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Research article

Energy and exergy simulation analysis and comparative study of solar ejector cooling system using TRNSYS for two climates of Iran

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ABSTRACT

This paper addresses hourly simulation of 3.5 kW Solar Ejector Cooling System (SECS) using R600a and R290 hydrocarbon refrigerants for application in two office buildings in semi-arid and hot-humid climates of Iran. During the period of the study, thermodynamics energy and exergy of the cooling systems when charged with the two refrigerants are fully assessed by simulation at the two study sites. The simulation studies of the entire cooling system indicate that the most irreversible process and hence the prime exergy destruction is related to the solar collector system followed by the ejector component in the cooling cycle. The ejector is a constant-area mixing (CAM) type which is mathematically modeled in Engineering Equation Solver (EES) software. Generator of the cooling cycle is modeled in EES using $\varepsilon - NTU$ method and a simulation program is developed on TRNSYS-EES co-simulator for dynamic study of the cooling cycle. For comparison of efficiency of the two refrigerants, working conditions are set to be the same. The systems are equipped with auxiliary heaters to provide constant inlet temperature of 85 °C for the generator when solar radiation is partially in phase with the building sites. The hourly and monthly simulation of both SECS in June, July, August and September 2019 demonstrate that R290 is more efficient for increasing the overall $COP (= 0.2844)$ of the system than R600a ($COP = 0.2797$) of the building office in the semi-arid region where the generator receives most of its thermal energy from solar radiation in July 17, 2019. Although, the same refrigerant is also more efficient than R600a in the hot-humid region system in the same day, but the system compensates shortage of its necessary solar thermal energy mostly from the auxiliary heater.

1. Introduction

Air conditioning and refrigeration technologies account for almost 20% of total electricity consumption of building cooling loads around the world and is projected to increase rapidly to more than 40 times in 2100 compared to 2000 in the hot regions of the world, such as Southern Asian and the Middle East countries [1]. Nowadays, majority of such technologies are utilized by vapor compression systems which consume significant electrical energy primarily generated by power plants which consume fossil fuels, including Mazut which are all harmful pollutants for the environment [2]. Solar cooling technology has been an attractive subject for various researches for replacing the common cooling systems due to its non-polluting character and compatibility with the environment [3]. Among such cooling technology, the solar ejector cooling system (SECS) is subjected to wide research activities because of its unique ability to operate with low thermal grade heat supply, such as solar energy. SECS has also unique advantages, such as high reliability, simplicity in structure and low maintenance cost [4, 5, 6]. Despite such advantages, SECS suffers from complexity of the detailed ejector design and its limited operating range, optimization of the complex flow field, and the overall coefficient of performance (COP), usually around 0.3 [7].

Iran is located in the Middle East Basin with a wide range of climatic conditions. Although Iran endowed with abundant renewable and non-renewable energy sources, but it exploits a great deal from the non-renewable sources which cause air pollution and climate change. This country has a unique geographical characteristic and can appropriately substitute clean sources of energy, such as solar to reduce its

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dependence on fossil fuels [8]. The annual-mean daily global solar irradiation in Iran is estimated at 4.5–5.4 kWh/m^2 in about four-fifths of the horizontal surfaces of the lands [9]. It has been reported that there is approximately 300 clear sunny days in a year in Iran and if only 1% of the total area could harness only 10% of this energy, then about 9 million-megawatt hour (MWh) of solar energy can be obtained daily in Iran [10]. The feasibility of the application of solar systems in different regions in Iran is thoroughly studied by means of proper economic criteria and a lifetime of 25 years for capital investment, and is reported that utilization of solar systems in most parts of this country is quite feasible [11, 12].

Early studies on ejector cooling system (ECS) were done by Huang et al. [13] to increase efficiency of the system in a 1-D analysis of ejector performance. The study used R141b as the refrigerant and the primary aim was to find the effective parameters for optimizing the ejector design. It was then demonstrated that the rate of primary and secondary superheat were crucial factors in the performance of the ejector and the overall performance of the ECS.

The types and properties of solar collectors on the performance of SECS were examined by Huang, et al. [14]. Three types of solar collectors, including conventional flat plates, high efficiency flat plates and vacuum tube collectors were examined in the study. It was found that the quality of the surface, proper insulation and high quality of flat plate type which was found to be the proper choice, were crucial factors in increasing the efficiency and also reducing of the cost of such cooling system.

A comparative thermodynamic study for finding the best environment-friendly refrigerant for application in SECS was conducted by Tashtoush and Bani-Younes [15]. Different refrigerants were examined in a fixed ejector geometry, using a set of standard operating conditions. The results of the study found that R1234yf was the best choice for the cycle in comparison with others and claimed that it was environment-friendly with thermophysical properties similar to R134a. In addition, when compared to other tested refrigerants, it has high entrainment ratio and is cheap, nonflammable and safe to be used in SECS.

A theoretical study was carried out by Varga et al. [16] on assessment of system and refrigeration efficiencies of a solar ejector cycle using water as the refrigerant. The modelling of the cooling system was based on 1-D ejector approach which included both the refrigeration and solar collector cycles. The aim of the study was to evaluate the ejector performance in different operating conditions. The results indicated that for reaching an acceptable COP of the cooling system, the temperatures of the generator should not fall below 90 °C. It was also found that if evaporator temperatures fall below 10 °C and condenser temperatures go above 35 °C, then the system efficiency significantly drops, and therefore, such temperature conditions were set to be the minimal design values.

Exergy analysis of ejector refrigeration system is used to find useful information for identifying the direction steps for the system improvement. A comprehensive study of an ejector refrigeration system using conventional and advanced exergy analysis at a given operating conditions was performed by Chen et al. [17]. The work demonstrated that the highest exergy destruction was occurred in the ejector with 53.6% of the total system exergy destruction, followed by the generator and then the condenser. The study found that splitting the exergy destruction happened into two parts, namely endogenous and exogenous, revealing strong interdependencies among the system components. It was then discussed that the destruction of exergy in the generator could be mainly reduced by improving of other components of the system and destruction in the condenser also could be reduced as such. The above points on splitting the exergy showed that 35% of the overall exergy destruction could be avoided in the system as well. Moreover, it was shown that combination of splitting could identify the order for system improvements if the ejector components would be the prime concern, followed by condenser and the generator at last.

A recent feasibility study of SECS in different weather conditions was conducted by Khalilpour et al. [18]. The thermodynamic analysis in this study was done on 24 Iranian cities with various geographical and climatic conditions. The refrigerants used in this study were some common hydrocarbons, such as R113, R114, R600, R600a, and R141b. It was shown that COP of the systems increase by increase the temperatures of the evaporator and the generator, but such increases caused a drop in the condenser temperature. It was then concluded that increase the system performance depends on exact controlling the condenser temperature to stay at a specified low degree and at the same time, when a specified refrigerant was used, the evaporator temperature should be controlled to stay at higher degrees. The reported ranges of COP of the systems for the 24 cities were reported between 0.43 to 0.66 from April to October, the time the study was performed. The study did not report that the reported COPs are overall or not. Applications of simulation software to study behaviors of various solar cooling and heating systems are so numerous and, the present paper is tried to refer to only few related previous works which their studies and techniques are somehow coherent to our current works.

The need to simulate a system over time is a very important step, especially when the solar energy is used which is a transient source. Vidal et al. [5] applied TRNSYS-EES as the co-simulator to simulate a 10.5 kW cooling cycle, using R141b as the refrigerant. The modelling work assisted to predict the type, area, size, angle and also the hot water mass flow rate of the solar flat collector. The work then concluded that an increase in the collector area from 20 to 80 m^2 resulted reduction of annual energy consumption of the solar cooling system.

Hourly dynamic simulation study on a 7.3 kW SECS for hot-dry Jordanian climate was done by Tashtoush et al. [19], using TRNSYS-EES co-simulator. The refrigerant choice was R134a hydrocarbon type and a one-year study resulted a minimum overall COP of 0.32. It was also demonstrated that solar thermal energy requirement for such SECS could be obtained from 60–70 m^2 evacuated-tube solar collector with a solar fraction of 0.52–0.542.

In typical theoretical studies on SECS, numerical method of analysis is used for simulation of the cooling system to select those parameters affect the performance of SECS. For example, combination of the lumped and dynamic methods is applied to predict effectiveness of the system in an office building during specific hours, using R134a refrigerant [20]. It is then demonstrated that the system can save 80% electric energy when is compared with the traditional compressor-based air conditioner by the same cooling capacity [21]. Such studies suggest that application of simulation software for evaluating performance of refrigerants in SECS is crucial for designing proper cooling ability [21, 22]. The simulation is usually done with the help of TRNSYS studio, but since the ejector component of such solar system is not included in the standard TRNSYS component library a mathematical model of this component is developed in Engineering Equation Solver, EES [23]. TRNSYS is a complete transient simulation studio which has worldwide applications by researchers and engineers. It is a powerful software which is primarily used in the areas of renewable energy and buildings for either active or passive designs [19].

Building spaces in Iran account for almost 36% of overall country's energy consumption and such spaces consume 33% of the total electric power [11]. A large part of electric energy in Iran is consumed for cooling and heating of building spaces and is dissipated due to non-intelligent design, poor materials and improper insulations of the buildings [24]. The cooling system in the semi-arid regions in Iran is customarily obtained from the old fashion water evaporative system and in the hot-humid parts, houses use conventional air conditioning, both of which consume high electric energy. It is reported that electrical energy consumption in Iranian buildings is more than double the world standard [25].

The work presented in this study focuses on application of dynamic simulation of SECSs using R290 and R600a refrigerants for two office building spaces with the same dimensions and cooling loads in Tehran

Table 1. The geographical coordinates of Tehran and Bandar Abbas.

Site	Latitude	Longitude
Tehran	35.34°	51.6°
Bandar Abbas	27.18°	56.27°

(semi-arid region) and Bandar Abbas (hot-humid region), using TRNSYS 16 and EES co-simulator. The simulation of both systems is focused on four selected days in June, July, August and September of 2019 when the solar radiation, G (W/m^2), is higher in both regions. Selection of the two refrigerants is based on their environmental safety characteristics, great cooling system efficiency and their wide application in household refrigeration and commercial cooling systems. The simulation work includes estimation of exergy destruction and points out any irreversibility and malfunctions occurring in the component of the refrigeration system when using the two refrigerants. The COP, transient U-value changes in the heat exchanger and also the thermal energy fraction for the two refrigerants are calculated and compared together.

2. Geographical and climatic characteristics of the SECSs sites

The office buildings are located in two cities with completely different climatic regions. Tehran is near the northern part, close to the Caspian Sea and Bandar Abbas is located in the southern part by the Gulf. The geographic coordinates of Tehran and Bandar Abbas are tabulated in Table 1. The maximum horizontal solar radiation received in Tehran was 1070 W/m^2, while Bandar Abbas received 960 W/m^2 during the study time in 2019 (22 June–23 September). The average humidity in Tehran and Bandar Abbas were 35% and 68% when the average temperatures of the cities reached 33 °C and 41 °C, respectively.

3. System description

The simulation is carried out for 3.5 kW solar systems for operating with the same cooling loads in buildings in both cities. The present research exploits of two hydrocarbon refrigerants, R290 and R660a. Such refrigerants are environmentally friendly and do not have any harmful effects on ozone depletion layer. Different properties of some typical characters of common hydrocarbon refrigerants are listed in Table 2. The properties comparison of R290 and R600a compared to R12 and R134a indicates that the selected two refrigerants in the present work have desired boiling points, critical temperatures and pressures. The running cycle for the SECS is schematically presented in Fig. 1. As depicted, the solar collector subsystem contains of solar flat collectors, a storage tank and a pump. The collector array (55 m^2) warms up the running water with the mass flow rate of 0.825 kg/s and transfers it to the storage tank (1.2 m^3). The hot water is then transferred with a pump to the generator unit causing the refrigerant in the ejector cooling cycle to vaporize. The refrigerants R600a and R290 vapors (T_f) then passes through an auxiliary heater for further warming and enters as a high-pressure primary fluid through the nozzle of the ejector where it starts to accelerate. The vapors are then mixed with the secondary low-pressure vapor flowing from the evaporator (T_e) in the mixing zone of the ejector to form a single stream, the transient supersonic stream. The single stream is then discharged through a diffuser into the condenser unit to transfer the heat to the air. The liquid fluid in the condenser unit is entered to the evaporator via a throttling valve and at the same time to the generator by help of a pump to complete the ejector cooling cycle. The generator specifications are listed in Table 3.

The COP of the ejector cooling cycle can be calculated as follows:

$$COP = \frac{Q_e}{Q_g} = m\frac{(h_9 - h_8)}{(h_3 - h_1)} \tag{1}$$

where h_9, h_8 are the enthalpy of the outlet and inlet of the evaporator, respectively, and h_3, h_1 are enthalpy of the outlet and inlet of the generator, respectively. In this equation, m is the relation of primary mass flow of the refrigerant (the evaporator) to the secondary mass flow of the refrigerant (the generator). In Equation (1), Q_e is the cooling capacity of the system and Q_g is the amount of heat that the refrigerant needs to absorb for reaching T_g. In our work, the temperatures of the evaporator (T_e) and the condenser (T_c) are 15 °C and 35 °C, respectively. The generator temperature, 85 °C, is assumed constant in our work, since any deviation from this value causes dropping of performance of the system and reduction of the compression ratio. The constant temperature of the generator is maintained with the help of an auxiliary heater when the solar thermal energy drops in each of the two regions.

4. Modeling of the components of SECSs

Fig. 2 presents schematic diagram showing a short description of TRNSYS model component set up of the solar cooling units. The red lines in the model present the solar heating subsystem and the blue ones show the ejector cooling cycle. In the TRNSYS diagram, the climatic condition and geographical data of Tehran and Bandar Abbas are shown as Type 109-TMY2. The data consist of solar irradiation for both horizontal and inclined surfaces and also the ambient temperature of the building sites. The active thermal energy for the units directs from the flat-collectors which is depicted as Type 1b in this figure. The throttling valve which adjusts the proper pressure for receiving hot water from the collector array allows only the hot water and not the water vapors to enter the storage tank (Type 1) and Type 4a is the storage tank. This unit in part circulates back the hot water by means of a pump, Type 3b, which completes the solar collector subsystem of the unit. The tank is specially designed for running the hot water to the ejector cooling cycle which is shown by T_s in Fig. 1. For transient analysis of the systems, the TRNSYS 16 model is coupled with the EES program and by using Type 66a, the ejector modeling is also developed.

4.1. Modeling of the generator

The generator heat exchanger is conjugated to the ejector component in the cooling cycle. This component in fact connects the solar collector subsystem and the cooling cycle. As a shell and tube type vapor generator, it is considered as an important inter connector part of the units. Modeling of this component hosts the EES software and the rate of heat transfer in the generator is modeled using the method of $\varepsilon - NTU$. If only hot and cold refrigerant inlet temperatures are specified, the $\varepsilon - NTU$ method can be considered as an effective tool to calculate the rate of heat transfer in the generator. The modeling of the generator in our study is carried out for R600a and R290 refrigerants.

Table 2. Different refrigerants characteristics.

Refrigerant	Molecular weight (g/mol)	Boiling point (°C)	Freezing point (°C)	Critical temperature (°C)	Critical pressure (MPa)	Latent heat (kJ/kg)
R12	120.9	−29.8	−158	112	4.14	166.2
R134a	102	−26.1	−104	101.1	4.06	217
R290	44.1	−42.1	−188	96.7	4.25	421.4
R600a	58.12	−11.7	160	134.7	3.64	364.4

Table 3. Specifications of the generator.

Heat exchanger material	Type of the heat exchanger	Inner diameter	Outer diameter	Length of the tubes
Copper	Double tube	0.635 cm	3.810 cm	3.75 m

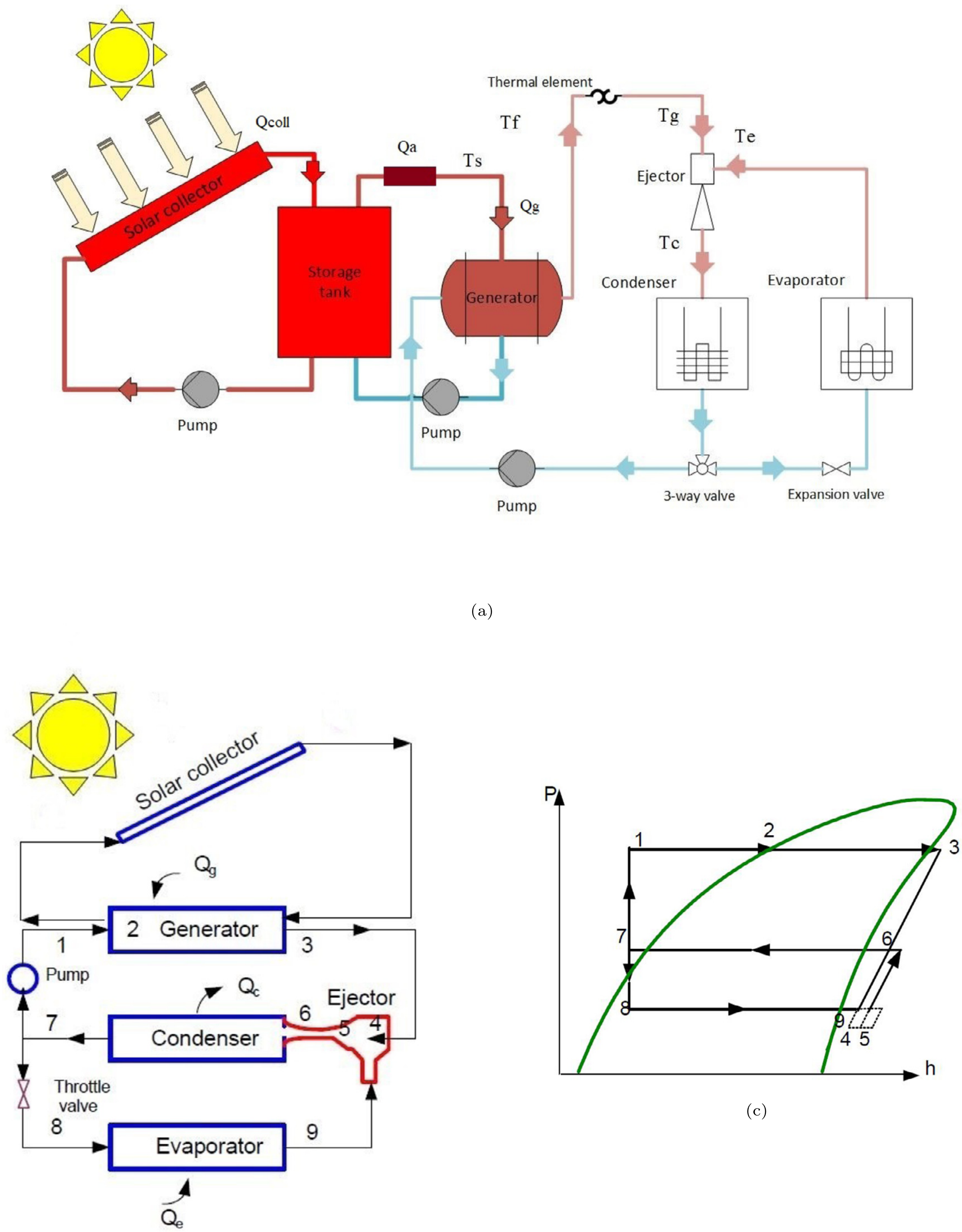


Fig. 1. (a) Schematic diagram of the solar ejector cooling system (SECS) components, (b) and (c) Ejector refrigeration cycle.

The velocities of the refrigerants in the pipe are calculated using Equation (2).

$$V = \frac{\dot{m}}{\rho A} \tag{2}$$

where $\dot{m}$, A, ρ are mass flow rate (kg/s), area of the solar collector (m^2), density (kg/m^3), respectively. By using V, then the Reynolds number (R_e) is obtained via Equation (3).

$$R_e = \frac{\rho V D}{\mu} \tag{3}$$

where D, μ are diameter (m), viscosity (kg/m.s), respectively.

R_e then helps to calculate the Nusselt number (Nu) via Equation (4).

$$Nu_D = 0.023 R_e^{\frac{4}{5}} Pr^n \tag{4}$$

The exponent of the Prandtl (Pr) number is $n = 0.4$ for heating of the fluid and $n = 0.3$ if the fluid is being cooled.

The heat transfer coefficient in the generator is calculated via Equations (5)–(9):

$$Nu_D = \frac{h.D}{K} \tag{5}$$

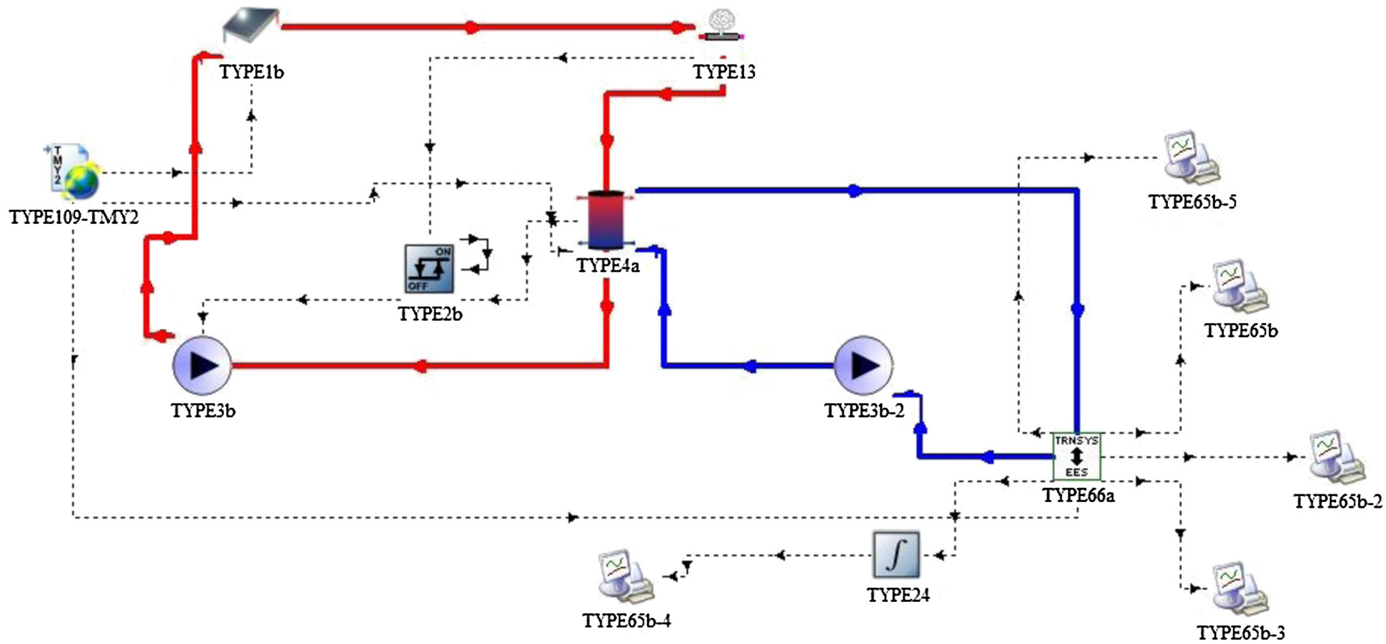


Fig. 2. TRNSYS project – simulation studio environment.

where h, K are convection heat transfer coefficient (W/m^{2}K), conductivity (W/m.K), respectively.

Equation (5) assists calculation of h which is then used to find the total heat transfer coefficient of the generator via Equation (6).

$$\frac{1}{UA} = \frac{1}{(hA)_c} + \frac{R''_{f.c}}{A_c} + R_w + \frac{R''_{f.h}}{A_h} + \frac{1}{(hA)_h} \tag{6}$$

where, $R''_{f.h}$ and $R''_{f.c}$ are fouling factor of the hot water and refrigerant in the pipe, respectively. The hot water fouling factor is 0.0002 and for both refrigerants is considered zero. R_w is the resistance of the pipe itself which can be calculated using Equation (7).

$$R_w = \frac{ln\frac{r_2}{r_1}}{2\pi LK} \tag{7}$$

where L, r_1, r_2 are length of the heat exchanger (m), inner radius of pipe (m), external radius of pipe (m), respectively.

Since the thickness of the tube is assumed insignificant, $r_2 \cong r_1$, then the value of R_w is considered as zero. By this assumption, the Equation (6) can be changed to the following equations:

$$\frac{1}{UA} = \frac{1}{(hA)_c} + \frac{1}{(hA)_h} \tag{8}$$

$$A = \pi \cdot Di \cdot L \tag{9}$$

Finally, by inserting the value of A into Equation (8), the total heat transfer coefficient U (kW/m^{2}K) of the generator is obtained.

4.2. Solar collectors and storage tank modeling

The solar energy conversion coefficient of the tilted flat plate collectors, having β angle with the horizon, is obtained via Equation (10) [26].

$$\bar{R}_b = \frac{\cos(\phi-\beta)\cos\delta\sin\omega'_s + (\frac{\pi}{180})\omega'_s\sin(\phi-\beta)\sin\delta}{\cos\phi\cos\delta\sin\omega_s + (\frac{\pi}{180})\omega_s\sin\phi\sin\delta} \tag{10}$$

where, ω'_s is the clock angle at sunset on the tilted surface, ω_s is the clock angle at sunset on the horizon surface, ϕ is the latitude, β is the slope and δ is the angle of deviation.

The energy of the cooling systems which is received by solar irradiation in each site can be obtained via Equation (11),

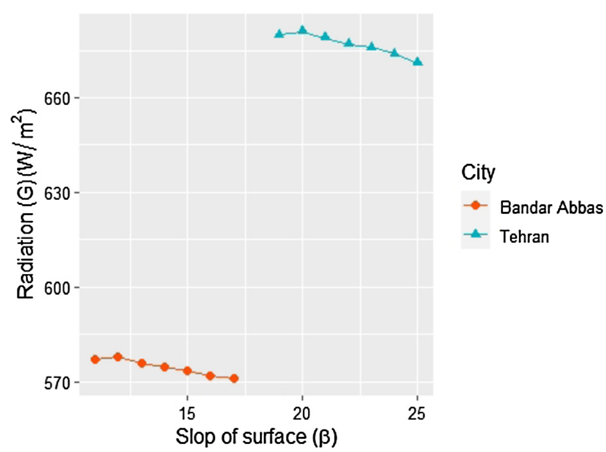


Fig. 3. Average radiation on the surface vs. surface angle.

$$Q_u = A_c F_R[\ s - U_L(T_i - T_a)] \tag{11}$$

Where, F_R is the heat removal factor, s is the solar energy, U_L is collector array heat transfer coefficient, T_i is the temperature of water and T_a is the ambient temperature. The energy balance of the storage tank can be expressed as:

$$(mC_p)_s \frac{dT_s}{dt} = Q_u - \dot{L}_s - (UA)_s(T_s - T_a) \tag{12}$$

where Q_u and $\dot{L}_s$, represent rates of entering solar heat energy from collector array to the tank and the loss of heat to the surrounding from the tank, respectively, while, T_a is the ambient temperature.

In order to collect maximum solar radiation for the systems in the two cities, solar energy calculation should be scheduled to be 10–15 degrees less than the latitudes of the two cities, as shown in Fig. 3. Therefore, the collector's tilt angles for Tehran and Bandar Abbas were adjusted to 20 degree and 12 degree, respectively for receiving maximum average solar energy of 681 W/m^2 (Tehran) and 578 W/m^2 (Bandar Abbas) on July 17. The solar collectors are flat-plates and the inlet temperature of water was 20 °C. The collector's array cross sectional area was considered 55 m^2 and the inlet flow rate of water to the collectors was taken 0.825 kg/s.

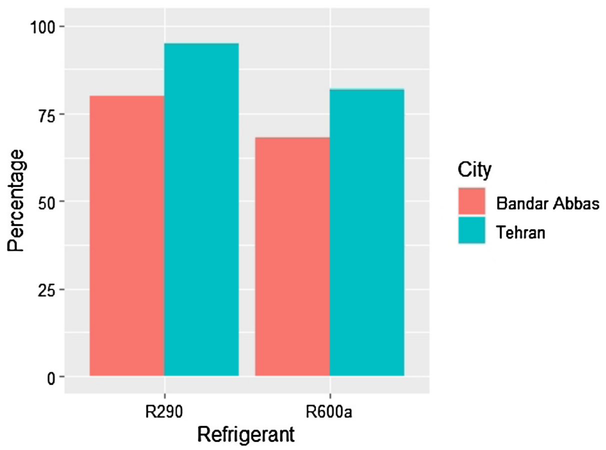


Fig. 4. Thermal energy fractions for R290 and R600a refrigerants.

The thermal energy fraction (S_R), is defined as the energy provided by the solar technology divided by the total energy required for the systems to operate. For the solar units, this fraction is defined in the following equation.

$$S_R = \frac{\dot{Q}_{hx}}{\dot{Q}_g} \tag{13}$$

where $\dot{Q}_{hx}$ is calculated according to the generator heating regime and $\dot{Q}_g$ is the amount of heat scheduled to be absorbed by the generator for reaching a specific temperature, which in the present study was 85 °C. Approaching the value of S_R to 1 can force the solar system to receive most of its thermal energy from the solar. The solar thermal fractions for the refrigerants for the two cities are illustrated in Fig. 4. As it is noticed in this figure, the thermal energy fraction for R290 is higher than R600a for both cities. This demonstrates that R290 refrigerant absorbs more energy from the collector array subsystem and therefore the thermal energy fraction for the solar cooling in Tehran was much higher than that for Bandar Abbas. This figure also depicts that the thermal fractions of both refrigerants are approximately 15 percent higher in Tehran than that for Bandar Abbas. The shortage of thermal fraction of refrigerants in the southern system is the effect of high humidity blocking of portion of solar radiation for receiving the collector arrays and as a result, the necessary thermal energy for the generator of the SECS is reduced.

4.3. Modelling of the auxiliary heater

As depicted in Fig. 1, the auxiliary heater (Q_a) is a component located between the storage tank and the generator which is an additional heat source to provide a constant inlet temperature (T_g) of the generator (85 °C). A mathematical model of the auxiliary heater is developed in EES. The required thermal input of this component is simply calculated as:

$$Q_a = Q_g - Q_{coll} \tag{14}$$

$$Q_{coll} = \eta_{coll} A_{coll} G \tag{15}$$

where η_{coll} is efficiency of the collector.

5. Exergy analysis of the SECS

Exergy analysis of SECS is used to identify the causes, locations and magnitude of the process inefficiencies in the system, including the solar flat plate collector subsystem and the ejector cooling cycle. The main purpose of the analysis is to examine energy and exergy balance in the solar system.

Governing equations which are applied in exergy analysis of the solar collector subsystem and the ejector cooling cycle of SECS (the detail elements in the exergy Equations (16)–(28) are defined in the text):

(a) Equations which are used in the exergy analysis for the solar collector subsystem in SECS: [27]

Exergy balance of the solar collector subsystem

$$E_{s,h} + E_e + W_{p,el} = E_{c,out} + I_{total} \tag{16}$$

where $E_{s,h}$, E_e, $W_{p,el}$, $E_{c,out}$, I_{total} are exergy input to the solar collector (kW), exergy cooling load (kW), electricity input to the pump (kW), exergy output from the condenser (kW), total irreversibility from all parts in the cooling cycle (kW), respectively.

Exergy (heat) input

$$E_{s,h} = Q_{ava} \cdot \left(1 - \frac{T_{ref}}{T_{coll}}\right) \tag{17}$$

where Q_{ava}, T_{ref}, T_{coll} are available heat for the process in the solar collector (kW), reference temperature (°C), average temperature of the solar collector (°C), respectively.

Loss (heat transfer)

$$I_{coll} = E_{s,h} - E_{su} \tag{18}$$

where I_{coll} and E_{su} are irreversibility of the solar collector (kW) and useful exergy input to the solar collector (kW), respectively.

Useful exergy gain

$$E_{su} = Q_u \cdot \left(1 - \frac{T_{ref}}{T_{coll}}\right) \tag{19}$$

where Q_u is useful heat absorbed by collector (kW).

(b) Equations which are used in the exergy analysis of different components in the ejector cooling cycle: [27]
This cycle includes the generator, ejector, condenser, pump, evaporator and the expansion device. The sub numbers in the equations for exergy analysis of the cooling cycle are depicted in Fig. 1.

Generator

Exergy available

$$E_g = Q_g \cdot \left(1 - \frac{T_{ref}}{T_g}\right) \tag{20}$$

where E_g is exergy input to the generator (kW).

Exergy loss

$$I_g = T_{ref} \cdot \left[(m_g(S_3 - S_1) + m_{coll}(S_{g,coll_{out}} - S_{g,coll_{in}})\right] \tag{21}$$

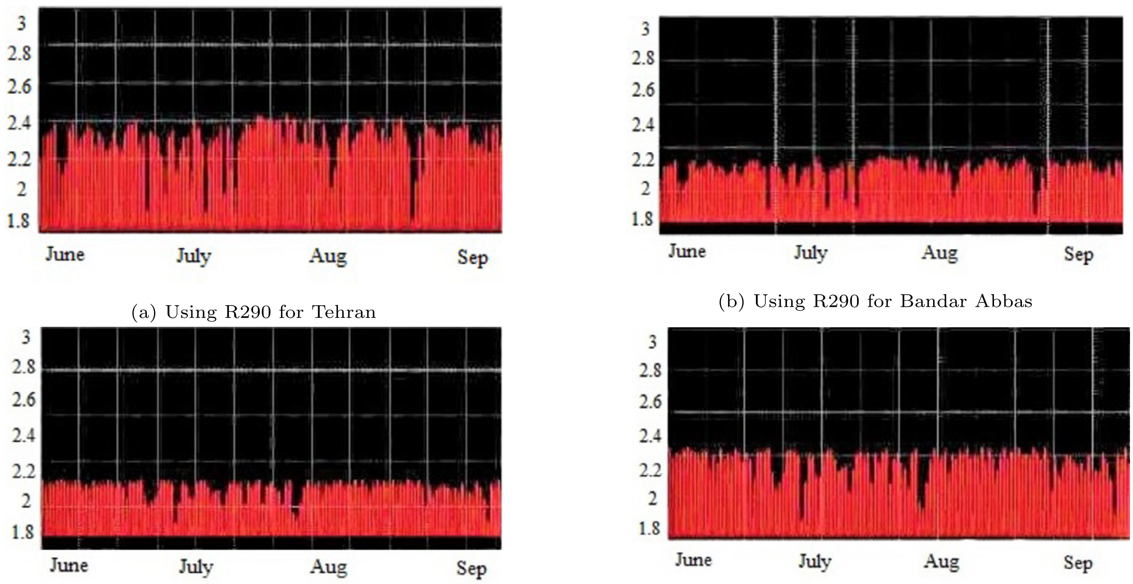
where I_g, m_g, S, m_{coll} are irreversibility in the generator (kW), mass flow of driving fluid from the generator (kg/s), entropy (kJ/kg.K), mass flow of driving fluid from the collector (kg/s), respectively.

Ejector

Exergy loss

$$I_j = T_{ref} \cdot \left[(m_e + m_g) \cdot S_6 - m_g \cdot S_3 - m_e \cdot S_9\right] \tag{22}$$

where I_j, m_e are irreversibility in the ejector (kW), mass flow of the entrained refrigerant from the evaporator (kg/s), respectively.



(a) Using R290 for Tehran

(b) Using R290 for Bandar Abbas

(c) Using R600a for Bandar Abbas

(d) Using R600a for Tehran

Fig. 5. Total heat transfer coefficient of the generator (kW/m^2.k).

Condenser

Exergy loss

$$I_c = T_{ref} \cdot \left[(m_e + m_g) \cdot (S_7 - S_6) + \frac{Q_c}{T_{ref}} \right] \tag{23}$$

where I_c is irreversibility in the condenser (kW).

Pump

Exergy loss

$$I_p = W_p + m_g \cdot \left[(h_1 - h_7) - T_{ref} \cdot (S_1 - S_7) \right] \tag{24}$$

where I_p is irreversibility in the pump (kW).

Evaporator

Exergy delivered

$$E_e = Q_e \cdot (1 - \frac{T_{ref}}{T_{room}}) \tag{25}$$

Exergy loss

$$I_e = T_{ref}(m_e \cdot (S_9 - S_8) - \frac{Q_e}{T_{room}}) \tag{26}$$

where I_e is irreversibility in the evaporator (kW).

Expansion device

Exergy loss

$$I_{exp} = m_e \cdot \left[T_{ref} \cdot (S_8 - S_7) \right] \tag{27}$$

where I_{exp} is irreversibility in the expansion device (kW).

Total irreversibility

$$I_{total} = I_{coll} + I_g + I_c + I_p + I_{exp} + I_e \tag{28}$$

6. Results and discussion

6.1. Exergy simulation and analysis of the SECSs

Exergy simulation assessments was developed in EES on irreversibility and hence destruction of the SECSs in two offices in Tehran and Bandar Abbas when loaded with R600a and R290 refrigerants. The detailed analysis was done in daily and hourly monitoring during June, July, Aug and Sep 2019, the results of which are tabulated in Table 4. The SECS in both Teran and Bandar Abbas offices suffered the most destruction in their solar collector components which was between 38.4 kW (Tehran SECS) to 39.09 kW (Bandar Abbas SECS). The ejector components in both SECSs was the next to suffer destruction with 3.449–1.257 kW (Tehran) when using R290 and R600a, respectively, and 3.43 kW (Bandar Abbas) when using both refrigerants. Total components destruction for the two systems was between 40.94–44.49 kW and the destruction was higher when the system charged with R290 refrigerant.

6.2. Ejector cooling cycle (ECC)

The ECC of the solar systems consist of generator, condenser, evaporator and generator. The standard TRNSYS library does not include the ejector and a mathematical model of this component is developed in the EES file. The ejector in 3.5 kW solar systems was CAM type and the properties of R290 and R600a as two hydrocarbon refrigerants were studied using EES. The hosted EES program for modeling of ejector cooling cycles has two separate parts, the first one models the cooling cycle and the ejector and the second assists modeling the generator heat exchanger. The overall performance of the EES depends on the operational design parameters, such as generator temperature and pressure (T_g and P_g), evaporator temperature (T_e), condenser temperature (T_c), the characteristics of refrigerants, cooling load of SECSs (3.5 kW) and the collector efficiency are fully discussed in coming sections.

6.3. Generator of the system

This component of the unit has a crucial application in the cooling system. The total heat transfer coefficient of this component was different for each of the refrigerants. Also, during the optimum operation of the solar cooling units, the total heat transfer coefficient of the generator varied continuously, since the temperature of inlet hot water to the

Table 4. Exergy destruction (kW)–Exergy efficiency (%).

	City			
	Tehran		Bandar Abbas	
Component	R290	R600a	R290	R600a
Ejector	3.449–55.34	1.257–69.27	3.43–55.88	1.248–69.81
Condenser	1.458–69.46	0.3057–89.21	1.115–67.25	0.07–97
Solar collector	38.4–6.8	38.4–6.8	39.09–5.26	39.05–5
Throttling valve	0.0344–99.27	0.02661–99.74	0.0352–99.27	0.027–99.74
Generator	0.62–48.58	0.77–31.53	0.5–54.31	0.6463–33.23
Pump	0.01699–97.88	0.01897–96.21	0.1–86.65	0.085–82.97
Evaporator	0.1628–88.64	0.1627–75.12	0.2236–85.33	0.22–69.4
Total	44.14	40.95	44.49	41.35

Table 5. U factor variation of refrigerants for the two cities.

	Refrigerant			
	R290		R600a	
City	Min U ($kW/m^2.k$)	Max U ($kW/m^2.k$)	Min U ($kW/m^2.k$)	Max U ($kW/m^2.k$)
Tehran	1.82	2.41	1.6	2.025
Bandar Abbas	1.8	2.34	1.58	1.98

Table 6. Thermal energy consumption of the auxiliary heater by the SECS charged with R600a and R290, kW/h (and %).

Refrigerants	June 11	July 17	Aug 16	Sep 15
Tehran				
R600a	13.6 (10.86)	12.2 (9.74)	15.8 (12.62)	13.6 (11.82)
R290	12.35 (10.03)	10.04 (8.44)	15 (12.18)	13.2 (10.72)
Bandar Abbas				
R600a	18.75 (14.97)	17.5 (13.97)	20 (15.97)	21.5 (17.17)
R290	18.4 (14.94)	16.85 (13.68)	19 (15.43)	21.08 (17.12)

Table 7. Comparison of the parameters of the working fluids on COP.

Refrigerant	COP	Primary mass flow (kg/s)	Secondary mass flow (kg/s)
R290	0.2844	0.03701	0.01177
R600a	0.2797	0.03353	0.01236

generator did not stay constant. The fluctuation of thermal coefficients of the generator for R600a and R290 for Tehran and Bandar Abbas are shown in Fig. 5. As depicted, the total heat transfer coefficient when using R290 was higher compared to R600a for both cities. The maximum value for the coefficient was 2.4 $kW/m^2.K$ for using R290 in the Tehran SECS. The variation range of U-value is listed in Table 5. The data listed in this table illustrate the effects of the two working refrigerants on the U-value modelled for the two cities. This factor was in its minimum value when using R600a in Bandar Abbas and when the system was charged with R290 in Tehran, the U-value approached its maximum.

6.4. Monthly and hourly simulation of SECSs

It has been illustrated that the 55 m^2 of the flat plate solar collector and 1.2 m^3 of storage tank at 0.825 kg/s water flow rate in the collector are necessary requirements for running the targeted 3.5 kW ejector cooling cycle in the model. Therefore, simulation was carried out to find out the overall performance of the solar systems within four months of June, July, August and September of 2019 when the solar radiation usually intensifies in both sites.

The simulation was found that solar radiation in days 11, 17, 16 and 15, respectively, in the four months was higher compared to other days and reached its maximum of 681 W/m^2 and 568 W/m^2 in Tehran and Bandar Abbas on July 17, respectively. The auxiliary heater thermal energy input was measured from 8:00 am to 6:00 pm in those days, results of which are tabulated in Table 6 in both kW/hr and percentage. The extra thermal inputs from the auxiliary heater to the Bandar Abbas cooling system compensate the shortage of the solar thermal in this region in comparison with the Tehran system in the named times were 5.15, 5.3, 4.2 and 7.1 kW/h and 6.05, 6.81, 4, and 6.88 kW/h when the operating systems were charged with R600a and R290 refrigerants, respectively. The lowest value of auxiliary heater input energy to the SECS in the hot-humid site when was charged with R290 was 16.85 kW/hr and was 17.5 kW/hr when was charged with R600a refrigerant and compared to the corresponding value in the semi-arid site, the heater input thermal energy to SECS in this region are 6.45, 5.3 kW/hr higher than the other SECS (Table 6). It means the SECS in the hot-humid site receives much less solar energy from solar than the other site and the shortage of such energy should be compensated from the auxiliary heater. The water drops in the high humid air in this region prevent effective transferring of solar radiation to the collectors in this site. The constant temperature of 85° of the generator, T_g, is maintained in the ejector cycle for R290 and R600a to be completely vaporize and therefore, the heater in the cycle compensate the shortage of the solar thermal energy in each study site. The simulation results on performance of the refrigerants listed in Table 6 indicate that R290 was more efficient refrigerant in both SECS in absorbing solar radiation than R600a during the study time. The data also indicate that maximum efficiency of charging the system with R290 was on July 17 when the system in Tehran site absorbed only 8.44 percent of its thermal energy from the heater.

The system provided higher overall COP, U-value of the generator and receives most of its necessary thermal energy directly from solar radiation than the input thermal energy from the auxiliary heater in the semi-arid region. By taking account all the necessary factors found so far, the simulation was carried out to evaluate the system overall performance and found that the overall COP of the system in the semi-arid region when charged with R290 refrigerant reached to 0.2844 higher than when R600a = (0.2797) used in the same operating conditions (Table 7). As listed in Table 7, the primary mass flow is the flow of the refrigerant from the generator into the ejector and the secondary flow is when the refrigerant enters the ejector from the evaporator. Fig. 6 de-

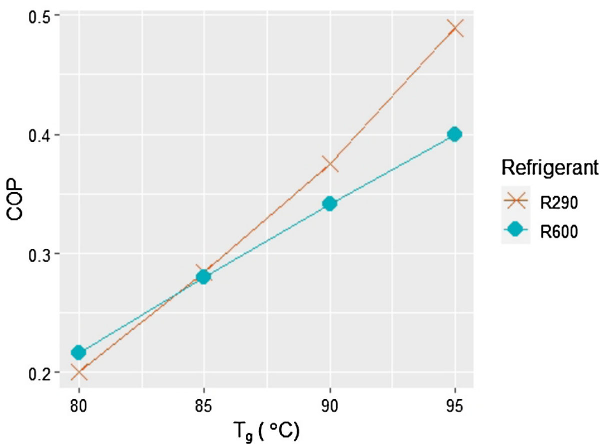


Fig. 6. Variation of COP of the study SECS with the generator temperature.

picts variation of COP of the SECS when charged by either refrigerant with different generator temperature. As it is seen in this figure the two curves of the refrigerants contact each other at 85 °C which is the temperature of T_g in the current study. The ability of R290 to stand higher generator temperatures is pronounced in the figure. The high thermal energy compensation of the heater into the southern cooling system was because the water droplets in the high humid air in the region prevent effective transferring of solar radiation to the collector system in this site. For obtaining reasonable performance from SECS in this region it is necessary to replace the refrigerants to those which can be vaporized faster in the ejector cycle when receives less solar thermal energy.

As discussed in previous chapter, the overall COP of SECS is usually around 0.3 and therefore, both of the overall COP obtained for the system operating in the semi-arid region are in the range of those reported in the literatures. It is noticed that R290 refrigerant is more suitable than R600a when was used in the SECS in the Tehran site. Therefore, application of the SECS using R290 or R600a refrigerants is feasible as a solar cooling system in the semi-arid region of Iran and moreover, R290 is an efficient choice as the more effective refrigerant in such system. The application of SECS in the hot-humid region in Iran, using R290 and R600a refrigerants is not feasible, because it receives considerable amount of its thermal energy from the auxiliary heater of the system, and therefore, the simulation analysis to find its performance was not carried out.

7. Conclusions

Dynamic simulation of 3.5 kW of the cooling systems comprise with R290 and R600a hydrocarbons as the working fluids are carried out using TRNYSIS-EES co-simulator in the summer of 2019. The systems are designed to provide solar air-conditioning in two housing spaces with the same dimensions in semi-arid and hot-humid regions of Iran. To evaluate the effect and compare the efficiency of the two refrigerants on overall COP of the cooling systems, solar fraction and energy consumption of the auxiliary heater of the solar systems are assessed. It is concluded that R290 refrigerant is more efficient working fluid when compared to R600a in the semi-arid site in Tehran than in Bandar Abbas. The solar system when is charged with R290 provides higher overall COP and heat transfer coefficient of the generator (U) and receives most of its thermal energy directly from solar radiation than the auxiliary heater in the semi-arid region. On the contrary, the system in Bandar Abbas obtains its necessary thermal energy mostly from the axillary heater since the high humidity of the air prevents complete solar radiation to reach directly to the solar collector array of SECS. It is concluded that the exergy efficiency for the throttling valve, solar collectors, and the ejector of the SECS in Tehran when using R600a are 99.74, 6.8 and 69.27 percentages, respectively and when using R290 for the same system, the percentages are 99.27, 6.8 and 55.34, respectively. It is also found that the energy destruction in Tehran SECS, when using R600a and R290 are 40.95 and 44.14 percentages, respectively.

This work offers valuable information for application of similar cooling system in many houses in Tehran which mostly use the old fashion evaporative cooling system (water cooler) and also it can be designed for using in other semi-arid regions having similar solar characteristics and geographical conditions to Tehran.

Declarations

Author contribution statement

H. Jadidi: Analyzed and interpreted the data; Wrote the paper.

M. Keyanpour-Rad: Conceived and designed the experiments; Analyzed and interpreted the data; Contributed reagents, materials, analysis tools or data; Wrote the paper.

H. Haghgou: Conceived and designed the experiments; Analyzed and interpreted the data; Contributed reagents, materials, analysis tools or data.

B. Chodani: Performed the experiments; Analyzed and interpreted the data.

S. Kianpour rad: Analyzed and interpreted the data.

M. Hasheminejad: Conceived and designed the experiments.

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The authors declare no conflict of interest.

Additional information

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