Heat Transfer Characteristics of Latent Heat Thermal Energy Storage for Floating Concentrated Solar Power Systems

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Abstract. The heat transfer characteristics of the latent heat thermal energy storage (LHTES) for floating concentrated solar power (CSP) systems were investigated experimentally and numerically. The understanding of natural convection heat transfer between the heat transfer fluid (HTF) and a tube bundle in the heat exchanger is important for the thermal design of the LHTES. As a primitive study, the analytical model of natural convection heat transfer was developed between the HTF and the tube bundle in the heat exchanger of LHTES. In this study, D-mannitol was used as the HTF for the LHTES. To simplify the tube bundle of the heat exchanger, a heated cylinder was installed in the liquid D-mannitol. In the experiment, the average temperature of the cylinder and the heat flux were measured, and the heat transfer coefficient under the natural convection was obtained. Meanwhile, the numerical analysis of natural convection heat transfer was conducted using the commercial computational fluid dynamics (CFD), PHOENICS. The temperature and velocity profiles around the heated cylinder were obtained in the numerical simulation. The heat transfer coefficient was compared with the author's experimental data. The numerical result was in good agreement with the measured heat transfer coefficient.

1. Introduction
The technology of concentrated solar power (CSP) systems have been developed [1,2]. Diendorfer et al. [3] proposed the floating offshore parabolic trough collector system. By applying the vertical collector system in the offshore area, the thermodynamic efficiency would increase due to the huge cooling seawater. Since their work was focused on the floating model and the optical performance as a function of time and location without the thermal energy storage (TES), the fundamental research of the TES was scarce for the CSP system. The TES is categorized as sensible heat systems and latent heat systems [4]. In general, the molten salts are applied for the TES of CSP systems. Since the temperature of molten salts is high, it is difficult to operate the TES. For example, the rate of mixture is 40 wt.% of KNO₃ to 60 wt.% of NaNO₃ as solar salts. The temperature of solar salts is ranged from 563 K to 823 K. Since the waste heat recovery of the CSP is needed below the melting temperature of molten salts (approximately 511K), the latent heat system using phase change material (PCM) is considered [5].

The latent heat thermal energy storage (LHTES) using PCMs is also implemented in various waste heat resources such as power plants, EVs and marine transportations, and industrial applications. The PCM depends on the fusion temperature and latent heat. High-temperature PCMs are utilized as energy conservation technologies. PCMs can be categorized as alkali hydroxide, alkali nitrate, hydrate salt,
organic material, and sugar alcohol [6-7]. As the PCM candidates for the LHTES of CSP, D-mannitol is an attractive material since the latent heat is relatively higher than the other materials in medium temperature ranges. Therefore, it is important to understand the heat transfer process of liquid D-mannitol for the design of the LHTES recovering waste heat from the CSP system. Shibahara et al. [8] investigated the heat transfer characteristics of the PCM such as a hydrate salt and sugar alcohol. As the PCM candidates for waste heat recovery of ship, the sodium acetate trihydrate and the D-mannitol were used since these materials have high latent heat. They clarified that the effect of heat inputs on the melting process of the PCM. Moreover, they researched the natural convection heat transfer of D-mannitol which is one of sugar alcohol for medium TES [9]. They measured the transient heat transfer coefficient of liquid D-mannitol at various liquid temperatures and exponential periods of heat generation rate. Then, the boiling incipience of liquid D-mannitol around the heated cylinder was investigated experimentally [10]. Although the effects of liquid temperatures and exponential periods on the natural convection and boiling heat transfer of D-mannitol were revealed in previous studies [9-10], the temperature and velocity profiles around the heated cylinder were not clarified numerically because the thermal properties of D-mannitol were not enough to conduct the numerical simulation. In this study, the numerical simulation of natural convection heat transfer for liquid D-mannitol was conducted using the measured thermal properties of D-mannitol. Moreover, the numerical result was compared with the experimental data. The purposes of this study are to clarify the heat transfer characteristics of D-mannitol numerically and to develop the analytical model of D-mannitol.

2. Experimental method

2.1. Experimental setup

Figure 1 shows the schematic of the experimental setup. The experimental apparatus is composed of a D.C power source, a standard resistance ($R_s$), a double bridge circuit, a silicon oil bath, a glass vessel, a platinum cylinder, a thermocouple, and a stirrer. In the experiment, the D-mannitol was heated in the silicon oil bath, and the temperature of D-mannitol was measured by the thermocouple. For visualization of the experiment, the video camera was set outside of the oil bath. The platinum cylinder (heater) was connected to the copper electrode, and it was annealed before using it in the experiment. Moreover, the electrical resistance ($R_f$) vs. the averaged temperature of the heater ($T_o$) was calibrated as follows:

$$R_T = R_0 (1 + c_1 T_o - c_2 T_o^2),$$

where $R_0$, $c_1$ and $c_2$ are the electric resistance at 0 °C, coefficient of Eq. (1), respectively.

Figure1. Experimental apparatus.
2.2. Measurement system and experimental conditions

The averaged temperature of the cylinder was calculated from \( V_T \) and \( I \) \((=V/I/R_s)\), based on the principle of Kelvin double bridge, as follows:

\[
R_T = R_2 + R_3 \frac{V_T}{R_s} + \frac{R_1 R_3}{R_2}.
\]

(2)

where \( V_T \) and \( I \) are the unbalanced voltage and the direct current, respectively. In the experiment, the heater was heated by an electric current from the D.C. power source \((=V_s I)\) after the liquid temperature was controlled by the silicon oil bath.

The heat flux can be expressed by the energy balance as follows:

\[
q = \frac{d}{4} \left( \dot{Q} - \rho_h c_h \frac{\dot{m}_h}{\dot{V}} \right),
\]

(3)

where \( d \), \( \dot{Q} \), \( \rho_h \), and \( t \) denote the diameter of the cylinder, the heat generation rate of the heater, density of the heater, and time, respectively. The surface temperature can be calculated from the unsteady heat conduction equation with boundary conditions as follows:

\[
\frac{\partial T}{\partial t} = \alpha \left( \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right) + \frac{\dot{Q}}{\rho_h c_h},
\]

(4)

\[
\frac{\partial T}{\partial r} \bigg|_{r=0} = 0,
\]

(5)

\[
-\lambda \frac{\partial T}{\partial r} \bigg|_{r=d/2} = q,
\]

(6)

\[
T_a = \frac{8}{d^2} \int_0^{d/2} T \, dr.
\]

(7)

The density of liquid D-mannitol was measured by the Archimedes’ principle as follows:

\[
\rho = \frac{m_{\text{air}} - m_{\text{liquid}}}{V} + \rho_{\text{air}},
\]

(8)

where \( m \) and \( V \) are a weight of float and volume of float, respectively. \( \rho_{\text{air}} \) is a density of air. The measurement result of density is shown in Figure 2. The density of liquid D-mannitol decreased in an increase of temperature. Based on the measurement result, the empirical correlation of density was obtained by the least squared method as follows:

\[
\rho = 1.478 \times 10^3 + 4.415 \times 10^{-1} T - 1.618 \times 10^{-3} T^2.
\]

(9)

The thermal expansion coefficient can be calculated by the following equation:

\[
\beta = -\frac{1}{\rho} \frac{\partial \rho}{\partial T}.
\]

(10)

The thermal conductivity of the D-mannitol was measured by the laser flash method (ULVAC-RIKO, TC-7000). To measure the specific heat of D-mannitol, the adiabatic scanning calorimetry was used (SHINKU-RIKO, SH-3000M). The experimental condition and the thermal properties of D-mannitol are summarized, as shown in Table 1. The experiment was carried out with various liquid temperature and exponential periods.

![Figure 2. Relationship between the density and the liquid temperature.](image-url)
Table 1. The experimental conditions and the thermal properties of D-mannitol.

| Experimental conditions | Thermal properties |
|-------------------------|--------------------|
| Heater configuration    | Cylinder           |
| Material                | Platinum           |
| Diameter \((d)\)        | 1.0 [mm]           |
| Effective length \((L)\)| 25.9 [mm]          |
| Exponential periods \((\tau)\)| 11.52 [s] |
| Liquid Temperature \((T_L)\)| 477.15 [K] |
| Test fluid              | D-mannitol         |
| Density \((\rho)\) at 470 K | 1325 [kg/m³] |
| Specific heat \((c_p)\) at 460 K | 2690 [J/kgK] |
| Thermal conductivity \((\lambda)\) at 420 K | 0.347 [W/mK] |
| Kinetic viscosity \((\nu)\) at 460 K [11] | 2.84×10⁻⁵ [m²/s] |
| Thermal expansion coefficient \((\beta)\) | 6.68×10⁻⁴ [1/K] |

3. Numerical method

3.1. Analytical model and boundary conditions

The analytical model is shown in Figure 3. The heated cylinder was arranged in the liquid D-mannitol. The diameter of the cylinder was 1.0 mm, and the effective length of the cylinder was 25.9 mm. The heat flux from the cylinder was set to a uniform heat flux with \(q = q_0 \exp(t/\tau)\) as the boundary condition. Based on the comparison of sensitivity analysis for the computational grid, the number of the computational grid for \(r, \theta,\) and \(z\) was 24, 20, and 25, respectively. In order to treat the near-wall condition between the solid region (cylinder) and fluid region (D-mannitol), the first mesh from the cylinder, \(\Delta r,\) was arranged in \(\Delta r < 144 \mu m.\)

![Figure 3. Analytical model in the numerical simulation.](image)

3.2. Numerical method

The numerical analysis of natural convection heat transfer was conducted using the commercial CFD code, PHOENICS ver.2013. To solve the pressure and velocity fields, the SIMPLE algorithm was applied in the CFD code. Governing equations of mass, momentum, and energy were discretized by finite volume (FVM) method in cylindrical geometry. Since this analytical model was a natural convection problem, the momentum equation was taken into account for the buoyancy term using the Boussinesq approximation. The time step was set to 1 s and the initial temperature was the same as the experiment for the transient analysis.

The surface temperature can be expressed by solving Fourier’s law as follows:

\[
T_S = \frac{q \Delta r}{2 \lambda} + T_{cel} \tag{11}
\]

where \(T_{cel}\) is the cell temperature of the first mesh on the cylinder. Since \(T_i\) is influenced by \(\Delta r\) in Eq. (11), the size of \(\Delta r\) was selected comparing the numerical result with the experimental data. It was 0.15 mm, approximately.
4. Experimental and numerical results

Figure 4 shows the relationship between heat flux and the surface temperature differences \((T_s - T_L)\) at the liquid temperature of 477.15 K. The symbol and dashed line were the experimental data and the numerical result. As shown in Figure 4, the numerical result is in good agreement with the experimental data. The surface temperature differences increased with an increase in heat flux. Figure 5 shows the relationship between the heat transfer coefficient and the heat flux. The heat transfer coefficient was defined as \(h = q/(T_s - T_L)\). The heat transfer coefficient slightly increased as the heat flux increased.

![Figure 4](image1.png)

Figure 4. Relationship between the heat flux and the surface temperature differences.

![Figure 5](image2.png)

Figure 5. Relationship between heat transfer coefficient and heat flux.

Figure 6 displays the temperature profiles in the liquid D-mannitol. The bulk temperature of the liquid D-mannitol was 204 K in the numerical simulation. As shown in Figure 6, the liquid D-mannitol was heated by the cylinder with the uniform heat flux, and the temperature increased with time. Since this heat transfer process was natural convection, the plum of liquid D-mannitol was formed as the heated time proceeded.
In order to understand the heat transfer characteristics between the heated cylinder and the liquid D-mannitol, the temperature distribution around the heated cylinder was illustrated in Figure 7. The analytical time was ranged from 10 to 30 s. $\theta=0^\circ$ and $\theta=180^\circ$ are the top and the bottom of the cylinder, respectively. As shown in Figure 7, it was understood that the temperature decreased as $\theta$ increased from $0^\circ$ to $90^\circ$. On the other hand, the temperature increased when $\theta$ was ranged from $270^\circ$ to $360^\circ$ due to the natural convection as mentioned above. Since the plum of natural convection was developed with the heated time, the temperature gradient of 30 s is higher than that of 10s. Moreover, each temperature was uniform at the range of $\theta$ from $90^\circ$ to $270^\circ$.

Figure 8 shows the velocity profiles at various times. The velocity derived from natural convection exists on the heated cylinder. As the heated time proceeded, the velocity increased around the cylinder. Since the velocity at the bottom of the cylinder was lower than that of the top, it was understood that the temperature distribution was uniform at the range of $\theta$ from $90^\circ$ to $270^\circ$ as shown in Figure 7.
5. Conclusions
The numerical simulation of natural convection heat transfer for liquid D-mannitol was conducted using the measured thermal properties of D-mannitol. Moreover, the numerical result was compared with the experimental data. The temperature and velocity profiles around the heated cylinder were obtained in the numerical simulation. The heat transfer coefficient was compared with the author’s experimental data. The numerical result was in good agreement with the measured heat transfer coefficient. The natural convection around the cylinder in liquid D-mannitol was clarified.

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