A comparative Thermal Analysis of conventional parabolic receiver tube and Cavity model tube in a Solar Parabolic Concentrator

S Arumugam1*, P Ramakrishna2 and S Sangavi3
1, 2 & 3Department of Mechanical Engineering, Sri Chandrasekharendra Saraswathi Viswa Mahavidyalaya, Enathur, Kanchipuram-631561, Tamilnadu, India

E-mail: 2ramakrishna.penumarthi@gmail.com; 3sangavisudhagaran@gmail.com
*Corresponding Author: aru_amace@yahoo.co.in

Abstract. Improvements in heating technology with solar energy is gaining focus, especially solar parabolic collectors. Solar heating in conventional parabolic collectors is done with the help of radiation concentration on receiver tubes. Conventional receiver tubes are open to atmosphere and loose heat by ambient air currents. In order to reduce the convection losses and also to improve the aperture area, we designed a tube with cavity. This study is a comparative performance behaviour of conventional tube and cavity model tube. The performance formulae were derived for the cavity model based on conventional model. Reduction in overall heat loss coefficient was observed for cavity model, though collector heat removal factor and collector efficiency were nearly same for both models. Improvement in efficiency was also observed in the cavity model’s performance. The approach towards the design of a cavity model tube as the receiver tube in solar parabolic collectors gave improved results and proved as a good consideration.

1. Introduction
Interests in the field of renewable energy are growing, especially in the field of solar applications. The major advantage is solar energy is spread everywhere and available freely. Solar Parabolic Collectors are one of the most promising technologies to substitute the fossil fuels for power generation and also used for industrial process heat, desalination, air-conditioning, refrigeration, chemistry production, and irrigation. Parabolic collectors reflect the sun’s radiation to the focal point, from that point the energy is transferred to a heat transfer fluid. Thermal performance of a parabolic collector depends on the solar radiation. Due to the inevitable interruptions by the atmosphere and varying solar radiation, output from the receiver gets effected. The errors created by these should be eliminated to find out the proper estimation of performance of the collector. Xu et al [1] used differential control volume methodology based on the time intervals of the data acquisition system, to model the solar irradiance in each volume. During the estimation of theoretical and practical performance there were considerable transient variations, even 100% instantaneous efficiency was observed. The effective solar radiation with the characteristic heat transfer fluid flow time matching involving the velocity of
the heat transfer fluid and the length of the parabolic trough solar collector row can solve the difficulties brought by the fluctuating solar irradiance. The results were predicted for the performance of the solar collectors by a valid selection of characteristic flow time and transient thermal performance was estimated. Due to temperature constraints on usage of liquid heat transfer fluid, Munoz-Anton et al [2] worked on usage of gas as heat transfer fluid which also had the advantage in solar power plants where the heat transfer liquid must work below 400°C. This limitation affects the performance of power plant due to poor Rankine cycle yield. Whereas gases provide the advantage of non-flammability and no environmental problems and can work at higher temperatures. Implementing gas as heat transfer fluid, during the first phase of usage the operating temperature reached was 400°C, which proved its feasibility for its usage in conventional purpose. But, the lower volumetric heat capacity and convective heat transfer of air compared to heat transfer fluid, require receivers with larger tube diameter and higher heat transfer area than those found in conventional receivers of solar parabolic trough collectors. Roman Bader et al [3] worked on a tubular cavity-receiver design with an increased tube diameter and higher heat transfer area, along with a linear secondary concentrator. A cavity-receiver reached four-fold solar concentration ratio compared to conventional solar receiver. Apart from industrial heating needs, solar energy could also be used for desalination purposes. There is an increase in water demand, and consequent reductions in ground water level, which can be balanced by desalination and wastewater treatment technologies. Arunkumar et al [4] worked on implementation of solar stills for desalination purposes. The production of de-saline water increases on increase in temperature of water and reduces if the glass temperature increases, and the single slope still was slightly effective than the pyramidal structure. Integration with different systems give augmented yields, but the cost increases. In compound, parabolic concentrators (CPCs) also improvements are being made with the studies on CPC with polygonal apertures. Studies done by Thomas Cooper et al [5] suggest that the performance of polygonal CPCs with reasonable numbers of sides was nearer to the revolved CPC. The polygonal CPCs have improved manufacturability than of revolving CPCs. Polygonal CPCs are less expensive alternatives to the revolved CPC for applications including stationary solar concentrators, and secondary concentrators for dish, tower, and trough primaries. Literature reviewed by the authors focused on the improvements in solar heating methods, in which receiver design modification was felt could find new space in this sector, in contemporary to the work done by various investigators[3, 8 & 9]. For this we designed a cavity model receiver for parabolic collectors which takes the help of vacuum for lowering the heat losses and the performance was compared with conventional parabolic collectors.

2. Methods

2.1. Experimentation

The conventional parabolic collector setup is the reference for this work. The conventional model has the receiver tube open to atmosphere and water is the heat transfer fluid used in this work. The details of the setup are listed in Table 1. A pump circulates the fluid through the receiver and also controls fluid flow. In present work, we maintained the fluid flow of 8.33×10⁻⁵ m³/s. The solar insolation from 11 AM to 4 PM was recorded. Simultaneously temperature of water at inlet, outlet of the receiver tube and in tank were also noted. The performance formulae for calculation are taken from Garg and Prakash [6].

2.2. Cavity Model Design

Cavity model is a modified conventional tube. The dimensions of the cavity model tube were given in Table 3 and is designed with the help of SOLIDWORKS software. It has a cavity with vacuum, covered by a glass plate. This glass plate allows solar radiation to heat the tube at cavity side. The external portion of the tube is enveloped by a glass enclosure with evacuation, the glass envelope allows the concentrated radiation to heat the outer side of the tube. The fluid is allowed to flow in a
concentric path in between the outer and inner tubes, which is enveloped both sides by vacuum. The pictorial representations of the cavity tube and its cross-sectional view is shown in Figure 1. The cavity model tube is designed by providing the same area of cross section for fluid flow as in the conventional model.

Table 1. Conventional Model and Material.

| Description                        | Value          |
|------------------------------------|----------------|
| Aperture of collector              | 1.2 m          |
| Inner diameter of absorber tube    | 0.043 m        |
| Tube thickness                     | 0.002 m        |
| Length of absorber tube and parabolic trough | 1.5 m        |
| Aperture area                      | 1.7325 m²      |
| Focal distance                     | 0.6 m          |
| Tube material                      | Copper         |
| Fluid                              | Water          |

Table 2. Properties of tube material.

| Description                        | Value |
|------------------------------------|-------|
| Reflectivity of concentrator       | 0.8   |
| Glass envelope transmissivity      | 0.8   |
| Absorber tube absorptivity        | 0.8   |
| Emissivity of absorber tube surface| 0.15  |
| Emissivity of glass               | 0.82  |

Table 3. Dimensions of Cavity model tube.

| Description                        | Value (m) |
|------------------------------------|-----------|
| Inner receiver tube diameter       | 0.043     |
| Outer receiver tube diameter       | 0.060811  |
| Glass envelope tube diameter       | 0.07      |
| Width of glass plate               | 0.01      |
| Length of the tube                 | 1.5       |

Figure 1. (a) Cavity Model Tube; (b) Cross-section of cavity model.
2.3. Performance Parameters and Formulae

Useful Energy ($Q_u$). The expression for the calculation of useful energy of the models are given in equations (1) and (2).

Conventional tube:

$$Q_u = A_F F R H - \frac{A_F}{A_p} \left( T_{f,i} - T_a \right)$$

(1)

Cavity model tube:

$$Q_u = A'_F F'_R H - \frac{A'_F}{A_p} \left( T_{f,i} - T_a \right)$$

(2)

Both the above expressions have the same parameters, but the parameters in the cavity model tube are subjected to some changes, as each term has to be found individually.

Collector heat removal factor ($F_R$). The expressions for collector heat removal factor for both the models are given in equations (3) and (4).

Conventional tube:

$$F_R = \frac{m C_f}{A F L_r} \left[ 1 - \exp \left( -\frac{A_F U L_F F'}{m C_f} \right) \right]$$

(3)

Cavity model tube:

$$F_R = \frac{m C_f}{A F L_r} \left[ 1 - \exp \left( -\frac{A'_F U L'_F F'}{m C_f} \right) \right]$$

(4)

Collector efficiency factor ($F$). The expression for collector efficiency factor is given in equations (5) and (6).

Conventional tube:

$$F = \frac{1}{U_L} \left( \frac{1}{U_L} + \frac{D_{r,o}}{h_{r,o} D_{r,i}} + \frac{D_{r,o} \ln \left( \frac{D_{r,o}}{D_{r,i}} \right)}{2K_r} \right)$$

(5)

Cavity model tube:

$$F' = \frac{1}{U_L} \left( \frac{1}{U_L} + \frac{A_{r,o}}{2\pi K_r L} + \frac{A_{r,o}}{h_{r,i} A_{r,i} + \frac{1}{h_{r,i}} + \frac{A_{r,o}}{2\pi K_r L}} \right)$$

(6)

Overall heat loss coefficient ($U_L$). The expression for finding the overall heat loss coefficient are given in equations (7) and (8).

Conventional tube:

$$U_L = \left[ \frac{1}{h_{c,v,r,a}} + \frac{1}{h_{r,d,r,a}} \right]^{-1}$$

(7)
Cavity model tube:

\[
U_L = \left( \frac{A_c}{A_g (h_{cv,g-a} + h_{rd,g-a})} + \frac{1}{h_{rd,r-g}} \right)^{-1}
\]  

(8)

The equations (7) and (8) are mainly dependent on convectional heat transfer coefficients \( h \). The heat transfer coefficients could be found from equations (9) to (13) for the respective models. As glass enclosure is present to the cavity model radiation losses inside the cavity were to be considered.

Conventional tube:

\[
h_{cv,a} = \frac{Nu_a K_a}{D_{r,o}}
\]

(9)

\[
h_{rd,a} = e_r \sigma (T_r + T_a) \left( T_r^2 + T_a^2 \right)
\]

(9.1)

For the calculation of Nusselt Number for convection by ambient air \( (Nu_a) \) formulae (9.2) is used. This equation (9.2) could also be used for calculating equation (10).

\[
Nu_a = \begin{cases} 
0.4 + 0.54Re^{0.52} & \text{for } 0.1 < Re < 1000, \\
0.3Re^{0.6} & \text{for } 1000 < Re < 50000.
\end{cases}
\]

(9.2)

\[
Re = \frac{\rho v D}{\mu}
\]

(10)

Cavity model tube:

\[
h_{cv,g-a} = \frac{Nu_g K_a}{D_g}
\]

(11)

\[
h_{rd,g-a} = e_g \sigma (T_g + T_a) \left( T_g^2 + T_a^2 \right)
\]

(12)

\[
h_{rd,r-g} = \frac{1}{e_r + \frac{A_r}{A_g} \left( \frac{1}{e_g} - 1 \right)}
\]

(13)

For the values of \( T_g \), heat conduction through the glass was taken into account by using the equation (2) of useful energy of cavity model. By the properties of glass form general material data and dimension of glass covering given in table 4, \( T_g \) was calculated by equation (13.1).

\[
Q_u = k_g A_g \left( \frac{T_g - T_a}{\delta} \right)
\]

(13.1)

**Efficiency of models \( (\eta_{th}) \).** The formulae for efficiencies are

Conventional tube:

\[
\eta_{th} = \eta_{op} - \frac{U_L (T_r - T_a)}{I_b C}
\]

(14)

Cavity model tube:

\[
\eta_{th} = \eta_{op} - \frac{U'_L (T_r - T_a)}{I_b C'}
\]

(15)
Both the models are dependent on common optical efficiency. They are also much dependent on concentration ratio.

### Table 4. Dimensions and Properties of Glass Enclosure.

| Property                              | Value   |
|---------------------------------------|---------|
| Thermal conductivity of glass ($k_g$) | 1.4 W/m K |
| Thickness of glass covering ($\delta$) | 0.00075 mm |
| Area of glass covering ($\mathcal{A}_g$) | 0.305 m² |

Optical efficiency ($\eta_{op}$). The optical efficiency depends upon optical properties of the materials involved [7].

$$\eta_{op} = \Gamma \rho \alpha$$

Concentration ratio ($C$). It is generally described as collector aperture area divided by the surface area of the receiver.

Conventional tube:

$$C = \frac{D - D_2}{\pi D_1}$$

Cavity model tube:

$$C' = \frac{D - D_2 + b}{\pi D_1 - b}$$

### 3. Results and Discussion

Different performance parameters were found out for both the models and the solar insolation values used were same for both the calculations. The temperature of water measured during experimentation at different points for the conventional model and other ambient conditions were also used for the cavity model calculations. Cavity model tube formulae were derived based on the conventional model formulae with the modifications according to dimensions and the provision of vacuum. All the performance calculations are carried out with the experimental data from the conventional parabolic collector and the plots were made for each of the parameters in terms of experimental time.

#### 3.1. Solar Insolation

Figure 2 shows the solar insolation as a function of experimentation time. It is the fundamental data that was collected during the test. The maximum radiation was showered plot shows that at 11:30 hours to 13:00 hours. The maximum radiation flux of 1217 W/m² was noticed at 12:30 hours. This data is the source for all the energy that could be used for the performance of the collector.
3.2. Heat Gained by Fluid

![Figure 3. Tube inlet, Outlet and Tank Water Temperatures as a function of experimentation time.](image)

Figure 3 provides the information about the heat gained by the fluid. The outlet temperature of the tube was high because of direct heating. Whereas for the tank water they mix with the outlet water, their temperature came down. During experimentation period, all the three temperatures increased in same manner. All the three temperatures increased to some extent and then its remains constant. By observing the solar insolation plot the insolation was less after 12:30 hours, and however, the temperatures are maintained constant by the system which implies that the retention level of the system was good and insulation worked properly.

3.3. Useful Energy

![Figure 4 (a). Useful energy plot of Conventional tube model](image)

![Figure 4 (b). Useful energy plot of Cavity tube model](image)

Figure 4 depicts that the plots of useful energy for both the models and they are of same trend, only a minute difference in the corresponding values of the models. The shape of the plot resembles that of solar insolation curve. As the solar insolation is the source of energy for heating, so does the useful energy values were resulted. Apart from the dimensional terms and solar radiation energy absorbed from equations (1) and (2), useful energy calculations are dependent on collector heat removal factor and overall heat loss coefficient.

3.3.1. Collector Heat Removal Factor. The collector heat removal factor, i.e., \( F_R \) and \( F'_R \) for both the models are 2.294 and 2.285 respectively. The calculated values are nearly same. This value is prominently dependent on flow rate of the fluid and overall heat transfer coefficient. The working fluid considered in this study is water and the flow rate was maintained constant at \( 8.33 \times 10^{-5} \text{ m}^3/\text{s} \). By observing equation (3), (4) the mass flow and specific heats are same, however, the dimensional term i.e., area is different and accordingly the specific heat capacity \( (C_f) \) is calculated and its corresponding value is \( 4.18 \times 10^3 \text{ kJ/kg K} \).
3.3.2. Collector Efficiency Factor. The collector efficiency factor, i.e., \( F \) and \( F' \) are 0.997 and 0.9938 respectively, for conventional tube and cavity tube model. The cavity tube has a concentric passage for a fluid flow, so the terms on either side should be considered. As conventional tube is a bare tube the equation (5) shows few terms and the cavity tube model has glass envelope and fluid flow between concentric tubes and hence the term in equation (6) for calculations are more. The collector efficiency factor is also dependent on overall heat loss coefficient.

3.4. Overall heat loss coefficient

Figure 5 shows the overall heat loss coefficient as a function of experimentation time. It is evident that the losses for the cavity model is always lower as compared to that of conventional tube model which is mainly due to the vacuum enclosure which reduced the convection heat losses. The pattern of both the curves is same due to both are calculated from same experimental data except the dimensional parameters. The heat transfer coefficient is needed for the calculations and are found out individually based on the fluid and surface material.

3.5. Efficiency of models

Figure 6 shows the thermal efficiencies of both models. During the course of time, the efficiency of both models is increased significantly. The efficiency of cavity tube model is slightly higher than the conventional tube model. During initial experimentation period, they maintained a difference is data pattern were observed and later which they were converging due to the lower solar insolation at this period. The efficiency also depends on optical efficiency, overall heat loss coefficient and concentration ratio. The concentration ratio for cavity tube model is higher than conventional tube model, this goes to the credit of the glass plate width (b). A 0.7 increment could be observed in cavity tube model than the conventional tube model. The overall heat loss coefficient is also less for cavity tube model. This helped for the increment in efficiency for cavity tube model.
3.5.1. Optical efficiency. The optical efficiency was calculated from the equation (16) and this value is 0.5350. From the equation (16) it is noticed that the maximum value for the theoretical efficiency that could be reached by both models is the optical efficiency. Whichever model has less losses and more concentration ratio performs with higher efficiency.

3.5.2. Concentration Ratio. From concentration ratio equations (17) and (18), $C'$ was found to be 8.868 and $C$ was 8.168. As $C'$ is greater than $C$, this was because the value glass plate width ($b=0.01$ m) is in addition in numerator of $C'$ and is being reduced in its denominator. This makes the value of $C'$ definitely higher than $C$, and the remaining terms are in common. From the above analysis, the cavity tube model showed a better performance as compared to conventional tube model. The general data for the models is same. The losses of the cavity tube model are less as compared to that of the conventional tube model, this has a major impact for its improvement over conventional tube model.

4. Conclusions

Cavity profile is designed in such a way that the cross-section area for fluid flow is same for both models i.e. conventional and cavity. The experimentation of conventional parabolic collector is done by taking water as heat transfer fluid and noting down the temperature changes of water. Based on the performance formulae of conventional model and design of cavity model the performance formulae of cavity model are derived. Comparing the performances, the useful energy for both models was similar as collector heat removal factor and collector efficiency factor were nearly same for both models. Overall heat loss coefficient is a very important parameter which was used in finding out most of other parameters. The comparison plot of overall heat loss coefficient for both models showed that, cavity model has less losses this could be attributed to vacuum, which reduces the convection losses. Though the optical efficiency of 0.53504 was same for both models, the efficiency of cavity model was more compared to conventional model. The concentration ratio for cavity model is 0.7 more than the conventional model, this is due to the opening over the cavity which augments the surface area for incoming solar radiation for heating. With theoretical estimations, the cavity tube model performance was comparable against conventional tube model. However, practical inevitabilities would definitely be there when cavity model tube is put into experimentation.

5. Appendices

| Symbol | Notation |
|--------|----------|
| $A$    | Area     |
| $T$    | Temperature |
| $H$    | Radiation energy absorbed by receiver |
| $I_i$  | Incident radiation per unit aperture area |
| $I_b$  | Solar insolation per unit area |
| $T$    | Transmittance |
| $\rho$ | Reflectivity, Density |
| $\nu$  | Velocity of ambient air |
| $\mu$  | Dynamic viscosity of air |
| $I$    | Intercept |
| $\alpha$ | Absorptivity |
| $\sigma$ | Emissivity |
| $\varepsilon$ | Stefan-Boltzmann constant (W/m$^2$K$^4$) |
| $k$    | Thermal conductivity (W/m K) |
| $D$    | Diameter |
| $l$    | Length of the receiver tube |
| $w$    | Width of the receiver tube |
| $b$    | Width of the glass plate of cavity tube |
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