STUDY ON EFFECTIVE USE OF THE ICE THERMAL STORAGE SYSTEM WITH SIMULATION

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Abstract: It is important to evaluate whether an ice thermal storage system can operate effectively as the way planned at the design stage. Ability and time of heat storage are the important indicators to evaluate the effective usability of the ice thermal storage system. It is necessary to consider them at the planning and design stage of air conditioning system.

The target system in this paper is an actual assembled ice thermal storage system in a research facility. While designing an ice thermal storage system, generally, the thermal storage time is calculated by using rated capacity of refrigerator or the heat exchange capacity of ice-on-coil, and the heating capacity time is calculated by using the average ice melting ability of the ice thermal storage tank. In this paper, taking into account the effect of the ice thickness on the ice making and melting ability, and the chiller COP variation due to the changes of the inlet and outlet brine temperature, the author develops a simulation program for multi-connected complete-blending ice thermal storage tank and discusses the effective usability of a practical assembled ice storage system by using the program.

Key Words: Multi-connected Complete-blending Tanks, Ice Thermal Storage System

1 INTRODUCTION

The simulation model of the external melting Ice-on-coil type of ice thermal storage system can be divided into direct and indirect simulation. In direct simulation, the ice-on-coil is modeled as a component of evaporator. In indirect simulation, the ice thermal storage tank can be modeled as a component of a heat exchanger.

A direct ice thermal storage simulation model that contains a detailed model for heat transfer from the ice thermal storage tank to the refrigerant was developed (R. Jacobson, 1986) by using a quasi-steady thermal resistance network heat transfer model. Coleman (1990) extended the work of Jacobson by developing a more complete model using the thermal resistance network technique applied to the entire coil. Silver et al. (1988) developed another model using the thermal resistance network technique. However, Silver et al. divided the coils in a finite number of segments. Generally, the refrigeration capacity and heat exchange amount of evaporator change depending on the ice thickness and the evaporating temperature. In the R. Jacobson's model, the evaporator capacity is set equal to the rated compressor capacity, so it can not exactly reproduce the dynamic performance of the ice-on-coil thermal storage system. The problem with the model is in the absence of information such as the tube length, so users must estimate the tank specifications, which destroys the accuracy of the detailed model. In addition, the execution time of the detailed model is unfavorably long for an annual simulation, which means poor applicability to a real ice-on-coil thermal storage system.

A dynamic ice-on-coil evaporator model was developed as a stand-alone simulation program considering the heat transfer and pressure drop correlations availability (Milbitz, 1987). In this model, based on the heat balance of the ice layer on the length of the tube in the steady state, to calculate amount of the ice making and melting. Calculation of the evaporation amount includes the effects of the refrigerant superheat, changes of the evaporating pressure and ice thickness. Jones and Shiddapur (1995) applied and improved Milbitz's direct ice-on-coil

evaporator model to an indirect ice-on-pipe components model and developed an indirect ice thermal storage system simulation program by using a finite difference approach. The results indicated that the model lacks some accuracy, mainly during the beginning of the charging period as well as during the ice melting period.

A simulation model of a temperature stratification ice thermal storage tank was developed (Nakahara, 1984) based on the experimental results of an indirect ice thermal storage tank with a finite difference approach. In this simulation model, the capacity of the chiller was modified by the brine temperature that changes due to the change of the water temperature and the heat balance between the amount of the heat transfer through the coil and chiller output. Zheng et. al (2001) modified Nakahara's model, and incorporated it into HVACSIM^{+} (J). Ibamoto et. al (1986) developed another simulation model of indirect ice thermal storage tank. In this model, although the changes of the external heat transfer coefficient of the coil along the tube length direction was taken into account, the capacity of the chiller was not modified.

However, these simulation model shown above, except for Nakahara model when given certain parameters as inputs, can not simulate an actually assembled multi-connected complete-blending ice thermal storage tank. Therefore, In this paper, taking into account the effect of the ice thickness on the ice making and melting ability, and the chiller COP variation due to the changes of the inlet and outlet brine temperature, the author develops a simulation program for multi-connected complete-blending ice thermal storage tank and discusses the effective usability of a practical ice storage system by using the program.

2 OUTLINES OF TARGET BUILDINGS AND ICE THERMAL STORAGE SYSTEM

The outlines of target buildings and ice thermal storage system are show in Table 1. And the outline of target heat source system and the Ice-on-coil placement is show in Figure 1. The target ice thermal storage system consists of two brine heat pumps and eleven ice-on-coils. In addition, because each ice-on-coil consists of 17 parallel columns of horizontal tube and each column of tube is connected to the heat pump chillers via the vertical headers (See Figure 2), almost no temperature difference exists for vertical direction in each tank during the ice making, so the target ice thermal storage tanks can be assumed of being a multi-connected complete-blending ice thermal storage tank.

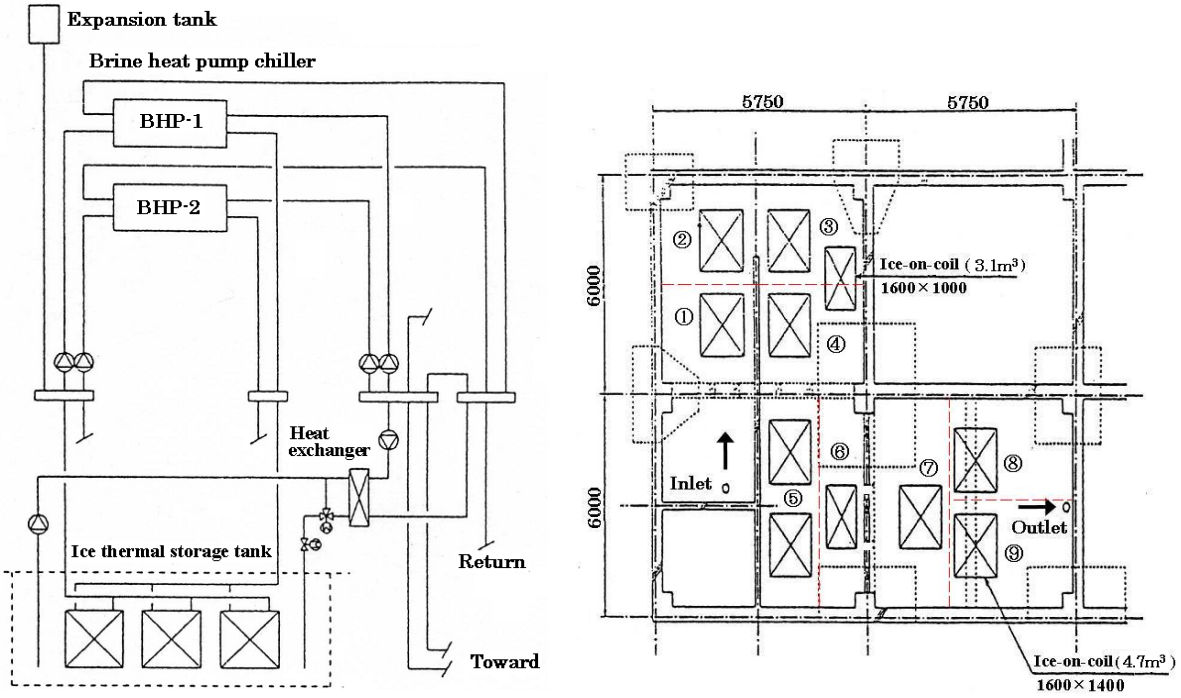


Figure 1: Outline of heat source system and ice-on-coil placement

Table 1: The outlines of the object buildings and ice thermal storage system

Buildings	Structure	Reinforced concrete made	**Ice thermal storage tanks**	Dimension of a tank Maximum ice volume	138m^2×3.5m (H) 30.1m^3
	Usage	Research facility		Number of ice-on-coil	4.7m^3×9,3.1m^3×2
	Total floor area	9,734m^2		Available water volume	208.9m^3
Heat source system	Brine heat pump chiller	100 Hp × 2		Material and concentration of brine	Ethylene glycol 60%
	Storage capacity Brine temperature range	440kW -7 °C ～ -3 °C		Material of brine, total length, inside and outside diameter of tube	Copper, 3956.8m 16mm, 19.05mm
	Chilled capacity Chilled water temperature range	560kW 7 °C ～ 12 °C		Water temperature range of storage tank Chilled water flow volume	5 °C ～ 2 °C 133.3 m^3/h

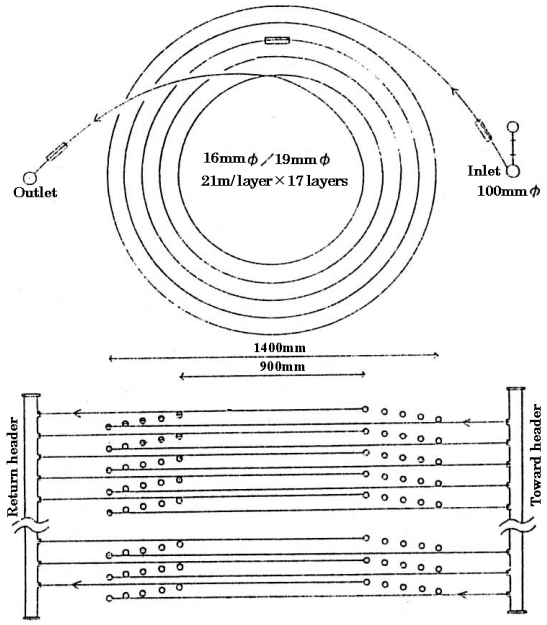


Figure 2: The structure of object ice-on-coil

3 CALCULATION OF CAPACITY AND TIME OF THERMAL STORAGE

When designing an ice thermal storage system, generally, the thermal storage time is calculated by the use of rated capacity of refrigerating machine or the heat exchange capacity of ice-on-coil, as shown in the following 3.1 and 3.2 sections, respectively.

3.1 Simple Calculation Method of Thermal Storage Time

Assuming the ability of heat storage equals to the rated capacity of the chiller during the thermal storage operation, the sensible and latent heat storage time can be calculated as following. The calculated results are shown in Table 2.

$$h_k = M_w C_w \rho_w T_m / NHP \quad (1)$$

$$h_s = M_{ice,\max} C_{ice} \rho_{ice} / NHP \quad (2)$$

Where, it was assumed that the brine temperature of outlet and inlet of chiller are their design values, and the heat exchanger temperature difference is the average temperature difference between the brine and water temperature. From Table 2, it can be understood that the total thermal storage time is 9.2 hours, less than the 10 hrs of nighttime duty between 22:00 and 8:00 of the cheaper electricity rate schedule set for the peak shift operation from the daytime.

3.2 Approximation Method of Thermal Storage Time

On the other hand, when the ice-on-coil storage system is in actual operation, the thermal storage capacity operates apart from the rated capacity of the chiller, but is limited to heat exchange capacity of coil. Moreover, the heat exchange capacity of ice-on-coil is decreasing with increase of ice thickness. In an approximation method, with an assumption that the rate of sensible heat exchange on ice-on-coil proportions to the average temperature difference between the brine and water, and ice formation is in proportion to the latent heat exchange rate as well, h_s and h_k are calculated by using Equation (3) and (4) respectively. The calculation results are also shown in Table 2.

$$h_k = M_w C_w \rho_w T_m / \{KL(T_{bm} - T_{wm})\} \quad (3)$$

$$h_s = M_{ice,\max} C_{ice} \rho_{ice} / \{KL(T_{bm} - 0)\} \quad (4)$$

In Equation (4), it is assumed that the heat exchange rate of ice-on-coil equals to the one when the ice volume is a half of the designed maximum ice volume. From Table 2, it is shown that the total thermal storage time is 15.5 hours that is 1.7 times of the result with a simple calculation method, which shows a big difference between the two methods.

3.3 Thermal Storage Time Simulation with the Present Tool

Two methods described above do not accurately calculate changes of the over-all heat transfer amount of ice-on-coil due to the change of ice thickness. Therefore, to calculate the thermal storage time more accurately, a simulation program that takes into account the changes of the ice thickness and the brine chiller COP changes due to the changes of the brine temperature that influence the ice making capacity of the ice thermal storage system.

3.3.1 Assumptions on the ice thermal storage simulation

a) The water temperature in the tank is in a fully mixed state and is 5 degree C in the beginning of thermal storage.
b) The water temperature in the tank is 0 degree C when the ice begins to produce.
c) The lower limit of the brine temperature in the chiller outlet is -10 degree C.
d) The chilled water temperature in the tank inlet is constant.

3.3.2 Heat transfer on the inside and outside surface of the tube

As the inside surface of the tube is in the forced convection state, the heat transfer coefficient α_i can be obtained by Equation (5) or Equation (6) (SHASEJ, 1995).

$$\alpha_i = \frac{0.023\ \mathrm{Pr}_b^{0.4}\ \mathrm{Re}_b^{0.8}\ \lambda_b}{2 r_i} \qquad \mathrm{Re_b} \geqq 2100 \quad (5)$$

$$\alpha_i = \left\{3.65 + \frac{0.0668\,\mathrm{Pr}_b\,\mathrm{Re}_b\,\frac{2r_i}{l_b}}{1 + 0.04(\mathrm{Pr}_b\,\mathrm{Re}_b\,\frac{2r_i}{l_b})^{2/3}}\right\}\frac{\lambda_b}{2r_i} \qquad \mathrm{Re_b} < 2100 \quad (6)$$

As the outside surface of the tube is in the natural convection, heat transfer coefficient α_o can be obtained by Equation (7) (SHASEJ, 1995).

$$\alpha_o = \frac{0.53\ (Gr_w)^{0.25}\ \lambda_w}{2 r_o} \qquad 10^4 < \mathrm{Gr_w Pr_w} < 10^8 \quad (7)$$

3.3.3 Simulation of thermal storage capacity

a) The over-all heat transfer rate of the coil (Q_c)

$$Q_c = 2\pi L_c K T_g \quad (8)$$

Where, T_g is the logarithmic mean temperature difference and can be calculated by Equation (9).

$$T_g = \frac{T_{bo,c} - T_{bi,c}}{\ln[(T_w - T_{bo,c})/(T_w - T_{bi,c})]} \tag{9}$$

K is the over-all heat transfer coefficient of tube, and can be calculated by Equation (10).

$$\frac{1}{K} = \frac{1}{\alpha_i r_i} + \frac{1}{\lambda_{coil}} \ln \frac{r_o}{r_i} + \frac{1}{\alpha_o r_o} \tag{10}$$

b) The brine temperature at the coil outlet ($T_{bo,c}$)

$$T_{bo,c} = T_{bi,c} - Q_c / (\rho_b C_b F_b) \tag{11}$$

c) The capacity of the chiller is modified using a theoretical COP formula (Nakahara, 1984).

$$NHP' = NHP \frac{T_{bo} + 273 - \Delta T_b}{T_{bo}' + 273 - \Delta T_b'} \times \frac{T_{cd}' - T_{bo}' + \Delta T}{T_{cd} - T_{bo} + \Delta T} \tag{12}$$

d) When the capacity of the chiller (NHP') is larger than the over-all heat transfer amount of the coil (Q_c), revise the brine temperature in the chiller inlet (T_{bi}) to downward until the T_{bo} equals to its lower limit value. And when NHP' is smaller than Q_c, adjust T_{bi} to upward.
e) The water temperature in the tank is represented with differential Equation (13).

$$dT_w = \frac{1}{M_w C_w \rho_w} L_c \times K \times T_g \, dt \tag{13}$$

f) When the water temperature in the tank reaches 0 degree C, ice formation begins. The ice volume is represented with differential Equation (14).

$$dM_{ice} = \frac{2\pi K_{ice} T_g L_c}{C_{ice} \rho_{ice}} dt \tag{14}$$

However, at the moment of the phase change from water to ice at the outer surface of the ice-on-coil, the heat transfer coefficient becomes almost infinite, so that the calculation of the thermal transmittance on the ice-on-coil (K_{ice}) will become as Equation (15).

$$\frac{1}{K_{ice}} = \frac{1}{\alpha_i r_i} + \frac{1}{\lambda_{coil}} \ln \frac{r_o}{r_i} + \frac{1}{\lambda_{ice}} \ln \frac{r_o + D_{ice}}{r_o} \tag{15}$$

3.3.4 The calculation of thermal storage time

a) The sensible heat changes of water in the tank (Q) can be calculated by Equation (16)

$$Q = C_w M_w \rho_w dT_w \tag{16}$$

b) The sensible heat storage time
As the over-all sensible heat transfer amount of the coil equals to the variation of sensible heat of water in the tank, linking Equation (8) and (16), and then integrating Equation (17) from initial water temperature in the tank to 0 degree C, the sensible heat storage time (h_k) can be obtained as Equation (17).

$$h_k = \int_{T_{wo}}^{0} \frac{C_w M_w \rho_w}{2\pi L_c K T_g} dT_w \tag{17}$$

c) The Latent heat storage time
When the tank water temperature becoming 0 degree C, the latent heat storage amount equals to the latent heat rate for making ice, and it can be obtained by integrating Equation (14) up to designed maximize ice volume, as shown in equation (18). Simulation results of the sensible and latent heat storage time are also shown in Table 2.

$$h_s = \int_{0}^{M_{ice,\max}} \frac{C_{ice} \rho_{ice}}{2\pi L_c K_{ice} T_g} dM_{ice} \tag{18}$$

3.3.5 Simulation flow

Because the capacity of the chiller (*NHP*') depends on the brine temperature in the chiller inlet, it is needed (a) to modify the capacity of the chiller with the brine temperature in the chiller inlet at first time; (b) to calculate α_i, α_0, K, T_w and M_{ice} with the average temperature of the brine and to calculate Q_c and T_{bo} by using them; (c) to modify T_{bi} by comparing *NHP*' with Q_c and to repeat calculations until *NHP*' becomes equal to Q_c; (d) The simulation shall be carried out until that the designed maximum ice volume is attained, and then h_k and h_s is obtained. Figure 3 shows the flow of simulation.

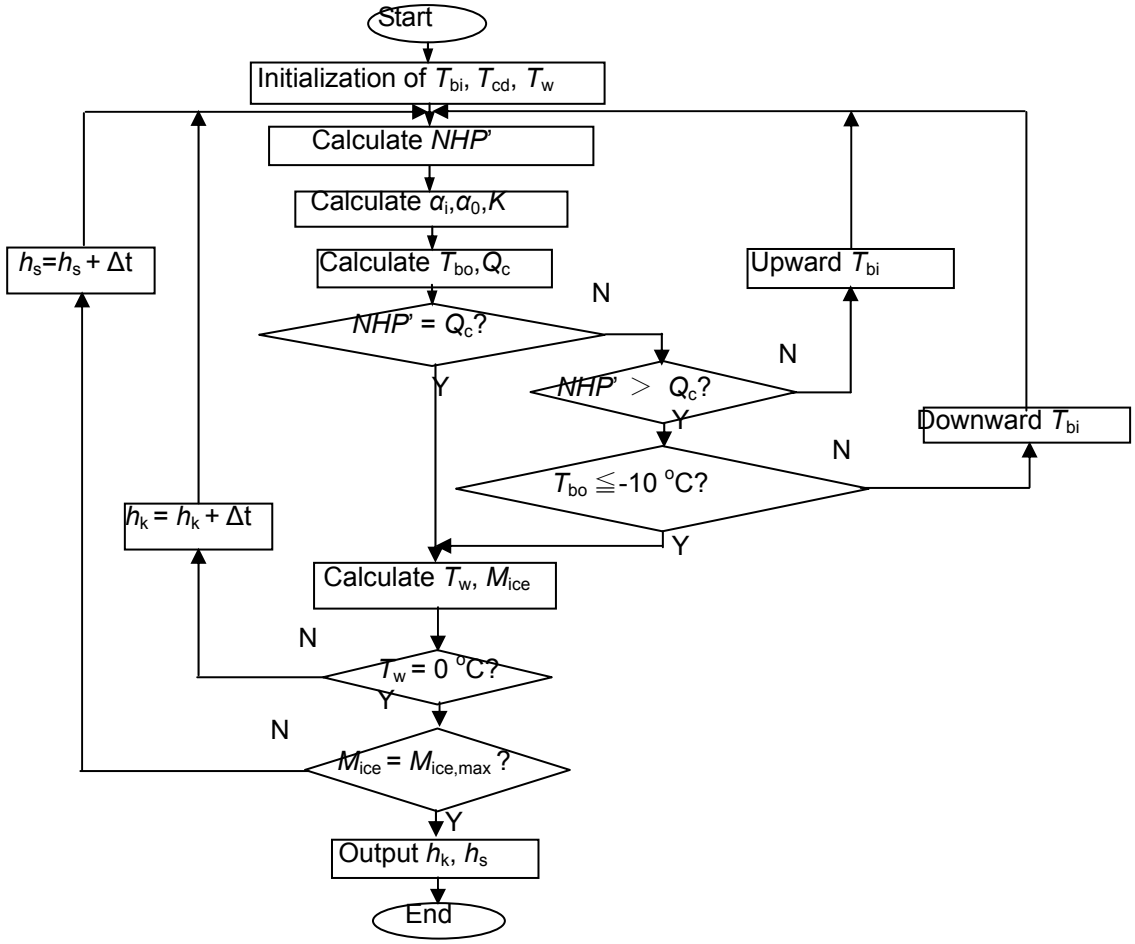


Figure 3: A flow chart of the simulation

3.3.6 Comparing results of three calculation methods

Figure 4 and 5 show the amount of sensible and latent heat storage in target ice thermal storage system and the brine temperature of the inlet and outlet in the chiller, respectively. From Figure 4, 5 and Table 2, the following findings are obtained.

a) Following the increase of the ice thickness, the amount of the thermal storage decreases.

b) The ice making time simulated is longer than that calculated by the simple method, and is less than that calculated by the approximation method. The differences among three methods can be explained as follows. (a) Increase of the thermal resistance according the ice thickness to be considered in simulation. (b) The average brine temperature was used in an approximation method. (c) The brine temperature in the chiller outlet was adjusted downward as the capacity of the chiller to maximize value in simulation.

c) After beginning to make ice, the brine temperature in the chiller outlet arrived at the lowest limit value, -10 degree C, and this indicates that total outer surface of coil in object ice thermal

storage system is too small.
d) According to the simulation result, the thermal storage time of object ice thermal storage system is 12.9 hours. If just 10 hours of nighttime operation is desired for ice storage operation, insufficient coil surface area is the cause of it. However, this will be allowable considering that this insufficiency only occurs at a few days of peak cooling season in summer.

Table 2: Comparing calculation results of thermal storage time (h) of three methods

Method	Sensible heat storage time	Latent heat storage time	Total time
Simplify	2.8	6.4	9.2
Approximate	3.9	11.6	15.5
Developed tool	2.8	10.1	12.9

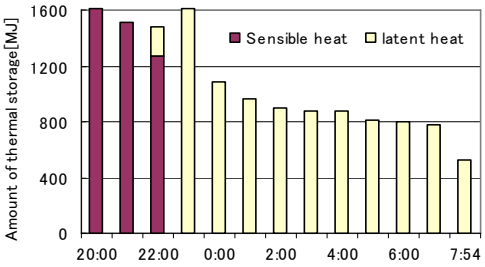


Figure 4: The amount of sensible and latent heat storage

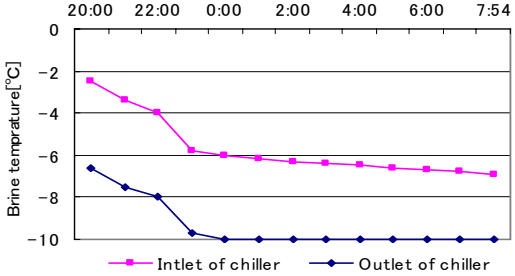


Figure 5: The brine temperature of outlet and inlet of the chiller

4 CALCULATION OF CAPACITY AND TIME REQUIRED FOR HEAT DISSIPATION

4.1 Approximation Method of Cooling Capacity Time

The cooling capacity is reduced according to reduction of ice surface area. The approximation method used by the designer assumed that there is a linear relationship between the cooling capacity and the residual ice volume, and calculated the cooling capacity time using a half of the designed maximum ice volume. The calculation methods and results are shown in Table 3. The ice melting time is 5.23 hours, and there is sufficient ice melting ability in the object ice thermal storage system. However, this method can not reproduce the effects of non-linear changes of the ice volume during the ice melting.

Table 3: The calculation formula and results of approximation method

Sign	Unit	Interpretation	Calculation methods	Results
M	m^3/m	Ice volume of unit coil length when the ice volume is 1/2 of the designed maximum volume	$M = 1/2 M_{ice,\max} / L$	3.8E-3
$d_{ice,m}$	m	Diameter of outer surface of the ice when the ice volume is 1/2 of the designed maximum volume	$d_{ice,m} = \left(4M/\pi + d_o^2\right)^{1/2}$	0.07215
A_{xm}	m^2	Outer surface area of ice when the ice volume is 1/2 of the designed maximum volume	$A_{xm} = \pi d_{ice,m} L$	717.51
T_{gm}	°C	Logarithmic mean temperature difference	$T_{gm} = \frac{T_{wo} - T_{wi}}{Ln(T_{wo}) - Ln(T_{wi})}$	3.28
Q_{ice}	MJ	Mean ability of the ice melting	$Q_{ice} = \alpha_o T_{gm} A_{xm} \quad (\alpha_o = 180)$	1,773
Q_s	MJ	Latent heating capacity	$Q_s = M_{ice} C_{ice} \rho_{ice}$	9,274
Q_k	MJ	Sensible heating capacity	$Q_k = M_w C_w \rho_w T_{gm}$	2,868
h_{ice}	h	Ice melting time	$h_{ice} = M_{ice} C_{ice} \rho_{ice} / Q_{ice}$	5.23

4.2 Simulation for Ability and Time of the Ice Melting

4.2.1 Assumptions of the ice melting simulation

a) The target ice thermal storage tank consists of eleven ice coil units and four tanks, and all ice-on-coil is not placed in series along the water flow direction. Therefore, four tanks were changed to nine tanks in the simulation, as indicated by thin dashed lines in Figure 1.
b) The water temperature in each tank is in a fully mixed state and is 0 degree C when beginning to melt ice.
c) The temperature and flow rate of the returning chilled water from the secondary side equal to the design values, 5 degree C and 133.3 m^3/h, respectively.
d) The ice volume equals to the designed maximum ice volume, and is evenly distributed on each coil.

4.2.2 The ice melting simulation model

a) If a large number of tubes are arranged in parallel to each ice coils, the convective heat transfer coefficient of the outer surface of the ice is calculated by Equation (19) (SHASEJ, 1995).

$$\alpha_0 = \frac{0.33\psi_h n \,\mathrm{Pr}_w^{0.3}\ \mathrm{Re}_w^{0.6}\ \lambda_w}{d_{ice}} \tag{19}$$

b) Heat balance of total cooling capacity of each ice thermal storage tank during the ice melting is represented by the differential Equation (20)

$$(1-IPF)V_w C_w \rho_w \frac{dT_w}{dt} = F_w C_w \rho_w (T_{in} - T_w) - A_x \alpha_o (T_w - 0) \tag{20}$$

c) Differencing and sort out Equation (20), Equation (21) is obtained and temperature of each tank can be calculated.

$$\begin{aligned} T_w^{i+1} &= \{(1-IPF^{i+1})V_w T_w^i + F_w T_{in}\Delta t\}/B && IPF \neq 0 \\ T_w^{i+1} &= \{V_m T_w^i + F_w T_{in}\Delta t\}/A && IPF = 0 \end{aligned} \tag{21}$$

Where,

$$B = (1 - IPF^{\,i+1})V_w + (F_w + \frac{\alpha_o A_x}{c_w \rho_w})\Delta t, \qquad A = V_w + F_w \Delta t \tag{22), (23}$$

IPF is a percentage of the ice volume in the tank, and is calculated by Equation (24).

$$IPF = \frac{\xi}{r_o^{\,2}}\left\{(r_o + D_{ice})^2 - r_o^{\,2}\right\} \tag{24}$$

d) The amount of ice melting in each tank is represented by the differential Equation (25).

$$\frac{dM_{ice}}{dt} = \frac{A_x \alpha_o (T_w - 0)}{C_{ice}\rho_{ice}} \tag{25}$$

e) The ice melting time
The ice melting time can be obtained by integrating Equation (25) from the designed maximum ice volume to 0 as shown in Equation (26).

$$h_{ice} = \int_{Mice,\max}^{0} \frac{C_{ice}\rho_{ice}}{A_x \alpha_o (T_w - 0)} dM_{ice} \tag{26}$$

4.2.3 The ice melting simulation results and discussion

a) Simulation on the step response of 5 degrees C water temperature input was executed and Figure 6 shows the temperature profiles of the object ice thermal storage tanks during the ice melting process. The water temperature of the last tank, from which the chilled water is supplied to the cooling circuit, gradually rises. Therefore, it is considered that the simulated sensible heating capacity is larger than the one with an approximation method.

b) Figure 7 shows the variation of residual ice volume in each tank during the ice melting

process. After four hours of cooling operation, no ice is found in the tank 1 and 2; but the tank 8 and 9 maintain the ice until the last moment as shown.

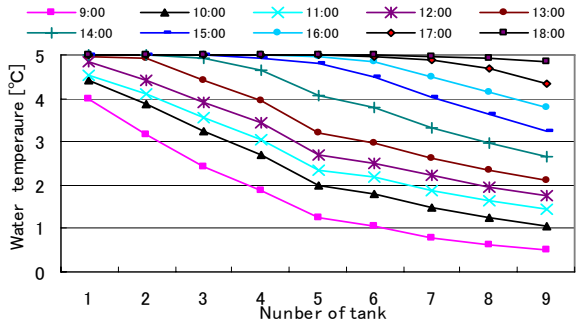


Figure 6: The temperature profile in tanks

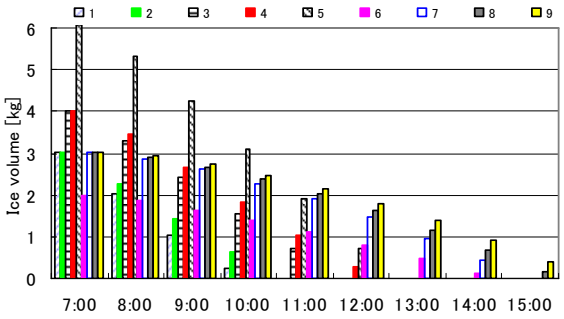


Figure 7: Changes of ice volume in each tank

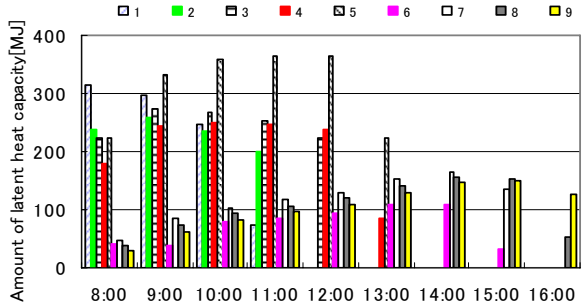


Figure 8: Changes of latent heating capacity in each tank

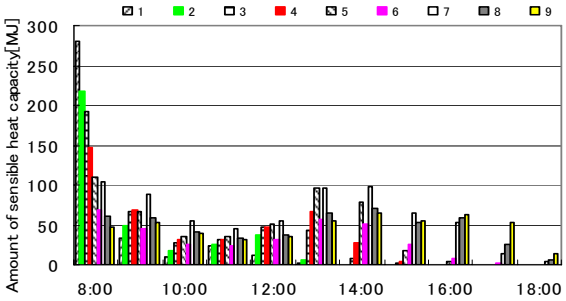


Figure 9: Changes of Sensible heating capacity in each tank

c) The variation of the sensible, latent and total cooling capacity during the ice melting are shown in Figure 8-Figure 10, respectively. From these figures, the following findings are obtained:

(a) The cooling capacity of each tank is different because the length of the tube of ice-on-coil in each tank is also different.
(b) Because the initial water temperature in the tank is 0 degree C, there is the sensible cooling capacity during all the ice melting period.
(c) The ice melting time is 9 hours, longer than that calculated by the approximation method.

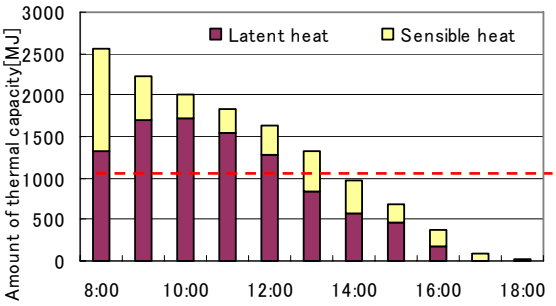


Figure 10: Changes of total heating capacity

4.3 Comparison between the Results of Two Calculation Methods

Comparison between the results of two calculation methods is shown in Table 4. The result of their latent heating capacity is same; however, the ice melting time obtained by each calculation method is different. This is because the average rates of the ice melting between the two calculate methods are different. The result of the sensible cooling capacity with simulation is greater than that by the approximation method. The difference appeared in the water temperature in the tank outlet, in which it raised up to 5 degree C for the simulation method, but it keeps 2 degree C, the designed value, for the approximation method. When a real ice thermal storage system operates on a condition of the ice melting, the abilities of the ice melting vary along with changes of the residual ice volume and the water temperature in tank outlet. For this reason, it is considered that the simulation results are more accurate than the one by using an approximation method.

Table 4: Comparison of the Results between Two Calculation Methods

Methods	Average abilities of the ice melting [MJ]	latent heating capacity [MJ]	Sensible heating capacity [MJ]	Total heating capacity [MJ]	The ice melting time[h]
Approximation	1,773	9,274	2,868	12,142	5.23
Simulation	1,070	9,274	4,079	13,353	9.0

5 DISCUSSION

In this paper, taking into account the effect of the ice thickness on the ice making and melting ability, and the chiller COP variation due to the changes of the inlet and outlet brine temperature, the author developed a simulation program for an assembled multi-connected complete-blending ice thermal storage tank and discussed the effective usability of a practical ice storage system by using the program. The following results were obtained.

1) Because the calculated capacity storage ability of the target ice thermal storage tank is smaller than the rated capacity of the chiller, the simulation value of the thermal storage time is longer than that by a simplify calculation, and shorter than that using the average overall heat transfer coefficient of the ice-on-coil. This shows that the simulation program reproduces the increase of the overall thermal resistance of the ice coil, due to freezing on the outer surface of the coil, and the actual chiller capacity changes and works on the conditions different from the rated.

2) In the simulation of the cooling capacity during the daytime, the simulation reproduces the changes of the ice melting ability due to changes of the water temperature and residual ice in each ice thermal storage tank. The simulation value of the ice melting time is longer than that calculated by an approximation method with the average ice melting ability of the ice-on-coil, the former of which obtained a value that is close to the actual one. Moreover, when a real ice thermal storage system operates on a condition of the ice melting, returned chilled water temperature from the secondary side varies depending on changes of the cooling load. However, according to the simulation described in this paper, 5 degree, the designed value of the chilled water, was applied. Therefore, it is considered there is a small difference between the simulation value of the ice melting time and actual value.

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Nomenclature

A_x Outer surface area of ice [m^2]
C Specific heat of brine [kJ/kg]
C_{ice} Heat of fusion of ice [kJ/kg]
d Diameter [m]
D_{ice} Ice Thickness [m]
F Flow rate [m^3/h]
G_r Grashof number
h Storage time [h]
h_{ice} The ice melting time [h]
IPF Ice packing factor [%]
K Over-all heat transfer coefficient of tube at non-freezing [kW/(m·K)]
K_{ice} Over-all heat transfer coefficient of tube at freezing [kW/(m·K)]
l_b Tube length of each layer in an ice coil [m]
L Total tube length in all tanks [m]
L_c Total tube length of each ice coil [m]
M Volume [m^3]
$M_{ice,max}$ Design maximum amount of ice [m^3]
n Correction factor based on Re and the ratio between the tube diameter and pitch of adjacency tubes
NHP Refrigerator capacity [kW]
P_r Prandtl number
Q Amount of latent heat storage [kJ]
Q_c Amount of over-all heat transfer of tube [kJ]
r Radius of tube [m]
R_e Reynolds number
t Time [h]
T Temperature [^oC]
T_{bm} Average brine temperature of storage [^oC]
T_g Logarithmic mean temperature difference at heat storage [^oC]
T_{in} Return chilled water temperature from the secondary side [^oC]
T_m The average temperature difference of heat exchange at sensible heat storage [^oC]
T_{wo} Initial water temperature in tank [^oC]
V_w water volume in each tank [m^3]
V_m Ice volume melted in each tank [m^3]
α Heat transfer coefficient [W/(m^2·K)]
ζ Coil percentage filling in tank [%]
λ Thermal conductivity [W/(m/K)]
ψ Correction factor based on the tubes number of columns along the direction of water flow
ρ Density [kg/m^3]
ΔT_b Temperature difference between brine outlet temperature and evaporating temperature [^oC]
ΔT_{cd} Temperature difference between the cooling water outlet temperature and condensation temperature [^oC]
ΔT $\Delta T_b+\Delta T_{cd}$ [^oC]
Δt Time step of difference calculation [h]

Subscripts

b: Value for brine
bi: Value of chiller inlet
bi,c: Value of coil inlet
bo: Value of chiller outlet
bo,c: Value of coil outlet
cd: Value about cooling water
coil: Value for coil
i: Value of the coil inter surface
ice: Value for ice
k: Value for sensible heat
o: Value of the coil outer surface
s: Value for Latent heat
w: Value for water
‘ : Value of any conditions