

Experimental investigation of heat transfer characteristics for a microencapsulated phase change material slurry

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ABSTRACT: A new microencapsulated PCM slurry (MEPCS) at high concentration (45% w/w) was developed based on microencapsulated Rubitherm RT6. Its heat storage and heat transfer characteristics have been experimentally investigated in order to assess its suitability for integration into a low temperature heat storage system for solar air conditioning applications. Differential scanning calorimetry tests have been conducted to evaluate the cold storage capacity and phase change temperature range. An experimental set-up was built in order to quantify the natural convection heat transfer occurring from a vertical helically coiled tube immersed in the MEPCS. First, tests were carried out using water in order to obtain natural convection heat transfer correlations and then a comparison was made with the results obtained for the MEPCS. It was found that inside the phase change interval the values of the heat transfer coefficient for the MEPCS were significantly higher than for water, under identical temperature conditions.

Keywords: Heat storage, Microencapsulated phase change material, Slurry, Natural convection heat transfer

1. INTRODUCTION

Recently, the use of phase change materials (PCMs) for heat storage has gained considerable interest with the development of micro-encapsulation techniques, in spite of the complexity of their thermodynamic and heat transfer behaviour. In this work, a new microencapsulated PCM slurry (MEPCS) at high concentration (45% w/w) was developed based on microencapsulated Rubitherm RT6. Its heat storage and heat transfer properties have been experimentally investigated in order to assess its suitability for integration into a low temperature heat

storage system for solar air conditioning applications.

Some introductory aspects of convective heat transfer in single-phase fluids and MEPCS materials are discussed in the next section.

1.1. Convective heat transfer in helically coiled tubes

Helically coiled tubes are effective heat transfer devices due to their simplicity, robustness and relatively high heat transfer rate to volume ratio. Helically coiled tubes are heat exchangers with liquid on both sides of the tube wall, and are extensively used in air conditioning, power generation, refrigeration and chemical engineering.

Most of the heat transfer studies are directed towards the inside tube, forced convection coefficient. The centrifugal forces that develop due to the tube curvature induce a secondary flow with a circulatory motion that causes the fluid particles to move towards the core region of the tube. This secondary flow reduces the thickness of the boundary layer, thus enhancing the heat transfer rate compared to a straight tube. The commonly accepted heat transfer correlation for forced convection inside helically coiled tubes is [1]:

$$Nu = 0.023 Re^{0.85} Pr^{0.4} (d_i / D)^{0.1} \quad (1)$$

The turbulent transition Reynolds number Re_t for flow in helically coiled tubes was established by Ito [2]:

$$Re_t = 2 \times 10^4 (d_i / D)^{0.32} \quad (2)$$

Naphon [3] reviewed the literature concerning flow and heat transfer in curved tubes. For helically coiled tubes, a large number of studies are reported for a wide range of conditions and working fluids. However, little investigation has been reported on the outside natural heat transfer coefficient. Ali [4] investigated experimentally natural convection occurring from vertical helically coiled tubes immersed in water. He studied two groups of coils with the exterior diameter of 8 and 12 mm, varying the pitch, the number of turns and the helix diameter. Based on the tube length (L) as the characteristic length, Ali [4] derived the following correlations for natural convection occurring from a helically coiled tube:

$$Nu_L = 0.685 (Ra_L)^{0.295} \quad (3)$$

for $d_o = 0.0012$ m, $3 \times 10^{12} \leq Ra_L \leq 8 \times 10^{14}$

$$Nu_L = 0.00044 (Ra_L)^{0.516} \quad (4)$$

for $d_o = 0.008$ m, $6 \times 10^{11} \leq Ra_L \leq 1 \times 10^{14}$

The following nomenclature was used in equations 1 – 4:

d_i internal diameter of the tube;

d_o external diameter of the tube;

D diameter of the helix.

To the authors' knowledge there isn't any general heat transfer correlation in the literature for natural convection occurring from a vertical helically coiled tube to a fluid taking into account the geometry of the tube and helical coil (diameter of the tube, diameter of the coil, the pitch and the number of turns). Most of the studies available in the literature refer to particular geometries and do not tend to generalization.

1.2. Convective heat transfer characteristics of microencapsulated phase change material suspensions

Phase change material slurries consisting of very fine particles of phase materials encapsulated in a polymer shell, have been increasingly used as heat storage medium. Compared to conventional fluids working in the sensible heat range, MEPCS exhibit high values of apparent specific heat capacity during the phase change process, that enhance the heat transfer rate between fluid and wall [6]. Proper design of cold/heat storage applications with PCS requires a good understanding of the heat transfer properties. Most studies available in the literature refer to forced convective heat transfer in tubes and micro-channels. Inaba [7] described a numerical solution and an experimental validation procedure for natural convection heat transfer (h_{nat}) occurring in a rectangular enclosure heated from below and cooled from the top. Inaba [7] found a significant increase of h_{nat} during the phase change. It was explained by a larger value of the temperature difference between wall and phase change slurry, that increased the heat flow as compared to a fluid without phase change. Inaba [7] found that increasing the mass concentration of PCM reduces the effect of heat transfer

enhancement, due to the significant increase of viscosity.

Zhang [6] developed a theoretical forced convection heat transfer model for a MEPCS under laminar flow conditions in a tube with constant wall temperature. Zhang found that the Stefan number and the concentration of PCM microcapsules were the most important parameters influencing heat transfer. Zhang's study reported an increase of the convective heat transfer coefficient (h_{conv}) as high as 4.5 times the value of a single phase fluid. Alvarado [8] investigated experimentally convective heat transfer of phase change slurry under turbulent flow, pointing out that h_{conv} for the MEPCM slurry is lower than for pure water at the same velocity values. For a smooth tube, Alvarado found values of the forced convection coefficient of approximately 4 times lower than for water (comparison was performed for Gnielinski correlation for forced convection inside circular tubes) for a PCM concentration of 16.5%. Conclusions similar to Inaba's [7] come out from Alvarado's investigation, that a significant increase of h_{conv} occurs during the phase change process. Inaba [9] investigated the natural convection heat transfer in rectangular enclosures filled with phase change slurry. Inaba derived generalized correlations for the natural convection coefficient for three regions: PCM in solid phase, in phase change region and in liquid phase. It was found that an increase of the

natural heat transfer coefficient (h_{nat}) up to 30% occurs in comparison to a fluid without phase change, depending on the pseudoplasticity index of power law model fluid.

The objective of this paper is to investigate the natural convection heat transfer occurring from a helically coiled tube immersed in a phase change material slurry. The main difficulty in developing a heat transfer correlation for MEPCS is the hysteretic behaviour of the thermal properties. The present study attempts a comparison between h_{nat} for MEPCS and for water, instead of developing a heat transfer correlation for MEPCS.

2. METHODOLOGY

2.1. Experimental setup.

The MEPCS investigated consisted of a 45% w/w aqueous dispersion of phase change material microparticles. The phase change temperature interval and the enthalpy change of the MEPCS listed in table I were determined by DSC analysis at two values of the heating rate: 0.1 °C/min and 0.01 °C/min.

The heat transfer coefficient of the MEPCM was measured experimentally and compared to the corresponding values for water, relative to the same bulk and wall temperatures. For water, a natural heat transfer correlation was derived for the particular geometry considered in this study.

Table I. Phase change range and enthalpy change of the MEPCS

Heating rate	Phase change temperature range Enthalpy change	Melting			Crystallization		
		Onset	End	t	Onset	End	t
0.1 °C/min	Characteristic temperatures in °C	4.1	7.5	3.4	5.8	2.2	3.6
	Enthalpy change in kJ/kg	53			56		
0.01 °C/min	Characteristic temperatures in °C	4.0	6.8	2.8	6.0	3.6	2.4
	Enthalpy change in kJ/kg	55			49.4		

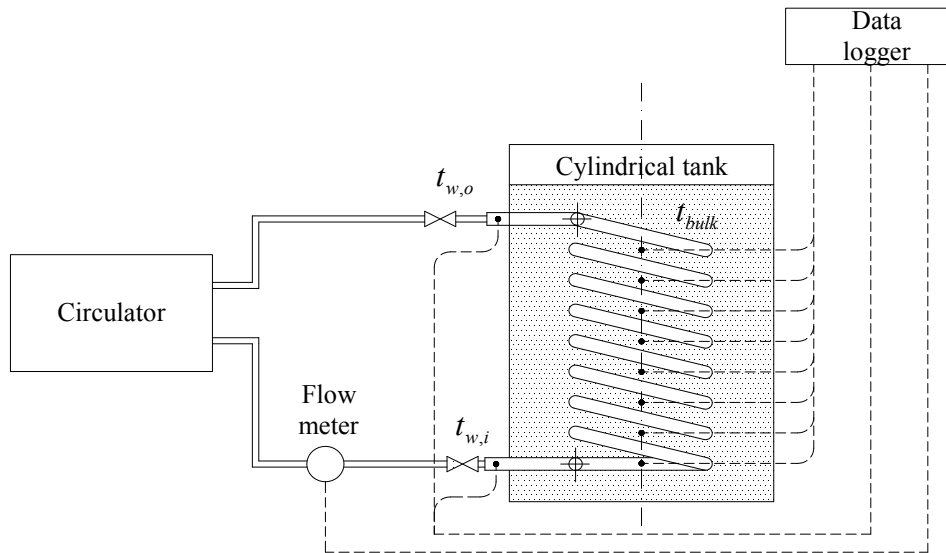


Figure 1. Schematic drawing of the experimental setup

A schematic drawing of the experimental setup is shown in figure 1. A heat exchanger consisting of a helically coiled copper tube immersed in a cylindrical tank was designed and built. The diameter of the helix – consisting of eight turns – was 0.14 m and the pitch was 0.037 m. The helical tube was fabricated from a 3.7 m copper tube with the interior/exterior diameter of 0.014/0.016 m. The coil was inserted coaxially into the tank having a diameter of 0.24 m and height of 0.55 m. Thermal insulation was achieved by using an elastomeric thermal insulation ($0.04 \text{ W}/(\text{mK})$) on the exterior surface. Distilled water was circulated through the coil at a controlled flow rate by using a circulating pump, included in a refrigerated/heated unit. The unit (Julabo HE, Germany) adjusted the temperature of distilled water within $\pm 0.01 \text{ }^\circ\text{C}$ to the set-point value. Eight type T thermocouples were tied up with metallic wires at equal distances in the axis of the system. Pt100 sensors were used for measuring the temperature of water circulating through the helical tube. The flow rate of water circulating through the helical tube was measured with an electromagnetic flow meter (ABB, Germany).

The thermo-physical properties of distilled water were assumed to be constant

and were evaluated at the average temperature. The bulk temperature inside the cylindrical tank was considered as an arithmetic average of the values recorded by the thermocouples. Before conducting a new set of measurements, the liquid in the bath was stirred in order to eliminate the stratification that occurs naturally during heating up or cooling down. Data recording was stopped when the vertical gradient of the temperature along the axis of the tank was higher than 1°C .

Experiments with water were followed by experiments with MEPCS.

2.1. Principles of analysis

The heat flow exchanged between the helical tube and the liquid in the tank $\dot{Q}$ was computed from the heat balance:

$$\dot{Q} = G c_w (t_{w,i} - t_{w,o}) \quad (5)$$

The overall heat transfer coefficient related to the external surface area of the helical tube can be computed from:

$$U_{ext} = \dot{Q} / (S_{ext} \cdot \Delta t_{med}) \quad (6)$$

The external heat transfer coefficient h_{ext} can be easily determined from the definition of the overall heat transfer coefficient relative to the external heat transfer surface area U_{ext} as:

$$\frac{1}{U_{ext}} = \frac{1}{h_{ext}} + \frac{d_{ext}}{k} \ln \frac{d_{ext}}{d_{int}} + \frac{d_{ext}}{d_{int}} \cdot \frac{1}{h_{int}} \quad (7)$$

The internal convection coefficient h_{int} can be evaluated using the heat transfer correlation for forced convection in helically coiled tubes, [1]:

$$Nu_{int} = 0.021 Re^{0.85} Pr^{0.4} (d_i / D)^{0.1} \quad (8)$$

The following nomenclature was used in equations 5 – 8:

G water mass flow rate through the helically coiled tube;

S_{ext} heat transfer surface of the tube relative to the outer diameter;

$t_{w,i}$, $t_{w,o}$ water temperature at the inlet and outlet respectively;

Δt_{med} logarithmic mean temperature difference;

k thermal conductivity of the tube wall.

3. RESULTS AND DISCUSSION

Experiments with water were conducted in order to develop a correlation for natural convection heat transfer for the geometry considered. The length of the tube was used as the characteristic length in defining Nusselt and Rayleigh numbers. The experimental data points are presented in figure 2. A total number of 107 data points were collected and the points for which the absolute value of the temperature difference between inlet and outlet was less than 0.15°C were discarded. The number of points considered in the analysis was equal to 57.

A least-squares fit yielded the following correlation:

$$Nu_L = 0.802 Ra_L^{0.278}, \quad 10^{11} < Ra_L < 10^{13} \quad (9)$$

The correlation coefficient is 0.87, the standard deviation of the fit 0.136 and the standard error 0.00238. This correlation can be considered satisfactory because of the relatively high value of the correlation coefficient and from statistical analysis of the residuals.

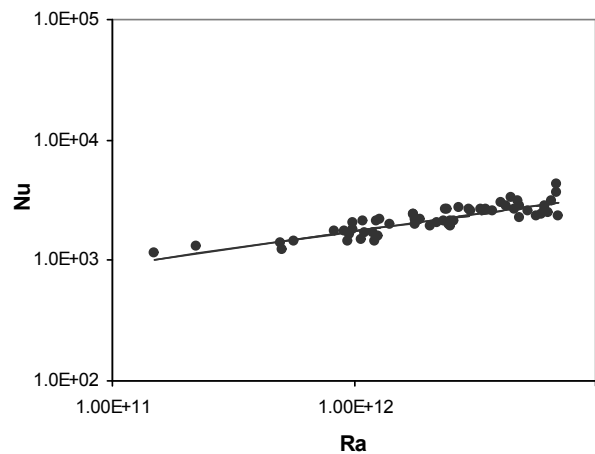


Figure 2. Natural heat transfer correlation for water

The correlation above was used in order to compare the heat transfer coefficient for water and for the PCMS, measured under identical conditions (bulk and wall temperature).

The principle of comparison was the following: for the temperature conditions (bulk and wall temperature) resulting from

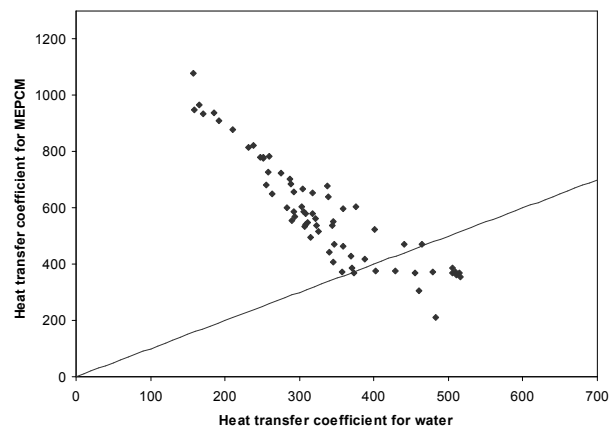


Figure 3. Heat transfer coefficient for MEPCM plotted against heat transfer coefficient for water

heat transfer coefficient measurements for MEPCS, the corresponding value of h_{nat} for water was computed, using the correlation developed for water. The values of the heat transfer coefficient for MEPCS were plotted against their water homologues in figure 3. The plot area is divided by the line $h_{\text{PCM}} = h_{\text{water}}$ in order to facilitate the comparison.

In an attempt to correlate the values of h_{nat} for MEPCS and for water, a regression analysis was performed (figure 4). Although the value of the Pearson's coefficient of regression, $R^2 = 0.805$, indicates a significant degree of correlation, no conclusion related to a functional dependence between the two heat transfer coefficients can be derived.

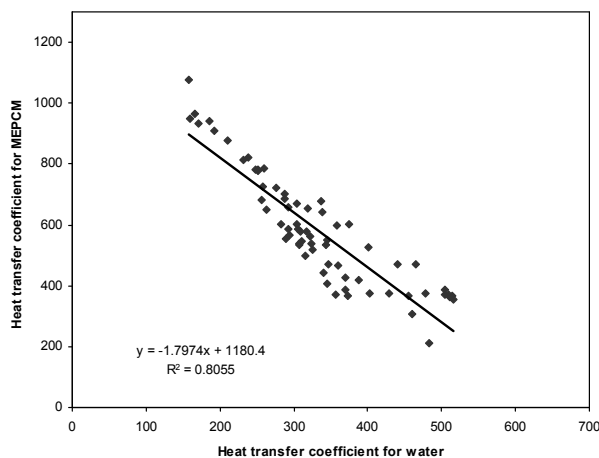


Figure 4. Correlation between heat transfer coefficient for water and for MEPCS

Between approximately 3°C and 6°C (phase change temperature range) the values of h_{nat} for MEPCS are significantly higher than the corresponding values for water, as shown in figure 5 (the points that lie inside the phase change temperature range were marked with triangles in order to distinguish them from the points outside the phase change range).

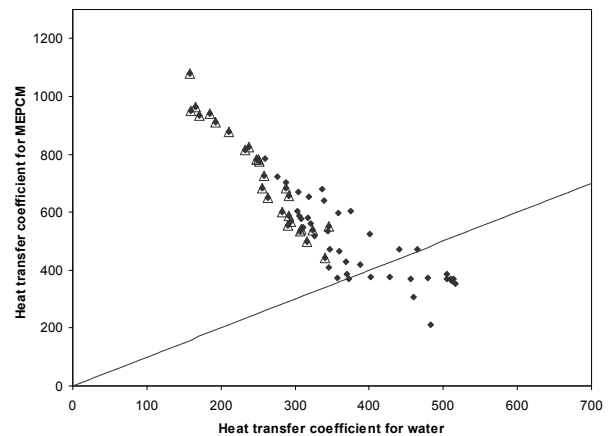


Figure 5. Data points inside the phase change interval

In order to distinguish between the temperature conditions that lead to values of h_{nat} for MEPCS higher and lower than for water, the following 3D representation was used: the values of h_{nat} for water were plotted taking as independent variables the bulk temperature and the wall-bulk temperature difference and the values of the heat transfer coefficient for MEPCS were overlapped as a set of points. By setting the view point along the positive and then the negative sense of the h_{nat} axis, one can identify the temperature conditions for which the heat transfer coefficient of the MEPCS is higher and lower than for water.

In figures 6 and 7 a color scale (ranging

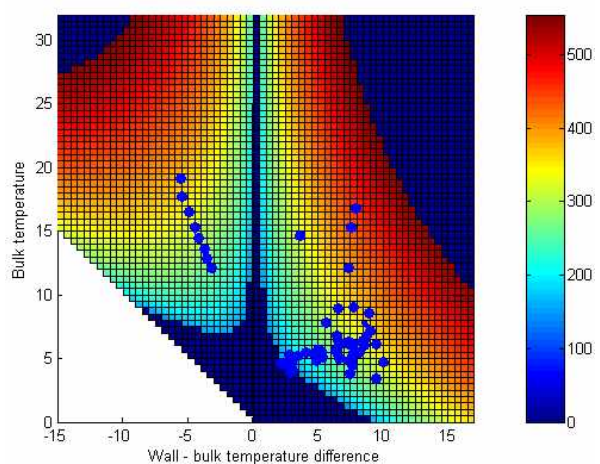


Figure 6. Data points with values of the heat transfer coefficient for MEPCS higher than for water

from 0 to $550 \frac{W}{m^2 \cdot ^\circ C}$) was used for the values of h_{nat} for water. Since the heat transfer correlation for water is only valid in the interval $10^{11} < Ra_L < 10^{13}$, the values of h_{nat} were set to 0 for temperature conditions that lead to Ra_L values outside the interval mentioned previously. This convention was used in order to facilitate the representation and it applies for all points in which the heat transfer coefficient is 0, except the points where the wall-bulk temperature difference is 0.

In figure 6 the view point is set above the surface represented by the values of h_{nat} for water in the negative sense of the h_{nat} axis. As a result, the surface obstructs the visibility of the points underneath and the only visible points are the ones with the value of h_{nat} higher than for water. In figure 7 the view point is set beneath the surface in the positive sense of the h_{nat} axis. The visible points are the ones with the value of the heat transfer coefficient lower than for water.

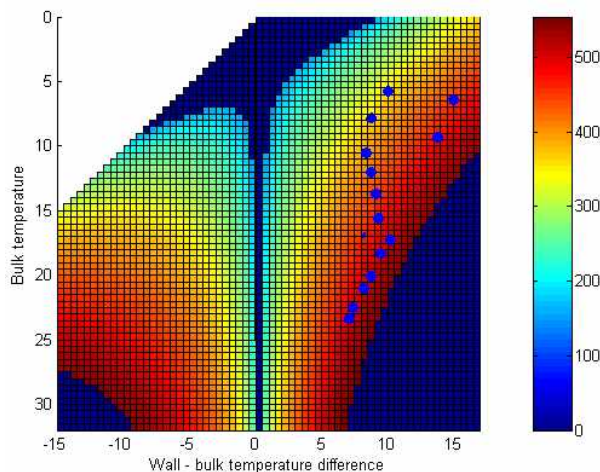


Figure 7. Data points with values of the heat transfer coefficient for MEPCS lower than for water

For values of the bulk temperature inside the phase change range, h_{nat} for MEPCS is higher than for water at values of the wall-bulk temperature difference up to approximately 10 °C. For values of the wall-

bulk temperature difference above 10 °C, h_{nat} for water becomes significantly higher than for MEPCS. At values of the bulk temperature above 20 °C, h_{nat} for water becomes higher than for MEPCS.

The difference between the values of h_{nat} for MEPCS and water could be generated by two causes: (1) the decrease of the heat transfer coefficient for water in the vicinity of 4°C (bulk temperature) and (2) an increase of h_{nat} for MEPCS within the phase change range. The weight of each cause can be easily assessed if the view point is set along the wall-bulk temperature difference axis (figure 8). In the phase change range the values of h_{nat} for MEPCS are significantly higher than the corresponding values for water.

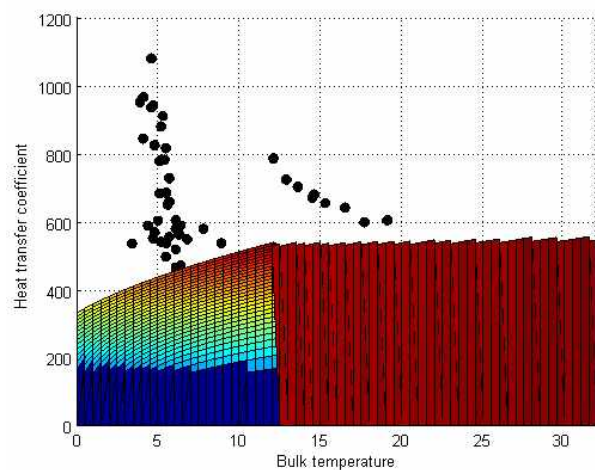


Figure 8. View point set along the wall – bulk temperature difference axis

4. CONCLUSION

Due to the hysteresis-like variation of the thermo-physical properties of the MEPCS, that corresponds to a non-univocal dependence of the properties on temperature, it is not possible to derive a heat transfer correlation. For the heat transfer geometry considered – helically coiled tube inserted vertically into a tank with stationary fluid – a natural heat transfer correlation was developed for water. The values of h_{nat} for MEPCS were measured for various temperature conditions. A comparison with

the corresponding values for water, for the same heat transfer geometry, was performed. Inside the phase change interval the MEPCS displays higher values of h_{nat} than water, that can go up to 5 times depending on temperature conditions. For values of the bulk temperature above 20 °C, h_{nat} for water becomes higher than for MEPCS.

A significant correlation was identified between the value of h_{nat} for MEPCS and for water, suggesting a functional dependence between the two.

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