

# MODELING OF A HEAT EXCHANGER UTILIZING A PHASE-CHANGE MATERIAL FOR SHORT-TERM STORAGE

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## ABSTRACT

A computer program was proposed to simulate the behavior of a low temperature phase change material (PCM) in a heat exchanger for short term storage. An experimental set-up of mainly a heat exchanger with staggered tube arrangement and air temperature control unit were used. In modeling approach, the PCM in the tube was divided into ten concentric cells of equal mass, and was logically treated in terms of energy balance. Experimental inlet air temperature (at three different flow rates) was utilized as input data to test the model output results and their reliability. Excellent agreement were obtained for the outlet air temperature when compared with the experimental measurements, and differences did not exceed 0.5 °C over the simulated period. The predicted PCM average temperature history showed good agreement with the experimental ones, and differences did not exceed 2 °C at the highest applied air flow rate. This modeling approach can be used for any PCM provided that its thermo-physical properties are available. The transient moving front for freezing or melting can be predicted, and consequently the mass fraction of either liquid or solid phase to the total PCM mass can be predicted as well.

## NOMENCLATURE

A outer surface area for a single tube, m<sup>2</sup>  
 $C_p$  specific heat, J/kg.°C  
d outer tube diameter, m  
h heat transfer coefficient, W/m<sup>2</sup>. °C  
 $h_{sf}$  heat of solidification, J/kg  
k thermal conductivity, W/m °C  
L tube length, m  
N number of the cell at which phase change takes place  
n number of tubes in a row  
Nu Nusselt number  
Q air volume flow rate, m<sup>3</sup>/s  
q rate of heat transfer, W  
 $Q_L$  heat absorbed by liquid phase, J  
 $Q_s$  heat conducted to the solid part, J  
 $Q_{sen}$  sensible heat, J  
 $Q_{sf}$  heat required for melting, J  
 $Q_1$  total heat convected from the tube surface =  $q \Delta t$ , J  
m mass, kg  
 $Pr_s$  Prandtl number, properties at tube surface temperature  
 $Pr_\infty$  Prandtl number, properties at incoming air temperature.  
R Thermal resistance (m °C)/W  
r cell radius, m  
Re Reynolds number =  $\rho v d / \mu$   
T temperature, °C  
t time,  
 $x_1$  ratio of heat used in melting to the total heat convected to the tube.  
v actual velocity, m/s.

## Subscript

a air  
o outer  
in inner  
s surface, solid phase  
t tube  
L liquid phase

## Greek symbol

$\rho$  density kg/m<sup>3</sup>  
 $\mu$  viscosity

## INTRODUCTION

Energy storage in the form of thermal energy is generally employed when it is to be used in the same form. It is generally stored either as sensible heat, or latent heat. The later option has the advantage of providing energy at almost constant temperature, since the medium undergoes to phase change.

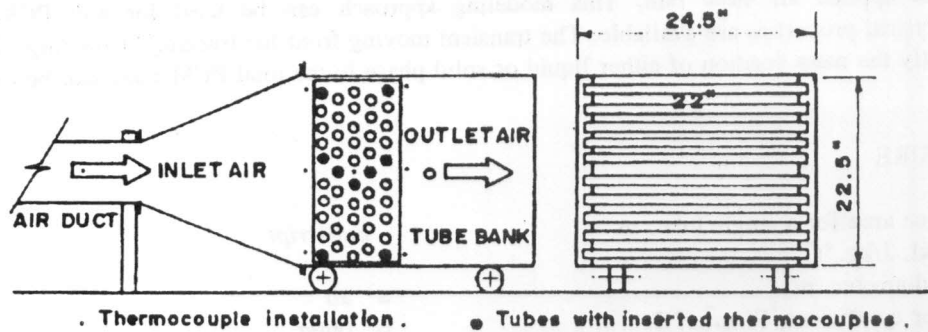
The properties of a phase change material (PCM) have to satisfy some thermal, kinetic, chemical and finally economic criteria. Information is available for numerous of PCMs [1,2,3,4]. McMullin and Ben-abdallah [5] carried out an extensive work on PCMs, and concluded that Calcium Chloride hexahydrate with added water was found to be the most promising storage media. It has a phase change at approximately 25 °C. Also, addition of a nucleating agent

would overcome the super-cooling problem. Johnstone [6] concluded that a mixture of  $\text{CaCl}_2 \cdot 5.5\text{H}_2\text{O}$  with 2 % of  $\text{Ca}(\text{NO}_3)_2$  as a nucleating agent could be selected as the most suitable PCM for phase transition temperature of 25 °C. Containerization of the PCM is highly questionable. In small heat storage, the surface area to volume ratio is large, and therefore the cost of packing material is an important factor. Different application of PCM as a storage media with various configurations were reported [7]. However; for small scale storage, phase change storages with higher energy densities are more attractive [6,8]. Modeling of heat transfer for problems involving PCM were proposed and applied, and extensive literature reviews are available [9,10]. The objective of the present work is to simulate the heat transfer from a heat exchanger utilizing a phase change material for short term storage.

EXPERIMENTAL SET-UP

An experimental heat exchanger with staggered tube configuration was constructed at the agricultural engineering department workshop, of the Technical University of Nova Scotia, Canada, during the summer of 1991. The specific information and dimensions are given in the schematic diagram in Figure (1). To formulate the PCM, a premixed proportions according to Johnstone [6] were prepared. The proportions were 51.8 % calcium chloride, 2 % Calcium nitrate and 46.2 % water by weight. A precalculated volume of the PCM was filled in the heat exchanger tubes to allow safe expansion at high temperatures.

The thermo-physical properties of such PCM is given in the following table as taken from Johnstone [6].



Tubes O.D = 2.85 cm. Total No. of tubes = 48. No. of rows = 5  
 Wall thickness = 0.28 cm. Total surface area = 2.41 Sq. m.  
 Tube spacing / Diam. = 2.0, No. of tubes per row = 10

Figure 1. Schematic of experimental se-up.

Table 1. Thermo-physical Properties of the PCM [6]

<u>Solid phase:</u>	
Density	1680 kg/ m <sup>3</sup> (15 -20 °C)
Specific heat	1303 J/ (kg. °C) (15 to 20 °C)
Melting point	25 °C
Heat of fusion	131 kJ/ kg
Thermal conductivity	1.11 W/ (m. °C)
<u>Liquid phase:</u>	
Volumetric thermal expansion coefficient	5.94 x 10 <sup>-4</sup> K <sup>-1</sup> (30 to 40 °C)
Viscosity	12.25 x 10 <sup>-3</sup> NSm <sup>-2</sup> , 50 °C
	16.48 x 10 <sup>-3</sup> NSm <sup>-2</sup> , 40 °C
	24.08 x 10 <sup>-3</sup> NSm <sup>-2</sup> , 30 °C
Density	1510 kg/ m <sup>-3</sup> , 40 °C
	1520 kg/ m <sup>-3</sup> , 40 °C
	1530 kg/ m <sup>-3</sup> , 40 °C
Specific heat	2244 J /kg. °C (25 to 40 °C)
Thermal conductivity	0.57 W/ m. °C (25 to 40 °C)

A controlled air source was used to supply air at desired temperatures. Temperature up to 35 °C was used for melting the PCM, while 15 °C was used for cooling. Air flow rates were varied by using different pulleys of different diameters to change the fan speed. Three levels of air flow rates were used. Values of 150, 300, and 450 m<sup>3</sup>/hr were applied. A set of T-type, thermocouple were utilized to monitor the following thermocouple: three for averaging the air temperature just before the tube banks and another set of three to average the outlet temperature just after the tube banks. One more thermocouple were placed at the center of heat exchanger. Nine thermocouple were placed in nine different tubes to monitor the PCM melting and solidifying temperature history, while an extra thermocouple was mounted on the middle tube external surface. All the thermocouple locations are shown in Figure (1). A fully programmable datalogger was utilized to record and store the temperature measurements.

MODELING APPROACH

As heat is extracted from a PCM, crystallization will occur at the walls and then progressively inward into the material. At the end of the crystallization, heat must be transferred across layers of solid to the container wall [11]. As a solidification material is heated, melting occurs first at the walls and then inward toward the center of the container as shown in Figure (2).

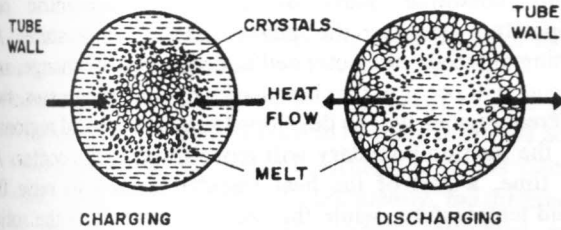


Figure 2. The heat flow in the phase change material in a horizontal tube.

In the present work the following assumptions are assumed:

- 1- The PCM inside the tube is divided into ten concentric cells of equal mass.
- 2- The volume change of the PCM during phase change will affect the length of the storage material inside the tube.
- 3- The change of heat content of the tube material is negligible. This is due to the fact that the average temperature of the tube material during phase change is almost constant, and hence the temperature variation with time is almost zero. In the same time, the heat capacity of the tubes is very small compared to the PCM (in the order of one tenth).
- 4- The heat transfer across each concentric cell is by pure conduction, no matter what phase the cell is in. The convection mode of heat transfer is simply ignored within the cell.

In the present work, three different cases were simulated to cover all different possibilities that the PCM is subjected to as follows:

- 1- Single phase (liquid or solid), in which the PCM is gaining or giving away sensible heat.
- 2- Melting case, in which the PCM is changing from solid to liquid phase by gaining latent heat.
- 3- Solidification case, in which the PCM is changing from liquid to solid phase by losing latent heat.

Case 1: Single Phase

In this case, the PCM is in a single phase (liquid or solid). If the total amount of heat convected from the tube surface is  $q$ , then each concentric cell will contribute to that  $q$  by one tenth. For the  $i$ th cell, the conducted heat transfer across

its surface facing the incoming direction of heat flow will be  $(i-1)/10$  from  $q$ , while the heat conducted from the other surface is  $(i/10) \times q$ , hence creating an instantaneous temperature gradient represented by  $T_i$  and  $T_{i+1}$  at some instant of time  $t$  as shown in Figure (3). During an interval time  $\Delta t$ , this cell will gain (or loose ) some sensible heat that changes that gradient to become  $T'_i$  and  $T'_{i+1}$ . This new temperature distribution could be considered as an initial condition for the next time interval. The central cell was considered to have a uniform temperature that will change with time. This procedure would keep on going until the outer cell temperature reaches the phase change temperature.

To calculate the thermal resistance of each cell its radii were calculated from:

$$r_i = [(r_{i+1})^2 - m/(10\pi L\rho)]^{1/2} \tag{1}$$

Then the thermal resistance would be

$$R_i = \ln \left[ \frac{r_{i+1}}{r_i} \right] / k_{PCM} \tag{2}$$

The total convected heat transfer rate from the outer surface of the tube during an interval time  $\Delta t$  would be :

$$q = h \cdot A \cdot (T_s - T_a) \tag{3}$$

The heat transfer coefficient  $h$  was calculated, as given in Ref.[12]

$$Nu = \frac{hd}{k_a} = C Re^n Pr^{0.36} \left[ \frac{Pr_\infty}{Pr_s} \right] \tag{4}$$

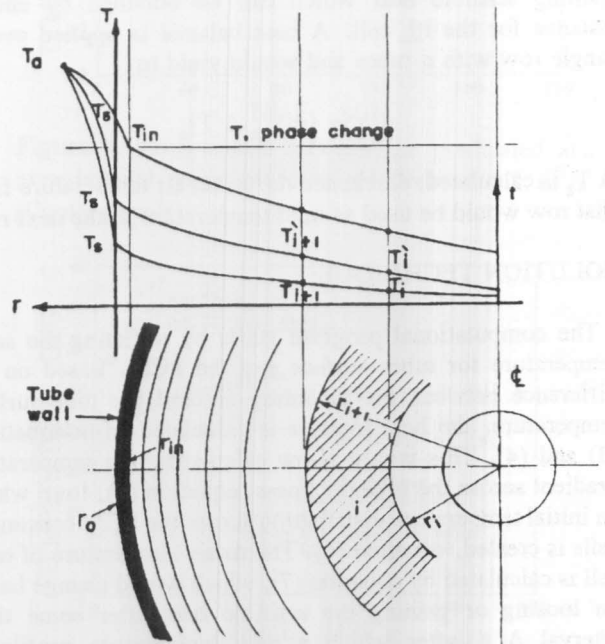


Figure 3. Schematic representation for the concentric cells and the transient temperature profiles (case 1).

C and n are constants depending on values of Reynolds number (Re). Values of air velocity measurements by a hot wire anemometer were corrected to calculate the actual air velocity over the tube banks and consequently Reynolds number calculations. Also, the air properties mainly density and viscosity, which are temperature dependent were calculated based on tabulated values in text books and correlation equation given in Ref.[13].

The surface temperature  $T_s$  at any time interval can be found from:

$$T_s = T_{in} - q \ln (r_o/r_{in})/(2\pi Lk_t) \quad (5)$$

The temperature distribution across the  $i^{th}$  cell would be given by:

$$T_i = T_{i+1} + \frac{q(\frac{i}{10})R_i}{2\pi L} \quad (6)$$

The mean bulk temperature for the  $i^{th}$  cell is obtained by integrating sensible heat contained in a finite ring within the cell over the inter cell, and can be given as:

$$T_{m,i} = T_i + (T_{i+1} - T_i) \left[ \frac{r_{i+1}^2}{r_{i+1}^2 - r_i^2} \right] - \left[ \frac{T_{i+1} - T_i}{2 \ln(\frac{r_{i+1}}{r_i})} \right] \quad (7)$$

This mean temperature will change with time by loosing or gaining sensible heat which can be obtained by energy balance for the  $i^{th}$  cell. A heat balance is applied over a single row with n tubes and would yield to

$$q.n = [m \cdot C_p \Delta T]_a \quad (8)$$

$\Delta T_a$  is calculated, and hence the outlet air temperature from that row would be used as inlet temperature to the next row.

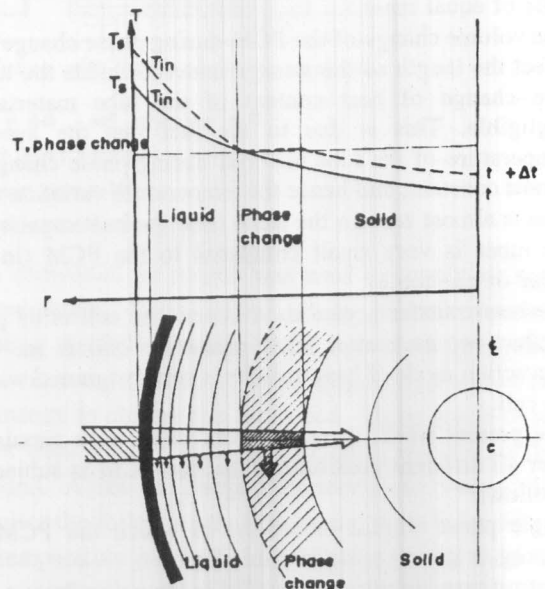
**SOLUTION TECHNIQUE:**

The computational program starts by initiating the same temperature for tubes surface and the PCM. Based on the difference between the incoming air and the tube surface temperature, the heat transfer is calculated from equations (3) and (4). This would allow calculating the temperature gradient across the tube thickness, equation (5), from which an initial temperature distribution across the PCM concentric cells is created, equation (6). The mean temperature of each cell is calculated by equation (7), which would change based on loosing or gaining the sensible heat after some time interval  $\Delta t$ , after which a new temperature profile is obtained. The later profile is utilized as an initial condition for the next time interval, for all tubes in that row. The outlet air temperature is obtained for that row from equation

(8), which in turn would be used as inlet temperature for the next row. This procedure is continued for the total number of rows and for the same time interval  $\Delta t$ . A repetitive procedure is applied for the successive time intervals until the outer cell of the PCM reaches the phase change temperature where a new case would be applied. Each row would reach this condition independently.

*Case 2: Melting the PCM:*

This condition starts as the outer concentric cell temperature reaches the phase change temperature. As heating proceeds, the outer cell undergoes phase change, and melting will occur at that boundary. As time progresses, two different regions will be developed (liquid and solid regions), and the melting boundary will move towards the center. At any time, a part of the heat transfer is used to raise the liquid temperature, while the rest is conducted to the solid. As heat being conducted the interface cell will utilize some of the heat to raise its temperature to phase change temperature (sensible), while some other part will be used in melting a part of it (phase change). The rest of the heat will be conducted toward the solid part raising its temperature as shown in Figure (4).



**Figure 4.** Schematic representation for the concentric cells showing the moving boundaries in the heating cycle (case 2).

At any time, the heat balance equation for the total mass of PCM would be

$$q \cdot \Delta t = Q_L + [Q_{sf} + Q_{sen}] + Q_s \quad (9)$$

heat convected to the tube liquid phase + heat absorbed by in melting dm + heat utilized to the solid phase + heat conducted to the solid phase

The sensible heat is calculated from:-

$$Q_{\text{sen}} = dm C_p \Delta T \quad (10)$$

where  $dm$  is newly melted part from the interface cell, and  $\Delta T$  is the temperature difference between its current temperature and the phase change temperature. On the other hand,  $dm$  is calculated from the amount of heat utilized in fusion ( $q_{\text{sf}}$ ) which can be given as

$$Q_{\text{sf}} = dm h_{\text{sf}} = x_1 q \Delta t \quad (11)$$

where  $x_1$  is a fraction representing the amount of heat used in melting, taking values from one to zero and is always automatically adjusted to satisfy the energy balance given in equation (9). The melted portion of mass is always added to the liquid mass, and would determine the mass fraction of either liquid or solid phase. Consequently, the different radii of the melted mass  $dm$  are simply calculated and the temperature profile across this mass before subjected to being melted is utilized to calculate the heat conducted to the solid part. This solid part is mathematically treated as in case 1. This is to obtain a temperature profile in the solid phase.

The part of heat that is used in melting ( $Q_{\text{sf}} + Q_{\text{sen}}$ ) and that conducted to the solid ( $Q_s$ ) had to pass through the liquid zone. This would create a new temperature profile across the liquid zone with boundary temperature equal to the phase change temperature. From that profile, a new temperature is calculated as in equation (7), from which its difference from the old mean temperature is obtained, and hence the sensible heat across the liquid zone is calculated ( $Q_L$ ).

#### Solution Techniques:

At any time interval a value of one for  $x_1$  is used to calculate  $Q_{\text{sf}}$ ,  $Q_{\text{sen}}$ ,  $Q_s$  and  $Q_L$  as previously explained. Applying equation (9) and adjusting  $x_1$  by decreasing its value with an increment of 0.01 until a balance is obtained at the end of time interval,  $\Delta t$ . At this point, the final value of  $x_1$  would lead to the exact melting mass that is added to the liquid phase, and hence, the fraction of the liquid (or solid) from the total PCM mass is obtained. The liquid zone is divided into an integer number of cells equal to ten time that fraction. The temperature profile across these cells has to be modified to allow each cell to absorb its share of  $Q_L$ . From that profile, a new mean temperature is calculated, and is used as initial mean temperature for the new time interval. The whole procedure is continuously repeated until the total mass of PCM is totally changed to pure liquid, which is then treated as case 1.

#### Case 3: Solidification of PCM

This case is typically as case 2 except that the solidification starts at the outer cell and moves towards the center. The mathematical treatment is similar as in case 2, taking into consideration the solidification front instead of the melting front.

#### RESULTS AND DISCUSSIONS

Analytical solutions for problems involving PCM are limited due to the mathematical complexity. This is because of the variable properties and moving boundaries of the PCM. With the current modeling approach, the computer program was first checked against its results reliability. The predicted outlet air, and the PCM temperature histories were compared with their corresponding experimental ones. Experimental inlet air temperature measurements were utilized as the driving force for the model. This was applied to both cooling and heating cases, and samples of comparison are given in Figures (5), (6) and (7).

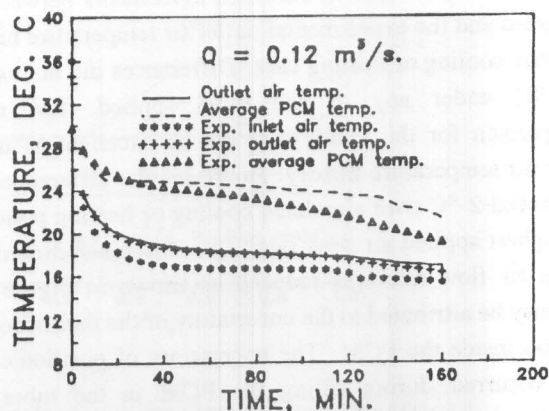


Figure 5. Comparison between the predicted and experimental temperature at air flow of  $1.2 \text{ m}^3/\text{s}$ . (Cooling cycle).

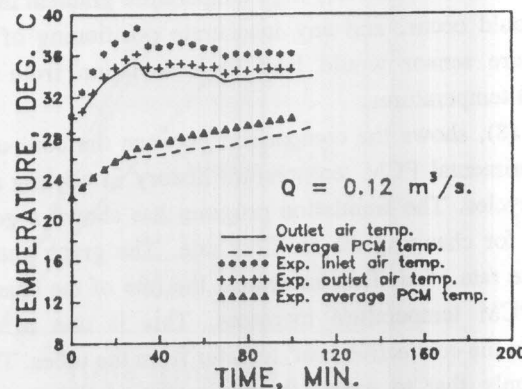


Figure 6. Comparison between the predicted and experimental temperature at air flow of  $0.12 \text{ m}^3/\text{s}$ . (heating cycle).

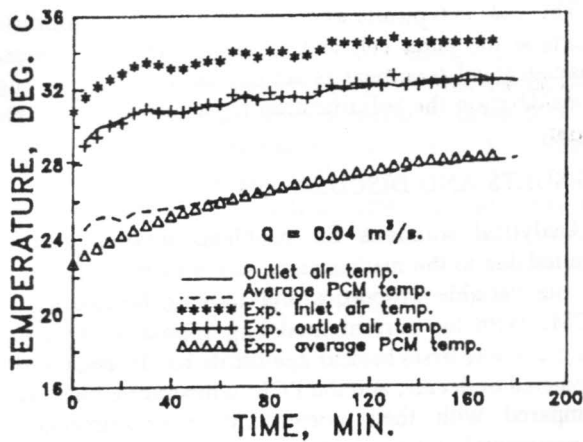


Figure 7. Comparison between the predicted and experimental temperature at air flow rate  $0.04 \text{ m}^3/\text{s}$ . (heating cycle).

The results have shown excellent agreements between the predicted and the experimental outlet air temperature history in either cooling or heating case. Differences did not exceed  $0.5 \text{ }^\circ\text{C}$  under any of the three applied flow rates. Comparison for the PCM were not as excellent as in the outlet air temperature history. However; the differences did not exceed  $2 \text{ }^\circ\text{C}$  over simulated cooling or heating period at the highest applied air flow rate. This difference diminishes as the air flow rate was reduced as shown in Figure (7). This may be attributed to the uncertainty of the thermocouple location inside the PCM. The uncertainty of position could have occurred during filling the PCM in the tubes. In addition, during volume change, the thermocouple junction might have moved from the tube center. This situation becomes more important when the air flow rate is high, a condition where a relatively high temperature gradient in the PCM would occur, and any inaccurate positioning of the temperature sensor would lead to a deviation from the predicted temperatures.

Figure (8), shows the comparison between the computed and experimental PCM temperature history in cooling and heating cycles. The simulation program has shown a good response for changing the air flow rate. The graph shows that as the rate of air flow increases, the rate of the change of the PCM temperature increases. This is due to the increase in the convective heat transfer from the tubes. This would imply that to extent the utilization of the storage energy in the PCM, the air flow rate has to be reduced. This also will provide energy at almost constant temperature.

The computer program was utilized to obtain some important information such as the transient temperature profile across the PCM in a single tube, and hence the prediction of moving front for freezing or melting can be determined. The variation of the transient temperature profiles for cooling or melting the PCM are given in Figure (9-a) and (9-b), when plotted against the dimensionless radius ratio  $r/r_0$ . The profiles are given for every 20 minutes intervals. The successive intersection of the profiles with the phase change temperature line gives the solidification or melting front movement toward the PCM center. Also the graph shows higher average slopes of the temperature profile for the heating cycle (in the liquid zone) than that of the cooling cycle (in the solid zone). This could be attributed to higher thermal conductivity values for the solid phase as given in Table 1. The case given in that figure is for the central tube in the heat exchanger (3rd row).

Also the variation of mass fraction of the liquid phase to the total PCM mass with time in the heat exchanger can be predicted as well. This variation is given to characterize the performance of the heat exchanger with the PCM when being considered by row. Figures. (10-a) and (10-b) show this response for both cooling and heating cases at air flow rate of  $0.12 \text{ m}^3/\text{s}$ . The graph showed a significant lag time for complete change of the PCM from liquid to solid or visa-versa with respect to the first row when compared with the rest of rows. This is mainly the effect of approach velocity on that row, which is lower than the maximum or average air velocity across the rest of the tube bank. This signifies the lower heat transfer rate to or from that row compared to the other rows. Differences in mass fraction ( $x$ ) between row 1 and 2 reached 80 minutes to reach a complete phase change. The differences of ( $x$ ) between row 2 to 5 were not significant, and did not exceed 10 minutes to reach complete phase change for either cooling or heating case.

This implies that when the last row reaches a complete phase change, about 40 - 50 % of the PCM mass in the first row yet has not been completely changed. For the current heat exchanger, this represents at least 10 % of the total stored energy that can still be utilized for an extended period of time. On the other hand the  $x$ - fraction of the PCM can be taken as a measure of energy stored in the system (cooling cycle) or energy consumed (heating cycle). Therefore graphs 10-a and 10-b can be used to predict how much energy is stored or consumed within the system at any time for the given air flow rate.

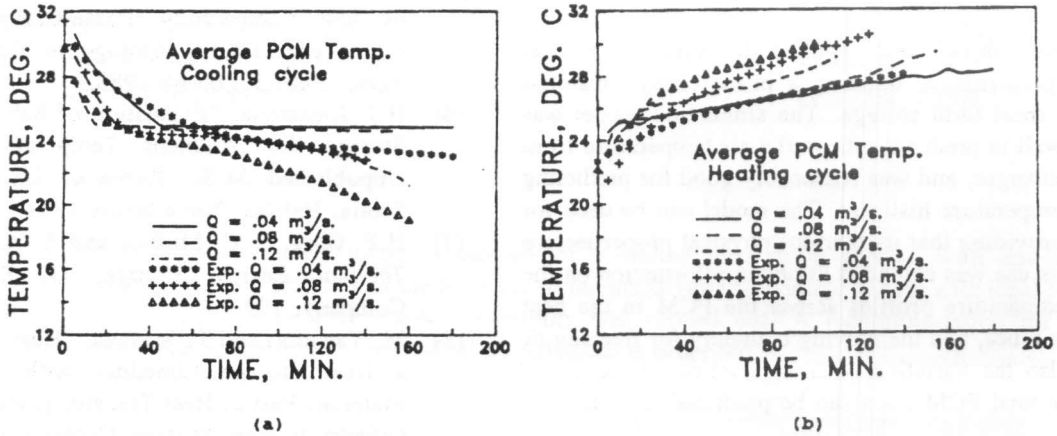


Figure 8. Comparison between the computed and experimental temperature histories at different air flow.

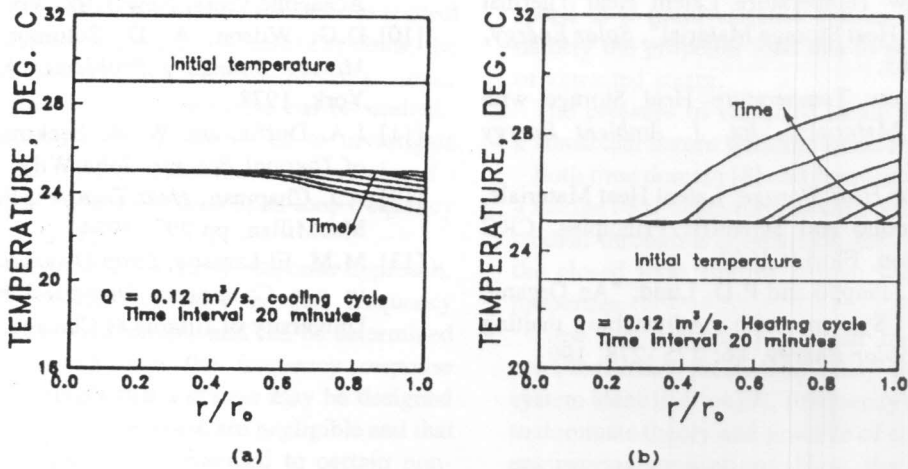


Figure 9. Variation of the transient PCM temperature profiles with the dimensionless ratio ( $r/r_0$ ).

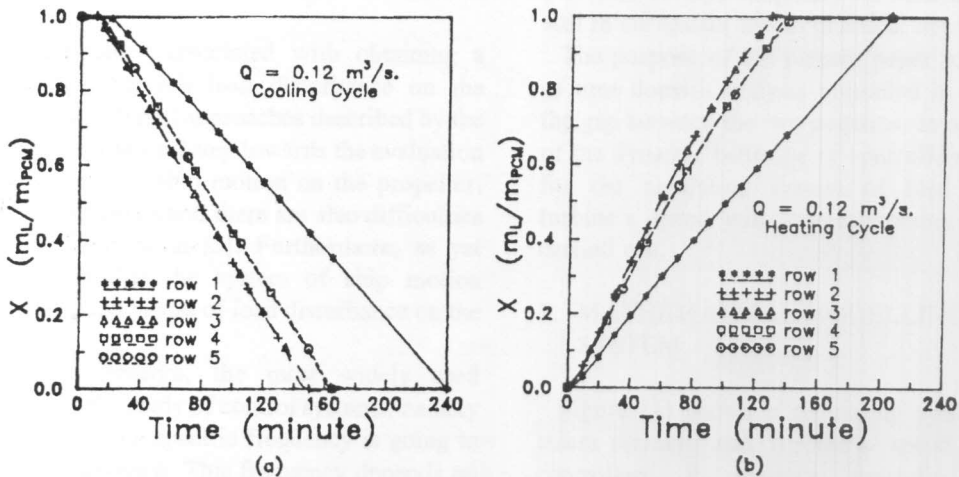


Figure 10. Variation of the liquid mass fraction to the total PCM mass ( $x$ ), with time.

## CONCLUSION

A proposed model was applied to simulate a heat exchanger performance utilizing a phase change material (PCM) for short term storage. The simulation model was extremely well in predicting the outlet air temperature from the heat exchanger, and was reasonably good for predicting the PCM temperature histories. The model can be used for any PCM providing that its thermo- physical properties are available. Its use was extended to obtain information on the transient temperature profiles across the PCM in the heat exchanger's tubes, and the moving boundary for freezing or melting. Also the variation of mass fraction of the liquid phase to the total PCM mass can be predicted as well.

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