International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:3, No:8, 2009

Optimisation of A Phase Change Thermal Storage System

Nasrul Amri Mohd Amin, Martin Belusko, Frank Bruno

Abstract**—**PCMs have always been viewed as a suitable candidate for off peak thermal storage, particularly for refrigeration systems, due to the high latent energy densities of these materials. However, due to the need to have them encapsulated within a container this density is reduced. Furthermore, PCMs have a low thermal conductivity which reduces the useful amount of energy which can be stored. To consider these factors, the true energy storage density of a PCM system was proposed and optimised for PCMs encapsulated in slabs. Using a validated numerical model of the system, a parametric study was undertaken to investigate the impact of the slab thickness, gap between slabs and the mass flow rate. The study showed that, when optimised, a PCM system can deliver a true energy storage density between 53% and 83% of the latent energy density of the PCM.

Keywords**—**Phase change material, refrigeration, sustainability, thermal energy storage.

I. Introduction

Electricity demand during the summer season [1-6] represents a significant impact to the cost of electricity [7]. One of the contributors to this demand during peak summer days is refrigeration. Therefore, there is a need for off peak storage for refrigeration. However, as with all commercial off peak storage systems, there is a need to prevent an increase in energy consumption [8]. Storing the 'coolness' from the off peak period using phase change material (PCM) is a potential option for refrigeration systems [9].

PCMs are a substance in which the heat at the solid-liquid phase transition point is used in applications for storing large amounts of thermal energy [10,11] at a certain temperature. A principle advantage of PCMs is the large energy storage density relative to sensible energy storage with rock or water (liquid) [12].

PCMs also have some disadvantages. The main drawbacks with PCMs are the low thermal conductivity and the encapsulation design. In a latent heat thermal energy storage system, the heat transfer fluid (HTF) passes through an area of encapsulated PCM. In a PCM, during melting or freezing, the thermal resistance increases as the solid/liquid boundary moves away from the heat exchange surface [10]. This resistance is significant and reduces the thermal capacity of the system [13]. This resistance changes with the design of the encapsulation.

In a PCM thermal storage unit, the percentage of the PCM volume per total system volume is defined as the compactness ratio. The challenge in PCM encapsulation design is to have a large compactness ratio so that the maximum amount of latent energy can be used. The compactness ratio can be as low as 50%, as in a sphere type PCM heat exchanger [14]. In this project, a plate type thermal storage system is used because of the high storage density of PCM which can be up to 95% of the total volume [15].

II. System Design

Optimisation involves varying the PCM slab thickness, gap width between slabs and the HTF mass flow rate. A PCM with a melting point of -27°C is used for this analysis because it is suitable for refrigeration applications [9] along with Dynalene HC-40 as the HTF [14].

The amount of PCM used in this model is fixed at $1 m^3$. Therefore, when the slab thickness is varied, the number of slabs is changed so that the total PCM volume is constant.

The actual performance of a PCM energy storage system depends on the effectiveness of charging and discharging, and the compactness ratio. These parameters can be incorporated into the true energy storage density factor, α. The value of α can be applied to the latent heat of fusion, H_L. The resulting product represents the true energy density of the PCM thermal storage system.

III. Theoretical Analysis

The storage system consists of several flat PCM slabs. No heat transfer to the surrounding is assumed. The HTF is flowing between these slabs as in Fig. 1.

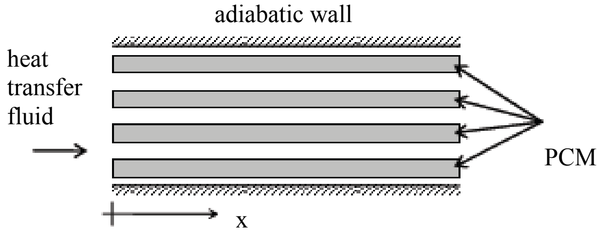

Fig. 1 Schematic diagram of the PCM model [9]

A latent heat thermal energy storage system can be described as a heat exchanger. Effectiveness is an important

N.A.M.A. was with School of Mechanical Engineering at the University of Adelaide. He is now with the Institute for Sustainable Systems and Technologies at the University of South Australia, Mawson Lakes, South Australia 5095, Australia. (phone: +618-83023230; fax: +618-83023380; e-mail: nasrulamri.mohdamin@yahoo.com.au).

Both of second and third author are with the Institute for Sustainable Systems and Technologies at the University of South Australia, Mawson Lakes, South Australia 5095, Australia. (phone: +618-83023767; fax: +618-83023380; e-mail: martin.belusko@unisa.edu.au).

parameter in heat exchanger design and can be applied to PCM storage systems as well. The effectiveness of a phase change storage system is equal to the ratio of the actual heat transfer to the maximum possible heat transfer [16]. In this work, the effectiveness of melting, ε_1 and freezing, ε_2 was calculated by using equations (1) and (2) over the phase change process. These equations ignore the sensible energy storage.

$$\varepsilon_1 = \sum \dot{m}\, Cp\,(T_{in} - T_{out}) / \sum \dot{m}\, Cp\,(T_{in} - T_{pcm}) \tag{1}$$

$$\varepsilon_2 = \sum \dot{m}\, Cp\,(T_{out} - T_{in}) / \sum \dot{m}\, Cp\,(T_{pcm} - T_{in}) \tag{2}$$

For ideal latent heat thermal energy storage, both values of ε_1 and ε_2 are 1. Theoretically, this means the exit temperature of the HTF must be the same as the PCM phase change temperature.

The most critical objective in thermal energy design is to get the highest α as possible. So the values of melting and freezing effectiveness must be high enough together with the proportion of PCM in the system, the compactness ratio, γ. Values of α and γ was calculated using equations (3) and (4).

$$\alpha = \varepsilon_1 \varepsilon_2 \gamma \tag{3}$$

$$\gamma = t/(w + t) \tag{4}$$

To consider pumping losses a modified effectiveness value was determined using equations (5) and (6), total losses of the system are then calculated using equations (7).

$$\varepsilon_{1*} = \varepsilon_1 - (Losses\ / Q_{th}) \tag{5}$$

$$\varepsilon_{2*} = \varepsilon_2 - (Losses\ / Q_{th}) \tag{6}$$

$$Losses = \Delta P\, \dot{V} / \eta_p \eta_{p.s} \tag{7}$$

The *Losses* define the primary energy of the pumping losses. Pump efficiency, η_p and power station efficiency, $\eta_{p.s}$ were fixed at 50% and 35% respectively. To consider these losses, α value became α^* as defined in (8).

$$\alpha^* = \varepsilon_{1*} \varepsilon_{2*} \gamma \tag{8}$$

IV. Simulation Software

In this work, the simulations were done using TRNSYS. TRNSYS is a transient systems simulation program with a modular structure [17]. The software has been used for analysing the time dependent numerical model for many heat exchanger types [11] in refrigeration systems [18].

In order to calculate the true energy storage density factor, α, the value of ε_1 and ε_2 values need to be known. TRNSYS was used to calculate these values.

V. PCM Model

An existing one dimensional computer model of latent heat thermal energy [9] in TRNSYS was used in this research. This experimentally validated model [19] consists of several flat PCM slabs which are then simulated in the computer using the finite-difference calculation method.

The ideal performance occurs when the inlet temperature is close to the melting temperature. Therefore a 1°C temperature different was applied between the inlet temperature and the PCM melting point. This temperature difference also minimised the sensible energy of the PCM. Simulations on the model are based on the following assumptions:-

1. No super cooling occurs,
2. Simulation stops when $T_{in} - T_{out}$ (melting) or T_{out} - T_{in} (freezing) @ $\Delta T < 0.01$. Results have shown that the effectiveness values for melting and freezing have minimal differences (comparing $\Delta T < 0.01$ and $\Delta T < 0.001$) and can be neglected,
3. Conduction in the PCM is ignored as it has been shown that the resistance to heat flow is small due to the thin slabs that are used [15,19,20],
4. Natural convection is ignored,
5. Sensible energy is ignored as it is only 2.5% of the latent energy.

Table 1 show the parameters which had to be set on the computer model. These values are fixed through simulations.

TABLE I
PRESET PARAMETERS

Parameter	Quantity	Unit
HTF inlet temperature	-25.7 (for melting)	ºC
	-27.7 (for cooling)	ºC
Solid thermal diffusivity	0.0000014176245	m^2/s
Liquid thermal diffusivity	0.0000001105538	m^2/s
Solid specific heat	1200	J/kg.K
Liquid specific heat	3650	J/kg.K
Solid thermal conductivity	2.2	W/m.K
Liquid thermal conductivity	0.55	W/m.K
Solid density	1305	kg/m^3
Liquid density	1363	kg/m^3
PCM melting temperature	-26.7	ºC
PCM freezing temperature	-30.6	ºC
Initial temperature	-26.9 (for melting)	ºC
	-26.5 (for cooling)	ºC
Latent heat of fusion	144000	J/kg
Wall thermal conductivity	0.02	W/m.K
Length of slabs	1	m
Width of slabs	1	m

The three main parameters investigated in this project are the gap between slabs, w, HTF mass flow rate, $\dot{m}$, and PCM slab thickness, t. Table 2 shows the range investigated.

TABLE II
CHANGING PARAMETERS

Parameter	Quantity	Unit
No of slabs	5 to 105	-
	Default is 45	-
Thickness of slabs	0.20056 to 0.009931	m
	Default is 0.02264	m
Gap between slabs	0.005 to 0.75	m
	Default is 0.01	m
Mass flow rate	50 to 7500	kg/hr
	Default is 100	kg/hr

VI. Simulation Findings

A. Gap width effects

The first simulation investigated the effect of the gap width between PCM slabs, w on the true energy storage density factor, α. The w value is increased from 0.005m to 0.75m through repeated simulations. The PCM slab thickness and mass flow rate of the HTF are fixed at 0.01m and 100kg/hr respectively. The result is shown in Fig. 2.

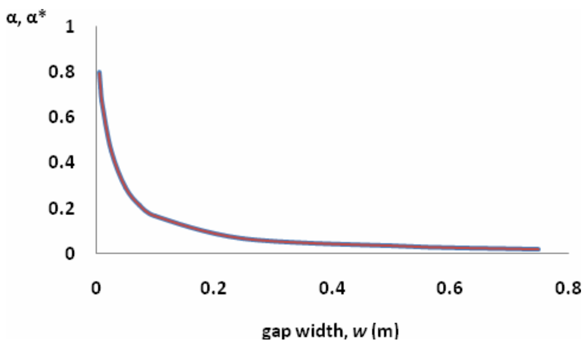

Fig. 2 True energy storage density factor, α and α^* versus gap width, w (m)

There are two lines in the graph. As the values for both theoretical, α and the one including losses, α^* are almost identical, it can be concluded that the pumping losses are negligible.

The α values declined sharply for w < 0.1m due to a decreasing compactness ratio, γ. As shown in Fig. 3, an increasing w had a small effect on the effectiveness. The small reduction was due to the decrease in overall heat transfer coefficient between the wall and the HTF that a larger gap would cause. The range of w < 0.1m is typical for many flat slabs type of PCM storage because of their high value of α. The system became in-efficient for larger gap widths.

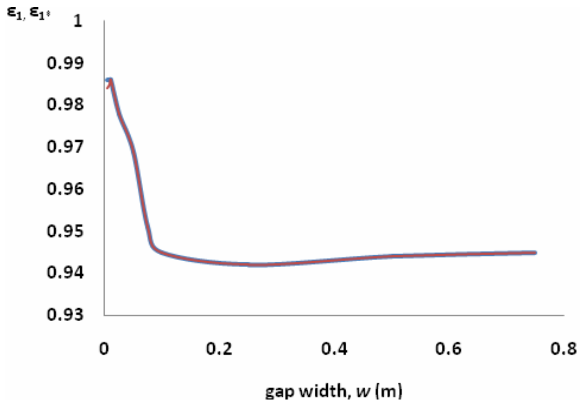

Fig. 3 Melting effectiveness, ε_1 and ε_{1^*} versus gap width, w (m)

B. Mass flow rate effects

The impact of HTF mass flow rate was investigated. The result is shown in Fig. 4.

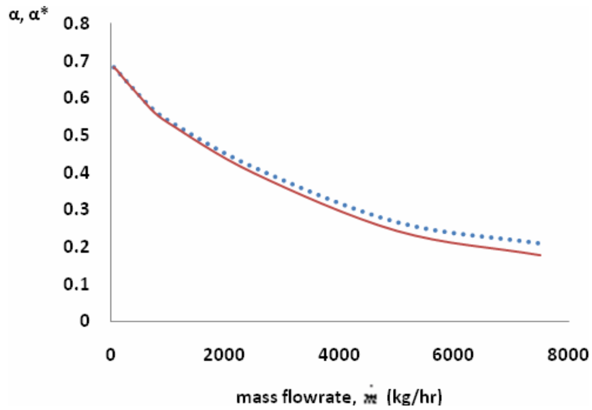

Fig. 4 True energy storage density factor, α and α^* versus mass flow rate, $\dot{m}$ (kg/hr)

The heat transfer from the HTF to PCM occurs best for low mass flow rates. This can be explained by referring to Fig. 5. The melting effectiveness, ε_1 and ε_{1^*} decreased with increasing flow rate as expected in heat exchangers. The impact of losses is small, increasing at higher flow rate. The same phenomenon applies to the freezing process.

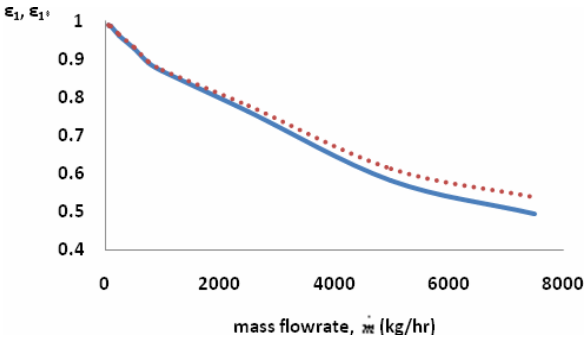

Fig. 5 Melting effectiveness, ε_1 *and* ε_{1^*} versus mass flow rate, $\dot{m}$ (kg/hr)

C. Slab thickness effects

The impact of the PCM slab thickness, *t* is shown in Fig. 6. The number of slabs was increased from 5 to 105, as the thickness decreased from 0.20056m to 0.009931m. The simulation is run repeatedly with 50, 100, 500, and 1000kg/hr mass flow rate of the HTF (from top to bottom in the graph).

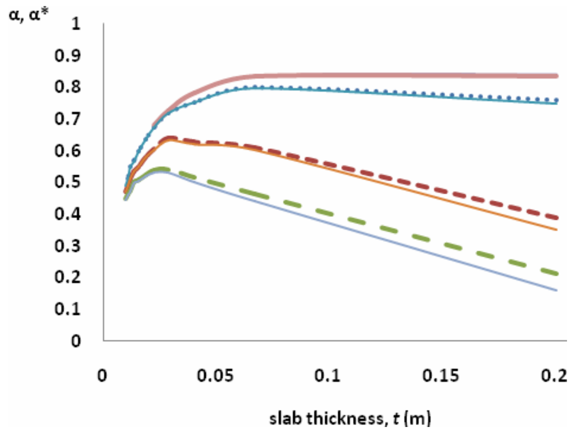

Fig. 6 True energy storage density factor, α and α^* versus slab thickness, *t* (m)

The graph shows an increasing α at small slab thicknesses. Given that the gap width, *w* is fixed, this increase is due to an increasing compactness ratio. Fig. 7 shows the melting effectiveness decreasing at larger slab thickness which is due to the decreasing heat exchanger area as the number of slabs decreased. Therefore an optimal slab thickness exists.

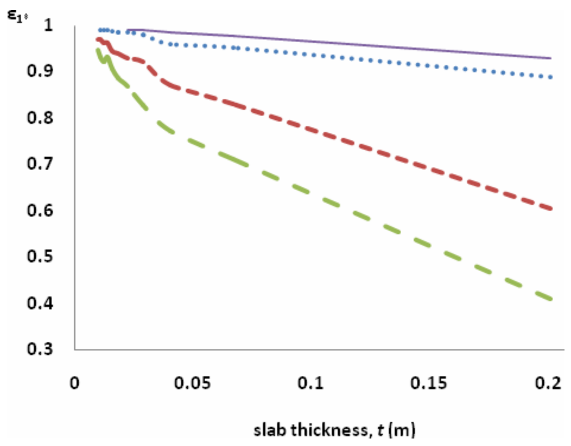

Fig. 7 Melting effectiveness, ε_1 and ε_{1^*} versus slab thickness, *t* (m)

This optimal thickness increases with decreasing flow rates. At the lowest flow rate, α reaches a constant maximum and this is due to the effectiveness also being relatively constant (Fig. 7). This occurs as with a decrease in area there is also an increase in the overall heat transfer coefficient between the wall and the HTF as the mass flow rate is fixed.

VII. OPTIMAL DESIGN

The most critical finding is depicted from Fig 6. It can be seen that the slab thickness plays an important role in thermal energy storage design. It is clearly shown that the optimum value for α is different depending on the mass flow rate supplied into the system. As the mass flow rate increased, the maximum α dropped off.

In Fig. 6, the top line represents α for 50kg/hr flow rate. Slab thickness greater than 0.06712m gives an almost constant α. It was concluded that for low flow rate, the optimum size of thickness is 0.06712m with 15 slabs in this thermal storage unit (with 0.01m gap between slabs). Another conclusion is that, for low flow rates, the α values are independent of losses. Overall an optimal α of 0.83 was obtained.

1000k/hr represents a typical high flow rate for the system. The optimum slab thickness is 0.02264m with 45 slabs. It can be seen that, the losses from pumping have a small effect. At this flow rate, the optimum α was 0.53.

Overall, it has been shown that with a plate design, the true energy storage density of a PCM thermal storage facility is heavily dependent on the design and the flow rates used. Optimisation has shown that the true energy storage density of a PCM system can be increased to within the range of 53% to 83% of the latent energy storage density of the PCM. With PCMs being many times more energy dense than sensible storage systems, it can be concluded that optimized PCM systems with plates can deliver true storage densities considerably higher than sensible systems.

APPENDIX

TABLE III
NOMENCLATURE

Symbol	Quantity	Unit
ε_1	melting effectiveness	-
ε_2	freezing effectiveness	-
ε_{1*}	modified melting effectiveness	-
ε_{2*}	modified freezing effectiveness	-
$\dot{m}$	mass flow rate of HTF between slabs	kg/hr
Cp	specific heat	J/kg.K
T_{in}	inlet temperature	ºC
T_{out}	outlet temperature	ºC
T_{pcm}	temperature of PCM	ºC
Q_{th}	theoretical stored energy	J/s
Q_{act}	actual stored energy	J/s
ΔP	pressure drop	kg/m^2
η_p	pumping efficiency	-
$\eta_{p.s}$	power station efficiency	-
f	relative roughness	-
l	length of PCM slab	m
ρ	density of the HTF	kg/m^3
v	velocity of the HTF	m/s
D_h	hydraulic diameter	m
Re	Reynolds number	-
w	width of gap	m
l	length of gap / PCM slab	m
t	thickness of PCM slab	m
μ	dynamic viscosity of the HTF	kg/ms
γ	proportion of PCM	-
α	true energy storage density factor	-
α^*	modified true energy storage density factor	-
$Q_{sensible}$	sensible heat	J
Q_{latent}	latent heat	J
H_L	latent heat of fusion	J/kg

ACKNOWLEDGMENT

The authors are very grateful to the Institute for Sustainable Systems and Technologies at the University of South Australia for providing technical supports. N.A.M.A. also be thankful for the financing grant by the Universiti Malaysia Perlis which governed by the Ministry of Higher Education, Malaysia.

REFERENCES

[1] F. Wang, G. Maidment, J. Missenden, and R. Tozer, "A review of research concerning the use of PCMS in air conditioning and refrigeration engineering", in *Proc. of the Int. Conf. on Advances Building Technology*, Hong Kong, China, 4–6 Dec. 2002, pp. 1273-1280.

[2] A. Pasupathy, L. Athanasius, R. Velraj, and R. V. Seeniraj, "Experimental investigation and numerical simulation analysis on the thermal performance of a building roof incorporating phase change material (PCM) for thermal management", *Applied Thermal Engineering,* vol. 28, no. 5-6, pp. 556-565, Apr. 2008.

[3] M. M. Farid, A. M. Khudhair, S. A. K. Razack, and S. Al-Hallaj, "A review on phase change energy storage: materials and applications", *Energy Conversion and Management,* vol. 45, no. 9-10, pp. 1597-1615, June 2004.

[4] S. M. Vakilaltojjar, "Phase change thermal storage system for space heating and cooling", Ph.D. thesis, School of AME, University of South Australia, 2000.

[5] A. M. Papadopoulos, S. Oxizidis, and N. Kyriakis, "Perspectives of solar cooling in view of the developments in the air-conditioning sector", *Renewable and Sustainable Energy Reviews,* vol. 7, no. 5, pp. 419-438, Oct. 2003.

[6] L. Jian-you, "Numerical and experimental investigation for heat transfer in triplex concentric tube with phase change material for thermal energy storage", *Solar Energy,* vol. 82, no. 11, pp. 977-985, Nov. 2008.

[7] W. Muriel, O. Hugh, S. Ted, M. Iain, O. Monica, and P. Scott, "Photovoltaics and peak electricity loads, Summer 2003-04", in *Report for BCSE*, May 2005.

[8] V. Butala and U. Stritih, "Experimental investigation of PCM cold storage", *Energy and Buildings,* vol. 41, no. 3, pp. 354-359, Mar. 2009.

[9] M. Liu, F. Bruno and W. Saman, "Thermal performance of a PCM thermal storage unit", in *Proc. of Int. Solar Energy Society for Solar World Congress*, Beijing, China, 18-21Sept. 2007.

[10] N. A. M. Amin, F. Bruno, and M. Belusko, "647-037- Maximizing the energy storage performance of phase change thermal storage systems", in *Proc. of the IASTED Int. Conf.*, Solar Energy (SOE 2009)", Phuket, Thailand, 16-18 Mar. 2009, pp. 55-60.

[11] H. Mehling and L.F. Cabeza, *Heat and Cold Storage with PCM: An up to date introduction into basics and applications.* Berlin Heidelberg: Springers-Verlag, 2008.

[12] F. Bruno, "Centralised PCM systems for shifting cooling loads during peak demands in buildings", in *Supplementary Technical Research Paper*, City of Melbourne, May 2006.

[13] H. Ettouney, H. El-Dessouky, and A. Al-Ali, "Heat transfer during phase change of paraffin wax stored in spherical shells", *Journal of Solar Energy Engineering,* vol. 127, no. 3, pp. 357-366, Aug. 2005.

[14] J. P. Bedecarrats, F. Strub, B. Falcon and J. P. Dumas, "Phase-change thermal energy storage using spherical capsules: performance of a test plant", *International Journal of Refrigeration,* vol. 19, no. 3, pp. 187-196, 1996.

[15] M. Belusko and F. Bruno, "140 - Design methodology of PCM thermal storage systems with parallel plates", in *Report for 1st Int. Congress on Heating, Cooling, and Buildings, EUROSUN 2008,* Lisbon, Portugal, 7-10 Oct. 2008.

[16] The Board of Regents of the University of Wisconsin System. (2006, November, 16). [Online]. Available: http://sel.me.wisc.edu/trnsys

[17] S. V. Dosky, D. Heinze, and J. Wolf, "Numerical simulation of a refrigeration cycle for scaling towards small geometries", *International Journal of Refrigeration*, vol. 31, no. 8, pp. 1384-1390, Dec. 2008.

[18] E. Halawa, F. Bruno, and W. Saman, "Numerical analysis of a PCM thermal storage system with varying wall temperature", *Energy Conversion and Management*, vol. 46, no. 15-16, pp. 2592-2604, Sept. 2005.

[19] E. Halawa, "Thermal performance analysis of a roof integrated solar heating system incorporating phase change thermal storage", Ph.D. thesis, School of AME, University of South Australia, 2005.

[20] D. J. Morrison and S. I. Abdel-Khalik, "Effects of phase-change energy storage on the performance of air-based and liquid-based solar heating systems", *Solar Energy*, vol. 20, no. 1, pp. 57-67, 1978.