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Optimization of Solar Absorption Cooling System Considering Hourly Analysis

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A B S T R A C T

Thermal modelling and optimal design of a solar absorption cooling system are presented, and hourly analysis is performed over the period of a year. Three design parameters are considered, and then the Real Parameter Genetic Algorithm (RPGA) is applied to obtain the minimum total annual cost. The optimization results show that the solar cooling optimum configuration needs 1630 square meter collectors, a storage tank with 15000L capacity as well as an absorption chiller with 300kW capacity. The hourly analysis shows that the space temperature fluctuates on average every 62 minutes during June and decreases to 51 minutes in September. In addition, the optimum number of collectors increases by 26.67% with a 50% increment in electricity price while it decreases by 20% with a 50% decrement in electricity price. Finally, a sensitivity analysis on RPGA parameters is performed and results are reported.

1. INTRODUCTION

About 40% of the primary energy usage in Iran is consumed by buildings [1]. Heating and air-conditioning facilities are responsible for most of the mentioned energy consumption.

Solar cooling and heating systems are one of the solutions proposed to meet the energy and environmental challenges associated with buildings. Furthermore, the combination of Solar Heating and Cooling (SHC) applications improves the efficiency of solar thermal systems compared to heating or cooling alone [2]. Many studies have been presented in the field of optimization of combined cooling and heating generation systems with a solar cycle to consider hourly analysis [3-6].

The fact that peak cooling demand in the summer is associated with high solar energy availability o□ers an opportunity to further exploit the solar energy for cooling.

Solar energy can be transformed either into electricity or into heat to power a refrigeration cycle. Since the efficiency of the solar photovoltaic collectors increases only slightly (10–15%) contrary to that of the solar thermal collectors, the electrically driven systems are characterized by the limited useful power that can be achieved by solar means. Given their fairly high initial cost, more interest has been paid to the solar thermal-driven refrigeration technologies [7]. The research in this direction attempts to match solar thermal technologies with thermally driven cooling equipment like absorption, adsorption, desiccant and ejector chillers. A comprehensive review of these technologies is given in several references [7-11].

In general, the energy performance of solar heating and cooling plants depends on the system components efficiency, layout and control logic. Up to now, single stage LiBr–H_2O absorption machines driven by hot water have been the most common selection in the system design and construction [12].

In an absorption refrigeration system, the refrigerant vapor is drawn from the evaporator by absorption into the absorbent. The addition of thermal energy to the generator liberates the refrigerant vapor from the strong solution. The refrigerant is condensed in the condenser by rejecting the heat. The liquid refrigerant is then expanded into the evaporator and the cycle is completed [13].

Experimental investigations on solar absorption cooling systems based upon medium and small-sized solar cooling systems have been reported [14-18]. Usually, the cooling capacity and COP of the solar cooling

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systems were tested under practical operating conditions.

The design of SHC systems is a complex problem, which has a high number of parameters (such as hot and cold storages volume, collector size, absorption chiller activation temperature, etc.), and therefore, the classical techniques of design may yield unsatisfactory results. Many authors have analyzed and simulated the system with the aid of simulation software. The Transient System Simulation program (TRNSYS) has been widely used as a simulation tool to identify the system energy performance [19-23]. The performance of this program has been experimentally validated [2, 24]. Several studies on SHC systems were carried out for different climatic areas [25, 26]. The investigated SHC systems are often based on flat plate or evacuated tube solar collectors; other types of solar collectors are used in some studies [12, 27].

In this paper, after thermo-economic modelling of a solar absorption cooling system (SAC), this equipment is optimized by minimizing the total annual cost (TAC). Nominal capacity of chiller, solar collector surface area and storage tank capacity are selected as design parameters and Real Parameter Genetic Algorithm (RPGA) is applied to provide the optimum solutions.

In this study, the hourly and monthly average variations of the ambient temperature during the four months of a year are considered. On the other hand, the different kinds of variables as well as optimization objective including the economic aspect of the model are selected.

As a summary, the followings are the contribution and novelty of this paper on the subject:

• The design and modelling of a solar absorption cooling system with absorption chiller by considering hourly analysis during the course of a year.

• Selecting nominal capacity of a chiller, solar collector surface area and storage tank capacity available in the market as design parameters (decision variables).

• Applying Real Parameter Genetic Algorithm to estimate the optimum values of design parameters by considering the total annual cost as the objective function.

2. THERMAL MODELLING

A schematic diagram of a solar cooling system with absorption chiller is shown in Fig. 1.

It mainly consists of a solar collector, storage tank with auxiliary heater, pumps and absorption chiller including the generator, condenser, evaporator, absorber, expansion valve, solution heat exchanger as well as a cooling tower. In this system, the required heat for the generator is provided by the solar collector. In addition, the storage tank and auxiliary heater are also used to increase the flexibility of the system especially in the absence of sufficient solar radiation during the night hours.

A simulation model is developed to predict the thermal performance of the solar absorption cooling system based on the following assumptions:

1- The system is at a quasi-steady state with reasonable time step.

2- Market available equipment including the absorption chiller, solar collector and storage tank are selected.

Using energy and mass balance equations, rate of heat transfer, cooling load and mass flow rate in the mentioned cooling system can be modelled and solved. Each component is described in the following sub-sections.

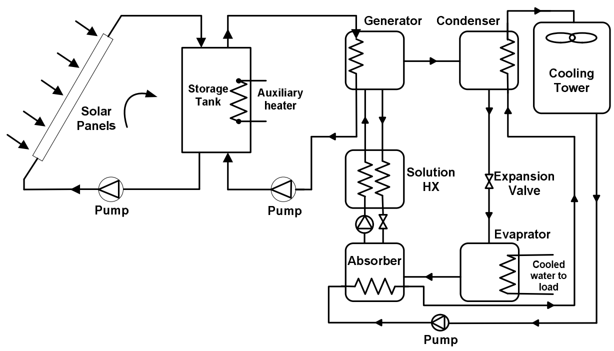

Figure 1. Schematic diagram of solar absorption cooling system

2.1. Solar collectors

The constant solar radiation received by earth's atmosphere is estimated as [28]:

$$\dot{G}_{sc} = 1.353\ (kW/m^2) \tag{1}$$

Due to the revolution of the earth around the sun and declination angle (δ), the received radiation on the earth's surface varies during the year and by latitude (λ) as follows [28]:

$$\dot{G}_{oh}(\frac{kW}{m^2}) = \frac{24}{2\pi}\dot{G}_{sc}\left[1+0.033Cos\left(\frac{360(N+81)}{365}\right)\right]\times \begin{bmatrix} \cos(\lambda)\cos(\delta)(\sin(\omega_2)-\sin(\omega_1))+\frac{2\pi}{360}(\omega 2-\omega 1) \\ \sin(\lambda)\sin(\delta) \end{bmatrix} \tag{2}$$

in which N and ω are index of each day in a year and hour angle, respectively.

The above radiation decreases by the clarity of the weather and is considered with a cloud less index (k_T) [28]:

$$\dot{H}_h = k_T \times \dot{G}_{oh} \tag{3}$$

To have optimum received radiation, solar collector should be installed at an angle. As a result of the slope, the rate of received radiation on the slope surface with angle β varies and is estimated as follows [28]:

$$\dot{H}_t = \overline{R}.\dot{H}_h \tag{4}$$

$$\overline{R} = \frac{\cos(\lambda-\beta)}{d\cos(\lambda)} \left[\begin{array}{l} (a-\xi)\left(\sin(w_s') - \frac{\pi w_s'}{180}\cos(w_s'')\right) \\ +\frac{b}{2}\left(\frac{\pi w_s'}{180} + \sin(w_s')(\cos(w_s') - 2\cos(w_s''))\right) \end{array} \right] + \xi\left(\frac{1+\cos(\beta)}{2}\right) + \rho_g\left(\frac{1-\cos(\beta)}{2}\right) \tag{5}$$

in which w_s', w_s'' a, b, d and ξ are some parameters which dependent on the declination angle, panel slope and latitude [29,30].

2.2. Thermal energy storage tank

Applying the first law of thermodynamic, one can find the formula for storage tank temperature variation in each step size [28]:

$$T_{stor}(t+1) = T_{stor}(t) + \left(\sum \dot{H}_{in}(t) - \sum \dot{H}_{out}(t) - \dot{H}_{los}(t)\right)\frac{\tau}{m_{stor}c_p} \tag{6}$$

when,

$$\sum \dot{H}_{in}(t) - \sum \dot{H}_{out}(t) = \dot{H}_{col} - \dot{H}_g = \dot{H}_{col} - \dot{m}\times c_p \times (T_{g,in} - T_{g,out}) \tag{7}$$

where τ, $m_{stor}c_p$ and T_g refer to the time step, total storage tank heat capacity and generator temperature. Furthermore, subscripts *in, out* indicate the rate of energy entering and leaving the Where the rate of energy loss is estimated as follows:

$$\dot{H}_{los} = (UA\Delta T)_{los} \tag{8}$$

where U, A and ΔT are overall heat transfer coefficient, tank heat transfer surface area and temperature difference between tank and tank outside temperature, respectively.

2.3. Absorption chiller

The chiller coefficient of performance (COP) which is defined as the ratio between the cooling and required heating load is a function of generator inlet temperature and is estimated as follows [31]:

$$\mathrm{COP} = \left(\begin{array}{l} -0.00137\mathrm{T}_{g,in}^3 + 1.11\mathrm{T}_{g,in}^2 \\ -66.38\mathrm{T}_{g,in} - 2.31 \end{array} \right) / \left(\mathrm{T}_{g,in}^2 - 58.43\mathrm{T}_{ig,in} - 0.4686\right) \tag{9}$$

where $\mathrm{T}_{g,in}$ is generator inlet temperature and above relation is valid for $\mathrm{T}_{g,in} > 65^{o}\mathrm{C}$.

The cooling load of chiller is a function of COP and nominal capacity of the chiller is predicted as follows [31]:

$$\dot{Q}_e = \left(2.8913COP^{4.286} + 0.0875\right)\dot{Q}_{nom} \tag{10}$$

where $\dot{Q}_e$ and $\dot{Q}_{nom}$ are evaporator load (chiller cooling load) and nominal capacity of chiller, respectively. The generator outlet temperature which is returned to the storage tank is also obtained using the first law of thermodynamic as shown below:

$$T_{g,out} = T_{g,in} - \dot{H}_g / (\dot{m}\times cp) \tag{11}$$

where $\dot{m}$ is generator mass flow rate and $\dot{H}_g$ is generator heating load which are determined as [31]:

$$\dot{m} = 0.03408\,\dot{Q}_{nom} - 0.1481 \tag{12}$$

$$H_g = \dot{Q}_e / COP \tag{13}$$

As a result, by using the chiller nominal capacity as well as generator inlet temperature, both chiller cooling capacity and generator outlet temperature could be evaluated using the above relations.

2.4. Auxiliary heater

During the chiller operation as well as in the absence of solar radiation, the generator inlet temperature decreases dramatically and as a result the performance and cooling capacity of the chiller decrease. To overcome this issue, the auxiliary heater is used to increase the storage tank heat when it drops to the boundary values as follows:

$$\dot{H}_{ax} = 0 \ \ when \ \ T_{stor} \geq T_{up,sp} \tag{14}$$

$$\dot{H}_{ax} = m_{stor}cp\left(T_{set,ax} - T_{stor}\right) \ \ when \ \ T_{stor} < T_{low,sp} \tag{15}$$

Where $T_{low,sp}$ and $T_{up,sp}$ are the lower and upper bounds of the comfortable space temperature. Moreover, $T_{set,ax}$ is the set point temperature of the auxiliary heater which should be adjusted in a case study.

2.5. Space cooling

The temperature of the space varies by outside air temperature and depends on the overall heat transfer coefficient. Applying the first law of thermodynamic, one can find the formula for the space temperature variation in each step size by considering the uniform variation of space temperature [31]:

$$T_{sp}(t+1)=T_{sp}(t)-\left[\frac{\dot{Q}_e+(UA)_{sp}\left(T_{sp}(t)-T_{amb}(t)\right)}{(m\times cp)_{sp}}\right]\tau \tag{16}$$

where U_{sp}, A_{sp}, $(m\times cp)_{sp}$ and T_{amb} are space overall heat transfer coefficient, total heat transfer surface area in the space, total space heating capacity and ambient temperature, respectively.

3. OBJECTIVE FUNCTION, DESIGN PARAMETERS AND CONSTRANTS

In this study, the total annual cost is considered as the objective function. Total annual cost includes investment cost (the annualized cost of absorption chiller, storage tank and solar collectors), operating cost of the auxiliary heater and the required electricity for some chiller electrical equipment [31]:

$$C_{total}=aC_{inv}+C_{op} \tag{17}$$

$$C_{inv}=C_{inv,ch}+C_{inv,stor}+C_{inv,col}=\left(b_1(\dot{Q}_{nom})^{d_1}+c_1\right)+b_2(m_{stor})^{d_2}+b_3(A_{col})^{d_3} \tag{18}$$

$$C_{op}=C_{op,ch}+C_{op,ax}=\sum_{i=1}^{N}\left(k_{el}\times\dot{E}_{ch}\times sens_1\right)_i\tau/3600+\sum_{i=1}^{N}\left(k_{el}\times\dot{E}_{ax}\times sens_2\right)_i\tau/3600 \tag{19}$$

here *b, d* and *c* are constant and vary based on the price of equipment available in the market. Moreover, N, k_{el}, τ, E_{ch} and E_{ax} are the number of operational seconds in a year, electricity unit price, step size, power demand by the chiller equipment (pump set, cooling tower and machine room) and auxiliary heater power consumption, respectively. It is worth mentioning that the chiller investment cost covers the entire cost of the chiller, pump set, and cooling tower as well as machine room. The $sens_1$ and $sens_2$ in Eq. (19) are a binary constant (0 or 1) for considering the operation hours of chiller and auxiliary heater.
In addition, a is the annual cost coefficient defined as:

$$a=\frac{i}{1-(1+i)^{-y}} \tag{20}$$

where i and y are interest rate and depreciation time, respectively. In this study the nominal capacity of chiller, solar collector surface area and storage tank capacity are considered as design parameters. In addition, a constraint is introduced to ensure that the maximum room temperature does not exceed the 25°C.

4. REAL PARAMETER GENETIC ALGORITHM (RPGA)

When binary coded GA is used to handle the problems having a continuous search space, a number of difficulties arise. One difficulty is the Hamming cliffs associated with certain strings from which a transition to a neighboring solution requires the alteration of many bits [32]. The other difficulty is the inability to achieve any arbitrary precision in the optimal solution. There exist a number of real parameter GA implementations, where crossover and mutation operators (0.9 and 0.03, respectively) are applied directly to real parameter values. Since real parameters are used directly, solving the real parameter optimization problems is a step easier when compared to the binary coded GAs. Unlike the binary coded GAs, decision variables can be directly used to compute the fitness values. Since the selection operator works with the fitness value, any selection operator used with binary coded GAs can also be used in the real parameter GAs.
As a summary, the followings are the steps of the RPGA used in this study for optimization of the solar cooling system:
1 - Initial population with *M* chromosome is randomly generated using upper and lower bounds of the design variables values, $x_{\min}$ and $x_{\max}$ which are listed in Table 1 are as follows:

$$x_0^t=x^{\min}+rand(x^{\max}-x^{\min}) \tag{21}$$

TABLE 1. Design parameters and their range of variation and step size

Parameter	From	To	Step size
Nominal capacity of the chiller (kW)	100	1000	5
Storage tank capacity (L)	5000	25000	1000
Solar collector surface area (m2)	500	2000	2

where *rand* is a uniformly distributed random.
2- Each chromosome is exported to the thermo-economic modeling and returned back with the value of objective function (total annual cost).
3- The selection operator is performed to determine the better chromosomes such as the binary GA [32].
4- Cross over operator is performed using the following relations [32]:

$$x_i^{(1,t+1)}=0.5\left\{\left(1+\zeta_i\right)x_i^{(1,t)}+\left(1-\zeta_i\right)x_i^{(2,t)}\right\} \tag{22}$$

$$x_i^{(2,t+1)} = 0.5\left\{(1-\zeta_i)x_i^{(1,t)} + (1+\zeta_i)x_i^{(2,t)}\right\} \tag{23}$$

where ζ_i is:

$$\zeta_i = \begin{cases} (2rand)^{1/(1+\alpha_c)} & if\ rand \le 0.5 \\ \left(\frac{1}{2-2rand}\right)^{1/(1+\alpha_c)} & otherwise \end{cases} \tag{24}$$

in which, α_c is cross over constant parameter.

5- Then, a mutation operator is performed on population as follows [32]:

$$x_i^{(1,t+1)} = x_i^{(1,t+1)} + \left(x_i^{\max} - x_i^{\min}\right)\sigma \tag{25}$$

where σ is:

$$\delta = \begin{cases} (2rand)^{1/(1+\alpha_m)} - 1 & if\ rand < 0.5 \\ 1-[2(1-rand)]^{1/(1+\alpha_m)} & if\ rand \ge 0.5 \end{cases} \tag{26}$$

in which, α_m is mutation constant parameter.

6- This procedure is repeated from step 2 until convergence is met.

5. CASE STUDY

The solar cooling optimization procedure is applied for a residential area located in Kerman, one of the province located in the south of Iran with the latitude of 25 degrees. The solar absorption cooling system provides the cooling load demand of residential apartments with a total area of 4000 m^2 with $(UA)_{sp} = 17.4\,kW/K$ and $(m \times cp)_{sp} = 36000\,kj/K$. The hourly and monthly average variations of the ambient temperature during the warm season including June-September for the studied case are shown in Fig. 2. The monthly average data is obtained from the data using $\overline{H} = \left(\sum_{i=1}^{M} H_i\right)/M$ where M is the number of time steps in a month.

The unit price of buying electricity from the grid, equipment lifetime and interest rate are considered as 0.2 \$/kWh, 20 years and 10%, respectively. Moreover, the constants of investment cost in Eq. 20 are taken as b=(1234,1.42,200), d=(0.7904,1,1) and c=(69630), based on the available equipment in the market. It is worth mentioning that, the chiller price includes all the cost of chiller as well as pump set, cooling tower and machine room. The results are obtained for N=122*24*3600 seconds for four months with $\tau = 12$ second as the step size. The mentioned step size is selected to have a reasonable result. Furthermore, $T_{low,sp}$, $T_{up,sp}$ and $T_{set,ax}$ are adjusted to 20, 24 and 70 ^{o}C, respectively.

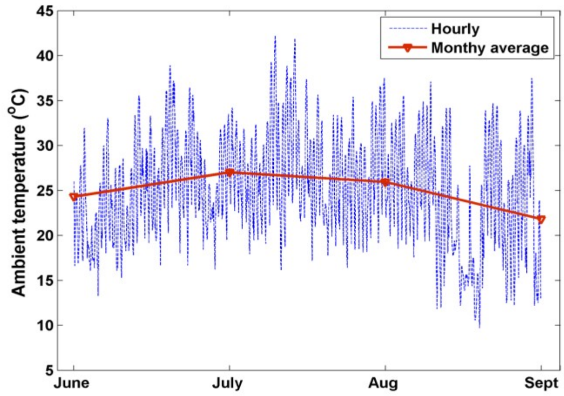

Figure 2. Hourly and monthly average variation of ambient temperature during the June to September for our case study [25].

6. RESULTS AND DISCUSSION

6.1. Optimization results

In this study, the total annual cost (TAC) is considered as the objective function and to minimize the TAC, three design parameters including the nominal capacity of the chiller, capacity of the storage tank as well as the collector surface area are selected. Design parameters (decision variables), the range of their variations and their step size are listed in Table 1. It is worth mentioning that, the market available equipment is selected. Moreover, the hourly received radiation during the June-September for our case study is obtained at optimum collector slope surface using Eq. 4 and shown in Fig. 3. Also the monthly average data is obtained and shown in Fig. 3 which has a good agreement (maximum 4.9% difference) with experimental data reported in [33]. Then, the RPGA is applied for 200 iterations, using 100 chromosomes, a mutation factor of $\alpha_m = 2$ and crossover factor $\alpha_c = 2$.

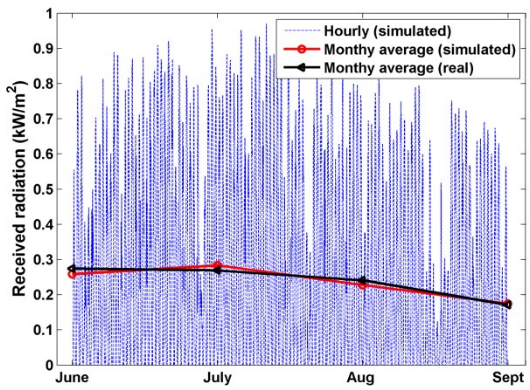

Figure 3. Hourly and monthly average variation of solar radiation during the June to September for our case study [25]

It is worth mentioning that the RPGA is run three times using a core-i7 -3200GHz processor for each case and the best results are presented here. Due to selecting the low step size ($\tau = 12$ second in this case) for analyzing the four months cooling during the warm season, each optimization needs 148 hours to be completed. To accelerate the optimization process, six separate optimization programs are run simultaneously on the mentioned processor.
The results of optimization including both optimum design parameters and objective function are listed in Table 2. An absorption chiller with nominal capacity of 300 kW with a 15000 L storage tank as well as a 1630 square meter solar collector is selected which leads to the minimum 65597 $/year annual cost. The main portion of cooling system price is for the solar collectors which is 79.94% more expensive than the optimum selected absorption chiller. Furthermore, 5.37% of the total annual cost is the operational cost (electricity price for auxiliary boiler, pump set, cooling tower and machine room).

TABLE 2. Optimum design parameters along with objective function

Parameter	Value
Nominal capacity of the chiller (kW)	300
Storage tank capacity (L)	15000
Solar collector surface area (m2)	1630
Total annual cost ($/year)	65597
Absorption chiller investment cost ($)	181170
Solar collectors investment cost ($)	326000
Storage tank investment cost ($)	21300
Annual cost of operation ($/year)	3523

Variation of space temperature during the warm seasons in the optimum point is depicted in Fig. 4 where the monthly average has the same trend with variation of ambient temperature and solar received radiation. As demonstrated, the space temperature significantly varied between the lower and upper bounds of comfortable space temperature which are selected to be 20 and 24 ^{o}C. Furthermore during some hours, the maximum temperature reaches the maximum allowable temperature which is selected to be 25 ^{o}C. Moreover, extending this temperature decreases both cost and comfort. In addition, the monthly average space temperature is varied in the range of 19.2-22 ^{o}C which is acceptable for the practical application.
Variations of generator inlet and outlet temperatures during the studied months as well as the monthly average results are shown in Fig. 5 and 6, respectively.

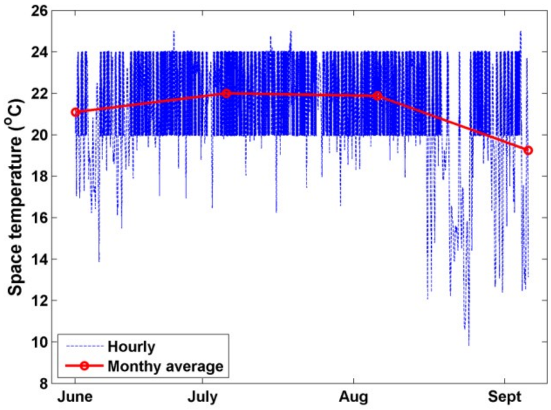

Figure 4. Hourly and monthly average variation of space temperature during the June to September in the optimum situation

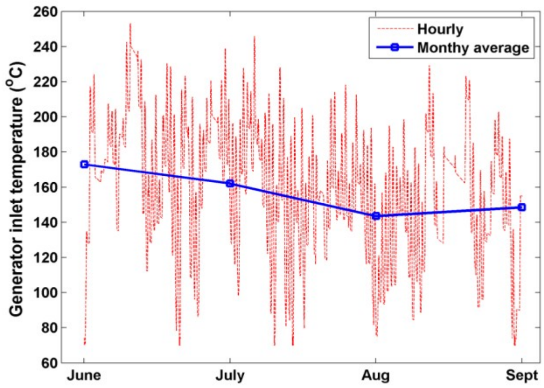

Figure 5. Hourly and monthly average variation of generator inlet temperature during the June to September in the optimum situation

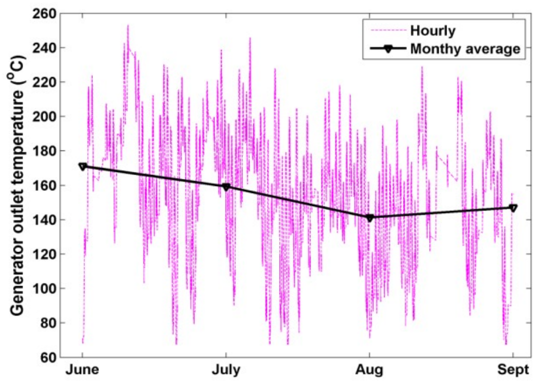

Figure 6. Hourly and monthly average variation of generator outlet temperature during the June to September in the optimum situation

The average inlet temperature is varied in the range of 144.5-173.9 ^{o}C while the generator outlet temperature is varied in the range of 141.3-171.1 ^{o}C. Actually, 3 ^{o}C is the average temperature difference between the inlet and outlet generator temperature. Due to the same trend of received radiation (higher radiation leads to the higher generator temperature) and ambient temperature (higher ambient temperature leads to the lower generator temperature), the trend of generator inlet/outlet temperature is not predictable. The hourly variation of space, generator inlet and outlet temperatures during the warm season for 2 days is depicted in Fig. 7. Days 15 and 16 for each month are selected.

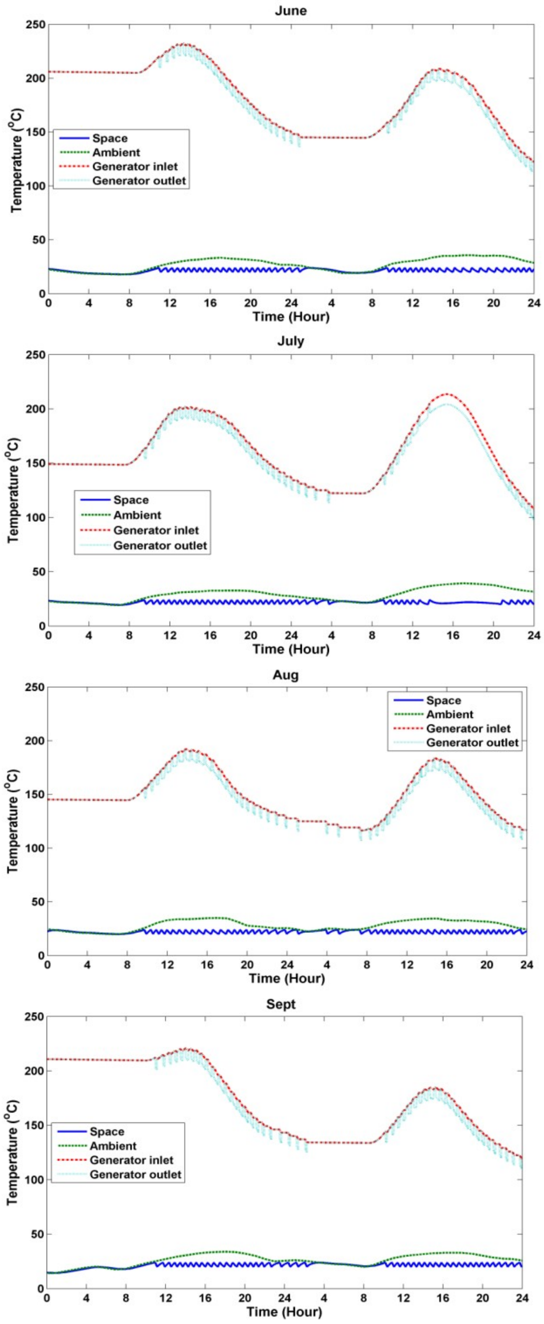

Figure 7. 48 hours variation of space, ambient, generator inlet and generator outlet temperature for the specific days during the June to September in the optimum point

In the absence of solar radiation, the ambient temperature as well as the rate of heat transfer to the space decreases. As a result, no chiller cooling is needed during the night hours (generally for the hours between 0 and 8) and generator inlet and outlet temperatures have the same value.

By the sunrise, both ambient temperature as well as the solar radiation increases. With an increase in the ambient temperature, the required space cooling load increases too. The chiller provides the demand load and as a result, the generator outlet temperature decreases compared with the generator inlet temperature. The space temperature fluctuates on average every 62 minutes during June and then decreases to 51 minutes in the September due to the decrease in the ambient temperature.

6.2. Sesitivity analysis on electricity price

Even though the primary required energy for absorption chiller is heat which is provided by solar collectors, some electricity is also needed for auxiliary equipment such as auxiliary boiler and pump sets. In fact, the electricity price is one of the important parameters in selection of optimum design parameters. To investigate the effect of electricity price (which was considered 0.2 $/kWh), the optimum objective function as well as the corresponding value of the design optimum design parameters for different prices of electricity in the range of 0.1-0.3 $/kWh were obtained and listed in Table. 3.

TABLE 3. Optimum design parameters and total annual cost for different electricity price

Parameter	kel=0.1 ($/kWh)	kel=0.2 ($/kWh)	kel=0.3 ($/kWh)
Nominal capacity of the chiller (kW)	275	300	305
Storage tank capacity (L)	12000	15000	19000
Solar collector surface area (m2)	1482	1630	1844
Total annual cost ($/year)	62763	65597	68871

With a 50% increase in electricity price, the optimum total annual cost just increases by 4.99% while a 4.32% decrease in the total annual cost is observed with a 50% decrease in the electricity price. This indicates that the electricity price does not have a significant effect on the value of the optimum objective function. It can be observed from Table 3 that the storage tank capacity as well as number of solar collector increases with an increase in electricity price. In fact, by increasing the electricity price, the number of solar collectors increases to provide the required heating for generator and this decreases the operational cost of electricity consumed by the auxiliary boiler. To have enough energy storage capacity for a larger number of solar collectors, the storage tank capacity is consequently increased by increase of solar collectors. In fact, the optimum number of collectors increases by 26.67% with a 50% increment in electricity price while it decreases by 20% with a 50% decrement in electricity price.

6.3. Sesivity analysis on rpga parameters

In the many optimization algorithms, particularly the evolutionary algorithms, some controlling parameters are used which affect the global optimum results and rate of convergence [34, 35]. These parameters should be selected appropriately to avoid algorithms trapping local optimum points. Furthermore, due to the semi stochastic nature of the evolutionary algorithms and unknown behavior of searching domain, generally it is not possible to estimate the best algorithm parameters and in many cases it should be evaluated by examination of the controlling parameters. Ideally, a suitable set of algorithm parameters leads to the same optimum result for each run. In this study, the results of the optimum total annual cost for six different algorithm parameters including population number, crossover and mutation constants are obtained and listed in Table. 4. The presented optimum results are the best of 3 runs. The results demonstrate that RPGA is more sensitive to the change in mutation constant rather than the crossover and the best results are achieved at levels 3 and 4 when the population number and mutation constant are 100 and 2, respectively, while the crossover constant varies between 2 and 2.2. Furthermore, the lower mutation constant is more suitable for the larger population size.

TABLE 4. Results of optimum total annual cost for various algorithm parameters

	Level 1	Level 2	Level 3	Level 4	Level 5	Level 6
Population size	50	50	100	100	150	150
Mutation constant (α_m)	2.2	2.2	2	2	1.8	1.8
Crossover constant (α_c)	2	2.2	2	2.2	2	2.2
Total annual cost ($/year)	65993	65891	65597	65597	65993	65668

7. CONCLUSIONS

An absorption cooling system assisted by solar collectors was designed optimally. After thermal modelling of the system by considering the hourly analysis during a year and selecting the nominal capacity of the chiller, solar collector surface area and storage tank capacity as decision variables, the Real Parameter Genetic Algorithm was used to obtain the minimum total annual cost. The optimization results showed that a 1630 square meter of solar collectors, a storage tank with a capacity of 15000L as well as an absorption chiller with a nominal capacity of 300kW were needed. The hourly analysis indicated that the space temperature was fluctuated on average every 62 minutes during June and decreased to 51 minutes in September due to the decreases in ambient temperature during September compared with that in June. Finally, the sensitivity analyses of change in the electricity price were also performed and results indicated that with a 50% increase in electricity price, the optimum total annual cost increases by 4.99% while a 4.32% decrease in total annual cost is observed with a 50% decrease in electricity price. In addition, the optimum number of solar collectors increased by 26.67% with a 50% increment in electricity price while it decreased by 20% with a 50% decrement in electricity price.

REFERENCES

1. Iran's Balance Sheet (2012), Tehran: Iran Department of Energy
2. Martínez, P.J., Martínez, J.C. and Lucas, M., "Design and Test Results of a Low-Capacity Solar Cooling System in Alicante (Spain)", ***Solar Energy,*** Vol. 86, (2012), 2950-2960.
3. Hajabdollahi, H., Ganjehkaviri, A. and Jaafar, M.N.M., "Thermo-economic optimization of RSORC considering hourly analysis", ***Energy,*** Vol. 87, (2015), 369-380.
4. Hajabdollahi, H., "Evaluation of cooling and thermal energy storage tanks in optimization of multi-generation system", ***Journal of Energy Storage,*** Vol. 4, (2015), 1-13.
5. Hajabdollahi, H., Ganjehkaviri, A. and Jaafar, M.N.M., "Assessment of new operational strategy in optimization of CCHP plant for different climates using evolutionary algorithms", ***Applied Thermal Engineering,*** Vol. 75, (2015), 468-480.
6. Hajabdollahi, H., "Investigating the effects of load demands on selection of optimum CCHP-ORC plant", ***Applied Thermal Engineering,*** Vol. 87, (2015), 547-558.
7. Fan, Y., Luo, L. and Souyri, B., "Review of Solar Sorption Refrigeration Technologies: Development and Applications", ***Renewable and Sustainable Energy Reviews,*** Vol. 11, (2007), 1758-75.

8. Sarbu, I. and Sebarchievici, C., "Review of Solar Refrigeration and Cooling Systems", ***Energy and Buildings,*** Vol. 67, (2013), 286-97.
9. Ferreira, C.I. and Kim, D.S., "Techno-Economic Review of Solar Cooling Technologies Based on Location-Specific Data", ***International Journal of Refrigeration,*** Vol. 39, (2013), 23-37.
10. Abdulateef, J.M., Sopian, K., Alghoul, M.A. and Sulaiman, M.Y., "Review on Solar-Driven Ejector Refrigeration Technologies", ***Renewable and Sustainable Energy Reviews,*** Vol. 13, (2009), 1338-1349.
11. Otanicar, T., Taylor, R.A. and Phelan, P.E., "Prospects for Solar Cooling–an Economic and Environmental Assessment", ***Solar Energy,*** Vol. 86, (2012), 1287-1299.
12. Buonomano, A., Calise, F. and Palombo, A., "Solar Heating and Cooling Systems by Cpvt and Et Solar Collectors: A Novel Transient Simulation Model", ***Applied Energy,*** Vol. 103, (2012), 588-606.
13. Chidambaram, L.A., Ramana, A.S., Kamaraj, G. and Velraj, R., "Review of Solar Cooling Methods and Thermal Storage Options", ***Renewable and Sustainable Energy Reviews,*** Vol. 15, (2011), 3220-3228.
14. Hammad, M. and Zurigat, Y., "Performance of a Second Generation Solar Cooling Unit", ***Solar Energy,*** Vol. 62, (1998), 79-84.
15. Syed, A., Izquierdo, M., Rodriguez, P., Maidment, G., Missenden, J., Lecuona, A. and Tozer, R., "A Novel Experimental Investigation of a Solar Cooling System in Madrid", ***International Journal of Refrigeration,*** Vol. 28, (2005), 859-871.
16. Marc, O., Praene, J.P., Bastide, A. and Lucas, F., "Modeling and Experimental Validation of the Solar Loop for Absorption Solar Cooling System Using Double-Glazed Collectors", ***Applied Thermal Engineering,*** Vol. 31, (2011), 268-277.
17. Marcos, J.D., Izquierdo, M. and Parra, D., "Solar Space Heating and Cooling for Spanish Housing: Potential Energy Savings and Emissions Reduction", ***Solar Energy,*** Vol. 85, (2011), 2622-2641.
18. Yin, Y.L., Song, Z.P., Li, Y., Wang, R.Z. and Zhai, X.Q., "Experimental Investigation of a Mini-Type Solar Absorption Cooling System under Different Cooling Modes", ***Energy and Buildings,*** Vol. 47, (2012), 131-138.
19. Florides, G.A., Kalogirou, S.A., Tassou, S.A. and Wrobel, L.C., "Modelling and Simulation of an Absorption Solar Cooling System for Cyprus", ***Solar Energy,*** Vol. 72, (2002), 43-51.
20. Assilzadeh, F., Kalogirou, S.A., Ali, Y. and Sopian, K., "Simulation and Optimization of a Libr Solar Absorption Cooling System with Evacuated Tube Collectors", ***Renewable Energy,*** Vol. 30, (2005), 1143-1159.
21. Argiriou, A.A., Balaras, C.A., Kontoyiannidis, S. and Michel, E., "Numerical Simulation and Performance Assessment of a Low Capacity Solar Assisted Absorption Heat Pump Coupled with a Sub-Floor System", ***Solar Energy,*** Vol. 79, (2005), 290-301.
22. Sanjuan, C., Soutullo, S. and Heras, M.R., "Optimization of a Solar Cooling System with Interior Energy Storage", ***Solar Energy,***Vol. 84, (2010), 1244-1254.
23. Calise, F., d'Accadia, M.D. and Vanoli, L., "Thermoeconomic Optimization of Solar Heating and Cooling Systems", ***Energy Conversion and Management,*** Vol. 52, (2011), 1562-1573.
24. Praene, J.P., Marc, O., Lucas, F. and Miranville, F., "Simulation and Experimental Investigation of Solar Absorption Cooling System in Reunion Island", ***Applied Energy,*** Vol. 88, (2011), 831-839.
25. Mateus, T. and Oliveira, A.C., "Energy and Economic Analysis of an Integrated Solar Absorption Cooling and Heating System in Different Building Types and Climates", ***Applied Energy,*** Vol. 86, (2009), 949-957.
26. Hang, Y., Du, L., Qu, M. and Peeta, S., "Multi-Objective Optimization of Integrated Solar Absorption Cooling and Heating Systems for Medium-Sized Office Buildings", ***Renewable Energy,*** Vol. 52, (2013), 67-78.
27. Mazloumi, M., Naghashzadegan, M. and Javaherdeh, K., "Simulation of Solar Lithium Bromide–Water Absorption Cooling System with Parabolic Trough Collector", ***Energy Conversion and Management,*** Vol. 49, (2008), 2820-2832.
28. Duffy, J., Beckman, W., Solar Engineering of Thermal Processes, Wiley&Sons, New York, (1991).
29. Sepehr, S. and Hassan, H., "Thermo-economic optimization of solar CCHP plant using Genetic and Particle swarm Algorithms", ***Journal of Solar Energy Engineering,*** Vol. 6, (2014), 430-442.
30. Khorasaninejad, E. and Hajabdollahi, H., "Thermo-Economic and Environmental Optimization of Solar assisted Heat Pump Plant by using Multi-Objective Particle Swarm Algorithm", ***Energy,*** Vol. 72, (2014), 680-690.
31. Broad, X., Non-Electric Chiller, Model Selection and Design Manual, Broad Central Air Conditioning (Absorption LiBr+ H20). Sept (2008).
32. Deb, K., Multi-objective optimization using evolutionary algorithms. Chichester: John Wiley and Sons Ltd, (2001).
33. Safaripour, M.H. and Mehrabian, M.A., "Predicting the direct, diffuse, and global solar radiation on a horizontal surface and comparing with real data", ***Heat and Mass Transfer,*** Vol. 47, (2011), 1537-1551.
34. Hajabdollahi, H., Ahmadi, P. and Dincer, I., "Modeling and Multi-Objective Optimization of Plain Fin and Tube Heat Exchanger Using Evolutionary Algorithm", ***International Journal of Thermophysics and Heat Transfer,*** Vol. 3, (2011), 424-431.
35. Hajabdollahi, H., Ahmadi, P. and Dincer, I., "Exergetic optimization of shell-and-tube heat exchangers using NSGA-II", ***Heat Transfer Engineering,*** Vol. 33, (2012), 618-628.