

# Performance of a swimming pool heating system by utilizing waste energy rejected from an ice rink with an energy storage tank



Muhammed Enes Kuyumcu<sup>a,\*</sup>, Hakan Tutumlu<sup>b</sup>, Recep Yumrutaş<sup>c</sup>

<sup>a</sup> Department of Mechanical Engineering, Kahramanmaraş Sutcu Imam University, 46100 Kahramanmaraş, Turkey

<sup>b</sup> Department of Mechanical Engineering, Firat University, 23119 Elazığ, Turkey

<sup>c</sup> Department of Mechanical Engineering, University of Gaziantep, 27310 Gaziantep, Turkey

## ARTICLE INFO

### Article history:

Received 30 March 2016

Received in revised form 9 May 2016

Accepted 18 May 2016

### Keywords:

Ice rink

Swimming pool

Chiller

Heat pump

Thermal energy storage tank

## ABSTRACT

This study deals with determining the long period performance of a swimming pool heating system by utilizing waste heat energy that is rejected from a chiller unit of ice rink and subsequently stored in an underground thermal energy storage (TES) tank. The system consists of an ice rink, a swimming pool, a spherical underground TES tank, a chiller and a heat pump. The ice rink and the swimming pool are both enclosed and located in Gaziantep, Turkey. An analytical model was developed to obtain the performance of the system using Duhamel's superposition and similarity transformation techniques. A computational model written in MATLAB program based on the transient heat transfer is used to obtain the annual variation of the ice rink and the swimming pool energy requirements, the water temperature in the TES tank, COP, and optimum ice rink size depending on the different ground, TES tank, chiller, and heat pump characteristics. The results obtained from the analysis indicate that 6–7 years' operational time span is necessary to obtain the annual periodic operation condition. In addition, an ice rink with a size of 475 m<sup>2</sup> gives the optimum performance of the system with a semi-Olympic size swimming pool (625 m<sup>2</sup>).

© 2016 Elsevier Ltd. All rights reserved.

## 1. Introduction

One of the most important conditions for the development and industrialization of countries is energy and the ability to use it efficiently. A large part of energy such countries use is obtained from polluting sources, such as coal and fossil fuels, leading to the increase of CO<sub>2</sub>, SO<sub>x</sub> and NO<sub>x</sub> emissions. Therefore, the national energy strategies of many countries should concentrate on the utilization of environmentalist, renewable and sustainable energy sources [1,2].

In addition, the number of sports facilities is increasing every day to promote public health, especially in developing countries. These facilities can contain several sections such as ice rinks, swimming pools, basketball courts, and volleyball courts. Widely, ice rinks and swimming pools are used for hockey, curling, figure skating, swimming races and water games. These swimming pools are commonly heated by conventional methods (e.g. coal or gas-fired boilers), and recently solar energy is also being used instead of these methods [3,4]. A swimming pool water temperature should be between 22 °C and 28 °C for comfortable conditions [5]. In an ice rink, in order to provide a necessary hardness of the

ice surface for different types of ice sports, the ice temperature should be kept between −6 °C and −1 °C by circulating a brine solution in pipes or tubes under the ice layer [6]. Excess energy from the ice rink is rejected from the condenser of a chiller unit into the environment as waste energy by using conventional air source chillers [7–9]. Furthermore, the instantaneous change in air temperature can cause irregularity of the system performance (COP) and conventional air source chillers work at low COP values when the weather temperature is high in summer. However, a ground couple chiller, which uses the buried thermal energy storage (TES) tank in the ground as a heat exchanger, can operate at more stable COP values. This is because the ground temperature does not fluctuate greatly during the whole year. It can be easily seen that underground TES tank can be a viable solution for saving the waste energy from the chiller unit.

Analytical and experimental investigations have been studied in the literature related to the design, analysis and optimization of TES for heating and cooling applications. Caliskan et al. [10] investigated energetic, exergetic, environmental and sustainability analyses of various TES systems (Latent, Thermochemical, and Sensible) for building applications at varying environment temperatures. They reported that the most sustainable system is the aquifer TES while the worst sustainable system is the latent TES. Rismanchi et al. [11] developed a computer model to determine

\* Corresponding author.

E-mail address: [m.enes.kuyumcu@gmail.com](mailto:m.enes.kuyumcu@gmail.com) (M.E. Kuyumcu).

## Nomenclature

|               |                                                                                     |                |                           |
|---------------|-------------------------------------------------------------------------------------|----------------|---------------------------|
| $A$           | area (m <sup>2</sup> )                                                              | $\psi$         | dimensionless temperature |
| $AF$          | activity factor                                                                     |                |                           |
| $C$           | specific heat (kJ/kg K)                                                             | <i>Indices</i> |                           |
| $COP$         | coefficient of performance                                                          | $b$            | building                  |
| $f$           | gray body configuration factor                                                      | $c$            | ceiling                   |
| $F$           | view factor                                                                         | $C$            | cooling                   |
| $h$           | convection heat transfer coefficient (W/m <sup>2</sup> K)                           | $comp$         | compressor                |
| $H$           | ceiling height (m)                                                                  | $cond$         | conduction                |
| $k$           | conduction heat transfer coefficient (W/m K)                                        | $condns$       | condensation              |
| $ln$          | length (m)                                                                          | $conv$         | convection                |
| $L$           | thickness (m)                                                                       | $eva$          | evaporation               |
| $LH$          | latent heat (kJ/kg)                                                                 | $flw$          | flood water               |
| $M$           | rate of mass transfer (kg/s)                                                        | $frzw$         | freezing water            |
| $N$           | number of daily repetition                                                          | $fw$           | supplementary feed water  |
| $p$           | pressure (kPa)                                                                      | $g$            | ground                    |
| $q$           | dimensionless heat energy                                                           | $H$            | heating                   |
| $Q$           | heat energy (kW)                                                                    | $i$            | indoor                    |
| $r$           | radial distance from the tank center (m)                                            | $ia$           | indoor air                |
| $R$           | tank radius (m)                                                                     | $ice$          | ice                       |
| $t$           | time (s)                                                                            | $IR$           | ice rink                  |
| $T$           | temperature (°C)                                                                    | $is$           | ice surface               |
| $v$           | air velocity (m/s)                                                                  | $j$            | component number          |
| $U$           | overall heat transfer coefficient (W/m <sup>2</sup> K)                              | $light$        | lighting                  |
| $V$           | volume (m <sup>3</sup> )                                                            | $lum$          | luminaries                |
| $w$           | dimensionless work                                                                  | $o$            | outdoor                   |
| $wd$          | width (m)                                                                           | $oa$           | outdoor air               |
| $W$           | work (kW)                                                                           | $ps$           | pool water surface        |
| $x$           | dimensionless radial distance                                                       | $pw$           | pool wall                 |
| $\alpha$      | thermal diffusivity of the ground (cm <sup>2</sup> /s)                              | $rad$          | radiation                 |
| $\gamma$      | dimensionless parameter                                                             | $ren$          | renovated feed water      |
| $\varepsilon$ | emissivity                                                                          | $rsurf$        | ice resurfacing           |
| $\eta$        | Carnot Efficiency (CE)                                                              | $SP$           | swimming pool             |
| $\rho$        | density of ground (kg/m <sup>3</sup> )                                              | $TES$          | thermal energy storage    |
| $\sigma$      | Stefan-Boltzmann constant ( $5.67 \times 10^{-8}$ W/m <sup>2</sup> K <sup>4</sup> ) | $w$            | water                     |
| $\tau$        | dimensionless time                                                                  | $\infty$       | infinity                  |
| $\phi$        | dimensionless temperature                                                           |                |                           |
| $\Phi$        | relative humidity                                                                   |                |                           |

the potential energy savings of implementing cold TES systems in Malaysia. They found that the overall energy usage of the cold TES storage strategy is almost 4% lower than the non-storage conventional system. Kizilkan and Dincer [12] presented a comprehensive thermodynamic assessment of a borehole TES system for a heating case at the University of Ontario Institute of Technology (UOIT). They performed energy and exergy analyses based on balance equations for the heating application.  $COP_{HP}$  and overall exergy efficiency of the studied system are calculated as 2.65% and 41.35%, respectively. Zhang et al. [13] analyzed a model of a space heating and cooling system of a surface water pond that has an insulating cover, which serves as the heat source in the winter and heat sink in the summer. They considered three running modes to analyze the interaction of the seasonal heat charge and discharge for heating and refrigeration individually. Yumrutaş and Ünsal [14] analyzed an annual periodic performance of a solar assisted ground coupled heat pump space heating system, which had a hemispherical surface tank as a ground heat source based on a hybrid analytical-numerical procedure, using analytical and computational models. Yumrutaş et al. [15] presented an analytical and a computational model for a solar assisted heat pump with an underground cylindrical storage tank. Yumrutaş et al. [16] developed a computational model for determining the annual periodic performance of a cooling system utilizing a ground coupled chiller and a spherical underground TES tank.

In this study, an analytical model and a computational program written in MATLAB were developed to obtain the annual variation of ice rink and swimming pool energy requirements, a periodic solution of the transient heat transfer of the underground TES tank, system performance (COP), and the optimum size of the ice rink. The program was executed to investigate the effects of the size of the ice rink and parameters such as ground type, Carnot efficiency, and TES tank volume. Results obtained from the computational program are given as figures and discussed in the study.

## 2. Description of the system

The simplified system shown in Fig. 1 is located in the city of Gaziantep in Turkey, which lies between 37°4' latitude N and 37°29' longitude E and has a Mediterranean climate. The system under investigation consists of five main sections: an ice rink, a swimming pool, an underground TES tank, a cooling unit (Chiller) and a heating unit (Heat pump). In the system, the ice rink and the swimming pool are coupled by the chiller and the heat pump to the underground TES tank, respectively. The swimming pool is semi-Olympic sized (625 m<sup>2</sup>) [17]. Different ice rink sizes (375–625 m<sup>2</sup>) are considered in order to obtain optimum performance of the system. Both of the systems are covered with 10 m high ceilings as well.

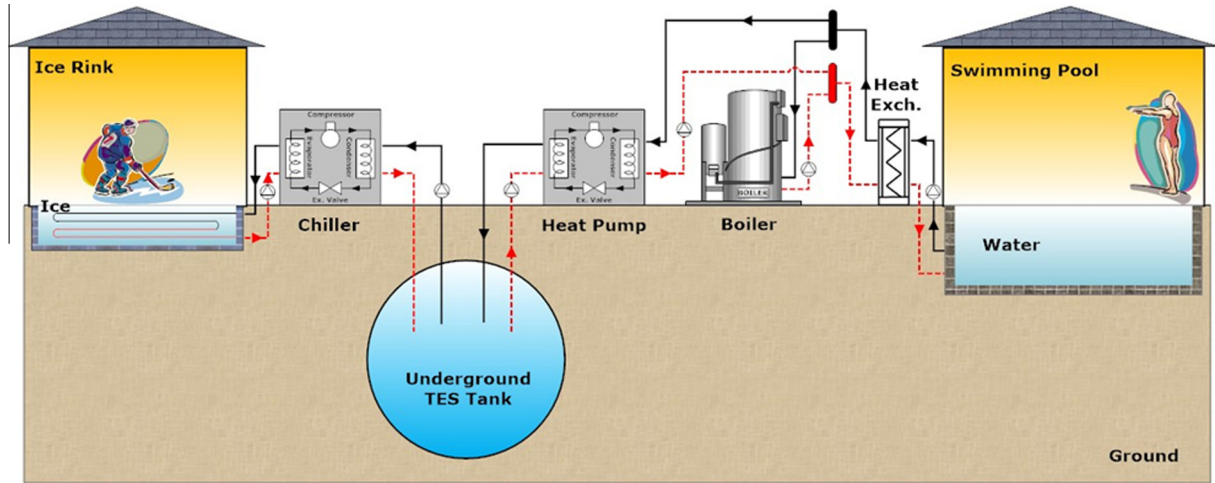


Fig. 1. The swimming pool heating system by utilizing waste energy rejected from an ice rink with an underground thermal energy storage tank.

An important part of the heating system is the underground TES tank which is used for long-term energy storage for energy saving. The TES tank is spherical and buried underground. Keeping the tank underground provides a large energy storage medium and less temperature fluctuation than ambient air temperature. The underground TES tank improves the performance of the system [14]. The TES tank is filled with water as a storage medium because it has high heat capacity and thermal charge–discharge rates.

In the ice rink, the brine solution (ethylene glycol, propylene glycol, and calcium chloride solutions are commonly used [7]) circulates as a secondary refrigerant through the pipes within a concrete slab and absorbs the heat from the ice sheet. The total heat energy absorbed, by means of the compressor of the Chiller using Refrigerant 134a from the brine system, is rejected into the water in the underground TES tank and stored. The thermal energy stored in the underground TES tank is extracted by means of the Heat pump's evaporator and transferred to the swimming pool. A standard gas-fired boiler is integrated into the system for pre-heating the pool water and balancing the energy demand of the pool for all seasons and weather conditions, if necessary.

### 3. Modeling of the thermal system

An analytical model was developed to determine the transient heat transfer of the underground TES tank, the cooling load of the ice rink, the heating load of the swimming pool, and the perfor-

mance of the system. The analysis of each component of the system is introduced in the following subsections.

#### 3.1. Transient heat transfer of the underground TES tank

The energy balance of the underground TES tank is shown in Fig. 2. The TES tank is spherical, filled with water and located deep underground. Water temperature in the TES tank is fully mixed and initially at the deep underground temperature,  $T_\infty$ . The ground is assumed to have constant thermal properties and homogeneous structure.

In spherical coordinate system, transient heat transfer of the underground TES tank and its initial and boundary conditions can be expressed as follows;

$$\frac{\partial^2 T}{\partial r^2} + \frac{2}{r} \frac{\partial T}{\partial r} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \quad (1)$$

$$T(R, t) = T_{TESw}(t) \quad (2)$$

$$T(\infty, t) = T_\infty \quad (3)$$

$$T(r, 0) = T_\infty \quad (4)$$

The energy transferred to the TES tank is equal to the difference between the energy increase of the TES tank and the conduction heat loss from the TES tank to the surrounding ground. This can be expressed by

$$Q = \rho_w C_w V_{TESw} \frac{dT_{TESw}}{dt} - k_g A_{TES} \frac{\partial T}{\partial r}(R, t) \quad (5)$$

where  $\rho_w$ ,  $C_w$ , and  $V_{TESw}$  are density, specific heat, and volume of the water in the TES tank, respectively.  $k_g$ ,  $A_{TES}$ , and  $R$  are, heat conduction coefficient of the surrounding ground, tank surface area, and tank radius, respectively. Dimensionless forms of Eqs. (1)–(5) can be obtained by using following dimensionless variables:

$$x = \frac{r}{R}, \quad \tau = \frac{\alpha t}{R^2}, \quad \phi = \frac{T - T_\infty}{T_\infty}, \quad q = \frac{Q}{4\pi R k_g T_\infty}, \quad p = \frac{\rho_w C_w}{3\rho_g C_g}, \quad \psi(x, \tau) = x\phi(x, \tau) \quad (6)$$

where  $x$ ,  $\tau$ ,  $\phi$  (or  $\psi$ ), and  $q$  are dimensionless parameters of radial distance, time, temperature, and net energy rate to the tank, respectively.  $\rho$  and  $C$  are the density and the specific heat of the ground, respectively.

The dimensionless temperature of the water in the TES tank at the  $n$ th time increment is obtained using Duhamel's superposition

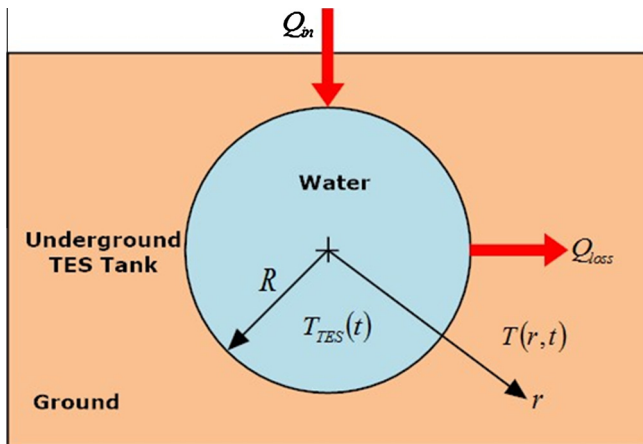


Fig. 2. The energy balance of the underground TES tank.

and similarity transformation techniques, which are explained in detail by Ref. [18]. Consequently, the water temperature of the spherical TES tank is given as:

$$\phi_{TESw}(\tau_n) = \frac{q(\tau_n) + \left(\frac{p}{\Delta\tau} + \frac{1}{\sqrt{\pi\Delta\tau}}\right)\phi_{TESw}(\tau_{n-1}) - \sum_{i=1}^{n-2} \frac{\phi_{TESw}(\tau_{i+1}) - \phi_{TESw}(\tau_i)}{\sqrt{\pi\Delta\tau}(n-i)}}{1 + \frac{p}{\Delta\tau} + \frac{1}{\sqrt{\pi\Delta\tau}}} \quad (7)$$

The term  $q(\tau)$  in Eq. (7) represents the dimensionless net energy input rate to the TES tank, which is given by:

$$q(\tau) = q_{IR}(\tau) - q_{SP}(\tau) + \frac{W(\tau)}{\gamma} \quad (8)$$

where  $q_{IR}(\tau)$  is a dimensionless heat energy rejected by the chiller unit of the ice rink to the TES tank,  $q_{SP}(\tau)$  is a dimensionless heat energy extracted by the heat pump unit of the swimming pool from the TES tank,  $W(\tau)$  is net Work required by chiller and heat pump and  $\gamma$  is a dimensionless parameter,  $[4\pi Rk/(UA)_h]$ .

### 3.2. Cooling load of the ice rink

Each component of the ice rink heat gains is shown in Fig. 3. The cooling load of the ice rink can be calculated by summing the heat gains which are convection, condensation, conduction, radiation, ice resurfacing and lighting. Thus, total cooling load of the ice rink is given as:

$$Q_{IR} = Q_{IRconv} + Q_{IRcondns} + Q_{IRcond} + Q_{IRrad} + Q_{IRrsurf} + Q_{IRlight} \quad (9)$$

The temperature of the air near the ice surface is higher than the ice temperature. This temperature difference induces the convection. The convection heat gain can be calculated as [19]:

$$Q_{IRconv} = h_{conv} A_{ice} (T_{IRia} - T_{ice}) \quad (10)$$

where  $h_{conv}$  is the convection heat transfer coefficient, which is given by [19]:

$$h_{conv} = 3.41 + 3.55v \quad (11)$$

The general condensation heat transfer can be calculated by:

$$Q_{IRcondns} = h_{condns} A_{ice} (T_{IRia} - T_{ice}) \quad (12)$$

where  $h_{condns}$  is the condensation heat transfer coefficient, which can be calculated by following equation [20]:

$$h_{condns} = 1740 \times h_{conv} \frac{(p_{ia} - p_{is})}{(T_{IRia} - T_{ice})} \quad (13)$$

where  $p_{ia}$  is the vapor saturation pressure in the air and  $p_{is}$  is the vapor saturation pressure on the ice surface. They can be calculated by following empirical equations [20]:

$$p_{ia} = e^{\left(12.03 - \frac{4025}{235 + T_{ia}}\right)} \quad (14)$$

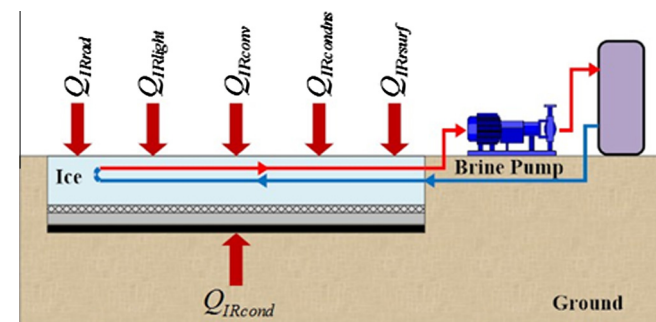


Fig. 3. Ice rink heat gains.

where  $T_{ia}$  is the indoor air temperature range from 0 °C to 50 °C.

$$p_{is} = e^{\left(17.391 - \frac{6142.83}{273.15 + T_{ice}}\right)} \quad (15)$$

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

Heat gain from the ground can be calculated as:

$$Q_{IRcond} = U_{IR} A_{ice} (T_{IRg} - T_{ice}) \quad (16)$$

where  $U_{IR}$  is the overall heat transfer coefficient of ice floor, which is given as:

$$U_{IR} = \frac{1}{\frac{1}{h_g} + \sum_{j=1}^n \frac{L_j}{k_j}} \quad (17)$$

A wide variety of flooring can be constructed depending on the sports to be performed on the ice rink. The thermal and physical properties of materials of the ice rink structure are given in Table 1 [21].

The heat radiation between the ceiling and the ice rink can be calculated on the basis of the Stefan–Boltzmann law. The ice rink arena can be taken as completely enclosed. The radiation equation is given below:

$$Q_{IRrad} = A_{ice} f_{IRc} \sigma (T_{IRc}^4 - T_{ice}^4) \quad (18)$$

where  $f_{IRc}$  is a gray body configuration factor and  $T_{IRc}$  is the ice rink ceiling surface temperature, which can be calculated as:

$$T_{IRc} = T_{IRia} - \left(\frac{Q_{IRc}}{h_i A_{IRc}}\right) \quad (19)$$

where  $Q_{IRc}$  is heat gain of the ice rink through the ceiling, which is given below:

$$Q_{IRc} = U_{IRc} A_{IRc} (T_{IRia} - T_{oa}) \quad (20)$$

where  $U_{IRc}$  is the overall ice rink ceiling heat transfer coefficient, which is given as:

$$U_{IRc} = \frac{1}{\frac{1}{h_i} + \sum_{j=1}^n \frac{L_j}{k_j} + \frac{1}{h_o}} \quad (21)$$

where  $h_i$  and  $h_o$  are indoor and outdoor convection heat transfer coefficient, and are taken to be 10 and 20 W/m<sup>2</sup> K, respectively;  $L_j$  is the thickness of ceiling insulation, and  $k_j$  is the conduction heat transfer coefficient of ceiling components, which is 0.035 W/m K for a wool insulated roof panel.

Gray body configuration factor ceiling to the ice rink interface can be calculated as follows [22]:

$$f_{IRc} = \frac{1}{\left[\frac{1}{F_{IRc}} + \left(\frac{1}{\epsilon_c} - 1\right) + \frac{A_{IRc}}{A_{ice}} \left(\frac{1}{\epsilon_{ice}} - 1\right)\right]} \quad (22)$$

where  $\epsilon_c$  and  $\epsilon_{ice}$  are the ceiling and the ice emissivity, which are important factors in radiation, and are 0.90 and 0.95, respectively [23].  $F_{IRc}$  is the view factor from ceiling to the ice surface, which

Table 1  
Thermal and physical properties of the ice rink structure [21].

| Materials           | $k$ (W/m K) | $L$ (m) | $h$ (W/m <sup>2</sup> K) |
|---------------------|-------------|---------|--------------------------|
| Ice                 | 2.220       | 0.025   | –                        |
| Second layer stucco | 0.645       | 0.020   | –                        |
| First layer stucco  | 1.032       | 0.030   | –                        |
| Concrete            | 1.290       | 0.250   | –                        |
| Bitumen             | 0.129       | 0.020   | –                        |
| Gas concrete        | 0.030       | 0.100   | –                        |
| Lean concrete       | 0.947       | 0.100   | –                        |
| Blockage            | 1.290       | 0.150   | –                        |
| Ground              | –           | –       | ∞                        |



depends on the ceiling dimensions, ice surface dimensions, and the arena height. The view factor can be calculated by the following equations [24]:

$$X = wd_{ice}/H \text{ and } Y = ln_{ice}/H \quad (23)$$

where  $wd_{ice}$  is ice rink width,  $ln_{ice}$  is ice rink length and  $H$  is ceiling height. The view factor is:

$$F_{IRC} = \frac{2}{\pi XY} \left\{ \begin{aligned} &\ln \left[ \frac{(1+X^2)(1+Y^2)}{1+X^2+Y^2} \right]^{1/2} + X(1+Y^2)^{1/2} \tan^{-1} \frac{X}{(1+Y^2)^{1/2}} \\ &+ Y(1+X^2)^{1/2} \tan^{-1} \frac{Y}{(1+X^2)^{1/2}} - X \tan^{-1} X - Y \tan^{-1} Y \end{aligned} \right\} \quad (24)$$

An ice resurfacing machine shaves the ice surface to maintain smoothness and rigidity and then sprays a thin layer of warm water, which is approximately 60–65 °C, on the ice. The ice resurfacing heat gain can be estimated by [19]:

$$Q_{IRsurf} = \frac{1000V_{flw}(LH_{flw} + 4.2T_{flw} - 2T_{ice})N_{rsurf}}{24t_{flw}} \quad (25)$$

Lighting is another heat source for the ice rink. The actual heat gain quantity 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 [19]:

$$Q_{IRlight} = 0.60Q_{lum} \quad (26)$$

### 3.3. Heating load of the swimming pool

Heat losses from the indoor swimming pool are schematically shown in Fig. 4. They 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 loss occurring between the surface of the pool and the ceiling, and energy requirements for renovating feed water heating. Total heating load of the swimming pool consists of the summation of each heat loss at design operating conditions, as in the following equation:

$$Q_{SP} = Q_{SPconv} + Q_{SPcond} + Q_{SPeva} + Q_{SPrad} + Q_{SPren} \quad (27)$$

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

$$Q_{SPconv} = h \cdot A_{ps} \cdot (T_w - T_{Spia}) \quad (28)$$

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

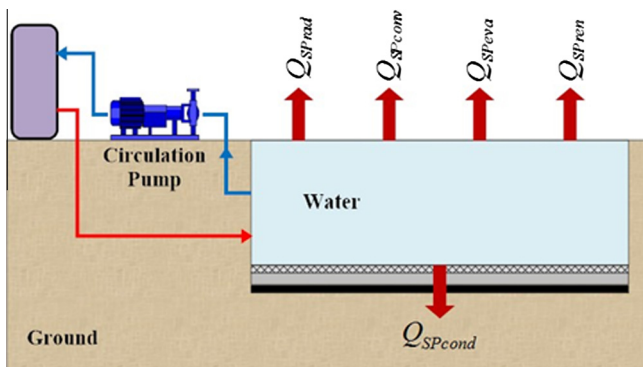


Fig. 4. Swimming pool heat losses.

$$Q_{SPcond} = U_{pw}A_{pw}(T_w - T_{SPg}) \quad (29)$$

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

$$U_{pw} = \frac{1}{\frac{1}{h_g} + \sum_{j=1}^n \frac{L_j}{k_j} + \frac{1}{h_w}} \quad (30)$$

In the construction of a swimming pool, structure materials should be selected to provide thermal insulation, durability for water pressure, and waterproofing. Thus, thermal and physical properties of suitable materials for swimming pool construction are shown in Table 2 [21].

The evaporation amount of the 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 [25].

$$M_{eva} = \frac{A_{ps}}{LH_{eva}} \cdot (p_w - p_{ia}) \cdot (0.089 + 0.0782 \times u)(AF) \quad (31)$$

where  $LH_{eva}$  is the latent heat required to water evaporation at the water temperature,  $p_w$  is the saturated vapor pressure at the water temperature, and  $p_{ia}$  is the vapor saturation pressure at the indoor air temperature, which can be calculated by Eq. (23), (AF) is the activity factor which is taken 1 for public and school pools [25].

The latent heat required to water evaporation for the temperature range from −40 °C to 40 °C is estimated by the following empirical cubic formula [26]:

$$LH_{eva} = 2500.8 - 2.36T_w + 0.0016T_w^2 - 0.00006T_w^3 \quad (32)$$

Swimmers and spectators are affected by relative humidity, and 50–60% relative humidity is most comfortable for swimmers [25]. The relative humidity can be defined as:

$$\Phi = \frac{p_{ia}}{p_w} \quad (33)$$

The swimming pool evaporation heat loss is then given by:

$$Q_{SPeva} = M_{eva}(LH_{eva}) \quad (34)$$

In swimming pools, the radiation heat loss can be calculated by Stefan–Boltzmann equation as given in Eq. (18). The swimming pool ceiling temperature  $T_{SPc}$  can be obtained by Eqs. (19)–(21). Gray body configuration factor  $f_{SPc}$  also can be calculated from Eqs. (22)–(24).

In addition to the evaporation of water in swimming pools, water losses occur from water splash and filtering system leaks. Also, a certain amount of water is refreshed in order to ensure hygienic conditions for swimmers and avoid microorganism proliferation. Therefore, 0.1 kg/s<sup>−1</sup> water for semi Olympic-sized swimming pool should be added to the pool [5]. The renovated feed water heating can be calculated as:

Table 2  
Thermal and physical properties of the pool structure [21].

| Materials           | $k$ (W/m K) | $L$ (m) | $h$ (W/m <sup>2</sup> K) |
|---------------------|-------------|---------|--------------------------|
| Pool water          | –           | –       | 429.92                   |
| BTB & BTB plaster   | 0.774       | 0.03    | –                        |
| Second layer stucco | 0.645       | 0.02    | –                        |
| First layer stucco  | 1.032       | 0.03    | –                        |
| Concrete            | 1.290       | 0.25    | –                        |
| Bitumen             | 0.129       | 0.02    | –                        |
| Gas concrete        | 0.030       | 0.10    | –                        |
| Lean concrete       | 0.946       | 0.10    | –                        |
| Blockage            | 1.290       | 0.15    | –                        |
| Ground              | –           | –       | ∞                        |

$$Q_{SPren} = (M_{eva} + M_{fw})C_w(T_w - T_{fw}) \quad (35)$$

### 3.4. Derivation of COP for the chiller and the heat pump

The chiller and the heat pump – consisting of an evaporator, a condenser, a compressor and an expansion device – work based on a vapor-compression cycle. The chiller absorbs heat from the ice rink cooling system and rejects it into the underground TES tank. Later on, the heat pump extracts heat from the TES tank and transfers it into the swimming pool heating system. The performance of the chiller  $COP_c$  and the heat pump  $COP_H$  can be expressed as:

$$COP_c = \frac{Q_c}{W_{comp}} = \frac{Q_c}{(Q_H - Q_c)} \quad (36)$$

$$COP_H = \frac{Q_H}{W_{comp}} = \frac{Q_H}{(Q_H - Q_c)} \quad (37)$$

Tarnawski [27] expresses the actual COP of the chiller and heat pump by multiplying the Carnot Efficiency (CE) factor  $\eta$ :

$$COP_c = \eta \frac{T_c}{(T_H - T_c)} \quad (38)$$

$$COP_H = \eta \frac{T_H}{(T_H - T_c)} \quad (39)$$

Energy requirements of the ice rink and the swimming pool can be expressed as a function of outdoor and indoor air temperature:

$$Q_{IR} = (UA)_{IRb}(T_{oa} - T_{IRia}) \quad (40)$$

$$Q_{SP} = (UA)_{SPb}(T_{SPia} - T_{oa}) \quad (41)$$

Energy requirements of the ice rink and the swimming pool can also be expressed as:

$$Q_{IR} = (UA)_{IR}(T_{IRia} - T_{ice}) \quad (42)$$

$$Q_{SP} = (UA)_{SP}(T_w - T_{SPia}) \quad (43)$$

Eqs. (40) and (42) are combined and solved for  $T_c$ ; and Eqs. (41) and (43) are combined and solved for  $T_H$ . When  $T_c$  and  $T_H$  are inserted into Eqs. (38) and (39), and using dimensionless parameters given in Eq. (6), we obtain following equations:

$$COP_c = \eta \left( \frac{u_{IR}[\phi_{IRia} - \phi_{oa}] + \phi_{IRia} + 1}{u_{IR}[\phi_{oa} - \phi_{IRia}] - \phi_{IRia} + \phi_{TESw}} \right) \quad (44)$$

$$COP_H = \eta \left( \frac{u_{SP}[\phi_{SPia} - \phi_{oa}] + \phi_{SPia} + 1}{u_{SP}[\phi_{SPia} - \phi_{oa}] + \phi_{SPia} - \phi_{TESw}} \right) \quad (45)$$

When Eqs. (40) and (44) are inserted into Eqs. (36), (41) and (45) are inserted into Eq. (37), the dimensionless works can be expressed as:

$$w_{IR} = \frac{(\phi_{oa} - \phi_{IRia})[u_{IR}(\phi_{oa} - \phi_{IRia}) - \phi_{IRia} + \phi_{TESw}]}{\eta[u_{IR}(\phi_{IRia} - \phi_{oa}) + \phi_{IRia} + 1]} \quad (46)$$

$$w_{SP} = \frac{(\phi_{SPia} - \phi_{oa})[u_{SP}(\phi_{SPia} - \phi_{oa}) + \phi_{SPia} - \phi_{TESw}]}{\eta[u_{SP}(\phi_{SPia} - \phi_{oa}) + \phi_{SPia} + 1]} \quad (47)$$

The parameters  $u_{IR}$  in Eqs. (44) and (46), and  $u_{SP}$  in Eqs. (45) and (47), respectively, are defined as:

$$u_{IR} = \frac{(UA)_{IRb}}{(UA)_{IR}} = \frac{T_{IRia} - T_{ice}}{T_{oa} - T_{IRia}} \quad (48)$$

$$u_{SP} = \frac{(UA)_{SPb}}{(UA)_{SP}} = \frac{T_w - T_{SPia}}{T_{SPia} - T_{oa}} \quad (49)$$

### 3.5. Meteorological data and properties of the ground structures

In this study, hourly weather temperature of the city of Gaziantep, Turkey is used for thermal analysis of the system. During numerical calculations, winter and summer outdoor air design temperature ( $T_{oa}$ ) were taken as  $-10^\circ\text{C}$  and  $39^\circ\text{C}$ , respectively. The ice rink ( $T_{IRia}$ ) and the swimming pool ( $T_{SPia}$ ) indoor air design temperature were  $15^\circ\text{C}$  and  $20^\circ\text{C}$ , respectively, and the ice ( $T_{ice} = -4^\circ\text{C}$ ) and the pool water ( $T_w = 26^\circ\text{C}$ ) design temperature were constant. Three types ground structure (coarse gravel, clay, and granite) are considered, and thermal and physical properties are given in Table 3. The deep ground temperature was taken as  $15^\circ\text{C}$ , and it is assumed that the initial water temperature of the TES tank is equal to ground temperature.

## 4. Results and discussion

The computer code written in MATLAB using the analytical model describing above was executed to investigate the effects of the size of the ice rink ( $375\text{--}625\text{ m}^2$ ) and parameters such as ground type (Coarse gravel, Clay, and Granite), the Carnot Efficiency (CE) factor (0.30, 0.40 and 0.50) and TES tank size (100, 200 and  $300\text{ m}^3$ ) on the system. In the following section, the results obtained from the computations are presented in figures and discussed.

Annual variation of the ice rink heat gain, the swimming pool heat loss and energy ratios of components are shown in Figs. 5 and 6. The ice rink energy ratios of heat gain components are obtained as follows: condensation (30%), convection (28%), radiation (27%), lighting (8%), ice resurfacing (5%) and conduction (2%). In addition, the swimming pool energy ratios of heat loss components are obtained as follows: evaporation (77%), radiation (16%), conduction (4%), renovated feed water (2%) and convection (1%). The highest cooling energy demand of the ice rink is in July, due to an increase in the total cooling load. In contrast, the lowest heating energy demand of the swimming pool is seen in July, due to a decrease in the total heating load.

In the first few years of system operation, the rate of heat energy exchange between the chiller, the heat pump and TES tank; and the rate of heat energy exchange between the TES tank and surrounding ground were not equal. Therefore, the temperature of water in the TES tank increased until the attainment of the annually periodic condition, and then the temperature of the water did not change. The variation of mean water temperature in the TES tank according to years for the different types of ground structure (coarse gravel, clay, and granite) is given in Fig. 7. It is clear that the water temperature rapidly increases in the first few years and after the sixth year remains almost stable. This indicates that the periodic condition is obtained for the water temperature in the TES tank.

The annual variation of water temperature in the TES tank during the sixth year for the coarse gravel, clay, and granite is shown in Fig. 8. It is seen that the water temperature in the TES tank surrounded with coarse gravel has the highest value when compared with clay and granite. The reason for this is that coarse gravel has a

**Table 3**  
Thermal and physical properties of the ground structures [28].

| Ground type   | Conductivity (W/m K) | Diffusivity ( $\text{m}^2/\text{s}$ ) | Specific heat (J/kg K) | Density ( $\text{kg}/\text{m}^3$ ) | Heat capacity ( $\text{kJ}/\text{m}^3\text{ K}$ ) |
|---------------|----------------------|---------------------------------------|------------------------|------------------------------------|---------------------------------------------------|
| Coarse gravel | 0.519                | $1.39 \times 10^{-7}$                 | 1842                   | 2050                               | 3772                                              |
| Clay          | 1.4                  | $1.1 \times 10^{-6}$                  | 848                    | 1500                               | 2250                                              |
| Granite       | 3.0                  | $14.0 \times 10^{-7}$                 | 811                    | 2640                               | 2164.8                                            |

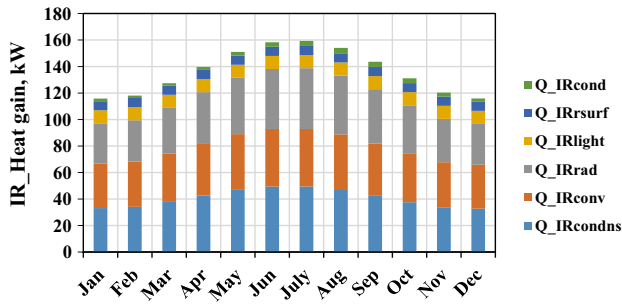


Fig. 5. Annual variation of ice rink heat gain components ( $A_{ice} = 625 \text{ m}^2$ ,  $T_{ice} = -4^\circ\text{C}$ ).

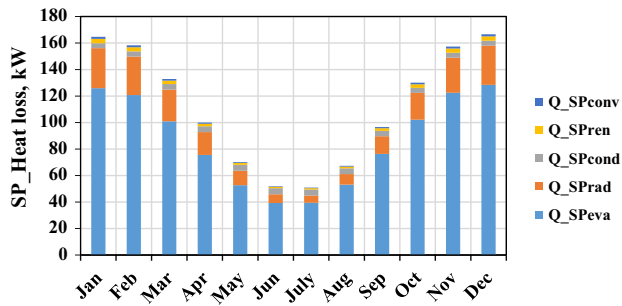


Fig. 6. Annual variation of swimming pool heat loss components ( $A_{ps} = 625 \text{ m}^2$ ,  $T_w = 26^\circ\text{C}$ ).

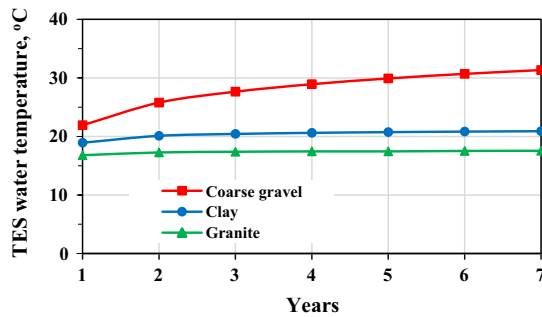


Fig. 7. The variation of mean water temperature in the TES tank according to years for the coarse gravel, clay, and granite. ( $A_{ice} = 625 \text{ m}^2$ ,  $CE = 40\%$ ,  $V = 100 \text{ m}^3$ ).

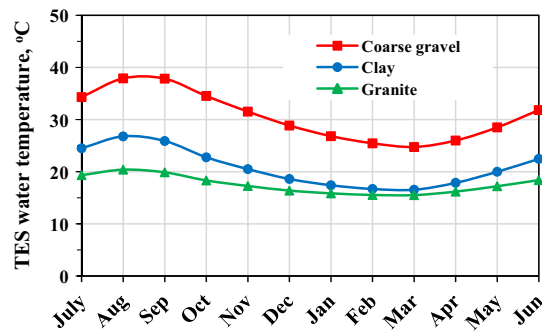


Fig. 8. The annual variation of water temperature in the TES tank during the sixth year for the coarse gravel, clay, and granite. ( $A_{ice} = 625 \text{ m}^2$ ,  $CE = 40\%$ ,  $V = 100 \text{ m}^3$ ).

lower thermal conductivity ( $0.519 \text{ W/m K}$ ) and diffusivity ( $1.39 \times 10^{-7} \text{ m}^2/\text{s}$ ) than the corresponding values of clay and granite, as listed in Table 3. Also, the coarse gravel has a higher heat capacity ( $3772 \text{ kJ/m}^3 \text{ K}$ ). This means that the heat transfer occurs

more slowly from the TES tank to the surrounding ground, and thus, the storage temperature is maintained at a higher value according to other types of ground structures. This indicates that thermal conductivity, diffusivity, and heat capacity have a great effect on the water temperature inside the TES tank.

The annual variation of water temperature, according to the first, second, sixth and seventh years in the TES tank buried in coarse gravel is presented in Fig. 9. It is clearly seen that the temperature of the water in the TES tank increases in the first few years and then the periodic condition is obtained after the sixth year. In addition, the highest water temperature is achieved at the end of summer, owing to the high energy input rate into the TES tank. In contrast, the lowest water temperature is achieved at the end of winter, owing to the low energy input rate. As expected, this trend can also be seen in other figures in this study.

The size of the TES tank affects the performance of the system. Fig. 10 shows the effect of the TES tank's size ( $100$ ,  $200$  and  $300 \text{ m}^3$ ) when surrounded with coarse gravel on the annual variation of water temperature during the sixth year. It is observed that the water temperature and amplitude of the water temperature both increase when the size of the TES tank is decreased. The results obtained for the periodic condition of the system shown in Figs. 7–10 are in good agreement with results in [13,14,18].

A reasonable Carnot Factor (CE) value is used to evaluate the actual efficiency of the system in this study. The COP depends on the real heat pump system types and sizes. The CE value ranges from  $0.30$ – $0.50$  for small electric heat pumps and  $0.50$ – $0.70$  for large, high-efficiency electric heat pumps [29]. Accordingly, in the present study, three CE values ( $0.30$ ,  $0.40$  and  $0.50$ ) are considered. The effect of the CE on the annual variation of water temperature in the TES tank during the sixth year is plotted in Fig. 11. It is clearly seen that the higher CE factor causes the maximum rate of heat energy exchange from the TES tank. Therefore, the higher CE factor leads to lower water temperatures in the TES tank. This is consistent with the results in [16].

The weather temperature of the city of Gaziantep and the water temperature in the TES tank buried in coarse gravel ranges between  $-10$  and  $39^\circ\text{C}$ , and between  $25$  and  $38^\circ\text{C}$  (with  $CE = 40\%$ ), respectively, during the whole year. It is clearly seen that the water temperature in the TES tank has less temperature fluctuation than the weather temperature. Therefore, using water stored in the underground TES tank rather than ambient air leads to a more stable operation condition and improves the performance of the system [14].

Fig. 12 shows the effect of CE on  $COP_C$  of the chiller and  $COP_H$  of the heat pump according to years. It is observed that the higher CE factor causes higher COP. As mentioned before, the temperature of the water in the TES tank increases in the first few years. This is caused by a gradual decrease in the  $COP_C$  of the chiller and a gradual increase in the  $COP_H$  of the heat pump, shown in Fig. 12. In

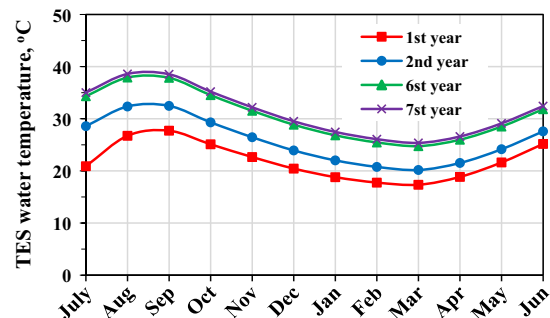
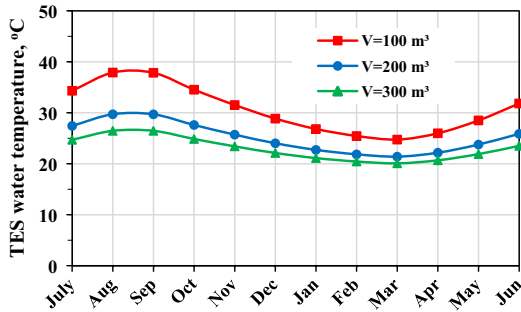
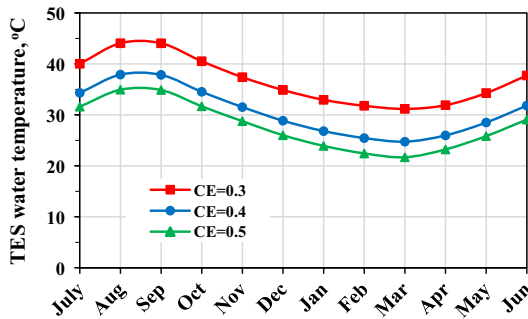


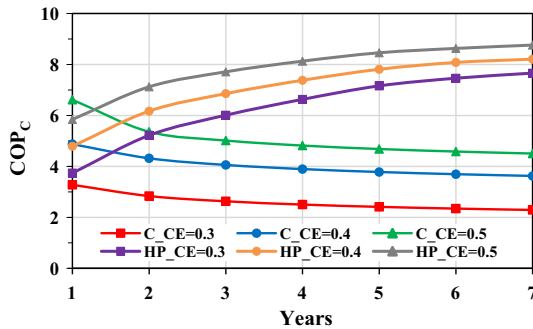
Fig. 9. The annual variation of water temperature in the TES tank buried in coarse gravel. ( $A_{ice} = 625 \text{ m}^2$ ,  $CE = 40\%$ ,  $V = 100 \text{ m}^3$ ).



**Fig. 10.** The effect of the TES tank's size when surrounded with coarse gravel on the annual variation of water temperature during the sixth year. ( $A_{ice} = 625 \text{ m}^2$ ,  $CE = 40\%$ ).



**Fig. 11.** The effect of CE on the annual variation of water temperature in the TES tank surrounded with coarse gravel during the sixth year. ( $A_{ice} = 625 \text{ m}^2$ ,  $V = 100 \text{ m}^3$ ).

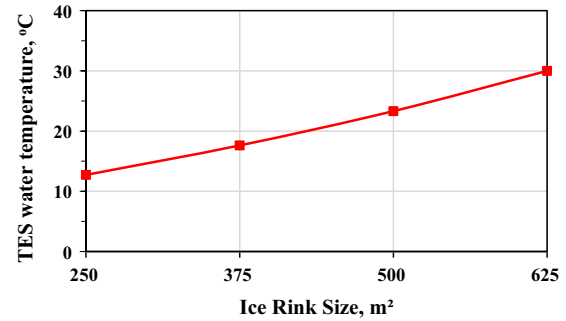


**Fig. 12.** The effect of CE on  $COP_C$  of the chiller and  $COP_H$  of the heat pump according to years. (Coarse gravel,  $A_{ice} = 625 \text{ m}^2$ ,  $V = 100 \text{ m}^3$ ).

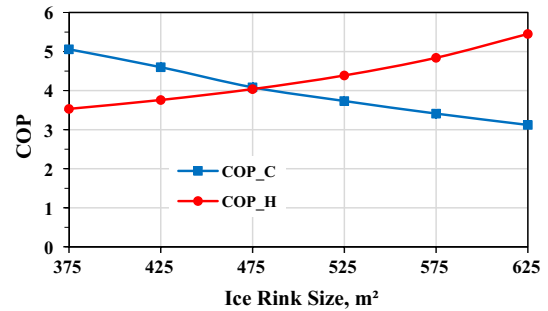
either case, after the sixth year,  $COP_C$  of the chiller and  $COP_H$  of the heat pump do not change and remain almost stable. This means that  $COP_C$  of the chiller and  $COP_H$  of the heat pump are attained from the annually periodic condition.

The size of the ice rink is another important parameter for the water temperature in the TES tank. The effect of the size of the ice rink on the water temperature during the sixth year is plotted in Fig. 13. The amount of heat energy rejected from the chiller to the TES tank increases with increasing the size of the ice rink. A higher amount of heat energy rejected means an increase in the water temperature. It is observed that  $125 \text{ m}^2$  increments in the size of the ice rink lead to an approximately  $5^\circ\text{C}$  increase in the water temperature in the TES tank.

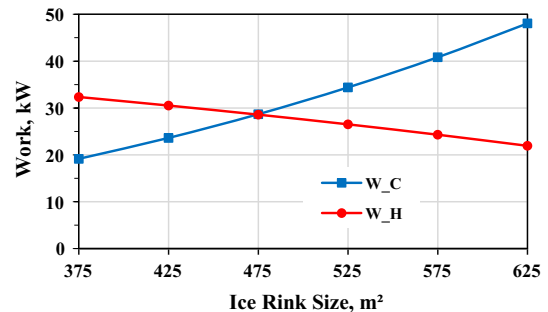
The temperature of the water in the TES tank has a great effect on the cooling performance of the chiller and the heating performance of the heat pump [30,31]. The temperature of the water in the TES tank increases with increasing the size of the ice rink.



**Fig. 13.** The effect of the size of the ice rink on the water temperature in the TES tank during the sixth year. (Coarse gravel,  $CE = 40\%$ ,  $V = 100 \text{ m}^3$ ).



**Fig. 14.** The effect of ice rink size on  $COP_C$  of the chiller and  $COP_H$  of the heat pump during the sixth year. ( $A_{ps} = 625 \text{ m}^2$ , coarse gravel,  $CE = 40\%$ ,  $V = 100 \text{ m}^3$ ).



**Fig. 15.** The effect of ice rink size on work consumption of the chiller and the heat pump compressors during the sixth year. ( $A_{ps} = 625 \text{ m}^2$ , coarse gravel,  $CE = 40\%$ ,  $V = 100 \text{ m}^3$ ).

The effect of the ice rink size on  $COP_C$  of the chiller and  $COP_H$  of the heat pump during the sixth year is shown in Fig. 14. It is observed that due to increasing the size of the ice rink, the increase in water temperature leads to a decrease of the cooling performance of the chiller. At the same time, this leads to an increase of the heating performance of the heat pump. The  $COP_C$  of the chiller and the  $COP_H$  of the heat pump intersect at a  $COP$  value of approximately 4 when the ice rink size reaches  $475 \text{ m}^2$ .

$COP$  is actually a result of work consumption of the compressor. The effect of ice rink size on work consumption of the chiller and the heat pump compressors during the sixth year is given in Fig. 15. The work consumption of the chiller compressor increases with the size of the ice rink while the work consumption of the heat pump compressor decreases, and they intersect at the ice rink size of  $475 \text{ m}^2$ . After this point, the work consumption of the heat pump increases approximately  $20 \text{ kW}$  and the work consumption of the chiller decreases approximately  $7 \text{ kW}$  when the ice rink size is reached to  $625 \text{ m}^2$ . Therefore, the compressor of the heat pump



consumes 13 kW more energy than the compressor of the chiller and as a result, total energy consumption of the system is increased. It is clearly seen that the ice rink with a size of 475 m<sup>2</sup> gives the optimum performance for a system with a semi-Olympic size swimming pool (625 m<sup>2</sup>).

## 5. Conclusions

In this study, an analytical model was developed to analyze the long-period performance of a swimming pool heating system by utilizing waste energy rejected from an ice rink with an underground thermal energy storage tank using Duhamel's superposition and similarity transformation technique. A computational model written in MATLAB program based on the transient heat transfer of the underground TES tank and energy requirements of the ice rink and the swimming pool was executed to investigate the effects of the size of the ice rink and other parameters such as ground type, the Carnot Efficiency (CE) factor and TES tank size on the system.

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

- The ice rink heat gain components: condensation (30%), convection (28%), radiation (27%), and the swimming pool heat loss components: evaporation (77%), radiation (16%) show a clear superiority when compared with other heat gain and heat loss components, respectively.
- Coarse gravel shows the best performance among the ground types considered in this study.
- 6–7 years' operational time span is necessary to obtain annual periodic operation condition.
- Carnot Efficiency (CE) factor has a great effect on COP<sub>C</sub> of the chiller and COP<sub>H</sub> of the heat pump and a value of CE = 40% is obtained as the ideal value.
- An ice rink with a size of 475 m<sup>2</sup> gives the optimum performance for a system with a semi-Olympic size swimming pool (625 m<sup>2</sup>).

## References

- [1] Nicoletti G, Arcuri N, Nicoletti G, Bruno R. A technical and environmental comparison between hydrogen and some fossil fuels. *Energy Convers Manage* 2015;89:205–13. <http://dx.doi.org/10.1016/j.enconman.2014.09.057>.
- [2] Ediger VŞ, Kentel E. Renewable energy potential as an alternative to fossil fuels in Turkey. *Energy Convers Manage* 1999;40:743–55. [http://dx.doi.org/10.1016/S0196-8904\(98\)00122-8](http://dx.doi.org/10.1016/S0196-8904(98)00122-8).
- [3] Buonomano A, De Luca G, Figaj RD, Vanoli L. Dynamic simulation and thermoeconomic analysis of a PhotoVoltaic/Thermal collector heating system for an indoor-outdoor swimming pool. *Energy Convers Manage* 2015;99:176–92. <http://dx.doi.org/10.1016/j.enconman.2015.04.022>.
- [4] Cunio LN, Sproul AB. Performance characterisation and energy savings of uncovered swimming pool solar collectors under reduced flow rate conditions. *Sol Energy* 2012;86:1511–7. <http://dx.doi.org/10.1016/j.solener.2012.02.012>.
- [5] Özyaman C. Calculation of heat load in swimming pools. *Install Eng* 2004;79:28–33.
- [6] IIHF. IIHF Official Rule Book 2014–2018; 2015.
- [7] İsbilen İ. Ice skating cooling-freeze installations. I Natl Install Eng Congr Exhib İzmir, Turkey 1993:335–53.
- [8] Khalid W, Rogstam J. Energy usage prediction model comparing outdoor vs. indoor ice rinks. *Energy Build* 2013;67:195–200. <http://dx.doi.org/10.1016/j.enbuild.2013.08.004>.
- [9] Seghouani L, Daoud A, Galanis N. Yearly simulation of the interaction between an ice rink and its refrigeration system: a case study. *Int J Refrig* 2011;34:383–9. <http://dx.doi.org/10.1016/j.iirefrig.2010.07.024>.
- [10] Caliskan H, Dincer I, Hepbasli A. Thermodynamic analyses and assessments of various thermal energy storage systems for buildings. *Energy Convers Manage* 2012;62:109–22. <http://dx.doi.org/10.1016/j.enconman.2012.03.024>.
- [11] Rismanchi B, Saidur R, Masjuki HH, Mahlia TMI. Modeling and simulation to determine the potential energy savings by implementing cold thermal energy storage system in office buildings. *Energy Convers Manage* 2013;75:152–61. <http://dx.doi.org/10.1016/j.enconman.2013.06.018>.
- [12] Kizilkan O, Dincer I. Borehole thermal energy storage system for heating applications: thermodynamic performance assessment. *Energy Convers Manage* 2015;90:53–61. <http://dx.doi.org/10.1016/j.enconman.2014.10.043>.
- [13] Zhang H-F, Ge X-S, Ye H. Modeling of a space heating and cooling system with seasonal energy storage. *Energy* 2007;32:51–8. <http://dx.doi.org/10.1016/j.energy.2006.02.007>.
- [14] Yumrutaş R, Ünsal M. Analysis of solar aided heat pump systems with seasonal thermal energy storage in surface tanks. *Energy* 2000;25:1231–43. [http://dx.doi.org/10.1016/S0360-5442\(00\)00032-3](http://dx.doi.org/10.1016/S0360-5442(00)00032-3).
- [15] Yumrutaş R, Kunduz M, Ayhan T. Investigation of thermal performance of a ground coupled heat pump system with a cylindrical energy storage tank. *Int J Energy Res* 2003;27:1051–66. <http://dx.doi.org/10.1002/er.932>.
- [16] Yumrutaş R, Kanoğlu M, Bolattürk A, Bedir MŞ. Computational model for a ground coupled space cooling system with an underground energy storage tank. *Energy Build* 2005;37:353–60. <http://dx.doi.org/10.1016/j.enbuild.2004.07.004>.
- [17] FINA. Fina Facilities Rules 2013–2017; 2013.
- [18] Yumrutaş R, Ünsal M. Energy analysis and modeling of a solar assisted house heating system with a heat pump and an underground energy storage tank. *Sol Energy* 2012;86:983–93. <http://dx.doi.org/10.1016/j.solener.2012.01.008>.
- [19] ASHRAE. ASHRAE handbook – refrigeration (SI); 2006 [Chapter 35].
- [20] Granryd E. Processes in moist air, frosting and defrosting. *Refrig Eng Part II* 2005 [Chapter 15].
- [21] Kincay O, Utlu Z, Akbulut U. Technical and economic performance analysis of utilization of solar energy in indoor swimming pools, an application. *J Sol Energy Eng* 2012;134:014502. <http://dx.doi.org/10.1115/1.4005106>.
- [22] Çengel YA. Heat and mass transfer: a practical approach. 3rd ed. USA: McGraw-Hill; 2007.
- [23] Bergman TL, Lavine AS, Incropera FP, DeWitt DP. Fundamentals of heat and mass transfer. 7th ed. USA: Wiley; 2011.
- [24] Narayana KB. View factors for parallel rectangular plates. *Heat Transf Eng* 1998;19:59–63. <http://dx.doi.org/10.1080/01457639808939915>.
- [25] ASHRAE. ASHRAE handbook – HVAC applications (SI). Ashrae; 2011 [Chapter 5.6].
- [26] Yau MK, Rogers RR. A short course in cloud physics. 3rd ed. Oxford: Butterworth-Heinemann; 1989.
- [27] Tarnawski VR. Ground heat storage with a double layer heat exchanger. *Int J Energy Res* 1989;13:137–48. <http://dx.doi.org/10.1002/er.4440130203>.
- [28] Ozisik MN. Heat transfer: a basic approach. New York: McGraw-Hill; 1985.
- [29] Zogou O, Stamatelos A. Effect of climatic conditions on the design optimization of heat pump systems for space heating and cooling. *Energy Convers Manage* 1998;39:609–22. [http://dx.doi.org/10.1016/S0196-8904\(97\)00050-2](http://dx.doi.org/10.1016/S0196-8904(97)00050-2).
- [30] Hou S, Li H, Zhang H. Open air-vapor compression refrigeration system for air conditioning and hot water cooled by cool water. *Energy Convers Manage* 2007;48:2255–60. <http://dx.doi.org/10.1016/j.enconman.2007.05.005>.
- [31] Sanaye S, Niroomand B. Thermal-economic modeling and optimization of vertical ground-coupled heat pump. *Energy Convers Manage* 2009;50:1136–47. <http://dx.doi.org/10.1016/j.enconman.2008.11.014>.