Recueil De Mécanique
Centre Universitaire
El Wancharissi de Tissemsilt

Research Paper DOI : 10.5281/zenodo.1490505 Open access

Numerical investigation of latent heat thermal energy storage system

Investigation numérique d'un système de stockage thermique par chaleur latente

Mohamed Lamine BENLEKKAM[*a,b,] **, *Driss NEHARI***[b]**, *Nassira CHERIET***[a]

[a] *Institute of Science and Technology University Center El Wancharissi of Tissemsilt, BP 180 Tissemsilt 38000, Algeria*

[b] *Smart Structure Laboratory, University Center of AinTemouchent,46000, Algeria*

A R T I C L E I N F O

Article history :

Received 06 Jully 18

Received in revised form 24 September 18

Accepted 27 September 18

Keywords:

Thermal storage; latent heat; PCM; Heat exchanger

Mots clés:

Stockage thermique; chaleur latente, MCP; échangeur de chaleur

A B S T R A C T

The present work aims to study numerically the performance of a latent heat thermal energy storage unit. Which is composed of shell and tube. The annular space is filled by a phase change material (PCM), however the water used as a heat transfer fluid (HTF) flows in the inner tube. The computational are based on an iterative numerical procedure that incorporates an enthalpy formulation for the modeling of the solid-liquid phase change. Then our numerical model was validated with experimental and numerical results of the literature, where a good agreement was obtained. A series of numerical computations was conducted to study the effect of different flow parameters: the inlet temperature (88°C, 75°C and 65°C) and the mass flow rate (0.072, 0.8 and 0.1 kg/mn) of HTF on the thermal energy storage. The obtained results shows that the variation of the inlet HTF temperature has a great effect on the thermal performance of the storage unit, relative to the variation of the mass flow in terms of the operating time of the storage (charging and discharging) and the outlet temperature of the heat transfer fluid.

RÉSUMÉ

Le présent travail représente une étude numérique des performances thermiques d'une unité de stockage thermique par chaleur latente. Cette dernière est composée par deux tubes concentriques ou l'espace annulaire est rempli par un matériau à changement de phase (MCP), tandis qu'un fluide caloporteur (eau) s'écoule dans le tube central. Les calculs sont basés sur une méthode numérique itérative qui adopte une formulation d'enthalpie pour la modélisation du changement de phase solide-liquide. Par la suite, ce modèle a été validé avec des résultats expérimentaux et numériques ou une bonne concordance a été obtenue pour les deux cycles de charge et décharge. Une étude paramétrique a été conduit dont le but est de voir l'influence de plusieurs paramètres de écoulements à savoir la température d'entrée de l'eau (88°C, 75°C et 65°C) et son débit massique (0.072, 0.8 et 0.1 kg/mn). Les résultats obtenus montrent que la variation de la température d'entrée du fluide caloporteur a grand effet sur les performances thermique de l'unité de stockage par rapport la variation du débit en terme du temps de fonctionnement de l'unité ainsi la température de sortie.

1 Introduction

During the last years, the increase of greenhouse gas emissions, it is important to adapt to renewable energies and use them more efficiently. Among the most effective solutions is to develop thermal energy storage systems. Indeed, the storage is a very important technique for managing the use of thermal energy in various fields (domestic or industrial). In general, there are three types of thermal energy storage: sensible heat, latent heat, and thermochemical heat. Among these types of energy storage, the most attractive form is latent heat storage using a phase change material (PCM) due to their thermophysical properties: latent heat, heat recovery with low temperature drop and chemical stability.

* *Corresponding author. Tel.: +213 780497444.*
E-mail address: mohamed_benlekkam@yahoo.fr

Indeed, latent heat thermal energy storage has received significant research attention in the recent last years due to the high storage density and minimal energy loss during its operation at almost constant temperature according to Kapsalis et al. [1]. However, the low thermal conductivity of phase change materials used in storage systems has hindered their widespread applications [2-4]. A significant amount of research has been devoted to improving heat transfer in storage systems through a better understanding of phase change heat transfer mechanisms, and subsequently to an improved design. Jegadheeswaran et al. [5] examined several approaches to improve the performance of storage systems and the different geometries proposed for such system. Shell and tube systems have been found to be the most frequently studied geometry in the literature. However, depending on the geometric forms of storage tank (rectangular or cylindrical), the cylindrical ones take a very short storage time for the same thermal conditions compared to the rectangular ones Tari et al. [6].

Kurklu et al. [7] have realized a new type of solar collector that has two sections: the first is filled with water and the other with a phase change material whose melting temperature is between 45-50°C. Experimental results showed that for a day with high sunlight, the tank temperature is kept at 30°C overnight. In addition, the values of the instantaneous thermal efficiency were between about 22% and 80%. To enhance the thermal conductivity, Zhang et al. [8] studied numerically the effect of the natural convection of a PCM in a spherical enclosure. Their results show that the PCM melts rapidly in the upper part of the storage unit, where heat transfer is dominated by convection compared to the lower part where conduction occurs, moreover Medrano et al. [9] experimentally studied the heat transfer characteristics of five latent heat storage systems for both charging and discharging cycles. The results indicate that the twin-tube configurations filled with graphite-embedded PCM have the best performance. In this study the effect of the inlet temperature and the mass flow rate on the performance of shell and tube latent heat storage unit were investigated numerically. The water used as heat transfer fluid (HTF) flows through the inner pipe, while the annular space filled by PCM. The focus of this study is the impact of the inlet temperature and mass flow rate of HTF on the charging and discharging process of the storage unit.

2 Numerical model

2.1 Description of the problem

The configuration of the thermal storage unit under investigation is presented in figure 1. We used the same geometrical configuration as shown in Tab.1 , the PCM and the initial boundary conditions with that of Kibria et al. [10]. The system composed of two coaxial tubes. Which the water considered as a heat transfer fluid (HTF) flows in the inner copper tube, while the annular space filled with PCM "Paraffin wax", which their properties have been reported in Tab.2 . In addition, the outer surface is thermally insulated.

Table 1 - Physical dimensions of the studied system

Inner diameter	D_i	0.0054 [m]
Outer diameter	D_0	0.006 [m]
Shell diameter	D	0.018 [m]
The length	L	1 [m]

During the charging phase of the storage system, the hot water entered at 88°C and a mass flow rate of 0.072 kg/min until this process is completed. Subsequently the HTF entered at 25°C to recover the thermal energy stored in the liquid PCM in discharging cycle.

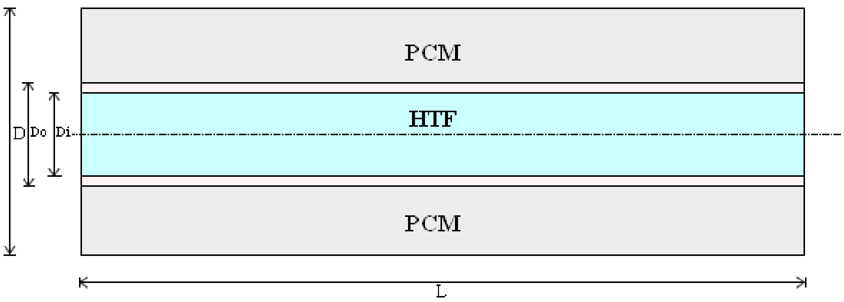

Fig. 1 – Schematic diagram of the latent heat storage unit.

Table 2 - Thermo-physical properties of PCM Kibria et al. [10]

Propriety	Density $Kg.m^{-3}$		Specific heat capacity $J.m^{-3}.K^{-1}$		Thermal conductivity $W.m^{-1}.K^{-1}$		Melting temperature °C	Latent heat $J.kg^{-1}$	Dynamic viscosity $Kg.m^{-1}.S^{-1}$	
	Solid	Liquid	Solid	Liquid	Solid	Liquid			Solid	Liquid
Paraffin Wax (PCM)	910	790	2000	2150	0.24	0.22	61	190000	--	0.004108

2.2 Governing equations

In the present simulation, 2D implicit finite volume method was used to solve the governing equations of a head transfer coupled with solid-liquid phase change process. The enthalpy-porosity formulation was used for simulation of the phase change phenomenon. In this method the interface between the solid and the liquid phases modeled as a porous medium. The liquid fraction varies smoothly across this porous, so-called mushy region. The mushy zone is modeled via the phase fractions, which are incorporated in the source terms in the governing equations to account for the phase change phenomena.

- The continuity equation:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{1}$$

- The momentum equation (x) axis:

$$\frac{\partial}{\partial t}(U)+\frac{\partial}{\partial X}(U^2)+\frac{\partial}{\partial Y}(UV)=-\frac{\partial P}{\partial X}+P_r\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right)+R_a\,P_r\sin(\alpha)+S'_x, \tag{2}$$

- The momentum equation (y) axis:

$$\frac{\partial}{\partial t}(V)+\frac{\partial}{\partial X}(UV)+\frac{\partial}{\partial Y}\left(V^2\right)=-\frac{\partial P}{\partial Y}+P_r\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)-R_a\,P_r\cos(\alpha)+S'_y, \tag{3}$$

- The energy equation :

$$\frac{\partial}{\partial t}(\rho h)+\frac{\partial}{\partial x}(\rho u h)+\frac{\partial}{\partial y}(\rho v h)=\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+S_h, \tag{4}$$

Where ρ is the density, k denotes the thermal conductivity, μ is the dynamic viscosity, S_i and S_h are the source terms, u_i is the velocity component in the i-direction, x_i is a Cartesian coordinate, and h is the specific enthalpy. The sensible enthalpy h_s is given by:

$$h_s = h_{ref} + \int_{T_{ref}}^{T} C_p dT \quad , \tag{5}$$

And the total enthalpy, *H* is defined as

$$H = h_s + \Delta H \quad , \tag{6}$$

Where ΔH = γL is the enthalpy change due to phase change, h_{ref} is the reference enthalpy at the reference temperature T_{ref}, Cp is the specific heat, L is the specific enthalpy of melting (liquid state) and γ is the liquid fraction during the phase change which occur over a range of temperatures

$T_{solidus} < T < T_{Liquidus}$ defied by the following relations:

If $T < T_{solidus}$

$$\gamma = \frac{\Delta H}{L} = 0 \quad , \tag{7}$$

If $T_{solidus} < T < T_{liquidus}$

$$\gamma = \frac{\Delta H}{L} = \frac{T - T_s}{T_l - T_s} \quad , \tag{8}$$

If $T > T_{Liquidus}$ (liquid state)

$$\gamma = \frac{\Delta H}{L} = 1 \, , \tag{9}$$

The source terms S_i and S_h are given by:

$$S_i = -A(\gamma) u_i \frac{C(1-\gamma)^2}{\gamma^3 + \varepsilon} u_i \quad , \tag{10}$$

$$S_h = \rho L \frac{\partial \gamma}{\partial t} \quad , \tag{11}$$

Where A(γ) is defined as the "porosity function" which governs the momentum equation based on Carman-Kozeny relationship for flow in porous media. The function reduces the velocities gradually from a finite value of 1 in fully liquid to 0 in fully solid state within the computational cells involving phase change. The epsilon ε = 0.001 infinity avoidance constant due to division by zero and C is a constant reflecting the morphology of the melting front where $C = 10^5$.

2.3 Numerical modeling

The geometry was created by the mesh generator GAMBIT for 35x300. Then, it was imported into FLUENT 6.3.26[11], the simulation was executed with the 2D solver in double precision (2ddp). The PRESTO scheme is used for pressure correction and the SIMPLE Algorithm (Semi-Explicit Pressure-Linked Equation)[12] is used for the pressure-velocity coupling. A grid of 35x300 was used for all simulation cases, also a time step of 0.1 seconds was adopted after a time step comparison test of 0.05 and 0.1 seconds which showed a slight difference which was neglected.

2.4 Assumptions

The following assumptions were adopted in the present numerical study:

- The thermophysical properties for PCM and HTF were constant with respect to temperature and pressure.
- The heat transfer inside the PCM was completely dominated by conduction.
- The HTF flow was laminar.
- The PCM and HTF considered as pure materials.

3 Results

3.1 Validation

The numerical model used in the present work has been validated with experimental and numerical results of Kibria et al. [10] as shown in Figures 2a and 2b for the same Assumptions, initial and boundary conditions. A good agreement was obtained for both charging and discharging phases.

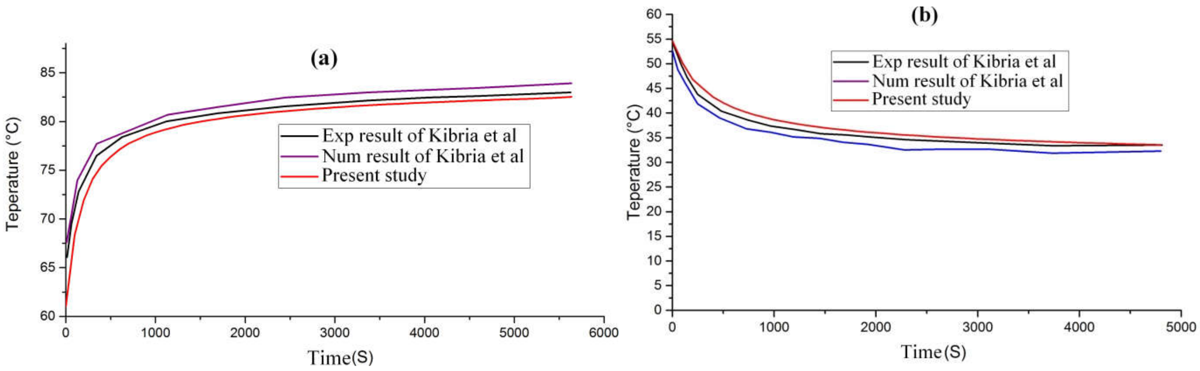

Fig. 2 – Numerical and experimental outlet temperatures for (a) the charging, (b) Discharging

3.2 Impact of the HTF inlet temperature

To illustrate the effect of the HTF inlet temperature variation on the heat storage performance, we took three different temperature values (88°C, 75°C and 65°C).

The obtained results show that the gradient of the inlet temperature plays a very important effect during the charging cycle fig. 3. For a higher inlet temperature, the melting time decreases due to the increase in the heat transfer rate. It can be seen that the total melting time was reduced about 60%, when the temperature of the inlet water increased from 65°C to 88°C, which means 26% temperature rise.

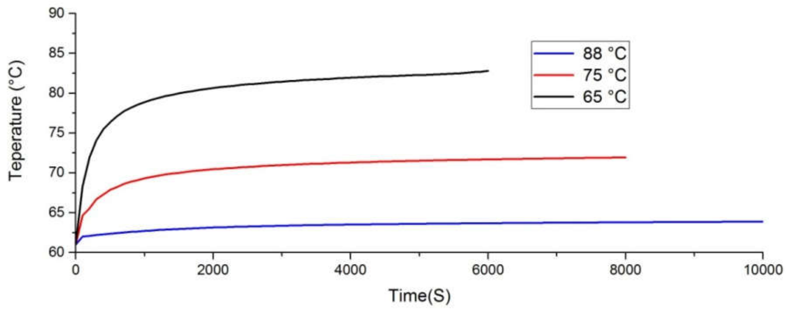

Fig. 3– Effect of the HTF inlet temperature

3.3 Impact of the HTF mass flow rate

The mass flow rate of the HTF has no significant effect on the operating time and the outlet temperature of the storage unit in both, the discharging and the charging cycles as shown in fig. 4. As the mass flow rate increases, keeping the same inlet temperature. The time required for solidification and fusion decreases slightly, because for a higher mass flow rate, the heat transfer rate increases in the PCM region. In addition, for a higher mass flow rate value, the heat transfer fluid outlet temperature approaches the inlet temperature in both the solidification and melting cycles.

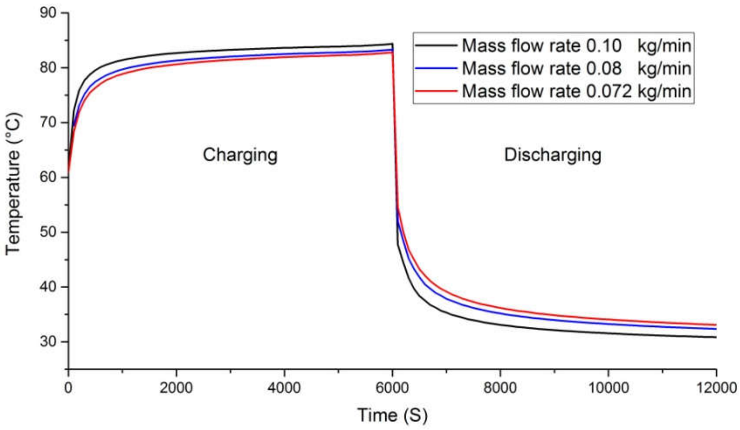

Fig. 4– Effect of the HTF mass flow rate

4 Conclusion

A two-dimensional numerical model has been developed with success, to solve the governing equations of the heat transfer between the PCM and the HTF in a latent heat storage unit for the charging and discharging cycles. The developed model has been validated with experimental and numerical results of Kibria et al. [10], or good agreement has been obtained.

A parametric study was conducted to evaluate the performance of the storage unit with different flow rate and inlet temperature parameters. The obtained results showed that the inlet temperature of the HTF has a great effect on the solidification and melting of PCM and consequently reduce the charging time about 60% for the inlet temperatures higher than its melting temperature by 30.9%. In addition, when the temperature approached the melting point of the PCM, the storage process time increased for the charging and the discharging cycles. On the other hand, the mass flow rate does not have a great influence.

NOMENCLATURE

H, h	Enthalpy, $J.Kg^{-1}$
L	Latent heat, J.kg-1
Cp	Specific heat at constant pressure, $J.kg^{-1}.K^{-1}$
K	Thermal conductivity, $W.m^{-1}.K^{-1}$
P	Pressure, Pa
T	Temperature, K
A	porosity function
U	Velocity component in the x-direction, $m.s^{-1}$
V	Velocity component in the y-direction, $m.s^{-1}$
C	Reflecting constant of melting front

Greek symbols

β	Thermal expansion coefficient, K^{-1}
γ	liquid fraction
ρ	Density of PCM/Water, $Kg.m^{-3}$
Δ	Difference
μ	Dynamic viscosity, $Kg.m^{-1}s^{-1}$

Subscripts

s	sensible
ref	Reference
m	Melting
S	Solidus
l	Liquids

REFERENCES

[1] V. Kapsalis, and D. Karamanis, Solar thermal energy storage and heat pumps with phase change materials. Appl. Therm. Eng. Vol.99 N°9 (2016) pp. 1212-1224.

[2] S.Sharma, and K. Sagara, Latent heat storage materials and systems: a review. Int. J. Green Energy. Vol.2 N°1 (2005) pp.1-56.

[3] S. Seddegh, X. Wang, A. Henderson, Z. Xing, Solar domestic hot water systems using latent heat energy storage medium: a review. Renewable Sustainable Energy Rev. Vol.49 (2015) pp.517-533.

[4] D. Groulx, A. Kheirabadi, L. Desgrosseilliers, M. Kabbara, M. Azad, A. Donaldson, A.Joseph, M A White, Working towards solving the Rate Problem: Geometric vs Nano-Enhanced PCM Solutions. in Proceedings of INNOSTORAGE Advances in Thermal Energy Storage Conference. 2016.

[5] S. Jegadheeswaran, and S.D. Pohekar, Performance enhancement in latent heat thermal storage system: a review. Renewable and Sustainable Energy Rev, Vol.13 N°9 (2009) pp.2225-2244.

[6] I. Tari, and M. Mehrtash, Natural convection heat transfer from inclined plate-fin heat sinks. Int. J. Heat Mass Transfer, Vol.2 N°1-2 (2013) pp.574-593.

[7] A. Kürklü, A. Özmerzi, and S. Bilgin, Thermal performance of a water-phase change material solar collector. Renewable Energy, Vol.26 N°3 (2002) pp.391-399.

[8] Y.Zhang, J.M. Khodadadi, and F. Kavicka. Effects of buoyancy-driven convection on melting within spheres. In ichmt digital library online. Begel House Inc.

[9] M. Medrano, M. Yilmaz, M. Nogués, I. Martorell, J. Roca, L F.Cabeza, Experimental evaluation of commercial heat exchangers for use as PCM thermal storage systems. Applied energy, Vol.86 N°10 (2009) pp. 2047-2055.

[10] M. Kibria, MR. Anisur, MH. Mahfuz, R. Saidur, IHSC. Metselaar, Numerical and experimental investigation of heat transfer in a shell and tube thermal energy storage system. Int. Commun. Heat Mass Transfer. Vol.53 (2014) pp.71-78.

[11] ANSYS Fluent 6.3.26, A.F., 2009.

[12] S. Patankar, Numerical heat transfer and fluid flow.CRC press, 1980.