Exergoeconomic optimization and sensitivity analysis of a commercial parabolic trough collector for the climate of Tehran, Iran

Hamed Hoseinzadeh1 | Alibakhsh Kasaeian2 | Mohammad Behshad Shafii3

1Department of Energy Engineering, Science and Research Branch, Islamic Azad University, Tehran, Iran
2Faculty of New Sciences and Technologies, University of Tehran, Tehran, Iran
3Faculty of Mechanical Engineering, Sharif University of Technology, Tehran, Iran

Correspondence
Alibakhsh Kasaeian, Faculty of New Sciences and Technologies, University of Tehran, Tehran, Iran.
Email: akasa@ut.ac.ir

Abstract
The exergoeconomic analysis is a powerful tool to study an energy system and provide rational decision-making for it. This paper is aimed at exergoeconomic optimization and sensitivity analysis evaluation of the results of a commercial parabolic trough collector (PTC). First, an analysis of optical, thermal, exergy, and economic equations of the PTC system is presented. Then, the objective function and variables are optimized through MATLAB software using the hybrid algorithm code. The objective function as the exergy loss rate is a combination of exergy loss and system cost. The evaluation of the results has been carried out considering the radiation intensity in different months of the year for Tehran, Iran. The commercial parabolic trough solar collector has been selected as the case study. Sensitivity analysis of exergy loss rate and exergy efficiency was conducted for important variables, namely collector length, the glass cover diameter, the outer diameter of absorber tube, the inner diameter of absorber, the fluid inlet temperature, the receiver aperture width, and fluid flow rate. The obtained results show that the exergy loss rate is 0.1905 kW/$, and the exergy efficiency is equal to 15.2% for the highest direct solar radiation of 674 W/m².

KEYWORDS
exergoeconomic, hybrid algorithm, parabolic trough collector, sensitivity analysis

1 | INTRODUCTION

Compared to numerous technologies employed for solar thermal energy, parabolic trough solar collectors (PTCs) are widely used in small and large scales. Concentrated solar power (CSP) is one of the leading systems among renewable energies, and PTC is one of the most widely used CSP systems, hence, improving its performance and efficiency is necessary for this technology. Exergy analysis is a powerful tool to evaluate the systems related to different types of energy. Exergy analysis detects the losses of an energy system where its combination with economic concepts provides more precise results from the technical-economic analysis of an energy system. The optimization of this technical-economic analysis can improve the decision-making process within the design. Many important studies have been carried out on the optical and thermal performance of the PTC system. However, further research is still needed to study this field thoroughly. In recent years, many studies and research works have been conducted on parabolic trough solar collectors using various methods to investigate their optical and thermal applications. Here, some of them are mentioned.

In 2012, optimizing the size of a solar heating system was investigated based on the genetic algorithm for the
aquatic system, by Atia et al. They used genetic algorithms and economic equations to optimize the size of a collector. Kalogirou examined a detailed thermal model of a PTC receiver. He analyzed the thermal loss conditions of the receiver. In 2013, Caliskan et al performed an analysis for the exergoeconomic and environmental effects of renewable energy based on hydrogen production. They examined seven refrigerants for the ORC in their study. Heat losses from parabolic trough solar collectors investigated by the thermal and thermodynamic analysis of a PTC were performed in different modes of concentration ratio and rim angle, by Mwesige and kalogirou. The exergy analysis of the PTC receiver was proposed by Padilla et al. Their main parameters were the inlet temperature, mass flow rate, wind speed, and the intensity of the sun’s radiation. Hernández-Román et al conducted the exergy and thermo-economic analyses of an air heater system in the PTC process. In 2015, the technical-economic evaluation of a Kalina cycle for the PTC was proposed by Ashouri et al. They examined the performance of the system, including a PTC, a storage tank, an auxiliary heater, and Kalina cycle, throughout the year in thermodynamic and economic terms. The thermal analysis and working fluid selection for the solar Rankine cycle were performed by Desai et al. They proposed their research based on the thermo-economic analysis and, also, a comparative diagram of the working fluid for the energy production cycle. The thermal analysis of a PTC, based on a nanofluid, was carried out using a hybrid algorithm by Zadeh et al. They used a combination of two GA and SQP algorithms for optimization. An analysis was carried out on exergetic and economic performance of photovoltaic, PTC, and wind power, by Sadati et al. In 2016, thermodynamic analysis and optimization of the combined cycle of Rankine power and nanofluid-based PTC were presented by Toghyani et al. Abid et al accomplished the energy and exergy analysis to compare the parabolic dish and the parabolic trough collector and the using nanofluid and molten salt. The results obtained show that the outlet temperature of the PD solar collector is higher in comparison to PT solar collector under identical operating conditions. They used four nano-materials for simulation, as well as a genetic algorithm for optimization. In the same year, the exergy analysis of solar thermal systems and a better understanding of their stability were provided by Kalogirou et al. Wang et al evaluated the heat flux distribution for the performance of PTC using Al2O3/synthetic oil nanofluid and finite element method (FEM) simulation. The exact investigation of working fluid for PTC was carried out by Bellos et al. They have examined fluids such as pressurized water, therminol VP-1 oil, molten salt, liquid sodium, air, carbon dioxide and helium. In 2017, Yüksel made the thermodynamic assessment of the organic Rankine cycle, which was integrated with the PTC system, for hydrogen generation. Thermo-economic analysis of the biomass Rankine cycle of a compression absorption system for LFR was conducted by Patel et al. The thermal performance of a PTC was evaluated by Conrado et al. They considered the thermal parameters: environmental conditions, temperature, thermal flux, and economic costs. The energy, exergy, and economic analysis and integration of a PTC with the Kalina cycle for high-temperature applications were investigated by Zare et al. They achieved a total exergy efficiency of 14%. Qiu et al have examined the thermal performance of PTC using the supercritical fluid CO2 as HTF. They have used the combination method of Monte Carlo ray tracing and the finite volume method (FVM). A study entitled “Exact Exergy Analysis from a PTC” was presented by Bellos et al. They used the LS-2 PTC in their study. Ghasemi et al numerically investigated the three-dimensional turbulent flow for heat transfer in a tube with porous rings. They performed numerical simulations with computational fluid dynamics (CFD). In 2018, the thermal and economic optimization of a solar heater with central heating and a series of flat plate collectors and PTC was conducted by Tian et al. The results demonstrated that the PTC was economically applicable in large sizes. The optical and thermal analysis of a PTC was carried out for thermal energy generation in different climatic conditions of Iran, by Marefati et al. Sadaghiyani et al have carried out the energy and exergy analysis of a PTC. They considered the LS-2 PTC in their research. The energy and exergy analysis of a PTC power plant has been performed using carbon dioxide power cycle by AlZahrani et al. Carballo et al have studied the energetic and exergetic analysis of a dynamic PTC model. Their model has been described and validated based on detailed physical principles. Also, a study has been carried out by Allouhi on the energy and exergy analysis of a PTC with nanofluid for the application in the medium and high temperatures. Fang et al have examined the thermodynamic evaluation of the solar thermochemical process using double-axis tracking PTC. In this system, the process of liquid methanol preheated and evaporated using PTC for a CHP system. An improved model was proposed to predict the performance of the solar parabolic collector performance by Agagna et al. Their proposed model performed better than experimental results. Ray et al developed a new method for analyzing the performance of interfacial space between the adsorbent tube and the glass cover for PTC using CFD. They have examined the effects of selective coating, vacuum, and semitransparent glass cover. Akbari Vakilabadi et al have carried out exergy analysis of a hybrid solar-fossil fuel power plant. They conducted their study of energy and exergy analysis and mathematical modeling for a 35 MW solar-fossil fuel power plant on a special day in the Mojave Desert, California.

According to research by other researchers on the optical and thermal performance of PTC systems, it has been observed that these investigations have been carried out separately for solar collector components, and have not been discussed integrally for all collector components. Some methods have been
used in previous research works, such as the FEM simulation, FVM, CFD, and so on. In previous research works, collectors such as LS 2, LS 3, and ET 150, have been used for optical and thermal analysis. The use of solar parabolic collectors has received attention from researchers due to Iran’s climatic conditions. Thus, several aspects of novelty can be pointed out in light of past research in the study. Firstly, the exergy test for all collector components in general with respect to economic parameters for optimization, that is, exergoeconomic optimization for the solar parabolic collector, has not been performed so far. Secondly, the method used in this study is a computational algorithm (hybrid algorithm) coded in Matlab environment that has not been used in previous research works to optimize parabolic solar collectors. Thirdly, the PT1‐IST type collector is used as a commercial collector for using in this study, and this type of collector has been less commonly used in previous research works. Fourth, the climatic conditions of Tehran, Iran, which have received less attention from researchers due to the suitability of conditions for the use of parabolic collectors. The present study is about optimizing a commercial collector for use in a region with different climatic conditions. This type of commercial collector (one of the most widely used small-scale and ground type), which is of interest to researchers, has been used for the city of Tehran for using with appropriate climatic conditions in this study. Optical and thermal analysis of this type of collector for future research is investigated in this study.

2 | SYSTEM DESCRIPTION

The solar parabolic collector system of commercial type has been employed for the analysis in the present study. The purpose of this study was to investigate the exergoeconomic and sensitivity analysis of a solar collector. For this reason, a commercial collector PT1‐IST (Industrial Solar Technology Corporation product) was selected for analysis in this study. Due to the choice of this type of collector, full information about this collector is provided in the report of the test results for the solar parabolic collector published by the National Laboratory of Sandia. Another reason for selecting this type of collector is to validate the results of this study. This type of collector is used more than other collectors due to its small size and ground type. Among other things, this type of collector has attracted the researchers’ interest in conducting past research and studies in the field of optical and thermal analyses.

The scheme of the parabolic trough collector system is demonstrated in Figure 1, and the main specifications of the system are presented in Table 1. As shown in the figure, the rays, after colliding to the reflector, are focused along the focal line of the parabola. The reflected ray ($\rho$) hits the absorber tube after passing through the glass cover. The radiation energy is absorbed by the heat transfer fluid (HTF) inside the absorber tube. The thermal energy of HTF is stored in a storage tank. The absorbed heat is used for various applications.

In this study, exergoeconomic analyses were performed based on the specifications of the solar collector, as provided in Table 1. This type of collector, which is a commercial module, has an aperture width of 2.3 m, a collector length of 6.1 m, an absorber tube with a diameter of 0.051 m, a rim angle of 72°, and a concentration ratio of 14.36. Also besides, the reflector is coated with acrylic aluminum or enhanced polished aluminum. Also, the absorber tube is a black nickel-coated tube.

3 | METHODOLOGY AND FORMULATIONS ANALYSIS

3.1 | Methodology

In this study, the technical-economic equations for the commercial parabolic trough solar collector have been developed.
The objective function is based on the exergy and economic model. The variables of this research are in two forms, with numerical limitations, and without numerical limitations. A hybrid algorithm (GA‐PSO) has been used to optimize and sensitivity analysis of the variables. This method has been performed by coding in MATLAB. The purpose of this study is to optimize the variables and the exergoeconomic objective function for a PTC system as a case study in the city of Tehran.

3.2 | Optical and radiation analysis

One of the most important parameters for analyzing a solar collector is the optical efficiency. The optical efficiency equation is written as:

\[ \eta_{opt} = \rho \tau a \gamma \left[ (1 - A_f \tan (\theta_i)) \cos (\theta_i) \right] \]  

The energy intercept factor (\( \gamma \)) is presented as followings:

\[ \gamma = \frac{1 + \cos (\varphi_r)}{2 \sin (\varphi_r)} \left\{ \frac{\sin (\varphi_r) \left[ 1 + \cos (\varphi_r) \right] \left[ 1 - 2d^* \sin (\varphi_r) \right] - \pi \beta^* \left[ 1 + \cos (\varphi_r) \right]}{\sqrt{2\pi \sigma^*}} \right\} \right\} \right\} d\varphi \]

(2)

The amount of the radiation flux, reaching the receiver tube, is defined as below:

\[ Q_S = \left[ I_b \left( \eta_{opt} + I_b (\tau a) \left( \frac{D_o}{W_a - D_o} \right) \right) \right] \]  

(3)

\[ Q_{S-r} = Q_S \left( W_a \cdot L_c \right) \]  

(4)

Also, \( Q_S \) is the radiation flux in W/m², and \( Q_{S-r} \) is the radiation on the receiver in W.

3.3 | Thermal analysis

There are essential factors for a parabolic trough solar collector, which could be changed to achieve an optimal mode. These factors include the receiver length, the inner diameter of the tube and the outer diameter of the absorber tube, the glass cover diameter, the fluid flow rate, the receiver temperature, the inlet fluid temperature, and the optical efficiency. Figure 2 shows a schematic of the structure and thermal operations of the PTC system. The radiation energy, entering the receiver, is converted into thermal energy. The optical efficiency is defined as the ratio of the energy absorbed by the receiver to the amount of the incoming energy to the collector. Moreover, the thermal efficiency refers to the ratio of the energy, absorbed by the fluid, to the amount of the energy absorbed by the receiver. The parameters affecting the thermal efficiency are the collector efficiency factor, loss factor, and the heat recovery factors of the collector. The thermal loss of the receiver is calculated in terms of the loss factor based on the receiver area.

The calculation of thermal losses requires structural information of the collector, whose equations are presented as below.

Due to the heat around the glass tube, the heat transfer coefficient of the radiation from the glass tube to the environment is obtained from the following equation:

\[ h_{r,c-a} = \varepsilon_g \cdot \sigma \cdot (T_g + T_{amb}) \cdot (T_g^2 + T_{amb}^2) \]

(5)
The radiation heat transfer coefficient between the receiver tube and the glass cover is obtained from the following equation:

$$h_{r,c} = \frac{\sigma \cdot (T_r + T_g) \cdot (T_r^2 + T_g^2)}{1 + \frac{1}{h_w} \cdot \left(\frac{1}{r_s} - 1\right)}$$  \hspace{1cm} (6)$$

Finally, the overall heat loss coefficient is obtained from the following equation:

$$U_L = \left[\frac{A_r}{h_w + h_{r,c}} + \frac{1}{h_{r,c}}\right]^{-1}$$  \hspace{1cm} (7)$$

Another parameter which is commonly used for parabolic trough collector analysis is the collector efficiency factor. The collector efficiency factor could be calculated from the following equation:

$$F_{eff,c} = \frac{1}{U_L} + \frac{D_i}{h_j \cdot D_i} + \left[\frac{D_i}{2} \ln \frac{D_o}{D_i}\right]$$  \hspace{1cm} (9)$$

The heat recovery factor is a dimensionless variable, known as the dimensionless rate of collector capacity, which is displayed as followings:

$$h_f = \frac{Nu \cdot k_f}{D_i}$$  \hspace{1cm} (10)$$

The tubular absorber absorbs a lower quantity of energy ($Q_{abs}$), due to the optical losses:

$$Q_{abs} = \eta_{opt} \cdot A_a$$  \hspace{1cm} (12)$$

Some of the energy, which exists as a useful energy, can be obtained by using the concept of the absorbed radiation:

$$Q_a = Q_{abs} - \left(A_r \cdot U_L \cdot (T_r - T_{amb})\right)$$  \hspace{1cm} (13)$$

where $T_r$ is the receiver temperature, which is calculated by following equation[

$$T_r = T_{in} + \frac{Q_a}{A_r \cdot U_L \cdot F_r} \cdot (1 - F_r)$$  \hspace{1cm} (14)$$

According to the fluid inlet temperature as well as the amount of the heat absorbed by the receiver, the fluid outlet temperature is achieved as:

$$T_{out} = T_{in} + \frac{Q_a}{m \cdot C_p}$$  \hspace{1cm} (15)$$

Also, the thermal efficiency of the collector is defined as followings:

$$\eta_t = F_r \cdot \left[\eta_{opt} - U_L \cdot \left(\frac{T_{in} - T_{amb}}{h_f \cdot C_r}\right)\right]$$  \hspace{1cm} (16)$$

### 3.4 | Exergy analysis

Exergy is defined as the maximum amount of the useful work, which could be obtained as the system is brought into equilibrium with the environment. Unlike energy, the exergy is not stored, but it is lost due to irreversibility. The loss of exergy in a process is relatively associated with the degree of the produced irregularity, due to irreversibility. In fact, exergy is actually the maximum useful work obtained by combining the system and environment. In other words, it is a portion of the thermal energy, which could be converted to effective work under the ideal conditions. In this section, the exergy loss equations are used to determine the exergy efficiency. In Figure 3, all the exergy losses are specified. These losses are generally optical and thermal, which are divided into different modes, as the exergy loss due to the optical losses, the exergy loss from the difference in the temperature of the sun and the receiver, the exergy loss due to the loss of fluid pressure inside the receiver, the exergy loss due to the thermal losses of the receiver, and the exergy loss due to the difference in the receiver's temperature and the temperature of the fluid inside the receiver. Later on, the above-mentioned items will be examined.

The incoming radiation exergy into the collector, that is, the amount of the exergy emitting from the sun's radiation to the Earth's surface, is obtained by the following equation:

$$\dot{E}_{in,r} = h_i \cdot A_a \cdot \eta_p$$  \hspace{1cm} (17)$$

Based on the theory of Petela, the maximum conversion efficiency as the ratio of the maximum work output to the energy is obtained as:

$$\eta_p = 1 - \frac{4T_{amb}^4}{3T_s^4} + \frac{1}{3} \left[\frac{T_{amb}}{T_s}\right]^4$$  \hspace{1cm} (18)$$

The useful exergy output could be calculated according to the following equation:

$$\dot{E}_u = Q_a \cdot m \cdot T_{amb} \cdot \ln \left(\frac{T_{out}}{T_{in}}\right) - m \cdot T_{amb} \cdot \left(\frac{\Delta P}{h_f \cdot T_f}\right)$$  \hspace{1cm} (19)$$

The mean fluid temperature ($T_f$) could be calculated as the mean value of the inlet ($\bar{T}_{in}$) and the outlet ($\bar{T}_{out}$) fluid temperatures:
The optical exergy loss, or the exergy loss of the solar radiation to the Earth’s surface, may be obtained by:

\[ \dot{E}_{\text{opt}} = \eta_{\text{opt}} \cdot \dot{E}_{\text{in,r}} \]  

Due to the existing of the heat between the receiver glass tube and its surroundings, the loss of the thermal exergy between the receiver and the environment is obtained by the following equation:

\[ \dot{E}_{\text{th}} = \frac{U_L \cdot A_r \cdot (T_r - T_{\text{amb}})}{T_r} \]  

Finally, the total exergy losses from the optical and thermal exergy losses could be obtained as:

\[ \dot{E}_{\text{loss}} = \dot{E}_{\text{opt}} + \dot{E}_{\text{th}} \]  

The destroyed exergy, due to the absorption of the radiation heat by the receiver, is obtained as below:

\[ \dot{E}_{\text{des,s-r}} = \eta_{\text{opt}} \cdot \dot{E}_{\text{in,r}} - Q_{\text{abs}} \cdot \left(1 - \frac{T_{\text{amb}}}{T_r}\right) \]  

The destroyed exergy due to the difference between the temperature of the receiver and its fluid is obtained from the following equation:

\[ \dot{E}_{\text{des,s-f}} = Q_u \cdot \left(1 - \frac{T_{\text{amb}}}{T_r}\right) - E_u \]  

Finally, the total exergy, destroyed by the pressure drop in the working fluid, the absorption of the radiation heat by the receiver, and the temperature difference between the receiver and its fluid are obtained by the following equation:

\[ \dot{E}_{\text{des}} = \dot{E}_{\text{des,s-r}} + \dot{E}_{\text{des,s-f}} \]  

The exergy efficiency is defined as the ratio of the useful exergy delivered to the exergy incident on the collector aperture and is given as:

\[ \eta_{\text{ex}} = \frac{\dot{E}_{\text{u}}}{\dot{E}_{\text{in,r}}} \]  

3.5 | Economic analysis

In the economic modeling, the economic costs including the initial investment, maintenance, operating, replacement, and equipment costs have been intended. The cost of the solar thermal systems could be obtained by the following equations:

\[ CS = C_a A_a + C_{\text{H&H}} A_a + C_{\text{st}} Q_{\text{st}} + C_{\text{M&O}} \]  

where \( CS \) is the cost of the solar thermal system, \( C_a \) is the cost dependent on the collector’s area (\( A_a \)), \( C_{\text{H&H}} \) is the cost of the HTF and hydraulic circuit dependent on the collector’s area (\( A_a \)), \( C_{\text{st}} \) is the cost dependent on the energy stored inside the tank (\( Q_{\text{st}} \)), and \( C_{\text{M&O}} \) is the cost of maintenance and operations (equivalent to 2% of the initial investment cost).

The capital recovery factor (CRF) is a ratio used to calculate the present value of an annuity (a series of equal annual cash flows). The equation for the CRF is:

\[ \text{CRF} = \frac{r \cdot (1+r)^n}{(1+r)^n - 1} \]  

where \( r \) is the interest rate equivalent to 15%, and \( n \) is system’s performance years equivalent to 20 years. The total annual system cost is defined as:

\[ \text{TCA} = \text{CRF} \left( CS + C_{\text{AUX}} \right) + C_{\text{AUX}} Q_{\text{AUX}} \]  

In the above equation, \( C_{\text{AUX}} \) is the fuel cost, and due to the lack of fuel consumption, this value is zero.

3.6 | Objective function

In this paper, the objective function is based on the exergy loss rate. The total exergy loss contains the total loss of exergy and the destroyed exergy. The loss of exergy is affected by the economic costs of the thermal system. Therefore, the objective function is a combination of these two functions of exergy loss and economic costs. The optimizer program will obtain the value of the objective function according to the constraints on the problem, and the range of the decision variables. The loss of exergy, which to be minimized, is obtained as:

\[ \dot{L}_{\text{ex}} = \sum_{i=1}^{n} \dot{E}_{\text{loss}} + \dot{E}_{\text{des}} \]
where $\dot{L}_{ex}$ is the total exergy loss in kW. Besides, the economic model of the solar thermal system, which to be minimized, is shown by the following equation:

$$
TCA = \sum_{i=1}^{n} \left( \frac{CRF \left( C_s + C_{AUX} \right) + C_{AUX} Q_{AUX}}{CS_i} \right) + \frac{\dot{C}_{AUX} Q_{AUX}}{CRF} \tag{32}
$$

where TCA is the total annual system cost in $, and the objective function model is formed considering the combination of the two above functions:

$$
\text{Minimize } R_{ex} = \frac{\dot{L}_{ex}}{TCA} \tag{33}
$$

In the above equation, $R_{ex}$ is the rate of exergy loss in kW/$.\textsuperscript{3}$

### 3.7 Hybrid algorithm

The hybrid algorithm is raised as one of the problem-solution among the conventional methods in artificial intelligence.\textsuperscript{39} The hybrid algorithm, as a computational optimization algorithm, effectively searches different areas of the solution space considering a set of response space points in each computational iteration. In fact, one can move faster in the space of a problem state to find possible solutions through this method. That is, we can achieve the desired solutions by not extending all the scenarios. In this study, the combination of two algorithms (GA-PSO) has been used for the exergoeconomic optimization. In fact, the genetic algorithm (GA) can be improved with the help of the particle swarm optimization (PSO) algorithm. Figure 4 shows the flowchart of the hybrid algorithm. The genetic algorithms act by using a specific basis, similar to the genetic structures and chromosomes, among a population of individuals. In summary, the genetic algorithm consists of the following operators: encoding, evaluation, combination, mutation, and decoding. There are a number of things in the particle swarm optimization algorithm, known as particles, which are distributed in the search function space. Each particle calculates the value of the objective function in its own location. In brief, the PSO algorithm consists of the following operators: initializing, estimating, and updating.

![FIGURE 4 Schematic of the flowchart of calculations with hybrid algorithm (GA-PSO)](image-url)
4 | RESULTS AND DISCUSSION

4.1 | Case study

In this study, a commercialized PTC system (the type of PT1-IST) with the weather conditions of Tehran has been presented. One of the most important parameters in determining the values of variables is the intensity of the solar radiation in that region. Figure 5 shows the intensity of solar radiation on a daily average per month, in terms of W/m² in the city of Tehran. In this figure, the highest direct radiation intensity is 674 W/m². Figure 6 shows the dry-bulb temperature in different months of the year in the city of Tehran, in terms of K.

The parabolic solar collector with the initial data and assumptions is illustrated in Table 2. Also, the working fluid is assumed as the Therminol 66 heat oil. The properties of this heating oil are represented in the ambient conditions, with the heat capacity of $C_p = 1550 J/kg K$, the density of $\rho = 1011 kg/m^3$, and the maximum working temperature of 345°C.

To calculate the total capital cost of the PTC system, the cost of each component should be calculated separately. The capital cost of these components is calculated based on data in Table 3.

4.2 | Design variables

The design variables for exergoeconomic optimization and sensitivity analysis of this research are divided into two forms: with numerical range and without numerical range. The variables with numerical ranges include the fluid flow rate $\dot{m}$ ($\dot{m}_f$), the input fluid temperature of the receiver $T_{in}$, the outer diameter of the absorber $D_o$, the inner diameter of the absorber $D_i$, the glass cover diameter of the receiver $D_g$, the length of collector $L_c$, and the collector’s aperture width $W_a$, which are defined as followings:

Variable with numerical range (down and upper limit)

\[
f_1(x): 0.1 \leq \dot{m} \left( \frac{kg}{s} \right) \leq 0.5
\]

\[
f_2(x): 300 \leq T_{in} (K) \leq 350
\]

\[
f_3(x): 0.045 \leq D_o (m) \leq 0.055
\]
The second type is the variables without a numerical range, including the concentration coefficient $C$, the intercept factor $\gamma$, the optical efficiency $\eta_{\text{opt}}$, the radiation heat to the receiver $Q_{\text{SR}}$, the receiver temperature $T_r$, the total heat loss coefficient $UL$, the heat recovery factor $Fr$, the useful energy of the fluid after the absorption of the radiation $Qu$, the fluid outlet temperature $T_{\text{out}}$, the thermal efficiency $\eta_t$, the exergy efficiency $\eta_{\text{ex}}$, the exergy loss rate $R_{\text{ex}}$, and the useful exergy output $Eu$. Therefore, there are 20 variables in this study.

### 4.3 The consequences of analysis

As already mentioned, the parabolic collector’s information is added in the code of the hybrid algorithm; the results are shown as the output of the software in Table 4, based on a maximum direct radiation intensity of 674 W/m².

Figure 7 shows the changes in the thermal flux and temperature of the receiver rim in different months of the year. In this figure, June, July, August, and September have the highest thermal flux and temperature on the receiver, as 10.97 kW and 412 K, respectively.

Figure 8 shows the changes of the useful energy and fluid outlet temperature in different months of the year. In this figure, the maximum amount of the useful energy is 8.44 kW, and the fluid outlet temperature is 427 K.

The numerical results for the exergy and exergoeconomic analysis of the solar parabolic collector are shown in Table 5. This table has been achieved based on the solar radiation intensity in different months of the year. The maximum outlet exergy outlet is 1.946 kW. According to the column of $R_{\text{ex}}$, it is evident that the amount of the exergy loss rate is increased, as the radiation intensity increases. Furthermore, according to the column $\eta_{\text{ex}}$, the amount of $\eta_{\text{ex}}$ is decreased, as the intensity of the sun’s radiation decreases. The numbers in the last row of the table represent the monthly average (over a year).

### 4.4 Validation

The results have been validated considering other studies conducted in this field. In order to confirm the accuracy of our results, the numerical results were compared with the results of two papers which had a common case study. Our numerical results along with the results of previous studies are shown in Table 6.

The exergy efficiency was one of the variables, which was used for the validation. The exergy efficiency was obtained as 14% in the study, conducted by Bellos et al. and Marefati.

### TABLE 2 The specifications and assumptions of the solar collector

| Parameter | Units | Value |
|-----------|-------|-------|
| $\varepsilon_r$ | – | 0.1 |
| $\varepsilon_g$ | – | 0.88 |
| $k_t$ | W/m K | 15 |
| $T_{\text{amb}}$ | K | 298 |
| $k_w$ | W/m K | 0.0276 |
| $Nu$ | – | 129.73 |
| $\sigma$ | W/m² K⁴ | 5.67 × 10⁻⁸ |
| $T_s$ | K | 5800 |
| $\Delta P$ | Pa | $10^5$ |
| $T_{g}$ | K | 350 |
| $\eta_{\text{opt}}$ | W/m² K | 330 |
| $\varphi_r$ | rad | 1.25 |
| $\varphi$ | rad | 0.785 |
| $\beta$ | rad | 0.02 |
| $\sigma_{\text{ext}}$ | mrad | 25 |
| $d_i$ | mm | 3.5 |
| $\rho$ | – | 0.9 |
| $\tau$ | – | 0.85 |
| $\alpha$ | – | 0.95 |
| $\eta_{\text{opt}}$ (%) | – | 76 |

### TABLE 3 Capital cost of solar field components

| Component | Capital cost |
|-----------|-------------|
| Solar collector | 150 $/m² |
| Heat transfer fluid and hydraulic circuit | 90 $/m² |
| Storage tank | 35 $/kWh |

### TABLE 4 The results at the optimum design point and operating conditions

| Variable | Value | Variable | Value |
|----------|-------|----------|-------|
| $r$ | 0.95 | $w_a$ (m) | 3 |
| $\eta_{\text{opt}}$ (%) | 77 | $T_r$ (K) | 412 |
| $\eta_t$ (%) | 15.2 | $T_{a}$ (K) | 350 |
| $C_r$ | 16.7 | $T_{\text{out}}$ (K) | 427 |
| $m$ (kg/s) | 0.1 | $D_i$ (m) | 0.070 |
| $K$ (w/kS) | 0.1905 | $U_s$ (m) | 0.052 |
| $L$ (m) | 7 | $L$ (m) | 0.055 |
| $Q_{\text{SR}}$ (W) | $1.097 \times 10^4$ | $Q_{s}$ (W) | $0.844 \times 10^4$ |
| $U_s$ (w/m² K) | $15.42$ | $F_r$ | 0.89 |
| $\alpha$ (%) | 62 | $Eu$ (W) | $0.1946 \times 10^4$ |

The second type is the variables without a numerical range, including the concentration coefficient $C$, the intercept factor $\gamma$, the optical efficiency $\eta_{\text{opt}}$, the radiation heat to the receiver $Q_{\text{SR}}$, the receiver temperature $T_r$, the total heat loss coefficient $UL$, the heat recovery factor $Fr$, the useful energy of the fluid after the absorption of the radiation $Qu$, the fluid outlet temperature $T_{\text{out}}$, the thermal efficiency $\eta_t$, the exergy efficiency $\eta_{\text{ex}}$, the exergy loss rate $R_{\text{ex}}$, and the useful exergy output $Eu$. Therefore, there are 20 variables in this study.

\[ f_4(x) : 0.042 \leq D_i (m) \leq 0.052 \]

\[ f_5(x) : 0.070 \leq D_g (m) \leq 0.080 \]

\[ f_6(x) : 6 \leq L_c (m) \leq 7 \]

\[ f_7(x) : 2 \leq W_a (m) \leq 3 \]
et al\textsuperscript{24} while it was obtained as 14\% in the current study with the same specifications of previous studies.

### 4.5 Sensitivity analysis

According to the sensitivity analysis of some variables with a hybrid algorithm, the results are presented as the following figures. Sensitivity analysis was carried out by the values of the variables with numerical range. Seven essential variables including the receiver length, the collector aperture width, the temperature of the inlet fluid to the receiver, the outer diameter of the receiver, the inner diameter of the receiver, the receiver glass cover diameter, and the mass flow rate were investigated in this study. Figure 9 shows the amount of exergy efficiency before and after the optimization for the different months of the year. In this figure, by optimizing the amount of exergy loss rate has been decreased.

Figure 11 shows the amount of exergy efficiency based on the down limit of variations for the different months of the year. Figure 12 shows the value of the exergy loss rate based on the down limit of variations for the different months of the year. Comparing the two Figures 11 and 12, it can be said that, by increasing the collector aperture width, the collector length, the absorber outer diameter, the absorber inside diameter, and the input fluid temperature are added to the amount of exergy efficiency, and with increasing the glass cover diameter and fluid flow rate, the exergy efficiency is reduced.

Figure 13 shows the amount of exergy efficiency based on the upper limit of variations for the different months of the year. Figure 14 shows the value of the exergy loss rate based on the upper limit of variations for the different months of the year. Comparing the two Figures 13 and 14, it can be said that, by increasing collector aperture width, the collector length, the absorber outer diameter, the absorber inside diameter, and the input fluid temperature, the value of the exergy...
loss rate decreased, and with increasing the glass cover diameter and fluid flow rate, the exergy efficiency is increased.

Table 7 shows the sensitivity analysis for the maximum direct radiation intensity of $(674 \text{ W/m}^2)$. In this table, sensitivity analysis numerically shows for the upper and down limit of the variables with numerical ranges to analyze the max and min amounts of the exergy efficiency and the exergy loss rate. Also in this table, the status of the exergy efficiency alterations and the exergy loss rate are represented from the down limit to the upper limit with a sign of increase or decrease.

Figure 15 shows the sensitivity percentage of each of the variables in the specified numerical range with respect to the changes in exergy efficiency and the exergy loss rate. In this figure, the fluid input temperature, fluid flow rate, collector opening width, absorber outer diameter, the receiver length, inner absorber diameter, and glass tube diameter have, respectively, the highest sensitivity percentage.

5 | CONCLUSIONS

In this research, an exergoeconomic optimization and sensitivity analysis of a commercial parabolic trough collector for a climate in Tehran, Iran, was conducted, using a
hybrid algorithm (GA-PSO). For the case study, the commercial PTC system of the type PT1-IST has been considered with Tehran’s weather conditions. The optimization method for the development of the optical, thermal, exergy, and economic equations, as well as the hybrid algorithm coding, have been performed in MATLAB software. Sensitivity analysis of the exergy loss rate and exergy efficiency has been carried out for some variables. The following conclusions have been made:

- The optimization results are shown for the highest direct radiation intensity of 674 (W/m²).
- The exergy loss rate and the exergy efficiency are 0.1905 kW/$ and 15.2%, respectively.
The results of the sensitivity analysis shows that as the fluid flow rate and the glass cover diameter increases, the exergy loss rate increases.

The results of the sensitivity analysis shows that as the fluid inlet temperature, the outer diameter of the absorber tube, the collector aperture width, the collector length, and the inner diameter of the absorber tube increases, the exergy loss rate decreases.

The results of the sensitivity analysis reveal that the changes in the fluid inlet temperature, the fluid flow rate, the outer diameter of the absorber tube, the collector aperture width, the collector length, and the inner diameter of the absorber tube increases, the exergy loss rate decreases.

The amount increases from down to upper limit.

The amount decreases from down to upper limit.

| Variable                  | Down limit | Upper limit | Max $\eta_{ex}$ (%) | Min $\eta_{ex}$ (%) | Status $\eta_{ex}$ | Max $R_{ex}$ (kW/$\circ$) | Min $R_{ex}$ (kW/$\circ$) | Status $R_{ex}$ |
|---------------------------|------------|-------------|----------------------|---------------------|----------------------|---------------------------|---------------------------|-------------------|
| Aperture width (m)        | 2          | 3           | 15.2                 | 14.5                | ↑                    | 0.1928                    | 0.1905                    | ↓                 |
| Collector length (m)      | 6          | 7           | 15.2                 | 14.7                | ↑                    | 0.192                     | 0.1905                    | ↓                 |
| Absorber outer diameter (m) | 0.045      | 0.055       | 15.2                 | 14.4                | ↑                    | 0.193                     | 0.1905                    | ↓                 |
| Absorber inside diameter (m) | 0.042     | 0.052       | 15.2                 | 15                  | ↑                    | 0.1915                    | 0.1905                    | ↓                 |
| Glass cover diameter (m)  | 0.070      | 0.080       | 15.2                 | 15.15               | ↓                    | 0.1911                    | 0.1905                    | ↑                 |
| Mass flow rate (kg/s)     | 0.1        | 0.5         | 15.2                 | 10.6                | ↓                    | 0.202                     | 0.1905                    | ↑                 |
| Input fluid temperature (K) | 300       | 350         | 15.2                 | 9.1                 | ↑                    | 0.206                     | 0.1905                    | ↓                 |

Note: ↑ The amount increases from down to upper limit.
↓ The amount decreases from down to upper limit.
diameter of the absorber tube, the collector aperture width, the collector length, the inner diameter of the absorber tube, and the glass cover diameter cause most changes on the values of the exergy loss rate and the exergy efficiency of the PTC.

Exergoeconomics is a powerful tool for the energy system's analysis, decision-making, development, and exploitation. Future research works are suggested to investigate the combination of different ray tracing methods to evaluate the appropriate thermal flux on the receiver.

**NOMENCLATURE**

| Symbol | Description |
|--------|-------------|
| A      | Area (m²)   |
| Aₐ     | Aperture area (m²) |
| A₋α    | Geometric factor |
| Aₐ₉     | Glass tube area (m²) |
| Aₐα     | Shadow surface area on the reflector (m²) |
| Aᵣ     | Receiver area (m²) |
| Cₚ     | Heat capacity (J/kg K) |
| Cᵣ     | Concentration ratio |
| Cₙ     | Cost of the solar thermal system ($) |
| Dᵣ      | Absorber inside diameter (m) |
| Dₒ      | Absorber outer diameter (m) |
| D₉      | Glass tube diameter (m) |
| dᵣ      | Displacement of the receiver from focus (m) |
| d*     | Universal nonrandom error to Receiver mislocation and reflector profile errors (m) |
| E      | Energy (W) |
| Ḕ      | Exergy (W) |
| Fᵣ     | Heat removal factor (%) |
| f      | Focal length (m) |
| h      | Heat transfer coefficient (W/m² K) |
| hₚ     | Parabolic vertical height (m) |
| hᵦ     | Heat transfer coefficient of fluid (W/m² K) |
| hᵦᵣ    | Heat transfer coefficient of air (W/m² K) |
| kᵣ      | Conductivity of air (W/m K) |
| kₖₙ     | Conductivity of absorber (W/m K) |
| Iᵦ      | Intensity of the sun's radiation (W/m²) |
| L      | Loss (W) |
| Lᵦ      | Collector length (m) |
| mᵦ(mᵦ) | Fluid flow rate (kg/s) |
| Nu     | Nusselt number |
| n      | System's performance years |
| ΔP     | Pressure drop (Pa) |
| Q      | Radiation flux (W/m²) |
| Qₛ     | Thermal energy around receiver (W) |
| Rₑₚ    | Energy Loss rate (kW/$) |
| Rₑₓ    | Exergy Loss rate (kW/$) |
| r      | Interest rate (%) |
| Tᵦₚ     | Ambient temperature (K) |
| Tᵦᵣ     | Inlet temperature (K) |
| Tᵦᵦ     | Outlet temperature (K) |
| Tᵦᵦᵦ     | Glass tube temperature (K) |
| Tᵦᵦᵦᵦ     | Receiver temperature (K) |
| Tₛ     | Sun temperature (K) |
| Uᵦᵦᵦ     | The overall collector heat loss coefficient (W/m² K) |
| v      | Fluid velocity (m/s) |
| wₐ      | Aperture width (m) |

**GREEK SYMBOLS**

| Symbol | Description |
|--------|-------------|
| α      | Absorptance of the receiver |
| β      | Misalignment angle error (rad) |
| β*     | Universal nonrandom error to angular errors (rad) |
| γ      | Intercept factor |
| εᵦᵦᵦᵦ     | Glass cover emissivity |
| εᵦᵦᵦᵦᵦ     | Receiver emissivity |
| ηᵦᵦᵦᵦ     | Optical efficiency (%) |
| ηᵦᵦᵦᵦᵦ     | Thermal efficiency (%) |
| ηᵦᵦᵦᵦᵦᵦ     | Petela efficiency (%) |
| θᵦ     | Incidence angle (rad) |
| μ      | Fluid viscosity (kg/m s) |
| ρ      | Reflectance of the mirror |
| ρᵦ      | Density (kg/m³) |
σ: Stefan–Boltzmann constant (W/m² K⁴)

σtot: Total random error (rad)

σ*: Universal random error (rad)

τ: Transmittance of the glass cover

φ: Angle between focus axis and reflector surface (rad)

φr: Rim angle (rad)

ABBREVIATIONS

Aux: Auxiliary

CRF: Capital recovery factor

GA: Genetic algorithm

HTF: Heat transfer fluid

LFR: Linear Fresnel reflector

PSO: Particle swarm optimization

PTC: Parabolic trough collector

SQP: Semistochastic quadratic bound

TCA: Total system cost annual.

SUBSCRIPTS

a: Aperture

amb: Ambient

en: Energy

ex: Exergy

f: Fluid

g: Glass

L: Loss

i, in: Inner

o, out: Outer

opt: Optical

r: Receiver

s: Sun

t: Thermal

ORCID

Alibakhsh Kasaeian https://orcid.org/0000-0002-4340-190X

REFERENCES

1. Atia DM, Fahmy FH, Ahmed NM, Dorrah HT. Optimal sizing of a solar water heating system based on a genetic algorithm for an aquaculture system. Math Comput Model. 2012;55:1436-1449.

2. Kalogirou SA. A detailed thermal model of a parabolic trough collector receiver. Energy. 2012;48:298-306.

3. Caliskan H, Dincer I, Hepbasli A. Exergoeconomic and environmental impact analyses of a renewable energy based hydrogen production system. Int J Hydrogen Energy. 2013;38:6104-6111.

4. Mwesigye A, Le Roux WG, Bello-Ochende T, Meyer JP. Thermal and thermodynamic analysis of parabolic trough receiver at different concentration ratios and rim angles. In: 10th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, Orlando, Florida; 2014:907-915.

5. Padilla RV, Fontalvo A, Demirkaya G, Martínez A, Quiroga AG. Exergy analysis of parabolic trough solar receiver. Appl Therm Eng. 2014;67:1-8.

6. Hernández-Román MáÁ, Manzano-Ramírez A, Pineda-Phión J, Ortega-Moody J. Exergetic and thermoeconomic analyses of solar air heating processes using a parabolic trough collector. Entropy. 2014;16:4612-4625.

7. Ashouri M, Vandani A, Mehrpooya M, Ahmadi MH, Abdollahpour A. Technoeconomic assessment of a Kalina cycle driven by a parabolic Trough solar collector. Energy Convers Manage. 2015;105:1328-1339.

8. Desai NB, Bandyopadhyay S. Thermo-economic analysis and selection of working fluid for solar organic Rankine cycle. Appl Therm Eng. 2016;95:471-481.

9. Zadeh PM, Sokhansefat T, Kasaeian AB, Kowsary F, Akbarzadeh A. Hybrid optimization algorithm for thermal analysis in a solar parabolic trough collector based on nanofluid. Energy. 2015;82:857-864.

10. Sadati S, Qureshi FU, Baker D. Energetic and economic performance analyses of photovoltaic, parabolic trough collector and wind energy systems for Multan, Pakistan. Renew Sustain Energy Rev. 2016;47:844-855.

11. Toghayani S, Baniasadi E, Afshari E. Thermodynamic analysis and optimization of an integrated Rankine power cycle and nano-fluid based parabolic trough solar collector. Energy Convers Manage. 2016;121:93-104.

12. Abid M, Ratlamwala T, Atikol U. Performance assessment of parabolic dish and parabolic trough solar thermal power plant using nanofluids and molten salts. Int J Energy Res. 2016;40:550-563.

13. Kalogirou SA, Karellas S, Badescu V, Braimakis K. Exergy analysis on solar thermal systems: a better understanding of their sustainability. Renewable Energy. 2016;85:1328-1333.

14. Yanjuan Wang JX, Liu Q, Chen Y, Liu H. Performance analysis of a parabolic trough solar collector using Al2O3/synthetic oil nanofluid. Appl Therm Eng. 2016;107:469-478.

15. Evangelos Bellos CT, Antonopoulos KA. A detailed working fluid investigation for solar parabolic trough collectors. Appl Therm Eng. 2016;114:374-386.

16. Yüksel YE. Thermodynamic assessment of modified Organic Rankine Cycle integrated with parabolic trough collector for hydrogen production. Int J Hydrogen Energy. 2017;43:5832-5841.

17. Patel B, Desai NB, Kachhwaha SS. Thermo-economic analysis of solar-biomass organic Rankine cycle powered cascaded vapor compression-absorption system. Sol Energy. 2017;157:920-933.

18. Conrado LS, Rodriguez-Pulido A, Calderón G. Thermal performance of parabolic trough solar collectors. Renew Sustain Energy Rev. 2017;67:1345-1359.

19. Zare V, Moallemian A. Parabolic trough solar collectors integrated with a Kalina cycle for high temperature applications: energy, exergy and economic analyses. Energy Convers Manage. 2017;151:681-692.

20. Qiu Y, Li MJ, He YL, Tao WQ. Thermal performance analysis of a parabolic trough solar collector using supercritical CO2 as heat transfer fluid under non-uniform solar flux. Appl Therm Eng. 2017;115:1255-1265.

21. Bellos E, Tzivanidis C. A detailed exergetic analysis of parabolic trough collectors. Energy Convers Manage. 2017;149:275-292.
22. Seyed Ebrahim Ghasemi A. Numerical thermal study on effect of porous rings on performance of solar parabolic trough collector. *Appl Therm Eng*. 2017;118:807-816.

23. Tian Z, Perers B, Furbo S, Fan J. Thermo-economic optimization of a hybrid solar district heating plant with flat plate collectors and parabolic trough collectors in series. *Energy Convers Manage*. 2018;165:92-101.

24. Marefatia M, Mehrpoya M, Shafii MB. Optical and thermal analysis of a parabolic trough solar collector for production of thermal energy in different climates in Iran with comparison between the conventional nanofluids. *J Clean Prod*. 2017;175:294-313.

25. Sadaghiyani OK, Boubakran MS, Hassanzadeh A. Energy and exergy analysis of parabolic trough collectors. *Int J Heat Technol*. 2018;36:147-158.

26. AlZahrani AA, Dincer I. Energy and exergy analyses of a parabolic trough solar power plant using carbon dioxide power cycle. *Energy Convers Manage*. 2018;158:476-488.

27. Jose JB, Carballo A, Berenguél M, Palenzuela P. Parabolic trough collector field dynamic model: validation, energetic and exergetic analyses. *Appl Therm Eng*. 2018;148:777-786.

28. Allouhi A, Amine MB, Saidur R, Kousksou T, Jamil A. Energy and exergy analyses of a parabolic trough collector operated with nanofluids for medium and high temperature applications. *Energy Convers Manage*. 2018;155:201-217.

29. Fang J, Liu Q, Liu T, Lei J, Jin H. Thermodynamic evaluation of a distributed energy system integrating a solar thermochemical process with a double-axis tracking parabolic trough collector. *Appl Therm Eng*. 2018;145:541-551.

30. Agagna B, Smaili A, Behar O. An improved model for predicting the performance of parabolic trough solar collectors. *Int J Energy Res*. 2018;42(14):4512-4521.

31. Ray S, Tripathy AK, Sahoo SS, Bindra H. Performance analysis of receiver of parabolic trough solar collector: effect of selective coating, vacuum and semitransparent glass cover. *Int J Energy Res*. 2018;42:1-15.

32. Vakilabadi MA, Bidi M, Najafi AF, Ahmadi MH. Exergy analysis of a hybrid solar-fossil fuel power plant. *Energy Sci Eng*. 2019;7(1):146-161.

33. Fernández-García A, Zarza E, Valenzuela L, Pérez M. Parabolic-trough solar collectors and their applications. *Renew Sustain Energy Rev*. 2010;14:1695-1721.

34. Hoseinzadeh H, Kasaeian A, Shafii MB. Geometric optimization of parabolic trough solar collector based on the local concentration ratio using the Monte Carlo method. *Energy Convers Manage*. 2018;175:278-287.

35. Giiven HM. Derivation of universal error parameters for comprehensive optical analysis of parabolic troughs. *J SolEnergy Eng*. 1986;108:275-281.

36. Kahrobaian MH. A exergy optimization applied to linear parabolic solar collectors. *J Faculty Eng*. 2008;42:131-144.

37. Tzivanidis C, Bellos E. The use of parabolic trough collectors for solar cooling – a case study for Athens climate. *Case Stud Therm Eng*. 2016;8:403-413.

38. Meyer JP, Bello-Ochende T, Ngo LC. Exergetic analysis and optimisation of a parabolic dish collector for low power application. In: *Proceedings of the Postgraduate Symposium 2012. Centre for Renewable and Sustainable Energy Studies (CRSES), Stellenbosch, South Africa*; 2012.

39. Kuo RJ, Han YS. A hybrid of genetic algorithm and particle swarm optimization for solving bi-level linear programming problem – a case study on supply chain model. *Appl Math Model*. 2011;35:3905-3917.

40. Bellos E, Tzivanidis C, Antonopoulos KA, Gkinis G. Thermal enhancement of solar parabolic trough collectors by using nanofluids and converging-diverging absorber tube. *Renewable Energy*. 2016;94:213-222.

41. Gorjian H, Ghobadian B. Estimation of mean monthly and hourly global solar radiation on surfaces tracking the sun case study: Tehran. In: *2012 Second Iranian Conference on Renewable Energy and Distributed Generation, Tehran*; 2012:172-177.

42. Dudley V. SANDIA. Report Test Results for Industrial Solar Technology Parabolic Trough Solar Collector. Albuquerque, USA: SAND94e1117; 1995.

How to cite this article: Hoseinzadeh H, Kasaeian A, Shafii MB. Exergoeconomic optimization and sensitivity analysis of a commercial parabolic trough collector for the climate of Tehran, Iran. *Energy Sci Eng*. 2019;7:2950-2965. [https://doi.org/10.1002/ese3.472](https://doi.org/10.1002/ese3.472)