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Research Paper

Energy storage in a complex heat storage unit using commercial grade phase change materials: Effect of convective heat transfer boundary conditions

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HIGHLIGHTS

- Latent energy storge in impure phase change material is studied.
- Non-orthogonal curvilinear coordinate system is used.
- A modified enthalpy-porosity approach is implemented.
- The concentric oblate-shaped inner tube stores the maximum energy in LHTES unit.
- Lower eccentric inner circular tube is more thermally efficient.

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ABSTRACT

For the storage of latent energy in an arbitrary-shaped double-pipe heat exchanger is considered in this study. The heat exchanger is numerically modeled considering the convective heat transfer boundary condition on the inner tube. The horizontal heat exchanger is composed of an insulated outer hexagonal tube and an inner tube. The commercial grade PCM which melts over a temperature range of 8.7 °C is placed in the annular gap between the two tubes. The flow rate and the inlet temperature of the heat transfer fluid (HTF) flowing through the inner tube are varied for the low-temperature solar energy storage. The cross-sectional shapes of the inner tube as well as the vertical position of the circular tube are varied, keeping the cross-sectional area of the annular gap constant. In order to correctly account for the arbitrary-shaped boundaries, a non-orthogonal boundary-fitted coordinate technique on a staggered grid system is used. A control volume based finite difference scheme is employed to solve the non-dimensional set of equations. The predicted results of the velocity and temperature fields, the surface-averaged Nusselt number on the heat transfer surface of the inner tube, the complete and total melt fractions, and the latent and total cumulative energy stored, all as a function of time are presented and discussed. The results show that the effect of the change of the bulk temperature is much more prominent on the storage of energy compared to the change in the mass flow rate of the HTF. For the identical conditions, the oblate-shaped inner tube stores the maximum amount of energy irrespective of the studied shapes and the vertical positions of the inner tube.

1. Introduction

Over the past few decades, the phase change materials (PCMs) have been used in many applications. These include storing and retrieving solar energy and industrial waste heat [1], in desalination [2], in heat recovery [3], to control the temperature in buildings [4], in spacecraft [5], for thermal control of electronic components [6], etc. The primary reason for using PCMs for the above purposes is that through solid-liquid phase change they can absorb or release a significant amount of heat during their melting and solidification processes, respectively. For their high energy density capacities, the PCMs have proven to the competing candidates for all of the above-cited applications.

In recent years, there have been intensive research efforts undertaken by various researchers worldwide to use PCMs for the utilization of renewable energy sources such as solar energy.

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Nomenclature

A^*	darcy coefficient (dimensionless)
$\overline{C}_{P_m}$	weighted average mixture specific heat, kJ kg^{-1} °C^{-1}
C	morphology constant (dimensionless)
φ_m	mixture gravitational acceleration, m s^{-2}
g_L	volume fraction of liquid (dimensionless)
g_S	volume fraction of solid (dimensionless)
f_L	mass fraction of liquid (dimensionless)
f_S	mass fraction of solid (dimensionless)
h_L	sensible enthalpy of liquid, kJ/m
h_S	sensible enthalpy of solid, kJ/m
h_m	sensible enthalpy for mixture, kJ/m
K'	permeability of porous media
P	hydrodynamic pressure, Pa
Pr	laminar Prandtl number = $\frac{\nu_{PCM}}{\alpha_{PCM}}$ (dimensionless)
D_i	diameter of inner cylinder, m
D_o	diameter of outer cylinder, m
Ra	Rayleigh number = $\frac{g\beta_{PCM}(T_{WALL}-T_{SOLIDUS})D_i^3\rho}{\mu_{PCM}\alpha_{PCM}}$ (dimensionless)
Ra^*	modified Rayleigh number = $\frac{Ra\times Pr}{Ste}$ (dimensionless)
Ste	Stefan number = $\frac{C_P(T_{WALL}-T_{SOLIDUS})}{\lambda}$(dimensionless)
Ste_i	initial Stefan number (dimensionless)
S	source term (dimensionless)
S_Φ	source term associated with Φ (dimensionless)
T	temperature, °C
T_{WALL}	inner tube wall temperature, °C
T_i	initial temperature of the solid PCM, °C
ΔT	temperature difference between the inner tube wall and solidus temperature of PCM, °C
T_{ref}	reference temperature, °C
t	time, min
h_m^*	$\frac{C_P(T-T_{SOLIDUS})}{\lambda}$ enthalpy at a grid point (dimensionless)
h_{WALL}^*	enthalpy at the inner wall (dimensionless)
h_i^*	initial enthalpy (dimensionless)
u_m	mixture velocity component along x direction, ms^{-1}
v_m	mixture velocity component along y direction, ms^{-1}
U_m^*, V_m^*	mixture velocity components along X and Y directions respectively (nondimensional)
U	contravariant velocity component in ξ -direction
V	contravariant velocity component in η -direction
$x_\xi, x_\eta, y_\xi, y_\eta$	metric derivatives
Nu_L	local Nusselt number (nondimensional)
$N\bar{u}_{avg}$	surface-averaged Nusselt number (nondimensional)

Greek symbols

α, β, γ	geometric relations between coordinate systems
ΔH	nodal latent heat, kJ
λ_{PCM}	latent heat of fusion, kJ kg^{-1}
Γ_Φ	diffusion coefficient associated with Φ
ρ_L	actual density of liquid, kg m^{-3}
ρ_S	actual density of solid, kg m^{-3}
μ_m	effective mixture viscosity, kg m^{-1} s^{-1}
μ_l	laminar viscosity, kg m^{-1} s^{-1}
Φ	generalized dependent variable
Γ_{eff}	effective diffusivity, m^2 s^{-1}
α_{PCM}	thermal diffusivity of PCM, m^2 s^{-1}
β	coefficient of thermal expansion, $-\frac{1}{\rho_{m\infty}}\left(\frac{\partial\rho_m}{\partial T}\right)_P$, K^{-1}
ν_{PCM}	kinematic viscosity, m^2 s^{-1}
F_o	Fourier number, $\frac{t\alpha_{PCM}}{D_i^2}$ (dimensionless)
ξ, η	axes of nonorthogonal curvilinear coordinate system

Subscripts

L	liquid
S	solid
i	inner cylinder
o	outer cylinder
REF	reference value

Superscripts

*	non-dimensional variables

Since solar energy is an intermittent supply source, hence for its effective and efficient use some form of thermal energy storage devices is necessary. Thermal energy can be stored when the solar energy is available, and the stored energy can be later utilized when the solar energy is not available or when there is a need for such thermal energy sources. Although, the solar thermal energy can be stored through the sensible heat storage systems or by utilizing suitable PCMs through the latent heat of fusion during their solid-liquid phase change transformations or through the thermo-chemical means of utilizing favorable thermo-chemical reactions. Among the above three broad thermal storage systems, the latent heat storage devices have drawn the attention of the researchers the most. The reason for such attention, as mentioned earlier, is their high energy density storage capacities over a narrow temperature range. Thus, they require smaller heat exchanging devices compared to the sensible and other energy storage methods.

The heat storage devices which utilize PCMs are usually called latent heat thermal energy storage (LHTES) systems. To store and later retrieve thermal energy from the LHTES devices, three basic components are necessary. They are the PCM, a suitable containment vessel for holding the PCM, and that will also participate in the heat transfer process and a fluid which would work as a carrier of thermal energy from the source to the PCM or from the PCM to the sink. The research efforts are being directed by the researchers in all of the above three sub-areas for the efficient use of thermal energies. A point to note here is that to maximize the thermal performance of an LHTES unit, a better understanding of the melting/solidification characteristics of the PCMs and the effect of the geometrical shape and orientation of the heat exchanging devices are necessary. The present study is carried out to improve the understanding of the above-stated issues.

In this study, the melting characteristics of a commercial grade paraffin wax which is acting as a PCM in a horizontal double-pipe heat exchanger are numerically modeled. The heat exchanger consists of an outer regular hexagonal-shaped tube and an inner tube. The outer tube is insulated and hence there is no heat loss from the system to the surroundings. The heat transfer fluid is flowing through the inner tube while the PCM is occupying the annular gap between the outer and inner tubes. Through the forced convective heat transfer process, the HTF is supplying heat to the PCM through the wall of the inner tube made of highly conductive material.

The reason for selecting the hexagonal cross-sectional ducts (also called double-trapezoidal ducts) are the facts that they are routinely encountered in compact heat exchangers, particularly of the lamella type heat exchangers [7,8]. These types of heat exchangers are extensively used in pulp and paper, alcohol, petrochemical, and other chemical industries [7,8]. In addition to the above usage, hexagonal-shaped heat exchangers are also used in nuclear reactors [9] and micro-scale fluidic devices [10].

Over the past few decades, many studies have appeared in the literature on the experimental, analytical and numerical modeling of LHTES systems. Unfortunately, most of such studies have dealt