Analysis of a latent heat thermal energy storage unit with metal foam insert in both the HTF and PCM sides

X Chen1,2, X L Xia2,4, F Q Wang3, C Sun2 and R Q Liu1
1School of Mechatronics Engineering, Harbin Institute of Technology, 92, West Dazhi Street, Harbin 150001, China
2School of Energy Science and Engineering, Harbin Institute of Technology, 92, West Dazhi Street, Harbin 150001, China
3School of Automobile Engineering, Harbin Institute of Technology at Weihai, 2, West Wenhua Road, Weihai 264209, China

E-mail: xiaxl@hit.edu.cn

Abstract. Latent heat thermal energy storage (LHTES) systems using phase change material (PCM) have received significant research attention in numerous engineering fields. The transient heat transfer phenomenon inside a vertical LHTES unit is numerically investigated, with the paraffin as the PCM and water as the heat transfer fluid (HTF). As a performance enhancement technique, the metal foam insert is applied in both the HTF and PCM sides. The conjugate thermal model for the HTF/foam−wall−PCM/foam system is built, considering the non-equilibrium effect between the solid and fluid phases with two-temperature energy equation and the natural convection inside the PCM with Boussinesq approximation. The enthalpy-based method is employed to account for the solid-liquid phase change problem. The overall performance is compared with other three cases: no foam insert, foam insert in HTF side and foam insert in PCM side. Besides, parametric study is also conducted on the melting features, including the foam structural parameters and inlet conditions of HTF. The results show that foam insert in both sides accelerates the PCM melting effectively. The foam porosity and HTF inlet temperature play important roles in the overall heat transfer, whereas the pore density and HTF inlet velocity have limited effects on the melting rate. The findings can provide referential information for the design of a LHTES system.

1. Introduction

Energy storage is an effective way to bridge the demand and supply for the intermittent renewable sources, and the thermal energy storage (TES) plays a vital role. There are three types of thermal energy storage: sensible heat, latent heat and reversible thermochemical reaction. Among the three technologies, the latent heat thermal energy storage (LHTES), using phase change material (PCM), has received a considerable observation and has been used in a wide range of applications due to its high energy storage density. It provides the possibility of an isothermal process along the solid-liquid phase change, and allows a more compact, efficient and economical system to realize [1]. However, the key issue limiting the use of PCMs is their intrinsically low thermal conductivity, which restricts the heat transfer rate seriously and delays the melting and solidification processes [2]. In order to improve the heat transfer, different enhancing techniques have been proposed, and they fall into three levels [3]: (1) material level, such as addition of highly thermal conductive particles and infiltrating the PCM in porous matrix (metal foam, carbon foam, etc.); (2) component level, such as adding fins,
using enhanced heat transfer tube and macro-capsulation; (3) system level, such as adopting cascaded latent heat storage.

As a promising approach to increase the thermal conductivity of PCM, foam insert has been extensively studied both experimentally and numerically. And the porous materials/foams are reported to be efficient for heat transfer/thermal conductivity enhancement by 3–500 times [4]. Most of the existing investigations focus on the melting and solidification processes in an enclosure filled with PCM-foam composite to outline the performance improvement. The effects of various factors have been considered, including heating mode, PCM filling forms, geometric design and system orientation [5]. All the conclusions show that the presence of foam matrix can lead to a more uniform temperature distribution and remarkably reduce the phase change time.

Since the experimental tests cannot provide the detailed local heat transfer information, numerical methods are developed to simulate the transient thermal transport, which could be generally classified into two categories: microscopic (pore-scale) approach and macroscopic (volume-averaging) approach [6]. One important prerequisite for the former calculation is a geometric model of the pore-level structure (usually as a representative elemental volume), and the geometric model can either be idealized cell or directly reconstructed from CT scans. Although pore-scale simulation is more reasonable, the simulation is too time consuming to implement parametric analysis and optimization. In contrast, the macroscopic approach typically uses the effective transport coefficients to characterize the flow and heat transfer properties inside the foam structure, and employs either thermal equilibrium or non-equilibrium based models for the interaction between the foam matrix and PCM. Owing to the difference in the thermal diffusivity between the foam matrix and the PCM, local thermal equilibrium assumption between the two phases may be not suitable. Based on the two-temperature energy model, the phase change process of the composite has been simulated and compared with the experimental data or pore-scale simulation, which results in good agreement, such as Tian and Zhao [7], Zhang et al. [8], and Yang et al. [9]. Besides, through pore-scale test, the same conclusion that existence of local thermal non-equilibrium phenomenon can be drawn [10]. On the other hand, some researchers pointed out that the local thermal equilibrium between foam matrix and PCM may be acceptable [11-13]. Until now, there is no specific illustration with the conditions that the local thermal non-equilibrium assumption should be applied.

Furthermore, many researches have been working on the design of heat transfer fluid (HTF)-PCM heat exchanger as a LHTES system. Since the tubular heat exchangers are widely used in real industrial applications, the triplex tube or shell and tube heat exchangers are mostly investigated [14]. There are various operating and design parameters, such as mass flow rate, inlet temperature of HTF, direction of HTF flow, orientation of heat exchanger, number of tubes in shell and tube heat exchanger, and diameters of tubes, etc. The prior investigations regard that the total thermal resistance mainly depends on that of PCM. With the application of foam matrix in the PCM side, several groups have conducted experiments on the dynamic thermal behavior in a LHTES unit, such as Yang et al. [15], Martinelli et al. [16], Atal et al. [17] and Fleming et al. [18]. Additionally, the numerical studies are simultaneously carried out, which can be divided to two types. One is that simplifying the computation in HTF side and changing the system into a 2D physical problem, which is based on a high HTF mass flow rate and the relatively small heat transfer surface of tube [19-21]. In real industrial applications, the influence by the HTF must be taken into account in the design and analysis of storage unit, such as the decrement of HTF temperature along the flow direction as the heat is transferred to the PCM. However, the corresponding works are limited. Liu et al. [22] developed a numerical model to analyze the effects of influential factors on the storage unit thermal performance, which included the convective heat transfer of the HTF and the melting process of the PCM in porous media. The overall thermal performance of a metal foam-enhanced cascaded thermal energy storage was analyzed by Tian and Zhao [23]. Meng and Zhang [24] investigated the performance of a tube-in-tank LHTES unit during the heat charging and discharging processes. Just as the double-pipe heat exchanger filled with foam matrix in both the hot and cold fluid sides [25,26], inserting the metal foam in the HTF side may further increase the heat transfer efficiency for the LHTES unit.
In this study, a HTF/foam−wall−PCM/foam configuration is adopted as the LHTES unit. A numerical model is constructed by using the finite volume method and two-temperature energy model to analyze the melting process. The enthalpy-porosity method is utilized to simulate the solid–liquid phase change process. The effects of several influential factors on the overall process are examined, including foam porosity and pore density, material of solid matrix, HTF inlet temperature and velocity. These results could contribute to the design of thermal energy storage unit.

2. Physical and mathematical model
A vertically oriented concentric tube heat exchanger is selected as the LHTES unit in this study, as presented in figure 1. The geometric design is 0.5 m in height with an inner tube diameter of 40 mm and an outer tube diameter of 104 mm. The solid wall of inner tube is 2 mm in thickness. The annular space is filled with PCM and water as the HTF which flows from top to bottom in the inner tube during the charging process. Copper foam is inserted in both the inner and outer tubes to improve the heat transfer capabilities. The Paraffin RT 58 is used as the PCM. The thermophysical properties of these materials are listed in table 1. The heat transfer process begins when the hot HTF moves downward in the tube. Heat is transferred into the PCM through the solid wall from the HTF, and the PCM melts after reaching the melting temperature. The thermal energy is stored in the PCM sensibly and latently.

![Diagram of the LHTES unit](image)

**Figure 1.** Schematic diagram of the LHTES unit considered in this study.

**Table 1.** Thermophysical properties of the materials.

| Property                           | Paraffin RT 58 | water | Cu  |
|------------------------------------|----------------|-------|-----|
| Density (kg m⁻³)                   | 840            | 998.2 | 8978|
| Specific heat capacity (J kg⁻¹ K⁻¹)| 2100           | 4182  | 381 |
| Thermal conductivity (W m⁻¹ K⁻¹)   | 0.2            | 0.6   | 387.6|
| Dynamic viscosity (kg m⁻¹ s⁻¹)     | 0.0269         | 0.001003 |   |
| Latent heat (kJ kg⁻¹)              | 181            | –     | –   |
| Thermal expansion coefficient (K⁻¹)| 1.1×10⁻⁴       | –     | –   |
| Melting temperature range (K)      | 321–335        | –     | –   |
The HTF and liquid PCM are considered as impressible fluid, and the fluid flow is assumed to be laminar. PCM and metal foam are homogeneous and isotropic. The volumetric expansion of the PCM during phase change is neglected. The natural convection is considered using the Boussinesq approximation, and the two-temperature model is used to represent the energy exchange between the HTF/PCM and metal foam.

A 2D axisymmetric numerical model is established to investigate the transient heat transfer inside the LHTES unit coupling the two domains according to the above assumptions. The enthalpy-porosity method and melting model are used to describe the phase change process of PCM. The governing equations are formulated as below, consisting of the continuity, momentum, and energy equations.

For the HTF/foam heat transfer in the inner tube,

\[
\nabla \cdot \mathbf{U} = 0 \quad (1)
\]

\[
\frac{1}{\phi} \frac{\partial (\rho_f U)}{\partial \tau} + \frac{1}{\phi^2} \nabla (\rho_f U \cdot U) = -\nabla p + \nabla \cdot \left( \frac{\mu_f}{\phi} \nabla U \right) - \frac{\mu_f}{K} U - \frac{\rho_f C_f}{\sqrt{K}} |U|U + \rho_f g
\]

\[
\frac{\partial (\phi \rho_f c_f T_f)}{\partial \tau} + \nabla \cdot (\rho_f c_f U T_f) = \nabla \cdot (k_{f,eff} \nabla T_f) + h_{sf} \alpha_{sf} (T_f - T_f) \quad (3)
\]

\[
\frac{\partial [(1 - \phi) \rho_f c_f T_f]}{\partial \tau} = \nabla \cdot (k_{f,eff} \nabla T_f) + h_{sf} \alpha_{sf} (T_f - T_f) \quad (4)
\]

For the inner tube wall,

\[
\rho_w c_w \frac{\partial T_w}{\partial \tau} = \nabla \cdot (k_w \nabla T_w) \quad (5)
\]

For the PCM/foam composite in the outer tube,

\[
\nabla \cdot \mathbf{U} = 0 \quad (6)
\]

\[
\frac{1}{\delta} \frac{\partial (\rho_p U)}{\partial \tau} + \frac{1}{\delta^2} \nabla (\rho_p U \cdot U) = -\nabla p + \nabla \cdot \left( \frac{\mu_p}{\delta} \nabla U \right) - \frac{\mu_p}{K} U - \frac{\rho_p C_f}{\sqrt{K}} |U|U + \rho_p g \beta (T_p - T_{ref}) - A \frac{(1 - \gamma)^2}{\gamma^3 + \zeta} \quad (7)
\]

\[
\frac{\partial (\phi \rho_p c_p T_p)}{\partial \tau} + \nabla \cdot (\rho_p c_p U T_p) = \nabla \cdot (k_{p,eff} \nabla T_p) + h_{sf} a_{sf} (T_s - T_p) - \phi \rho_p \Delta H \frac{\partial \gamma}{\partial \tau} \quad (8)
\]

\[
\frac{\partial [(1 - \phi) \rho_p c_p T_p]}{\partial \tau} = \nabla \cdot (k_{p,eff} \nabla T_p) + h_{sf} a_{sf} (T_p - T_s) \quad (9)
\]

where \( \delta \) is the liquid fraction of PCM in the volume element and can be denoted by \( \delta = \phi' \), \( \gamma \) is the liquid fraction in the pore space, \( \phi \) is the porosity; \( \rho, c, \mu, \beta \) are the density, specific heat, dynamic viscosity and thermal expansion respectively; \( \mathbf{U}, T, p \) separately denote the velocity, temperature and pressure; \( k_w, k_{p,eff}, K \) and \( C_f \) are the thermal conductivity of tube wall, effective thermal conductivity, permeability and inertial coefficient for metal foam; \( a_{sf} \) and \( h_{sf} \) are the interfacial surface area and interstitial heat transfer coefficient; \( \tau \) and \( \Delta H \) is the heat transfer time and latent heat of PCM; \( A \) is the mushy zone constant, which is value between \( 10^4 \) and \( 10^7 \); \( \zeta \) is a small constant of about \( 10^{-4} \) to
avoid being divided by zero; the subscripts \( f \), \( p \), \( s \) and \( w \) represent the HTF, PCM, foam matrix and solid wall respectively.

The liquid fraction is determined by the following equation, and \( \gamma = 0 \) at solid phase and \( \gamma = 1 \) at liquid phase. When the paraffin is in the mushy region, the liquid fraction is within 0 and 1.

\[
\gamma = \begin{cases} 
0, & T_p < T_{m1} \\
(T_p - T_{m1}) / (T_{m2} - T_{m1}), & T_{m1} \leq T_p \leq T_{m2} \\
1, & T_p > T_{m2}
\end{cases}
\]  

(10)

where \( T_{m1} \) and \( T_{m2} \) are the lower and upper limits of the melting temperature.

The effective transport parameters related to the structure of metal foam are important for the simulation, such as permeability, heat transfer coefficient and effective thermal conductivity. There are several empirical expressions commonly used in literatures.

\[
K = \frac{0.00073(1 - \phi)^{-0.224}}{d_p^{1.11}} 
\]

(11)

\[
C_F = \frac{0.00212(1 - \phi)^{-0.132}}{d_p^{1.63}} 
\]

(12)

where the pore diameter is computed as \( d_p = 0.0254 / \omega_{p,\text{eq}} \) and \( \frac{d_f}{d_p} = 1.18 \sqrt{\frac{1 - \phi}{3\pi \left(1 - e^{-(1 - \phi)0.04}\right)}} \).

Besides, the tetrakaidecahedron cell model proposed by Boomsma and Poulikakos \[27\] is adopted to describe the effective thermal conductivity in the present study.

For the forced convection in the inner tube, the interstitial heat transfer coefficient is determined by the following equation.

\[
Nu_f = \frac{h_f d_f}{k_f} = \begin{cases} 
0.76Re^{0.4}Pr^{0.37}, & 1 \leq Re_d \leq 40 \\
0.52Re^{0.5}Pr^{0.37}, & 40 \leq Re_d \leq 10^3 \\
0.26Re^{0.6}Pr^{0.37}, & 10^3 \leq Re_d \leq 2 \times 10^5 
\end{cases} 
\]

(13)

where \( Re_d = \rho_d \mu d_f / \mu_f \), the equivalent pore diameter is \( d = (1 - e^{-(1 - \phi)0.04}) d_f \), the interfacial surface area is calculated as \( a_f = 3\pi d_f (1 - e^{-(1 - \phi)0.04}) / (0.59 d_f)^2 \).

The above expression of interstitial heat transfer coefficient deduced based on forced flow yields \( Nu_f = 0 \) as the \( Re_d \to 0 \), which does not correctly represent the conduction limit. For the natural convection in the annular space, the interstitial heat transfer considering the natural convection or pure conduction in the PCM-foam composite is determined by the following correlation.

\[
Nu_f = \frac{h_f d_f}{k_p} = \left[ 1 + \frac{4(1 - \phi)}{\phi} \right] + \frac{1}{2} (1 - \phi)^{0.5} Re^{0.6} Pr^{1/3} 
\]

(14)

where \( Re_p = \rho_p \mu d_f / \mu_p \).

The initial temperature of the storage unit is 300 K. Inlet velocity and temperature of the HTF are 0.05 m/s and 350 K. The outlet of the inner tube is set as \( p = 0 \) (with the reference pressure of 101 325 Pa).

3. Numerical method and validation

The governing equations combined with the initial and boundary conditions are discretized using the finite volume method in the unsteady solver of Fluent software, applying a two dimensional
axisymmetric model. The SIMPLE algorithm is employed in pressure–velocity coupling. The PRESTO scheme is applied as the discretization method for pressure. Several UDFs are used in the software to define the heat transfer coefficient, effective thermal conductivity, momentum source, etc. The convergence criterion for the residuals for continuity equation, velocity and energy equations are set as $10^{-6}$, $10^{-6}$ and $10^{-10}$, respectively. Three different meshes of 150×75, 200×85 and 250×105 grids and three time steps of 0.1 s, 0.2 s, 0.5 s are tested. After independent check, a grid system of 200×85 and time step of 0.2 s are used to balance the accuracy and the computational time.

The numerical method is checked against the literature using two cases. One is the melting simulation of the PCM-foam composite employing the two-temperature model and considering the natural convection. The problem under investigation is depicted in figure 2. The main parameters are $\phi_1=0.8$, $d_p/H=0.135$, $T^*=(T-T_c)/(T_h-T_c)$, $\tau^*=(\tau_k)/(\rho_p c_p H^2)$ and further details about the model are available in Krishnan et al [28]. The temperature profile versus time at the mid-height of domain is compared in the figure.

**Figure 2.** Comparison of temperature variation with results in [28]. (a) physical model and (b) temperature comparison.

The second case is the coupling heat transfer in a double-tube heat exchanger with both sides filled with metal foam. This simulation is to check the treatment of foam-solid interface at the inner tube.
wall. With the same operating parameters, the cross-sectional mean temperatures of inner fluid, outer fluid, and the inner pipe wall are calculated and presented in figure 3.

From the above figures, the results show a good agreement between the numerical predictions and the literature data.

4. Results and discussion

4.1. Performance comparison of four configurations
In order to access the contribution of foam insert to the heat transfer augmentation, four cases shown in figure 4(a) are compared, namely case I: foam insert in both HTF and PCM sides, case II: foam insert in HTF side, case III: foam insert in PCM side, and case IV: no foam insert. The structural parameters of copper foam are $\phi = 0.9$, 10 PPI. The time evolution of the liquid fraction is displayed in figure 4(b). Compared to the cases II and IV, the melting rate is much quicker for cases I and III where the PCM is embedded in metal foam. It demonstrates that the overall thermal resistance is dominated by that of PCM. The complete melting times of Case II and case IV have the same magnitude, as 24750 s for case II and 26850 s for case IV. Extra inserting metal foam in the HTF side has further accelerated the melting process (nearly 72.2%), as the complete melting time is 770 s for case I and 2770 s for case III. This phenomenon indicates that the insert of foam in the HTF side can also reduce the thermal resistance, being an effective way to shorten the melting process.

![Figure 4](image)

**Figure 4.** Comparison of four cases with different configurations of foam insert. (a) four cases considered and (b) liquid friction change with time.

For the case I, the variation of PCM temperature at different locations versus time is exhibited in figure 5. The points are A (0.037, 0.15), B (0.037, 0.25), C (0.037, 0.35), D (0.027, 0.25) and E (0.047, 0.25). The three points A, B and C have the same radius, and the similar temperature change trend can be observed. Point A reaches the melting temperature first due to the flow injects from top to bottom. The decrease of HTF temperature along the flow direction contributes to the difference in the temperature of three points (nearly 6 K), which means that the complete coupling heat transfer between the HTF and PCM sides should be considered. Therefore, the simplification of 2D problem may cause a deviation in the performance prediction. The points D, B and E have the same height, and the temperature decreases from D to E (with maximum difference of 16 K) which represents the heat flow direction.
4.2. Effect of the metal foam material
The influence of the material of metal foam on the melting fraction is shown in figure 6. The material investigated are copper, aluminum and nickel, which are widely used [4]. The foam parameters are $\phi=0.9$ and 10 PPI. Concerning the solid phases, the thermal diffusivities are respectively $1.1\times10^{-4}$ m$^2$/s, $8.5\times10^{-5}$ m$^2$/s and $2.2\times10^{-5}$ m$^2$/s, which leads to the least time needed for the PCM melting in Cu foam. The complete melting times are 770 s, 1265 s and 2543 s, which means that the Cu foam presents the best performance among the three materials.

4.3. Effect of the HTF inlet conditions
Two operating conditions, namely mass flow rate and inlet temperature of HTF, are the main influencing factors on the melting rates. As the mass flow rate which is related to velocity increases, the volumetric heat transfer between HTF and foam matrix increases, consequently improving the heat transfer to the PCM side by foam-tube wall conduction and HTF-tube wall convection. Increase in inlet velocity of HTF enhances the heat transfer rate and shortens the melting time, as shown in figure 7(a). For example, the melting times are 990 s at $u_\text{in} = 0.02$ m/s, 874 s at $u_\text{in} = 0.03$ m/s, 811 s at $u_\text{in} = 0.04$ m/s, and 770 s at $u_\text{in} = 0.05$ m/s. From the results, it indicates the melting time does not
present a noticeable decline.

In contrast, the difference between the phase change temperature of the PCM and inlet temperature of HTF is the driving force of heat transfer between PCM and HTF. Thus, the HTF inlet temperature has a strong effect on the heat transfer rate. The figure 7(b) demonstrates that as the temperature difference increases, the total melting time reduces apparently. For the four inlet temperatures (340 K, 350 K, 360 K and 370 K), the melting times are 1405 s, 770 s, 555 s and 442 s. The melting time is reduced by 68.5% as the inlet temperature increases from 340 K to 370 K. In comparison with the effect of HTF inlet velocity, the inlet temperature has greater effect on the over heat transfer.

![Graph](image)

**Figure 7.** Effect of inlet conditions of HTF on the melting process. (a) effect of the inlet velocity and (b) effect of the inlet temperature.

### 4.4. Effect of the foam structural parameters

Figure 8 displays the liquid fraction variation history with time with different foam structural parameters. With the pore density fixed at 10 PPI, the melting rate increases with the decrease of porosity. In the phase change, the heat conduction and natural convection are the two dominant factors. Lower porosity holds higher effective thermal conductivity, resulting in a fast melting process, despite the suppression of the natural convection because of low fraction of pore volume. The total melting times are 271 s, 415 s, 770 s and 1558 s for the porosities of 0.7, 0.8, 0.9 and 0.95. It achieves a substantial reduction in melting time (up to 82.6%) as the porosity decrease from 0.95 to 0.7. Besides, the difference in melting time becomes smaller as porosity decreases.

As shown in the figure, different pore densities with porosity of 0.9 have nearly the same melting times, which are 770 s, 750 s, 745 s and 744 s for 10 PPI, 20 PPI, 30 PPI and 40 PPI. The overall variation of liquid fraction versus time has no considerable change (only a little reduction) as the pore density increases from 10 PPI to 40 PPI. A higher value of PPI means a smaller pore size of metal foam, which leads to higher values of surface area per volume and flow resistance. As the pore density increase, the volumetric heat transfer between the liquid PCM and foam matrix increases, which speeds the melting process. However, the pore size descends, which restricts the natural convection. To summarize, the porosity of foam has a huge impact on the overall performance compared to pore density.
Figure 8. Effect of structural parameters of metal foam on the melting process. (a) effect of porosity and (b) effect of pore density.

5. Conclusions
A transient thermal model of a LHTES unit with metal foam insert in both the PCM and HTF sides is established, using the two-temperature model and Boussinesq approximation. Different design and operating parameters include inlet HTF conditions, foam structural parameters, material of foam, foam filling forms are analyzed. Based on the simulation results, the following conclusions can be drawn.

- Foam insert in both the HTF and PCM sides further reduces the overall thermal resistance, achieving a substantial reduction in the melting time up to 72.2% compared with foam insert only in the PCM side.
- Increasing the inlet HTF velocity and temperature accelerates the melting process, while the inlet temperature has a more significant influence, as a decrease by 68.5% in melting time is observed when the inlet temperature increases from 340K to 370K.
- As the foam porosity decreases, the melting time greatly shortens (nearly 82.6% is found here); moreover, the pore density has a minor impact on the melting rate.

Acknowledgments
This research is supported by the National Natural Science Foundation of China (No. 51536001, No. 51806046) and the China Postdoctoral Science Foundation (No. 2018M630350).

References
[1] Zhang H et al 2016 Thermal energy storage: Recent developments and practical aspects Prog. Energy Combust. Sci. 53 1-40
[2] Lin Y et al 2018 Review on thermal conductivity enhancement, thermal properties and applications of phase change materials in thermal energy storage Renew. Sustain. Energ. Rev. 82 2730-42
[3] Zhao Y et al 2018 Experimental study on the thermodynamic performance of cascaded latent heat storage in the heat charging process Energy 157 690-706
[4] Tauseef-ur-Rehman et al 2019 A critical review on heat transfer augmentation of phase change materials embedded with porous materials/foams Int. J. Heat Mass Transf. 135 649-73
[5] Zhu Z et al 2018 Transient performance of a PCM-based heat sink with a partially filled metal foam: Effects of the filling height ratio Appl. Therm. Eng. 128 966-72
[6] Abishek S 2018 Effect of microstructure on melting in metal-foam/paraffin composite phase change materials Int. J. Heat Mass Transf. 127 135-44
[7] Tian Y and Zhao C Y 2011 A numerical investigation of heat transfer in phase change materials
(PCMs) embedded in porous metals Energy 36 5539-46

[8] Zhang P et al 2017 Melting heat transfer characteristics of a composite phase change material fabricated by paraffin and metal foam Appl. Energy 185 1971-83

[9] Yang X et al 2018 Comparison of direct numerical simulation with volume-averaged method on composite phase change materials for thermal energy storage Appl. Energy 229 700-14

[10] Yao Y et al 2018 Pore-scale visualization and measurement of paraffin melting in high porosity open-cell metal foam Int. J. Therm. Sci. 123 73-85

[11] Feng S et al 2015 Unidirectional freezing of phase change materials saturated in open-cell metal foams Appl. Therm. Eng. 88 315-21

[12] Feng S et al 2015 Pore-scale and volume-averaged numerical simulations of melting phase change heat transfer in finned metal foams Int. J. Heat Mass Transf. 90 838-47

[13] Yang X et al 2018 Thermal and economic analysis of charging and discharging characteristics of composite phase change materials for cold storage Appl. Energy 225 585-99

[14] Kalapala L and Devanuri J K 2018 Influence of operational and design parameters on the performance of a PCM based heat exchanger for thermal energy storage – A review J. Energy Storage 20 497-519

[15] Yang J et al 2016 Experimental study on enhancement of thermal energy storage with phase-change material Appl. Energy 169 164-76

[16] Martinelli M et al 2016 Experimental study of a phase change thermal energy storage with copper foam Appl. Therm. Eng. 101 247-61

[17] Atal A et al 2016 Effect of porosity of conducting matrix on a phase change energy storage device Int. J. Heat Mass Transf. 93 9-16

[18] Fleming E et al 2015 Experimental and theoretical analysis of an aluminum foam enhanced phase change thermal storage unit Int. J. Heat Mass Transf. 82 273-81

[19] Esapour M et al 2018 Melting and solidification of PCM embedded in porous metal foam in horizontal multi-tube heat storage system Energy Convers. Manage. 171 398-410

[20] Mahdi J M and Nsofor E C 2017 Solidification enhancement in a triplex-tube latent heat energy storage system using nanoparticles-metal foam combination Energy 126 501-12

[21] Taghilou M et al 2018 Solid-liquid phase change investigation through a double pipe heat exchanger dealing with time-dependent boundary conditions Appl. Therm. Eng. 128 725-36

[22] Liu Z, Yao Y and Wu H 2013 Numerical modeling for solid–liquid phase change phenomena in porous media: Shell-and-tube type latent heat thermal energy storage Appl. Energy 112 1222-32

[23] Tian Y and Zhao C Y 2013 Thermal and exergetic analysis of metal foam-enhanced cascaded thermal energy storage (MF-CTES) Int. J. Heat Mass Transf. 58 86-96

[24] Meng Z N and Zhang P 2017 Experimental and numerical investigation of a tube-in-tank latent thermal energy storage unit using composite PCM Appl. Energy 190 524-39

[25] Chen X et al 2019 Conjugated heat transfer analysis of a foam filled double-pipe heat exchanger for high-temperature application Int. J. Heat Mass Transf. 134 1003-13

[26] Xu H J, Qu Z G and Tao W Q 2014 Numerical investigation on self-coupling heat transfer in a counter-flow double-pipe heat exchanger filled with metallic foams Appl. Therm. Eng. 66 43-54

[27] Boomsma K and Poulidakos D 2001 On the effective thermal conductivity of a three dimensionally structured fluid-saturated metal foam Int. J. Heat Mass Transf. 44 827-36

[28] Krishnan S, Murthy J Y and Garimella S V 2005 A two-temperature model for solid-liquid phase change in metal foams J. Heat Transf.-Trans. ASME 127 995-1004