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Numerical investigation of heat transfer performance of a rotating latent heat thermal energy storage

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HIGHLIGHTS

- A rotating LHTES is evaluated based on heat transfer and parasitic load of rotation.
- Rotating LHTES has higher heat transfer and more uniform temperature distribution.
- Rotational speed influences the heat transfer performance of a rotating LHTES.
- The developed model can be used as performance evaluation and design tools of LHTES.

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ABSTRACT

During charging and discharging process, due to natural convection, latent heat thermal energy storage (LHTES) experiences a non uniform heat transfer process in which higher heat transfer generally occurs at the upper area of LHTES. To overcome this issue, a rotating LHTES is proposed and is expected to have higher and more uniform heat transfer. Hence, this study is conducted to evaluate the potential of heat transfer enhancement of latent heat thermal energy storage by rotation. A computational fluid dynamics (CFD) model for conjugate heat transfer between heat transfer fluid (HTF) and phase change material (PCM) in the latent heat thermal energy storage which experiences charging-discharging process is developed and validated against the experimental measured data. An enthalpy-porosity formulation is adopted to take into account the melting and solidification process of the PCM. For performance evaluation, thermal enhancement ratio (TER) is introduced and defined as the ratio of heat transfer rate enhancement due to rotation to the heat transfer rate of the base case (stationary). This ratio takes into account the parasitic energy needed for rotation. The results reveal that rotation does increase the heat transfer performance of LHTES with up to 25% and 41% enhancement can be achieved during charging and discharging, respectively. In addition, it was found that rotational speed posses significant influence on the performance of rotating LHTES where, for the studied cases, higher rotation speed results in higher heat transfer rate. This study serves as a guideline in designing innovative high performance rotating LHTES.

1. Introduction

With the growing interest to reduce energy-related emission and dependence on fossil fuel, global community has been looking towards renewable energy. Accordingly renewable energy research and development has been intensified over the last decades. Nevertheless, the adoption of renewable energy worldwide is rather low. One key factor that hinders the effective utilization of renewable energy is its intermittent nature. Most of renewable energy such as solar and wind energy can only be collected within the limited period during the day which triggers discrepancy between energy supply and demand. As such efficient heat storage is essential in bridging the mismatch between energy supply and demand. Currently several energy storages are available: sensible heat energy storage, latent heat thermal energy storage and chemical energy storage [1]. Among them, latent heat thermal energy storage has gained considerable attention due to its capability of storing large amount of energy with marginal increase in temperature, hence minimizing heat loss from the storage to the environment [2,3]. This capability is mainly attributed to the phase change material (PCM) used as storage medium which has high latent heat of fusion. Despite its high storage capability, PCM has inherent drawback of slow heat transfer which is mainly due to its low thermal conductivity.

To address this problem, intensive studies have been conducted to

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enhance the heat transfer performance of LHTES and various enhancement methods have been proposed and evaluated. Increasing the heat transfer area is one of the most widely adopted and studied methods since it offers simplicity in design, easiness in fabrication and low cost of construction [4]. Heat transfer enhancement in LHTES due to installation of extended heat transfer surface such as longitudinal fin [5,6], circular fin [7] and perforated partition plate [8] have been reported in literature. Extended surface can be introduced on either fluid side or PCM side or both side. However, in most cases, the extended surface installed on PCM side where heat transfer is relatively low. Other enhancement methods include insertion of metal matrix to PCM [9–12], application of multitubes [13], addition of high conductivity micro/nano particle to PCM [14,15] and utilization of multiple PCM with different melting temperature [16–18]. Insertion of metal matrix into the PCM was found to enhance heat transfer especially during discharging. However, metal matrix may obstruct natural convection during charging hence it may reduce total heat transfer rate during charging. The use of multiple PCM allows for sufficient driving force for heat transfers which otherwise is not possible when single PCM is used. Other than multiple PCM, these enhancement methods have similar basic principle of increasing heat conduction.

During charging and discharging, depending on the state of the material (solid/liquid), two heat transfer processes are generally taking place: conduction and natural convection. Liu and Groulx [19] experimentally found that conduction is the dominant heat transfer mechanism during initial stage of charging. Once the amount of liquid PCM is sufficiently high, the heat transfer is dominated by the natural convection. While conduction is primarily depend on the material properties, natural convection is significantly affected by the gravity. Higher heat transfer is generally observed at the location above pipe containing the heat transfer fluid. This is supported by the numerical finding of Tao and He [6] where higher temperature is observed at the location upper the HTF pipe, indicating higher heat transfer. Yazici et al. evaluated the effect of eccentricity of the working liquid tube and PCM tube on the heat transfer performance of tube-in shell storage unit. It was found that eccentricity to enhance heat transfer in a horizontal LHTES [20] is attributed to the intensified natural convection inside PCM. Noting the important effect of natural convection on heat transfer performance of LHTES, it is therefore of interest to study the potential heat transfer enhancement by continuously change the PCM located at the upper area of LHTES, which can be achieved by slowly rotating the LHTES. By implementing this method, higher heat transfer due to enhanced natural convection is expected. Despite the promising heat transfer enhancement expected by slowly rotating the LHTES, based on our literature review, no study investigating effect of rotation on heat transfer performance of latent heat thermal energy storage has been conducted and reported.

The present study is thus conducted with the objective to evaluate the heat transfer performance of the rotating LHTES during charging and discharging as compared to the performance of the stationary counterpart. The effect of rotational speed are examined and discussed in light of the numerical result. Heat transfer performance evaluation will be conducted by examining the temperature, liquid volume fraction and required power for rotation. The results are expected to elaborate the heat transfer performance of rotating LHTES and provide general design guidelines for rotating LHTES.

2. Mathematical formulation

Here, heat transfer between PCM as storage medium and water as heat transfer fluid and fluid flow are investigated by formulating a three dimensional CFD model. The model is based on the enthalpy-porosity formulation, similar to the validated model presented in our previous study for planar thermal energy storage [16]. To accommodate rotation, a moving reference frame approach is adopted. The schematic representation of the physical model is presented in Fig. 1. The PCM investigated in this study is paraffin wax (n-octadecane) which has melting temperature around 28 °C and the heat transfer fluid (HTF) is water.

2.1. Governing equations

The model takes into account transport phenomena in HTF and PCM as well as heat transfer between them. For rotation case, the whole domain is rotating. Thus, an absolute velocity formulation is chosen in this study. The conservation equations for mass, momentum and energy for the heat transfer fluid (HTF) flow inside the tube can then be expressed as [21]

$$\frac{\partial \rho_{htf}}{\partial t} + \nabla\cdot(\rho_{htf}\mathbf{v}_r) = 0, \tag{1}$$

$$\frac{\partial}{\partial t}(\rho_{htf}\mathbf{v}) + \nabla\cdot(\rho_{htf}\mathbf{v}_r\mathbf{v}) + \rho_{htf}[\omega\times(\mathbf{v}-\mathbf{v}_t)] = -\nabla p\mathbf{I} + \nabla\cdot[\mu_{htf}(\nabla\mathbf{v} + (\nabla\mathbf{v})^{\mathrm{T}})] + \rho_{htf}\mathbf{g}, \tag{2}$$

$$\frac{\partial}{\partial t}(\rho_{htf}c_{p,htf}T) + \nabla\cdot(\rho_{htf}\mathbf{v}_r c_{p,htf}T) = \nabla\cdot(k_{htf}\nabla T). \tag{3}$$

Here, ρ_{htf} is the heat transfer fluid density, p is the pressure, $\mathbf{I}$ is the identity tensor, μ_{htf} is the fluid dynamic viscosity, $\mathbf{g}$ is gravity acceleration, $c_{p,htf}$ is the specific heat of the fluid, k_{htf} is thermal conductivity of the fluid and T is the temperature. The relative velocity, $\mathbf{v}_r$, is given by $\mathbf{v}_r = \mathbf{v}-\mathbf{u}_r$ where $\mathbf{v}$ is the absolute velocity and $\mathbf{u}_r$ is the moving frame velocity which comprises two components, i.e. $\mathbf{u}_r = \mathbf{v}_t + \omega\times\mathbf{r}$. Here, $\mathbf{v}_t$ is the translational frame velocity and $\boldsymbol{\omega}$ is the angular velocity.

Meanwhile, for the phase change materials (PCM), the conservation of mass, momentum and energy can be expressed as [21]

$$\frac{\partial \rho_{pcm}}{\partial t} + \nabla\cdot(\rho_{pcm}\mathbf{v}_r) = 0, \tag{4}$$

$$\frac{\partial}{\partial t}(\rho_{pcm}\mathbf{v}) + \nabla\cdot(\rho_{pcm}\mathbf{v}_r\mathbf{v}) + \rho_{htf}[\omega\times(\mathbf{v}-\mathbf{v}_t)] = -\nabla p\mathbf{I} + \nabla\cdot[\mu_{pcm}(\nabla\mathbf{v} + (\nabla\mathbf{v})^{\mathrm{T}})] + \rho f\mathbf{g} + \mathbf{S}_{mom}, \tag{5}$$

$$\frac{\partial}{\partial t}(\rho_{pcm}H_{pcm}) + \nabla\cdot(\rho_{pcm}\mathbf{v}_r H_{pcm}) = \nabla\cdot(k_{pcm}\nabla T), \tag{6}$$

where ρ_{pcm}, μ_{pcm}, and k_{pcm} are the PCM density, dynamic viscosity and thermal conductivity, respectively. Addition source term $\mathbf{S}_{mom}$ is introduced in the above equations to account for reduced porosity in the mushy zone and H_{pcm} is the enthalpy of PCM, respectively.

For the HTF tube, no mass and momentum transfer occurs and the heat is transferred via conduction.

$$\frac{\partial}{\partial t}(\rho_{tube}c_{p,tube}T) = \nabla\cdot(k_{tube}\nabla T). \tag{7}$$

where ρ_{tube}, $c_{p,tube}$ and k_{tube} are the HTF tube density, specific heat and thermal conductivity, respectively.

2.2. Constitutive relation

The thermal properties of the water depend on its temperature. Accordingly, to take into account the dependency of water properties to temperature, polynomial function and power-law function are adopted. The temperature dependent properties of water were obtained from Kays et al. [22] for temperature of 298–373 K. The water density, viscosity and thermal conductivity are given by [16]

$$\rho_{htf} = C_{\rho,htf1}T^2 - C_{\rho,htf2}T + C_{\rho,htf3} \tag{8}$$

$$\mu_{htf} = C_{\mu,htf1}\times 10^{\frac{C_{\mu,htf2}}{T-C_{\mu,htf3}}} \tag{9}$$

$$k_{htf} = C_{k,htf1}T^2 + C_{k,htf2}T + C_{k,htf3} \tag{10}$$