INTRODUCTION

The energy received by the Earth from the sun in 1 day, provides energy for more than 20 years. Because the amount of energy that comes from the sun to Earth’s surface is $120 \times 10^5$ Solar energy development can expand the level of energy security because it is an energy source that is independent and infinite. In addition, the use of solar energy reduces the environmental impacts. Concentrating solar thermal power plant (CSP) is one of the best choices among clean energies. Also, it can be easily coupled with a fossil fuel boiler to solve variability problem. Wang et al. studied the future strategy for using concentrating solar power in China. Duan et al. Simulated a solar collector connected to an auxiliary fossil fuel heater to generate power and reduce fossil fuel consumption. Behar et al. developed a novel parabolic trough collector and they compared obtained results with experimental data. Bellos and Tzivanidis studied...
In 2016, Ahmadi and Toghraie performed exergy and energy simultaneously provide the heat needed for the power plant. In this study, fossil fuel boiler and solar collectors supplied the heat needed for the power plant economically. The energy analysis obtained from the first thermodynamic law, gives quantitative evaluation of different energy losses taking place in all the components. Therefore, exergy analysis is needed to analyze energy losses qualitatively as well as quantitatively. Ibrahim and Rahman studied a combined power plant with thermal analysis. Mehrooza et al investigated a solar chimney power plant with exergy analysis. They calculated the power output of the power plant throughout the year and concluded that exergy efficiency is varied from 3.5% to 93.3%. Abuelnuor et al studied exergy analysis on a combined cycle power plant. Sukumaran applied exergy analysis on a solar powered airport. Fontalvo et al investigated a power plant that its power generation system is the same as present study and is intended as a hybrid power plant. Sheu and Mitsos optimized a hybrid power plant. At this power plant, fossil fuel boiler and solar collector provide the heat needed for the power plant and is considered as a hybrid fossil-solar power plant. Peng et al analyzed a hybrid fossil-solar power plant. In this study, fossil fuel boiler and solar collectors simultaneously provide the heat needed for the power plant. In 2016, Ahmadi and Toghraie performed exergy and energy analysis on a thermal power plant. They used energy analysis and showed that 96.8% of the total lost energy occurs in the condenser. Also with exergy analysis, they concluded 85.66% of the total exergy is lost in boiler. Lu Yao et al performed exergy analysis for a regenerative turbine in a power plant with a capacity of 1000 MW. They showed exergy destruction in the regenerative turbine system decreases with increasing output power of plant. Eboh et al investigated a new model for calculating the chemical exergy of solid waste. Aljundi studied exergy analysis for a steam power plant. The primary goal of this study was to analyze the various components to determine which component has the most exergy destruction. They concluded that the maximum energy dissipation happens in the condenser and the maximum exergy destruction occurs in the boiler. Ali et al investigated a combined power plant with exergy analysis. They showed that the maximum energy destruction occurs in the boiler and maximum heat loss happens in the condenser. Regularagadda analyzed exergy destruction in the different components in a power plant. They found the maximum exergy destruction is in boiler. Pattanayak et al performed exergy analysis on a combined power plant. They calculated exergy destruction in different components and concluded that boiler has the most exergy destruction. Also Sengupta et al and Zhao et al in separate studies, calculated exergy destruction in different components and showed that the maximum exergy destruction occurs in the boiler. Ameri et al in 2008 carried out an exergy analysis for a thermal power plant. In 2017, Ibrahim et al modeled thermal operation of gas turbine power plant with exergy analysis. They calculated exergy destruction for the boiler, the gas turbine and the compressor. They showed that exergy efficiency of combustion chamber, air compressor and gas turbine are 67.5%, 94.9%, and 92%, respectively. In 2017, Sharma and Singh studied the influence of different variables such as fin density on exergy analysis in HRSG. They investigated exergy losses in a low pressure evaporator and concluded that fin density has low effect on exergy efficiency in HRSG. Exergy analysis in shell and helically coiled tube heat exchangers was investigated by Alimoradi. He considered water as a heat transfer fluid in shell and tubes. Geometrical variables effects on the exergy destruction were studied and the results showed that, with 50% increase of coil radius, it causes a 10.7% reduction in the exergy efficiency. In addition, a 50% increase in the coil length causes a 8.9% reduction in the exergy efficiency. Yildirim and Gene applied the first and second laws of thermodynamic analysis for a milk powder manufacture process. The results showed that the first and second law efficiencies for the overall system are 85.4% and 57.45%, respectively. In 2017, Vandani et al investigated exergoeconomic effect of an additional feed water heater after condensate pump, in a steam cycle. The results showed that increasing the feed water in this cycle, the final cost rate will decrease by 0.16% and the exergy efficiency increase by 0.33%. Mohtaram et al investigated the effect of compressor pressure ratio on ammonia water combined cycle. Bellos et al performed a parametric study on a gas turbine coupled with solar collectors. They analyzed gas turbine and solar collectors separately and results showed that the usage of collector in gas turbine leads to 64% fuel saving, but decreases by 2.8% electricity production. In 2016, Lior studied the exergy and energy analysis of hydro-fractured shale gas. The study calculated the amount of exergy loss in each component and showed that exergy loss increases with time. Also, in the same year, Hofmann and Tsatsaronis performed exergy analysis on a binary Rankine cycle. In this study, exergy analysis was completed on a double Rankine cycle with the working fluids of Potassium and water. They concluded that the amount of efficiency of a Binary Rankine process is considerably higher than a conventional coal-fired. Neri et al calculated the solar radiation exergy from the real radiation data. In this study, several methods were investigated in order to increase power efficiency with solar radiation. Srinivas and Reddy optimized a cogeneration plant. In this study, the coupling of the Kalina cycle and Vapor absorption refrigerator system, a simultaneous cooling unit was developed. The plant characteristics were studied by changing the ratio of mass fraction to produce cooling or power generation; this study showed that the
characteristics are optimal for a mass fraction ratio of 45%. Also, the characteristic power and characteristic cooling in these conditions are 62 and 72 kJ/kg, respectively. Gonçalves studied exergy analysis of a gas turbine unit with two reheaters and two intercoolers. The effect of design parameters, turbine pressure ratio on the operation characteristics and exergy efficiency, has been studied. The research concluded that exergy efficiency increases with increasing the turbine pressure ratio. Hadizadeh et al. improved efficiency of steam power plants applying chillers. They concluded that the steam extraction of high pressure turbine approach is more efficient than other approaches. Khaliq and Kausik evaluated the second law of thermodynamics for the combined cycle of Rankine and Brayton. They found that exergy destruction in combustion chamber is over 50% of the total exergy destruction in the overall cycle. Reddy et al. studied the exergy analysis and performance evaluation on a power plant with 50 MW capacity. They studied the effects of pressure and plant location on the exergy efficiency and showed that exergy efficiency increases by 1.51% with increasing pressure from 90 to 105 bar. Singh and Kausik studied a fossil fuel steam power plant. They understand that maximum exergy destruction occurs in boiler, which is about 62.03%, and that it decreases with the reduction of the excess air combustion or reduction of the gas outlet temperature. Exergy analysis in a solar power plant is investigated by Al-Sulaiman. He showed that more than 50% of the inlet exergy dissipates in collector, and 70% of the total exergy loss happens in collector. Velmurugan et al. studied energy and exergy analysis for solar heaters with different geometries and concluded that wire mesh dual-pass solar air heater has the higher efficiency. In 2016, Deniz did an exergy and energy analysis on a desalination system with a flat solar collector. Its results showed that the maximum exergy and energy efficiencies for optimum flow rate values of 2.76% and 48.1%, respectively. In 2016, Zhu et al. performed studies on exergy analysis of a solar tower coupled with a fossil fuel power generator. In this study, exergy performance and exergy losses analysis were done for each component. Results showed that the boiler and solar tower have minimum exergy performances that together the exergy losses of these two components constitute 85% of the total exergy losses of all components. In 2017, Terhan and Comakli studied energy and exergy analysis of boilers with gas as fuel. They found out that, the part with the most irreversibility in boilers is the combustion chamber. In 2015, Zheng et al. studied thermodynamically a thermal power plant with a solar tower. They studied the effects of various parameters on the exergy performance of a power plant. They concluded that with the rise of the operating temperature of the receiver, the exergy and energy performance would be increased till an optimum temperature and then after that it reduces. Said et al. investigated using of nanoparticles in Heat Transfer Fluid (HTF). They concluded with nanoparticles, the temperature difference between the HTF and the environment decreases and exergy destruction in the collector is reduced. The aim of this research is to perform a dynamic simulation and Exergy analysis on SEGS VI power plant. This is a solar-fossil fuel power plant. In dynamic simulation, the amount of absorbed sun radiation, the effect of flow rate and temperature of oil on the power plant performance, effect of turbine inlet pressure on power plant output power is carried out. Due to the variability of the sun’s radiation, in order to produce a constant power for the power plant, the variable boiler fuel consumption is considered. To improve the performance of the power plant, it is necessary to find the components with the highest exergy destruction. With Exergy analysis, exergy destruction in different components and their effects on power plant performance is analyzed. The effect of turbine inlet pressure, oil flow rate, and oil temperature on exergy efficiency is investigated.

2 | MATHEMATICAL MODELING

Figure 1 schematically shows the entire cycle of solar-fossil fuel power plant. This cycle includes several parabolic trough collectors, auxiliary heater (boiler) and steam power cycle. The existence of a boiler is essential, in order to assure sustainable working conditions in steam cycle and the supply of a stable output power throughout the day.

2.1 | Modeling of parabolic trough collector

In this study, the modeled collectors are LS-2 manufactured by LUX Company (Albuquerque, NM, USA). In the previous study, Ashouri et al. Stuetzle et al. and Fontalvo used this type of collector in their research. Therminol oil VP-1 is considered as heat transfer fluid (HTF). Therminol oil VP-1 is a common fluid for transferring heat that extensively being used in industry. Its specific heat capacity is written as follows:

\[ C_p(T) = 1000(1.509 + 0.002496T + 0.0000007888T^2) \]  \( \text{J/kg k} \)  

where in the Equation 1 \( T \) is in Celsius and \( C_p \) is in J/kg k.

Flux of heat absorbed by collector \( (S_b) \) depends on four parameters: (a) the amount of direct normal radiation \( (I_b) \). (b) Optical efficiency from radiation characteristics \( (\epsilon_{opt}) \). (c) Different amount of dirt on the mirrors \( (\gamma) \). (d) Incident angle \( (\theta) \). Flux of heat absorbed by collector is calculated as follow:

\[ S_b = I_b \epsilon_{opt} \gamma F(\theta) \]  \( \text{W/m}^2 \)

The direct normal radiation \( (I_b) \), can be obtained from Typical Meteorological Year (TMY) file that is presented by
National Solar Radiation Database. γ is a factor that is different for each day and is depend on dirty of collector surface. In this study, γ is assumed to 0.95. F(θ) is incidence modifier function that describe dependence of θ to $S_b$ and is obtained from the expression below:

In the condition that $\cos(\theta) > 0.9$:

$$F = -938564.84377331\cos^6(\theta) + 5222972.5393731\cos^5(\theta) - 12093484.903502\cos^4(\theta) + 14912235.279499\cos^3(\theta) - 10327122.89884\cos^2(\theta) + 3808006.9842855\cos(\theta) - 584041.2051114$$ (3)

And if $\cos(\theta) < 0.9$, the corresponding equation is as follows:

$$F = 7995.6488341455\cos^4(\theta) - 45016.702352137\cos^3(\theta) + 110302.75784952\cos^2(\theta) - 153602.39131907\cos(\theta) + 132938.65779691\cos^4(\theta) - 73211.270566734\cos^3(\theta) + 25050.730094871\cos^2(\theta) - 4867.542978969\cos(\theta) + 411.23466109821$$ (4)

$\epsilon_{opt}$ is optical efficiency from radiation characteristics. In the annulus between the absorber tube and the glass envelope, the radiation, which is not absorbed by the absorber but instead reflected back to the glass envelope, is partially reflected at the glass envelope back to the absorber again where part of it may now be absorbed. That’s why there is a slight increase of the absorbed solar beam radiation compared to the radiation from single absorption, accounted through the definition of the transmittance-absorptance product ($\tau\alpha$). For most practical solar collectors, a reasonable approximation is as below:

$$\epsilon_{opt} = 1.01\rho\tau\alpha$$ (5)

That $\rho$, $\tau$, and $\alpha$ are constants and are equal to 0.95, 0.915, and 0.94, respectively.

$\theta$ is the incident angle. That is the angle between the line perpendicular to the collector and the solar beam radiation vector. Cosine of this angle is calculated as follow:

$$\cos(\theta) = \cos(\theta_z)\cos(\omega) + \cos(\delta)\sin^2(\omega)$$ (6)

That in Equation 6, $\theta_z$, $\omega$, and $\delta$ define the position of the sun throughout the day.

$\theta_z$ is Zenith angle. It is projection of incidence angle onto a horizontal plane.

$$\cos(\theta_z) = \cos(\varphi)\cos(\delta)\cos(\omega) + \sin(\varphi)\sin(\delta)$$ (7)

$\omega$ is the angular displacement of the sun east or west of the local meridian due to rotation of the earth on its axis at 15° per hour.

$$\omega = 15(t_s - 12)$$ (8)
\( \delta \) is declination angle, the angular position of the sun when the sun is on the local meridian and is defined as follows:55

\[
\sin \delta = 23.45 \cos (0.986(N + 284))
\]  

(9)

In above equation, \( N \) is the day of year, is 1 for January 1st and is 365 for December 31st.

\( \phi \) is the latitude angle, the location of SEGS VI plant is \( \varphi = 35^\circ N \). \( t_i \) is solar time. Time based on the apparent angular motion of the sun across the sky.57

\[
t_s = t + \left( 4 \left( L - L_{l o c} \right) + E_i \right) \frac{1}{60}
\]

(10)

\( L_{st} \) and \( L_{loc} \) are standard longitude and local longitude, respectively. The SEGS VI plant is located at the standard longitude and local longitude of 120°W and 117.022°W, respectively.

The parameter \( E_i \) is the equation of time, which takes into account the perturbations in the earth’s rate of rotation which affect the time the sun crosses the observer’s meridian:57

\[
E_i = 229.18(0.000075 + 0.001868 \cos (D) - 0.032077 \sin (D) - 0.014615 \sin (2D))
\]

(11)

\( D \) is the day angle and is defined from the below equation:57

\[
D = \frac{360}{365}(N - 1)
\]

(12)

In this plant, there are 50 collectors that are connected in parallel. Each collector is made up of 16 smaller collectors. These 16 collectors are connected in series. Due to the high variation in oil temperature from entering to the first collector till exiting from the 16th collector, there is a great deal of error if we want to obtain the heat loss of the oil using the average input and output temperature. Thus, we need to divide the total length of the absorber tube into smaller segments and obtain the heat loss by using the average input and output temperature of each segment and with the summation of heat loss in each segment; the output oil temperature from the 16th collector is obtained. With this method the error is very low. The received heat for each segment is:

\[
Q_{a}(\text{segment}) = F_R A_s (S_b - \frac{A_t}{A_s} U_L (T_1 - T_b))
\]

(13)

\( T_1 \) is the HTF temperature inside the absorbent tube and \( T_b \) is the environment temperature.

\( A_t \) and \( A_s \) are defined as follow:

\[
A_t = \pi D_t L_c
\]

(14)

That \( W_o, D_o, D_t \) and \( L_c \) are aperture width of collector, outlet diameter of absorber, outlet diameter of envelope, and Length of element of absorber tube, respectively.

In Equation 13, the first term is the absorbed heat by Absorber and the second term is the heat loss from that. \( F_R \) is calculated as follows:

\[
F_R = \frac{\dot{m}_c C_p}{\pi D_o L_c} (1 - \exp \left( -F' \frac{\pi D_o L_c U_L}{\dot{m}_c C_p} \right))
\]

(16)

\[
F' = \frac{1}{U_L \left( \frac{1}{U_L} + \frac{D_o}{D_{o,pol}} \right)}
\]

(17)

\( U_L \) is calculated based on heat loss for each segment between absorber tube and environment.

\[
U_L = \frac{q_{\text{loss}}}{2\pi D_L \left(T_3 - T_6\right)}
\]

(18)

That:

\[
q_{\text{loss}} = \frac{2\pi L_c k_{\text{eff}}}{\ln \left( \frac{D_t}{D_3} \right)} + \frac{\pi D_3 L_c \sigma(T_3^4 - T_4^4)}{\frac{1}{\epsilon_3} + \left( \frac{1}{\epsilon_4} - 1 \right) \frac{D_t}{D_4}}
\]

(19)

Now, for calculation of \( U_L, T_3, \) and \( T_4 \) are needed. To obtain these temperatures, it is necessary to obtain temperature of different layers in Figure 2 that is calculated in the Appendix.

With energy balance equation, the useful absorbed heat by the oil is obtained as below:

\[
\dot{Q}_a = \dot{m}_c C_p(T_{c2} - T_{c1})
\]

(20)

In order to calculate the transferred heat from absorber to the HTF, collector is divided into small elements. The heat loss to the environment is calculated with high precision.

In Figure 2, different heat transfer mechanisms, conduction, convection, and radiation, that occur in absorber tube are depicted. The requested characteristics of PTC have been mentioned in the table 1:

The assumptions made in order to modeling the collector are given as follow:

1. The space between the absorber tube and the envelope is filled by air.
2. The glass envelope is assumed to have no radial temperature gradients.
3. \( \gamma \) is assumed to be 0.95. This is a factor varying from 1 day to the other. Different amount of dirt on the mirrors and the number of broken collectors influence this factor.
4. Ambient temperature and pressure is assumed to be 25°C and 100 kPa, respectively.

2.2 Steam cycle

In order to derive the thermodynamic characteristics of the steam cycle, energy and mass balance is written for each component.

In Figure 3, a schematic of the steam section of the power plant is shown.

Some assumptions are considered in order to simplify the model:
1. It is assumed that the system performance is time independent.
2. The pressure loss is negligible in the pipes.

| Table 1 | Technical properties of collector |
|---------|----------------------------------|
| Heat transfer fluid (HTF) | Therminol oil VP-1 |
| Total length of a collector | 794.24 m |
| Absorber tube material | Steel |
| Inside diameter of the absorber tube | 0.066 m |
| Outside diameter of the absorber tube | 0.07 m |
| Inside diameter of the glass envelope | 0.112 m |
| Outside diameter of the glass envelope | 0.115 m |
| Aperture width of collector | 4.823 m |
| Fudge factor ($\gamma^*$) | 0.95 |
| Transmittance (\(\tau\)) | 0.915 |
| Absorptance (\(\alpha\)) | 0.94 |
| Latitude angle (\(\phi\)) | 35°N |

3. Heat loss is assumed to be zero in the pipes, pumps and turbines.
4. The output stream from the condenser is assumed as saturated liquid.

The power generated by turbine is calculated as follows:

\[
W_T = m_7 h_7 - m_8 h_8 + m_{10} h_{10} - m_{11} h_{11} + m_{12} h_{12} - m_{13} h_{13} + m_{15} h_{15} - m_{16} h_{16} + m_{18} h_{18} - m_{19} h_{19} \]

(21)
where \( h_1, h_2, h_3, h_4 \) are calculated with regard to the high pressure and low pressure turbine efficiency as follows:

\[
\eta_{\text{HPT1}} = \frac{h_7 - h_8}{h_7 - h_{8,\text{rev}}} \quad (22)
\]

\[
\eta_{\text{HPT2}} = \frac{h_{10} - h_{11}}{h_{10} - h_{11,\text{rev}}} \quad (23)
\]

\[
\eta_{\text{LPT1}} = \frac{h_{12} - h_{13}}{h_{12} - h_{13,\text{rev}}} \quad (24)
\]

\[
\eta_{\text{LPT2}} = \frac{h_{15} - h_{16}}{h_{15} - h_{16,\text{rev}}} \quad (25)
\]

\[
\eta_{\text{LPT3}} = \frac{h_{18} - h_{19}}{h_{18} - h_{19,\text{rev}}} \quad (26)
\]

And for pumps, the efficiency is defined as follow:

\[
\eta_{\text{FWP}} = \frac{h_4 - h_{5,\text{rev}}}{h_4 - h_5} \quad (27)
\]

\[
\eta_{\text{CP}} = \frac{h_1 - h_{2,\text{rev}}}{h_1 - h_2} \quad (28)
\]

The balance of energy for LPFH:

\[
\dot{m}_3 h_2 - \dot{m}_3 h_3 + \dot{m}_1 h_{17} - \dot{m}_2 h_{22} = 0 \quad (29)
\]

The enthalpy \( h_{23} \) can be estimated using the following energy balance:

\[
\dot{m}_4 h_{14} + \dot{m}_2 h_{24} - \dot{m}_3 h_{23} = 0 \quad (30)
\]

To calculate \( h_3 \), the energy balance in the open feed water heater (dearator) can be rewritten as follows:

\[
\dot{m}_3 h_3 + \dot{m}_2 h_{23} - \dot{m}_4 h_4 = 0 \quad (31)
\]

HPFH is a closed feedwater heater and energy conservation is written as follow:

\[
\dot{m}_3 h_5 - \dot{m}_6 h_6 + \dot{m}_9 h_9 - \dot{m}_2 h_{21} = 0 \quad (32)
\]

The drain enthalpy, \( h_{21} \), is calculated from an effectiveness equation for HPFH:

\[
\delta_{\text{HPFH}} = \frac{h_9 - h_{21}}{h_9 - h_5} \quad (33)
\]

The incoming stream to the condenser, \( h_{20} \), is defined as Equation 52.

\[
\dot{m}_1 h_{19} + \dot{m}_5 h_{25} - \dot{m}_2 h_{20} = 0 \quad (34)
\]

The amount of heat that is absorbed by the condenser of the cycle is calculated as follows:

\[
\dot{Q}_{\text{cond}} = \dot{m}_2 h_{20} - \dot{m}_1 h_1 \quad (35)
\]

Isentropic efficiency of turbines and pumps are shown in the Table 2.

The mathematical model is carried out by MATLAB and the thermodynamic properties of the working fluids are obtained from REFPROP 8.0.\(^6\)

### 2.2.1 | Backup energy modeling of the power plant

All CSP power plants require a backup system. This backup system is used to initiate the system, or shut it down, or to be used in order to provide a constant output and increase the operational hours of the power plant. In this power plant, the backup system is a heater with a natural fuel gas. During the time that solar energy cannot provide the heat to supply superheated vapor, this system connects to the circuit in parallel. Hence, no heat storage energy is needed.

\[
\dot{q}_{\text{Aux}} = m_{\text{HTF}} C (T_{\text{out}} - T_{\text{in}}), \quad (36)
\]

where \( \dot{q}_{\text{Aux}} \) is the heat given to the fluid by the fossil-fuel-fired furnace and \( T_{\text{out}} \) is the fluid temperature after heat absorbing, \( T_{\text{in}} \) is the fluid temperature before entering into the backup system.

In this solar-fossil power plant the existing boiler supplies the required energy conditions of insufficient solar energy.

### 2.3 | Energy analysis

In this analysis, the heat loss from different components will be studied and also for a fossil-solar power plant the amount of absorbed energy will be analyzed. Energy balance in an energy system is written as below:

| TABLE 2 | Isentropic efficiency of turbines and pumps |  
|----------------|-------------------|-------------------|
| Turbine         | Efficiency        |
| High pressure turbine 1 (\( \eta_{\text{HPT1}} \)) | 0.84 |
| High pressure turbine 2 (\( \eta_{\text{HPT2}} \)) | 0.85 |
| Low pressure turbine 1 (\( \eta_{\text{LPT1}} \)) | 0.86 |
| Low pressure turbine 2 (\( \eta_{\text{LPT2}} \)) | 0.92 |
| Low pressure turbine 3 (\( \eta_{\text{LPT3}} \)) | 0.94 |
| Low pressure turbine 4 (\( \eta_{\text{LPT4}} \)) | 0.88 |
| Condensing pump efficiency (\( \eta_{\text{CP}} \)) | 0.64 |
| Feedwater pump efficiency (\( \eta_{\text{FWP}} \)) | 0.7 |

- High pressure turbine efficiency 1 (\( \eta_{\text{HPT1}} \))
- High pressure turbine efficiency 2 (\( \eta_{\text{HPT2}} \))
- Low pressure turbine efficiency 1 (\( \eta_{\text{LPT1}} \))
- Low pressure turbine efficiency 2 (\( \eta_{\text{LPT2}} \))
- Low pressure turbine efficiency 3 (\( \eta_{\text{LPT3}} \))
- Low pressure turbine efficiency 4 (\( \eta_{\text{LPT4}} \))
- Condensing pump efficiency (\( \eta_{\text{CP}} \))
- Feedwater pump efficiency (\( \eta_{\text{FWP}} \))
\[
\sum Q + \sum \dot{m}_{in}h_{in} = \sum \dot{W} + \sum \dot{m}_{out}h_{out} \quad (37)
\]

### 2.4 Exergy analysis

Exergy analysis calculates the amount of destruction of the useful work and the irreversibility of each component. In this paper, exergy analysis for a 35 MW solar fossil power plant, the amount of exergy loss in each component and also the amount of the total exergy performance at different hours are evaluated. For a single component, the balance of exergy is written as follows:

\[
\dot{E}^Q + \sum \dot{m}_{in}e_{in} = \dot{E}^w + \sum \dot{m}_{out}e_{out} + \dot{E}_{Dist}. \quad (38)
\]

That:

\[
\dot{E}^w = \dot{W}, \quad (39)
\]

\[
\dot{E}^Q = \left(1 - \frac{T_{amb}}{T_r}\right) \dot{Q} \quad (40)
\]

In Equation 60, \(T_r\) is the temperature of heat source. By neglecting the potential and kinetic exergy, the flow exergy is equal to the physical exergy.

Exergy in each stream defined as:

\[
e_{ph} = (h - h_{amb}) - T_{amb}(s - s_{amb}) \quad (41)
\]

Exergy performance in solar application is the total output work divided by exergy received from the sun:

\[
\eta_{ex,d} = \frac{\sum \dot{W}_{net}}{\sum \dot{E}_{in}}, \quad (42)
\]

where \(\sum \dot{E}_{in}\) is the sum of the sun exergy and the auxiliary heater exergy. The sun exergy is calculated by the following equation:

\[
\dot{E}_{sun} = A_c J_b \left(1 + \frac{1}{3} \left( \frac{T_{amb}}{T_{sun}} \right)^4 - \frac{4}{3} \left( \frac{T_{amb}}{T_{sun}} \right) \right) \quad (43)
\]

In Equation 63, \(T_{sun}\) is the temperature of sun surface that is considered 6000 K.

The rate of the exergy destruction for each component \(Y_D\) is written as below:

\[
Y_D = \frac{\dot{E}^D}{\dot{E}_{total,in}} \quad (44)
\]

The contribution of each component in total exergy destruction is defined as follows:

\[
Y_{D}^* = \frac{\dot{E}^D}{\dot{E}^D_{total}} \quad (45)
\]

The potential to improve a component to be assessed as follows:

\[
\text{IP} = \left(1 - \frac{W_{e}}{100}\right) \dot{E}^D \quad (46)
\]

### 3 DISCUSSION

In this study, exergy analysis is performed on a SEGS VI solar-fossil power plant that to the authors’ knowledge, there is no data available in the literature which evaluates exergy analysis of the SEGS VI power plant. The amount of exergy destruction is calculated in different components and the effect of various parameters such as turbine inlet pressure and HTF flow rate on exergy efficiency and output power is done. So it is a useful study to optimize the performance of the SEGS VI power plant. According to the author’s knowledge, the amount of irreversibility in each component and effect of key parameters on the exergy efficiency has not been studied in any research (as shown in Figure 11). Stuetzel et al.\(^{53}\) studied automatic control for SEGS VI power plant. Saracoglu\(^{64}\) modeled SEGS VI power plant with SAM (Solar Advisory Model) software. Rolim et al.\(^{65}\) performed an analytical model for SEGS VI power plant. To apply energy and exergy analyzes, the power plant was simulated using MATLAB and REFPROP softwares. Then, having the thermodynamic properties for each stream, the exergy and energy destruction was calculated in each component.

In Figure 4, the changes of the fuel flow rate, exergy efficiency and energy efficiency are shown. When an insufficient solar energy is available, an auxiliary heater with fossil fuel supplies the required energy. As it is shown, in the hours of the day that the solar energy is insufficient, fuel flow rate is higher.

Also in Figure 4, energy performance of power plant on the 20th of June is shown. It is shown that solar mode energy efficiency of power plant is lower than fossil fuel mode. By increasing the amount of solar radiation, the amount of energy dissipation increases due to rising HTF temperature. Therefore, the energy efficiency is low in the hours with the highest radiation. In the hours that most of the required thermal energy is provided through the auxiliary heater, the energy efficiency is higher due to the proper insulation for the heater. The exergy efficiency is described as the output work of the power plant divided by the gained solar and fossil fuel energy of the plant. With the change of the absorbed energy by the parabolic trough collector and the fuel flow rate throughout the day, the power plant exergy efficiency would be changed. As seen in Figure 4, during daytime hours the amount of the absorbed solar energy is maximum, the exergy efficiency during those hours are minimal. This means that the existing collectors only absorb a specific amount of solar
energy. Hence, in order to optimize the power plant at first, the amount of solar energy absorbed by the collectors must be increased. In the hours that the plant uses the boiler, as a result of high performance of fossil fuel, the amount of exergy performance of the power plant is high.

Exergy loss of each component is shown in Figure 5. As shown in Figure 5, the loss of exergy in the solar collector has the maximum value due to high temperature difference between HTF and environment. Hence, optimization of this component has the greatest impact on exergy efficiency and a good design of solar collector is needed to reduce the losses. After the collector, the greatest exergy loss happens in the heat exchanger, and the condenser, respectively. The large exergy destruction in the heat exchanger and the condenser is due to the high temperature difference and high energy dissipation in these components.

Exergy losses in pumps are negligible due to the generated pressure difference in the fluid.
In Figure 6 the exergy and energy performances for different components in solar mode are shown. The minimum exergy efficiency happens in the collector, because the oil is heated and its entropy increases. Therefore according to definition of exergy, its exergy efficiency decreases. The minimum energy efficiency occurs in the condenser, because vapor is converted to saturated water. Therefore it has the highest energy dissipation and thus the lowest energy efficiency.

In Table 3, the exergy loss in main components of a power plant is shown when an axillary boiler and solar energy are used simultaneously. During the simultaneous use of two heat sources, the maximum exergy loss happens in collector and boiler. The cause of this exergy loss in these two components is the high temperature gradient in these components. Therefore, for optimizing the power plant, first these two components should be thermodynamically optimized.

In Figure 7, the exergy and energy performances of different components in solar mode are shown. The exergy efficiency and gross power in terms of different inlet pressures of the high pressure turbine are given. In this diagram, when turbine inlet pressure increases, exergy efficiency increases initially and then decreases. As turbine input pressure increases, the input temperature to the turbine and the output power increases simultaneously. Exergy efficiency increases with increasing output power, but with the rising the entrance temperature to the turbine, entropy of the steam stream increases. So the efficiency of the exergy reduces. Thus, initially with increasing turbine inlet pressure, the increase in the amount of output power is dominated by the increase in the input temperature to the turbine. But after a pressure of about 95.8 MPa, the increase in turbine inlet temperature prevails over the output power. Therefore, the variation in exergy efficiency will be on the upside and then on the downside, based on the turbine input pressure.

In Figure 8, the output power and output temperature from collector and output power at different hours on 20th June in solar mode are shown. With rising the amount of sun radiation, the output power increases and the output temperature decreases.
the exergy rate of the exhaust stream from the collector increases, so the amount of useful energy that is given to the steam cycle rises. The increase in the useful energy provided to the steam cycle results in an increase in the exergy of the steam entering the turbine, which ultimately produces more power. It also appears in Figure 8, in the hours of the day that the radiation is high, the output temperature of the collector increases which is due to the large amount of radiation entering to the collector. In this figure, it is understood that in the period of time that the sun’s radiation is the maximum value, one would have the maximum output power.

In Figure 9, the output power variation in term of HTF flow rate is shown. With increasing the flow rate of HTF, the amount of entrance exergy to the heat exchanger rises which gives more exergy to the steam cycle which leads to an increase in the output power. Also in Figure 10, the exergy efficiency is a linear function of the HTF flow rate that with the rise of the flow rare the exergy efficiency augments. By increasing the HTF flow rate, the exergy value of output HTF stream from the collector increases. Thus the total exergy efficiency, which is proportional to the exergy value of the output HTF stream from the collector, is rised. Also it is appears from Figure 10, in a constant HTF flow rate, exergy efficiency increases with rising HTF temperature. Because with increasing the HTF temperature, the amount of stream energy is rised and according to the definition of exergy, the amount of exergy efficiency increases.

In solar collectors, only a certain amount of the existed solar energy would be absorbed by the collector, and that

FIGURE 9  The changes of the output power in terms of the heat transfer fluid’s flow rate

FIGURE 10  Exergy efficiency versus HTF flow rate in different HTF temperature

FIGURE 11  Changes of the solar energy absorbed and normal incident solar energy at different hours and comparison of the results with the existed data
the absorbed energy varies along the day. In addition, the amount of available solar energy will vary according to time. These changes are shown in Figure 11 and show a good agreement with the existing data.

4 | CONCLUSION

In this research, dynamic simulation and Exergy analysis of a solar-fossil fuel power plant is studied. In dynamic simulation, the amount of absorbed sun radiation, the effect of flow rate and temperature of oil on the power plant performance, effect of turbine inlet pressure on power plant output power is carried out. Due to the variability of the sun’s radiation, in order to produce a constant power for the power plant, the boiler fuel consumption is also variable. The simulation results show that maximum boiler consumption is 1.8 kg/s. Exergy analysis, exergy destruction in different components and their effects on power plant performance are analyzed. To improve the performance of the power plant, it is necessary to find the components with the highest exergy destruction. The results show that collector has the highest exergy destruction among all of components. Therefore, in order to rise the performance of solar thermal power plants, one should focus on this component (solar collector). The steam cycle has little impact on the entire exergy loss, so by changing the physical and optical properties of the collector, the exergy efficiency and cycle thermal efficiency of the cycle will be increased. The effect of turbine inlet pressure on exergy efficiency shows that when this pressure is 95.8 bar, the exergy efficiency is maximum. Variations of Exergy efficiency in the length of day show that the minimum exergy efficiency (32.78%) occurs when we have the highest radiation levels. And when there is no solar radiation in the night, fossil fuel-fired auxiliary boiler, and exergy efficiency is in its highest value (44.94%). Also, variation of flow rate and temperature of oil on exergy efficiency shows that increasing oil flow rate and oil temperature, exergy efficiency increases linearly. The results have been compared with valid results, which have a good agreement.

NOMENCLATURE

| Symbol | Definition | Subscripts | Greek |
|--------|------------|------------|-------|
| $T$    | Temperature| $K$        | $\sigma$ |
| $s$    | Entropy    | $N$        | $\nu$ |
| $K$    | Thermal conductivity | $\rho$ | $\epsilon$ |
| $N$    | Number of day | $P$ | $\phi$ |
| $A$    | Area       | $Et$       | $\tau$ |
| $Et$   | Equation of time | $Y_D$ | $\alpha$ |
| $Y_D$  | Irreversibility ratio | $E_D$ | $\omega$ |
| $E_D$  | Exergy loss | $E_{sun}$ | $\delta$ |
| $E_{sun}$ | Sun exergy | loss | $\delta$ |
| $U_L$  | Overall heat transfer coefficient | $m$ | $\phi$ |
| $m$    | Mass flow rate | $u$ | $\phi$ |
| $F_R$  | Heat transfer ratio | $Abbreviations$ | $\omega$ |
| $\delta$ | Declination angle | $Conv$ | $\omega$ |
| $C_p$  | Heat capacity | $Cond$ | $\omega$ |
| $Sb$   | Heat absorbtion | $Kn$ | $\omega$ |
| $lb$   | Beam radiation | $HPFH$ | $\omega$ |
| $\alpha$ | Absorptance | $HRSG$ | $\omega$ |
| $t$    | Local time | $W_a$ | $\omega$ |
| $\sigma$ | Stefan-Boltzmann constant | $\omega$ | $\omega$ |

ORCID

Mohammad H. Ahmadi [ID] https://orcid.org/0000-0002-0097-2534

REFERENCES

1. Chu Y, Meisen P. Review and Comparison of Different Solar Energy Technologies. San Diego, CA: Global Energy Network Institute (GENI); 2011.
2. Devabhaktuni V, Alam M, Depuru SSSR, Green RC II, Nims D, Near C. Solar energy: trends and enabling technologies. Renew Sustain Energy Rev. 2013;19:555-564.
3. Ahmadi MH, Ghazvini M, Sadeghzadeh M, et al. Solar power technology for electricity generation: a critical review. Energy Sci Eng. 2018;6(5):340-361.
4. Wang J, Yang S, Jiang C, Zhang Y, Lund PD. Status and future strategies for concentrating solar power in China. Energy Sci Eng. 2017;5(2):100-109.
5. Duan L, Yu X, Jia S, Wang B, Zhang J. Performance analysis of a tower solar collector-aided coal-fired power generation system. Energy Sci Eng. 2017;5(1):38-50.
6. Behar O, Khellaf A, Mohammedi K. A novel parabolic trough solar collector model–Validation with experimental data and comparison to Engineering Equation Solver (EES). Energy Convers Manage. 2015;106:268-281.
7. Bellos E, Tzivanidis C. A detailed exergetic analysis of parabolic trough collectors. Energy Convers Manage. 2017;149:275-292.
8. Boukelia T, Arslan O, Mecibah M. ANN-based optimization of a parabolic trough solar thermal power plant. Appl Therm Eng. 2016;107:1210-1218.
9. Ortiz C, Chacartegui R, Valverde J, Alosiova A, Becerra J. Power cycles integration in concentrated solar power plants with energy storage based on calcium looping. Energy Convers Manage. 2017;149:815-829.
10. Alashkar A, Gadalla M. Thermo-economic analysis of an integrated solar power generation system using nanofluids. Appl Energy. 2017;191:469-491.

11. Wu H, Li S, Gao L. Exergy destruction mechanism of coal gasification by combining the kinetic method and the energy utilization diagram. J Energy Res Technol. 2017;139(6):062201.

12. Şöhret Y, Açıkkalp E, Hepbasli A, Karakoc TH. Advanced exergy analysis of an aircraft gas turbine engine: splitting exergy destruction into parts. Energy. 2015;90:1219-1228.

13. Ibrahim TK, Rahman M. Optimum performance improvements of the combined cycle based on an intercooler–reheated gas turbine. J Energy Res Technol. 2015;137(6):061601.

14. Mehrpooya M, Shahsavam M, Shartizadeh MMM. Modeling, energy and exergy analysis of solar chimney power plant-Tehran climate data case study. Energy. 2016;115:257-273.

15. Abuelnuor A, Saqr KM, Mohieldein SAA, Dafallah KA, Abdullah MM, Mogoud YAM. Exergy analysis of Garri "2" 180 MW combined cycle power plant. Renew Sustain Energy Rev. 2017;79:960-969.

16. Sakumaran S, Sudhakar K. Performance analysis of solar powered airport based on energy and exergy analysis. Energy. 2018;149:1000-1009.

17. Fontalvo A, Garcia J, Sanjuan M, Padilla RV. Automatic control strategies for hybrid solar-fossil fuel power plants. Renew Energy. 2014;62:424-431.

18. Sheu EJ, Mitsos A. Optimization of a hybrid solar-fossil fuel plant: solar steam reforming of methane in a combined cycle. Energy. 2013;51:193-202.

19. Peng S, Wang Z, Hong H, Xu D, Jin H. Exergy evaluation of a typical 330 MW solar-hybrid coal-fired power plant in China. Energy Convers Manage. 2014;85:848-855.

20. Ahmadi GR, Toghrane D. Energy and exergy analysis of Montazeri steam power plant in Iran. Renew Sustain Energy Rev. 2016;56:454-463.

21. Layao Z, Cheng X, Gang X, Pu B, Yongping Y. Exergy analysis and economic evaluation of the steam superheat utilization using regenerative turbine in ultra-supercritical power plants under design/off-design conditions. Energy Sci Eng. 2017;5(3):156-166.

22. Eboh FC, Ahlström P, Richards T. Estimating the specific chemical exergy of municipal solid waste. Energy Sci Eng. 2016;4(3):217-231.

23. Aljundu IH. Energy and exergy analysis of a steam power plant in Jordan. Appl Therm Eng. 2009;29(2–3):324-328.

24. Ali MS, Shafique QN, Kumar D, Kumar S, Kumar S. Energy and exergy analysis of a 747-MW combined cycle power plant Guddu. Int J Amphli Energy. 2018;39:1-10.

25. Regulargadda P, Dincer I, Naterer G. Exergy analysis of a thermal power plant with measured boiler and turbine losses. Appl Therm Eng. 2010;30(8–9):970-976.

26. Pattanayak L, Sahu JN, Mohanty P. Combined cycle power plant performance evaluation using exergy and energy analysis. Environ Prog Sustain Energ. 2017;36(4):1180-1186.

27. Sengupta S, Datta A, Duttagupta S. ExERGY analysis of a coal-based 210 MW thermal power plant. Int J Energy Res. 2007;31(1):14-28.

28. Zhao H, Bai Y. Thermodynamic performance analysis of the coal-fired power plant with solar thermal utilizations. Int J Energy Res. 2014;38(11):1446-1456.

29. Ameri M, Ahmadi P. Khanmohammadi S. Exergy analysis of a 420 MW combined cycle power plant. Int J Energy Res. 2008;32(2):175-183.

30. Ibrahim TK, Basrawi F, Awad OI, et al. Thermal performance of gas turbine power plant based on exergy analysis. Appl Therm Eng. 2017;115:977-985.

31. Sharma M, Singh O. Exergy analysis of the dual pressure HRSG for varying physical parameters. Appl Therm Eng. 2017;114:993-1001.

32. Alimoradi A. Investigation of exergy efficiency in shell and helically cooled tube heat exchangers. Case Stud Thermal Eng. 2017;10:1-8.

33. Yildirim N, Genc S. Thermodynamic analysis of a milk pasteurization process assisted by geothermal energy. Energy. 2015;90:987-996.

34. Vandani AMK, Joda F, Ahmadi F, Ahmadi MH. Exergoeconomic effect of adding a new feedwater heater to a steam power plant. Mech Indus. 2017;18(2):224.

35. Mohtram S, Chen W, Zargar T, Lin J. Energy-exergy analysis of compressor pressure ratio effects on thermodynamic performance of ammonia water combined cycle. Energy Convers Manage. 2017;134:77-87.

36. Bellos E, Tzivanidis C, Antonopoulos KA. Parametric analysis and optimization of a solar assisted gas turbine. Energy Convers Manage. 2017;139:151-165.

37. Lior N. Exergy, energy, and gas flow analysis of hydrofractured shale gas extraction. J Energy Res Technol. 2016;138(6):061601.

38. Hofmann M, Tsatsaronis G. Exergy-based study of a binary Rankine cycle. J Energy Res Technol. 2016;138(6):062003.

39. Neri M, Luscietti D, Pilotelli M. Computing the exergy of solar radiation from real radiation data. J Energy Res Technol. 2017;139(6):061201.

40. Sríñivas T, Reddy BB. Thermal optimization of a solar thermal cooling cogeneration plant at low temperature heat recovery. J Energy Res Technol. 2014;136(2):021204.

41. Gonca G. Exergetic and ecological performance analyses of a gas turbine system with two intercoolers and two re-heaters. Energy. 2017;124:579-588.

42. Hadizadeh A, Paknafs B, Javaherdeh M, Naghashzadegan H. Evaluating and improving the efficiency of steam power plants using chillers. Mech Indus. 2017;18(2):213.

43. Khalilq A, Kaushik S. Second-law based thermodynamic analysis of Brayton/Rankine combined power cycle with reheat. Appl Energy. 2004;78(2):179-197.

44. Reddy VS, Kaushik S, Tyagi S. Exergetic analysis and performance evaluation of parabolic trough concentrating solar thermal power plant (PTCSTPP). Energy. 2012;39(1):258-273.

45. Singh OK, Kaushik S. Variables influencing the exergy based performance of a steam power plant. Int J Green Energ. 2013;10(3):257-284.

46. Al-Sulaiman FA. Exergy analysis of parabolic trough solar collectors integrated with combined steam and organic Rankine cycles. Energy Convers Manage. 2014;77:441-449.

47. Deniz E, Cinar S. Energy, exergy, economic and environmental (4E) analysis of a solar desalination system with humidification-dehumidification. Energy Conversion and Management. 2016;126:12-19.

48. Zhu Y, Zhai R, Peng H, Yang Y. Exergy destruction analysis of solar tower aided coal-fired power generation system using exergy and advanced exergetic methods. Appl Therm Eng. 2016;108:339-346.

49. Terhan M, Comakli K. Energy and exergy analyses of natural gas-fired boilers in a district heating system. Appl Therm Eng. 2017;121:380-387.
APPENDIX

As shown in Figure 12, HCE is divided into $J_0$ different parts of length $\Delta z$ where:

$$\Delta z = \frac{\text{Length}}{J_0}$$  \hspace{1cm} (47)

The following approximation is provided for obtaining temperature at each node:

$$\frac{\partial T_{\text{HTF}}(z,t)}{\partial z} \approx \frac{T_{\text{HTF},j}(t) - T_{\text{HTF},j-1}(t)}{\Delta z}, \quad j = 1, 2, \ldots, J_0$$  \hspace{1cm} (48)

By this approximation, partial differential equation for the HTF temperature is as follow:

$$\frac{dT_{\text{HTF}}(t)}{dt} = -\frac{V_{\text{HTF}}(t)}{\rho_{\text{HTF}} c_{\text{HTF}}} T_{\text{HTF}}(t)$$
$$+ \frac{\rho_{\text{HTF}} c_{\text{HTF}}}{\rho_{\text{HTF}} c_{\text{HTF}}} \psi_{\text{HTF}}(T_{\text{HTF}}) \rho_{\text{HTF}} c_{\text{HTF}} \frac{\Delta z}{\lambda_{\text{HTF},j}} q_{12}(T_{\text{HTF}}(t), T_{\text{ABS},j}(t))$$  \hspace{1cm} (49)

Boundary condition is presented by Equation 50:

$$T_{\text{HTF,0}}(t) = T_{\text{HTF},\text{inlet}}(t)$$  \hspace{1cm} (50)

And initial condition is as follow:

$$T_{\text{HTF}}(0) = T_{\text{HTF},\text{init}}(z_j, z_j = j \cdot \Delta z, \quad j = 1, 2, \ldots, J_0$$  \hspace{1cm} (51)

For each part of CE, absorber temperature is calculated as follow:

$$\frac{dT_{\text{ABS},j}(t)}{dt} = \frac{1}{\rho_{\text{L}} c_{\text{L}} \frac{A_{\text{COLLECTOR}}}{A_{\text{L}}} \frac{\Delta z}{A_{\text{L}}}} \left[ q_{34}(T_{\text{ABS},j}(t), T_{\text{ENV},j}(t)) - q_{12}(T_{\text{ABS},j}(t), T_{\text{HTF},j}(t)) \right]$$  \hspace{1cm} (52)

Initial condition for Equation 52 is obtained from Equation 53:

$$T_{\text{ABS},j}(0) = T_{\text{ABS,init}}(z_j), \quad z_j = j \cdot \Delta z, \quad j = 1, 2, \ldots, J_0$$  \hspace{1cm} (53)

For each part of envelope, the temperature is obtained as follows:

$$\frac{dT_{\text{ENV},j}(t)}{dt} = \frac{1}{\rho_{\text{ENV}} c_{\text{ENV}} A_{\text{ENV}}} \left[ q_{34}(T_{\text{ABS},j}(t), T_{\text{ENV},j}(t)) - q_{56}(T_{\text{ENV},j}(t), T_{\text{amb},j}(t)) \right]$$  \hspace{1cm} (54)

That for Equation 54, the initial condition is presented by Equation 55:

$$T_{\text{ENV},j}(0) = T_{\text{ENV,init}}(z_j), \quad z_j = j \cdot \Delta z, \quad j = 1, 2, \ldots, J_0$$  \hspace{1cm} (55)

Based on Figure 2, between first and second layers, convection heat transfer occurs. Due to steady state conditions, conduction heat transfer between second and third layers is equal to convection heat transfer between first and second layers.

$$q_{12\text{conv}} = q_{23\text{cond}}$$  \hspace{1cm} (56)
Between third and fourth layers, simultaneously radiation and convection heat transfer occurs.

\[ \dot{q}_{23\text{conv}} = \dot{q}_{34\text{rad}} + \dot{q}_{54\text{conv}} \]  

(57)

Heat transfer between inner and outer surface of envelope is conduction so:

\[ \dot{q}_{34\text{rad}} + \dot{q}_{34\text{conv}} = \dot{q}_{45\text{conv}} \]  

(58)

Outer surface of envelope transfers heat to ambient and sky in the form of convection and radiation respectively. Therefore:

\[ \dot{q}_{45\text{cond}} = \dot{q}_{56\text{conv}} + \dot{q}_{57\text{rad}} \]  

(59)

Convection heat transfer between 1-2 layers is as follow:

\[ \dot{q}_{12\text{conv}} = h_1 D_2 L_c \pi \left( T_2 - T_1 \right) \]  

(60)

That \( h_1 \) is calculated from Equations (27) and (29).

Nusselt number is a function of Reynolds number and Prandtl number. Thus:

\[ Nu_f = \frac{0.023 (Re_D)^{0.8} \cdot (Pr)^{0.4}}{\left( \frac{D_3}{D_4} \right)^{0.78} \left( \frac{D_2}{D_3} \right)^{0.7}} \]  

(64)

Conduction heat transfer between 2-3:

\[ \dot{q}_{23\text{cond}} = \frac{2 \pi k_{23} L_c (T_2 - T_3)}{\ln \left( \frac{D_3}{D_2} \right)} \]  

(65)

That \( k = 0.013 \) \( T_23 = 15.2 \) for tube material.  

Convection heat transfer between 3-4:

\[ \dot{q}_{34\text{conv}} = \frac{2 \pi L_c k_{\text{eff}} (T_3 - T_4)}{\ln \left( \frac{D_4}{D_3} \right)} \]  

(66)

The space between the absorber tube and the envelope is filled by air. Thus in this space, we have free convection heat transfer. In Equation 31, \( k_{\text{eff}} \) is as below:

\[ k_{\text{eff}} = k_{34} \max \left[ 1, 0.386 \left( \frac{pr_{34} \text{Ra}_{34}^*}{pr_{34}} \right)^{0.25} \right] \]  

(67)

That:

\[ \text{Ra}_{34}^* = \frac{\left( \ln \left( \frac{D_3}{D_4} \right) \right)^4}{\left( \frac{D_3 - D_4}{2} \right)^3 \left( D_4^{0.6} + D_3^{0.6} \right)^5} \text{Ra}_{34} \]  

(68)

\( \text{Ra}_{34} \) is Rayleigh number that is a dimensionless number to calculation of natural convection.

\[ \text{Ra}_{34} = 9.81 \left( \frac{D_3 - D_4}{2} \right)^3 \frac{\beta_{34} (T_3 - T_4)}{\theta_{34} a_{34}} \]  

(69)
Parameters used in Equation 34, is defined as follow:

\[
\theta_{34} = \frac{\mu_{34}}{\rho_{34} \cdot \alpha_{34}}; \frac{k_{34}}{C_p \cdot \rho_{34}} \cdot \text{Pr}_{34} = \frac{\theta_{34}}{\alpha_{34}}; \beta_{34} = \frac{1}{T_{34} + 273.15} \quad (70)
\]

Convection heat transfer between 3-4:

\[
\dot{q}_{34\text{conv}} = \frac{2.425 k_{34} L_c (T_3 - T_4)}{\ln \left( \frac{D_5}{D_4} \right)} \left( \frac{Pr \cdot Ra_{A3}}{0.861 + Pr_{34}} \right)^{1/2} \quad (71)
\]

Conduction heat transfer between 4-5:

\[
\dot{q}_{45\text{cond}} = \frac{2\pi k_{45} L_c (T_4 - T_5)}{\ln \left( \frac{D_5}{D_4} \right)} \quad (72)
\]

That \( k_{45} = 1.04 \) for glass.\(^{42}\)

Convection heat transfer between 5-6:

\[
\dot{q}_{56\text{conv}} = h_{56} D_5 \pi L_c (T_5 - T_{\text{amb}}) \quad (73)
\]

Radiation heat transfer between 5-7:

\[
\dot{q}_{57\text{rad}} = \sigma \pi D_5 \varepsilon_5 \left( (T_5 + 273.15)^4 - (T_7 + 273.15)^4 \right) \quad (74)
\]

By solving the Equations 49, 52, and 54 with the corresponding boundary conditions in the MATLAB software, the heat transfer between the different layers of the HCE is obtained.