INVESTIGATION OF THE DYNAMIC CHARACTERISTICS OF A THERMAL ENERGY STORAGE UNIT FILLED WITH MULTIPLE PHASE CHANGE MATERIALS

by

Xiaoyan LI *, Rongpeng HUANG, Xinyue MIAO, Xuelei WANG, Yabin LIU, and Xiaoya ZHANG

School of Energy and Building Engineering, Harbin University of Commerce, Harbin, China

Original scientific paper
https://doi.org/10.2298/TSCI171114037L

In order to improve the thermal performance of thermal energy storage systems, a packed bed thermal energy storage systems unit using spherical capsules filled with multiple phase change materials (multi-PCM) for use in conventional air-conditioning systems is presented. A 3-D mathematical model was established to investigate the charging characteristics of the thermal energy storage systems unit. The optimum proportion between the multi-PCM was identified. The effects of heat transfer fluid-flow rate and heat transfer fluid inlet temperature on the liquid phase change materials volume fraction, charging time and charging capacity of the thermal energy storage system unit are studied. The results indicate that the charging capacity of multi-PCM units is higher than that of the conventional single-PCM (HY-2). For proportions 0:1:0, 2:3:3, 3:2:3, 3:3:2, 4:1:3, and 4:2:2, the charging capacity decreases by approximately 24.84%, 14.69%, 6.47%, 3.82%, and 1.13%, respectively, compared to the 4:2:2 proportion. Moreover, decreasing the heat transfer fluid inlet temperature can obviously shorten the complete charging time of the thermal energy storage systems unit.

Key words: *cold storage air-conditioning, latent heat thermal energy storage, multiple phase change materials, dynamic characteristics*

Introduction

Conventional central air-conditioning systems equipped with thermal energy storage systems (TES) can use cheaper electric power at night to reduce electrical costs and alleviate the on-peak electricity load [1-3]. Latent heat storage (LHS) is considered as one of the most promising TES methods due to its much higher energy storage density when compared with the sensible heat storage [4]. At present, LHS units using ice have gained considerable attention in conventional air-conditioning systems. However, with the melting point of ice being 0 °C, the evaporation temperature and the coefficient of performance (*COP*) of these units are lower than those of conventional air-conditioning systems [5]. To deal with this issue, Li *et al.* [6] prepared a novel phase change material (PCM) whose melting temperature (8.5 °C) is higher than ice. However, PCM suitable for air-conditioning systems suffer from the disadvantages of low latent heat and the poor thermal conductivity, which limits the systems' charging capacity [7]. To over-

*Corresponding author, e-mail: mylxy6168@sina.com

come this problem, the use of multi-PCM with different melting temperatures in TES systems has been considered.

The multi-PCM TES systems have been applied in various fields, such as solar-based thermal power plants [8], waste heat recovery systems [9], and energy-efficient buildings [10]. The heat transfer in multi-PCM TES systems determines their thermal performance of such systems. Aldoss and Rahman [11] investigated the performance of a TES system based on a multi-PCM design of two and three stages. Chiu and Martin [12] analyzed the performance of a multistage latent heat thermal energy storage (LHTES) system using a cascade design of multi-PCM at various phase change temperatures. Peiro *et al*. [13] experimentally analyzed the advantages of multi-PCM compared to single PCM in TES systems at a pilot plant scale. However, these studies did not provide a further discussion on the effects of the operating parameters on the charging performance of the multi-PCM TES systems. Shaikh and Lafdi [14] investigated a combined convection-diffusion phase change heat transfer process in various configurations of composite PCM slabs based on a 2-D control volume based numerical method. Hu *et al*. [15] developed a 2-D control volume based numerical model to investigate the effects of the arrangement of multi-PCM, their melting temperature distribution and type on melting behavior and heat transfer. Wang *et al*. [16] developed a 2-D numerical simulation model of a LHTES unit with three high temperature PCM, simulating the effects of operating and geometric parameters on the LHTES. The result showed that the melting times decrease with an increase in the air inlet temperature. Meanwhile, an optimum length of the LHTES exists for achieving a high melting rate.

Several studies reported that the improvement of charging processes is a key issue to be solved for the development of TES systems. However, 2-D mathematical models can neither predict the local temperatures in the PCM well nor show the complete charging time when compared to 3-D analyses. Futhermore, few studies have dealt with the charging process of multi-PCM TES systems using 3-D modelling. Therefore, this paper presents a TES unit using spherical capsules filled with three kinds of PCM. More specifically, the multi-PCM TES unit is fitted to conventional air-conditioning systems. A 3-D simulation model is developed using ANSYS FLUENT to investigate the dynamic characteristics of multi-PCM TES unit. An optimum proportion between the multi-PCM is deeply studied and identified. The effect of different heat transfer fluid (HTF) inlet temperature and flow rates on the volume fraction and the charging capacity are investigated.

Physical and mathematic models

Physical model of the LHTES

The LHTES tank is composed of spherical capsules filled with three kinds of PCM. Water is used as the HTF. The multi-PCM are used and those with a higher phase change temperature are placed higher in the storage tank. The initial temperature of the PCM was set at 12 °C and the HTF temperature was kept at 2 °C during the charging process, while the fluid was flowing at different flow rates. The charging process was considered as finished once the volume fraction of the PCM was 0. Figure 1 illustrates the TES unit.

Mathematical model

The energy conservation equation is:

$$\frac{\partial}{\partial t}(\rho h) \qquad (\rho \vec{v} h) \qquad (k \quad T) \quad q_{\mathrm{r}} \tag{1}$$

where ρ is the density, h – the enthalpy, $\vec{\mathrm{v}}$ – the velocity of the fluid, k – the thermal conductivity. The q_r is the source term, and it is set to zero. Enthalpy h can be written:

$$h = h_0 + \int_{T_0}^{T} c\mathrm{d}T + \alpha L \tag{2}$$

where c is the specific heat, L – the latent heat of the PCM, and the volume fraction α is:

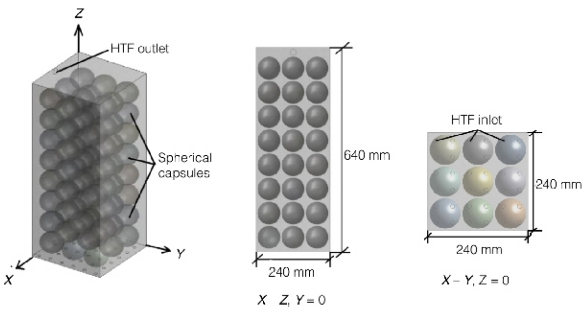

Figure 1. Physical model of the TES unit

$$\begin{aligned} a &= 0 & T &< T_\mathrm{solidus} \\ a &= 1 & T &> T_\mathrm{liquidus} \\ a &= \frac{T - T_\mathrm{solidus}}{T_\mathrm{liquidus} - T_\mathrm{solidus}} & T_\mathrm{solidus} &< T < T_\mathrm{liquidus} \end{aligned} \tag{3}$$

The momentum equation is:

$$\frac{\partial}{\partial t}(\rho\vec{\mathrm{v}}) + \nabla\cdot(\rho\vec{\mathrm{v}}\vec{\mathrm{v}}) = -\nabla p + \nabla\cdot[\mu(\nabla\vec{\mathrm{v}} + \nabla\vec{\mathrm{v}}^T)] + \rho\vec{\mathrm{g}} - \frac{(1-\alpha)^2}{\alpha^3 + \varepsilon}\vec{\mathrm{v}}A_\mathrm{mush} \tag{4}$$

where t is the time, p – the pressure, μ – the dynamic viscosity, $\rho\vec{\mathrm{g}}$ – the gravitational body force, A_mush – the mushy zone constant (1) [17], and ε – a constant (0.001) to avoid division by zero.

The charging capacity of the PCM in the TES tank can be calculated using:

$$Q_\mathrm{PCM} = \sum_{i=1}^{n} \rho_{\mathrm{PCM},i} V_{p,i} L_i + \int_{T_\mathrm{initial}}^{T_\mathrm{finally}} [\rho_\mathrm{PCM}(1-\alpha)V_\mathrm{p}c + \rho_\mathrm{PCM}\alpha V_\mathrm{p}c]\mathrm{d}T \tag{5}$$

where $\rho_{\mathrm{PCM},i}$ is the density of the i^th PCM, and L_i – the latent heat value of the i^th PCM, T_initial and T_final are the initial and final temperatures, respectively, and $V_{p,i}$ – the volume of the spherical capsule.

Results and discussion

Effects of different proportions of PCM on the thermal performance of the TES tank

In this paper, the PCM considered are HY-1, HY-2, and HI-1. Among them, HI-1 and HY-2 were prepared by our team. The phase change temperatures of HI-1 and HY-2 are 5.3 and 6.5 °C, and latent heat values are 271.2 and 226.2 kJ/kg, respectively. The HY-1 is the already existing material paraffin C15, whose phase change temperature and latent heat are 10 °C and 205 kJ/kg, respectively. Meanwhile, multi-PCM designs with volume ratios of 2:3:3, 3:2:3, 3:3:2, 4:1:3, and 4:2:2 (HI-1:HY-2:HY-1) were studied.

Figure 2 shows the effects of different proportions of PCM on the charging capacity of the TES tank. According to the figure, the charging capacity of TES tank using only HY-2 (*i. e.* volume ratio 0:1:0) is smaller than that of the other proportions. To find the optimum proportion between PCM, the complete charging time (*i. e.* the time needed to achieve a charging capacity of 95%) and the corre-

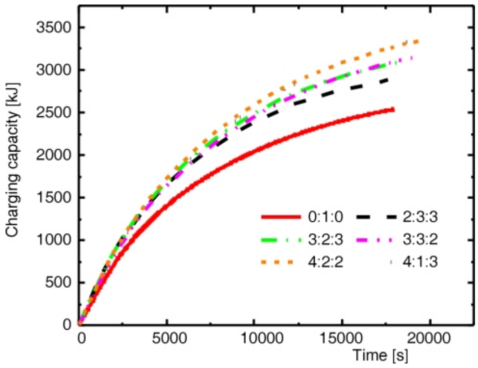

Figure 2. Charging capacity of different proportions of PCM

sponding charging capacity values are shown. When the proportions were 0:1:0, 2:3:3, 3:2:3, 3:3:2, 4:1:3, and 4:2:2, the corresponding complete charging times were 17941.8, 17580.2, 18469.3, 18992.5, 19521.8, and 19745.9 seconds, while the charging capacities were 2539.59, 2882.55, 3108.89, 3147.43, 3340.43, and 3379.11 kJ, respectively. For proportions 0:1:0, 2:3:3, 3:2:3, 3:3:2, 4:1:3, and 4:2:2, the charging capacity decreases by approximately 24.84%, 14.69%, 6.47%, 3.82%, and 1.13%, respectively, compared to the 4:2:2 proportion. The results indicate that the proportion 4:2:2 have the largest charging capacity. Therefore, the optimum proportion 4:2:2 is selected to investigate the dynamic characteristics of multi-PCM TES unit.

Effects of different HTF flow rates on charging time and charging capacity

Figure 3 shows the charging process of the PCM under different HTF flow rates, with the HTF inlet temperature being 2 °C. The figure illustrates the temperature fields of the cross-sections at different times. In fig. 3(a), the temperature of the PCM remains below 10 °C throughout, indicating that HY-1, whose phase change temperature is 10 °C, is completely frozen. Due to the

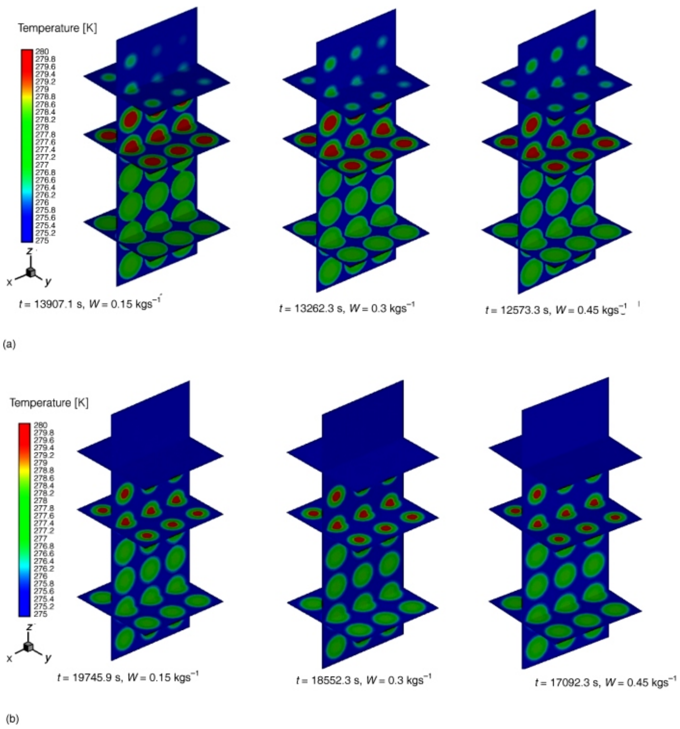

Figure 3. Effects of different HTF flow rates on the charging time; temperature distributions

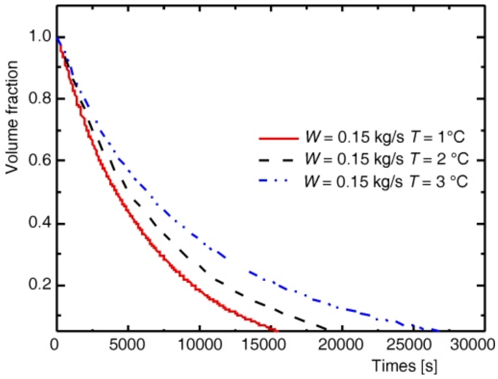

Figure 4. Volume fraction at different HTF inlet temperatures

phase change temperature of HY-1 being higher than that of HY-2 and HI-1, HY-1 is the fastest to completely freeze. Figure 3(b) is the temperature fields of the cross-sections at 95% of the total charging time. It takes 19745.9, 18552.3, and 17092.3 seconds to complete the charging process at the flow rates of 0.15, 0.3, and 0.45 kg/s, respectively. For the 0.3 kg/s and 0.45 kg/s cases, the time to reach the complete charging process is shorter by approximately 6.04%, and 13.44%, respectively, compared the 0.15 kg/s case. The results indicate that increasing the HTF flow rate contributes to the faster freezing of the PCM and shortens the necessary charging time, but the effect is not obvious.

Effects of different HTF inlet temperatures on the volume fraction

Figure 4 shows the effect of inlet temperature on the volume fraction. An HTF flow rate of $W = 0.15$ kg/s condition indicates that the volume fraction increases more rapidly when the inlet temperature is smaller during the charging process, and the volume fraction is a steady value during the final period. When the HTF inlet temperature are 1, 2, and 3 °C with an HTF flow rate of 0.15 kg/s, the charging processes end (*i. e.* reach a charging capacity of 95%) at 15518.9, 19745.9, and 25299.9 seconds, respectively. The complete charging time is reduced by decreasing the HTF inlet temperature; differences of 27.24% and 63.03% are, respectively, observed when the inlet temperature of HTF is 2 °C and 3 °C compared to 1 °C results.

Conclusions

The numerical results indicate that the 4:2:2 proportion yields the largest charging capacity. For this proportion, the charging capacity is increased by approximately 33.06%, 17.23%, 8.69%, 7.36%, and 1.16%, compared to the 0:1:0, 2:3:3, 3:2:3, 3:3:2, and 4:1:3 configurations, respectively.

Increasing the HTF flow rate can shorten the complete charging time. For the 0.3 kg/s and 0.45 kg/s cases, the time to reach the complete charging process is shortened by approximately 6.04%, and 13.44%, compared to 0.15 kg/s case. However, in order to reduce energy consumption, the HTF flow rate should remain within a certain range in practical applications.

The charging time decreases as the HTF inlet temperature decreases. The complete charging time is reduced by decreasing the HTF inlet temperature. Differences of 27.24% and 63.03% are respectively observed when the inlet temperature of the HTF is 2 °C and 3 °C compared to 1 °C results.

Acknowledgment

The authors acknowledge the support provided by the National Natural Science Foundation of China (51476049).

Nomenclature

A_{mush} – mushy zone constant
$\vec{g}$ – gravity vector, [ms^{-2}]
c – constant-pressure specific heat, [kJkg^{-1} °C^{-1}]
h – enthalpy, [kJ]
k – thermal conductivity, [Wm^{-1} °C^{-1}]
L – PCM latent heat, [kJkg^{-1}]
p – pressure, [Nm^{-2}]
Q – charging capacity of the TES unit, [kJ]
T – temperature, [°C]
t – time, [s]
$\vec{v}$ – velocity, [ms^{-1}]
W – flow rate, [kgs^{-1}]

Acronyms

COP – coefficient of operation performance
HTF – heat transfer fuid
LHS – latent heat storage
LHTES – latent heat thermal energy storage
multi-PCM – multiple phase change materials
PCM – phase change material
TES – thermal energy storage

Greek symbols

α – volume fraction
μ – dynamic viscosity coefficient, [Nm^{-2}s^{-1}]
ρ – density, [kgm^{-3}]

Subscripts

l – liquid
s – solid

References

[1] Rismanchi, B., *et al*., Energetic, Economic and Environmental Benefits of Utilizing the Ice Thermal Storage Systems for Office Building Applications, *Energy & Buildings, 50* (2012), 7, pp. 347-354
[2] Stathopoulos, N., *et al*., Air-PCM Heat Exchanger for Peak Load Management: Experimental and Simulation, *Solar Energy, 132* (2016), July, pp. 453-466
[3] Li, X. Y., *et al*., Investigation on the Dynamic LHTES
– latent heat thermal energy storagCharacteristics of a Direct Contact Thermal Energy Storage Charging Process for Use in Conventional Air-Conditioning Systems, *Applied Thermal Engineering, 91* (2015), Dec., pp. 172-180
[4] Mosaffa, A. H., *et al*., Thermal Performance of a Multiple PCM Thermal Storage Unit for Free Cooling, *Energy Conversion & Management, 67* (2013), 2, pp. 1-7
[5] Bourne, S., Novoselac, A., Improved Performance in Tube-Encapsulated Phase Change Thermal Energy Stores for HVAC Applications, *Building & Environment, 98* (2016), Mar., pp. 133-144
[6] Li, X. Y., *et al*., Investigation of the Dynamic Characteristics of a Storage Tank Discharging Process for Use in Conventional Air-Conditioning System, *Solar Energy, 96* (2013), 9, pp. 300-310
[7] Li, Z., Wu, Z. G., Analysis of HTF, PCM and Fins Effects on the Thermal Performance of Shell-Tube Thermal Energy Storage Units, *Solar Energy, 122* (2015), 2, pp. 382-395
[8] Seeniraj, R. V., Narasimhan, N. L., Performance Enhancement of a Solar Dynamic LHTS Module Having Both Fins and Multiple PCM, *Solar Energy, 82* (2008), 6, pp. 535-542
[9] Tessier, M. J., *et al*., Exergy Analysis of an Adiabatic Compressed Air Energy Storage System Using a Cascade of Phase Change Materials, *Energy, 106* (2016), July, pp. 528-534
[10] Liu, S., *et al*., Numerically Study the Performance of an Air-Multiple PCM Unit for Free Cooling and Ventilation, *Energy & Buildings, 151* (2017), Sept., pp. 520-533
[11] Aldoss, T. K., Rahman, M. M., Comparison between the Single-PCM and Multi-PCM Thermal Energy Storage Design, *Energy Conversion & Management, 83* (2014), 4, pp. 79-87
[12] Chiu, J. N. W., Martin, V., Multistage Latent Heat Cold Thermal Energy Storage Design Analysis, *Applied Energy, 112* (2013), 16, pp. 1438-1445
[13] Peiro, G., *et al*., Experimental Evaluation at Pilot Plant Scale of Multiple PCM (Cascaded) *vs*. Single PCM Configuration for Thermal Energy Storage, *Renewable Energy, 83* (2015), Nov., pp. 729-736
[14] Shaikh, S., Lafdi, K., Effect of Multiple Phase Change Materials (PCM) Slab Configurations on Thermal Energy Storage, *Energy Conversion & Management, 47* (2006), 15-16, pp. 2103-2117
[15] Hu, Z., *et al*., Enhanced Heat Transfer for PCM Melting in the Frustum-Shaped Unit with Multiple PCM, *Journal of Thermal Analysis & Calorimetry, 120* (2015), 2, pp. 1407-1416

[16] Wang, P., *et al*., Thermal Energy Charging Behaviour of a Heat Exchange Device with a Zigzag Plate Configuration Containing Multi-Phase-Change-Materials (m-PCM), *Applied Energy, 142* (2015), Mar., pp. 328-336
[17] Brent, A. D., *et al.*, Enthalpy-Porosity Technique for Modeling Convection-Diffusion Phase Change: Application to the Melting of a Pure Metal, *Numerical Heat Transfer Applications, 13* (1988), 3, pp. 297-318

Paper submitted: November 14, 2017
Paper revised: November 25, 2017
Paper accepted: November 26, 2017