Enhancing concentration ratio of solar concentrators

Application of a flat secondary reflector in parabolic trough collectors

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Mechanical and Manufacturing Engineering

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Declaration

I certify that except where due acknowledgement has been made, the work is that of the author alone; the work has not been submitted previously, in whole or in part, to qualify for any other academic award; the content of the thesis is the result of work which has been carried out since the official commencement date of the approved research program; any editorial work, paid or unpaid, carried out by a third party is acknowledged; and, ethics procedures and guidelines have been followed.

David Rodriguez Sanchez

February 2017
To my grandfather, who will not see these lines. Every time we spoke, he used to ask:

'¿Cómo va la faena?' / How's the work going?

And now, I could finally tell him

'Ya he acabado la faena, yayo' / I have finished the work, grandpa.

And to my grandmother, my 'yaya' Encarna, who sadly passed away just two weeks ago, while I was working on this thesis final version.

I love you two so much... I will miss you

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Abstract

Solar irradiation is approximately $1\text{kW/m}^2$ on the surface of the Earth and utilising only 1 % of global solar resources could be enough to achieve the recommendations of international organisations to achieve a long-term climate stabilisation. Concentration of solar energy is a possible solution to overcome the low density disadvantage and to achieve a more efficient conversion to other types of energy such as electricity.

Concentration is limited by the Sun-Earth geometry. The maximum limit of concentration for an ideal concentrator is established by the laws of thermodynamics for those concentrators which concentrate the Sun onto a point-like receiver (3D concentrators). In the past, a simplification of this limit was conducted to calculate the limits of those concentrators which concentrate the Sun onto a linear receiver (2D), and it was called the thermodynamics limit of 2D receivers. This limit was not directly calculated as a product of thermodynamics laws but as a simplification of the 3D case and in this work, it will be demonstrated how this limit was underestimated by 20 %.

Solar concentrators have been designed in the past considering that no rays are to be missed in the receiver after being reflected (or refracted) in the primary concentrator. This assumption makes it possible to analyse concentrators by considering a cross-section along the symmetry plane of the systems. By applying the edge-ray theorem while maintaining the etendue of the solar bundle, the receiver size can be calculated for a given concentrator. Most of solar concentrators are based on parabolic reflectors and the assumption that no rays can miss the receiver further limits the maximum concentration achievable. This limitation depends on whether the parabolic concentrator is a dish (3D) or a trough (2D) and ranges between $\frac{1}{4}$ and $\frac{1}{2}$ the theoretical limits of concentration.

In this work, an increase of concentration ratio is explored by sacrificing optical efficiency. If rays are missed in the absorber, concentrations up to the theoretical limits are found for parabolic concentrators by maintaining the simplification of the cross-section analysis. Further research demonstrates how this simplification, though practical for designing 100 % optical efficient concentrators and accurate if no rays are missed, is not accurate if rays are missing the target. Calculations of concentration for parabolic concentrators without cross-section simplifications conducted in this work lead to the conclusion that the thermodynamic limits of concentration are achievable for ideal 3D parabolic concentrators while there can be a concentration surpass of a 20 % the previously stated as limit of concentration for 2D systems. These new limits are reached with a reduction in optical efficiency and they would be not practical for real concentrators. However, it is demonstrated how concentration ratios around 80 % the previous theoretical limit can be achieved for parabolic troughs with cylindrical receivers while maintaining optical efficiencies above 70 % with concentrations close to the previous limit can be achieved with troughs and flat receivers maintaining optical efficiency above 80 %. Ray tracing simulations conducted to validate the theoretical development matched the results obtained in this thesis.

The missed radiation can be redirected to the receivers by the addition of a second-stage optics. Previous works have addressed complicated geometries for the secondary optics, and concentration and thermal improvements have been achieved. However, due to the complicated geometries required, the large dimensions of the secondary mirrors or to a combination of both, the proposed secondary optics have not become a practical solution. In this thesis, a flat secondary reflector is proposed as an alternative to increase the concentration ratio while maintaining simplicity. A theoretical analysis to calculate the appropriate
dimensions of a secondary flat reflector for a given primary mirror is developed in this thesis and ray tracing is conducted to validate the equations obtained. A flat reflector will have a minimal economic impact in the cost of a parabolic trough and it allows larger concentration ratios for identical primary mirror areas compared to a standard parabolic trough. Increases of concentration ratio up to 80% are observed when a secondary flat reflector is inserted in a commercial system, while the shadow area introduced in the primary mirror is usually less than 15% of the primary mirror area. The increase in pumping power is offset by the increase in system efficiency.

The inclusion of the flat secondary reflector changes the trough tolerance to misalignments and the flux distribution along the receiver's surface. Both effects are analysed with ray tracing simulations considering the flat secondary mirrors and the original absorber of representative benchmark parabolic mirrors. In some cases, the required dimensions for the flat secondary reflector would make it a non-realistic solution due to the impossibility of encapsulating it within a glass cover. To overcome this issue a proposal of a shortened version of the secondary mirror is evaluated in the simulations. A shortened secondary flat reflector will decrease the pernicious effects of the shading that the secondary mirror projects over the primary mirror, but will change the tolerance to misalignments of the troughs as well.

An additional advantage of the inclusion of a secondary optics in the parabolic trough is the increase of uniformity of the flux around the absorber; that will reduce temperature gradients and thermal stress and could have implications on the thermal performance of the troughs. Literature is not completely clear about the effect of the flux distribution on the performance of parabolic trough receivers and to offer a better understanding of them CFD simulations are conducted for an evacuated and a non-evacuated absorber under different flux profile conditions, considering as well the influence of flow rate in the heat transfer fluid and wind velocity on the glass cover. The influence of the flux distribution on the thermal performance of the absorbers is demonstrated, but the decrease of performance caused by irregular fluxes can be compensated with an increase of flow rate which will not cause a noticeable increase in pumping power.

The increase of the concentration achieved with the secondary flat reflector is significant, but the associated shading will decrease the energy input to the mirror. CFD simulations for two representative benchmark absorbers and their modifications with a secondary flat receiver show a potential increase of performance, especially for an evacuated tube and a shortened flat secondary reflector. A similar performance was obtained when evaluating the receivers with no wind, maximum flow rates, and low emissivities. However, the secondary flat reflector receiver will improve the performance of parabolic trough receivers at maximum flow rates up to 3% if emissivity and wind velocity increases and the increases are higher if the flow rate is decreased. As power plants are required to work at different flow rates depending on the solar irradiation available, the lower drop of performance when the flow rate is decreased would help to increase the efficiency of the plants.

In the case of the non-evacuated absorber, the increase of performance does not happen in the range of working temperatures of those collectors, which is lower than the evacuated ones, if low emissivities are considered. However, the emissivity of non-evacuated absorbers is normally higher than the evacuated ones. The inclusion of an insulation layer in the upper part of the receiver or the modification of the secondary reflector to act as a flat plate collector in its upper surface to recover the shadow losses would make the secondary flat reflector receiver offer higher performance than the standard one.

In any case, it is shown how the secondary flat reflector can improve the thermal performance of the collectors if the output temperature is increased. This would help to increase the efficiency of the Rankine cycle for electricity generation or increase the applications for industrial heat. If alternative fluids such as Molten Salts allow working temperatures near 500 °C the flat secondary receivers could become a promising
solution to increase parabolic troughs performance.

Finally, an experimental comparison was conducted for both a standard and a secondary flat non-evacuated absorbers. Due to limitations in the experimental facility the temperature range of the test was limited to 100 °C. In this case, due to real misalignments and inaccuracies on the primary mirror, the flat secondary reflector offered a higher performance than the standard one, and it is proof that, in real conditions, the optical efficiency of those collectors can be higher than the standard ones due to a better tolerance to misalignments.
## Contents

**Nomenclature**

Chapter 1: Introduction

1.1: Thesis outline  
1.2: Scientific contributions of this thesis

Chapter 2: Literature review

2.1: Generalities  
2.2: Parabolic trough collectors. State-of-the-art  
2.3: Parabolic trough collectors. Background  
2.4: The flux distribution on a parabolic-trough  
2.5: Secondary optics on parabolic-trough collectors  
2.6: Definitions  
2.7: Summary

Chapter 3: Limits of concentration on parabolic concentrators

3.1: Introduction  
3.2: Thermodynamics limits of concentration for solar collectors  
3.3: Concentration ratio of parabolic surfaces if no rays are missed  
3.3.1: Non-planar receiver  
3.3.2: Planar absorbers  
3.4: Maximum concentration ratio if rays are to be missed  
3.4.1: Maximum concentration ratio on parabolic concentrators with bundle plane simplification  
3.5: Maximum concentration ratio without bundle-plane simplification  
3.5.1: Missed rays on a 3D receivers  
3.5.2: Concentration limits of 2D concentrators if no cross section simplification is applied  
3.6: Ray tracing simulations  
3.7: Optical efficiency versus concentration ratio  
3.8: Summary and discussion.

Chapter 4: Improving the concentration ratio with a secondary reflector

4.1: Introduction  
4.2: Theoretical development
Chapter 6: The prototypes

6.4.1: The prototypes 141
6.4.2: Thermal performance calculation 144
6.5: Results 146
6.6: Summary 151

Chapter 7: Conclusions and future work 153

7.1: Conclusions 153
7.2: Future work and recommendations 156

Chapter 8: References 158

Appendix A: Validation simulations results 167
Appendix B: Mesh independence study 169
Appendix C: Material properties 174
Appendix D: Trigonometrical identities 176
Appendix E: Derivation 177
Appendix F: Uncertainty analysis 178
List of Figures

Figure 2.1: Solar cone radial distribution measured from different locations. Adapted from [8]. Different lines represent different locations.

Figure 2.2: Acceptance angle of the absorber for various circumsolar ratios (CSR) vs the overall optical efficiency of a line focus imaging concentrator plotted. Adapted from [11].

Figure 2.3: General schematic of a CSP plant. Copyright free images downloaded from [28-32].

Figure 2.4: Focused receiver during the preheating step (left), and during the steam generation step (right) [53].

Figure 2.5: PTC showing the acceptance angle, $\theta$, and the rim angle $\psi$.

Figure 2.6: Analysis of angular twist and radiative spillage as result of wind load on the Solar Collector Assembly Eurotrough collector with 100 and 150 m (ET-4 ET-6), and the reference collector (LS-3) [71].

Figure 2.7: Flux profile in half a cylindrical absorber obtained from Jeter's formulation [76].

Figure 2.8: Absorber displaced from the focal line of the parabolic mirror due to thermal expansions [87].

Figure 2.9: Three types of LTP solar concentrators classified by the degrees of freedom of the secondary concentrator stage: a) fixed secondary optics; b) discrete-switching secondary optics; c) continuous-tracking secondary optics [111].

Figure 2.10: Schematic cross-section of solar air receiver based on an array of cross-flow cavities (left) and CAD view (right) [45].

Figure 2.11: Sketches of secondary reflectors addressed adapted from those found in the literature [116-120].
   a) Helmet concentrator; b) Single mirror two stage with snail reconcentrator; c) Seagull concentrator; d) Snail concentrator on parabolic trough.

Figure 2.12: Secondary reflectors evaluated in [121] a) Reflective glass surface; b) Reflective annulus insulation; c) Aplanatic secondary reflector; d) Tailored seagull reflector.

Figure 2.13: Comparison between the secondary reflectors proposed by Canavarro et al. [125].

Figure 3.1: Sun - Earth Geometry.

Figure 3.2: Section of a parabolic reflector surface and a non-planar receiver.

Figure 3.3: Planar receiver projections for a single point of the parabola.

Figure 3.4: Planar absorber projections.
Figure 3.5: Parabolic reflector with a non-planar receiver which misses part of the incoming radiation.

Figure 3.6: Parabolic concentrator cross section and planar receiver.

Figure 3.7: Difference of the two flat mirror dimensions compared with the focal distance.

Figure 3.8: Intercept factor for different , bundle-plane simplification considered. a) Parabolic dish with spherical absorber. b) Parabolic dish with planar absorber. c) Parabolic trough with cylindrical absorber. d) Parabolic trough with planar absorber.

Figure 3.9: Concentration ratio for parabolic troughs with non-planar receivers missing rays. a) Relative to the thermodynamics limit concentration ratio for different design rim angles b) Size comparison of an absorber designed to not miss rays and an absorber with . Note that the solar cone has been exaggerated.

Figure 3.10: Relative to the thermodynamics limit concentration ratio for parabolic trough with planar receivers missing rays.

Figure 3.11: Apollonius’ cone.

Figure 3.12: Projections of the solar bundle after being reflected in a point of a parabolic dish.

Figure 3.13: Intercepted rays projection.

Figure 3.14: Concentration ratio achievable with a parabolic dish and a spherical receiver with \( \eta_{\text{opt}} < 1 \).

Figure 3.15: Parabolic dish with rays reflected in a point of the reflector for which the absorber misses rays.

Figure 3.16: Illumination of the receiver for a point of the parabola close to the design point.

Figure 3.17: Ray tracing conducted with ten million rays to prove the uniformity of the flux at the focal plane. a) 3D model of the dish, only the reflected rays are shown b) Flux distribution at the focal plane, colour bar represents number of hits.

Figure 3.18: 3D schematics of the missed energy of one point of the reflector with \( \psi > \psi_d \).

Figure 3.19: Schematic of the intercepted energy of a differential point of reflector.

Figure 3.20: Schematic of the circular and triangular sector.

Figure 3.21: Projection of the rays reflected in a differential element of area of the parabolic mirror.

Figure 3.22: Intercepted rays sketch.

Figure 3.23: Analysis of the simplifications assumed in the ellipse section calculations.

Figure 3.24: Comparison between analytical equations (line) and Zemax simulations (points). a) Parabolic dish and spherical receiver, b) parabolic dish and planar receiver, c) parabolic trough and
cylindrical receiver, d) parabolic trough and planar receiver.

Figure 3.25: Optical efficiency vs concentration ratio for a mirror focusing light on a receiver missing rays using the bundle-plane simplification. a) Parabolic dish with spherical absorber. b) Parabolic dish with planar absorber. c) Parabolic trough with cylindrical absorber. d) Parabolic trough with planar absorber.

Figure 3.26: Optical efficiency vs concentration ratio for a mirror focusing light on a receiver missing rays without using the bundle-plane simplification. a) Parabolic dish with spherical absorber. b) Parabolic dish with planar absorber. c) Parabolic trough with cylindrical absorber. d) Parabolic trough with planar absorber.

Figure 3.27: Ray tracing simulations of a parabolic trough with a low \( \psi_d \) and at different \( \psi \) a) Cylindrical receiver \( \psi = \psi_d \) b) Flat receiver \( \psi = \psi_d \) c) Cylindrical receiver \( \psi > \psi_d \) d) Flat receiver \( \psi > \psi_d \) e) Cylindrical receiver \( \psi > \psi_d \) f) Flat receiver \( \psi > \psi_d \)

Figure 3.28: trade-off between optical efficiency and relative concentration ratio for a parabolic trough with cylindrical absorber.

Figure 4.1: PTC with a secondary flat reflector and gap losses.

Figure 4.2: Parabola geometrical parameters.

Figure 4.3: a) Secondary reflector projections. b) Detail of the projections’ dimensions and their relevant angles.

Figure 4.4: Sketch of the five rays necessary in the study to determine the absorber’s size and position.

Figure 4.5: Rays 2 and 4 projections.

Figure 4.6: Minimum dimension for an absorber capable of capturing rays 1 and 2; b) Minimum dimension for an absorber capable of capturing rays 1 and 3.

Figure 4.7: Ray tracing validation simulations.

Figure 4.8: Comparison of the concentration ratio of a PTC collector for different \( \psi \). a) \( F=647.5 \)mm; \( \theta = 1.02^\circ \) b) \( F=680 \)mm; \( \theta = 0.96^\circ \) c) \( F=1400 \)mm; \( \theta = 0.8^\circ \) d) \( F=1710 \)mm; \( \theta = 0.69^\circ \). The vertical discontinuous line represents the the commercial collector’s \( \psi \).

Figure 4.9: Energy received in the absorber relative to that for no secondary mirror for different misalignments and secondary mirror reflectivities. a) NEP collector; b) LS1 collector; c) LS2 collector; d) LS3 collector.

Figure 4.10: Flux distribution around the absorber of the NEP and LS3 collectors with and without a secondary reflector for different mirror misalignments. a) NEP, perfect alignment. b) LS3, perfect alignment. c) NEP, half maximum misalignment. d) LS3, half maximum misalignment. e) NEP, maximum misalignment. f) LS3, maximum misalignment. 0° represents the bottom of the absorber.
Figure 4.11: Misalignments analysis for the NEP mirror and its standard and SFR receivers. a) Horizontal misalignments b) Vertical misalignments c) Rotational misalignments.

Figure 4.12: Misalignments analysis for the LS3/ET mirror and its standard and SFR receivers. a) Horizontal misalignments b) Vertical misalignments c) Rotational misalignments.

Figure 5.1: Heat transfer mechanisms considered in the CFD model.

Figure 5.2: Local concentration ratio obtained in Zemax and its approximation with Fourier series five to seven terms.

Figure 5.3: Flux profiles obtained with Zemax and their Fourier approximations. a) NEP receiver b) NEP with SFR receiver c) LS3 receiver d) LS3 with SFR receiver.

Figure 5.4: Isothermal concentric cylinders.

Figure 5.5: Velocity contours in the glass cavity for vacuum conditions.

Figure 5.6: Simulation conducted to validate CFD radiation model, theoretical results obtained with equation 5.11.

Figure 5.7: Characteristic numbers for air at different temperatures. a) Prandtl number b) Rayleigh number

Figure 5.8: Product of Rayleigh and the geometric factor with NEP’s dimensions concentric cylinders geometry.

Figure 5.9: CFD validation for the natural convection model using different air properties.

Figure 5.10: Heat transferred to the fluid for $v_{wind} = 0$ m/s and $\varepsilon = 0.1$. a) Realistic flux profile; b) Uniform flux profile; c) Max concentration in a sector; d) Realistic flux vs Uniform profile.

Figure 5.11: Heat transferred to the fluid for $v_{wind} = 15$ m/s and $\varepsilon = 0.1$. a) Realistic flux profile; b) Uniform flux profile; c) Max concentration in a sector; d) Realistic flux vs Uniform profile.

Figure 5.12: Heat transferred to the fluid for $v_{wind} = 0$ m/s and $\varepsilon = 0.2$. a) Realistic flux profile; b) Uniform flux profile; c) Max concentration in a sector; d) Realistic flux vs Uniform profile.

Figure 5.13: Heat transferred to the fluid for $v_{wind} = 15$ m/s and $\varepsilon = 0.2$. a) Realistic flux profile; b) Uniform flux profile; c) Max concentration in a sector; d) Realistic flux vs Uniform profile.

Figure 5.14: Temperature on the absorber’s outer surface. a) $\dot{m} = 0.1$ kg/s; $\varepsilon = 0.1$; $T_{fluid} = 290$ °C b) $\dot{m} = 2.5$ kg/s; $\varepsilon = 0.2$; $T_{fluid} = 425$ °C.

Figure 5.15: Heat transfer in the non-evacuated absorber for different flux profiles a) $\varepsilon = 0.1$; $v_{wind} = 0$ m/s; $\dot{m} = 0.05$ kg/s b) $\varepsilon = 0.1$; $v_{wind} = 15$ m/s; $\dot{m} = 1$ kg/s c) $\varepsilon = 0.2$; $v_{wind} = 0$ m/s; $\dot{m} = 0.05$ kg/s d) $\varepsilon = 0.2$; $v_{wind} = 15$ m/s; $\dot{m} = 1$ kg/s.
Figure 5.16: Temperature distribution on the absorber's surface a) $\dot{m} = 0.05 \text{ kg/s}; \varepsilon = 0.1; T_{\text{fluid}} = 250 \degree \text{C}$

b) $\dot{m} = 1 \text{ kg/s}; \varepsilon = 0.1; T_{\text{fluid}} = 250 \degree \text{C}$.

Figure 5.17: Schott PR70 and CFD simulations thermal losses. The CFD model considered a uniform flux, $\varepsilon = 0.1$, zero wind speed and an ambient temperature of 20 \degree \text{C}.

Figure 5.18: HTF heat transfer coefficients calculated with Petukov equations for: a) LS3 absorber.

b) LS3-SFR absorber.

Figure 5.19: Standard and SFR receiver for the LS3 primary mirror. a) $\varepsilon=0.1; v_{\text{wind}} = 0 \text{ m/s}$. b) $\varepsilon = 0.1$; $v_{\text{wind}} = 0 \text{ m/s}$. c) $\varepsilon = 0.1$; $v_{\text{wind}} = 15 \text{ m/s}$. d) $\varepsilon = 0.1$; $v_{\text{wind}} = 15 \text{ m/s}$.

Figure 5.20: Standard and SFR receivers for the LS3 primary mirror. a) $\varepsilon=0.2; v_{\text{wind}} = 0 \text{ m/s}$. b) $\varepsilon = 0.2$; $v_{\text{wind}} = 0 \text{ m/s}$. c) $\varepsilon = 0.2$; $v_{\text{wind}} = 15 \text{ m/s}$. d) $\varepsilon = 0.2$; $v_{\text{wind}} = 15 \text{ m/s}$.

Figure 5.21: Temperature profile around the LS3 absorber's surface: a) $\dot{m} = 0.1 \text{ kg/s}$ b) $\dot{m} = 2.5 \text{ kg/s}$.

Figure 5.22: Sketch of the two models of SFR receiver simulated for the non-evacuated receiver a) without insulation b) with insulation.

Figure 5.23: HTF heat transfer coefficients calculated with Petukov equations for: a) NEP receiver.

b) NEP-SFR receiver.

Figure 5.24: Heat transfer coefficient between glass and air for the NEP absorbers; $v_{\text{wind}} = 0 \text{ m/s}$.

Figure 5.25: Standard and SFR receivers for the NEP primary mirror. a) $\varepsilon = 0.1$; $v_{\text{wind}} = 0 \text{ m/s}$. b) $\varepsilon = 0.1$; $v_{\text{wind}} = 0 \text{ m/s}$. c) $\varepsilon = 0.1$; $v_{\text{wind}} = 15 \text{ m/s}$. d) $\varepsilon = 0.1$; $v_{\text{wind}} = 15 \text{ m/s}$.

Figure 5.26: Standard and SFR receivers for the NEP primary mirror. a) $\varepsilon = 0.2$; $v_{\text{wind}} = 0 \text{ m/s}$. b) $\varepsilon = 0.2$; $v_{\text{wind}} = 0 \text{ m/s}$. c) $\varepsilon = 0.2$; $v_{\text{wind}} = 15 \text{ m/s}$. d) $\varepsilon = 0.2$; $v_{\text{wind}} = 15 \text{ m/s}$.

Figure 5.27: Sketch of the SFR receiver (without glass cover) showing a possible thermal connection if the top face of the SFR is selective coated.
Figure 6.8: SFR receiver installed on the NEP primary mirror.

Figure 6.9: End of collector losses.

Figure 6.10: Inlet temperature over time with cooling activated and external heating deactivated. Test conducted to investigate the minimum set temperature which would allow stable conditions.

Figure 6.11: Test conducted over time with automatised tracking.

Figure 6.12: Glare of the concentrated flux reflected on the primary mirror during the tests.

Figure 6.13: Measurement of two different set points.

Figure 6.14: Tests conducted with the two prototypes and CFD comparison. a) Standard receiver b) SFR receiver.

Figure B.1: Two of the meshes tested in the mesh independence study and the imaginary line to obtain the temperature and velocity plots (in orange) a) NEP Standard b) NEP SFR.

Figure B.2: NEP standard mesh independence analysis a) Temperature b) Velocity.

Figure B.3: NEP SFR mesh independence analysis a) Temperature b) Velocity.

Figure B.4: LS3/ET standard mesh independence analysis a) Temperature b) Velocity.

Figure B.5: LS3/ET mesh independence analysis a) Temperature b) Velocity.

Figure C.1: Thermal properties of Therminol 66.

Figure C.2: Thermal properties of air.
Index of Tables

Table 2.1: PT Solar power plants size. Modified from [35, 40]: 31
Table 2.2. PTC used in this work. Data extracted and modified (where required) from [35, 40, 66]: 32
Table 3.1: Overview of limits of concentration: 75
Table 4.1: Calculated $r_{1,2}$ and $r_{1,3}$ for different PTCs: 85
Table 4.2: Concentration ratio of different PTCs with and without secondary reflector: 87
Table 4.3: Primary mirrors and SFR modifications considered in the misalignments analysis: 92
Table 5.1: Pumping power requirements estimation for the NEP absorber: 98
Table 5.2: Pumping power requirements estimation for the LS3 absorber: 99
Table 5.3: $C$ and $n$ to calculate heat transfer coefficient in equation 5.6. Fluid evaluated at 50°C: 104
Table 5.4. Different air thermal properties tested in the natural convection models: 112
Table 5.5: Heat transfer coefficients calculated for the LS3 absorbers' glasses: 122
Table 5.6: Heat transfer coefficients calculated for the NEP receivers' glasses: 128
Table 6.1: Uncertainties of the relevant variables used in the experiment: 138
Table 6.2: Emissivities obtained for the 4 samples coated with the cylindrical absorbers: 140
Table A.1: Radiation model validation case 1. $\varepsilon_{\text{inner}}=0.95; \varepsilon_{\text{outer}}=0.2; T_{\text{out}}=25$ °C and mesh independence analysis: 155
Table A.2: Radiation model validation case 2. $\varepsilon_{\text{inner}}=0.25; \varepsilon_{\text{outer}}=0.9; T_{\text{out}}=25$ °C: 155
Table A.3: Radiation model validation case 3. $\varepsilon_{\text{inner}}=0.25; \varepsilon_{\text{outer}}=0.2; T_{\text{out}}=25$ °C: 155
Table A.4: Radiation model validation case 4. $\varepsilon_{\text{inner}}=0.95; \varepsilon_{\text{outer}}=0.9; T_{\text{out}}=25$ °C: 156
Table B.1: Dimensions of the NEP standard receiver model: 158
Table B.2: Quality statistics of the NEP standard receiver meshes: 158
Table B.3: Dimensions of the NEP standard receiver model: 159
Table B.4: Quality statistics of the NEP SFR receiver meshes: 159
Table B.5: Dimensions of the LS3/ET standard receiver model: 160
Table B.6: Quality statistics of the LS3/ET standard receiver meshes: 160
Table B.7: Dimensions of the LS3/ET SFR receiver model:  

Table B.8: Quality statistics of the LS3/ET SFR receiver meshes:
Nomenclature

Abbreviations

CFD: Computing Fluid Dynamics
CPC: Compound Parabolic Concentrator
CSP: Concentrating Solar Power
DNI: Direct Normal Irradiation

HTF: Heat Transfer Fluid
LTP: Line to point
PTC: Parabolic trough collector

Roman symbols

A: Area
A\text{p}^\circ: Aperture area
a: Absorber area
B,: Bias error
C: Concentration ratio
C_0^\circ: Constant coefficient
C_p^\circ: Specific heat
c_0^\circ: Speed of light at vacuum
D: Diameter
d_{\text{sun}}: Sun distance, evaluated as \(1.4895\times10^{11}\) m \(\pm 1.7\%\)
E_{b\lambda}: Blackbody emissive power
F: Focal distance
F_{cyf}: Geometric factor for concentric cylinders
F_{i,j}: View factor
F_m^\circ: Average focal distance
f: Skin friction coefficient
f_{in}^\circ: Moody's friction coefficient
Gr: Grashof number
g: Gravity
h: heat transfer coefficient
h_{0}^\circ: Planck's constant
I: Intercept factor
k_{0}^\circ: Boltzmann's constant
k_{\text{eff}}^\circ: Effective thermal conductivity
k_f^\circ: Fluid thermal conductivity
k_c^\circ: Thermal conductivity
L: Length
L_c^\circ: Characteristic length
m: Constant exponent
N: Day of the year
N_{\text{Nu}}^\circ: Nusselt number
m_c^\circ: Constant exponents
P: Pressure
Pr: Prandlt number
P_e^\circ: Precision error
PV: Photovoltaic
\dot{Q}: Radiation heat transfer
\dot{Q}_{ij}^\circ: Heat exchange between the body i and the body j
\dot{Q}_{\text{con}\circ}^\circ: Conduction heat transfer
R: Sun radius, evaluated as \(1.39\times10^9\) m
R_{\text{A}L}^\circ: Rayleigh number
r: Radius
r^\circ: Indirect measurement
S_e^\circ: Semi – minor axis of an ellipse
S_x^\circ: Standard deviation of measurement x
T: Temperature
T: Semi-major axis of an ellipse  
\( t \): t-value  
U: Uncertainty  
v: Velocity  
\( \dot{V} \): Volumetric flow rate  
W: Mirror width

\textit{Subscripts}

\texttt{abs}: Absorber  
\texttt{cyl}: Cylindrical receiver  
\texttt{dish}: Parabolic dish concentrator  
\texttt{eff}: Effective  
\texttt{fl}: Film temperature  
\texttt{flat}: Flat receiver  
\texttt{opt}: Optical  
S: Sun  
\texttt{sph}: Spherical receiver

\textit{Greek symbols}

\( \alpha \): Generic angle  
\( \alpha_s \): Sun elevation angle  
\( \beta \): Generic angle  
\( \beta_e \): Thermal expansion coefficient  
\( \gamma \): Generic angle  
\( \delta \): Solar declination  
\( \varepsilon \): Emissivity  
\( \zeta \): Incidence angle  
\( \eta \): Efficiency  
\( \theta \): Half-acceptance angle  
\( \theta_s \): Solar cone semi-angle  
\( \lambda \): Wavelength  
\( \lambda_T \): Latitude, change  
\( \mu \): Dynamic viscosity  
\( \rho \): Density  
\( \sigma \): Stefan Boltzmann constant  
\( \nu \): Kinematic viscosity  
\( \psi \): Rim angle  
\( \psi_s \): Design rim angle  
\( \omega \): Solar hour angle
1.1 Thesis outline

Solar energy is one of the most widespread sources of renewable energy nowadays. Solar energy production, however, is limited by the low density of solar energy, which is approximately 1 kW/m\(^2\). This limitation may reduce the effectiveness of solar energy as a source of energy when compared with non-renewable sources, such as fossil fuel electricity generation. Achieving a higher energy density would help to increase solar energy competitiveness. Concentration increases solar energy density which, in turn, increases the efficiency of conversion of solar energy to electricity or another type of energy. If concentration is employed, higher temperatures can be reached in the solar fields, increasing exergetic levels, and solar energy can then be applied to a wider range of applications, such as solar cooling and different industrial heat processes.

A fraction of the solar energy has a well-known trajectory from the Sun to the Earth surface. Two main physical phenomena can be employed to concentrate solar energy: reflection and refraction. The most common family of concentrators are those based on parabolic reflector surfaces, which reflect the solar energy to a focus if the aperture areas of the surfaces are normal to the solar radiation. Those concentrators, whose focus can be approximated to a point are called 3D concentrators, while those concentrators whose focus approximation is a line are called 2D. An absorber is placed on the focus of the concentrator, so that it will receive the energy and convert it into a usable source of energy - typically heat - by transferring the energy to a fluid or electricity through the installation of photovoltaic cells at the focus.

The parabolic trough is the best-known 2D concentrator; it has been used in power plants for longer than 20 years and they typically receive the energy onto a cylinder which contains a heat transfer fluid. The operating range of parabolic troughs is limited to temperatures around 400 °C due to two main reasons: thermal losses, and heat transfer fluid stability. Increasing their working temperature will increase their competitiveness against other sources of energy. This thesis is devoted to those concentrators, and it evaluates alternatives to improve parabolic troughs performance. Chapter 2, Literature review, develops a state-of-the-art of solar concentration that focuses in parabolic troughs and the use of secondary optics to increase the amount of concentration on them. Despite recent works that state the potential benefits of using a second reflector in the proximities of the absorber, the application of secondary optics has not been extensively explored for parabolic troughs. Further research on the effectiveness of secondary optics is needed in order to assess its potential to increase solar energy concentration.

The investigation developed through this thesis can be divided into two main topics. First the theoretical limits of solar concentration are discussed and later a proposal to enhance concentration in commercial parabolic troughs is presented. The literature demonstrated how any ideal concentrator is limited by thermodynamics, and the maximum concentration achievable is defined by the Sun-Earth geometry. The so-called thermodynamic limit has been evaluated as 46396 for 3D concentrators and as 215.4 for 2D. When a particular concentrator is considered, geometrical limitations decrease the concentration limit. For parabolic concentrators, the geometry of the concentrator will reduce the ideal concentration by a factor between 2 and 4. In the literature, those limits were defined by assuming that all rays reflected by the concentrator were to be received in the absorber. Chapter 3: Limits of concentration on parabolic concentrators develops the limits of concentration for parabolic concentrators without the assumption that all the energy is received in the absorber. Removing such design condition allows for ideal parabolic concentrators to reach the so-called thermodynamic limits in three of the four scenarios considered. Those limits were calculated using a cross-section analysis, simplification that was used to analyse solar concentrators in the literature. A further analysis without the cross-section analysis is conducted to find the limits of concentration of parabolic concentrators that are missing part of the solar radiation. Finally, as the increase of concentration was achieved by a loss of incoming energy, the ratio between optical efficiency and concentration was evaluated.
Past works have not addressed widely the use of secondary optics for parabolic troughs. Research studies that examine secondary mirrors for the troughs are typically based on complicated geometries that, despite improving optical efficiency, do not offer practical solutions. For this reason the second half of this thesis focuses on the use of a simple secondary optics stage to increase the concentration of parabolic trough that could be implemented in existing absorbers without unduly increasing manufacturing costs. The use of a secondary optics mirror has an associated loss of energy due to the shadow projected by the secondary mirror on the primary concentrator, which explains why the total performance of a parabolic trough with secondary optics decreases if compared with a standard absorber on a parabolic primary mirror of equal dimensions.

Chapter 4: Improving the concentration ratio with a secondary reflector develops a theory to calculate the appropriate dimensions to minimise the absorber and the secondary optics size and position on a given primary mirror by maintaining the trough tolerance to misalignments and optical aberrations. A wide range of commercial troughs are analysed, and the potential improvement of concentration is evaluated. Ray tracing simulations were employed to validate the theory developed in this chapter, which exemplifies the capability of the secondary flat reflector.

Later, considering four representative collectors, the flux pattern around the absorber is evaluated. A new flux profile could be employed to increase thermal performance by reducing hot spots and, consequently, thermal losses. More importantly, a new flux profile would reduce thermal stresses that sometimes lead to a drop of optical efficiency or, in a more dramatic scenario, a breakage of the absorber due to an excessive bending. The flux profiles were obtained by ray tracing simulations, and a well-known profile case was used to evaluate the accuracy of the ray tracing models.

Finally, a proposal to reduce shadowing based on the development conducted in chapter 3 is developed for the currently most used mirror dimensions (5.76 m width) and for a smaller mirror (1.2 m width). Both mirrors have a very different rim angle, which is a representative angle related to the ratio defined by the distance from the focus to the mirror and its width. These mirrors will be representative of two different scenarios. Also, the concentration ratio of those mirrors is very different (the 5.76 m wide mirror and its original collector are the absorber/mirror combination with a higher concentration ratio available in the market), while the second mirror has a much lower concentration.

The flux distribution around the absorber surface is affected by the inclusion of a secondary optics. As the flux distribution is defined by the mirror dimensions in a standard parabolic trough, previous research studies have not analysed the benefits of a modified flux distribution - even though some undesirable effects, such thermal stress and bending on the absorber, are caused by the temperature gradients created by the flux distribution along the absorber's surface. In chapter 5, CFD simulations are conducted for the two standard absorbers considering the procedures carried out in chapter 4 to find the effects of flux distribution in the heat transferred to the fluid, as well as the temperature gradients created at the absorber's surface. For that purpose, three different flux profiles around the absorber's surface were simulated: a completely uniform flux, a realistic flux obtained with ray tracing, and an extreme flux profile, which considers that all the incoming energy is received in a narrow sector of the tube.

To predict changes in thermal performance caused by the inclusion of the secondary flat mirror when compared with the standard absorber, CFD simulations were conducted for both the absorbers considering that one of them is evacuated (as in the majority of commercial plants) and the other one is not (as in some commercial collectors, focused in working at lower temperatures). Realistic fluxes were applied for the standard absorbers and those smaller absorbers obtained from the theoretical development conducted in the previous chapter. The secondary flat reflector was considered in two ways: a full size scenario - which will produce a bigger shadow - and a shortened version of the mirror - which will produce an optical loss compensated by the lower shading on the primary mirror. The simulations span over the temperature range available for current commercial oils and evaluate different mass flow rates. The appropriate selection of mass flow rate will enhance heat transfer. Pumping power was analysed to assure that it would not be a
Enhancing concentration ratio of solar concentrators

relevant percentage of the collector's production.

A prototype of a secondary flat reflector receiver was tested in the RMIT university facilities. Its performance was compared with a standard absorber built with the same methodology. The experiment is used as a means to assess whether a secondary flat reflector is capable of redirecting energy effectively to a smaller receiver. The tests couldn’t be conducted at high temperatures due to the impossibility of pressurising the fluid circuit. The experiment development and subsequent conclusions are presented in chapter 6.

Finally, thesis conclusions and recommendations of future work necessary to fully develop the flat secondary reflector receivers for parabolic trough are addressed in chapter 7.

1.2 Scientific contributions of this thesis

The work developed in this thesis can be categorised into two main themes. This thesis presents a new theoretical approach to maximum concentration as an alternative to previous approaches found in the literature. The investigation of a geometrically simple second-stage optics shows that parabolic through concentration can be improved. Some original scientific contributions to the solar energy field can be extracted from this work.

Firstly, this work presents a better understanding of the so-called thermodynamics limits for concentrations, as well as a discussion of the reasons why - considering that no energy is missed - the limits of parabolic concentrators drop far from the theoretical limit.

The analytical equations that demonstrate how parabolic concentrators will approach thermodynamics limits if an optical efficiency is considered can be employed to design parabolic concentrators trying to maximise the optical efficiency – concentration relationship, which can be useful in some applications which require a maximisation of concentration (e.g. achieving higher temperatures).

More importantly, the development of maximum concentration ratio for linear parabolic concentrators without simplifications presents a new concentration limit for 2D concentration, beyond the limits calculated in previous literature.

The optical development conducted to design a secondary flat reflector can be used as a new methodology to design such receivers, showing how an easy geometry can increase the concentration ratio. The equations developed in chapter 4 will allow as well to increase the size of primary mirrors with the use of available absorbers, increasing the overall energy collection. To do so, the equations should be addressed by fixing the radius of the absorber and calculating then the new size of the parabolic mirror.

Even though the influence on the flux analysis was addressed in previous literature - and conclusions on the temperature gradient effects on the absorber's surface were obtained - the effects of flux distribution on the thermal performance were not clear. In particular, such effects have not been studied in conjunction with a change on the fluid flow rate. Moreover, the effect of fluid flow rate is unclear. While some authors state that uniform flux could increase thermal performance of solar collectors, other authors have found a thermal performance enhancement for localised fluxes. This thesis shows how flux distribution affects thermal performance, although the effects are negligible if appropriate flow rates are chosen.

Finally, this thesis offers a practical solution to increase the global performance of parabolic trough evacuated receivers, especially at high temperatures. This solution offers better tolerance to misalignments and aberrations. Moreover, if combined with fluids such as Molten Salts, the output temperature of parabolic trough collectors can be increased, thus becoming a promising alternative to current commercial absorbers.
Chapter 2: Literature review

2.1 Generalities

Solar irradiation is approximately 1 kW/m$^2$ on the surface of the Earth [1], which is a low energy density when compared with fossil fuels. However, solar energy has a huge potential as utilising only 1% of global solar resources could be enough to achieve the recommendations of the United Nations' Intergovernmental Panel on Climate Change for long-term climate stabilisation [2]. Concentration of solar energy is a possible solution to overcome the low density disadvantage and to achieve a more efficient conversion from solar energy to other types of energy such as electricity [3].

Due to atmospheric conditions, not all the solar energy reaching the Earth surface can be concentrated. The radiation reaching the Earth can be divided in those rays which have been not altered indirection during their travel through the atmosphere, known as Direct Normal Irradiance (DNI), and those rays that have been deviated before reaching the Earth's surface, known as diffuse radiation [4]. The DNI direction is known from the Earth/Sun geometry and it depends on the date and time of the year, as well as the terrestrial coordinates [5, 6]. The diffuse radiation direction is random due the solar rays experiencing reflections, scattering and diffractions before reaching the Earth's surface. In order to achieve concentration ratios higher than 10 (the ratio between the receiver size and the effective solar energy caption area size) only DNI can be considered.

An important effect to consider in any concentration system is the size of the solar cone. The Sun is a sphere that emits radiation isotropically in every direction. The Earth is a much smaller sphere at a distance much greater than its radius or the Sun's. For that reason, it can be considered that the Earth's surface is a plane sector on the isotropic distribution of the solar energy and therefore, the Sun radiation is seen from the Earth surface as a cone, with a solid semi-angle of 4.5 mrad. This value can be obtained with geometrical calculations as developed by [7] and explained in detail in section 3.2. This semi-angle is vital in any concentration system since it limits the amount of concentration in a solar receiver. In classic literature, the angular distribution of solar radiation has been considered as a pillbox extending 4.5 mrad [7]. However, Buie [8] developed a model of the effective solar cone based on the Lawrence Berkeley Laboratory's database which contains measurements during ten years in eleven different locations in the United States in the late 1970s and early 1980s as well as the data published by Neumann [9], containing 2300 different measurements taken in the German Aerospace Center (Cologne, Germany) the Plataforma Solar de Almeria (Almeria, Spain) and the CNRS Solar Furnace in Odeillo (France). The data and methodology that Buie used to conduct his work has been published in several works, and a detailed list of the works published from this measurements is offered in [10]. Buie found that approximately 20% of the incoming energy is not located within the pillbox but in a greater angular distribution due to atmospheric effects. The amount of energy observed outside the solar cone depends on the latitude and longitude and it could be important in the optimal design of concentration systems [11]. Figures 2.1 and 2.2 show some of the results obtained by Buie for the solar cone size and the influence of this on the fraction of solar energy that a concentration solar system can receive. The database used by Buie includes solar data measurements for fourteen different locations in the United States and Europe and a range of atmospheric characteristics was obtained from factors such as altitude, humidity, climates and even sources of large particulates due to industrial environment [8].
Solar energy concentration is achieved by redirecting the DNI to a receiver using reflectors or lenses. The focused solar energy is then transformed into a usable form of energy directly on the receiver or transferring the energy to an energy converter. Considering the energy obtained after the transformation there are two main categories for concentrating solar systems, photovoltaic (PV) and thermal. There are also hybrid systems that try to increase the overall performance by obtaining both types of energy in the same
In PV concentrating systems, the energy conversion is done by the PV cells [12]. The advantage of concentrating the energy in this case is mainly economical, since the concentration systems used are cheaper than the PV cells used as receivers. The advantage of the PV cells is that they directly produce electricity that can be stored in batteries or inserted in the electrical network and even the efficiency of PV cells can be increased if they are designed to work with concentrated flux [16].

Concentrating thermal systems focus the energy of the Sun on a thermal receiver which contains a heat transfer fluid (HTF). The HTF can be used directly as heat in a process or to generate steam to feed a Rankine cycle to produce electricity as in a fossil fuel electric power plant. Solar thermal electricity, also known as Concentrating Solar Power or CSP is expected to be an alternative to classical power plants such as nuclear, fossil fuel or hydroelectric. Nevertheless, a cost reduction is mandatory for CSP plants in order to be considered a competitive alternative for cheaper production methods such as fossil fuel, hydroelectric or nuclear power plants. Even photovoltaic and wind energy are currently cheaper electricity sources [17-21].

CSP plants have a great potential in reducing CO₂ emissions. A solar field of 1m² produces around 400 kWhe per year. One megawatt of concentrating solar thermal power can save up to 688 tons of CO₂ per year if compared with a conventional combined natural gas cycle plant and up to 1360 tons per year if compared with a classical coal plant. In addition to the savings in CO₂ emissions, it also would save 2.5 tons of fossil fuel in the lifetime of the plant (estimated 25 years) [22].

A CSP plant can be divided into two main parts, the solar field and the power block (Rankine cycle), which is identical to the one in a fossil fuel power plant. Detailed examples of a Rankine cycle in a solar power plant can be found in [23, 24]. The generalised concept of a solar power plant is shown in figure 2.3. If a thermal storage is added to the solar field, a CSP plant can dispatch electricity even if there is no solar energy available at the moment (cloudy days or nights). The storage can be done in thermal oil, molten salts or even phase change materials. Examples of CSP plants with thermal storage in different materials can be found in [25-27].

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**Figure 2.3:** General schematic of a CSP plant. Copyright free images downloaded from [28-32].
The main advantage of CSP plants when compared to the most typical renewable energy plants, PV and on-shore wind, is its dispatchability. A CSP plant dispatches energy in the same way as a fossil fuel plant such that they can adjust their production to the electrical market’s instantaneous demand. If the solar field is oversized compared with the Rankine cycle turbine, a CSP plant can store the excess heat and about 2000 to 3500 hours per year of operation can be achieved [22]. In comparison, PV and on-shore wind energy plants have a low index of operating hours per year [33, 34] while off-shore wind plants have a similar range of operation [33]. An increase on the amount of operational hours per year of up to 5000 hours would be desired to make CSP capable of fully substituting fossil power plants [22]. However, another advantage of CSP plants against other renewable energy sources is its easy hybridisation, since the Rankine cycle can be used in parallel with fossil fuels [35].

A general classification on the main CSP plants can be done considering their solar fields. There are four main classifications; power towers or central receiver systems, linear Fresnel collectors, parabolic dishes with Stirling engines and parabolic trough collectors (PTC). All of them are based on the basics of reflection and Snell’s law [7] and they are built with parabolic shapes and a surface facing the Sun that effectively reflects the Sun’s radiation onto the parabola focus. The basic concepts of the reflection are based on Fermat’s principle of reflection and in the optics analyses conducted in this work, the edge-ray principle, that can be derived from the application of Fermat’s to certain rays as detailed in section 4.2.3, was applied [34]. Depending on the type of receiver installed, two main classifications can be made; those systems where a receiver is placed along a focal line (PTC and Fresnel) called 2D systems and those with a receiver placed on an area surrounding a focal point (towers and dishes) called 3D systems. 3D systems are required to track the Sun on two axes, while 2D do only require tracking in one direction, although this causes a lower concentration ratio.

2.2 Parabolic trough collectors. State-of-the-art

Out of all the concentrated solar thermal power technologies, parabolic trough collectors are the most developed, and the closest to being competitive in the electrical market. There are some commercial examples running now for longer than 25 years, such as the SEGS plants in the Mojave desert [36]. Thus PTCs are a proven and reliable technology, especially when compared to other concentrating solar technologies [37]. This is why most of the worldwide solar concentrating power plants are built with this technology. As an example, in Spain, the country where most solar concentrating power plants are built [38], 94% of the already installed power and most of the projected plants use this technology [39]. Despite the fact that a government’s moratorium on feed tariffs for new plants have decreased the number of new projects in Spain, it remains as the worldwide leader in cumulative power installed (2.3 GW out of 4.35 GW installed worldwide at the end of 2014) [38]. Meanwhile, diversification in other countries, due governmental support in solar towers, such as the US government and the Ivanpah plant (377 MW) will make the percentage of parabolic troughs new plants to decrease [38]. However, US government already approved parabolic troughs projects in 2015 for a total power of 375 MW with the Mojave plant, the second phase of the Genesis project (finished both of them during 2015) and the future installation of the Crescent dunes plant [38].

PTCs are linear focus concentrating systems. A long parabolic mirror concentrates the direct normal irradiation onto its linear focus where generally a tube containing a heat transfer fluid is located. The concentration is effective only if the aperture area of the mirror is perpendicular to the Sun, therefore PTCs need to track the Sun. Although there have been some attempts in the past to track the Sun in two axis [40, 41], the inclusion of the two-axis tracker complicates the installation of the troughs due to the shadows projected between adjacent collectors and a mobile piping required as the collector tracks the Sun. Generally, to increase simplicity, PTCs are installed with a one axis-tracker which tracks the Sun North-South or East-West. The main inconvenience of one axis tracking is that a geometrical loss, called end loss, appears due to
the incidence angle of the Sun [42]. A sketch of these losses and the equations to calculate them are shown in section 6.4.2, as they were relevant in obtaining the performance of the collectors in the outdoors testing conducted.

Geometrically, a parabolic mirror is a reflective surface obtained from the extrusion of a parabola along an axis. An interesting property of these mirrors, directly extracted from Snell’s law of reflection [7] is that they concentrate the Sun’s rays into their extruded focus if its aperture area \( A_p \) is perpendicular to the solar radiation. A typical PTC plant is composed of several rows of parabolic mirrors that concentrate the beam radiation into a cylindrical absorber placed along the focal line of the mirrors while tracking the Sun. Normally, inside this tube, there is a heat transfer fluid that absorbs the thermal energy and can be used in an industrial process. However, more commonly it is used to generate steam that feeds a Rankine generator to produce electricity [43]. Some alternatives, such as using CO\(_2\) in the Rankine cycle have been attempted [43, 44] and also air [45]. For all the applications it is desirable either to obtain the highest output temperature possible, although with higher temperatures the heat losses increase [46, 47], or to achieve a fixed temperature for applications such as solar cooling. If the collector’s efficiency is increased, a smaller mirror field could be sufficient to achieve the target temperature.

Electricity PTC commercial plants operate at temperatures of approximately 400 °C, which lead to “Sun to electricity” performances around 20 % [48]. Most of these plants operate with thermal oils such as VP-1 and Dowterm [49]. These oils are based on a mixture of diphenil oxide (C\(_{12}\)H\(_{10}\)O) and biphenyl (C\(_{12}\)H\(_{10}\)) and they have several limitations which have to be considered in the operation of the plant. The main one is that the oil chemicals degrade rapidly at temperatures higher than 425 °C, and this degradation becomes the main limitation in increasing the output temperature of the plant. They have also a high solidification temperature (12 °C) which makes an auxiliary heating system necessary in the plant to avoid the oil freezing overnight. The boiling points of the oils are around 257 °C and therefore pressurisation is necessary to achieve higher temperatures [50]. Lastly, the high viscosity at low temperatures could be an issue in the start-up of the plants, especially if it has to be done at times of the day that don't have high solar resources.

Higher output temperatures on the solar field would increase Rankine cycle thermodynamic efficiency [51]. There are alternatives for the heat transfer fluid that could lead to higher operation temperatures, such as water. Due to the high pressure necessary to maintain saturated water at high temperatures several attempts in generating steam directly in the absorber have been attempted. There are several advantages to using water in the solar field; the main ones are that the heat exchanger can be removed, increasing the solar field efficiency and that water has increased heat transfer coefficients compared to thermal oils. Additionally higher temperatures can be achieved, as once steam is generated the pressure inside the solar receivers won't be a limiting factor on the design of the absorber. The use of water instead of synthetic oils is also an environmental friendly option as it won't generate a waste product after the lifetime of the plant [52].

However, disadvantages arise from the two-phase flow that occurs when steam is being generated. To maintain two phase flows on the same row of collectors can compromise the reliability of the absorbers. In those absorber regions where two phase flow is present, high temperature gradients can appear, which leads to a dramatic bending of the absorber [53]. Figure 2.4 shows those dramatic bending reported by Lentz et al. during the two phase flow.
Enhancing concentration ratio of solar concentrators

An increase of flow rate will avoid the stratification of the two phase fluid flow, and intermittent or annular flow will occur. However, if the mass flow rate increases to much, the increase of temperature will be reduced and it would be necessary a higher heat flux impacting on the absorber to guarantee steam generation. This necessity will decrease the range of hours that the solar plant can work in a day.

The DISS project tried to separate the two phases of the flow with a water/steam separation tank. However, pressures up to 110 bars are necessary on the solar field. After more than 3500 hours of operation, the possibility of a direct steam generation parabolic-trough plant was demonstrated. If components such as compact steam/water separators and high temperature joints become a reality, those plants could increase the output temperature of the solar fields up to 500 °C.

Molten salts have been also investigated as an alternative for the heat transfer fluid. Their main advantages are their thermal stability at higher temperatures than thermal oils and their low vapour pressure (less than 1 Pa at temperatures around 500 °C). However, they have high freezing temperatures, typically between 120 and 220 °C. An alternative not explored in much detail as thermal fluid is air. Air won't freeze and it could operate with thermal stability and relative low pressures at higher temperatures than any other thermal fluid. However, air receivers should be insulated in order to achieve high temperatures and it makes difficult their construction. As shown in section 2.5, the use of secondary optics is helpful to overcome the insulation challenge for air collectors.

Higher temperatures increase thermal losses so the heat loss area of the absorber should be as small as possible. However, the pressure drop is higher in smaller absorbers with the same flow rate, and thus more pump energy is required. If the pressure drop increases too much, the pumping power required could influence the viability of the field. This effect has been given very little attention in the literature.

Although parabolic troughs are a mature technology, there are potential improvements to develop in this field and for that reason, the main core of this thesis consists of a proposal to increase the performance of existing parabolic trough by the use of a secondary optics. Next section focuses on the background of parabolic troughs, necessary to present their typical dimensions and the conditions which allow parabolic troughs to work. Section 2.4 will review the flux distribution around a parabolic trough, which cannot be changed without the use of secondary optics but has an influence on the thermal performance of parabolic troughs. Finally, section 2.5 shows the attempts found in the literature of using secondary reflectors to improve parabolic troughs performance.
2.3 Parabolic trough collectors. Background

Without accounting for any optical losses, e.g. attenuations in the mirrors and reflections or absorption losses in the receiver [57], the fraction of the radiation reflected by the parabolic mirror which is received by the absorber can be defined by considering only the geometry of the system. The concentration ratio is the ratio of area of input beam divided by the area of output beam [7, 58]. In this way, considering that the incoming radiation emanates from a disc shape placed at an infinite distance with a semi angle $\theta_s$, the theoretical maximum concentration ratio for a PTC with planar receivers is reduced by a factor of 2 compared with the thermodynamic limit of concentrators, defined as in equation 2.1:

$$C_{\text{max}} = \frac{1}{2 \sin \theta_s} \quad \text{(Equation 2.1)}$$

Since $\theta_s \approx 0.266^\circ$ for the earth-sun geometry, the theoretical maximum concentration ratio for a PTC with a planar receiver is roughly 107. In a real PTC system, it is necessary to consider some additional effects regarding energy capture, reducing more the concentration ratio of the troughs.

For PTCs with cylindrical absorbers, if only the sun shape is considered, the maximum concentration ratio achievable will be reduced compared with the thermodynamic limit by a factor of $\pi$ due to the absorber being cylindrical instead of flat. Then, the theoretical maximum concentration ratio for a standard PTC is limited to roughly 70. A complete overview of the thermodynamic and geometrical limits of concentration is developed in the next chapter.

The parabolic mirrors in commercial PTCs normally have a large aperture area with high wind resistance making them susceptible to vibrations [59] and deformations on the mirror [60]; even in a perfectly aligned system, the thermal expansion of the primary mirror and the absorber can change the geometry and the alignment of the components [61-64]. Other uncertainties include deviation in shape from an ideal parabola and scattering of the beam radiation on the reflective surface due to surface imperfections [65]. Before the absorber's dimensions are defined, all these effects must be considered, but as some of these effects are random it will result in partial cancellations [7]. The half-acceptance angle, $\psi$, defines the width of an absorber capable of capturing all the solar radiation despite the spread of the sun's disc and all the other possible errors and it is shown in figure 2.5.

It is also necessary to consider that the dimensions of the parabola are finite. The rim angle, $\psi$, is defined as the angle formed by the symmetry axis of the system and the line that joins the edge of the mirror with the focus of the parabola as shown in figure 2.5. Now, the concentration ratio for a PTC, when the receiver is cylindrical, can be defined as in equation 2.2 [66].
From equation 2.2 it is easy to find that the maximum concentration ratio for a parabolic trough is achieved with a $\psi = 90^\circ$. Nevertheless, there are no physical reasons not to have higher $\psi$ (although commercial collectors are around this value). If the absorber is not cylindrical, as the geometrical concentration ratio can be also defined as the ratio between the areas of the concentrator and the absorber [7], an equivalent equation (2.3) valid for every $\psi$ and absorber shape can be defined from another parameter, the aperture width of the mirror (W).

$$C = \frac{\sin \psi}{\pi \sin \theta}$$  \hspace{1cm} (Equation 2.2)

And in the case of a cylindrical absorber can be simplified further as in equation 2.4

$$C = \frac{A_p}{A_{absorber}} = \frac{WL}{\pi DL} = \frac{W}{\pi D}$$  \hspace{1cm} (Equation 2.3)

From this equation, it is clear that increasing the size of the mirrors (and maintaining, if possible, the size of the absorbers) increases the amount of thermal energy around the receiver and thereby the performance of the system is increased. Over the years, the size of the mirrors for PTCs has increased, with current commercial PTCs having a mirror aperture around 6m and a tubular receiver with a diameter of 70 mm[67, 68]. For a constant absorber diameter, there is a limit on the aperture width of the parabolic mirrors that will depend on how much the half-acceptance angle can be decreased. However, larger diameter absorbers will be available soon and absorbers up to 90 mm have been considered [68]. A PTC with an 80 mm absorber has been designed recently [69]. Increasing the size of the absorber can increase the concentration ratio by allowing a size increase of the primary mirror without affecting the half acceptance angle, but it will increase the heat losses as well.

There is recent work trying to increase the size of the primary mirrors [68, 70]. Although bigger receivers can be built, there are problems associated with the mirror aperture becoming bigger. The wind load increases and stronger tracking systems and structures are necessary [59, 60]. Commercial parabolic troughs operation have to be cancelled at velocities higher than 14 m/s and even at parking position the structural safety is compromised at velocities higher than 40 m/s [71]. An increase of weight and size of the primary mirror can decrease both ranges and become a limiting factor. In addition to the reliability of the structure, the wind speed induces some twisting on the mirrors, which becomes relevant in big collectors such as the
LS-3 or Eurotrough at speeds around 5 m/s that can produce an optic loss close to 25% [71]. Recently, a pilot plant that encloses the PTCs inside a modified agricultural glasshouse was presented [72]. Enclosing the PTCs eliminates the effects of wind as well as protects the collectors from dust, reducing cleaning requirements. The enclosure environment also allows the reduction in the structural cost of the mirrors and reduces the tracking requirements due to the reduction of weight.

An increase in the primary mirror implies either an increase on the absorber diameter (therefore reducing concentration ratio and increasing thermal losses) or a decrease on $\theta$. If $\theta$ decreases too much, the tracking system has to be highly accurate and the system is exposed to any misalignment which could decrease the intercept factor (I) of the collector [73]. In this work, the intercept factor is defined as the ratio between the energy reflected in the primary mirror and the energy striking the receiver. Last, but not least, with bigger mirrors the shadows projected by adjacent mirrors increase, so the distance between rows has to be increased, so the ground usage, the piping length and the plant size become higher.

Although the problems involving the increase of mirror size are solvable problems, the ground usage can become a problematic limitation for solar power plants, especially for rooftop applications where the space is limited by the size of the roof and it is desirable to have the highest percentage of ground usage possible. From the data shown in [43] a typical ground usage in a solar power plant is between 2 and 5 ha per Mwe. That is a low energy density if compared with a typical fossil fuel power plants and even with Fresnel collectors [74]. The low density is in part due to the 'dead' area between adjacent rows of mirrors, necessary to avoid shadow between them. In Table 2.1, some commercial solar power plant dimensions are shown. An increase on the mirror size will scale the plant size as the space between adjacent rows should also be increased to avoid an excessive shadow of the primary mirrors.
Enhancing concentration ratio of solar concentrators

Table 2.1: PT Solar power plants size. Modified from \[37, 43\]

<table>
<thead>
<tr>
<th>Plant</th>
<th>Location</th>
<th>Operating since</th>
<th>Net output capacity (MW)</th>
<th>Land area (ha)</th>
<th>Solar field aperture area (ha)</th>
<th>Area/MW (ha/MW)</th>
<th>Type of collectors</th>
</tr>
</thead>
<tbody>
<tr>
<td>Segs I</td>
<td>Dagett (USA)</td>
<td>1984</td>
<td>13.8</td>
<td>29.0</td>
<td>8.3</td>
<td>2.1</td>
<td>LS1, LS2</td>
</tr>
<tr>
<td>Segs IV</td>
<td>Kramer Junction (USA)</td>
<td>1986</td>
<td>30.0</td>
<td>87.0</td>
<td>23</td>
<td>2.9</td>
<td>LS2</td>
</tr>
<tr>
<td>Segs IX</td>
<td>Harper Lake (USA)</td>
<td>1990</td>
<td>80.0</td>
<td>169.0</td>
<td>48.4</td>
<td>2.1</td>
<td>LS3</td>
</tr>
<tr>
<td>Saguaro Solar</td>
<td>Red Rock (USA)</td>
<td>2006</td>
<td>1.0</td>
<td>4.7</td>
<td>1.0</td>
<td>4.7</td>
<td>SGX1</td>
</tr>
<tr>
<td>Generating Station</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Nevada Solar One</td>
<td>Boulder City (USA)</td>
<td>2007</td>
<td>64.0</td>
<td>162.0</td>
<td>35.7</td>
<td>2.5</td>
<td>SGX2</td>
</tr>
<tr>
<td>Andasol</td>
<td>Granada (Spain)</td>
<td>2009</td>
<td>49.9</td>
<td>200.0</td>
<td>51.0</td>
<td>4.0</td>
<td>654 SKAL-ET150/AS1</td>
</tr>
<tr>
<td>Ibersol</td>
<td>Puertollano (Spain)</td>
<td>2009</td>
<td>50.0</td>
<td>135.0</td>
<td>29.0</td>
<td>2.7</td>
<td>352ET150</td>
</tr>
<tr>
<td>Solnova</td>
<td>Sanlucar la Mayor (Spain)</td>
<td>2009</td>
<td>50.0</td>
<td>120.0</td>
<td>29.4</td>
<td>2.4</td>
<td>360 ET150</td>
</tr>
</tbody>
</table>

Table 2.1 shows how the ground usage is similar for different size of collectors, for example the LS1, LS2 and LS3 collectors. Increasing the space between rows does not imply increasing the ground usage. However, it will increase the length of the piping which has two main undesirable effects as a consequence. It increases the piping and insulation costs; that can be a significant percentage of the total cost of a solar power plant and with a longer fluid circuit, the pumping power increases.

All these limitations mean PTCs are designed far from the geometrical concentration limit. The biggest commercial PTCs have a concentration ratio around 30 \[37, 43, 69\]. A representative range of commercial and prototype PTCs with different dimensions are shown in table 2.2.

Table 2.2. PTC used in this work. Data extracted and modified (where required) from \[37, 43, 69\]. Description of the dimensional parameters was developed in Figure 2.5.

<table>
<thead>
<tr>
<th>Collector</th>
<th>Aperture (mm)</th>
<th>Focal distance (mm)</th>
<th>Radius of the absorber (mm)</th>
<th>$\psi$ (°)</th>
<th>$\theta$ (°)</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEP</td>
<td>1208</td>
<td>647.5</td>
<td>14</td>
<td>50</td>
<td>1.02</td>
<td>13.73</td>
</tr>
<tr>
<td>NEP2</td>
<td>1844</td>
<td>647.5</td>
<td>14</td>
<td>70.9</td>
<td>0.83</td>
<td>20.96</td>
</tr>
<tr>
<td>LS1</td>
<td>2500</td>
<td>680</td>
<td>21</td>
<td>85</td>
<td>0.96</td>
<td>18.95</td>
</tr>
<tr>
<td>LS2*</td>
<td>5000</td>
<td>1400</td>
<td>35</td>
<td>83.5</td>
<td>0.8</td>
<td>22.73</td>
</tr>
<tr>
<td>LS3*</td>
<td>5760</td>
<td>1710</td>
<td>35</td>
<td>80</td>
<td>0.69</td>
<td>26.19</td>
</tr>
<tr>
<td>Helioman 3/32</td>
<td>1810</td>
<td>640</td>
<td>17</td>
<td>70.5</td>
<td>1.01</td>
<td>16.85</td>
</tr>
<tr>
<td>PT1</td>
<td>2300</td>
<td>800</td>
<td>25.5</td>
<td>71.4</td>
<td>1.21</td>
<td>14.36</td>
</tr>
<tr>
<td>Solitem</td>
<td>1800</td>
<td>780</td>
<td>19</td>
<td>59.9</td>
<td>1.05</td>
<td>15.08</td>
</tr>
<tr>
<td>Acurex 3001</td>
<td>1830</td>
<td>457</td>
<td>15.9</td>
<td>90</td>
<td>1</td>
<td>18.32</td>
</tr>
<tr>
<td>Acurex 3011</td>
<td>2130</td>
<td>533</td>
<td>15.9</td>
<td>90</td>
<td>0.85</td>
<td>21.32</td>
</tr>
<tr>
<td>Sener trough 2</td>
<td>6868</td>
<td>2000</td>
<td>40</td>
<td>81.3</td>
<td>0.66</td>
<td>27.3</td>
</tr>
<tr>
<td>Solar Kinetics T-700</td>
<td>2130</td>
<td>559</td>
<td>31.75</td>
<td>87</td>
<td>0.85</td>
<td>21.32</td>
</tr>
<tr>
<td>Solar Kinetics T-800</td>
<td>2360</td>
<td>483</td>
<td>20.7</td>
<td>101.4</td>
<td>0.99</td>
<td>18.19</td>
</tr>
<tr>
<td>Suntec Systems IV</td>
<td>3050</td>
<td>838</td>
<td>19</td>
<td>84.6</td>
<td>0.72</td>
<td>25.48</td>
</tr>
<tr>
<td>Solel IND-300</td>
<td>1300</td>
<td>272</td>
<td>11</td>
<td>100</td>
<td>0.98</td>
<td>18.64</td>
</tr>
</tbody>
</table>

*The relevant dimensions for these collectors in this work match with Duke and Eurotrough collectors, so the results obtained will remain the same. The differences between LS2 and Duke or LS3 and Eurotrough can be checked at \[21\]. Flabeg recent designs \[75\] focal distances match with LS3 collector. However their aperture areas may change.
2.4 The flux distribution on a parabolic-trough

As the parabolic mirror of a PTC sits below the absorber and reflects the sun beam radiation back to it, the flux profile received in the absorber is not uniform. Jeter [76] developed a theoretical calculation on the flux profile of a parabolic-trough with ideal optics considering different solar cone semi-angles (16' and 32') for the focal plane of the trough and for $\psi = 90^\circ$ angle and 20 times concentration trough with a cylindrical receiver and a solar radius of 0.0075 mrad (25°47′). The flux distribution obtained in Jeter's is shown in figure 2.7.

One-dimensional numerical simulations have been addressed in literature considering the glass-absorber interface [77, 78]. Two-dimensional models have been also developed even for non-evacuated tubes, including the effects of natural convection in the glass enclosure [79]. However, most of this works assumed an uniform flux distribution around the absorber. In cases such as non-evacuated absorbers, including a non-uniform flux profile could help to increase the accuracy of the convective heat losses.

The irregular flux distribution induces a temperature distribution on the absorber wall that could affect the PTC performance. The heat transfer models and performance simulations conducted in the literature to calculate the thermal performance of solar collectors consider only uniform solar flux or absorber wall temperature [73, 74]. There is recent work that calculated theoretically, and by simulations, the influence on the thermal performance of the irregular flux on evacuated receivers [80]. They found an increase of heat losses for a non-uniform model although it did not affect significantly the performance of the absorber. However, they did not consider parameters such the mass flow rate, which will have a tremendous impact on the thermal performance with different fluxes, as will be shown in section 5.5 of this work. Lu et. al. [80] concluded that the inhomogeneous flux distribution increases heat loss on the absorber while Ghomrassi et al.'s simulations [81] concluded that a higher outlet temperature can be achieved if there is an increase of flux concentration on the bottom of the tube. An increase of local concentration on the bottom on the absorber would have associated a thermal stress and a possible bending as it is shown later, but this flux distribution will increase the local temperatures and it does not seem possible to increase thermal performance by increasing local temperatures, since it would lead to higher thermal losses according with Steffan-Boltzmann laws [82]. As the explored literature did not seem conclusive about the effects of the flux distribution on the solar collectors' performance, part of this thesis will include CFD simulations trying to explain this phenomenon.
CFD simulations and a validation experiment were conducted in [83] considering an irregular flux profile on the absorber to calculate the performance of the parabolic troughs, although effects such as natural convection in the case of a non-evacuated tube were ignored. Experiments conducted in the Plataforma Solar de Almería [84] show that the CFD estimations for the temperature distribution on an absorber tube containing superheated steam were overestimated. The most plausible reasons for this overestimation were those involving a lower DNI reaching the absorber in the real experiment, due to lower optical quality than expected or lack of cleanness of the glass and mirrors.

Other effects such as scattering, misalignments, diffractions on the glass envelope or wind forces will also change the flux distribution. Very recent work conducted by Zhao et al. [85] considered those effects in the flux distribution calculations with ray tracing. Those results align with the ones used in this thesis and published in [86] and shown in section 4.3.

Apart from the direct influence of flux distribution on the thermal performance, the temperature gradient adds an undesirable effect on the parabolic trough receiver, a thermal stress due to the temperature gradients on the absorber's surface. The temperature gradients are proportional to the mirror size since bigger mirrors increase the local concentration ratio, which can be calculated by ray tracing or with Jeter's formulation [72]. This stress will cause the absorber to bend and in the worst case scenario if the bending is large enough, the thermal stress will break the glass cover, due to the different expansion coefficients between glass and steel. Akbarimoosavi and Yaghoubi predicted the breakage to be possible at a bending greater than 20 mm considering the parabolic troughs used in the Shiraz Solar Power Plant [87]. The bending observed in the Shiraz power plant during its operation is shown in figure 2.8. The bending is dependent on the mirror size and the operation conditions; with the maximum bending reported by Akbarimoosavi and Yaghoubi being 19.42 mm [87], quite close to the critical 20 mm stated.

![Figure 2.8: Absorber displaced from the focal line of the parabolic mirror due to thermal expansions [87].](image-url)

More commonly, the thermal stress wouldn't produce the glass breakage and the bending of the tube will only affect its axis being displaced from the focus on the parabola. If the bending is high enough, it will cause some rays to miss the absorber, decreasing the energy production of the trough.

Several studies have reported bending due to the thermal gradient. Figure 2.4 shows the bending reported by Lentz et al. [53] in an absorber for direct steam generation. This study reported a maximum bending of 25 mm on the tube. Khanna [88] evaluated the bending on the biggest absorber available on the market [89] finding bending ranges from 1.7 mm up to 13.7 mm considering different optical errors and $\psi$ on the primary
Chapter 2: Literature review

They discovered that not only the flux distribution but also the tube support number and spacing have an influence on the bending of the absorbers [61, 90], although the total length of the absorber is not an issue if the support spacing is maintained constant. Fuqiang et al. [91] suggested that using a corrugated tube as an absorber can increase the heat transfer coefficient of thermal absorbers by 8.4% while reducing thermal stress at the same time by 13.1% under certain flow conditions.

From the works found in the literature, it is obvious that flux distribution will affect the reliability of parabolic troughs and the mass flow rate range in which the solar power plants can operate. Achieving a more uniform flux distribution around the absorber surface can help to reduce the thermal stress and will reduce the risk of bending in the absorbers. The introduction of a secondary optics addressed in the next section, will minimise the pernicious effects of flux distribution and at the same time would help to increase the concentration ratio of the troughs.

2.5 Secondary optics on parabolic-trough collectors

Parabolic trough absorbers receive light after one reflection on the parabolic mirror. The quality of the mirror as well as the manufacturing tolerances of every element and the tracker accuracy have to be added to the Sun radius in order to calculate the minimum absorber size required to capture the incoming energy. Details of how the size of an standard absorber will define the misalignments sensitivity of the parabolic troughs are given in next section which addresses the limits of concentration of ideal parabolic concentrators. The half-acceptance angle, $\theta$, shown in figure 2.2, defines the minimum width of the absorber to make it receive the whole reflected bundle incoming from the reflector, despite all the possible misalignments. In order to increase concentration, $\theta$ should be as small as possible.

There are other types of parabolic trough collectors, such as compound parabolic concentrators (CPCs) [7, 58, 92] that focus on increasing $\theta$ as much as possible instead of high concentration. CPCs require several reflections inside the two truncated parabolas before reaching the absorber although there are some designs of CPC that try to minimise the number of reflections [93]. In the case of CPCs, to achieve a large concentration ratio implies very deep reflectors. These deep reflectors are not easy to manufacture and still don’t offer enough concentration to be a practical solution for solar thermal plants. Nonetheless, they can become a promising solution for water heating, commercial, domestic and industrial applications up to 300 °C such as methanol reforming [94] or low concentration PV-T systems [95]. CPCs can be designed for any planar receiver applications, bi-facial [96] or one-sided [97], or for a tubular absorber [98]. An interesting option to increase CPCs performance is to encapsulate a U-pipe absorber inside a bi-facial planar receiver, to decrease gap losses [99]. For some applications, the truncated parabolas don’t have to be symmetrical [100]. If CPCs are designed with low concentration ratios but higher $\theta$, they are capable of concentrating the solar radiation without any tracking while still receiving a fraction of the diffuse radiation [99]. This absence of tracking makes CPCs an interesting solution for architectural integration [101, 102].

A secondary reflector can be added to the concentration systems in order to further concentrate the solar radiation and improve the collector optical efficiency. High increases of solar concentration have been achieved for concentrating systems with the use of secondary reflectors [103]. In some cases, they can even increase flux uniformity by creating a collimated beam at the same time that they approach the thermodynamics limit [104]. Linear Fresnel concentrators normally include secondary optics [105], the most common secondary reflector in this case are cavity receivers [106] and CPCs [107], although other shapes such ellipticals [108], asymmetrical parabolas [109] or semicircular [110] have been considered.

An interesting application of secondary reflectors on linear reflector is the line-to-point concentrators (LTP) [111]. The original idea of those collectors was, apart from increasing concentration ratio, to get a cost reduction by replacing solar cells by lower cost materials such as Fresnel lenses or mirrors [112]. These concentrators use a linear primary optics combined with a secondary optics stage that allows the solar bundle to be concentrated in a point, increasing the concentration at limits even above the maximum limit of
linear concentrators [112]. ‘LTP concentrator differs from a traditional line-focus trough concentrator in that the focal line is split into a number of point-like foci spanning along the length of the trough concentrator’ [111]. Maintaining the one axis tracking in linear concentrators, a point focus can be used if secondary optics is installed to divide the linear flux reflected by the primary mirror. Several attempts of LTP collectors are found in the literature and they can be divided in three main groups according to Cooper et al. [111] which included the sketch of the three of them as shown in figure 2.9.

- LTP with fixed secondary optics [112, 113]
- LTP with discrete switching optics [114]
- LTP with continuous tracking optics [111]

Figure 2.9: Three types of LTP solar concentrators classified by the degrees of freedom of the secondary concentrator stage: a) fixed secondary optics; b) discrete-switching secondary optics; c) continuous-tracking secondary optics [111].

The use of a row of CPCs as the optics second-stage allowed concentration ratios of around 300 to be reached in the BICON concept [112, 113]. Cooper et al. [111] explored LTP collectors with concentration ratios ranging between 500 and 1500 and they stated that the concentration limit of LTP collectors is around 4000 for hollow secondary reflectors and it can be increased up to 6000 with the inclusion of dielectric materials. In that project, the primary mirror was an asymmetric parabolic mirror to reduce the shadows between consecutive rows of primary mirrors. As this prototype was a LTP with fixed secondary optics, $\theta$ and dimensions of the secondary mirrors depended on the Sun position and it was defined considering the winter and the summer solstice. The concentration ratio changes depending on the latitude, making this type of collector non suitable for certain locations.

A way of improving the concentration ratio of the LTP concentrators and making them independent from latitude while maintaining the simplicity of the system is to make the secondary stage pivot while the receiver is fixed [114, 115]. The switching secondary optics allows higher concentration ratios at high latitudes but the rows of secondary concentrators are then limited due to their movement.

LTP with continuous tracking optics overcomes the issue of the switching secondary optics by making the receiver also to track the Sun. As the secondary optics tracks the Sun it is also not dependent on the latitude as does the fixed type. It has been demonstrated for planar receivers and is limited to $\psi < 90$. Nevertheless, in order to obtain a good concentration ratio, parabolic troughs with low $\psi$ were used [111].

Secondary concentrators were used in [45] as a way to reduce the heat exchange surface and making possible the insulation of an air receiver. In this case, the parabolic trough has been split in two halves and their foci
have been slightly displaced. A pair of secondary mirrors for each half re conducts the Sun radiation into the cavity receiver. A schematic of the proposed receiver is shown in figure 2.10.

![Figure 2.10: Schematic cross-section of solar air receiver based on an array of cross-flow cavities (left) and CAD view (right) [45].]

Commercial PTCs with tubular receivers have not included secondary optics so far although several designs have been studied in the past [116-120]. Normally they have included complicated secondary mirror geometries that, while increasing the concentration ratio considerably, did not become practical solutions due to the difficulty in manufacturing them at a competitive cost. Figure 2.11 shows sketches of typical secondary receivers found in literature.

![Figure 2.11: Sketches of secondary reflectors addressed adapted from those found in the literature [116-120]. a) Helmet concentrator; b) Single mirror two stage with snail reconcentrator; c) Seagull concentrator; d) Snail concentrator on parabolic trough.]

Recent work states the potential improvement of secondary reflectors in PTC and analyses geometries previously investigated [121]. The selected geometries were (reflective glass surface [116], reflective annulus insulation [122], aplanatic mirrors [104, 123] and a tailored seagull reflector [120]) and they shown a thermal efficiency potential improvement between 0.8 (Seagull) and 1.6% (Aplanat) if compared with a benchmark
A new method for the design of the secondary mirror has been carried out by Cannavaro et al in the XX-SMS concentrator [68]. They emphasise increasing the primary mirror area as much as possible while keeping the receiver as small as possible. They also state that some of the present limitations of the second stage solutions are that the absorber may sometimes touch the secondary mirror, increasing the thermal losses, or that they can even introduce a new optical loss due to the necessity to accommodate the gap between the secondary reflector and the absorber (gap losses) [124]. In their particular scenario, the aperture area of the primary mirror could be increased up to 11 m, but the great dimensions of the secondary optics (110 cm tall and 79 cm wide) produce a big shading and will make difficult its manufacture. However, the concentration ratio obtained, 51, is the highest concentration ratio reported in literature for a parabolic trough which does not miss rays. Very recent work from these authors [125] reduced the dimensions of the secondary optics by installing a novel compound parabolic elliptical type secondary stage with smaller dimensions (34x37 cm) but also a smaller concentration ratio (42.6). However, in both proposals, $\theta = 0.44^\circ$, a much lower $\theta$ that those considered a standard 70 mm absorber with a LS3/ET primary mirror ($0.69^\circ$). Figure 2.13 sketches the dimensions and shapes of both secondary optics.
2.6 Definitions

Some terms, such as intercept factor, concentration ratio or optical efficiency may vary slightly in literature depending on the effects considered in the cases studied. In some cases, these terms are quite similar and they could lead to confusion. To avoid confusion, the exact definition of how this terms are used in this work are explained in this section.

Concentration ratio, $C$, is usually defined as the ratio between the aperture and the absorber's areas and represents the amount of concentration achieved in a solar system [7]. This terminology is purely geometrical and it does not consider the possibility of a concentrator designed not to receive all the energy reflected in the mirror. In this work, the concentration ratio is calculated as the combination of the geometrical concentration ratio and the effects that could reduce it (such as shadows or missed rays due to the geometry of the system). As an example, in the flat secondary reflector development, the effects of the shadow are considered and the concentration ratio won't be equal to the geometrical concentration ratio. See equation 4.6.

Relative concentration ratio is defined as the concentration ratio achieved in the system divided by the maximum concentration achievable in a system of its characteristics. As explained in detail during chapter 3, the maximum concentration was defined in the literature and it depends on if the absorber considered is point-like or linear [7].

Intercept factor, $I$, is the ratio of rays reflected by the mirror that strike the absorber. Effects such as misalignments or diffractions in the glass can reduce the intercept factor of a solar collector. For ideal optics the intercept factor is one for those solar receivers designed to capture all the reflected radiation. In this work, the glass cover effect is not considered..

Local intercept factor, $I(\psi)$, is the amount of rays reflected in a point of the mirror that strike the absorber. In sections 3.4 and 3.5 the local intercept factor is used to calculate the total intercept factor of those ideal concentrators which absorber is not big enough to receive all the radiation reflected in the primary mirror.
Optical efficiency, \( \eta_{opt} \), is defined as the ratio of energy received in the receiver and the energy entering the aperture area of the concentrator. It should take into account the amount of rays missed in non-ideal reflections, the intercept factor, and shadowing of elements such as secondary reflectors or the upper part of a planar receiver which is not capable of absorbing the energy from above (e.g., the back of a PV cell). If ideal optics are considered, optical efficiency matches the intercept factor.

2.7 Summary

Parabolic troughs are the type of concentrator most used in solar concentrating plants and they are a reliable and mature technology. Commercial plants are normally based on the use of parabolic troughs in combination with thermal oils which will degrade rapidly for temperatures above 425 °C. That limits the output temperature of solar plants and alternatives such as direct steam generation and molten salts have been explored in literature.

Classic parabolic troughs are composed of the parabolic mirror and a cylindrical receiver and this setup defines a characteristic flux around the absorber which can increase thermal stress on the receiver due to different expansion coefficients between the glass cover and the metallic absorber. That configuration creates a characteristic flux pattern around the absorber which affects the parabolic troughs reliability and which effect in the thermal performance is not completely explained in literature. However, it is impossible to change the flux profile in a standard parabolic trough.

The inclusion of a secondary optics would allow parabolic troughs to modify its flux distribution, and that could allow to reach safely higher temperatures, since the temperature gradients along the absorber's surface will be decreased. More importantly, there is a potential thermal efficiency improvement by the use of secondary optics and an enormous potential in increasing their concentration ratio, which has not been exploited in more than a few recent works. Increasing the concentration ratio while maintaining efficiency is one of the main challenges to overcome with parabolic trough collectors to make them commercially competitive against fossil fuels.

However, the concentration ratio cannot be increased over certain limits, defined by the laws of thermodynamics, that were defined in literature for 3D concentrators and obtained with some simplifications for 2D concentrators. For parabolic concentrators, those limits are unreachable and the maximum concentration ratio of parabolic troughs has been found to be \( \pi \) times lower than the thermodynamics limit itself.

Solar concentrators have been designed in the past with the assumption that every solar ray reflected by the primary mirror is going to reach the absorber. That limits the maximum size of primary mirrors that could be installed for a given absorber, since their size are limited to 70 mm for commercial absorbers nowadays, although prototypes of 80 mm and 90 mm are being studied. The inclusion of secondary optics would allow the use of bigger primary mirrors or smaller absorbers, leading to higher concentration ratios.

Improving the concentration ratio or simply decreasing thermal losses, the secondary optics for parabolic troughs found in literature have the main inconvenient of being complicated to manufacture, due to complicated shapes or big dimensions.

Taking into account the state-of-the-art of solar concentrators in general and parabolic troughs in particular. This thesis will investigate two main topics. The real maximum concentration achievable by a parabolic concentrator if some rays can be missed and the potential improvement in concentration ratio and thermal efficiency of a parabolic trough by the inclusion of a simple secondary optics geometry.
Chapter 3: Limits of concentration on parabolic concentrators

The maximum amount of solar energy that any system can concentrate is restricted due to thermodynamics. The second law of thermodynamics limits the amount of concentration achievable by an ideal 3D system. For 2D systems, a concentration limit was calculated as a simplification of the 3D concentrator scenario [7]. In real concentrators, such as parabolic troughs, the real limit is much lower than the thermodynamic limit, and that is a limiting factor on their performance. In previous literature, the limits of concentration for parabolic surfaces have been calculated considering that no concentrated sunlight misses the absorber. In some cases, optical efficiency can be compromised to increase concentration. In this section, the limits of concentration for parabolic surfaces are explored with the assumption that a certain optical loss is acceptable in order to increase concentration. Theoretical concentration limits equal to the thermodynamics limit are achieved if some rays are missed. The formulation of the concentration ratio of parabolic surfaces which some of the rays reflected are missed in their targets are developed in this work and then it has been validated with a ray tracing software.

3.1 Introduction

Increasing the amount of energy concentration around solar receivers is necessary to achieve high temperatures in solar thermal applications. The maximum concentration ratio that an ideal solar system can achieve is limited by the laws of thermodynamics. This limit depends on the Sun-Earth geometry and it finds its maximum for 3D concentrators. For 2D concentrators, a limit related to the 3D thermodynamics limit has been developed in the literature [7]. Real systems will have their own limitations due to misalignments, scattering and aberrations of the mirrors, refraction on the glass cover of the receivers, etc. These limitations make real systems to have concentration ratios far below the thermodynamics limit [58, 126].

Even if a system is considered ideal, the geometry of the system itself will impose a lower limit than the thermodynamic one.

In previous work, two assumptions were made in order to calculate the concentration ratio of ideal systems. The first one is that no rays should miss the absorber, and the second one is the assumption that, due to symmetry, a cross-section of the concentrators is enough to calculate the concentration ratio of the systems applying edge-ray theorem [58]. The optical performance of both parabolic dishes and troughs can be analysed in two dimensions by assuming that the bundle of rays contained in the solar cone spreads only in two dimensions [116]. In this thesis, when this simplification is taken, it will be referred as cross-section simplification or bundle-plane simplification. With these assumptions, parabolic troughs fall below the fundamental limit of concentration by factors of two or \( \pi \) depending on if the receiver is a planar surface or a cylinder respectively, while parabolic dishes have their concentration limit fall by a factor of four [58, 126]. In the classic literature the maximum concentration ratio of parabolic mirrors has been calculated such that no one single ray misses the absorber [7]. In this chapter, the maximum concentration ratio achievable is calculated considering that some rays can miss the absorber in order to increase the concentration ratio, accessing higher exergy levels. New equations for finding the concentration limits of a given parabolic mirror are developed in this chapter, assuming a cross-section simplification, and the new concentration limits are explored.

The development of the maximum concentration ratio shows that the limits of concentration for parabolic concentrators are closer to the thermodynamics limit if rays can be missed. The next step was to calculate the maximum concentration ratio achievable without the cross-section simplification. The new concentration
limits found differ from those with the cross-section simplification, and they were validated conducting ray tracing simulations. Finally, an overview of the relationship between concentration ratio and optical efficiency was addressed for all the parabolic surfaces studied with and without bundle-plane simplification.

In this section, it was proven that the concentration limits for parabolic surfaces were underestimated in the literature. Concentration limits up to the thermodynamics limit were found for 3D concentrators. It was also demonstrated that the cross-section simplification is not accurate enough if rays are to be missed and finally it is shown how the theoretical limit for 2D concentrators was also underestimated due to the cross-section simplification. However, this simplification makes it much easier to calculate the concentration ratio of solar collectors and is accurate for those collectors designed to capture the whole bundle of rays incoming from the Sun.

To develop the calculations in this chapter, it is assumed that the concentrators and receivers are ideal and there are no tracking error or misalignments. The optical efficiency is related then as a ratio between the available solar flux and the one received in the absorber. In a realistic system, effects such as the real reflectivity of the mirror or the transmissivity of the glass should be taken into account. As ideal limits are explored in this work, the only optical losses taken into account here are geometrical. The solar cone has been represented in literature in different ways, at it was shown in the literature review chapter. For the simplicity of this work, the Sun is considered here as a uniform pillbox distribution over a range of angles \( \theta < 4.5 \text{ mrad} \). If any other Sun model is to be considered for the calculation of concentration limits, the equations developed in this work should be re-adapted considering the new Sun model chosen.

### 3.2 Thermodynamics limits of concentration for solar collectors

The Sun is not a perfect linear source of energy but a gigantic sphere radiating energy isotropically. For this reason, solar radiation reaching the Earth surface is not a perfectly collimated beam. From the Earth surface, the Sun is seen as a sphere with a width of 32' (9 mrad). In ideal conditions, the radiation incoming from the Sun will be contained within this nine mrad solid angle. This is called solar cone angle, but normally, solar concentration is based on half this angle, \( \theta_e \). This geometry between the Earth and the Sun will limit the amount of concentration achievable in any concentrating system. However, there is recent work [8] that claims than the effective solar cone is actually bigger due to the effect of the Earth's atmosphere. According to Buie's measurements [11], solar concentrating collectors could be overestimating their capability for concentrating by a margin of 20 \%. The solar cone angle is dependent on the distance between the Sun and the Earth and it changes during the year. In terms of simplicity, the Sun-Earth distance was considered as a constant in this work and therefore the solar cone was considered as constant. The effects that could make the effective solar cone bigger than the ideal nine mrad were also ignored.
Chapter 3: Limits of concentration on parabolic concentrators

The second law of thermodynamics allows to calculate the maximum concentration achievable by any linear (2D) or point-like (3D) concentrating system. In this section, the highest concentration ratio achievable was evaluated for concentrating systems in the same way as others authors in the past have done [7, 58]. The purpose of this development is to offer a comprehensive review of the thermodynamics limit based on the one shown in [7]. The previous limits of concentration were found by a thermodynamic approach by Rabl [7] in the past, and this development is shown here as it follows.

Although there are works that demonstrated that solar radiation is a non-uniform extended light in intensity and angular distribution [8, 127], in this work it is assumed that the solar radiation is uniformly distributed over the range of angles ($\theta < \theta_s$) defined by the solar cone. Realistic profiles can be added to the equations developed in this chapter, but a pillbox distribution make it easier to analyse the limits of concentration, as it is the ideal distribution and any other distribution would make the concentration ratio to drop. In the past, authors exploring limits of concentration worked with pillbox distributions [7].

A given concentrator has a concentrating element with an aperture area, $A$, normal to the solar flux, and it concentrates the radiation entering its area onto a receiver with a smaller surface area, $a_{abs}$. As it is an ideal concentrator, no shadows are considered and both concentrator and absorber are considered ideal (no losses). All the rays that pass through the concentrator’s aperture are then assumed to be received by the absorber. In this case, the amount of concentration is a geometrical relationship between the two areas, known as the geometrical concentration ratio $C_g = A/a_{abs}$.

The Sun can be considered a sphere of radius, $R_{sun}$, located at a known distance from the Earth, $d_{sun}$, that isotropically emits energy, at an equivalent blackbody temperature of $T_S$. The aperture of the concentrator is normal to the imaginary line joining the centre of the Sun and Earth. As the area of the concentrator, $A$, is much smaller than the distance from Sun to Earth, the limit $A/d_{sun} \rightarrow 0$. The Sun only transfers a fraction of its total energy to the receiver depending on the geometrical relationship between both of them. If the optical system is ideal, no rays are missed and there are no heat losses in the concentrator or the receiver, and the heat received in the absorber becomes:

$$Q_{Sun \rightarrow Receiver} = \frac{A R^2}{d_{Sun}^2} \sigma T_S^4$$  \hspace{1cm} (Equation 3.1)

If an absorber of area $a_{abs}$ is considered a perfect black body as well, it will emit energy isotropically at a temperature $T_{abs}$ calculated as:

$$Q_{abs} = a_{abs} \sigma T_{abs}^4$$  \hspace{1cm} (Equation 3.2)

A fraction of this energy will be received back at the Sun. ($E_{abs \rightarrow Sun}$) is an exchange factor explained in
Enhancing concentration ratio of solar concentrators

[128]. The fraction of heat radiated to the Sun from the receiver is:

$$\dot{Q}_{\text{abs}} = a_{\text{abs}} \sigma T^4_{\text{abs}} F_{\text{abs}\rightarrow \text{Sun}}$$

(Equation 3.3)

At the limit, the temperatures of both bodies will be equal and as the second law of thermodynamics states, the net heat exchange between two bodies at the same temperature has to be zero. As the system is considered ideal, the concentration ratio is $A/a_{\text{abs}}$. Merging equations 3.1 and 3.3 the maximum concentration ratio in a concentrating system will happen for an exchange factor of 1.

$$C_{\text{max}} = \frac{A}{a_{\text{abs}}} = \frac{d^2_{\text{Sun}}}{R^2_{\text{Sun}}} F_{\text{abs}\rightarrow \text{Sun}} = \frac{1}{\sin^2 \theta_S}$$

(Equation 3.4)

In a common case, such as parabolic troughs, the concentration does not happen in a point but into a line. The geometrical analysis above will change due to both the collector and the absorber having a linear dimension, L. From Figure 3.1 a similar analysis can be conducting obtaining:

$$C_{\text{max, linear}} = \frac{1}{\sin \theta_S}$$

(Equation 3.5)

However, as mentioned previously this limit comes from assuming a linear symmetry on both the concentrating system and the Sun. Assuming no rays are missed, then this simplification is true, since the most distant ray is within the cross sectional plane. If rays are to be missed, it leads to an underestimated concentration limit.

From this development, the limits of concentration can be found if the solar semi-angle is known. Given the value of 32° (9 mrad) the limits of the concentration ratios become 46396 for 3D collectors and 215.4 for 2D collectors using equations 3.4 and 3.5.

Parabolic surfaces can form 3D or 2D concentrators. They can typically have also two type of receivers, planar or non-planar. A parabola rotated along its axis (3D concentrator) will form a parabolic dish. The revolution along the same axis of a planar receiver will form a circle, meanwhile the rotation of a non-planar one forms a sphere. The extrusion of the parabola (2D) forms a parabolic trough; the extrusion of the planar receiver forms a rectangular receiver and the extrusion of the non-planar a cylinder.

Planar receivers are typically photovoltaic cells that produce electricity. Cylindrical receivers are widely used in PTCs as a thermal receiver and fluid conduit. A spherical receiver does not exist by itself, but some applications such as Stirling engines can be an approximation of a spherical (or non-planar) receiver [129]. In every case, a real geometry of the receiver makes a parabolic concentrator fall below the maximum theoretical concentration ratio. A real absorber will need supports, connections, etc, and those elements cause the optical efficiency of the system to decrease. In 2D systems an end of collector loss appears, as described in section 6.4.2. This loss can be ignored if it is considered that the aperture is negligible compared to the receiver length of the collectors.

In this work, the classic equations for the concentration limits on parabolic surfaces with ideal receivers with no rays missed [7] are re-examined.

### 3.3 Concentration ratio of parabolic surfaces if no rays are missed

When a solar concentrator is designed, it is logical to define its absorber in a way that it is capable of receiving all the energy from the concentrating element and is as small as possible to maximise the energy concentration around its surface thus minimising thermal losses. That scenario is the typical case considered in literature and the concentration ratio equations for those absorbers are well known. In this work, a review of these equations is shown, as they become useful in exploring the maximum concentration ratio of parabolic surfaces when rays may be missed. The receivers considered in this work were cylindrical or
spherical (dependant on the parabolic concentrator being 2D or 3D) and planar. In the case of the planar receiver, the concentration ratio equations [7] were reformulated in an alternative form in this work. The equivalent equations developed here for the planar receiver were found useful in the latter case of a concentrator with missed rays.

It is necessary to consider that the shape of the Sun will have an impact on how the light is spread in an optical system. The etendue [58] is the property that defines how the light is “spread out” in an optical system and it has to be considered in the design of solar concentrator receivers. For a bundle of rays incoming from a source, the etendue is obtained by integrating all the rays included in the bundle. For practical purposes, simplifications have been done in literature assuming that the sunlight spreads only in one plane. This assumption allows the study of solar concentrators by the optical analysis of their cross sections and the limits of concentration for solar concentrators have been found in literature assuming that all the incoming rays are contained within this cross-sectional plane.

Etendue has to be conserved. In an optical system this “spread” of light can not be lowered without a loss of energy. Further details of etendue and conservation of etendue can be found in [58, 66]. In the systems considered in this thesis, the rays with a “higher” etendue or a higher angular spread are considered as those ones defining the size of the absorbers. If the reflection at a point is considered, studying the two extreme rays will allow the calculation of reflection projection onto an absorber. Conservation of etendue shows that, as further the receiver is located from the reflector, the area necessary to receive the whole bundle of rays increases. As a practical consequence, in a parabolic mirror, studying the rays reflected at the edge of the parabola is enough to define the size of the absorber. This consequence is a basic conclusion of the edge-ray theorem, which is explained in detail and demonstrated in [58].

Rabl, [7] defined the concentration ratio equations for parabolic concentrators considering that $\theta$ would be different from the angular radius of the Sun, $\theta_s$, the maximum etendue expected in a ray considering the solar cone and any inaccuracy in the concentration system such as tracking errors, misalignments, scattering in the mirror, etc. In most applications $\theta_s$ is increased to account for those effects. In this work, it was considered that at its design point, the concentrator will have $\theta = \theta_s$ and the receiver was capable of capturing all the incoming radiation. For finding the concentration ratio of a real collector, $\theta_s$ should be replaced by the real $\theta$ considered necessary for the concentrator being able to capture all the radiation considering real conditions.

3.3.1 Non-planar receiver

The first receiver considered is of non-planar design. A non-planar receiver becomes a cylinder if the parabolic reflective surface is considered to be extruded along the longitudinal axis. A cylindrical receiver is the typical shape used in parabolic troughs for thermal applications. If the parabolic surface is considered to be rotated along the axis of revolution (parabolic dish), the non-planar receiver becomes a sphere. Both cases can be analysed by considering the cross section of the system represented in Figure 3.2.
Considering the circumference of the receiver cross-section, there is no $\psi$ limitation in the parabolic mirror due to the receiver and the upper limit for the $\psi = 180^\circ$ that is an infinitely high parabola. This is an unrealistic and non-practical scenario but a very high parabola with a large $\psi$ can be designed. The concentration ratio for a parabolic trough collector (PTC) with a cylindrical receiver and an acceptance angle, $\theta$, is then:

$$C_{\text{PTC,cyl}} = \frac{\sin \psi}{\pi \sin \theta}$$  \hspace{1cm} (Equation 3.6)

and the equivalent equation for a spherical receiver in a parabolic dish (PD) becomes:

$$C_{\text{PD,sph}} = \frac{\sin^2 \psi}{4 \sin^2 \theta}$$  \hspace{1cm} (Equation 3.7)

Deriving both equations in terms of $\psi$ to find the optimum size of the parabolic mirror is straightforward and shows that it happens at $\psi = \pi/2$. For an ideal concentrator of acceptance angle $\theta = \theta_0$, the maximum concentration ratio obtained falls below the thermodynamics limit of concentration by $\pi$ in the case of a parabolic trough and by 4 in the case of the parabolic dish. In real case scenarios this concentration will drop even more and is one of the main limitations for parabolic troughs.

### 3.3.2 Planar absorbers

Some typical absorbers, such as photovoltaic cells, can be assumed to be a flat surface. It can be considered that the surface can receive the sun only by one or two faces. If the receiver is capable of receiving energy only in one of its faces it will produce a shadow the size of its own area over the parabolic surface. If a two-sided receiver is considered, its area doubles up, and then the concentration ratio will drop by the relationship shown in \cite{7}, for that reason, only one-sided receivers are considered in this work:

$$C_{\text{1sided}} = 2C_{\text{2sided}} - 1$$  \hspace{1cm} (Equation 3.8)

But a one-sided absorber will limit the reflector's, rim angle $\psi$. If $\psi \geq 90^\circ$, the rays will strike the non-absorbing area of the receiver, and will be considered missed. Now, the reflector can be considered a parabolic trough, and its receiver will become a rectangle, or a parabolic dish, and its receiver will become a circle. In both cases the problem can be studied as a cross section, but in this case, it is necessary to consider that as the receiver is not a circle, the projection of the incoming rays from one point of the parabola is not entirely symmetrical. Figure 3.3 shows this phenomenon.

If no rays are to be missed, the receiver should be chosen as the biggest of the two flat planar projections, Figure 3.4 shows detailed dimensions of both projections to obtain equations 3.9 and 3.10.
Chapter 3: Limits of concentration on parabolic concentrators

The size of the absorber will depend on the mirror dimensions but for \( \psi < \pi/2 \), \( W_{s1} \geq W_{s2} \) and the planar receiver becomes \( 2W_{s2} \). The concentration of the system can be defined as the ratio between receiver's and absorber's area if no rays are to be missed. As the two components would have the same lengths, the ratio can be analysed as the ratio between their widths.

For linear reflectors, some authors such as Winston [58] consider that, as the Sun's radius is very small, the concentration ratio can be defined for one-side receivers as:

\[
C = \frac{\sin 2\psi}{2 \sin \theta} \quad \text{(Equation 3.11)}
\]

which is the most extended equation in literature but it does not consider the shadow projected or all the effects of the Sun shape. If the optimal rim angle, \( \psi_{opt} \), is analysed equation 3.11, it can be concluded that \( \psi_{opt} = \pi/4 \) and if an ideal Sun shape is considered the maximum concentration ratio obtained is 107.7, just half the limit extracted from thermodynamic studies. These results are correct for practical purposes but not entirely accurate. Rabl [7] developed an alternative equation considering the effects of the sun shape and the shadow.

\[
C_{PTC,planar} = \frac{\sin \psi \cos(\psi + \theta)}{\sin \theta} - 1 \quad \text{(Equation 3.12)}
\]

Equation 3.8 reaches its maximum at \( \psi = \pi/4 - \theta/2 \). Now, \( \psi_{opt} \) would have to be chosen as a function of the sun shape and with analysis of an ideal scenario the maximum concentration ratio achievable for a planar receiver if no rays are to be missed becomes 106.2. In this work, an equivalent expression is used to find \( \psi_{opt} \), as its derivative becomes simpler. Obtaining this equation from equation 3.12 is not straight forward and the complete derivation is included in Appendix E.

A similar development can be conducted for a parabolic dish with a non-planar receiver. The maximum concentration ratio is found at the same \( \psi \) that in the trough case and its concentration ratio becomes:
Enhancing concentration ratio of solar concentrators

\[ C_{\text{dish,planar}} = \frac{\sin^2 \psi \cos^2(\psi + \theta)}{\sin^2 \theta} - 1 \]  
(Equation 3.13)

which shows that the maximum concentration achievable by parabolic concentrators is approximately 1/4 of the thermodynamics limit.

### 3.4 Maximum concentration ratio if rays are to be missed

The above developments consider that no rays are to be missed in the receiver. In those cases, a real receiver should be designed with \( \theta \) bigger than the sun's radiation optical extent at the receiver for every point of the concentrator. For an absorber designed with an acceptance angle, \( \theta \), matching exactly the biggest optical extent observed in the concentrator, any misalignment or inaccuracies on the system will cause rays to be missed, lowering the optical performance of the system. Also, if an ideal sun angle of 4.5 mrad is considered, a considerable percentage of energy is going to be missed due to the real effective Sun shape [11]. If no rays are to be missed and the effective solar angle after considering the real sun disc and aberrations of the concentrator or misalignments is much larger, wider receivers become necessary and the concentration ratio drops quickly. That is why, in order to maintain a high intercept factor, \( \theta > \theta_s \) \( \theta \) is normally below 1° in commercial parabolic trough collectors as it was shown in table 2.2. Even for those bigger acceptance angles, a fraction of energy is going to be missed if those Sun models shown by Buie and Monger [8] are considered.

For parabolic dishes, Gordon and Freuermann [104] proposed the idea of increasing the concentration ratio of the systems by allowing them to miss some rays at its focal plane. Missing a fraction of energy lowers the optical efficiency of the system, but smaller receivers can be placed at the focus of the system or even secondary reflectors can re-direct the missed radiation to the target. Concentration higher than the geometrical limit of parabolic concentrators shown in previous section can be achieved if rays are missed. In this work, the limits of concentration for 2D and 3D parabolic concentrators that miss some rays are studied analytically and the equations obtained are validated with a ray tracing software. In both 2D and 3D concentrators, a planar and a non-planar receiver are considered.

Missing a percentage of the incoming radiation can be useful if the maximisation of concentration is desired. If the receiver of a given concentrator is reduced, the reduction of intercept factor is not proportional to the increase of concentration and high increases of concentration can be achieved with a relative low loss of optical performance. To find the maximum concentration ratio, the parabolic surface is studied as though it is “divided” on two smaller mirrors. The first case considered is a parabolic mirror which will not miss any rays and the receiver size is then calculated to fulfil this requirement as those that can be obtained from Figures 3.2 and 3.3 for a planar and a non-planar absorber respectively. This mirror shape corresponds to a parabola up to a point at which acceptance angle is the same than the minimum acceptance angle considered. In this work ideal collectors are studied so the minimum \( \theta = \theta_c \). This mirror will define the size of the receiver and its concentration can be calculated using the equations shown in the previous section.

The biggest rim angle that does not produce missed rays at the absorber for a given mirror and absorber will be called the design rim angle, \( \psi_d \) from now on when developing concentrators that miss a part of the radiation. The mirror will be 'divided' in two parts, the points of the mirror in which \( \psi \leq \psi_d \) will reflect all the rays impacting on them effectively to the absorber, while those points which \( \psi \geq \psi_d \) will reflect some rays with a deviation high enough to be missed by the absorber.

In any concentrator, if for a certain acceptance angle, and given its dimensions, if \( \psi < \psi_d \) for all the points of the mirror, it is considered that the absorber dimensions are bigger than the necessary to capture all the radiation reflected in the primary, despite misalignments, and in this work the absorber is then re-calculated for this smaller reflector, and the concentration ratio is calculated with this smaller absorber.

Any receiver could be placed on the focus of the system, and one smaller than the re-calculated would cause the mirror to have \( \theta < \theta_s \) at some points of the parabola. For simplicity, it is assumed that there is always at
least one point of the mirror for which \( \theta \geq \theta_s \) and at the limit, the worst case scenario would be a mirror with \( \psi_d = 0 \) and \( \theta_{\psi=0} = \theta_s \) that will make all its surface to reflect rays outside the boundaries of the absorber. Figure 3.5 shows graphically a parabolic concentrator with a \( \psi_{\text{max}} > \psi_d \) considering in this case a non-planar absorber. A similar analysis can be done for the planar absorber as shown in Figure 3.6.

For any point of the mirror with \( \psi > \psi_d \), only a fraction of the reflected rays will impact the receiver, this fraction can be defined as a local intercept factor, \( I(\psi) \), and it depends on the absorber shape and the reflector’s point chosen. The intercept factor will decrease as the rim angle increases, and it will tend to 0 at the limit of the concentrator (\( \pi/2 \) for planar receivers and \( \pi \) for non-planar ones). As the reflectors and concentrators considered in this section are ideal, the total intercept factor will represent the optical efficiency, \( \eta_{\text{opt}} \), of the system.

![Figure 3.5: Parabolic reflector with a non-planar receiver which misses part of the incoming radiation.](image)

Similarly to previous literature, a cross section of the parabola is considered as shown in Figure 3.5. Several dimensions necessary to find the concentration ratio in the cases of non-planar receivers are extracted.

The half-aperture can be found as a function of the focal distance and the rim angle, \( \psi \), of the point of the parabolic surface to be studied.

\[
X = 2F \tan \frac{\psi}{2} \quad \text{(Equation 3.14)}
\]

The parabolic surface can be a trough of length \( L \) or a parabolic dish, the aperture areas are then the rectangle or the circumference formed by the parabolic surface and it will represent the amount of incoming energy to the collector.

\[
A_{\text{PTC}} = 2XL = 4FL \tan \frac{\psi}{2} \quad \text{(Equation 3.15)}
\]

\[
A_{\text{Dish}} = \pi X^2 = 4\pi F^2 \tan^2 \frac{\psi}{2} \quad \text{(Equation 3.16)}
\]

If a non-planar receiver is placed onto the focus of the parabolic concentrator its radius can be found as a function of the critical angle chosen to miss no rays. When the radius is chosen, the acceptance angle can be found for any point. \( \theta \geq \theta_s \) if \( \psi \leq \psi_d \) and it will gradually decrease as the rim angle increases, reaching 0 at the limit of the concentrator. In the case of non-planar receivers, the limit is located at \( \psi = \pi \), which represents an infinitely tall parabola. If the radius is known, the area of the receiver can also be calculated. The non-planar receiver is a cylinder for parabolic trough and a sphere for dishes of radius \( r \) given below.

\[
r = F \sec^2 \frac{\psi_d}{2} \sin \theta_s \quad \text{(Equation 3.17)}
\]
Enhancing concentration ratio of solar concentrators

\[ \theta = \arcsin \left( \sin \theta_s \sec^2 \frac{\psi}{2} \cos^2 \frac{\psi}{2} \right) \]  \hspace{1cm} \text{(Equation 3.18)}

\[ a_{cylinder} = 2\pi L = 2\pi F L \sec \frac{\psi_d}{2} \sin \theta_s \]  \hspace{1cm} \text{(Equation 3.19)}

\[ a_{sphere} = 4\pi R^2 = 4\pi F^2 \sec^4 \frac{\psi_d}{2} \sin^2 \theta_s \]  \hspace{1cm} \text{(Equation 3.20)}

Flat receivers such as photovoltaic cells can be used in concentrating systems. Depending on whether the planar receiver is capable of capturing solar radiation on one or two faces, a shadow may need to be considered as mentioned previously. If the flat surface is one-sided, the parabolic reflector surface rim angle will be limited to \( \pi/2 \), as higher rim angle will reflect the radiation to the side that the receiver is unable to capture the energy. If it is two-sided, the area of the receiver is doubled, and the concentration ratio is therefore halved. In this work, as the maximum concentration ratio is explored, only one-sided absorbers are considered. Figure 3.6 represents a cross section of a parabolic concentrator with a \( \psi \geq \psi_d \) and a planar receiver that misses some rays. If the concentrator is extruded (parabolic trough) the receiver becomes a rectangle, if the reflector is a parabolic dish, the receiver becomes a circle.

For planar receivers, when the dimensions of the absorber are chosen, it has to be considered that the problem now is not entirely symmetrical. Figure 3.6 shows this phenomena and it can be checked with equations 3.9 and 3.10 that \( W_{s2} > W_{s1} \) for any \( \psi < \pi/2 \). The receiver to choose then is \( 2W_{s2} \) as the opposite edge of the parabola will give an opposite distribution of the solar energy around the focus.

This lack of symmetry will have an influence when the intercept decreases. If the absorber is symmetrical, those rays with a “positive” deviation as shown in Figure 3.4, will be intercepted with a higher ratio than those ones with a “negative” one. As the sun semi-angle is very small, both dimensions are similar and this effect can be ignored in benefit of simplicity. In Figure 3.7, the difference between the two dimensions is shown as a ratio of the focal distance and a function of rim angle. The difference between both dimensions is minuscule compared to the total parabolic mirror dimensions for rim angles far from the limit. The dimension mismatch becomes relevant for \( \psi \approx \pi/2 \), but as the intercept factor is negligible for those angles, the solution is expected to be accurate as the product of the mismatch of the dimensions and the intercept factor becomes zero. If a more exact solution is desired to be explored, the development done here should be done for both “divisions” of the pillbox, and the total concentration ratio is to be calculated as the average of them.
Chapter 3: Limits of concentration on parabolic concentrators

The absorber’s size can be calculated from any point of the parabola if the rim angle and the acceptance angle are known. Any concentrator receiver can be designed for a certain mirror dimension, that will have an acceptance angle, $\theta$, equal to the sun semi-angle, $\theta_s$, and it will allow us to find an expression for the acceptance angle of any point on the parabola. Equations 3.21 and 3.22 can be extracted from Figure 3.6.

\[
\frac{\sec^2(\psi/2) \sin \theta}{\cos(\psi + \theta)} = \frac{W_{s2}}{F} = \frac{\sec^2(\psi_{id}/2) \sin \theta_s}{\cos(\psi_{id} + \theta_s)}
\]  
(Equation 3.21)

\[
\theta = \arctan \frac{\frac{W_{s2}}{F^2} \cos \psi}{\sin \psi + \sec^2 (\psi/2)}
\]  
(Equation 3.22)

And considering that the possible absorbers are a rectangle in the case of a parabolic trough and a circle in the case of a parabolic dish, the absorbers' areas can be calculated as:

\[
a_{\text{planar,PT}} = 2W_{s2}L = 2FL \frac{\sec^2 \frac{\psi}{2}}{\cos(\psi_{id} + \theta_s)}
\]  
(Equation 3.23)

and

\[
a_{\text{planar,dish}} = \pi W_{s2}^2 = \pi F^2 \left( \frac{\sec^2 \frac{\psi}{2}}{\cos(\psi_{id} + \theta_s)} \right)^2
\]  
(Equation 3.24)

3.4.1 Maximum concentration ratio on parabolic concentrators with bundle plane simplification

The receiver of a solar concentrator that misses no rays reflected from the reflector has to be big enough to capture those rays with the highest etendue. Applying the edge-ray theorem [58], it can be extracted that if those rays reflected in the points of a parabolic mirror with a bigger rim angle are received on a receiver placed on the focus of the parabola any other ray reflected from the mirror will also be received on the absorber. In those mirrors with areas not capable of capturing all the rays, the absorber dimensions are defined by the use of the edge-ray theorem at the design point, $\psi_{id}$. In this work it was assumed that every collector has a design point $\psi_{id}$ comprised in the ranges $[0 - \frac{\pi}{2}]$ for both planar and non-planar receivers. As the concentrators are assumed to be ideal, no rays will be missed in the range $[0 - \psi_{id}]$.

For a given receiver, capable of capturing all the incoming rays reflected by a parabolic surface of a given
design rim angle, it is possible to extend the reflective surface to a bigger rim angle, \( \psi_2 \geq \psi_d \). Any differential element of area of the concentrator will have a given acceptance angle that will depend on the absorber’s shape and dimensions. The intercept factor of that element of area will be the fraction of energy that the absorber receives and it is 1 for those points with \( \psi \leq \psi_d \) angle due to non-planar receivers having no shadowing caused by the absorber. Calculating the amount of the absorbed and missed rays the differential element aperture of the mirror allows the calculation of the fraction of rays impacting the receiver and therefore the concentration ratio that this element of area adds to the receiver. Considering the definition of concentration as in equation 2.4, a differential element of mirror that reflects some rays out of the absorber will have a concentration of:

\[
\frac{dC}{d\psi} = \frac{dA}{ad\psi} \eta_{\text{opt}} \tag{Equation 3.25}
\]

and as a differential element of aperture can be expressed as terms of the rim angle, if troughs and dishes are considered, the differential element of aperture leads to a differential element of area. Therefore from the areas of a trough and a dish defined in equations 3.15 and 3.16:

\[
\frac{dA_{\text{T}}}{d\psi} = 2FL \sec^2 \frac{\psi}{2} \tag{Equation 3.26}
\]

\[
\frac{dA_{\text{dish}}}{d\psi} = 4\pi F^2 \tan \frac{\psi}{2} \sec^2 \frac{\psi}{2} \tag{Equation 3.27}
\]

The integration of equation 3.25 along the mirror surface gives a general formula for the concentration ratio of any concentrator. If the intercept factor is 1, the formula leads to those found in literature for collectors which capture all the incoming bundle of rays.

\[
C = \int_{\psi_1}^{\psi_2} \frac{dA}{ad\psi} \eta_{\text{opt}} d\psi \tag{Equation 3.28}
\]

As described before, for an ideal concentrator, the intercept factor matches \( \eta_{\text{opt}} \) and it represents the fraction of rays reflected from the element of mirror that are received in the absorber. The parabolic trough is studied as a section, but both its absorber and its reflector have an equal length, L. The intercept factor of the parabolic trough becomes the ratio between \( \theta \) of the concentrator and the spread of the bundle of rays within the plane.

\[
\eta_{\text{opt,PTC}} = \frac{\theta L}{\theta_0 L} = \frac{\theta}{\theta_0} \tag{Equation 3.29}
\]

For 3D concentrators, the optical performance is the ratio of the circumferences that are obtained after rotating the cross-section studied. It is considered that as \( \theta_0 \) is very small and \( \theta < \theta_0 \):

\[
\eta_{\text{opt, dish}} = \frac{\pi \theta^2}{\pi \theta_0^2} = \frac{\theta^2}{\theta_0^2} \tag{Equation 3.30}
\]

Figure 3.8 shows the intercept factor for dishes and parabolic trough with planar and non-planar receivers. For \( \psi \leq \psi_d \), \( \eta_{\text{opt}} = 1 \), and as it is considered that the concentrators are ideal the effect of the shadow is excluded in the optical performance in the cases of a planar receiver, but as it is considered that the receiver is one sided and it will shade a certain area of the mirror, it will be considered in calculating the total concentration ratio by subtracting it.
Chapter 3: Limits of concentration on parabolic concentrators

3.4.1.1 Non-planar receivers

The integral shown in equation 3.28 can be divided on two parts. The first one will represent the part of the mirror that is not missing rays ($\eta_{\text{opt}} = 1$) and it will lead to the corresponding equation in previous section for each case, the second part of the integral $[\psi_d, \psi_2]$, represents those parts of the concentrator which will not reflect all the incoming rays to the reflector.

Depending on the type of concentrator, the non-planar receiver will become a cylinder (parabolic trough) or a sphere (parabolic dish). As it is assumed that the solar flux is contained within a plane, both cases can be studied as the cross section shown in Figure 3.5. The receiver is designed to capture all the rays reflected in any point $\psi < \psi_d$ while it will miss a fraction of energy of the rays reflected in the remaining mirror’s area. As the receiver is non-planar, the mirrors can be extended up to a rim angle, $\psi_2 = \pi$ that represents an infinitely tall parabola. This is an unrealistic scenario and real parabolas will have a lower final rim angle, $\psi_2$. 

Figure 3.8: Intercept factor for different $\psi$; bundle-plane simplification considered. a) Parabolic dish with spherical absorber. b) Parabolic dish with planar absorber. c) Parabolic trough with cylindrical absorber. d) Parabolic trough with planar absorber.
Considering the case of a non-planar receiver with a parabolic trough, the total concentration can be expressed as:

\[ C = \frac{\sin \theta_d}{\pi \sin \theta_d} + \int_{\psi_d}^{\psi_d+\theta} \frac{\theta}{\theta_d} \frac{dA}{d\psi} \]  

(Equation 3.31)

Inserting equations 3.17, 3.18 and 3.23 into equation 3.31, the concentration ratio equation obtained for any parabolic trough with a non-planar receiver becomes:

\[ C_{2D,cyl} = \frac{\sin \psi_2}{\pi \sin \theta_d} + \frac{\cos^2 \frac{\psi_2}{2}}{\pi \theta_d \sin \theta_d} \int_{\psi_d}^{\psi_d+\theta} \frac{\sec^2 \frac{\psi_2}{2}}{2} \arcsin \left( \frac{\sin \theta_d \sec^2 \frac{\psi_2}{2} \cos^2 \frac{\psi}{2} \theta_d}{2} \right) d\psi \]  

(Equation 3.32)

Where the first term on the integral corresponds to the concentration achievable with a parabolic trough which does not miss any rays (equation 3.6) and the second term represents the extra energy received in the absorber by extending the concentrator, despite missing part of the energy reflected by the primary mirror.

To calculate the maximum concentration ratio it is necessary to obtain the derivative of the concentration equation. As each part of the mirror (the mirror part causing all the radiation being received at the absorber and the mirror part that is not) can be optimised, the derivative is found considering the design and the final angles as independent. After some calculation it can be stated that:

\[ C'_{2D,cyl} = \frac{dC}{d\psi} = \frac{\cos \psi_2}{\pi \sin \theta_d} + \frac{\cos^3 \frac{\psi_2}{2}}{\pi \theta_d \sin \theta_d} \left[ \sec^2 \frac{\psi_2}{2} \arcsin \left( \frac{\sin \theta_d \sec^2 \frac{\psi_2}{2} \cos^2 \frac{\psi}{2} \theta_d}{2} \right) \right] \frac{\psi_2}{\psi_d} \]  

(Equation 3.33)

As any additional part of the mirror adds flux to the receiver, the maximum concentration ratio occurs if the mirror is extended up to a final rim angle, \( \psi_2 = \pi \) (with zero optical efficiency) and an infinite aperture area. The derivative obtained above does not have a solution \( C' = 0 \) on the domain \( [0, \pi] \) but it approaches 0 if \( \psi_d \to 0^\circ \) and if \( \theta_d = 0 \), previous equations can be simplified. The first term in equation 3.32 becomes zero since there are not areas of the mirror with \( \theta_d = 0 \). Therefore:

\[ C_{2D,cyl,max}^{\psi_2} = \frac{1}{\pi \theta_d \sin \theta_d} \int_0^{\pi} \sec^2 \frac{\psi_2}{2} \arcsin \left( \frac{\sin \theta_d \cos^2 \frac{\psi}{2}}{2} \theta_d \right) d\psi = 215.4 \]  

(Equation 3.34)

The maximum concentration ratio achievable then in a parabolic trough with a cylindrical receiver matches the one developed by adapting the 3D thermodynamics limit to a 2D case. In Figure 3.9 a, the concentration ratio achieved for several design points, \( \psi_d \), in a parabolic trough with a cylindrical receiver is shown. As stated before, if \( \psi_2 < \psi_d \), the absorber is considered to be re-designed for that angle. If the absorber is not considered to be redesigned for those cases, the concentration ratio obtained will be lower than the one obtained in the case of no missed rays, as the size of the receiver is bigger than the one necessary to capture all the rays. The y axis represents the concentration relative to the maximum concentration considered in previous literature reported in equation 3.5 and evaluated as 215.4. Figure 3.9 b represents a parabolic-trough with \( \psi = 90^\circ \) and two absorbers, one considering that no rays are to be missed, and a second one that misses part of the radiation and has \( \psi_d = 0^\circ \). In both cases the solar cone has been exaggerated in the drawing.
Chapter 3: Limits of concentration on parabolic concentrators

Figure 3.9: Concentration ratio for parabolic troughs with non-planar receivers missing rays. a) Relative to the thermodynamics limit concentration ratio for different design rim angles b) Size comparison of an absorber designed to not miss rays and an absorber with $\psi_d = 0^\circ$. Note that the solar cone has been exaggerated.

Proceeding in an analogous way for a 3D receiver, a similar equation can be obtained. It has to be considered now that the receiver is a whole sphere and the reflector’s aperture area becomes a circle. Re-adapting equations 3.32 to 3.34 the concentration ratio for any $\psi$ can be found as:

$$C_{\text{3D,sph}} = \frac{\sin^2 \psi_d}{4 \sin^2 \theta_s} + \left[ \frac{\cos^4 \frac{\psi_d}{2} \theta_s^2}{\theta_s^2 \sin \theta_s^2} \int_{\psi_d}^{\psi} \tan \frac{\psi}{2} \sec^2 \frac{\psi}{2} \arcsin^2 \left( \sin \theta_s \sec^2 \frac{\psi_d}{2} \cos^2 \frac{\psi}{2} \right) d\psi \right]$$

(Equation 3.35)

$$C_{\text{3D,sph}} = \frac{\sin(2\psi_d)}{4 \sin^2 \theta_s} + \left[ \frac{\cos^4 \frac{\psi_d}{2} \theta_s^2}{\theta_s^2 \sin \theta_s^2} \tan \frac{\psi}{2} \sec^2 \frac{\psi}{2} \arcsin^2 \left( \sin \theta_s \sec^2 \frac{\psi_d}{2} \cos^2 \frac{\psi}{2} \right) \right]_{\psi_d}^{\pi}$$

(Equation 3.36)

The derivative of the second mirror is indeterminate at the limit, but:

$$\lim_{\psi \to \pi} \left[ \tan \frac{\psi}{2} \sec^2 \frac{\psi}{2} \left( k \cos^2 \frac{\psi}{2} \right) \right] = 0$$

(Equation 3.37)

Considering that $k$ is a constant, the derivative becomes 0 for $\psi_d = 0^\circ$, and the maximum concentration achievable by a parabolic dish missing some rays at the absorber corresponds to an scenario in which all the areas of the mirror are making the absorber to miss some rays.

$$C_{\text{3D,sph, max}} = \frac{\cos^4 \frac{\psi_d}{2} \theta_s^2}{\theta_s^2 \sin \theta_s^2} \int_0^\pi \tan \frac{\psi}{2} \sec^2 \frac{\psi}{2} \arcsin^2 \left( \sin \theta_s \sec^2 \frac{\psi_d}{2} \cos^2 \frac{\psi}{2} \right) d\psi = 46396$$

(Equation 3.38)

The maximum concentration ratio obtained matches the thermodynamics limit for an infinitely tall parabola, and it demonstrates that concentration ratios higher than the limit of those collectors with no rays missed can be achieved, due to those falling by four the thermodynamics limit as shown in equation 3.7.
3.4.1.2 Planar receivers

For a planar receiver is considered, $\psi_{\text{max}} = \pi/2$ and shadows are to be considered. The geometrical concentration ratio of a differential element in mirror area is the ratio between the differential mirror's surface and the absorber's area. Integrating the surface in the interval $[\psi_d, \psi_2]$ the geometrical concentration ratio of the parts of the mirror with a non-ideal optical efficiency is obtained, and combining it with the intercept factor which is the ratio of the acceptance angle, $\theta_a$, obtained with equation 3.22, and the sun semi-angle, $\theta_s$, the concentration ratio of those parts of the mirror with $\eta_{\text{opt}} < 1$ is expressed as:

$$C_{\text{2D, planar}} = \int_{\psi_d}^{\psi_2} \frac{dX}{\theta_s W_{x2} \psi} = \frac{1}{\theta_s W_{x2}} \int_{\psi_d}^{\psi_2} \frac{\cos \theta_a \cos^2 \frac{\psi}{2}}{2} \arctan \frac{W_{x2} \cos \psi}{W_{x2} \sin \psi + \sec^2 \frac{\psi}{2}} d\psi \quad (\text{Equation 3.39})$$

And considering the no missed rays region of the mirror and the shadow projected by the receiver, the total concentration ratio for any parabolic trough with a planar absorber is obtained.

$$C_{\text{2D, planar}} = \sin \psi_d \cos \psi_d \tan \theta_s - \sin^2 \psi_d - 1 + \frac{\cos(\psi_d + \theta_a) \cos^2 \frac{\psi}{2}}{\theta_d \sin \theta_a} \int_{\psi_d}^{\psi_2} \frac{\sec^2 \frac{\psi}{2}}{2} \arctan \frac{W_{x2} \cos \psi}{W_{x2} \sin \psi + \sec^2 \frac{\psi}{2}} d\psi \quad (\text{Equation 3.40})$$

As in the case of non-planar receivers, the maximum concentration ratio occurs at $\psi_2 = 90$, which is now the maximum $\psi$ that the mirror can have without the rays being blocked by the planar absorbers non receiver face. From equation 3.22 it is possible to evaluate that $\theta_a = \pi/2 = 0$. The optical efficiency for $\psi \approx \pi/2$ is very small, as the optical efficiency quickly drops, but in order to explore the maximum concentration ratio they should still be considered. To find the maximum concentration ratio achievable, the derivative of equation 3.40 has to be calculated considering that the two parts of the mirror (the one that does not miss rays and the one that does) can be optimised.

$$C'_{\text{2D, planar}} = \frac{d(C)}{d\psi} = \csc \theta_s \cos(2\psi_d + \theta_a) + \left[ \frac{\cos(\psi_d + \theta_a) \cos^2 \frac{\psi}{2}}{\theta_d \sin \theta_a} \frac{\sec^2 \frac{\psi}{2}}{2} \arctan \frac{W_{x2} \cos \psi}{W_{x2} \sin \psi + \sec^2 \frac{\psi}{2}} \right]_{\psi_2} \quad (\text{Equation 3.41})$$

Equation 3.41 can be simplified since the final rim angle, $\psi_2$, term of the derivative becomes 0 for $\psi_2 = \pi/2$ and the maximum can be found as a function of the design rim angle, $\psi_d$.

$$C'_{\text{2D, planar}} = \frac{d(C)}{d\psi} = \csc \theta_s \cos(2\psi_d + \theta_a) + \left[ \frac{\cos(\psi_d + \theta_a) \cos^2 \frac{\psi}{2}}{\theta_d \sin \theta_a} \frac{\sec^2 \frac{\psi}{2}}{2} \arctan \frac{W_{x2} \cos \psi}{W_{x2} \sin \psi + \sec^2 \frac{\psi}{2}} \right]_{\psi_d} \quad (\text{Equation 3.42})$$

The expression does not have an exact numerical solution and it will depend not only on the design rim angle, but also on the sun semi-angle, as it will define the width of the absorber, $W_{x2}$. However, the function is continuous in the range $[0, \pi/2]$ and as the sun semi-angle is considered to be small enough, these particular solutions can be obtained.

$$C_{\text{2D, planar}}(\psi_d = 0) = \sin \theta_s \quad (\text{Equation 3.43})$$

$$C'_{\text{2D, planar}}(\psi_d = \theta_a) = \sin(2\theta_a) - \sin(3\theta_a) \quad (\text{Equation 3.44})$$

For small values of $\theta_s$ as the ideal solar cone, equation 3.43 is positive and equation 3.44 is negative. So applying the Bolzano – Weierstrass theorem [130] we can conclude that the maximum concentration occurs when $0 \leq \psi \leq \theta$. In any case which $0 \leq \psi_d \leq \theta$, the mirror is designed not to miss any rays is blocked by the
Chapter 3: Limits of concentration on parabolic concentrators

shadow of the receiver. If $\psi_d = 0$, the area of the primary mirror is 0 and it will not have any concentration ratio. If $\psi_d = \theta$, the shadow of the receiver will be projected exactly in the whole mirror area, and the “-1” term of the equation will negate this term. The maximum concentration ratio expected was 214.4, as the limit extracted from adapting the thermodynamics limit of 3D concentrators was 215.4. Iterating for several values between 0 and $\theta$, the maximum concentration found was 214, which is comparable to the expected result. Figure 3.10 shows the relative concentration ratio (compared with the 2D limits of concentrators) achievable for different $\psi_d$.

Figure 3.10: Relative to the thermodynamics limit concentration ratio for parabolic trough with planar receivers missing rays.

Applying the same procedure to a dish with a planar receiver, the concentration and the maximum concentration found are:

\[
C_{3D,\text{planar}} = C_1 + C_2 \tag{Equation 3.45}
\]

\[
C_1 = \frac{\sin^2 \psi_d \cos^2(\psi_d + \theta_a)}{\sin^2 \theta_d} - 1 \tag{Equation 3.46}
\]

\[
C_2 = \frac{4 \cos^2(\psi_d + \theta_a) \cos^4 \frac{\psi_d}{2}}{\theta_d^2 \sin^2 \theta_a} \int_{\psi_d}^{\psi} \tan \frac{\psi}{2} \sec^2 \frac{\psi}{2} \arctan^2 \left( \frac{W_{\psi} \cos \psi}{\frac{W}{\cos \psi} \sin \psi + \sec^2 \frac{\psi}{2}} \right) d\psi \tag{Equation 3.47}
\]

And the derivative becomes:

\[
C'_1 = \frac{2 \sin \psi_d \cos(\psi_d + \theta_a) \cos(2\psi_d + \theta_a)}{\sin^2 \theta_a} \tag{Equation 3.48}
\]

\[
C'_2 = \frac{4 \cos^2(\psi_d + \theta_a) \cos^4 \frac{\psi_d}{2}}{\theta_d^2 \sin^2 \theta_a} \left[ \tan \frac{\psi}{2} \sec^2 \frac{\psi}{2} \frac{W_{\psi} \cos \psi}{\frac{W}{\cos \psi} \sin \psi + \sec^2 \frac{\psi}{2}} \right]_{\psi_d} \tag{Equation 3.49}
\]
At $\psi = \pi$, the derivative of $C_2$ becomes 0, and the maximum can be calculated considering that angle as:

$$C_{\text{3D,planar}}' = \frac{2 \sin \psi d \cos(\psi d + \theta_s) \cos(2\psi d + \theta_s)}{\sin^2 \theta_s} + \frac{4 \cos^2(\psi d + \theta_s)}{\theta_d^2 \sin^2 \theta_s} \sin \frac{\psi d}{2} \cos \frac{\psi d}{2} \left[ \arctan^2 \frac{W}{E^2 \cos \psi d} \right]$$

(Equation 3.50)

The derivative becomes null if $\psi_d = 0$. In that case $C_1 = 0$ and the maximum concentration ratio achievable can be found as:

$$C_{\text{3D,planar, max}} = \frac{4 \cos^2(\psi d + \theta_s) \cos^4 \frac{\psi d}{2}}{\theta_d^2 \sin^2 \theta_s} \int_0^\pi \tan \frac{\psi}{2} \sec^2 \frac{\psi}{2} \arctan^2 \frac{W}{E^2 \cos \psi} \sin \psi + \sec^2 \frac{\psi}{2} \, d\psi - 1 \approx 30787$$

(Equation 3.51)

Which is approximately 65% of the thermodynamics limit.

### 3.5 Maximum concentration ratio without bundle-plane simplification

In the previous section the limits of concentration were explored for 2D and 3D parabolic concentrators considering that the sunlight has angular spread in a plane containing a cross-section of the concentrator only. Previous literature demonstrated this method to be accurate for solar collectors designed to capture all the solar flux [131]. If rays can be missed in order to increase concentration, parabolic trough concentrators can achieve the expected limits of concentration, while parabolic dishes with planar receivers are still far from the theoretical limits. The limit of parabolic trough concentrators was extracted by simplifying the parabolic dish case, calculated with thermodynamic laws in section 3.2. In this section, the limits of both parabolic dishes and trough concentrators are explored considering the effects of the whole solar cone, without assuming that it is contained within a plane. The limits of concentration found for parabolic dishes agree with the bundle-plane simplification for a non-planar receiver and they differ in the case of a planar receiver. In the case of parabolic troughs, the limits presented here are higher than the theoretical limit found in literature. This is thermodynamically possible, since the simplification made to find the limit of 2D concentrators was based on the bundle-plane assumption, and it will be demonstrated to be invalid for those collectors which are missing rays.

A bundle of rays impacting a differential element of area of a parabolic surface is reflected as a cone with an axis connecting the reflection point and the focus of the parabola; the angle formed between the generatrix and its axis is equal to the half solar cone if the reflector is considered ideal. If a receiver is placed at the focus, the bundle of rays is intercepted when the projected area of the receiver is bigger than the solar cone projected area. That case corresponds to the cases studied in literature and it is accurate if the sun is considered to be contained within a plane [7]. If the projected area of the solar cone is bigger than the receiver's area, some rays will be missed.

A cone intersection onto a plane defines a conic curve depending on the angle formed between the cone axis and the plane. The different possibilities were studied by Apollonius de Perga in ancient Greece and they depend on the angle between axis and plane; there are four possibilities as shown in Figure 3.11.

- The axis and the plane are perpendicular: The section is a circle.
- The axis and the plane are parallel: The section is an hyperbola.
- The plane angle is parallel to the generatrix: The section is a parabola.
- None of this cases: The section is an ellipse
Chapter 3: Limits of concentration on parabolic concentrators

3.5.1 Missed rays on a 3D receivers

In the case of a parabolic dish, its non-planar receiver becomes a sphere when the plane is rotated. Any plane that passes through the centre of the sphere will section the receiver in a circle. Therefore there will always be a plane that intersects the solar cone in a circle for the reflected rays from any point of the parabola. The section of the circle circumscribed by the circumference represents the bundle of rays intercepted by the absorber. If the cone section is bigger than the absorber's, there is a circular crown section corresponding to the missed rays. The dimensions of the intercepted solar flux corresponds to the area of the circumference circumscribed on the sphere and therefore it depends on the absorber dimensions and the dimensions of the solar cone base which depends on the point chosen in the parabola.

Figure 3.12 represents the reflections of the solar bundle on two points of a parabolic dish and shows the amount of rays that the receiver is not able to absorb, a perpendicular plane to the bundle that contains the receiver sphere's centre can be defined for each one of these points. The receiver and the solar bundle projection onto these planes are both a circle.
The receiver radius ($r$) is defined by $\psi_d$ as in equation 3.17, for the point of the parabola to be studied, the projection of the solar bundle is contained in a circle of radius, $R$. For every point which solar bundle's projection is bigger than the sphere section, the intercept factor can be expressed as the ratio between the two projected areas. If the point to be studied is considered, the radius of the solar bundle projection at the plane can be calculated as a function of the parabolic mirror dimensions as:

$$R = F \sec^2 \frac{\psi}{2} \tan \theta$$  \hspace{1cm} (Equation 3.52)

Figure 3.13 shows that the intercepted rays projection does not have the same dimensions than the absorber's projection on that plane. All the intercepted rays strike the absorber before the perpendicular plane, and at the limit the last rays intercepted are tangent to the receiver. The projection of those tangent rays onto the perpendicular plane have an effective radius ($r'$) bigger than the receiver's:

$$r' = F \sec^2 \frac{\psi}{2} \tan \theta$$  \hspace{1cm} (Equation 3.53)
The intercept factor has to consider the area of both projections and the optical performance becomes:

\[
\eta_{\text{opt}} = \frac{\pi r^2}{\pi R^2} = \frac{r^2}{R^2} = \frac{\tan^2 \theta}{\tan^2 \theta_s},
\]

(Equation 3.54)

where \( \theta \) for any point of a parabolic mirror and a non-planar receiver was defined in equation 3.18.

The incoming radiation onto a parabolic dish is equal to its aperture area, therefore a circle. Knowing the absorber dimensions the geometrical concentration ratio of a differential element of area of a point of parabolic dish can be found as:

\[
\frac{\text{d}C}{\text{d}\psi} = \frac{\text{d}A}{\text{d}\psi} = \frac{4\pi F^2 \tan \psi \sec^2 \frac{\psi}{2}}{4\pi F^2 \sin^2 \theta_s \tan^2 \theta_s \frac{4\tan^2 \frac{\psi}{2}}{\sin^2 \psi_d}}
\]

(Equation 3.55)

And the total concentration ratio for any parabolic dish which is designed for capturing all the rays up to certain \( \psi_d \) but bigger dimensions, \( \psi_{2\text{L}} \), can be calculated combining equations 3.7 and 3.55.

\[
C = C_1 + C_2 = \frac{\sin^2 \psi_d}{4 \sin^2 \theta_s} + \frac{\cos^4 \frac{\psi_d}{2}}{\sin^2 \theta_s \tan^2 \theta_s} \int_{\psi_d}^{\psi_{2\text{L}}} \tan \frac{\psi}{2} \sec^2 \frac{\psi}{2} \tan^2 \theta \text{d}\psi
\]

(Equation 3.56)

The maximum is located, as in the bundle-plane simplification, at \( \psi_d = 0 \) and matches the thermodynamics limit, as expected. Figure 3.14 shows the concentration ratio achievable for different rim angles, \( \psi \), depending on \( \psi_d \).

The planar absorber for a parabolic dish is a flat surface and becomes a circle. In this case, the projected rays of the solar cone over the plane of the receiver will form an ellipse. As it was shown in equations 3.9 and 3.10, the projections of a cross section are not entirely symmetrical, therefore the ellipse is not centred at the
focus of the parabola. As it was shown in Figure 3.7, the difference is negligible for most of the range of the mirror and in order to increase simplicity the two projections are considered identical. With this simplification considered, the minor and major axis of the ellipse will be centred at the focus of the concentrator and the dimensions of the two axes and the area of the projection can be calculated from the dimensions at point of the parabola chosen:

\[
S = \frac{F \sec^2 \frac{\psi}{2} \sin \theta_s}{\cos(\psi + \theta_s)} \quad \text{(Equation 3.57)}
\]

\[
T = F \sec^2 \frac{\psi}{2} \tan \theta_s \quad \text{(Equation 3.58)}
\]

\[
Ac = \pi ST = \pi F^2 \sec^4 \frac{\psi}{2} \sin \theta_s \tan \theta_s \frac{\tan \theta_s}{2 \cos(\psi + \theta_s)} \quad \text{(Equation 3.59)}
\]

For those points close to the design point, \(\psi_d\), but with a bigger rim angle, the solar cone projected by a differential element of the reflector will not illuminate a certain section of the non-planar receiver as shown in Figure 3.16 while at the same time some rays are starting to be missed.

This region is larger as \(\psi_d\) increases, but the effect is not considered here to increase simplicity. This effect does only happen when the mirror point’s rim angle is slightly bigger than the design rim angle. After that,
Chapter 3: Limits of concentration on parabolic concentrators

the absorber is “contained” inside the flux distribution and the ratio of both areas represents the intercept factor. If this region is ignored and as the receiver is contained on the plane studied, the intercept factor for a differential element of area becomes the ratio between the absorber area and the projection of the solar cone.

\[ C = C_1 + C_2 = \frac{\sin^2 \psi \cos^2 (\psi + \theta)}{\sin^2 \theta} + \frac{4}{\tan \theta_1 \sin \theta_1} \int_{\psi_1}^{\psi_2} \sin \frac{\theta}{2} \cos \frac{\psi}{2} \cos (\psi + \theta_2) \, d\psi \]  

(Equation 3.60)

Equation 3.60 is much simpler the one obtained if the effect in Figure 3.16 is considered, but as the intercept factor is overestimated in a portion of the parabolic dish the total concentration ratio will be overestimated as well. However, in an ideal scenario, the ellipse would become a circle and the intercept factor would not be overestimated, so it is considered that the maximum concentration ratio achieved here is a good reference as a maximum concentration ratio for parabolic dishes with a planar receiver. The fact that the intersection becomes an ellipse wouldn’t affect in the flux distribution, that will remain specially uniform. To corroborate that, a ray tracing simulation was conducted. The simulation considered a point source impacting a random point of a parabolic trough, and the flux distribution in the focal plane was calculated. Figure 3.17 shows the results of the simulation, which consisted of ten millions rays and a square detector of 7.5x7.5 mm and 2000x2000 pixels.

![Figure 3.17: Ray tracing conducted with ten million rays to prove the uniformity of the flux at the focal plane. a) 3D model of the dish, only the reflected rays are shown b) Flux distribution at the focal plane, colour bar represents number of hits.](image)

The maximum concentration achieved was approximately 46000, slightly below thermodynamics limit. The ray tracing simulations conducted to validate the calculations shown a concentration ratio of approximately 45200, a 98.2 % of the concentration predicted. As expected, there was some overestimation on the maximum concentration, due to the simplifications on the missed rays in the elliptical section.

3.5.2 Concentration limits of 2D concentrators if no cross section simplification is applied

In the previous section, the limits of concentration were approached for 2D concentrators considering a cross section of the system. This consideration simplifies the calculations and it is accurate if it is considered that all the rays are to be captured in the receiver. The incoming bundle of rays is considered to be also distributed in a plane and that assumption makes the limits of concentration to be underestimated if it is
Enhancing concentration ratio of solar concentrators

Considered that some rays can be missed. If a given point of a mirror, reflects the bundle of rays to a receiver that is unable to absorb all the incoming rays, the 2D receiver (planar or non-planar) can be considered infinitely long. The projection of the bundle of rays has to be considered as a circumference for non-planar receivers and as an ellipse for planar ones. No matter which acceptance angle is chosen, those rays projected in the longitudinal plane of the system will be captured, meanwhile those ones in different planes will be dependant on the acceptance angle, $\theta$. It is necessary then to calculate the projections of the solar bundle at the relevant plane and in the intercept factor it will be considered that, as the receiver is long enough, only a sector of circumference or ellipse will be missed.

3.5.2.1 Cylindrical receiver

For the cylindrical receiver, for any point of the parabola there is a plane that contains the cylinder's central axis and it is parallel to the base of the solar cone reflected from that point of the reflectors. The section of the reflected bundle of rays is then a circle and the rays intercepted by the receiver are those within the intersection of the circle and the rectangular projection of those rays tangent to the cylindrical receiver.

![Diagram](image)

Figure 3.18: 3D schematics of the missed energy of one point of the reflector with $\psi > \psi_d$

The cross-section of the system matches with the one shown for the spherical receiver in Figure 3.13, therefore $R$, $r$ and $r'$ can be calculated using equations 3.17, 3.52, and 3.53. Figure 3.14 shows how the intercepted energy can be divided on two circular sectors and two triangular sectors. If the dimensions of those sectors are found, the optical performance for a differential element of mirror area can be calculated. Figure 3.19 shows a detail of the dimensions of the circular and the triangular sector.
Chapter 3: Limits of concentration on parabolic concentrators

Figure 3.19: Schematic of the intercepted energy of a differential point of reflector.

\[
\eta_{\text{opt}} = 2 \frac{\text{Area triangle} + \text{Area circular sector}}{\pi R^2} \tag{Equation 3.61}
\]

\[
\alpha = \arcsin \frac{r'}{R} = \frac{\tan \theta}{\tan \theta_s} \tag{Equation 3.62}
\]

Figure 3.20: Schematic of the circular and triangular sector.

The total area of the circular and triangular sectors (it is necessary to notice that there are two of each, can be calculated as:

\[
A_{\text{circular}} = 2\alpha R^2 \tag{Equation 3.63}
\]

\[
A_{\text{triangular}} = 2r'R\cos\alpha \tag{Equation 3.64}
\]
Therefore, the optical performance of a differential element of area \( \frac{dA}{d\psi} \) can be found after some derivation;

\[
\eta_{opt} = \frac{2}{\pi} \left( \arcsin \frac{\tan \theta}{\tan \theta_s} + \frac{\tan \theta}{\tan \theta_s} \sqrt{1 - \arcsin^2 \frac{\tan \theta}{\tan \theta_s}} \right)
\]

(Equation 3.65)

The acceptance angle can be found with equation 3.18, and its tangent is:

\[
\tan \theta = \frac{\sin \theta_s \sec \frac{\psi}{2} \cos \frac{\psi}{2}}{\sqrt{1 - \left( \sin \theta_s \sec \frac{\psi}{2} \cos \frac{\psi}{2} \right)^2}}
\]

(Equation 3.66)

Combining equations 3.65 and 3.66 with the ratio between the differential element of area and the absorber’s dimensions and integrating along the aperture of the reflector from its design point up to the end of the mirror, the concentration ratio of the section of the concentrator that is missing radiation in its absorber. If it is added to the concentration achieved for those parts of the reflector not missing any rays, the total concentration can be found as:

\[
C_{2D,cyl} = \frac{\sin \psi_d}{\pi \sin \theta_s} + \frac{2 \cos^2 \frac{\psi_d}{2}}{\pi^2 \sin \theta_s} \int_{\psi_d}^{\psi_s} \sec^2 \frac{\psi}{2} \left( \arcsin \frac{\tan \theta}{\tan \theta_s} + \frac{\tan \theta}{\tan \theta_s} \sqrt{1 - \arcsin^2 \frac{\tan \theta}{\tan \theta_s}} \right) d\psi
\]

(Equation 3.67)

The derivative of the concentration ratio shows again that the maximum concentration ratio achievable for a parabolic trough corresponds to a parabolic trough which receiver is designed at \( \psi_d = 0 \) and it is extended up to \( \psi_2 = \pi \).

\[
C'_{2D,cyl} = \frac{\cos \psi_d}{\pi \sin \theta_s} + \frac{2 \cos^2 \frac{\psi_d}{2}}{\pi^2 \sin \theta_s} \left[ \sec^2 \frac{\psi}{2} \left( \arcsin \frac{\tan \theta}{\tan \theta_s} + \frac{\tan \theta}{\tan \theta_s} \sqrt{1 - \arcsin^2 \frac{\tan \theta}{\tan \theta_s}} \right) \right]_{\psi_d}^{\pi}
\]

(Equation 3.68)

\[
C_{2D,cyl,max} = \frac{2}{\pi^2 \sin \theta_s} \int_0^{\pi} \sec^2 \frac{\psi}{2} \left( \arcsin \frac{\tan \theta}{\tan \theta_s} + \frac{\tan \theta}{\tan \theta_s} \sqrt{1 - \arcsin^2 \frac{\tan \theta}{\tan \theta_s}} \right) d\psi \approx 254
\]

(Equation 3.69)

The maximum concentration ratio achievable by a parabolic trough with a cylindrical absorber is 254, which is higher that the limit of concentration estimated in previous literature for any 2D concentration system. That demonstrates that the previous formulation for the parabolic trough limits, though practical when designing parabolic troughs, is not accurate if \( \psi_2 > \psi_b \).

### 3.5.2.2 Planar receiver

The intersection of the solar cone with a plane orthogonal to the focal line is an ellipse, as seen in previous developments. The ellipse scenario can be overcomplicated if no assumptions are done. For maintaining simplicity, it is assumed that the ellipse is symmetrical along the longitudinal axis of the system.

The projection of the bundle of rays over a plane which contains a planar receiver was explained in equations 3.57 to 3.59. In Figure 3.21 it can be seen how some of the reflected rays will be missed by the mirror. Calculating the area of the projection contained within the receiver the fraction of rays intercepted in the absorber can be calculated.
Applying symmetry, the rays intercepted can be found as the sum of a triangular sector and an elliptical sector as in Figure 3.22. To calculate these areas, it is necessary to calculate the y coordinate of the parabola. It is necessary to use an auxiliary circumference, circumscribed to the major axis of the ellipse to calculate the intersection points of the rectangular absorber and the solar cone section as shown in Figure 3.22. The width of the receiver and the acceptance angle, $\theta$, were defined in equations 3.10 and 3.22.
Enhancing concentration ratio of solar concentrators

\[ y_c = \theta \sin \arccos \frac{\theta}{\theta_s} = \theta \sqrt{1 - \left( \frac{\theta}{\theta_s} \right)^2} \]  
(Equation 3.70)

\[ y_e = \frac{S_e}{T_e} y_c = \frac{\theta \cos \theta_s}{\cos(\psi + \theta_s)} \sqrt{1 - \left( \frac{\theta}{\theta_s} \right)^2} \]  
(Equation 3.71)

\[ \alpha = \arccos \frac{W_s}{S_e} = \arccos \frac{\sin \theta \cos(\psi + \theta_s)}{\sin \theta_s \cos(\psi + \theta)} \]  
(Equation 3.72)

As the acceptance angle is always lower than the sun semi-angle, and both of them are assumed to be very small, the following simplifications are considered:

\[ \frac{\cos(\psi + \theta_s)}{\cos(\psi + \theta)} = 1 \]  
(Equation 3.73)

\[ \frac{\sin \theta}{\sin \theta_s} = \frac{\theta}{\theta_s} \]  
(Equation 3.74)

Both simplifications are accurate for a wide range of \( \psi \) but the cosine simplification loses accuracy when the \( \psi \rightarrow \pi/2 \). However, as shown in Figure 3.8 in the previous section, the intercept factor at high rim angles will approach 0, and therefore the loss of accuracy of the cosine simplification at high numbers is irrelevant.

Taking into account both simplifications, equation 3.72 is simplified as

\[ \alpha = \arccos \frac{\theta}{\theta_s} \]  
(Equation 3.75)

which leads to simplified versions of equations 3.70 and 3.71, necessary to define \( \beta \) to calculate the area of the elliptical sector.

\[ y_c = S \sin \alpha \]  
(Equation 3.76)
Chapter 3: Limits of concentration on parabolic concentrators

\[ \beta = \arctan \frac{y_e}{W_{s2}} = \arctan \left[ \frac{\theta_e \sin (\arccos \frac{\theta}{\theta_e})}{\theta \cos (\psi + \theta_e)} \right] \]  
(Equation 3.78)

Finally, if \( \beta \) is known, the area of the elliptical sector can be calculated as [132]:

\[ A_{\text{elliptical sector}} = F(\beta) - F\left(\frac{\pi}{2}\right) \]  
(Equation 3.79)

\[ F(\gamma) = \frac{S_e T_e}{2} \left[ \gamma - \arctan \left( \frac{(T_e - S_e) \sin 2\gamma}{T_e + S_e + (T_e - S_e) \cos 2\gamma} \right) \right] \]  
(Equation 3.80)

\[ A_{\text{elliptical sector}} = ST \left[ \beta - \arctan \left( \frac{(T_e - S_e) \sin 2\beta}{T_e + S_e + (T_e - S_e) \cos 2\beta} \right) \right] - \frac{\pi S_e T_e}{2} \]  
(Equation 3.81)

The area of the triangular sectors is a function of the width of the receiver and the \( y \) coordinate of the ellipse.

\[ A_{\text{triangular sector}} = 2W_{s2}y_e \]  
(Equation 3.82)

And the optical efficiency can be calculated as:

\[ \eta_{\text{opt}} = \frac{A_{\text{triangular sector}} + A_{\text{elliptical sector}}}{\pi S_e T_e} \]  
(Equation 3.83)

The maximum concentration ratio, located at \( \psi_d = 0 \) is found from adapting equation 3.28 a with the efficiency found by 3.83 and it is approximately 240, a 11% higher than the maximum concentration calculated for any 2D concentrator with the simplifications found in the literature.

### 3.6 Ray tracing simulations

Disregarding the case of a parabolic dish with a spherical receiver, which is completely symmetrical, the rest of the concentration limits predicted changed depending on whether the bundle-plane simplification was applied or not. In the case of a parabolic trough, the limits obtained if the simplification were not conducted are higher than the maximum limit established in previous literature.

In order to validate the results of the previous section, ray tracing simulations were conducted with Zemax software. The ray tracing simulations were conducted without simplifications either in the source or the concentrating system since it is not possible to simulate a light source within a plane in the software chosen.

The solid-cone Sun semi-angle was defined with a maximum deviation of 16'. To approximate a complete pillbox distribution in the software used, the total energy was divided in 100 bins. Each one of these bins takes a deviation from 0 to \( \theta \) and it has an intensity of 1/100 the total intensity chosen. A ray independence study was conducted to assure that the number of rays used in the simulations had no influence on the final result.

The reflectors simulated are considered ideal, and its location was also defined without any deviation. The focal distance for both the parabolic trough and the parabolic dish used in the simulations was 500 mm and the absorber dimensions were calculated using previous equations for those design angles considered and it
was defined as an ideal absorber. It was checked that for those angles lower than the $\psi_d$ the optical efficiency of the system was 100%.

In the case of the parabolic dish, the entire dish was simulated and a source of the same size as its aperture area was defined. In the case of the parabolic trough, to avoid the end of collector losses, a 10 m long parabolic trough and cylindrical receiver was simulated, meanwhile the solar source was much narrower, matching the aperture length of the mirror in each case.

Planar receivers were simulated up to $\psi_d = \pi/2$. For non-planar receivers the maximum rim angle chosen was 140°. For angles bigger than 150° an erratic behaviour was shown in the simulations, since the optical efficiency of the system was observed to remain at a constant value of 93.1% and therefore the concentration ratio obtained was determined to be incorrect. This error could be due to the relatively large dimensions of the source and mirror when rim angle approaches 180°. The height of the parabola increases rapidly at those $\psi$, tending to infinite. The primary mirror and absorber length had to be increased massively to avoid end of collector losses and the amount of rays had to be also increased a lot to make the software being able to properly distribute the rays onto the mirror. Errors in the approximation of the parabola that the CAD software used is implementing at such big rim angle could also be important. At these large dimensions, rounding also becomes relevant and the accuracy of the simulations were found to be insufficient. A refining on the code implemented and the use of a more powerful computer that would let to increase the amount of rays to a much higher number could help to solve this issues. However, simulating until $\psi = 140^\circ$ is enough to validate the analytical equations.

Figure 3.24 shows the comparison between the analytical equations developed and the ray tracing simulations conducted in order to validate them. There was a good agreement shown for the 2D concentrators and for the parabolic dish with a spherical receiver. However, for big design angles, $\psi_d$ there is a notable mismatch between the equations and the ray tracing simulations for the dish concentrator with flat receiver. The cause for this is the ellipse intersection simplification explained in Figure 3.16. As the design rim angle increases, a bigger region of the ellipse is not “illuminated” for those differential elements of the mirror close $\psi_d$ and the formula represented in equation 3.60 overestimates the concentration achieved. However, the agreement was greater for small design rim angles, and it was found acceptable at the limits of concentration. The relative concentration ratio obtained in this case with the analytical equations was 0.992 while the one obtained with ray tracing was 0.975. As equation 3.60 is intended to find the new concentration limits and it is simpler than those without any simplification, it was considered acceptable in the scope of this work. In future works, the accurate intercept factor of a parabolic dish with a planar receiver should be found to explore the limits of concentration for any $\psi_d$. 

Enhancing concentration ratio of solar concentrators
Chapter 3: Limits of concentration on parabolic concentrators

3.7 Optical efficiency versus concentration ratio

As it was shown in previous sections, an increase of concentration ratio approaching values close to the theoretical limits are possible for parabolic concentrators if rays are missed. The optical efficiency of these collectors will be low for cases with $\psi > \psi_D$. It is necessary to consider that for cylindrical receivers, reaching the limit implies a concentrator of infinite dimensions if $\psi \rightarrow \pi$. Figures 3.9 and 3.10 shows that the decrease of maximum concentration is low for mirrors with a low design rim angle and the limit falls rapidly for those design rim angle bigger than the optimal $\psi$ of collectors not missing any rays, which was approximately $\pi/2$ for planar and $\pi$ for non-planar receivers.

If mirrors are considered to be extended up to a realistic $\psi < \pi$ for cylindrical receivers and close to $\pi/2$ for planar receivers, the drop in concentration compared with the maximum achievable is not significant, as the optical performance of collectors falls rapidly as they tend to the limit as shown in Figure 3.8. In the case of non-planar receivers, a decrease of the rim angle implies almost a linear concentration drop, but for those...
areas close to the limit, an increase of rim angle implies a massive increase of area, and it is mandatory to reduce it in order to achieve practical mirrors.

Freuermann et al. [133] developed a way to choose the appropriate dimensions for a parabolic dish that misses some rays at its focal plane by plotting the optical efficiency versus the concentration ratio. The total optical efficiency of an ideal system was calculated as the ratio between the average flux concentration at the receiver divided by the geometrical concentration ratio that represents the concentration ratio that would be achieved if the optical efficiency is 1. Concentrator limitations such as reflectivity being less than unity, aberrations or misalignments were ignored.

Freuermann et al. [133] considered the final dimensions of $\psi$ as fixed and they modified the absorber dimensions from 0 to the minimum dimension that intercepted the whole bundle of rays. They studied a case of a parabolic dish and a planar receiver, since they considered only the focal plane of the system. The shadow of the receivers was not considered as the maximum optical efficiency shown is 1. In any case, as the receiver size was adapted to small $\psi$, and as it is a parabolic dish, the shadow is negligible.
Figure 3.25: Optical efficiency vs concentration ratio for a mirror focusing light on a receiver missing rays using the bundle-plane simplification. a) Parabolic dish with spherical absorber. b) Parabolic dish with planar absorber. c) Parabolic trough with cylindrical absorber. d) Parabolic trough with planar absorber.

Figure 3.25 represents the relative concentration ratio versus the optical efficiency achievable for 2D and 3D parabolic concentrators if the plane section simplification is considered. Additionally, Figure 3.26 shows concentration and optical efficiency without the cross-section simplification considered. In this case, for each design rim angle, the absorber dimensions are considered to be fixed for that design point independent of the rim angle being smaller or bigger. In the case of the planar receiver, the absorber’s shadow is considered and the optical efficiency is lower than 1 even for those regions with $\psi < \psi_{th}$. For a given $\psi_{th}$, the plots show how, by extending $\psi_2$, a concentration closer to the limit can be reached by obtaining a lower optical efficiency.
Figure 3.26: Optical efficiency vs concentration ratio for a mirror focusing light on a receiver missing rays without using the bundle-plane simplification. a) Parabolic dish with spherical absorber. b) Parabolic dish with planar absorber. c) Parabolic trough with cylindrical absorber. d) Parabolic trough with planar absorber.

Figure 3.27 shows a ray tracing conducted for a parabolic mirror with both a cylindrical and a planar receiver. Sub-figures a and b show the analysis conducted for the mirror at its design point, and no rays are missed. Sub-figures c to e show the effects of extending the mirror up to bigger rim angles. The solar cone is not to scale in the figure, as a bigger solar-cone angle was used to make the number of missed rays increase rapidly in the simulations to make the missed rays noticeable in the figure, which only plots 250 rays per simulation despite 250 k rays being calculated. The planar receivers width is also exaggerated to make it visible in the plots, although their real dimensions were used to calculate the losses.
Chapter 3: Limits of concentration on parabolic concentrators

3.8 Summary and discussion.

In this chapter, the limits of concentration for solar parabolic collectors were discussed. The limit of concentration of any concentrating system depends on the physics of the Sun and it is limited by the second law of thermodynamics. In previous literature, the parabolic concentrators were studied considering two constraints. The first one is the bundle-plane simplification, that states that any concentration system can be studied as though the solar cone is contained within a plane, and therefore it is only necessary to study a symmetry plane's section of a solar system to calculate its concentration ratio. The second one was the requirement that the receiver dimensions should be big enough to not miss any ray reflected on the concentrator, and therefore the optical efficiency of a ideal system is always 1 for cylindrical receivers and the geometrical concentration ratio minus 1 for planar receivers due to the absorber shadowing a certain area of the reflector.

Figure 3.27: Ray tracing simulations of a parabolic trough with a low $\psi_d$ and at different $\psi$ a) Cylindrical receiver $\psi = \psi_d$ b) Planar receiver $\psi = \psi_d$ c) Cylindrical receiver $\psi > \psi_d$ d) Planar receiver $\psi > \psi_d$ e) Cylindrical receiver $\psi > \psi_d$ f) Planar receiver $\psi > \psi_d$. Note that the thickness of the planar absorbers is not real but a representation of a planar absorber to make it visible in the plot. The solar cone size is also exaggerated in the figure.
Enhancing concentration ratio of solar concentrators

Considering those restrictions, the limits of concentration of parabolic concentrators were found in literature to fall below the limit by a factor comprised between two and four the limits of 2D and 3D concentration systems. The exact fall depends on the type of absorber considered and the optimal rim angle, $\psi_{t}$, of parabolic concentrators were found at $\pi$ for non-planar receivers and close to $\pi/2$ for planar receivers.

In this work, it was considered that some rays could be missed in order to increase concentration as much as possible. It was assumed that the absorber of the concentrator is designed for rim angles, $\psi_{t}$, between 0 and the maximum achievable by a parabolic surface ($\pi/2$ for planar receivers and $\pi$ for non-planar receivers). The optical efficiency of the concentrator would be the same as those found in literature for those $\psi \leq \psi_{t}$ and it will tend to zero at the limit of the parabolic surface ($\pi/2$ for planar receivers and $\pi$ for non-planar receivers). Considering the bundle-plane simplification, the limits of concentration were analytically found for 3D and 2D receivers with planar and non-planar receivers. It was shown how it is possible to reach the thermodynamics limit of concentration with a parabolic dish with a spherical receiver. In the case of the 3D concentrator with a planar receiver, a new limit was found to be much greater than the one defined in literature previously, but still only approximately 66% of the thermodynamics limit. For the 2D concentrators, the limit matched the limit derived from the 3D case, that in that case was double the limit found in literature for parabolic troughs.

Also, the concentration limits of collectors without considering a cross-section simplification were studied. The limits of 2D concentrators extracted from the thermodynamic 3D limits found in literature shown to be lower than the ones found in this work. As 2D receivers are not axisymmetric, the bundle-plane simplification is not accurate if rays are to be missed. The new limits of concentration for parabolic troughs are found to be approximately 20% greater than the limit defined in literature assuming the bundle-plane simplification.

### Table 3.1: Overview of limits of concentration

<table>
<thead>
<tr>
<th>Concentrator/Receiver</th>
<th>Thermodynamic limit</th>
<th>$C_{\text{max}}$, no missed rays</th>
<th>$C_{\text{max}}$, bundle simplification</th>
<th>$C_{\text{max}}$, no simplification</th>
</tr>
</thead>
<tbody>
<tr>
<td>3D/ Spherical</td>
<td>46396</td>
<td>11599</td>
<td>46396</td>
<td>46396</td>
</tr>
<tr>
<td>3D/ Planar</td>
<td>46396</td>
<td>11490</td>
<td>30787</td>
<td>45200</td>
</tr>
<tr>
<td>2D/ Cylindrical</td>
<td>215.4</td>
<td>69</td>
<td>215.4</td>
<td>254</td>
</tr>
<tr>
<td>2D/ Planar</td>
<td>215.4</td>
<td>107</td>
<td>214</td>
<td>240</td>
</tr>
</tbody>
</table>

The limits of concentration for a 3D concentrator with a planar receiver was finally found to be close to the thermodynamic limit of concentration, although some simplifications were assumed in order to obtain a simple concentration equation.

The development shown in this chapter shows how much higher concentration ratios can be achieved by dropping the optical efficiency of the collectors. Which configuration is best is not an obvious answer and it will depend on external factors such as the output temperature to obtain. In future stages, an optimisation of the trade-off between optical efficiency and concentration ratio should be addressed in detail. Figure 3.28 represents the product of relative concentration ratio and optical efficiency for a parabolic trough with different rim angles. This product allows to find the rim angle which allows to maximise the combined ratio for both concentration and efficiency. Bigger rim angles will drop efficiency faster than the gain of concentration. In future work, these configurations will be studied in detail.

Validation simulations in a ray tracing software were conducted to validate the equations developed and they have shown an acceptable match between simulation and calculation. Only in the case of a 3D concentrator with a planar receiver and large $\psi_{t}$ a mismatch was observed, due to the simplifications
Chapter 3: Limits of concentration on parabolic concentrators

The concentration ratio of 3D systems with planar receivers is overestimated in this work. Regardless, the discrepancy between analytical and ray tracing is 1.2 % for $\psi_d = 0$, that corresponds to the new limit of concentration and due to the simplicity of the equation obtained here, it was considered to be an acceptable approximation to the real limit.

Three major conclusions can be extracted from this chapter:

- There is an enormous potential in improving the concentration of commercial collectors by slightly compromising their optical efficiency. For ideal collectors, increases of 50 % the limit of concentrations for those collectors not missing any rays can be achieved with optical efficiencies of more than 50 % in all cases, reaching 80 % of optical efficiency for a parabolic trough with a planar receiver. However, the trade-off between the increase of concentration and the decrease of efficiency should be studied in detail in the future to optimise the design and total rim angle of concentrators.

- The bundle-plane simplification is not valid if rays are considered to be missed, except in the case of a parabolic dish with a spherical receiver.

- The limits of concentration for 2D concentrators are underestimated in literature.

Figure 3.28: trade-off between optical efficiency and relative concentration ratio for a parabolic trough with cylindrical absorber.
Chapter 4: Improving the concentration ratio with a secondary reflector

Increasing the concentration ratio of parabolic troughs is one of the challenges to make this technology economically competitive against fossil fuels. Parabolic troughs with large concentration ratios face several problems such as difficulty capturing all the solar direct radiation, and structural issues associated with thermal expansions and wind resistance amongst others. For larger mirrors it may be necessary to use a bigger absorber in order to capture all the radiation, thus increasing the thermal losses. A second stage reflector helps to increase the concentration ratio without increasing the primary mirror size. In this work, a theoretical analysis of a parabolic trough with a secondary flat reflector (SFR) is developed and ray tracing is conducted in order to validate the equations obtained. A flat reflector will have a minimal economic impact in the cost of a parabolic trough and it allows larger concentration ratios for identical primary mirror areas compared to a standard parabolic trough. Increases of concentration ratio up to 80% are observed when a secondary flat reflector is inserted in a commercial system, while the shadow area introduced in the primary mirror is usually less than 15% of the primary mirror area. The increase in pumping power is offset by the increase in system efficiency. An estimation of the pump requirements for two commercial absorbers and their variations with a SFR are addressed in section 5.1.

4.1 Introduction

As previously studied secondary reflectors where not practical for the reasons shown in the section 2.5 of the literature review, the main part of this thesis is devoted to studying an easy to manufacture flat secondary reflector to increase the geometrical concentration ratio of several commercial PTCs considering the range of deviations and misalignments that the systems can handle. The flat second stage mirror could be a practical and cost-effective solution to increase the concentration ratio of the collectors. The cost of installation of PTCs can be decreased if the concentration ratio is increased; an increase of concentration would lead to a reduction of thermal losses and a reduction of the primary mirror area to achieve the output temperature required. However, the ground usage and structures, tracking systems and primary mirrors (per unit of primary mirror) remain the same. This method can be used for new designs as well. However, the objective of this work is to analyse the theoretical gain of using a secondary reflector. Further work will include a full cost analysis.

As an additional advantage, the secondary flat reflector helps to distribute the energy flux around the tube more evenly than with a classic PTC where all the concentrated flux falls on one side of the receiver tube. This improved flux distribution could help to minimise the deflections and the stress of the absorber tube due to different thermal expansions [61] and could also stabilise internal two phase flow and pressure drop. Also, a reduction of the absorber dimensions and the local maximum temperature will reduce the thermal losses. However, a detailed analysis of the temperature distribution and the thermal losses around standard absorbers and SFR absorbers is conducted in next section and the temperature profiles around two commercial absorbers are shown in section 5.6.
Unlike other studies using secondary reflectors in linear absorbers [68, 118], in this work the absorber is not located on the parabola focus because the secondary reflector is placed on it. As a consequence, the absorber has to be located at a point near the focus, but not on it. The new location of the absorber is chosen to ensure there is no thermal short-circuit between it and the second-stage reflector, while not producing gap losses due to the distance between these two components. In figure 4.1 a schematic of a PTC with a secondary flat reflector with gap losses is shown.

In this work, the theoretical equations for designing a flat secondary mirror in order to reduce the absorber size for a given parabolic mirror and acceptance angle, $\theta$, are developed. Following this, ray tracing and CFD simulations as well as experiments are developed to check the potential improvement of a parabolic trough collector performance by using this secondary mirror. Maintaining the primary mirror aperture makes this solution suitable for currently designed or built plants, since only the receiver has to be altered and the tracking system and the location of the collectors can remain the same. However, replacing the receiver in already constructed plants has to be justified with a complete economic study, including the cost of replacing the absorbers, possible structure modifications to fix the new receivers, change of pumping power, etc. This is out of the scope of this work and it will be addressed in future stages. As the pumping requirements can be a limiting factor on the installation of the new absorbers, some estimations of its effects on the performance of the absorbers are conducted in section 5.1. This method is also a promising solution for rooftop applications, where the collectors are relatively small and the ground usage is critical due to limited roof top spaces. A higher concentration could make easier for these small collectors to achieve higher output temperatures to make them suitable for solar cooling or industrial heat applications.

The secondary reflector dimensions and the new geometrical concentration ratio are analytically found for the commercial collectors shown in Table 2.2, and ray tracing was conducted in order to check the validity of the new configuration. Emphasis was placed in looking for possible gap losses and observing the new flux distribution around the receiver tube. The intercept factors of the original collectors are maintained in the secondary mirror versions. It was assumed that the properties of the mirrors, the misalignments, the tracking accuracy, etc. remain the same.

This analysis shows how a smaller absorber can receive all the solar flux from a given mirror. However, the problem can be addressed in the opposite way. The size of the absorber can be chosen first and then a larger primary mirror can be designed for that particular absorber.
The edge ray principle [58] is used in the theoretical calculations. As the absorber is not placed on the focus, the necessary rays to study are explained in detail, since they are different from classic scenarios. The intercept factor is kept the same when the secondary mirror is inserted, so the maximum possible deviation is applied to the edge ray. The same assumptions are used in the ray tracing simulations. For the flux distribution, the solar disc and its effects are included. How rays are treated is discussed in section 3.3.

Using the theoretical equations, the absorber dimensions for any aperture area when the focal distance and the acceptance angle are given is calculated and the concentration ratio for the standard and the secondary reflector PTC is shown taking four representative focal distances. Finally, the energy absorbed by the proposed collector is compared with that of the original one in the same four scenarios.

4.2 Theoretical development

4.2.1 Parabola definitions

Optically, a PTC is a two dimensional parabola extruded along an axis that concentrates the sun into its extruded focus when its aperture area is perpendicular to the sun. As a PTC is symmetric along the long side of the mirror, it was demonstrated in the previous chapter how, if it is considered that the absorber’s dimensions are great enough to the collector being able to capture all the incoming radiation, a 2D analysis can be conducted to analyse its optics using a cross section of the system and the rays contained in it [131]. In this section, it is considered that the absorber will capture all the radiation incoming from the reflector and therefore a 2D analysis is considered to be accurate enough.

The reason for avoiding a 3D study of the PTC is that, in a real system, depending on the orientation of the PTC rows, an end of collector loss will appear due to the incident angle. This loss will remain the same when a secondary flat reflector is added, so it is not relevant for this work. Nevertheless, the reader can find about these losses in [134].

In the previous chapter, the concept of a cross section of a classic parabolic trough and the definition of rim angle, \( \psi \), was introduced in Figure 3.2. A parabola can be also defined by its focal distance, \( F \), and (if it is symmetrical) its half aperture width, \( X \), as defined in equation 4.1. The definition of the parameters of the parabola necessary to analyse the flat secondary reflector later in this section is shown in Figure 4.2. With these two parameters, the rim angle is also defined as in equation 4.2. Also, if the distance to the focal axis of any point of the parabola, \( x_i \), is known, its rim angle, \( \psi_i \), can be obtained.

\[
Y = F - \frac{X^2}{4F} \quad (\text{Equation 4.1})
\]

\[
\psi = \arctan \frac{X}{Y} = \arctan \frac{X}{F - \frac{X^2}{4F}} \quad (\text{Equation 4.2})
\]

\[
y_i = F - \frac{x_i^2}{4F} \quad (\text{Equation 4.3})
\]

\[
\psi_i = \arctan \frac{x_i}{y_i} = \arctan \frac{x_i}{F - \frac{x_i^2}{4F}} \quad (\text{Equation 4.4})
\]

The maximum deviation from the normal to the half aperture that can be allowed in the collector while still receiving on the absorber all the energy reflected from the primary mirror is defined by the acceptance angle, \( \theta \). In this section the acceptance angle is considered greater than the solar cone as the absorbers of real collectors are considered. The higher the acceptance angle, the easier it is to capture all the energy, but since the absorber dimensions increase, the concentration ratio decreases. Typical acceptance angles for current
commercial parabolic troughs are lower than 1° and they were summarized in Table 2.2. If the radius of the absorber is defined, the acceptance angle can be calculated from the parameters of the parabola as defined in equation 4.5:

\[ \theta = \arcsin \left( \frac{r}{\sqrt{X^2 + Y^2}} \right) = \arcsin \left( \frac{r}{\sqrt{X^2 + \left( F - \frac{x_i^2}{2F} \right)^2}} \right) \]  

(Equation 4.5)

4.2.2 The secondary flat reflector

The main aim of this chapter is to study the use of a secondary flat reflector in already designed collectors and analyse how it helps to increase their concentration ratios. A flat reflector is easy to manufacture and it will represent a lower additional cost to a PTC than the implementation of bigger primary mirrors.

The secondary reflector is designed in a way that the acceptance angle of the original commercial collector is maintained. That allows maintaining the same tracking system and primary mirrors in already installed collectors. It is considered that the introduction of a flat secondary reflector in the proximities of the absorber will have a negligible effect in the increase of the acceptance angle. In new collectors, if the desired rim angle, focal distance and acceptance angle are defined, the secondary reflector and the absorber dimensions can also be calculated. As the secondary reflector is a flat surface perpendicular to the symmetry axis of the parabola, a shadow is projected from it on the primary mirror, which has to be considered when the concentration ratio is calculated. If the new absorber diameter is small enough, the concentration ratio is increased despite the shadow projected from the secondary reflector on the parabolic mirror. Adding this shadow to equation 2.3 the concentration ratio of a PTC partially shaded by its own secondary reflector can be found as:

\[ C = \frac{(W - W_s) L}{\pi d' L} = \frac{W - W_s}{\pi d'} \]  

(Equation 4.6)

Equation 2.2 defined the concentration as a function of \( \psi \) and in the previous chapter it was explained how the maximum concentration ratio if no rays are to be missed is found for \( \psi = \pi/2 \). However, there are no \( \psi \) limitations for a classic parabolic trough with cylindrical absorber. In equations 2.2 and 2.3 the shadow of the tube was not considered, as the upper part of the absorber will intercept that radiation. If using a secondary flat mirror is desired, there appears a limit for the dimensions of the primary mirror. If \( \psi > \pi/2 \), the rays proceeding from those parts of the primary mirror with \( \psi > \pi/2 \) will impact the back of the secondary mirror, and then they will be missed. Figure 4.1 represents that phenomenon in detail. There have been some prototypes of PTCs with \( \psi > \pi/2 \) as it was shown in Table 2.2 but they are unusual nowadays. In this work some collectors with \( \psi > \pi/2 \) are included. When the secondary reflector is calculated for one of these PTCs, its aperture area is decreased in order to obtain a \( \psi = 85^\circ \) to avoid any ray impacting the back of the secondary reflector. However, the change of concentration ratio if a secondary flat reflector is inserted.
was compared with the concentration ratio obtained for the original reflector without secondary optics.

4.2.3 Application of the edge ray principle

The edge ray principle [58] is used in order to define the secondary flat reflector dimensions. The secondary mirror is then treated as a receiver big enough to receive all the rays proceeding from the primary mirror. As the parabola is symmetrical, it is enough if only one of its two edges is studied.

If a perfect collimated beam impacts an ideal parabolic reflective surface, the beam will be reflected onto the focus of the parabola. Any misalignment of a component (tracking error, vertical misalignments of the absorber, etc.) can be treated as an angular deviation of the beam, and in its reflection it will be deviated from the focus. The edge rays will represent the worst case scenario. By giving a deviation of $\pm \theta$ to an edge ray, the deviation of this ray will become the maximum in the whole parabola. Thus, if this ray is intercepted by the absorber, all the inner rays will be intercepted as well.

This method can determine whether the absorber is capable of intercepting all the rays, but cannot be used to estimate the flux distribution as the sun shape is a cone with an angle of revolution of $\alpha = 4.5$ mrad and the solar beam will not be perfectly collimated. The reflected beam will have an even wider spread due to the scattering of the reflective surface, $\sigma$. Thus, the absorber will receive a cone of radiation bigger than the sun shape. When the flux distribution was analysed, this effect was included.

From the primary mirror edge, two rays have to be studied, one with a positive $\theta$ deviation and another ray with a negative one. Each ray defines a different secondary reflector width and the bigger projection has to be chosen to ensure that no rays are missed. In figure 4.3, a representation of a general case is shown. The widths of the secondary mirror can be calculated as in equations 4.7 and 4.8:

\[
W_{s1} = \frac{r}{\cos(\psi - \theta)} = \frac{\sin \theta \sqrt{X^2 + \left(\frac{F - X^2}{4F}\right)^2}}{\cos(\psi - \theta)} \quad \text{(Equation 4.7)}
\]

\[
W_{s2} = \frac{r}{\cos(\psi + \theta)} = \frac{\sin \theta \sqrt{X^2 + \left(\frac{F - X^2}{4F}\right)^2}}{\cos(\psi + \theta)} \quad \text{(Equation 4.8)}
\]

As $\psi < \pi/2$, $\cos(\psi - \theta) > \cos(\psi + \theta)$ and then $W_{s2} > W_{s1}$. Nonetheless, the appropriate dimension for the
Chapter 4: Improving the concentration ratio with a secondary reflector

secondary reflector is, actually, \( W_{s1} \); the new absorber has to be placed somewhere in the symmetry axis, so the rays which should impact the secondary mirror at \( W_{s2} \) must be intercepted by the absorber before reaching the secondary mirror. If the absorber does not intercept them, the secondary reflector would reflect them to the opposite part of the primary mirror and finally they will be reflected back to the sky, never reaching the absorber. The width of the secondary mirror is chosen then as \( W_s = 2W_{s1} \).

Once the secondary reflector dimensions are defined, the new absorber dimensions and location can be calculated. The original parabola is now 'divided' by the shadow of the secondary mirror into two non-symmetrical parabolas, but the two parabolas are symmetric along their longitudinal axis. The centre of the new absorber has to be placed on this symmetry axis, in order to maintain the symmetry of the system. As long as the system is kept symmetric, only one of the two non-symmetrical parabolas has to be studied.

As the receiver should now capture radiation both from primary and secondary mirror, there is not a uniform incoming flux to apply the edge-ray principle any more. Nevertheless, both mirrors have a well-known shape and they can be studied applying the edge-ray method for each one separately. If all the edge rays are analysed when they go through the symmetry axis and the two with a bigger distance between them are picked, the resulting absorber should be able to capture all the incoming radiation. If only the deviated rays are studied, the calculations could lead to a solution where, apparently, all the radiation impacts the absorber. However, a gap loss could occur for a ray reflected from the edge of the primary mirror with no deviation from the ideal. To avoid this loss, a ray with no deviation proceeding from the edge of the primary mirror, ray 1, is included in the study. In Figure 4.4, the five necessary rays to study are defined. Ray 1 is a ray with no deviation reflected in the edge of the parabolic mirror that will impact the centre of the secondary mirror (placed in the focus of the parabola). Rays 2 and 4 are also reflected in the edge of the parabola, but with maximum deviations \((\pm \theta)\). Rays 3 and 5 have also maximum possible deviations, but their origin is the edge created by the projected shadow on the primary mirror. The new absorber is then, the smallest possible absorber than can absorb these 5 rays or their reflections on the secondary reflector.

\[ d_{m,f} \] is defined as the distance of a ray to the focus of the parabola when it reaches the axis of symmetry where \( m \) is the ray number. By definition, as ray 1 is a ray with no deviation, \( d_{1,f} = 0 \) as the ray will impact the geometrical focus. The absorber will be defined then by ray 1 and the most distant from the focus of the other four rays.

Analysing the distance of rays 2r and 4 and the distance of 3r and 5 to the focus, it can be demonstrated that, for \( \psi < \pi/2 \), rays 2r and 3r always have longer distances to the focus than rays 4 and 5 respectively. In

\[ \text{Figure 4.4: Sketch of the five rays necessary in the study to determine the absorber’s size and position.} \]
this work, the fact that the $d_{2r,f}$ is bigger than $d_{4,f}$ is demonstrated. In the same way, $d_{3r,f}$ and $d_{5,f}$ can be analysed if $\psi$ is substituted by $\psi_3$. With this demonstration, it is proven that there is no need for studying rays 4 and 5 when the new absorber size and position are calculated. So, only studying rays 1, 2 and 3 will allow calculating the new absorber.

As ray 2 is a ray with $\psi_2 = \psi - \theta$ that comes from the edge of the parabola, it will impact the secondary mirror near its edge before reaching the symmetry axis. As it impacts a flat surface, ray 2 is reflected with an angle equal to $\psi$. The vertical distance to the focus of the parabola is 0 in the moment of the reflection. Then, the vertical distance of the reflection of ray 2 to the focus when it crosses the symmetry axis is equal to the distance that it would have travelled through the focus without a secondary reflector as shown in figure 4.5. Therefore:

$$d_{2r,f} = \frac{X}{\tan(\psi - \theta)} - \frac{X}{\tan \psi}$$  \hspace{1cm} (Equation 4.9)

In a similar way, ray 4 is a ray with $\psi_4 = \psi + \theta$ that comes from the edge of the parabola. As ray 4 passes through the axis of symmetry before it impacts the secondary reflector, its vertical distance to the focus when it goes through the symmetry axis can be defined as:

$$d_{4,f} = \frac{X}{\tan \psi} - \frac{X}{\tan(\psi + \theta)}$$  \hspace{1cm} (Equation 4.10)

If the distance of the reflection of ray 2 to the focus is always bigger than the distance of ray 4 to the focus when they go through the focal line, any absorber that intercepts ray 2 will intercept ray 4 as well. The distance between rays 2 and 4 in the focal line can be defined as:

$$d_{2,f} - d_{4,f} = \frac{X}{\tan(\psi - \theta)} - \frac{X}{\tan \psi} - \frac{X}{\tan \psi} + \frac{X}{\tan(\psi + \theta)}$$  \hspace{1cm} (Equation 4.11)

And since $X$ is constant for a given parabolic mirror and it is known that $k/\tan 90 = 0$ and that $\tan(90 + \beta) = -\tan(90 - \beta)$. Then, for a $\psi = \pi/2$ (the maximum rim angle achievable for the flat reflector configuration) the distance between the two rays becomes:

$$d_{2,f} - d_{4,f} = \frac{X}{\tan(\psi - \theta)} - \frac{X}{\tan(\psi + \theta)} = 0$$  \hspace{1cm} (Equation 4.12)

So the distance of rays 2 and 4 to the focus is the same for $\psi = \pi/2$. Now, if the first and second derivatives of $k/\tan \psi$ are calculated:

$$\frac{d(k)}{\tan(\psi)d\psi} = \frac{-k(1 + \tan^2 \psi)}{\tan^2 \psi}$$  \hspace{1cm} (Equation 4.13)
Chapter 4: Improving the concentration ratio with a secondary reflector

\[
\frac{d}{\tan^2(\psi)} \left[ k \left( 1 + \tan^2 \psi \right) \right] = 2k \cot \psi \csc^2 \psi = \frac{8k \cos \psi}{3 \sin \psi - \sin(3\psi)} \quad \text{(Equation 4.14)}
\]

And from the analysis of the two derivatives it can be concluded that, in the range of 0-90°, \( k / \tan \psi \) is decreasing and concave without any maximums, minimums or inflexion points in the range studied. As \( \theta \) is small compared to \( \psi \), and, as the curve is concave, \( \frac{X}{\tan(\psi - \theta)} - \frac{X}{\tan(\psi)} > \frac{X}{\tan(\psi + \theta)} \). So, \( d_{2r, t} \) is always bigger than \( d_{4, t} \) in the range considered. With \( d_{2r, t} \) defined, the position and the size of the smallest absorber that is able to collect all the rays in the cone defined by rays 2 and 4 is defined by a circumference tangent to both ray 1 and ray 2r, as shown in figure 4.6

![Figure 4.6: Minimum dimension for an absorber capable of capturing rays 1 and 2 b) Minimum dimension for an absorber capable of capturing rays 1 and 3.](image)

From the figure above, the following equations to find the radius and the position of the absorber can be defined:

\[
a_{1,2} + b_{1,2} = d_{1,2} = \frac{W_{a1}}{\tan(\psi - \theta)} \quad \text{(Equation 4.15)}
\]

\[
r_{1,2} = a_{1,2} \sin \psi \quad \text{(Equation 4.16)}
\]

\[
r_{1,2} = b_{1,2} \sin(\psi - \theta) \quad \text{(Equation 4.17)}
\]

Substituting in equations 4.6-4.5, the position of the centre of the absorber \( a_{1,2} \) capable of intercepting all the rays within a cone formed by rays 2 and 4 and its radius can be found as:

\[
a_{1,2} = \frac{d_{1,2} \sin(\psi - \theta)}{\sin \psi + \sin(\psi - \theta)} = \frac{W_{a1} \sin(\psi - \theta)}{\tan(\psi - \theta) (\sin \psi + \sin(\psi - \theta))} = \frac{\sqrt{X^2 + (F - \frac{X^2}{4F})^2} \sin \theta}{\sin \psi + \sin(\psi - \theta)} \quad \text{(Equation 4.18)}
\]

\[
r_{1,2} = \frac{\sqrt{X^2 + (F - \frac{X^2}{4F})^2} \sin \theta \sin \psi}{\sin \psi + \sin(\psi - \theta)} \quad \text{(Equation 4.19)}
\]

Ray 3 is a ray emanating from the inner edge of the parabola, this is, the point where the shadow projected over the secondary mirror ends, and it has the maximum deviation allowed \( (-\theta) \). This ray will be reflected back with an angle, \( \psi_3 \), and the point where it impacts the secondary reflector \( W_{a3} \) can be defined using the general properties of the parabola:
\[
\psi_3 = \arctan \left( \frac{W_{s1}}{F - \frac{W_{s1}^2}{4F}} \right) = \arctan \left( \frac{\frac{\sin \theta \sqrt{x^2 + \left(\frac{F - x^2}{2F}\right)^2}}{\cos (\psi - \theta)}}{F - \frac{\sin \theta \sqrt{x^2 + \left(\frac{F - x^2}{2F}\right)^2}}{4F \cos (\psi - \theta)}} \right)
\]  
(Equation 4.20)

Now the position and dimensions of an absorber capable of capturing all the rays in the cone defined by rays 3 and 5 and capturing ray 1 at the same time, is the same as for an absorber placed in the axis of symmetry and tangent to ray 1 and the reflection of ray 3 in the secondary reflector. Proceeding in the same way as in the previous case with rays 1 and 2:

\[
d_{1,3} = \frac{W_{s1}}{\tan(\psi_3 - \theta)} - \frac{W_{s1}}{\tan \psi_3}
\]  
(Equation 4.21)

\[
a_{1,3} = \frac{d_{1,3} \sin(\psi_3 - \theta)}{\sin \psi + \sin(\psi_3 - \theta)}
\]  
(Equation 4.22)

\[
r_{1,3} = a_{1,3} \sin \psi
\]  
(Equation 4.23)

The largest radius between \(r_{1,2}\) and \(r_{1,3}\) has to be chosen to obtain an absorber capable of obtaining all the radiation proceeding both from the primary and secondary mirrors. Which one of them is bigger is not a simple question as it changes depending on the cases studied. In Table 4.1, the dimensions of both radii are shown for different commercial parabolic troughs. It can be seen how sometimes the larger radius corresponds to \(r_{1,2}\) and other times to \(r_{1,3}\) as it depends on the focal distance, the aperture of the mirror and \(\theta\). It can also be seen how much the absorber is reduced from the case without a secondary reflector.

**Table 4.1: Calculated \(r_{1,2}\) and \(r_{1,3}\) for different PTCs**

<table>
<thead>
<tr>
<th>Collector</th>
<th>Aperture (mm)</th>
<th>Focal distance (mm)</th>
<th>Original absorber radius (mm)</th>
<th>(r_{1,2}) (mm)</th>
<th>(r_{1,3}) (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEP</td>
<td>1208</td>
<td>647.5</td>
<td>14</td>
<td>7.07</td>
<td>11.3</td>
</tr>
<tr>
<td>NEP2</td>
<td>1844</td>
<td>647.5</td>
<td>14</td>
<td>7.08</td>
<td>8.94</td>
</tr>
<tr>
<td>LS1</td>
<td>2500</td>
<td>680</td>
<td>21</td>
<td>10.51</td>
<td>9.87</td>
</tr>
<tr>
<td>LS2</td>
<td>5000</td>
<td>1400</td>
<td>35</td>
<td>17.57</td>
<td>17.25</td>
</tr>
<tr>
<td>LS3</td>
<td>5760</td>
<td>1710</td>
<td>35</td>
<td>17.61</td>
<td>18.78</td>
</tr>
<tr>
<td>Helian 3/32</td>
<td>1810</td>
<td>640</td>
<td>17</td>
<td>8.48</td>
<td>10.68</td>
</tr>
<tr>
<td>PT1</td>
<td>2300</td>
<td>800</td>
<td>25.5</td>
<td>12.86</td>
<td>15.8</td>
</tr>
<tr>
<td>Solitem</td>
<td>1800</td>
<td>780</td>
<td>19</td>
<td>9.58</td>
<td>13.85</td>
</tr>
<tr>
<td>Acurex 3001*</td>
<td>1830</td>
<td>457</td>
<td>15.9</td>
<td>7.34</td>
<td>6.91</td>
</tr>
<tr>
<td>Acurex 3011*</td>
<td>2130</td>
<td>533</td>
<td>15.9</td>
<td>7.27</td>
<td>6.86</td>
</tr>
<tr>
<td>Sener</td>
<td>6868</td>
<td>2000</td>
<td>40</td>
<td>20.02</td>
<td>21.04</td>
</tr>
<tr>
<td>Solar Kinetics T-700</td>
<td>2130</td>
<td>559</td>
<td>15.9</td>
<td>7.91</td>
<td>7.42</td>
</tr>
<tr>
<td>Solar Kinetics T-800*</td>
<td>2360</td>
<td>483</td>
<td>15.9</td>
<td>7.69</td>
<td>7.23</td>
</tr>
<tr>
<td>Suntec Systems IV</td>
<td>3050</td>
<td>838</td>
<td>19</td>
<td>9.63</td>
<td>9.23</td>
</tr>
<tr>
<td>Solel IND300*</td>
<td>990</td>
<td>272</td>
<td>11</td>
<td>4.28</td>
<td>4.03</td>
</tr>
</tbody>
</table>

*Collectors with \(\psi \geq \pi/2\) receive special consideration in this thesis as the secondary solar collector is not suitable for them. Their \(\psi\) are reduced to 85° to make it suitable to have a secondary reflector installed and \(\theta\) is maintained for this \(\psi\). Then, the secondary reflector is calculated and the new concentration ratios obtained. Nevertheless, they are compared with the original collector where no secondary reflector is installed.
4.3 Ray tracing

Ray tracing with Zemax software was conducted in order to prove the validity of the equations obtained in this work. An interface where the user can choose the aperture of the primary mirror, its focal distance and the desired was designed. As the purpose of these simulations is to prove that the new radius chosen for the absorber is able to capture all the energy proceeding from both the primary and the secondary mirror, the receiver is considered only as an absorber tube. In future work, glass should be added in order to simulate a more realistic PTC.

Three series of simulations were made, changing the shape of the source. In the first one, the radiation source is considered as a radial, with an angular distribution of 4.5 mrad, corresponding to the solar disc. Both the classic PTC and the PTC with the secondary reflector are simulated for a range of misalignment angles between $\theta - 0.266^\circ$ and $0^\circ$. We then calculate the smallest absorber radius for both collectors that can capture all the radiation and we determine the missed energy for both systems and the shadow projected by the flat secondary mirror. The concentration ratio of both collectors is then calculated using equations 2.3 and 4.4. In the second series, the radiation source is modified to simulate the effect of the scattering from the mirrors, with $\sigma = 0.23^\circ$ [121] in order to recreate a more realistic flux distribution around the absorber. The third series was used to investigate gap losses in detail, as if the absorber is not perfectly placed, the edge rays could impact the centre of the secondary mirror and then miss the absorber. In this one the source was simulated as ideal (no angular spread) to increase the possibilities of gap losses. Half mirror was simulated and a detector was located after the gap to detect rays being reflected in the secondary mirror and not impacting the absorber. However, with a correct alignment of the absorber no gap losses were observed.

To validate the analytical equations, the dimensions obtained with them for the SFR and the new absorber and its new position were inserted in the ray tracing software. Several series of simulations were conducted, checking in every case that the only energy missing the absorber was the one received by the back of the secondary reflector.

In this thesis, the commercial PTCs shown in Table 2.2 were examined by maintaining their primary mirror dimensions and extracting the exact acceptance angle from equation 4.3 from the absorber radius. For those collectors with $\psi > \pi/2$ the aperture area was reduced in order to obtain $\psi = 85^\circ$. Nevertheless, the new concentration ratio is compared with the concentration ratio that these PTCs had for the original rim angle.
The data from the ray tracing simulations was processed in Matlab to obtain the flux distribution around the receiver. A validation simulation was conducted and the results were compared with those obtained by Jeter [76]. The comparison between the two methods is shown in figure 4.7. The agreement is quite clear, however, there is a mismatch in the maximum local concentration ratio around 60°. Discrete intervals of 0.25° and 0.5° were adopted in Matlab to recreate the flux distribution around the absorber from the ray tracing data and the results obtained in both cases were identical. Smoothing functions were applied to remove the noise caused by the discretisation. The small difference shown in the peak between Jeter’s result could be caused as an effect of these smoothing filters conducted in Matlab to recreate the curve from the ray tracing results. However, the total concentration found by integration, shows that the two results are very similar (19.82, compared to 20). Finally a ray independence study was also conducted. Simulations with less than 150 k rays were not repeatable and thus results depended on the number of rays. From simulations with more than 200 k rays, the results did not change with the introduction of more rays. It was decided to conduct the simulations with 250 k rays, as a solution of compromise between accuracy and speed in the simulations.

4.4 Results

For the collectors in Table 2.2, the original concentration ratio, the secondary reflector dimensions, the new absorber dimensions, the new concentration ratio, the increase of concentration ratio and the percentage of primary mirror area shadowed by the secondary mirror are shown in table 4.2.
Chapter 4: Improving the concentration ratio with a secondary reflector

The increase of concentration and the percentage of parabolic mirror shadowed changes for different collector dimensions. Nevertheless, the concentration ratio is always increased with the inclusion of the secondary reflector when already existing PTCs are analysed. Even for those PTC with a $\psi > \pi/2$ the concentration is increased. However, the aperture area of those collectors was modified to obtain a $\psi = 85^\circ$ and not only the shadow, but an additional aperture area reduction has to be considered.

For already built plants it is desirable to keep the primary mirrors and ground layout so the $\psi$ remains unchanged. However, if a new installation is designed, maximum concentration ratio is generally desirable. In Figure 4.8, the aperture areas of the NEP and LS collectors are varied such that their $\psi$ angles change from 0° to 90° and the concentration with and without secondary reflector are shown. The increase of concentration for a given aperture area and the percentage of primary mirror is shown as well. Focal distance and acceptance angle are kept as the original collectors, and the radius of the absorber is then calculated to achieve the desired acceptance angle. The secondary flat reflector increases the concentration ratio of the PTCs with a $\psi$ ranged roughly between 20° and 88° depending on the focal distance and acceptance angle of the original collector. This maximum increase of concentration ratio happens when $r_{1,2} = r_{1,3}$ and it depends then on the focal distance and acceptance angle. This maximum occurs for $81^\circ < \psi < 83^\circ$, for focal distances and acceptance angle in the range of commercial PTCs.

### Table 4.2: Concentration ratio of different PTCs with and without secondary reflector

<table>
<thead>
<tr>
<th>Collector</th>
<th>Original concentration ratio</th>
<th>Secondary reflector width (mm)</th>
<th>New absorber radius (mm)</th>
<th>New C</th>
<th>C increase (%)</th>
<th>Shadowed area (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEP</td>
<td>13.7</td>
<td>42.8</td>
<td>11.3</td>
<td>16.4</td>
<td>19.7%</td>
<td>3.5%</td>
</tr>
<tr>
<td>NEP2</td>
<td>21.0</td>
<td>82.9</td>
<td>8.9</td>
<td>31.3</td>
<td>50.9%</td>
<td>4.5%</td>
</tr>
<tr>
<td>LS1</td>
<td>19.0</td>
<td>416.9</td>
<td>10.5</td>
<td>31.5</td>
<td>66.5%</td>
<td>16.7%</td>
</tr>
<tr>
<td>LS2</td>
<td>22.7</td>
<td>554.5</td>
<td>17.6</td>
<td>40.2</td>
<td>77.6%</td>
<td>11.1%</td>
</tr>
<tr>
<td>LS3</td>
<td>26.2</td>
<td>386.7</td>
<td>18.7</td>
<td>45.2</td>
<td>72.5%</td>
<td>6.7%</td>
</tr>
<tr>
<td>Heliomax 3/32</td>
<td>16.9</td>
<td>96.7</td>
<td>10.7</td>
<td>25.5</td>
<td>50.0%</td>
<td>5.3%</td>
</tr>
<tr>
<td>PT1</td>
<td>14.4</td>
<td>151.3</td>
<td>15.8</td>
<td>21.6</td>
<td>51.5%</td>
<td>6.6%</td>
</tr>
<tr>
<td>Solitem</td>
<td>15.1</td>
<td>73.8</td>
<td>13.9</td>
<td>19.8</td>
<td>31.9%</td>
<td>4.1%</td>
</tr>
<tr>
<td>Acurex 3001*</td>
<td>18.3</td>
<td>282.4</td>
<td>7.3</td>
<td>30.2</td>
<td>66.2%</td>
<td>16.9%</td>
</tr>
<tr>
<td>Acurex 3011*</td>
<td>21.3</td>
<td>286.0</td>
<td>8.0</td>
<td>36.4</td>
<td>70.0%</td>
<td>14.6%</td>
</tr>
<tr>
<td>Sener</td>
<td>27.3</td>
<td>491.6</td>
<td>21.0</td>
<td>48.2</td>
<td>76.5%</td>
<td>7.2%</td>
</tr>
<tr>
<td>Solar Kinetics T-700</td>
<td>21.3</td>
<td>499.5</td>
<td>7.9</td>
<td>35.0</td>
<td>63.1%</td>
<td>23.5%</td>
</tr>
<tr>
<td>Solar Kinetics T-800*</td>
<td>18.2</td>
<td>297.1</td>
<td>7.7</td>
<td>30.5</td>
<td>70.0%</td>
<td>16.8%</td>
</tr>
<tr>
<td>Suntec Systems IV</td>
<td>25.5</td>
<td>361.0</td>
<td>9.6</td>
<td>44.4</td>
<td>76.2%</td>
<td>11.8%</td>
</tr>
<tr>
<td>Solel IND 300*</td>
<td>18.6</td>
<td>166.1</td>
<td>4.3</td>
<td>30.9</td>
<td>66.0%</td>
<td>16.6%</td>
</tr>
</tbody>
</table>

The increase of concentration and the percentage of parabolic mirror shadowed changes for different collector dimensions. Nevertheless, the concentration ratio is always increased with the inclusion of the secondary reflector when already existing PTCs are analysed. Even for those PTC with a $\psi > \pi/2$ the concentration is increased. However, the aperture area of those collectors was modified to obtain a $\psi = 85^\circ$ and not only the shadow, but an additional aperture area reduction has to be considered.
Although the concentration ratio of the collector is increased, the total amount of energy received in the absorber when the secondary mirror is installed is lower than in the original PTC. This decrease of received energy is due to the projected shadow and to the attenuation of the radiation that impacts the secondary mirror. The shadow is constant, as the secondary mirror is always perpendicular to the sun, but the amount of radiation impacting the secondary mirror changes for different misalignment angles. The amount of energy received by the absorber for the NEP and LS collectors with secondary mirror is shown in Figure 4.9 for different secondary mirror reflectivities from a perfect alignment until the maximum deviation allowed in each case. These different reflectivities try to simulate not only the reflectivity of the mirror itself, but also a possible degree of soiling that could happen during the operation of the PTC. 100% represents the amount of energy received by the original PTC.
Chapter 4: Improving the concentration ratio with a secondary reflector

Figure 4.9: Energy received in the absorber relative to that for no secondary mirror for different misalignments and secondary mirror reflectivities. a) NEP collector; b) LS1 collector; c) LS2 collector; d) LS3 collector.
Figure 4.10: Flux distribution around the absorber of the NEP and LS3 collectors with and without a secondary reflector for different mirror misalignments. a) NEP, perfect alignment. b) LS3, perfect alignment. c) NEP, half maximum misalignment. d) LS3, half maximum misalignment. e) NEP, maximum misalignment. f) LS3, maximum misalignment. θ° represents the bottom of the absorber.
In worst case scenarios, almost a 25% of energy is not reaching the absorber (LS1 collector). In this particular case, the increase of concentration was 66% and the absorber is half the dimensions than in the original collector. As stated by Rabl [7]: 'The heat losses from a collector are proportional to the absorber area (to a good approximation) and hence inversely proportional to the concentration'. So, although this 25% less energy reaching our absorber, a better thermal performance is expected due to a 40% decrease in the energy losses. Heating on the secondary mirror is not expected to be relevant. The primary mirror will absorb some radiation, typically between 5 and 10% of the solar spectrum and this is wavelength dependent. Thus spectrum than the secondary mirror will receive is contains wavelengths where it is almost 100% reflective. Ray tracing showed that, in worst case scenario, 40% of the incoming flux will impact the secondary reflector. For a clean mirror we estimate a reflectivity of, at least, 97% due to the changed spectrum. The absorbed flux will be around 1.2% of the incoming radiation and thus heating will not be significant. However, a further study to investigate the durability of the mirror should be addressed in the future. Effects such as delamination or reliability of the mirror due to high temperatures should be considered in the coming stages before manufacturing.

Figure 4.10 shows results from the ray tracing conducted for the NEP and LS3 collectors in two different scenarios, one consider that the beam is distributed in a cone formed only by the sun's shape and all the other possible misalignments considered as angular deviations and a second one considering that the beam is distributed in a bigger disc due to the scattering of the mirror. The maximum deviation considered in each case is the maximum angle allowed without missing any rays, that is, the acceptance angle minus the effects of solar cone and scattering. It is clear that the secondary reflector produces a more uniform flux distribution around the absorber, especially for low deviations. When a deviation is simulated, a higher local concentration peak is observed, especially in the case of LS3 collector. This is expected, as the concentration ratio in this mirror had been increased from 26.2 to 45.6. However, the flux is still distributed over a larger area than that without the secondary reflector thus helping to reduce the thermal stress on the tube. Moreover, when the scattering effect is considered, the peak of concentration is reduced and the flux distribution uniformity is increased compared with the standard collector.

4.5 Misalignments sensitivity.

In the calculations of the secondary flat reflector, it was assumed that all the deviations were a result of rotational misalignments and a scattering produced in the reflection of the primary mirror. The rotational misalignments made the reflected cone to be reflected to a point near the focus while the scattering increased the angular distribution of the solar cone.

In a real system, misalignments can appear in the parabolic troughs. A classification of the possible misalignments was developed by Guven [135], the different misalignments were classified on three different categories, materials, manufacture and operation.

The quality of the mirror can make scatter the rays reflected on it which will increase the reflected cone dimensions. The manufacturing and assembling process of the parabolic troughs can lead to several errors that will reduce the optical performance of the troughs, some of those errors can produce random reflections, as in profile errors in the parabola surface due to distortions during the manufacture of the primary mirror. Some other errors can displace in a linear direction the focus of the system, such as misalignments of the primary mirror or the absorber during the installation of the trough. A rotational error can also appear as an effect of a poor assembling of the tracking system or the primary mirror.

Last group of misalignments described by Guven were those ones produced during the trough operation. The main ones are the tracking errors, temperature and the wind loads, that can twist the primary mirror.
wind load effects were analysed in the Plataforma Solar de Almeria as shown in Figure 2.6 in Section 2.3. Some other effects, related with the aging of the troughs can increase the misalignments over time, the mirror can lose properties due to wind particles erosion, increasing its scattering, and the thermal stress on the absorber can produce a permanent expansion of the absorber tube, finally, load stress can decrease the parabolic profile quality over time.

In this section, a linear and rotational sensitivity analysis of the impact factor of two benchmark mirrors have been conducted for a standard absorber and a secondary flat reflector one. The two benchmark mirrors chosen are the NEP and the LS3. The NEP was chosen due to its small dimensions on both absorber and $\psi$, that will cause a higher sensitivity to misalignments. This higher sensitivity is due to the local $\theta$, as shown in Figure 3.9, as $\psi_d$ is lower, the inner points of the mirror will have a local $\theta$ similar to the design angle, while it will be higher than the design for those mirrors with a big $\psi_d$. This change of $\theta$ can be directly extracted from equation 3.18. The LS3 was chosen as it is the most common mirror in PTC power plants. As the LS3 mirror has a big $\theta$, its inner points will have a much bigger $\psi$ in comparison, making it less sensitive to misalignments.

Ray tracing simulations were conducted for those mirrors and the different absorbers considering a glass envelope with a diffractive index of 1.5 and no optical inaccuracies and a scattering on the primary mirror of 4 mrad [121]. The intercept factor was found as the fraction of the energy available in the primary mirror aperture considering that no absorptions happen in the primary or secondary mirror. The effects of the secondary mirror absorptivity were shown in figure 4.2.

The secondary mirror was considered to be encapsulated within the glass envelope for the NEP mirror. That makes the glass to be slightly bigger in the case of the NEP collector but it would be necessary an evacuated glass of 20 cm radius for the LS3 absorber, which is not considered feasible. However, a partial encapsulation of the LS3 SFR absorber was considered. A partial encapsulation would produce an additional optical loss corresponding to the width of the glass envelope plus the space considered between the sectors of the secondary mirror and the glass. This optical loss will depend on the misalignments considered in the system that would make the incident flux in the void area created by the glass. For the glass thickness considered (1.8 mm) and no misalignments, ray tracing simulations show little dependence of that gap and it was decided to be ignored to increase simplicity.

As shown in the previous chapter, an increase of concentration can be achieved by placing smaller absorbers and decreasing the optical efficiency of the systems. For that reason, it was included in the analysis of both primary mirror an absorber tube of the same dimensions that the SFR absorbers but without including the secondary reflector and its optical efficiency was calculated. Finally, in order to reproduce a feasible absorber for the LS3 mirror, a smaller SFR was analysed as well, with a width of 62 mm, that could be encapsulated on the current 125 mm glasses available on the market. This will change the shadow projected by the secondary flat reflector, that due to the big $\psi$ of the LS3 mirror was of 6.7 % and it will reduce it to 2.15 %, although the optical efficiency will drop due to some rays missing both the secondary mirror and the absorber. Table 4.3 shows the absorbers studied in this section and the concentration ratio of all of them.

<table>
<thead>
<tr>
<th>Case</th>
<th>Mirror</th>
<th>Absorber radius (mm)</th>
<th>SFR dimensions (mm)</th>
<th>Shadow (%)</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>NEP</td>
<td>14</td>
<td>-</td>
<td>0</td>
<td>13.73</td>
</tr>
<tr>
<td>2</td>
<td>NEP</td>
<td>11.3</td>
<td>-</td>
<td>0</td>
<td>17.01</td>
</tr>
<tr>
<td>3</td>
<td>NEP</td>
<td>11.3</td>
<td>42.8</td>
<td>3.56%</td>
<td>16.41</td>
</tr>
<tr>
<td>4</td>
<td>LS3</td>
<td>35</td>
<td>-</td>
<td>0</td>
<td>26.2</td>
</tr>
<tr>
<td>5</td>
<td>LS3</td>
<td>18.93</td>
<td>-</td>
<td>0</td>
<td>48.4</td>
</tr>
<tr>
<td>6</td>
<td>LS3</td>
<td>18.93</td>
<td>386.7</td>
<td>6.71</td>
<td>45.2</td>
</tr>
<tr>
<td>7</td>
<td>LS3</td>
<td>18.93</td>
<td>124</td>
<td>2.15</td>
<td>47.4</td>
</tr>
</tbody>
</table>
Figure 4.11 shows the results for the three different absorbers considered for the NEP absorber. The flat secondary reflector helps to improve the sensitivity of all misalignments and due to that even smaller absorbers could be installed in order to decrease the shadow projected by the secondary reflector and increase further the concentration ratio.

The NEP mirror shows a better tolerance to misalignments if a secondary reflector is installed. At certain point, the tolerance to misalignments is high enough to compensate the negative effects of the shadowing. The sensitivity to misalignments decreases specially in the case of a negative vertical misalignment. This high tolerance to sensitivity would make possible to install an even smaller absorber, that could increase concentration ratio and decrease the secondary reflector shadow without decreasing the optical performance of the absorber.

In the case of the LS3 mirror, with $\psi = 80^\circ$, the sensitivity to misalignments is not significantly increased by the inclusion of a secondary flat reflector and it does not compensate the negative effects of the shadow in any scenario, even considering big misalignments. The effects of the misalignments in the four absorbers considered in this scenario are shown in Figure 4.12.
Enhancing concentration ratio of solar concentrators

Figure 4.12: Misalignments analysis for the LS3/ET mirror and its standard and SFR receivers. a) Horizontal misalignments b) Vertical misalignments c) Rotational misalignments.

However, the increase of concentration in this case was of a 73.4 % as shown in Table 4.2 and the decrease of thermal losses could make the global thermal performance to increase as it will be explored in next chapter. In this case, it is shown how a smaller secondary reflector will increase the optical performance of the collector, since the increase of radiation produced by the decrease of the shadow (from 6.7 to 2.1 %) is higher than the decrease of optical performance due to rays missing the target.

Figure 4.12 b shows a potential further improvement of optical performance on the short SFR if a vertical displacement is inserted in the absorber position. The potential improvement is close to a 5 % but a further analysis of the sensitivity of this new position to horizontal and rotational misalignments as well as the new flux distribution should be conducted in a future stage.
Chapter 4: Improving the concentration ratio with a secondary reflector

4.6 Summary and discussion.

The main objective of this work was to demonstrate the increase of the concentration ratio in a PTC by using a secondary flat reflector, and to develop the equations necessary to choose the appropriate size and position of the secondary reflector and the absorber for given primary mirror dimensions. This method is an alternative for improving the performance of already designed parabolic mirrors, since it shows that the new absorber configuration improves the concentration ratio of the studied commercial collectors. This increase of concentration without increasing the mirror area could be a promising solution for rooftop applications, where the space available is critical and the collectors used have a low concentration ratio.

The calculations and the simulations showed an increase of the concentration ratio for a PTC when a secondary flat reflector is used compared to a classical single mirrored PTC. The increase of concentration ratio reaches a maximum for \( \psi \approx 80^\circ \) with shadowing factors of approximately 10%.

The increase in concentration ratio is higher in collectors with a larger focal distance, although the gap between absorber and mirror is smaller in the cases of large primary mirrors. Increases close to 80% in the concentration ratio can be achieved. The absorbers radii are reduced by around 50%, meaning the heat loss from the absorber is reduced by the same proportion.

Another advantage of the use of secondary collectors is that the flux around the receiver is distributed more uniformly around the tube. When using a secondary flat reflector, more than half the absorber will receive concentrated flux even in the worst case scenario, which may help in increasing the heat transfer between absorber and heat transfer fluid. This new distribution will also help in minimising the harmful effects of the difference of temperatures in the absorber surface. The effect of the heat transfer and pressure drop changes associated with changing the receiver size and flux distribution are addressed in the next chapter.

As the size of the receiver is decreased, this method could be valid to increase the aperture area of the mirrors as well, since the current available biggest receivers are limited to diameters of 90 mm. The equations shown in this work can be adapted and, if the diameter of the absorber and the acceptance angle are fixed, a bigger primary mirror can be designed without losing intercept factor.

In some cases, the inclusion of a secondary reflector within the glass envelope will be difficult due to the big dimensions necessary to capture all the reflected rays. A reduction of the secondary flat reflector dimensions in those cases would make it possible to encapsulate them, and the decrease of optical efficiency is compensated by the drop of the shadow losses, especially if there are not big misalignments.

The collectors with a flat secondary reflector in its absorber show a higher sensitivity to misalignments than the standard absorbers. However, there are necessary linear misalignments bigger than 1.5 cm or rotational misalignments bigger than 0.5° to have an impact on the optical performance of the parabolic troughs when a scattering factor of 0.23° (4 mrad) was already considered. The misalignments on a real system are expected to be lower than that [121, 136].

In collectors with a lower \( \psi \), the secondary reflector improves the tolerance to misalignments. For linear misalignments of 1.2 cm, the drop of optical efficiency of the standard absorber is higher than the drop on the secondary reflector absorber, even with the shadow considered. Those misalignments are not expected to happen in a real system. However, smaller misalignments will make the secondary reflector absorber performance to be closer to the standard one. The SFR absorber tolerance to misalignment in this case, especially in the vertical direction, makes it possible to design a smaller absorber, that will help to reduce even further the optical losses.
Enhancing concentration ratio of solar concentrators

An interesting possibility for the use of a secondary flat reflector is to install it in already built plants since no ground redistribution of the primary mirrors is required. For this purpose, it would be recommended to develop an economic analysis that includes the cost of changing the receivers, possible structure modifications to place the new absorber and new pumping power requirements in the plant. If the economic analysis shows that there is no reason for the substitution of the absorbers in already built plants, still new absorbers can be designed for the primary mirrors available in the market.

The major conclusions obtained from this chapter are:

- There is a potential in improving the optical performance of parabolic trough with a secondary flat reflector.

- In those mirrors with a big \( \psi \), a reduction of the shadow by decreasing the secondary reflector has a positive impact on the global optical performance of the trough.

- The flux distribution on a parabolic mirror absorber with secondary reflector is more uniform than in a standard absorber which will reduce thermal stress on the absorbers.
Chapter 5: CFD of a parabolic trough with secondary flat reflector

5.1 Introduction

In the previous chapter, a theoretical development and ray tracing analysis demonstrated how commercial parabolic troughs increase their concentration ratio by replacing their absorbers with smaller thermal receivers and flat secondary reflectors.

The total amount of energy received in the absorber of a trough with a secondary flat reflector decreases due to the shadow projected by the secondary mirror, as the new receiver is placed beneath it. For commercial parabolic troughs with $\psi < 85^\circ$ increases of concentration up to 77% were achieved and the total loss of energy due to the shadow was 11% for the worst scenario. It was shown how a higher increase of concentration increased the shadow projected on the primary mirror. A further analysis conducted for two benchmark parabolic troughs shown how the secondary reflector improved the tolerance of the collector to misalignments for a primary mirror with low rim angle. A further increase of concentration was achieved if the secondary mirror was designed to be shorter than the one needed to capture all the radiation due to a decrease on the shadow projected. A concentration increase of 74% was achieved with a shadow projected of 6.7% for the LS3/ET mirror by replacing its original absorber, a decrease of the secondary mirror leaded to a concentration increase of an 81% caused by the projected shadow reduction.

The main objective of this chapter is to analyse if the benefits of increasing the concentration ratio would have a positive effect in the thermal performance of a parabolic trough. The energy input in the absorber becomes lower due to the shadow projected, but the thermal losses decrease as an effect of reducing the thermal receiver area. In non-evacuated absorbers the appropriate design of the glass encapsulating both the thermal absorber and the secondary mirror can reduce the natural convection, further reducing the thermal losses. Finally, a different radius in the absorber will have an impact in the mass flow rate of the thermal fluid running inside the tube, so the pumping power will be altered. Pumping losses are not an objective to analyse in this chapter, but some considerations about them were taken in order to check the cases which could make the pumping power required relevant in the global performance of the collectors.

To predict the performance of the proposed absorbers, CFD simulations were conducted for the receivers of two benchmark parabolic troughs, the NEP collector and the LS3/Eurotrough collector. The first receiver is not evacuated, while the second one is evacuated. The non-evacuation of the absorber suggests higher thermal losses so that the maximum working temperature will be lower. A validation model consisting on two concentric cylinders was conducted prior the thermal performance analysis and the results obtained were compared with a model found in literature. This model was analysed first considering vacuum in the cavity, as in the LS3 receiver, and convection effects were added later, as in the NEP collector. The CFD models built matched the experimental results found in literature and the physical model was considered to be representative of the problem studied.

In addition to the change of concentration ratios and the reduction of the absorber size shown in the previous section, the flux distribution around the absorber changes with the introduction of the secondary reflector. A new temperature profile will appear on the cylindrical tube's surface due to the change of concentrated energy distribution along the receiver's surface. Having a more uniform temperature distribution along the tube surface will reduce thermal stress that in some cases can lead to plastic deformation on the metallic tube of the absorber.
Additionally, a different flux profile of radiation and temperature on the absorber surface could have an effect on the thermal performance of the parabolic troughs itself. Considering the receiver's tube size to be constant, CFD simulations were conducted in order to analyze the change of the thermal losses for different profiles of radiation on the receiver. To maintain simplicity, the optics and alignments of the components of the collectors were assumed ideal and a constant DNI of 800 W/m² was considered.

The main advantage of the inclusion of a secondary flat reflector was the reduction of the absorber diameter and the consequential increase of concentration. However, the absorber's diameter reduction will increase the pumping power required as the pressure drop increases with a diameter reduction. The pumping losses can be calculated from the pressure drop on the tube obtained with equation 5.1:

\[
\Delta P_{\text{tube}} = \frac{f_m}{2} \rho v^2 \frac{L}{D}
\]

(Equation 5.1)

The Moody's friction coefficient, \( f_m \), can be obtained from correlations. It depends on the Reynolds number and the tube's relative roughness. It was assumed a typical value of 0.06 to estimate the pumping power required. The electrical power required on the pump can be calculated from the pressure drop and the flow conditions adding a performance factor (electrical + mechanical).

\[
P_e = \dot{V} \frac{\Delta P_{\text{tube}}}{\eta_{\text{pump}}}
\]

(Equation 5.2)

It was assumed a total pump efficiency of 0.35 (mechanical to electrical) to establish an estimation of the pumping power required per meter in all the absorbers studied. Table 5.2 shows the maximum electric pumping power calculated per meter of absorber under the different flow rates studied for the NEP absorbers. For the DNI considered in this work, it is assumed that the available energy on the receiver (before thermal losses) will be of 960 W/m.

The maximum flow rate was chosen as 1 kg/s due to the rapid raise of pumping power. However, working flow rates around 2 kg/s have been reported for this collector [137], but even in the case of a standard absorber, the pumping power would have a noticeable impact on the performance of the plant and only if the thermal performance enhancement is high enough this flow rate would be recommendable. Table 5.2 shows the results of the power requirement estimations for the LS3 absorbers studied in this work. In this case, the available energy per meter of absorber will be 4608 W/m.
In this case, the maximum flow rate considered is suitable for the standard absorber, but it could represent a noticeable percentage of the produced energy for a secondary flat reflector absorber.

To calculate the pumping power of the whole plant, an equivalent length factor should be applied, in order to consider all the piping in the plant. However, the equivalent length of the plant won’t be the same for the standard and the secondary flat reflector absorber, since the diameter of the tubes joining collectors will change. Additionally in these calculations, the fluid viscosity was considered at low temperatures, which in the case of the thermal oil will drop considerably at high temperatures as shown in Appendix C, further reducing pumping power. Finally, very poor performances of the pump and the mechanical to electrical energy conversion were used as a worst case scenario. An accurate choice of all the parameters will decrease the difference in pumping power between the secondary flat reflector and the standard absorber. Nevertheless the advantage of working with lower viscosity fluids in the case of secondary reflectors is demonstrated, since the pumping power requirements will become a relevant percentage of the energy production at high flows.

### 5.2 Theoretical model

In order to analyse the flux distribution effects on the parabolic trough absorbers and the thermal performance of the absorbers and their modifications with secondary flat reflectors, two dimensional CFD analyses were conducted. Figure 5.6 represents the 2D models analysed in Ansys Fluent and all the main thermal losses considered in the model. To simplify the study only the parabolic trough absorber was simulated, assuming that its surface receives a given amount of flux depending on a fixed value of DNI and the primary mirror size. An ideal tracking and no misalignments were assumed with no shadow projected from other elements but the secondary flat reflector and no end of collector losses or conduction losses between the absorber and the glass were considered. The absorption of the glass will have a minor influence on the collector thermal losses, and such was not considered in the model.
Enhancing concentration ratio of solar concentrators

The absorber models were composed of a glass cover, the absorber tube, the gap between them and the heat transfer fluid inside the absorber in the case of the standard absorbers, with the addition of the secondary flat reflector in the case of the modified absorbers. The flux input was defined in a very narrow hollow cylinder joined to the absorber surface and with ideal conductivity between them. The details of each component are described in detail below, all the solid and fluid material properties can be found in Appendix C.

5.2.1 The absorber

In every case, the absorber was defined by the external radii of the different thermal absorbers considered, which can be extracted from Tables 2.2 and 4.2. In all cases, a thickness of 1.6 mm was considered for the absorber and steel for the material. The absorber was considered to receive a volumetric heat flux representing the solar flux impacting the thermal receiver, considering a DNI of 800 W/m². A volumetric flux was used instead of a surface heat flux to make it possible to recreate realistic flux distributions as it is explained in detail later. The length of the absorber was considered as 1 meter and the optics of all the components were considered ideal, therefore, except when a secondary flat reflector is inserted, the whole bundle of radiation entering the area of the mirror is assumed to be perfectly reflected in the mirror and received in the absorber.

In a real scenario, the heat flux impacts the surface of the absorber, not the entire volume. To simulate that effect in the CFD software, it was assumed that there was a $10^{-4}$ m hollow cylinder joint to the external face of the thermal absorber that was used as a volumetric heat flux source. For the secondary flat reflector thermal comparisons it was assumed that the amount of flux in this volumetric cavity depends on the flux distribution caused by the combination of primary and secondary mirrors.

The absorber perimeter is a circumference and its position can be represented as an angle considering as 0° the point of the circumference located at the line joining vertex and focus of the parabola. The local concentration ratio versus the absorber position can be considered a single cycle of a periodic function with period $2\pi$.

To analyse the effect of the flux distribution, different flux profiles were simulated for standard receivers, a uniform profile, an extreme concentrating profile and a realistic profile. To simulate the realistic fluxes around the absorber in the CFD model, the ray tracing profiles obtained were adapted as a Fourier series of eight terms that approaches the discrete data obtained from ray tracing simulations. Different Fourier
approximations with a lower number of terms and its comparison with the discrete data obtained from Zemax are shown in Figure 5.2. The agreement between the seven and eight terms degree Fourier transform and the set of discrete data is notorious. Defining the flux profile as a continuous function increases simplicity in implementing it in the CFD software.

As the Fourier series is an approximation to a periodic function obtained by a sum of sinusoidal functions, for those regions of the absorber with an intensity close to zero the Fourier series shows a small region with a negative intensity as shown in figure 5.2. To avoid parts of the absorber having a negative energy input, the profile was corrected with post processing to match the negative sectors of the series to zero. Figure 5.3 shows the flux profiles obtained with the ray tracing software and the eight terms Fourier series approximation used as a profile in the CFD simulations for all the absorbers studied in this section.
The emissivity of thermal receivers [138] depends on the absorber temperature and the material chosen. In the simulations conducted in this section, emissivity values of 0.1 and 0.2 were considered. Typical commercial absorbers emissivities are approximately 0.1 at operation temperatures [89]. At higher temperatures, the emissivity of the absorbers will increase, although quality materials could provide emissivities around 0.14 at 580 °C [138]. There are other effects such as aging or cheaper materials that could increase the emissivity of the solar absorber.

The energy received in the absorber is transferred to a working fluid flowing inside the tube. To simulate that effect in the CFD models, the absorber body was set up with 'convection thermal' conditions and an internal emissivity of 1. This assumption is based on considering that the heat transfer fluid will be opaque at the temperatures and wavelengths considered. The steel tube will then "conduct" energy from the fake hollow cylinder considered as the heat flux source and it will transfer the heat to the fluid, defined with a heat transfer coefficient and a free stream temperature, as it is described on section 5.2.5.
5.2.2 The secondary flat reflector

In those cases which consider a flat secondary reflector it was considered to be a 2 mm thick aluminium sheet. In the NEP model, the whole secondary flat reflector was considered to be encapsulated within the glass. In the LS3 collector, it was considered a 'partial' encapsulation of the secondary mirror by a glass with the same dimensions than the LS3 standard absorber. In a realistic scenario, this partial encapsulation will produce an optical loss due to the gap between the encapsulated part of the SFR and the non-encapsulated part, that will be equal to the flux impacting the area occupied by the glass. This loss has been ignored in terms of simplicity.

The emissivity considered for the flat secondary reflector was 0.1 and no optical losses or scattering effects were considered in it.

5.2.3 The glass cover

The glass used in the CFD model was considered as 1.8 mm thick wall, which matches the thickness of the prototypes built. The glass cover wall will simulate the interaction between the absorber and the atmosphere by a heat transfer coefficient.

The thermal boundary conditions for the glass cover were set up as 'mixed via system coupling' and 'via mapped interface'. The free stream temperature, representing the ambient temperature, was considered 20 °C and the temperature of the sky (that will define radiation losses) was considered of 0 °C. The glass internal and external emissivity were considered as 0.9.

A heat transfer coefficient was considered as a parameter input during the CFD simulations. It simulates the interaction between the surrounding air and the glass cover, and depends on the wind speed and the temperature of the glass. For low wind velocities, the glass temperature has a big influence, that becomes less dominating if compared with the high wind velocities influence. To simulate the heat transfer coefficient at extreme cases, the glass temperature was considered as constant at 50 °C, and the wind velocity was ranged between 0 and 15 m/s, since the operation of the solar power plants will be compromised for higher wind velocities [71].

The heat transfer coefficient was calculated as in [139] and it considered two cases. The first one considers a zero air velocity, and natural convection is the heat transfer mechanism. In the second case, velocities up to 15m/s were considered, and the forced convection heat transfer coefficient was calculated as a function of wind velocity. To maintain simplicity, it was assumed a constant temperature on the glass of 50 °C and an ambient temperature of 20 °C since the wind speed is much more relevant in the heat transfer coefficient that the air temperature.

In the case of still air conditions, the correlations found in literature are based on the equation 5.3, in which the subscript fl means that the dimensionless numbers are evaluated at the film temperature:

$$N_{u_{fl}} = C_b (Gr_{fl}Pr_{fl})^m$$

(Equation 5.3)

The coefficients $C_b$ and $m$ shown by Holman [139] and used in this section were calculated in several experimental works by other authors [140, 141]. for the range of Grashof and Prandtl numbers in which the solar thermal receivers were analysed in this chapter, the coefficients recommended by Holman are the ones calculated by Morgan [141] $C_b=0.53$ and $m=0.25$. If the coefficients and the air thermal properties are known, and considering that the length of the absorber simulated is one meter, the heat transfer coefficient can be calculated if the Grashof number, the Prandlt number, and the thermal conductivity, $k$, are known as:
Enhancing concentration ratio of solar concentrators

\[ h = \frac{k_f C_b (Gr B P_{fl})^n}{L_c} \]  
(Equation 5.4)

Lc is a characteristic length that will depend on the geometry studied. In the case of a cylinder surrounded by air this length is equivalent to the glass diameter \[139\]. For those cases in which air velocity is not zero, forced convection appears as the heat transfer mechanism between the glass and the atmosphere. The correlations used for the forced convection case are shown in equations 5.5 and 5.6. Again, the subscript \(fl\) means that the air properties are evaluated at the film temperature, maintained as a constant at 50 °C in all the simulations.

\[ Nu_f = C_b \left( \frac{u_x D}{\nu_f} \right)^n Pr_f^{1/3} \]  
(Equation 5.5)

\[ h = \frac{k_f C_b (\rho Re)^n}{L_c} Pr_f^{1/3} \]  
(Equation 5.6)

\(C_b\) and \(n\) depend on the Reynolds number and they can be obtained from table 5.3 [139], which values in the particular range studied in this work were obtained by [142]. As the velocities considered in this work, ranged between 1 and 15 m/s and the diameters of the glass covers considered, ranged between 36.6 and 125 mm, the Reynolds numbers obtained are comprised between 2241 and 117563.

<table>
<thead>
<tr>
<th>( Re )</th>
<th>( C_b )</th>
<th>( n )</th>
</tr>
</thead>
<tbody>
<tr>
<td>40-4000</td>
<td>0.683</td>
<td>0.466</td>
</tr>
<tr>
<td>4000-40000</td>
<td>0.193</td>
<td>0.618</td>
</tr>
<tr>
<td>40000-400000</td>
<td>0.027</td>
<td>0.805</td>
</tr>
</tbody>
</table>

5.2.4 The absorber-glass enclosure.

The absorber glass enclosure was considered to be either filled with air or evacuated. Air thermal properties were extracted from [82] and they can be checked in Appendix C.

It is not possible to define a 'vacuum' material in the CFD software used for the simulations neither the working fluid in the enclosure could be 'turned off' to simulate an evacuated collector. For this reason, a material which properties would make it behave similarly to vacuum was implemented. To model the vacuum, a gas material with the following properties were defined:

- Density: \(10^4\) kg/m³
- Thermal conductivity: \(10^4\) W/mK
- Viscosity: \(10^4\) kg/m s

During the CFD simulations, it was checked that there were no velocity profiles within the cavity when vacuum was selected as the material contained within the enclosure and that no convection losses appeared between absorber and glass.
5.2.5 The heat transfer fluid

The heat transfer fluid was modelled as the convective boundary conditions inside the absorber tube. Its magnitude depends on two parameters defined as independent inputs of the simulations, the free stream temperature, representing the heat transfer fluid temperature and the heat transfer coefficient, that depends on the heat transfer fluid thermal properties and its mass flow rate.

The working fluid considered was Therminol V66 \[143\]. Its thermal properties, shown in Appendix C are similar to other oils used in commercial power plants such as \[49\]. Those mineral oils degrade rapidly with temperatures over 425 °C and that was decided to be the maximum fluid temperature simulated. Water was not used despite having better thermal properties due to the difficulty of operating with high pressures maintaining saturated liquid water. As seen in the literature review section, there are cases which produce steam directly in the absorbers using water as transfer fluid, but in terms of simplicity it was decided to avoid this scenario, as well as the consideration of molten salts despite they can operate at higher temperature.

The heat transfer coefficient for the boundary condition was found by calculating the Nusselt number defined as:

\[
\text{Nul} = \frac{hL_e}{k_f}
\]

(Equation 5.7)

For turbulent flow, defining the fluid temperature and velocity, the Nusselt number was obtained using the Petukov equations \[144\]. The Petukov equations allows the calculation of Nusselt number as a function of the skin friction coefficient, Reynolds and Prandtl numbers and it is valid for \(4000 \leq \text{Re} \leq 10^5\) and \(0.6 \leq \text{Pr} \leq 60\).

\[
f = \frac{1}{4 \left(1.82 \log_{10} \text{Re} - 1.64\right)^2}
\]

(Equation 5.8)

\[
\text{Nu} = \frac{f/2 \left(\text{RePr}^{1/2}\right)}{1.07 + 12.7 f/2^{1/2} \left(\text{Pr}^{2/3} - 1\right)}
\]

(Equation 5.9)

The range of velocities analysed assured a turbulent regime for both the standard and the secondary flat reflector absorbers. The maximum velocity analysed was defined by the Reynolds number limitation on the Petukov equations. There is a second Petukov equation valid for a bigger range of Reynolds numbers. However, this first equation is more accurate for the Reynolds number range studied and it was decided to maintain it defining the maximum velocity within the appropriate range.

5.3 Radiation and convection heat transfer models

As the absorber heats up and its thermal losses by radiation increase. In evacuated collectors, despite the improvements on the materials, the maximum operation temperature of the plant is given (among other aspects such oil reliability) by the radiation thermal losses. Smaller collectors, such as the NEP receiver, suitable for mid-range temperature applications, use non-evacuated absorber, and convection between the absorber and its surrounding air increases thermal losses and lowers the maximum output of the solar collector.

To simulate the thermal performance of the thermal receivers, a natural convection and a Discrete Ordinates (DO) radiation model were set up in Fluent. Before analysing the thermal performance and the flux distribution effects on the receiver tube, it is necessary to prove that the models built in CFD represent the reality, and two validation models were analysed against experimental models found in literature to validate the model correctness.
Enhancing concentration ratio of solar concentrators

The validation cases considered an inner and an outer isothermal cylinder surface with different temperatures and emissivities. In this section, the radiation and natural convection mechanisms are explained in detail, and in the next, the validation CFD models compared with the theoretical models explained are shown.

5.3.1 Radiation heat transfer

Even if a solar thermal absorber is evacuated, it will radiate part of the received energy to the glass due to the temperature difference between the two bodies. The amount of radiation that a body emits depends on the temperature and the material itself. There are techniques, such as selective surfaces, that maximise solar absorbance and minimise thermal radiation losses. The CFD models built in this work include radiation and therefore a well-known model was used to validate the simulations against it.

The radiation heat exchange ratio between two surfaces will depend on the temperature difference between these two surfaces and their materials, but also on the orientation of such surfaces. The view factor $F_{ij}$ is a factor that takes into account how much energy is transferred from a body “i” to a body “j” depending on the orientation between them. As defined in [82], the view factor is a purely geometric quantity and it is independent of the surface properties and temperature. The view factor is based on the assumption that the surfaces are diffuse emitters and diffuse reflectors.

The value of the view factor ranges between zero and one. If $F_{ij} = 0$, body “i” “doesn’t see” body “j” and therefore there is no radiation heat exchange between them. If $F_{ij} = 1$, it means that body “j” surrounds completely body “i” and therefore all the energy radiated from the body “i” is received by the body “j”. To reach a view factor of 1, it is necessary that there are not other bodies that could absorb, emit or scatter radiation. For this reason, it is necessary to consider that the two bodies are separated by a non-participating medium such as air or vacuum and that the two bodies are diffuse emitters and isothermal.

There are well-known view factors for typical geometries, such as the case of two concentric cylinders. In the case of a solar thermal collector, it is necessary to calculate the view factor from the inner (absorber) to the outer (glass) cylinders and other elements such as lids, supports, etc. As the length of the absorber is much bigger than its width, no other elements than the glass and the absorbers were considered in the analysis.

Considering a 1 meter long thermal absorber with the dimensions of the NEP and the LS3 collector, the view factor obtained is 1 as it can be assumed that the length is much bigger than the width.

In the case of the two concentric cylinders shown in figure 5.4, if their dimensions, temperatures, view factor and emissivities, $\varepsilon_i$ are known, it is possible to calculate the radiation heat exchange between the two bodies as shown in equation 5.11 [82]:

$$\frac{A_1}{A_2} = \frac{r_1}{r_2}$$

(Equation 5.10)

$$\dot{Q}_{1,2} = \frac{A_1\sigma(T_1^4 - T_2^4)}{\frac{1}{c_2} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{T_1}{T_2}\right)^2}$$

(Equation 5.11)

These analytical correlations are only applicable for a case with two isothermal cylinders and it is not possible to analyse a realistic flux with this equations, since the temperature of the absorber won’t be uniform any more. However, the case of isothermal cylinders is useful to validate the CFD model constructed in this work.
5.3.2 Natural convection heat exchange

Natural convection occurs in the surroundings of a surface surrounded by a gas at a different temperature and it depends also on the surface geometry as well as its orientation. The same phenomena will appear inside an enclosure filled with a gas.

The complexity of this phenomena makes impossible to find an analytical relationship for the heat transfer around a surface by solving the known motion and energy equations. However, there are several numerical correlations for typical geometries based on experimental studies, based on the average Nusselt number. The Nusselt number can be calculated as in equation 5.12:

\[ Nu = \frac{h L_c}{k_t} = C (Gr T L \Pr)^n = CRa_{le}^n \]  

(Equation 5.12)

Where \( h \) is the convective heat transfer coefficient and \( k_t \) is the thermal conductivity of the working fluid at the surface. The constants \( C \) and \( n \) depend on the surface geometry and the flow regime. \( C \) is normally lower than 1 while for natural convection flow \( n \) is usually \( \frac{1}{3} \). The Rayleigh number, \( Ra_{le} \), is an dimensionless number related to the fluid motion which is the product of the Prandtl number, \( \Pr \), which describes the ratio between momentum diffusivity and thermal diffusivity and the Grashof number which is the ratio of the buoyancy to viscous force acting on a fluid.

If \( Ra_{le} > 1708 \) the buoyant force is strong enough to initiate a laminar flow while if \( Ra_{le} > 3 \times 10^5 \) the fluid will become turbulent. The fluid properties are evaluated at its average temperature \( \frac{T_1 + T_2}{2} \). The heat transfer through the enclosure can be calculated as:

\[ \dot{Q} = k_{eff} A \frac{T_1 - T_2}{L_c} \]  

(Equation 5.14)

In which, the effective thermal conductivity, \( k_{eff} \), can be calculated as \( k_{eff} = k_t Nu \). If there is no motion in
Enhancing concentration ratio of solar concentrators

5.3.3 Natural convection in a concentric cylinder enclosure.

Considering two concentric cylinders as in a typical solar thermal receiver, the natural convection of the air within the enclosure can be defined with equations 5.12 to 5.14 if the characteristic length is defined. The characteristic length of a cylindrical enclosure can be found from the diameters of the inner and outer cylinders as $L_c = \frac{D_{\text{out}} - D_{\text{in}}}{2}$. The rate of heat transfer through the annular space per unit length of the tubes is expressed in the literature as:

$$\dot{Q} = \frac{2\pi k_{\text{eff}}}{\ln\left(\frac{D_o}{D_i}\right)}(T_i - T_o)$$  \hspace{1cm} (Equation 5.15)

The relationship between effective thermal conductivity and thermal conductivity within a cylindrical enclosure was found by experimental correlations by Raithby and Hollands [145]. The relationship found was as described in equations 5.7 and 5.4.

$$\frac{k_{\text{eff}}}{k_i} = 0.386 \left(\frac{Pr}{0.861 + Pr}\right)^{\frac{1}{3}} (F_{\text{cyl}}Ra_L)^{\frac{1}{3}} \text{ if } k_{\text{eff}} > k_i$$ \hspace{1cm} (Equation 5.16)

$$1 = 0.386 \left(\frac{Pr}{0.861 + Pr}\right)^{\frac{1}{3}} (F_{\text{cyl}}Ra_L)^{\frac{1}{3}} \text{ if } k_{\text{eff}} \leq k_i$$ \hspace{1cm} (Equation 5.17)

The geometric factor for concentric cylinders, $F_{\text{cyl}}$, is a variable that depends on the dimensions of both inner and outer cylinder that can be expressed as:

$$F_{\text{cyl}} = \frac{\ln\left(\frac{D_o}{D_i}\right)}{L_c^3 \left(D_i^{-\frac{3}{2}} + D_o^{-\frac{3}{2}}\right)^{\frac{5}{2}}}$$ \hspace{1cm} (Equation 5.18)

Equation 5.15 shows the rate of heat transfer of natural convection in a cylindrical annulus and the correlation is proven to be true within the range $0.7 \leq Pr \leq 6000$ and $10^2 \leq F_{\text{cyl}}Ra_L \leq 10^7$. If $F_{\text{cyl}}Ra_L \leq 100$, natural convection forces are negligible and thus the heat transfer mechanism in the cavity is conduction. In some cases, when the receiver temperature is low and the glass temperature is high (although lower than the absorber's), the air will have not enough energy to start natural convection and therefore $Nu=1$ and $k_{\text{eff}}=k_i$ and the heat loss mechanisms will become conduction as well.

As it was shown above, the dimensions of the cylinders will have an influence in natural convection as well as their temperatures. An analysis of the natural convection temperature range is developed in the next section prior to the validation CFD simulations.

5.4 Validation simulations

In this work, several models representing a parabolic trough receiver were simulated using ANSYS Fluent 14.5. To prove the models accuracy, two validation simulations based on two isothermal concentric cylinders were conducted. The results obtained were compared with those found in [82], obtained from experimental correlations.

In this work, both evacuated and non evacuated receivers are analysed. The first validation model considers that the space between the concentric cylinders is evacuated, therefore radiation was the only thermal exchange between both cylinders. The second case considers that the gap between cylinders is filled with air at atmospheric pressure and natural convection occurs inside the cavity formed by the two cylinders. Several
air models were compared with an analytical model based on the air properties found in [82] and shown in Appendix C.

5.4.1 The validation models

A model based on two concentric isothermal cylinders was built in Ansys considering the dimensions of the NEP absorber Table 2.2. The two isothermal cylinders were used to validate the physical model configured in Ansys Fluent against the analytic equations shown above. Three different meshes were simulated to assure the independence of the results from the number of nodes considered in the meshes.

The characteristics of the three meshes built are shown in Appendix B. The first one is the mesh used in the thermal performance simulations while the other ones were used to check the validity of the results obtained with the first one.

5.4.2 Radiation model validation

In an evacuated solar receiver, if the conduction losses at the end of the receiver are considered negligible, the only heat loss mechanism observed is radiation. If the two walls are considered isothermal and their temperatures are known, the heat exchange between them can be calculated with equation 5.11. The amount of heat exchanged does not only depend on the surfaces temperature but also on their emissivities. An ideal solar receiver would have zero emissivity from the absorber tube to the glass. However, real absorbers have emissivities that depend on the temperature of the receiver, generally increasing as the temperature increases.

To validate the radiation model four scenarios which consider different emissivities for both inner and outer cylinders and a range of temperatures ranging from 70°C and 490°C were simulated. The outer cylinder represents the glass enveloping the absorber and it will be in contact with the atmosphere, and it will keep the vacuum in the gap formed by the glass and the absorber. In order to simulate the vacuum, a gas with the properties shown in section 5.2.4 was used. As it is shown in Figure 5.5, the velocity of the gas inside the chamber is negligible and there was no convection or conduction between the isothermal cylinders and the gas. Therefore, the material model was considered valid as a vacuum model.

![Figure 5.5: Velocity contours in the glass cavity for vacuum conditions.](image-url)
Several radiation models were tested in this validation stage and the discrete ordinates (DO) model was chosen as the most appropriate for representing the radiation between the absorber and the glass cover. Figure 5.24 shows the correlation between the theoretical results and two of the meshes for an ambient temperature of 20 °C and the NEP dimensions.

![Figure 5.24](image)

Tables A.1 to A.4 attached in Appendix A show the calculated values for the radiation heat transfer and the total and radiation heat transfer obtained with the CFD simulations. The total radiation obtained in CFD matches the radiation heat transfer, showing that the gas modelled as “vacuum” has no influence on the heat transfer between surfaces and that the radiation model is correct. Only two meshes were considered in this validation, to check that there was an independence of the mesh in the radiation model. However, a more detailed mesh independence study was conducted in the next case, which considers natural convection. A third mesh with added in that scenario and different diameter sizes and meshes with and without secondary flat reflector were considered then. Velocity and temperature profiles along an imaginary line were used to validate the meshes.

**5.4.3 Natural convection model validation**

Prandtl number and Rayleigh number depends only on the fluid properties and the temperature of the cavity surfaces. Both numbers were evaluated for air at different ambient temperatures for temperatures in the inner cylinder between 50 and 500 °C. Prandtl number shows little difference at low ambient temperatures, being approximately 0.725 and it falls under the critical value $Pr>0.7$ for temperatures close to 350 °C. However, the Prandtl number at 500 °C was 0.699 in all the scenarios studies and it was considered that the Prandtl number won't be a limitation for natural convection in this analysis.

Figure 5.7 shows how, for a range of temperatures between 90 and 500 °C and considering air as the fluid inside the cavity, Rayleigh number falls within the limits that allow natural convection to occur in the cavity.
Chapter 5: CFD of a parabolic trough with secondary flat reflector

Chapter 5: CFD of a parabolic trough with secondary flat reflector

The product of Rayleigh number and the geometric factor relates the fluid properties within a cavity and the cavity dimensions suitable for natural convection. Natural convection will happen if the product of the geometric factor of the cavity and the Rayleigh, \( \text{Ra}_c \), number falls between 100 and \( 10^7 \). Figure 5.8 shows how natural convection will be the heat transfer mechanism in a NEP’s dimensions cavity for a range between 100 and 500 °C.

Different properties were tested to model the 'air' material in the CFD software and they were compared with the correlation experiments found in literature shown in previous section. The natural convection model used, was compared with the theoretical correlation shown in equation 5.15 using different sets of properties for the air. Figure 5.9 shows the results obtained for the 7 different air properties tested and its comparison with the analytical equations shown in the previous section.

![Figure 5.7: Characteristic numbers for air at different temperatures. a) Prandtl number b) Rayleigh number.](image)

![Figure 5.8: Product of Rayleigh and the geometric factor with NEP’s dimensions concentric cylinders geometry.](image)
Table 5.4 shows the air properties chosen in the CFD natural convection models. The polynomials used in some of the models are shown in Appendix C, and demonstrated to offer the highest accuracy in the range of temperatures and for the cavity dimensions studied. The polynomials values were extracted from the values found in [82].

Table 5.4. Different air thermal properties tested in the natural convection models

<table>
<thead>
<tr>
<th>CFD Air Model</th>
<th>Density (kg/m$^3$)</th>
<th>$C_p$ (J/Kg K)</th>
<th>$\nu$ (m$^2$/s)</th>
<th>$k_t$ (W/m K)</th>
<th>$\beta_t$ (1/K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chosen model</td>
<td>Polynomial</td>
<td>Polynomial</td>
<td>Polynomial</td>
<td>Polynomial</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Bousinesq (1.18)</td>
<td>Poly</td>
<td>Polynomial</td>
<td>Polynomial</td>
<td>0.00255</td>
</tr>
<tr>
<td>3</td>
<td>Bousinesq (1.18)</td>
<td>Predefined$^\ast$</td>
<td>Predefined$^\ast$</td>
<td>Predefined$^\ast$</td>
<td>0.00255</td>
</tr>
<tr>
<td>4</td>
<td>Bousinesq (1.18)</td>
<td>1050</td>
<td>2.15E-005</td>
<td>0.031005</td>
<td>0.00255</td>
</tr>
<tr>
<td>5</td>
<td>Polynomial</td>
<td>1050</td>
<td>2.15E-005</td>
<td>0.031005</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Bousinesq (1.18)</td>
<td>Polynomial</td>
<td>Polynomial</td>
<td>Polynomial</td>
<td></td>
</tr>
</tbody>
</table>

$^\ast$Thermal expansion is calculated from the density if density is a polynomial and it cannot be selected

$^\ast\ast$Predefined are the default parameters in Ansys

5.4.4 Mesh independence study

For the four models analysed in this chapter, the NEP and the LS3/ET with and without secondary mirror, three different meshes were analysed in order to prove the independence of the results from the mesh used in the simulations. In every case, a non-evacuated absorber was simulated in the mesh independence analysis and the temperature and velocity profiles around a line were obtained.

The constructive details of every mesh and the results obtained in the mesh independence analysis can be found in Appendix B.
5.5 Influence of the flux distribution

Adding a secondary reflector stage on a parabolic trough will change the flux profile around the absorber. The flux distribution for a standard absorber and an absorber with secondary flat reflector was obtained in section 4.3 by ray tracing simulations and the flux obtained in the case of a standard tube was compared with examples found in literature. It was shown how the misalignments and optics aberrations modify the flux distribution around the absorber, but the secondary flat reflector improves uniformity in every scenario. The increase of uniformity will reduce hot spots on the absorber, minimising thermal stress that could lead to a bending in the absorber, reducing optical efficiency.

Although there are several works that considered realistic flux distributions in their models to study effects such as the change of flux distribution due to misalignments and optics aberrations [85], or the thermal stress and bending on a solar receiver [87], its influence on the absorber's efficiency is not entirely clear. Lu et al. [80] found an increase of heat loss on the absorber while comparing a uniform and a non-uniform flux distribution around the absorber, although effects as the mass flow rate were not included as variables in the study. Ghomrassi et al.'s simulations [81] found an increase of outlet temperature on the absorber for those absorbers receiving a higher flux on the bottom. In this section, the flux influence on the performance of two commercial absorbers has been analysed with CFD comparing a realistic, an ideal, and an extreme concentration flux (in which the absorber receives all the incoming energy in a narrow section) with the same total intensity.

The realistic flux was obtained as described in 5.2.1, the ideal flux is assumed to be completely uniform, so the flux is distributed evenly around the absorber surface. However, this ideal flux distribution is impossible to achieve without secondary optics if the primary mirror has finite dimensions. Lastly, if there are vertical misalignments in the absorber that make its centre to be more distant from the centre of the parabola than the focal line, the area of absorber illuminated will decrease as the bottom part of the tube will be closer to the focus. As a worst case scenario, it was considered a case which has all the flux concentrated in a narrow sector at the bottom of the receiver. This flux will cause a maximum thermal stress since the absorber temperature will be dramatically influenced by this extreme concentration.

Figures 5.10 to 5.13 show the percentage of heat transferred to the heat transfer fluid for the different flux profiles considering different wind speeds and emissivities on the LS3/ET evacuated absorber.
Enhancing concentration ratio of solar concentrators

Figure 5.10: Heat transferred to the fluid for \( v_{\text{wind}} = 0 \) m/s and \( \varepsilon = 0.1 \). a) Realistic flux profile; b) Uniform flux profile; c) Max concentration in a sector; d) Realistic flux vs Uniform profile.
Figure 5.11: Heat transferred to the fluid for $v_{\text{wind}} = 15$ m/s and $\varepsilon = 0.1$. a) Realistic flux profile; b) Uniform flux profile; c) Max concentration in a sector; d) Realistic flux vs Uniform profile.
Figure 5.12: Heat transferred to the fluid for $v_{\text{wind}} = 0$ m/s and $\varepsilon = 0.2$. a) Realistic flux profile; b) Uniform flux profile; c) Max concentration in a sector; d) Realistic flux vs Uniform profile.
Chapter 5: CFD of a parabolic trough with secondary flat reflector

Figure 5.13: Heat transferred to the fluid for $v_{\text{wind}} = 15$ m/s and $\varepsilon = 0.2$. a) Realistic flux profile; b) Uniform flux profile; c) Max concentration in a sector; d) Realistic flux vs Uniform profile.

If compared with a realistic flux distribution, a uniform flux profile increases the thermal performance of the evacuated absorber at low flow rates while a extreme flux distribution will decrease it. The performance difference between realistic and uniform flux distribution is compensated at high flow rates and it becomes negligible for the maximum flow rate considered. The performance difference decreases for the extreme concentration scenario as well, but it does not get compensated at high flows, emissivities and wind velocities considered in the simulations. A minimum decrease of 1% was observed for the extreme flux conditions performance at high temperatures and flows while it decreases further for lower temperatures and flow rates.

The performance decreases at low temperature for all the fluxes considered. Again, the decrease is lower at high mass flow rates and it is lower again in the case of a uniform distribution. The decrease of performance is caused by the high viscosity of the oil at those temperatures that decreases the heat transfer coefficient between the absorber and the heat transfer fluid.
Finally, the absorber surface temperature was evaluated for all the scenarios considered and the most representative results are displayed on Figure 5.14 considering the three different flux distributions. Figure 5.14 a shows a worst case scenario found in the simulations for the temperature distribution, with a cold heat transfer fluid and the minimum flow rate while Figure 5.14 b represents the temperature distribution at maximum flow rate and maximum fluid temperature.

![Figure 5.14: Temperature on the absorber's outer surface. a) $\dot{m} = 0.1 \text{ kg/s; } \tau = 0.1; T_{\text{fluid}} = 290 \, ^\circ \text{C}$ b) $\dot{m} = 2.5 \text{ kg/s; } \tau = 0.2; T_{\text{fluid}} = 425 \, ^\circ \text{C}$](image)

At low flow rates, as shown in Figure 5.14 a, the temperature profile in an absorber can reach differences around 400 \(^\circ\text{C}\). This temperature difference can lead to bending and misalignments due to expansions that could decrease the optical performance of the trough or put stress on the absorber that could compromise its long term reliability. The effect of flux distribution on the thermal performance of the parabolic trough absorber can be neglected by an increase of flow rate. However, the temperature profile at high flow rates with a realistic flux shows a maximum difference of 50 \(^\circ\text{C}\) and the thermal stress would be much lower.

A similar analysis was conducted for the NEP non-evacuated absorber. The influence of the flux is expected to change in this absorber since natural convection will reduce the temperature gradient on the absorber’s surface. The position of the Sun will be relevant in this scenario; as the parabolic trough tracks the Sun the “0” position of the absorber, which is the point of the absorber joined to the vertex of the parabola by its focal line, will change its position in respect to a vertical axis. Three different positions have been considered for the NEP absorber to study the gravity effects on the natural convection. The first one assumes that the focal line of the mirror is aligned in a vertical axis, which will occur in a parabolic trough north-south oriented at midday, the second one assumes that the angle between the focal line and the vertical axis is 30° and the third one assumes that the angle between them is 60°. These two cases correspond to a parabolic trough East-West oriented and the angles represent midday Sun positions in winter and Summer for a typical location suitable for concentrating power systems (approximately 30° of latitude).

Due to the effect of natural convection, in this case the difference on thermal performance is not relevant, and all the fluxes considered have similar thermal performance at high flow rates. At low flow rates, a maximum difference close to 1 % appears between the extreme cases, but it is considered that the flux distribution has no influence in the thermal performance of non-evacuated absorbers. Figure 5.15 shows four of the scenarios considered, with minimum and maximum flow rates and minimum and maximum wind velocities. The realistic flux at 60° position was removed from the graphs, as no difference was obtained with the realistic fluxes at 0 and 30°.
Figure 5.15: Heat transfer in the non-evacuated absorber for different flux profiles a) $\dot{m} = 0.05 \text{ kg/s; } \varepsilon = 0.1; v_{\text{wind}} = 0 \text{ m/s}$ b) $\dot{m} = 1 \text{ kg/s; } \varepsilon = 0.1; v_{\text{wind}} = 15 \text{ m/s}$ c) $\dot{m} = 0.05 \text{ kg/s; } \varepsilon = 0.2; v_{\text{wind}} = 0 \text{ m/s}$ d) $\dot{m} = 1 \text{ kg/s; } \varepsilon = 0.2; v_{\text{wind}} = 15 \text{ m/s}$.

A similar trend is observed if the temperature profile is evaluated. The maximum temperature difference is decreased due to natural convection for the realistic and the extreme case flux profiles. Again, the extreme flux considered can be relevant in terms of thermal stress while the thermal difference calculated would not cause an undesired bending in with a realistic flux distribution.
Figure 5.16: Temperature distribution on the non-evacuated absorber's surface a) $\dot{m} = 0.05 \text{ kg/s; } \varepsilon = 0.1; T_{\text{fluid}} = 250 \degree C$ b) $\dot{m} = 1 \text{ kg/s; } \varepsilon = 0.1; T_{\text{fluid}} = 250 \degree C$.

The irregularity shown in the temperature profile for a maximum concentration sector is due to the approximation realised to obtain the profile.

5.6 Thermal performance of a solar collector with secondary flat reflector

In this section, CFD simulations were conducted for the NEP and LS3 absorber with and without secondary flat reflector to compare the total thermal performance of the original receiver and the one modified with the flat secondary reflector. Two additional proposals to enhance the efficiency of the absorbers were also evaluated. For the NEP collector, with a secondary reflector that does not represent a big percentage of the primary mirror area, but which absorber is non-evacuated, an absorber with insulation filling the upper gap between the secondary flat reflector and the glass was simulated. This insulation will remove natural convection from the upper part of the absorber-glass envelope, decreasing thermal losses. For the LS3 absorber, the shorter secondary mirror evaluated in the previous chapter was simulated considering a perfect alignment. That’s not the optimal position for this absorber, since in this case a positive vertical displacement increases the optical performance. An optimal location of the absorber can increase the thermal input by another 1.6 %, as shown in Figure 4.12.

The decrease in the diameter of the absorber when a secondary flat reflector is inserted modifies the heat transfer coefficient from the absorber to the fluid that is also dependant on the flow rate and the fluid temperature. The comparisons were taken considering an identical mass flow rate for both the original absorber and the ones modified with the secondary flat reflector. Flow rates were chosen high enough to assure a turbulent flow of the thermal oil inside the absorber and in an appropriate range to fit in Petukov equations. Each flow rate was simulated from the lowest temperature that will assure a turbulent flow for the sections of both absorbers and 425 °C which is the maximum operation temperature allowed for commercial thermal oils.

Despite the small influence of the flux profile shown for high mass flow rates in the previous sections, a realistic flux profile obtained with ray tracing simulations was inserted in every model. As in previous simulations, the flux considered ideal optics in both primary and secondary mirror, no scattering and no
misalignments of any component and a DNI of 800 W/m². The flux profiles obtained can be checked in Figure 5.3.

Wind velocity and thermal emissivity on the absorbers were varied to simulate their influence on the thermal performance. Both values were considered as the minimum and the maximum expected in the normal operation of a parabolic trough. Wind velocity was ranged between 0 m/s and 15 m/s. Although intermediate wind velocities were analysed, the graphs in this section show only minimum and maximum wind.

The absorbers emissivity was considered as 0.1 as commercial collectors emissivity is around this value at their working temperatures. Figure 5.17 compares the results obtained with that emissivity and the highest flow rate simulated the thermal losses to those reported by Schott absorbers for its 4th generation of solar absorbers [82] and tested in NREL (US National Renewable Energy Laboratory) under laboratory conditions.

The Schott 4th generation absorber is the one with a lowest emissivity found in the market, and the emissivity for temperatures above 425 °C is bigger than 0.1, effects such as aging can increase further the thermal emissivity. In the CFD simulations a emissivity of 0.2 was also considered as a worst case scenario. The heat transferred to the fluid was calculated as a percentage of the total energy reaching the absorber. No end of the collector or other optical losses such as glass diffraction were considered as they can be expected to be equal in both absorbers. The secondary flat reflector was considered to have ideal optics. The influence of the mirror optics was discussed in section 4.4 and it was ignored in this CFD analysis to increase simplicity. The percentage of energy transferred for the secondary flat reflector cases is obtained as a ratio between the energy received in the absorber and the total energy impacting the mirror in standard absorber cases in order to consider the shadow effects.
5.6.1 LS3 absorber

The LS3’s primary mirror has a length of 5.76 m. For a DNI of 800 W/m², an incoming flux of 4608 W/m is expected to impact the standard absorber while 4299.26 W/m and 4422 W/m are expected for both the original and the shortened flat reflectors with an ideal position. Due to the software rounding while creating the flux profiles, the fluxes obtained were approximately 4572 W/m, 4297 W/m and 4407 W/m for the three absorbers considered. The absorbers’ diameters were of 35 mm in the case of the standard absorber and of 18.7 mm for the secondary reflector ones. Figure 5.18 shows the heat transfer coefficients calculated for the absorber-oil interface for the Petukov approximations, the markers show the actual points that were used later as the fluid temperature in the CFD simulations.

As the absorber is evacuated it is not expected that the wind has a very strong influence on the thermal losses of the absorber and the glass was considered of equal dimensions for both absorbers. As explained in the optics section this will cause a partial encapsulation of the secondary flat reflector within the glass for the secondary reflector with original dimensions and it will allow to fully encapsulate the shortened version. A partial encapsulation would imply some issues in the operation of a secondary flat reflector absorber, such as a loss of optical properties due to being scratched over time by particles of even dust being stuck to it or the necessity of additional supports to fix the secondary reflector parts that cannot be encapsulated. For the glass diameter considered (62.5 mm) the wind heat transfer coefficient obtained with the methodology developed in [139] is shown in table 5.5.

<table>
<thead>
<tr>
<th>Wind velocity (m/s)</th>
<th>Heat transfer coefficient (-)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>5.9</td>
</tr>
<tr>
<td>2</td>
<td>16.6</td>
</tr>
<tr>
<td>5</td>
<td>25.7</td>
</tr>
<tr>
<td>15</td>
<td>61.8</td>
</tr>
</tbody>
</table>

Figure 5.18: HTF heat transfer coefficients calculated with Petukov equations for: a) LS3 absorber. b) LS3-SFR absorber.
Figures 5.19 and 5.20 show the results obtained for the CFD simulations for the LS3 absorber and its SFR modifications considering maximum and minimum absorber emissivities and wind velocity.

Figure 5.19: Standard and SFR receiver for the LS3 primary mirror. a) $\varepsilon=0.1$; $v_{\text{wind}}=0$ m/s. b) $\varepsilon=0.1$; $v_{\text{wind}}=0$ m/s. c) $\varepsilon=0.1$; $v_{\text{wind}}=15$ m/s. d) $\varepsilon=0.1$; $v_{\text{wind}}=15$ m/s.
Both secondary reflector options increase the performance of a standard absorber at low flow rates for every scenario considered. However, in the same way that an increase of flow rate compensated the pernicious effects of the flux distribution, the performance of the standard absorber increases if compared with the standard absorber at low temperatures for high flow rates.

In the range of operation of a power plant, assuming high flow rates, the full sized secondary reflector would only offer a higher performance for high emissivities and temperatures close to the outlet temperature. As the thermal emissivity increases with temperature, a detailed study of the emissivity of the collector during operation should be conducted since effects such as aging can increase the emissivity of the absorber over time. However, the plots show how the performance of a standard absorber will drop quicker than the secondary flat reflector's and for higher temperatures of operation, which can be achieved with the use of alternative heat transfer fluids such as molten salts [27]. The secondary flat reflector could be a promising option if the output temperature becomes closer to 550 °C.
However, in the range of temperatures analysed, a shortened secondary flat reflector can be an alternative to standard absorbers. The decrease of the shadow size means higher performance if no misalignments are present even with high flow rates and low emissivities. At a maximum flow rate and low emissivities the increase of performance could be enough to enhance the performance of parabolic troughs, but it is important to consider some additional aspects that could make the shortened secondary reflector have higher performance than the one shown in this work.

Although the increase of pumping power was not considered relevant for the LS3 concentrator, the results shown a very small increase of performance from 1 kg/s to 2.5 kg/s representing around 1 % for the standard absorber. A real plant won't operate at a fixed flow rate since it will need to couple the flow rate to the DNI as the output temperature is desired to be constant and the DNI will vary over time. Maintaining the efficiency of the absorber when the flow rate is changed will help to operate more efficiently power plants and it will help to increase the performance of the plants over time.

As shown in Figure 4.11, the optimal position of the shortened secondary flat reflector absorber is not the focus of the parabola, but a point with a positive y deviation between 15 and 20 mm. This positive positioning can increase the intercept factor by 1.6 %. At maximum flow rate, the maximum performance difference obtained was 1.9 % at 290 °C with an emissivity of 0.1 . This performance of both absorbers at 425 °C was the same. The emissivity is expected to be higher than 0.1 at maximum operation temperature even for the 4th generation Schott absorber, as shown in Figure 5.25. The combination of an optimal positioning of the absorber and the higher emissivity at outlet temperature would allow the performance of the shortened flat reflector to surpass the standard absorber's.

Improvements on the secondary flat reflector can help to decrease the thermal losses further. The emissivity of the secondary flat reflector was considered to be 0.1 , but a proper design of the mirror would decrease it, lowering thermal losses. In the shortened SFR, the shadow was still a 2.15 % of the primary mirror area. As it is fully encapsulated, it can be coated to absorb the flux impacting its upper part, and if it becomes thermally connected to the cylindrical absorber, it would transfer the flux loss in the shadow to the receiver as a flat plate thermal collector.

Although it does not provide a perfectly uniform flux around the absorber, the secondary flat reflector increases the flux uniformity around the absorber. Figure 5.21 shows the temperature profiles in the absorber surface with no wind conditions, considering the minimum and the maximum HTF temperature in a power plant (assumed as 290 and 425 °C) and the minimum and maximum flow rates considered in this work for the LS3 absorber (0.1 and 2.5 kg/s). The uniformity on the absorber surface temperature will reduce thermal stress on the absorber, helping it to improve its durability and decrease the possible bending.
5.6.2 NEP collector

In this case, the primary mirror has a width of 1.2 m and as the DNI was maintained at 800 W/m², the fluxes obtained for both the standard and the secondary flat reflector absorbers were 960 and 926 W/m². The NEP collector was analysed considering its current design which is a non-evacuated absorber with a diameter of 28 mm encapsulated in a glass of 42 mm diameter. Its secondary flat reflector absorber has a diameter of 22.6 mm and it was analysed considering the flat secondary reflector completely encapsulated. The encapsulation of the secondary flat reflector necessitates an increase in the glass diameter, from 42 mm to 51 mm. Two models were considered for the secondary flat reflector absorber, the first one considers the gap above the secondary flat reflector contains air while the second model considers that there is an insulation material $k=0.04$ W/mK that could help to reduce the heat losses. A sketch for both the secondary reflector models is shown in Figure 5.22.
As in the case of the LS3 absorber, Figure 5.23 shows the heat transfer coefficients calculated for the absorber-oil interface for the Petukov approximations. Again, the marks show the points that were considered later as the fluid temperature in the CFD simulations.

In this case it is expected that, as the absorber is not evacuated, the wind will cause a higher thermal loss. The wind heat transfer coefficients were obtained for the different glass dimensions as in the LS3 case. If there is wind velocity, the ambient and the glass temperatures do not have a strong influence on the heat transfer coefficient between glass and the surrounding air, but the glass surface temperature becomes relevant if no wind velocity is considered. The CFD simulations showed a higher glass temperature on the standard absorber model, but as the no wind conditions represents the case with the lowest heat losses on the absorbers, it was decided to maintain the glass temperature on both cases as 50 °C for an ambient temperature of 20 °C. Figure 5.16 shows the heat transfer coefficients for both absorbers with no wind velocity at different glass temperatures while table 5.6 shows the heat transfer coefficients considered in the simulations and calculated with equation 5.6.

### Table 5.6: Heat transfer coefficients calculated for the NEP receivers' glasses. $T_{\text{glass}} = 50$ °C; $T_{\text{amb}} = 20$ °C

<table>
<thead>
<tr>
<th>$V_{\text{wind}}$ (m/s)</th>
<th>$h$ NEP Standard receiver</th>
<th>$h$ NEP SFR receiver</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>7.7</td>
<td>7.3</td>
</tr>
<tr>
<td>2</td>
<td>22.2</td>
<td>20.6</td>
</tr>
<tr>
<td>5</td>
<td>39.0</td>
<td>36.2</td>
</tr>
<tr>
<td>15</td>
<td>77.0</td>
<td>73.6</td>
</tr>
</tbody>
</table>
In this case, as shown in section 4.4, an increase of concentration of 19.7% was achieved with a projected shadow of 3.5%. The intermediate flow rate (0.25 kg/s) is not shown in the figures to improve readability. The results obtained from the CFD simulations are shown in Figures 5.25 and 5.26.

Figure 5.24: Heat transfer coefficient between glass and air for the NEP absorbers; $v_{\text{wind}} = 0 \text{ m/s}$. 
Figure 5.25: Standard and SFR receivers for the NEP primary mirror. a) $\varepsilon=0.1$; $v_{\text{wind}}=0$ m/s. b) $\varepsilon=0.1$; $v_{\text{wind}}=0$ m/s. c) $\varepsilon=0.1$; $v_{\text{wind}}=15$ m/s. d) $\varepsilon=0.1$; $v_{\text{wind}}=15$ m/s.
Enhancing concentration ratio of solar concentrators

The main advantage of the secondary flat reflector shown in the CFD simulations is a better behaviour with the increase of wind velocity as the standard absorber loses a higher performance if wind is considered. The simulations shown that insulation has be included in the secondary flat reflector for non-evacuated absorbers. Even insulated, the inclusion of a secondary flat reflector does only show a performance improvement for temperatures above 250 °C at high flow rates. Except in those rooftop applications that could need low flow rates the inclusion of a secondary mirror does not help to increase the performance of the non-evacuated collectors. The insulation considered had a thermal conductivity of 0.04 W/mK, a better insulation could help to increase the performance of the absorber.

The increase of pumping power in this absorber was relevant even for the standard absorber at the highest flow rate and a lower flow rate is recommended. At 0.5 kg/s the difference of performance between the standard absorber and the non-evacuated one is 2 % in the case of a 0.1 emissivity. A modification of the secondary flat reflector could allow it to use its face normal to the Sun as a flat plate collector would eliminate the 3.5 % losses caused by the shadow increasing its performance.
Chapter 5: CFD of a parabolic trough with secondary flat reflector

5.7 Summary

In this chapter, CFD simulations were conducted to analyse the influence of the flux distribution on a parabolic trough receiver performance and to compare the thermal efficiency of two benchmark parabolic trough absorbers and their modifications with a secondary flat reflector. The benchmark troughs considered were the ET/LS3 primary mirror and its evacuated absorber and the NEP mirror and its non-evacuated absorber. The first one is the most common mirror used for electricity production while the second one is a smaller mirror suitable for rooftop applications.

To analyse the influence of the flux distribution, three different flux profiles were implemented. A realistic profile obtained from ray tracing, a uniform profile and an extreme profile with all the incoming flux impacting a narrow sector of the absorber. In the case of the non-evacuated absorber, the influence of the position of the trough was also considered as the flux profile changes its position respect to a vertical axis and natural convection is influenced by gravity.

In evacuated absorbers it was demonstrated that the influence of the flux is almost negligible at high flow rates, due to the increase on the heat transfer coefficient between the absorber inner interface and the heat transfer fluid, while it becomes relevant at lower flow rates which shown differences bigger than 5 % for the LS3 absorber performance. This difference is augmented when an extreme flux pattern is studied.

Additionally, the temperature profiles around the absorber surface were studied. The thermal stress produced by the temperature gradients caused by irregular fluxes can lead to bending in the tubes and it was shown how more uniform heat flux distributions will minimise the temperature gradients. Again, the thermal gradients obtained were reduced with an increase of flow rate.

The thermal losses were reduced with the inclusion of a flat secondary reflector, as well as the energy input due to the mirror shadow. Despite the loss reductions, and in the range of operation of current power plants, the inclusion of a secondary flat reflector as designed in previous chapter does not improve the performance of standard absorbers in general for both evacuated and non-evacuated absorbers. The lower thermal performance is caused by the amount of energy lost in the shadow. However, for evacuated absorbers there are some scenarios which will make the secondary flat reflector produce higher performance than standard absorbers. Results shown how secondary flat reflector absorbers will increase the performance of parabolic trough if high emissivities are considered in the absorber as will be expected at high temperatures. For applications such as molten salts, which are expected to increase the range of operation of parabolic troughs up to 550 °C the secondary flat reflector would offer a better thermal performance than a standard absorber. An enhancement of energy input by adding a flat plate collector located at the upper surface of the secondary mirror would be a possible solution to further increase standard absorbers performance as the flat plate collector would eliminate the shading effects of the secondary mirror.

As the cause of the thermal performance drop for secondary flat reflectors absorbers is its projected shadow, a shorter secondary flat reflector was also simulated for the evacuated absorber. This secondary flat reflector will produce a lower shadow, that in the original flat reflector was close to 7 % due to the big rim angle of the primary mirror. The reduction of the shadow and its associated optical losses were addressed in the previous chapter. The increase of concentration obtained suggest that the shorter version of the secondary flat reflector can increase the performance of the parabolic troughs in the range of operation of parabolic trough power plants, and this effect will be increased further for higher temperatures.

However, the results obtained for a non-evacuated absorber simulations do not suggest a higher thermal performance at the expected temperature range for those parabolic troughs (150-250 °C). Simulations shown an increase of performance if the upper part of the glass cover is insulated. A different model considering insulation in the upper part of the glass did not show enough thermal improvement to overcome the standard absorber's. The small $\psi$ of the primary mirror causes a high tolerance to misalignments in the secondary flat reflector absorber, as shown in Figure 4.11. A bigger decrease of the absorber and flat mirror
sizes can be attempted in order to increase thermal performance while maintaining a good misalignments tolerance.

Other actions such improving the thermal insulation or the modification of the secondary flat reflector to make it work at its upper surface as a flat plate absorber, which should be then connected thermally to the cylindrical absorber would compensate the loss efficiency of the non-evacuated secondary flat absorbers. Figure 5.27 shows this concept; the thermal connectors can also be selected coated to capture the radiation impacting them.

Figure 5.27: Sketch of the SFR receiver (without glass cover) showing a possible thermal connection if the top face of the SFR is selective coated.

CFD simulations shown a reduction of the temperature differences on the absorber surface, specially at low flow rates. This reduction can be helpful in reducing misalignments increasing the intercept factor of parabolic troughs as addressed in the previous chapter.

The difference in the pumping power required was considered irrelevant for evacuated tubes; although the pumping required is four times bigger in the secondary flat reflector absorber, the maximum pumping power per meter of collector was estimated as estimated to be lower than 5 % of the production of the secondary mirror evacuated absorber at the highest flow rate. The pumping power becomes irrelevant at low flow rates. A decrease of the highest flow rate shown little effect on the thermal performance of secondary flat reflector receivers. However, a thermal performance loss between 0.5 and 1.2 % is shown for the standard absorbers, depending on wind velocity and absorber emissivity when the flow rate was reduced. The capability of reducing the flow rate without losing thermal efficiency would help to maximise production in a yearly basis as power plants need to maintain a constant output with variable DNI.

In the non-evacuated absorber, the maximum pumping power required estimated was close to 10 % for the standard absorber and 30 % for the secondary flat reflector. This suggest that the maximum flow rate studied is not effective, despite being considered in power plants which operate with this collector [137]. A reduction of the flow rate causes a lower difference of performance between the secondary flat reflector and the standard absorber, and the suggestions proposed to enhance the thermal performance of the secondary flat reflector could make its thermal performance to be higher than the standard one.

The major conclusions extracted from the CFD simulations addressed in this chapter are:

- There is a dependence on the flux distribution for the thermal performance of parabolic trough receivers, but it can be compensated with increases of the flow rate.

- Secondary flat reflectors shortened to maximise the intercept factor, despite missing some radiation, are an effective way to increase thermal performance of evacuated absorbers, specially if future applications lead to working temperatures above 400 °C.

- Secondary flat reflectors can improve the thermal performance of non-evacuated tubes, but alternatives
to increase its thermal performance such as better insulation or a flat plate absorber on top the secondary mirror and connected to the cylindrical absorber should be included to minimise the effects of the shadow.

- Secondary flat reflectors will reduce the dependence of the flow rate in the thermal performance of the collector. This lower dependence will improve energy production of a power plant along the year.
6.1 Introduction

In the previous chapter, the performance of parabolic-trough absorbers with and without secondary reflectors was predicted by CFD simulations. An experimental performance analysis for a non-evacuated absorber with and without secondary mirror was conducted to compare the thermal performance of both absorbers in real conditions. In this chapter, the results of the experiment are shown, and a CFD simulation considering the conditions of the tests (DNI, wind, flow rate, HTF, etc.) is added to prove the validity of the CFD model. This section includes the description of the experimental facility, the manufacturing process of the absorbers, and the set up of the experiment, the methodology and the analysis of experimental results.

The production of both absorbers was conducted within the university facilities due to the impossibility of acquiring a coated absorber of appropriate dimensions for the flat secondary reflector receiver since the tube diameter does not fit any available coated receiver in the market. The two prototypes were attached to a commercial primary mirror, with arms adapted to fit absorbers of different sizes and to locate them in their appropriate position. This is needed as the SFR receiver has not the centre of the cylindrical absorber located at the parabola focus, as it was demonstrated in section 4.2.3.

The prototypes were manufactured at the university facilities and a lower performance for both the absorbers if compared with commercially coated absorbers can be expected, mainly due to the difficulty of achieving a high-quality selective coating. However, the main purpose of this work is to analyse the performance difference between similar receivers when a secondary reflector is included, and a low efficiency won't affect the experiment conclusions.

The experiment was conducted at the Laboratory for Innovative Fluid Thermal Systems (LIFTS) located in the RMIT city campus in Carlton (Australia), 37°48'20" S, 144°57'56" E. The primary mirror was located with an east-west orientation that assured a normal solar incidence at midday every day of the year. As it will be explained in detail during this section, to avoid a significant influence of the end of collector losses, the tests were conducted at midday, from 11 am to 1 pm.

An outdoors solar temperature and flow controlled liquid delivery system existing in the RMIT University was used to conduct the experiment. The facility includes a closed loop hydraulic circuit to recirculate a heat transfer fluid and a heat and cooling system was installed in the circuit to control the absorber's input temperature. Two pyranometers mounted on a two-axis tracker allow the instantaneous DNI to be measured. All the sensors are connected to a data acquisition system; that records every signal with a sample rate of 100 samples/s. For the performance analysis, the absorbers were installed on a NEP polytrough-1200 primary mirror, which dimensions are listed in Table 2.2.

6.2 The experiment set up

The primary mirror was installed with an east-west orientation and it was planned to track the Sun using its commercial tracking system. As it will be described later, a bias error appeared on the tracking system when the system was working for an extended time and for that reason and to assure a correct absorber position, a start-up of the tracking system was conducted before each measurement time frame instead of using the tracking system in a continuous mode. Figure 6.1 shows the parabolic mirror with one of the absorbers installed during one of the tests.
The input and output absorber temperatures were measured with 4-wire DIN Class AA thermocouples which were calibrated within the facilities. The temperature measurements maximum deviation expected was found to be of ±0.125 °C. A heater and a cooling system composed of a liquid-liquid heat exchanger and a variable speed fan were installed in the fluid circuit to control the input temperature of the absorber by a proportional-integral-derivative (PID) controller commanded by the reading of a third thermocouple placed at the heater outlet. The mass flow rate was achieved by a variable speed pump which allowed fluid flow rates between 0.05 and 0.2 kg/s and was PID controlled by the measurement obtained by a micro Coriolis effect flow meter with a total uncertainty of ±0.5 %.

An anemometer was mounted close to the collector to record the wind speed during the tests and two class A Middleton EQ08 pyranometers installed on a two-axes tracker were used to obtain the total and diffuse radiation in a free of shadows location close to the primary mirror. One of the pyranometers recorded the total radiation received in a plane normal to the Sun radiation while the other obtained the diffuse radiation. To avoid DNI reaching this second pyranometer, a shading disk with dimensions calculated to project a shadow on the sensor of equal size to the direct beam of radiation, was installed on top of it. The spacing between shading disk and the sensor was of 50cm to allow diffuse radiation reach the pyranometer. Figure 6.2 shows both pyranometers during one of the tests conducted. The effectiveness of the shadow disk can be appreciated as the glare is removed in the shadowed pyranometer but no extra shadow is projected on the pyranometer box.
Solar radiation is a critical measurement for the analysis of the tests results. For that reason, a calibration test was conducted in addition to the uncertainty analysis which is described in next section. Both pyranometers were calibrated by measuring the total radiation at different planes and comparing their measurements against the ones obtained from a certified pyranometer with an uncertainty of 4%. The analysis was conducted for both absorbers under low solar radiation (cloudy day) and at high solar radiation (sunny day) conditions. The two-axis tracker was moved along the east-west axis to make different normal irradiations impacting the sensors during the tests. Figure 6.3 shows the comparison test against the calibrated pyranometer conducted for each sensor; the DNI can be obtained by subtracting both pyranometer measurements.

The calibration test showed a good agreement between the pyranometers at low radiation, and, at high radiation, the agreement between the calibrated pyranometer and the one used to record total radiation was found to be within the range of uncertainty of the calibrated pyranometer. However, there is a mismatch between measurements for the diffuse pyranometer and the calibrated one at high radiation fluxes. As the diffuse pyranometer was going to be shadowed by its shading disk and low radiation readings were expected from this sensor, it was considered to offer a sufficient accuracy to conduct the experiment.

The experiments were controlled by a SCADA developed in LabVIEW which recorded continuously every sensor measurement. The SCADA was installed in a NI9188 Compact DAQ controller, and the experiment was controlled by a computer connected to the DAQ which allowed the user to select the relevant variables to conduct the experiment. Mass flow rate, collector input temperature and the recording time were defined by the user. The controller recorded every sensor measurement with a sampling rate of 100 samples/second (NI9219 module) and calculated the average value for the selected record time. Each set temperature was analysed after reaching a steady state input on the absorber for at least five minutes. The mean value of every variable was recorded every 10 seconds taking into account all the samples recorded in the SCADA. By recording every 10 seconds during 5 minutes a sample size of at least 30 samples was guaranteed, which leads to a confidence level of a 95% on the measurements as it will be demonstrated in the next section. The real sample size was 1000 times bigger since the sampling rate was 100 samples per second. However, the standard deviation of the samples in every interval was not recorded and for that reason, the sample size is considered 30. Figure 6.4 shows the SCADA control panel during one of the experiments.
Uncertainty analysis

The performance of the thermal absorbers was calculated from the measurements obtained by different sensors in the experimental facility. There is an error associated with any experimental measurement, and its magnitude, known as uncertainty, has to be calculated to evaluate the confidence level of the experiment.

Uncertainty depends on several aspects of the experiment, such as instrumentation, number of measurements, repeatability of results, etc. All those aspects can be classified in systematic (bias) errors and random (precision) errors. The combination of every possible bias and precision error will contribute to a mismatch between measurement and real value that has to be weighted to report the credibility of the results. The uncertainty analysis conducted in this work was developed using the methodology proposed by Coleman [147].

The total uncertainty of a variable measured directly, $U_r$, can be calculated from its bias error ($B_r$) and its precision error ($P_r$) as their root-sum-square (RSS)

$$U_r = \sqrt{B_r^2 + P_r^2}$$  \hspace{1cm} \text{(Equation 6.1)}

The systematic error cannot be removed and it has to be evaluated from tolerances, data sheets, instrument accuracy, etc. The bias errors can be provided from the manufacturer or they can be a combination of bias errors associated with a certain measurement. As an example, the position of the mirror tracker does not only depend on the mechanical parts accuracy or misalignments but also on its algorithm that calculates the position and the refresh rate, as the position of the Sun changes continuously. For a direct measurement `$x$' that can be affected by `$i$' bias errors, the total bias can be calculated as:

$$B_x = \sqrt{\sum_{j=1}^{k} B_j^2}$$  \hspace{1cm} \text{(Equation 6.2)}
The random error is caused by unpredictable phenomena. Therefore, it cannot be measured. An infinite number of measurements will sit along a normal distribution. This allows decreasing random errors by taking several measurements and evaluating the mean value and the standard deviation of the sample recorded. The standard deviation represents the difference between the individual measurements taken respect to the mean value. A lower standard deviation value will be a reflection of a lower precision error. For a sample size of \( n \) measurements, the standard deviation can be obtained with equation 6.3 and the precision error of the mean value is calculated with equation 6.4.

\[
S_x = \left \{ \frac{1}{n} \sum_{i=1}^{n} (X_i - \bar{X})^2 \right \} \quad \text{(Equation 6.3)}
\]

\[
\pm P_{\bar{X}} = t \frac{S_x}{\sqrt{n}} \quad \text{(Equation 6.4)}
\]

The value of \( t \) is a scaling factor that can be obtained from the Student's t-distribution and it shows how the sample size approaches a normal distribution. As higher the \( t \)-value is, the closer to a normal distribution is the sample. For the sample size of this experiment, it can be assumed \( t=2 \). This \( t \)-value gives a confidence level of 95\% which means that it is expected that the sample measured is representative of the real measurement with a 95\% confidence.

In most of the experiments several direct measurements have to be taken to calculate indirect variables as in the case of the heat transferred to the fluid in the parabolic-trough which cannot be measured directly. The general expression for an indirect result is shown in equation 6.5 in which \( r \) is the desired result and \( X_1 \) to \( X_n \) the measurable variables necessary to calculate the result.

\[
r_i = r_i(X_1, X_2, X_3, \ldots, X_{n-1}, X_n) \quad \text{(Equation 6.5)}
\]

If the uncertainties of the '\( n \)' direct measurements are known, the total uncertainty for the calculated measurement can be calculated as shown in equation 6.2:

\[
U_r^2 = \sum_{i=1}^{j} \left ( \frac{\delta r}{\delta X_i} \right )^2 U_{X_i}^2 \quad \text{(Equation 6.6)}
\]

The uncertainties associated with the relevant variables measured and calculated for this chapter are resumed in Table 6.1. A detailed analysis of every individual variable and the complete calculation of their associated uncertainties are developed in Appendix F.
6.4 Experiment description

The main objective of this chapter is to evaluate the change of performance of a parabolic trough when the original absorber is substituted by a flat secondary reflector receiver with its secondary mirror encapsulated together with the cylindrical receiver. For this purpose, a NEP polytrough 1200 with its commercial tracking system was installed on the experimental facility available in RMIT University. The polytrough primary mirror is a parabolic absorber with a focal distance of 647.5 mm, a width of 1.2m and a length of 2 m. The reflective surface is composed of MIRO-SUN 90, which reflective characteristics can be found in the website of the manufacturer. The reflectivity of MIRO-SUN 90 is around 90 %, although there are variations depending on the wavelength and the angle of incidence of the solar energy [148]. However, the assumption that the reflectivity of the mirror is 90 % won't affect the comparison between receivers and will increase simplicity.

6.4.1 The prototypes

Two different absorbers were built for the thermal comparison conducted in this chapter. The first one, a standard absorber, has the same dimensions that the commercial NEP polytrough 1200 absorber, with an external diameter of 28 mm. The second one incorporates a secondary flat reflector with a width of 42 mm, calculated with the methodology developed in section 4.2.2, and an absorber with an external diameter of 22 mm which was the most approximate available commercial steel tube dimension to the one obtained in the optics chapter. However, there is not any commercially coated absorber matching those dimensions, and a steel tube was coated with a selective coating product (Solkote Hi/Sorb II) [149].

The emissivity and the absorptivity of Solkote range from 0.2 to 0.49 and from 0.88 to 0.94 respectively depending on the thickness, substrate and application method. Due to the big range of both parameters, especially the emissivity, and to obtain similar properties in both absorbers, it was decided to coat both of them to ensure a fair comparison. The coating was done manually with an atomisation spray, and a natural cure was applied due to the impossibility of getting an oven of appropriate dimensions. The absorbers were coated manually with a atomisation spray due to the impossibility of getting specialised machinery which

| Table 6.1: Uncertainties of the relevant variables used in the experiment |
|-----------------|-----------------|
| Primary variables | ±0.0005 kg/s |
| Flow rate        | ±0.125 K       |
| Temperature      | ±54.83 W/m²    |
| Solar radiation  | ±0.155 m/s     |
| Wind speed       | ±0.0025 h      |
| Mirror length    | ±0.005 m       |
| Mirror width     | ±0.002 m       |

<table>
<thead>
<tr>
<th>Calculated variables</th>
<th>±0.0375°</th>
</tr>
</thead>
<tbody>
<tr>
<td>Incidence angle</td>
<td>±77.54 W/m²</td>
</tr>
<tr>
<td>DNI</td>
<td>±67.17 W/m²</td>
</tr>
<tr>
<td>Effective area</td>
<td>±0.045 m²</td>
</tr>
<tr>
<td>ΔT</td>
<td>±0.177 K</td>
</tr>
<tr>
<td>Heat transferred</td>
<td>±111.16 W</td>
</tr>
<tr>
<td>Incoming energy</td>
<td>±172.40 W</td>
</tr>
<tr>
<td>Thermal Performance</td>
<td>±8.5 %</td>
</tr>
</tbody>
</table>
would assure an uniform coating. Due to the manual coating, imperfections in the coating were expected to appear, and even a visual inspection of the absorbers showed irregularities on their surface.

As the emissivity of the coating is a critical parameter on the collector performance and as even with an appropriate coating process the range of emissivities of Solkote has a wide variation, six samples, consisting of small flat steel plates coated with the selective paint were tested to obtain their emissivities. Two samples were coated with a brush, trying to obtain a more uniform coating to compare the effects of the coating irregularities on the samples' emissivities. Four samples were coated at the same time that the cylindrical absorbers, with the atomisation spray, as each absorber was coated separately, samples 3 and 4 were coated at the same time than the standard receiver, while 5 and 6 were coated with the SFR receiver. Figure 6.5 shows the selected samples (1 and 2 are the ones coated with a brush) while Figure 6.6 shows the results obtained for the emissivity obtained by testing the samples in a Perkin Elmer Spectrum 1 series FT-IR spectrometer.

![Figure 6.5: Flat samples measured in the spectrometer.](image)

![Figure 6.6: Flat samples' emissivities, 1 and 2 brush coated, 3-6 spray coated.](image)
Despite the optical irregularities, the emissivity obtained for the sprayed samples was lower than the one obtained with a brush, and the repeatability of the results was high enough to consider that the absorbers' emissivities can be considered equal. As expected, the emissivity of the samples ranged between 0.2 and 0.49. The spectral blackbody emissivity power $E_{b\lambda}$ was developed by Max Planck, and its relationship, represented in equation 6.7 allows to calculate the emissivity of the absorber along the whole spectrum [75].

$$E_{b\lambda}(\lambda, T) = \frac{C_1}{\lambda^5 \left[ e^{(C_2/\lambda T) - 1} \right]} \quad \text{(Equation 6.7)}$$

The blackbody was evaluated at a temperature of 353 K (80 °C) and $C_1$ and $C_2$ are constants found with equations 6.8 and 6.9:

$$C_1 = 2\pi h c_0^2 = 3.74177 \times 10^8 \text{(W/μm}^4\text{m}^2\text{)} \quad \text{(Equation 6.8)}$$
$$C_2 = h c_0 / k_b = 1.43878 \times 10^4 \text{(μmK)} \quad \text{(Equation 6.9)}$$

$c_0$ is the speed of light, $k_b$ is the Boltzmann's constant and $h_b$ is the Planck's constant. Integrating the blackbody power and the product of emissivity and blackbody power as shown in equation 6.10, the average emissivity of the absorber at the specified temperature was calculated for the four samples coated at the same time that the absorbers (samples 3 to 6).

$$\epsilon = \frac{\int_{\lambda_1}^{\lambda_2} \epsilon(\lambda) E_{b\lambda}(\lambda, T) \, d\lambda}{\int_{\lambda_1}^{\lambda_2} E_{b\lambda}(\lambda, T) \, d\lambda} \quad \text{(Equation 6.10)}$$

The average emissivities obtained for the four spray coated samples are represented in table 6.2.

<table>
<thead>
<tr>
<th>Sample</th>
<th>$\epsilon$</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>0.23</td>
</tr>
<tr>
<td>4</td>
<td>0.27</td>
</tr>
<tr>
<td>5</td>
<td>0.24</td>
</tr>
<tr>
<td>6</td>
<td>0.24</td>
</tr>
</tbody>
</table>

In the CFD simulations conducted to compare with the experimental results, an emissivity of 0.25 was considered.

The glass covers were built from glass tubes with a length of 1.5 m [150] and welded in the workshop of the University of Melbourne. The welded area could produce an optical loss due to a loss of glass properties in the welded area as well as a curve area in the junction of the two welded tubes that can increase refraction losses. This optical loss was not taken into consideration as it was expected to be similar in both absorbers and it won’t affect the thermal efficiency comparison. In the standard receiver, the glass cover outer diameter was chosen the same as the commercial product for the standard receiver (42 mm). For the secondary flat reflector receiver, a 54mm outer diameter glass was chosen, which allowed encapsulating the secondary mirror maintaining the distance between absorber and glass cover of the original absorber.

The secondary flat reflector consisted of a polished flat aluminium reflective surface of the same material than the primary mirror. Eight aluminium supports with an insulation layer on top were added to the absorber tube to attach the secondary mirror to it and to avoid thermal conduction between them. The supports were coated as well with the selective paint to increase their absorptivity and minimise optical losses, the thickness of the insulation layer was calculated to match the mirror-absorber gap as calculated in
section 4.2.2. Figure 6.7 shows the assembling of one of the flat secondary mirrors on top of the thermal receiver and a visual inspection of the gap between mirror and receiver. Figure 6.8 shows the SFR receiver installed on the primary mirror during one of the tests conducted.

Figure 6.7: Assembling of the flat secondary mirror.

Figure 6.8: SFR receiver installed on the NEP primary mirror.

6.4.2 Thermal performance calculation

The thermal efficiency of a solar collector can be calculated as the ratio between the heat transferred to the heat transfer fluid and the energy entering the primary mirror area, which depends on the Sun position.

To calculate the heat transferred to the fluid, it is necessary to know the mass flow rate and the inlet and output temperatures of the fluid inside the absorber. To obtain the temperature readings, the RTDs were installed within a probe as near as possible to the absorber inlet and outlet with “T” junctions which were insulated to avoid heat losses. As the fluid regime is turbulent (Re>1.5x10^5 in both absorbers) the temperature of the fluid can be considered uniform for a section of the tube. Assuming a constant specific heat for water in the range of temperature studied, the heat transfer can be calculated as:

\[ \dot{Q} = \dot{m}C_p \Delta T. \]  

(Equation 6.11)
As the primary mirror tracks the Sun in one axis, it appears an end of collector loss caused by the angle formed by the DNI radiation and the primary mirror area. This loss depends on the mirror orientation and the Sun position and will cause the “effective area” of the parabolic trough to be lower than the primary mirror area. This loss only appears once in a row of collectors, and, if several mirrors are connected in series, the end of collector losses decrease, as the loss at the end of one mirror can be recovered by the adjacent one.

A sketch of those losses is shown in figure 6.9, in which the grey area of the parabolic mirror represents the area which reflected rays would be missed as the reflection goes further than the length of the absorber. End of collector losses become relevant with short collectors and high latitudes [134] and they depend on the average distance from the mirror to the absorber and the solar incidence angle, which depends on the position of the Sun. There are several methods to calculate the Sun position; equation 6.12 for a PTC with the axis oriented East-West and with equation 6.13 for a PTC in which the axis is oriented North-South [6, 42, 151].

![Figure 6.9: End of collector losses.](image)

\[
\zeta = \arccos \left( \sqrt{1 + \cos^2 \delta \left( \cos^2 \omega_e - 1 \right)} \right)
\]  
\quad \text{(Equation 6.12)}

\[
\zeta = \arccos \left( \cos \delta \sqrt{\cos \lambda_T \cos \omega_e + \tan \delta \sin \lambda}^2 + \sin^2 \omega_e \right)
\]  

\quad \text{(Equation 6.13)}

For an East-West oriented trough, the angle of incidence depends only on declination and solar hour angle while in a North-South oriented depends also on the latitude, \(\lambda_T\). The declination angle, \(\delta\), can be obtained as a function of the day and year analysed, there are different approximations [48, 152]. In this work, the declination angle was calculated for a given day of the year, \(N\), as:

\[
\delta = \arcsin \left( \sin (\delta_0) \sin (N - 80) \frac{360}{365.25} \right),
\]  
\quad \text{(Equation 6.14)}

considering that the initial declination angle is obtained as in equation 6.14 that depends on the year, \(Y\):

\[
\delta_0 = 23.4523 - \left( \frac{0.46845 (Y - 1900)}{3600} \right)
\]  
\quad \text{(Equation 6.15)}

The solar hour angle, \(\omega_e\), is defined as 0 at midday and as the Earth completes a rotation in 24 hours, the solar angle (in degrees) can be obtained from the solar time, \(ST\), defined in hours as:

\[
\omega_e = 12 - (15 \times ST),
\]  
\quad \text{(Equation 6.16)}

and if the incidence angle is known, the end of collector losses can be calculated if the width of the primary mirror and the average distance between absorber and mirror for trough cross-section are known. The average distance of a mirror cross-section to the absorber can be calculated with equation 6.13 while the effective area (the primary mirror area minus the end of collector loss) can be obtained from equation 6.11.
Enhancing concentration ratio of solar concentrators

\[
F_m = F + \frac{FW^2}{48F^2}
\]  
(Equation 6.17)

\[
A_c = A - WF_m \tan \zeta
\]  
(Equation 6.18)

The end of collector losses is a geometrical loss that occurs in every parabolic trough and it is time dependant as demonstrated. As the inclusion of a secondary reflector won't have any impact on end of collector losses, they were removed from the efficiency calculations and the efficiency was calculated taking into consideration the radiation entering the effective area of the parabolic trough. Calculating the effective area allows to compare both thermal collectors without having a time dependence, and then it won't matter the time of the day used to obtain the temperature measurements. However, a decrease of effective area implies a lower temperature increase between inlet and outlet and a bigger uncertainty in the temperature measurements. For that reason, all the tests were conducted in the central hours of the day, from 11 am to 1 pm, in which \( \omega_p \) is low and the end of collector losses minimum.

The one-axis tracker adds one more energy loss to the parabolic trough, also time dependant. This loss is due to the fact that the aperture area of the primary mirror is not normal to the solar radiation. As the DNI was obtained with two pyranometers installed in a two axis tracker, a correction factor on the DNI has to be added to calculate the incoming energy on the primary mirror, in the case studied, East-West axis, that correction factor is calculated from the hour angle as

\[
\text{DNI}_{\text{eff}} = \text{DNI} \cos \omega_p
\]  
(Equation 6.19)

while in a North-South oriented collector the effective DNI can be calculated from the elevation angle of the Sun, \( \alpha_s \), which can be calculated as a function of latitude, declination and hour angle.

\[
\text{DNI}_{\text{eff}} = \text{DNI} \cos \alpha_s
\]  
(Equation 6.20)

The elevation angle can be obtained with known formulae which can be found in literature [6]. However, as the orientation of the collector used in this work is not North-South, the equations to calculate this angle are not shown in this section. Finally, if the incoming energy and the heat transferred to the fluid are known, the performance of the absorber can be calculated as the ratio between them, as shown in equation 6.6.

\[
\eta = \frac{\dot{Q}}{\text{DNI}_{\text{eff}} A_{\text{eff}}}
\]  
(Equation 6.21)

6.5 Results

The experiments were conducted in sets of 5 to 10 minutes. An inlet set temperature was defined and the measurements were recorded once a steady state input was achieved. The necessity of obtaining a fixed set temperature made impossible to measure under a certain inlet temperature since the heat dissipation was not high enough to guarantee a stable inlet temperature. Figure 6.10 shows a test conducted over time, in which the heater was not enabled and the heat transferred to the fluid by the solar collector was partially dissipated at low temperatures, making the inlet temperature increase over time. The test allowed choosing the minimum set inlet temperature that permits achieving stable conditions.
The maximum temperature in the outlet was decided to be set below 95 °C to avoid the possibility of water starting to boil, which could produce a latent heat exchange that would lead to an erroneous calculation of the thermal performance. Another major issue observed during the tests was a bias error on the primary mirror tracker over time. If the tracker was set in automatic mode, the tracking was initially correct but a misalignment appeared over time, making heat gained to drop, despite a continuous DNI. For that reason it was decided to set the tracker in a 'fixed' mode. Once that a set temperature was fixed, the tracker was set up to assure a correct tracking (since the tracking error appeared over time) and measurements were conducted then for elapsed times of five minutes. The sun altitude at midday won't change significantly over 5 minutes, and the acceptance angle of the collector is big enough to avoid a drop of optical efficiency due to the absorber being fixed for that short time. Figure 6.11 shows how, if the automatic tracking mode was activated, the bias error would produce a drop in the outlet temperature, due to part of the radiation being missed. Every time that the drop was detected during this test, the tracker was reactivated to remove the bias error. It is possible to see how, after a tracking correction, the absorber will maintain a correct tracking for a time longer than the test time intervals (5 min).
The measurements were not recorded during the time between set points, in which the heater was activated, and the cooling deactivated to increase the inlet temperature. Once the next steady state was achieved at the collector inlet and before recording the measurements, the tracker was re-set to remove its bias error, and a visual inspection of the absorber was conducted to check the correct tracking. Figure 6.12 shows the reflection of the absorber on the primary mirror, which allows viewing of the concentrated flux on the standard absorber during one of the measurements. Figure 6.13 shows the measurements obtained during two different set points. It can be checked how the inlet temperature remains stable, so an appropriate number of measurements could be taken to calculate the performance of the collectors at such point.
Chapter 6: Experimental comparison of a PTC absorber with and without SFR

For both prototypes, standard and flat secondary reflector, two different tests, conducted in different days are shown in Figure 6.14. The performance is calculated with the methodology developed in section 6.4.2.

The CFD simulations were conducted using the models constructed in the previous chapter. The heat transfer coefficient of water was evaluated at the mean temperature of each set point. An emissivity of 0.25 was used for the absorber and an optical efficiency of 0.9 for both the glass cover and the mirror were used. Finally, an ambient temperature of 15 °C (the average ambient temperature during the tests oscillated between 12 and 16 °C) and a constant wind velocity of 2 m/s were assumed in the simulations. The heat transfers coefficients for the glass cover was calculated with equation 5.6. The result obtained was 22.16 W/m²K in the case of the standard absorber and 20.58 W/m²K for the SFR.

Table 6.3 shows the heat transfer coefficient assumed for the inner surface of the absorber (HTF-absorber interface) considering $m = 0.14$ kg/s and obtained from equations 5.7 to 5.9.
The agreement between CFD simulations and the tests conducted on the secondary flat receiver is acceptable, although the simulations predicted around a 3% higher performance. In the second SFR test, the first measurement falls off the expected range, and the performance at that low-temperature set point is lower than the next set points considered during that experiment. The analysis of the wind velocities showed a higher wind velocity during the two first set points, but it is very unlikely to be the reason for that lower performance. A more likely explanation could be a wrong tracker set-up during those measurements, leading to higher optical losses.

However, there is not enough agreement between the standard receiver CFD simulations and the experimental measurements. Even more, the thermal performance of the standard receiver is lower at every set point if compared with the secondary flat reflector. After the evaluation of the results, there are some possible explanations for that low performance.

The primary mirror is a commercially available parabolic-trough and to fix the prototypes the arms of the mirror were modified with the addition of a metallic clamp, used to fix the absorbers. When the absorbers were fixed, insulation material was inserted between the clamps and the absorber, to reduce thermal conduction between the absorber end metallic supports and the arms of the mirror. As the end supports were of different sizes for different mirrors, the clamps were more tightened for the standard receiver, as its glass diameter is lower, and as a result of that it is possible that the insulation material was more compressed, resulting in a higher thermal conduction from the absorber to the mirror. This conductivity would also explain the rapid drop of thermal performance of both receivers if compared with the simulations as a heat-sink effect would appear, increasing the losses at high temperatures.

After one of the tests with the standard absorber, it was noticed some condensation within the glass cover in the area next to the outlet of the absorber. Condensation will increase the scattering of the glass and increase reflection losses, decreasing optical efficiency. It is possible that humidity in the laboratory, which is located in a room next to the rooftop facility was high during the assembling. That would make the standard absorber contain a higher percentage of water within its encapsulation. It is also possible that the O-ring used in that receiver to seal the glass was not tight enough, leading to some heat exchange of air flowing in and out the enclosure. However, condensation was detected in a small area of the receiver and it is not expected to cause this significant difference.

But the evaluated as the most plausible reason is that the optical aberrations of the mirror were high enough to make the concentrator miss part of the rays as described in section 4.5. An inspection of the primary mirror shown some damage on its surface, and it could be possible that this damage made the mirror have a higher scattering than the expected. As shown in Figure 4.11, the sensitivity to misalignments of the NEP mirror with a standard absorber is much higher than if a flat reflector is inserted and it could explain the difference between both performances.

A ray tracing simulation conducted for both absorbers. It was assumed a higher scattering than expected (0.5° compared to the 0.23° expected) and 2.5 mm misalignments in vertical and horizontal location and a 0.1° tracking error. The results obtained showed an approximately 7% decrease in the expected optical performance.

### Table 6.3: Heat transfer coefficients used in the CFD experiment validation simulations

<table>
<thead>
<tr>
<th>HTF Temperature (°C)</th>
<th>$h_{\text{fluid}}$ Standard Absorber (W/m²K)</th>
<th>$h_{\text{fluid}}$ SFR Absorber (W/m²K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>65</td>
<td>2338</td>
<td>3632</td>
</tr>
<tr>
<td>70</td>
<td>2396</td>
<td>3720</td>
</tr>
<tr>
<td>75</td>
<td>2419</td>
<td>3802</td>
</tr>
<tr>
<td>80</td>
<td>2499</td>
<td>3878</td>
</tr>
<tr>
<td>85</td>
<td>2546</td>
<td>3950</td>
</tr>
</tbody>
</table>
Chapter 6: Experimental comparison of a PTC absorber with and without SFR

efficiency on the standard absorber, which is similar to the performance difference observed in the experiment, while the secondary flat reflector receiver optical loss was 1%. Although those possible misalignments were impossible to quantify, this ray tracing simulation reinforces the hypothesis of the misalignments during the tests.

6.6 Summary

Two prototypes, a standard, and a secondary flat reflector receiver, were built in the university facilities to compare the thermal performance of the receivers when a secondary flat mirror is inserted. The emissivity of the prototypes was expected to be higher than in commercial absorbers, and it was measured on flat samples coated with the same selective paint at the same time that the prototypes. Both receivers were tested on a two-meters long commercial parabolic trough, but its tracker shown a bias error over time. For that reason, time frames of 5 to 10 minutes were elapsed for each set point analysed which led to sample sizes bigger than 30 samples per set point.

The heat transfer fluid used was water, and due to the impossibility of pressurising the prototypes built, a maximum analysis temperature of 95 °C was considered to avoid evaporation and latent heat transfer. The minimum set temperature was evaluated as the minimum temperature that allowed the cooling system of the test facility to cool down the heat transfer fluid rapidly enough to assure a constant inlet temperature.

One of the main issues in measuring the thermal performance of the prototypes was the impossibility of achieving a high-temperature difference between inlet and outlet, as the mass flow rate had to be set high enough to assure turbulent flow to enhance heat transfer. Temperature increases around 2 °C were achieved, that led to high uncertainties in the measurements due to the accuracy of the temperature sensors. A complete uncertainty analysis was conducted to evaluate the uncertainties of the measurements taken.

The performance obtained for the SFR receiver was close to 60 %, and CFD simulations conducted under similar conductions shown an acceptable agreement. However, a more rapid drop in the performance for higher temperatures was obtained for the measured prototype if compared with the simulations. The most likely reason was an undesired thermal conduction between the absorber end and the arm of the primary mirror. The commercial primary mirror was adapted to support different absorbers, and although thermal insulation was installed between the absorber extremes and the clamps used to fix the receivers to the mirror, tightening the clamp made the insulation layer to become narrower, increasing thermal conduction to the arm.

The performance obtained for the standard receiver was lower than the one expected by simulations and even lower than the one obtained for the secondary flat reflector. Two possibilities were hypothesised to explain this phenomenon. Condensation was detected on one of the absorber's ends at the end of one of the test, and it is possible that humidity conditions in the lab at the time of assembling the receiver caused a higher percentage of water inside the enclosure. Evaporation of this water caused the condensation, that decreased the transparency of the glass and increased optical losses. A visual inspection of the primary mirror discovered some damage on its surface which could led to higher scattering and lower reflectivity. Ray tracing simulations assuming a higher scattering that the one predicted and some small misalignments predicted up to a 7 % thermal performance drop for the standard absorber, while the SFR would drop by 1 % under the same conditions. The combination of this two possibilities could explain the mismatch observed in the tests. These hypotheses would offer practical proof of how a secondary flat reflector could behave better under real conditions.

For the aforementioned reasons, the performance tests of the standard and secondary flat receivers was not completely satisfactory. Several improvements should be addressed in the future to obtain a more accurate evaluation. The main improvements to be implemented are:
- To improve experimental uncertainty, by reducing the DNI and temperature sensors uncertainty.

- To test the prototypes with a longer primary mirror, to reduce end of collector losses influence and to increase the temperature difference between inlet and outlet which will reduce temperature measurements uncertainties.

- To build the thermal prototypes with a commercially coated absorber (the absorbers were coated by manual spray due to the impossibility of acquiring commercial receivers of the required dimensions).

- To improve the sealing of the prototypes by manufacturing an end support with a low thermal conductivity material that could bear high temperatures. By doing this, the clamps used to fix the prototypes to the primary mirror would be clamped directly to a non-conductive material, reducing thermal conduction.

- To add pressurised fittings to the inlet and the outlet of the collector, that would allow to test the prototypes at higher temperatures and to adapt the fittings of the thermal rig to work with pressurised water.

- To conduct a flux map on the focal plane to assure that the tracking is appropriate and no energy is missed.

- Improve the tracking accuracy, recalibrate the tracking system to remove bias error.

However, the main conclusion of the experiment is that inserting the flat secondary reflector onto a thermal receiver for parabolic troughs is achievable and that in a real scenario, the highest tolerance to misalignments can make the thermal efficiency of a parabolic trough being enhanced by the inclusion of the second-stage mirror.
Chapter 7: Conclusions and future work

7.1 Conclusions

In this thesis, a new approach to the ideal maximum concentration of parabolic concentrators was developed. It was taken into consideration that there is not need to effectively reflect all the solar radiation to the receiver. This new approach makes the optical efficiency of solar concentrators drop, but concentration ratios up to the thermodynamics limit can be reached with ideal parabolas. Increasing the concentration ratio of solar collectors could increase the exergy level at the output of concentrators allowing to increase the efficiency of the energy conversions to transform solar energy onto electricity or heat. If this increase of concentration is significant in comparison with the drop of optical efficiency, the efficiency of parabolic troughs with less than 100% optical efficiency could be higher than that obtained in classic concentrators.

The first approach to concentration limits of parabolic concentrators that are not 100% optical efficient was conducted considering a cross-section simplification which has been used extensively in the literature. This simplification states that the study of a cross-section on the symmetry plane of the concentrators is enough to calculate their concentration ratio. Applying the edge-ray theorem and maintaining the etendue of the solar radiation allows the design of concentrators. The results obtained with this methodology showed an increase of concentration of parabolic concentrators up to the previously stated thermodynamics limits in three cases, the parabolic dish with spherical receiver, and the parabolic trough with both a planar and a cylindrical receiver. However, the limits obtained for a parabolic dish with a planar circular receiver did not reach the thermodynamics levels, and its new limit was found at an 80% of the thermodynamics limit.

A second approach was conducted taking into consideration the real projection of the rays reflected in the concentrators, avoiding then the cross-section simplification. For a parabolic dish with a spherical receiver the concentration calculated with this method matched the thermodynamics levels, as it did with the simplification. That was expected since the thermodynamics limit of 3D concentrators cannot be surpassed, and as explained in section 3.5.1 this scenario is completely symmetrical. As a result of this symmetry, any ray with a given etendue will be received on a point in the cross-sectional plane at the focus equally distant to the focus since there is always a section of the absorber normal to the direction of the reflected bundle. For the parabolic dish with a planar circular receiver the concentration limits found were closer to the thermodynamics limits when the simplifications were not applied, but the maximum concentration found was of 45200, a 97% of the thermodynamics limit. The 3% difference could be due to some simplifications conducted when calculating the intersection of the reflected bundle on the receiver’s plane, and the percentage of rays missed.

The evaluation of parabolic trough collectors not using the cross-section simplification revealed that the so-called thermodynamics limit for 2D concentrators, that was not directly extracted from thermodynamics but by a simplification of the 3D limit, was indeed below the real limits achievable by 2D concentrators. A concentration increase close to a 20% for a parabolic trough concentrator with a cylindrical receiver and of 12% for a trough with a planar receiver were discovered. This increase demonstrates that the so-called thermodynamics limits of 2D collectors were underestimated due to the cross-section simplification.

In this thesis it was demonstrated that the cross-section simplification is valid only for those 2D collectors with a 100% optical efficiency in which the absorber is designed to be big enough to receive all the incoming radiation. The simplification becomes invalid for non-100% optical efficient absorbers due to a lack of symmetry, as explained in section 3.5.2. If rays are missed in linear receivers, the rays with the highest etendue contained in different planes that the cross-sectional, could be received or missed in the absorber as they are spread in a cone that is intersected by the linear absorber, which can be considered infinitely long, intersecting any ray contained in this axis but which its width dimension, comparable to the conical spread.
size, makes a fraction of the rays contained in different axial projections to be missed.

Finally, an optical efficiency versus concentration analysis was developed for those collectors missing radiation. These curves would allow obtaining the concentration ratio of a parabolic concentrator above the limits established when no rays are missed while maintaining an acceptable optical efficiency. As an example, Figure 3.26 shows how concentrations around 80% of the previously stated as the 2D concentration limit can be achieved for parabolic troughs with cylindrical receivers while maintaining an 80% optical efficiency. The limit for those mirrors while maintaining a 100% optical efficiency was 31.8% the 2D limit. In the case of planar receivers, which previous concentration limit with a parabolic trough was 50% the 2D limit, concentrations up to the previously stated limit can be achieved with optical efficiencies of 80%.

The literature in the field demonstrated a potential of improvement of parabolic troughs with the addition of a secondary optics, but the works found in literature focused on complicated shapes that, though improving the concentration theoretically, do not seem a practical solution for real receivers. For that reason, a simple geometry was proposed as the objective of a second-stage receiver and chapter 4 found how, for a given primary mirror, the correct design of a flat secondary reflector could increase the concentration ratio of commercial parabolic troughs up to 80% while maintaining their \( \theta \). A secondary flat receiver will not imply a noticeable increase in manufacturing costs, but, in some cases, the dimensions necessary for this mirror can cause two problems: A big shadow, up to a worst case scenario of 15% the area of the primary mirror and the impossibility of encapsulating it inside a realistic glass cover, especially if the collector has to be evacuated.

Chapter 3 demonstrated how a less than 100% optically efficient trough would also achieve a higher concentration ratio. In that scenario or if the secondary reflector dimensions are reduced to fit inside the glass cover, a reduction of the optical efficiency or the tolerance misalignments of the trough will appear. In section 4.5 ray tracing simulations were conducted for two benchmark primary mirrors. The first one, with a small rim angle, had a secondary flat receiver of appropriate dimensions to fit inside the glass cover, and in this case, the standard absorber, the full-size secondary mirror, and the reduced dimensions standard absorber were analysed. Ray tracing simulations were conducted for the three receivers by adding horizontal, vertical and rotational misalignments to the already existing scattering effect, which made the effective cone of the reflected light bigger than the solar cone distribution as it will happen in a non-ideal specular surface. For that mirror, a higher tolerance to any type of misalignment was shown for the flat secondary mirror, and the shadow loss would be compensated for linear misalignments around 1 cm.

The second primary mirror considered was the LS3/ET primary mirror, which is the biggest primary mirror available in the market. In this case, the secondary full-sized mirror required a width of around 40 cm, which will be impossible to encapsulate in an evacuated glass cover, increasing the manufacturing process of the absorber. For that reason, a reduced-sized secondary mirror which could be encapsulated within a glass cover was added to the analysis. Additionally, in this case, the reduction of the secondary mirror decreases the shadow from a 6.7% to 2.15%. The decrease of efficiency for horizontal and vertical misalignments was not high enough in the standard absorber to compensate the effects of the shadow on the secondary full receivers, but an increase of optical efficiency was observed for the reduced-sized secondary mirror if a vertical misalignment was considered. This increase suggests that, in this case, the optimal location of the secondary flat receiver is displaced 2 cm from the focus, and the optical efficiency at that point will be of 98%, while the increase of concentration achieved was 70%.

An additional advantage of the inclusion of the secondary optics is the increase of uniformity on the flux distribution, as shown in Figure 4.10. The flux distribution on an standard absorber is defined by the primary mirror dimensions and it produces a different local flux distribution along the absorber's surface. This distribution is known for causing thermal stresses on the troughs receivers that can cause bending and optical losses associated with those bending. In a worst case scenario, the thermal stress could be high enough to compromise the reliability of the trough. An increase of uniformity will decrease the stress.
Additionally, if the local concentration is reduced, the maximum temperature on the absorber will decrease, and the thermal losses are expected to decrease further. There were found some works in literature aligned with this theory, while other work claimed that a localised concentration ratio on the bottom of the absorber could enhance thermal efficiency. CFD simulations conducted in chapter 5 demonstrated how there is a theoretical enhancement of thermal efficiency if an uniform flux is achieved, but an increase of flow rate makes this increase in performance negligible. An increase of flow rate has associated an increase of pump power requirements, but an analysis of pumping power requirements shown that there is no reason to not increase pumping power within a range to compensate the effects of the irregular flux distribution on the thermal efficiency. However, increasing the uniformity of the flux makes the temperature distribution along the absorber to be more uniform and the thermal stress will be reduced.

The thermal efficiency of an evacuated and a non-evacuated absorber with and without secondary flat reflector was evaluated by CFD simulations considering different emissivities on the absorber and considering the range of working temperatures of commercial thermal oils. In an evacuated absorber, the inclusion of a full-sized secondary optics will only increase thermal efficiency at low flow rates or at high emissivities. However, a reduced-sized SFR can increase the performance of parabolic troughs at high flow rates, especially if wind velocities are high, or if the emissivity is higher than 0.1 which is likely to happen at high temperatures. In both cases, the trend of the efficiency over temperature suggests that the inclusion of a secondary flat reflector will help to obtain better thermal performance at higher temperatures, which can be achieved in the future with the use of alternative heat transfer fluids such as molten salts.

For a non-evacuated absorber, the inclusion of thermal insulation in the volume created in the glass enclosure above the secondary reflector was evaluated in the CFD simulations, as well as the standard receiver and the SFR without any insulation. The inclusion of the secondary mirror will help to reduce natural convection, reducing the heat interchange between receiver and glass cover. The working temperature of those receivers is normally lower than evacuated receivers since the heat losses increase rapidly due to natural convection. However, the range of scenarios which offer an increase of performance is not wide enough and some further improvements, such as a flat plate absorber on top the secondary mirror and connected to the cylindrical absorber should be included to minimise the effects of the shadow.

An additional advantage observed in every scenario simulated obtained is a reduced dependence of the flow rate in the thermal performance of the flat reflector receivers if compared with a standard receiver. The secondary flat reflector receivers would have a lower drop of performance if the flow rate is decreased and that will allow to improve the operation of the plants along the year since the output temperature is desired to be maintained constant for the different radiation available during the year.

In chapter 6, the results of an experimental comparison between a non-evacuated absorber with and without secondary flat reflector were shown. A commercial parabolic mirror was adapted to support different receivers changing its distance to the trough to accommodate the cylindrical receiver or the flat mirror at the focus. The thermal performance of both prototypes was obtained after setting different temperatures and analysing them when steady states were achieved at the inlet of the collector.

The CFD model constructed in chapter 5 was used to compare the results obtained during the tests. The flux distribution in the CFD model was calculated from the DNI measurements by scaling it considering an optical efficiency of 90% for both the primary mirror and the glass, a scattering factor of 0.23° in the primary mirror and an emissivity of 0.25, which was obtained by measuring flat samples coated with the same selective paint used to coat the cylindrical absorbers. The agreement between CFD and experiment was found to be within the range of uncertainties expected for the SFR receiver. The CFD model predicted a efficiency between 63.1 and 61.9% in the range of temperatures considered, and the ones obtained in the test were between 61.3 and 55.7%. When considering the standard receiver, a performance between 64.6 and 63.3% was expected, but the performance obtained was between 54 and 50%. It shows two effects which
were not expected, a rapid drop of performance in both absorbers with the increase of temperature and a lower performance for the standard collector than for the SFR.

Several hypothesis were raised to explained the mismatch between experiment and simulations, most of them related to the own-manufacturing of the absorbers. The most possible explanation for the global difference of performance raises from the primary mirror, its tracking system, and the modifications implemented in the arms of the collector to fix the different receivers. The tracking system used in the parabolic trough was the commercial tracker included in this mirror, but it was observed a bias error over time shown in Figure 6.11. To avoid this error, the tracker was set up before starting the measurements at each set point, but it is possible that the combination of tracking error and a slight deviations on the position of the receivers onto the modified arms couldn't make the intercept factor of the receivers to drop. A ray tracing simulation conducted for both receivers assuming 2.5 mm horizontal and vertical misalignments as well as a 0.1° tracking error (which was in the range of uncertainty) and a modified scattering from $0.23^\circ$ (4 mrad) -which was assumed from literature since it is not reported by the manufacturer- to $0.5^\circ$ showed a intercept factor drop of 5% for the standard absorber and an optical loss lower than 1% for the secondary flat receiver, which would make the simulations results much closer to the measured ones. However, it was impossible to really quantify this misalignments to fully demonstrate this theory.

The mirror used in the experiment has an optimal reflectivity of 89-90%. However, it has some defects due to storms and weather conditions, and there were scratches due to dust and erosion which influence on the optical efficiency couldn't be evaluated. Those effects could cause a higher scattering than expected and although the mirror was cleaned during the experiments, its real reflectivity could be lower than the assumed 0.9. The glass used for receivers was not a solar degree glass, with low-iron content and anti-reflective coating and it is very likely that its optical efficiency was overestimated.

The combination of ray tracing and CFD simulations and the implementation of a secondary flat reflector in a real receiver suggests that in a real application it would be possible to increase the thermal performance of parabolic troughs with secondary flat reflectors, and especially in high-temperature applications they will contribute to enhancing the performance of parabolic troughs. However, further research should be addressed to help SFR receivers to become a reality. Suggestions to achieve that are made in the next section.

### 7.2 Future work and recommendations

To further the research conducted in this thesis, some recommendations are addressed in this section. Those recommendations are mainly related to the experiments, since the manufacturing of the collectors lead to inconclusive results for the experiment. But also some suggestions to further improve the performance of SFR receivers are proposed here.

In order to improve the quality of the measurements, the first action to consider should be to improve the manufacturing process of the prototypes. Despite being coated in the laboratory, the emissivity obtained in the receivers did not seem critical in the poor performance of the receivers. Modifying the receiver ends and building them using a high insulation material that can resist temperatures above 150 °C will reduce the heat-sink effect to the arm of the mirror. By doing that, the insulation between clamps and receiver won't be necessary.

The commercial parabolic trough structure was used to place the receiver prototypes which can cause small deviations from the focus that further ray tracing demonstrated as a likely cause of performance loss. A system to insert millimetre scale positioning in the absorbers should be implemented within the arms of the commercial absorber, to assure a perfect alignment of the collectors. The commercial tracking algorithm should be recalibrated to remove the biasing error over time, eliminating the necessity of setting up the tracker every time a measurement was desired to be taken. And finally, it was demonstrated the difficulty of measuring accurately the temperature increases due to the short length of the absorbers. By increasing the
length of the primary mirrors, the temperature measurements uncertainty would decrease drastically as an effect of obtaining a higher increase of temperature along the absorber.

The author would also recommend a real experiment to validate the development of maximum concentration equations conducted in chapter 3. It is recommendable to validate this equations in a solar simulation, since the optics accuracy necessary to conduct this experiment should be maximum.

Figure 3.28 shown the trade-off between optical efficiency and concentration ratio and a possible maximisation of the product of the two parameters. Further research in exploring the advantages of designing parabolic concentrators at this design point should be addressed.

From the simulations it was demonstrated that it is necessary to recalculate the thermal performance of the shortened-SFR for the LS3 mirror with its optimal position which was demonstrated to be 2 cm above the focus. The shortened versions of the SFR are a promising solution for commercially available parabolic trough receivers, but a complete analysis to find the optical positions and $\psi_d$ in those receivers should be developed.

The main disadvantage of SFR is the input energy loss due to the shadow projected onto the primary mirror. A flat plate collector could be installed on top of the secondary mirror, and by transferring the heat collected to the receiver the shadowing effect would be eliminated.

And finally, a rearrangement of the equations developed in chapter 4 would allow designing primary mirrors unreachable currently due to the reduction of acceptance angle and the limitations imposed by thermal receivers dimensions. Implementing larger primary mirrors will increase the energy input of the receivers, increasing the energy production. Rough estimations on the maximum size achievable for a 70 mm diameter receiver with an SFR suggest that primary mirrors up to 30 m wide could be implemented maintaining acceptance angles in values similar to current receivers. Such big increases of primary mirrors are not an obvious task since new structures should be designed to bear such primary mirror dimensions. Also, it is likely that higher acceptance angles have to be considered. But even with increases of acceptance angles there is room for improvement in allowing wider mirrors being used in parabolic troughs plants.


Enhancing concentration ratio of solar concentrators


97. Tchinda, R. and N. Ngos, A theoretical evaluation of the thermal performance of CPC with flat one-


Enhancing concentration ratio of solar concentrators


integration in Swiss dairies. in Energy Procedia. 2014.


Appendices
## Appendix A: Validation simulations results

Radiation model validation.

### Table A.1: Radiation model validation case 1. $\varepsilon_{\text{inner}} = 0.95$; $\varepsilon_{\text{outer}} = 0.2$; $T_{\text{out}} = 25 \, ^\circ\text{C}$ and mesh independence analysis.

<table>
<thead>
<tr>
<th>Inner T</th>
<th>$Q_{\text{radiation theoretical}}$ (W/m)</th>
<th>$Q_{\text{total CFD}}$ (W/m)</th>
<th>$Q_{\text{rad CFD}}$ (W/m)</th>
<th>$Q_{\text{rad CFD mesh independence}}$ (W/m)</th>
</tr>
</thead>
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<tr>
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<td>8.01</td>
<td>8.01</td>
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<td>110</td>
<td>18.30</td>
<td>18.31</td>
<td>18.31</td>
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<tr>
<td>150</td>
<td>32.40</td>
<td>32.41</td>
<td>32.41</td>
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<td>51.12</td>
<td>51.12</td>
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<td>230</td>
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<td>75.37</td>
<td>75.37</td>
<td>75.35</td>
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<td>270</td>
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<td>106.15</td>
<td>106.15</td>
<td>106.12</td>
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<td>310</td>
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<td>144.53</td>
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<td>430</td>
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### Table A.2: Radiation model validation case 2. $\varepsilon_{\text{inner}} = 0.25$; $\varepsilon_{\text{outer}} = 0.9$; $T_{\text{out}} = 25 \, ^\circ\text{C}$.

<table>
<thead>
<tr>
<th>Inner T</th>
<th>$Q_{\text{radiation theoretical}}$ (W/m)</th>
<th>$Q_{\text{total CFD}}$ (W/m)</th>
<th>$Q_{\text{rad CFD}}$ (W/m)</th>
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<td>110</td>
<td>16.710</td>
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<td>150</td>
<td>29.576</td>
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<td>29.555</td>
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<tr>
<td>190</td>
<td>46.657</td>
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<td>230</td>
<td>68.787</td>
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<td>96.873</td>
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<td>131.900</td>
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<td>430</td>
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<td>289.377</td>
<td>289.377</td>
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</table>

### Table A.3: Radiation model validation case 3. $\varepsilon_{\text{inner}} = 0.25$; $\varepsilon_{\text{outer}} = 0.2$; $T_{\text{out}} = 25 \, ^\circ\text{C}$.

<table>
<thead>
<tr>
<th>Inner T</th>
<th>$Q_{\text{radiation theoretical}}$ (W/m)</th>
<th>$Q_{\text{total CFD}}$ (W/m)</th>
<th>$Q_{\text{rad CFD}}$ (W/m)</th>
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<td>10.216</td>
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<td>18.076</td>
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<td>190</td>
<td>28.513</td>
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<tr>
<td>230</td>
<td>42.036</td>
<td>42.036</td>
<td>42.036</td>
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<td>270</td>
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<td>106.900</td>
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<td>106.887</td>
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<tr>
<td>390</td>
<td>138.775</td>
<td>138.760</td>
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<tr>
<td>430</td>
<td>176.972</td>
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<td>176.952</td>
</tr>
</tbody>
</table>
Table 8A.4: Radiation model validation case 4. $\varepsilon_{\text{inner}} = 0.95; \varepsilon_{\text{outer}} = 0.9; T_{\text{out}} = 25 \degree C.$

<table>
<thead>
<tr>
<th>Inner T (°C)</th>
<th>$Q_{\text{radiation theoretical}}$ (W/m)</th>
<th>$Q_{\text{total CFD}}$ (W/m)</th>
<th>$Q_{\text{rad CFD}}$ (W/m)</th>
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</thead>
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<td>110</td>
<td>60.422</td>
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<td>150</td>
<td>106.945</td>
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<td>106.706</td>
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<tr>
<td>190</td>
<td>168.709</td>
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<td>168.334</td>
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<tr>
<td>230</td>
<td>248.727</td>
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<td>270</td>
<td>350.285</td>
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<td>476.939</td>
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<td>632.519</td>
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<td>430</td>
<td>1047.133</td>
<td>1044.821</td>
<td>1044.820</td>
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</table>
Appendix B: Mesh independence study

Once that the radiation and convection models were developed and validated by comparing them with experimental correlations for the case of two concentric isothermal cylinders, the different models of absorbers to study (with and without secondary flat reflector) were built. The different meshes obtained were analysed under the same extreme conditions changing the number of elements on them to assure there was no relationship between the number of elements meshed and the results.

The conditions used in that study were considered as:

- Temperature of the fluid: 400 °C
- Heat transfer coefficient between absorber and fluid: 5000 W/mK
- Heat transfer coefficient between glass and atmosphere:
- Ambient temperature: 25 °C
- Temperature of the sky: 0 °C
- Uniform flux around the absorber.
- Thermal emissivity of the absorber: 0.1
- Thermal emissivity of glass: 0.9
- Thermal emissivity of SFR: 0.1
- Thickness of the glass: 0.018 m

For the different models, the different meshes built are shown in this appendix as well as the velocity and temperature profile along an imaginary section of the absorber. The imaginary line was set in a way which intersects the secondary flat reflector in those case which had one.

In order to build different meshes, different sizing parameters on the global mesh and in the relevant edges were configured. In order to improve the quality of the mesh, inflation layers, defined by its first layer thickness were implemented in the absorber, glass and secondary flat reflector edges. The thickness of the first layer was defined with a similar dimension compared with its respective edge sizing, to avoid a low element quality in the elements placed in the firsts inflation layers.

![Figure B.1](image.png)

Figure B.1: Two of the meshes tested in the mesh independence study and the imaginary line to obtain the temperature and velocity plots (in orange) a) NEP Standard b) NEP SFR.
Enhancing concentration ratio of solar concentrators

NEP standard dimensions model

Table B.1: Dimensions of the NEP standard receiver model

<table>
<thead>
<tr>
<th>Element</th>
<th>Characteristic dimension (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glass diameter</td>
<td>42</td>
</tr>
<tr>
<td>Glass thickness</td>
<td>18</td>
</tr>
<tr>
<td>Absorber external diameter</td>
<td>28</td>
</tr>
<tr>
<td>Absorber thickness</td>
<td>1.6</td>
</tr>
<tr>
<td>Absorber wall thickness</td>
<td>0.1</td>
</tr>
</tbody>
</table>

Table B.2: Quality statistics of the NEP standard receiver meshes

<table>
<thead>
<tr>
<th></th>
<th>Final mesh</th>
<th>Mesh independence 1</th>
<th>Mesh independence 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of elements</td>
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<tr>
<td>Quality statistics</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Minimum element quality</td>
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<td>0.25</td>
<td>0.33</td>
</tr>
<tr>
<td>Average element quality</td>
<td>0.77</td>
<td>0.83</td>
<td>0.76</td>
</tr>
<tr>
<td>Minimum orthogonal quality</td>
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<td>0.6</td>
<td>0.72</td>
</tr>
<tr>
<td>Average orthogonal quality</td>
<td>0.98</td>
<td>0.99</td>
<td>0.99</td>
</tr>
<tr>
<td>Maximum skewness factor</td>
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<td>Average skewness factor</td>
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<td>Global max face size</td>
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<td>Global growth rate</td>
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<tr>
<td>Edge sizing on Absorber</td>
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<td>1.50E-005</td>
<td>1.00E-004</td>
</tr>
<tr>
<td>Edge sizing on Glass</td>
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</table>

Inflation layer details

<table>
<thead>
<tr>
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<th>Number</th>
<th>Growth rate</th>
<th>Number</th>
<th>Growth rate</th>
<th>Number</th>
<th>Growth rate</th>
</tr>
</thead>
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<tr>
<td>Inflation layers on absorber</td>
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<td>6</td>
<td>1.1</td>
<td>10</td>
<td>1.05</td>
</tr>
<tr>
<td>Inflation layers on glass</td>
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<td>1.1</td>
<td>6</td>
<td>1.1</td>
<td>6</td>
<td>1.05</td>
</tr>
</tbody>
</table>

Figure B.2: NEP standard mesh independence analysis a) Temperature b) Velocity.
NEP with secondary flat reflector dimensions model

Table B.3: Dimensions of the NEP standard receiver model

<table>
<thead>
<tr>
<th>Element</th>
<th>Characteristic dimension (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glass diameter</td>
<td>36.6</td>
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<tr>
<td>Glass thickness</td>
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</tr>
<tr>
<td>Absorber external diameter</td>
<td>22.6</td>
</tr>
<tr>
<td>Absorber thickness</td>
<td>1.6</td>
</tr>
<tr>
<td>Absorber wall thickness</td>
<td>0.1</td>
</tr>
<tr>
<td>SFR length</td>
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</tr>
<tr>
<td>SFR Width</td>
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</table>

Table B.4: Quality statistics of the NEP SFR receiver meshes

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<th></th>
<th>Final mesh</th>
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<th>Mesh independence 2</th>
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<td>Minimum element quality</td>
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<td>0.22</td>
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<tr>
<td>Average element quality</td>
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<td>Minimum orthogonal quality</td>
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<tr>
<td>Average orthogonal quality</td>
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<td>Maximum skewness factor</td>
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<td>Average skewness factor</td>
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<td>5.00E-004</td>
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</tr>
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<td>Global max face size</td>
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<td>Global growth rate</td>
<td>Default</td>
<td>Default</td>
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<tr>
<td>Edge sizing on Absorber</td>
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<td>1.00E-004</td>
</tr>
<tr>
<td>Edge sizing on Glass</td>
<td>1.00E-004</td>
<td>1.00E-004</td>
<td>1.00E-004</td>
</tr>
<tr>
<td>1.00E-004</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inflation layers on absorber</td>
<td>6</td>
<td>1.1</td>
<td>6</td>
</tr>
<tr>
<td>Inflation layers on glass</td>
<td>6</td>
<td>1.1</td>
<td>6</td>
</tr>
<tr>
<td>Inflation layers on SFR</td>
<td>6</td>
<td>1.1</td>
<td>6</td>
</tr>
</tbody>
</table>

Figure B.3: NEP SFR mesh independence analysis a) Temperature b) Velocity.
Enhancing concentration ratio of solar concentrators

LS3 standard dimensions model

Table B.5: Dimensions of the LS3/ET standard receiver model

<table>
<thead>
<tr>
<th>Element</th>
<th>Characteristic dimension (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glass diameter</td>
<td>125</td>
</tr>
<tr>
<td>Glass thickness</td>
<td>18</td>
</tr>
<tr>
<td>Absorber external diameter</td>
<td>70</td>
</tr>
<tr>
<td>Absorber thickness</td>
<td>1</td>
</tr>
<tr>
<td>Absorber wall thickness</td>
<td>0.1</td>
</tr>
</tbody>
</table>

Table B.6: Quality statistics of the LS3/ET standard receiver meshes

<table>
<thead>
<tr>
<th></th>
<th>Final mesh</th>
<th>Mesh independence 1</th>
<th>Mesh independence 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of elements</td>
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<td>76707</td>
<td>150152</td>
</tr>
<tr>
<td>Quality statistics</td>
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</tr>
<tr>
<td>Minimum element quality</td>
<td>0.29</td>
<td>0.22</td>
<td>0.26</td>
</tr>
<tr>
<td>Average element quality</td>
<td>0.8</td>
<td>0.73</td>
<td>0.89</td>
</tr>
<tr>
<td>Minimum orthogonal quality</td>
<td>0.59</td>
<td>0.62</td>
<td>0.4</td>
</tr>
<tr>
<td>Average orthogonal quality</td>
<td>0.98</td>
<td>0.98</td>
<td>0.99</td>
</tr>
<tr>
<td>Maximum skewness factor</td>
<td>0.72</td>
<td>0.75</td>
<td>0.89</td>
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<tr>
<td>Average skewness factor</td>
<td>0.08</td>
<td>0.08</td>
<td>0.05</td>
</tr>
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</table>

Sizing parameters (m)

<table>
<thead>
<tr>
<th></th>
<th>Final mesh</th>
<th>Mesh independence 1</th>
<th>Mesh independence 2</th>
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<tbody>
<tr>
<td>Global minimum size</td>
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<td>5.00E-006</td>
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<td>Global max face size</td>
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<td>5.00E-004</td>
<td>5.00E-004</td>
</tr>
<tr>
<td>Global growth rate</td>
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<td>5.00E-004</td>
<td>5.00E-004</td>
</tr>
<tr>
<td>Edge sizing on Absorber</td>
<td>5.00E-006</td>
<td>5.00E-006</td>
<td>2.50E-004</td>
</tr>
<tr>
<td>Edge sizing on Glass</td>
<td>No</td>
<td>No</td>
<td>No</td>
</tr>
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</table>

Inflation layer details

<table>
<thead>
<tr>
<th></th>
<th>Number</th>
<th>Growth rate</th>
<th>Number</th>
<th>Growth rate</th>
<th>Number</th>
<th>Growth rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inflation layers on absorber</td>
<td>6</td>
<td>1.1</td>
<td>10</td>
<td>1.1</td>
<td>6</td>
<td>1.1</td>
</tr>
<tr>
<td>Inflation layers on glass</td>
<td>6</td>
<td>1.1</td>
<td>10</td>
<td>1.1</td>
<td>6</td>
<td>1.1</td>
</tr>
</tbody>
</table>

Figure B.4: LS3/ET standard mesh independence analysis a) Temperature b) Velocity.
Appendix B: Mesh independence study

LS3 with SFR dimensions model

Table B.7: Dimensions of the LS3/ET SFR receiver model

<table>
<thead>
<tr>
<th>Element</th>
<th>Characteristic dimension (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glass diameter</td>
<td>125</td>
</tr>
<tr>
<td>Glass thickness</td>
<td>18</td>
</tr>
<tr>
<td>Absorber external diameter</td>
<td>70</td>
</tr>
<tr>
<td>Absorber thickness</td>
<td>1.6</td>
</tr>
<tr>
<td>Absorber wall thickness</td>
<td>0.1</td>
</tr>
<tr>
<td>SFR length</td>
<td>120</td>
</tr>
<tr>
<td>SFR width</td>
<td>2</td>
</tr>
</tbody>
</table>

Table B.8: Quality statistics of the LS3/ET SFR receiver meshes

<table>
<thead>
<tr>
<th></th>
<th>Final mesh</th>
<th>Mesh independence 1</th>
<th>Mesh independence 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of elements</td>
<td>42313</td>
<td>49438</td>
<td>124218</td>
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<tr>
<td>Quality statistics</td>
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</tr>
<tr>
<td>Minimum element quality</td>
<td>0.28</td>
<td>0.27</td>
<td>0.16</td>
</tr>
<tr>
<td>Average element quality</td>
<td>0.87</td>
<td>0.88</td>
<td>0.62</td>
</tr>
<tr>
<td>Minimum orthogonal quality</td>
<td>0.45</td>
<td>0.44</td>
<td>0.47</td>
</tr>
<tr>
<td>Average orthogonal quality</td>
<td>0.97</td>
<td>0.97</td>
<td>0.98</td>
</tr>
<tr>
<td>Maximum skewness factor</td>
<td>0.73</td>
<td>0.83</td>
<td>0.81</td>
</tr>
<tr>
<td>Average skewness factor</td>
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<td>0.11</td>
<td>0.1</td>
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<tr>
<td>Sizing parameters (m)</td>
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<tr>
<td>Global minimum size</td>
<td>Default</td>
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<td>1e-4</td>
</tr>
<tr>
<td>Global max face size</td>
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<td>1e-3</td>
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<tr>
<td>Global growth rate</td>
<td>Default</td>
<td>3.00E-003</td>
<td>Default</td>
</tr>
<tr>
<td>Edge sizing on Absorber</td>
<td>2.00E-004</td>
<td>5e-5</td>
<td>5e-5</td>
</tr>
<tr>
<td>Edge sizing on Glass</td>
<td>3.00E-004</td>
<td>5e-5</td>
<td>5e-5</td>
</tr>
<tr>
<td>Inflation layer details</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inflation layers on absorber</td>
<td>6</td>
<td>1.05</td>
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</tr>
<tr>
<td>Inflation layers on glass</td>
<td>6</td>
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</tr>
<tr>
<td>Inflation layers on SFR</td>
<td>6</td>
<td>1.05</td>
<td>6</td>
</tr>
</tbody>
</table>

Figure B.5: LS3/ET mesh independence analysis a) Temperature b) Velocity.
Appendix C: Material properties

Therminol 66

The properties for the thermal oil used in the CFD simulations are shown in the following graphs. The graphs were obtained by obtaining a correlation from the data extracted from the technical bulletin of the manufacturer. In every graph, the correlation obtained is shown. Note that the expression shown in the graphs is rounded, but in the calculations a higher number of decimals was considered in order to increase accuracy.

Figure C.1: Thermal properties of Therminol 66.
Appendix C: Material properties

Air

The thermal properties of air have been extracted from [82]. The expansion coefficient has been considered as 0.00255 in order to maintaining simplicity in the CFD simulations.

Figure C.2: Thermal properties of air.
Appendix D: Trigonometrical identities

\[
\frac{2 \tan \frac{\alpha}{2}}{\sin \alpha} = \sec^2(\alpha) \quad \text{(Equation D.1)}
\]

\[
\cos \arcsin \alpha = \sqrt{1 - \alpha^2} \quad \text{(Equation D.2)}
\]

\[
\sin \arccos \alpha = \sqrt{1 - \alpha^2} \quad \text{(Equation D.3)}
\]

\[
\tan \arcsin \alpha = \frac{\alpha}{\sqrt{1 - \alpha^2}} \quad \text{(Equation D.4)}
\]

\[
2 \sin \alpha \cos \alpha = \sin(2\alpha) \quad \text{(Equation D.5)}
\]

\[-2 \sin^2 \alpha = \cos(2\alpha) - 1 \quad \text{(Equation D.6)}
\]

\[
2 \sin \alpha \cos \alpha = \sin(2\alpha) \quad \text{(Equation D.7)}
\]

\[-2 \sin^2 \alpha = \cos(2\alpha) - 1 \quad \text{(Equation D.8)}
\]
Appendix E: Derivation

\[ \frac{\sin \psi}{\sin \theta} = \frac{\sin \psi [\cos \psi \cos \theta - \sin \psi \cos \theta]}{\sin \theta} = \frac{\sin \psi \cos \psi}{\tan \theta} - \sin^2 \psi \]  
(Equation E.1)

The optimal rim angle of a parabolic trough with cylindrical receiver can be found with the derivative of equation E.1. The following trigonometrical changes are made in the above equation to obtain a simple derivative afterwards

\[ C = \frac{\sin \psi \cos \psi \cos \theta}{\sin \theta} - 2\psi = \csc \theta \sin \psi \cos \theta \cos \phi - \sin^2 \psi = \frac{1}{2} \csc \theta \left[ 2 \left( \sin \psi \cos \psi \cos \theta - \frac{\sin^2 \psi}{\csc \theta} \right) \right] \]  
(Equation E.2)

\[ C = \frac{1}{2} \csc \theta \left[ 2 \sin \psi \cos \psi \cos \theta - 2 \sin^2 \psi \sin \theta \right] \]  
(Equation E.3)

And considering that \( 2 \sin \psi \cos \psi = \sin(2\psi) \) and \(-2 \sin^2 \psi = \cos(2\psi) - 1 \)

\[ C = \frac{1}{2} \csc \theta \left[ \sin(2\psi) \cos \theta + \sin \theta (\cos(2\psi) - 1) \right] = \frac{1}{2} \csc \theta \left[ \sin(2\psi) \cos \theta + \cos(2\psi) \sin \theta - \sin \theta \right] \]  
(Equation E.4)

\[ C = \frac{1}{2} \csc \left[ \sin(2\psi + \theta) - \sin \theta \right] = \frac{1}{2} \left[ \csc \theta \sin(2\psi + \theta) - \sin \theta \csc \theta \right] \]  
(Equation E.5)

And finally the concentration ratio can be alternatively presented as:

\[ C = \frac{1}{2} \left[ \csc \theta \sin(2\psi + \theta) - 1 \right] \]  
(Equation E.6)

And the concentration ratio of a parabolic trough with a planar receiver can be found by its derivative

\[ C' = \frac{dC}{d\psi} = \frac{d}{2d\psi} \left[ \sin (2\psi + \theta) - 1 \right] = \csc \theta \cos(2\psi + \theta) \]  
(Equation E.7)

Which has a 0 value at \( \psi_{C_{\text{max}}} = \pi/4 - \theta/2 \)
Appendix F: Uncertainty analysis

This appendix shows the uncertainty analysis of those direct and calculated variables relevant in the experiment. In any case, a worst case scenario was considered to find the maximum uncertainty expected in the measurements.

Direct measurements

The uncertainties associated with direct measurements were extracted from the technical specifications of the sensors. In some cases, such the pyranometers, the uncertainty considered ignores installation bias errors (e.g. 2 axis tracker vibrations) since it was not possible to quantify them. Note that some variables were ignored in this analysis as it was impossible to analyse the uncertainty. In those cases, it was also checked that the expected uncertainty of that parameter would not affect the calculated variables uncertainty. As an example, in the range of temperatures studied, the water specific heat can be considered constant and its effect on the uncertainty analysis was considered negligible to improve simplicity as the uncertainty magnitude of temperature and mass flow rate are around 4000 times higher.

- Mass flow rate

The flow meter data sheet reported a bias error of 0.2 % of the measurement. The random error was obtained with readings of the flow meter without any heating element on the hydraulic circuit and calculated as twice the standard deviation of the readings. The random error obtained was of 0.000362 kg/s and the maximum flow rate analysed was of 0.15 kg/s. The maximum error was calculated as:

\[ U_{in} = \sqrt{(0.002 \times 0.15)^2 + 0.000362^2} = 0.0005 \text{ kg/s} \]

- Temperature readings

The RTDs used in the fluid loop were calibrated before the experiment, the maximum systematic error found was of 0.125 K and the random errors, considered as twice the standard deviation of the calibration measurements was found as 0.001 K. The total uncertainty of the RTDs used to measure fluid temperatures were obtained as:

\[ U_{RTD_{\text{Fluid}}} = \sqrt{0.125^2 + 0.001^2} = 0.125 \text{ K} \]

The RTD used to measure ambient temperature shown a systematic error of 0.25 K and a random error of 0.01 K. Approaching the error of this RTD in the same way than the fluid RTDs the analysis of the uncertainty gives a maximum error of 0.25 K

- Radiation

The systematic errors listed by the pyranometer’s manufacturer are:

\[ B_0 = 3 \text{ [W/m}^2\text{]} \]
\[ B_{\text{stability}} = 0.015 \times G \text{ [W/m}^2\text{]} \]
\[ B_{\text{non-linearity}} = 0.005 \times G \text{ [W/m}^2\text{]} \]
\[ B_{\text{directional}} = 15 \text{ [W/m}^2\text{]} \]
\[ B_{\text{spectral}} = 0.03 \text{ [W/m}^2\text{]} \]
\[ B_{\text{temperature}} = 0.02 \times G \text{ [W/m}^2\text{]} \]
\[ B_{\text{bias}} = 0.0025 \times G \text{ [W/m}^2\text{]} \]
\[ B_{\text{calib}} = 46 \text{ [W/m}^2\text{]} \]
Assuming the worst case scenario $G = 1000 \text{ W/m}^2$, and as the errors show no dependence between them, the maximum uncertainty can be evaluated as:

$$B_{\text{radiation}} = \sqrt{B_0^2 + B_{\text{stability}}^2 + B_{\text{nonlinear}}^2 + B_{\text{direct}}^2 + B_{\text{spectral}}^2 + B_T^2 + B_{\text{tilt}}^2 + B_{\text{calib}}^2} \approx 54.83 \text{ W/m}^2$$

- **Time**

Time is necessary to calculate the solar time, relevant in the end of collector losses. The 2 axis tracker clock was set using a gps clock. However, an uncertainty of 0.0025 h (9 seconds) was assumed.

- **Mirror dimensions**

The uncertainty on the primary mirror dimensions were obtained from the mirror manufacturer as:

$$U_L = 0.005 \text{ m}$$
$$U_W = 0.002 \text{ m}$$

Secondary flat mirror was cut in the university workshop from big aluminium sheets with a tolerance of 0.005 m in both width and length

- **Wind speed**

The wind meter's systematic error is 0.11 m/s, obtained from the data sheet. The random error was assumed to be of equal magnitude as a worst case scenario. Maximum uncertainty can be calculated in this case as:

$$U_{\text{wind}} = \sqrt{\text{ }^2 + \text{ }^2} = \sqrt{0.11^2 + 0.11^2} = 0.155 \text{ m/s}$$

Despite the random error assumption, the wind speed accuracy won't affect significantly the heat transfer coefficient estimation between glass and ambient as the real wind velocity changes that can happen instantaneously will have a greater weight on the heat transfer coefficient than the uncertainty of the measurement.

**Ignored uncertainties**

- **Declination angle**

The declination angle is calculated from astronomical correlations, and it is a fixed value for a certain day of the year. There are not uncertainties associated with that other than the accuracy of the correlation used. The declination angle is used in the calculation of the incidence angle, in combination with the solar hour angle that depends on the accuracy of the time recorded in the measurement (assumed to be within 0.01 hours). As the hour angle will have a much higher uncertainty and to maintain simplicity, declination angle uncertainty was ignored.

- **$C_p$**

$C_p$ of water is used to calculate the heat transferred to the fluid. In the range of temperatures used to evaluate the thermal performance of the collectors and considering water the heat transfer fluid, $C_p$ is almost constant and obtained from empirical correlations.

As the maximum flow rate is evaluated as 0.15 kg/s and the maximum temperature increase was 3 K, the uncertainty of the heat transferred associated with the $C_p$ uncertainty will have a magnitude 10-4 times lower than those associated with mass flow rate and temperature difference.
Calculated variables

The calculated variables uncertainties are calculated with equation 6.2. As there are no correlations between any of the sensors employed in the experiments, total uncertainty of a calculated variable, \( z \), can be obtained from the direct variables:

- Solar hour Angle

\[
\omega_s = 15(\text{ST} - 12)
\]

Solar time (ST) is the time of the day in hours, with an uncertainty of 0.0025.

\[
\frac{\delta \omega_s}{\delta \text{ST}} = 15
\]

\[
U_{\omega_s} = \sqrt{\left(\frac{\delta \omega_s}{\delta \text{ST}} U_{\text{ST}}\right)^2} = 0.0375^\circ
\]

- Incidence angle

As the primary mirror is aligned in an East-West axis, the incidence angle is calculated from equation 6.8 which shows its dependence from solar time and declination angle.

The declination angle is calculated as shown in equation 6.1 from astronomical correlations and the solar hour angle. Those correlations are expected to be highly accurate and it is impossible to evaluate the uncertainty. To maintain simplicity, it was assumed that the uncertainty of the incidence angle was only caused by the uncertainty on the hour angle and therefore the magnitude of their uncertainties was equal.

- Effective area

\[
A_{\text{eff}} = W_{\text{mirror}} L_{\text{mirror}} \cos \zeta
\]

\[
U_{W_{\text{mirror}}} = 0.002 \text{ m} \quad U_{L_{\text{mirror}}} = 0.005 \text{ m} \quad U_\zeta = 0.0375^\circ
\]

\[
\frac{\delta A_{\text{eff}}}{\delta W_{\text{mirror}}} = L_{\text{mirror}} \cos \zeta \quad \frac{\delta A_{\text{eff}}}{\delta L_{\text{mirror}}} = W_{\text{mirror}} \cos \zeta \quad \frac{\delta A_{\text{eff}}}{\delta \omega_s} = W_{\text{mirror}} L_{\text{mirror}} \sin \zeta
\]

Considering the length as 2 m, the width as 1.2 and a maximum solar angle of 30° (approximately 2 hours before and after midday), the worst case scenario for the effective area is calculated as:

\[
U_{\text{Area-\text{eff}}} = \sqrt{\left(\frac{\delta A_{\text{eff}}}{\delta W_{\text{mirror}}} U_{W_{\text{mirror}}}\right)^2 + \left(\frac{\delta A_{\text{eff}}}{\delta L_{\text{mirror}}} U_{L_{\text{mirror}}}\right)^2 + \left(\frac{\delta A_{\text{eff}}}{\delta \omega_s} U_\omega\right)^2} \simeq 0.045 \text{ m}^2
\]

- DNI

\[
\text{DNI} = \text{Total Radiation} - \text{Difusse radiation}
\]

\[
\frac{\delta \text{DNI}}{\delta \text{Total}} = 1 \quad U_{\text{pyranometer}} = 54.83 \text{ W/m}^2 \quad \frac{\delta \text{DNI}}{\delta \text{Difusse}} = 1
\]
Appendix F : Uncertainty analysis

- Effective DNI

As the DNI is measured in a two axis tracker, the radiation normal to the aperture plane of the mirror is evaluated as:

\[
U_{DNI} = 77.54 \text{ W/m}^2 \\
U_{\text{DNI}} = \frac{\delta \text{DNI}}{\delta \text{Total}} U_{\text{Total}} + \frac{\delta \text{DNI}}{\delta \text{Diffuse}} U_{\text{Diffuse}} = 77.54 \text{ W/m}^2
\]

- Increase of temperature

\[
\delta \Delta T \quad \delta T = 1\\nU_{\text{RTDFluid}} = 0.125 \text{ K}
\]

- Heat transferred to the fluid

\[
U_{\Delta T} = \sqrt{\left( \frac{\delta T}{\delta \text{out}} U_{\text{out}} \right)^2 + \left( \frac{\delta T}{\delta \text{in}} U_{\text{in}} \right)^2} = 0.177 \text{ K}
\]

\[
\frac{\delta Q}{\delta m} = 4180 \Delta T \\
\frac{\delta Q}{\delta \Delta T} = 4180 \bar{m} \\
\frac{\delta Q}{\delta C_p} = \bar{m} \Delta T \simeq 0.45
\]

In the range of temperatures considered in the experiment, \( C_p \) is almost constant for water and it was assumed as 4180 KJ/Kg K from the empirical correlations shown in Appendix C. Its uncertainty will be multiplied by approximately 0.45 when the uncertainty of the heat transferred to the fluid is calculated, assuming a flow rate of 0.15 kg/s and a temperature difference of 3 °C. As the magnitude of its uncertainty is much smaller than those for the mass flow rate and the temperature difference, \( C_p \) uncertainty was ignored to increase simplicity. The heat transfer uncertainty can be obtained as:

\[
U_\dot{Q} = \sqrt{\left( \frac{\delta Q}{\delta m} U_m \right)^2 + \left( \frac{\delta Q}{\delta \Delta T} U_{\Delta T} \right)^2 + \left( \frac{\delta Q}{\delta C_p} U_{C_p} \right)^2} \simeq 111.16 \text{ W}
\]
Available energy

The available energy is calculated as the incoming energy at the primary mirror aperture area.

\[ \dot{Q}_a = \text{DNI}_{\text{eff}} A_{\text{eff}} \]

\[ U_{A_{\text{eff}}} = 0.045 \text{ m}^2 \quad U_{\text{DNI}_{\text{eff}}} = 67.17 \text{ W/m}^2 \]

\[ \frac{\delta Q_a}{\delta A_{\text{eff}}} = \text{DNI}_{\text{eff}} \quad \frac{\delta Q_a}{\delta \text{DNI}_{\text{eff}}} = A_{\text{eff}} \]

It is assumed that the worst case scenario will happen at midday, as at that moment the aperture area will be completely perpendicular to the DNI and the effective DNI will be maximum. It is assumed a DNI of 1000 W/m\(^2\) and the area is calculated as 2.4 m\(^2\).

\[ U_{Q_a} = \sqrt{\left( \frac{\delta Q_a}{\delta \text{DNI}_{\text{eff}}} U_{\text{DNI}_{\text{eff}}} \right)^2 + \left( \frac{\delta Q_a}{\delta A_{\text{eff}}} U_{A_{\text{eff}}} \right)^2} \approx 172.4 \text{ W} \]

Thermal efficiency

\[ \eta = \frac{\dot{Q}}{\dot{Q}_a} \]

\[ U_{\dot{Q}} = 111.16 \text{ W} \quad U_{\dot{Q}_a} = 172.4 \text{ W} \]

\[ \frac{\delta \eta}{\delta \dot{Q}} = \frac{1}{\dot{Q}_a} \quad \frac{\delta \eta}{\delta \dot{Q}_a} = -\frac{\dot{Q}}{\dot{Q}_a^2} \]

For the worst case scenario, both heat received in the absorber and entering the primary mirror were assumed as 2400 W, which is the product of a DNI of 1000 W/m\(^2\) and the aperture area.

\[ U_{\eta} = \sqrt{\left( \frac{\delta \eta}{\delta \dot{Q}} U_{\dot{Q}} \right)^2 + \left( \frac{\delta \eta}{\delta \dot{Q}_a} U_{\dot{Q}_a} \right)^2} \approx 8.5 \% \]