Experimental and numerical study on a lab–scale latent heat storage prototype for cooling applications

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Abstract. Latent thermal energy storage systems using phase change materials (PCMs) represent an effective way of storing thermal energy because of high-energy storage density and the isothermal nature of the storage process. In the current study, the charging and discharging characteristics of a lab-scale latent heat storage (LH TES) prototype for cooling applications are experimentally and numerically studied. Two numerical models are developed to analyse the performance characteristics of the LH TES prototype: a conductive model and a conductive-convective model. Effective heat capacity (EHC) method is implemented to consider the latent heat of the phase change material. The governing equations involved in the models are solved using the finite element based software product, COMSOL Multiphysics, and the initial and the boundary conditions are determined on the basis of the data obtained from the experimental tests. Numerically predicted temperature variations of the models during charging and discharging processes are compared with the experimental data extracted from the lab-scale LH TES prototype, and a good agreement between them is found when the conductive-convective model is used, while high deviation is observed in case of use of the conductive model. Other results are presented in terms of the performance parameters such as charging/discharging time, energy storage charge/discharge rate, and melt fraction.

1. Introduction

Latent thermal energy storage systems (LHTES) using phase change materials (PCMs) are one of the most efficient methods to store thermal energy. The use of PCM provides higher heat storage capacity and more isothermal behavior during charging and discharging processes, compared to sensible heat storage systems [1]. Therefore, many authors have evaluated experimentally and numerically the LHTES systems performance, analyzing different storage configurations and comparing different numerical models [2-11]. Charvát et al. [12] studied the effect on a solar air-based thermal system of the use of a paraffin-based PCM as thermal energy storage material. Allouhi et al. [13] numerically studied the melting and solidification processes of a PCM integrated in a solar collector. Zhao et al. [14] analyzed different operation strategies for a solar heating system, including a thermal energy storage tank based on PCM. Bejarano et al. [15] numerically studied an innovative cold energy storage system based on PCMs. Cheng and Zhai [16] numerically and experimentally studied a cold thermal energy storage system composed of a packed bed with multiple PCMs. Aljehani et al. [17] simulated a phase change composite consisting of a paraffin wax and expanded graphite for cold thermal energy.
storage in air conditioning applications. Niyas et al. [18] analyzed experimentally two latent thermal energy storage systems based on PCM for medium and high temperature applications. In another work [19], the same authors studied numerically the performance of a lab-scale LHTES prototype for heat storage applications. Neumann et al. [20] developed a simplified approach for simulating latent thermal energy storage systems based on PCMs consisting of a fin-and-tubes heat exchanger. They compared the results from numerical simulations to the ones of experimental tests carried out on two different storages varying their geometric configuration. Seddegh et al. [21] developed and compared two numerical models for studying a shell-and-tube LHTES system: a conduction model and a conduction-convection model. They observed that the results obtained with the conduction-convection model agreed with the experimental results.

The aim of the current work is to study the charging and discharging characteristics of a novel lab-scale latent heat storage prototype for cooling applications, both by experimental and numerical simulation. In order to analyze the performance characteristics of the LHTES prototype, two numerical models are developed: a conductive model and a conductive-convective model. Effective heat capacity (EHC) method is implemented to consider the latent heat of the phase change material. The governing equations involved in the models are solved using the finite element based software product, COMSOL Multiphysics, and the initial and the boundary conditions are determined on the basis of the data obtained from the experimental tests. Numerically predicted temperature variations of the models during charging and discharging processes are compared with the experimental data extracted from the lab-scale LHTES prototype. Results of combined conduction-convection model agree well with the experimental data, while in case of conductive model, results show a high deviation between experimental and numerical data. Moreover, results show that, in the current application, the discharging process is a natural convection dominated process, while the charging one is a conduction dominated process.

2. Description of the experiment test
The employed experimental apparatus consists of a LHTES unit, a climatic chamber, five thermocouples, a data acquisition unit, and a computer. The storage unit consists of an aluminium cylindrical container of 25.0 cm height and radius equal to 6.9 cm filled with 2.40 kg of a biological PCM, which characteristics are shown in Table 1, as given by the producer. T-type thermocouples with an accuracy of ±0.5°C are used to measure the module local temperature at five points located on the same horizontal cross section placed at 9 cm from the container bottom: on the axis, and at four points at a distance of 3.45 cm from the axis arranged to form a cross. Fig. 1 shows the LHTES prototype filled with liquid PCM, and the climatic chamber used to carry out the experimental tests. The temperature data acquisition is conducted by means of the National Instruments NI 9213 acquisition module, using the NI cRIO 9066 controller. The experimental measurement data are recorded with sample time of 1 second.
The experimental tests are carried out by varying the air temperature inside the climatic chamber. For the charging process, starting from the uniform temperature condition $T_0=23.8^\circ$C, at which all the PCM is liquid, one hour temperature ramp is set up to bring the climatic chamber internal temperature to the charging temperature $T_c=7^\circ$C; then the temperature inside the climatic chamber is kept at $T_c$ for 72h. For the discharging process, one hour temperature ramp is set up to bring the climatic chamber internal temperature to the discharging temperature $T_f=23^\circ$C; then the temperature inside the climatic chamber is kept at $T_d$ until all the measured temperature inside the unit are well below to the phase change temperature.

2.1. Experimental results

Fig. 2 shows the temperature variation of the LHTES system obtained from the experiments. In detail, the variation of the local temperature at the central position and of the average of the temperatures measured with the four lateral thermocouples, for both charging and discharging process are reported. From these results, it can be seen that, during the charging process, the PCM undergoes solidification at a temperature slightly lower than 15°C. Similarly, it can be seen that, during the discharging process, the PCM starts melting at temperature lower than 15°C. From these figures, it can be also seen that the PCM temperature decrease during charging is slower than the PCM temperature increase during discharging, due to the addition of convective heat transfer model that increases the overall heat transfer rate inside the PCM during the discharging process.
3. Numerical modelling
The geometry of the storage prototype and the symmetric boundary conditions of the experimental tests allow to implement 2D axisymmetric numerical models. Therefore, two 2D axisymmetric numerical models are developed for simulating the charging and discharging process of the considered storage unit: a conductive-mechanism-based model, and a conductive-convective-mechanism-based model.

3.1. Conductive model
In order to perform the numerical simulation of the considered storage unit, in case of conductive model, the modelling is carried out through the energy equation of eq. (1), valid for both solid and liquid phases, and the following assumptions are considered [18]: (i) PCM is homogenous and isotropic; (ii) phase change during solidification/melting occurs in a temperature range; (iii) negligible convective mechanisms.

\[
\rho_{\text{PCM}} c_{p,\text{PCM}} \frac{\partial T}{\partial t} = \nabla \cdot (k_{\text{PCM}} \nabla T)
\]  

In eq. (1), \(\rho_{\text{PCM}}\) is the density of the PCM, \(T\) is the temperature, \(c_{p,\text{PCM}}\) is the heat capacity, and \(k_{\text{PCM}}\) is the thermal conductivity.

The PCM properties, such as \(k_{\text{PCM}}, \rho_{\text{PCM}},\) and \(c_{p,\text{PCM}}\) are considered constant over the entire domain, and fixed equal to the average value between the values of the liquid and solid phase.

To simulate the phase change process, the effective heat capacity method (EHC) is implemented, according to which the material effective heat capacity \(c_{p,\text{PCM}}\) depends on the latent heat of fusion \((L_h)\) of the PCM.

\[
c_{p,\text{PCM}} = c_p + L_h \frac{d\varphi(T)}{dT}
\]  

In eq. (2), \(\varphi(T)\) is a non-dimensional parameter that varies during the phase transition as follows:
\[ \varphi(T) = \begin{cases} 
0, & T < (T_M - \Delta T_M); \\
\frac{T - T_M + \Delta T_M}{2\Delta T_M}, & (T_M - \Delta T_M) \leq T \leq (T_M + \Delta T_M); \\
1, & T > (T_M + \Delta T_M) 
\end{cases} \] 

where \( \Delta T_M \) defines the temperature interval in which the phase change takes place.

In detail, \( \varphi(T) \) is 0 in the solid phase; \( \varphi(T) \) is 1 in the liquid phase, and \( 0 < \varphi(T) < 1 \) in the transition zone.

### 3.2. Conductive-convective model

In case convective mechanisms are considered, the developed thermal model is based on the following assumptions [18]: (i) PCM is homogenous and isotropic; (ii) phase change during solidification/melting occurs in a temperature range; (iii) the volume expansion during phase change is ignored; (iv) liquid PCM is Newtonian and treated with the laminar flow model. The modelling is carried out through the following equations:

- **Energy equation**
  \[
  \rho_{PCM} c_{p,PCM} \left( \frac{\partial T}{\partial t} + \vec{v} \cdot \nabla T \right) = \nabla \cdot (k_{PCM} \nabla T) \tag{4}
  \]

- **Continuity equation**
  \[
  \nabla \cdot \vec{v} = 0 \tag{5}
  \]

- **Momentum equation**
  \[
  \rho_{PCM} \left( \frac{\partial \vec{v}}{\partial t} + \vec{v} \nabla \cdot \vec{v} \right) = -\nabla p + \mu_{PCM} \nabla^2 \vec{v} + \vec{F}_b \tag{6}
  \]

where \( p \) is the pressure, \( \mu_{PCM} \) is the dynamic viscosity, and \( \vec{v} \) is the velocity vector.

In eq. (6), \( \vec{F}_b \) represents the Boussinesq approximation added to the momentum equation for including the buoyancy effects, and it is evaluated according to eq. (7).

\[
\vec{F}_b = \rho_{PCM} \bar{g} \beta (T - T_M) \tag{7}
\]

In eq. (7), \( \bar{g} \) and \( \beta \) are the gravitational acceleration and the thermal expansion coefficient, respectively.

Also in this case, the PCM properties, such as \( k_{PCM}, \rho_{PCM}, \) and \( c_{p,PCM} \) are considered constant over the entire domain, and to simulate the phase change process, the effective heat capacity method, presented in section 3.1, is implemented. As concerns the PCM dynamic viscosity \( \mu_{PCM} \), it is evaluated according to eq. (8), in order to impose a zero velocity in the solid region of the PCM.

\[
\mu_{PCM} = \mu_L (1 + S(T)) \tag{8}
\]

\[
S(T) = C \frac{(1 - \varphi(T))^2}{\varphi(T)^3 + \delta} \tag{9}
\]
In eq. (9), $C$ and $\delta$ are arbitrary constant: $\delta$ is usually equal to $10^{-3}$, and $C$, that defines the velocity variation into the phase transition zone, is usually between $10^3$ and $10^7$. $\delta$ is fixed according to ref. [18], while in order to fix the values of $C$, several tests are performed, for both charging and discharging process, by varying $C$ from $10^3$ to $10^{12}$. Among the considered values of $C$, the one that allows the best overlap of the numerical simulation results with the experimental ones is $10^{10}$ for the charging process and $10^{3.7}$ for the discharging one.

3.3. Initial and boundary condition

Initially ($t=0$), the temperature is fixed equal to the temperature measured at the start-up of the experimental tests, and, obviously only in the case in which convective mechanisms are included, a no-flow condition is set, i.e. the liquid PCM velocity is given zero over the entire domain. For each $t>0$, the boundary conditions, for charging and discharging process, are set according to the experimental test, described in section 2. The bottom and top surfaces of the storage unit are considered adiabatic, while the boundary condition relative to the lateral surface is set according to eq (10):

$$q_l = h_l (T - T_\infty), \quad r = r_{\text{max}}, 0 \leq l \leq l_{\text{max}}$$

In eq. (10), $q_l$ is the heat flux relative to the lateral surface of the storage unit, $h_l$ is the heat transfer coefficient relative to the lateral surface of the storage unit, $T_\infty$ is equal to the air temperature inside the climatic chamber, and $r$ and $l$ are the radius and the height of the storage unit, ranging from zero to a maximum value depending on the storage unit dimension.

The convective heat transfer $h_l$ is set at 27 W/(m$^2$K), evaluated according to the empirical correlation for vertical cylinder subjected to transverse flow under forced convection conditions of ref. [22].

3.4. Numerical treatment and mesh generation

The governing equations are solved with the finite element simulation software COMSOL Multiphysics 5.3a. The non-linearity of the problem is resolved through a segregated approach. The backward differentiation formula is adopted for the time stepping in both charging and discharging, by setting the simulation initial time step to $10^{-4}$ s and by using a no-fixed maximum time step. Triangle physics-controlled meshes are used, and for both developed model, a grid independence test is conducted. The simulations are performed with a Dell Precision T7610 workstation, equipped with two Intel Xeon E5-2687 w2 processors and a RAM of 64 GB and 1866 MHz clock. Fig. 3 shows the 2D meshed computational domain of the LHTES prototype in case of conductive-convective model. Moreover, the figure shows the points at which the temperature measurements are carried out to conduct the validation, or rather the central point, placed on the container axis, and indicated with $T_C$ and the lateral one, indicated with $T_L$. 
4. Results

In the following sub-sections, the validation, the grid independence test and various results obtained from the numerical simulations of the lab-scale LHTES prototype during charging and discharging process are presented and discussed.

4.1. Models validation

To validate the developed numerical models for the charging and discharging process, the variation of the local temperature at position $TC$ in the LHTES prototype, from simulations, is compared with the corresponding temperature values obtained from the experiments, while the variation of the local temperature at position $TL$ in the LHTES prototype, from simulations, is compared with the average temperature variation of the PCM obtained from the experiments.

For the charging process, the numerical simulations are carried out by setting the solidification temperature ($T_S$) at $13^\circ C$, while, for the discharging process, a melting temperature ($T_M$) of $15^\circ C$ is used. As for the parameter $C$, in order to fix the value of $\Delta T_{M}$, several tests are carried out, for both charging and discharging process, by varying $\Delta T_{M}$ from $0.5^\circ C$ to $6^\circ C$. Among the considered values of $\Delta T_{M}$, the one that produces the best overlap of the numerical simulation results and the experimental ones is $2^\circ C$ for the charging process and $5.5^\circ C$ for the discharging one.

Figures 4 and 5 show the comparison between the experimental and numerical values for charging and discharging process, respectively. From these figures, it can be noted that the results are much better, namely closer to the experimental ones, with the conductive-convective model, especially for the discharging process, where the convective heat transfer plays a major role.

It is clear that the conductive model is not suitable for describing the heat transfer phenomena for both charging and discharging process. For this reason, only the main results obtained in the case of the use of conductive-convective model are reported in the following.
4.2. Grid independence test

In order to test the independence of the numerical simulations results on the element size of the mesh, several simulations with different mesh element sizes are carried out for both charging and discharging process and for both the developed numerical models. In detail, meshes consisting of 948, 1780, 2788 and 5080 elements are considered for simulations. Fig. 6 shows the average temperature variation of the LHTES prototype during charging in case of conductive - convective model with the different mesh elements sizes. It is observed from Fig. 6 that the numerical simulations results are grid independent, being all the results quite overlapped. In this work, simulations are carried out considering a mesh composed of 2788 elements.
4.3. Melt fraction and charging/discharging time

Fig. 7 (a) and (b) shows the average melt fraction variation of the LHTES system during the charging and discharging process, respectively, evaluated as the volumetric average melt fraction of all mesh elements of the numerical model. Melt fraction is a key performance parameter which depicts the latent heat storage and discharge characteristics during the charging and discharging process. Moreover, the velocity variation of the melt fraction as function of temperature provides information about the velocity and the effectiveness of the charging and discharging processes.

It can be seen from Fig. 7 (a) and (b) that the average melt fraction of the LHTES system is 1/0 at the start of the charging/discharging processes, at \( t = 0 \text{ h} \), then it starts to decrease/increase. The solidification rate, during the charging process, is initially fast, then it decreases, whereas, due to the formation of the solid PCM on the prototype walls, that acts reducing the heat transfer rate. During the discharging, the melting rate is relatively fast for all process. The difference between the trends is mainly due to the contribution of natural convection heat transfer that occurs during the discharging process. As it can be seen from Fig. 7 (a) and (b), the charging process is much slower than the discharging one, indeed, the LHTES system is fully charged at about 50h, and discharged at about 15h.
4.4. Cooling thermal energy storage and discharge

Fig. 8 (a) and (b) shows the total cooling energy storage rate of the LHTES system during the charging and discharging process, respectively. For convention, the incoming cooling energy is assumed positive, whereas the outcome cooling energy is assumed negative. The total cooling energy stored ($E_{\text{stored}}$) and discharged ($E_{\text{discharged}}$) from the PCM, and its sensible ($E_{\text{sens,c}}$ and $E_{\text{sens,d}}$) and latent ($E_{\text{lat,c}}$ and $E_{\text{lat,d}}$) contributions, are key parameters for evaluating the thermal performances of the storage unit, and are calculated as the integral of the heat flux across the prototype boundaries.

Initially cooling energy gets stored/discharged in the form of sensible cooling energy only. Once the PCM reaches near the phase change temperature, cooling energy gets stored/discharged in the form of latent cooling energy. Similarly, after phase change of PCM, cooling energy is stored/discharged in the form of sensible cooling energy. The total cooling energy stored by the PCM is about 530 kJ, while the total cooling energy discharged by the PCM is about -488 kJ. As to the
sensible and latent contributions of the total cooling energy stored and discharged by the PCM, they are shown in Table 2.

Table 2. Sensible and latent contributions of cooling energy stored and discharged.

| Energy stored (kJ)          |       |
|----------------------------|-------|
| Sensible contribution      | 93    |
| Latent contribution        | 437   |
| Energy discharged (kJ)     |       |
| Sensible contribution      | -51   |
| Latent contribution        | -437  |

5. Conclusions

In this work, the charging and discharging characteristics of a lab-scale latent heat storage prototype for cooling applications are experimentally and numerically studied. Two numerical models are developed to analyse the performance characteristics of the LHTES prototype: a purely conductive model and a conductive-convective model. Performance parameters such as melt fraction, charging/discharging time, and energy storage/discharge rate are evaluated, and the following conclusions are derived:

- A better agreement with the experimental data are found with the conductive-convective model.
- Charging is a conduction dominated process, while discharging is a convection dominated process. For that, charging process is much slower than discharging one.
- The LHTES system takes about 50h to be fully charged, and about 15h to be fully discharged.
- The total amount of energy stored during charging is 530 kJ and during discharging is -488 kJ.

Future development of this work will concern the integration of the developed simulation models in a model for the simulation of cold water storage systems, in order to simulate cold storage systems using water and macro-encapsulated PCM as storage material.

6. Nomenclature

\( C \) Constant in eq. (11) \( \rho \) Density (kg/m³)
\( c_p \) Heat capacity (J/kg/K) \( c \) Charging
\( h \) Heat transfer coefficient (W/m²/K) \( c \) Condutive
\( k \) Thermal conductivity (W/m/K) \( c \) Convective
\( l \) Length of module (m) \( d \) Discharging
\( L_h \) Latent heat of fusion (kJ/kg) \( d \) Experimental
\( r \) Radius of module (m) \( L \) Liquid
\( t \) Time (h) \( l \) Lateral
\( TL \) Lateral position \( lat \) Latent
\( TC \) Central position \( M \) Melting
\( \Delta T \) Range of fusion (°C) \( num \) Numerical

Greek symbols

\( \delta \) Constant in eq. (9) \( S \) Solid
\( \mu \) Dynamic viscosity (Pa·s) \( s \) Superior
\( \phi \) Melt fraction \( sens \) Sensible

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References

[1] Oró E, de Gracia A, Castell A, Farid MM, Cabeza LF 2012 Review on phase change materials (PCMs) for cold thermal energy storage applications Applied Energy 99 513–533.

[2] Farid MM, Hamad FA, Abu-Arabi M 1998 Melting and solidification in multidimensional geometry and presence of more than one interface Energy Conversion and Management 39 809–818.

[3] Lacroix M 1993 Numerical simulation of a shell-and-tube latent heat thermal energy storage unit Solar Energy 50 357–367.

[4] Ng KW, Gong ZX, Mujumdar AS 1998 Heat transfer in free convection-dominated melting of a phase change material in a horizontal annulus International Communications in Heat and Mass Transfer 25 631–640.

[5] Lamberg P, Lehtiniemi R, Henell AM 2004 Numerical and experimental investigation of melting and freezing processes in phase change material storage International Journal of Thermal Science 43 277–287.

[6] Esapour M, Hosseini MJ, Ranjar AA, Pahamli Y, Bahrampouri R 2016 Phase change in multi-tube heat exchangers Renewable Energy 85 1017–1025.

[7] Zhang Y, Du K, He JP, Yang L, Li YJ 2014 Impact factors analysis of the enthalpy method and the effective heat capacity method on the transient nonlinear heat transfer in phase change materials (PCMs) Numerical Heat Transfer Part A 65 66–83.

[8] Allouche Y, Varga S, Bouden C, Oliveira AC 2016 Validation of a CFD model for the simulation of heat transfer in a tubes-in-tank PCM storage unit Renewable Energy 89 371–379.

[9] Dutil Y, Rousse DR, Salah NB, Lassue S, Zalewski L 2011 A review on phase-change materials Mathematical modeling and simulations Renew. Sustain. Energy Rev. 15 112–130.

[10] De Gracia A, Cabeza LF 2017 Numerical simulation of a PCM packed bed system: A review Renew. Sustain. Energy Rev. 69 1055–1063.

[11] Al-abidi AA, Mat SB, Sopian K, Sulaiman MY, Mohammed AT 2013 CFD applications for latent heat thermal energy storage: A review. Renew. Sustain. Energy Rev. 20 353–363.

[12] Charvát P, Klimeš L, Ostry M 2014 Numerical and experimental investigation of a PCM-based thermal storage unit for solar air systems Energy and Buildings 68 488–497.

[13] Allouhi A, Ait Msaad A, Benzakour Amine M, Saidur R, Mahdaoui M, Kousksou T, Pandey AK, Jamil A, Moujibi N, Benbassou A 2018 Optimization of melting and solidification processes of PCM: Application to integrated collector storage solar water heater, (ICSSWH) Solar Energy 171 562–570.

[14] Zhao J, Ji Y, Yuan Y, Zhang Z, Lu J 2017 Seven Operation Modes and Simulation Models of Solar Heating System with PCM Storage Tank Energies 10 21-28.

[15] Bejarano G, Suffo JJ, Vargas M, Ortega MG 2018 Novel scheme for a PCM-based cold energy storage system. Design, modelling, and simulation Applied Thermal Engineering 132 256–274.

[16] Cheng X, Zhai X, Wang R 2018 Thermal performance analysis and optimization of a cascaded packed bed cool thermal energy storage unit using multiple phase change materials Applied Energy 215 566–576.

[17] Aljehani A, Razack SAK, Nitsche L, Al-Hallaj S 2018 Design and optimization of a hybrid air conditioning system with thermal energy storage using phase change composite Energy Conversion and Management 169 404–418.
[18] Niyas H, Rao CRC, Muthukumar P 2017 Performance investigation of a lab-scale latent heat storage prototype - Experimental results Solar Energy 155 971-984.

[19] Niyas H, Prasad S, Muthukumar P 2017 Performance investigation of a lab–scale latent heat storage prototype – Numerical results Energy Conversion and Management 135 188–199.

[20] Neumann H, Palomba V, Frazzica A, Seiler D, Wittstadt U, Gschwander S, Restuccia G 2017 A simplified approach for modelling latent heat storages: Application and validation on two different fin-and-tubes heat exchangers Applied Thermal Engineering 125 41-52.

[21] Joybari MM, Haghigfat F, Seddegh S 2017 Natural convection characterization during melting of phase change materials: Development of a simplified front tracking method Solar Energy 158 711-720.

[22] Baehr HD, Stephan K 2006 Heat and Mass Transfer, second edition Springer.