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Numerical simulation and exergetic performance assessment of charging process in encapsulated ice thermal energy storage system

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ABSTRACT

The solidification process in encapsulated ice thermal energy storage (EITES) system is simulated for water-filled capsules while neglecting storage tank wall effects and heat penetration. Energy and exergy efficiencies were calculated while varying capsule shape, inlet Heat Transfer Fluid (HTF) temperature as well as HTF flow rate. 105 test cases are conducted including seven geometries, five inlet HTF temperatures, and three HTF flow rates. It was found that the energy efficiencies did not accurately reflect system performance, and in all cases, were found to be above 99.96%. However, exergy efficiencies ranged from 78 to 92%, and provided better insight into system losses. The results suggest that an effective way to increase system efficiency is to increase inlet HTF temperature; considerable efficiency gains are possible by setting inlet HTF temperature slightly below solidification temperature. Varying capsule geometry had inconsistent effects on the efficiency, different geometries being optimal in different situations. Surprisingly, viscous dissipation had very little effect on the exergy efficiency and was a source of very little entropy generation. Thus, EITES designers could increase both flow rate and inlet HTF temperature in order to achieve full system charging in an acceptable amount of time.

1. Introduction

Rising global population and rising urbanization, especially in desert and semi desert areas will require an equally increasing energy demand, particularly for HVAC purposes. In 2008, residential and commercial buildings accounted for 40% of the U.S. primary energy use, with space heating and cooling alone accounting for about 36% of this total [1]. As a result of this increasing energy demand, there has been much effort devoted to reducing the cost and amount of energy use in heating and cooling applications, both in residential and commercial buildings. For example, many buildings or communities now use combined heat and power (CHP) which can drastically reduce energy use by utilizing waste heat from power production to cover a thermal load such as space heating. Furthermore, advances in cooling technologies, such as adsorption and absorption chillers, have facilitated the use of combined cooling, heating, and power to even better utilize low grade as well as waste heat for cooling purposes.

In addition to the aforementioned strategies adopted to increase thermal efficiency of heating and cooling processes, thermal energy storage (TES) has been gaining popularity as well, especially in cooling applications. This is due to the ability of TES to help better match power supply with demand, while simultaneously reducing operational cost. This is done through the loading of a cold thermal storage during low demand, i.e., low cost times, generally at night, and unloading the same storage during high demand, i.e., high cost times. Not only does this strategy reduce costs associated with cooling, it also helps reduce energy use during peak times. This is particularly relevant since less efficient means of electricity production, such as gas turbines, are used often during peak hours. Thus, ultimately, TES can help reduce greenhouse gas emissions and fossil fuel use, particularly if the stored energy comes from renewable sources.

For cooling purposes, one of the most common forms of TES is ice storage, due to its high latent heat, non-volatility and low cost. The two most common ways of achieving ice storage are ice-on-coil [2–4] and encapsulated ice [5–7]. Due to the low cost associated with the encapsulated ice method, it has seen the most commercial success to date. In this method, water is contained in capsules located in an insulated storage tank. During low electricity demand times (usually at night), chillers use cheaper electricity to cool a Heat Transfer Fluid (HTF) to below freezing temperatures, which in turn flows through the storage tank, effectively freezing the water inside the capsules. During high demand times (usually in

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mid-day), the latent energy in the ice is used as a heat sink to alleviate electricity usage. This not only reduces costs associated with space cooling but also helps to reduce the peak demand for electricity.

In the open literature, studies involving encapsulated ice TES are limited. Kousksou et al. [8] utilize a numerical model to thermal behavior of a cylindrical tank filled with ice capsules in both vertical and horizontal positions. MacPhee and Dincer [9] use the porous medium concept, similar to [8], to assess the energy and exergy efficiencies of charging such a cylindrical storage tank.

Most other studies use numerical methods to assess charging/discharging times and efficiencies of a uniformly packed bed of spherical capsules. For example, Erek and Dincer [10] consider the variable thermal/fluid interactions around successive rows of capsules to determine the effects of capsule diameter and HTF mass flow rate and temperature on the solidification process of water in a spherical capsule. Ismail et al. [11] use a novel finite difference approach, foregoing fluid calculations and instead utilizing Churchill's Nusselt number approximation [12] to assess the charging process in a spherical capsule. The model was used to investigate many capsule parameters on complete solidification time.

The literature on efficiency of the latent TES process is even scarcer, with only a select few investigations to report. MacPhee and Dincer [13] numerically investigate the charging and discharging processes in a bed of spherical ice capsules, while varying HTF mass flow rate and temperature. MacPhee and Dincer [14] have also investigated that the discharging process of encapsulated ice TES system in which they consider other geometries besides spheres, including rectangular slab and cylindrical capsules. The discharging times were greatly reduced in some capsule configurations, meaning that the spherical capsules may not always be the best choice for capsules.

This paper serves as a progression of the above works [13,14]. While the previous study by the authors [14] investigated the discharging process of the latent ice TES system, this study is concerned with the charging process. There are very few works involving numerical or analytical analyses of encapsulated ice TES system. This study is unique in that it concerned with the numerical simulation of the charging process of a single capsule in a storage tank. By varying the inlet HTF temperature and flow rate, as well as capsule geometry, the effectiveness of either spherical, slab, or cylindrical capsules in latent TES applications is investigated. The results provide insight into what factors most influence efficiency and charging times for storage of ice in a tank of such capsules.

2. Mathematical model

In this study, we wish to address the thermodynamic performance of the charging process in a bed of phase change material (PCM) capsules, and to compare these results in terms of capsule geometry, HTF inlet temperature and flow rate. In order to accomplish this, we must first outline the governing equations and numerical procedures used by Fluent 6.3 to obtain data sufficient for thermodynamic calculations. Then, the equations used to address both first and second law efficiencies are presented. Finally, these procedures are used to gain insight into the effectiveness of the solidification process in a bed of PCM capsules through various test cases.

The following assumptions are made:

- Each PCM has wall thickness of 5 mm and inner volume 268 ml, similar to that used in industry [15].
- Constant thermophysical properties for HTF and capsule material.
- Negligible effects from storage tank wall.
- Negligible potential energy and gravitational effects.
- Negligible changes in density of PCM during solidification.
- The HTF acts as a Newtonian fluid.

Additionally, the thermophysical properties of the materials used in this study are identical to those in [14], and are listed in Table 1. The properties of the PCM (i.e., water) are taken just above and below, respectively, its freezing point at atmospheric pressure. The HTF is assumed to be ethylene glycol (30% by mass) [16], a common liquid used in subzero cooling applications, while the capsule is assumed to be made of polyvinyl chloride (PVC).

2.1. Numerical simulation of heat transfer and fluid flow

The solution strategy for the numerical simulations can be simplified by considering Fig. 1. The left side of Fig. 1 shows the conceptual schematic for the simulations; the subcooled HTF flows over a capsule containing a PCM, allowing it to freeze. The problem is simplified mathematically by observing the right side of Fig. 1, by splitting the domain of interest into Ω_i and boundaries into Γ_i, where $i = \{1,2,3\}$, to be described later. This greatly facilitates the presentation of governing equations.

Concerning the HTF, which occupies the domain Ω_1, the momentum, continuity and energy equations, subject to the aforementioned assumptions, must be solved:

$$\rho\left(\frac{\partial u_i}{\partial t}+u_j\frac{\partial u_i}{\partial x_j}\right) = -\frac{\partial p}{\partial x_i}+\mu\frac{\partial^2 u_i}{\partial x_k \partial x_k} \quad \text{in } \Omega_1 \tag{1}$$

$$\frac{\partial u_i}{\partial x_i} = 0 \quad \text{in } \Omega_1 \tag{2}$$

$$\rho\frac{Dh}{Dt} = \frac{Dp}{Dt}+k\frac{\partial^2 T}{\partial x_i^2}+\Phi \quad \text{in } \Omega_1 \tag{3}$$

where the energy equation (Eq. (3)) uses the substantial derivative and viscous dissipation functions, i.e.:

$$\frac{D}{Dt} = \frac{\partial}{\partial t}+u_j\frac{\partial}{\partial x_j} \tag{4}$$

$$\Phi = \frac{\mu}{2}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)^2 \tag{5}$$

The capsule region, domain Ω_2, contains no flowing particles, thus the only equation to be solved is the energy equation – in this case a simple relationship between enthalpy and temperature:

$$\rho\frac{Dh}{Dt} = k\frac{\partial^2 T}{\partial x_k \partial x_k} \quad \text{in } \Omega_2 \tag{6}$$

Table 1
Material thermophysical properties.

Material	c [kJ/kg·K]	ρ [kg/m^3]	k [W/m·K]	μ [mPa·s]
PCM (liquid)	4.20	1000	0.558	1.52
PCM (solid)	2.11	917.4	2.108	N/A
HTF	3.57	1053	0.422	5.03
Capsule	0.90	1380	0.160	N/A