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The impact of refrigerant inlet temperature on the ice storage process in an ice-on-coil storage plate

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Abstract

The ice-on-coil storage plate is one of the core devices in the latent heat cold storage system, which benefits both energy and operation cost savings in refrigerated warehouses, especially under the peak-valley price mechanism. Although the structure design of coils and the selection of cold storage agent materials have attracted much research attention, very few research has studied the impact of refrigerant inlet temperature on the thermal performance of an ice-on-coil storage plate. To facilitate the application of ice-on-coil storage plates in the latent heat cold storage system and improve the performance, this study developed a detailed three-dimensional CFD model in ANSYS to investigate the ice storage process of the ice-on-coil storage plate with different refrigerant inlet temperatures. The results showed that a lower refrigerant inlet temperature contributes to a higher heat exchange efficiency, a bigger ice thickness in the same period of time, and a shorter time required for ice storage. And a lower refrigerant inlet temperature can also lead to a higher cold storage rate and cold storage capacity.

Keywords: Ice-on-coil storage plate; Ice storage; Inlet temperature; CFD; Modelling and simulation

1. Introduction

Latent heat cold storage (LHCS) is the technology to store low-temperature energy temporarily for later use, and the storage cycle, depending on requirements, can be daily, weekly or seasonal [1]. It is the supplement and adjustment

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of refrigeration technology and an economical and feasible way to coordinate the mismatch between the supply and demand of thermal energy in time and intensity. Under a peak-valley price mechanism, it is inevitable to apply latent heat cold storage to refrigerated warehouses.

The ice-on-coil storage plate is one of the core devices in the latent heat cold storage system, and the icing and melting performance is the key to the success of the system. A recent study compared the performance of two heat transfer enhancement methods including the usage of thin rings and annular fins around coils in an ice-on-coil storage plate; the results showed that ice formation would be correspondingly 21% and 34% higher with respect to the bare tube [2]. A numerical study conducted on a shell and tube latent heat storage system whereby the inlet heat transfer fluid direction was periodically reversed during charging and discharging resulted in a lower temperature gradient in space and time and consequently higher exergy recovery [3]. A versatile latent heat storage tank capable of working with organic phase change materials was presented recently, of which the enthalpy–temperature curve, the specific heat and density were also measured [4]. A recent study presented the results of an experimental investigation carried out on a tube-in-tank design filled with PCM for cold storage applications. From the experimental measurements, the average heat exchange effectiveness of the storage tank was determined and a characteristic design curve has been developed as a function of the measured average NTU [5]. Although the structure design of coils and the selection of cold storage agent materials have attracted much research attention, very few research has studied the impact of refrigerant inlet temperature on the thermal performance of an ice-on-coil storage plate. This paper aims to facilitate the application of ice-on-coil storage plates in the latent heat cold storage system and improve the performance. To achieve this aim, the paper first develops a detailed three-dimensional CFD model in ANSYS to investigate the ice storage process of the ice-on-coil storage plate with three different refrigerant inlet temperatures and then evaluate the thermal performance with several criteria.

2. Methodology

2.1. Assumptions

The key assumptions used to develop the model are summarized as follows:

(1) The impact of the material and the thickness of the coil on the heat transfer is ignored.

(2) There is no heat transfer between the ice-on-coil storage plate and the external environment; namely, the plate surface is adiabatic with no heat loss.

(3) The temperature of the cold storage agent in the ice-on-coil storage plate is well-distributed at the initial time.

(4) The thermophysical properties of the refrigerant and the cold storage agent are assumed as constant.

2.2. Governing equation

The method used for modelling the solidification/melting process in ANSYS is the enthalpy-porosity technique [6, 7]. In this approach, the temperature and the enthalpy are used to obtain the energy equation containing the two variables, and then to obtain the temperature distribution by the relationship between the temperature and the enthalpy.

The enthalpy model considers the whole calculation area as a porous medium, instead of tracking the phase change interface. And a quantity called the liquid fraction, which represents the fraction of the cell volume that is in liquid form, is associated with each cell in the domain [8]. The enthalpy of the material is computed as:

$$H = h + \Delta H \tag{1}$$

$$h = h_{ref} + \int_{T_{ref}}^{T} c_p dT \tag{2}$$

$$\Delta H = \beta L \tag{3}$$

where h is the sensible enthalpy; h_{ref} is the reference enthalpy; c_p is the specific heat; ΔH is the phase change enthalpy which varies between 0 (for a solid) and L (for a liquid); L is the fusion heat which refers to the latent heat property of the material, and β is the liquid fraction.

2.3. Physical model

In this study, an ice-on-coil storage plate with the dimension of $1 \times 0.5 \times 0.05$ m^3 was chosen as the research object. Fig. 1 presents the physical model of the ice-on-coil storage plate. The plate and the coil were filled with cold storage agent and refrigerant, respectively. Water was chosen as the cold storage agent and ethylene glycol in volume fraction of 30% the refrigerant. In the ice storage process, heat exchange happened between the ethylene glycol flowing through the coil and the water inside the plate.

Table 1. Thermophysical properties of the refrigerant at different inlet temperatures.

Temperature (K)	Density (kg/m^3)	Specific heat (J/kg K)	Thermal conductivity (W/m K)	Dynamic viscosity (Pa s)
263.15	1054.31	3560	0.415	0.00619
265.65	1053.71	3567	0.419	0.00561
268.15	1053.11	3574	0.422	0.00503

Table 2. Thermophysical properties of the cold storage agent.

Parameters	Values
Density (kg/m^3)	998.2
Specific heat (J/kg K)	4182
Thermal conductivity (W/m K)	0.6
Dynamic viscosity (Pa s)	0.001003
Pure solvent melting heat (J/kg)	333146
Freezing point (K)	273.15

A detailed three-dimensional CFD model of the half ice-on-coil storage plate was developed in ANSYS, because the plate can be divided into two same sections in the thickness direction. After the grid sizes of 846375, 1521570 and 2469153 cells and the time step sizes of 5, 1 and 0.2 s were tested for the independent verification, 1521570 cells and 1 s were chosen in this study. This study focused on the impact of refrigerant inlet temperature on the ice storage process in the ice-on-coil storage plate. So three different refrigerant inlet temperatures were studied, which were 263.15, 265.65 and 268.15 K, respectively. And the thermophysical properties of the refrigerant at different inlet temperatures and the cold storage agent are listed in Table 1-2.

2.4. Initial conditions and boundary conditions

In order to figure out the temperature distribution and the liquid fraction of the ice-on-coil storage plate during the ice storage process, a transient analysis was carried out. Designed test conditions were implemented for the simulation, where the inlet flowrate of the refrigerant and the initial temperature of the cold storage agent were set as 2 m/s and 274 K respectively.

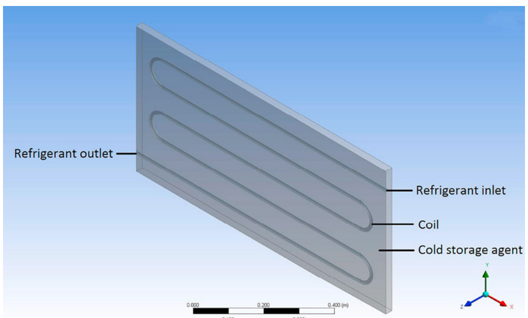

Fig. 1. Physical model of the ice-on-coil storage plate.

The turbulence specification method was set as intensity and hydraulic diameter, where the hydraulic diameter is 0.012 m and the turbulent intensity can be calculated by:

$$I = 0.16Re^{-\frac{1}{8}} \tag{4}$$

where Re is the Reynolds number that can be calculated by:

$$Re = \frac{vd\rho}{\mu} \tag{5}$$

where v is the inlet flowrate of the refrigerant; d is the hydraulic diameter; ρ is the density of the refrigerant, and μ is the dynamic viscosity of the refrigerant.

3. Results and discussion

According to China's peak-valley price policy, the valley period is 23:00-7:00, which is the charging time of a latent heat cold storage system. Therefore, this study implemented the simulation for 8 hours in a single day. The results are reported as follows.

3.1. Liquid fraction contours

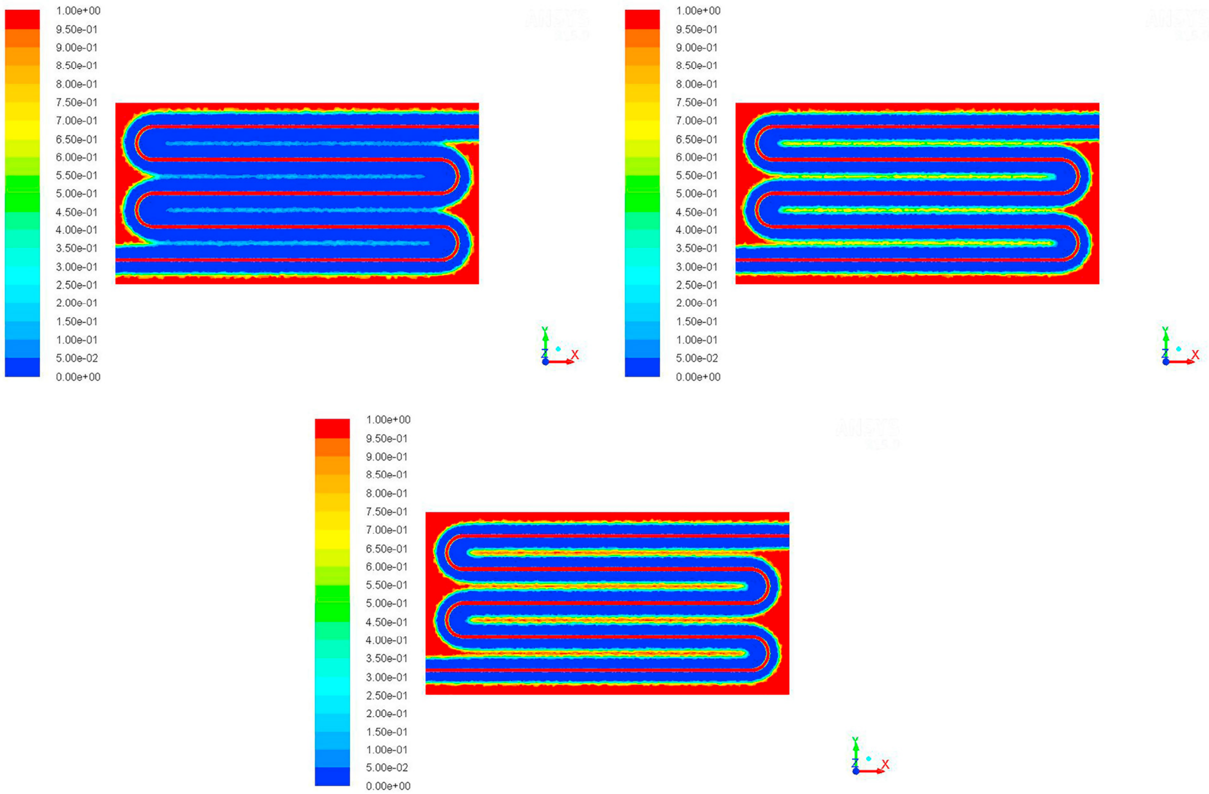

Fig. 2. Liquid fraction contours of (a) T_i=263.15 K; (b) T_i=265.65 K; and (c) T_i=268.15 K at 2400 s.

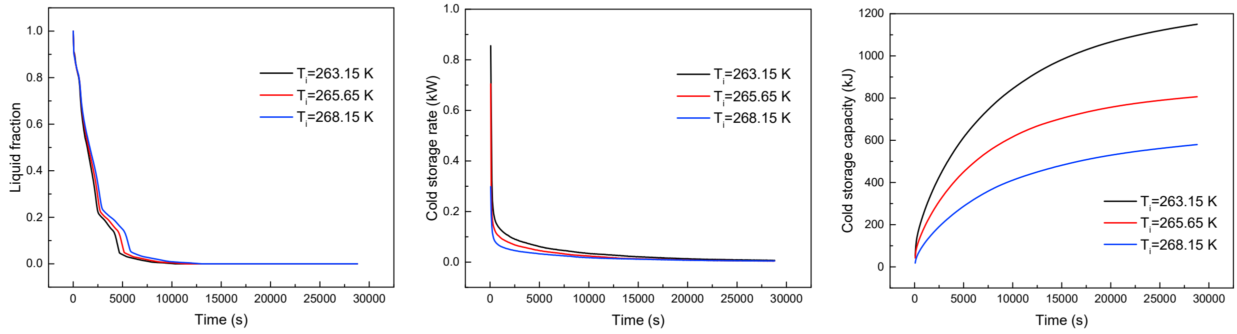

Fig. 3. (a) Liquid fraction; (b) Cold storage rate; and (c) Cold storage capacity of three different refrigerant inlet temperatures.

Fig. 2 shows the liquid fraction contours of T_i=263.15, 265.65 and 268.15 K at 2400 s. It is noted that the ice thickness of the low refrigerant inlet temperature is bigger than that of the high refrigerant inlet temperature. This is because that the lower inlet temperature of the refrigerant contributes to the higher heat exchange temperature difference, which causes the higher heat exchange efficiency, resulting in the bigger ice thickness in the same period of time. It can also be observed that the solid-liquid phase interface is constantly changing along the flow direction of the refrigerant, and the thickness of the ice layer outside the coil varies with different locations at the same time. Besides, the ice thickness near the inlet is slightly bigger than that near the outlet, due to the gradual increase in the temperature of the refrigerant along the flow direction.

Fig. 3 (a) indicates the liquid fraction of the cold storage agent inside the plate, varying between 0 (for solid phase) and 1 (for liquid phase). It is noted that the time for the complete solidification of the cold storage agent is 10440, 11520 and 13140 s with the refrigerant inlet temperature of 263.15, 265.65 and 268.15 K, respectively. It is obvious that the lower the inlet temperature of the refrigerant is, the shorter time required for the ice storage process is.

3.2. Cold storage rate and capacity

The cold storage rate refers to the amount of the cold energy stored in the ice-on-coil storage plate per unit time, which can be calculated by:

$$q_i = \rho q_v c_p \Delta T_i \tag{6}$$

where ρ is the density of the refrigerant; q_v is the volume flow of the refrigerant; c_p is the specific heat of the refrigerant, and ΔT_i is the temperature difference between the inlet and the outlet of the ice-on-coil storage plate.

Fig. 3 (b) presents the cold storage rates of three different refrigerant inlet temperatures. It is noted that the cold storage rate drops dramatically at the beginning, and then becomes mild gradually. Because there is a huge temperature difference between the refrigerant and the cold storage agent at the initial time. With the rising of the ice thickness around the coil, the thermal resistance increases, and the natural convection heat transfer weakens. It is also obvious that the lower the refrigerant inlet temperature is, the greater the cold storage rate is during the latent heat cold storage stage, which also explains that the lower the inlet temperature of refrigerant is, the shorter time required for ice storage is.

The cold storage capacity is the total amount of the cold energy stored in the ice-on-coil storage plate since the beginning, which can be calculated by:

$$Q_i = \int q_i \cdot dt = Q_{i-1} + \frac{q_{i-1} + q_i}{2} \Delta t \tag{7}$$

where Δt is the time interval between the moment i-1 and the moment i.

Fig. 3 (c) shows the cold storage capacities of three different refrigerant inlet temperatures. It is noted that the lower the inlet temperature of the refrigerant is, the more cold energy the ice-on-coil storage plate stores in the same period of time. Taking 15000 s as an example, the cold storage capacity is 981.64, 703.75 and 481.57 kJ with the refrigerant inlet temperature of 263.15, 265.65 and 268.15 K, respectively. The cold storage capacity with the refrigerant inlet temperature of 263.15 and 265.65 K shows a growth by 103.84% and 46.14% respectively, compared with that of 268.15 K. So the lower inlet temperature of the refrigerant contributes to the higher heat transfer temperature difference, and causes the quicker accumulation of the cold storage capacity. It can also be observed that the cold storage capacities increase dramatically at the first 10000 s and become mild after that. This is because the initial mode of cold storage is latent heat, and that becomes sensible heat after the complete solidification of the cold storage agent inside the plate.

4. Conclusions

In this study, the impact of refrigerant inlet temperature on the ice storage process in an ice-on-coil storage plate was investigated. A detailed three-dimensional CFD model of a half ice-on-coil storage plate was developed in ANSYS. Calculations were carried out for the plate with three different refrigerant inlet temperatures. The simulation results showed that lower refrigerant inlet temperature contributes to higher heat exchange efficiency and bigger ice thickness in the same period of time, but shorter time required for ice storage. And lower refrigerant inlet temperature can also lead to higher cold storage rate and cold storage capacity. One thing worth noting is that every 1 K drop in the refrigerant inlet temperature can cause a decrease by about 1.5% in the unit coefficient of performance (COP). Therefore, it is inadvisable to reduce the inlet temperature blindly for a better thermal performance. The overall efficiency of the system needs to be considered in practical engineering. The simulation results achieved from this work could be used to predict and optimize the thermal performance of the ice-on-coil storage plates in different sizes and uses, by changing various parameters.

Acknowledgements

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