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DESIGN OF A SMALL-CAPACITY ICE THERMAL ENERGY STORAGE SYSTEM

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ABSTRACT
In this study, ice storage tank of an air-conditioning system based on ice storage to cool a room located in the Laboratories of Mechanical Engineering Department of Çukurova University, Turkey was modeled, designed and tested. Considering the cooling load distribution of the room, a model was written to calculate the heat transfer process for internal melt ice-on-coil system. Comparison of the model with the experimental results showed that the agreement between the two is good.

INTRODUCTION
Due to high cooling demand, electricity consumption peaks during hot summer days in some provinces of Turkey for which cooling degree days are high [1]. However, demand is reduced and utilities have electricity to spare during the nighttime. Cool thermal energy storage is one of the technologies for solving the problem of electrical power imbalance between nighttime and daytime.

Cool thermal energy storage is a technology that shifts electricity demand from on-peak period to off-peak hours, reducing peak demand. It also reduces electricity bills due to use of electricity for cooling during the periods when electricity prices are lower.

These systems utilize a storage medium to store cooling produced during off-peak hours. The stored cooling is later used during peak hours to meet an air-conditioning or process-cooling load. Various media such as chilled water, ice, or phase-change-materials can be used to store cooling material [2]. Chilled water storage and ice storage are the most common storage designs. Ice storage tanks have much smaller volumes than chilled water tanks of the same cooling capacity.

A conventional vapor compression chiller or an industrial-grade ice-making unit is used to charge the storage tanks during off-peak hours. The chilled liquid from the storage tanks is circulated through the building's fan coils or a secondary heat exchanger during on-peak times for cooling.

Operating strategies for cool thermal energy storage systems are generally classified as either "full storage" or "partial storage", referring to the amount of cooling load transferred from on-peak period to off-peak. In full storage thermal energy storage strategy, the refrigeration equipment is off during on-peak hours, and all cooling load is provided from the storage that is charged during the off-peak period [3].

Although thermal energy storage systems for cooling were first used commercially in the 1940s and have gaining popularity in the USA and Europe in recent years, application of these systems are very limited in Turkey. Although there has been an interest in these systems in recent years, barriers still exist in Turkey regarding the application of these systems; such systems are not well known, and they are notorious for troublemaking. Another barrier is the lack of designers, contractors and operators who can properly design, install and operate these systems economically and reliably. Cool storage tanks, which are one of the key components in cool thermal energy storage systems, are not manufactured at all in the country. They are imported from abroad and this makes these systems economically unattractive.

Turkish Government has introduced a law on 25 December 1997 about electricity tariffs to shave the peak-load. According to the new law, the day is divided into three periods: 17^{00} to 22^{00} (peak period), 22^{00} to 6^{00} (off-peak period), and, 6^{00} to 17^{00} (normal period). At the present time, variable tariffs are optional for all electricity consumers. Compared with the constant tariffs (daylong constant prices) for office buildings, the cost of electricity is 82% higher for the peak period, and 60% and 5% lower for the off-peak and the normal periods, respectively according to the variable tariffs [4].

In this study, an ice thermal energy storage system with approximately 42 kWh cooling capacity was built and tested in the Laboratories of Mechanical Engineering Department of Çukurova University, Turkey. In the study, focus was on the ice storage tank. An internal melt ice-on-coil tank was modeled, designed, constructed and tested. Comparison of the model with the experimental results showed that there is a good agreement between the two [3].

DESIGN OF ICE STORAGE TANK
In this study an ice storage tank was designed to store the daily cooling energy requirement of a selected space whose details are given below. Among the various cool storage technologies available, internal melt ice-on-coil storage system was selected. In this technology, a transport fluid is utilized for the heat transfer. The transport fluid that is cooled to a temperature between -3 to -6 ^{o}C in the chiller is circulated through the coils submerged in a water-filled tank during ice making mode (charging). Ice is formed from the water on the outside of the coils. When cooling is needed, the transport fluid is circulated through the coils and then directed to the load (discharging). It melts the ice from the inside out during which temperature of the working fluid reduces. Water-glycol solution is usually used as transport fluid.

In this study, the heat transfer during the charging between the water inside the storage tank and the transport fluid circulating inside the coil tubes was modeled for sizing the coil. The heat transfer process is shown schematically in Fig. 1.

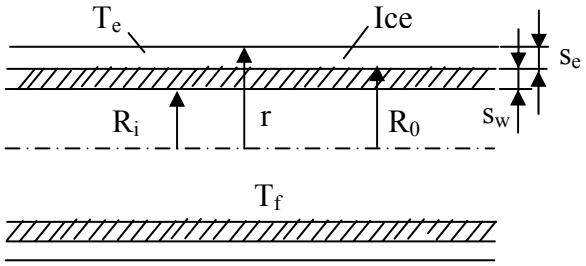


Fig. 1. Schematic of the ice production process.

If only a small length of the tube is considered, the temperature of the transport fluid (T_f) flowing through this part can be assumed constant. At this axial position of the tube, the amount of heat transferred per length of the tube ($\dot{Q}_x$) can be calculated from;

$$\dot{Q}_x = \frac{T_e - T_f}{R_p} \quad \textbf{(1)}$$

where T_e is the temperature of the ice and R_p is the thermal resistance based on the peripheries, instead of the surface areas;

$$R_p = \frac{1}{h_i P_i} + \frac{s_w}{k_w P_{wm}} + \frac{s_e}{k_e P_{em}} \quad \textbf{(2)}$$

in which k_w and k_e are the thermal conductivity of the tube material and the ice, respectively. h_i is the convective heat transfer coefficient inside the tube. The peripheries (P_i, P_{wm} and P_{em}) are obtained from the following equations;

$$P_i = 2\pi R_i \quad \textbf{(3)}$$

$$P_{wm} = \frac{2\pi(R_o - R_i)}{\ln \frac{R_o}{R_i}} \quad \textbf{(4)}$$

$$P_{em} = \pi(2R_0 + s_e) \quad \textbf{(5)}$$

In the last equation, it was assumed that the ice thickness s_e is very small compared with R_0, so that the arithmetic mean value can be used instead of logarithmic one. Defining;

$$R_{p0} = \frac{1}{h_i P_i} + \frac{s_w}{k_w P_{wm}} \quad \textbf{(6)}$$

and

$$R_p^* = R_p / R_{p0} \quad \textbf{(7)}$$

the following equation is obtained from Eq. (2);

$$R_p^* = 1 + \frac{s_e}{k_e R_{p0} \pi (2R_0 + s_e)} \quad \textbf{(8)}$$

The heat transferred from water to the transport fluid flowing inside the tube causes ice production on the outer surfaces of the tubes. Therefore Eq. (9) can be written;

$$\dot{Q}_x \, dt = \rho_e h_{ew} 2\pi r dr \quad \textbf{(9)}$$

where ρ_e is the density of the ice, h_{ew} is the latent heat of freezing of water and t is the time. Using Eqs. (1) and (9) the following equation is obtained;

$$r \frac{dr}{dt} = \frac{T_e - T_f}{2\pi . h_{ew} . \rho_e . R_p} \quad \textbf{(10)}$$

Utilizing the following relations,

$$r = (R_0 + s_e) \quad \textbf{(11)}$$

$$s_e^* = s_e / R_0 \quad \textbf{(12)}$$

$$t^* = t \frac{T_e - T_f}{2\pi R_0^2 h_{ew} \rho_e R_{p0}} \quad \textbf{(13)}$$

Eq. (14) can be derived from Eq. (10);

$$(1 + s_e^*) \left[1 + \frac{s_e^*}{R_{p0}^* (1 + s_e^* / 2)} \right] ds_e^* = dt^* \quad \textbf{(14)}$$

where

$$R_{p0}^* = 2\pi R_{p0} k_e \quad \textbf{(15)}$$

Integrating differential equation (14) with the following boundary condition;

$$s_e^* = 0 \quad : \qquad t^* = 0 \quad \textbf{(16)}$$

Eq. (17) can be derived for the relationship between dimensionless time t^* and dimensionless ice thickness s_e^*;

$$t^* = s_e^* \left(1 - \frac{2}{R_{p0}^*} \right) + \frac{s_e^{*2}}{2} \left(1 + \frac{2}{R_{p0}^*} \right) + \frac{4}{R_{p0}^*} \ln(1 + s_e^* / 2) \quad \textbf{(17)}$$

For the calculation of P_{em}, If we use

$$P_{em} \approx 2\pi R_0 \quad \textbf{(18)}$$

instead of Eq. (5), the following equation for the relationship between t^* and s_e^* is obtained;

$$t^* = s_e^* + \frac{s_e^{*2}}{2}\left(1+\frac{1}{R_{p0}^*}\right) + \frac{s_e^{*3}}{3R_{p0}^*} \quad \textbf{(19)}$$

The term $\left(1+s_e^*/2\right)$ in Eq. (14) can be approximated with;

$$\left(1+s_e^*/2\right) \approx 1 - \frac{s_e^*}{2} + \frac{s_e^{*2}}{4} \quad \textbf{(20)}$$

In this case, the following solution is obtained:

$$t^* = s_e^* + \frac{s_e^{*2}}{2}\left(1+\frac{1}{R_{p0}^*}\right) + \frac{s_e^{*3}}{6R_{p0}^*} \quad \textbf{(21)}$$

In Fig. 2, s_e^* is demonstrated as a function of t^* for different values of parameter R_{p0}^* (1, 3 and 10) using the exact solution (Eq. 17) and the approximate solutions (Eqs. 19 and 21). As can be seen from the Fig., dimensionless ice thickness s_e^* increases with dimensionless time t^* and also with the parameter R_{p0}^*. The approximate solutions (Eqs. 19 and 21) produce smaller ice thickness than the exact solution (Eq. 17). However, the differences between the equations reduce with the increase of parameter R_{p0}^*. For R_{p0}^*=10, the differences between the equations are insignificant.

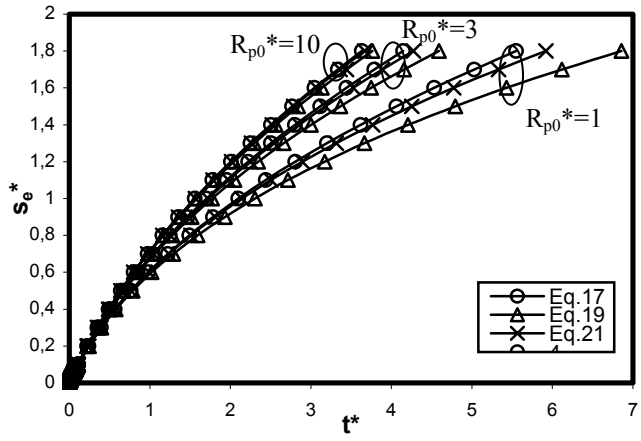


Fig. 2. Variation of dimensionless ice thickness s_e^* with dimensionless time t^* for different values of R_{p0}^*.

EXPERIMENTAL SETUP

Three equations (Eqs. 17, 19 and 21) were developed above for the calculation of dimensionless ice thickness s_e^* at any axial location of the coil tube. The models developed were tested experimentally by designing a small-sized cool storage system. For this purpose, a room located inside the Laboratories of Mechanical Engineering Department of Çukurova University, Adana, Turkey was chosen for the application of the ice storage based air-conditioning.

The dimensions of the room chosen are 6x5.8x2.2 m and the walls are not insulated. Only the south and the north walls of the room face to outside. The room has two windows on the south and two on the north. Since the room is an office room, it is used between 8^{00} and 17^{00} h. Therefore, all the size calculations were carried out for this period. The maximum daily total cooling load requirement in a year between 8^{00} and 17^{00} h (Q_d) was calculated to be 42 kWh.

General view of the ice storage based air-conditioning system is shown in Fig. 3. Although the system can be operated both under full-storage and partial storage load leveling operating strategies, only full-storage is considered in this study. The experimental system included a water chiller, an ice storage tank, interconnecting piping, controllers (two and three way valves), measuring equipment, and data collection system. The glycol-water solution with 30% glycol was used as the transport fluid.

To cool the transport fluid, an air-cooled chiller (McQuay MHP 04P) with a nominal capacity of 11.3 kW was chosen [5]. The chiller comes with a standard pumping unit.

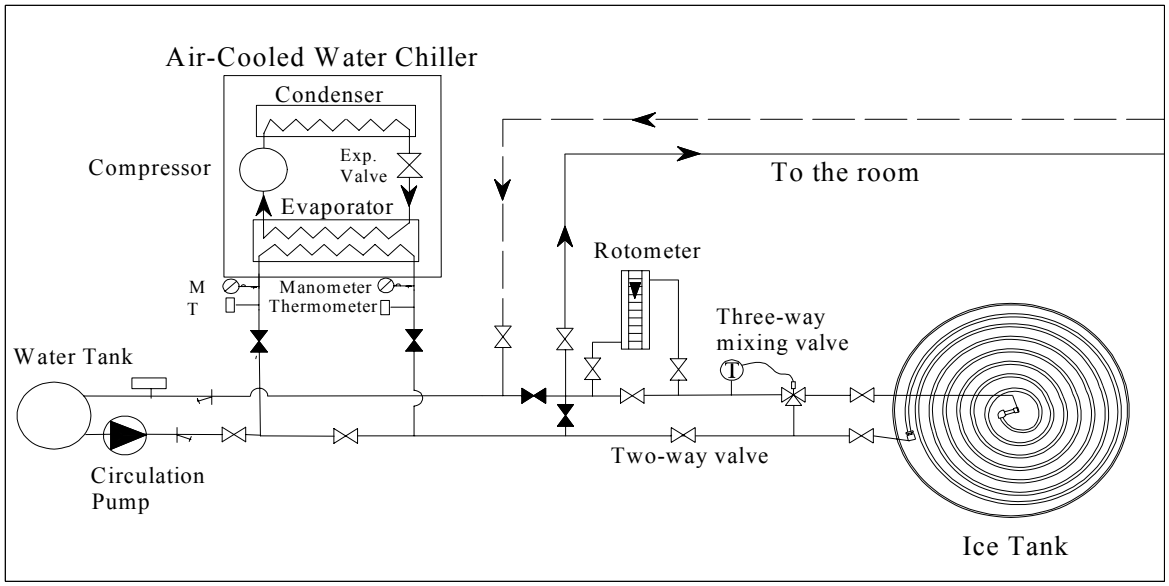


Fig. 3. Schematic representation of the experimental setup.

The ice storage tank should be sized to meet the daily total cooling requirement. The amount of ice (m_e) that should be produced during the charging cycle to meet the daily total cooling requirement can be calculated from:

$$m_e = \frac{Q_d}{h_{ew}} \qquad \textbf{(22)}$$

With Q_d=42 kWh and h_{ew}=0.093 kWh/kg, m_e is found to be 452 kg.

For the sizing of the coil needed for storing the daily cooling demand, variation of ice thickness with time is needed. For this purpose, the equations (Eqs. 17, 19 and 21) developed above were used.

For the construction of the coils of the ice storage tank, polyethylene pipe with 16 mm outside diameter (R_o) and 2 mm thickness (s_w) was used. Thermal conductivity of the pipe material (k_w) is 0,36 W/mK. The pipe was selected among the various pipes that are commercially available in Turkey considering the heat transfer characteristics, easy construction and cost. It is commercially sold for construction of under-floor heating systems. For the ice, thermal conductivity (k_e) is 2.2 W/mK, and density (ρ_e) is 917 m^3/kg.

For the calculation of R_{po}, the convective heat transfer coefficient inside the tube h_i is needed. It can be calculated using the well-known Dittus-Boelter equation for the turbulent flow inside the tubes, if the velocity (Reynolds number) of the transport fluid is known. It can be estimated from the properties of the chiller pumping unit (≅0.6 l/s).

Using the data given above and the definitions of s_e^* (Eq.12) and t^* (Eq. 13), variation of ice thickness with time can be obtained from Eqs. (17, 19 and 21) for any temperature difference between the ice and the transport fluid flowing inside the tube (T_e-T_f), which varies along the tube. During the ice formation, temperature of the water is constant around 0 °C, and temperature of the transport fluid increases along the tube. Therefore, temperature difference decreases along the tube. T_f at the inlet of the coil tube ($x=0$), will be slightly higher than the set temperature at the exit of the chiller unit and T_f at the exit of the coil tube ($x=L_t$) will be a function of the tube length (L_t). In practice, temperature difference (T_e-T_f) will be around 6 °C at the tube inlet ($x=0$), and 3 °C at the tube exit ($x=L_t$).

In Fig. 4, variation of ice thickness with time is shown for three different (6, 4.5, and 3 °C) values of (T_e-T_f). The s_e values were obtained from Eq. (17). As can be seen from the Fig., ice formation will be faster for high temperature differences (T_e-T_f). 13700 s (≅ 4 h) after the start of the ice formation, approximately 14.5 mm ice will form around the tube at the inlet part of the tube (high T_e-T_f), whilst ice thickness will be approximately 9.5 mm at the exit of the tube (low T_e-T_f). It should be noted that the total charging period is 8 h, but during the first a few hours, the chiller unit works for reducing the temperature of the water inside the tank down to 0 °C from the initial water temperature. It can be seen from Fig. 4 that at the end of 8 h operation of the chiller during the off-peak period, approximately 15 mm of ice will be formed around the coil tube.

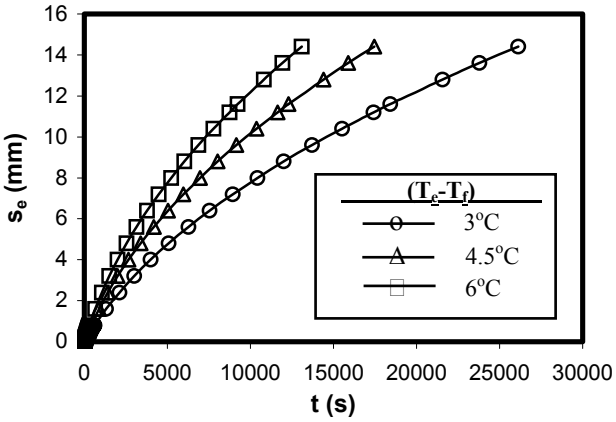


Fig. 4. Variation of ice thickness with time for different values of (T_e-T_f).

The total length of the coil tube (L_t) required for producing m_e can be calculated from :

$$L_t = \frac{(m_e / \rho_e)}{\pi(r^2 - R_o{}^2)} \qquad \textbf{(23)}$$

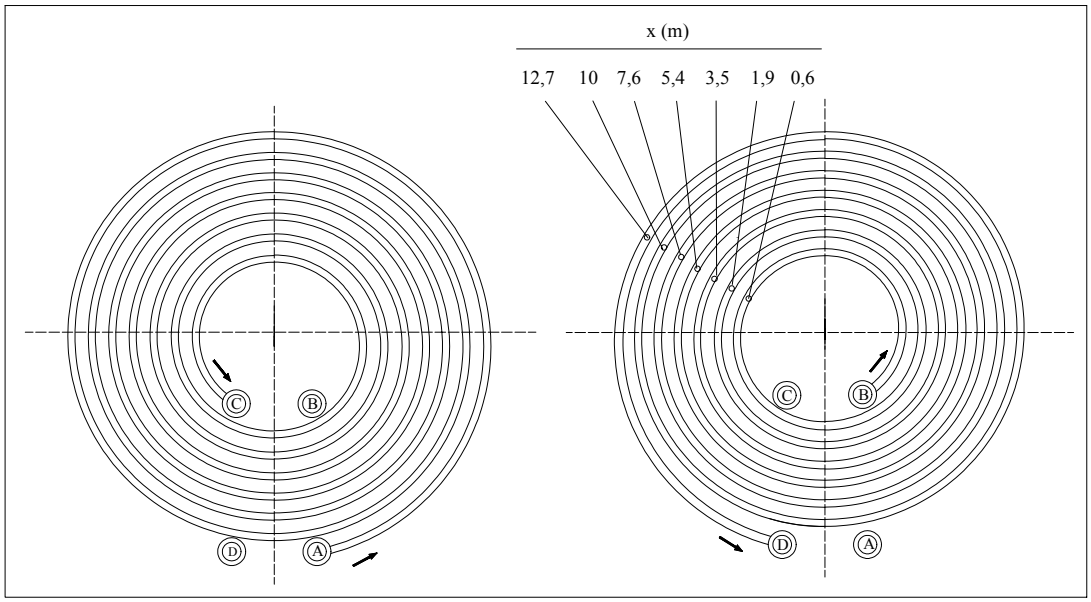
Using the data given above L_t is found be approximately 340 m with an average temperature difference of 4 °C.

A circular polyethylene tank having a diameter of 1 m and an overall height of 1,5 m was used for the construction of ice storage tank. The tank was insulated using 9 mm thick expanded nitrile rubber. 340 m long polyethylene coil pipe was divided into 26 parts and each part (13 m long) was wounded (Fig. 5) to form a coil. The horizontal distance between two wounds was chosen to be 30 mm. The coils were replaced into the storage tank with a vertical distance of 30 mm between them. The horizontal and vertical spaces between the pipes were chosen for enough ice formation on each pipe (15 mm on each pipe).

The coils were connected from the both ends to polypropylene collector tubes having 42 mm inner diameter. Two of the collectors serve as supply and the other two as return collectors (Fig. 5). For the 1st, 3rd, 5th, … coils, the transport fluid cooled in the chiller unit is supplied from the outer side of the coil (A) and the warm fluid is collected from the inner side (B). For the 2nd, 4th, 6th … coils, supply is from the inner side (B) and collection is from the outer side (D). This arrangement allows a uniform ice formation around the coils.

RESULTS

The internal melt ice-on-coil system constructed was charged during off-peak period (22^{00}-06^{00}). The transport fluid (70% water + 30% glycol) was cooled in the air-cooled chiller unit and circulated through the coils located inside the ice storage tank. The transport fluid firstly reduces the temperature of the water inside the tank down to 0 °C and then ice formation starts. Fig. 6 shows the variation of measured ice thickness with time at the locations shown in Fig. 5(b). The initial water temperature was 15 °C for this experiment and ice formation started approximately 8000 s after the start of the experiment. After ice formation started, (T_e-T_f) was almost constant during the charging period and it was approximately 6 °C at the inlet of the coils (*x=0*) and 3 °C at the exit (*x=13* m). At the end of the charging period, approximately 15 and 10 mm ice was formed at the inlet and the exit of the coils, respectively. The change in the ice thickness was insignificant for *x>7.6* m [6].



(a) **(b)**

Fig. 5. Coil distribution arrangement in the tank. (a) 1st, 3rd, 5th, … coils (b) 2nd, 4th, 6th … coils.

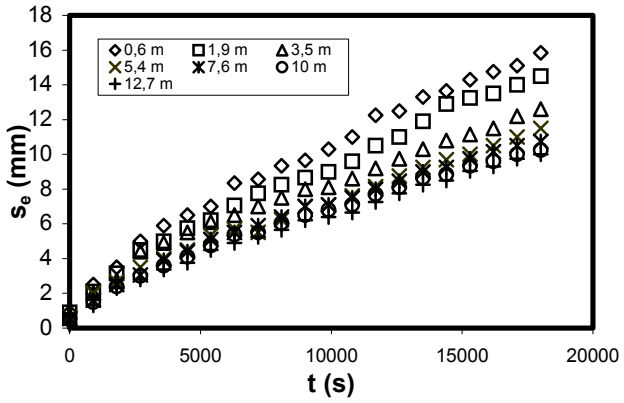


Fig. 6. Measured ice thickness at different locations (x) along a coil.

Experimentally obtained ice thickness values were compared with the result of the model in Fig. 7. Using the model given above, ice thickness was calculated for two different values of (T_e-T_f) that were observed during the experiment at the inlet (6 ^oC) and at the exit (3 ^oC). As can be seen from the Fig., the agreement between the measurement and the results of the model is good, although the model predicts slightly higher ice thickness at the exit.

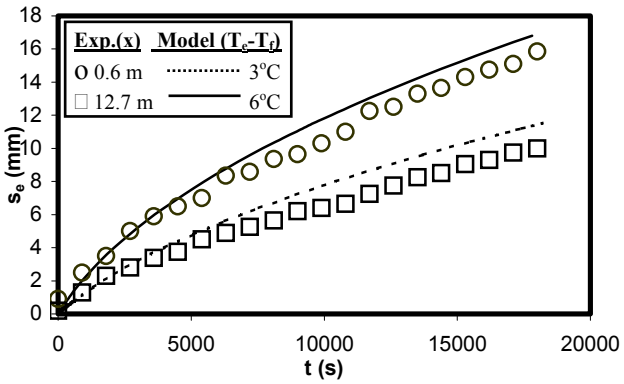


Fig. 7. Comparison of the ice thickness obtained from measurements and from the model.

CONCLUSIONS

The heat transfer process between the ice and the transport fluid flowing inside the coil tubes of an internal melt ice-on-coil system was modeled with some simplifying assumptions. Using the model, variation of dimensionless ice thickness with dimensionless time was obtained for a given temperature difference between the ice and the transport fluid. Calculating daily total cooling load of the room and selecting a suitable tube, variation of ice thickness with time was obtained. This information was used for the calculation of tube length. Results of the experiments were compared with that of the model and it was seen that the simple model predicts the ice formation with a good approximation.

Considering the rapid growth of Turkey's energy consumption and imports in recent years, use of cool storage systems for cooling and use of ice as cool storage medium in these systems should be encouraged.

NOMENCLATURE

$\dot{Q}_x$	heat transfer along the tube, W
Q_d	maximum daily total cooling load, kWh
c_s	specific heat of transport fluid, kJ/kg^{0}C
h	heat transfer coefficient, W/m^{2}K
k	thermal conductivity, W/mK
L	length, m
m_e	amount of ice, kg
P	periphery, m
R_p	thermal resistance based on peripheries
r	radius, m
R	tube radius, m
s	thickness, m
s_e^*	dimensionless ice thickness
t^*	dimensionless time
t	time, s
T	temperature , ^{0}C

Greek Letters

ρ	density , kg/ m^3

Subscripts

e	ice
f	transport fluid
i	inner
m	wall mean
o	outer
w	wall

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