Heat Transfer in Practical-Scale Thermal Energy Storage Tanks Using Carbon Fibers
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ABSTRACT

Latent heat thermal energy storage tanks, where carbon-fiber brushes are inserted to improve the heat transfer rates in the phase change materials, are installed in an air-conditioning system of a building as a space heating resource. The measured outlet fluid temperatures are compared with the numerical ones predicted by the previously developed three-dimensional heat transfer model. The numerical results have unallowable prediction errors, which probably result from poor contact between the brushes and the heat transfer tubes due to an installation problem of the brushes. However, the numerical results predicted by a corrected model agree well with the experimental ones under various operating conditions. The effect of the brushes on the thermal outputs of the tanks is then investigated using the corrected model. The result shows that the brushes contribute to saving space and cost of the tanks.

KEYWORDS
Latent heat thermal energy storage, Carbon fiber, Numerical, Experiment

INTRODUCTION

Thermal energy storage systems play an important role in solving many energy conservation programs (Hasnain, 1998, Dincer and Rosen, 2002, Dincer, 2002, Zalba et al., 2003). Accordingly, phase change materials (PCMs) have lately attract great interest as thermal storage materials because the thermal energy is stored with a high density. A typical phase change material is water/ice. The latent heat thermal energy storage (LHTES) systems using water/ice, i.e., an ice storage system, are commonly used for peak shifts in electrical demand. In this system, the thermal energy stored as ice using surplus electric power in the nighttime is released for the space cooling of buildings in the daytime. Many types of the ice storage tanks have been developed and have become widespread. On the other hand, the LHTES systems for space heating are not widely used. One of the reasons is because the PCMs whose melting points are above room temperature are more expensive than water. However, the LHTES systems for high temperature levels will be demanded for the efficient utilization of renewable energy, waste heat and so on.

The most important technical problem when using the PCMs is their low thermal conductivities (Jong and Hoogendoorn, 1981, Sasaguchi et al., 1986, Hasnain and Gibbs, 1988, Hoogendoorn and Bart, 1992, Lacroix, 1993, Zhang and Faghri, 1996, Valraj et al., 1999, Py et al., 2001, Fukai et al., 1997, 2000). The low thermal conductivity frequently makes it impossible to obtain a designed thermal output. To solve this problem, the authors noticed the high thermal conductivity of pitch-derived carbon fibers, and then developed brushes made of carbon fibers as an additive (Fukai et al., 2000). They packed the brushes with paraffin wax into shell-and-tube heat exchangers on a laboratory scale, and experimentally and theoretically demonstrated that the brushes essentially improve the thermal characteristics of the equipment (Fukai et al., 2002, 2003, Hamada et al. 2003).
The purpose of this paper is to investigate the effect of the carbon-fiber brushes on the thermal outputs of the practical-scale LHTES tanks, which are installed in an air-conditioning system of a building as a resource of space heating. The experimental investigation using the practical-scale equipment is limited because the operating conditions are not arbitrarily established contrary to the laboratory-scale equipment. Accordingly, the thermal outputs calculated using the previously reported three-dimensional heat transfer model (Fukai et al., 2003) are compared with the experimental ones. The effect of the carbon-fiber brushes on the thermal outputs of the tanks is then numerically discussed.

**EXPERIMENTAL PROCEDURE**

Figure 1 shows the flow diagram of the air-conditioning system for the Kyudenko R&D Institute of Kyudenko Co., Ltd. (three floors, reinforced concrete structure, total floor area = 3000 m²). This system principally consists of two latent heat thermal energy storage tanks (LHTES tank A and B), an ice storage tank, a heat pump and some heat exchangers. The heat pump supplies cool water to the ice storage tank using nighttime power. Hot water heated by the exhaust heat of the heat pump is simultaneously pumped into tanks A and B. As a result, this system provides energy conservation when both cooling and heating are in demand. An individual operation is also possible. The heat operation using LHTES tanks is only mentioned below because the present paper focuses on it.

Thermal energy is stored in the two tanks from 22:00 to 8:00. The total flow rate of water, $V_{tot}$, as the heat transfer fluid was constant. The tanks started releasing the thermal energy at 8:00 for heating. The fluid supplied to the load, and the fluid returning branch off at point BP and join at point JP. The fluid pumped into the LHTES tanks was heated while one flowing in another line (bypass) was not heated. To keep the fluid at a constant temperature (= 43 °C) at point CP, the flow rate of water passing through the bypass was controlled using valve V. This valve was closed when the temperature of the fluid from the LHTES tanks was below 43 °C. The thermal energy release process proceeded until the outlet fluid temperature of the LHTES tanks decreased to a preset temperature (= 35 °C). The flow rates and temperatures of the fluid were measured using electromagnetic flowmeters (accuracy: ±1%) and resistance bulbs (accuracy: ±0.1°C), respectively, every five minutes.

The schematic diagram of the LHTES tank is shown in Fig. 2. A heat exchanger consists of 14 paths and 18 columns of heat transfer tubes (27.2-mm inner diameter and 23.6-mm outer diameter) made of SC steel (density: $\rho_t = 7850$ kg/m³, specific heat: $c_{p,t} = 465$ J/kg K, thermal conductivity: $k_t = 43$ W/m K). Two heat exchangers were placed side by side in a tank (W2.29 m × L4.55 m × H2.05 m). The carbon-fiber brushes with the diameter $d_b = 110$ mm were inserted in the whole space on the shell side.
along the tubes. The thermal conductivity and the diameter of the carbon fibers are 190 W/m K and 10 µm, respectively. Paraffin wax (melting temperature = nearly 49 °C, latent heat = $1.8 \times 10^5$ J/kg) as the thermal storage material filled the tanks to a height of 1840 mm in the liquid state. The thermal conductivities of the paraffin wax are 0.21 W/m K in the solid state and 0.12 W/m K in the liquid state.

The photograph of the carbon-fiber brushes is shown in Fig. 3. The brushes are manufactured by twisting two wires between which the carbon fibers are vertically aligned. The brushes with $X_{fa} = 0.4$ and 0.8 vol.% fibers were provided and inserted into tanks A and B, respectively. As a result, the volume fractions of the fibers to the PCM (15.5 m³) are 0.5 vol.% for tank A and 1.0 vol.% for tank B because of overlapping of the brushes.

The flow modes are changeable by switching the exterior line connecting two sets headers H1 and H2 as shown in Fig.4:

1. mode 1b: the fluid was supplied from the bottom headers to the top headers
2. mode 1t: the fluid was supplied from the top headers to the bottom headers
3. mode 2b: the exterior line between H1 as the top and H2 at the bottom are connected. As a result, the fluid flowed in H1 at the bottom and out H2 at the top.

The results for mode 1b will be shown because there were little differences among the thermal characteristics for these modes.
NUMERICAL PROCEDURE

The accuracy of the mathematical model three-dimensionally describing the heat transfer in shell-and-tube heat exchangers was demonstrated by comparing it to the laboratory-scale experiments (Fukai et al., 2003). A feature of this model is that the cross sections of the tubes are assumed to be square. Another is that the anisotropic heat transfer caused by the brushes is simply modeled. These techniques result in mesh-size independence solutions with fewer meshes and a shorter CPU time than the model where they are accurately modeled. The numerical procedure is briefly mentioned below because its details are found in previous reports (Fukai et al., 2002, 2003).

A three-dimensional calculation domain is schematically described in Fig. 5. Assuming the adiabatic surfaces of the tanks, the periodicity of the tube arrangement on the z-x plane reduces the computational domain to the area of 105 mm × 4360 mm × 1840 mm, where two columns of the heat transfer tube are included. The headers are neglected because their heat transfer area is much smaller than that of the heat transfer tubes.

The three-dimensional energy conservation equations in the brush/PCM composite (c) and the tube wall (t) are given by

\[
\frac{c_p \rho}{T_c} \frac{\partial T_c}{\partial t} = \nabla \cdot (k_c \nabla T_c) \quad (1)
\]

\[
\frac{c_p \rho}{T_t} \frac{\partial T_t}{\partial t} = \nabla \cdot (k_t \nabla T_t) \quad (2)
\]

where \(T\) is the temperature, and \(t\) the time. The thermal conductivity tensor \(k_c\) in Eq. (1) is given based on a simple model (Fukai et al., 2003). The convective heat transfer term does not appear in Eq. (1) because the brushes obstruct the natural convection. For the heat transfer fluid \(h\), the following one-dimensional conservation equation is given by:

\[
\left( c_p \rho \right)_h \left\{ \frac{\partial T_h}{\partial t} + u_h(t) \frac{\partial T_h}{\partial s} \right\} = \frac{\partial}{\partial s} \left( k_h \frac{\partial T_h}{\partial s} \right) \quad (3)
\]

where \(s\) is the coordinate along the center of the tube measured from the inlet of the tube, \(u_h\) the mean fluid velocity, and \(T_h\) the bulk temperature of the fluid. Eq. (3) is independently adapted to the fluid.
flowing in the two tubes.

The initial conditions are
\[ T_j = T_0 \quad \text{at } t = 0 \quad (j = c, t, h) \quad (4) \]

The boundary conditions are given by:
\[ T_h = T_{in}(t) \quad \text{at the inlets of the tubes} \quad (5) \]
\[-k_i \frac{\partial T}{\partial n} = h_i(T_i - T_h) \quad \text{on the tube-wall/fluid interface} \quad (6)\]
\[ T_c = T_i , \quad k_i \frac{\partial T}{\partial n} = k_i \frac{\partial T}{\partial n} \quad \text{on the composite/tube-wall interface} \quad (7) \]
\[ \frac{\partial T_j}{\partial n} = 0 \quad \text{on the exposed surface } (j = c, t) \quad (8) \]
\[ T_c|_{\kappa=0} = T_{\infty}|_{\kappa=0} , \quad k_c \frac{\partial T_c}{\partial \kappa} |_{\kappa=0} = k_c \frac{\partial T_c}{\partial \kappa} |_{\kappa=\infty} \quad \text{on the periodic surface} \quad (9) \]

where \( n \) is the coordinate normal to the boundary surface, and \( h \) the heat transfer coefficient of the heat transfer fluid.

The heat capacity and thermal conductivity of the composite is given as a function of the local volume fraction of the fibers \( X_f(r) \). The local volume fraction \( X_{fi}(r) \) for the \( i \)-th brush is approximated by (Fukai et al., 2000)
\[
\begin{cases} 
1 & \text{at } 0 \leq 2r / d_b \leq X_{fa} / 2 \\
X_{fa} d_b / 4r & \text{at } X_{fa} / 2 < 2r / d_b \leq 1 \\
0 & \text{at } 2r / d_b > 1
\end{cases}
\]

(10)

where \( d_b \) is the diameter of the brush, and \( r \) the local radial distance from the center of the brush. Considering that a part of a brush overlaps other brushes, the local volume fraction of the fibers \( X_f(r) \) is generally calculated from the sum of \( X_{fi}(r) \):
\[ X_f(r) = \sum_i X_{fi}(r) \quad (11) \]

The mean fluid velocity \( u_0(t) \), the initial temperatures \( T_0 \) and the inlet temperatures \( T_{in}(t) \) are given based on the experiments. The governing equations are numerically solved using the control volume method (Patankar, 1980). The number of typical meshes is \( 18 \times 36 \times 229 \) and the time step is 20 s. The thermal storage/release energy from the tanks is calculated from the difference between the inlet and outlet fluid temperatures. Another value is calculated from the time variation in the thermal energy within the computational domain. They agreed within 0.1 % for each time step.

RESULTS

Comparison between calculations and experiments

A typical experimental result during the discharge process for mode 1b is shown in Fig. 6. \( T_{out} \) is the outlet fluid temperature. The releasing rate of the thermal energy, \( q_{dis} \), generally shows a peak at the initial stage because the space in the building cooling during the nighttime has to be quickly heated. However, the mean fluid velocity of water flowing in the tanks is not high at this stage because of the sufficient difference between the outlet and inlet fluid temperatures. The mean fluid velocity increases as the temperature difference decreases, and finally reaches a maximum preset.

For the laboratory-scale thermal energy storage tank whose structure was almost the same as the present tanks, the present model predicted well the thermal output (Fukai et al., 2003). However,
unallowable prediction errors are found in the numerical $T_{\text{out}}$ and $q_{\text{dis}}$ as shown in Fig. 6 when the diameter of the actually installed brushes is given, i.e. $d_b = 110$ mm. These prediction errors may be due to incomplete contact between the tubes and the fibers. That is, the brushes are fixed along the tubes by pulling the wires of the brushes from both sides of the frames. For the laboratory-scale tanks, the tension of the wire is enough strong to locate the wire nearly at the center of the four tubes because the brushes (40-mm diameter and 450-mm length) are short and lightweight. On the other hand, for the practical-scale tanks, the tension is too weak to do this because the brushes (110-mm diameter and 2000-mm length) are longer and heavier, resulting in a longitudinal warp in the wires due to their own weight. Such a warp produces a heat transfer surface where the brush tips do not attach. Because it is difficult to construct a model considering such a structural defect, the diameter of the brushes $d_b$ is chosen as an empirical parameter.

The value of $d_b$ fitting the numerical results to the experiments was found using the trial-and-error technique. As a result, the values of $d_b = 75$ - 80 mm were chosen as demonstrated in Fig. 6. In addition, the accumulations of the thermal release energy defined by Eq. (12) are compared in Fig. 7 to evaluate the accumulative errors with time;

$$Q_{\text{dis}}(t) = \int_0^t q_{\text{dis}}(t) \, dt \quad (12)$$

The error bar in the figure indicates the experimental errors considering the accuracy of the electromagnetic flowmeters ($\pm 1\%$) and the resistance bulbs ($\pm 0.1$ K). The values of $d_b = 75$ - 80 mm are found to be reasonable results though the optimum value might strictly depend on the tanks. Fig. 8 compares the calculations for $d_b = 80$ mm with the experiments for other runs. The figure shows that the value of $d_b = 80$ mm is applicable for the present equipment used here, being independent of the load conditions, the volume fractions of the fibers and the flow modes.

Fig. 6 The time variations in the mean fluid velocity and the outlet fluid temperatures during discharge. Comparison between the experiments and the calculations ($X_{fa} = 0.008$, mode 1b).

Fig. 7 Effect of the brush diameter on the accumulation of the thermal release energy during discharge. Comparison between the experiments and the calculations (mode 1b).
The distances between the brush tips and the tube walls for $d_b = 80$ mm is 2 mm. The corresponding interfacial heat transfer coefficients are 100 W/m² K. This value is much lower than that for the laboratory-scale equipment (= 340 W/m² K) (Hamada et al., 2003). The numerical results for no carbon fibers ($d_b = 0$) are plotted in Figs. 6 and 7. A comparison between the numerical results for $d_b = 0$ and $d_b = 80$ mm shows that the brushes efficiently work during the energy discharge processes, nevertheless, the interfacial heat transfer coefficient is low.

The free conventional term is not considered in the conservation equation in the liquid phase (Eq. (1)). In spite of this fact, the present model predicts well the experiments because the free convection heat transfer does not play an important role during the solidification process, i.e., the discharge process. Contrary to this process, it plays an important role during the melting process, i.e., the charge process. The numerical and experimental results during the charge process are compared in Fig. 9. The calculated results agree well with the experimental before 120 min probably because the PCM has not yet fully melted. After that, the difference between the numerical $Q_{ch}$ and experiment increases

![Fig. 8 Comparison between the experiments and the calculations under various operating conditions during discharge.](image)

![Fig. 9 Time variations in the outlet fluid temperatures and the accumulation of the thermal storage energy during charge. Comparison between the experiments and the calculations ($d_b = 80$ mm, mode 1b).](image)
with time. Although the carbon fibers are concluded to obstruct the natural convection in the laboratory-scale thermal energy storage tanks (Fukai et al., 2003), the natural convection seems to contribute to the heat transfer in the large equipment. Consequently, the present model is applicable to the discharge process, but not to the charge process on a practical scale.

**Effect of brushes on thermal outputs**

The effect of the carbon-fiber brushes on the thermal outputs was discussed based on the experiments where \( X_{fa} \) was chosen as a parameter. However, it is impossible to experimentally have the same discussion for the practical-scale tanks because the tanks using no brushes have to wastefully be constructed to obtain reference data. Although the usefulness of the brushes is shown in Figs. 6, 7 and 8, detail discussions during the discharge process are numerically done. In the calculation, the releasing rate of the thermal energy, \( q_{dis} \), is assumed to maintain a constant. To satisfy this assumption, \( q_{dis} \) is calculated from the difference between the inlet and outlet fluid temperatures at each time-step and then the mean fluid velocity of the heat transfer fluid is changed to yield the preset \( q_{dis} \). The dimensions of the equipment are the same as the experiments.

The time variations in the flow velocities, the outlet fluid temperatures and the thermal release energy are shown in Figs. 10 \( (q_{dis} = 42 \ \text{kW}, \ T_{h,in} = 35 \ ^\circ\text{C}, \ d_b = 80 \ \text{mm}, \ \text{mode 1b}) \) and 11 \( (q_{dis} = 63 \ \text{kW}, \ T_{h,in} = 35 \ ^\circ\text{C}, \ d_b = 80 \ \text{mm}, \ \text{mode 1b}) \). The mean fluid velocity increases to maintain a constant \( q_{dis} \) as the outlet fluid temperature decreases. It is obvious that an increase in \( X_{fa} \) extends the time at which \( T_{out} \) fully declines. The phase change of the PCM causes the time period where \( T_{out} \) is maintained a constant. \( T_{out} \) at this period increases as \( q_{dis} \) increases. This is because a thinner solid layer around the tubes is required to yield higher heat transfer rates, approaching the temperature of the tube wall to the interface temperature, i.e., the melting temperature \( \approx 49 \ ^\circ\text{C} \), due to a lower overall thermal resistance of the solid layer.

\( T_{out} \) generally limits the thermal energy effectively used. Thus, the effective thermal energy, \( Q_{eff} \), released until \( T_{out} \) reaches a critical temperature,
$T_{\text{eff}}$, is computed from the numerical results. A $T_{\text{eff}}$ value of 40 °C is assumed according to the actual operating condition. Fig. 12 shows the relationship between $Q_{\text{eff}}$, normalized with the total thermal energy, $Q_{\text{tot}}$, stored in the tank, and the releasing rate $q_{\text{dis}}$. $Q_{\text{tot}}$ is defined as the thermal energy held by the PCM in the range between $T_{\text{eff}}$ and the initial temperature $T_0$ (= 53 °C). $Q_{\text{tot}}$ is 1.86 GJ under the present condition ($T_{\text{eff}} = 40$ °C and $T_0 = 53$ °C). As easily expected, $Q_{\text{eff}}/Q_{\text{tot}}$ decreases with increasing $q_{\text{dis}}$. The difference between the calculated curves for different values of $X_{fa}$ becomes as high as $q_{\text{dis}}$ increases because of improving the effective thermal conductivity of the solid phase. $Q_{\text{eff}}/Q_{\text{tot}}$ in a high-$q_{\text{dis}}$ region is not saturated but gradually decreases. In this region, a high flow rate of the fluid compensates a low heat exchange rate to keep a high $q_{\text{dis}}$. Consequently, the numerical results in such a region are not reality because there is a limit in the flow rate in the practical operation.

The average thermal output demanded in the present air-conditioning system was about 40 kW as shown in Fig. 6. $Q_{\text{eff}}/Q_{\text{tot}} > 80\%$ may be generally required for the design of the tanks. For this restriction, $q_{\text{dis}}$ for no brushes is no more than 24 kW while the tanks for $X_{fa} = 0.008$ is satisfied with this request. Accordingly, a couple of tanks must be roughly stood in a row when no brushes are used. When there is no request on $Q_{\text{eff}}/Q_{\text{tot}}$, $Q_{\text{eff}}/Q_{\text{tot}}$ for no brushes is 0.4 at $q_{\text{dis}} = 40$ kW while that for $X_{fa} = 0.008$ is 0.8. In this case, a couple of tanks for no brushes have also been developed to obtain the same total thermal energy as $X_{fa} = 0.008$. Consequently, the brushes save a space where the tank is installed. With respect to cost performance, the carbon fibers are expensive. However, the brushes with $X_{fa} = 0.008$ brings about the 75 % cost of the tank using no brushes, including those of brushes and construction, though the details are not described here.

**CONCLUSIONS**

A three-dimensional heat transfer model, which is applicable to laboratory-scale equipment, does not provide a good prediction for the present practical-scale equipment. This may be relevant to production control regarding installation of the brushes. Using spacers or other methods will probably solve this problem. In this study, the model is corrected by choosing the diameter of the brushes as an empirical parameter. The corrected model predicts well the thermal outputs of the tanks under various experimental conditions. The effect of the brushes on the thermal outputs is discussed using the numerical results.
Consequently, the brushes remarkably improve the thermal outputs of the thermal energy storage tanks, resulting in reduction in cost and space.

REFERENCES