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HEAT EXCHANGER PERFORMANCE FOR LATENT HEAT THERMAL ENERGY STORAGE SYSTEM

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Abstract

An experimental and theoretical study was carried-out to investigate the heat transfer in a component of a low temperature thermal energy storage (TES) system using latent heat of fusion of a phase change material (PCM). The heat storage container is a double pipe heat exchanger where the energy source fluid (hot water) flows through the inner tube, while the heat sink (storage PCM) fills the annulus gap. The tested phase change material is the paraffin wax of melting temperature of 50^{o}C. Measurements were made of the temperature change of hot water as well as the paraffin wax. The heat transfer rate and the accumulative energy stored as a function of time is presented for different hot water flow rates and inlet temperature. An analytical model is developed for the prediction of the storage heat transfer rate and accumulative energy. The predictions of the model are compared with the experimental results and show good agreement. The developed model is also used to study the effect of convection in the molten zone, initial PCM subcooling, and heat exchanger dimensions on heat transfer rate and accumulative energy.

Nomenclature

A	Surface area	m^2
C	Specific heat	$kJ/kg.^oc$
d	Diameter	m
E	Accumulative energy stored	kJ
h	Coefficient of heat transfer	$W/m^2.^oc$
h_{sL}	Latent heat Of fusion	kJ/kg
k	PCM Thermal conductivity	$W/m.^oc$
L	Length	m
m	Mass Flow rate	kg/s
Nu	Nusselt number, Nu=hd/k	
P	Surface perimeter	m
Pr	Prandtl number, $Pr= \nu / \alpha$	
Q	Heat transfer rate	W
R	Radius	m
Ra	Rayleigh number , $Ra =(\beta g \Delta T d^3)/\nu\alpha$	
t	time	s
T	Temperature	^oc
U	Overall heat transfer coefficient	$W/m^2.^oc$
V	Volume	m^3
x	Coordinate	

Greek Letters

α	Thermal diffusivity	m^2/s
β	Coefficient of Thermal expansion	$1/^oc$
ρ	density	kg/m^3

Subscripts

ave	Average
e	Equivalent
ii	Inner tube inner radius
io	Inner tube outer radius
int	Interface boundary
in	Inlet
out	Outlet
oi	Outer tube inner radius
P	Paraffin
sL	Soild-liquid

1. Introduction

Storage of thermal energy is very important in many engineering applications. For example, among the practical problems involved in solar energy systems is the need for an effective means by which the excess heat collected during periods of bright sunshine can be stored, preserved, and later be released for utilization during night or other periods. Similar problems arises for waste heat recovery systems. Traditionally such available heat has been stored in the form of sensible heat (typically by raising the temperature of rocks, water, ...etc), Carter [1]. Attempts have been mode to make container small in size and large in capacity, particularly at low storage temperature, by using latent heat of fusion of phase change materials (PCM) such as paraffin wax and salt hydrates. Simulation studies have been carried by Morrison and Abdel-Khalik [2] for TES systems using both PCM and sensible heat. The authors have revealed that both systems were comparable with regard to overall system performance. Other previous studies were concerned with the choice of (or the development of new) storage materials and systems in order to improve the TES system performance [3 to 5]. Thermal and economical comparisons between different PCM were given by Hedman [3]. Farid [4] used TES unit contains two sections each filled with a PCM having different melting temperatures resulting in an improvement of the overall heat transfer rate. The introduction of additional "heat path" provided by the conducting container walls was shown by Smith et.al[5] to reduce the PCM heat transfer resistance. Heat exchanger design aspects and performance have also received attension. For example, a one dimensional model was developed by Shamsundar and Sparrow [6] and Bailey et.al. [7] to estimate heat exchanger performance in a paraffin was storage system.

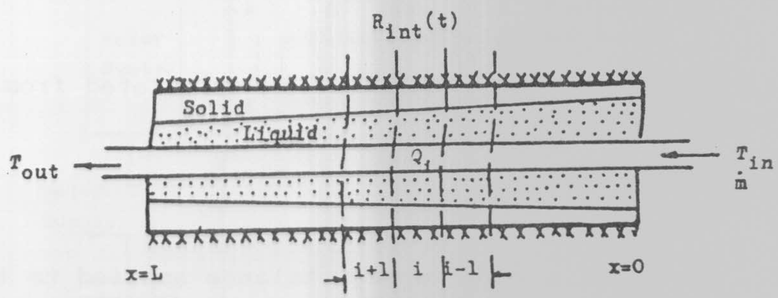
The aim of the present investigation is to experimentally study the heat exchanger transient behaviour as a TES system and to predict the system performance analytically. The selected PCM is the paraffin wax. The thermophysical properties of paraffin wax are given in table (1) as compared to water.

Table (1) Thermophysical Properties Of Paraffin Wax

Materials / Properties	Paraffin Wax	Water
Melting Temperature (^{o}C)	50	
Latent heat of fusion h_{sL} (kJ/kg)	190	
Density ρ_s ρ_L (kg/m^3)	930,830	997
Thermal conductivity k_s,k_L ($W/m.^{o}C$)	0.21,0.21	0.65
Specific heat C_s, C_L ($kJ/kg.^{o}c$)	2.1,2.1	4.18

2. Analysis

A schematic drawing of the storage heat exchanger container is shown in Figure (1). A fluid (hot water) of temperature T_{in} enters the



Figure(1) Heat Exchanger TES Schematic

heat exchanger at a flow rate $\dot{m}$ and is cooled to an outlet temperature $T_{out}(t)$ by heating and then melting the PCM, which is intially at temperature T_i. The heat transferred rate Q(t) to the heat exchanger is calculated as:

$$Q(t) = \dot{m}.\ C_w.\ [T_{in} - T_{out}(t)] \qquad (1)$$

The accumulative heat storage E(t) is obtained from the integral of equation (1) over time. The TES heat exchanger performance is then representable as $\dot{Q}(t)$ and E(t) as a function of time. The heat transfer from the fluid to the PCM may be assumed to be in the direction normal to the flow direction. Also, in this model a quasi-steady state is assumed, therfore the local heat transfer rate and the rate of PCM melting is given by:

$$Q(x,t) = A_{int} \cdot h_{sL} \cdot \frac{\partial R_{int}(x,t)}{\rho\, \partial t} = U_x A_{ii} [T_{ave}(x,t) - T_m] \qquad (2)$$

$$U_x A_{ii} = \frac{1}{h A_{ii}} + \frac{\ln R_{io}/R_{ii}}{2\pi k_s L} + \frac{\ln R_{int}/R_{oi}}{2\pi k_p L} \qquad (3)$$

The coefficient of heat transfer in calculated from the following:

$$Nu = 0.036\ Re^{0.8}\ Pr^{\frac{1}{3}}\ (\frac{d}{L})^{0.055} \qquad (4)$$

Likewise, a quasi-steady energy balance applied to the fluid in the flow channel yields:

$$\dot{m}.C. \frac{\partial T_{ave}(x,t)}{\partial x} = U_x P_{ii} [T_{ave}(x,t) - T_m] \qquad (5)$$

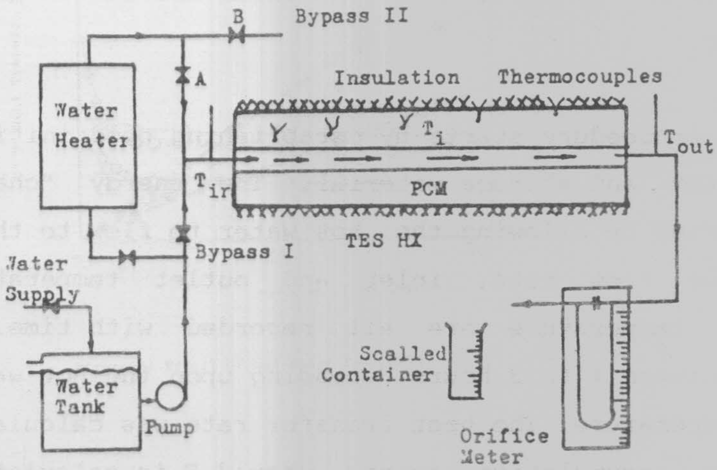
The initial and boundary conditions for R_{int} and T_{ave} are:

$$R_{int}(x,0) = R_{io} \text{ and } T_{ave}(x,o) = T_i$$
$$T_{ave}(0,t) = T_{in} \qquad (6)$$

The computation starts by calculating the coefficent of heat transfer, the local heat transfer rate , the rate of interface velocity, equation (2), and then the fluid temperature gradient, equation (5). The heat transfer rate Q(t), equation (1), is then calculated and integrated to get the accumulative heat storage E(t). The computation steps is then repeated by advancing time with an incremental time step until the specified computation time is over.

3. Experiment

A schematic drawing of the experimental apparatus is shown in Figure (2). The heat transfer fluid (water) was fed to a reservoir which had



Figure(2) Experimental Setup

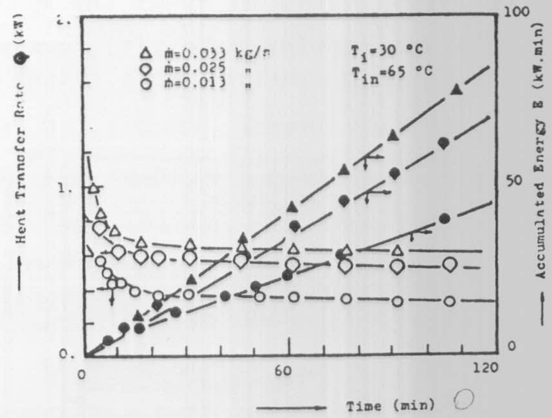
an overflow connection to maintain a constant water level. A pump fed the water from the reservoir to a water heater which was capable of controlling the temperature outlet within $\pm$ 0.5^{o}C. The cold water could by-pass the heater directly to the heat exchanger. The water flow rate was controlled by the control valves A and B permitting the water to flow through the heat exchanger or into a bypass (II). Flow rate of water was measured by scaled container and a stop watch (with orifice-meter as a guide). The TES system is a double pipe heat exchanger were the heat transfer water flows through the exchanger inner tube (copper, of inner radius R_{ii} = 7 mm, and outer radius

R_{io}= 7.7 mm and length L=260 cm). The PCM (paraffin wax) filled the annulus gap of the exchanger (outer radius R_{oi} = 80 mm). The stored energy was preserved by well insulating the heat exchanger outer surface with 3.0 cm thick of glass wool. Water inlet and outlet temperatures were measured by mercury thermometers ($\pm$ 0.2^{o}c accuracy) while the paraffin temperature was measured by five thermocouples located different axial and radial position along the exchanger.

The test proceedure starts by establishing cold initial condition for the system and storage material. The energy "charging" test was then started by allowing the hot water to flow to the heat exchanger. The water flow rate, inlet and outlet temperatures, as well as paraffin temperature are all recorded with time. Changing periods varies between 1 to 3 hours depending upon the hot water flow rate and inlet temperature. The heat transfer rate was calculated from equation (1). The accumulative energy stored E is calculated by integrating equation (1) with respect to time. All tests were carried-out during may-july months (room temperature was about 30 ^{o}C) and therefore the calculated energy losses during the test period were less than 3 %. Similarly, the heat capcity of the heat exchanger walls was much less than the total heat capacity of the storage material and therefore was neglected.

4. Results and Discussion

Figure (3) presents the experimentally measured heat transfer rate Q(t) and accumulative energy E(t) for different values of hot water flow rates. These results indicate that the heat transfer rate Q(t) drops very rapidly within the first 30 minutes (due to the thermal resistance of the low conductivity PCM) after which the reduction in Q(t) is slow and"almost constant " (the PCM thermal resistance is partially compensated by (i) larger interface area and (ii) higher



Figure(3) Effect Of Hot Water Flow Rate

influence of convection in the molten zone which increases the effective thermal conductivity of PCM). Increasing the hot water flow rate from 0.013 kg/s to 0.033 kg/s (90%) increases the accumulated energy stored by about 90 % due to higher temperature difference and coefficient of heat transfer . The effect of increasing the hot water inlet temperature is shown in Figure (4). The results show higher heat transfer rate Q(t) and accumulative energy E(t) as the inlet temperature increases, mainly due to the higher $(T_{ave}-T_m)$ temperature difference. Comparison between the experimental results and the present theoretical model prediction as shown in Figure (5). The effect of convection in the molten zone is represented in the Figure by the equivalent thermal conductivity ratio k_e/k, which is a function of temperature difference $(T_{ave}-T_m)$. The resultant comparison is good for the shown values of k_e/k. The assumed values of k_e/k are however obtained from the experimental results of Rieger et. at. [8] and Kemink and Sparrow [9].

The theoretical model is then used to study the effect of various parameters influencing the heat transfer rate Q(t), accumulated energy

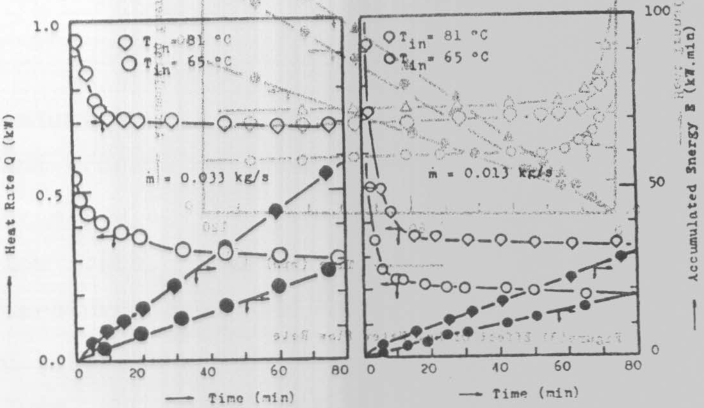


Figure (4) Effect Of Inlet Temperature T_{in}

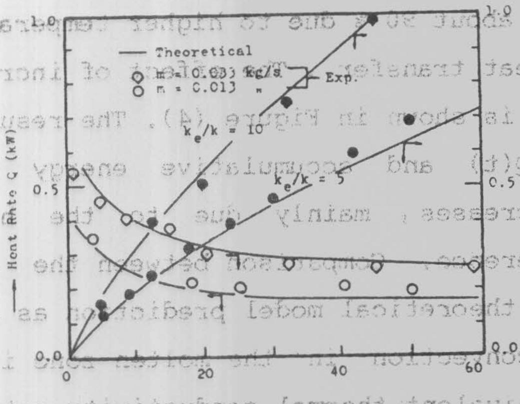


Figure (5) Comparison Between Theoretical Predictions And Experimental Measurements

E(t), and the liquid-solid interface radius growth rate $R_{int}(t)$. The effect of convection is shown in Figure (6). Increasing k_e/k increases both Q(t) and E(t) due to lower thermal resistance between the hot water (heat source) and the interface (heat sink) boundary. Higher heat transfer rate increases the rate of liquid-solid interface radius growth as shown. These results agree with other published investigations, for example Rieger et.al [8] and Farid [4]. The effect of decreasing the initial PCM temperature is shown in Figure (7) to increase both Q(t) and E(t) as expected. The effect of increasing the heat exchanger length L is shown in Figure (8). Increasing the length for the same tube diameter increases the heat transfer area and consequently increases both Q(t) and E(t). However it is expected that there will be a limit of tube length after which it will have no effect. This limit occurs when the average fluid temperature aproaches the PCM melting temperature.

An interesting case was investigated, that is, one heat exchanger of inner tube L,d dimensions and two similar parallel heat exchangers (half fluid flow rate) and of L/2, d dimensions (having the same storing PCM heat capacity). The results however indicated that the values of the stored energy are higher for the long single heat exchanger over the two parallel half length exchangers, Figure (9) . There is still a need for more detailed experimental and theoretical study to see whether it will decrease the TES capacity if (n) heat exchangers of (1/n) length are connected in parallel? or not? This question should also address the two systems pressure drop for overall performance.

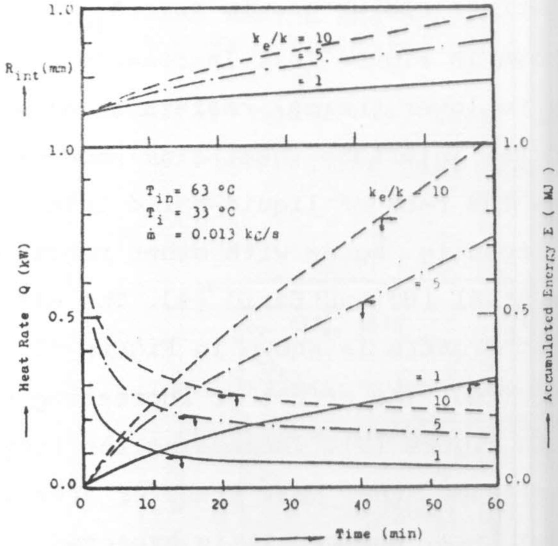


Figure (6) Effect Of Convection (k_e/k)

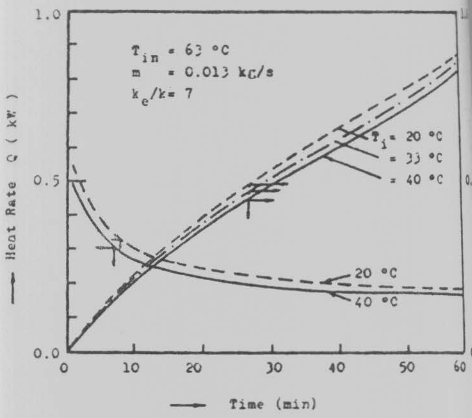


Figure (7) Effect Of Initial Temperature T_i

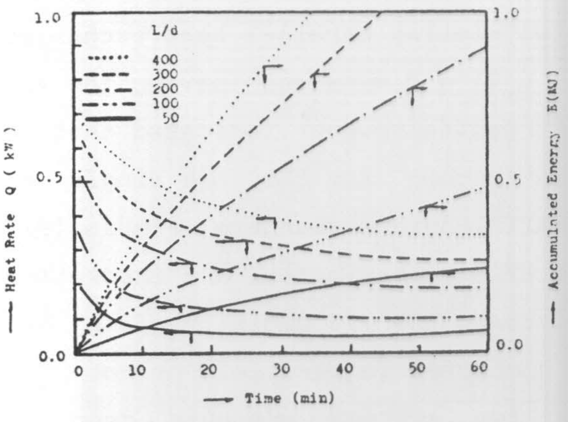


Figure (8) Effect Of HX Dimensions (L/d)

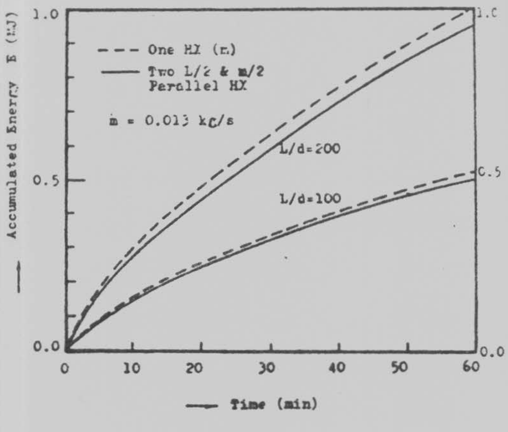


Figure (9) Compareson Between One (full length) HX And Two (half length) Parallel HX