed model

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1. Introduction

In many energy applications, the mismatch in the quantity, timing and location of energy supply and demand is often encountered, which prevents efficient and optimal use of the available resources. These energy applications include solar energy utilization, waste heat recovery, load leveling for power generation, building energy conservation, combined cooling, heating & power system and air conditioning applications. In particular, solar energy, a major renewable energy resource, is time-dependent, with intermittent characteristics, which requires an auxiliary system for smoothening the energy supply instability [1]. Energy storage provides a valuable solution for adjusting the mismatch and can incorporate great flexibilities into the energy system. Especially for the solar energy, its effective utilization is significantly dependent on the efficiency and effectiveness of the energy storage systems. The thermal energy storage techniques available include sensible and latent thermal energy storage (LTES) or both as in hybrid systems. LTES is particularly attractive due to its ability to provide high energy storage capacity and additionally, it has a narrow operating temperature range corresponding to the phase change [2]. Various LTES techniques have been developed and various encapsulations have been used in LTES systems which involve bulk storage, macro-encapsulation and micro-encapsulation. As a compromise between the heat storage density and the heat transfer rate, the macro-encapsulation is the dominant form the phase change material (PCM) may be encapsulated in cans, spheres and other commercially available geometries. In particular, the spherical geometry is one of the most preferred encapsulations because it can make the storage tank possess not only a higher packing density but also a larger area for heat transfer.

Several mathematical models have been reported in the literature for predicting the thermal and hydrodynamic performance of a packed bed LTES system. These models can be subdivided into two major groups. The first group is the single phase model where the solid (PCM spheres) and fluid phases in a packed bed are regarded as one phase. The second group is the two phase model where the packed bed is represented by two distinct phases, i.e., solid and fluid. Examples of the two phase model are the continuous solid phase model and the concentric dispersion model [3].

Details of these models are discussed below.

- Single phase model

In the single phase model, the instantaneous temperature of the solid and fluid phases is the same. The governing equation in the 2-D case is given as:
\[
\left[(1 - \epsilon)\rho_c c_s + \varepsilon\rho_f c_f \right] \frac{\partial \theta}{\partial t} + \rho_f c_f \frac{\partial T}{\partial x} = ks \frac{\partial^2 \theta}{\partial x^2} + h_{fp} \left( \theta|_{\eta=\eta_0} - T \right) - U_W a_W (T - T_0)
\]
for Eq. (1), \( x \) and \( r \) are the axial and radial coordinates inside the packed bed LTES tank, respectively.

- Concentric dispersion model

The concentric dispersion model considers the thermal conduction inside the solid spheres (i.e. thermal gradient in the solid spheres). The energy equation of the fluid phase can be written as:

\[
\varepsilon \rho_f c_f \left( \frac{\partial T}{\partial t} + \frac{\partial T}{\partial x} \right) = ks \frac{\partial^2 \theta}{\partial x^2} + h_{fp} \left( \theta|_{\eta=\eta_0} - T \right)
\]
whereas for the solid phase, it can be written in the following form:

\[
\rho_s c_s \frac{\partial \theta}{\partial t} = ks \frac{\partial^2 \theta}{\partial x^2} + \frac{2 \partial \theta}{\eta \eta_0}
\]
where \( T \) and \( \theta \) are the temperatures of the fluid phase and the solid phase, respectively. For Eq. (3), \( \eta \) is the radial coordinate inside the sphere.

When considering a temperature gradient inside the solid sphere, two additional boundary conditions are needed as shown as follows:

\[
k_s \frac{\partial \theta}{\partial \eta} = h_{fp} \left( T - \theta \right)|_{\eta=\eta_0}
\]
\[
\frac{\partial \theta}{\partial \eta} = 0 \quad \text{at} \quad \eta = 0
\]
which assumes a thermal equilibrium between the PCM spheres and the flowing fluid.

- Continuous solid phase model

In continuous solid phase model, the PCM spheres are assumed to behave as a continuous porous medium and not as a medium composed of independent and individual spheres. The energy equations for the fluid and solid phases can be respectively written as:

\[
\varepsilon \rho_f c_f \left( \frac{\partial T}{\partial t} + \frac{\partial T}{\partial x} \right) = ks \frac{\partial^2 T}{\partial x^2} + k_f \left( \frac{\partial^2 \theta}{\partial x^2} + \frac{\partial \theta}{\partial r} \right) + h_{fp} a_p (\theta - T)
\]
\[
\left(1 - \epsilon\right)\rho_c c_s \frac{\partial \theta}{\partial t} = k_s \frac{\partial^2 \theta}{\partial x^2} + k_s \left( \frac{\partial^2 \theta}{\partial r^2} + \frac{\partial \theta}{r \partial r} \right) + h_{fp} a_p (T - \theta)
\]
when the variation of the porosity with the radial distance from the wall is taken into account, a monotonic exponential expression is used to represent the radial distribution of the bed porosity [4]:

\[
\varepsilon(r) = \varepsilon_0 \left[ 1 + \left( \frac{0.87}{\varepsilon_\infty} - 1 \right) \exp \left( -\frac{5D}{2r} \right) \right]
\]
the radial distribution of the fluid velocity is calculated using the extended Brinkman equation [4]:

\[
\frac{\partial p}{\partial x} = -A(1 - \varepsilon(r))^2 \frac{\mu}{\varepsilon(r)^3} \frac{d^2 \varepsilon(r)}{r^2} - B(1 - \varepsilon(r)) \frac{\rho}{\varepsilon(r)^3} + \frac{\mu_{el}}{r} \frac{\partial}{\partial r} \left( \frac{\partial \varepsilon}{\partial r} \right)
\]

the fluid flow in the porous material can be also solved by using the momentum conservation equation, which is formulated as:

\[
\frac{\partial \varepsilon}{\partial t} \frac{\partial \varepsilon}{\partial x_t} + \frac{\rho}{\varepsilon} \frac{\partial \varepsilon}{\partial x_t} \frac{\partial \varepsilon}{\partial x_t} = -\frac{\partial p}{\partial x_t} + \frac{\partial \varepsilon}{\partial x_t} - \rho(\varepsilon(T - T_R))_t - \varepsilon \frac{\mu_{el}}{\rho} \frac{\partial \varepsilon}{\partial r} \]

Erek and Dincer [5] undertook a comprehensive numerical analysis using the concentric dispersion model with the empirical heat transfer coefficient correlation for studying the heat transfer behavior of an encapsulated ice LTES system. Arkar and Medved [4] developed a cylindrical LTES system containing spheres filled with paraffin. They used the continuous solid phase model to investigate the influence of PCM thermal property on the thermal response of the LTES system [6]. Benmansour et al. [7] used a continuous solid phase model to numerically investigate the behavior of an encapsulated ice LTES system. Seeniraj and Narasimhan [8] also used a continuous solid phase model to numerically investigate the influence of various parameters such as porosity, Stanton number and Stefan number on temperature, solidified fraction and energy storage. Comparing the utilization of these three models, the continuous solid phase model is more convenient than the concentric dispersion model and more accurate than the single phase model, so it has been extensively used to study the energy storage and retrieval in the packed bed [9-11].

In fact, it is quite difficult to investigate the fluid flow and heat transfer characteristics in such a complex arrangement of PCM spheres. Therefore, all the three models assume fluid flow among PCM spheres as a kind of flow in the porous material, where the variation in the cross-sectional area of flow passages can not be accurately reflected and an average flow rate of the fluid along the flow passages is used. As well known, accurate description of the flow field is the premise to correctly calculate the heat transfer between the PCM spheres and fluid. None of these models can provide the information of the shape of the flow passages and the accurate flow field as the fluid flows through the voids among the PCM spheres.

These packed bed models, except the concentric dispersion model, all fail to account for the thermal gradients inside the PCM spheres, and can not reflect the influence of the shape and the size of the spheres on the fluid flow and heat transfer. Especially, for the single phase model, it is obviously not accurate to assume the uniform instantaneous temperature of the solid as that of the fluid, because in the practical LTES systems, there is a large temperature difference between the solid and the fluid phases due to the large difference in the thermal conductivities. In addition, as for the single phase model and continuous solid phase model, since the PCM spheres are assumed to behave as a continuous medium instead of independent spheres, the thermal resistance of the encapsulation could not be properly dealt with. It is most notable that none of these models can describe the differences in the arrangement of PCM spheres if the packed bed LTES systems have the same loading ratio of the PCM.

Moreover, the accuracy of these models intensively depends on two parameters: the effective thermal conductivity of the porous material and total heat transfer coefficient between fluid and the porous PCM matrix. Usually, they are calculated by experimental correlations which are generally limited to a particular bed elements used in developing them and not versatile for all the packed beds.

In the present work, a packed bed model, which is different from the previous ones, was developed. This model, named ‘effective packed bed model’, can investigate the flow field as the fluid flows through the voids among the PCM spheres and can also account for the thermal gradients inside the PCM spheres. In particular, the model can describe the arrangement of PCM spheres in different LTES systems. The model does not involve any experimental correlations or empirical expressions (except for the treatment of the thermal conductivity of the liquid PCM during heat storage). Thus, the model can be regarded as a universal one, which may be applied to all the packed bed systems rather than limited to particular cases. The accuracy of the model was validated by comparing the numerical results with the experimental data in the references. As a demonstration of the effectiveness of the developed model, the model was used to investigate the influence of the arrangement of the PCM spheres and encapsulation of the PCM on the heat transfer performance of LTES bed, which is difficult or even impossible to be performed using the previous packed bed models.

2. Mathematical model

A typical packed bed LTES system consists of a cylindrical tank containing spheres filled with PCM and a fluid flows through the voids among the PCM spheres. For a clear illustration, a schematic of the packed bed LTES system is shown in Fig. 1.

Detailed modeling of the fluid flow and heat transfer in such a complex configuration is difficult. Therefore, the following assumptions are applied:

Fig. 2. Schematic of the transformation from 3-D model to 2-D model.
Due to the axial symmetrical configuration and boundary conditions in the packed bed LTES system, 2-D model is adopted. However, the direct treatment as a 2-D model will lead to no flow passage for the fluid due to the complex packing of the spheres and the contact between spheres. Hence, a further modification should be done based on the next assumption.

The heat transfer between spheres is negligible because the heat is only transported through the point-to-point contact between spheres. Thus, the voids among the PCM spheres, through which the fluid flows, are transformed from 3-D passages to similar 2-D ones, as shown in Fig. 2. This transformation should guarantee that the porosity in the 2-D physical model is equal to that in the practical 3-D packed bed. The porosity of packed bed in 3-D and 2-D physical models is defined by the following equations, respectively:

\[
\varepsilon_{3D} = \frac{V_{\text{PCM}}}{V_{\text{tank}}} = \frac{n_{3D} \cdot 4/3 \cdot \pi \cdot (d/2)^3}{\pi \cdot (D/2)^2 \cdot H} \tag{11}
\]

\[
\varepsilon_{2D} = \frac{S_{\text{PCM}}}{S_{\text{tank}}} = \frac{n_{2D} \cdot \pi \cdot (d/2)^3}{\pi \cdot (D/2)^2 \cdot H} \tag{12}
\]

• According to previous researches on the melting and freezing of PCM, thermal conduction is the major mechanism of heat transfer during freezing, whereas natural convection plays an important role during melting [12,13]. Due to the complexity in describing the natural convection during melting of PCM, most of the models only considered the thermal conduction in PCM, which resulted in a large deviation from the experimental results. Here, natural convection during the melting of PCM is taken into account by using the effective thermal conductivity of liquid PCM defined by the following equation [14,15]:

\[
k_{p,\text{cov}} = \frac{k_{p,\text{l}}}{C \cdot R_a^m} \tag{13}
\]

the values of constants C and m used for the numerical predictions are 0.18 and 0.25, respectively.

• The heat transfer fluid (HTF) is incompressible and it can be considered as a Newtonian fluid.

• The PCM is homogeneous and isotropic.

Based on the above assumptions, a mathematical model, describing the transient phase change process coupled with a forced convection heat transfer of HTF, is formulated in 2-D coordinates (as shown in Fig. 1):

For the HTF:

Continuity equation:

\[
\frac{\partial u_f}{\partial x} + \frac{\partial v_f}{\partial y} = 0 \tag{14}
\]
The enthalpy method is the most suitable and advantageous for solving the phase change problems, especially when the phase change does not take place at a temperature point but in a temperature range [17], because in this method the phase change interface need not be explicitly treated and can be determined from the enthalpy distribution. The energy equation based on enthalpy method assumes the liquid fraction ($\gamma$) to be zero inside the fully solid PCM regions, unity inside the fully liquid PCM regions, and between zero and one inside the phase change regions.

The parameter $\gamma$ is defined by the following relations:

$$\gamma = 0 \quad \text{if } T_p < T_s$$
$$\gamma = 1 \quad \text{if } T_p > T_s$$
$$\gamma = \frac{T_p - T_l}{T_s - T_l} \quad \text{if } T_s < T_p < T_l$$

In the energy equation (18), $h_p$ is the specific enthalpy defined as the sum of the sensible enthalpy $h_s$ and the enthalpy change $\gamma L$ during phase change, where $L$ is the latent heat of PCM.

The initial condition ($t = 0$) in the model is a constant temperature for the fluid and the PCM ($T_f = T_p = T_0$), while the boundary conditions ($t > 0$) are as follows:

At the entry to the storage bed, the temperature and velocity of the fluid can be written as:

- $T_f = T_{in}$, $u_f = u_{in}$ if entry, $t > 0$;
- At the exit, the outflow boundary is used which assumes the diffusion fluxes in the direction normal to the exit plane to be zero:

$$\frac{\partial T_f}{\partial y} = 0$$ at exit, $t > 0$;

The outside wall of the heat storage tank is adiabatic and the boundary condition can be written as:

$$\frac{\partial T_f}{\partial y} = 0$$ at exit, $t > 0$. 

In the study, the standard $k-\varepsilon$ model [16] is used for investigating the influence of the turbulent flow.
\[
\frac{\partial T_f}{\partial y} = 0, \frac{\partial T_f}{\partial x} = 0; \quad v = 0, u = 0 \quad \text{at the wall, } t > 0.
\]

In this study, a commercial CFD program Fluent 6.2 was used to carry out the numerical study. The governing integral equations for the conservation of mass and momentum and for energy and other scalars such as turbulence were solved through control-volume-based technique. Pressure and velocity fields were computed by Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) algorithms. The effects of time step and grid size on the solution were carefully examined in the preliminary calculations, where three time steps viz.: 0.005, 0.01 and 0.1 s, and three different grid sizes viz.: 0.001, 0.002 and 0.005 m, were tested. As a compromise between the calculating time and the accuracy, 0.01 s and 0.002 m were adopted as time step and grid size, respectively. Convergence of the solution was checked at each time step, with the convergence criterion of $10^{-6}$ for the velocity components and continuity equation and criterion of $10^{-8}$ for energy equation.

3. Results and discussion

3.1. Validation of the effective packed bed model

3.1.1. LTES system used for hot water supply

In order to prove the credibility of the developed mathematical model, a comparison of the numerical predictions and experimental data was conducted. A LTES system integrated with a constant temperature water bath/solar flat plate collector, built for hot water supply by Nallusamy et al. [18], was selected and their experimental results were used to validate the effective packed bed model. The stainless-steel LTES tank, insulated with glass wool of 50 mm thickness, had a capacity of 47 L (360 mm diameter and 460 mm height) to supply hot water to a family of 5–6 persons, in which 264 spheres with the 55 mm outer diameter were uniformly packed in eight layers and each layer was supported by a wire mesh. The paraffin with a melting temperature of 60 ± 1 °C was used as the PCM and water was used as the HTF. Firstly, a numerical analysis was performed to calculate the thermal response of the LTES system during heat storage. The initial temperature of the packed bed was 32 °C, and the inlet temperature of the hot water was constant at 70 °C and the mass flow rate was 2 L min⁻¹. Due to the small mass flow rate and the moderate packed density of the spheres, the flow of water was regarded as laminar flow with low Reynolds number ($Re_{\text{max}} = \frac{\rho v_{\text{max}} d}{\mu} = 250$), the maximal velocity presented at the minimal sectional area of the flowing passage between the PCM spheres.) During heat storage (i.e. during melting of the PCM), the effect of natural convection in the melted PCM was taken into account by replacing the thermal conductivity of liquid PCM with the effective thermal conductivity. A comparison of the measured and the calculated PCM temperature during the melting is shown in Fig. 3. It is seen that the PCM temperature increases gradually at the beginning of the sensible heat storage period, and
then remains nearly constant during phase change which is followed by a rapid temperature increase during heating the liquid PCM. The effective packed bed model allows for thermal gradients inside the PCM spheres, as shown in Fig. 4a, which could not be shown by the continuous solid phase model and the single phase model. Moreover, it can describe the flow of water through the voids among the PCM spheres, as shown in Fig. 4b, which could not be conducted by any previous models.

Moreover, the heat retrieval of the LTES system was also analyzed. The initial temperature of the packed bed was 70 °C and the inlet temperature of the cooling water was constant at 32 °C, and the mass flow rate was still 2 L min⁻¹. The water flow was also regarded to be laminar flow. During heat retrieval (i.e. during freezing of the PCM), the effect of natural convection is negligible. A comparison of the measured and the modeled PCM temperature during the freezing is shown in Fig. 5. It is seen that the temperature decreases fast until it reaches the PCM phase change temperature, then the temperature remains nearly constant for more than 30 min as the PCM releases its latent heat. The temperature decreases rapidly again during the following sensible heat retrieval of the solid PCM.

When the HTF flow was laminar, there is a good agreement between the present numerical results and the experimental data, and the deviation of them is within 3 °C. Moreover, it is worth mentioning that during the heat storage, only the analysis involving natural convection heat transfer mechanism can show better agreement with experimental measurements, whereas natural convection inside the PCM was always neglected in most of the previous literature.

3.1.2. LTES system used for ventilation

The numerical model was also validated with the experimental results of a ventilation system with LTES developed in Ref. [4]. In this system, the heat storage tank was cylindrical, with a diameter of 340 mm and a height of 1520 mm; it had 35 rows of spheres with a diameter of 50 mm and each was filled with RT20 paraffin. The average porosity of the packed bed was 0.388. Experiments were conducted at different flow rates: 166 m³ h⁻¹ during the freezing and 215 m³ h⁻¹ during the melting. Due to the higher Reynolds-number turbulent flow (Re_max = 8300 during heat storage, Re_max = 6400 during heat retrieval) in this LTES system, the assumption of laminar flow caused significant discrepancy between the numerical results and the experimental data, so the standard k-ε model was added into the effective packed bed model.

During heat storage, the initial temperature of the packed bed was 11 °C and the inlet temperature of air was constant at 35 °C. The comparison between the experimental measurement and numerical results of the effective packed bed model is shown in Fig. 6, where the temperature of the centre of two spheres: one in the 16th row and the other in the 35th row was monitored during the melting. When the k-ε model was integrated in the model, there is a good agreement between the present numerical results and the

![Fig. 8. Temperature profiles in the packed bed during heat storage after (a) 1 h, (b) 2 h, (c) 3 h, (d) 4 h.](image-url)
experimental data. However, the numerical results show a better agreement for the sphere in the 16th row than that in the 35th row. This is because the model neglected the heat leak near the outlet of the packed bed and consequently the calculated temperature of the sphere in the 35th row rose more rapidly than the experimental one. The numerical results of the effective packed bed model were also compared with those calculated by the continuous solid phase model in Ref. [4] (Fig. 6), which shows that the effective packed bed model had higher accuracy than the continuous solid phase model for revealing the thermal behaviors of the packed bed.

The numerical results of the flow field and the profile of the pressure drop are shown in Fig. 7, where the flow field (Fig. 7a) can be a reference for optimal arrangement of the PCM spheres, and the distribution of the pressure drop (Fig. 7b) can indicate the pressure loss as the HTF flows through the packed bed (in this case, the pressure drop is about 159 Pa). As shown in Fig. 7, a non-uniform radial distribution of bed porosity influences the radial distribution of the flow rate (Fig. 7a) and consequently affects the temperature distribution. The bed porosity near the wall is higher than that in the inner part of the bed, hence the flow rate of the fluid near the wall is higher compared with that in the centre of bed. Therefore, the spheres near the wall are heated intensely and the temperature rises more rapidly than those in the centre of the bed, as shown in Fig. 8. The temperature difference between PCM sphere and fluid shown in Fig. 8 is so large due to the lower thermal conductivity of PCM, which indicates the assumption of the single phase model to be not valid.

During heat retrieval, the initial temperature of the packed bed was 27 °C and the inlet temperature of air was constant at 9 °C. The comparison among the experimental measurement, numerical results of the effective packed bed model and the continuous solid phase model is shown in Fig. 9, in which the temperature of two spheres: one in the 16th row and the other in the 35th row was also monitored. A good agreement between the present numerical results and the experimental data is also observed (the maximal deviation between them do not exceed 1.5 °C), and the effective packed bed model once again shows better accuracy than the continuous solid phase model during heat retrieval. Since the thermal conduction in the PCM spheres is dominant and natural convection nearly does not play any role during the heat retrieval, the heat transfer rate in the LTES system is reduced and the duration of heat retrieval is longer than that of heat storage. The temperature profiles of the packed bed during heat retrieval at different time are shown in Fig. 10. Because flow field during the heat retrieval and storage is similar, the spheres near the wall are cooled intensely and the temperature drops more rapidly than in the centre of the bed.

It can be seen that the numerical results of the effective packed bed model were not only in good agreement with the experimental data but also the capability to capture the details of the fluid flow and heat transfer characteristics in the packed bed LTES system. So it can be used to optimize the design of the packed bed LTES system and to investigate the influence of some parameters on the performance of the packed bed LTES system, such as: the sphere size, flow rate, storage material, HTF inlet temperature, void fraction, heat loss, and operating mode of LTES system. Especially, it can be applied to investigate some parameters which are difficult for the other models to perform, such as arrangement of the spheres in the packed bed, encapsulation of PCM, among others.

3.2. Application of the effective packed bed model

The effective packed bed model developed in the present study was used to do a parametric investigation of a LTES bed. In the investigations, the same working media was adopted, i.e., the paraffin with a melting temperature of 60 ± 1 °C was used as the PCM and water was used as the HTF. The HTF flow was regarded as laminar flow.

3.2.1. Influence of the arrangement of spheres on the heat transfer performance of LTES bed

When the PCM spheres occupy the same tank volume, various arrangements of spheres will influence the heat transfer performance of LTES bed. Two arrangements: the random and the special packing with the same quantity of the PCM spheres were studied in an LTES system with a cylindrical tank of 500 mm diameter and 1000 mm height. The ratio of the tank volume occupied by PCM spheres with a diameter of 46 mm was 50%. Numerical calculations were conducted at a flow rate of 6 L min⁻¹ and an inlet fluid temperature of 70 °C. The flow field in the tank with randomly packed spheres, shown in Fig. 11a, was significantly different from that in the tank with specially packed spheres (Fig. 11b), due to the difference in the flow passages. The random packing reduced the sectional area of the flow passages, which led to an increase in the velocity of the HTF. As shown in Fig. 11, the maximal velocity of the HTF in the randomly packed bed was 7.63 × 10⁻³ m s⁻¹ which was higher than that 4.25 × 10⁻³ m s⁻¹ in the specially packed bed. The velocity of the HTF can further influence the heat transfer between the HTF and the surface of PCM spheres, and consequently the temperature of the HTF at the outlet of the two packed beds was significantly different (Fig. 12), with a maximal
Fig. 10. Temperature profiles of the packed bed during heat retrieval after (a) 1 h, (b) 3 h, (c) 5 h.

Fig. 11. Flow field of the randomly packed bed (a) and specially packed bed (b).
temperature difference of 5 °C. Moreover, the heat retrieval rate in the randomly packed bed was higher than that in the specially packed bed. Overall, the random packing has better heat transfer performance and is more favorable for heat retrieval in a LTES system.

3.2.2. Influence of the encapsulation of PCM on the heat transfer performance of LTES bed

The material and the thickness of the encapsulation drastically influence the thermal resistance (the relationship among them is expressed by Eq. (20) [19]), and consequently influence the heat transfer performance of the LTES system [20].

\[ R_{en} = \frac{\delta_{en}}{4\pi k_{en} r_{en,inner} r_{en,outer}} \] (20)

In this study, four types of encapsulations with different thermal resistance, shown in Table 1, were used to encapsulate the PCM and their effects on the heat transfer performance of the packed bed LTES system were investigated.

In this case, the tank was a metallic cylinder whose useful height was 1420 mm and diameter was 900 mm. The encapsulations had the same inner diameter of 70 mm and were filled with paraffin and the PCM spheres were randomly arranged in the tank. During heat retrieval, the flow rate of HTF was 19.5 L min\(^{-1}\) and the inlet temperature was constant at 70 °C. The fluid outlet temperature and the PCM temperature for the different encapsulations were compared and are shown in Fig. 13a and b, respectively. The heat transfer performance of the packed bed was influenced significantly by the encapsulations.

- The influence of the encapsulation material on the heat transfer performance of LTES system

When the encapsulation of the PCM had the same thickness, the stainless steel encapsulation always indicated a better heat transfer performance than the polyolefin one due to its higher thermal conductivity, showing a maximal increase of 5 °C in the temperature of water at the outlet, as seen from Fig. 13a. Moreover, in Fig. 13b, the freezing of the PCM in the stainless steel encapsulation is faster than that in the polyolefin encapsulation and the freezing duration of the former one has shown a nearly 15% reduction compared with the latter one, which is also attributed to the difference in the thermal conduction for different encapsulation materials.

- The influence of the encapsulation thickness on the heat transfer performance of LTES system

As for the polyolefin encapsulation, its thickness had a significant influence on the heat transfer performance of the LTES system, whereas the influence of the thickness was not so evident for the stainless steel encapsulation, as shown in Fig. 13. Because the thermal conductivity of the stainless steel was much higher than PCM, i.e., the mainly thermal resistance lay in the PCM, and as a result the decrease in the thickness of the stainless steel encapsulation could not cause the great decrease in the total thermal resistance and the significant improvement in the heat transfer performance.

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Four types of the encapsulations.</th>
</tr>
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<tbody>
<tr>
<td>Material</td>
<td>polyolefin</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>0.4 W m(^{-1}) K(^{-1})</td>
</tr>
<tr>
<td>Thickness</td>
<td>3 mm</td>
</tr>
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</table>

Fig. 12. Water temperature at the outlet of the packed bed with randomly and specially packed spheres.

Fig. 13. Influence of the encapsulation of PCM on the heat transfer performance of LTES bed (a) water temperature at the outlet of the packed bed (b) PCM temperature in the center of the LTES bed.
performance of the LTES system. However, the low thermal conductivity of polyolefin led to a large thermal resistance of the polyolefin encapsulation which accounted for a considerable part in total, hence, the decrease in the thickness of the polyolefin encapsulation could cause the great increase in the heat transfer performance of the LTES system.

4. Conclusions

An effective model to investigate the fluid flow and heat transfer in a packed bed was developed and validated with the experimental data available in the literature. This effective packed bed model was used to study the performance of a LTES system under various operating conditions viz.: randomly or specially packed bed, laminar or turbulent flow, heat storage or retrieval among others. The results indicated that the effective packed bed model can always provide accurate analytical results. In addition, the effective packed bed model is superior to other models in that it is versatile for various packed bed LTES systems; it is capable of showing in detail the flow field in the packed bed and the thermal gradients inside the PCM spheres.

Moreover, the effective packed bed model can be used to investigate the influence of some factors on the heat transfer performance of LTES bed which is difficult to be conducted by the previous packed bed models, such as the arrangement of PCM spheres and the encapsulation of PCM. The numerical results done with the effective packed bed model revealed the follows: 1. The heat retrieval rate in an LTES system with randomly packed bed is higher than that with specially packed bed. 2. The encapsulation of the PCM has a significant influence on the heat transfer of the LTES system. The freezing duration of the PCM spheres with stainless steel encapsulation was nearly 15% shorter than that with polyolefin encapsulation. The thickness of the polyolefin encapsulation has significant influence on the heat transfer performance of the LTES system, whereas it is not evident as for the stainless steel encapsulation.

The effective packed bed model would be one of the most preferred tools to optimize the design and operation of the packed bed LTES systems.

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