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Thermal Charging and Discharging of Sensible and Latent Heat Storage Packed Beds

Mehmet Sözen,* Kambiz Vafai,† and Lawrence A. Kennedy‡
Ohio State University, Columbus, Ohio 43210

Nomenclature

c_p = specific heat at constant pressure, J/kg.K
h_{sf} = specific latent heat of fusion, J/kg
k = thermal conductivity, W/m.K
L = length of the packed bed, m
P = pressure, N/m^2
R = gas constant for Refrigerant-12, J/kg.K
T = temperature, K
t = time, s
μ = absolute viscosity, kg/m.s
ρ = density, kg/m^3

Subscripts
in = inlet
o = initial
s = solid
v = vapor

Introduction

Contemporary applications of packed beds include the use of encapsulated phase change materials as energy storage media for large energy storage densities. The transient response of sensible heat storage packed beds with incompressible transport fluid has been studied in investigations such as those reported by Riaz,[1] Beasley et al.,[2] and Gross et al.[3] Similar studies for packed beds with encapsulated phase change material (PCM) as storage medium have been reported by Pitts and Hong,[4] and Ananthanarayanan et al.[5] A model for transport phenomena in sensible heat storage packed beds with compressible working fluid, modeled as an ideal gas, has been reported by Vafai and Sozen.[6]

In the present work, the thermal energy storage characteristics of a sensible heat storage and a latent heat storage packed bed are investigated. The energy transporting fluid is Refrigerant-12 which is modeled as an ideal gas. Local thermal equilibrium assumption is not used, and the inertia effects are considered in the vapor phase momentum equation.

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*Postdoctoral Research Associate.
†Professor of Mechanical Engineering. Member AIAA.
‡Professor of Mechanical Engineering. Associate Fellow AIAA.

Analysis

The physical system considered consists of a horizontal channel filled with randomly packed fixed particles of regularly sized and shaped spheres. Initially the void volume of the packed bed is filled with quiescent working fluid which is at uniform temperature and pressure and in thermal equilibrium with the bed particles. Vapor from a reservoir at a higher temperature and pressure is allowed to flow through the packed bed in the thermal charging mode. In the thermal discharging mode, vapor at a lower temperature is considered to flow through the packed bed.

The governing equations for the present problem were developed by use of the well-known "local volume averaging" technique and they follow from the previous work of Vafai and Sozen.[6] It should be emphasized that in modeling the physical phenomena, the thermophysical properties of the encapsulation material were assumed to be essentially the same as those of the PCM in the case of latent heat storage packed bed. Due to the insulated boundary conditions employed in this investigation, the problem essentially reduces to be one dimensional. The governing equations are given in Ref. 6. The model developed forms a system of five equations in five unknowns.

The initial conditions of the problem were as follows:

$$P_v(x, t = 0) = P_o$$
$$T_v(x, t = 0) = T_s(x, t = 0) = T_o$$
$$u_v(x, t = 0) = 0 \tag{1}$$

and the corresponding initial values of ρ_v are calculated from the equation of state.

The boundary conditions of the problem were as follows:

$$P_v(x = 0, t) = P_{\rm in}$$
$$P_v(x = L, t) = P_o$$
$$T_v(x = 0, t) = T_{v\rm in} \tag{2}$$

where P_o = 100 kPa, $P_{\rm in}$ = 106.83 kPa, T_0 = 300 K and $T_{v\rm in}$ = 350 K for thermal charging mode, and $T_{v\rm in}$ = 300 K for thermal discharging mode. The particle diameter was taken to be 2 mm and the average porosity of the packed bed was taken to be 0.39. The nominal particle Reynolds number for these boundary conditions was 1000.

The modeling of the effective thermal conductivities, the fluid-to-particle heat transfer coefficient, the permeability, and the physical and geometric parameters in the vapor phase momentum equation has been explained in detail in Ref. 6.

The thermophysical properties used in the present investigation were as follows:

Refrigerant-12	*1% Carbon-steel*	*Myristic acid (PCM)*
c_p = 710 J/kg.K	c_p = 473 J/kg.K	c_p = 1590 J/kg.K (solid)
k = 0.0097 W/m.K	k = 43 W/m.K	c_p = 2260 J/kg.K (liquid)
$\mu = 12.6 \times 10^{-6}$ kg/m.s	ρ = 7800 kg/m^3	k = 0.1 W/m.K
		ρ = 860 kg/m^3
R = 68.7588 J/kg.K		$h_{sf} = 200.5 \times 10^3$ J/kg

The numerical solution scheme for the sensible heat storage packed bed (SHSPB) is detailed in Ref. 6 and is used in the present work too. It should be noted that for the solution of the problem for latent heat storage packed bed (LHSPB), the solution algorithm is the same as for the SHSPB except that during the thermal charging process, once the temperature of the PCM reaches its melting temperature, it remains constant until melting of the PCM is complete. This condition has been incorporated into the numerical algorithm such that at a given point once the melting temperature is reached, the PCM temperature is kept constant while the net rate of heat flowing into the associated volume is continuously monitored and integrated over time to give the energy that is used to melt the PCM until it becomes equal to the amount required for melting. After this stage the PCM temperature is again computed from the solid phase (or PCM) energy equation, this time with the liquid properties of the PCM incorporated into the equation. Similar provisions are made for the thermal discharging process of the LHSPB which involves freezing of the PCM.

Results and Discussions

The two problems considered in the present work were the complete thermal charging and thermal discharging of two types of packed beds: one with sensible heat storage material, chosen to be 1% Carbon-steel, and the other one with latent heat storage material, chosen to be myristic acid (a PCM with a melting point of 331 K). Thermal charging or discharging was assumed to be complete when the difference between the incoming and exiting vapor temperatures was less than 0.01 K.

In the solution of the thermal charging problem, it was found that two distinct stages were present, namely, the *early stage* and the *later stage*. This has been discussed in detail in a previous work of the authors.[6] In this work, emphasis is given to the thermal charging and discharging characteristics of the two types of packed beds using a compressible working fluid.

Figure 1 depicts the variation in the temperature distribution of the vapor and solid phases during the thermal charging and discharging processes for the LHSPB. Corresponding variation for the case of SHSPB has been presented elsewhere[6] for thermal charging, and, therefore, for purposes of space economy it is not repeated here.

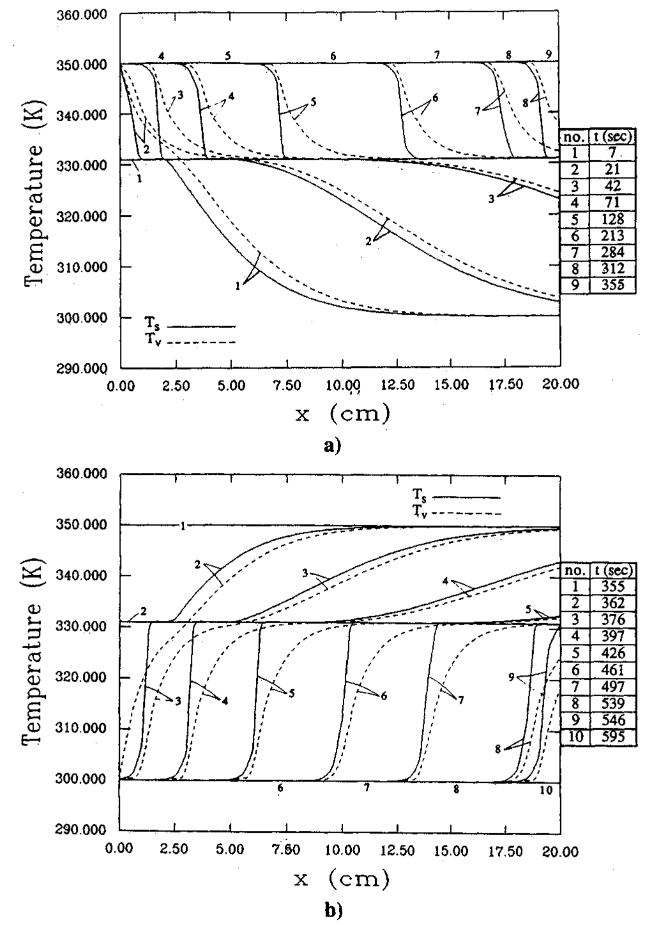

Fig. 1 Temperature distribution in the LHSPB during a) thermal charging and b) thermal discharging.

The time histories of the thermal charging and discharging of the SHSPB and the LHSPB are shown in Fig. 2. This figure depicts the variation of the net energy stored per unit width of the packed beds. In the case of the SHSPB the thermal charging and discharging times of the bed are practically equal. However, the LHSPB shows different characteristics. A few interesting points can be observed in the thermal charging and discharging of the LHSPB. Both during the charging and the discharging modes there is a major portion of each process during which there is a quite uniform difference between the heat fluxes into and out of the LHSPB which is not seen in the case of the SHSPB. The reason for this kind of behavior is the fact that during these portions of the charging or the discharging, the temperature of the PCM in the particles downstream of the packed bed remain constant at the melting/freezing temperature. Hence, for these portions of the charging or discharging processes, the vapor exit temperature is practically equal to the melting/freezing temperature of the PCM and, therefore, the difference between the inlet and exit temperatures for the vapor phase remains practically unchanged. This can be seen more clearly by considering Fig. 3 which depicts the time histories of the rates of heat flow into and out of the SHSPB and LHSPB under similar conditions during thermal charging and discharging. Another interesting observation can be made with respect to the charging and discharging times of the LHSPB. From Fig. 3, it can easily be observed that the thermal discharging of the LHSPB lasts

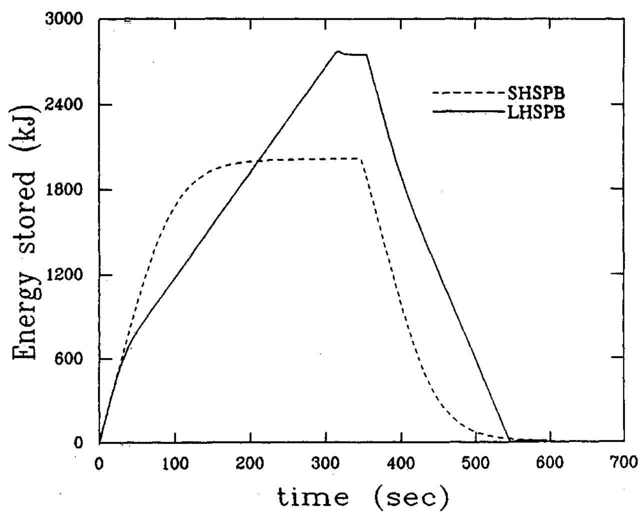

Fig. 2 Thermal charging and discharging of the SHSPB and the LHSPB.

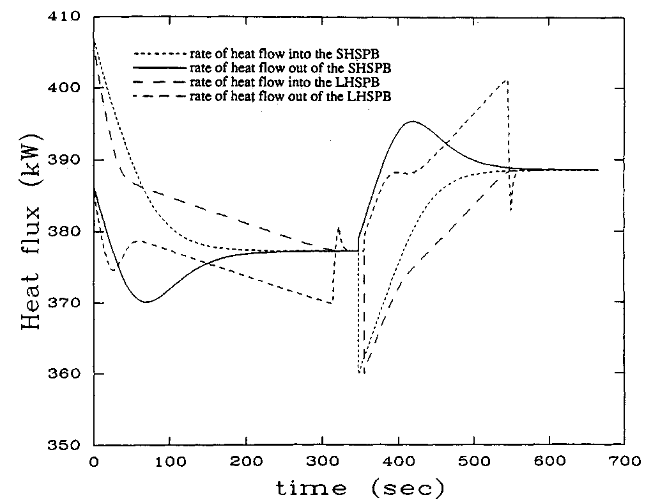

Fig. 3 Time histories of the rates of heat flow into and out of the SHSPB and LHSPB.

considerably shorter than the thermal charging. The reason for this is that for the major portion of the discharging process during which the temperature of the PCM downstream of the LHSPB remains at the melting/freezing point, there is a constant temperature difference between the vapor inlet and vapor exit temperatures. This difference is larger than the corresponding difference in the charging process, i.e., the difference between the vapor inlet temperature and the melting point of the PCM during the charging process is only 19 K, whereas during the discharging process the difference between the vapor inlet temperature and the freezing point of the PCM is 31 K. Hence, energy is stored at a lower rate in the charging process than energy is removed during the discharging process for the portions of these processes under discussion. Again, this fact can easily be seen from Fig. 3 in which the difference between the rate of heat flow into and out of the packed bed is shown.

Conclusions

The transient processes of thermal charging and discharging of a SHSPB and a LHSPB with a compressible working fluid were simulated. The investigations showed distinctly different energy storage characteristics for these two kinds of packed beds. The high-energy storage density of the LHSPB was clearly observed from the studies carried out. For the two energy storage materials considered, although the density of the sensible heat storage material (1% Carbon-steel) was approximately nine times larger than that of the PCM (myristic acid), the total energy storage capacity of the LHSPB was higher.

Use of a compressible working fluid and accounting for the inertia effects in the vapor phase momentum equation resulted in a time-dependent mass flow rate through the packed bed.

Also, in the case of the LHSPB it was observed that the thermal charging and discharging times differed considerably. The main reason for this difference is that for a major portion of the charging or discharging process the PCM temperature downstream of the packed bed remains constant at the melting/freezing temperature of the PCM. An important conclusion which was obtained from our results is that the closer the PCM melting/freezing temperature is to the charging temperature (vapor inlet temperature during thermal charging mode) the longer will be the time taken for charging and the shorter will be the time taken for discharging of the packed bed. Likewise, the closer the PCM melting/freezing temperature is to the discharging temperature (vapor inlet temperature during thermal discharging mode) the longer will be the time taken for discharging and the shorter will be the time taken for charging of the packed bed.

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Convective Heat Transfer Across a Duct with Asymmetric Blowing

H. J. Deacon Jr.* and D. A. Wallace†
The Aerospace Corporation,
El Segundo, California 90009

Introduction

A THERMAL protection system for advanced reusable hypersonic vehicles uses the evaporation of ice in a wicking material to protect the substructure from excessive heating. In the concept, refractory metal tiles supported by low-conductance thermal standoffs are subjected to the aerodynamic forces and heating of re-entry. The wick, which lies between the tiles and substructure, is heated by radiation from the tiles and convection through steam released from upstream evaporation. The steam convection problem may be modeled as the flow between an impermeable upper wall and a porous lower wall with fluid injection. No previous solutions to this problem were found. In an initial attempt to model the flow and heat transfer, similarity solutions of the incompressible, constant property Navier-Stokes and energy equations were sought. Preliminary design analysis suggested that the Reynolds number based on the mass blowing rate of the steam and the Peclet number would be of order unity. Therefore, a wide range of solutions were sought at higher and lower values of these parameters. The basic approach of White et al.[1] and Berman[2] was adopted to obtain solutions to the Navier-Stokes equations. Similarity solutions to the energy equation are found for the boundary layer approximation with viscous dissipation neglected for constant wall temperatures. The present problem is different from the work of Refs. 1 and 2 because only one wall is porous in this work and because solutions to the energy equation are also obtained. These solutions are used to obtain heat transfer rates to the upper and lower walls. The computed results can then be used for improved design analysis of the thermal protection system.

Solution of Navier-Stokes Equations

In this work, the flow is assumed to be two dimensional, incompressible, steady, and laminar with negligible body forces. The evaporation of the ice is simulated by a porous lower plate having a constant and uniform normal velocity.

The mass and momentum conservation equations are simplified to

$$\frac{\partial u}{\partial x} + \frac{1}{h}\frac{\partial v}{\partial \lambda} = 0$$

$$u\frac{\partial u}{\partial x} + \frac{v}{h}\frac{\partial u}{\partial \lambda} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \nu\left(\frac{\partial^2 u}{\partial x^2} + \frac{1}{h^2}\frac{\partial^2 u}{\partial \lambda^2}\right) \quad (1)$$

$$u\frac{\partial v}{\partial x} + \frac{v}{h}\frac{\partial v}{\partial \lambda} = -\frac{1}{\rho h}\frac{\partial p}{\partial \lambda} + \nu\left(\frac{\partial^2 v}{\partial x^2} + \frac{1}{h^2}\frac{\partial^2 v}{\partial \lambda^2}\right) \quad (2)$$

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*Engineering Specialist, P.O. Box 92957, Los Angeles, CA 90009. Member AIAA.

†Member of the Technical Staff.