

# Simulation of a linear Fresnel solar collector concentrator

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## Abstract

A trapezoidal cavity receiver for a linear Fresnel solar collector is analysed and optimized via ray-trace and computational fluid dynamics (CFD) simulations. The number of receiver absorber tubes and the inclination of lateral walls in the cavity are checked with simplified ray-trace simulation. The CFD simulation makes possible to optimize cavity depth and rock wool insulation thickness. The simulated global heat transfer coefficient, based on primary mirror area, is correlated with a power-law fit instead of a parabolic fit. The correlation results are compared with heat transfer coefficients available for linear Fresnel collector prototypes.

*Keywords:* concentrating solar collector; linear Fresnel; ray-trace; CFD; simulation

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## 1 INTRODUCTION

When looking at reducing CO<sub>2</sub> emissions, the greatest task in creating viable solar energy conversion systems is that of reducing system cost. The solution to this problem does not necessarily lie on creating the most efficient system, but more on the development of a system that has the lowest lifetime cost per unit of electricity converted from solar energy. A linear Fresnel solar collector concentrator may have a lower efficiency than the other concentrating geometries, but the likely reduced cost may well compensate that, providing a solution for cost-effective solar energy collection on a large scale [1]. The linear Fresnel collector concept uses a number of rows of relatively small one-axis tracking mirrors that concentrate the radiation on a linear receiver.

The advantages of linear concentrating Fresnel collectors include relatively simple construction, low wind loads, a stationary receiver and high ground usage [2].

In this paper, the optical and thermal performances of a new trapezoidal cavity for a small linear Fresnel receiver are analysed, using simplified ray-tracing and computational fluid dynamics (CFD). Natural convection inside the cavity, thermal radiation between surfaces and conduction through the walls are simulated, and the overall heat loss coefficient is evaluated. The system uses 10 rows of 4 reflector mirrors with a north-south tracking axis (see Figure 1). The primary mirrors are cylindrical with different small curvatures. The mirror width is 0.4 m, the length is 3 m and mirror spacing is 0.15 m. The total mirror area is approximately 48 m<sup>2</sup>. The linear receiver is composed of six pipes with an inside diameter of 0.5 in.,

placed 2.5 m above the mirrors, as shown in Figure 1, inside an insulated trapezoidal cavity.

The maximum temperature achieved in the receiver tubes was fixed through the operating temperature of an organic Rankine cycle (power cycle) driven by the linear Fresnel collectors, designed to operate at 230°C.

## 2 RAY-TRACE SIMULATION

Ray-trace was used for optical efficiency simulations of the concentrating collector. The process consists in following the paths of a large number of rays of incident radiation throughout the system. For reflecting surfaces, the direction and point of intersection of an incident ray with the reflecting surface are determined. The surface normal in each point is also determined, and the reflected ray follows the principle that the angle of reflection equals the angle of incidence. The input data for the simulations are solar geometry and normal beam radiation intensity. Exhaustive ray-trace simulation enables to study the sensitivity of delivered energy to height and width of the receiver, collector tracking orientation, climate and design modifications. We concentrated our attention in design optimization and behaviour of the receiver cavity, which means that a simplified ray-trace analysis was carried out.

The tracking angle  $\psi_i$  (in degrees) of the  $i$ -th reflector was calculated according to Rabl [3]:

$$\psi_i = \frac{\phi_i - \theta}{2} \quad (1)$$

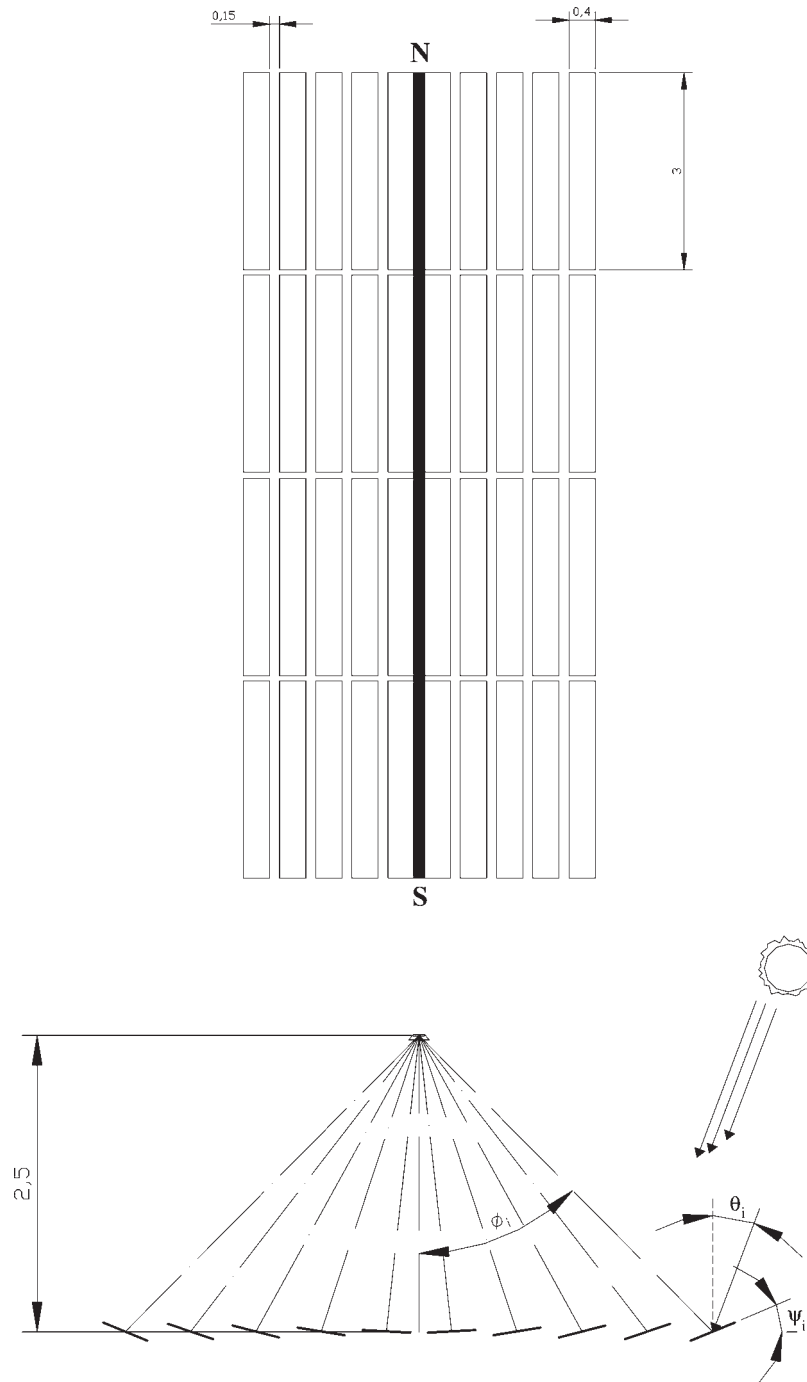


Figure 1. Schematic representation of the collector system (view from above and East–West plane view).

where  $\phi_i$  is the angle between optical axis (in degrees) and the line from focus to reflector,  $\theta$  is the incident angle of the sun relative to the aperture normal (in degrees).

It is known that the parabolic concentrator is the unique reflector shape that focuses beam radiation into a single point. However, the manufacture of a parabolic reflector is too expensive. In this system, we adopted cylindrical mirrors with different curvature. The mirror radius of curvature  $r_i$  (in m)

depends on the focal length  $f$  (in m) of the mirror, and the tracking angle:

$$r_i = \frac{2f}{\cos(\psi)} \quad (2)$$

For radius of curvature calculation (equation 2), we adopted the tracking angle for perpendicular incident radiation ( $\theta = 0$ )

and for focal length the distance from mirror centre to absorber centre.

Table 1 presents the calculated radius of curvature for the mirrors (numbered from centre to extremity).

Figure 2 presents ray-trace simulation results for an incidence angle of  $30^\circ$ . The inclination of the lateral absorber walls' cavity was fixed at  $50^\circ$ , which is about the complementary angle of  $\phi_5$ . The radiation is more or less evenly distributed in the absorber tubes. Some rays are intersected by the absorber shade and by adjacent mirror shade before reaching the mirrors. The geometry of the cavity, with six tubes and an inclination of  $50^\circ$ , optically fulfils the requirements of the Fresnel collector.

### 3 HEAT LOSS COEFFICIENT

The efficiency parameters usually used in solar collector analyses are: the optical efficiency, the incidence angle modifier and the heat loss coefficient. To calculate the heat losses from the absorber tubes, CFD simulations were carried out, taking into account all heat transfer mechanisms: radiation, convection and conduction. To calculate the other two parameters, radiation properties of the reflectors, absorbers and cover are needed. A three-dimensional exhaustive ray-trace analysis must also be carried out, which is out of the scope of this work.

CFD has greatly been developed over recent years, mostly due to the rapid advance of computer technology. It is now possible to solve scientific problems in complex geometries. Natural convection inside the cavity, thermal radiation between surfaces and conduction through the walls was simulated in this work, using Fluent software [4].

Several simplifying assumptions were used:

- steady state;
- laminar flow;
- equal temperature of all receiver pipes;
- symmetry across the vertical mid-plane;
- the cavity cover has negligible thermal mass; it was modelled as having a uniform temperature, considering radiation (emissivity) on both sides and a fixed external convection coefficient;
- the cavity cover is opaque to long-wave radiation;
- the effect of cavity window heating, due to absorptance of the glass, was neglected;
- the pipe temperatures were fixed and resulting heat losses calculated.

The cavity geometry is presented in Figure 3. To simulate the actual cavity is complicated. The methodology was first to separate the simulated geometry in two parts: the geometry situated below the pipes and the geometry above the pipes, which is repeated 10 times in the receiver. The circular pipes were approximated by regular hexadecagons, to avoid elements with

Table 1. Radius of curvature for the different mirrors

Mirror	$\phi$ [°]	$\psi$ [°]	$f$ [m]	$r$ [m]
1	6.28	3.14	2.52	5.04
2	18.26	9.13	2.63	5.33
3	28.81	14.41	2.85	5.89
4	37.60	18.80	3.16	6.67
5	44.71	22.36	3.52	7.61

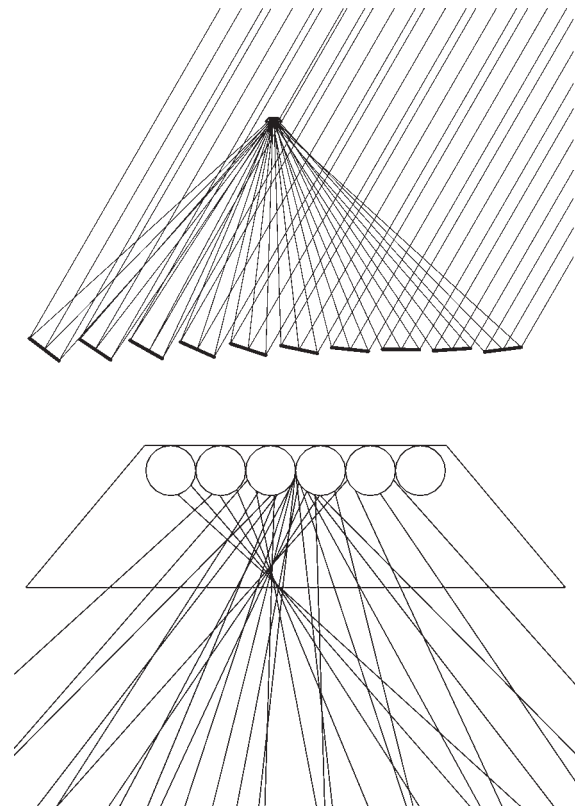


Figure 2. Ray-trace simulation for an incidence angle of  $30^\circ$ ; cross section and zoomed absorber area.

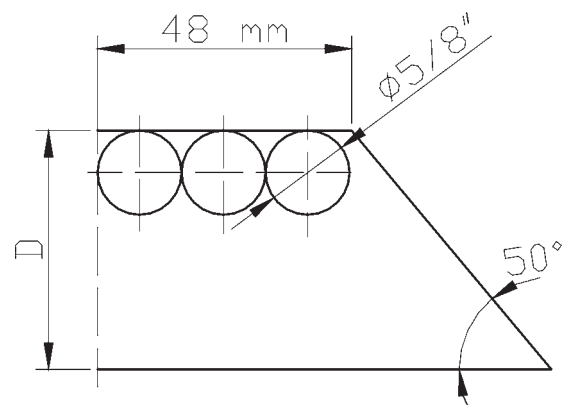


Figure 3. Trapezoidal geometry of the Fresnel receiver cavity.

high skewness, which lead to convergence difficulties and inaccuracies in the numerical solution [5].

The grid was set up with Fluent's grid generation package, Gambit. A hybrid grid was adopted, with an elementary volume size of 0.2 mm for the geometry below the tubes, and a triangular one with an interval size of 0.05 mm for the geometry above the tubes. To check grid independence, a model with a resolution increased by a factor of 2, in all directions, was tested. This represents an increase in the number of cells of four times, and also an increase in computation time of approximately four times [5]. The heat loss coefficients with both grid sizes had a relative difference of 0.0011%, which means that the solution can be considered independent of the grid size.

The density of air was approximated by the Bousinesq model. This model treats the density as a constant value in all equations, except for the buoyancy term in the momentum equation. The air temperature considered in the Bousinesq model was 350 K, and the thermal expansion coefficient was  $0.002857 \text{ K}^{-1}$ .

The model used to simulate thermal radiation was the discrete transfer radiation model. The main assumption is that the radiation leaving the surface element in a certain range of solid angles can be approximated by a single ray. The polar (theta) divisions and azimuthal (Phi) divisions control the number of rays being created from each surface cluster. The number of theta divisions was set to 4, and the number of phi divisions was set to 16. These values were changed from the default values, until the total heat transfer rate from the tubes was equal to the total heat transfer rate in walls and cover.

Table 2. Influence of receiver depth;  $t_{insulation} = 35 \text{ mm}$ ,  $T_{tubes} = 230^\circ\text{C}$ ,  $T_a = 20^\circ\text{C}$  and  $\alpha = 5 \text{ W}/(\text{m}^2 \text{ K})$

$D$ [mm]	30	45	60
$U$ [ $\text{W}/(\text{m}^2 \text{ K})$ ]	0.2405	0.2400	0.2438
Radiation loss [%]	68	73	75

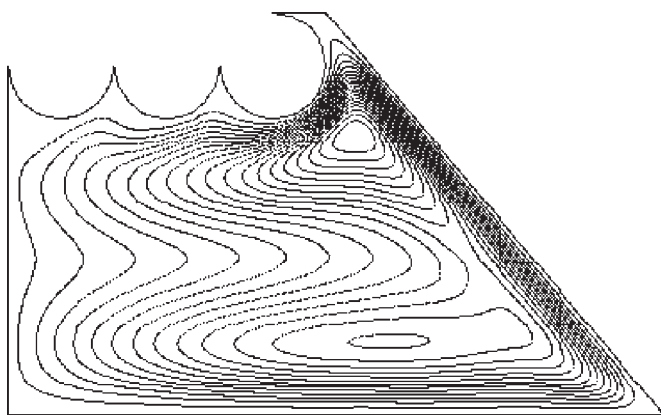


Figure 4. Contours of stream function inside the cavity for a receiver depth of 60 mm; tube temperature of  $230^\circ\text{C}$ , external air temperature of  $20^\circ\text{C}$  and external heat transfer coefficient of  $5 \text{ W}/(\text{m}^2 \text{ K})$ .

The air properties were correlated by quadratic polynomials, as a function of temperature.

Three temperatures of the tubes were simulated: 110, 170 and  $230^\circ\text{C}$ . The tube emissivity was fixed at 0.49, similar to the value used by Pye [6]. External convection was simulated with two different values of the heat transfer coefficient: 5 and  $10 \text{ W}/(\text{m}^2 \text{ K})$ , considered representative of two different wind speeds. External air temperatures considered were 15, 25 and  $35^\circ\text{C}$ . The wall internal emissivity was taken as 0.1 [5]. In the cover, a mixed thermal boundary condition was considered, with external and internal emissivities of 0.9 [5], and two alternative values for the external heat transfer coefficient: 5 or  $10 \text{ W}/(\text{m}^2 \text{ K})$ ; the same three values of external air temperature were considered, with a sky temperature  $5^\circ\text{C}$  lower than air temperature.

In the CFD simulations, the solution was considered as converged when residuals were lower than  $10^{-8}$  for the energy equation and  $10^{-5}$  for the other equations.

Two geometrical parameters were changed and analysed, in order to choose the best geometry: the rock wool insulation thickness and the receiver depth  $D$  (in mm). Table 2 presents

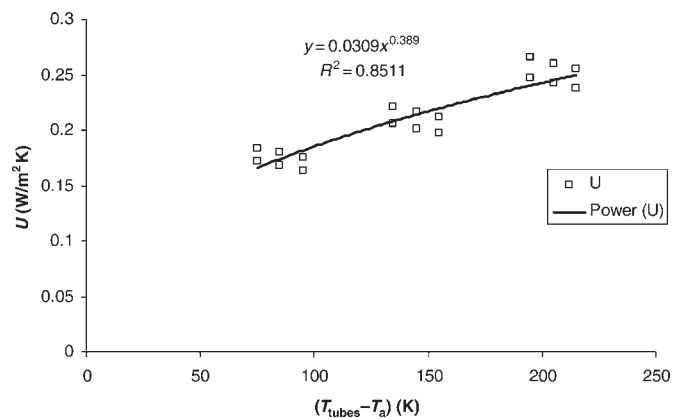


Figure 5. Simulated heat loss coefficient on Fresnel receiver, as a function of temperature difference;  $t_{insulation} = 35 \text{ mm}$  and  $D = 45 \text{ mm}$ .

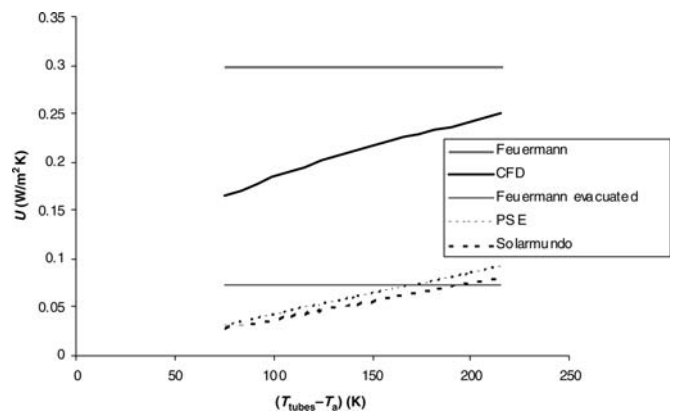


Figure 6. Simulated (CFD) heat transfer coefficient compared with literature values.

the global heat transfer coefficient for three receiver depths: 30, 45 and 60 mm. The receiver with 45 mm presents the lowest heat transfer coefficient, although the differences are very small. Radiation losses dominate heat transfer, compared with convection losses, as shown in Table 2. Radiation losses increase with receiver depth, because cover and lateral wall surfaces are increased. Flow patterns in the cavity are presented in Figure 4. Thermal stratification is observed in the cavity, confirming the small convection losses.

The influence of rock wool insulation thickness was also analysed. As expected, the heat transfer coefficient decreases with insulation thickness. The change from 20 to 35 mm represents a 7% reduction in the heat transfer coefficient (from 0.2565 to 0.2400 W/m<sup>2</sup> K), while from 35 to 50 mm it represents a 4% reduction (to 0.2317 W/m<sup>2</sup> K); 35 mm should be chosen, taking into account the heat transfer coefficient value and that the increase in insulation thickness leads to more significant shading in the Fresnel collector.

Eighteen combinations were simulated and the heat loss coefficient based on primary mirror aperture area was calculated as a function of the difference between average tube temperature and ambient air temperature, as presented in Figure 5. A power-law trendline was chosen to fit to the results. Its correlation coefficient is 0.85.

The correlation obtained for the Fresnel receiver cavity in analysis was compared with the available literature correlations, for this type of collectors. The new cavity presents a simulated heat transfer coefficient that is smaller than for the non-evacuated receiver used by Feuermann and Gordon [7], but larger than the others available in the literature (see Figure 6). Two possible causes for this difference are: the cavity under analysis is non-evacuated compared with others that are evacuated (Feuermann and Gordon [7] and PSE [2]) and the total primary mirror area taken for the new system. The heat transfer coefficient varies inversely with total primary mirror area. Here, an aperture width of 4 m was considered, equal to the aperture of Feuermann and Gordon [7]. The prototype of PSE [2] had 5.5 m, whereas Solarmundo [8] had 24 m.

## 4 CONCLUSIONS

A trapezoidal cavity receiver for a linear Fresnel solar collector concentrator was designed and numerically simulated.

Fixing the geometry of the collector and regarding only the cavity, simplified ray-trace simulations concluded that the

cavity with six absorber tubes of 0.5 in. internal diameter (5/8 in. outside diameter) collects all the concentrated beam radiation. The 50° inclination of the lateral cavity walls was also found to be optically acceptable.

To evaluate the overall heat transfer coefficient of the Fresnel collector, CFD simulations were applied. Natural convection inside the cavity, thermal radiation between surfaces and conduction through the walls were modelled. Two geometrical parameters were analysed: receiver depth and insulation thickness. It was concluded that the cavity with a 45 mm depth presents the lowest global heat transfer coefficient. Regarding insulation thickness, 35 mm of rock wool presented a good compromise between insulation and shading.

Correlating the simulated heat transfer coefficient (based on primary mirror area) with the temperature difference between tubes and ambient air, a power-law fit was obtained. The simulated heat loss coefficient for the new cavity showed larger values, when compared with values presented in the existing literature for linear Fresnel collectors. Two possible causes for this difference are that the new cavity is not evacuated and the larger aperture width of the system when compared with the available prototypes.

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