EFFECT OF THERMAL ENERGY STORAGE IN ENERGY CONSUMPTION REQUIRED FOR AIR CONDITIONING SYSTEM IN OFFICE BUILDING UNDER THE AFRICAN MEDITERRANEAN CLIMATE

by

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In the African Mediterranean countries, cooling demand constitutes a large proportion of total electrical demand for office buildings during peak hours. The thermal energy storage systems can be an alternative method to be utilized to reduce and time shift the electrical load of air conditioning from on-peak to off-peak hours. In this study, the Hourly Analysis Program has been used to estimate the cooling load profile for an office building based in Tripoli weather data conditions. Preliminary study was performed in order to define the most suitable operating strategies of ice thermal storage, including partial (load leveling and demand limiting), full storage and conventional A/C system. Then, the mathematical model of heat transfer for external ice storage would be based on the operating strategy which achieves the lowest energy consumption. Results indicate that the largest rate of energy consumption occurs when the conventional system is applied to the building, while the lowest rate of energy consumption is obtained when the partial storage (demand limiting 60%) is applied. Analysis of results shows that the new layer of ice formed on the surface of the existing ice lead to an increase of thermal resistance of heat transfer, which in return decreased cooling capacity.

Key words: *thermal storage, heat transfer, energy consumption,*

Introduction

Libya, in the African Mediterranean climate zone, that has an electric energy demand is expected to grow extremely rapidly. Libyan Energy makers and government organizations expect that demand for electrical power will double by 2014 and it will be more than two and half by the end of the year 2020. [1] A major portion of electricity demand is required mainly for A/C systems in commercial and office buildings as a result of extremely high ambient temperature, which sometimes reaches 46 ^{o}C during the daytime, thus increasing the high cooling demined in Libya. The TES technology has great potential to become one of the primary solutions to shifting the peak electrical loads to off peak periods as result of imbalance between production and demand energy [2]. In general, TES may take place either as a sensible process: chilled water, in which the energy is stored in a change in temperature of a material, or a latent process, ice-making, in which the energy changes in a substance as it undergoes a

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change in phase. The advantage of latent storage is that the energy released or absorbed can be done so at a constant temperature. Although sensible cold TES is quite simple and cost effective, the size of these devices is quite large when compared to ice storage, therefore the latent TES has received much more attention in recent years. The latent TES, ice-making types are classified by the heat transfer process in which the storage tank is either a static process, in which heat transfer takes place via a solid surface which includes both external and internal melt ice-on-coil and encapsulated ice thermal storage. The ice slurry and ice-harvesting storage systems belong to the dynamic process, in which the heat transfer medium and storage medium are in direct contact. Most of the latent TES, ice-making types that are incorporated with conventional A/C system are static. The review of various types of TES techniques is presented in [3]. The external melt ice on coil storage system is widely used because it can provide supply water of low temperature equal to dynamic ice storage systems. It is also more easily controlled than dynamic systems during charge and discharge processes [4]. The TES has been researched extensively worldwide; however, none of the existing studies uses Libya as the typical design day conditions. Hasnain et al. [5. 6] investigated the prospects of using cold thermal storage for office buildings in the hot weather of Saudi Arabia. M.J. Sebzali et al. [7] analyzed the ice cool thermal storage system in a clinic building based on the typical design day conditions of Kuwait. Rosen and Dincer [8] highlighted two critical factors. Firstly using appropriate TES efficiency measures and secondly, the importance of temperature in TES evaluations by considering energetic, exergetic, environmental and sustainability aspects of TES. Chen et al. [9]. In their economic analysis, the objective function included initial cost and energy cost of the system, and optimal parameters of the system were ultimately obtained. Many electrical utilities have adopted time of use tariff, which opened a wide door for the use of energy storage systems. Therefore, there are many studies related to the theoretical modeling and cost analysis of TES. Habeebullah [10] investigated the economic feasibility of both building an ITS and structure a time of rate tariff for the unique A/C plant of the Grand Holy Mosque of Makkah, Saudi Arabia. Despite the extensive literature reported and published worldwide regarding TES systems technology, none of the efforts consider the effect of TES on electrical energy consumptions in office buildings in Libya. This paper presents an analysis of TES that applies to a conventional A/C system in a Libyan office building for typical design day conditions. The conventional A/C system was compared with different (TES) operating strategies in order to determine the case which achieves the highest potential of energy saving for our case study. The mathematical model of heat transfer for the chosen operating strategy is developed using the Math (CAD) program in order to predict the chiller capacity of storage and to satisfy the cooling demand of an office building for design day conditions. The capacity of chiller is modified by the evaporating temperature that changes due to the change of the thickness of ice on the horizontal tube.

Preliminary study

The operating strategies of TES are often classified as either full storage or partial storage. Partial storage systems can be sized for load-leveling or demand- limiting operation. These terms refer to the amount of on-peak cooling load that is shifted to off-peak. The amount of the considered case's load met by the storage is dependent on the operation strategies of TES. Therefore, operating strategy plays an important role in the energy consumption in each case. Hence, the following preliminary study was performed in order to determine which strategy has the lowest energy consumptions in our case study. This chosen operating strategy which achieves lowest energy consumptions would be adopted in the mathematical model of the heat transfer for storage.

Case study

An office building modeling with conventional A/C system is situated in a hot humid climate zone (Tripoli) where reliance on A/C increases significantly. The weather and building construction data have been modified in hourly analysis program (HAP) in order to estimate the cooling load profile for the considered building in the selected city. The HAP takes into account the entire building envelope, internal loads such as occupancy, lighting, electric loads, and building thermal mass to calculate cooling load profile. The office building consists of three floors, each with 1152 m^2 of floor surface area and 3.2 m of height. The fan coil system is designed to satisfy the cooling demand necessity of office building during occupied hours (from 7.00 a.m. to 7.00 p.m.) only. The maximum building load reaches the value of 484 kW on July 21st at 4 pm, when outdoor temperature is 42°C and the total integrated system cooling load is 4953 kWh. The conventional A/C system chiller capacity has to be determined for comparison with TES. In common practice sizing of the chiller is selected equal to maximum cooling load which is 484kW in this study. The chiller operates by four screw compressors with swept volume and compressor capacity for each 295.7m3/h and 122.7 kW respectively.

Evaluation of storage and chiller capacities for different operating strategie

The cost of electricity in Libya does not differentiate between on-peak and off-peak periods, therefore, the selection of charging and discharging time must be based on the profiles of both the cooling load of the office building and outdoor temperature. This means that selection of charging storage period must be at night (after 10 pm) where outdoor temperature is relatively low for better chiller performance. The total integrated load is equal to the total cooling loads of the office building during operating hours for the design day. According to the cooling load profile of the selected building and the applied operating strategies (charging and discharging time), the sizes of chiller and storage capacities were obtained by applying eq. (1) and eq. (2). [11].

$$q_{chil} = \frac{\text{total intgrated system cooling load kWh}}{H_{char} \times CR_{char} + H_{dirt} \times CR_{dirt}} \tag{1}$$

$$\text{Storage capacity} = q_{chil} \times H_{char} \times CR_{char} - TC_{chair} \tag{2}$$

Where the chiller capacity ratio CR_{char}, CR_{dirt} can be expressed as the capacity of each cooling mode percentage to its nominal capacity. Since the typical temperature of refrigerant R134a during the period of charging and discharging is usually in the range of -6.5- -2.5°C the capacity ratios of the chiller are therefore assumed equal to 1, and TC_{char} is the total integrated system cooling load during charging time only for sizing storage capacity. BITZER compressors selection software was used to select the required compact screw compressor that could be used in the air cooled chiller and to determine its capacity for different condensing temperatures (40°C, 50°C). A mean evaporating temperature of -4.5°C and useful superheat of 3 K were adapted for day and night modes. The hourly cooling capacity profile for different operating strategies is shown in fig.1. In the full storage operation strategy, the refrigeration equipment does not run during on-peak period and all cooling loads are met by the storage. In this operating strategy the chiller with a capacity of

381kW operates for only 13 hours during off-peak period when there is no cooling load. This means that the entire cooling load is directly met by the storage during on-peak hours as shown in fig.1. On-peak hours refer to the period of building occupied hours while off-peak hours refer to unoccupied hours.

Using a load-leveling operation strategy, the refrigeration equipment runs on full capacity for 24 hours. However, the chiller capacity in this case is only 237kW which starts to charge the storage at about 07:00 pm and ends at about 07:00 am. The refrigerant R134a circulated with a variation of evaporating temperature from -2.5- -6.5°C for approximately 13 h running at chiller full load to fully charge the storage. Besides, during night hours the performance of the chiller is enhanced due to decreased of outdoor temperature which results in an increased of cooling capacity. In 55%, 60% and 65% load ratio of demand limiting partial storage operation strategy, the refrigeration equipment operates at demand level during the on-peak period. The portion of integrated cooling load during on-peak hours will be met by the storage and the rest will met by the chiller. In this case, the charging time during off-peak hours is varying from 5 to 9h depending on the amount of cooling load which would be covered by the storage during on peak hours. The conducted results of sizing the chiller and storage capacities for different operating strategies are summarized in tab.1.

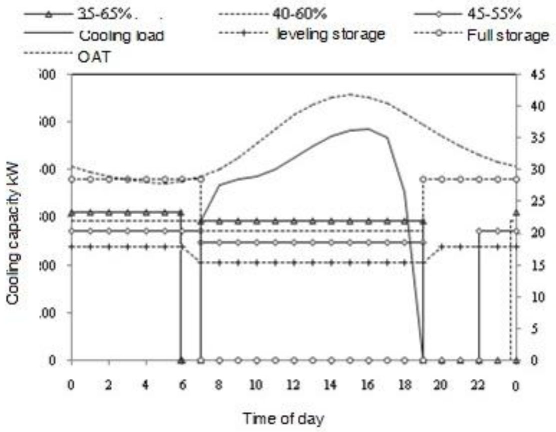

Figure 1. Hourly cooling capacity profile for different operating strategies, outdoor temperature and cooling load.

Estimation of energy consumption for different operating strategies

Generally, the input power requirement of the chiller increases as the load on the chiller rises. For a conventional system as can be seen in table 1, the maximum power demand occurs during the peak load, which reaches 198 kW at about 04:00 pm. The lowest power demand during occupied hours of the building occurs in the first and last hours of occupancy (7-8 am and 6-7 pm). However, the opposite trend occurs when the chiller operate with the full storage strategy. In the conventional system the maximum cooling load occurred during afternoon hours when outdoor conditions were extremely high and consequently, the power consumption was high as well.

In the full storage operating strategy, the chiller operates during night hours when outdoor conditions are considerably lower compared with day conditions which resulted in a decrease in the required input power for the chiller. Furthermore, as can be seen in the table 1, 1939.6 kWh of energy was required to operate the chiller with a capacity of 381kW during off-peak hours to produce a sufficient amount of ice to meet the building is cooling load during on-peak hours, while a value of 2059.9 kWh of energy was required when applying the conventional system only.

Table 1 Chiller and storage capacities sizes and energy consumption for different operating strategies

Items		Conventional	Full storage	Partial storage			
		-	-	Load leveling	Demand Limiting		
					55%	60%	65%
load met by chiller kWh		4953	0	2244	2724.1	2971.8	3219.4
Storage capacity kWh		0	4953	3081	2228	1981.2	1733.5
Charging hours h		0	13	13	9	7.33	5.9
Discharging hours		11	11	11	11	11	11
Chiller capacity kW	Night	0	381	237	271	294.6	312.4
	Day	484	0	204	247.6	270.6	292.6
Power input	Night	0	149.2	69.2	83.4	87.8	95
	Day	At max load 198	0	84.9	102.2	107.2	116.4
COP	Night	0	2.29	3.29	3.12	3.21	3.15
	Day	2.4	0	2.3	2.18	2.25	2.2
Energy consumption kWh		2059.9	1939.6	1833.5	1874.8	1823	1843

Condensing temperature: 50°C during day, 40°C during night. Evaporating temperature: -4.5°C during day& night.

For the load leveling operating strategy, the value of the input power varies from day to night due to different outdoor conditions as mentioned previously. The total required energy for the chiller during day and night is 1833.5 kWh.

Required chiller power input values were obtained in the scenarios where 55%,60% and 65% of the cooling load was supplied by storage, and the remainder supplied by chiller. As can be observed, the lowest value of power input can be achieved with 55% scenario of the demand limiting strategy because the cooling capacity of chiller is less during day and night compared with other designs of demand limiting scenarios. However, the longest charging period is for 55% demand limiting storage. The total energy required to operate the chiller day and night in the three scenarios 55%, 60% and 65% were 1874.8, 1823 and 1843 kWh, respectively. A preliminary study was conducted (which it is summarized in tab.1) the 60% scenario of partial storage demand limiting strategy achieves the lowest energy consumption compared to other the operating strategies that were considered in the study. Therefore, the following modeling of heat transfer will be based on this type of operating strategy for the partial storage demand limiting.

Heat transfer modeling of the 60% limiting partial thermal storage

A schematic diagram of the external ice thermal storage (EITS) system considered in this study is demonstrated in fig.2. This system includes two main cycles. The primary cycle includes refrigeration cycle equipment with an ice storage tank while the secondary cycle includes air handling unit (AHU), heat exchanger, ice storage tank and circulating pumps for ice and chilled water. In this system, R134a refrigerant is used as the heat transfer fluid and it is circulated in the pipes throughout the tank with temperatures from -2.5 to -

6.5°C during operating strategies. The dimensions of the rectangular storage tank were designed as 2.31m height, 2.17m width and 4.068m. The tank has two sections, each section has 14 pipes, and each pipe is 5m in length which consists of 19 rows in horizontally stacked. The pipes have inner and outer diameters of 0.034m and 0.038m respectively. The vertical distance between the center lines of the pipes is 0.118m with total length of pipes is 2660 m.

Charging Process

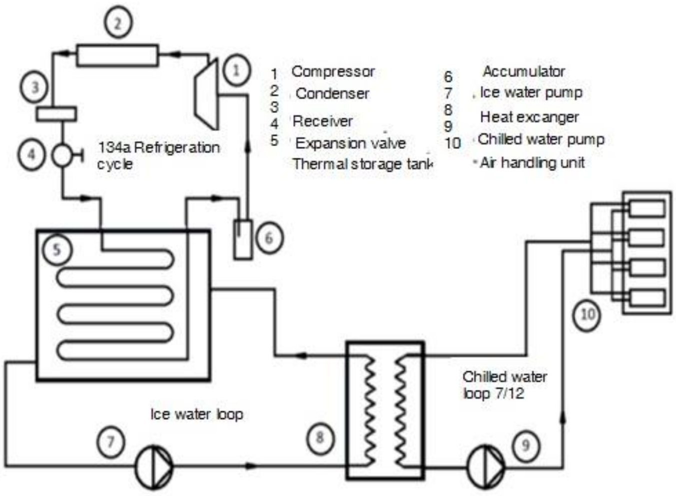

Figure 2. Schematic diagram of the external ice thermal storage system

The charging cycle of the ice storage tank is used during off-peak demand hours (at night after 10 pm) and requires the inlet temperature of the R134a refrigerant to be below 0 °C when it passes through pipes submerged inside storage tank. Ice starts to be accumulated on the outside of the submerged pipes as soon as the temperature of water inside the tank reaches close to 0°C. The compressor of the chiller is switched off once the thickness of ice around the pipes reaches the desired set-point thickness of ice. It must be noted that other two cycles in the system should be turned off during the charging process.

Discharging Process

In the discharging cycle, the storage is discharged by circulating the ice return water over the pipes thus, melting the ice. The ice return water exchanges the heat with the chilled water return from the building load by means of the heat exchanger. However, in this study the refrigeration cycle is on simultaneously with the discharging process.

Mathematical analysis

In order to thermodynamically model the EITS system, a set of assumptions must be taken into consideration based on the procedure of developing the system model.

- The storage tank temperature distribution was assumed as 0.5°C.
- The states of the refrigerant R134a at the EITS and condenser outlets were considered as saturated vapor and saturated liquid, respectively.
- No heat transfer exists from the storage to the environment and adjacent pipes.

- The ice growth is assumed to be uniform along the length of the tube.
- Vertical Agitators create a flow rate of water inside the tank of 0.1 ms^{-1}.
- The total volume of ice formed around pipes was calculated by using the following equation.

$$V_{ice} = \left[d_{ice\max}^2 - d_0^2 \right] \cdot \frac{\pi}{4} \cdot L_t \tag{3}$$

The total storage capacity can be determined by the eq. (4).

$$Q_{ice} = \frac{1}{3600} \cdot \left(m_{ice} \cdot Q_f \right) \tag{4}$$

where m_{ice} is the mass of ice, which is equal to volume of the ice formed around the horizontally stacked pipes multiply by the density of ice [920 kg m^{-3}]. The total area available for water flow can be obtained by eq. (5).

$$A_{\mathrm{w}} = \left[W_{\mathrm{s}} \cdot 2 \right] \cdot H_{\mathrm{w}} - N_{\mathrm{z1}} \cdot n_{\mathrm{y}} \cdot d_{\mathrm{icemax}}$$

The volume of water flow in the storage tank is equal to the available area of water multiplied by velocity of water. The thermo physical properties of the water are calculated based on a principle suggested by *ASHRAE 2009*. [12], where it is assumed that the starting temperature of ice formation around the pipes is 0.5°C. The mathematical model is divided into the following two subsections: sensible charging process and latent charging process. The sensible charging process lowers the temperature of the water to freezing temperature. It is assumed that no latent charging occurs in this period. The overall heat flux between the refrigerant R134a and the water flow based on the outside area of pipes is given in the following equation (cooling capacity of evaporator).

$$Q_{\mathrm{o}} = \frac{1}{100} \cdot \left[L_{\mathrm{c}} \cdot U_{\mathrm{tot}} \cdot \Delta T_{\mathrm{mw}} \right] \tag{6}$$

where U_{tot} is the total heat transfer coefficient can be obtained by eq. (7):

$$U_{\mathrm{tot}} = \left[\frac{1}{d_{\mathrm{u}} \cdot \pi \cdot \alpha_{\mathrm{uw}}} + \frac{1}{2 \cdot \pi \cdot \lambda_{\mathrm{cel}}} \cdot L_{\mathrm{n}} \cdot \left(\frac{d_{\mathrm{s}}}{d_{\mathrm{u}}} \right) + \frac{1}{d_{\mathrm{s}} \cdot \pi \cdot A_{\mathrm{lfaw}}} \right]^{-1} \tag{7}$$

The first term in eq. (7) represents the convective resistance inside the tube, while the second term represents the resistance of the tube, and the third term is the convective resistance outside the tube. The heat transfer coefficient inside the pipe α_{uw}, was determined using the following equation.

$$\alpha_{\mathrm{uw}} = C_{\mathrm{1w}} \cdot \left(1000 \cdot m_{\mathrm{zR}} \cdot \frac{q_{\mathrm{uz}}}{\mathrm{d_u}} \right)^{0.5} \tag{8}$$

The correlation C_{1w} for turbulent flow -2.5°C refrigerant R134a in a circular pipe was defined as following [13].

$$C_{\mathrm{1w}} = 0.13 + \left(\frac{0.06}{40} \right) \cdot \left(40 + T_{\mathrm{ow}} \right) \tag{9}$$

The m_{zR} is mass velocity of refrigerant inside the pipe and q_{uz}, the heat flux per one square meter of internal surface of the pipe, are obtained by eq. (10) and eq. (11) respectively.

$$M_{\mathrm{zR}} = \frac{\mathrm{Q_{oc}} \cdot n_{\mathrm{cR}}}{\frac{n_{\mathrm{z}}}{2} \cdot h_{\mathrm{v}} \cdot d_u^2 \cdot \frac{\pi}{4}} \tag{10}$$

$$\mathrm{q_{uz}} = \frac{Q_{\mathrm{oc}}}{\frac{\mathrm{A_u}}{2}} \tag{11}$$

The heat transfer coefficient of the water A_{lfaw}, is determined using eq. (12).

$$\mathrm{A_{Lfaw}} = \frac{N_{\mathrm{uD}} \cdot \lambda_{\mathrm{w}}}{\mathrm{L_s}} \tag{12}$$

Where NuD, Re and Pr are calculated based on a principle suggested by *ASHRAE 2009* [12]. The mean temperature ΔT_{mw} of water is defined by eq. (13).

$$\Delta T_{\mathrm{mw}} = \frac{T_{\mathrm{wr}} - T_{\mathrm{w}}}{\mathrm{Ln} \cdot \left(\frac{T_{\mathrm{wr}} - T_{\mathrm{ow}}}{T_{\mathrm{w}} - T_{\mathrm{ow}}} \right)} \tag{13}$$

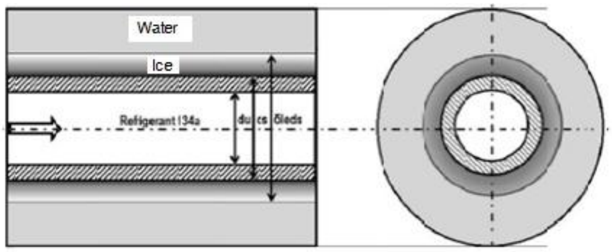

Figure 3. Schematic diagram for ice formation around pipes

T_{wr} is the return ice water temperature, which is assumed to be 4°C. By applying eq. (7) and eq. (13) in eq. (6), the cooling capacity of the evaporator during the sensible charging process (prior to ice layer formation) can be obtained. When water temperature approaches 0°C, the latent charging process begins and causes the onset of ice formation around the storage pipes. As a consequence, a new resistance factor (that of the ice layer to the heat flux) must be included in the U_{tot} expression. The overall heat transfer coefficient U_{totL} during latent charging process between refrigerant R134a and the water based on the outside area of ice is given in the following equation:

$$U_{\mathrm{tot\,L}} = \left[\frac{1}{d_{\mathrm{u}} \cdot \pi \cdot \alpha_{\mathrm{uw}}} + \frac{1}{2 \cdot \pi \cdot \lambda_{\mathrm{cel}}} \cdot L_{\mathrm{n}} \cdot \left(\frac{d_{\mathrm{s}}}{d_{\mathrm{u}}} \right) + \frac{1}{d_{\mathrm{s}} \cdot \pi \cdot A_{\mathrm{LedL}}} \cdot L_{\mathrm{n}} \cdot \left(\frac{\delta_{\mathrm{Ledsp}}}{d_{\mathrm{s}}} \right) + \frac{1}{\delta_{\mathrm{Ledsp}} \cdot \pi \cdot \alpha_{\mathrm{sw}}} \right]^{-1} \tag{14}$$

The outer diameter of the ice δ_{Ledsp} shown in fig.3.The third term in eq. (14) is the resistance of the ice that is formed on the outside of the pipes. The α_{sw} is the heat transfer coefficient of the water of 5000Wm^{-1}.K^{-1}.The total heat transfer coefficient is calculated using the same correlations as the sensible charging process in eq. (8). The evaporating temperature and the thickness of ice reached the values of -6.5°C and 0.035m, respectively. However, since the ice geometry is cylindrical, the new formed ice diameter can be found using the following equation.

$$d_{x1} = \left[d_x^2 + \frac{4 \cdot Q_\mathrm{o} \cdot \Delta t}{\pi \cdot \rho \cdot Q_\mathrm{f} \cdot l_c} \right]^{1/2} \tag{15}$$

where d_{x1} and d_x are the new and old diameter of the ice respectively, Δt is the time step. Since the different temperature of refrigeration in the given interval (-2.5 - -6.5°C) is small, it is acceptable to calculate the new ice layer diameter with an average heat flux in the same temperature interval.

Results and Discussion

The chiller power input and capacity for charge and discharge rates in kW for conventional A/C and TES operating strategy systems for the office building in Libya are summarized in tab. 1. For the conventional A/C system, the chiller consumes more energy as the cooling load increases during the design day due to the increase in internal loads and outside weather conditions. As expected, the full storage operating strategy has the biggest energy consumption compared to other operating strategies, due to the bigger size of the chiller and storage and longer charging time. Consequently more power input is required. The total amount of energy consumption for full storage strategy is 5.5%, 3.4%, 6% and 4.9% higher compared to partial storage load leveling, and demand limiting 55%, 60% and 65% respectively. On the other hand, the load leveling operation strategy has the lowest chiller capacity, which leads to lower energy consumption by 11% compared with a conventional A/C system. However, its storage capacity is higher than the other partial storage demand limiting load ratios while less than the full storage strategy. Further analysis has been conducted based on the selection of the charging hours in an attempt to reduce energy consumption of the chiller using different partial storage demand limiting load ratios. The chiller capacity in the partial storage demand limiting operation with 60% load ratios is larger than 55% load ratio and lower than 65% load ratio of demand limiting. Nevertheless, in the case of the storage capacity comparison, the 55% load ratio of demand limiting has the biggest capacity while the 65% load ratio of demand limiting has the smallest capacity. Although, the 65% load ratio has the lowest charging time due to the smaller capacity of storage, the 60% load ratio of demand limiting strategy has the lowest energy consumption required compared to all other operating strategies and conventional A/C systems.

The mathematical model was presented to studying performance of external ice during the charging process. Two steps of calculations were carried out to determine heat transfer between water and refrigeration in the storage. These steps including heat transfer inside the tube, through layer of tubes (Sensible) and through the ice layer that built on the external diameter of the tubes (Latent). Regarding the sensible heat transfer process, the thickness of the ice is assumed to be zero at the starting point, so the transfer fluid is in direct contact with the water. Therefore, the evaporating is temperature modified to be -2.5°C. When the water temperature approaches zero, the ice layers start to build around the pipes, which means that the latent heat transfer process takes place and the evaporating temperature has decreased to -6.5°C in order build the required thickness of the ice layer.

Clearly as shown in fig.4, an increase in outdoor temperature causes an increase in compressor power. For instance, at 45°C condensing temperature, the required power input is 95 kW, while at 60°C condensing temperature; the power input is about 135 kW. In short, the thicker the ice layer, the greater its thermal resistance, which therefore significantly slows the rate of ice build-up and further decrease of evaporating temperature is thus required.
Ideally, so as to maximize the rate of ice build-up, the ice layer thickness should be kept as thin as possible. However this would reduce the amount of latent energy storage per tube, thus requiring a greater total number of tubes to be installed in a package to maintain the same overall total thermal storage capacity of the unit. Therefore the maximum ice thickness was modified to be 0.035 m to store the desired amount of energy required to meet the cooling load of the building.

Analyzing the thickness of ice at different condensing temperatures is shown in fig. 5 and fig. 6. As can be seen in fig. 5, the heat transfer rate between the water and the refrigerant is higher at lower outdoor temperature. Consequently, the ice thickness increased with decreasing outdoor temperature for the same capacity of the chiller as shown in fig. 6. This is due to the enhancement of chiller performance at lower outdoor temperature. Conversely, in higher outer temperatures more energy is transferred to the ice layers. As shown in fig. 6, the increasing of thermal resistance of the ice layer as it built on the heat exchanger tubes results in cooling capacity reduction.

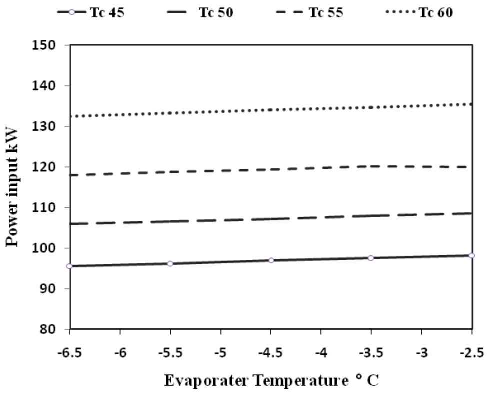

Figure 4. Increasing power input of compressor

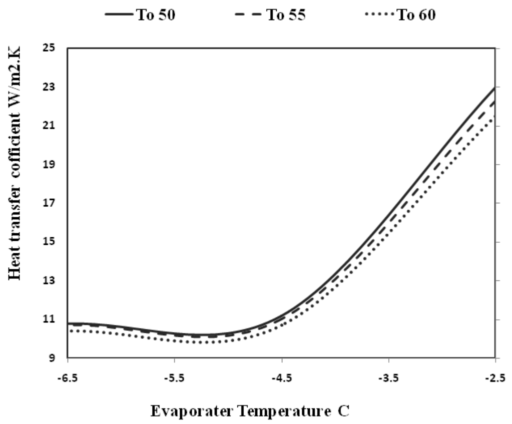

Figure 5. Decreasing cooling capacity with thickness of ice

Furthermore, fig. 6 observes that lower outdoor temperatures, less charging time is required to reach the optimum ice layer thickness of 0.035m. As the ice thickness on the heat exchanger tubes increases, further decreasing of the evaporating temperature is required which in return further decreases of cooling capacity as observed in fig. 7. This caused a decrease in the refrigerant mass flow rate.

Conclusion

The amount of reduction in energy consumption depends on the operating strategies that are considered in this study. The partial storage demand limiting 60% scenario achieves the lowest energy consumption compared to other TES operating strategies and the conventional A/C system. It was discovered from mathematical modeling of the 60% demand limiting scenario that the ice thickness increased slightly when increasing the period of charging hours due to the decreasing evaporating temperature. Optimum ice layer thickness is achieved faster when outer temperature is lower. It can be concluded that any new layer of ice must be formed on the surface of existing ice. This leads to increased thermal resistance of heat transfer between the refrigerant and water, which in return decreases the cooling capacity.

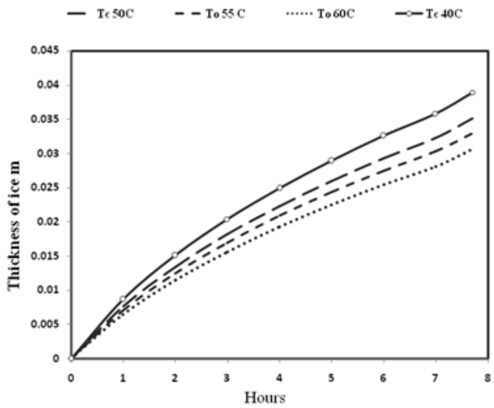

Figure 6. Increasing thickness of ice with time

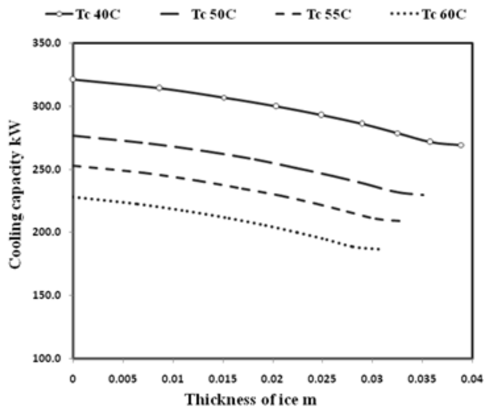

Figure 7. Decreasing cooling capacity with thickness of ice

Nomenclature

a_w	Thermal diffusivity of water[m^2s^{-1}]
d_{icemax}	Maximum diameter of ice layer[m]
hv	latent heat vaporization [kJ kg^{-1}]
H_{char}	Number of charging hours [h]
H_{dirt}	Number of direct cooling hours[h]
H_w	Height of water in the storage [m]
Lc	Total length of pipes [m]
L_s	Length of one section [m]
L_t	Length of pipes[m]
NcR	Circulation number of brine[-]
Nz	Number of parallel pipes [-]
N_{z1}	Number of pipes in one section [-]
Nu	Nusselt number [-]
q_{chil}	Cooling capacity of chiller[kW]
Qoc	Capacity of compressor [kW]
Q_f	Latent heat of fusion of water [$kJkg^{-1}$]
Pr	Prandtl number [–]
Re	Reynolds number [-]
W_s	Width of one section of storage [m]
W_w	Speed of water [m s^{-1}]

Greek letters

$\boldsymbol{\lambda}_{cel}$	Thermal conductivity of pipes material [$Wm^{-1}K^{-1}$]
$\boldsymbol{\lambda}_w$	Thermal conductivity of water [$Wm^{-1}K^{-1}$]
$\boldsymbol{\lambda}_{ledl}$	Thermal conductivity of ice [$Wm^{-1}K^{-1}$]
$\boldsymbol{\nu}_w$	Kinematic viscosity of water [m^2s^{-1}]
$\boldsymbol{\delta}_{ledsp}$	Outer diameter of ice[m]
$\boldsymbol{\rho}$	Density of water[kg m^{-3}]

References

[1] Mohamed, *et al*., An investigation into the current utilization and prospective of renewable energy resources and technologies in Libya, *Renewable Energy,50* (2013), Feb., pp.732-740
[2] ASHRAE *Handbook of Applications* (SI).Chapter33 (1999)
[3] I.Dincer, M.Rosen, *Thermal Energy Storage Systems and Application,* John Wiley and Sons Ltd., Chichester, England,2002
[4] A.H.W. Lee, J.W. Jones., Modeling of an ice-on-coil thermal storage system, Energy Conversion and Management, 37 (1996), 10, pp. 1493-1507

[5] S.M. Hasnain, et al., Prospects of cool thermal storage utilization in Saudi Arabia, Energy Conversion and Management, 41 (2000), 17, pp. 1829-1839
[6] S.M. Hasnain, N.M. Alabbadi., Need for thermal energy storage in Saudi Arabia, *Applied Energy,65* (2000), pp. 153-164
[7] M.J. Sebzali, P.A. Rubini., Analysis of ice cool thermal storage for a clinic building in Kuwait, Energy Conversion and Management,47 (2006), 18-19, pp. 3417-3434
[8] M.A. Rosen, I. Dincer., Exergy methods for assessing and comparing thermal storage systems, International Journal of Energy Research,27,(2003), pp. 415-430
[9] H.-J. Chen, et al., Optimization of an ice-storage air conditioning system using dynamic programming method, Applied Thermal Engineering, 25, (2005), 4, pp. 461-472
[10] B.A. Habeebullah., Economic feasibility of thermal energy storage systems, Energy and Buildings, 39, (2007), pp. 355-363
[11] C.E. Dorgan, J.S. Elleson, Design Guide for Thermal Storage, ASHRAE, 1994
[12] ASHRAE Handbook of Fundamentals (SI). Chapters 1, 4 (2009).
[13] S. Vujic, Refrigeration (in Serbian), Beograd, Serbia, 1990

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