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THERMAL ANALYSIS AND MODELING OF A SWIMMING POOL HEATING SYSTEM BY UTILIZING WASTE ENERGY REJECTED FROM A CHILLER UNIT OF AN ICE RINK

by

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This study deals with the thermal analysis and modeling of a swimming pool heating system in which the waste energy rejected from the chiller unit of an ice rink is used as an energy source. The system consists of a swimming pool and an ice rink coupled by a chiller unit. The swimming pool and the ice rink both indoor types and were constructed in Gaziantep, Turkey. The thermal energy requirement for each section is determined by thermal analysis of each component of the system. Effects of different design parameters such as ceiling insulation thickness, ceiling emissivity, Carnot Efficiency (CE) factor and size of the ice rink on the thermal energy requirements and coefficient of performance (COP) of the chiller unit are investigated. As a result of analyses of the system, the minimum ice rink area is determined in order to meet annual total heat energy demand of the Olympic-sized swimming pool.

Key words: Swimming pool, Ice rink, Chiller unit, Waste energy.

1. Introduction

Nowadays, recreational areas that contain sports complexes are gradually increasing with population growth, especially in developing countries. These sports complexes consist of several sections, such as swimming pools and ice rinks. The swimming pools are commonly heated with conventional methods (e.g. Coal or gas fired boilers). In addition, solar energy can be used for heating of the swimming pools [1]. The water temperature of a swimming pool should be between 22 °C and 28 °C [2]. In an ice rink, ice temperature should be kept between –6 to –1 °C to provide the necessary hardness of the ice surface [3]. During this cooling operation, a higher energy rate than the ice rink's energy consumption is rejected from the chiller unit into the environment [4].

Various theoretical and experimental studies have focused on the analysis, design, and optimization of swimming pool heating and ice rink cooling systems. For instance, Kincay et al. [5] analyzed the technical and economic performance of solar energy utilization in indoor swimming pools. They determined the optimum collector surface area for indoor swimming pools. Chow et al. [6] examined a solar-assisted heat pump system for heating indoor swimming pools and spaces. Özyaman [7] investigated the selection criteria of hot water supply, heat exchanger, installations, and design parameters for heated swimming pools. Mun and Krarti [8] presented an ice rink floor thermal model suitable for whole-building energy simulation analysis. Karampour and Rogstam [9] studied modeling of ice rink heat loads and measurements of the refrigeration system's performance, which they evaluated in two ice rinks and subsequently discovered different heat load shares by analytical modeling. Caliskan and Hepbasli [10] studied energy and exergy analysis of ice rink buildings.

The aim of this study is utilizing waste energy from the condenser of an ice rink chiller unit in order to heat a swimming pool. An analytical model is developed to investigate the heating load of a swimming pool, the cooling load of an ice rink, the energy ratio of each component of these systems, and the chiller unit's coefficient of performance. Therefore, a computational model written in Matlab is prepared to explore the effects of some system parameters such as ceiling insulation thickness, ceiling emissivity, Carnot Efficiency (CE) factor and size of the ice rink on the thermal energy requirements and on the performance of the system.

2. Description of the heating and cooling system

The heating and cooling system under investigation is shown in Fig. 1. The system consists of three main sections, which are a swimming pool, an ice rink and a chiller unit. The system is located in the city of Gaziantep in Turkey. The swimming pool is Olympic-sized (1250 m²) with a 2 m depth. In addition, different ice rink sizes (400 m², 500 m², and 600 m²) are considered in order to meet the swimming pool heat energy demand. Both of the systems have 10m high ceilings.

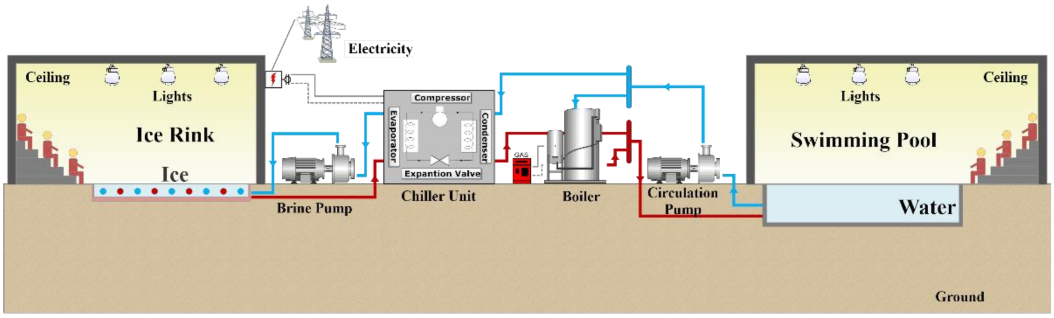

Figure 1. The swimming pool heating system by utilizing waste energy rejected from the ice rink.

In the system consisting of three cycles. The first cycle occurs between the ice rink and the evaporator of the chiller unit, and through this cycle, a brine solution (calcium chloride [3]) is circulated. Since the brine solution temperature is lower than the ice temperature, the brine solution absorbs heat from the ice sheet and delivers it to the evaporator of the chiller unit. The second cycle is the chiller cycle in which Refrigerant 134a is circulated by means of the compressor of the chiller unit. The R134a absorbs heat through the evaporator and sends it to the other hot fluid circulating in the third cycle by means of the condenser. The third cycle takes place between the condenser and the swimming pool and for which water-glycol mixture is commonly used as a hot fluid [1]. A standard gas-fired boiler is integrated into the system for pre-heating the pool water and balancing the heating energy if necessary.

3. Modeling of the thermal system

3.1. *Estimation of heating load for the swimming pool*

Heat losses from the indoor swimming pool occur in five different ways, which are convection heat loss, conduction heat loss from bottom surface and side wall to the ground, latent heat loss due to evaporation from the surface of the water, radiation heat that loss occurs between the surface of the pool and the ceiling, and energy requirements for renovating feed water heating. Total heating load of the swimming pool is expressed by the following equation;

$$Q_{SP} = Q_{SPconv} + Q_{SPcond} + Q_{SPeva} + Q_{SPrad} + Q_{SPren} \tag{1}$$

Convection heat loss is proportional to the difference between ambient air and pool water temperatures. Forced convection occurs when ambient air is not stationary ($u \neq 0$). Convection heat loss can be calculated on the basis of Newton's formula given below:

$$Q_{SPconv} = h \cdot A_{ps} \cdot (T_w - T_{SPia}) \tag{2}$$

where h is the convection heat transfer coefficient, which is given by:

$$h = Nu \times k_a / L_{ch} \tag{3}$$

where k_a is the conduction heat transfer coefficient of air, which is taken as 0.0262 [Wm^{-1}K^{-1}]; L_{ch} is characteristic length, and Nu is Nusselt number. For forced convection, the Nu is given [11];

$$\text{For } Re \leq 5 \times 10^5 \quad Nu = 0.664 \times Pr^{1/3} \times Re^{1/2} \tag{4}$$

$$\text{For } Re > 5 \times 10^5 \quad Nu = 0.037 \times Pr^{1/3} \times Re^{4/5} \tag{5}$$

where Pr is the Prandtl number and Re is the Reynolds number, which is;

$$Re = \frac{u \times L_{ch}}{\nu} \tag{6}$$

where u is the air velocity above the water surface which is considered to be 0.15 m/s [12].

Heat loss by conduction through the poolside and bottom surfaces can be calculated as:

$$Q_{SPcond} = U_{pw} A_{pw} (T_w - T_{SPg}) \tag{7}$$

where U_{pw} is the overall pool wall heat transfer coefficient, which is given as:

$$U_{pw} = \frac{1}{\dfrac{1}{h_g} + \sum_{j=1}^{n} \dfrac{L_j}{k_j} + \dfrac{1}{h_w}} \tag{8}$$

The evaporation amount of water from water surface depends on the difference between the saturated vapor pressure on the surface of the water and the indoor air saturation pressure. Fluctuations also have an effect on the amount of evaporation from the swimming pool's water surface. The equation below can be used to find the rate of evaporation [12].

$$M_{eva} = \frac{A_{ps}}{LH_{eva}} \cdot (p_w - p_a) \cdot (0.089 + 0.0782 \times u)(AF) \tag{9}$$

where AF is the activity factor which is taken 1 for public and school pools [12].

The vapor pressure can be calculated by following empirical equation for 0 °C - 50 °C [13]:

$$p = e^{\left(12.03 - \frac{4025}{235 + T}\right)} \tag{10}$$

The swimming pool evaporation heat loss is then given by;

$$Q_{SPeva} = M_{eva}(LH_{eva}) \tag{11}$$

The heat exchange by radiation between the ceiling and the swimming pool can be calculated on the basis of the Stefan-Boltzmann law. For an indoor swimming pool, the pool area can be taken as completely enclosed. Radiation equation is given below:

$$Q_{SPrad} = A_{ps} f_{pc} \sigma \left((T_w + 273)^4 - (T_{SPc} + 273)^4 \right) \tag{12}$$

where f_{pc} is a gray body configuration factor, swimming pool interface to the ceiling and T_{SPc} is the swimming pool ceiling surface temperature. T_{SPc} can be calculated as:

$$T_{SPc} = T_{SPia} - \left(\frac{Q_{SPc}}{h_i A_{SPc}} \right) \tag{13}$$

where Q_{SPc} is building heat loss of the swimming pool through the ceiling, which is given below:

$$Q_{SPc} = U_{SPc} A_{SPc} \left(T_{SPia} - T_{oa} \right) \tag{14}$$

where U_{SPc} is the overall swimming pool ceiling heat transfer coefficient, which is given as:

$$U_{SPc} = \frac{1}{\frac{1}{h_i} + \sum_{j=1}^{n} \frac{L_j}{k_j} + \frac{1}{h_o}} \tag{15}$$

where h_i and h_o are indoor and outdoor convection heat transfer coefficient, and are 10 and 20 W/m²K, respectively; L_j is the thickness of ceiling insulation; k_j is conduction heat transfer coefficient of ceiling components, which is 0.035 $Wm^{-1}K^{-1}$ for a wool-insulated roof panel [11].

Gray body configuration factor pool interface to the ceiling can be calculated as follows [11]:

$$f_{pc} = \frac{1}{\left[\frac{1}{F_{pc}} + \left(\frac{1}{\varepsilon_p} - 1 \right) + \frac{A_{ps}}{A_{SPc}} \left(\frac{1}{\varepsilon_c} - 1 \right) \right]} \tag{16}$$

where F_{pc} is the view factor from pool surface to the ceiling, which is determined based on the ceiling dimensions, pool surface dimensions and the pool height; ε_p and ε_c are the pool water and the ceiling emissivity, which are the important factors in radiation and are taken to be 0.98 and 0.90, respectively [14]; A_{ps} is the swimming pool surface area; and A_{SPc} is the swimming pool ceiling area.

The view factor from pool surface to the ceiling can be calculated by following equations [15];

$$X = w_p / H \quad \text{and} \quad Y = l_p / H \tag{17}$$

where w_p is swimming pool width, l_p is swimming pool length, and H is swimming pool ceiling height. The view factor is:

$$F_{pc} = \frac{2}{\pi XY} \left\{ \begin{array}{l} \ln\left[\frac{\left(1+X^2\right)\left(1+Y^2\right)}{1+X^2+Y^2} \right]^{1/2} + X\left(1+Y^2\right)^{1/2} \tan^{-1} \frac{X}{\left(1+Y^2\right)^{1/2}} \\ + Y\left(1+X^2\right)^{1/2} \tan^{-1} \frac{Y}{\left(1+X^2\right)^{1/2}} - X \tan^{-1} X - Y \tan^{-1} Y \end{array} \right\} \tag{18}$$

Some amount of water splashes and evaporates from the pool, and this should be substituted by adding water. Additionally, a certain amount of water is refreshed in order to maintain a certain pool water quality. The renovated feed water heating can be calculated as;

$$Q_{SPren} = \left(M_{eva} + M_{fw} \right) C_w \left(T_w - T_{fw} \right) \tag{19}$$

where M_{eva} is evaporation rate, M_{fw} is splashing and refreshing water rate which is taken to be 0.2 kg/s^{-1} for Olympic-sized swimming pool [7], C_w is the specific heat of water, T_w is water temperature and T_{fw} is the supplementary feed water temperature which is taken as 10 $^{\circ}$C [7].

3.2. *Estimation of cooling load for the ice rink*

The cooling load of the ice rink can be calculated by summing the heat gains which can be given as convection, condensation, conduction, radiation, ice resurfacing and lighting. Total cooling load of the ice rink is given as:

$$Q_{IR} = Q_{IRconv} + Q_{IRcondns} + Q_{IRcond} + Q_{IRrad} + Q_{IRrsurf} + Q_{IRlight} \tag{20}$$

The temperature of the air at near surface of the ice surface is higher than the ice temperature. The temperature difference induces the convection. The convection heat gain can be calculated by equation [16]:

$$Q_{IRconv} = h_{conv} A_{is} \left(T_{IRia} - T_{ice}\right) \tag{21}$$

where h_{conv} is the convection heat transfer coefficient, which is given by:

$$h_{conv} = 3.41 + 3.55u \tag{22}$$

The general condensation heat transfer can be calculated by:

$$Q_{IRcondns} = h_{condns} A_{is} \left(T_{IRia} - T_{ice}\right) \tag{23}$$

where h_{condns} is the condensation heat transfer coefficient, which can be calculated by following equation [13]:

$$h_{condns} = 1740 \times h_{conv} \frac{\left(p_{ia} - p_{is}\right)}{\left(T_{IRia} - T_{ice}\right)} \tag{24}$$

where p_{ia} is the water vapor saturation pressure in the air, which can be calculated by Eq. (10), and p_{is} is the water vapor saturation pressure on the ice surface, which can be calculated by following empirical equation [13]:

$$p_{is} = e^{\left(17.391 - \frac{614283}{273.15 + T_{ice}}\right)} \tag{25}$$

where T_{ice} is the ice temperature range from -40 °C to 0 °C.

Heat gain from the ground can be calculated by Eqs. (7-8). The radiation heat gain from the ceiling can be calculated by the Stefan–Boltzmann equation, as given in Eq. (12). The ice rink ceiling temperature is T_{IRc} and can be obtained by Eqs. (13-15). Gray body configuration factor f_{ci} also can be calculated as given in Eqs. (16-18).

To maintain the smoothness and rigidity of an ice rink surface, an ice resurfacing machine shaves the ice surface and then sprays a thin layer of warm water, which is approximately 60-65 °C, onto the ice. The ice resurfacing heat gain can be estimated by [16]:

$$Q_{IRrsurf} = \frac{1000 V_{flw} \left(LH_{frzw} + 4.2 T_{flw} - 2 T_{ice}\right) N_{rsurf}}{24\ t_{flw}} \tag{26}$$

The quantity of lighting heat gain depends on the type of lighting and its applied style. The heat gain component of the lighting can be 60% of the power of luminaries and can be expressed by [16]:

$$Q_{IRlight} = 0.60\ Q_{lum} \tag{27}$$

3.3. *Coefficient of Performance (COP) for the chiller unit*

The chiller unit has two desired purposes: cooling of the ice rink and heating of the swimming pool. Only one chiller unit is used for both of these operations of cooling and heating. Therefore, two performance parameters can be defined for cooling and heating operations. The first one is the Coefficient of Performance of the chiller unit for cooling operation (COP_C), and the second one is the Coefficient of Performance of the chiller unit for heating operation (COP_H). They can be expressed as;

$$COP_C = \frac{Q_C}{W_{comp}} = \frac{Q_C}{(Q_H - Q_C)} \tag{28}$$

$$COP_H = \frac{Q_H}{W_{comp}} = \frac{Q_H}{(Q_H - Q_C)} = COP_C + 1 \tag{29}$$

where Q_C is total heat absorbed from the ice rink, Q_H is total heat rejected to the swimming pool and W_{comp} is work consumption of the compressor.

The actual COP of the chiller unit can be calculated using the approach given by Tarnawski [17]. In this approach, the actual COP is defined by multiplying the COP of a chiller unit by a Carnot Efficiency (CE) factor. Thus, COP_C and COP_H can be expressed as:

$$COP_C = \beta \frac{T_C}{(T_H - T_C)} \tag{30}$$

$$COP_H = \beta \frac{T_H}{(T_H - T_C)} = COP_C + 1 \tag{31}$$

where T_C is the temperature of brine solution in the ice rink brine system, T_H is the temperature of circulating water in the swimming pool circulation system and β is the Carnot Efficiency (CE) factor. Total cooling load of the ice rink and total heating load of the swimming pool may be expressed as a function of outdoor and indoor air temperature which changes with time. Energy requirements of the ice rink and swimming pool in the whole year may be expressed as:

$$Q_{IR} = (UA)_{IR}(T_{oa} - T_{IRia}) \tag{32}$$

$$Q_{SP} = (UA)_{SP}(T_{SPia} - T_{oa}) \tag{33}$$

where $(UA)_{IR}$ and $(UA)_{SP}$ are the overall heat transfer coefficient, T_{IRia} and T_{SPia} are ice rink and swimming pool indoor air temperature, respectively, and T_{oa} is the outdoor air temperature. Energy requirements of the ice rink and swimming pool may also be expressed as:

$$Q_{IR} = (UA)_C(T_{ice} - T_C) \tag{34}$$

$$Q_{SP} = (UA)_H(T_H - T_w) \tag{35}$$

where $(UA)_C$ and $(UA)_H$ are the overall heat transfer coefficient for the brine system of the ice rink and for circulation system of the swimming pool, respectively, T_{ice} is the ice temperature, T_w is the water temperature, and T_C and T_H are the temperature of brine solution and temperature of circulating water, respectively.

Formulations presented by Eqs. (32) and (33) are "quasi-steady formulation," which neglects thermal capacity effects of the ice rink and swimming pool structure. In the quasi-steady assumption, the energy of the system is maintained on a time-scale. The heat transfer process between the subsystems occurs slowly enough for the entire system to remain in internal equilibrium. In this case, the heat energy flowing into the system is considered equal to the heat energy flowing from the system into its surroundings. This simplification was used in previous literature [18] and the same simplification is used in the present study. Eqs. (32) and (34) are combined and solved for T_C, and Eqs. (33) and (35) are combined and solved for T_H. When T_C and T_H are substituted into Eq. (30) we obtain:

$$COP_C = \beta \frac{u_C(T_{IRia} - T_{oa}) + T_{ice}}{u_C(T_{oa} - T_{IRia}) + u_H(T_{SPia} - T_{oa}) + T_w - T_{ice}} \tag{36}$$

The parameters u_C and u_H in Eq. (36) are defined as;

$$u_C = \frac{(UA)_{IR}}{(UA)_C} = \frac{T_{ice} - T_C}{T_{oa} - T_{IRia}} \tag{37}$$

$$u_H = \frac{(UA)_{SP}}{(UA)_H} = \frac{T_H - T_w}{T_{SPia} - T_{oa}} \tag{38}$$

The total heat energy absorbed from the ice rink is rejected into the swimming pool by means of the condenser of the chiller unit. This energy is used to meet the heat energy demand of the swimming pool. The rejected heat energy, which depends on COP_C and total cooling load of the ice rink, is expressed as follows:

$$Q_{SP} = Q_{IR}\left(1 + \frac{1}{COP_C}\right) \tag{39}$$

In this study, the winter and summer outdoor air design temperature (T_{oa}) are taken to be -9 °C and 39 °C, respectively for the city of Gaziantep. Swimming pool (T_{SPia}) and ice rink (T_{IRia}) indoor air design temperature are assumed to be 20 °C and 15 °C, respectively. These temperatures are used to obtain the parameters u_C and u_H in COP_C Eq. (36). In addition, the temperature of brine solution (T_C) and temperature of circulating water (T_H) are taken to be -10 °C and 55 °C, respectively for the system.

4. Results and discussion

The effects of system parameters such as - ceiling insulation thickness, ceiling emissivity, Carnot Efficiency (CE) factor and size of the ice rink - on the energy requirements and the system performance are investigated by executing the computer code written in Matlab. In this section, results obtained from the computational model are given as figures and discussed.

Initially, the following system parameters values are considered: swimming pool area is A_{ps} =1250 m², ice rink area is A_{is} =600 m², ceiling insulation thickness is L=3 cm, ceiling emissivity is ε =0.90, and Carnot Efficiency factor is CE=0.40. During numerical calculations, pool water (26 °C) and ice (-4 °C) temperatures are considered constant. In addition, swimming pool and ice rink indoor air temperatures are considered to change between 10-15 °C and 20-25 °C, respectively, during the whole year.

Fig. 2(a) and 2(b) show the yearly average energy and energy fraction of swimming pool heat loss and ice rink heat gain components. It can be clearly seen that the evaporation (77.00 kW, 77%) and radiation (16.42 kW, 16%) heat losses are more significant than the other heat loss components in the swimming pool energy fractions. On the other hand, the effect of convection, conduction and renovated feed water heat loss on the energy analysis of swimming pool is found to be minimal, and they can be disregarded in the energy analysis calculations. In Fig. 2(b), it is observed that condensation (29.43 kW, 30%), convection (27.67 kW, 28%) and radiation (27.43 kW, 27%) heat gain shows a clear superiority when compared with the conduction, ice resurfacing and lighting heat gain in the ice rink. They should be taken into consideration in the energy analysis calculations.

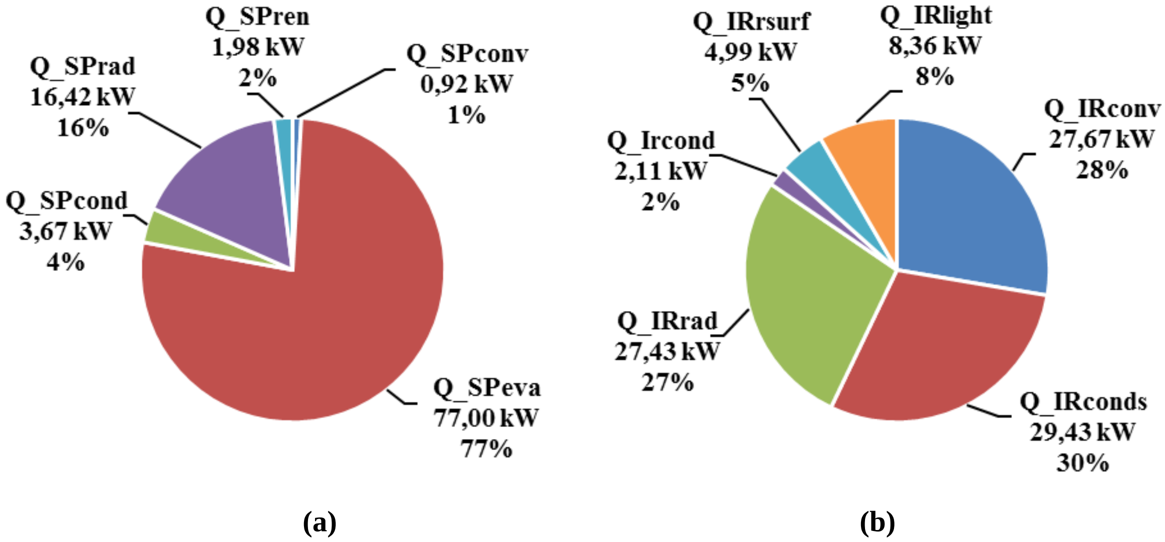

Figure 2. Yearly average energy and energy fraction of (a) swimming pool heat loss and (b) ice rink heat gain components.

Fig. 3(a) and 3(b) were prepared to emphasize the effect of the different ceiling insulation thickness on the swimming pool and ice rink radiation heat transfer. It is seen that the highest radiation heat transfer is obtained for the swimming pool in January and for the ice rink in July, respectively. The radiation heat transfer varies based on the thickness of ceiling insulation. The radiation heat transfer dramatically increases when the thickness of ceiling insulation decreases. It is observed that the heat transfer by radiation increases remarkably when the thickness of ceiling insulation is 1 cm. But, the same reduction ratio is not observed when the thickness of ceiling insulation is 5 cm. Based on this information, the optimum ceiling insulation thickness is found 3 cm.

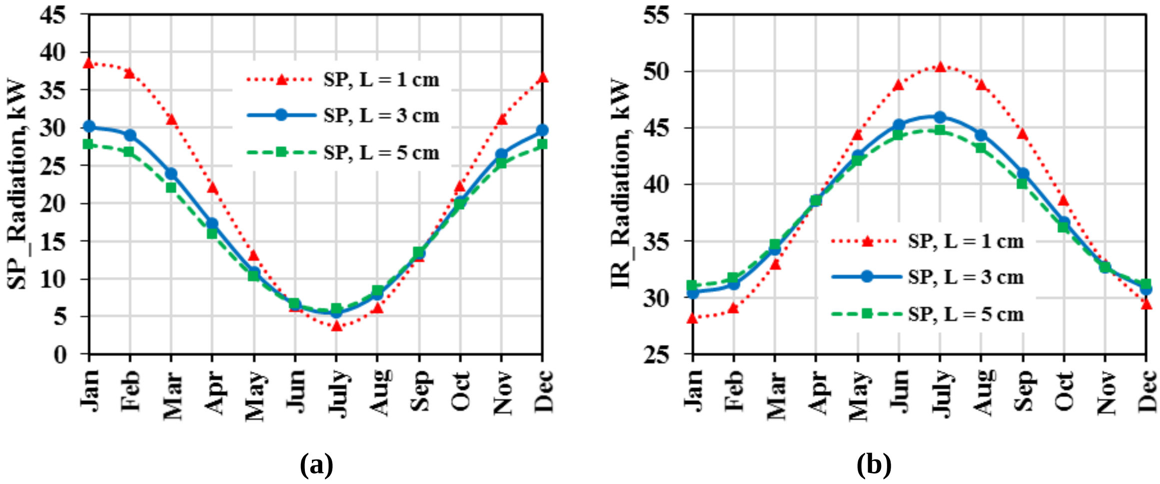

Figure 3. Effect of ceiling insulation thickness (L) on the annual variation of monthly average radiation heat transfer of (a) the swimming pool and of (b) the ice rink.

The effects of different ceiling emissivity on the radiation heat transfer for the swimming pool and ice rink are shown in Fig. 4(a) and 4(b), respectively. They show that the swimming pool radiation heat loss and the ice rink radiation heat gain increases when ceiling emissivity is increased. The ceiling can be covered with aluminum foils or paint metallic color in order to reduce the emissivity. Materials that have a low emissivity can also be selected for ceiling construction. Thus, the swimming pool radiation heat loss and the ice rink radiation heat gain are reduced.

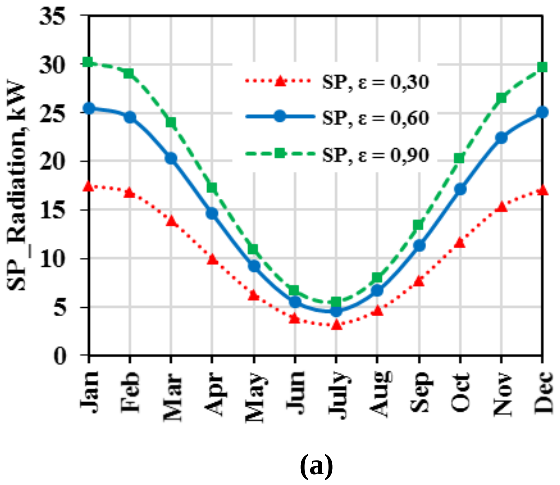

Figure 4. Effect of ceiling emissivity (ε) on the annual variation of monthly average radiation heat transfer of (a) swimming pool and of (b) ice rink.

A reasonable CE value was used to evaluate the actual efficiency of the system in the present study. It is known that actual COP depends on the real heat pump system types and sizes. Thus, the CE value ranges from 0.30 to 0.50 for small electric heat pumps and 0.50 to 0.70 for large, high-efficiency electric heat pumps [19]. Accordingly, in the present study, three CE values (0.30, 0.40 and 0.50) were considered. Results obtained for actual COP_C and COP_H are plotted in Fig. 5. It is clearly seen that COP_C and COP_H increase with the CE factor and reach the maximum and the minimum values in August and January, respectively.

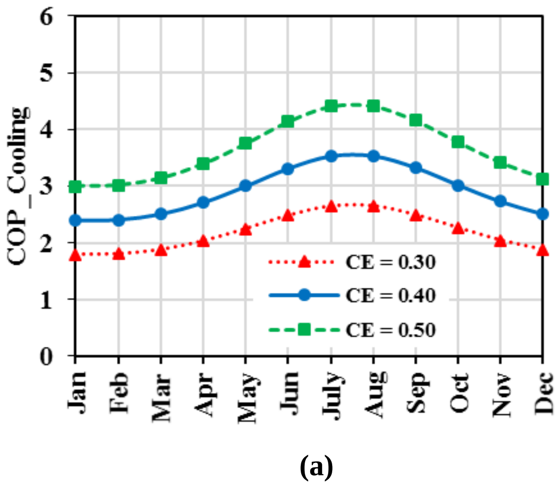

Figure 5. Effect of CE on the annual variation of monthly average (a) COP_C and of (b) COP_H.

Based on the results, following system parameters values are considered to meet the energy demands of the pool: ceiling insulation thickness L=3 cm, ceiling emissivity ε =0.90, and Carnot Efficiency factor CE=0.40. Fig. 6 emphasizes the effect of the different ice rink sizes (400 m², 500 m², and 600 m²) on the rejected heat energy into the swimming pool. It can be easily seen that the amount of heat energy rejected increases when the ice rink size increases. Rejected heat energy begins to cover the energy demands of the swimming pool when the ice rink size is 600 m². Therefore, a 600m² ice rink is needed to meet all the annual energy demand of the Olympic-sized swimming pool (1250m²).

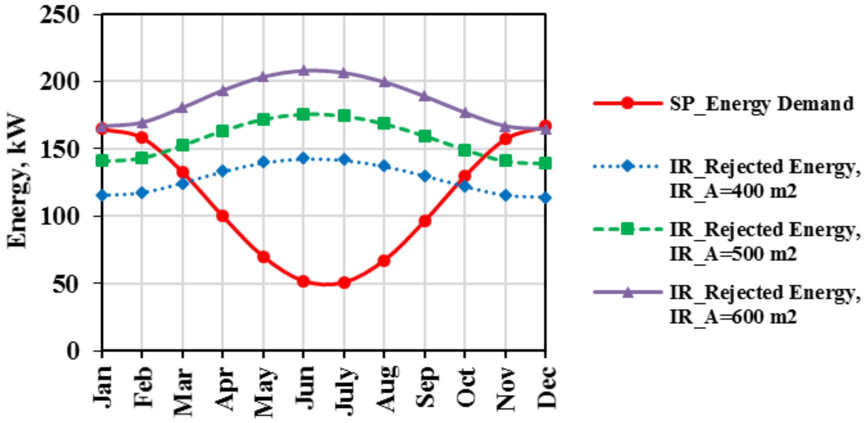

Figure 6. Effect of the different ice rink size on the rejected heat energy to the swimming pool.

5. Conclusions

In this study, the effects of design parameters on the system were investigated. Finally, required ice rink size was also determined for heating of the Olympic-sized swimming pool.

The conclusion from the results of the present study may be listed as follows:

- The optimum ceiling insulation thickness was found to be 3 cm for both ceilings. The ceiling structure can be covered with aluminum foils or paint metallic color in order to reduce the emissivity of the ceiling. Thus, the radiation can be decreased considerably.
- Using a higher efficient chiller unit (higher CE factor) can increase the COP of the system, but higher COP causes a lower amount of rejected heat energy. Therefore, the chiller unit should be selected considering the energy demand of the swimming pool; a CE value of 40% is ideal.
- At least 600 m² ice rink is needed to meet the energy demand of an Olympic-sized swimming pool (1250 m²).

Nomenclature

A	Area [m²]	P	Pressure [kPa]
AF	Activity factor	Q	Heat energy [kW]
C	Specific heat [$kJkg^{-1}K^{-1}$]	t	Time [s]
COP	Coefficient of performance	T	Temperature [°C]
f	Gray body configuration factor	u	Air velocity [m/s]
F	View factor	U	Overall heat transfer coefficient [$Wm^{-2}K^{-1}$]
h	Convection heat transfer coefficient [$Wm^{-2}K^{-1}$]	V	Volume [m³]
H	Swimming pool ceiling height [m]	W	Work [kW]
k	Conduction heat transfer coefficient [$Wm^{-1}K^{-1}$]	ε	Emissivity
		ν	Kinematic viscosity of air [m²/s]
L	Length [m]	σ	Stefan-Boltzmann constant [5.67x10^{-8} $Wm^{-2}K^{-4}$]
LH	Latent heat [$kJkg^{-1}$]		
M	Rate of mass transfer [kgs^{-1}]		

Indices

a	Air	*IR*	Ice rink
c	Ceiling	*is*	Ice surface
C	Cooling	*j*	Component number
ch	Characteristic	*light*	Lighting
ci	Ceiling to ice surface	*lum*	Luminaries
comp	Compressor	*o*	Outdoor
cond	Conduction	*oa*	Outdoor air
condns	Condensation	*p*	Pool
conv	Convection	*pc*	Pool surface to ceiling
eva	Evaporation	*ps*	Pool water surface
flw	Flood water	*pw*	Pool wall
frzw	Freezing water	*rad*	Radiation
fw	Supplementary feed water	*ren*	Renovated feed water
H	Heating	*rsurf*	Resurfacing
i	Indoor	*SP*	Swimming pool
ia	Indoor air	*w*	Water
ice	Ice		

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