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Validation of a mathematical model for encapsulated phase change material flat slabs for cooling applications

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ABSTRACT

A one-dimensional liquid-based model for a flat slab phase change thermal storage unit was developed. The model allows for varying wall temperatures along the direction of flow and integrates a convective boundary layer using a previously developed algorithm, which is employed to iteratively calculate the liquid fraction and the temperature of each phase change material (PCM) node. The melting and freezing processes are analysed based on the temperatures of the heat transfer fluid nodes, the wall nodes and the PCM nodes during melting and freezing. The mathematical model developed was validated using two sets of experimental data obtained from tests using a PCM having a melting point of −26.7 °C and a liquid glycol based heat transfer fluid. The numerical results show a good agreement with the experimental ones.

1. Introduction

Due to the advantages offered by phase change thermal energy storage such as low temperature variation during charging and discharging cycles, small unit size and low weight per unit storage capacity, PCMs have been applied into numerous applications including solar heating and cooling system [1–6], conventional air conditioning system [7–11], underfloor heating system [12–14] and building envelop [15–19].

Various studies [20–22] have provided detailed illustrations about the PCM's classification, characteristics, measurement techniques and their benefits and limitations. In most phase change thermal storage systems, the PCM needs to be encapsulated into containers in order to hold the liquid phase PCM and also to avoid the PCM contacting with the environment [22]. The three configurations of phase change thermal storage units (PCTSUs) currently in use are flat slabs, shell-and-tube and spherical capsules. The paper analysed a flat PCTSU due to its simple construction and high ratio of surface area to volume. The ratio of surface area to volume of the flat containers used in the experiments is around 110 m^2/m^3.

Morrison and Abdel-Khalik [1] studied the performance of a flat slab thermal storage unit in air-based and liquid-based solar heating systems using a transient simulation program. Zalba et al. [23] tested a storage module with the thickness of PCM slabs of 15 mm and 25 mm. The results show that the solidification and the melting time of a flat slab storage unit are significantly affected by the thickness of the slab, the inlet temperature and the flow rate of the heat transfer fluid (HTF). Lin et al. [13] developed a new structure of underfloor electric heating system with shape-stabilized PCM slabs (thickness of 15 mm) and it has been tested in an experimental house. Ismail et al. [24] experimentally and numerically studied a parallel plate ice bank. The results revealed that the gap between the plates and the wall temperature of the plate have a strong effect on the freezing process. Also, the frosting phenomenon in a flat slab cold storage unit was investigated by Simard and Lacroix [25]. The results indicated that the unit performs best when the thickness and distance separating the PCM slabs are 50 mm and 30 mm respectively.

Modelling of phase change heat transfer in melting and solidification processes has attracted considerable attention of researchers. The difficulty of solving the heat transfer problem during the phase change process is due to the nonlinear movement of the solid–liquid interface, namely moving boundary problems (MBPs) or Stefan problem [21,26]. Furthermore, the position of this boundary is not known a priori and is a part of the solution. Lacroix [26], Verma and Singal [27] and Zalba et al. [21] reviewed various models of PCTSU.

Vakilaltojjar [8] and Vakilaltojjar and Saman [9] integrated a flat slab PCTSU employing two PCMs ($CaCl_2 \cdot 6H_2O$ and $KF \cdot 4H_2O$) in different sections into conventional air conditioning. He analytically modelled the two-dimensional heat transfer problem by using the "Neumann Solution" presented by Carslaw and Jaeger [28]. The

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PCM was initially at the melting temperature. The parametric study revealed that the thermal performance of the PCTSU is better with smaller air gaps and thinner PCM slabs.

Saman et al. [4] employed a two-dimensional numerical model based on enthalpy formulation and analysed the thermal performance of a flat PCTSU. The outlet air temperatures and heat transfer rates predicted by this model and Vakilaltojjar's model were compared with experimental data. The model showed a close agreement with the experimental results.

Halawa [29] and Halawa et al. [30] developed a one-dimensional model to study the heat transfer problem in the PCTSU with the same arrangement as above. The natural convection effect was not included in the model. They proposed a phase change processor (PCP) algorithm to solve the phase change of the PCM in melting and freezing processes and also to determine the liquid faction of the PCM node. The model has been verified with experimental results and air is the HTF. Due to the small thickness and the high length to thickness ratio of the slabs under investigation, the two-dimensional model previously developed is redundant [29]. The one-dimensional model was shown to be adequate to simulate the phase change problem in flat slabs of thickness of 20 mm and below. The one-dimensional model has the advantage of quick computation time.

In this paper, a one-dimensional liquid-based model with varying wall temperatures along the direction of the HTF was developed. Unlike previous models, the heat capacity of the HTF is included in this model. The model integrates a convective boundary layer into a PCP algorithm, which is employed to iteratively calculate the liquid fraction and the temperature of each PCM node. Two melting tests were carried out and the results obtained in the melting tests were used to validate the mathematical model. A PCM with a low melting temperature (−26.7 °C) and a glycol based HTF is utilized in the model validation. The PCTSU used in the experiment has direct applications in refrigerated truck and stationary refrigeration and cooling applications, where the refrigerated space needs to be maintained at temperatures as low as −18 °C.

2. The mathematical model

The PCTSU consists of a number of flat PCM slabs surrounded by an adiabatic rectangular wall. Liquid HTF flows along the passages between the slabs (Fig. 1).

2.1. Modelling assumptions and governing equations

The PCTSU is analysed based on half a thickness of the PCM slab and the HTF passage, because the boundary and the initial conditions at the top and bottom surfaces of the container are identical. The mathematical model developed is based on the following assumptions:

1. thermophysical properties of the PCM are different for the solid and liquid phases but they are constant within one phase;
2. thermal conduction in the PCM in the axial direction is ignored;
3. temperature variation of HTF normal to the flow direction is ignored;
4. heat losses to the surroundings are ignored;
5. supercooling of the PCM does not take place;
6. the PCM container wall is regarded as a single node;
7. the thermal resistance between the inner surface of the container and the liquid PCM is ignored.

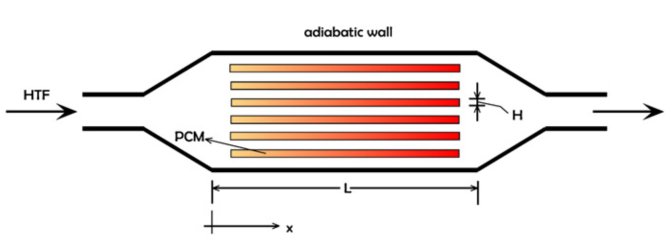

Fig. 1. Schematic diagram of the PCTSU showing the PCM containers and circulating fluid channels.

Based on these assumptions, the energy balance for the HTF can be present as [1]:

$$\frac{\partial T_f}{\partial t} + \frac{\dot{m}}{\rho_f A_f}\frac{\partial T_f}{\partial x} = \frac{UP}{\rho_f A_f c_f}\left(T_w - T_f\right) \tag{1}$$

Where T_f, ρ_f, c_f, A_f and $\dot{m}$ are the temperature, density, specific heat, flow area and mass flow rate of the HTF, T_w is the temperature of the container wall, x is the distance in flow direction, P is the wetted perimeter and U is the overall heat transfer coefficient between the HTF and the container wall. The terms $\partial T_f/\partial t$, $(\dot{m}/\rho_f A_f)\ \partial T_f/\partial x$ and $UP(T_w - T_f)/\rho_f A_f c_f$ present the variation of the temperature of the HTF, thermal capacitance of the HTF and the convection between the HTF and the wall respectively.

$\partial T_f/\partial t$ and $\partial T_f/\partial x$ can be discretized by the backward time upwind scheme:

$$\frac{\partial T_f}{\partial t} = \frac{T_f(i) - T_f^0(i)}{\Delta t} \tag{2}$$

$$\frac{\partial T_f}{\partial x} = \frac{T_f(i) - T_f(i-1)}{\Delta x} \tag{3}$$

Then Eq. (1) becomes:

$$\left(\Delta x \rho_f A_f c_f/\Delta t\right)\left(T_f(i) - T_f^0(i)\right) + \dot{m}c_f\left(T_f(i) - T_f(i-1)\right) = \Delta x UP\left(T_w - T_f\right) \tag{4}$$

On rearranging, Eq. (4) becomes:

$$T_f(i) = \left(aT_f^0(i) + bT_f(i-1) + cT_w(i)\right)\Big/(a+b+c) \tag{5}$$

Where, $a = \Delta x \rho_f A_f c_f/\Delta t$; $b = \dot{m}c_f$ and $c = \Delta x UP \approx UA$.

Heat convection to the PCM container wall by the HTF, q_{conv}, is:

$$q_{conv} = UA\left(T_f(i) - T_w(i)\right) \tag{6}$$

Heat conduction to the PCM node through the wall, q_{cond}, is:

$$q_{cond} = k_w A(T_w(i) - T(i))/dX_w \tag{7}$$

Since the heat transferred by the HTF to the wall is equal to the heat conducted to the PCM, an equation for evaluating $T_w(i)$ can be obtained:

$$T_w(i) = \left(cT_f(i) + dT(i)\right)\Big/(c+d) \tag{8}$$

Where, $d = k_w A/dX_w$.

2.2. Heat transfer coefficient

Generally, the existing heat transfer correlations for the flow between parallel plates are for two cases: *constant and equal wall temperatures* and *constant and equal wall heat fluxes*. The case of *constant and equal wall temperatures* is more appropriate to analyse the heat transfer in this PCTSU, because when the PCM nodes are storing or releasing latent heat of fusion, the wall temperatures are