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HEAT TRANSFER STUDY IN THE LOW TEMPERATURE THERMAL ENERGY STORAGE MATERIAL (MULTI-COMPONENT MATERIAL) BETWEEN TUBES ARRAY DURING MELTING

Suditama Sulaiman[1]
Megat Mohamad Hamdan Megat Ahmad[2]
Shamsuddin Sulaiman[2]
Normariah Adam[2]

[1]Department Chemical and Mechanical
Terengganu Advanced Technical Institute (TATI)
Kemaman 24000, Terengganu D.I., Malaysia

[2]Department Mechanical and Manufacturing
Faculty Engineering
University Putra Malaysia
Serdang 43400, Selangor D.E., Malaysia

ABSTRACT

This paper is to present a simple and efficient numerical technique for solving transient multidimensional heat transfer problem of melting processes and comparison to an experiment result. The proposed technique comprises an enthalpy-based method for solving the problems by a finite difference scheme, lump system behavior being assumed for each node. The multi-dimensional transient heat transfer characteristics in term of temperature distribution history, the temporal storage heat and moving rate of solid-liquid interface in a multi-component phases change material (PCM) heated by the tube array under inline and staggered arrangement that have been studied numerically. Finite difference scheme for both time and space variable has been applied to solve the governing differential equation for obtained algebraic equation in explicit forms. The results are presented and discussed in this paper.

***Keywords**: Phase Change Material (PCM), thermal storage, transient, arrangement tubes.*

1.0 INTRODUCTION

The thermal energy storage has long been considered as one among the potential method for the solution of many problem encountered in the field of energy and utilization [1,2]. The melting problem is non linear in the mathematical sense due to the existence of a moving boundary (interface) between the two phases associated with the release of latent heat. Neither the position nor the velocity of the interface can be predicted in advance. Mathematical analysis is more complicated as the physical properties of the phase change material (PCM) are

temperature dependent and when the boundary conditions are the same as the type that applied to convection.

Phase change problems have a limited number of analytical solutions. Most of those available from the literature applies to simplified and idealized systems and is one-dimensional. The analytical solution have been reviewed by Lunardini [3]. Numerical methods appear to be a more practical approach for solving phase change problems and the moving interface can be traced. Reviews of numerical solution have been presented by Viscanta [2]. Among the possibilities of storing the thermal energy, the thermal energy storage by the solid-liquid PCM has already been considered as the most favorable compared to other methods, particularly for the reasons of high specific heat storage capacity, better volume stability and offering the possibility of passive temperature control. In general, the main problem faced in the PCM application as thermal energy storage is the fact that the heat transfer properties of the PCM are poor which creates the problem in the thermal charging and discharging. Increasing the specific heat transfer contact surface of the storage is one among several possible solutions to the problem. Basically, this numerical study is an attempt to contribute to the effort of improving the transfer characteristics of a low temperature thermal energy storage using paraffin as PCM.

The objective of this paper is to present an efficient numerical and experimental methods for solving multidimensional phase change, for system with diverse boundary condition.

2.0 MATHEMATICAL FORMULATION

The problem under consideration deals with the heat transfer mode by conduction in multi-component PCM, initially at a uniform ambient temperature and heated by a tubes bundle 9 in-line and staggered arrangement. Therefore, the rate of the energy transfer by conduction into a unit volume plus the volumetric rate of the thermal energy generation must equal the rate of change of thermal energy storage within the volume. The heat balance equations that describe the transfer of heat during phase change can be separate into there equations describing each of there regions, the solid, liquid, and the solid/liquid interface.

In the solid phase:

$$\frac{\partial}{\partial x}\left(k_s \frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k_s \frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k_s \frac{\partial T}{\partial z}\right)+\dot{q}=\rho_s C_P \frac{\partial T}{\partial t} \qquad T\langle T_m \qquad (1)$$

In the liquid phase:

$$\frac{\partial}{\partial x}\left(k_l \frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k_l \frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k_l \frac{\partial T}{\partial z}\right)+\dot{q}=\rho_l C_P \frac{\partial T}{\partial t} \quad T\rangle T_m \qquad (2)$$

At the solid and liquid interface:

$$k_l \frac{\partial T_l}{\partial \bar{n}} - k_s \frac{\partial T_S}{\partial \bar{n}} + \dot{q} = \rho_m L \frac{\partial h}{\partial t} \qquad T = T_m \tag{3}$$

Where $\bar{n}$ is a direction normal to the interface, L latent heat and h is the interface location. The heat source $\dot{q}$ can describe radiation into the PCM, or any other heat source/sink, such as an electric current flowing through the PCM. Let us assume that the density and the specific heat are:

$$\rho_m C_{p_m} = \frac{1}{2}(\rho_s C_{p_s} + \rho_l C_{p_l}) + \rho_m \frac{L}{2\Delta T_m} \tag{4}$$

Where C_{ps}, ρ_s, C_{pl}, and ρ_l are constant value in the temperature interval $T_m \pm \Delta T_m$, with the new properties. The storage energy in a unit volume of the PCM may be expressed as:

$$E = \int_{T_0}^{T_m} \rho_s C_{p_s} dT + \rho_m L + \int_{T_m}^{T_l} \rho_l C_{p_l} dT) \tag{5}$$

The thermal conductivity coefficient can be estimated as an arithmetic average

$$k_m = \frac{1}{2}(k_s + k_l) \qquad T \in T_m \pm \Delta T_m \tag{6}$$

3.0 NUMERICAL SOLUTION

A finite difference method was applied for the numerical computation. For this purpose, as shown in Figure 1 the space of the PCM was seek into volume elements and that a two-dimensional heat transfer (xy-plane). Thermal conductivity, density, specific heat of the PCM as well as thermo-physical properties of the heat transfer fluid (HTF) are dependent of temperature. The following typical formulated was applied for that purpose,

$$T^{n+1}{}_{(i,j)} = T^n{}_{(i,j)} + \frac{q^n{}_{(i,j)} \Delta t}{C^n{}_{(i,j)}} \tag{7}$$

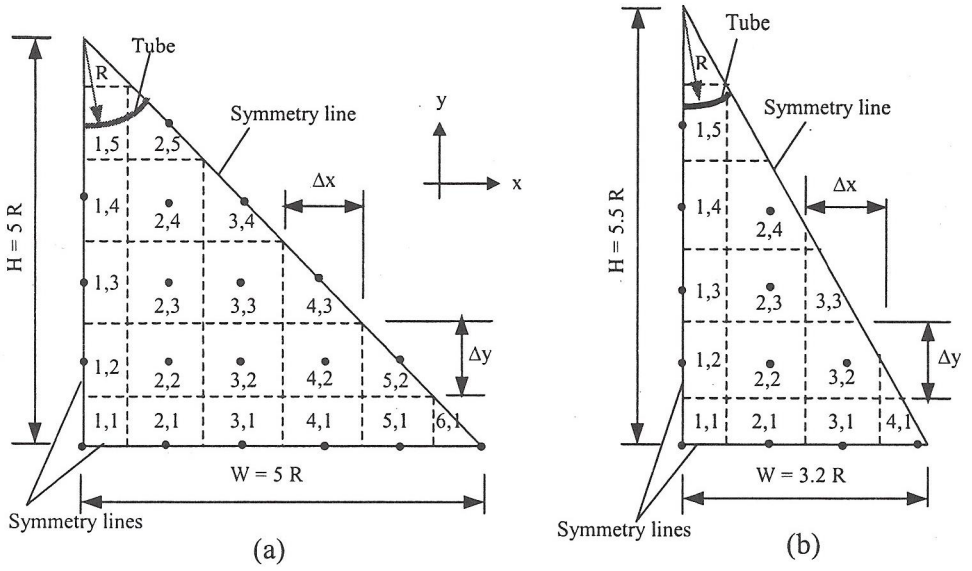


Figure 1: The finite different grids of the symmetry region PCM for (a) inline arrangement and, (b) staggered arrangement

The thermal resistance $R_{i,j}$ to heat transfer by conduction from the an element to its neighbors, in the x direction in a Cartesian system is:

$$R_{i\pm1,j} = \frac{0.5\,\Delta x_i}{\Delta y_i\,\Delta z_i\,k_i} + \frac{0.5\,\Delta x_{j\pm1}}{\Delta y_{j\pm1}\,\Delta z_{j\pm1}\,k_{j\pm1}} \tag{8}$$

The heat rate stored for each node was also calculated using the equation:

$$q_{(i,j)} = \frac{T_{(i+1,j)} - T_{(i,j)}}{R_{(i+1,j)}} + \frac{T_{(i-1,j)} - T_{(i,j)}}{R_{(i-1,j)}} + \frac{T_{(i,j+1)} - T_{(i,j)}}{R_{(i,j+1)}} + \frac{T_{(i,j-1)} - T_{(i,j)}}{R_{(i,j-1)}} \tag{9}$$

As presented by Carnahan *et al.* [5], the numerical scheme has the following stability criterion:

$$\frac{\alpha\,\Delta t}{\Delta x^2} \le \frac{1}{4} \tag{10}$$

The flow chart of computational procedure is:

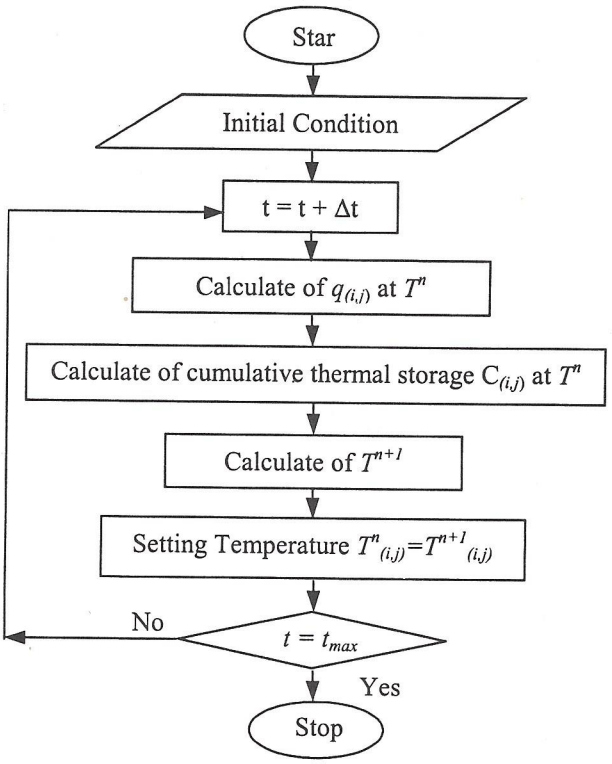


Figure 2: Computational flow chart

4.0 EXPERIMENTAL APPARATUS

An experimental model of paraffin as PCM for a latent thermal energy storage system was design and built for investigation of the concept and the mathematical model in the melting (charging) process. A Schematic representation of the latent thermal energy storage using *n*-paraffin as PCM unit is provided in Figure 3. The unit consists of 9-aluminum (staggered and inline arrangement) with heat transfer fluid (HTF) flowing inside, immersed in *n*-paraffin (multi component material) as a PCM. Hot air is used as HTF in the experiment as on Figure 5. Inside diameter of the tube is 0.017 m. The container size is (0.29x0.29x0.43) m. The temperature of PCM, tubes surface and HTF were measured by copper-constantan (t type) thermocouples of 0.2 mm in diameter. Twelve thermocouples were used embedded at radial distance around the tube selected (on middle tube) to measured the temperature profile of the PCM as can be seen on Figure 4. Two thermocouples in each were placed for measuring the inlet and outlet temperature of HTF and tube surfaces.

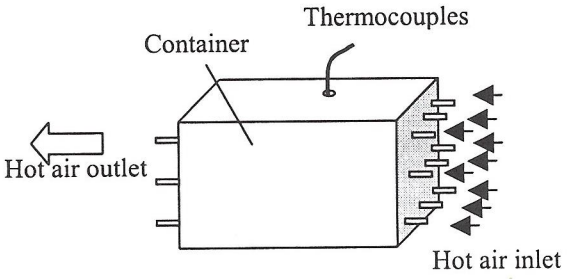


Figure 3: Structure of test section

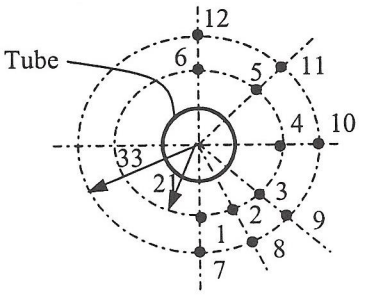
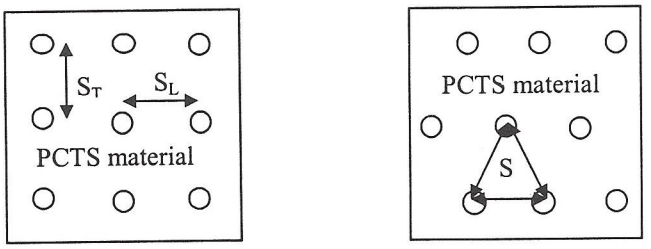


Figure 4: Thermocouples position at PCM



(a) (b)

Figure 5: (a) Inline $S_T = S_L = 5D$, D = 19 mm and
(b) Staggered S = 5.5 D, D = 19 mm

5.0 RESULTS AND DISCUSSION

5.1 The Temperature Profiles of PCM

Two types of tubes arrangement i.e. inline and staggered configuration were investigated. To account the energy input (charged) during melting of the PCM, the tube wall temperature was considered as a possible time dependent boundary condition. The temperature profiles of PCM at thermocouples no. 1, 6 and some selected nodes for both types are shown on Figures 6 and 7. The time dependent of wall temperature for each respective computational set and the experimental results are plotted together. The observed discrepancies between computational and experimental data which are in general less than 3°C is within the acceptable tolerance and these might be due to numerical truncation error, the accuracies of measurements and applied assumptions.

The liquid region around the cylinder indicates that heat transfer from the cylinder is primarily by conduction. Natural convection develops and intensifies, influencing the melt shape in general and the melt region above the cylinder in particular. At higher imposed wall heat fluxes and longer time, the flow pattern in the melt is sufficiently intense to affect the lower half of the melt region too. Maximum affect is at the top of the cylinder ($\theta = 180^{\circ}$) for the inline and staggered tubes heater arrangement.

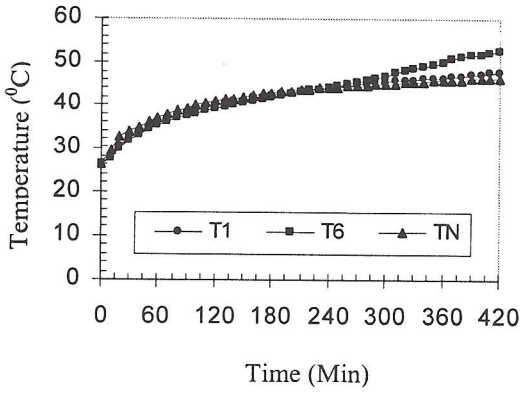


Figure 6: The temperature profiles of PCM for inline tubes arrangement at thermocouples no. 1 and 6, and numerical result (T_N) of node no. [1,4]

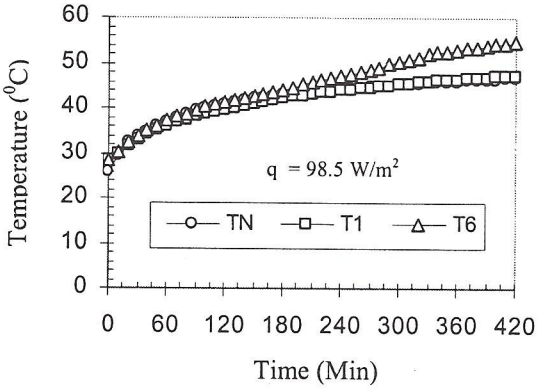


Figure 7: The temperature profiles of PCM for tubes staggered arrangement at thermocouple no. 1 and 6, numerical result (T_N) of node no. [1,4]

5.2 Heat Transfer at The Solidliquid Interface

Based on experiment with constant heat flux boundary condition at the cylinder surface, heat transfer coefficients at the solid-liquid interface were determined during melting. This was accomplished by analyzing interface motion during the phase change process. Energy balance at the interface can be expressed as:

$$\rho \Delta h_f \frac{dr}{dt} = h(T_w - T_f) + k_S \frac{\partial T_s}{\partial r} \tag{10}$$

The heat transfer coefficient at the melt boundary h can be determined from the knowledge of the interface velocity (dr/dt) and the knowledge of the cylinder surface temperature T_w if heat conduction into the solid is small and can be neglected. This would occur when the temperature of the solid is close to its fusion temperature, i.e. negligible sub-cooling. In each experiment the interface position was measured as the melting progressed and the data was analyzed to determine

dr/dt as function of angular position θ and a time. Since the remaining parameters in equation (10) are known or measured quantities, the local heat transfer coefficient was determined from the equation:experiment the interface position was measured as the melting progressed and the data was analyzed to determine *dr/dt* as a function of angular position θ and a time. Since the remaining parameters in equation (10) are known or measured quantities, the local heat transfer coefficient was determined from the equation:

$$h_{(\theta,t)} = \frac{\rho \Delta h_f}{(T_w - T_f)} \left(\frac{dr}{dt} \right)_{\theta,t} \tag{11}$$

The dependence of the local heat transfer coefficient with the angular position and the boundary condition at the cylinder surface can be approximate by empirical equation:

$$h_{(\theta)} = \frac{a}{Ste_l} + b\, exp[-c(\pi - \theta)^2 (Ste)^{1/2}] \tag{12}$$

Where a = 0.35 W/m^2.K and c = 0.078 are empirical constants and the constant b depend on the boundary condition at the cylinder surface, b = 40 W/m^2.K for inline and staggered tubes arrangement. The Stefan number of liquid phase (Ste_l) in equation (12) is independent of the boundary condition the cylinder surface. The local heat transfer coefficient at the solid-liquid interface determined for the constant heat flux, boundary condition are little dependent on Ste_l and are almost dependent of angular position θ and time.

5.3 Correlation of Natural Convection Heat Transfer at The Solid-liquid Interface

Natural convection heat transfer from a horizontal cylinder and in horizontal annuli has been studied and correlations have been proposed. Almost all the correlations use the gap width δ as characteristic length in the Grashof and Nusselt Numbers. Itoh *et al.* [6] have proposed $\sqrt{r_1 - r_2\, ln(\, r_2 \,/\, r_1\,)}$ as a characteristic length for the Grashof number, where the subscripts 1 and 2 refer to inner and outer radii of the annulus, respectively.

Analysis of the data obtained showed that the experimental results use by based of the Nusselt and Grashof number on the characteristic radius R_C defined as $R_C = (r_m R_0)^{1/2}\, ln(r_m / R_0)$, r_m is the mean interface radius based on perimeter ($P = 2\,\pi\, r_m$) of the solid-liquid interface which defines the outer boundary of the liquid region. By choosing an abscissa which takes the Stefan number at the solid-liquid interface into account, determine of the local Nusselt number at the interface for the prescribed boundary condition at the cylinder surface are correlated by straight lines. The symbol indicates experimental data while the solid lines represent empirical equations of the form:

$$Nu_{Rc} = C(Ra / Ste)^{1/3} \tag{13}$$

where the constant C depends on the boundary condition. The average of Nusselt numbers at the solid-liquid interface is given in Figure 8 for experiment and Figure 9 for numerical result. For both the inline and staggered tubes arrangement boundary conditions the experimental results can be approximated by empirical equation:

$$\overline{Nu}_{Rc} = K(Ra / Ste)^{1/3} \tag{14}$$

where K = 0.01305 for inline and K = 0.01084 for staggered tubes heater arrangement of heat flux constant boundary conditions. The correlating equations with numerical results for mean Nusselt numbers at the solid-liquid interface is given in Figure 8. For both the inline and staggered tubes arrangement boundary conditions the numerical results can be approximated by empirical equation (4.6) is:

$$\overline{Nu}_{Rc} = 0.00968(Ra / Ste)^{1/3} \tag{15}$$

For staggered and inline tubes heater arrangement, the melt shape is symmetrically around the tube.

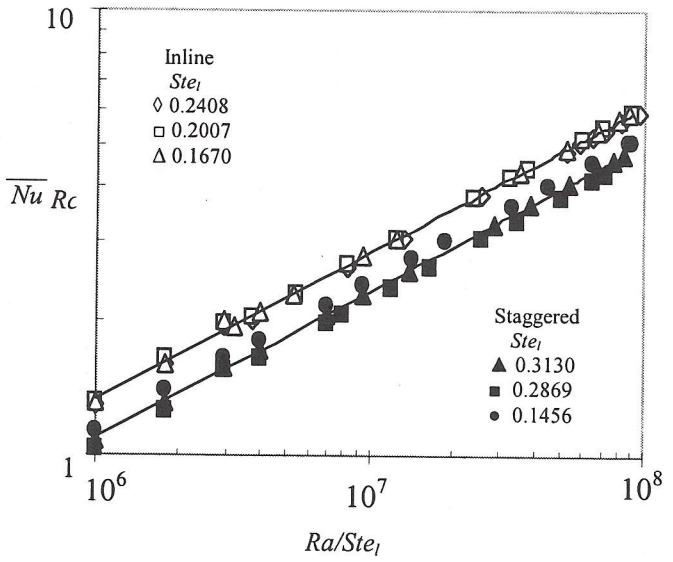


Figure 8: Correlation of average Nusselt number at the solid-liquid interface, constant surface heat flux, the experiment result

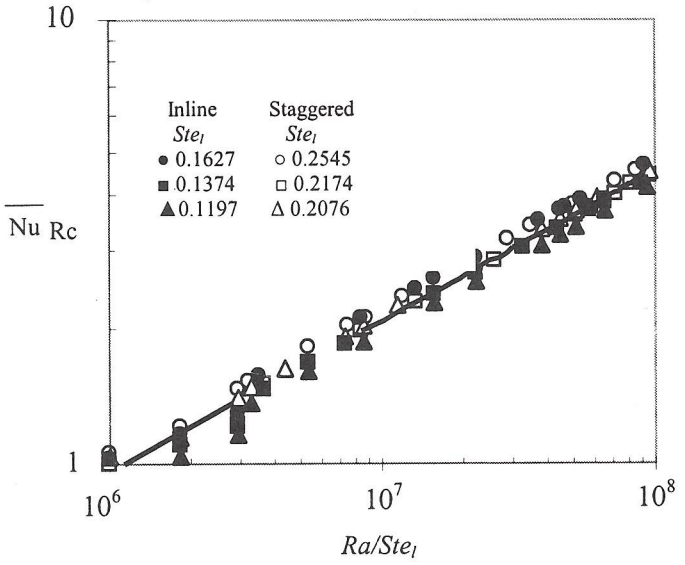


Figure 9: Correlation of average Nusselt number at the solid-liquid interface, constant surface heat flux, the numerical result

5.4 Thermal Energy Storage

The heat exchanger unit used for the simplified model of the thermal storage material is considered to be infinite in the direction perpendicular to the flow. This experiment deals with the measurement of the transient response of simplified model latent thermal storage. The process is transient, the temperature of the HTF entering and out are a periodic function of time (t) with a flow of liquid at a constant mass flow rate. The cumulative heat storage volumetric capacity are given in Figure 10 for experiment and Figure 11 for numerical and the experimental results can be approximated by empirical equation

$$E = A\, q\, t \tag{15}$$

where A = 413.43 for inline and A = 413.63 for staggered. For the numerical results A = 404.03 for inline and A = 404.33 for staggered tubes heater arrangements.

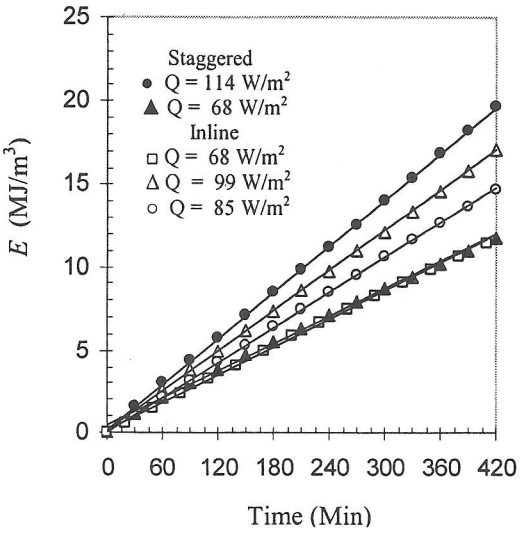


Figure 10: Cumulative heat storage volumetric capacity for the experiment result

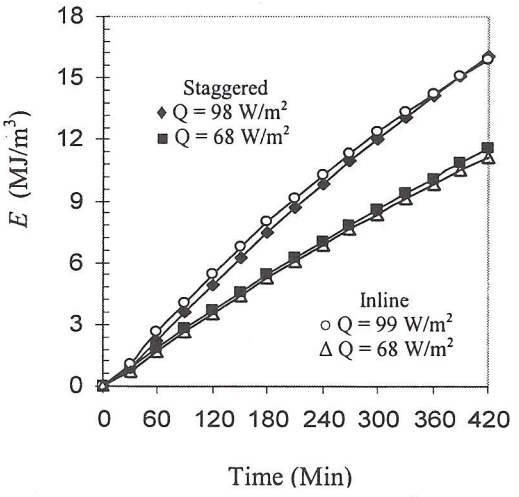


Figure 11: Cumulative heat storage volumetric capacity for the numerical result

6.0 CONCLUSION

Numerical scheme is suitable of solved phase change problem with boundary condition of constant heat flux and convection. The results scheme is simple and straightforward and requires neither iteration nor tracing of the interface location.

Melting from the horizontal tubes (cylinder) arrangement (inline and staggered) immersed in *n*-paraffin (multi component material) produce an unsymmetrical zone around the heat source (tubes). The unsymmetrical zone is due to the

development of natural convection flow in the *n*-paraffin (multi component material). The local heat transfer coefficients at the solid-liquid interface were determined from interface position.

The experiment results show that natural convection effects during melting are of first order importance and should be considered in any analysis of system involving phase change.

NOMENCLATURE

C_p	specific heat
D	tube diameter
E	energy per unit volume
F	radiation Shape factor
Gr	Grashof number
h	convection heat transfer coefficient
k	thermal conductivity
L	latent heat
Nu	Nusselt number
$\overline{n}$	normal
Pr	Prandl number
q	heat flux
R	tube radius
Ra	Rayleigh number
Rc	characteristic radius
r_o	radial distance
r_m	mean radius
q	heat source per unit volume
Ste	Stefan number
t	time
T	temperature
v	volume
x,y	Cartesian Coordinate

Greek Symbol

α	thermal diffusivity
Δ	interval
ρ	density

Subscripts

i	calculated element
j	neighbor element to the i element
l	liquid
m	melting
s	solid
w	wall
o	initial

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