Parabolic Trough Collector with Internal Fractal Fins in Receiver to Increase Thermal Performance in Working Fluid

Angélica PALACIOS PhD student¹, Darío AMAYA Ph. D², Olga RAMOS PhD ³
{u1801712 1 dario. amaya2, olga. ramos³} @unimilitar.edu.co

Universidad Militar Nueva Granada, Colombia

Abstract: Concentrated solar power technologies have been studied in recent years as a potential solution for energy production industry, however this kind of systems face different challenges in order to increase thermal performance and efficiency. This paper presents the results of a thermal analysis on a parabolic trough collector with different receivers internal fins configurations, systems were studied in CFD Solidworks® flow simulation software and compared with a traditional parabolic collector with cylindrical receiver. Results shows that the lowest thermal performance was achieved with a cylindrical pipe, instead fractal fins receiver with 5 internal fins achieves a temperature increase of 27% and the best pipe configuration was fractal Descartes with 12 internal pipes which the cylindrical temperature in 29%.

Keywords: Solar radiation, parabolic collectors, fractal receivers, coupled collectors, thermal heat collectors.

I. INTRODUCTION

Concentrated technologies for energy production have been developed in many knowledge fields, specifically in countries energy industry. Many of them plan to develop this technology on a high level for the next years [1], is the case of Morocco, Saudi Arabia and India, where planned to install 2GW, 25GW and 200GW in the next two years [2], [3]. Even though, the industry of these countries are still in a commercial demonstration the efforts to achieves their purpose is evident [4]. In contrast, countries as Spain and America already pass the border and achieved a great experience from 2007 to 2016 and now are in commercial operation stage [5], [6].

On solar energy production field, the most developed technologies are photovoltaic and concentrated solar power (CSP), first one offer a decentralized power generation and second one offer energy plants with storage system, both are the most suitable sources for energy production on a high scale [7], [8].Within the advantages of concentrator solar power is producing dispatchable renewable electricity with a balancing intermittent of renewable sources,[9] as is the case of wind power and solar photovoltaic sources which are intermittent, so require an additional storage system. CSP has the option to integrate a thermal storage and at the same time an energy storage when sun hours are available, so later can be used after generation time or even in the night [10].

CSP systems works in three phases, first consist on solar radiation transformation to concentrated radiations heat, trough rays concentrated on system surface to a focal point. Second phase relates absorbance in central receiver from radiation to thermal conversion. Finally, thermal energy is transformed to kinetic energy and transformed in electrical energy on a power generator, [11].

Thermal performance analysis of a CSP system has been developed in certain researches, many of them use a computational fluid dynamics (CFD) algorithms to predict physical interaction between solar systems and working fluids. Is the case of the work presented in [12], which identify the convective flows produced within the melted phase by temperature gradients and gravity, applied to high temperature concentrated solar power plants. Heat transfer was simulated and analyzed by Navier–Stokes equations, as results was possible to enhanced heat flux, associated to natural convective flow and system reduced around 30% the time needed to charge the heat storage. Furthermore, in [13] is presented a model which solves a heat transfer problem in receiver pipe conduction with a sodium internal flow. In the analysis was studied under different speeds of wind, also convective losses were calculated through empirical correlation based on literature and then evaluated by CFD study. As results, was evident a significant difference in temperature distribution in receivers and empirical correlations from literature.

Thermal enhancement analysis trough plate fin was developed in [14], the principal aim of this works was studied with FLUENT characteristics and thermal performance of phase change material (PCM) in different receivers and covers configurations; different designs were contrasted with plate fins design. Result presented an increase in heat transfer rate with a vertical array of plate fins in contrast with a counter flow cover and receivers configurations. The design proposed leads to less redundant PCM, as well as a smaller and more cost effective PCM system as a heat storage unit. Similarly, the geometry optimization of a phase change material (PCM) heat storage system is presented in [15] with
the purpose of balance and counter solar radiation variations effect on a heat exchanger a PCM-fins located in backside of a absorber receiver is proposed. Based on a geometrical optimization was possible to calculate an optimum ratio, width and length of the PCM. Results shows a numerically model validated with CFD study obtaining a good approximation of the real optimum.

The modeling approach for 1D microstructure absorbing multi-layers for Concentrated Solar Power (CSP) receivers is described in [16], the studied was use an optimized multilayer structure to achieve a higher absorption in receiver a Chandezon method was used as theoretically model to optimize grating profiles. In experimental results a total of 96.5% was obtained for absorption in the visible and UV range, which means an enhancement of 2% in comparison to non-structured coatings. On the other hand, in [17] different arrangements of an open cavity receivers using Monte-Carlo ray-tracing technique was studied with ray tracing Opti Works software, the purpose was predict a flux distribution, decrease the optical and thermal losses of a receivers with different shape, diameter and helical coil pitch. As results, a zero-pitch evidence a better performance in both optically and thermal analysis also an enhancement up to around 7% in the overall thermal performance was achieved when the receiver aperture area was covered by glass.

Based on the above, this paper presents the results of a thermal analysis on a parabolic trough collector with different receivers internal fins configurations, systems were studied in CFD Solidworks® flow simulation software and compared with a traditional parabolic collector with cylindrical receiver. Results shows that the lowest thermal performance was achieved with a cylindrical pipe, instead fractal fins receiver with 5 internal fins achieves a temperature increase of 27% and the best pipe configuration was fractal Descartes with 12 internal pipes which e the cylindrical temperature in 29%.

### Nomenclature

| Symbol | Description               |
|--------|---------------------------|
| Q      | Heat                      |
| T      | Temperature               |
| h      | Convective Coefficient    |
| K      | Conductivity Coefficient  |
| Nu     | Nusselt Number            |
| W      | Weight                    |
| f      | Focal Length              |
| L      | Length                    |
| d      | Diameter                  |
| A      | Area                      |
| C      | Concentration Ratio       |
| σ      | Boltzmann Coefficient     |
| εc     | Cover Emissivity          |
| B      | Radius Variations         |
| θ      | Angle                     |
| a      | Aperture                  |
| r      | Receiver                  |
| rd     | Fluid-Internal Receiver   |
| h      | Hydraulic                 |
| f      | Radiation                 |
| r      | Internal Receiver-External Receiver |
| cl     | External Receiver-Internal Cover |
| c      | Internal Cover-External Cover |
| env    | External Cover-Environment |
| sky    | External Cover-Sky         |
| hl     | Heat Loss                 |
| ab     | Absorbance                |
| n      | Iterations                |

### II. METHODS

#### Parabolic Through Collector

A parabolic through collector consist on a solar energy concentrator, the principal source of energy is Sun radiation which achieves maximum values of $1.74 \times 10^6 \text{W} \text{m}^{-2}$ on earth atmosphere [18], [19]. As show Figure 1, system is composed by a concentrator surface which has a parabolic geometry with the purpose of concentrate all solar rays on a focal point, same location of the receiver pipe where working flow is conducted and finally a glass cover over receiver pipe to reduce convection and radiation losses between receiver and environment.

![Figure 1. Parabolic through collector.](image)

In this paper a parabolic through collector with dimensions registered in Table 1, was studied with different types of pipes. External diameter, length and focal length were the same for all receivers, so concentration ratio remains in 10.24 in all applications.

#### Mathematical Description

#### Energy Balance Model

Radial heat transfer terms are related on energy balance model, where conduction, convection and radiation phenomena’s between each system are involved. Energy balance model for a parabolic trough collector is presented in [Eq.(1)-Eq.(5)]. Heat flux absorbed since glass cover ($Q_{ab}$) in [Eq.(2)]groups convection heat flux from receiver to internal cover($Q_{cc,ro-cl}$), radiation heat flux from external receiver to internal cover ($Q_{rd,ro-cl}$), conduction heat flux of pipe ($Q_{cd,pipe}$) and conduction heat flux from internal receiver to external receiver ($Q_{cd,ri-ro}$). Heat loss flux ($Q_{hl}$) in [Eq.(5)] combine conduction un pipe, radiation from external glass...
cover to sky \( (Q_{rd,c-sky}) \) and convection from external glass cover to environmental \( (Q_{co,co-env}) \).

\[
\begin{align*}
Q_{co,fr} & = Q_{cd,ri-ro} \\
Q_{ab} & = Q_{co,ro-ci} + Q_{rd,ro-ci} + Q_{cd,ri-ro} + Q_{cd,pipe} \\
Q_{co,ro-ci} + Q_{rd,ro-ci} & = Q_{cd,ci-c} \\
Q_{cd,ci-c} + Q_{a} & = Q_{cd,ci-env} + Q_{rd,c-sky} \\
Q_{hl} & = Q_{co,ci-env} + Q_{rd,c-sky} + Q_{cd,pipe}
\end{align*}
\]

(1) Radiative Heat Transfer

Radiative heat transfer is fundamental in thermal performance analysis of a parabolic trough collector. Radiation between glass cover and sky can be represented by [Eq.(6)], which relates Boltzmann coefficient, cover emissivity \( (\varepsilon) \), cover diameter \( (D_c) \) and delta temperature between two environments (pipe and sky). In this case, sky is considered as a blackbody at \( (T_{sky}) \).

\[
Q_{rd,c-sky} = \sigma \varepsilon D_c (T_{sky}^4 - T_{sky}^4)
\]

(6) Sky temperature \( (T_{sky}) \) can be assumed as 6°C lower that environment temperature. If environment temperature is the range from 20°C to 22°C, \( T_{sky} \approx [14°C - 16°C] \).

(2) Convection heat Transfer

Heat transfer by convection phenomena, is described through Newton’s law of cooling. For a glass cover to environment the convection heat transfer can be related as [Eq.(7)].

\[
Q_{conv(co-env)} = h_{cv,co-env} \times \pi D_c (T_{co} - T_{env})
\]

(7) Convective coefficient \( h_{cv,co-env} \) in [Eq.(8)] depends on conductivity coefficient of air \( (K_{air}) \), cover diameter \( (D_{co}) \) and Nusselt number \( (Nu) \).

\[
h_{cv,co-env} = \frac{K_{air}}{D_{co}} Nu
\]

(8) Simulations and Boundary Conditions

For the analysis was selected aluminum as material for surface concentrator, copper for receivers and glass for the cover. As radiative properties, surface collector was defined as symmetry and receiver pipe as blackbody, the total radiation on collector was establishment in 800W/m² and the working flow was air at a flow mass of 0.0016 Kg/s. Simulation tool used in this application was flow simulation from Solworks® software. To resolve a CFD analysis this software applied Navier-Stokes equations [Eq.(9) -Eq.(11)] which relates conservation laws for mass, angular momentum and energy in the Cartesian coordinate system.

\[
\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho u_i) = 0
\]

(9) \[
\frac{\partial \rho u_i}{\partial t} + \frac{\partial}{\partial x_j} (\rho u_i u_j) + \frac{\partial}{\partial x_j} (\tau_{ij} + \tau_{ij}^r) + S_i
\]

(10) \[
\frac{\partial H}{\partial t} + \frac{\partial}{\partial x_i} (u_i (\tau_{ij} + \tau_{ij}^r) + q_i) + \frac{\partial}{\partial x_j} (\tau_{ij} \frac{\partial u_i}{\partial x_j} + \rho e)
\]

(11) Fractal Description

Fractal Fins

Receivers with internal fractal fins increase the number of fins in each iteration, the internal fins have a specific organization with the purpose to guarantee the same distribution in all external pipe, which means that besides number of fins is associate a variation angle formed for all fins intercepted in receiver center. Hydraulic diameter of a regular pipe can be defined as [Eq.(12)], which involve area and perimeter and is finally reduced until internal diameter.

\[
D_h = \frac{4A}{2\pi r} = \frac{4\pi r^2}{2\pi r} = \frac{D}{2}
\]

(12) In fractal fins receiver hydraulic diameter is modified and can be calculated as circular section mathematical relation in [Eq.(15)] for fractal iterations \( n \) will depends the hydraulic diameter at angle \( (\theta) \) and radius variations \( (B) \).

\[
D_{hf} = \frac{2B \theta}{1 + \theta} \frac{D_{hf}}{1 + \theta} = \frac{2B \theta}{1 + \theta} \frac{D}{1 + \theta}
\]

(13) Radius, angle and hydraulic diameter for each fractal fins receiver are listed in Table 2.

| n  | B   | \( \theta \) | \( D_h \) |
|----|-----|--------------|----------|
| 3  | 11.43mm | 120°        | 0.0464m  |
| 4  | 11.43mm | 90°         | 0.0559m  |
| 5  | 11.43mm | 72°         | 0.0636m  |
| 6  | 11.43mm | 60°         | 0.0702m  |
| 7  | 11.43mm | 51.43°      | 0.0757m  |
| 8  | 11.43mm | 45°         | 0.0804m  |
| 9  | 11.37mm | 40°         | 0.0841m  |
| 10 | 11.27mm | 36°         | 0.0870m  |

Figure 2. Fractal Fins receivers.
Fractal Descartes
Fractal Descartes consist on a circular shaped as Apollonian fractal, the internal geometry was constructed with fractal array based on Descartes circle theorem, [20], [21].

\[(a + b_1 + b_2 + b_3)^2 - 2(a^2 + b_1^2 + b_2^2 + b_3^2) = 0\]  \(\quad\) (14)

System can be solved in \(a\) as a quadratic equation , this system has two roots as [Eq.(15) describes.

\[a_{1,2} = (b_1 + b_2 + b_3) \pm 2\sqrt{D}\]

\[D = b_1 b_2 + b_2 b_3 + b_3 b_1\]  \(\quad\) (15)

This relations conduct to (Vieta formulas) in [Eq.(16).

\[a_1+a_2=2(b_1 + b_2 + b_3) \quad a_1 a_2 = b_1^2 + b_2^2 + b_3^2 - 2(b_1 b_2 + b_2 b_3 + b_3 b_1)\]  \(\quad\) (16)

In Descartes configuration of four mutually tangent circles, one can select one of the circles and replace it by another circle, tangent to the same three circles (replace \(a_1\) by \(a_2\)). Continuing this process results in an Apollonian packing of circles (Apollonian gasket) [22]. Figure 4 presents 5 receivers designed based on Descartes theorem of an Apollonian fractal, starting in iteration 1 (one inner pipe) Figure 4.A until 5 iteration (12 inner pipe) Figure 4.E.

III. RESULTS
All collectors with different receiver as cylindrical, fractal fins and fractal Descartes models were simulated in Solidworks® flow simulation tool. A CFD analysis below to identify the best configuration based on thermal performance in the working flow and external surface in pipe. First, the results of a parabolic trough collector with a conventional receiver (cylindrical with out pipes or fins inside) are presented in the section down below.

Parabolic Trough Collector
(Conventional Receiver)

With an initial temperature of 21°C the parabolic trough collector with a conventional receiver achieved a maximum temperature in the internal fluid of 80.61°C without a cover. Instead the same system with a glass cover over cylindrical pipe achieves a maximum temperature of 102°C, which means an increase of 26%. Regard to inlet temperature the first system obtain an increase of 59.61°C and the last an increase of 81.09°C, results are described in Table 3.

| Maximum Temperature | Pipe F0 |
|---------------------|--------|
| **Internal Fluid**  | 80.61°C| No cover |
| **External Solid**  | 80.61°C|        |
| **Internal Fluid**  | 102.09°C| Cover   |
The differences between the use and non-use of an external cover can be visualized in graphs of Figure 5. In all iterations fluid temperature with cover is always higher than the fluid temperature without a cover; as is presented the delta temperature between both curves is almost 22°C. A total maximum increase of 385% was achieved in cylindrical receiver with a glass cover.

Parabolic Trough Collector (Fractal Fins Receiver)
Second system proposed on a parabolic trough collector, consisted on a fractal fins inside the cylindrical receiver, in this case as the last one, the temperature performance was analyzed. The first group of pipes is conformed by receiver with 3, 4 and 5 fins, in each one the fluid and solid temperature was obtained by different configurations in flow areas and external covers. In flow areas configuration was determined the fluid conduction in all or certain areas generated by the fins.

In all cases the fluid conduction by all areas were studied and in the most cases a number between 2 to 1 area was studied. As presents Table 4 the temperatures (fluid and solid) in F3, F4 and F5 receivers are lower with a high number of area conduction, while an area decrease the temperature are maximum. Also was possible to evidence that a glass cover achieved a higher temperature that the same system without a cover. The maximum temperature obtained in F3 receiver was 112°C in the internal fluid with a single area conduction and a glass cover, in this pipe a total increase of 433% regard to inlet temperature was achieved and a total increase of 9.8% regard to cylindrical receiver.

F4 receiver obtained a maximum temperature of 121°C which means 10°C more that F3 receiver, in this case a total of area conduction was 2 and the pipe was covered with a glass arrangement. Almost the same pattern was evidenced in F5 receiver where a total of 129.4°C was achieved as maximum temperature with a single area conduction and a cover system. Regard to cylindrical pipe F5 receiver has a higher temperature with an order of 27%.

Figure 5. Fluid temperatures in CCP with cylindrical pipe.

| Surface         | Maximum Temperature [°C] | Average Temperature [°C] | Fins | Inlet Flow | Pipe    |
|-----------------|--------------------------|--------------------------|------|------------|---------|
| **F3 Pipe**     |                          |                          |      |            |         |
| Internal Fluid  | 85.18                    | 75.547                   | 3    | all        | No cover|
| External Solid  | 85.18                    | 75.459                   |      | No cover   |         |
| Internal Fluid  | 88.5                     | 80.495                   | 1    | No cover   | Cover   |
| External Solid  | 88.5                     | 80.424                   |      |            |         |
| Internal Fluid  | 111.98                   | 102.357                  | 2    | Cover      |         |
| External Solid  | 112.13                   | 102.492                  |      |            |         |
| **F4 Pipe**     |                          |                          |      |            |         |
| Internal Fluid  | 89.42                    | 81.607                   | 4    | all        | No cover|
| External Solid  | 89.42                    | 81.554                   |      | No cover   |         |
| Internal Fluid  | 95.59                    | 83.853                   |      |            | Cover   |
| External Solid  | 95.59                    | 83.800                   |      |            |         |
| Internal Fluid  | 120.95                   | 108.702                  |      |            |         |
| External Solid  | 120.95                   | 108.879                  |      |            |         |
| **F5 Pipe**     |                          |                          |      |            |         |
| Internal Fluid  | 88.22                    | 79.722                   | 5    | all        | No cover|
| External Solid  | 88.22                    | 79.655                   |      | No cover   |         |
| Internal Fluid  | 91.49                    | 82.665                   |      |            | No Cover|
| External Solid  | 91.49                    | 82.732                   |      |            | Cover   |
| Internal Fluid  | 104.53                   | 93.775                   |      |            |         |
| External Solid  | 104.53                   | 93.708                   |      |            |         |
| Internal Fluid  | 129.39                   | 119.004                  |      |            |         |
| External Solid  | 129.75                   | 119.303                  |      |            |         |
As Table 5 reference, the next receiver with 6, 7 and 8 fins were studied, in this case F6 pipe achieved a maximum temperature of 127.21°C and a mean temperature of 117°C with two areas and a pipe covered. A decrease of 2°C was identify regard to F5 pipe and an increase of 6.3°C and 15°C regard to F4 and F3 pipes respectively. Similarly, a temperature decrease was evidence in F7 receiver where the maximum temperature was 117°C almost 10°C less that F6 and 12°C that F5. On the other hand, F8 pipe achieved a maximum temperature of 120.35°C similarly to F4 receiver. This performance show that the number of fins has not a relation of the temperature increased, however the number of area conduction influence on thermal results. In contrast of cylindrical receiver, the increase between F6 receiver and the conventional model was a total of 25%.

| Surface               | Maximum Temperature [°C] | Average Temperature [°C] | Fins | Inlet Flow | Pipe  |
|-----------------------|--------------------------|--------------------------|------|------------|-------|
| **F6 Pipe**           |                          |                          |      |            |       |
| Internal Fluid        | 87.08                    | 78.437                   | 6    | all        | No cover |
| External Solid        | 87.08                    | 78.369                   |      |            |       |
| Internal Fluid        | **103.79**               | 92.492                   |      | 2          | No cover |
| External Solid        | 103.79                   | 92.422                   |      |            |       |
| Internal Fluid        | 92.93                    |                          | 1    |            | No cover |
| External Solid        | 93.93                    |                          |      |            |       |
| Internal Fluid        | **127.21**               | 116.654                  | 2    | Cover      |       |
| External Solid        | 127.32                   | 116.763                  |      |            |       |

| Surface               | Maximum Temperature [°C] | Average Temperature [°C] | Fins | Inlet Flow | Pipe  |
|-----------------------|--------------------------|--------------------------|------|------------|-------|
| **F7 Pipe**           |                          |                          |      |            |       |
| Internal Fluid        | 85.22                    | 77.701                   | 7    | all        | No cover |
| External Solid        | 85.22                    | 77.738                   |      |            |       |
| Internal Fluid        | 88.80                    | 80.824                   |      | 2          | No cover |
| External Solid        | 88.80                    | 80.760                   |      |            |       |
| Internal Fluid        | 91.32                    | 83.122                   | 1    |            | No cover |
| External Solid        | 91.32                    | 83.050                   |      |            |       |
| Internal Fluid        | **117.40**               | 109.865                  | 1    | Cover      |       |
| External Solid        | 117.41                   | 109.884                  |      |            |       |

| Surface               | Maximum Temperature [°C] | Average Temperature [°C] | Fins | Inlet Flow | Pipe  |
|-----------------------|--------------------------|--------------------------|------|------------|-------|
| **F8 Pipe**           |                          |                          |      |            |       |
| Internal Fluid        | 87.88                    | 79.888                   | 8    | all        | No cover |
| External Solid        | 87.88                    | 79.824                   |      |            |       |
| Internal Fluid        | 91.92                    | 83.725                   |      | 2          | No cover |
| External Solid        | 91.92                    | 83.661                   |      |            |       |
| Internal Fluid        | 94.26                    | 85.703                   | 1    |            | No cover |
| External Solid        | 94.26                    | 85.769                   |      |            |       |
| Internal Fluid        | **120.35**               | 111.069                  | 1    | Cover      |       |
| External Solid        | 120.58                   | 111.256                  |      |            |       |

Finally, the last fractal fins pipes were analyzed in the first F9 pipe the maximum temperature obtained during CFD study was 96°C with a single area for fluid conduction and a glass cover on the receiver. This pipe didn’t exceed the temperature obtained in the cylindrical pipe, conversely the difference was almost 6°C which means a decrease of 5.88%. Furthermore, the last pipe (F10) achieved a maximum temperature of 129.32°C very similarly to F5 temperature, so was possible to identify that the best results were obtained with F5 and F10 both on a system with a single area conduction and with a pipe covered. Results for F9 and F10 receiver are consigned in Table 6.

| Surface               | Maximum Temperature [°C] | Average Temperature [°C] | Fins | Inlet Flow | Pipe  |
|-----------------------|--------------------------|--------------------------|------|------------|-------|
| **F9 Pipe**           |                          |                          |      |            |       |
| Internal Fluid        | 88.81                    | 82.734                   | 9    | all        | No cover |
| External Solid        | 88.81                    | 82.190                   |      |            |       |
| Internal Fluid        | 93.03                    | 86.140                   |      | 2          | No cover |
| External Solid        | 93.03                    | 86.095                   |      |            |       |
| Internal Fluid        | 95.81                    | 86.947                   |      |            |       |
| External Solid        | 95.81                    | 86.916                   |      | 1          | No cover |
| Internal Fluid        | **95.89**                | 86.945                   | 1    | Cover      |       |
| External Solid        | 95.89                    | 86.914                   |      |            |       |
The thermal performance of each fractal receiver is shown in Figure 6, is possible to evidence temperature distribution inside pipes, inside cover and external environment. Figure 6.A describe thermal performance in cylindrical or conventional pipe with a total temperature of 102°C. Figure 6.B related results of F3 receiver where three fins inside the pipe separate the conduction area in three sections, as shown the area where flow conduction occurs remain with a lower temperature that the other sections. On the other hand, Figure 6.C show the results of F4 pipe in this case two sections are clearly defined as flow conduction areas, also presents a best flow distribution inside pipe and cover. Then in Figure 6.D the best results obtained under fins configuration with F5 receiver are shown. F6 receiver thermal results can be visualized in Figure 6.E with a total of 127°C, this receiver achieved one of the higher temperature in this study, however has the particularity that best results were obtained with two areas and not with a single fluid conduction. So, a better thermal distribution was obtained in pipe with six fins that pipes with five or ten fins.

F7 and F8 receivers with a single conduction is presented in Figure 6.F and Figure 6.G respectively. The lowest temperature achieved during the study in F9 pipe is related in Figure 6.H however the system was covered with glass was not evidenced a difference between system with cover and system without. Finally, F10 receiver results of thermal performance is presented in Figure 6.I as was described this system was one of the best, with a single area of conduction and a pipe covered up with glass arrangement.
During the analysis was possible to establish that the number of fins inside receiver pipe has not relation with the increase of temperature, due to F9 achieves the lowest temperature and F5, F6 and F10 receiver achieved the highest results. Instead was possible to evidence that the number of areas for fluid conduction influence on thermal performance, for a minor area the temperature will be higher and for a major area the temperature will be lower. However, it’s not only depends of the number of areas, also is related with which kind of areas and that means that it can not be consecutive or side matching; for a best flow distribution inside pipe the best configuration is areas opposed as F6 receiver presents.

In Figure 7, is presented the relation between temperature and fins inside pipe. On the range of 3 to 5 Fins the temperature is increased in each receiver, however in the range of 6 to 9 fins the temperature decreases and finally increase in the last receiver with 10 fins.

On the other hand, was possible to determinate that a couple of pipe achieves similarly temperatures with different numbers of fins inside, was the case of F5 and F10 both obtain 129°C in fluid and F4 y F8 which both obtain 120°C in outlet temperature. Graph with all results of fractal fins receivers es related in Figure 8.

The curves in dotted lines indicate fluid temperature results in all fractal systems without cover, as shown the systems with a glass cover increased temperatures due to environment losses.

Comparing results of cylindrical receiver and the best results of fractal fins receiver as graphs in Figure 9, was evidenced a total difference of 27°C between cylindrical temperature and F5 temperature, means that changing flow conduction inside pipe allow to improve thermal performance on a solar concentrator system as parabolic trough collector.
Parabolic Trough Collector (Fractal Descartes Receiver)
Fractal Descartes receiver consist on an external and internal cylindrical pipe, the number of internal pipes define each architecture; based on Descartes theorem of tangent circles all pipes were constructed. As fractal fins analysis different kind of flow conduction was evaluated in fractal Descartes analysis. First pipe D1 is confirmed by an external cylindrical pipe and one cylindrical pipe inside achieved a maximum temperature of 127°C with a flow conduction only in the internal receiver, this system has a cover glass to reduce environment losses. In Table 7 results for all flow conduction as external pipe (PE), internal (PI) or all areas which relates both internal and external pipes transversal areas.

| Maximum Temperature | Flow | Descartes 1 |
|---------------------|------|-------------|
| Fluid Temperature °C | 87.68 | all | No Cover |
| Solid Temperature °C | 87.68 | PE | No Cover |
| Fluid Temperature °C | 87.46 | PI | No Cover |
| Solid Temperature °C | 95.28 | PI | Cover |
| Fluid Temperature °C | 126.87 | PI | Cover |
| Solid Temperature °C | 126.94 | |

Thermal performance in D1 receiver with a flow conduction in the external and internal pipe, can be observed in Figure 10.A fluid temperature achieved a value of 88°C, delta temperature regard to inlet temperature was 67°C. In Figure 10.B flow conduction was only in the external pipe and temperature was 87.5°C similarly to the first configuration, thermal performance with a single internal conduction and not covered is visualized in Figure 10.C a maximum temperature of 95°C was obtained, almost 8°C more that the external flow. The same system with the glass cover obtains the highest temperature of 127°C as shown in Figure 10.D.

As presents the graphs in Figure 11, the lowest temperature were systems with flow conduction in all pipes and only in the external pipe, instead a flow conduction in internal pipe without cover and covered up achieved the maximum temperature. Regard to cylindrical receiver the fractal D1 receiver increase outlet temperature in 24.5%.

| Maximum Temperature | Flow | Descartes 2 |
|---------------------|------|-------------|
| Fluid Temperature °C | 87.19 | all | No Cover |
| Solid Temperature °C | 87.19 | PE | No Cover |
| Fluid Temperature °C | 87.22 | PE | Cover |
| Solid Temperature °C | 87.22 |
|----------------------|-------|
| Fluid Temperature °C | 91.64 |
| Solid Temperature °C | 91.64 |
| Fluid Temperature °C | 123.66 |
| Solid Temperature °C | 123.66 |

**Figure 12.A** relate thermal distribution in D2 receiver with a flow conduction in all pipes, the maximum temperature achieved in this case was 87°C, with a flow conduction only in both internal pipes the temperature increased to 92°C, which means a difference of 5°C, **Figure 12.B**. A bigger difference was evidenced between systems with cover and without, the covered systems achieved a maximum temperature, **Figure 12.C**.

The thermal performance of flow conduction in all pipes and the external pipes is nearly, a bigger difference is visualized with internal flow conduction results, specifically with a covered system. Those results can be observed in the graph of **Figure 13**. Regard to cylindrical receiver the fractal D2 receiver increase outlet temperature in 21.57%.

![Figure 13. Fluid temperature in D2 receiver.](image)

The next iteration of fractal Descartes receiver consist by one external pipe and three internal pipes. As is registered in **Table 9**, the lower temperature obtained in this pipe was 85°C with a flow conduction in the external flow, however the maximum temperature was 116°C with an internal flow and a covered system. In contrast of Descartes D1 and Descartes D2 pipes, the difference between D3 and the first was 11°C and with the second one was 8°C.

**Table 9. Temperature results of D3 receiver.**

|                      | Maximum Temperature | Flow | Descartes 3 |
|----------------------|---------------------|------|-------------|
| Fluid Temperature °C | 85.76               | PE   | No Cover    |
| Solid Temperature °C | 85.76               | PE   | No Cover    |
| Fluid Temperature °C | 84.99               | PI   | No Cover    |
| Solid Temperature °C | 84.99               | PI   | No Cover    |
| Fluid Temperature °C | 89.4                | PE   | No Cover    |
| Solid Temperature °C | 89.4                | PE   | No Cover    |
| Fluid Temperature °C | 115.69              | PI   | Cover       |
| Solid Temperature °C | 115.72              | PI   | Cover       |

Temperature distribution obtained with fractal Descartes D3 pipe can be observed in **Figure 14.A**, specifically results of flow conduction in all pipes related is presented in **Figure 14.A**. The maximum temperature achieved in this scheme was 85.21°C, furthermore results of flow conduction only in external pipe achieved a maximum temperature of 89°C, 4°C higher that the first one, **Figure 14.B**. Finally, thermal distribution of internal flow conduction configuration coupled with a glass cover is related in **Figure 14.C**.

![Figure 14. Thermal performance in D3 receiver.](image)

As shown in graph of **Figure 15** fluid temperature performance in pipe D3 under a flow conduction in all pipes and in external pipe are so similarly a small difference can be observed, however the maximum value vary just for a few decimals. Fluid temperature in system when the mass flow was established only in the internal pipes was superior that obtained with external flow, this behavior it has been evidenced in all Descartes systems. Also, the influence of a cover is observed in all kind of collectors studied in this paper, in Descartes D3 pipe the difference between receiver covered and not covered was 27°C, means an increase with the cover almost of 30%.

For the first three iterations temperature results are decreased in an incremental behavior, nevertheless all fluid and solid temperatures are better that results obtained with cylindrical receiver. Regard to cylindrical receiver the fractal D3 receiver increase outlet temperature in 13.45%.
In D4 CFD study a temperature increased was evidenced in contrast of D3 receiver, the maximum temperature obtained by D4 was 122°C as fluid as solid. A lower temperature was achieved by the same system but has not glass cover, in this case temperature was 93°C; in both cases, a mass flow was defined inside of the small internal pipes (PIS). On the other hand, temperature decreased in the big internal pipes (PIB) resulting in a value range of 91.8°C, difference was almost 1.5°C between both internal pipes. Table 10 presents results for D4 fractal receiver analyzed in this paper.

| Maximum Temperature | Flow | Descartes 4 |
|---------------------|------|-------------|
| Fluid Temperature °C | 87.95| all No Cover |
| Solid Temperature °C | 87.95| PE No Cover |
| Fluid Temperature °C | 88.21| PI all No Cover |
| Solid Temperature °C | 88.21| PI B No Cover |
| Fluid Temperature °C | 90.18| PI S No Cover |
| Solid Temperature °C | 90.19| PI S Cover |
| Fluid Temperature °C | 91.75| PI S Cover |
| Solid Temperature °C | 91.76| PI S Cover |
| Fluid Temperature °C | 93.08| PI S Cover |
| Solid Temperature °C | 93.08| PI S Cover |

As show Figure 16.A and Figure 16.B temperature performance was similarly with a flow conduction in all pipes and in the external pipe, however temperature distribution seems more uniform inside all pipes when flow conduction is carried out in external receiver. A small increase is evident in all internal pipes which the maximum temperature in fluid was 90.1°C as Figure 16.C present. In the same way, an increase around 1.5°C or 2°C was obtained in the next configurations, where air flow inside of the big internal pipes Figure 16.D, of the small internal pipes Figure 16.E and the same small pipes but covered Figure 16.F.

In contrast to the first three iteration the fourth iteration increase the temperature of the previous iteration in this case the third one, delta temperature was 6°C between both receivers. Regard to cylindrical receiver the fractal D4 receiver increase outlet temperature in 19.61%. The small temperature increases from external to all internal pipes can be observed in graph of Figure 17, a higher increase is evident between the external pipe and the small internal pipe that difference in near to 5°C. Also, the highest difference was in systems with cover or without cover, a total of 31% was the increase of cover system regard to not cover system.

![Figure 16. Thermal performance in D4 receiver.](image)

![Figure 17. Fluid temperature in D4 receiver.](image)
defined inside of the smallest pipe related in the receiver, also the receiver was covered with a glass arrangement. Results for the last pipe in each flow condition is described in Table 11.

Table 11. Temperature results of D5 receiver.

| Maximum Temperature | Flow | Descartes 5 |
|---------------------|------|-------------|
| Fluid Temperature °C| 88.59| all         |
| Solid Temperature °C| 88.59| No Cover    |
| Fluid Temperature °C| 84.84| PE          |
| Solid Temperature °C| 84.84| No Cover    |
| Fluid Temperature °C| 90.27| PI all      |
| Solid Temperature °C| 90.27| No Cover    |
| Fluid Temperature °C| 91.83| PI B        |
| Solid Temperature °C| 91.83| No Cover    |
| Fluid Temperature °C| 92.84| PI S        |
| Solid Temperature °C| 92.84| No Cover    |
| Fluid Temperature °C| 97.82| PI ES       |
| Solid Temperature °C| 97.82| No Cover    |
| Fluid Temperature °C| 131.36| PI ES      |
| Solid Temperature °C| 131.36| Cover      |

Finally, fluid temperature graphs of each configuration in D5 receiver study are presented in the scheme of Figure 19. Just like, Fractal Descartes pipes a high difference is evident between results obtained in systems with cover or un-cover. Regard to cylindrical receiver the fractal D5 receiver increase outlet temperature in 29% the highest increment in all study.

From Figure 18.A to Figure 18.G thermal performance and distribution inside each pipe and cover can be observed. The first Figure 18.A relates system with a flow conduction in all pipes and not a cover, second Figure 18.B illustrate system with a flow conduction only inside the external pipe and without a cover. Figure 18.C shows system with the flow conduction in all internal pipes without an external cover, then Figure 18.D represents results for system the bigger internal pipes flow conduction and not cover. Figure 18.E relates temperatures of system with flow conduction in the small internal pipes, similarly Figure 18.F and Figure 18.G presents system with flow conduction in the smallest internal pipes without a cover and cover respectively.

Analyzing the number of internal pipes in the cylindrical receiver and how influence temperature results, was possible to determinate that at the first third iteration temperature decrease regard to the previous iteration, however when the number of internal pipes increase and free areas are filled the thermal performance of the collector increase, as show graph in Figure 20.

Contrasting Descartes model with cylindrical receiver, a delta temperature of 17.2°C was calculated between D5 pipe
and cylindrical pipe, on the other hand, comparing systems covered delta temperature increase to 29.3°C, as show Figure 21.

Figure 21. Cylindrical-Fractal Descartes.

The best result of each proposed model is related in Figure 22, as was describes the maximum temperatures as fluid as solid were obtained with fractal Descartes pipes and fractal fins pipes. For both cases, reducing internal areas in a conventional cylindrical receiver is possible to increase thermal performance, the areas that are not available works like an energy capacitor or thermal storage which keep those zones under thermal conditions were losses are reduced.

Figure 22. Contrast systems.

Resuming all results obtained in this study, Table 12 (cylindrical, fractal fins and fractal Descartes receivers) the lowest thermal performance was achieved with a cylindrical pipe, instead fractal fins receiver with 5 internal fins achieves a temperature increase of 27%, finally the best pipe configuration was fractal Descartes with 12 internal pipes which e the cylindrical temperature in 29%.

| Pipe       | Internal-Flow | Number of Fins | Average Temperature | Maximum Temperature |
|------------|---------------|----------------|---------------------|---------------------|
| Cylindrical| All Fins      | 10             | 85.01°C             | 91.29°C             |
| Fractal Fins| 2 Fins       | 6              | 97.61°C             | 103.79°C            |
| Fractal Fins| 1 Fins       | 5              | 97.96°C             | 104.53°C            |
| Fractal Fins| Fin-cover    | 5              | 122.26°C            | 129.39°C            |
| Descartes  | All pipes    | 5              | 79.79°C             | 88.59°C             |
| Descartes  | External     | 4              | 79.71°C             | 88.21°C             |
| Descartes  | Internal     | 5              | 92.04°C             | 97.82°C             |
| Descartes  | Internal cover | 5            | 118.43°C            | 131.36°C            |

IV. CONCLUSIONS

Thermal performance analysis of a parabolic through collector with receivers internal fractal fins were studied and compared to a regular parabolic collector with cylindrical receiver, in the studied was possible to obtain a lowest thermal performance was achieved with a cylindrical pipe, instead fractal fins receiver with 5 internal fins achieves a temperature increase of 27%, finally the best pipe configuration was fractal Descartes with 12 internal pipes which e the cylindrical temperature in 29%.

A glass cover on receivers shows a significant increase in fluid and solid temperature in all systems analyzed, for cylindrical receiver delta temperature between cover and not cover was almost 22°C. Cover influence in fractal 5 fins receiver delta temperature was almost 25°C and 33.54°C. in Descartes D5 receiver.
ACKNOWLEDGMENT
The authors would like to thank the Nueva Granada Military University research center for financing this work (research project IMP-ING-2656, 2019).

V. REFERENCES

[1] A. Kumar, O. Prakash, and A. Dube, “A review on progress of concentrated solar power in India,” Renew. Sustain. Energy Rev., vol. 79, pp. 304–307, Nov. 2017.

[2] T. Bouhal et al., “Technical feasibility of a sustainable Concentrated Solar Power in Morocco through an energy analysis,” Renew. Sustain. Energy Rev., vol. 81, pp. 1087–1095, Jan. 2018.

[3] A. Kassem, K. Al-Haddad, and D. Komljenovic, “Concentrated solar thermal power in Saudi Arabia: Definition and simulation of alternative scenarios,” Renew. Sustain. Energy Rev., vol. 80, pp. 75–91, Dec. 2017.

[4] R. Ling-zhi, Z. Xin-gang, Z. Yu-zhuo, and L. Yan-bin, “The economic performance of concentrated solar power industry in China,” J. Clean. Prod., vol. 205, pp. 799–813, Dec. 2018.

[5] H. Martín, J. de la Hoz, G. Velasco, M. Castilla, and J. L. García de Vicuña, “Promotion of concentrating solar thermal power (CSP) in Spain: Performance analysis of the period 1998–2013,” Renew. Sustain. Energy Rev., vol. 50, pp. 1052–1068, Oct. 2015.

[6] U. Pelay, L. Luo, Y. Fan, D. Stitou, and M. Rood, “Technical data for concentrated solar power plants in operation, under construction and in project,” Data Br., vol. 13, pp. 597–599, 2017.

[7] L. El-Katiri, A roadmap for renewable energy in the Middle East and North Africa. Oxford institute for energy studies, 2014.

[8] G. Picotti, P. Borghesani, G. Manzolini, M. E. Cholette, and R. Wang, “Development and experimental validation of a physical model for the soiling of mirrors for CSP industry applications,” Sol. Energy, vol. 173, pp. 1287–1305, Oct. 2018.

[9] J. Lilliestam et al., “Policies to keep and expand the option of concentrating solar power for dispatchable renewable electricity,” Energy Policy, vol. 116, pp. 193–197, May 2018.

[10] F. Trieb, T. Fichter, and M. Moser, “Concentrating solar power in a sustainable future electricity mix,” Sustain. Sci., vol. 9, no. 1, pp. 47–60, Jan. 2014.

[11] R. Ling-zhi, Z. xin-gang, Y. Xin-xuan, and Z. Yu-zhuo, “Cost-benefit evolution for concentrated solar power in China,” J. Clean. Prod., vol. 190, pp. 471–482, Jul. 2018.

[12] F. Fornarelli et al., “CFD analysis of melting process in a shell-and-tube latent heat storage for concentrated solar power plants,” Appl. Energy, vol. 164, pp. 711–722, Feb. 2016.

[13] M. Cagnoli, A. de la Calle, J. Pye, L. Savoldi, and R. Zanino, “A CFD-supported dynamic system-level model of a sodium-cooled billboard-type receiver for central tower CSP applications,” Sol. Energy, vol. 177, pp. 576–594, Jan. 2019.

[14] S. Riahi, W. Y. Saman, F. Bruno, M. Belusko, and N. H. S. Tay, “Performance comparison of latent heat storage systems comprising plate fins with different shell and tube configurations,” Appl. Energy, vol. 212, pp. 1095–1106, Feb. 2018.

[15] A. Solé, Q. Falcoz, L. F. Cabeza, and P. Neveu, “Geometry optimization of a heat storage system for concentrated solar power plants (CSP),” Renew. Energy, vol. 123, pp. 227–235, Aug. 2018.

[16] M. Bichotte et al., “High efficiency concentrated solar power plant receivers using periodic microstructured absorbing layers,” Sol. Energy Mater. Sol. Cells, vol. 160, pp. 328–334, Feb. 2017.

[17] A. M. 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,” Appl. Therm. Eng., vol. 122, pp. 626–641, Jul. 2017.

[18] T. Li, R. Wang, J. Kiplagat, and Y. Kang, “Performance analysis of an integrated energy storage and energy upgrade thermochemical solid–gas sorption system for seasonal storage of solar thermal,” Energy, vol. 50, pp. 454–467, 2013.

[19] X. Meng, X. Xia, C. Sun, and X. Hou, “Adjustment, error analysis and modular strategy for Space Solar Power Station,” Energy Convers. Manag., vol. 85, pp. 292–301, 2014.

[20] R. Ghatak, B. Biswas, A. Karmakar, and D. R. Poddar, “A Circular Fractal UWB Antenna Based on Descartes Circle Theorem With Band Rejection Capability,” Prog. Electromagn. Res. C, vol. 37, pp. 235–248, 2013.

[21] S. Gupta, M. Chauhan, and B. Mukherjee, “Fractal on Hemispherical DRA by Descarte’s Circle Theorem for wideband application,” in 2018 Conference on Information and Communication Technology (CICT), 2018, pp. 1–6.

[22] R. E. Schwartz and S. Tabachnikov, “Descartes Circle Theorem, Steiner Porism, and Spherical Designs,” Nov. 2018.