RESEARCH ARTICLE

Sensitivity analysis of a parabolic trough concentrator with linear V-shape cavity

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Abstract
In the present investigation, a solar parabolic trough concentrator (PTC) with a linear cavity was examined by using a developed numerical model. More specifically, a V-shape cavity receiver was studied in this work as a new and alternative design. The use of a cavity in a PTC is a very promising idea because it leads to high efficiency and low investment cost. Up today, the emphasis in the use of cavities is given in dish concentrators and so there is a need for investigation of cavities also in linear concentrators. Different dimensional and operational parameters of the PTC with a V-shape cavity were considered. More specifically, the position, aperture area, and the tube diameter of the V-shape cavity, solar irradiance, inlet temperature level, and mass flow rate were evaluated based on sensitivity analysis. The results revealed that the highest optical performance was attained when the cavity was placed at the PTC focal line. Moreover, it is found that there is an optimum cavity aperture wide to determine the greatest thermal efficiency. For the optical errors of 5, 10, 15, 20, and 35 mrad, the optimum cavity aperture wide was calculated at 5, 5, 6, 7, and 8 cm, respectively, with the tracking error to be 0°. The highest thermal performance has found when the collector operates with low inlet temperature, high flow rate, high solar irradiation level, and small cavity tube diameter.

KEYWORDS
linear V-shape cavity, parabolic trough concentrator, sensitivity analysis, thermal analysis, thermal oil
1 | INTRODUCTION

Nowadays, the use of sustainable energies is a very attractive choice due to a series of issues such as the increasing greenhouse gas emissions (CO₂), global warming, the depletion of the ozone layer, and the increasing worldwide energy demand.¹,² Diverse types of renewable energy resources such as wind, biomass, and solar could be exploited to mitigate the aforementioned crises.³,⁴ Solar irradiation is accounted as a very promising energy source due to its abundance in numerous countries and its possibility to be converted to either heat or electricity. Solar collectors are the proper devices harvesting incoming solar irradiance and converting it into thermal energy at different temperature levels. The absorbed thermal energy by solar collectors can be employed for various processes such as electricity production, cooling, heating, hydrogen production, desalination, and chemical processes.⁵,⁶ Evacuated tube, solar dish concentrator, flat plate, compound parabolic concentrator, and parabolic trough concentrator (PTC) have been recognized as the practical solar collectors. The PTC is a linear concentrating technology that is one of the mature solar solutions for operating at high temperatures, and it could be considered as the most developed solar concentrating technology.⁷ Generally, two kinds of receivers namely conventional and cavities are employed for the PTCs. The cavities are proposed as an efficient approach to raise the thermal performance of solar systems regarding their special structure and cost-effective designs.⁸,⁹

The thermal performance of the PTC has been studied in numerous studies in the present literature. Akbarzadeh and Valipour¹⁰ reviewed different methods for enhancing the thermal performance of the PTCs including the application of nanofluids, and modified optical and structural parameters. Bellos and Tzivanidisd¹¹ and Hafez et al¹² presented review papers related to different methods for designing PTCs. They suggested some methods for designing PTC yielding the highest performance and the most economical cost, as well as they gave the emphasis in alternative PTC designs to enhance the operation and to reduce the investment cost. In another work, Bellos et al¹³ investigated the daily performance of an integrated PTC under energy and exergy aspects. The application of the storage tank has a low and high influence on the energetic and exergetic aspects of the unit, respectively. Lamrani et al¹⁴ performed a numerical work about the operation of a PTC with thermal oil as the working fluid. The obtained outcomes were validated according to some experimental outcomes. The highest thermal efficiency was calculated as 76%. Agagna et al¹⁵ considered a PTC throughout a numerical and experimental study. They validated the developed model based on the measured experimental results. They reported that the collector performance can be predicted by the developed numerical models.

On the other side, cavities were numerically and experimentally taken into consideration as receiver of solar concentrators.¹⁶,¹⁷ Numerous studies were conducted on the usage of the cavities as the desirable absorber of dish concentrators.¹⁸ Lee et al¹⁹ considered heat losses from a cylindrical-shaped cavity with regard to the variation of aspect ratio and wind direction. It was observed that increasing aspect ratio caused the cavity performance to fall. A conical cavity receiver and a spiral cavity receiver coupled to a dish concentrator were investigated under energy and exergy aspects by Loni et al.²⁰ They found that the conical cavity design is more efficient than the spiral design, and it is better in both thermal and optical terms. Yang et al²¹ considered the convective cavity thermal losses, and they suggested a new air circulation method for reducing heat losses. In some other literature studies, Loni et al²²-²⁴ performed a detailed experimental study about the influence of the different cavity shapes such as cubical, cylindrical, and hemispherical. Moreover, they examined different working fluids such as pure thermal oil, thermal oil/Al₂O₃, thermal oil/SiO₂, and thermal oil/nanocarbon on the performance of the unit. They found that the dish concentrator with nanofluids yielded higher thermal performance than the pure oil.

Also, some researchers have evaluated the usage of cavity in linear solar concentrators including linear Fresnel concentrator, and parabolic trough concentrators under numerical and experimental approaches.⁸,²⁵ Qiu et al²⁶ numerically studied a linear Fresnel concentrator whose capacity shape was trapezoidal, in thermal and optical terms. They found that there are a nonuniform solar flux and temperature distributions on the cavity and also they concluded the lower emissivity of the cavity resulted in higher thermal efficiency and in enhanced optical performance. The performance of a linear Fresnel reflector coupled to the linear cavity with trapezoidal shape was dynamically considered from optical and thermal points of view by Roostae and Ameri.²⁷ They optimized the effective properties of the solar system such as dimensional and optical ones for attaining the best performance. Dabiri et al²⁸ considered a linear Fresnel reflector with a linear trapezoidal-shape cavity through a numerical investigation. The thermal operation of the solar unit was investigated including heat transfer and heat losses. Moreover, the influence of the cavity tube, and cavity institute angle were evaluated. They found that the cavity angle was the most effective parameter for enhancing the thermal efficiency of the unit compared with the cavity tube diameter. Li et al²⁹ considered the thermal performance of a PTC with a novel kind of cavity with arc shape. Different optical and dimensional characteristics were analyzed such as cavity aperture, inclination angle, and emittance of the cavity. They indicated the proposed cavity would lead to the higher performance of the PTC compared with a conventional one. Chen et al³⁰ studied a new shape of a PTC unit with a cavity under the numerical and experimental
aspects. They found that the system efficiency can be enhanced using the incorporation of fins and glass cover. Also, Liang et al.\textsuperscript{31} considered the thermal performance of a PTC with the cavity. The validity of the model was proven based on the measured experimental results.

Throughout the reviewed articles, it can be observed that some types of studies have been carried out on the application of cavity in the PTCs. In this research, the parametric analysis of the linear V-shape cavity receiver as a PTC receiver was done as a novel subject in the solar concentrating scientific area. The use of the cavity yields high efficiency and also decreases the price of the PTC compared to the usual design with the expensive evacuated tube receiver. Generally, there is some significant research on the literature about the use of cavities in PTC, but there are studies about the utilization of cavities in solar dishes. Thus, this work presents important novelty and it has to add something new in the literature. It is also worth stating that the cavity of the PTC is linear, while in the dish it is focal, and thus, there is a need for detailed investigation. In this direction, this work performs a detailed sensitivity analysis by investigating different dimensional and operational parameters of the PTC with a V-shape cavity. More specifically, the position of the cavity, the aperture of the cavity, and cavity tube diameter were examined as well as some operating parameters including solar irradiation, working fluid mass flow rate, and inlet temperature level were studied in detail. The outcomes of this work can make way for the design of the future PTC with cavities with high efficiency so as to create competitive solar collector devices.

2 | Modeling and Description

In this work, a PTC coupled to a linear cavity with V shape was examined under optical and thermal simulations and it is depicted in Figure 1. It should be mentioned that a V-shape cavity receiver was investigated based on exergy and economic aspects by the authors.\textsuperscript{32} The thermal oil passes the cavity tube from one side, and after circulation in the cavity tube and harnessing, solar irradiance is excited from another side of the cavity tube. The cavity was insulated by mineral wool for decreasing the heat waste from the cavity. The optical simulation was conducted by the SolTrace software. Different optical and structural parameters of the system were considered such as optical error, tracking error, cavity position, cavity aperture area, and PTC aperture area. Moreover, the thermal model of the proposed system was done by coding in the Maple software. The thermal modeling was conducted regarding thermal energy-balance equations and the use of the thermal resistance methodology. The impact of the different structural and operational characteristics on the thermal operation of the system was studied such as the aperture area of the cavity, the diameter of the cavity tubes, the inlet temperature level, and the working fluid mass flow rate. Furthermore, this work uses thermal oil in the solar loop as the proper working fluid in order to operate in higher temperatures. The optical modeling and thermal modeling of the PTC will be presented in the next sections in detail.

2.1 | Optical modeling

The PTC unit coupled to a V-shape cavity is analyzed via an optical simulation done by the SolTrace tool which is an accurate software for optical modeling of solar concentrating collectors such as dish concentrator and PTC. Figure 2A,B shows a view of the designated PTC and cavity, respectively. Some assumptions were used for the optical modeling including sun shape and half-angle width as a pillbox, and 4.65 mrad during the simulation, respectively. The number of ray intersections was assumed to equal to 10 000, whereas the reflectance coefficient of the cavity walls was 0.15 based on black cobalt properties as the selective cavity coating. Moreover, the optical error of the PTC ranged from 5 mrad to 35 mrad. It should be noted that the tracking error of the PTC varied between 0° and 2° around axial direction of the PTC. It has to be said that the mean values of the optical properties have been selected which are reasonable and in accordance with the wavelength distribution of the sun irradiation. The dimensions of the structure of the PTC are presented in Table 1 as default parameters. Optical analyses were done for varying amounts of the cavity position, cavity aperture diameter, and cavity tube diameter that will be discussed in the results section in detail. The absorbed solar irradiance by the internal surfaces of the cavity will be considered as the input parameters of the thermal modeling which has been presented in the next section in detail.
At this point, it is useful to highlight that the PTC with a V-shape cavity was optically simulated by the SolTrace without the reflector surface below the cavity for achieving accurate results. Figure 3A,B displays a view of the simulated system including and without reflector surface below the linear V-shape cavity, respectively. All of the presented results in this research were conducted based on the presented schematic in Figure 3B for the first time in the world. Also, the cavity was simulated whereas the solar irradiance was captured by the internal faces of the cavity, and reflected by the outside of the cavity for calculating accuracy results. A view of absorbing solar irradiance by the internal surface of the cavity and reflected solar irradiance by the outside cavity walls is presented in Figure 4A,B, respectively.

2.2 | Thermal modeling

Figure 5 illustrates the cross section of the examined cavity receiver. As seen, mineral wool was used to insulate the outside of the cavity for reducing the dissipated heat from the cavity. Also, physical properties of the cavity including cavity aperture wide, cavity tube diameter, cavity angels, and insulation thickness were presented as “a,” “d_{tube},” “θ,” and “t_{ins},” respectively.

The absorbed thermal energy would be wasted from the V-shape cavity through different heat transfer mechanisms including convection, conduction, and radiation. Figure 6 shows the flow of the various thermal losses from the cavity to the ambient. The thermal resistance method was used to simulate the thermal behavior of the examined receiver.
FIGURE 4 A view of (A) absorbing solar irradiance by the internal surface of the cavity and (B) reflected solar irradiance by the outside cavity walls.

FIGURE 5 Cross section of the examined cavity with the main dimensions explained.

FIGURE 6 Thermal losses depiction of the examined receiver.

FIGURE 7 Thermal resistance depiction for (A) internal dissipated heat and (B) external dissipated heat from the cavity to the ambient.
Internal dissipated heat and external dissipated heat from the V-shape cavity were calculated by using the thermal resistance method presented in Figure 7A,B. The thermal resistance which is associated with the internal dissipated heat from the V-shape cavity is assumed to be the thermal resistance which is related to the heat waste from the glass cover to the surrounding \((R_1, \text{ and } R_2 \text{ in Figure 7A}),\) and thermal resistance is related to heat losses from the internal space of the cavity to the glass cover \((R_3 \text{ and } R_4 \text{ in Figure 7A}).\) Thermal resistance is related to the heat losses from the glass cover to the ambient condition including thermal resistance related to convection heat losses as displayed as \(R_1\) in Figure 7A, and thermal resistance is related to radiation thermal waste as displayed as \(R_2\) in Figure 7A. Thermal resistance is related to the thermal losses from the internal space of the cavity to the glass cover including thermal resistance related to internal convection heat losses as shown as \(R_3\) in Figure 7A, and internal radiation thermal waste as shown as \(R_4\) in Figure 7A. Thermal resistance related to the internal dissipated heat equations is presented in detail as follows:

- Thermal resistance related to convection heat losses from glass cover to the surrounding:

\[
R_1 = \frac{1}{A_{gc}h_{conv,gc}}
\]  

where

\[
h_{conv,gc} = \frac{0.27\lambda_a (Gr \cdot Pr)^{1/4}}{w_{gc}}, \quad w_{gc} = w_{ap}
\]

\[
Gr = \frac{ga\Delta Td_0^3}{v^2}
\]

- Thermal resistance related to radiation thermal waste from glass cover to the surrounding:

\[
R_2 = \frac{1}{A_{gc}h_{rad,gc}}
\]

where

\[
\dot{Q}_{rad,gc} = \delta \varepsilon_{gc} A_{gc} \left( T_s^4 - T_{gc}^4 \right)
\]

- Thermal resistance related to conduction thermal losses from the mineral wool layer:

\[
R_5 = \frac{t_{ins}}{k_{ins}A_{cavity}}
\]

Moreover, it has to be said that the external dissipated heat from the cavity is described as heat waste from the insulation layer of the cavity to the ambient. The external dissipated heat including conduction thermal losses from the insulation layer is shown as \(R_5\) in Figure 7B, and convection and radiation thermal waste from the exterior of the mineral wool layer to the ambient condition are presented as \(R_6\) and \(R_7\) in Figure 7B, respectively. All of the thermal resistance related to external dissipated heat will be computed as follows:

- Thermal resistance related to convection heat losses from the mineral wool layer:

\[
R_6 = \frac{1}{A_{cavity}h_{rad,ext}}
\]

where

\[
h_{conv,ext} = \frac{0.59\lambda_a (Gr \cdot Pr)^{1/4}}{d_0}, \quad d_0 = \text{cavity depth}
\]

\[
Gr = \frac{ga\Delta Td_0^3}{v^2}
\]
where
\[ \dot{Q}_{\text{rad,ext}} = \delta \epsilon A_{\text{cavity}} \left( T_s^4 - T_\infty^4 \right) \] (14)

- Thermal resistance related to convection thermal waste from mineral wool layer to ambient:

\[ R_0 = \frac{1}{A_{\text{cavity}} h_{\text{conv,ext}}} \] (15)

The determination of the convection heat transfer coefficient on the insulation layer external surface has to take into account the combination of the natural convection and of the forced convection owing to the wind. To calculate the Nusselt number when the wind direction is parallel to the receiver insulation layer, the Equation (16) is used:\(^3\)

\[ Nu = 0.664 Re_L^{0.5} Pr^{1/3} \quad Re < 5 \times 10^5, \quad 0.6 \leq Pr \leq 60 \] (16)

This value for the natural convection on the cavity is:\(^3\)

\[ Nu = 0.59 Ra_L^{1/7} \cos \theta \quad 10^4 < Ra < 10^9 \] (17)

The Nusselt number could be estimated by considering the combined natural and forced convection according to:\(^3\)

\[ Nu_{\text{combined}} = \left( Nu_{\text{forced}}^{3.5} + Nu_{\text{natural}}^{3.5} \right)^{1/7} \] (18)

The overall convection heat transfer coefficient is calculated as below:

\[ h_{\text{conv,ext}} = \frac{1}{A_{\text{total}}} \sum_i h_i A_i \] (19)

where \( A_{\text{total}} = \sum_i A_i \)

With regard to the thermal resistance method and presented schematics in Figure 7, overall interior thermal waste can be computed as follows:

\[ \dot{Q}_{\text{loss,int}} = \frac{T_s - T_\infty}{R_{\text{total,int}}} \] (20)

where

\[ R_{\text{total,int}} = \frac{R_1 R_2}{R_1 + R_2} + \frac{R_3 R_4}{R_3 + R_4} \] (21)

The overall exterior thermal waste can be estimated as below:

\[ \dot{Q}_{\text{loss,ext}} = \frac{T_s - T_\infty}{R_{\text{total,ext}}} \] (22)

\[ R_{\text{total,ext}} = R_5 + \frac{R_6 R_7}{R_6 + R_7} \] (23)

Cavity net heat gain is calculated as follows:

\[ \dot{Q}_{\text{net}} = \dot{Q}^* - \dot{Q}_{\text{loss,ext}} - \dot{Q}_{\text{loss,int}} \] (24)

where \( \dot{Q}^* \) (W) is received solar irradiance by the cavity walls evaluated by the SolTrace software. The previous equation is very important because it practically connects the two used tools in order to perform a proper thermal and optical study.

The collector thermal efficiency and the solar irradiation inputs are defined as below:

\[ \eta_{th} = \frac{\dot{Q}_{\text{net}}}{Q_{\text{solar}}} \] (25)

\[ Q_{\text{solar}} = I_{\text{sun}} A_{\text{ap,PTC}} \] (26)

Finally, solving Equations (27) and (28) simultaneously by employing the Newton-Raphson method leads to determine the surface temperature \( T_{s,n} \) and the net heat transfer rate \( \dot{Q}_{\text{net,n}} \) at different positions of the cavity tube.\(^3\) The cavity tube was separated into a smaller length along the cavity tube. The length of each element was assumed as 10 cm for calculation accuracy results.

\[ \dot{Q}_{\text{net,n}} = \frac{\left( T_{s,n} - \sum_{i=1}^{n-1} \left( \frac{Q_{\text{wall}}}{m c_p} \right) - T_{\text{inlet,0}} \right)}{\left( \frac{1}{h x_i} + \frac{1}{2 m c_p} \right)} \] (27)

and

\[ \dot{Q}_{\text{net,n}} = \dot{Q}^* - \frac{A_n}{R_{\text{total}}} \left( T_{s,n} - T_\infty \right) \] (28)

The thermal properties of working fluid are acquired as:\(^6\)

\[ k_f = 0.1882 - 8.304 \times 10^{-5} \left( T_f \right) \left( \frac{W}{mK} \right) \] (29)

\[ c_{pf} = 0.8132 + 3.706 \times 10^{-3} \left( T_f \right) \left( \frac{kJ}{kgK} \right) \] (30)

\[ \rho_f = 1071.76 - 0.72 \left( T_f \right) \left( \frac{kg}{m^3} \right) \] (31)

\[ Pr = 6.73899 \times 10^{21} \left( T_f \right)^{-7.7127} \] (32)

It should be said that the surface temperature of the cavity walls was assumed as 120°C during thermal resistance
calculations.\textsuperscript{35} However, the author will investigate the influence of variation of constant surface temperature during the thermal resistance method in the future works. Also, they will plan to develop their model for using the calculated $T_{s,n}$ for each element as assumed surface temperature during thermal resistance calculations.

\section*{2.3 \quad Validation of the proposed model}

The parabolic trough concentrator was used as the solar liner focal reflector. The dimension of a built PTC collector at Tehran University, Tehran province (Iran), was used in this study.\textsuperscript{36} The designed PTC solar system in reference\textsuperscript{36} was equipped using a conventional receiver with a glass tube. In the current study, the results from the developed numerical model were compared and validated with those reported by an experimental study in Ref.36 Table 1 indicates the physical dimensions of the proposed PTC.

Figure 9 shows the efficiency results of the present numerical work and of an experimental literature study of Ref.36 As seen in Figure 8, a promising agreement could be observed between both collected data about the collector efficiency.

\section*{3 \quad RESULT AND DISCUSSION}

In section 3, the optical and thermal performance results of the examined solar collector are presented and discussed in order to determine the best structural and operational parameters. Firstly, the location of the cavity in comparison with the PTC focal line is evaluated. Secondly, the results of PTC performance are reported under the variation of the cavity aperture area. Finally, the PTC performance results under variation of the cavity tube diameter as well as operational parameters are presented.

\subsection*{3.1 \quad Cavity position optimization}

Optical analyses of the PTC with V-shape cavity at different positions of the cavity from the PTC focal line including $\pm a$, $\pm a/2$, and 0 are presented in Figure 9. In these analyses, “$a$” is assumed equal to cavity aperture wide. It is worth mentioning that the SolTrace software was used to analyze such a system. The shown solar beam by yellow color presents the absorbed solar beam by the internal space of the cavity wall, whereas the shown solar beam by red color depicts the nonabsorbed solar beam by the internal space of the cavity wall. In other words, the shown solar beam by yellow color is accounted for calculating the optical efficiency of the cavity at different positions. As seen in Figure 9, the PTC presents the highest optical performance when the cavity is placed at the focal line of the parabolic trough. It can be concluded that the position of the cavity at the focal line of the PTC resulted in the highest optical performance. Also, Figure 10 presents solar flux distribution on internal walls of the V-shape cavity at different positions of the cavity from the PTC focal line including $\pm a$, $\pm a/2$, and 0. As mentioned, the cavity at the focal line of the parabolic trough has the maximum solar beam absorption.

Figures 11 and 12 displayed variation of cavity absorbed solar irradiance, and optical efficiency of the examined solar collector vs different amounts of the cavity position in comparison with the PTC focal line including $\pm a$, $\pm a/2$, and 0, whereas “$a$” is assumed equal to cavity aperture wide of 0.06 m, respectively. The optical analyses were conducted at different optical errors including 5, 10, 15, 20, and 35 mrad at the tracking error of 0°. As seen in Figures 11 and 12, the
highest amounts of the cavity absorbed solar irradiance and optical efficiency were achieved when the cavity was placed at the parabola focal line. It could be observed that the optical performance of the solar system is getting lower when there is a rise in the optical error. Also, it was found that the optical efficiency trend has reflected the pattern of the absorbed solar irradiance by the receiver.

Variation of cavity absorbed solar irradiance of the PTC with V-shape cavity vs different amounts of the cavity position at different amounts of tracing errors including 0°, 1°, and 2° and optical error equal to 5 mrad is depicted in Figure 12. Different positions of the cavity in comparison with the PTC focal line were studied including ±a, ±a/2, and 0, whereas “a” is assumed equal to cavity aperture wide of 0.06 m. As resulted from Figure 12, the highest amounts of the cavity absorbed solar irradiance were calculated when the cavity is located at the focal line of the PTC. Also, it was found that the optical efficiency trend has reflected the pattern of the absorbed solar irradiance by the receiver.

FIGURE 9 Optical analysis results at different positions of the cavity from the PTC focal line including (A) −a, (B) −a/2, (C) 0, (D) +a/2, and (E) +a, where “a” is equal to cavity aperture wide

the PTC with V-shape cavity at the PTC focal line with the lowest optical error and lowest tracking errors has the highest optical performance.

3.2 Cavity aperture optimization

In the second stage, the cavity aperture area was optimized based on optical and thermal analyses. Figure 13A,B presents variations of cavity absorbed solar energy by the cavity walls and collector optical performance vs different cavity aperture wide, respectively. The analyses were conducted for different optical errors including 5 mrad, 10 mrad, 15 mrad, 20 mrad, and 35 mrad at zero tracking error. It is worth mentioning that the cavity was placed at the parabola focal line. As seen from Figure 13A, to achieve the maximum absorbed solar energy for each level of the determined optical errors, an optimum cavity aperture area exists. This is due to the decreasing amount of incoming solar irradiance by increasing the cavity aperture area. Also, it could be resulted from Figure 13B, the optical efficiency of the PTC grows as increasing the cavity aperture area until a level of investigated cavity aperture area,
and then, it remains constant. On the other side, the optical performance of the collector decreased with the growing optical error.

Variation of the net absorbed power by the linear cavity and the trade-off between the thermal efficiency and the various cavity aperture dimensions for various optical errors at zero tracking error are presented in Figure 14A,B, respectively. As understood from Figure 14A,B, there is an optimum amount of the cavity aperture area to attain the highest harvested solar energy for each amount of the optical errors. The reason for this issue is that the optical error is getting lower with a decrease in the aperture area of the cavity and the thermal efficiency of the PTC decreases with an increase in the aperture area of the cavity. Also, it could be seen that the optimum amounts of the cavity aperture area increase with the increasing optical error of the solar system. Optimum amounts of the
absorbed heat, thermal efficiency, and cavity aperture wide at different optical errors are reported in Table 2. Optimum amounts of the cavity aperture wide were calculated equal to 5, 5, 6, 7, and 8 cm for the optical error of 5, 10, 15, 20, and 35 mrad, respectively. Optimum thermal efficiencies of the PTC were calculated equal to 68%, 64%, 59%, 58%, and 51%, for the optimum amounts of the cavity aperture wide as 3, 4, 5, 6, and 6 cm, respectively. Finally, a variation of the net captured thermal energy by the thermal oil vs variation of the cavity aperture wide has shown a similar pattern with a variety of the collector thermal performance vs variation of the cavity aperture wide at different optical errors.
In section 3.3, the optical and thermal performance of the examined system was investigated with 3 different diameters of the cavity tube including 5 mm, 10 mm, and 25 mm. The cavity was considered with an optimum cavity aperture area of 5 cm, the optical error of 10 mrad, and tracking error of 1°, whereas the linear cavity was placed at the parabola focal line. It should be noted that the distribution of solar irradiance was assumed nonuniform in the current research. The solar flux distribution was calculated using the SolTrace software along the cavity tube as shown in Figure 15. Also, Figure 16 presents the absorbed heat distribution along the cavity tubes. Solar heat flux distribution along the cavity tube is presented in Figure 15 for three investigated cavity tube diameters including 5 mm, 10 mm, and 25 mm at the optical error equal to 10 mrad, and tracking error equal to 1°. As mentioned in section 2, the cavity was separated into smaller elements along the cavity tube. The length of each element was assumed to equal to 10 cm. As seen from Figure 15, the solar heat flux amount has enhanced with increasing cavity tube diameter. This issue is due to more surface area of the cavity tube for hitting solar irradiance with increasing cavity tube diameter. Also, the variation of the absorbed heat along the cavity tube for various diameters of the cavity tube is shown in Figure 16. As seen, the absorbed heat amount has improved with increasing cavity tube diameter.

The collector thermal performance under the variation of solar irradiance for three cavity tube diameters is reported in Table 3. Variation of thermal efficiency, cavity heat gain, cavity surface temperature, the outlet temperature of the working fluid, and pressure drop were presented regarding the solar intensity variation and cavity tube diameter. It
should be noted that the solar working fluid for all analyses was thermal oil with a constant inlet temperature level of 50°C and a constant working fluid flow rate of 50 mL/s. Table 3 shows that the net absorbed heat and thermal efficiency of the receiver have enhanced with a growth in the solar irradiance for three investigated diameters of the cavity tube. Also, the cavity surface temperature and working fluid outlet temperature have resulted in improving with increasing solar irradiance for three investigated cavity tube diameters. Moreover, the PTC thermal performance has enhanced with decreasing cavity tube diameter. On the other hand, the pressure drop of the solar collector increases when the cavity tube diameter decreases as seen in Table 3. The highest amount of thermal performance was calculated as 72.08% for the cavity tube diameter equivalent to 5 mm, solar irradiation equal to 1000 W/m², working fluid inlet temperature level equal to 50°C, and thermal oil flow rate equal to 50 mL/s.

Table 4 presents the collector thermal performance under the variation of the fluid inlet temperature for three cavity tube diameters. The constant flow rate and solar irradiance were supposed 50 mL/s and 800 W/m², respectively. Variation of different characteristics such as the cavity useful heat production, collector thermal efficiency, cavity surface temperature, thermal oil outlet temperature, and pressure drop was reported for different amounts of working fluid inlet temperature and cavity tube diameter. Three amounts of working fluid inlet temperature were investigated including 50°C, 100°C, and 150°C. As concluded from Table 4, the net harvested heat and thermal efficiency reduced with a rise in the thermal oil inlet temperature level. On the other side, the cavity surface temperature and the thermal oil outlet temperature level have shown increasing by a rise in the inlet temperature of the solar working fluid. Furthermore, the PTC thermal performance has resulted in enhancing by decreasing cavity tube diameter. Conversely, the pressure drop of the solar system increases with a decrease in cavity tube diameter as well as decreasing the inlet temperature of the thermal oil. The maximum amount of thermal efficiency was calculated as 72.04% for the cavity tube diameter equivalent to 5 mm, and solar working fluid inlet temperature equal to 50°C at solar irradiance of 800 W/m² and a flow rate of 50 mL/s.

Finally, the impact of the fluid flow rate on the collector thermal performance is depicted in Table 5 for different cavity tube diameters. The inlet temperature was selected at
50°C and the solar irradiation level at 800 W/m² in this analysis. Three levels of flow rate were studied including 30 mL/s, 210 mL/s and, 530 mL/s. Different parameters including the cavity useful heat production, collector thermal efficiency, cavity surface temperature, outlet temperature of the working fluid, and pressure drop were investigated under variation of the working fluid flow rate and cavity tube diameter. As resulted from Table 5, the net absorbed heat and thermal efficiency have increased by an increase in the thermal oil flow rate. Also, the cavity surface temperature and working fluid outlet temperature resulted in a reduction by increasing the flow rate of the solar working fluid. The highest thermal efficiency of the proposed PTC was acquired as 72.47% when the cavity tube diameter, the flow rate, the thermal oil inlet temperature, and the solar irradiance were equal to 5 mm, 530 mL/s 50°C, and 800 W/m², respectively.

4 | CONCLUSIONS

In this research, the optical performance and the thermal performance of a parabolic trough collector with V-shape cavity

### TABLE 3 Thermal performance of the examined collector with a variation of solar irradiance for three cavity tube diameters

| $I_{\text{sun}}$ (W/m²) | $d_{\text{tube}} = 5$ mm | $\dot{Q}_{\text{net}}$ (W) | $\eta_{\text{th}}$ | $T_s$ (K) | $T_{\text{out}}$ (K) | $\Delta P$ (MPa) |
|------------------------|---------------------------|--------------------------|-----------------|----------|-----------------|----------------|
| 600                    | 475.00                    | 0.7197                   | 346.74          | 328.63   | 2.14            |
| 800                    | 633.95                    | 0.7204                   | 354.68          | 330.51   | 2.14            |
| 1000                   | 792.88                    | 0.7208                   | 362.63          | 332.40   | 2.14            |

| $I_{\text{sun}}$ (W/m²) | $d_{\text{tube}} = 10$ mm | $\dot{Q}_{\text{net}}$ (W) | $\eta_{\text{th}}$ | $T_s$ (K) | $T_{\text{out}}$ (K) | $\Delta P$ (MPa) |
|------------------------|---------------------------|--------------------------|-----------------|----------|-----------------|----------------|
| 600                    | 468.92                    | 0.7105                   | 366.51          | 328.56   | 0.06            |
| 800                    | 626.04                    | 0.7114                   | 381.09          | 330.42   | 0.06            |
| 1000                   | 782.92                    | 0.7117                   | 395.64          | 332.28   | 0.06            |

| $I_{\text{sun}}$ (W/m²) | $d_{\text{tube}} = 25$ mm | $\dot{Q}_{\text{net}}$ (W) | $\eta_{\text{th}}$ | $T_s$ (K) | $T_{\text{out}}$ (K) | $\Delta P \times 10^{-3}$ (MPa) |
|------------------------|---------------------------|--------------------------|-----------------|----------|-----------------|-----------------------------|
| 600                    | 454.32                    | 0.6884                   | 422.76          | 328.39   | 0.69            |
| 800                    | 605.89                    | 0.6885                   | 456.05          | 330.18   | 0.69            |
| 1000                   | 756.29                    | 0.6875                   | 489.10          | 331.96   | 0.69            |

### TABLE 4 Thermal performance of the examined collector with a variation of inlet temperature for three cavity tube diameters

| $T_{\text{in}}$ (°C) | $d_{\text{tube}} = 5$ mm | $\dot{Q}_{\text{net}}$ (W) | $\eta_{\text{th}}$ | $T_{\text{avg}}$ (K) | $T_{\text{out}}$ (K) | $\Delta P$ (MPa) |
|----------------------|---------------------------|--------------------------|-----------------|---------------------|-----------------|----------------|
| 50                   | 633.95                    | 0.7204                   | 354.68          | 380.13              | 2.14            |
| 100                  | 628.49                    | 0.7142                   | 403.92          | 380.13              | 0.56            |
| 150                  | 622.90                    | 0.7078                   | 453.21          | 429.82              | 0.25            |

| $T_{\text{in}}$ (°C) | $d_{\text{tube}} = 10$ mm | $\dot{Q}_{\text{net}}$ (W) | $\eta_{\text{th}}$ | $T_{\text{avg}}$ (K) | $T_{\text{out}}$ (K) | $\Delta P$ (MPa) |
|----------------------|---------------------------|--------------------------|-----------------|---------------------|-----------------|----------------|
| 50                   | 626.04                    | 0.7114                   | 381.09          | 380.01              | 0.17            |
| 100                  | 617.87                    | 0.7021                   | 429.56          | 379.72              | 0.07            |
| 150                  | 608.62                    | 0.6916                   | 478.01          | 429.66              | 0.07            |

| $T_{\text{in}}$ (°C) | $d_{\text{tube}} = 25$ mm | $\dot{Q}_{\text{net}}$ (W) | $\eta_{\text{th}}$ | $T_{\text{avg}}$ (K) | $T_{\text{out}}$ (K) | $\Delta P \times 10^{-3}$ (MPa) |
|----------------------|---------------------------|--------------------------|-----------------|---------------------|-----------------|-----------------------------|
| 50                   | 605.89                    | 0.6885                   | 456.05          | 330.18              | 0.69            |
| 100                  | 592.68                    | 0.6735                   | 501.62          | 379.72              | 0.18            |
| 150                  | 576.98                    | 0.6557                   | 546.78          | 429.32              | 0.08            |
were numerically calculated. Different dimensional parameters such as cavity position in comparison with the PTC focal position, cavity aperture area, and cavity tube diameter were considered. Moreover, some operational parameters such as solar irradiance, inlet temperature level, and working fluid mass flow rate were studied in this investigation. The main results of this research could be highlighted as below:

- The maximum collector optical performance was found when the cavity is located in the parabola focal line.
- The collector optical efficiency shows a reducing trend with the increase in the tracking and optical errors.
- The collector optical efficiency increases as increasing the cavity aperture area until a level of investigated cavity aperture area, and then, it remains constant.
- To achieve the maximum absorbed solar energy for each level of the determined optical errors, an optimum cavity aperture area exists. The optimum values of the cavity aperture wide were calculated equal to 5, 5, 6, 7, and 8 cm for the optical error of 5, 10, 15, 20, and 35 mrad, respectively, at zero tracking error.
- Optimum thermal efficiencies of the PTC were calculated equal to 68%, 64%, 59%, 58%, and 51%, for the optimum values of the cavity aperture wide as 5, 5, 6, 7, and 8 cm, respectively.
- The highest collector thermal performance has been found when the system operates with the minimum inlet temperature, the highest volumetric flow rate, the highest solar beam irradiation level, and the smallest cavity tube diameter.
- The greatest thermal efficiency was calculated as 72.08% for the cavity tube diameter of 5 mm and solar irradiance of 1000 W/m² with the solar working fluid inlet temperature level at 50°C and a volumetric flow rate at 50 mL/s.

### Nomenclature

- $a$: cavity aperture wide, m
- $A$: area, m²
- $c_p$: specific heat capacity, J kg⁻¹ K⁻¹
- $d_0$: cavity depth, m
- $d$: diameter of the receiver tube, m
- $f$: focal distance, cm
- $Gr$: dimensionless Grashof number
- $G$: acceleration of the gravity, m s⁻²
- $h$: heat transfer coefficient, W m⁻² K⁻¹
- $I$: solar beam direct irradiation level, W m⁻²
- $k$: material thermal conductivity, W m⁻¹ K⁻¹
- $L_c$: parabola length, m
- $m$: working fluid mass flow rate, kg s⁻¹
- $N$: number of tube sections
- $Nu$: dimensionless Nusselt number
- $P$: pressure level, Pa
- $Pr$: dimensionless Prandtl number
- $Q_{net}$: useful heat production in the cavity, W
- $Q^*$: available solar irradiation at the cavity, W
- $Q_{loss}$: heat losses in the cavity, W
- $Q_{solar}$: available solar irradiation at the collector, W
- $R$: thermal resistance, K W⁻¹
- $Ra$: dimensionless Raleigh number
- $Re$: dimensionless Reynolds number
- $T$: temperature level, K
- $t$: thickness, m

### Table 5

| $\dot{m}$ (mL/s) | $d_{tub} = 5$ mm | $Q_{net}$ (W) | $\eta_{th}$ | $T_{s,avg}$ (K) | $T_{out}$ (K) | $\Delta P$ (MPa) |
|------------------|----------------|---------------|-------------|----------------|-------------|-----------------|
| 30               | 633.69         | 0.7201        | 357.70      | 335.52         | 1.28        |
| 210              | 634.26         | 0.7207        | 351.24      | 324.79         | 9.02        |
| 530              | 637.75         | 0.7247        | 324.06      | 323.71         | 85.90       |

| $\dot{m}$ (mL/s) | $d_{tub} = 10$ mm | $Q_{net}$ (W) | $\eta_{th}$ | $T_{s,avg}$ (K) | $T_{out}$ (K) | $\Delta P$ (MPa) |
|------------------|--------------------|---------------|-------------|----------------|-------------|-----------------|
| 30               | 625.69             | 0.7110        | 384.04      | 335.36         | 0.04        |
| 210              | 626.45             | 0.7119        | 377.72      | 324.77         | 0.28        |
| 570              | 635.77             | 0.7225        | 325.86      | 323.66         | 1.88        |

| $\dot{m}$ (mL/s) | $d_{tub} = 25$ mm | $Q_{net}$ (W) | $\eta_{th}$ | $T_{s,avg}$ (K) | $T_{out}$ (K) | $\Delta P \times 10^{-3}$ (MPa) |
|------------------|--------------------|---------------|-------------|----------------|-------------|-----------------------------|
| 30               | 605.45             | 0.6880        | 458.18      | 334.96         | 0.41        |
| 210              | 606.39             | 0.6891        | 453.63      | 324.71         | 2.90        |
| 530              | 606.48             | 0.6892        | 453.17      | 323.68         | 7.32        |
Greek symbols

\(w\) concentrator wide, m

\(\alpha\) absorbance

\(\varepsilon\) emittance

\(\eta\) efficiency factor

\(\lambda\) thermal conductivity, W m\(^{-1}\) K\(^{-1}\)

\(\rho\) density, kg m\(^{-3}\)

\(\delta\) Stefan-Boltzmann constant, W m\(^{-2}\) K\(^{-4}\)

\(\theta\) cavity angels, °

\(\varphi\) rim angle, °

\(\nu\) kinematics viscosity, m\(^2\) s\(^{-1}\)

Subscripts

0 initial inlet to receiver

a air

ap aperture

cavity cavity

combined natural and forced convection

cond conduction

conv convection

ext external

f fluid parameter

forced forced convection

gc glass

int internal

inlet inlet

ins insulation

n tube section number

natural natural convection

net net quantity

PTC parabolic trough collector

r receiver part

rad radiation

s inner tube surface

sun sun outer surface

th thermal

total total

tube tube

\(\infty\) infinitive

Abbreviation

PTC parabolic trough concentrator

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