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Characterization of a Micro Thermal Cavity Receiver- Experimental and Analytical Investigation

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Abstract

The objective of this work is to characterize the optical and thermal performance of a micro-scale cylindrical cavity solar receiver for the Brayton gas power cycle at various solar radiation levels through experimental and analytical investigation. A thermal receiver consisting of a 300 mm deep and 200 mm diameter cylindrical cavity equipped with an 8 mm diameter helical copper tube was studied. An advanced ray-tracing technique using OptisWorks software was used to predict the distribution of solar radiation inside the cavity. Also, computational fluid dynamics simulations were carried out using ANSYS CFD software to predict the temperature distributions of the coil surface and the compressed air outlet temperature. Having satisfying conformity between the numerical and the experimental results, the current results demonstrated that a competent flux and temperature distributions were directed on the receiver's tube. Moreover, an outlet temperature up to 70°C, based on the available compressed air flow rate. This point leads to the probability of operating a micro-scale dish concentrator for Concentrated Solar Power in a domestic application. The results of a parametric study indicated that a cavity's receiver depth and width of 180 mm and 240 mm give the best thermal operation.

Keywords

Solar thermal dish, micro-scale, optical analysis, cavity receiver, thermal analysis

1. Introduction

Solar energy utilization is vital for producing clean energy and so to face important issues such as global warming [1]. Solar concentrating power is the most promising solar technology in order to produce high amounts of useful heat in medium and high temperatures [2-3]. Micro Scale Solar Powered Cycles comes to be one of the most important technologies for sustainable, reliable and clean electricity generation based on the reports of the Environmental Protection Agency (EPA 2013). To make this new technology commercially available and to add to its overall performance, the trend of this technology has recently attracted more awareness as a consequence of the increasing interest of global warming.

Recently, Bashir et al. [4] have developed a cylindrical cavity receiver with U-shaped tubes and phase change material inside it. The proposed receiver was designed to fit in a micro-scale solar-powered cycle. The results showed that the optimum tube number and dimensions can be achieved based on the amount of heat absorbed. Milind et al. [5] performed experimental work for a hemispherical cavity receiver using a 2 kW electrical source and water as a working fluid in order to assess the natural convection mode of heat losses for the cavity receiver of 0.54 m diameter and for various inclinations between 90° and 0°. The authors investigated the three types of heat losses, convection, conduction

and radiation. Different inlet temperatures, between 50°C to 75°C, were studied to find their effect on the heat losses. The convective mode of heat loss decreased with increasing the receiver inclination angle and that was the case for all temperature difference that was taken and that led to a decrease in the total heat losses. The Australian National University (ANU) has a clear contribution to this field [6-8]. For example, Hughes et al. [7] played on the technique of air flow inside and around the cavity receiver (by enhancing the receiver design) in order to decrease the both; the convection and radiation modes of heat losses using both; simulation and experimental analyses. The result showed that up to 50% reduction is applicable while the optimum design was not even reached. Similarly, some other interesting works [9-11] have been achieved by the (ETH) among them; the pressurized air receiver was reported in [10]. Reducing the forced convection heat losses for a cavity receiver using two different measures was a study conducted by Flesch et al. [12]. In that study, the researcher used a special type of curtain and partial window to cover about 33% of the receiver aperture and that contributed to about 50% enhancement. Measuring the forced convection for an open cylindrical cavity receiver and optimizing the cavity by considering various parameters like the receiver efficiency, lifetime and reliability for a solar tower were reported in [13-14] respectively.

Some interesting studies on the optical and thermal analysis both; experimentally and numerically at various operating conditions and working fluids, of a spiral cavity receiver with a parabolic dish concentrator were conducted by Pavlovic et al. [15-17]. The overall conclusion indicated that water is the best working fluid for low-temperature values due to its high heat transfer coefficient. Moreover, they highlighted the importance of using the correct type of material for manufacturing the concentrator the low-cost materials could lead to manufacturing errors, which causes a relatively low intercept factor. A cylindrical cavity receiver filled with helically coiled tubes with different values of coil height and diameter was analyzed by Kanatani et al. [18]. The authors concluded that the receiver efficiency decreased if its temperature was high and when the total incident energy was low for the receiver temperature. Moreover, they indicated that the ceiling absorptivity has a slight impact on the outlet temperature when the difference of the absorptivity values between infrared and visible light is insignificant.

In order to enhance the heat transfer between the receiver surfaces and the working fluid, Ngo et al. [19] developed a finned cavity receiver. The analysis focused on computing both; natural convection and radiation heat losses at various operating temperatures, surface emissivity, orientation and geometrical parameters of the receiver. The results indicated that using the finned cavity contributed to about a 20% decrease in natural convection loss with only around 5% of the radiation losses. Furthermore, fins shape and position have an impact on the heat loss amount.

The effect of the cavity receiver shape on the optical and thermal performance of a small scale solar powered Brayton cycle was investigated by Daabo et al. [20-21]. The results showed that the geometry, in terms of its shape, focal point position and other surface properties, has a significant effect on the overall Concentrated Solar Power CSP system efficiency. The effect of pitch value of the helical tube of the cavity receiver both; optically and thermally on the performance of three different geometries: cylindrical, spherical and conical receivers, coupled with a parabolic dish reflector was also evaluated by Daabo et al. [22]. Two various types of software were employed in their study and the results showed that the best pitch value is influenced by both the cavity configuration and the helical tube diameter. Using R141b and thermal oil were as two different types of the working fluid, the effect of the concentrator reflectivity, shape, optical and solar tracking errors on the performance of solar closed tube open cylindrical coated copper receiver were studied by Loni et al. [23]. Their results indicated that the receiver height has a vital role in attaining high overall system efficiency and lower inlet temperature for the thermal oil is also necessary.

Recently, interesting work on designing and optimizing a cylindrical cavity receiver by improving a thermal model for the cavity shape has been published by Zou et al. [24]. Their results showed the possibility of predicting the real performance and sensitivity analysis of the receiver geometry design. The effect of some geometric parameters on the amount of thermal efficiency and heat losses was also discussed. This paper provides a comprehensive description for simulating a cylindrical shaped cavity receiver by simultaneously integrating SolidWorks, OptisWorks and Fluent solver of ANSYS software. To validate this numerical simulation, an experimental work, including both; the optical and the thermal analysis has also been carried out. The results of this work can be used for the proper design of the cavity receivers in order to achieve high efficiency. The results can be extended also for greater scale systems which can be commercialized.

2. Thermal System

This section describes the design of the cavity receiver and the parabolic reflector. They together, as shown in Figure 1, represent the thermal cavity receiver system that receives, reflects and concentrates the received flux to supply the compressed airflow with the essential thermal energy. As a result, the optical and thermal analysis of the system has been carried out to identify the relevant amount of losses and then the development of the system performance.

Figure 1: (a): The cylindrical shape receiver including the helical tube and (b): The parabolic concentrator dish.

3. Numerical Simulation of the Receiver

OptisWorks software is commercial software that has been employed to simulate the optical performance of various solar concentrator structures using ray-tracing. This tool has been extensively employed by many studies, for example [25-28]. With its new features particularly the three-dimensional detectors, this software was able to accurately predict ray distribution in the cavity receiver and at the same time how the reflected irradiance is spread on the cavity walls.

In the current work, the cylindrical cavity receiver geometry as presented in Figure 2 is evaluated. The main goal of this step is to determine the irradiance obtained by the cavity receiver and simultaneously the amount of the lost flux that occurs in the cavity receiver. Consequently, the cylindrical receiver was modeled in OptisWorks to determine the quantity of received irradiance by the receiver aperture using a light source which acts as the sun. The optical properties of the parabolic dish like the reflectivity and the material type were identified. Then the detector, which records the quantity of incoming flux from the source and also the reflected irradiance to the receiver aperture area, is set. Similarly, the relevant material property of the cavity receiver as well as the focal point, which was determined by the parabolic concentrator's rim angle and diameter, was also marked. Figure 2 shows the parabolic dish combined with the cavity receiver both are under the ray-tracing analysis. Furthermore, Figure 3 shows a flow chart outing the steps followed in the OptisWorks simulation.

Figure 2: Ray tracing analysis achieved by OptisWorks (a): Parabolic dish together with the cavity receiver and (b): Cylindrical receiver (The different colors indicate different paths of the sun rays) [22].

Figure 3: Modeling process of parabolic concentrator and receiver using the ray-tracing technique.

4. Experimental Set up

The experimental work is aimed at evaluating optical and the thermal performance of the cylindrical shape cavity receiver. The detailed experimental results have been presented and parametrically compared with the numerical results of the relevant simulation through the designed models.

4.1 Optical Test Facility

The optical rig contained the following parts:

- 1) The thermal cavity receiver was selected to be a cylindrical shape.
- 2) The copper tube of an 8 mm diameter was helically assembled and inserted to shield the surface of the cylindrical shape internally.
- 3) The heat source was selected to be a Halogen bulb which was placed at different distances concerning the receiver aperture, Figure 4a.
- 4) Ten sensors (three of them were flux sensors) were spread out longitudinally and circumferentially on the helical tube surface inside the cylindrical shape receiver to measure the received flux and the temperature values, Figure 4b.

Figure 4: (a): The Halogen bulb light and (b): The seven surface sensors.

4.2 Optical Experiments:

The optical experimental work was conducted in the solar lab by employing the model and the instruments mentioned in section 5.1. That was established in order to determine the surface temperature and the flux distribution on the helical tube inside the cavity receiver. Experiments were carried out at various input irradiance of values i.e. 500, 600, 700, 800, 900 and 1000 W/m² obtained by varying the light source position.

4.3 Results of the Optical Analysis

4.3.1 Input Flux at Receiver Aperture

The input flux at the receiver aperture was evaluated by measuring the flux at 64 positions which were identified by plotting 64 equal squares on A4 paper placed on the receiver shown in Figure 5. In this experiment, the irradiance flux, coming from the halogen source, was varied to be from 500 W/m² to 1000 W/m² and the considered distance was between the receiver aperture and an imaginary plane located on the light source surface.

A flux sensor was manually put on each of the 64 (6.76 cm²) squares drawn on the A4 paper and the measured flux values were compared with their pair's pattern in the simulation achieved using the OptisWorks software. Figures 6a, 6b, 6c, 6d and 6e present the flux, distribution over the receiver aperture at various distance values i.e. 7 cm, 5 cm, 3 cm, 1 cm and 0 cm between the electrical source and the aperture of the cavity receiver. Moreover, the difference between the analytical and experimental results is given in Figure 6f. It can be concluded from these figures that the input flux at the aperture is uniform. Also, the difference between the experimental and the simulation results for all investigated distance values is less than 10%. The simulation results are obtained from [25-26] representing a solar input to the parabolic dish as shown in Table (1).

Figure 7 presents the flux uniformity at the aperture at three distance values; of 18, 7 cm and 0cm, determined based on three uniformity correlations established in previous studies [29-31]. It is clear from the figure that the peak uniformity was at a distance of 7 cm. The highest investigated distances, 18 cm, came with a relatively good uniformity level while the zero distance had the lowest uniformity.

Table 1: The selected distances with their related analytical and experimental flux values.

Irradiation (W/m^2)	Simulation (W/m^2)	Experiment (W/m^2)	Distance (m)
1050	5359	5599	0.000
1000	5093	5235	0.010
800	4056	3967	0.035
600	3059	3165	0.050
400	2102	1935	0.070

Figure 5: The experimental and simulation optical analysis carried.

Figure 6: The flux distribution at zero, 1 cm, 3.5 cm, 5 cm and 7 cm distance (a), (b), (c), (d) and (e) respectively, (f): is the average experimental and theoretical irradiance at the aperture.

Figure 7: The uniformity, using three uniformity laws, at the aperture for 0 cm, 7 cm and 18 cm distance.

4.3.2 Irradiance at Receiver Body

Figure 8 presents the model set up using the OptisWorks in order to evaluate the flux value and distribution on the helical tube surface as well as the 3D detector model of the real reflector. By contrast, the cylindrical cavity receiver at two various distances between the receiver and the source is produced in Figure 9.

Figure 8: All the models and the 3D detector for the two reflectors.

Figure 9: The flux distribution for the cylindrical receiver at two extreme distance values (a): 0 and (b): 18 cm.

Flux and temperature sensors were distributed, longitudinally and circumferentially, on the helical tube to evaluate both; the flux and the surface temperature along all the helical tubes. The average experimental and analytical flux values were compared and the results are shown in Figure 10.

Figure 10: The average flux values on the aperture at the three investigated height values.

Figures 11 shows the measured flux at various positions of the copper coil surface at various light source position. Point 1 refers to the deepest location inside the cylindrical receiver. It is clear that by decreasing the distance between the thermal receiver and the light source, the received flux increases. Also, it can be seen that the received flux increases with the coil positions where the tubes close to the aperture have the highest flux.

Figures 11 and 12 show the measured flux and temperature at various positions of the copper coil surface at various light source position. Point 1 refers to the deepest location inside the cylindrical receiver. For the sake of precision, more investigations were done in order to exactly evaluate the amount of flux at every single sensor's position at all investigated distance values and the results are displayed in the next few figures.

This was achieved on the model by introducing a model that is similar to the ruler and contains seven micro squares with the same size as the real sensor and inserted inside the cylindrical receiver. The comparison of the measured and simulated results are established and figured out in Figure 13 at three different distance values, 1, 5 and 7 cm, where the maximum percentage error values were noticed to be around 7%, 9% for (a) and (b) respectively and the highest at (c) with 15%.

Figure 11: The values of average flux along with the cavity receiver at various distances.

Figure 12: The temperature values along with the cavity receiver at various distances.

Figure 13: Numerical and experimental flux values at several gaps stuck between the receiver aperture and the source; 1 cm, 5 cm and 7 cm (a), (b) and (c) respectively.

5. Thermal Analysis

Thermal analysis was carried out on the receiver taking into account the three types of heat losses, convection, radiation and conduction as follows:

5.1 Heat Losses

The total heat losses and the heat balance are defined by equations (1 and 2) respectively.

$$Q_T = Q_{Cond} + Q_{Conv} + Q_{Rad} \quad (1)$$

$$Q_T = m \cdot c_p \cdot (T_{fin} - T_{fout}) \quad (2)$$

Whereas each; the conduction mode, convection mode and radiation, are defined in equations (3-5) respectively [32-33]:

$$\frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial Y^2} + \frac{\partial^2 T}{\partial Z^2} + \frac{g}{k} = \frac{1}{\sigma} \frac{\partial T}{\partial t} \quad (3)$$

$$Q_{Conv} = h \cdot A \quad (4)$$

$$\text{Where } h = \frac{k \cdot Nu}{L}$$

$$Q_{Rad} = A \cdot \varepsilon \cdot \sigma \cdot (T_{Sur}^4 - T_{Amb}^4) \quad (5)$$

Where (σ) is the Stefan-Boltzmann constant ($5.67 \cdot 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}$) and (ε) the emissivity respectively.

Computing the h based on the Nusselt number requires certain factors to be taken into consideration. These parameters are the model characteristic length, working fluid temperature, acceleration, as a result of the gravity, receiver wall surface temperature and fluid dynamic viscosity, specific heat and thermal conductivity.

In general, for laminar flow, Nusselt numbers can be calculated using correlations (6) and similarly, the turbulent free convection is computed using correlations (7) [34-35]:

$$Nu_{natural} = 0.508 \cdot \left(\frac{Pr}{0.952 + Pr} \right)^{1/4} Ra^{1/4} \quad (6)$$

$$Nu_{natural} = 0.15 \cdot \left(\frac{Pr^{9/16}}{0.671 + Pr^{9/16}} \right)^{16/27} Ra^{1/3}, \text{ where } Pr = \frac{\nu}{\alpha} \quad (7)$$

Many correlations are available for calculating Nusselt number, based on cavity receivers' applications, which are recommended by many researchers, according to their experimented models and boundary conditions. Some of the most familiar correlation for calculating Nusselt number are those found in reference [36], using equations (8-10).

$$Nu = 0.52 \cdot P(\varphi)^{1.75} \cdot l_c \cdot (Gr \cdot Pr)^{0.25} \quad (8)$$

Where:

$$P(\varphi) = (\cos \varphi)^{3.2} \quad \text{For } 0^\circ \leq \varphi \leq 45^\circ \quad (9)$$

And

$$P(\varphi) = 707 \cdot (\cos \varphi)^{2.2} \quad \text{For } 45^\circ \leq \varphi \leq 90^\circ \quad (10)$$

$$l_c = \left(\frac{D_{ap}}{D_{cav}} \right) \quad (11)$$

Equation (8) was derived depending on the high temperatures of $550^\circ\text{C} \leq T_w \leq 900^\circ\text{C}$

An explicit model accounts for the merged effects of aperture size, operating temperature and tilt angles was derived in [37-39]:

$$Nu = 0.088 Gr^{1/3} (T_w/T_\infty)^{0.18} (\cos \varphi)^{2.47} (D_{ap}/L_c)^s \quad (12)$$

Where:

$$s = 1.12 - 0.98 \cdot (D_{ap}/L_c), 0^\circ \leq \varphi \leq 90^\circ \quad (13)$$

And L_c = average internal dimension of the cavity; $T_{prop} = T_\infty$

By contrast, the radiative mode of heat losses can be computed depending on the view factor, radiosity and the emissivity of the relevant shape. The receiver's inner surface is divided into a satisfying number of elements while each one of those elements has the same, or sometimes different, values of emissivity and temperature. In order to determine the radiosity of each element, equation (14) is commonly used:

$$\sigma \cdot T_{w,1}^4 = J_1 + \left(\frac{1 - \varepsilon_1}{\varepsilon_1} \right) \cdot \sum_{j=1}^N F_{i,j} \cdot (J_i - J_j) \quad (14)$$

Here F and J are the view factor and the radiosity respectively; while i and j can be any two surfaces inside the receiver. The next step is the presentation of the total radiative heat losses which are given in equations (15- 17) [38-40]:

$$Q_{Rad\ i} = \sum_{j=1}^N F_{i,j} \cdot (J_i - J_j) \quad (15)$$

$$Q_{Rad} = \varepsilon_{eff} \cdot \sigma \cdot A_{ap} \cdot (T_{wall}^4 - T_{amp}^4) \quad (16)$$

$$\varepsilon_{eff} = \frac{1}{\left(\frac{1-\varepsilon}{\varepsilon}\right) \frac{A_{ap}}{A_w}} \quad (17)$$

5.2 Numerical and CFD Modeling

The receiver geometry is then exported to the ANSYS CFD software [41] (see Figure 14) to determine the energy that the Heat Transfer Fluid HTF (compressed air) has absorbed. This can be obtained based on the principles of heat losses which are governed equations (18-20). The cavity receiver and filled with 8 mm diameter helical tube coil with pitches value equals to 0 mm have been extensively studied using the numerical solution in order to accurately obtain the amount of the absorbed energy. Moreover, Figure 15 shows the grid independence assessment for the selected element type with respect to the considered cavity surface temperature values.

Figure 14: (a): The cavity receiver model and its helical tube, as an air domain, modeled in ANSYS/ Fluent and (b): The receiver shape with the sensors' positions.

Figure 15: Mesh independence for the tube surface temperature.

The energy, continuity and Navier-Stokes momentum equations (18-20), shown below, were simultaneously used in order to govern the fluid flow in the curve tube [42-46].

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0 \quad (18)$$

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_i}(\rho u_i u_j) = \frac{\partial}{\partial x_j} \left[-\rho \delta_{ij} + \mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + \rho g_i \quad (19)$$

$$\frac{\partial}{\partial t}(\rho C_p T) + \frac{\partial}{\partial x_i}(\rho u_i C_p T) - \frac{\partial}{\partial x_i} \left(\lambda \frac{\partial T}{\partial x_i} \right) = S_T \quad (20)$$

The tetrahedral type of element was employed with a smooth medium size; the essential quality of mesh was achieved [47]. In the modeling, several assumptions were made as below:

- 1) The compressed air is the working fluid that was chosen as HTF and the solar irradiance is distributed uniformly on the internal surfaces filled in the helical tube of the cavity receiver [48]. The air inlet temperature was 24°C and the mass flow rate was varied between 0.009 and 0.02 kg/s.
- 2) Insulation material was added to shield the external walls of the cavity receivers [47] with the aim of reducing the conductive heat losses.
- 3) The outlet condition was kept to be at atmospheric pressure. Moreover, as the natural convection heat loss was only evaluated ignoring the wind influence [47].
- 4) The model was at 90° only since this orientation achieves the lowest amount of convective heat losses [47-48]. Moreover, this position does not affect the optical analyses which were achieved in a position of -90° [49].

As for the solution type, the surface to surface, S2S, radiation model which necessitates the calculation of the view-factors in order to calculate the radiation type of heat losses [49-50]. Also, the steady-state analysis was utilized and the three types of heat losses were figured out. The solution was converged when the scaled-residuals were less than 10^{-4} for the velocity and the continuity equations. On the other hand, in order to attain more precise results, the residual of the energy equation was fixed to 10^{-7} [51-52].

5.3 Results of the Thermal Analysis

Figure 16 is a picture of the test rig that was built to evaluate the cavity receiver's thermal behavior at different boundary conditions. The test rig consists of the cylindrical cavity receiver positioned

vertically above the light source. The receiver is fitted with 8 mm copper tube helical coil with zero spacing between the tube's six surfaces thermocouples (type K) are placed at various positions on the surface of the coil. The air circuit includes a pressure gauge to measure inlet pressure, flow meter to measure the air volume flow rate, control valve to the air mass flow rate, thermocouples probes at inlet and outlet from the receiver coil and differential pressure transducer to measure the pressure drop.

Initially, the light source was switched on and the coil surfaces temperatures were monitored till reach to steady-state, without air flow. Then the compressed air delivered to the thermal system, at three flow rates, to evaluate its outlet temperatures. The exit air temperature and the surface temperature values were recorded until they reach steady-state conditions. The experiment was repeated with 7 cm, 3.5 cm and 0 cm distance between the receiver aperture and the source.

Figures 17a, 17b and 17c show the temperature distribution with time at various flow rates and spacing between the light source and the receiver aperture of 0, 3.5 and 7 cm respectively. The main outcomes from Figure 17 are given below:

- As the distance between the light source and receiver increased the surface and the exit air temperature values decreased. The main reason for that is the high energy scattered outside the aperture and as a result, not all the incoming energy from the source will be received by the receiver.
- The tube surface temperatures increased dramatically when there was no air supplied to the system and the transient state remained for about two hours while the maximum temperature achieved was strongly influenced by the distance between the light source and receiver.
- Once the air was supplied to the system, the surface temperature was decreased and at the same time, the exit air temperature was significantly enlarged and was continuing for around 7 minutes when both of the mentioned temperature values reached almost the steady values.

Finally, with decreasing the mass flow rate to the system, both; the surface and the exit air temperature increased reaching the steady-state and achieve maximum values.

Figure 16: The thermal cavity receiver test facility.

Figure 17: The compressed air exit temperature and the surface temperature values without supplying the compressed air and when it was supplied at several values and at several gaps from the light source: 0 cm, 3.5 cm and 7 cm (a), (b) and (c) respectively.

6. Experimental Evaluation for the Thermal model

The 3D model with its shape and dimensions was firstly drawn using SolidWorks and then exported to the fluent/ ANSYS in order to evaluate and validate the achieved thermal outcomes, in terms of the fluid outlet temperature. Similar mass flow rate and irradiance values to those obtained experimentally were used in the modeling and the predicted values were compared with the values attained experimentally. Figure 18 shows how the predicted temperature distribution over the helical tube of the cavity model. The temperature of the compressed air starts at 297 K; then it increased through each turn of the helical tube because of the radiated heat flux, to achieve its highest value in the last part of the tube length, receiver bottom. At this instant, it is essential to state that only the extreme levels (0.01 and 0.02 kg/s) of the investigated cases have been presented in this figure. Similarly, Figures 18 shows the temperature distribution for the compressed air at the extreme investigated levels of both; the flow rate values and the gap between the light source and the thermal cavity receiver.

Figure 18: The distribution of temperature, using fluent/ ANSYS, over the cavity receiver at; (a): 0.02 kg/s and (b): 0.01 kg/s and smallest possible gap.

Figures 19a, b and c displayed the measured surface temperature for seven positions in terms of the cavity depth, when the compressed air was supplied to the system and the steady-state was achieved, at four supplied values of mass flow rate and three different distance values. It can be seen that the temperature difference along the depth was low and became lower when the flow rate values were increased. The highest temperature of the tube surface was at the bottom of the receiver where heat is gathered and trapped (because of the lower density of the heated air domain inside the receiver cavity). By contrast, the lowest temperature values were noticed to be at the upper position in the vicinity of the receiver aperture which is open to the atmosphere.

Figure 20a shows the predicted temperature distribution on the surface of the helical coil while Figure 20b compared these predicted temperatures with the measured values. It is clear from Figure 20b that the predicted results are in good agreement with the experimental values with a maximum difference of around 13%.

Figure 19: The sensors' positions using a schematic figure for the receiver and the surface temperature values at several helical tube positions captured at different periods (With flow) at several distances between the cylindrical receiver and the light source; (a): 1 cm, (b): 3.5 cm and (c): 7 cm.

Figure 20: (a): The helical tube surface temperature at the seventh sensors' positions and (b): the numerical and the experimental results at several compressed air flow rates and 1 cm distances between the cylindrical receiver and the light source.

In Figure 21a the maximum experimentally achieved temperature for the exit was compared with their equivalent from the simulation, at different flow rates and minimum gap between the receiver and the electrical source. It is clear that the temperature varied from around 40°C to 75°C when the compressed air flow rate was reduced from 0.02 to 0.01 kg/s. Similarly, the two types of results for the compressed air outlet temperature values at several amounts of airflow values and 7 cm and 3.5 cm gap values between the receiver aperture and the light source are respectively presented in Figures 21b and Figure 21c. Again the highest air temperature was at a minimum compressed air flow rate and the lower gap between the receiver and source because of the amount of the concentrated flux which reaches the tube surface. This fact is normal, as the difference between the two results is with the acceptance limit of 20%. One of the main reasons for this difference could be the difficulty of having the same level of the supplied compressed air from the storage tank which obviously affects the output condition from the receiver like the number of heat losses and as a result the outlet air

temperature. At the moment, it is important to draw attention to the differences between the two results' values, got from the experimental and the simulation, which was increased with increasing the gap distance. The main reasons behind this might be the just mentioned above or even could be the high values of accumulated losses, the optical and the thermal, which might have not been accounted in a very precise way.

Figure 21: The numerical and the experimental temperature at different compressed air flow rates and 1 cm, 3.5 cm and 7 cm distances, (a), (b) and (c), respectively between the cylindrical aperture and the light source.

7. Receiver Aspect Ratio

In this section, the validated numerical model was used to investigate the effect of receiver Aspect Ratio AR, which is defined as the ratio of the cylinder cavity length to its overall width. The investigation is dependent on the amount of the natural convection, which has been represented by (h) and translated by using the amount of received flux inside the receiver cavity. In this analysis, seven different values for the aperture diameter with six various values for the cavity depth were considered with the aim of finding their effect on the thermal performance and eventually the best shape of the cavity (based on its thermal behavior). This analysis has been achieved using a parametric study that was established using ANYS/Fluent when more than 45 separate design points were evaluated. Figure 22, shows the effect of various values of AR, in terms of the cavity depth and width, on the heat transfer coefficient, which reflects the amount of the natural convection heat losses. As it is shown in this figure, there is no constant path or specific (direct) curve that can produce the relationships that connect the three parameters. Having said that, it can be seen that for fixed AR, the convective heat transfer coefficient increases with the decrease of the receiver diameter. Also, for the same diameter, the convective heat transfer coefficient decreases with the increase of AR. This could be attributed to the low cavity depth which can be achieved at high AR which results in a micro area or space that can be used as a trapped for the heat inside the receiver cavity.

Figure 23 shows the best two design points or models (with various AR) in terms of the amount of heat flux in the internal surface of the cavity receiver where most of the cavity has been covered with flux. It is clear from this figure that each cavity width and its depth are the difference and despite that those two configurations showed an excellent thermal performance. By contrast, in Figure 24 two bad design points were presented which were also at different values of the AR at the same boundary conditions. It can be noticed that most of the heat was lost to the surroundings which were represented by the air domain of the enclosure.

Figure 22: The coefficient of heat transfer at different values of AR and Receiver diameter of; (a) 180: mm, (b) 200 mm, (c): 220mm, (d) 260 mm, (e): 280 mm and finally (f): 300 mm.

(a)

(b)

Figure 23: The heat flux on the internal surface (a): at 180 mm with 0.6 AR and (b): at 200 mm and 1 AR.

(a)

(b)

Figure 24: The heat flux on the internal surface (a): at 300 mm with 1 AR and (b): at 300 mm and 1.37 AR.

8. Conclusions

This paper is an experimental and theoretical study for a micro-scale cavity receiver. This paper included both; the optical and thermal studies and the achieved results were confirmed by using the relevant model where two different types of software were employed. Moreover, a 3D model was developed and used to simulate the experimental work in terms of both; the optical and the thermal performance and the achieved results were in good agreement. After that, the influence of the receiver aspect ratio AR was investigated with the aim of knowing its overall effect on the receiver performance.

The micro-scale parabolic dish for CSP was successfully replaced with electrical light, halogen, a source with its frame and micro-scale cylindrical cavity receiver, which was designed and manufactured with its copper helical tube. So, a steel cylindrical receiver and a copper helical tube were assembled and experimented in a solar lab per the objective of figuring out its behavior in terms of temperature and flux distributions as well as the exit air temperature which was circulated over the receiver as a working fluid.

The Aspect Ratio, at different aperture diameter values of the thermal receiver, has a significant role in the overall thermal performance, represented by the heat transfer coefficient. Moreover, the following conclusions can be summarized below:

- Each cavity depth and its width have a direct influence on the thermal accomplishment of the thermal receiver.
- There is no constant path or specific (direct) curve that can produce the relationships that connect the three parameters.
- For each receiver aperture (width) there is an optimum value or sometimes values for the depth which can produce the lowest amount of heat transfer coefficient. It has to be said that the lowest heat transfer coefficient leads to the minimization of the cavity thermal losses and thus this is the goal of this analysis.
- The best three configurations for the cavity were at 180 mm width and 0.75 AR, 200 mm width and 1 AR and 220 mm and around 0.43 AR. On the other hand, the worst three configurations were found to be at 200 mm and about 0.91, 280 mm and around 1.08 and finally 300 mm and 1.5 AR.

Nomenclature

A	Area, m ²
A _{ap}	Aperture, m ²
c _p	Specific heat capacity, J/kgK
d	Diameter, m
F	View factor, -
g	Gravity, m/s ²
Gr	Grasshof number, -
h	Heat transfer coefficient, W/m ² K
J	Radiosity, -
k	Thermal conductivity, W/mK
m	Mass flow rate, kg/s
Nu	Nusselt number, -
Pr	Prandtl number, -
Q	Heat rate, W
u	Velocity, m/s
T	Temperature, K
x	Coordinate, m

Greek symbols

ε	Emittance, -
μ	Dynamic viscosity, kg/ms
φ	Inclination angle, °

Subscripts and Superscripts

A _p	Aperture
AR	Aspect ratio (=Width/Height)
Cav	Cavity
Cond	Conduction
Conv	Convection
i	Coordinate, inlet
f	Fluid
o	Outlet
Rad	Radiation

Abbreviations

AR	Aspect ratio
CFD	Computation fluid dynamics
CSP	Concentrating solar power

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References

- [1] A. Mehmood, A. Waqas, Z. Said, S. Mohammad A. Rahman, M. Akram, Performance evaluation of solar water heating system with heat pipe evacuated tubes provided with natural gas backup, *Energy Reports* 2019;5:1432-1444
- [2] Z. Said, M. Ghodbane, A.A. Hachicha, B. Boumeddane, Optical performance assessment of a small experimental prototype of linear Fresnel reflector, *Case Studies in Thermal Engineering* 2019;16:100541
- [3] A.A. Hachicha, B.A.A. Yousef, Z. Said, I. Rodríguez, A review study on the modeling of high-temperature solar thermal collector systems, *Renewable and Sustainable Energy Reviews* 2019;112:280-298
- [4] M.A. Bashir, A. Giovannelli, H. Muhammad Ali, Design of high-temperature solar receiver integrated with short-term thermal storage for Dish-Micro Gas Turbine systems, *Solar Energy* 2019;190:156-166.
- [5] M. Patil, R. Jahagirdar, E. Deore, Experimental investigation of heat loss from hemispherical solar concentrator receiver, *Frontiers in Heat and Mass Transfer (FHMT)* 2012;3:033008.
- [6] J. Pye, et al. "Development of a higher-efficiency tubular cavity receiver for direct steam generation on a dish concentrator." *AIP Conference Proceedings*. Vol. 1734. No. 1. AIP Publishing LLC, 2016.
- [7] G. Hughes, et al. "Reduction of convective losses in solar cavity receivers." *AIP Conference Proceedings*. Vol. 1734. No. 1. AIP Publishing LLC, 2016.
- [8] K. Tsuda, et al. "Development of high absorption, high durability coatings for solar receivers in CSP plants." *AIP Conference Proceedings*. Vol. 2033. No. 1. AIP Publishing LLC, 2018.
- [9] Casati, Emiliano, and Aldo Steinfeld. "Direct absorption of concentrated solar radiation by glass above 1000°C." *AIP Conference Proceedings*. Vol. 2126. No. 1. AIP Publishing LLC, 2019.
- [10] Poživil, Peter, et al. "A pressurized air receiver for solar-driven gas turbines." *Energy Procedia* 49 (2014): 498-503.
- [11] Hischier, Illias, et al. "Heat transfer analysis of a novel pressurized air receiver for concentrated solar power via combined cycles." *Journal of Thermal Science and Engineering Applications* 1.4 (2009).
- [12] Flesch, Robert, et al. "Reducing the convective losses of cavity receivers." *AIP Conference Proceedings*. Vol. 1734. No. 1. AIP Publishing LLC, 2016.
- [13] Siegrist, Silvan, Hannes Stadler, and Bernhard Hoffschmidt. "Measurements of the forced convective heat loss from open cylindrical cavities of multi-MW scale solar central receiver systems." *AIP Conference Proceedings*. Vol. 2033. No. 1. AIP Publishing LLC, 2018.
- [14] Frantz, Cathy, Andreas Fritsch, and Ralf Uhlig. "ASTRID©—Advanced Solar Tubular Receiver Design: A powerful tool for receiver design and optimization." *AIP Conference Proceedings*. Vol. 1850. No. 1. AIP Publishing LLC, 2017.
- [15] S.R. Pavlović, V.P. Stefanović, S.H. Suljković, Optical modeling of a solar dish thermal concentrator based on square flat facets, *Thermal Science* 2014;18(3):989-998.
- [16] S. Pavlovic, E. Bellos, W.G. Le Roux, V. Stefanovic, C. Tzivanidis, Experimental investigation and parametric analysis of a solar thermal dish collector with spiral absorber, *Applied Thermal Engineering* 2017;121:126-135.
- [17] E. Bellos, E. Bousi, C. Tzivanidis, S. Pavlovic, Optical and thermal analysis of different cavity receiver designs for solar dish concentrators, *Energy Conversion and Management: X* 2019;2:100013.
- [18] K. Kanatani, T. Yamamoto, Y. Tamaura, H. Kikura, A model of a solar cavity receiver with coiled tubes, *Solar Energy* 2017;153:249-261.
- [19] L. Ngo, T. Bello-Ochende, J.P. Meyer, Numerical modelling and optimisation of natural convection heat loss suppression in a solar cavity receiver with plate fins, *Renewable Energy* 2015;74:95-105.
- [20] A.M. Daabo, S. Mahmoud, R. K. Al-Dadah, The effect of receiver geometry on the optical performance of a small-scale solar cavity receiver for parabolic dish applications, *Energy* 2016;114:513-525.
- [21] A.M. Daabo, S. Mahmoud, R.K. Al-Dadah, The optical efficiency of three different geometries of a small scale cavity receiver for concentrated solar applications, *Applied Energy* 2016;179:1081-1096.
- [22] A.M. Daabo, S. Mahmoud, R.K. Al-Dadah, A. Ahmad, Numerical investigation of pitch value on thermal performance of solar receiver for solar powered Brayton cycle application, *Energy* 2017;119:523-539.

- [23] R. Loni, A.B. Kasaeian, E. Askari Asli-Ardeh, B. Ghobadian, Optimizing the efficiency of a solar receiver with tubular cylindrical cavity for a solar-powered organic Rankine cycle, *Energy* 2016;112:1259-1272.
- [24] C. Zou, Y. Zhang, Q. Falcoz, P. Neveu, C. Zhang, W. Shu, S. Huang, Design and optimization of a high-temperature cavity receiver for a solar energy cascade utilization system, *Renewable Energy* 2017;103:478-489.
- [25] A. Daabo, S. Mahmoud, R. AL-Dadah, Effect of open cavity configuration on solar receiver thermal performance, Conference: Heat Powered Cycles 2016 (Ext. Abstr.).
- [26] I.M.S. Ali, et al. "Optical performance evaluation of a 2-D and 3-D novel hyperboloid solar concentrator." World renewable energy congress XI. 2010.
- [27] L. Xiao, et al. "Optical-thermal conversion characteristics of cylindrical receiver with built-in helically coiled tubes." *Sustainable Energy Technologies and Assessments* 37 (2020): 100626.
- [28] S. Pavlovic, A.M. Daabo, E. Bellos, V. Stefanovic, S. Mahmoud, R.K. Al-Dadah, Experimental and numerical investigation on the optical and thermal performance of solar parabolic dish and corrugated spiral cavity receiver, *Journal of Cleaner Production* 2017;150:75-92.
- [29] F. Reis, Development of photovoltaic systems with concentration, (2013).
- [30] I. Moreno, Illumination uniformity assessment based on human vision, *Opt Lett* 2010;35(23):4030-4032.
- [31] A. Daabo, A. Ahmad, S. Mahmoud, and R. K. Al-Dadah, "Parametric analysis of small scale cavity receiver with optimum shape for solar powered closed Brayton cycle applications," *Applied Thermal Engineering*, vol. 122, pp. 626-641, 2017.
- [32] T.L. Bergman, F.P. Incropera, D.P. DeWitt, A.S. Lavine, Fundamentals of heat and mass transfer: John Wiley & Sons, 2011
- [33] A. Bejan, Heat Transfer John Wiley & Sons, New York, 1993.
- [34] E. Eckert, R. Drake Jr, Introduction to the Transfer of Heat and Mass McGraw-Hill, New York, New York, 1950.
- [35] S.W. Churchill, H.H. Chu, Correlating equations for laminar and turbulent free convection from a vertical plate, *International journal of heat and mass transfer* 1975;18:1323-1329.
- [36] A.A. Koenig, M. Marvin, M., Convection heat loss sensitivity in open cavity solar receivers, Final Report, DOE Contract No: EG77-C-04-3985, Department of Energy, USA, 1981.
- [37] W.B. Stine, C.G. McDonald, Cavity receiver convective heat loss. In: Proceedings of the International Solar Energy Society (ISES) Solar World Conference, Kobe, Japan, 1989.
- [38] W.M. Rohsenow, J.P. Hartnett, Y.I. Cho, Handbook of heat transfer vol. 3: McGraw-Hill New York, 1998.
- [39] R. Jilte, S. Kedare, J. Nayak, Natural convection and radiation heat loss from open cavities of different shapes and sizes used with dish concentrator, *Mechanical Engineering Research* 2013;3:25.
- [40] Y. Wu, L. Wen, Solar receiver performance of point focusing collector system, American Society of Mechanical Engineers, Winter Annual Meeting, San Francisco, Calif., Dec. 10-15, 1978.
- [41] Ansys, ANSYS FLUENT theory guide, Knowl. Creat. Diffus. Util 2011;15317:724-746.
- [42] A.M. Daabo, A. Al Jubori, S. Mahmoud, R.K. Al-Dadah, Development of three-dimensional optimization of a small-scale radial turbine for solar powered Brayton cycle application, *Applied Thermal Engineering* 2017;111:718-733.
- [43] A. Daabo, S. Mahmood, R.K. Al-Dadah, Structural Analysis of Small Scale Radial Turbine for Solar Powered Brayton Cycle Application, ASME 2018 12th International Conference on Energy Sustainability collocated with the ASME 2018 Power Conference and the ASME 2018 Nuclear Forum. American Society of Mechanical Engineers;2018.
- [44] A.M. Daabo, S. Mahmoud, R.K. Al-Dadah, A.M. Al Jubori, A. Bhar Ennil, Numerical analysis of small scale axial and radial turbines for solar powered Brayton cycle application, *Applied Thermal Engineering* 2017;120:672-693.
- [45] O. Ansys, "Ansys CFX-Solver Theory Guide Ansys CFX Release 10.0. Ansys," ed: Inc, 2005.
- [46] L. Ngo, T. Bello-Ochende, J.P. Meyer, Three-dimensional analysis and numerical optimization of combined natural convection and radiation heat loss in solar cavity receiver with plate fins insert, *Energy Conversion and Management* 2015;101:757-766.
- [47] M. Prakash, Numerical study of natural convection heat loss from cylindrical solar cavity receivers, *ISRN Renewable Energy* 2014;2014:104686.

- [48] N.S. Kumar, K. Reddy, Numerical investigation of natural convection heat loss in modified cavity receiver for fuzzy focal solar dish concentrator, *Solar Energy* 2007;81:846-855
- [49] Z.-G. Shen, S.-Y. Wu, L. Xiao, Y. Qiu, K. Wang, Effect of aperture size on free convection and radiation heat transfer in isoflux upward-facing cylindrical cavities, *Experimental Thermal and Fluid Science* 2017;87:1-14.
- [50] U.U. Rehman, Heat transfer optimization of shell-and-tube heat exchanger through CFD studies, Master Thesis, Chalmers University of Technology, 2012.
- [51] A.M.A. Daabo, Stress analysis and aerodynamic optimization of small-scale compressed air turbines. Diss. University of Birmingham, 2018.
- [52] A.M. Daabo, Y.S. Al-Mola, A.Y. Al-Rawy, T. Lattimore, State of the art single-objective optimization of small scale cylindrical cavity receiver, *Sustainable Energy Technologies and Assessments* 2019;35:278-290.

Author statement

A.M. Daabo, M.A. Bashir, S. Mahmoud and R.K. Al-Dadah had contribution about the experimental investigation and the methodology.

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Declaration of interests

☒ The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

☐ The authors declare the following financial interests/personal relationships which may be considered as potential competing interests:

Highlights

Experimental investigation of a small scale cavity receiver is performed.

A numerical model is developed for the thermal analysis of the receiver.

The operating working fluid is compressed air.

The aspect ratio of the cavity is an important parameter of this work.

The thermally optimum cavity depth is 180 mm and width 240 mm.