MODELING OF INTERNAL MELT ICE-ON-COIL TANK

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ABSTRACT

Internal melt ice-on-coil tank with built-in horizontal tubes is a kind of ice tank used widely. Its discharge process is effected greatly by density different between ice and water. The existing models developed for internal melt ice-on-coil tank are concentric cylinder models which are applicable to tank with built-in vertical tube or to case when all water in tank is frozen, but is not applicable to describing the effect of density difference between ice and water on discharge. This paper introduces an eccentric cylinder model developed especially for discharge process in this kind of tank with built-in horizontal tubes. The model has simple form and is suitable for system simulation. Experimental validation proved the reliability of the model.

INTRODUCTION

Internal melt ice-on-coil thermal storage tank is a kind of widely used ice tank in HVAC system. In this kind of tank, brine (usually glycol solution) flows through coils, and charges and discharges the tank on the inside of the coils. Well understanding of the charging and discharging characteristics of thermal storage tank is very important to both HVAC system design with thermal storage installations and system operation to reduce the risk. There were quite lots of published literatures reporting researches of thermal characteristics and modeling of ice-on-coil tank, of which most were on external melt ice-on-coil tank, and some were on internal melt ice-on-coil tank.

Jekel (1993)[1] and B. Vick (1996)[2][3] built the theoretical model for internal melt ice-on-coil tank. The models treated the ice cylinder, water cylinder and tube as concentric cylinders during both charge and discharge process, i.e. ice cylinder grows over the tube at the same speed in different directions during charging process, while the water cylinder grows inside the ice concentrically with the ice cylinder during discharge. Neto (1997)[4][5] reported the study on a built-in spiral coil tank in which overlapping phenomenon of ice in charge and of water in discharge were took into account. A theoretical model of the tank was developed. It was also a concentric cylinder model.

H. Li (1997)[6][7] made a series of experiments to study the thermal response of an internal melt ice-on-coil tank. It was a kind of built-in horizontal coil tank as shown in Figure 6(a). He found that heat flux through the coil wall decreased while the thickness of the ice layer growing up during charge. However, the response of discharge process is very different from charge. In the most part of the discharge period, the heat flux through the coil keeps steady rather than decreasing with water layer growing inside in ice cylinder. And before discharge ending, the heat flux increased obviously in a relatively short period. It can be explained that the ice cylinder floated up during discharging due to the density difference between ice and water, then the real heat flux change is far from the concentric cylinder assumption.

Similar phenomena pointed out by H. Li existed in experiments of other tanks. Both the manufacturers' catalogues of BAC and CALMAC ice-on-coil tanks show a quite long steady period of outlet secondary fluid temperature during discharge when inlet temperature is fixed. The difference between them are: outlet temperature of BAC's tank (same as Figure 6(a)) keeps steady longer with slight decrease than CALMAC's tank (with built-in spiral coil), while outlet temperature of CALMAC's tank keeps fix before increasing. In fact, Neto's experimental results[5] of Figure 9 (experimental brine and coil wall temperature) also shown the similar phenomena.

Of course, concentric cylinder model for discharge process is quite suitable to some kind of coil and some kind of discharging cases. For example, if the coil is installed in vertical way or very close to vertical way, during melting process, the ice cylinder will float up along the axial direction of the tube, and the water layer inside the ice keeps approximately concentric with the ice cylinder and the coil. It is why some kind of U-type coils installed in declivitous way such as the products of FAFCO shows the similar properties as concentric cylinders model, i.e. outlet temperature of secondary fluid increases through all discharge process while inlet temperature keeps constant. Besides, if the tank is so fully charged that all the ice cylinders are overlapped and bound together, no displacement will happen during discharge although the coils are installed in horizontal way. In this case, concentric cylinders model is also applicable.

However, partial charging with slight or without ice overlapping is a very important and normal case in real operation. Therefore, ice floating up during discharging always happens in built-in horizontal coil tank and mustn't be ignored in thermal storage system simulation. Obviously, description of the thermal properties of such a process is more complicated than concentric cylinder model.

TSTORS, a Thermal STORage System dynamic simulation software was developed in Tsinghua University, of which an internal melt ice-on-coil ice tank model taking into account of ice floating up in discharging was developed. The model introduced in this paper was developed combining theoretical analysis and experimental study. The experimental analysis bases on H. Li's experimental data, and Li's experimental input/output secondary fluid temperature, secondary fluid flow rate, inventory of ice and heat released rate [6] were used as validation data. The model needs only the geometric dimensions of the tank as the input parameters. Besides, in order to keep higher simulation accuracy, only experimental data of two discharge cases from manufacturer's catalogues were required to regress out the thermal properties of the tank which cannot be reflected by general geometric dimensions.

CHARGE PROCESS ANALYSIS AND MODELING

Charge process can be divided into 4 stages: (1) Water cooling down stage, only sensible heat exchange happens; (2) Unconstrained ice formation stage without ice overlapping, ice cylinder grow concentric with tube, see Figure 1(a); (3) Constrained ice formation stage with ice overlapping, starts from Figure 1(b); (4) Ice cooling down stage, all water is frozen, see Figure 1(c).

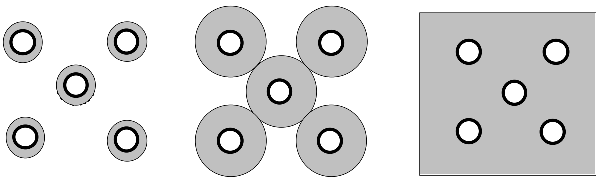


Figure 1 Stages of charge process

Therefore, concentric cylinder model well describes the ice formation process. For stage (1) and (2), thermal resistance can be described using well-developed thermal conduction formula. In stage (3), ice overlapping restricts the heat transfer so that charge rate decreases obviously with time. Therefore, correction factor f [1] is employed here to modify the thermal resistance of the ice:

$$R_{ice} = \ln (d_{crit}/d_o)/ (2\pi\lambda_{ice} lf) \qquad (1)$$

$$f = -1.441Mr + 2.455\sqrt{Mr} + (3.116Mr - 3.158\sqrt{Mr})\, d_o/d_{crit} \qquad (2)$$

$$Mr = 1 - 4\cos^{-1}(d_{crit}/d_{ice})/\pi \qquad (3)$$

where d_{crit} is the diameter of ice cylinder when the ice cylinder touch each other and equals to one-half the tube spacing, and d_{ice} is the possible diameter if there is no constraint. The convective coefficient inside the tube is determined by empirical formula for forced transition flow (2200<Re<10000)[9], and the convective coefficient for the water in the tank outside of the tube is determined using the empirical formula for natural laminar convection on external surface of horizontal tube[9]:

$$\alpha_o = 0.53(Gr_w \Pr_w)^{1/4} \lambda_w / d_o \qquad (4)$$

$$\alpha_{in} = 0.012(\mathrm{Re}^{0.87} - 280)\Pr^{0.4} \lambda_f / d_{in} \qquad (5)$$

Figure 2 shows the experimental data and simulation results of one of the experimental cases. It is found that the charge rate decreases with ice cylinder growing, and the simulation results agree quite well with experiment. Comparison between simulations and experiments of many operation cases of same tank led to the same conclusion.

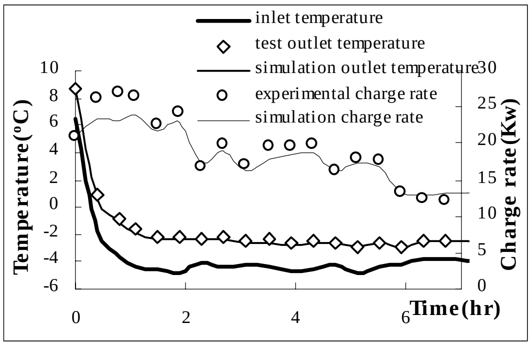


Figure 2 Experimental and simulation result of charge process (w = 9m^3/h)

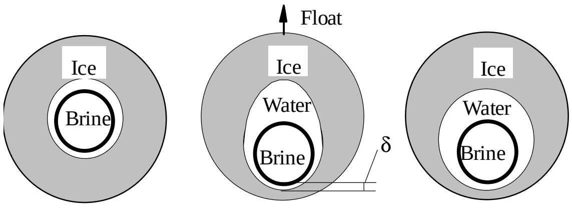
DISCHARGE PROCESS ANALYSIS

Thermal process description

If the temperature of secondary fluid is higher than freezing point, the ice close to the tube starts to melt due to the heat exchange between ice and secondary fluid. If the tube is horizontal, ice will float up due to the density difference between ice and water after a part of ice melted. Hence the bottom of the internal surface of ice cylinder is pressed close to the bottom of the tube, so that both the internal and the external surfaces of the ice cylinder are eccentric to the tube cross section, see Figure 3(b). The secondary fluid transfer heat to the ice mainly by water conduction in the bottom and synthetical effect of conduction and natural convection in the upper part of the water layer. The heat exchange rate of the bottom part change no much, while increasing natural convection in the upper water space countervails the water conductance decreasing when water space expands with melting. Therefore, it is possible that the discharge rate keeps approximately steady during a quite long period when both inlet temperature and flow rate of secondary fluid are fixed.

No matter the ice cylinder melts concentrically or eccentrically to the tube, the volume of the water layer inside the ice cylinder will shrink due to the density difference between ice and water, so that stress is caused on the ice shell. Ice cylinder will be broken from the weakest point if it cannot stand, then the ice shell will leave the tube, float up and contact the upper tube, see Figure 4. The surrounding environment of the tube in this stage is very similar to that of the eccentric stage except the upper water space of the tube is enlarged, so the discharge rate is very similar to that of the eccentric stage.

Ice shell will be broken in to small pieces gradually after the top part of the ice shell is melted away with further melting, as shown in Figure 5. Heat exchange during this stage will be considerably strengthened because ice shell and pieces' moving upward in the water promotes the convection in the tank and speeds up the ice melting. Therefore, discharge rate should be increased considerably in this stage.



(a)Quasi-concentric stage (b) Eccentric stage (c) Eccentric cylinders model

Figure 3 Stage of discharge process before ice cylinder leaves the tube

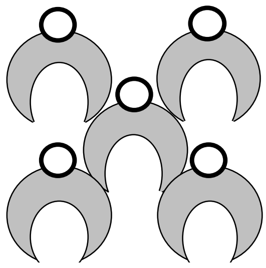


Figure 4 Ice floating stage

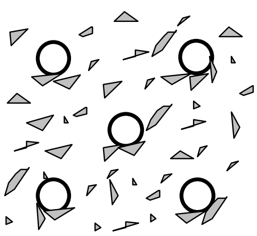


Figure 5 Ice pieces stage

It can be reasoned out that ice shell leaves tube earlier when ice cylinder melts eccentrically to the tube than that when ice melts concentrically with tube, because the weakest point appears at bottom in the former case. In order to describe the progress of discharge, a discharge ratio *fa* is defined as:

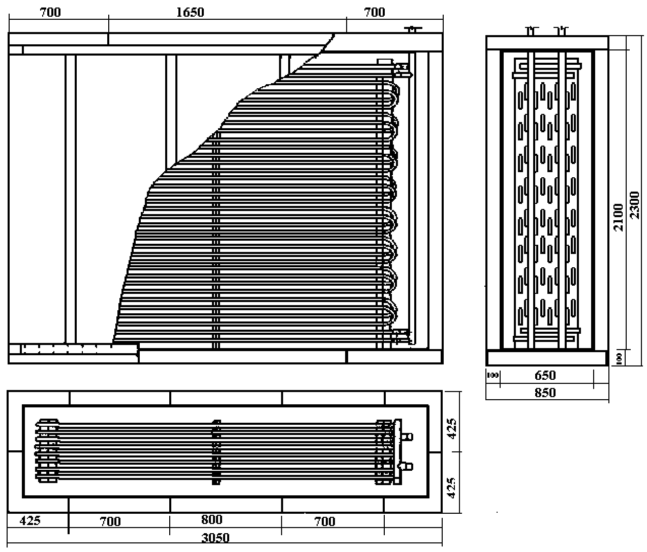
fa=(ice mass melted)/(ice mass before discharge)

All ice is regarded melted away when *fa*=1.0. The charging level before discharge starting decides the discharging way. There are three possibilities. First, if there is no ice overlapping, ice cylinders will float up independently with melting. Second, if ice cylinders overlap partially, all ice cylinders will move up together, with the displacement constrained by the ice cylinder of the tube in the outlet part that melts the least. Third, if all water in the tank is frozen or all remained water is sealed in ice, no displacement will happen in the discharge process. In this case, ice cylinder melts approximately concentrically with the tube.

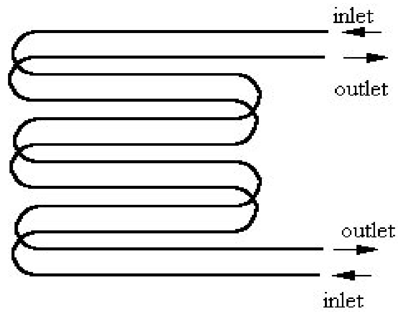
H. Li's experimental data are used here for further analysis on melting process with ice cylinders floating up. The second case mentioned above is the test case of H. Li, because the ice cylinders contact each other as shown in Figure 1(b) before melting.

Experimental installation

Figure 6(a) shows Li's experimental tank. It was a tank with 8 staggered built-in horizontal tubes of 48m length. Each tube has 18 passes. The rated capacity of the tank was 50RTh. Horizontal distance between axis of two adjacent tubes was 53mm while vertical distance was 55mm. The external diameter of the tube was 25mm, and internal diameter was 21mm. The volume of the water at 0℃ was 3.4m^3 while the water line was higher than the top tube. The water level can be observed and measured through a gauge glass.



(a)



(b)

Figure 6 The experimental tank

In order to uniform water temperature in the tank, the secondary fluid in the adjacent tube flow in opposite directions, see Figure 6(b).

The experimental system composed of chiller, electric heater, pump, valve, pipe, controller and data collecting system. The concentration of the glycol solution was 30%. Both inlet temperature and flow rate of secondary fluid were controlled by the control system. In fact, inlet temperature of the secondary fluid had to take a period to reach steady due to the thermal mass of the system. Therefore, the term of 'inlet temperature of secondary fluid' in the following text infra indicates the steady temperature.

According to reference [7], the accuracy of measurement systems were: platinic thermal resistor ±0.05℃, flowmeter 5%, gauge glass 0.25%. Time step of data collector was 30sec. The heat leakage through the tank insulation is lower than 8% of the tank thermal capacity.

DISCHARGE MODELING

The average thickness of water film under the bottom of the tube

When ice is forced float up by density difference between water and ice, the bottom of the ice cylinder will contact the bottom of the tube. The part of ice contacting the tube will melt quickly, so that there keeps a water film between tube and ice, as shown in Figure 3(b). Melting rate of this part of ice depends on the heat flux from secondary fluid to the ice through the tube and the water film, while the thickness of the water film depends on the melting rate and water buoyancy to the ice cylinder. The more slowly the ice melts, or the larger the ice volume is, the thinner the water film is. Therefore, the film thickness will increase slightly with melting progress because water buoyancy to the ice cylinder is getting smaller when ice volume is being reduced.

Considering the geometry of the tube and ice, the contact area of ice and tube will change no much through out the process before ice shell leaves the tube. Assuming that the contact area keeps constant, and the heat conducted through the tube and water film is the only reason for the ice under the tube melts away, an estimated average film thickness can be derived as follows.

One of H. Li's experimental cases was used for this estimation. The initial external diameter of the ice was 76mm, the ice mass under the tube was m_{ice}=0.8999kg/m, and the area of tube bottom contacting the ice was A=0.03925m^2/m. The average temperature of secondary fluid was 7℃, and time interval from melting starting to ice starts becoming piece (when discharge rate increased suddenly) was τ=380min. During this interval, 2 times of the thickness of the ice cylinder was melted. The average temperature difference between ice and secondary fluid Δt can be calculated according to the recorded inlet and outlet temperature.

Energy balance for the ice melting under the tube was $2m_{ice}r = AU\Delta t\ \tau$. The average thickness of the water film can be estimated as: $\delta = \lambda_w/U = 5.9mm$. It indicates that the thickness of the water film is about several millimeters.

Heat transfer coefficient for discharge process

The heat transfer coefficient of melting process can be expressed as:

$$AU=(R_w+R_{coil}+R_{in})^{-1}$$

Determining thermal resistance of water layer outside the tube R_w is quite difficult, so empirical formula have to be derived from the experimental data.

(1) Quasi-concentric cylinder phase

Based on the above estimation, it can be described that when difference between external radius of tube and equivalent radius of ice-water interface is less than the average thickness of water film δ, the tube, ice-water interface and ice external surface keep as concentric cylinders, as shown in Figure 3(a). Figure 5 gives the thermal resistance of water layer inside ice both regressed from experimental data and from concentric cylinder model. It is found that, in the beginning of discharge (e.g. when fa<0.2), difference between test data and concentric model is acceptable small. Therefore, concentric cylinder model is applicable for this quasi-concentric cylinder phase. Heat exchange coefficient between ice and secondary fluid depends on the conductance of the water layer inside the ice.

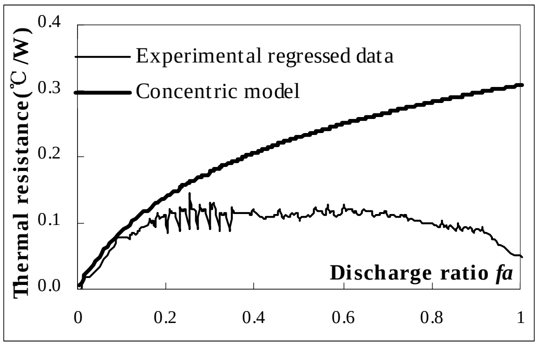


Figure 7 Comparison between concentric model and experimental data (T_{inlet}=8℃, w=9m^3/h)

(2) Eccentric phase

Figure 3(b) shows the cross section of ice and tube after a period of discharge. In fact, the cross section of the ice-water interface looks like an elliptical cross section, because the lower part of the ice cylinder close to the tube will melt more than the upper part.

However, both description of the shape of the ice-water interface and calculating the thermal resistance of water layer are very difficult. In order to simplify the problem, the ice-water interface is described as a cylinder approximately, and the cylinder is eccentric to the tube and the ice external surface, see Figure 3(c). The diameter of the cylinder can be easily determined according to the discharging ratio fa.

Defining δ_{cri} as the criterion for quasi-concentric phase closing and eccentric phase starting, it is regarded that when $(d_w - d_o)/2 \geq \delta_{cri}$, eccentric model is applicable.

Comparing the experimental regressed average water conductance and the simulation conductance of concentric model, *fa* at the moment when concentric stage ends for many cases can be determined. Thus, the thickness of water film can be determined as well. Using this thickness as δ_{cri}, the empirical formula for relation between δ_{cri} and average temperature of secondary fluid T_{fave} is expressed as:

$$\delta_{cri} = a_1 \ (a_2 - T_{f\,ave}) \qquad (6)$$

where T_{fave} equals to the average of inlet and outlet temperature of glycol, a_1 and a_2 are the regressive coefficients from test data, and a_1=0.000452, a_2=26.0. The result indicates that the higher the average temperature of glycol, the shorter the concentric phase is.

Although eccentric cylinder conduction can be solved analytically, effect of natural convection on heat exchange in the water space inside the ice should be taken into account. The theoretical solution for thermal resistance of eccentric cylinder conduction is[10]:

$$R_w = \frac{1}{2\pi\lambda_{eff}\, l}\ln\left(\frac{2N_1/d_w + \sqrt{4(N_1/d_w)^2 - 1}}{2N_2/d_o + \sqrt{4(N_2/d_o)^2 - 1}}\right) \qquad (7)$$

where

$$N_1 = \frac{(d_w/2)^2 - (d_o/2)^2 - \varepsilon^2}{2\varepsilon}$$

$$N_2 = \frac{(d_w/2)^2 - (d_o/2)^2 + \varepsilon^2}{2\varepsilon}$$

$$\varepsilon = (d_w - d_o)/2 - \delta$$

λ_{eff} is the equivalent conductivity of water layer accounting the effect of natural convection. In order to determine λ_{eff}, the empirical formula for determine the effect of natural convection on conductance in concentric cylinder is taken as an exemplar [8]:

$$\lambda_{eff} = 0.386\,\lambda\left(\frac{\Pr}{\Pr + 0.861}\right)^{0.25} Ra_c^{\,0.25} \qquad (8)$$

where

$$Ra_c = \frac{[\ln(d_{ext}/d_{int})]^4}{(d_{ext}^{\,-0.6} + d_{int}^{\,-0.6})^5 L^3} Ra_l \qquad (9)$$

When water temperature is in the range of 0~10°C, it is found that the value of $\left(\frac{\Pr}{\Pr + 0.861}\right)^{0.25}$ varies within 5%, so it can be considered constant. Then the empirical formula for the effect of natural convection on conductance in eccentric cylinder can be taken in following form as:

$$\lambda_{eff} = a\lambda_w Ra_c^{\,b} \qquad (10)$$

$$Ra_c = \frac{[\ln((d_w + \delta)/d_o)]^{1/b}}{(d_o^{\,-0.6} + (d_w + \delta)^{-0.6})^5 l^3} Ra_l \qquad (11)$$

where the reference length of Ra_l is $L=(d_{ext}-d_{int})/2$, the reference temperature is the average temperature of secondary fluid. The coefficients of *a* and *b* are regressed from the experimental data, and the results are: *a*=0.9，*b*=0. 05。

(3) Ice floating stage

Examining the surrounding environment of the tube during ice floating stage, it is found that the heat transfer mechanism of this stage is almost the same as the former eccentric stage. Therefore, discharge rate of this stage should be almost the same as that of eccentric stage. The only difference is that the water space over the tube is larger, and more 0°C free water can enter this space. It is why even during the steady discharge rate phase shown in Figure 8~10, the discharge rate of later period is always bit higher than earlier one.

Therefore, this stage can be considered as the same stage as the former eccentric stage. The same formula of heat transfer coefficient can be used for it.

(4) Ice pieces stage

Assuming that the formalized ice external diameter $D_{max}=d_{icemax}/d_o$ is fixed, the critical discharge ratio fa_{cri} when ice pieces stage starts depends on the temperature of the secondary fluid. The higher the glycol temperature, the lower the value of fa_{cri} is, the earlier the ice contacting the tube is melted away, and the earlier the ice pieces stage starts. The empirical formula for fa_{cri} is:

$$fa_{cri} = b_1 - b_2 T_{f\,ave} \qquad (12)$$

where b_1= 1.2495, b_2=0.0488。Ice pieces stage starts when $fa=(D_w^2-1^2)/(D_{max}^2-1^2)=fa_{cri}$. The formalized diameter of ice-water interface is expressed as: $D_w=d_w/d_o$。

The value of fa_{cri} should decrease when D_{max} increases, because the high buoyancy accelerates the melting in the bottom of ice cylinder and bring the ice cracking forward. Therefore, further experiments for other D_{max} are required to support this conclusion.

In this stage, unrestricted space outside the tube is enlarged. Therefore, enhanced natural convection improves the heat transfer and ice melting. The heat flux through the tube wall depends on discharge ratio *fa*. The less the ice contacting the tube, the less the thermal resistance. Following is the regressive relationship between thermal resistance, discharge ratio and average glycol temperature:

$$R_w = (c_1 - c_2 fa)\,(c_3 - c_4 T_{f\,ave}) \qquad (13)$$

where c_1=0.1956, c_2=0.1117, c_3=1.579, c_4=0.0579。Average temperature of secondary fluid ranged from 6～12℃ for empirical formula (6), (10), (12) and (13).

Heat transfer model for the tank

(1) Thermal balance for secondary fluid side

The secondary fluid flows in the tube, and exchanges heat with water or ice outside of the tube through the tube wall. Length of the tube is usually more than 1000 times of the diameter. Therefore, flow in the tube can be considered as a one-dimensional piston flow. The model of the heat transfer process can be expressed as:

$$\frac{\partial T_f}{\partial \tau} = -u\frac{\partial T_f}{\partial x} + \frac{AU(T - T_f)}{C_f \rho_f V_f} \tag{14}$$

(2) Thermal balance for external side of the tube

In the sensible heat stage, the secondary fluid exchanges heat with water. The secondary fluid in the adjacent tubes flow in opposite directions, so the water temperature in the tank is quite uniform.

In ice-water phase change stage, the secondary fluid exchange heat with ice, and the ice temperature can be considered at freeze point uniformly.

Therefore, lumped capacity model is used to describe the thermal balance of outside of the tube. The expression is as follows:

$$m\frac{dh}{d\tau} = AU(T_f - T) \tag{15}$$

In equation (14) and (15), T represents the bulk temperature in the tank for the first sensible heat stage, and represents the bulk temperature of ice in phase change stage. The left side of equation (15) represents the energy increment of water in sensible heat stage, and represents the energy increment of ice in phase change stage.

EXPERIMENTAL VALIDATION

In order to examine the model's reliability, some experimental data are used for model validation. The empirical coefficients in formula (6), (10), (12) and (13) are regressed from two groups of data obtained from two different tests. The glycol flow rate of both experiments were 9m^3/h, and the inlet temperatures of glycol were 12 ℃ and 10 ℃ respectively.

The glycol flow rate of validation cases ranged within 5~12m^3/h, and the inlet glycol temperatures ranged within 8~12℃. The measured inlet secondary fluid temperatures and flowrate were used as the input data of the simulation. Figure 8~10 show parts of the simulation results and experimental data. From the comparisons, it is found that the simulation results agree well with the experimental data. However, the difference between experimental discharge rate and simulation discharge rate is larger than that of temperatures. It is caused mainly by flowrate measurement error, because the flowmeters responded not as sensitively as temperature sensors.

From the validation results, it is found that the simulation model reflects well the characteristics of the discharge process, and reasonedly explains the mechanism of the variation in the discharge process. The form of the model and the coefficient regression are reasonable. Information obtained from only two experimental cases are sufficiently applicable for other operation cases prediction. Therefore, this model is practicable for real project of thermal storage system design.

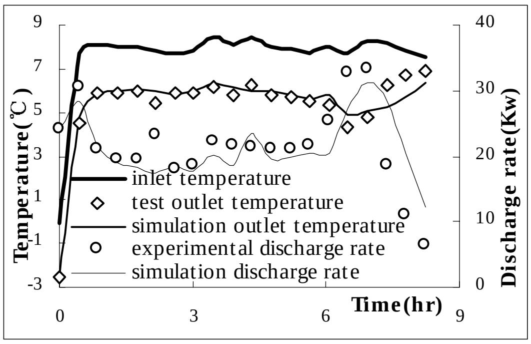


Fig. 8 Experimental and simulation result of discharge process (w=9m^3/h)

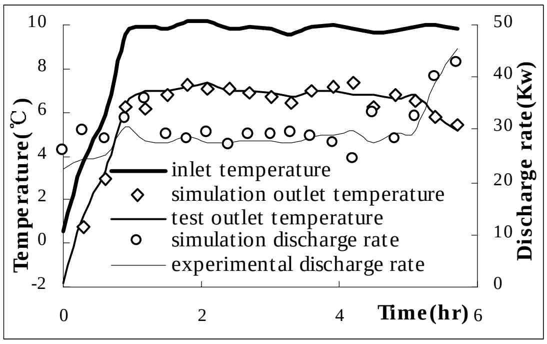


Fig. 9 Experimental and simulation result of discharge process (w=7m^3/h)

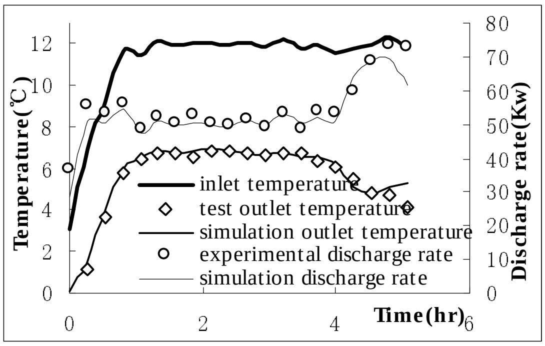


Fig. 10 Experimental and simulation result of discharge process (w=5m^3/h)

CONCLUSIONS

This paper introduced a dynamic simulation model for internal melt ice-on-coil tank with built-in horizontal tubes. The model has following properties:

(1) The form of the model is simple. The tube side is a one-dimensional time-dependent model, and the tank side is a lumped capacity time-dependent model. Therefore, it is applicable for HVAC system simulation software such as TSTORS.

(2) Different from the existing concentric cylinder models, it takes account of the effect of density different between ice and water on the thermal process of discharge, and uses an eccentric cylinder model to approach the ice-water interface in discharge process.

(3) The model explains reasonedly the principle of discharge process, and reflects well the variation in temperature and discharge rate of discharge

process.

(4) The model needs test data of only two different operation cases to obtain the peculiarities of the tank, which is difficult to be fully described by geometric dimensions of the tank. Information from the two experiments is sufficient to simulate the other operation cases and to get results accurate enough for system design application.

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NOMENCLATURE

A	area of tube external surface (m^2)
C	specific heat (J/Kg.℃)
d	diameterof tube or ice cylinder (m)
D	formalized diameter
fa	discharge ratio
Gr	Grashof number
h	enthalpy (J/kg)
m	mass (kg)
L	reference length of Ra_l (m)
T	temperature (℃)
Pr	Prandtl number
R	thermal resistance (℃/W)
Ra	Rayleigh number
Re	Reynolds number
r	latent heat of ice formation (J/Kg)
U	heat transfer coefficient (W/m^2℃)
u	velocity of secondary fluid (m/s)
V	volume (m^3)
w	secondary fluid flow rate(m^3/h)
α	convection coefficient (W/m^2℃)
δ	thickness of water film (m)
ε	eccentricity (m)
λ	thermal conductivity (W/m℃)
ρ	density (Kg/m^3)
τ	time (sec)

SUBSCRIPTS:

f	secondary fluid
w	water
ice	ice
o	outside of the tube
in	inside of the tube
ext	external of the cylinder
int	internal of the cylinder
inlet	inlet of secondary fluid
ave	average
cri	critical
max	maximum
coil	coil