Design of PCM Thermal Storage System using the Effectiveness-NTU Method

Frank Bruno*, **Martin Belusko and N.H. Steven Tay**

Sustainable Energy Centre, University of South Australia, GPO Box 2471, Adelaide, South Australia 5001, Australia, Ph +61 8 8302 3230, Fax +61 8 8302 3380

* Corresponding Author, Frank.Bruno@unisa.edu.au

Abstract

Thermal storage systems with phase change materials are predominantly designed through numerical modelling. This can be a time consuming process. An alternative simplified method is being proposed for the design of these phase change thermal storage systems. The method is based on the effectiveness-Number of Transfer Units (ε-NTU) technique. A mathematical model has been developed using the ε-NTU technique for a cylindrical tank filled with phase change material (PCM), with heat transfer fluid flowing through tubes inside the tank. Experiments have been carried out on a cylindrical tank filled with PCM and with one, two and four coils of tubes to validate the models. Experimental results compare well with those calculated from the model. The results show that this technique can readily be used as a design tool for sizing and optimising a thermal storage unit with phase change materials. From this study, it may be concluded that the model based on the ε-NTU technique can accurately predict the effectiveness of the thermal storage system during charging and discharging.

1. Introduction

Considerable research work has been reported on modelling and experiments on latent heat storage systems [1-4]. Work has also been conducted on latent heat thermal storage systems using multiple phase change materials (PCMs) to improve the heat transfer rate usually hindered by the PCM's low thermal conductivity [5-8].

Numerous mathematical models of PCM in thermal storage units (TSU) have been developed over the years. These models have been used to determine the performance of the TSU for design and simulation purposes [9]. However, little attention has been placed on using these models to develop generic representations which can be readily used for the characterisation and ultimately the design and optimisation of a TSU with PCM.

Halawa et al. [9] presented the numerical analysis of melting and freezing of a PCM thermal storage unit (TSU) with varying wall temperature. The TSU comprises of several layers of thin slabs of PCM subjected to convective boundary conditions where air flows between the slabs. The numerical model considered the variations in wall temperature along the direction of air flow as well as sensible heat. Models created using the ε-NTU methodology have been conducted by several researchers. Browne and Bansal [10] presented a new steady-state model for vapour-compression liquid chillers. The principle of this model is on physical laws and heat transfer coefficients. In order to better predict the heat transfer, the heat exchanger is divided into elements by applying the elemental ε-NTU methodology. Mathew and Hegab [11] developed a thermal model of a parallel flow microchannel heat exchanger subjected to external heat transfer. In the laminar flow regime, the effectiveness of the fluids in parallel flow microchannel heat exchangers and the axial temperature can be predicted. When subjected to external heating for a specific NTU, the effectiveness of the hot fluid would decrease while the cold fluid increased. In the presence of external cooling, the effectiveness of the hot fluid was observed to improve while the cold fluid degraded. A one dimensional equation of the effectiveness of a TSU based on the ε-NTU approach has been formulated by Belusko and Bruno [12]. The effectiveness of one and two dimensional phase change within a PCM slab was defined in term of phase change fraction. Employment of phase change fraction characterizes the TSU into a single effectiveness chart and thus developing a useful method for determining the size of the TSU.

In this paper, a mathematical model based on the ε-NTU approach [12] is developed for coils in a thermal energy storage system. The tubes are coiled inside a cylindrical tank filled with PCM. The heat transfer fluid (HTF) passes through the tube during the charging and discharging processes. The model assumes that the phase change of the PCM will ultimately form a square. Experiments have been conducted to validate the numerical analysis. Using the experimental parameters on the mathematical model, theoretical graphs of the mathematical model have been plotted and compared with the experimental graphs.

2. Experiment equipment and test procedure

Four experiments were carried out for this work. The first three experiments used a salt hydrate PCM with a phase change temperature of -27°C.The fourth experiment used water as the PCM. The first experiment was conducted with a 5.46 meters length of tube coiled in a cylindrical tank filled with PCM (Figure 1 (a)) while the second experiment had two coils of tubes with lengths of 5.61 and 6.01 meters respectively (Figure 1 (b)). The third and fourth experiments were conducted with four coils of tubes with lengths of 5.951, 6.047, 5.789 and 6.04 meters respectively (Figure 2 (a)). The compactness factor (CF) is the ratio of the volume of PCM to the volume of the tank. The first experiment provides a higher CF of up to 98% but a lower heat transfer surface area. The second experiment in turn has a higher heat transfer surface area but a lower CF of 96%. The third and last experiments have the highest heat transfer surface area and lowest CF of 90%.

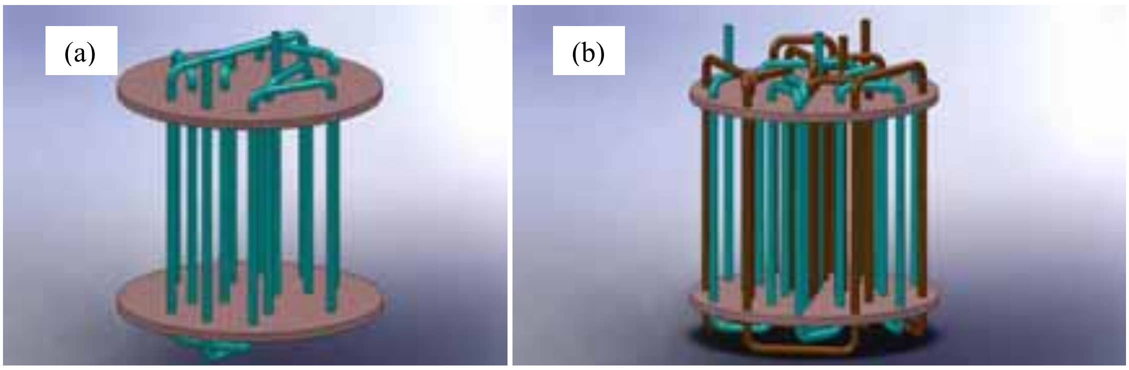


Figure 1. Schematic of One-coil Tank (a) and Two-coils Tank (b).

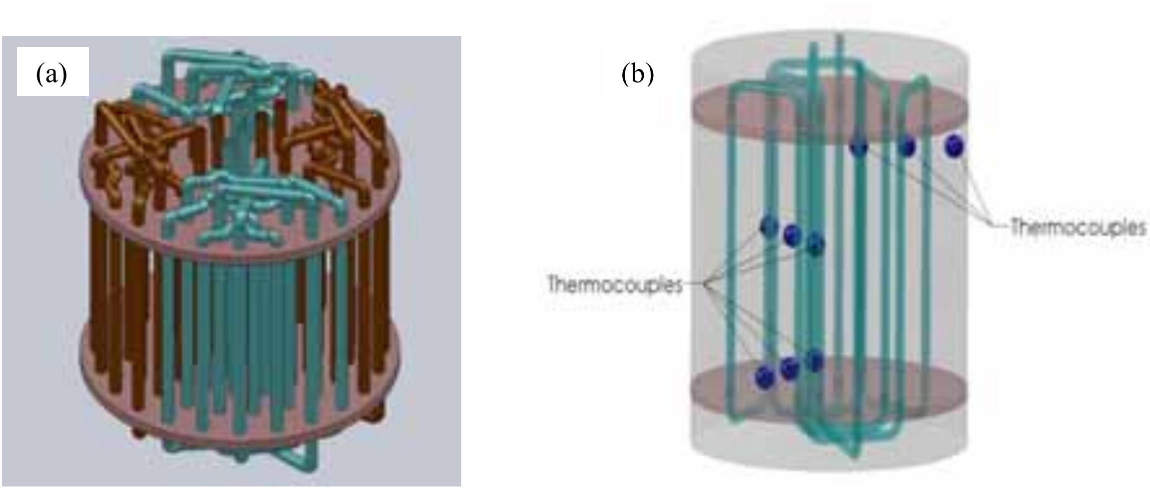


Figure 2. Schematic of Four-coils Tank (a) and schematic of the thermocouples location (b).

Table 1 shows the parameters for the four experiments. The mass flow rates, together with the inlet and outlet temperatures of the HTF were read and recorded, while nine thermocouples were located inside the tank to register the PCM temperatures at different positions (Figure 2 (b)). Figure 3 shows a schematic of the experimental set-up.

Table 1. Parameters for the experiments conducted

	One Coil	Two Coils	Four Coils
CF (-)	0.98	0.96	0.90
Heat transfer surface area (m^2)	0.172	0.365	0.823
PCM mass (kg)	30.02	29.38	27.62
Energy storage capacity (kJ)	4322	4231	3977
Tank diameter (m)	0.29	0.29	0.29
Tank height (m)	0.35	0.35	0.35
Total HTF pipe length (m)	5.46	11.62	23.83

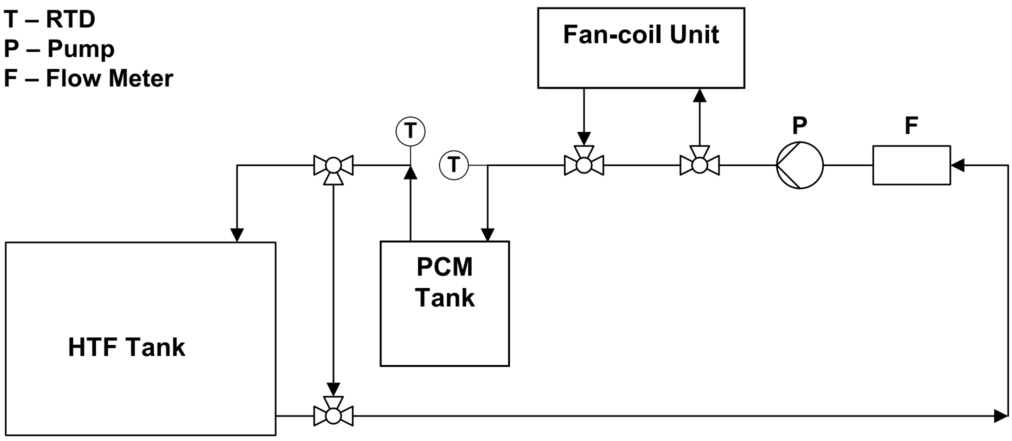


Figure 3. Schematic of the experimental setup.

3. Mathematical formulation

As expressed in [14], the effectiveness of a thermal storage system with PCM at time, t, can be defined by the effectiveness equation for boilers or condensers [15] as shown below:

$$\varepsilon = 1 - e^{-NTU} \tag{1}$$

Where $NTU = \frac{UA}{\dot{m}\, C_p} = \frac{1}{R_T \dot{m} C_p}$ (2)

In Equation (2), U refers to the overall heat-transfer coefficient and A is the heat transfer area for the heat flow. Thus, in order to formulate the mathematical models for effectiveness, the total thermal resistance, R_T, needs to be determined. Thermal resistance is a function of the overall heat transfer coefficient between the fluid and PCM, and the heat transfer area [12]. Considering a tube surrounded by a volume of PCM, the total thermal resistance, R_T, is given in Equations (3) and (4) where R_{HTF} is the resistance of the HTF, R_{WALL} is the resistance of the tube wall and R_{PCM} is the resistance of the PCM.

$$R_T = R_{HTF} + R_{WALL} + R_{PCM} \tag{3}$$

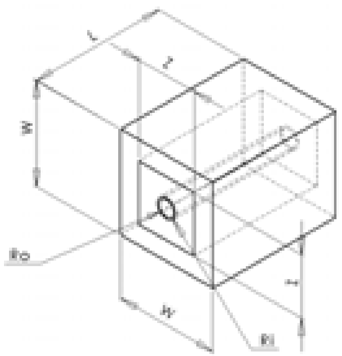
$$R_T = \frac{1}{2\pi R_i\, L h_f} + \frac{ln\left({R_o}/{R_i}\right)}{2\pi k_w\, L} + \frac{1}{S k_{PCM}} \tag{4}$$

In Equation (4), R_i is the inner radius of the tube while R_o is the outer radius of the tube. L is the length of the tube, h_f is the heat transfer coefficient of the HTF, k_w is the thermal conductivity of the tube wall, k_{PCM} is the thermal conductivity of the PCM and S is the shape factor of the PCM.

The mathematical model assumes that the phase change of the PCM will ultimately form a square (Figure 4 (a)). Figure 4 (b) shows the thermal circuit for the model. Convection is not be considered in this model, therefore the heat transfer through the PCM will only be by conduction. The formula of R_{PCM} will thus be utilizing the shape factor. The formula for R_T is shown in Equation (5). Equation (6) shows the formula for the shape factor, S found in ref. [15]. The 'Z' dimension in Figure 4 (a) is the changing width of the PCM during phase change while the 'W' dimension is the maximum width of the PCM after phase change.

$$R_T = \frac{1}{2\pi R_i\, L h_f} + \frac{ln\left({R_o}/{R_i}\right)}{2\pi K_w\, L} + \frac{1}{\frac{2\pi L}{ln\left(\frac{0.54Z}{R_o}\right)} . K_{PCM}} \tag{5}$$

$$S = \frac{2\pi L}{ln\left(\frac{0.54Z}{R_o}\right)} \tag{6}$$



(a) (b)

Figure 4. Simplified model (a) and thermal circuit (b).

The NTU for the heat exchanger analysis is constant for a given mass flow rate. The NTU, however, for a thermal storage system with a constant mass flow rate changes with time as the solid to liquid line, which defines U and A, are affected due to phase change. The phase change fraction is the proportion of material that has yet to change phase and relates directly to the solid to liquid line within the PCM. This can be used as the defining variables instead of time [12]. The phase change fraction (δ) for a tube surrounded by a square volume of PCM is defined as follows:

$$\delta = \frac{Z^2 - \pi r_o{}^2}{W^2 - \pi r_o{}^2} \tag{7}$$

Where $Z = \left\{\delta\left(W^2 - \pi R_o{}^2\right) + \pi R_o{}^2\right\}^{1/2}$ (8)

Substituting Equation (8) into (5), the total thermal resistance, R_T is defined as:

$$R_T = \frac{1}{2\pi R_i\, L h_f} + \frac{ln\left({R_o}/{R_i}\right)}{2\pi k_w\, L} + \frac{ln\left[\frac{0.54\left[\left\{\delta\left(W^2 - \pi R_o{}^2\right) + \pi R_o{}^2\right\}^{1/2}\right]}{R_o}\right]}{2\pi L k_{PCM}} \tag{9}$$

Therefore, the heat exchanger effectiveness can be written in terms of R_T:

$$\varepsilon = 1 - e^{\frac{-1}{\dot{m}\, C_p R_T}} \tag{10}$$

Equation (10) defines the effectiveness at a given mass flow rate and phase change fraction with respect to the width W.

The following formulae are utilized for the calculation of h_f:

$$Pr = \frac{\mu_f . C_p}{k_f} \tag{11}$$

$$Re = \frac{\dot{m} d_i}{A \mu_f} \tag{12}$$

For laminar flow, $Nu = 3.66 + \frac{0.0668\left({d_i}/{L}\right) R_e . P_r}{1 + 0.04\left[\left(\frac{d_i}{L}\right) . R_e . P_r\right]^{2/3}}$ (13)

For turbulent flow, $Nu = 0.023 . Re^{0.8} Pr^n$ (14)

Where n = 0.3 for cooling and n = 0.4 for heating,

$$h_f = \frac{Nu . K_f}{d_i} \tag{15}$$

Average effectiveness, $\bar{\varepsilon}$ can be defined as:

$$\bar{\varepsilon} = \frac{\int_{\delta=0}^{\delta=1} \varepsilon \; d\delta}{1-0}$$

$$\bar{\varepsilon} = \int_0^1 \left(1 - e^{\frac{-1}{\dot{m}\, C_p R_T}}\right) d\delta \tag{16}$$

As these equations could not be computed explicitly, $\bar{\varepsilon}$ needs to be computed numerically. To solve $\bar{\varepsilon}$ numerically, we need to find the area under the curve of ε.

Taking the δ interval to be 0.01,

$$\bar{\varepsilon} = Sum\ of\ \left[\varepsilon_{\delta=0.005} . (0.01) + \varepsilon_{\delta=0.015} . (0.01) + \cdots + \varepsilon_{\delta=0.995} . (0.01)\right] \tag{17}$$

4. Results and discussion

The freezing and melting processes in the PCM tank were conducted for each experiment and the results were analyzed. As the refrigeration system in the experimental set-up could only provide the HTF at a temperature slightly below the freezing point of the salt hydrate PCM, the difference between the tank inlet and outlet temperatures of the HTF during freezing were found to be too small leading to large experimental errors. The difference between the inlet and outlet temperatures for the melting process was however large enough for an accurate analysis to be conducted.

As mentioned by Belusko and Bruno [12], a PCM storage system can be seen and analyzed as a heat exchanger where the HTF exchanges heat with the PCM at the phase change temperature. Therefore, the effectiveness (ε) of the PCM storage system can be defined as that of a heat exchanger. Effectiveness of the storage system is the ratio of the actual heat transfer to the maximum possible heat transfer. The maximum effectiveness of the system arises when the outlet temperature of the HTF is the same as the phase change temperature.

A comparison between the numerical and the experimental values of effectiveness against the ratio of mass flow rate to area, during the discharge process for the one coil tank experiment using salt hydrate is shown in Figure 5 (a), while Figure 5 (b) and Figure 6 (a) show similar comparison for the two coils and four coils tank experiments, respectively. The graphs shows an agreement between the predicted values and the experimental values.

It can be observed from Figure 5 and Figure 6 (a) that the experimental values are higher than the theoretical values. This is due to natural convection that occurs during the melting process of the experiments. Natural convection will expedite the melting process thus causing a higher experimental effectiveness compared to the theoretical. The difference between the experimental and theoretical values is observed to decrease as the length of the tube increases (Figure 5 (b) and Figure 6 (a)). As the length of the tube increases, the tube to tube distance will decrease and thus reducing the resistance of the PCM. With the reduction of the PCM, the effect of natural convection will not be as significant. The one coil tank experiment (Figure 5 (a)) in this case, has the largest effect of natural convection due to the shortest length of tube used.

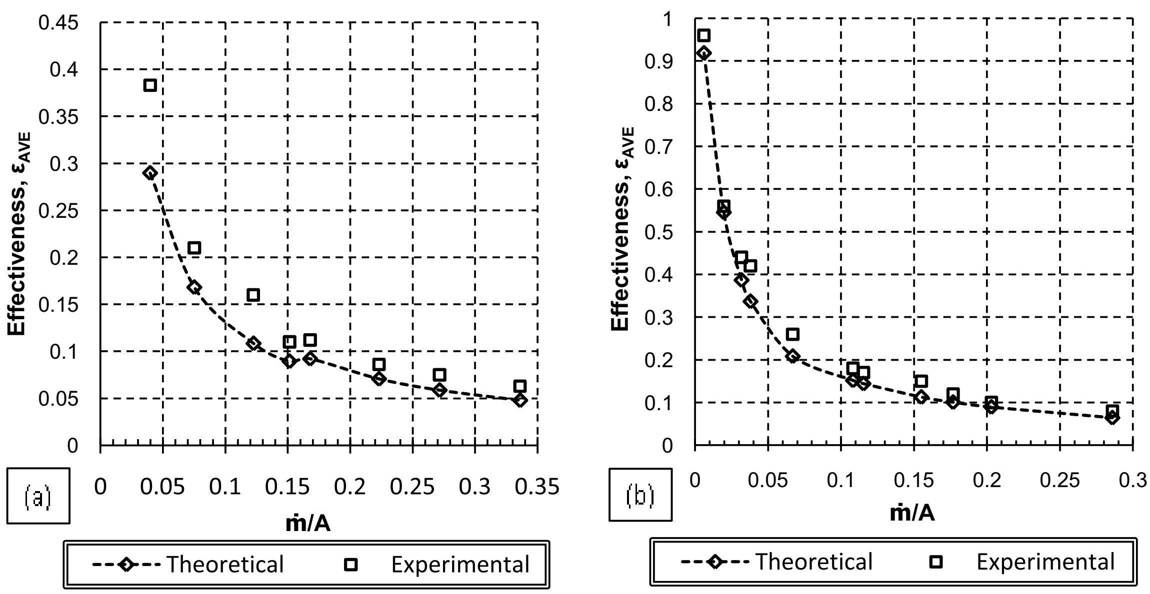


Figure 5. Comparison of Effectiveness over the ratio of ṁ/A for Experimental and Theoretical Melting Results for One Coil Tank (a) and Two Coils Tank (b) for Salt hydrate PCM.

Figure 6 (b) shows the comparison between the numerical and the experimental values of the effectiveness against the ratio of the mass flow rate to area, during the discharging process for the one, two and four coils tank experiments. It can be seen from the graph that both the experimental and numerical values form a single characteristic curve.

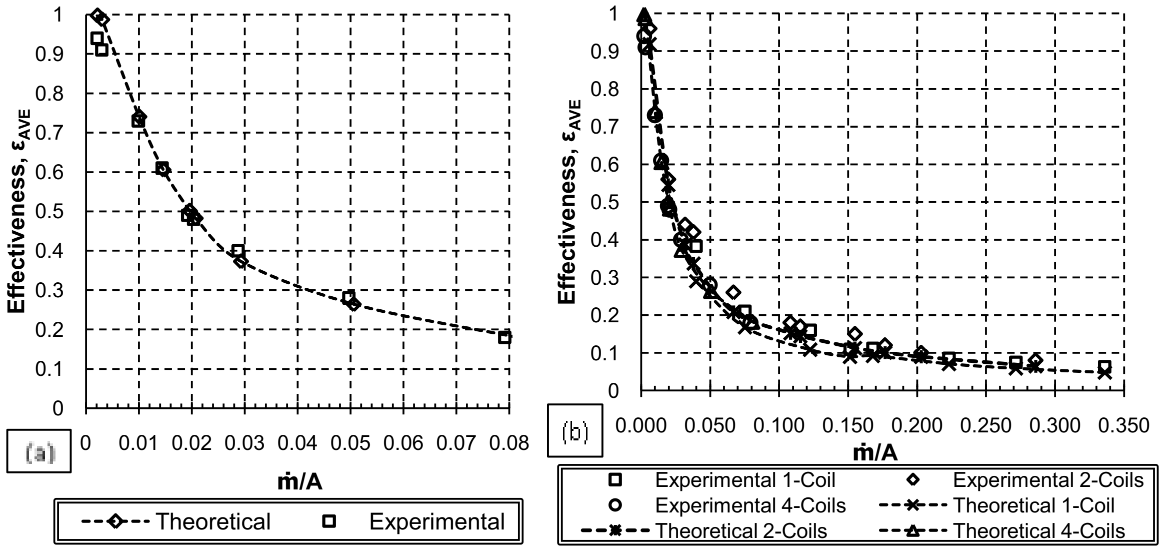


Figure 6**.** Comparison of Effectiveness over the ratio of $\dot{m}$/A for Experimental and Theoretical Melting Results for Four Coils Tank (a) and One, Two and Four Coils Tank for salt hydrate PCM (b).

Figure 7 show the comparison between the numerical and the experimental values of the effectiveness against the ratio of the mass flow rate to area, during the charging and discharging processes for the four coils tank experiment using water as PCM. It can be observed that the graphs revealed an agreement between the predicted and experimental values. Figure 7 (b) is also observed to have a sudden change in curvature at $\dot{\mathbf{m}}$/A = 0.057. This is owing to the change of flow of the HTF from laminar flow to turbulent flow.

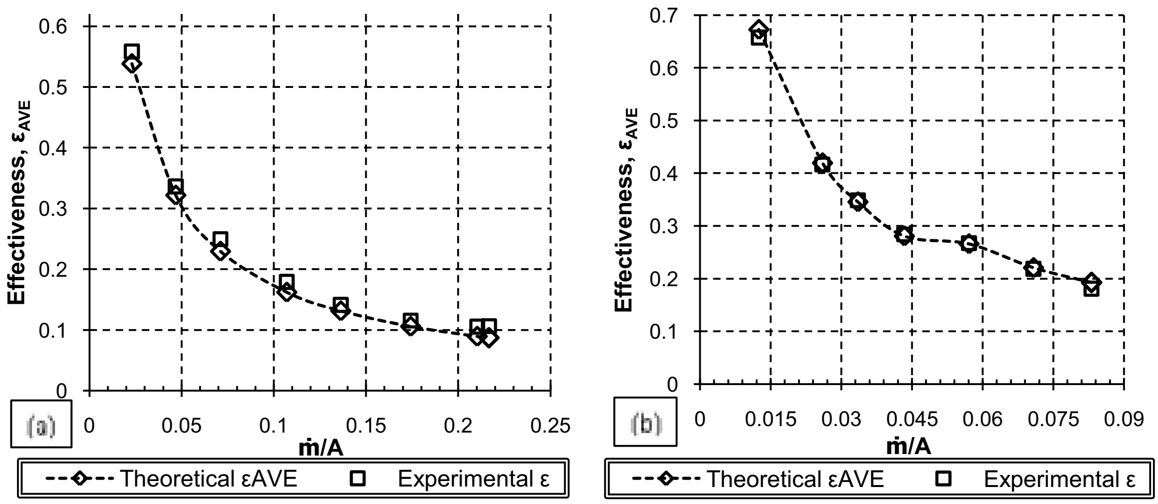


Figure 7**.** Comparison of Effectiveness over the ratio of $\dot{m}$/A for Experimental and Theoretical Freezing Results (a) and Melting Results (b) for Four Coils Tank (water as PCM).

5. Conclusion

A theoretical and experimental investigation was performed for a thermal energy storage system with coils of tube inside a PCM filled cylindrical tank. From this study, it may be concluded that the numerical model developed is able to accurately predict the effectiveness of the storage system during the charging and discharging processes.

References

[1] I. Dincer, M.A. Rosen. Thermal energy storage: Systems and applications. Wiley; 2002.

[2] K.A.R. Ismail, J.R. Henríquez. Solidification of pcm inside a spherical capsule. Energy Conversion and Management. 2000;41(2):173-87.

[3] Z. Ait Hammou, M. Lacroix. A hybrid thermal energy storage system for managing simultaneously solar and electric energy. Energy Conversion and Management. 2006;47(3):273-88.

[4] H. Ettouney, I. Alatiqi, M. Al-Sahali, K. Al-Hajirie. Heat transfer enhancement in energy storage in spherical capsules filled with paraffin wax and metal beads. Energy Conversion and Management. 2006;47(2):211-28.

[5] Adine HA, El Qarnia H. Numerical analysis of the thermal behaviour of a shell-and-tube heat storage unit using phase change materials. Applied Mathematical Modelling. 2009 4;33(4):2132-44.

[6] M.M. Farid, A. Kanzawa. Thermal performance of a heat storage module using PCM's with different melting temperatures: Mathematical modeling. Journal of solar energy engineering. 1989;111(2):152-7.

[7] J.F. Wang, Y.X. Ouyang, G.M. Chen. Experimental study on charging processes of a cylindrical heat storage capsule employing multiple-phase-change materials. Int J Energy Res. 2001;25(5):439-47.

[8] Fang M, Chen G. Effects of different multiple PCMs on the performance of a latent thermal energy storage system. Appl Therm Eng. 2007 [cited 3 July 2009];27(5-6):994-1000.

[9] Halawa E, Bruno F, Saman W. Numerical analysis of a PCM thermal storage system with varying wall temperature. Energy Conversion and Management. 2005 9;46(15-16):2592-604.

[10] Browne MW, Bansal PK. An elemental NTU-ε model for vapour-compression liquid chillers. Int J Refrig. 2001 7;24(7):612-27.

[11] Mathew B, Hegab H. Application of effectiveness-NTU relationship to parallel flow microchannel heat exchangers subjected to external heat transfer. International Journal of Thermal Sciences;In Press, Corrected Proof.

[12] Belusko M, Bruno F. In: Design methodology of PCM thermal storage systems with parallel plates. 7-10 October; Lisbon, Portugal. ; 2008.

[13] Mehling H, Cabeza LF. Heat and cold storage with PCM
an up to date introduction into basics and applications. Springers-Verlag Berlin Heidelberg; 2008.

[14] El-Dessouky H, Al-Juwayhel F. Effectiveness of a thermal energy storage system using phase-change materials. Energy Conversion and Management. 1997 4;38(6):601-17.

[15] Holman J. Heat transfer 9 th edition". McGraw Hill, London; 2002.

[16] Ismail KAR, Gonçalves MM. Thermal performance of a pcm storage unit. Energy Conversion and Management. 1999 1;40(2):115-38.