SUPPRESSION OF NATURAL CONVECTION AND RADIATION HEAT LOSSES IN SOLAR CAVITY RECEIVERS: A NOVEL APPROACH

T. Bello-Ochende

Department of Mechanical Engineering Department, University of Cape Town, Private Bag X3, Rondebosch South Africa.
tunde.bello-ochende@uct.ac.za

Abstract:
The concentrating solar power technology has great potential to be used for energy production and it is a promising alternative to conventional fossil fuel-based energy technologies, such as coal power plants, due to the abundance of solar energy as an energy resource, as well as its minimal impact on the environment. The parabolic dish receiver assembly is one such promising concentrating solar power technology. It usually consists of a reflector in the form of a dish with a downward-facing receiver at the focus of the dish. A cavity receiver is used to maximise the absorption of the concentrated flux. However, the receiver is subjected to environmental variations, as well as changes in receiver inclination angle, which lead to heat losses that affect the overall receiver’s performance. The need for the commercialisation of economically viable parabolic dish systems necessitates further in-depth investigation into cavity receiver designs. As the cavity receiver plays a critical role in transferring solar heat to the engine, any heat loss from the cavity receiver can significantly reduce the efficiency and, consequently, the system’s cost effectiveness. It is therefore essential to assess and effectively minimise heat loss in the cavity receiver to improve the thermal performance of the system, which can contribute to the commercialisation of this type of technology. This novel approach of suppressing natural convection heat loss in a cavity receiver was investigated. The proposed model has not been observed in literature. A cavity receiver with plate fins attached to the inner aperture surface was investigated as a possible low-cost means of suppressing natural convection heat loss in a cavity receiver. Employing air as the working fluid, laminar natural convection heat transfer from the cavity receiver with plate fins attached to the inner aperture surface was investigated for a range of Rayleigh numbers, inclination angles, and fin heights and thicknesses.

Key words: Parabolic dish; Cavity receiver; Natural convection; plate fin; Raleigh number.

1. Introduction
The use of concentrating solar thermal technology has great potential for generating power. The parabolic dish-receiver assembly is one such promising system among several others. It normally consists of a reflector in the form of a parabolic dish with a downward facing receiver at the focus of the dish. These systems are continuously subjected to changes in environmental conditions such as wind, solar insolation, and ambient temperature. These environmental variations coupled with changes in receiver inclination angle affect the overall receiver performance leading to energy loss. A cavity receiver is preferred in such a system to maximize the absorption of the concentrated flux. The total energy loss of solar receivers which include conduction through the receiver insulation, convection and radiation through the aperture...
opening to the ambient environment plays a dominant role in the light-heat conversion. Heat loss through radiation heat is dependent on the temperature, emissivity/absorptivity and the shape factors of the receiver walls, while conduction heat loss is dependent on the receiver wall temperature and the material of the insulation used. Natural convection through the receiver aperture contributes a significant fraction of the energy loss and hence it is essential to effectively minimise it in order to improve the system efficiency.

Research of flow and heat transfer for cavity receivers can greatly help to estimate the thermal performance and optimise the design of the receivers [1-3]. With cavity receivers radiation and conduction can readily be determined analytically. However, this is not the case for natural convection. The complexity of geometry, temperature and velocity fields in and around the receiver makes it difficult to use existing analytical models for predicting convective heat loss. Therefore, many significant investigations have been conducted on natural convection heat transfer in open cavities. For instance, Le Quere et al. [4] investigated heat loss characteristics of two different sized cubical cavities. They considered variations in receiver operating temperature and angle, in their study. They found convection heat loss to be strongly dependent on the cavity inclination. Harris and Lenz [5] presented a study performed by Koenig and Marvin that empirically derived a correlation for convective heat loss from cylindrical cavity type receivers, including the effects of variation in operating temperature and angle. An analytical model for convective heat loss for an open cubical cavity receiver was presented by Clausing [6]. The Clausing model was developed for a central receiver operating at much higher temperatures.

Siangsukone and Lovegrove [7] presented work on modelling and simulation of the Australia National University (ANU) 400m² paraboloidal dish concentrator system with a direct steam generating cavity receiver and the steam line. Taumoefolau and Lovegrove [8] presented an experimental investigation based on an isothermal electrically heated model cavity receiver. Kumar and Reddy [9] presented a two-dimensional model to estimate the approximate natural convection heat loss from the modified cavity receiver of a fuzzy focal solar dish concentrator. Both insulation conditions and no insulation conditions are used for estimation of heat loss. The analysis of the receiver carried out was based on the assumption of the uniform and maximum solar flux distribution in the central plane of the receiver. Kumar and Reddy [10] performed a comparative study to predict the natural convection heat loss from the cavity, modified cavity and semi-cavity receivers. Reddy and Kumar [11] presented a numerical study of combined laminar natural convection and surface radiation heat transfer in a modified cavity receiver of solar parabolic dish collector. A comparison of 2-Dimensional and 3-Dimensional natural convection heat loss from a modified cavity receiver was carried out by Reddy and Kumar [12]. Prakash et al. [13] has reported experimental and numerical studies of the steady state convection heat losses occurring from a downward-facing cylindrical cavity receiver. From all the data points Nusselt number correlations as a function of receiver aperture diameter were proposed for the natural convection heat losses. Wu et al. [14] performed a three-dimensional numerical study of a heat-pipe receiver to investigate the influence of aperture size and position.
on natural convection heat loss taking into account the effects air property variation with temperature. Le Roux et al. [15] used the second law of thermodynamics to optimally size a modified cavity receiver under steady-state so that the parabolic dish system can have maximum net power.

Minimizing natural convection is seen as an effective method to improve the thermal efficiency of cavity receivers. Some investigations have been reported on the heat loss reduction of the cavity receiver. Kribus et al. [16] designed and demonstrated the operation of the multistage receiver under elevated temperatures, which divided the aperture into separate stages according to the irradiation distribution, to achieve high working temperature and thermal efficiency. Karni et al. [17] designed a volumetric solar receiver, nicknamed Porcupine. They demonstrated the capability of the Porcupine to endure concentrated solar flux of up to 4 MW/m2 and producing exist working fluid temperatures of up to 940 °C. Reddy and Kumar [18] analyzed the heat transfer behaviour of the modified cavity receiver with cone, CPC and trumpet reflector. The results showed that the trumpet one had the better performance than other ones. Fuqing Cui et al. [19] presented a cavity receiver with quartz glass cover for the dish concentrating system. They proposed the use of a quartz glass cover to separate the receiver cavity from the ambient air and its selective coating layer to intercept the infrared radiation emitted from the inner-surface of the cavity receiver, which greatly reduced the natural convection and surface radiation heat losses. Heat transfer rate through the enclosures can be controlled by means of fin’s configuration and literature is available [20-40].

This keynote presents a novel approach [47, 48] of suppressing natural convection heat loss in a cavity receiver. In the first case, a cavity receiver with plate fins attached to the inner aperture surface is presented as a possible low cost means of suppressing natural convection heat loss in a cavity receiver. This numerical study employs air as the working fluid, and laminar natural convection heat transfer from cavity receiver with plate fins attached to the inner aperture surface has been investigated for a range of Rayleigh number, inclination angle, fin height and thickness. In the second case a 3D numerical model of a modified cavity receiver with fin plates has been developed and investigated for combined laminar natural convection and surface radiation heat loss using the non-Boussinesq method. The effects of surface emissivity, operating temperature, orientation and the geometric parameters on the total heat loss from the receiver was investigated.

2. Physical and Mathematical model (Case one)

In this study, a three-dimensional cavity receiver is considered [47]. Figure 1 shows the schematics of the cavity receiver and the proposed cavity with plate fins in two-dimension. Compared with the prototype cavity receiver (Figure 1a), isothermal circular plate fins were installed on the inner side of the aperture surface on the proposed receiver (Figure 1c) to suppress natural convection. Both receivers are made of copper tubing with an opening aperture diameter \( d \) and cavity diameter \( D \). The diameter of the receiver and its aperture are defined as 180mm and 100mm respectively. The copper tubes are wound spirally to get the respective
shape of receiver. The outer surface of the cavity receiver is completely covered with opaque insulation. Some assumptions are made for modelling the cavity receiver: (a) there is uniform and maximum solar flux distribution in the cavity receiver (b) the surfaces of the tube are uniform and smooth for the prototype cavity receiver; (c) the plate fins are made of copper and installed on the inner side of the aperture surface between the copper tubing; (d) the temperature of air flowing through the copper tube is the same as the surface temperature of the tube. And to simply the model, the copper tubes were not considered in the simulation; (e) effects of wind are not considered (i.e. no wind case).

![Diagram](image)

(a)

(b)

(c)

Figure 1. Schematic of modified cavity receiver with and without plate fins [47, 48]

The idea in this work is aimed at reducing natural convection losses in the cavity receiver thus improving its performance. The numerical study employs air as the working fluid \((Pr = 0.71)\). The design variables which greatly affect the hydrodynamic performance of plate fins are the geometric parameters fin heights \((H_1, H_2, \text{ and } H_3)\) the fin thickness \((t_1, t_2 \text{ and } t_3)\) and the number of fins \(N\) as depicted in Figure 1b. The fin height and thickness are defined as dimensionless parameters \(H_j/W\) and \(t_j/W\) respectively in this study (for \(j = 1, 2, 3\)). Natural convection heat loss was estimated at different Raleigh numbers, inclination angles, plate fin
heights and thicknesses. The dimensionless fin heights were varied in the following range: \(0.025 \leq H_f/W \leq 0.15\). The dimensionless thicknesses of the fins were varied in the range: \(0.025 \leq t_f/W \leq 0.125\) and the value of \(N\) was varied from zero to three, where \(N = 0\) indicate no fin condition. In this study the effects of the distance between two plate fins \(S\) was not considered. The Raleigh Number was varied in the range: \(10^4 \leq Ra \leq 10^7\).

For natural convection in the cavity receiver, the flow and heat transfer simulations are based on the simultaneous solution of equations describing the conservation of mass, momentum and energy of the system.

Continuity equation:

\[ \nabla \cdot (\rho \mathbf{V}) = 0 \]  \hspace{1cm} (1)

Momentum equation:

\[ \mathbf{V} \cdot \nabla (\rho \mathbf{V}) = \rho \mathbf{X} - \nabla p + \nabla^2 (\mu \mathbf{V}) \]  \hspace{1cm} (2)

Energy equation:

\[ \mathbf{V} \cdot \nabla (\rho c_p T) = \nabla^2 (kT) \]  \hspace{1cm} (3)

where \(\rho\) is the density of air, \(\text{kg/m}^3\), \(\mathbf{V}\) is the velocity vector of air, \(\text{m/s}\), \(\mathbf{X}\) is the mass force vector, \(\text{N/kg}\), \(p\) is the pressure, \(\text{Pa}\), \(\mu\) is dynamic viscosity, \(\text{kg/(m.s)}\), \(c_p\) is the specific heat capacity at constant pressure, \(\text{J/(kg.K)}\), \(k\) is thermal conductivity of air, \(\text{W/(m.K)}\) and \(T\) is temperature in Kelvin.

The Boussinesq approximation which has been applied in previous numerical investigations for modified cavity receivers [9-12], lead to considerable deviations at high operating temperatures, and can no longer be applicable to such receivers, because the air properties change significantly with the remarkable operating temperature increments. Polynomial relationships for density, specific heat capacity at constant pressure, dynamic viscosity and thermal conductivity are used to account for air property variation with temperature [41].

\[ \rho = 7.4992 \times 10^3 T^3 + 1.6487 \times 10^5 T^2 - 1.2366 \times 10^7 T + 3.6508 \]  \hspace{1cm} (4)

\[ c_p = 1.3864 \times 10^{15} T^4 - 6.4747 \times 10^{14} T^3 + 1.0234 \times 10^{14} T^2 - 4.3282 \times 10^9 T + 1.0613 \]  \hspace{1cm} (5)

\[ \mu = 1.3864 \times 10^{15} T^2 - 1.4346 \times 10^{11} T^2 + 5.0523 \times 10^6 T + 4.1130 \times 10^{-6} \]  \hspace{1cm} (6)
The convective heat transfer coefficient can be expressed as:

\[ h_c = \frac{Nuk}{D} \quad (8) \]

where \( Nu \) is the Nusselt number and \( D \) is the receiver cavity diameter. The Nusselt numbers were calculated using CFD code. The convective heat loss from the modified cavity receiver is given as:

\[ Q_c = h_c A(T_r - T_a) \quad (9) \]

The Raleigh number for all results in this study is based on the cavity receiver diameter \( D \) and is defined as:

\[ Ra = \frac{g \beta (T_r - T_a) D^3}{\nu \alpha} \quad (10) \]

where \( g \) is the acceleration due to gravity, m/s², \( \beta \) is the thermal expansion coefficient, 1/K, \( \nu \) is the kinematic viscosity, m²/s, and \( \alpha \) is the thermal diffusivity, m²/s.

3. Numerical procedure and validation

3.1 Numerical procedure.

A finite volume based CFD code, STAR-CCM+ 7.06 was employed in the 3D simulation of the natural convection through the aperture of the cavity receiver. Figure 2 schematically represents computational grid of the cavity receiver. In reality, the receiver is surrounded by an infinite atmosphere with a limiting temperature equal to ambient air temperature. In the numerical analysis, the region outside the cavity is represented by a spherical enclosure (Figure 2a). The size of the enclosure was increased until it had an insignificant effect on fluid and heat flows in the vicinity of the receiver. It was found that in STARCCM+, the diameter of the spherical enclosure should be about ten times the diameter of the receiver to achieve this.

The core volume mesh contains polyhedral cells. The cells were refined on the walls of the cavity receiver as a percentage of base cell and prism layers cells were used on the walls of the cavity receiver (Fig. 2b). This led to cells being very small in the region inside the cavity and nearby the receiver but increase in size gradually toward the spherical enclosure wall. The prism layer mesh model was used with a core volume mesh to generate orthogonal prismatic cells next to wall boundaries. This layer of cells is necessary to improve the accuracy of the flow solution [42]. An enlarged portion of the mesh is shown in Figures 2b and 2c.
A mesh refinement was performed on the cavity receiver, investigating the average Nusselt number on the hot inner surfaces of the cavity receiver ($T_i = 800$ K and $Ra = 10^6$). Table 1 presents the average Nusselt numbers obtained for four different grids at two different inclination angles of the receiver ($\theta = 0^\circ$ and $\theta = 30^\circ$). The relative deviation for the Nusselt number between grid 1 and 2 was less than 1%. Since the differences between the two were minor, we chose grid 2 for all the simulations presented in this work. This was considered as a good trade-off between accuracy and cost of time.

| Cells    | Nusselt number $\theta = 0^\circ$ | Relative Deviation | Nusselt number $\theta = 30^\circ$ | Relative Deviation |
|----------|-----------------------------------|--------------------|-------------------------------------|--------------------|
| 801,337  | 10.1675                           |                    | 18.2941                             |                    |
| 320,000  | 10.1529                           | 0.001436           | 18.2734                             | 0.001132           |
| 241,336  | 10.2338                           | -0.00797           | 18.2008                             | 0.003973           |
| 218,272  | 10.2947                           | -0.00595           | 18.2096                             | -0.00048           |
3.2 Boundary Conditions.

The model “without insulation at the aperture plane” proposed by Reddy and Kumar [12] is used in this investigation. It is assumed that the surface of the cavity receiver attains constant or almost constant surface temperature when exposed to reflected solar radiation of the parabolic concentrator and as such isothermal boundary condition was applied to the internal receiver surfaces, outer surface on the aperture plane as well as the fins. The temperature stated in the current model is stagnation temperature which corresponds to the no flow condition when no useful energy is being collected for constant insolation and the system attains peak temperature [12]. The temperature was varied from 400-1000 K. The outer spherical walls of the receiver were treated as adiabatic since it was covered with insulation to prevent heat loss. The outer domain was treated as a pressure outlet boundary condition. The wall temperature of the entire spherical enclosure was set to an ambient temperature of 300K.

3.3 Validation

The numerical procedure was validated using the experimental work reported by Yasuaki et al. [43] as shown in Table 2. The enclosed hemisphere was experimentally studied under steady state, laminar conditions. The curved portion and bottom surface were taken as cold and hot surfaces respectively. The area weighted average Nusselt numbers of hot surface of the enclosed hemisphere was obtained for different Rayleigh numbers. It was observed that the present numerical procedure is in good agreement with the experimental data with maximum deviation approximately 2.8%.

| Rayleigh number | Nusselt number  | Percentage deviation |
|-----------------|-----------------|----------------------|
|                 | Yasuaki et al.  | Present work         |
| 1.9437×10^5     | 6.91            | 7.09                 |-2.6|
| 2.8523×10^6     | 7.60            | 7.39                 |2.8 |
| 4.9288×10^6     | 8.72            | 8.62                 |1.1 |
| 7.8266×10^6     | 9.78            | 9.86                 |-0.8|
| 1.1683×10^7     | 10.82           | 10.84                |-0.2|
| 2.2818×10^7     | 12.79           | 12.44                |2.7 |

4. Optimization problem formulation
The optimization problem was tailored towards finding the best plate fin geometric parameters, which give the least natural heat loss from the modified receiver cavity for different Raleigh number, and inclination angle at a given temperature. The design variables greatly affect the hydrodynamic performance of plate fins are the plate fin heights and thicknesses. The dimensionless fin heights were varied in the following range: $0.025 \leq H_f/W \leq 0.15$. The dimensionless thicknesses of the fins were varied in the range: $0.025 \leq t_f/W \leq 0.125$. The type of optimisation considered in this study is the placement of individual plate fins to form channels. This leads to the treatment of an array of fins in which each fin is operating in an optimum manner [44]. The solution process for an automated design optimization study is illustrated in Figure 3 [47, 48].

![Automated design optimization process flowchart](image)

**Figure 3.** Automated design optimization process flowchart [47, 48]

5. **Case one: Natural Convection heat loss suppression in a solar Cavity Receiver with plate fins**

In this section, the optimization SHERPA algorithm was applied to obtain the best geometric configuration of the plate fins that will optimally suppress natural convection heat loss in cavity
receiver. The optimal geometric parameters have significant influence on the performance of the receiver cavity as reducing natural convection will improve the performance of the cavity receiver. For each simulation, 30 optimization numerical evaluations were conducted automatically within the constraint ranges given in previous section and convergence was attained after approximately 14 hours. The results are presented in table 3.

Table 3. Optimization results [47]

| Variable | Fin Height/Thickness (m) |
|----------|--------------------------|
|          | Initial | Optimised |
| H1       | 0.005   | 0.006     |
| H2       | 0.005   | 0.006     |
| H3       | 0.005   | 0.006     |
| t1       | 0.003   | 0.005     |
| t2       | 0.003   | 0.005     |
| t3       | 0.003   | 0.005     |

Figure 4 shows optimized results of the cavity receiver plate fin heights at all inclination angles. The optimum results is plotted together with the effect of cavity receiver inclination angle on natural convection for the unfinned and finned aperture cases having 3 plate fins and $H_j/W = 0.075$ and 0.125 respectively. For the range of inclination angles, Figure 4 shows that natural convection heat loss is highly dependent on fin height and decreases with increasing plate fin height until the optimum is reached.
Figure 4. Variation of convection heat loss with receiver inclination angle including optimized results, $Ra = 10^6$

Figure 5 shows optimized results of the cavity receiver plate fins thicknesses at all inclination angles. The optimum results is plotted together with the effect of cavity receiver inclination angle on natural convection for the unfinned and finned aperture cases having 3 plate fins and $t_f/W = 0.075$ and 0.125 respectively. Figure 5 show that natural convection heat loss does not significantly depend on plate fin thickness and varying $t/W$ shows insignificant change in natural convection heat loss. However, we observe from the optimization study that natural convection suppression is achieved at optimum when $t_f/W = 0.125$. Thus, it can be concluded that plate fins geometric configuration in cavity receiver could be important for optimum performance of cavities in practical applications.

![Figure 5](image-url)

Figure 5. Variation of convection heat loss with receiver inclination angle optimized results, $Ra = 10^6$ [47]

Figure 6 shows the effects of the minimized natural convection heat loss $Q_{\text{min}}$ as function of the Raleigh number for the various receiver inclination angles. It was observed that minimized natural convection heat loss increases with the increase in Raleigh number for all receiver inclination angles. It was also observed that at lower Raleigh numbers ($10^4$ and $10^5$); the
variation of minimized natural convection heat loss at different inclination angles of the cavity receiver is lower as compared to that at high numbers ($10^6$ and $10^7$).

**Figure 6.** Effect of Raleigh number on minimized natural convection heat loss from the proposed modified cavity [47].

**Figure 7.** Effect of Raleigh number on optimized dimensionless plate fin heights and thickness [47].
6. Case Two: Combined natural convection and radiation heat loss in Solar Cavity Receiver with plate fins insert.

In the case of combined natural convection and radiation heat loss in Solar Cavity Receiver with plate fins insert, the numerical formulation, boundary conditions and validations are given in references [48] only the important results are presented in this section.

Figure 8 shows the effects of fins on combined natural convection and radiation heat loss from the cavity receiver. Combined natural convection and radiation heat loss is plotted for different inclination angles for the cavity receiver without fins and the one with fins at 800 K operating temperature. It is observed from Figure 8, that the inclusion of fins decreases both the natural convection and radiation heat loss from the cavity receiver. It is also observed in both cases that the radiative heat loss is more dominant and constant at all inclination angles. The decrease in convection heat transfer in the cavity receiver with plate fins is due to the decrease in the velocity of the fluid flowing through the cavities created by the insertion of plate fins. The fins increases the resistance of the convection cell movement between the plate fins thus further decreasing the convective heat loss. The conclusion is that the combined natural convection and radiation heat loss from the cavity receiver with fins was reduced compared to the case without fins. Convective heat loss is observed to have significantly reduced by approximately 20% at 0° and 10% at 90° with the introduction of fin plates. Radiation heat loss minimally reduced by 5% at all inclination angles.

![Figure 8](image.png)

**Figure 8** Variation of combined natural convection and radiation heat loss with receiver cavity inclination angle 800 K operating temperature [48].
Figure 9 shows the dependence of the combined natural convection and radiation heat loss on the number of plate fins on the aperture of the cavity receiver. Combined natural convection and radiation heat loss is plotted against the number of plate fins for different inclination angles, emissivity of 1.0 and fixed fin height and thickness. It is observed from Figure 9 that, there is a decrease in combined natural convection and radiation heat loss at all inclination angles of the cavity receiver, when the number of plate fins is increased. This decrease is due to the decrease in the flow intensity with the insertion of plate fins. Increasing the number of fins leads to weaker flow intensity due to the expected increased resistance of motion of the rotating convection cells. Weaker flow intensity lowers the heat transfer rate thus decreasing the natural convection heat loss and ultimately the total heat loss. The effect of surface emissivity on total heat loss from the cavity receiver was investigated for different inclination angles.

For a given Rayleigh number and temperature, the total heat loss varies linearly with emissivity for all inclination angles as seen from Figure 10.

![Figure 9](image-url)

**Figure 9** Variation of combined natural convection and radiation heat loss with number of fins [48]

There is a sharp increase in the total heat loss with increasing surface emissivity. This can be attributed to the fact that radiative heat loss increases with increasing emissivity in the receiver. Surface radiation is directly affected by the surface emissivity of the cavity receiver thus increasing the radiative heat loss and the total heat loss as a whole. The most heat loss occurs at 90° and 1.0 emissivity and the least total heat loss is estimated at 0° when the emissivity is 0.2. Based on this observation, it would therefore be important to use selective coating on the tubing and inner walls of the cavity receiver which would help reduce the radiative heat loss and ultimately the total heat loss.
To observe the effects of temperature on combined natural convection and radiation heat transfer, the temperature of the cavity receiver was varied from 800 K to 1200 K. For a given dimensionless fin height \((H/W = 0.125\text{m})\), fin thickness \((t/W = 0.075\text{m})\), emissivity \((1.0)\) and Rayleigh number \((1 \times 10^6)\), the temperature contours of the cavity receiver at a given inclination angle and surface temperatures are shown in Figures 11a. and 11b. It is observed from the cavity receiver at 800 K and at 60° inclination angle (Figure 11a) that, the thickness of the stagnation zone in the top part of the cavity is higher compared to that of the corresponding cavity at 1200 K (Figure 11b). It is seen in both cases that the upper parts of the walls have higher temperatures compared to the lower parts of the cavity receiver walls. As air at ambient temperature is driven into the cavity receiver by the natural convective currents, the air adjacent to the receiver surface heats up and becomes light after absorbing heat from the receiver surfaces and consequently flows up along the cavity wall. This results in stagnant hot air only appearing at the top of the cavity receiver. Eventually, the hot air exits the cavity through its aperture and thereafter cooled by the ambient. The temperature contours are distorted by the flow and the fluid penetrates through into the cavity. It is observed that the fluid penetrates more into the cavity at high temperature (Figure 11b) compared to the cavity at lower temperature (Figure 11a). It is also noted that the thickness of the hydrodynamic boundary layer adjacent to the hot wall decreases in both cases as the temperature increases.
Figure 11 Temperature contours for cavity receiver with three plate fins at 60 degrees for Ra = 10^6 with surface temperature at 800 K and 1200 K [48]

Figure 12 shows the variation of combined natural convection and radiation heat loss with temperature for a given emissivity, Rayleigh number at different inclination angles of the cavity receiver. It is observed from the graph that radiation heat loss varies greatly with increasing temperature but remains constant for all inclination angles while natural convection heat loss varies steadily with increasing temperature and also depends on the inclination angle of the receiver. It is also observed that natural convection is minimum at 90° and highest at 0°. Because of the higher radiation heat loss which is constant at all inclination angles, the total heat loss is also high and varies greatly with temperature. It is important therefore to reduce the temperature of the receiver surfaces to avoid large radiation and convection losses from these surfaces and this can be achieved by using selective coating with low emissivity for the cavity receiver wall surfaces.

In this section optimization algorithm SHERPA was used to get the best geometric shape of the plate fins that will optimally suppress combined natural convection and radiation heat loss in a cavity receiver. The optimal geometric parameters have significant influence on the performance of the receiver cavity as the reduction of total heat loss will improve the performance of the cavity receiver. For each simulation, 30 optimization numerical evaluations were conducted automatically within the constraint ranges given in the previous section and convergence was attained after approximately 14 hours. The results are presented in Table 4.
Figure 12 Variation of combined natural convection and radiation heat loss with temperature [48]

Table 4 Optimization results [48]

| Variable | Fin Height/Thickness (m) |
|----------|-------------------------|
|          | Initial | Optimized |
| H1       | 0.005   | 0.006     |
| H2       | 0.005   | 0.006     |
| H3       | 0.005   | 0.006     |
| t1       | 0.003   | 0.005     |
| t2       | 0.003   | 0.005     |
| t3       | 0.003   | 0.005     |

Figures 13 and 14 illustrate the change trends of overall cavity receiver efficiency when the optimum operating temperature varies from 800 K to 1200 K. The results show that the overall cavity receiver efficiency increases with increasing operating temperature. It is observed from Figure 13 that the overall cavity receiver efficiency varies marginally with increasing inclination angle. The cavity receiver at 90° is seen to have a slightly higher efficiency compared to when the cavity receiver is at 0°.
It is observed from Figure 14 that, the increase of overall cavity efficiency is marginally at approximately 2% with the insertion of fin plates although the convective heat loss decreases by 20%. This is due to the fact that radiation heat loss dominates at high operating temperatures compared to convective heat loss and this supports the conclusion by other researchers. From
Figure 14, we observe that the cavity overall efficiency increases with the introduction of fin plates and optimum geometry exits as the results show in Table 4 for maximum cavity receiver efficiency.

7. Conclusions

The geometric parameters examined were optimized using HEEDS search algorithm called SHERPA. The major finding is that significant reduction on the natural convection heat loss from the cavity receiver can be achieved by using the plate fins, which act as heat transfer suppressor and that optimal plate fin geometries exist for minimized natural convection heat loss. Reduction up to a maximum of 20% at 0° receiver inclination was observed. The results obtained provide a novel approach for improving the design of cavity receivers for optimal performance. In the second case the key conclusion is that substantial reduction in natural convection heat loss of about 20% from the cavity receiver was attained with the use of plate fins while radiation heat loss was marginally reduced by about 5%. The overall cavity efficiency marginally increased by approximately 2% with the insertion of fin plates although the convective heat loss was suppressed by about 20%. This is due to the fact that radiation heat loss dominates at high operating temperatures compared to convective heat loss and this supports the conclusion by other researchers that radiation heat loss dominates at higher temperatures. The results obtained in this study provide a novel approach for improving the design of cavity receivers for optimal performance.

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Nomenclature

| Alphabetical Symbols | Greek symbols |
|----------------------|--------------|
| D                    | ϕ            |
| d                    | ρ            |
| Cp                   | µ            |
| hc                   | ε            |
| H                    | H/W          |
| H/W                  | N            |
| K                    | Kg           |
| J                    | m            |
| k                    | N             |
| kg                   | Nu           |
| m                    | P             |
| N                    | Pr           |
| Pa                   | Q             |
| Q                    | Ra           |

| Subscript            | Abbreviation |
|----------------------|--------------|
| c                    | ANU          |
| s                    | CFD          |
| ∞                    | CPC          |

Alphabetical Symbols:
- Cavity diameter, m
- Aperture diameter, m
- Specific heat capacity, J/(kg.K)
- Convective heat transfer coefficient, W/(m²K)
- Plate fin height, m
- Dimensionless plate fin height
- Joules
- Thermal conductivity, W/(m K)
- Kilogram
- Metre
- Number of fins
- Nusselt number
- Pressure, Pa
- Prandtl number
- Pascal
- Heat loss, W

Greek symbols:
- Receiver inclination angle, degree
- Density of air, kg/m³
- Dynamic viscosity, kg/(m.s)
- Fin effectiveness
- Convection
- Surface
- Ambient

Abbreviation:
- ANU: Australia National University
- CFD: Computational Fluid Dynamics
- CPC: Compound parabolic concentrator
S

Fin space, m

 t

Plate fin thickness, m

 t/W

Dimensionless plate fin thickness

 T

Temperature, K

 T_s

Surface temperature, K

 V

Velocity vector, m/s

 W

Watt

X

Mass force vector, N/kg

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