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Power, Efficiency, and Irreversibility of Latent Energy Systems

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The object of this paper is to present a simple model for the latent storage systems by using a phase-change material encapsulated in the spherical capsules. When designing phase-change-material systems, it is important to minimize thermal irreversibilities, because it works best when irreversibilities are done. In this paper, second law analysis is used to evaluate the irreversibility of the system. The effects of different parameters are also discussed. The proposed model can be applied to the charge and the discharge mode. Exergy efficiency is also evaluated.

Nomenclature

A = superficial particle area per unit bed volume, m^{-1}
c_f = specific heat of the heat-transfer fluid, $J\,kg^{-1}\,K^{-1}$
c_{PCM} = specific heat of the PCM, $J\,kg^{-1}\,K^{-1}$
D = outer diameter of the capsule, m
$\dot{E}x_{cv}$ = rate of exergy change of the control volume, w
k_f = thermal conductivity of the working fluid, $W\,m^{-1}\,K^{-1}$
k_{PCM} = thermal conductivity of the PCM, $W\,m^{-1}\,K^{-1}$
k^*_{PCM} = effective thermal conductivity of the PCM, $W\,m^{-1}\,K^{-1}$
L_F = heat of fusion of the PCM, $J\,kg^{-1}$
m = mass of the PCM inside the tank, kg
$\dot{m}$ = mass flow rate of the heat-transfer fluid, $kg\,s^{-1}$
N_s = entropy generation number
Nu = Nusselt number
$P(t)$ = useful power, W
Pr = Prandtl number
R = specific gas constant, $J\,kg^{-1}\,K^{-1}$
Re = Reynolds number
$\dot{S}_{gen}$ = entropy generation rate, $W\,K^{-1}$
s = specific entropy of the fluid, $J\,kg^{-1}\,K^{-1}$
T_f = temperature of the working fluid, K
T_M = melting temperature of the PCM, K
T_{ini} = initial temperature in the charging process, K
T_{int} = inlet temperature of the fluid, K
T_{out} = outlet temperature of the fluid, K
T_{PCM} = temperature of the PCM, K
t^* = dimensionless time
U_f = heat-transfer coefficient, $W\,m^{-2}\,K^{-1}$
V_e = velocity of the working fluid $m\ s^{-1}$
x = axial coordinate, m
β = fraction of the liquid PCM inside the capsules, m
ΔP = pressure drop, Pa
ΔT = difference between the inlet and the melting temperatures, K
ε = porosity of the tank
μ = dynamic viscosity of the fluid, $K\,gm^{-1}\,s^{-1}$
ρ_f = density of the heat-transfer fluid, $kg\,m^{-3}$
ρ_{PCM} = density of the PCM, $kg\,m^{-3}$
τ = duration of the thermal storage, s

I. Introduction

LATENT-HEAT thermal energy storage (LHTES) systems provide an effective way to make better use of the thermal energy (Khudhair and Farid [1], Zalba et al. [2], Saito [3], Hasnain [4], and Hall et al. [5]). These systems are based on the use of phase-change materials (PCM).

Different PCM systems are reviewed and reported in research (Zalba et al. [2]). Several characteristics are desired from a PCM to be used for energy storage. Among them, melting point and supercooling are particularly considered here, because the application related to encapsulated PCM systems has been discussed in previous research (Kousksou et al. [6], Ismail et al. [7], Arnold [8], Eames and Adref [9], Saitoh and Hirose [10], Laybourn [11], and Ismail et al. [12]). The data reported are mainly based on the first law of thermodynamics. From a first law perspective, the efficiency of a thermal energy storage system can be assessed in terms of how much thermal energy the system can store.

The conventional energy analysis (based only on first law analysis) is not sufficient enough to account for the actual performance of the storage process, because it does not consider the effect of irreversibilities on the system efficiency. That is why the exergy analysis [based on second law analysis (Bejan [13])] is used to get a clear picture of the various losses quantitatively as well as qualitatively.

In the present work, we apply models based on the first and second laws of thermodynamics to analyze the latent energy storage. The influences of different parameters on entropy generation during thermal energy storage are also investigated. Exergy efficiency is also evaluated.

II. Phase-Change Energy System

Figure 1 presents a latent-heat thermal energy storage system. The system consists in using the spherical containers of PCM, randomly set into a tank. It should be used for storing solar energy using PCM, for which the melting point can be changed from 18 to 60°C and which does not present supercooling. The working fluid can be air, in solar applications, or gas in other industrial processes. During the charge mode, the working fluid from the heat source flows through the tank, causing the melting of the encapsulated PCM, whereas during the discharge mode, the cooled fluid, which passes through the capsules, crystallizes the PCM.

The fluid is then used to provide heating either directly or through a heat exchanger. The spherical capsules were chosen because of their performance and easy introduction into various tanks.

III. Physical Model and Derivation

The tank filled with the PCM is considered as a porous medium. In this model, the following assumptions are considered:

1) The PCM capsules form a continuous medium. We do not consider the capsules as independent particles.

2) Thermal gradients within the capsules are negligible. In general, a temperature gradient within the capsule can be negligible when the Biot number $(U_f D/k_{PCM})$ is less than 0.1. In the present study, this condition is verified.

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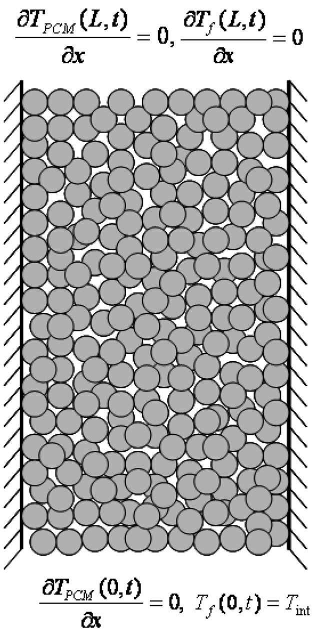

Fig. 1 Phase-change system.

3) The thermophysical properties of the PCM and of the heat fluid flow are temperature-independent.

4) Overall heat-transfer coefficients between the capsules and the coolant are constant during the charge period.

5) Fluid flow is considered unidirectional.

Based on the preceding assumptions, the energy equations for the fluid can be written as

$$\varepsilon(\rho c)_f\left(\frac{\partial T_f}{\partial t} + V_e\frac{\partial T_f}{\partial x}\right) = \frac{\partial}{\partial x}\left(k_f\frac{\partial T_f}{\partial x}\right) + U_f A(T_{\mathrm{PCM}} - T_f) \quad (1)$$

During sensible heat storage, when the PCM is completely solid or liquid, the corresponding governing equation can be written in this form:

$$(1-\varepsilon)(\rho c)_{\mathrm{PCM}}\frac{\partial T_{\mathrm{PCM}}}{\partial t} = \frac{\partial}{\partial x}\left(k^*_{\mathrm{PCM}}\frac{\partial T_{\mathrm{PCM}}}{\partial x}\right) + U_f A(T_f - T_{\mathrm{PCM}}) \quad (2)$$

In the latent-heat storage, when the PCM is changing phase, the corresponding model takes this form:

$$\frac{\partial \beta}{\partial t} = \frac{AU_f}{(1-\varepsilon)\rho_{\mathrm{PCM}}L_F}(T_f - T_M) + \frac{k^*_{\mathrm{PCM}}}{\rho_{\mathrm{PCM}}L_F}\frac{\partial^2 T_{\mathrm{PCM}}}{\partial x^2} \quad (3)$$

where ρ_f and c_f are, respectively, the density and the specific heat of the fluid; U_f is the overall constant heat-transfer coefficient; A is the superficial particle area per unit bed volume; V_e is the mean heat-transfer fluid flow velocity; ε is the void fraction of the tank; β is the fraction of the liquid PCM inside the capsules; T_M is the melting temperature of the PCM; and L_F is the latent heat resulting from the melting of PCM.

In the preceding equations, the effective thermal conductivity k^*_{PCM} is evaluated from the Schmidt correlation (Donnadieu [14]):

$$k^*_{\mathrm{PCM}} = 1.72 k_f\left(\frac{k_f}{k_{\mathrm{PCM}}}\right)^{0.26} \quad (4)$$

The initial and boundary conditions are specified by

$$T_f(x,0) = T_{\mathrm{PCM}} = T_{\mathrm{ini}} \quad (5)$$

$$T_f(0,t) = T_{\mathrm{int}} \quad (6)$$

$$\frac{\partial T_f(L,t)}{\partial x} = 0 \quad (7)$$

$$\frac{\partial T_{\mathrm{PCM}}(0,t)}{\partial x} = \frac{\partial T_{\mathrm{PCM}}(L,t)}{\partial x} = 0 \quad (8)$$

It is noted that $\beta = 0$ if the capsules are in a completely solid state, and $\beta = 1$ in the case in which the whole containers are in a liquid state.

The convection heat-transfer condition at the external surface of the capsules can be treated by using the mean Nusselt number (Gunn [15]), given as

$$Nu = (7 - 10\varepsilon + 5\varepsilon^2)(1 + 0.7Re^{0.2}Pr^{0.33}) + (1.33 - 2.44\varepsilon + 1.2\varepsilon^2)Re^{0.7}Pr^{0.33} \quad (9)$$

The Reynolds number Re can be obtained from the relation

$$Re = \frac{\rho_f V_e D}{\mu} \quad (10)$$

where D is the diameter of the sphere, and μ the dynamic viscosity of the fluid. The pressure drop can be calculated by using the Ergun equation (Bird et al. [16]) :

$$\frac{\Delta P}{L} = \frac{P_{\mathrm{out}} - P_{\mathrm{int}}}{L} = \frac{150\mu_f u_0}{D^2}\frac{(1-\epsilon)^2}{\epsilon^3} + 1.75\frac{\rho_f u_0^2}{D}\frac{(1-\epsilon)}{\epsilon^3} \quad (11)$$

where P_{out} and P_{int} are the outlet and inlet pressures of thermal storage tank. The length of the thermal storage tank and the capsule diameter are L and D, respectively. The viscosity of the fluid is μ_f. The superficial velocity of the coolant flowing through the thermal storage tank with no paraffin capsules is u_0.

The finite difference equations are obtained by integrating the conservation equations over a control volume. The resulting finite difference scheme is solved iteratively at a given time step, with a tridiagonal matrix solver.

IV. Results and Discussion

In this section, only the charge mode is considered. In all calculations, we suppose that the inlet temperature T_{int} of the working fluid remains at a constant value (Fig. 2). The charging process is terminated when the PCM temperature at the outlet of the tank reaches the value of the inlet temperature T_{int}. The effects of the Reynolds number of the working fluid and the difference ΔT (between the inlet temperature of the heat-transfer fluid throughout the heat storage system and the melting temperature) on the behavior of the storage tank are discussed.

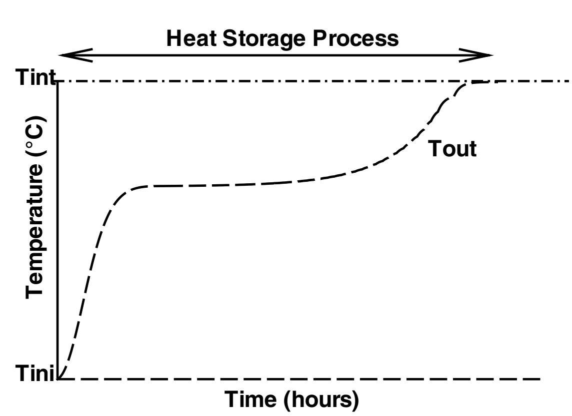

Fig. 2 Temperature variation during the storage period.

A. First Law Analysis

Figure 3 presents the effect of the Reynolds number Re on the duration of the total completion of the storage process for various values of ΔT. We note that an increase in the Reynolds number or ΔT leads to a decrease in the charging time. These results are evident because the increase in both the Reynolds number and the inlet temperature of the working fluid, involves an increase in the heat transfer between the PCM and the fluid.

To evaluate the performance of the storage tank, the useful power $P(t)$ versus time was calculated:

$$P(t) = \dot{m} c_f (T_{\mathrm{int}}(-T_{\mathrm{out}}(t))) \tag{12}$$

where $\dot{m}$ is the mass flow rate of the fluid, and T_{int} and T_{out} are the inlet and the outlet temperatures of the heat-transfer fluid.

Figures 4 and 5 show the effect of ΔT and the Reynolds number on the useful power as a function of the percentage of the stored latent energy, respectively. At the initial period of storage, a sharp drop of the useful power is observed. This is due to the sudden increase of the inlet temperature, whereas the outlet remains constant at the beginning of the process. The duration of this increase represents the time required for the fluid to flow through the tank from the inlet to the outlet. Then the useful power stabilizes as a result of melting. Before this stabilization, the useful power decreases. In general, any increase of ΔT or the Reynolds number leads to an increase in the useful power.

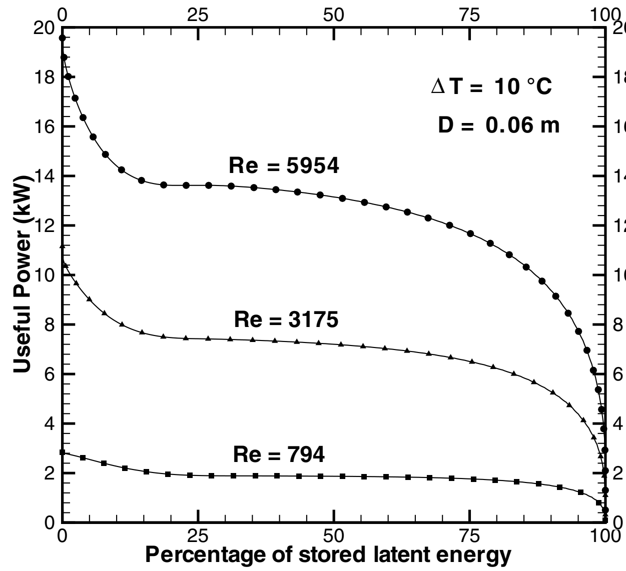

Fig. 5 Variation of the useful power versus the percentage of latent stored energy for various Reynolds numbers.

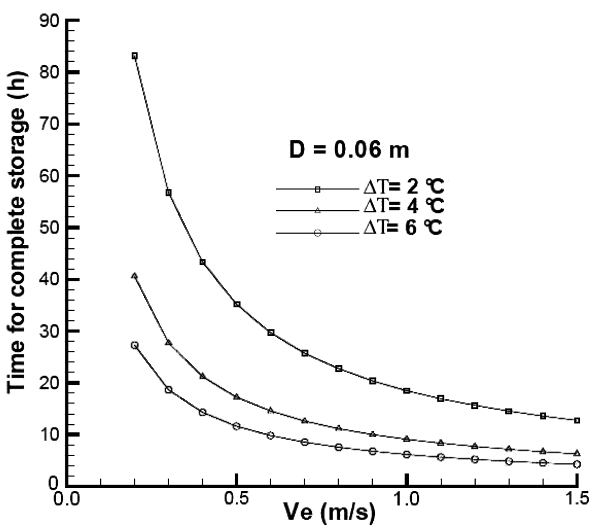

Fig. 3 Effect of the fluid velocity on the charging time.

The tank can be regarded as a heat exchanger for which the efficiency is defined as the relationship between transferred heat flow and maximal transferable heat flow:

$$\eta(t) = \frac{\dot{Q}(t)}{\dot{Q}_{\mathrm{max}}} = \frac{\dot{m} c_f (T_{\mathrm{int}} - T_{\mathrm{out}}(t))}{\dot{m} c_f (T_{\mathrm{int}} - T_{\mathrm{ini}})} = \frac{T_{\mathrm{int}} - T_{\mathrm{out}}(t)}{T_{\mathrm{int}} - T_{\mathrm{ini}}} \tag{13}$$

The influences of ΔT and the Reynolds number on the efficiency of the storage tank versus the percentage of latent stored energy are shown in Figs. 6 and 7. As the time of storage increases (time required for the total completion of the charging process), the system efficiency decreases during sensible heating of the solid PCM and it remains constant during the phase change. After phase change, the efficiency then further decreases when the liquid PCM is significantly heated. This is due to the fact that as charging proceeds, the temperature difference between the operating fluid and the PCM in the storage tank decreases.

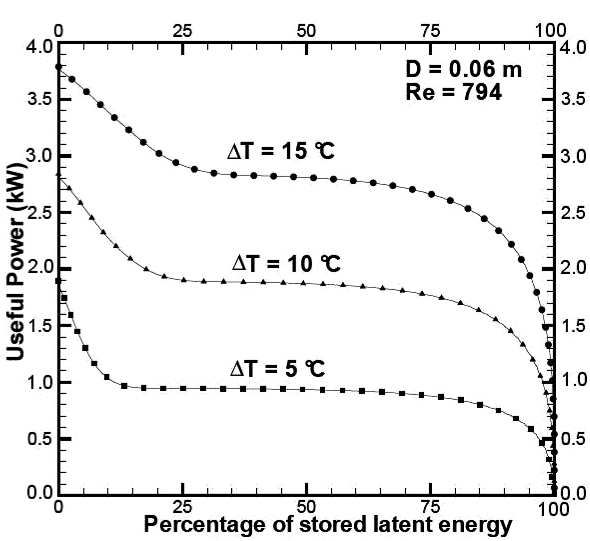

Fig. 4 Variation of the useful power versus the percentage of latent stored energy for different ΔT.

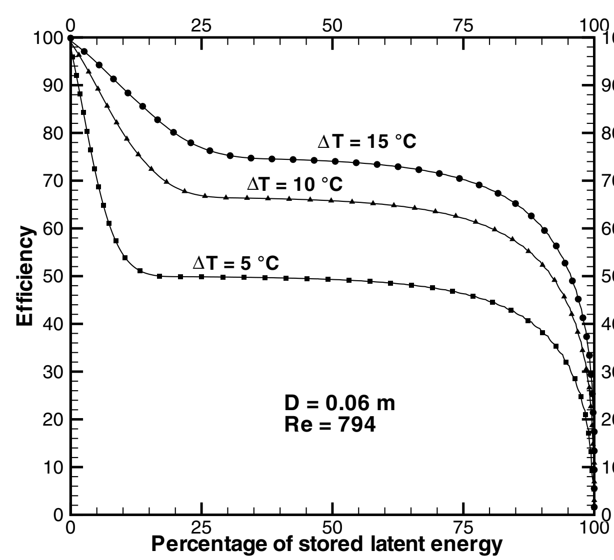

Fig. 6 Effect of the inlet temperature on the thermal efficiency of the storage tank.

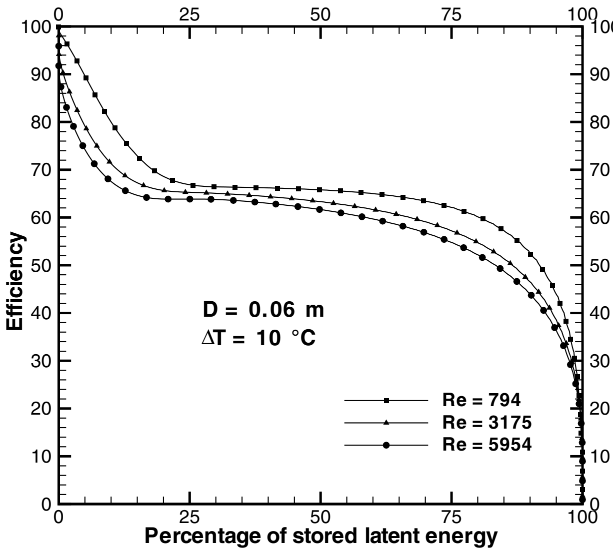

Fig. 7 Effect of the Reynolds number on the thermal efficiency of the storage tank.

We can also note that the system efficiency increases as the inlet temperature of the working fluid goes up. However, it decreases when the Reynolds number increases. The dependence of the efficiency at ΔT is more pronounced than with the Reynolds number.

We have also studied the effect of the capsule diameter D on the pressure drop inside the tank. Figure 8 shows the variation of the pressure drop through the tank versus D and for various fluid velocities. The pressure drop increases strongly with velocity and is much higher for cases with lower capsule diameters. An optimized design must balance the heat-transfer performance for the given porous configuration with the pressure drop penalty. An interesting performance-to-penalty measure is the ratio of the heat-transfer coefficient to pressure drop. This is shown as a function of velocity in Fig. 9. This ration is much higher at lower velocities and decreases rapidly with velocity, indicating the benefits of system optimization for the lowest possible velocity. Figure 9 also shows the ration decrease sharply with decreasing capsule diameter. This indicates the benefit of using the largest diameter possible for the capsule.

The results presented up to now are based on a first law of thermodynamics model. From these results, we can note that a combination of a low Reynolds number and a high inlet temperature of the operating fluid seems to be the ideal combination for the range of the study parameters investigated.

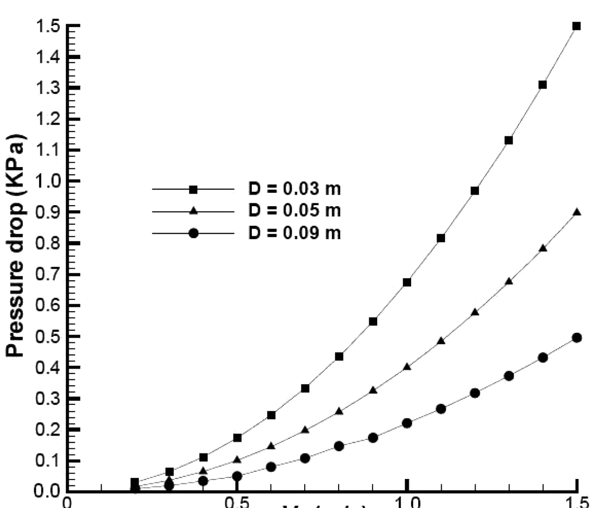

Fig. 8 Variation of pressure drop versus the fluid velocity for different capsule diameters.

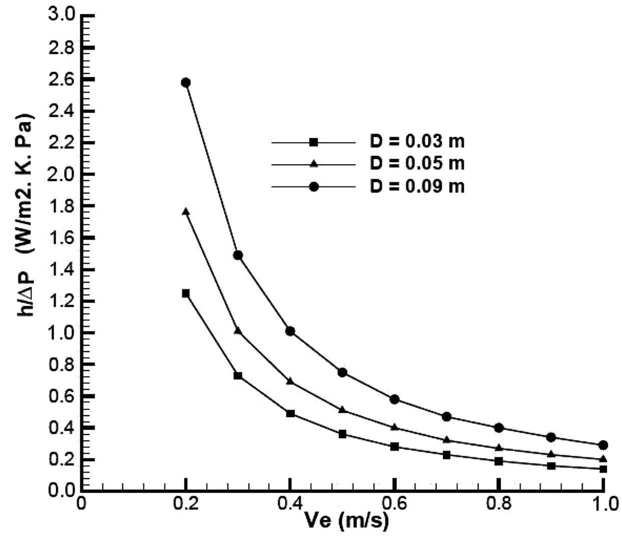

Fig. 9 Ratio of heat-transfer coefficient to pressure drop as a function of velocity for various capsule diameters.

B. Second Law Analysis

To draw conclusions with more precision, we applied a second law analysis. Usually, the increase of ΔT and the Reynolds number leads to an increase in two kinds of irreversibilities: one is due to heat transfer through the temperature gradient, and the second is due to the fluid flow through the tank.

In fact, global energetic efficiency depends principally on the value of these two parameters. For high values of ΔT, energy storage is possible for a short duration, but it would be necessary to have a source of heat (for example, high-efficiency solar collectors). On the other hand, high Reynolds numbers also lead to a short storage duration; however, these cause mechanical irreversibilities and additional pumping work.

To optimize the irreversibility inside the storage system during the charge period, the entropy generation number N_s is considered (Bejan [13]). This number is defined as the ratio of the destroyed exergy (during the time interval [0,t]) to the exergy stored by the system during the same interval; that is,

$$N_s = \frac{T_0 \int_0^t \dot{S}_{\mathrm{gen}}\,\mathrm{d}t}{\int_0^t \dot{E}x_{\mathrm{cv}}\,\mathrm{d}t} \tag{14}$$

Tank (control volume) is considered as an open system. During the charge mode, the fluid enters the system, flows through the nodules and exits colder. $\dot{S}_{\mathrm{gen}}$ is the instantaneous rate of entropy. The second law of thermodynamics states

$$\dot{S}_{\mathrm{gen}} = \frac{\partial S}{\partial t} - \dot{m}(s_{\mathrm{int}} - s_{\mathrm{out}}) \geq 0 \tag{15}$$

where $\partial S/\partial t$ is the instantaneous variation of the entropy of the system, and both s_{in} and s_{out} are the specific entropy of the fluid at the inlet and the outlet of the bed. Assuming that air is an ideal gas, the specific entropy change of the fluid can be calculated:

$$s_{\mathrm{in}} - s_{\mathrm{out}} = c_f \ln\left(\frac{T_{\mathrm{int}}}{T_{\mathrm{out}}}\right) + R \ln\left(\frac{P_{\mathrm{out}}}{P_{\mathrm{int}}}\right) \tag{16}$$

Assuming that the pressure of the inlet air is equal to the atmospheric pressure P_a, Eq. (15) can be written as

$$s_{\mathrm{in}} - s_{\mathrm{out}} = c_f \ln\left(\frac{T_{\mathrm{int}}}{T_{\mathrm{out}}}\right) + R \ln\left(1 + \frac{\Delta P}{P_a}\right) \tag{17}$$

The entropy variation of the system is the sum of the entropy variation of all the parts in each control volume:

$$\frac{\partial S}{\partial t} = \sum_{i=1}^{N}\left[\frac{\partial S_{\mathrm{air},i}}{\partial t} + \frac{\partial S_{\mathrm{PCM},i}}{\partial t}\right] \tag{18}$$

For the fluid flow,

$$\frac{\partial S_{\mathrm{air},i}}{\partial t} = \rho c_f \epsilon V_i \frac{\partial}{\partial t}(\ell n\, T_i) \tag{19}$$

In this model, we neglect the entropy variation of the plastic capsules.

For the solid or liquid PCM, the corresponding equation is

$$\frac{\partial S_{\mathrm{PCM},i}}{\partial t} = \rho_{\mathrm{PCM}} c_{\mathrm{PCM}}(1-\epsilon)V_i \frac{\partial}{\partial t}(\ell n\, T_{\mathrm{PCM},i}) \tag{20}$$

During the melting process, the equation can be written as

$$\frac{\partial S_{\mathrm{PCM},i}}{\partial t} = \frac{(1-\epsilon)V_i \rho_l L_F}{T_m}\frac{\partial f_i(t)}{\partial t} \tag{21}$$

$\dot{E}x_{\mathrm{cv}}$ is the time rate of change exergy of the control volume. It is defined as

$$\dot{E}x_{\mathrm{cv}} = \overbrace{\dot{E}x_{\mathrm{inl}} - \dot{E}x_{\mathrm{out}}}^{1} - \overbrace{T_0 \dot{S}_{\mathrm{gen}}}^{2} \tag{22}$$

where the first and second terms correspond, respectively, to the rate of exergy transfer and exergy time rate.

The rate of exergy transfer can be expressed as

$$\dot{E}x_{\mathrm{inl}} - \dot{E}x_{\mathrm{out}} = \dot{m}c_f(T_{\mathrm{inl}} - T_{\mathrm{out}}) - T_0 \dot{m} c_f\, \ell n\left(\frac{T_{\mathrm{inl}}}{T_{\mathrm{out}}}\right) \tag{23}$$

Figures 10 and 11 show the effect of ΔT and Reynolds number on the fraction of destroyed exergy. The variations of N_s are represented versus the dimensionless time (this is the time required for the total completion of the process).

During the initial period of the latent storage, the entropy generation number N_s increases when the dimensionless time increases. This increase is due to the fact that during this period, the temperature of both the operating fluid and the PCM increase at a faster rate. Moreover, the temperature difference between them also increases continuously until the PCM reaches its melting temperature. This causes an increase in the irreversibility inside the tank. During this period, N_s increases, thus increasing either ΔT or the Reynolds number.

After the initial period (during the phase-change period and the sensible heating of PCM), the entropy generation number N_s remains nearly constant. This is due to the decrease in both the irreversibility and the temperature gradient between the working fluid and PCM within the storage tank. Unlike the initial period, N_s decreases, therefore increasing ΔT and the Reynolds number.

We can also note that the irreversibility within the LHTES is strongly affected by the inlet temperature of the operating fluid. Based on the obtained results (second law analysis), the availability losses of the system can be reduced by proper selection of the Reynolds number and the inlet temperature. Particular attention should be given to the inlet temperature, because the results have shown that the Reynolds number Re has less influence on availability losses.

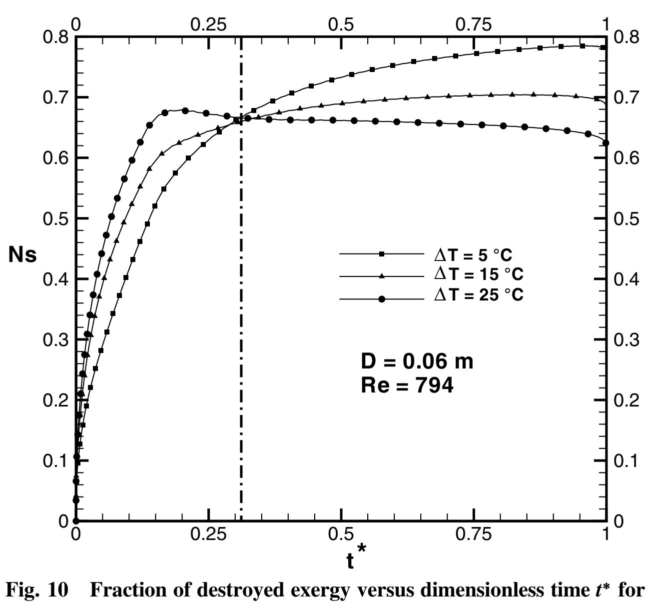

Fig. 10 Fraction of destroyed exergy versus dimensionless time t^* for different ΔT.

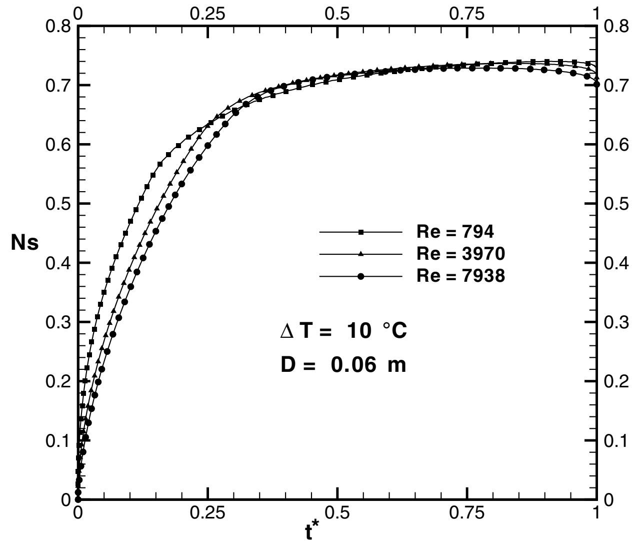

Fig. 11 Fraction of destroyed exergy versus dimensionless time t^* for different Reynolds numbers.

V. Exergy Efficiency

After the storage operation, it is desirable to release the working fluid at a constant temperature as long as possible and with a higher power. The tank works best during this period, as was studied by applying the basis of exergy principle. In the discharge process, the enthalpy efficiency is defined as

$$\eta = \frac{Q_f}{Q_L} = \frac{c_f \dot{m}(T_{\mathrm{out}} - T_{\mathrm{int}})\Delta t}{m L_F} \tag{24}$$

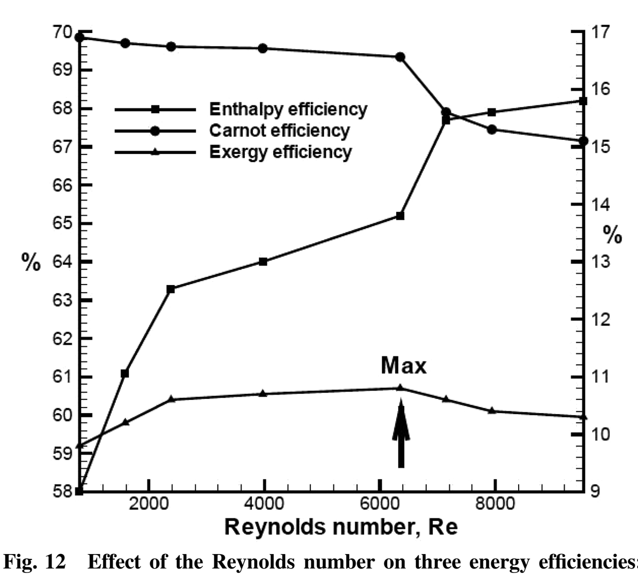

Fig. 12 Effect of the Reynolds number on three energy efficiencies: enthalpy efficiency, Carnot efficiency, and exergy efficiency.

where Q_f is the enthalpy of outlet fluid at the period of constant temperature, and Q_L is the latent heat of PCM inside the tank. Exergy efficiency is defined as enthalpy efficiency times Carnot efficiency $[1 - (T_{\mathrm{int}}/T_{\mathrm{out}})]$. The enthalpy efficiency expresses the total quantity of energy, whereas the exergy efficiency reflects the quality of the energy. Therefore, it is desirable to operate the latent storage system under maximum exergy efficiency.

Figure 12 illustrates the effect of the Reynolds number on three energy efficiencies. With increase of the Reynolds number, enthalpy efficiency increases; however, Carnot efficiency decreases because outlet fluid temperature lowered. As a result, exergy efficiency shows maximum value.

VI. Conclusions

This paper analyzed the performance of a thermal storage tank during the charging process by using spherical capsules with varying inlet fluid temperature and fluid velocities. This study revealed that an increase in the inlet temperature of the coolant fluid to reduce the duration of the latent storage leads to significant exergy destruction within the overall system. The irreversibilities are mainly affected by the inlet temperature. This results cannot be expected from a first law analysis. The storage tank works at its optimum during the discharging period, which is determined from its exergy efficiency.

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References

[1] Khudhair, A. M., and Farid, M. M., "A Review on Energy Conservation in Building Applications with Thermal Storage by Latent Heat Using Phase Change Materials," *Energy Conversion and Management*, Vol. 45, No. 2, Jan. 2004, pp. 263–275. doi:10.1016/S0196-8904(03)00131-6

[2] Zalba, B., Marin, J. M., Cabeza, L. F., and Mehling, H., "Review on Thermal Energy Storage with Phase Change: Materials, Heat Transfer Analysis and Applications," *Applied Thermal Engineering*, Vol. 23, No. 3, Feb. 2003, pp. 251–283. doi:10.1016/S1359-4311(02)00192-8

[3] Saito, A., "Recent Advances in Research on Cold Thermal Energy Storage," *International Journal of Refrigeration*, Vol. 25, No. 2, Mar. 2002, pp. 177–189. doi:10.1016/S0140-7007(01)00078-0

[4] Hasnain, S. M., "Review on Sustainable Thermal Energy Storage Technologies, Part 1: Heat Storage Materials and Techniques," *Energy Conversion and Management*, Vol. 39, No. 11, Aug. 1998, pp. 1127–1138. doi:10.1016/S0196-8904(98)00025-9

[5] Hall, C. A., Glakpe, E. K., Cannon, J. N., and Kerslake, T. W., "Modeling Cyclic Phase Change and Energy Storage in Solar Heat Receivers," *Journal of Thermophysics and Heat Transfer*, Vol. 12, June 1998, pp. 406–413.

[6] Kousksou, T., Bédécarrats, J. P., Dumas, J. P., and Mimet, A., "Dynamic Modelling of the Storage of an Encapsulated Ice Tank," *Applied Thermal Engineering*, Vol. 25, No. 10, July 2005, pp. 1534–1548. doi:10.1016/j.applthermaleng.2004.09.010

[7] Ismail, K. A. R., Henriquez, J. R., and Silva, T. M., "A Parametric Study on Ice Formation Inside a Spherical Capsule," *International Journal of Thermal Sciences*, Vol. 42, No. 9, Sept. 2003, pp. 881–887. doi:10.1016/S1290-0729(03)00060-7

[8] Arnold, D., "Dynamic Simulation of Encapsulated Stores, Part 1: The Model," *ASHRAE Transactions*, Vol. 96, 1990, pp. 1103–1110.

[9] Eames, I. W., and Adref, K. T., "Freezing and Melting of Water in Spherical Enclosures of the Type Used in Thermal (Ice) Storage Systems," *Applied Thermal Engineering*, Vol. 22, No. 7, May 2002, pp. 733–745. doi:10.1016/S1359-4311(02)00026-1

[10] Saitoh, T., and Hirose, K., "High-Performance Phase-Change Thermal Energy Storage Using Spherical Capsules," *Chemical Engineering Communications*, Vol. 41, Nos. 1–6, 1986, pp. 39–58. doi:10.1080/00986448608911711

[11] Laybourn, D. R., "Thermal Energy Storage with Encapsulated Ice," *ASHRAE Transactions*, Vol. 94, No. 1, 1988, pp. 1971–1988.

[12] Ismail, K. A. R., and Henriquez, J. R., "Numerical and Experimental Study of Spherical Capsules Packed Bed Latent Heat Storage System," *Applied Thermal Engineering*, Vol. 22, No. 15, Oct. 2002, pp. 1705–1716. doi:10.1016/S1359-4311(02)00080-7

[13] Bejan, A., *Entropy Generation Minimization*, CRC Press, Boca Raton, FL, 1996.

[14] Donnadieu, G., "Transmission de la Chaleur dans les Milieux Granulaires," *Revue de l'Institut Français du Pétrole*, Vol. 16, No. 9, 1961.

[15] Gunn, D. J., "Transfer of Heat or Mass to Particles in Fixed and Fluidized Beds," *International Journal of Heat and Mass Transfer*, Vol. 21, No. 4, Apr. 1978, pp. 467–476. doi:10.1016/0017-9310(78)90080-7

[16] Bird, R. B., Stewart, W. E., and Lightfoot, E. N., *Transport Phenomena*, Wiley, New York, 1960, Chap. 6, pp. 196–200.