*Analysis of a solar heating industrial plant
based on a parabolic dish collector*

*Doctorate in Renewable Energy and
Energy Efficiency*

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Abstract

With the aim of reducing costs and weight and increasing thermal efficiency of parabolic dish collectors (PDCs), a modified cavity receiver for a PDC is proposed and analysed. The application is generation of steam at 150 °C in an industrial pilot plant of thermal treatment. The diameter of the PDC is 4.5 m. The proposed cavity receiver does not use molten salts due to the maze-like design of its interior.

The performance in the absorption of solar radiation of the proposed cavity receiver is studied with a ray-tracing model. It is found that there is a range of positions where the amount of absorbed radiation does not significantly change. Energy losses by natural and forced convection are studied by using a CFD model. Two correlations for natural and forced convection are developed and it is found that the use of a window can help to suppress convection. The interior is analysed with a CFD model for different HTF mass flow rates. An accurate distribution of radiation in the walls is applied, computed by a ray-tracing model for a fixed E_{DNI} . It is found that the proposed cavity receiver has some zones of low flow speed which produce peaks of high temperature.

The thermal performance over a typical meteorological year (TMY) of the PDC is studied for three different receivers: the proposed cavity receiver (OCR), an equivalent flat-plate receiver (FPR), and the proposed cavity receiver with a window (CCR). The results show that in comparison with the thermal efficiency of the FPR, the OCR gives an improvement of 5 %-7 %, but the CCR gives a significant improvement of 17 %-20 %. And, in addition, the temperatures reached by the FPR are significantly higher than the ones reached by the OCR and the CCR. By itself, this may be a rejecting factor for FPR in practice (like in the case considered in this work where the limit temperature of the HFT is 350 °C).

Finally, a solar plant is designed for convenient use of the energy collected at the PDC in the cogeneration of steam at 150 °C in an industrial pilot plant for thermal treatment. The system integration is done at boilers. For the cogeneration, a solar boiler is used in parallel with an existing electrical boiler. The two boilers are connected to the steam main of the pilot plant. The plant is regulated to meet a constant demand of steam. A lumped-element model of the solar and industrial pilot plants is used to estimate the performance on two selected days of the TMY. It is found that the solar boiler supplies part of the steam demand when there is solar energy available. In particular, the mass flow rate of steam meets the required demand and it is the sum of steam generated at the electrical and solar boilers.

Resumen

Con el objetivo de reducir costes y peso e incrementar la eficiencia térmica de los colectores de disco parabólico (PDC, por sus siglas en inglés), se propone y analiza un receptor de cavidad modificado. El PDC tiene un diámetro de 4.5 m. El PDC se aplica en una planta piloto de tratamiento térmico industrial para generar vapor a una temperatura de 150 °C. El receptor de cavidad propuesto no usa sales fundentes debido al diseño de laberinto del interior de la cavidad.

El rendimiento en la absorción de la radiación solar del receptor de cavidad propuesto se estudia con un modelo de trazado de rayos. Se encuentra que el receptor puede ser desplazado en un intervalo alrededor del foco de la parábola sin que varíe prácticamente la radiación absorbida. Se analiza el interior de la cavidad con un modelo CFD para diferentes situaciones de velocidad de flujo másico del fluido caloportador (HTF, por sus siglas en inglés) y para una temperatura fija de 150 °C. Se usa una distribución precisa del flujo de radiación en las paredes, calculada en análisis previos para un E_{DNI} dado. Se encuentra que el receptor de cavidad propuesto tiene zonas de baja velocidad del HTF, lo que genera picos de alta temperatura.

El rendimiento térmico del PDC durante un año meteorológico típico (TMY, por sus siglas en inglés) se estudia para tres receptores diferentes: el receptor de cavidad propuesto (OCR), un receptor plano equivalente (FPR), y el receptor de cavidad propuesto con una cubierta de cristal (CCR). Para calcular la eficiencia térmica media anual, se desarrolla un modelo compartimental del sistema. Los resultados muestran que en comparación con el FPR, el OCR da una mejora de 5 %-7 %, pero el CCR da una mejora significativa de 17 %-20 %. Y, además, las temperaturas alcanzadas por el FPR son significativamente más altas que la alcanzadas por el OCR y el CCR. Por sí mismo, esto puede ser un factor de rechazo del FPR en la práctica (como en el caso considerado en este trabajo donde el límite de temperatura del HTF es 350 °C).

Finalmente, se diseña una planta solar simple para el uso de la energía solar obtenida en el PDC en la cogeneración de vapor a 150 °C en una planta piloto industrial de tratamiento térmico. La integración de sistemas se hace en las calderas. Para la cogeneración, se usa una caldera solar en paralelo con la caldera eléctrica existente. Las dos calderas se conectan a la línea principal de vapor de la planta. La planta se regula para satisfacer una demanda constante de vapor. Un modelo compartimental de la planta solar y de la planta piloto se usa para estimar el rendimiento durante dos días seleccionados del TMY. Se encuentra que la caldera solar suministra parte del vapor que se demanda cuando hay energía solar disponible. En particular, el flujo de vapor suministrado se mantiene constante en el nivel requerido y es la suma de los vapores generados en las calderas eléctrica y solar.

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Chapter 1

Introduction

1.1 Preliminaries

Climate change, sustainable development, energy scarcity, and energy security make it interesting for the global energy scheme to accelerate the development of solar energy technologies [52, 53].

Concentrating solar thermal technologies can significantly contribute to the delivery of clean, sustainable energy worldwide. They have proved capable of providing heat effectively at any temperature level. For power generation, as stated in [40]: “Tower technology comes second [...] to parabolic dishes with respect to concentration ratio and theoretical efficiency [...]”.

The parabolic dish collector (PDC) is a concentrating solar thermal technology. It has traditionally delivered the best performance amongst the concentrating solar power (CSP) technologies for the conversion of solar energy into heat at any temperature level. PDC/Stirling systems have the greatest efficiency in electricity generation. A PDC also has the greatest efficiency in chemical energy generation. A review, history, current applications, and evolution of PDCs is given in [15].

Today, in some applications, this technology yields a costly energy compared to other solar collectors, and it is generally accepted that [15]: “[...] economic not technical barriers limit the widespread uptake of dish (and more broadly CSP technologies).”

In particular, in this thesis it is considered a PDC applied to an industrial pilot plant for pasteurization. This application is a solar heat for industrial process (SHIP) application. SHIP applications are interesting and they are already

deployed in industries [24]. The application of solar heat in industry is interesting since industry represents 32 % of the global energy demand. Of the energy used in industry, 75 % is in the form of heat. Renewable sources only supply 9 % of such heat. Industrial heat is 50 % high-temperature heat. In the 2DS forecast scenario, over the period to 2060 the industry and transport sectors will become the largest source of CO₂ emissions with a share of 59 % [24]. The 2DS (2 °C Scenario) gives the world an 80 % chance of keeping average global temperature rise below 2 °C compared to pre-industrial levels. This is a main goal of the Paris Agreement. Additional technology development and breakthroughs will increase the likelihood of reaching this very-long-term target.

A collection of industrial plants using SHIP can be found in [24] with further references to more exhaustive databases. Four suppliers of concentrating dish are reported in [24]. To improve the uptake of PDCs the design and analysis of the thermal receiver is very important [15, 30, 81]. The receiver must be designed to intercept the energy effectively, absorb it efficiently, and turn it into thermal energy at the temperature required by the application, while being limited by system costs. The design must account for thermal losses and should be such as to mitigate them to the extent possible [81].

In a PDC, the solar radiation is absorbed in an apparatus called the receiver. When solar radiation is transferred into thermal energy it is called a thermal receiver. Receivers are classified based on its geometry. Flat-plate receivers have a flat surface as receiving wall. Cavity receivers have a cavity intended for reducing the concentration and thermal losses for high temperatures [81]. One benefit of this geometry is that the surfaces tend to form a black-body radiative enclosure which enhances the radiation absorption. However, it also enhances the emission of radiation. Flat-plate receivers, on the other hand, present high local temperatures that develop due to high concentration of solar radiation when they are placed at the focal plane. The heat exchanger material of which the receiver is made must be able to withstand high heat loads, and the properties of the heat-transferring fluid (HTF) must be such as to allow the required thermal energy transport.

For the designs of the thermal receivers considered in this thesis, some trade-offs are made based on the following items:

- Nominal operating temperature is usually set by application needs, but total system efficiency (and cost-effectiveness) depends on the receiver type.
- Material degradation increases with temperature.

- The choice of HTF limits the maximum allowable flux density.
- Cavity receivers may be more efficient in high temperature, while in low temperature flat-plate receivers can be more efficient. Today, the emphasis is set on modified receivers. Cavity receivers have higher convective loss, while flat-plate receivers present higher local temperatures.
- A larger receiver area reduces spillage loss but increases thermal loss.

While there are many options and alternatives, usually the application (defining the operating temperature) and selection of the HTF (defining allowable flux levels and corrosion problems) are the main independent choices. Other selections can largely be made on economic grounds following the results of trade studies [81].

An strategy found in one commercial design is to use a flat-plate displaced from the focal plane in order to reduce the concentration. However more area would be needed and thermal losses would be increased. In this thesis, we focus on cavity receivers, and a detailed analysis of several designs will be performed in comparison with a flat-plate receiver.

1.2 Parabolic Dish Collectors

The variety of current solar collector technologies may be classified as in the roadmap shown in Fig. 1.1 from [87]. In concentrating solar power plants, we have the technologies shown in Fig. 1.2 [40]. The PDC technology belongs to the concentrating solar collectors category [81, 73, 30]. The PDC technology is characterized by the solar concentrator, or dish. The dish is a paraboloid and it gathers the solar radiation that is parallel to its axis. The resulting beam of concentrated sunlight ends in a receiver that absorbs the radiative heat. The dish is mounted on a structure that tracks the sun continuously throughout the day to reflect the highest percentage of sunlight possible onto the receiver.

The PDC technology is beneficial for the following characteristics compared with the other solar collectors: greatest efficiency, highest temperature, reduced receiver area, and modularity. A PDC can reach temperatures of 1500 °C while maintaining the best efficiency [81]. As it is noted by [15]: “Parabolic dishes are commonly accepted as the most efficient concentrating solar power (CSP) technology for the conversion of solar energy into electric or chemical energy”.

Figure 1.1: Classification map of solar collectors [87].

Figure 1.2: Typical CSP Plants schemes from [40], where a PDC is shown.

However, flexible parts, costly materials, structural aspects, foundations, and ultimately high costs make this technology sometimes unattractive.

Reduction of costs can make PDCs more attractive for low to medium temperatures. One strategy is to use cheaper mirrors of low optical quality. In this situation the receiver has to be modified which increased area for reception since concentrators of low optical quality usually stretch the spot of radiation. In addition, cheaper mirrors are usually lighter so that their use can bring the opportunity of reducing weight.

1.3 Thermal Receivers

In a typical parabolic dish collector (PDC) as those considered in [81], a (thermal) receiver is a key device in that it transforms the energy of solar radiation into internal thermal energy.

As it has been already mentioned, the costs of the PDC technology are a concern. A strategy to reduce costs is to use cheaper mirrors. According to [32] it is realistic to assume that the receiver cost is 9 % of a parabolic dish collector of 16 m². It is smaller than the concentrator cost. Thus, it might be interesting to increase the cost on the receiver if this brings the opportunity of using cheaper mirrors. This strategy is also observed in [16].

Two types of receivers are considered in PDC systems [80]: flat-plate receivers and cavity receivers. Although different designs of cavity receivers can be found in the literature, most of them are equivalent in terms of thermal efficiency [27], and in fact it has been shown that, for temperatures around 700 °C, flat-plate and cavity receivers have similar efficiency: more specifically, cavity receivers outperform flat-plate receivers by a modest 5 %. This small difference is mainly due to the increased natural convection in the cavity which is up to three times greater than in the flat-plate receiver [80]. In the cited efficiency analysis, convection losses do not include forced convection, which could change the picture even worse. Thus, the apparent advantages of a cavity receiver compared to a flat-plate one are called into question [80, pp. 173-197] when they are operated in a temperature regime below those studied in the literature cited above. In summary, it is generally assumed [80] that both flat-plate and cavity receivers are adequate and have similar efficiencies for temperatures over 600 °C. For purposes of this work, which is focused on low to medium temperature range, besides the fact that the materials must accommodate the exceptional conditions of radiant flux and temperature in a flat-plate receiver, there is other reasons to

use cavity receivers such as reduction of thermal loss specially in reradiation and convection. Besides efficiency, temperature distribution over the receiver, and in particular maximal temperature are limiting parameters that must be taken into account in practice when deciding between the two receiver designs.

One commercial PDC with flat-plate receiver for low temperature regimes is SolarBeam Concentrator of [63], which is designed for heating water for domestic use. Generally, a PDC module is used on-site where the application is located. As claimed by the manufacturer, it has a peak temperature of 93°C with an average thermal efficiency around 64 %. The SolarBeam Concentrator has a pay-back of less than six years in Australia. One SolarBeam equals 15 flat-plate collectors, supposedly of standard size. The system is designed for more than 20 years of low maintenance operation.

1.4 Perspectives for Parabolic Dish Collectors

Today, PDCs are not exploited despite their potential and applicability. One main reason is that they are costly compared to other solar technologies for low to medium temperatures. As stated in [41]: “Dishes cost more but give you more in terms of concentration ratio. This concentration is effectively used for very high temperatures (over 700°C). Lower temperatures do not justify the concentration costs.”. Other reasons for which PDC are discouraged are unreliable flexible parts, high weight, and a high area and structural requirements for foundations.

As it was already mentioned, the Stirling dish system has the highest efficiency in the conversion from heat to electricity, with a net average annual yield rate that is 23 % higher than any other solar energy system. PDCs also have the best optical efficiency [40]. However, PDCs have been abandoned due to their drawbacks, and specifically due to technical difficulties and the lack of reliable engines for large power plants [60, 40].

An important advantage of CSP technology is that radiation can be converted efficiently into heat and large amounts of heat can be easily stored unlike electricity [57, 83]. However, for small PDCs, the option of storage would increase cost, which makes it more difficult to compete with PV, including CPV [40]. In this work, no storage will be considered.

SHIP technologies are reviewed in [66]. The current picture is that the vast majority of projects use non-concentrating technologies such as flat-plate collectors or evacuated tubes, but they usually do not deliver usable heat above 100°C . However linear concentrating technologies can reach or even exceed 400°C , while

maintaining a high performance. The food and beverage industry, the service industry, and the textile industry, all of which mostly need low-temperature heat, are the main areas in which solar heat has been deployed [53]. Scheffler dishes are the most widespread systems [2].

The collector type called Scheffler dish, after the name of its inventor, is being used as a source of heat in community kitchens and other service or small industry facilities in India [52]. A Scheffler dish is less efficient than a PDC but more convenient as it concentrates the sun's rays on a fixed receiver.

1.5 Overview of the solar energy resource

The Sun transfers radiative heat to the Earth's surface. This phenomenon is familiar to us since it is an everyday experience to see daylight, feel this heat in our bodies, observe the formation of shadows, etc. With the right technology, it can also be used as an effective source of energy for electricity and heat generation. These technologies constitute the so-called solar energy sources.

The solar radiation reaching the Earth's surface has two components: direct or beam radiation, which comes directly from the sun's disk; and diffuse radiation, which comes indirectly. Direct radiation creates shadows, diffuse does not. Direct radiation is casually experienced as sunshine, a combination of bright light and radiant heat. Diffuse irradiance is experienced as daylight. On any solar device one may also account for a third component the diffuse radiation reflected by ground surfaces. The term global solar radiation refers to the sum of the direct and diffuse components.

The spectra of solar radiation has the bulk of energy in the visible light and near infrared. Thus we not only have to consider visible light, but also part of the infrared spectrum for effective solar collection [52].

In [52], the global technical potential of direct solar energy is estimated. The technical potential is a measure of the expected quantity of energy that can be collected annually with a solar technology. As it is noted, the lowest estimate of the technical potential for direct solar energy is not only greater than the global primary energy supply by 10 times; it is also greater than the highest estimate of any other renewable energy potential. Thus, with only solar energy our energy demand is potentially fully covered.

Global irradiance is a measure of energy density (flux) of different collectors. It is expressed as the annually available solar resource per surface area. The following definitions are used [52]

- Global horizontal irradiance (GHI): Global irradiance on horizontal surfaces.
- Global normal irradiance (GNI): Global irradiance on tracking surfaces. It is the maximum solar resource that can be used.
- Direct normal irradiance (DNI): Direct beam irradiance received on a normal surface to the solar direction. It is usually used to quantify the direct irradiance.

With respect to GHI, tilting collectors increase the irradiance (in this case it is equal to the GNI) up to 35 % or about 500 kWh/m²/y, especially for latitudes below 30°S and higher than 30°N.

Note that, in concentrating systems, only DNI is considered, while in non-concentrating collectors, GHI is usually used. When comparing concentrating and nonconcentrating collectors, the efficiency and other values per unit of the total input solar energy should be renormalized to GHI for a fair comparison.

Non-concentrating technologies take advantage of the global radiation, direct and diffuse (including the reflections from the ground or other surfaces) and do not require tracking. If sun-tracking is used with non-concentrating solar devices, it need not be very precise and therefore costly, while it increases the amount of collected solar energy. It allows taking advantage of the best possible resource. This is worthwhile in some cases, but not too many, as expanding a fixed receiver area is often a less costly solution for collecting as much energy.

Concentration is desirable when the following characteristics are required:

- high fluxes,
- high temperatures regimes,
- high efficiency, and
- reduced receiver size.

1.6 Motivation and goals

1.6.1 Motivation

To reduce the costs of PDCs, in [16] a modified cavity receiver is presented. It has a coil for the HTF and the coil is in a bath of molten salt. The walls for

absorbing radiation are a cavity and the aperture of the cavity is surrounded by a ring. The rest is the surface of the insulation that do not absorb solar radiation.

The main disadvantage of the solar plant of [16] for SHIP applications is that it generates steam at the receiver which causes water hammer in some cases. On the other hand, the modified receiver of [16] is heavy and it has a high thermal inertia due to the molten salts.

The main advantages are related to its geometry for absorbing radiation, i.e., the geometry of the receiving walls. The cavity is placed exactly at the focus of the parabola where the radiative heat flux is high. The flat ring is placed to receive low radiative heat flux. The low optical quality of the parabola stretches the spot of radiation in the focal plane, and the ring can help to intercept more radiation without increasing the size of the cavity. This configuration seems to take the benefits from both the flat-plate and cavity receivers at the same time.

From the above considerations, in this thesis it is proposed a modified cavity receiver based on the modified cavity receiver of [16].

The use of PDCs in SHIP applications is scarce despite the potential of PDCs. In this thesis, a proof of concept of the use of a PDC in a pilot plant for pasteurization is done. The system integration consists of the use of a solar plant for generating steam with a solar boiler connected to the steam main of the pilot plant.

On the other hand, the proposed cavity receiver is compared with a flat-plate receiver. Flat-plate receivers are receivers used in PDCs which have a flat receiving wall. The flat-plate receiver is considered for its simple geometry. Simple geometries or designs can translate in a reduction of manufacturing costs.

Finally, in preliminary analyses, it was found that the proposed cavity receiver has a lot of energy loss by convection due to its open cavity. Glazing the aperture is a strategy to suppress convection loss in cavity receivers [76]. However the thermal efficiency is impacted by the reflections in the glass. A third receiver is considered which is the proposed cavity receiver with a glass window or cover.

1.6.2 Objectives

This thesis presents an analysis of a parabolic dish collector (PDC) for solar heat applications in the lower bound of medium temperature range. The diameter of the PDC is 4.5 m. The temperature ranges are [53]: low ($< 100^{\circ}\text{C}$), medium (100°C to 400°C), and high ($> 400^{\circ}\text{C}$). The first objective is to estimate the thermal efficiency of the PDC with the proposed cavity receiver, the flat-plate receiver, and the proposed cavity receiver with a window. The thermal

efficiency is an average over a typical meteorological year (TMY). In the analysis, forced convection is included, which is not done in previous analysis like [27].

The analysis of the modified cavity receiver can also be helpful to gain more insight into the potential of cost reduction for PDCs, as presented in the previous section.

The modified cavity receiver of [16] for the PDC of [63] is used in this work. A main contribution is the new design of the receiver interior that does not need molten salts to function properly. When the HTF is passed through a coiled tube, molten salts are used as a heat conductor. Using molten salts makes the receiver heavy and with high thermal inertia. Leakages are also a concern. The design also avoids stagnation zones and tries to quickly equilibrate the transfer of heat.

The exterior geometry of the proposed modified receiver is constructed based on the following design criteria:

- Simpler hybridization between flat-plate and cavity receivers because it helps to reduce the cavity area effectively.
- No use of apertures.
- Maintaining the simplicity of cavity and flat-plate designs. It can be viewed as the subtraction of flat and cavity surfaces.
- The cylindrical cavity geometry allows for varying the height and diameter of the cavity independently without affecting the manufacturing complexity.
- The bottom is slightly modified to form a dome that tries to enhance the effective absorption.

The second objective is the design of the system integration of the PDC in a conventional pilot plant for thermal treatment. The PDC will provide heat in the range of temperatures 150 °C to 250 °C for an effective heat transfer in the solar boiler.

1.7 List of publications

Most of the works that are exposed in this thesis have been already published to several international conferences and journals. The list of publications is presented:

- Ovidio López, Aurelio Arenas, and Alfonso Baños. Radiation performance of a cavity receiver for a parabolic dish solar concentrator system. *Renewable Energy Power Quality Journal*, 1(14), 2016. Paper 370.
- Ovidio López, Aurelio Arenas, and Alfonso Baños. Convective heat loss analysis of a cavity receiver for a solar concentrator. *Renewable Energy Power Quality Journal*, 1(17), 2016. Paper 457.
- Ovidio López, Alfonso Baños, and Aurelio Arenas. Heat transfer and flow friction analysis of a modified solar cavity absorber. *Event-Based Control, Communication and Signal Processing 2018*, 2018.
- Ovidio López, José F. Sáez, Alfonso Baños, and Aurelio Arenas. Modeling and control of steam generation with a parabolic dish collector. *Event-Based Control, Communication and Signal Processing 2019*, 2019.
- Ovidio López, Alfonso Baños, and Aurelio Arenas. On the thermal performance of flat and cavity receivers for a parabolic dish concentrator and low/medium temperatures. *Solar Energy*, 2019. In Press.

Chapter 2

Background

2.1 Elements of solar power plant engineering

Many aspects of solar power plant (SPP) engineering can be used for the design and optimization of a thermal PDC. For this reason, in this section we show the general aspects of solar power plant engineering with special emphasis on the elements applying to PDCs. The elements are taken from [81].

2.1.1 Collection

Several methods for the collection and conversion of solar radiation into heat are possible [81]. Concentrating configurations in combination with primary heat transfer fluids (HTF) have been favoured concepts in most of the thermal solar collectors which exist today. The high temperature heat achievable with radiation concentration is useful immediately for industrial process heat purposes, and is attractive for driving chemical reactions. Furthermore, efficient thermodynamic conversion becomes feasible in high temperature regimes, and prime movers can be used which are established technology in conventional power plants.

Nonetheless, the potential of lower investment costs may make non- or low-concentrating configurations attractive thermal SPP concepts, provided operating viability and economy of such systems can be verified under field conditions.

The assembly of collectors, consisting of either point-focussing parabolic dishes, line-focussing parabolic troughs, vacuum tube or non-concentrating flat-plate collectors, constitutes a *collector field*. Thermal SPPs using such fields of collectors are referred to as distributed collectors, distributed receivers, or farm

systems.

Point-focussing thermal SPPs concentrate direct irradiation on an absorber (i.e. on a receiver) of relatively compact volume. Whereas thermal energy is collected by a multitude of such receivers in parabolic dish farm systems, a field of mirrors (heliostats) with large focal length is used with one centrally located receiver in central receiver or tower SPPs. Another solar-thermal concept, utilizing a completely different non-concentrating principle, is the solar chimney consisting of translucent, large-area above-ground panes with a centrally attached vertical pipe. Energy is extracted by a wind turbine at the neck of the tunnel from the upward motion of air heated underneath the translucent panes.

2.1.2 Energy conversion

Currently, electricity generation is the primary output objective of SPPs. However, solar process heat facilities are spreading.

Radiation-to-thermal processes limit the attainable and practical thermal SPP system efficiency. Efficiency regions are calculated in [81] under the assumptions specified, for the three major combinations of thermal SPP concepts (parabolic trough, central receiver, parabolic dish). The upper edge of these regions is indicative for peak thermal SPP performance and steady-state operation. Temperature limits of the HTF or materials determine the upper process temperatures.

2.1.3 Characterization and Physical Properties of Solar Power Plants

Solar collectors can be classified by their temperature ranges. In the following descriptions, we follow the definitions of [53]. Heat-temperature ranges will be defined as low (below 150 °C), medium (150 °C-400 °C), and high (above 400 °C). Temperature levels are important to define the suitability of different supply technologies to meet specific heat requirements in the various end-use sectors.

System diagram. Any power plant can be reduced to and functionally represented as a block diagram. The functional blocks correspond individually or in combination, depending on plant concept, to the major subsystems. The block diagram is characteristic, although quite similar to those of thermal and photovoltaic SPPs.

Radiation is the primary energy input to SPPs (the solar fuel). Optionally, an

auxiliary fossil-fueled source may provide backup in case of inadequate or lacking solar energy input. Usually a gas boiler (auxiliary boiler) provides the alternate source for thermal input in a thermal SPP. In large photovoltaic SPPs, a diesel generator may be the backup for electrical plant output. If a Diesel/generator is used in a thermal SPP, exhaust heat of the diesel engine can be useful in addition for covering part of parasitic thermal stand-by requirements.

The collecting, concentrating and radiation-to-heat conversion elements constitute the collector/receiver subsystem of a thermal SPP. In photovoltaic SPPs, the assembly of several photovoltaic modules constitutes an array, one or several arrays being the array field.

Once converted into sensible and/or latent heat of a suitable heat transfer fluid (HTF), energy in thermal SPPs is processed downstream in the heat transfer subsystem (similar to practice in standard thermal power plants). No equivalent subsystem exists in the photovoltaic SPPs, where, after conversion to electricity, plant-internal energy transport takes place by electric conduction.

Intermediate storage of solar-derived energy (storage subsystem) is quite feasible in thermal SPPs, but only in principle in photovoltaic SPPs. Its obvious main function is to shift utilization of solar energy into off-sunshine time periods. It provides an essential second function in stand-alone operations in steadying plant output under fluctuating radiation input conditions. In thermal SPPs, the beneficial further function of storage is the shielding of thermodynamic cycle operation from fluctuations in solar energy input, thus stabilizing operating conditions and improving conversion efficiency. Additionally, the thermal inventory provided by storage can serve as a ready reservoir for covering parasitic thermal needs, for instance for thermal SPP start-up.

Energy flow diagrams. Absorption, transport, and conversion of energy is unavoidably associated with losses. It is instructive therefore to examine the flow of energy through the system in the form of a Sankey (loss tree) diagram, or in a cascade (waterfall) diagram.

Several factors influence the performance of a SPP and must be accounted for in a design process. Such factors are, for example,

- inoperability for reasons of planned or unexpected outages and repairs, because irradiance stays below a certain threshold needed for operation, or because the energy balance is negative, i.e. if irradiance stays below the level needed for generating enough output to cover system losses and auxiliary energy requirements (over the time period considered).
- Restrictions or losses incurred for physical and/or technical reasons, such

as

- the ability to utilize only direct irradiation in concentrating thermal and photovoltaic SPPs,
- optical losses (i.e. imperfect reflection caused by dirty reflectors, or reduced absorption at large incidence angles), atmospheric losses (e.g. beam attenuation by haze or dust under large focal lengths), or spillage losses (e.g. imperfect tracking and beam aiming under windy conditions);
- thermal reradiation, conductive and convective losses at receivers/absorbers in thermal SPPs,
- thermal conductivity (heat dissipation) and non-recoverable heat inventory in the heat transport and storage subsystems in thermal SPPs, or ohmic losses in the field cabling in photovoltaic SPPs,
- cosine losses in fixed-collector or I-axis-tracking SPPs,
- thermodynamic and electromechanical conversion losses (in thermal SPPs, and radiation- to-electricity conversion losses in the semiconductor material together with power conditioning losses in photovoltaic SPPs).
- The thermal and electrical parasitic needs for operating a SPP, or for keeping it operational over periods when solar input is not, or only inadequately, available.

The existence, amount, and relative significance of these losses may vary in individual cases, depending on the collection principle employed, on full or part-load operating regimes, on the time period considered, and on plant type, design, and size. For instance, immediate thermal end use obviates thermodynamic conversion losses, or low plant availability may overshadow the significance of the other loss elements.

Efficiency. Efficiency in general terms implies - and efficiency measurements require - steady, non-changing boundary conditions. Although energies are used in the definition, time is not an explicit factor. Under steady-state conditions, ratios of energy and ratios of power become the same.

It is established practice in engineering, therefore, to define and to interpret efficiency as the ratio of output power (kW_{out}) to input power (kW_{in}). The tacit assumption remains, however, that steady-state operating conditions prevail, independent of any length of time which might be considered. The attractiveness

of these assumptions is that system-level efficiency becomes readily calculable, i.e. by multiplying the efficiencies of successive subsystems which make up the total system. The efficiencies thus used and defined would more appropriately be termed power ratios. Power ratios can be zero but can never become negative.

In any practical setting, however, time and time-dependent events are indeed factors which influence the energy throughput of a system. Under transient regimes, efficiency in terms of energy (energy efficiency) may differ significantly from power ratio; specifically, system-level efficiency no longer is the simple product of subsystem power ratios. Energy-based efficiency under transient conditions becomes ambiguous because technical factors such as energy inventories and storage, thermal inertia, operating parameters, etc. and the time period over which energy flow is accumulated, all influence the numerical value of the energy input-output ratio.

Any energy balance of a thermal SPP has to include its thermal inventory. For example, energy flows into the system but no output is generated during morning warm-up of a thermal SPP. In terms of input and output, power ratio and efficiency would be zero over this time period. At the other extreme, output can be produced solely by depleting a thermal inventory, i.e. the thermal storage (solar energy input being absent). Such an operation mode cannot be described in simple terms of power or energy input-output ratios. A third example might be the case where the amount of energy output generated over some time period is less than the parasitic energy needed for keeping the system operational; efficiency in terms of energy output to solar energy input would be negative, and hence would be without meaning. Using the term efficiency as an indicator for the capability of a SPP to convert and to transfer energy hence requires qualifying annotations⁴

A further distinction has to be made when assessing SPP performance: the time periods on which energy efficiency are based may be restricted to periods when output actually has been generated; to periods of sunshine hours (whether or not the plant is operating); or to the entire 24-hour day period. This distinction is important because net generation is affected noticeably by auxiliary and parasitic energy needs for plant operation and stand-by. Four bases for calculating input/output relationships are in use [3] and are to be differentiated:

- Gross or net power efficiency. Ratio of output power, divided by the active collector area.
- Operating hours energy efficiency. Ratio of energy output, divided by the active collector area

- Ratio of 24-hour average energy output to the total or direct irradiation.
- Annual energy efficiency. Ratio of the total of output generated over a year

For given site-specific meteorological conditions, the annual net energy efficiency (or annual energy yield) is the most significant criterion for evaluating SPP technical/operational performance at the design stage, and to assess operational performance/reliability in relation to predicted performance. It is also the key performance baseline figure for judging value and cost-effectiveness of a SPP scheme. As different meteorological conditions/statistics will result in different net annual energy efficiencies, the same SPP configuration may perform and produce differently at various sites.

2.1.4 Annual output performance calculation

In detailed designs, annual output performance is predicted by computational simulation, using a model of the SPP dynamic characteristics and irradiation data representative of the plant location. Measured long-term irradiation data, averaged over appropriate small time increments (minutes, hours), are the best basis for such a numerical analysis. The performance predicted on this basis is well suited for analytically comparing the calculated performance of alternate SPP configurations for the same irradiation conditions at the same site, even for extended time periods.

In thermal receivers current analyses couple radiative heat transfer and fluid dynamics effects in a mathematical system of equations (the model). Numerical methods for ordinary differential equations are then used to solve the model, see for example [10, 22].

Averaging irradiation daily (or monthly) and establishing a distribution of the number of days falling within certain categories of daily average irradiation, a first-order approximation of yearly plant output performance can be estimated using the model. The approach produces upper-bound performance estimates, as the influence of plant availability on performance is neglected; the impact of weather and meteorological statistics on performance can only be estimated in the case of thermal SPPs, for instance the effects due to weather-induced irradiation transients, or due to prolonged periods of energy input below the threshold for net output.

2.1.5 Design aspects

Design of conventional power plants, in general, starts with a decision about supply needs, plant size and operational strategy (e.g. peak, intermediate, base load or load-following supply). The next steps focus on the selection of the primary energy source and on means and options for heat rejection (in case of thermodynamic cycles). Definition of the heat transfer and/or working medium, as well as the selection of a prime mover unit which best satisfies supply needs, are other key decisions. Hence the procedures for correlating size and capacity of subsystems with requirements, and the evaluation of system performance, are well established; in comparison, design procedures for SPPs are still being refined and are not yet standardized. There is no question as to the primary energy source for a SPP. A key decision will be whether intermediate energy storage and/or auxiliary (fossil) energy sources must be incorporated to satisfy output requirements. However, because of the nature and quality of solar radiation, SPP design procedures differ in some essential aspects from the conventional design approach. Concentrating thermal SPPs are dependent on good direct normal irradiation (DNI) conditions, typically found in desert locations. In a desert environment, however, availability of water for cycle cooling can become critical. If wet cooling is not feasible for lack of water, heat must be rejected at ambient air temperatures, diminishing the advantage of a desert location with good DNI availability.

Terminology

Design point. A conventional thermal plant facility is usually sized on the basis of nominal (nameplate) output conditions. Operation at these conditions may be maintained for extended time periods (base load supply). As solar energy input is periodic and fluctuating, however, design of a SPP must start by specifying energy input conditions, i.e. assuming a specific irradiance to be present at a specific time during a particular day of the year. This so-called design point along with a specification of the output power capacity under rated conditions provides the basis for technically sizing a SPP, its subsystems and elements. The design point irradiation is frequently chosen at 10:00 a.m. or at noon at equinox, but sometimes also at noon of summer solstice. Rated (nameplate) power capacity of SPPs is usually stated in terms of design point operating conditions. Comparison of power capacity between SPP alternatives on the basis of nominal performance, therefore, is only of limited significance as design points usually are not identical,

and because irradiation and ambient conditions may be different at the locations where SPPs are sited.

Performance. SPP power capacity under design point conditions conveys only an idea of the potential of a SPP to provide energy output over extended time periods. Actual SPP energy output performance (over a year, month, week, or a day) must be calculated by adding up average energy output over incremental time intervals (e.g. minutes). Key factors influencing SPP energy output performance are the irradiation as determined by the meteorological situation, the SPP operating strategies aiming at economizing plant parasitics and losses including those for storage (if any), as well as the readiness of a SPP to operate whenever solar input is available. Plant output performance may be improved significantly if an auxiliary energy source can be exploited for providing energy output in the absence of (or in combination with) solar energy input, or, in the thermal SPP case, if thermal storage can be used as a thermal buffer or as a tool for time-shifting solar energy input.

Solar multiple. To obtain better SPP performance on average irradiation days and over the year, a larger-than-nominal collector subsystem may be desirable, even if energy storage capacities are small, or if output possibilities are exceeded under (short-term) favorable input conditions. Furthermore, if plant output and charging of thermal storage are to be accomplished simultaneously in a thermal SPP, the collector subsystem must be oversized relative to that needed for rated (design point) output.

By definition, solar multiple (SM) is the ratio of collector subsystem output power at design point conditions, to that needed for generating nominal output.

Capacity factor. For reasons of maintenance, service or cost, power plants are never operated at all times. The ratio of yearly average output to the hypothetical yearly output if operating continuously at nameplate rating is defined as the capacity factor (CF). The higher the CF, the better the utilization of the plant. A high CF improves the revenue situation for the financial investment which a SPP represents. Three factors principally affect the CF, the location-specific irradiation conditions, the SPP type and system configuration, and the SPP operating reliability during times of irradiance availability. Measured CFs, if different from calculated ones, are performance indicators for the maturity of the SPP in question

Field-receiver ratio. To increase the number of hours at which the receiver can operate at its design point rating, the converter may be oversized at the design point by some factor. While some energy must be spilled for a few hours on the best solar days, the receiver will be used more effectively on an annual

basis.

Design objectives

Any power plant constitutes a considerable capital investment risk which is to be recouped from revenues accumulated over the life of the facility. Hence, the overriding objective in a SPP design is to generate output at the least possible unit cost. The goal of SPP development efforts is to reduce the cost of production so as to render SPPs more competitive with other alternatives.

There are several widely accepted models and techniques for calculating and judging the attractiveness of investment propositions. One basic approach is to relate costs and expenditures with benefits, suitably aggregated over lifetime and converted to a yearly basis. This cost/benefit approach compares, in the case of power plants, the cost for capital investment plus present value of recurrent costs over plant lifetime (for operations, maintenance and consumables) with the present value of lifetime revenues from the sale of plant output (net present value).

For the relative ranking between power plant alternatives, frequently specific energy cost or levelized energy cost (LEC) is the preferred indicator, defined as the average annual cost divided by the average expected yearly energy output. Ranking by LEC presupposes that economic assumptions used in the calculations are consistent, and that service provided by the SPP alternatives is equivalent. LEC may also be viewed as the constant amount of revenue per unit of energy output which is necessary to recover the full cost of the system over its lifetime, provided of course that the stipulated amount of annual energy is actually produced, and that plant lifetime predictions are actually achieved.

Definition, detailed design and comparative assessment of SPP configurations require a techno-economic optimization with the objective functions either of

- least LEC (LLEC) if SPP energy output can be utilized whenever it is produced, tacitly assuming that revenue produced per unit of energy output is fixed, or of
- least cost/benefit ratio for specific applications of interest, in the case when SPP energy output can be controlled (by intermediate storage of solar energy, or by auxiliary fossil energy input) and shifted to the time of day when highest revenue per unit of energy output can be achieved.

The two objectives - least energy cost or highest energy value - may not

necessarily result in the same design solution because of different time-of-day and/or time-of-year energy value variations.

Assumptions about technical reliability, operational availability and lifetime, as well as amount of O&M over lifetime, influence the calculated energy unit cost. Precision in quantifying these parameters increases with accumulating SPP construction and operating experience. On the other hand, energy value strongly depends on the application environment and on the economic outlook, i.e. the value of energy depends on whether it is available during periods of need, and on future energy price expectations. If energy value is the driving function, this objective, possibly in combination with time-related output delivery and assurance requirements, may lead to quite distinct SPP designs. For example, expectation of a low escalation of fossil fuel cost may favor hybrid SPPs with minimal SM and minimal storage capacity, while high future fuel cost expectations may lead to quite opposite SPP solutions.

Design process and parameters

Since many parameters and factors must be considered which influence the SPP design and the cost of annual energy collected, analytical design tools are developed and refined for the conceptual design of most types of SPP and SPP technologies.

Design process. Optimum cost/performance is the principal optimization criterion in a conceptual design process. For comparing design alternatives, quantifiable factors such as system efficiency at design point, annual energy yield, or cost and value of the energy produced may be used. Qualitative assessments may enter the process, for instance the assessment of technical risk and reliability.

The conceptual design process usually involves distinct stages. These stages may be illustrated using the central receiver systems as the most complex example for a thermal SPP design. The design stages are in succession:

- Plant definition - defining and specifying the basic plant configuration and the nominal performance at design point;
- Energy collection optimization - for given insolation statistics, sizing of the tower and receiver, layout and configuration of the heliostat field, with particular regard to output temperature and receiver and heat transfer fluid characteristics, with LEC as objective functions for the thermal energy produced in the system (in this case at the base of the tower); and

- Energy utilization optimization - selecting the optimal turbine-generator power block, thermal storage capacity and/or auxiliary energy input. Depending on dispatch strategy, the objective function may be optimal service by assuring, to a high degree, that output can be made available when needed, or it may be maximum revenue for one year and/or over lifetime.

Design parameters. In each design stage the parameters influencing the plant have to be specified. For comparative studies of design alternatives, a multitude of designs must be prepared and the results compared.

Energy collection optimization requires, in addition to the plant parameters discussed, specification of the performance and cost factors associated with transforming site-specific irradiation into thermal energy.

The last stage of the design process is the optimization analysis to select the best combination of parameters values.

By this conceptual design approach, performance and cost figures for each technical alternative can be compared. However, important factors influencing the analysis exist which generally are not considered explicitly. These are

- detailed O&M cost,
- operator/item plant availability and reliability,
- plant lifetime,
- technical risk.

2.2 Advanced receiver designs

A variety of evolved, advanced thermal receivers are found in the literature and in commercial systems seeking to improve the performance and eligibility of concentrating solar technologies [81, 31, 19]. For the parabolic dish collector of this work, a selection of the most promising advances is presented in the following.

An analysis of a variety of cavity receivers can be found in [27], where it is shown that varying concentrator rim angle and cavity geometry can greatly affect the cavity power profile without a large effect on system efficiency. The analysed geometries of cavity receiver are: cylindrical, hemispherical, spherical, elliptical, and conical.

In [47], the heat loss by convection is reduced up to 20 % by adding plate fins in the modified semi-cavity receiver of [34].

To reduce the high cost of a PDC, in [32] the cost of the concentrator is reduced by using an aluminium frame of a paraboloidal satellite communication dish. The PDC is applied to generate steam at 450 °C. It is pointed out that the concentrator has the highest cost of all the components of a PDC. The inherent strength of the paraboloidal shell involving a deep dish wherein no frame is required is helpful to reduce the weight and cost. The wire mesh of the paraboloidal satellite communication dish is replaced by a mirror made of polymer film coated with a silver-aluminum (Ag-Al) alloy. The paraboloidal satellite communication dish has a low optical quality with important slope errors. The dish is mounted with a modified cavity receiver. The results show a thermal efficiency in the range 70 % to 80 % at a working temperature of 450 °C. The cost of the modified receiver is 12 % of the total system cost. A system cost of 200 €/m² to 225 €/m² is given. It is shown that the reflectance of the mirror was reduced by 26 % during a period of 6 years.

In [32] the low optical quality of the dish would require a cavity receiver of relatively large aperture which, in turn, would involve large energy losses. Because of this, two additional types of cavity receivers are proposed: a modified cavity receiver, and a semi-cavity receiver. The receivers are portions of a sphere with an external surface covering partially the aperture. In the case of the modified cavity receiver this external surface is flat, while in the case of the semi-cavity receiver it is spherical. The receivers are analysed and optimized for maximum thermal efficiency. The results show that the modified cavity and semi-cavity receivers are more efficient than the cavity receiver, with a relative change in thermal efficiency of around 33 %.

To reduce the high cost of a PDC, in [16] a low cost concentrator is developed. The PDC is applied to generate steam at 150 °C. The concentrator is made of fiberglass, which is chosen because of its ability to form to any mold. To maximize the concentrator size, small sections were made and fitted together to form a continuous dish. The cost reductions are achieved by cheap concentrator materials. A modified cavity receiver is developed to accommodate the low optical quality of the mirror. This modified cavity receiver has a cylindrical cavity with a flat annulus surrounding the cavity aperture which extends the receiving area. The annulus is added to accommodate the tubes inside the cavity. The size of the receiver is optimized for increasing the thermal efficiency. The thermal efficiency is 62 %. The annulus has an efficiency of 42 % while the cavity has 70 %. The cost of the receiver is 16 % of the total system.

Natural convection on the receivers of [32] is analysed in [34]. Natural convection is computed for a variety of inclinations in the range from 0 °C (cavity

aperture facing sideways) to 90°C (cavity aperture facing down). The results point out that the their proposed modified cavity receiver has less heat loss by natural convection than the cavity and semi-cavity receivers. For each receiver, convection is very high at 0° and decreases monotonically as the angle increase up to 90° .

In [33] a high-temperature solar thermal receiver for an heliostat fields is analysed. The heliostat field consists of 12 units of area 56 m^2 . Operating temperature is 1000°C and pressure 30 bar. The receiver is a cavity with the aperture divided into separate stages according to the irradiance distribution. This design has been shown theoretically to significantly reduce emission and convection losses. In such a partitioned system, the working fluid (air) is gradually heated as it passes through a sequence of receiver elements with increasing irradiance levels. An experiment to demonstrate this principle using two heating stages has been conducted. A low temperature receiver (preheater) is the first stage. The high-temperature receiver stage is the Directly Irradiated Annular Pressurized Receiver (DIAPR). Air exit temperatures of up to 1000°C were obtained, with the low-temperature stage supplying up to 750°C . The power output was up to 55 kW_{th} . The application is modern power generation and fuel production cycles. The convection losses are compared to the values given by a correlation for infinite cylinder arrays. Discrepancies around 50 % are found. As stated by the authors, this is a very reasonable result, considering the highly simplifying assumptions employed in the estimates. In this receiver, a window made of fused silica is used that separates the receiver cavity from the ambient air and allows operation at high pressure, while admitting incoming solar radiation with minimal reflection loss. A modification over the original design was to introduce cold air that bypasses the preheaters and serves for cooling of the window. In [33] the pressure drop over the receiver preheaters is about 5 % of the overall system pressure loss. In order to reduce pressure losses, extended surfaces with lower Reynolds numbers are pointed. Most of the pressure loss is due to the inlet and exit conduits.

An advanced design of helical conical cavity receiver is found in [49] in order to be used in a low-cost PDC. The concentrator and the receiver are made from low-cost materials in order to reduce the total investment. The total final cost of the system is about 7000€, with the receiver amounting to 500€. A flat-plate receiver is used which is a corrugated spiral tube located inside an aluminum housing. The flat-plate receiver is created from stainless steel and it has not a selective surface. Despite the low thermal efficiency of 34 %, the optimum exergetic performance was observed for a working temperature of 155°C . As it

is pointed out in that work, the proposed conical configuration has the potential to offer a higher performance than the flat-plate shape.

In work [62], a modified pear cavity receiver is computed from a shape optimization algorithm which optimizes for a uniform radiation flux distribution. The uniformity performance of the wall flux is compared with five traditional geometries: cylindrical, dome, heteroconical, elliptical, spherical, and conical. In this work, it is also found that cavity geometry can greatly affect the cavity power profile or radiation flux distribution.

2.2.1 Convection suppression on cavity receivers

Cavity receivers have more energy loss by convection than flat-plate receivers due to the higher area for thermal loss [27, 81], due to which convection suppression on cavity receivers has been an actively researched topic.

It can be assumed that heat loss by convection in a (open) cavity receiver of a parabolic dish concentrator (PDC) system is 20 % of total loss *in the receiver* [27], being 5 % of the total solar radiation input. Heat loss by forced convection can also be very important [80, 27, 19] for low to medium temperatures, being roughly two times higher than heat loss by natural convection only for a light breeze [27].

Strategies to reduce convection on cavity receivers are found in the works [20, 76]. An interesting strategy used in solar power towers is the use of air curtains or aerowindows. In work [20], an air curtain is used and it is obtained that convection losses are reduced by 50 % in the best case. This is in concordance with the reduction of losses found in [86]. In [69] it is shown that an air curtain with a wind speed of 8 m/s can be used to effectively increase efficiency of a cavity receiver by 5 % points. It is also pointed out that the air curtain enhances the cavity efficiency especially when the external wind speed is not very high.

A cover window is a cheap alternative for PDCs with the main drawback of the limited transmittance of some materials. In [75] a silica window is able to work in the solar broadband with a transmittance of 0.99. In this work, the cover window strategy is selected.

2.3 The parabolic dish collector and the proposed cavity thermal receiver

2.3.1 The parabolic dish collector

The studied PDC studied in this thesis is shown in Fig. 2.1. It is a commercial PDC [63] with a diameter of 4.5 m and a focal distance of 2.2 m. This PDC works properly on a rooftop and is designed for using a flat-plate receiver for water heating. We will use the location of this system as reference being 38.0204°N and 1.1709°W .

The geometry is drawn in Fig. 2.2. The parabolic dish is made of petals of fused-glass mirrors. This architecture allows for ease of transport. Four bars support the receiver, one of which is selected for the HTF pipes. The HTF pipes are made flexible and long enough to allow for the movement of the dish, which are visible in Fig. 2.1. The parabola tracks the sun by a 2-axis system comprised of two motors, a PLC (Programmable Logic Controller) and a PU (Power Unit) located nearby. The PLC runs a solar position algorithm fixing the direction (open loop configuration).

The functioning of the system is represented in Fig. 2.3 which is different from other solar thermal collector technologies [81]. The paraboloidal mirror orients to the Sun and concentrates solar rays around the focal point. Then, the radiation interacts with the receiver and a considerable portion of photons are absorbed by the thin film of solar paint applied to the surfaces of the receiver. This absorption produces a heat transfer from the Sun to the steel of the receiver. Inside the receiver, heat is transferred by conduction through the steel and then it is transferred to the HTF by convection. In steady conditions, between the inlet and outlet the HTF gains (internal) energy and increases its temperature.

2.3.2 The proposed cavity thermal receiver

The geometry of the proposed cavity receiver is shown in Fig. 2.4, with dimensions given in Table 2.1. This receiver was initially motivated by the receiver shown in the work [16]. Some important drawbacks were identified and then a new design was proposed. The cavity exterior geometry was retained from the receiver of [16]. The dimensions were optimized for a parabolic dish similar to the one considered here. On the other hand the bottom is a spherical cap that is intended to reflect solar rays to the cylindrical walls if the black coating would have a

Figure 2.1: Experimental PDC. Placed in the rooftop of a building. The HTF pipes are mounted in one of the bars and reach the connection point of the post with a stretch of flexible tubes, which is visible in the picture. The collector is based on a commercial parabolic dish [63], which is for heating water around 60°C using a flat receiver. It is deployed at University of Murcia, Spain (38.0204°N , 1.1709°W , altitude 98 m).

specular component which can take advantage of the specular component of some solar black paints. It also helps to reduce the irradiance flux due to higher inclined area with respect to the impinging rays. In the work [77] a curved bottom is analysed and, at least, an optical efficiency of 2% is expected for an elevation equal to that of the bottom of the receiver with respect to a flat bottom. Nevertheless, it increases the area involved in thermal losses.

The cavity interior has been redesigned completely. The interior consists of a series of concentric tubes and the exterior shell. These parts are welded to a rigid cylindrical base. The oil enters by the outermost pipe, and reaches the outlet from outer to inner cylinders. The cylinders in the bottom force the oil to pass along the bottom of the cavity, avoiding dead zones. The small circular passes between tubes avoid stagnation zones as well. With this configuration, the HTF is forced to circulate and mix while passing through all the volume.

Salt bath is avoided which makes the cavity lighter and with low thermal inertia. The maze-like flow distribution avoids dead zones while filling the entire volume with HTF. Moreover coil tubes are avoided without complicating the

Figure 2.2: Sketch of the PDC showing the main components. Post/HTF indicates that the HTF pipes are installed inside the post. As it can be observed the mirror surface is constructed from petals.

manufacturing very much.

Figure 2.3: Block functional diagram and energy flow characterizing the PDC studied, see [80]. Collection is performed in the reflector/concentrator, also called the parabola. Absorption/conversion is performed in the receiver.

H	27 cm	F	6 cm
D	27 cm	H_{ins}	32 cm
D_{ins}	38 cm	F_{ins}	5 cm
H_{cav}	15.4 cm	D_{cav}	15.4 cm
D_{bottom}	11.4 cm		
A_{cav}	0.096 m ²	A_{cap}	0.019 m ²
A_{front}	0.04 m ²	A_{ins}	0.49 m ²
A_{insint}	0.39 m ²		

Table 2.1: Dimensions of the proposed cavity receiver.

Figure 2.4: Geometry of the proposed cavity receiver. This view of the cross section of the receiver is selected for illustrative purposes. It is a thermal receiver in that it operates as a recuperative heat exchanger. A heat transfer fluid (HTF) is used. The Front and Cavity labels identify the active walls intended for collection of solar radiation. They are cured with a solar paint to enhance the solar radiation absorption. The intended design flow pattern is also shown in the interior of the receiver in the HTF volume.

Chapter 3

Analysis of the proposed cavity receiver

In this chapter the proposed cavity receiver, presented in section 2.3.2, is analysed in detail for the following aspects. First, solar radiation on the PDC is analysed. Ray tracing is used to compute the optical performance. Second, a CFD analysis of the convective loss in the cavity is done. Natural and forced convection are studied in detail. And third, a CFD analysis of the interior is done. Some of these studies will also serve as preliminary comparisons between the proposed cavity receiver and a flat-plate receiver receiver.

3.1 Radiation performance of the cavity receiver

The PDC is designed with the parabola, also known as concentrator, and the cavity receiver. The two components have a centre axis that coincide when mounted together. The receiver can be displaced along the center axis of the parabola. Here, the final position of the receiver is be based on the radiation performance. For this a ray tracing simulation can be done for different positions. The analysis is extended to explore different geometrical parameters as the diameter of the cavity or the diameter of the aperture.

It is well understood that displacing the receiver from the focal point changes the flux distribution on the walls and the radiation performance [81]. In particular, moving away the receiver from the focal point can help to reduce the concentration locally in the walls of the cavity. Lower concentration means lower temperature and a more uniform temperature distribution. Thus, displacing the

receiver can be identified as an interesting strategy for reducing temperature. An important advantage of this strategy is that it also applies to simple geometries like a flat-plate receiver. It is speculated that some commercial designs already use this strategy [63]. However, little works are found in the literature on this strategy.

On the other hand, other design parameters like the diameter of the cavity and the diameter of the aperture influence the radiation performance [62, 27]. The used methodology is easily extended to analyse these parameters as well.

Recall that for high temperatures, cavity receivers are preferred over flat-plate ones because cavity receivers are specifically designed to reduce concentration and they have a higher absorption of radiation than flat-plate receivers [27, 62]. However, for medium temperatures the discrepancies get blurred. In particular, it is not well understood whether the proposed cavity receiver would be clearly superior in comparison with the radiation performance of an equivalent flat-plate receiver. Thus, the above points need a detailed investigation for the receiver at hand, which is done in this section. Concretely, the radiation performance of the proposed cavity receiver is analysed. The effect of displacing the cavity receiver along the focal axis is studied. On the other hand, the influence of the diameter of the cavity and the diameter of the aperture are also studied. For comparison, an equivalent flat-plate receiver is defined and also analysed.

3.1.1 System Structure and Design Approach

The PDC is described in section 2.3. For this analysis it is used the simplified geometry shown in Fig. 3.1.

3.1.2 Cavity receiver

The simplified geometry of the proposed cavity receiver is shown in Fig. 3.2. For this analysis an annulus at the aperture with a small thickness is added. The front surface is not a receiving surface here since we want to investigate the absorption of radiation in the cavity exclusively. The receiving walls are those shown in gray in Fig. 3.2. They are the bottom wall, cylindrical walls, and the annulus wall that is inside the cavity.

The receiving walls are treated with a solar paint. This paint makes the wall opaque and an ideal, diffusely reflector and absorber. The walls in blue are perfect absorbers. They are not taken into account in the computation of the energy absorbed by the receiver.

Figure 3.1: Sketch of the simplified geometry for the system of the PDC. The length unit is the meter.

The bottom is a dome designed to increase absorption of radiation. It is speculated that the dome reflects some rays to the cylindrical walls instead of reflecting them back to the aperture. On the other hand, the annulus should help to block rays. A smaller aperture (greater annulus) will absorb more rays than a greater one. Thus, it is hypothesised that there is an optimal diameter of the aperture for maximum radiation performance.

3.1.3 Process modelling

The radiative heat transfer in the system is modelled by geometrical or ray optics [44]. The assumptions of ray optics hold since the range of wavelengths of the solar radiation is from 300 to 2000 nm (the light intensity in the infrared-C (IRC) sub-spectrum from $3\mu\text{m}$ to 1mm is negligible [18]), which are small values compared with the smallest features of the system.

The model is simulated by Monte Carlo ray tracing [44].

3.1.4 Concentrator optics

The effects of concentrator imperfections and the solar rays are important for good results. They are modelled as described in the following.

Figure 3.2: Cross section of the cavity receiver. The length unit is the meter.

The solar rays are not parallel and they have a nonuniform intensity depending on the angular position in the solar disk from where they originate. The limb darkening of the Sun reflects the fact that the limb of the Sun appears darker than its center. These effects are important in most CSP plants and can be observed as a distortion of the flux distribution of an ideal parabolic mirror. These effects are modelled by the sunshape function defined in [9, 81]. It is assumed the sunshape of [11]. The Sun's angular radius is θ_{max} , given by the following expression

$$\theta_{max} = \tan^{-1} \left(\frac{r_{Sun}}{d_{EarthSun}} \right) = 4.65 \text{ mrad}, \quad (3.1)$$

where r_{Sun} is the solar radius and $d_{EarthSun}$ is the distance from the Sun to the Earth. Values can be found in [81].

Typically, a concentrator for a PDC has important imperfections originating from the microscopic level to the system level whose overall effect is a distorted, fuzzy focal flux [81]. Specular errors are due to waving of the surface. They may be increased over time due to hail, etc.

A proper description of all the errors can become quite complex. However, it is a standard approach to approximate the entire degradation function of the concentrator by a Gaussian, with little loss in accuracy [81]; but still some care is required particularly at big angles [50]. The nature of tracking collectors leads to a reasonably representation by the error function of the concentrator [81]. This

function can be used to generate random errors in the angle of reflection of each ray sampled from a standard probability distribution given by the error function.

The standard deviation σ of the error function is variable among different PDCs found in the literature. Similar PDCs are found in the works [62] considering a value of $\sigma = 3.45\text{mrad}$, and [16] considering a value of $\sigma = 6.7\text{mrad}$. These works are calculated from experiments. For this work, the selected value is $\sigma = 3.45\text{mrad}$ as it has been considered more representative.

It is assumed that the system is always tracking perfectly the Sun. However, in real operating conditions a small error is introduced, which could be modelled as a source of error in the concentrator [81].

It is assumed a clear sky. The structure supporting the receiver is neglected [27]. This is the reason for the simplification of the supporting bars in the system considered here.

3.1.5 Methodology

The model gives a stationary value of the optical efficiency. It is solved using the Monte Carlo Ray Tracing method [44].

The radiation performance of the cavity receiver is computed for the following parameters shown in Fig. 3.1 and Fig. 3.2:

- *Position to the focal point.* The receiver is positioned along the center-line of the system.
- *Aperture diameter.* This parameter can be varied up to the cavity diameter.
- *Cavity diameter.* The bottom shape does not change when varying this parameter.

For comparison a flat-plate receiver was also considered. Its geometry is a square surface with side 25.4cm. It is placed at the focal point perpendicular to the axis of the parabola. The square surface is the receiving wall, which has the same properties of the cavity walls.

The proportion of the energy absorbed in the receiving walls to the total incoming solar energy is the optical efficiency. The optical or radiation performance is quantified by this figure. For all the studies the direct normal irradiance is $E_{\text{DNI}} = 1000\text{W m}^{-2}$.

3.1.6 Simulations

The solar absorptance, α , of the solar paint was set to 0.9, a representative value for black coatings. Moreover, as mentioned above, these walls were considered totally diffusive.

The concentrator surface was set to an emitting surface. Thus, the rays are released as they were coming from the Sun direction.

The remaining domain walls were set to a freeze condition, thus removing a ray from the simulation when it reaches the domain limits. This condition is applicable when a ray moving away from the receiver will not return back. Although this might not be true if we considered reflections on the parabolic dish from rays leaving the receiver.

For an accurate computation of the flux distribution on the walls of the receiver we needed a finer element size on them. This, in turn, increased the memory usage, so a balance had to be found. The minimum length of a surface element was set approximately to 2mm.

Specific study

The flux concentration ratio distribution C_f on the focal plane is computed to qualitatively validate the assumptions made above. C_f is defined as in [81]. Intuitively, it is the “Suns” that give the same irradiation at the point considered. Fig. 3.3 shows the C_f distribution on the focal plane along a radius. The number of rays was set to 10^7 for all the simulations. This distribution is similar to the one found in previous works [62, 23, 27].

3.1.7 Results

Simulations were performed for different positions of the cavity receiver. The ratio of the efficiency of the cavity receiver to the flat-plate receiver is obtained. This ratio assumes a fixed value of the total flux absorbed by the flat-plate receiver. The focal position is selected for which the total flux absorbed by the flat-plate receiver is 12.3kW. Fig. 3.4 shows this ratio for different positions of the cavity receiver around the focal point. It can be observed a range of positions where the ratio is flat. The ratio is above the unity because the cavity receiver is more efficient than the flat-plate receiver. At the focal point the ratio is 2.34%.

In the cavity receiver, the C_f distribution is symmetrical on the cylindrical walls. Thus C_f only depends on the position in the central line. The values

Figure 3.3: Total flux C_f in the focal plane of the parabola. The remaining parameters are set to their nominal values. The receiver position is measured relative to the focal point of the concentrator. A positive value means a position further from the focus.

of C_f along this line are shown in Fig. 3.5. The different C_f functions are for different positions of the receiver. This result is in concordance with the results of [62].

The C_f distribution is symmetrical on the bottom and it only depends on the radial distance from the central line. The C_f distribution along a radius is shown in Fig. 3.6. The different C_f functions are for the same positions considered in Fig. 3.5. It is observed that the maximum C_f is $C_{f,\max}^{\text{OCR}} = 4.3 \times 10^3$ so the cavity receiver reduces in a factor of 35% the maximum concentration ratio of the flat receiver.

For the flat-plate receiver, the C_f distribution is given by the values of the distribution of Fig. 3.3 times the absorptance of the solar paint. The maximum C_f is $C_{f,\max}^{\text{FPR}} = 1.19 \times 10^4$.

The cavity receiver at different aperture diameters was considered. Simula-

Figure 3.4: Ratio of the optical efficiency of cavity receiver to flat-plate receiver. In the figure this ratio is shown for different cavity receiver positions along the focal axis. The flat-plate receiver is fixed at the focal point. The origin is the focal point. A movement in the positive direction means to move away from the parabola.

tions were done in each one of them to compute the total absorbed power. The results are shown in Fig. 3.7. It can be observed that the power decreases as the diameter reduces. It is not observed any trade-off. Thus, the annulus does not increase noticeably the optical efficiency of the cavity receiver. The annulus could help to reduce convection losses [65]. From the results, it is speculated that the reflected radiation (radiation going out from the cavity after some reflections) is of little importance.

The cavity receiver is considered for different diameters of the cylindrical walls. Fig. 3.8 shows the total absorbed power for each one diameter. It is not observed any tradeoff.

Figure 3.5: C_f distribution on the cylindrical walls of the cavity receiver along the center line for different positions of the receiver. The unit for the positions is the centimeter.

3.2 Energy loss by convection from the cavity receiver

Energy loss by convection from cavity receivers is usually high which degrades considerably the thermal efficiency of some PDCs [27, 81]. The problem is increased under wind. A considerable variety of strategies for reducing convection have been already developed [76].

In the proposed cavity receiver, it would be interesting to determine how important is convection in the overall loss of the receiver. This receiver has a cavity smaller than a comparable cavity receiver, but the cavity works similarly to a conventional cavity receiver, thus the same pitfalls are expected. Nevertheless it is hypothesised that the reduced size should help to reduce loss. Thus, a detailed study can bring the opportunity to determine whether convection is important in comparison to a flat-plate receiver and it can help to assess some approaches to

Figure 3.6: C_f distribution on the bottom wall of the cavity receiver along a radius line for different positions. The unit for the positions is the centimeter.

reduce convection.

3.2.1 Process Modelling

The PDC is described in section 2.3. A flat-plate receiver is considered for comparison shown in Fig. 3.10. For this study the proposed cavity receiver is simplified with the geometry shown in Fig. 3.9.

A computational fluid dynamics (CFD) analysis of convection is done. The model is constructed from the assumption of fluid mechanics, where it is assumed a continuum medium, the equation for mass conservation, momentum equations, and the first law of thermodynamics [6]. The flow equations thus obtained constitute the model equations. The following fields describe the flow: velocity v , density ρ , pressure p , temperature T , and enthalpy h .

Figure 3.7: Total absorbed power in the cavity receiver for different aperture diameters.

Turbulence modelling

For the two receivers, the flow is turbulent even for moderate wind speeds (around 15 km/h). However, the computational complexity of the flow equations is huge in this condition. For this, turbulence can be modelled by means of some additional properties accounting for the turbulent effects in the mean flow [79]. As the mean flow is expected to be smoother, it can help to reduce the computational complexity while retaining a reasonable accuracy on the quantities of interest. Concretely, the Reynolds-Average Navier-Stokes (RANS) approach is chosen here due to being the lowest computationally-demanding one.

In a RANS model for a position r and instant t , the value of a flow variable ϕ is expressed as $\phi(r, t) = \bar{\phi}(r) + \phi'(r, t)$, where $\bar{\phi}(r)$ is the time-mean value at position r and $\phi'(r, t)$ is the deviation from the mean value. For the cases considered here the turbulent motion is a stationary turbulence. Thus the time-average of the flow field $\bar{\phi}(r)$ exists.

In particular the $k - \omega$ SST (Shear Stress Transport) turbulence model is

Figure 3.8: Total absorbed power in the cavity receiver for different diameters of the cylindrical walls.

employed [42, 43]. The main reasons are the following. Firstly this model can be integrated down to the wall which is appropriate for wall heat transfer calculations as it depends directly on the thermal boundary layer close to the wall. Secondly it also predicts recirculation and reattachment regions, which affect to the overall heat loss rate by the dragging of the heated air inside the cavity.

This model adds to the fundamental flow equations two new turbulent flow variables: the turbulent kinetic energy k , and the specific energy dissipation rate ω , together with two new transport equations governing them. It also adds new flow-dependent parameters that ideally would have to be fined-tuned for every application; however the general values are used since they are shown to be suitable for many applications.

Figure 3.9: Cavity receiver. The length unit is the millimetre.

Air model

In the range of the operating conditions studied air can be approximated as an ideal gas. Its dynamic viscosity is modelled by the Sutherland's law [6]. Thermal expansion coefficient, thermal conductivity and heat capacity are considered constant and uniform throughout the domain at the normal pressure p_0 and the film temperature defined as $T_{\text{film}} = (T_w + T_{\text{amb}})/2$, where T_w is the wall temperature and T_{amb} is the ambient temperature.

Geometry of the computational domain

The receiver geometry is placed inside a finite domain filled with air. The outermost geometry of the domain is a rectangular prism. The receiver is placed at a fixed position. The box is subtracted with the receiver volume, retaining the outermost walls of the receiver. The silhouette of the receiver remains and conforms new additional walls of the computational domain.

The atmosphere can be modelled placing the outermost boundaries of the computational domain very far from the receiver. In practice, this requires a huge computational effort and models with closer boundaries still yield a good approximation. For natural convection, the chimney approach is used [5]. Given a characteristic length of the receiver, L , the bounding box is constructed with

Figure 3.10: Flat receiver. The length unit is the millimetre.

the following dimensions: The upward distance (the distance from the centre to the roof) is $5L$. The downward distance is $3L$. And the side distances are also $3L$. These dimensions should reduce the effects of a finite boundary, and the solution should be approximately equal to that obtained with unlimited boundaries distance.

For forced convection, the chimney approach with the same considerations as above are used. This results in the same geometry.

Boundary conditions

The boundary conditions are described in the following.

Velocity

For natural convection, the atmospheric patches are set to a zero gradient for velocity (Neumann type). This condition represents an external boundary where the air is able to freely enter or exit the domain.

For wind conditions, the atmospheric patches are distinguished. At the inlet boundary the velocity is fixed (Dirichlet type), otherwise it is set to zero gradient.

For the walls of the receiver the velocity is set to 0.

Pressure

For pressure the atmospheric patches are set to a fixed total pressure equals to the normal pressure. For inlets and walls a zero gradient condition is used.

Temperature

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The temperature on the receiving walls is fixed at a nonuniform field that depends on the direction of the HTF flow and the distribution of flux of solar radiation in the walls. The distribution of flux of solar radiation was computed in the section 3.1 for the cavity and flat-plate receivers.

In the cavity receiver, the working fluid flows from the external wall to the bottom uniformly. Assuming a stationary condition, the energy flux can be related to the temperature variation along the centre axis z by [12]

$$\dot{Q} = C \frac{dT}{dz}, \quad (3.2)$$

where C is a constant that can be determined by fixing the operating temperatures of the receiver at the inlet and outlet. Thus, given an inlet temperature T_i , and an outlet temperature T_o , the following expression can be used to calculate the temperature distribution:

$$T(z) = T_i + \frac{\Delta T}{\int_0^L \dot{Q}(z') dz'} \int_0^z \dot{Q}(z') dz', \quad (3.3)$$

where $\Delta T = T_o - T_i$, and L is the total length of the cavity.

For the flat receiver, it is considered that the working fluid flows from side to side. The above assumptions can be applied here. The resulting expressions are similar to those presented above.

The other walls correspond to the insulator surface. They are approximated as adiabatic.

Heat transfer computation

The heat transfer at the wall F is calculated based on the following expression:

$$\dot{Q} = \int_F \alpha_{\text{eff}} \frac{\partial h}{\partial n} ds, \quad (3.4)$$

where h is the enthalpy, n is the normal to F and ds is the area element. α_{eff} is the effective thermal diffusivity. This expression is derived from the Fourier's thermal conductivity law assuming a zero gradient pressure normal to the wall, which is the pressure boundary condition used here for fixed walls.

The effective thermal diffusivity is

$$\alpha_{\text{eff}} = \alpha + \alpha_{\text{turbulent}}, \quad (3.5)$$

where α is the thermal diffusivity and $\alpha_{\text{turbulent}}$ is the thermal diffusivity given by the turbulence model.

3.2.2 Methodology

The following set of altitude angles are chosen $\{40, 70\}^\circ$ which are representative of the altitudes of the Sun for the latitude of the PDC.

The wind direction is assumed to be parallel to the concentrator plane. It is speculated that these directions play a major role than the perpendicular ones since the concentrator speeds down the wind in those cases. The two speeds considered are $\{15, 30\}$ km/h [69]. They are representative of a light breeze and a normal breeze.

The operating temperatures are given in Table 3.1. These values are representative of the intended range of working temperatures of the PDC.

Temperature	Value/ $^\circ\text{C}$
T_o	250
T_i	150
T_{amb}	27
T_f	152

Table 3.1: Operating temperatures

The distributions of temperatures for both receivers are shown in Figs. 3.11 and 3.12.

Figure 3.11: Distribution of temperature for the flat-plate receiver.

The representative pressure is the normal pressure, 1 atm.

The cases and the corresponding parameters are summarized in Table 3.2. Each case represents a simulation case. The Grashof and Reynolds numbers, Re and Gr respectively, are defined as in [6]. For the computation of these values the characteristic lengths used are given now. For the cavity receiver, it is the outermost diameter. For the flat receiver the length of one side. As it can

Figure 3.12: Distribution of temperature for the proposed cavity receiver.

be observed the Grashof numbers fall below the turbulent region (10^9). Thus, natural convection can be modelled without the need of modelling the turbulence. The Reynolds numbers for forced convection are of order 10^4 to 10^5 falling in the transition to turbulence regime. This is a low level of turbulence which falls outside of the range of applicability of the model of turbulence. However, it is found that the model is also used at low Reynolds numbers and it gives reasonably accurate results. This is specially true in the study of heat transfer by convection [1, 58].

Id.	Rec	v/km/h	θ /degree	Re	Gr
1	Flat	0	40	-	9×10^7
2	Flat	0	70	-	9×10^7
3	Flat	15	-	4×10^4	9×10^7
4	Flat	30	-	8×10^4	9×10^7
5	Cavity	0	40	-	10^8
6	Cavity	0	70	-	10^8
7	Cavity	15	-	2×10^4	10^8
8	Cavity	30	-	7×10^4	10^8

Table 3.2: Cases

Mesh independence study

To validate the numerical scheme used, a particular case is simulated which experimental data are available for. The case 1 is chosen for this purpose.

Table 3.3 shows the series of meshes studied. In this case the representative magnitude for mesh independence criterion is the rate of change of the total power over the number of elements.

Elements/ 10^5	FLH/mm	IL	GR	ES/mm
3.86	0.1	20	1.15	7.5
4.76	0.1	20	1.15	5
11.5	0.1	15	1.15	2.5
26.5	0.1	10	1.15	1.5

Table 3.3: Mesh independence study. The number of elements and the convective power and the relative change are shown. Moreover some indicators of the mesh are also detailed. They are abbreviated as: FLH for First Layer Height, IL for Inflation layers, GR for Growth Rate, and ES for Element Size. Continued on Table 3.3.

Power/W	Relative change/W
114	-
97	-18.9
87	-1.48
86	-0.067

Table 3.4: Mesh independence study. This is the part 2 of Table 3.3.

It is considered that the third test is mesh independent which will be employed for the construction of the meshes for the remaining cases. For extrapolation and other recommended techniques see [46].

Validation with experiments

Experimental data are available for inclined walls given as correlations that can be found in the literature [7]. The selected correlation corresponds to an inclined wall in the laminar regime that can be found in [7]. In this case the gravity-inclination correction is used: $g \cos \theta$. The inclination angle is $\theta = 70^\circ$ and the mean temperature of the heated wall is $T_w = 500$ K. This results in an experimental heat transfer rate of 82.7 W. It represents a 6-4% of relative deviation from the value 86 W of table 3.4, which is considered to match the experimental value.

3.2.3 Simulation

The OpenFOAM ®[48] software has been used.

Id.	Power/W	Flux/W/m ²
1	87	1.39
2	148	2.39
3	300	4.65
4	472	7.31
5	124	1.06
6	62	0.529
7	352	3.01
8	534	4.56

Table 3.5: Results of the simulation cases.

Natural convection

Simulation of the cases of natural convection is done with the *PIMPLE* algorithm. Concretely the solver *buoyantPimpleFoam* is used. For these simulations a time step of 0.1 s is used.

The heat transfer value is obtained when the flow reaches a stationary behaviour. It is found to occur at typically above 100 s of simulation time.

Forced convection

Simulation of the cases of forced convection are done with the solver *SIMPLE* algorithm. The *buoyantSimpleFoam* solver is used. The convergence criteria are a relative residual of 10^{-3} for momentum and continuity, and 10^{-6} for energy.

The total heat transfer rate at the receiving walls is also monitored. It is observed that this value oscillates in the first iterations. The amplitude is reduced when the solution is near to converge. Despite of the final oscillations only the value at the final iteration is taken as the total heat transfer.

3.2.4 Results

For each case, the rate of heat loss and the flux are computed. The results are given in Table 3.5.

The relative difference between the cavity receiver and the flat-plate receiver is considered. The relative difference of the flux is computed. The results are shown in Table 3.6.

Id 1.	Id. 2	Rel
1	5	27 %
2	6	75 %
3	7	35 %
4	8	38 %

Table 3.6: Comparison between the cavity and flat-plate receivers. The Id. 1 is the base case and the Id. 2 is the comparison case. Rel is its relative difference.

The interesting flow fields are the temperature and the velocity. It would be insightful to know the flow field distribution, for which streamlines are useful. For natural convection, only the cavity receiver is considered. The flow in the flat-plate receiver is known from [7]. The flow fields are shown in Figs. 3.13 and 3.14. For forced convection, both receivers are shown. The flow fields are shown in Figs. 3.15 and 3.16.

Figure 3.13: Temperature field for natural convection in the cavity receiver; case 6.

3.2.5 Discussion and Conclusions

As it can be observed from Table 3.6, the cavity improves the flux, as it was already observed. The cavity is more efficient as the angle is increased from 40° to 70° , which reflects the fact that it retains more heated air when it approaches to a vertical position. This behaviour is also found in [34]. Under forced convection this advantage is reduced since the wind drags the heated air, but with a less speed than in the flat-plate receiver. This can be observed in Fig. 3.16. From Table 3.6 it is shown that the improvement is kept at approximately 35 %.

However, in absolute terms, the rate of heat loss in the cavity receiver is

Figure 3.14: Temperature field for natural convection in the cavity receiver; case 5.

Figure 3.15: Temperature field for forced convection at the state for a wind of 30 km/h for the flat receiver. Flow streamlines are also shown. This is the case 4.

greater than the one of the flat-plate receiver. In general, this is not true, with only the exception of the case 6 (the two receivers are facing down). This behaviour of cavity receivers is also pointed out in [81] for higher temperatures.

Thus, compared with an equivalent flat-plate receiver, the proposed cavity receiver has more energy loss by convection. To reduce convection, a handy strategy is to cover with a window pane the cavity aperture. This could also help to reduce the energy loss by thermal radiation the glass used were of one-way type.

Nevertheless, it is pointed out that it is desirable to consider more inclination angles and wind speeds for further testing of the trends observed.

Figure 3.16: Temperature field for forced convection for a wind of 30 km/h in the cavity receiver. Flow streamlines are also shown. This is the case 8.

3.3 Correlations of natural and forced convection

Although Nusselt numbers for cavity receivers in parabolic dish collectors are already available in the literature, see for example [82], they are limited to specific operating conditions and geometry for which the proposed cavity receiver was not found applicable. Moreover, only limited studies are found for forced convection. The aim is to study convection for the cavity wall with a detailed computational fluid dynamics (CFD) model, and obtain Nusselt numbers both for natural and forced convection. A computational fluid dynamics model is a detailed and accurate model which is found to resemble pretty well experimental data in thermal processes. We will use this model for obtaining both correlations for natural and forced convection in the cavity geometry.

3.3.1 Natural convection

A similar modelling methodology of 3.2 is used here, however we give some details about the new simulations. The medium is air using the ideal gas equation of state with properties obtained from [6, p. 633].

The geometry of the computational domain is rather simple. It contains the shape of the receiver geometry which is the OCR geometry with some minor simplifications for suppressing small corners or small features. The extent of this domain is delimited by a cylindrical geometry with a diameter of $6L$ and height $8L$, with L being the characteristic length. This boundary is referred to as ambient boundary. This geometry is a chimney-like geometry. These dimensions are used to reduce the influence of the ambient boundary conditions in the solution while keeping a small number of elements of the computational

grid. Note that ideally the receiver would be placed in an unlimited air reservoir. The receiver is placed at height $3L$ from the bottom. The value of L is the outermost diameter of the receiver $L = 36$ cm.

The insulator and front walls are set as adiabatic walls. The cavity wall is set to a constant temperature of 200°C . For pressure and velocity the usual boundary conditions for walls are used.

The ambient boundary is set to an entrainment boundary condition or pressure inlet boundary condition. For velocity we have a zero gradient in the normal direction n

$$\frac{\partial v}{\partial n} = 0.$$

For pressure we have

$$p = \begin{cases} p_0 & \text{for outflow} \\ p_0 - \frac{1}{2}U^2 & \text{for inflow} \end{cases}$$

In this case p_0 is the pressure without the column static pressure, $p' = p - \rho gh$. We set $p_0 = p_{\text{normal,amb}} = 1$ atm. For the temperature boundary condition we set a zero gradient for outflow conditions. The ambient temperature is used for inflow conditions, which is 26.85°C .

Although turbulence is not expected, it has been observed some instabilities in preliminary studies and the model could not have a steady solution. For this purpose a turbulence model is used. The $k-\omega$ SST (Shear Stress Transport) turbulence model is used [42, 43, 79]. This turbulence model is widely used for heat transfer predictions [58, 1]. This model fails in some flows while performs well in others. In particular, in [1] it is found that the model yields accurate predictions for natural convection in an enclosure even in the laminar to turbulence transition.

In the ambient boundary, the turbulence intensity is set to 0.05% . The turbulent viscosity ratio is set to 10^{-4} . The above model is reproduced for 10 equidistant inclinations from 0° to 90° . These cases will be simulated and their results presented in the following.

The finite volume method is used [45] to solve numerically the equations. The grid is constructed with ANSYS Meshing, see [3]. It uses tetrahedral elements. The receiver walls have a spacing of 1 cm and 10 boundary layers with a transition ratio of 0.272. In the cavity wall the elements grow to the domain with a rate of 1.1. The remaining faces grow with a rate of 1.2. The far field elements have a length of 10 cm.

Case Id	Elements	Heat loss (W)
1	10^5	53
2	2.8×10^5	53.3
3	3.3×10^5	53.1

Table 3.7: Mesh independence for the case for natural convection in the OCR.

The numerical procedure is the SIMPLEC method [45] with 1 skewness correction. The software used is ANSYS Fluent, see [3]. The gradient discretization is least squares cell based. The pressure discretization is second order. The momentum discretization is second order upwind. The turbulent kinetic energy and specific dissipation rate discretizations are first order upwind. Finally the energy discretization is second order upwind.

The absolute criteria for reaching convergence for residuals are 10^{-3} , except for energy which is 10^{-6} . The heat loss is also monitored until no changes are observed for 500 iterations with a precision of 0.01 W. The under-relaxation factors are left to the default values except for energy which is lowered to 0.9.

The grid has been checked for independence with the solution, whose results are shown in Table 3.7. The computational model is tested for the $\theta = 0^\circ$ with global grid refinement of about 1.5 times each step. Although it is not shown in detail, the boundary resolution in the ORC walls has been checked and a relatively good resolution has been found, in spite of the relatively small number of boundary layers used in the grid construction described above. The mesh of the case 2 of Table 3.7 is used for the simulation of the CFD models, which is the mesh described above.

From each simulation we obtain the heat flow rate in the cavity wall, from which the Nusselt number Nu_{nat} is computed for several values of the OCR inclination θ , by using the procedure detailed in [6]. The result is shown in Fig. 3.17. The temperature field for some selected inclinations θ (0° , 30° , 60° , and 90°), are shown in Fig. 3.18, 3.19, 3.20, and 3.21, respectively. The increase of stagnant zone is related to the decrease in the convective loss.

A good fitting of Nu_{nat} to the simulated data is given by the following expression:

$$Nu_{nat} = 0.072(1 + 5.75(\cos \theta)^{1.95})Ra^{1/4}. \quad (3.6)$$

It has been checked that this expression holds for the working range of temperatures, from 150°C to 250°C .

Figure 3.17: Inclination and Nusselt number results.

3.3.2 Forced convection

Many of the elements for natural convection can be reused for performing a CFD simulation of side-on wind in the OCR. Only the side-on direction is considered since it is believed that it is more frequent than other directions, because the collector could shield the other directions. This direction is generally found to be one which yields more heat losses by forced convection, see for example [56].

The computation domain contains the same OCR geometry used for natural convection. It is placed in a cylinder that simulates a wind tunnel, with a diameter $10L$ and height $15L$, [54, 56, 45]. The receiver is placed at $3L$ from the inlet.

The inlet is set to a prescribed speed. The outlet is set to an output boundary condition at the normal pressure. The side wall is set with zero shear stress.

The computation grid is constructed in the following way. The cavity wall have a length of 5 mm. The inflation layers are 20. The insulated and front walls have a length of 1 cm. An edge refinement is applied in all the edges except the one of the dome of the bottom, with a length of 5 mm. Finally three spheres of influence are set to refine the downwind volume. They are placed in the downwind axis direction at distances 0 m, 0.75 m, and 1.5 m from the center of the cavity aperture. Each sphere has a diameter of 1 m, and an element size of 5 cm. Where the spheres of influence affect the local sizing control elements, a hard behaviour is necessary to keep the desired local sizing.

Figure 3.18: Temperature contours in the symmetry plane for natural convection for an inclination of $\theta = 0^\circ$.

Case Id	Elements	Heat loss (W)
1	5×10^5	202
2	8×10^5	219
3	1.3×10^6	210

Table 3.8: Mesh independence for the case for forced convection in the OCR.

The selected wind speeds are 9 equidistant points in the interval from 10 km/h to 60 km/h. We assume that for no-wind condition no heat loss is produced. This is an approximation since heat by conduction should be present. However in realistic conditions, natural convection is present for very low speeds.

Based on the above grid, a grid refinement study is carried out. The selected case is the maximum speed. The results are shown in Table 3.8. We select the grid of the case 2 which is the grid described above. As with natural convection the boundary layer resolution of the proposed grid has also been checked. It has been found that temperature and velocity are relatively well represented with 20 boundary layers. Moreover a maximum y^+ value of about 5 is obtained which we consider to be appropriate since the enhanced wall treatment model is used.

The obtained simulation data are shown in Fig. 3.22. From these numerical data, it has been found a good fitting for the Nusselt number Nu_{for} versus wind

Figure 3.19: Temperature contours in the symmetry plane for natural convection for an inclination of $\theta = 30^\circ$.

speed. It is given by

$$Nu_{for} = 0.0522 Re_L^{0.7} Pr^{1/3}. \quad (3.7)$$

The flow streamlines for some selected speeds (22.5 km/h and 47.5 km/h) are shown in Fig. 3.23 and 3.24. The vortex generated in the cavity interior is responsible for the heat loss. Interestingly, a similar pattern is found experimentally in the work [84]. Comparable temperature patterns can also be observed in [56]. The temperature contours for the same cases are shown in Fig. 3.25 and 3.26.

3.4 Energy transfer in the proposed cavity receiver

In the proposed cavity receiver, the solar radiation that is absorbed in the walls, is eventually transferred to the HTF. The solar radiation that is absorbed in the tiny thickness of the wall where the solar paint is present, is distributed through the interior of the receiver consisting of the solid and fluid parts. The characteristics of this energy transfer process are key to understand how the geometry and materials of the propose receiver perform and to detect pitfalls without the need of experimenting with the actual system [81, 61]. In particular,

Figure 3.20: Temperature contours in the symmetry plane for natural convection for an inclination of $\theta = 60^\circ$.

a detailed knowledge of the field of temperature in the cavity receiver is of interest, since it will help to detect temperature peaks, temperatures above the safety temperature of the HTF, temperatures near the softening point of the steel, etc. The flow field is also of interest, since it determines the zones of low speeds, pressure loss, flow distribution, etc. The aim of this study is that from a such detailed knowledge, better designs can be proposed in further studies.

The models used in CFD are highly detailed and they can give an accurate picture of the reality. They are specially accurate at describing the energy transfer process of solar thermal concentrators [28, 25]. In fact, CFD is increasingly used in the literature of solar thermal concentrators [60]. Thus, it is expected that a CFD simulation of the proposed cavity receiver of this thesis will provide a detailed understanding of the energy transfer process. This is the topic of this section.

3.4.1 Modelling

In the following, the convective heat transfer problem for the cavity interior is considered. The receiver interior is shown in Fig. 2.4 of section 2.3.2.

Unless otherwise stated, it is assumed that the insulator part is made from foil batts insulation. The supporting bars are suppressed for the sake of simplicity since heat loss by conduction can be neglected for small contact areas.

Figure 3.21: Temperature contours in the symmetry plane for natural convection for an inclination of $\theta = 90^\circ$.

The heat transfer fluid (HTF) is a synthetic oil [59]. The physical properties of the materials used are shown in Table 3.9.

The model is stationary. Four system volumes can be distinguished: HTF, Steel, Insulator and Ambient. They can be distinguished in Fig. 2.4.

In the HTF volume, the wetted volume filled with HTF, the incompressible Navier-Stokes equations are posed. This is where convective heat transfer occurs. No generation of energy by friction is included. The turbulence model k- ω SST (Shear Stress Transport)[79, 42, 43] is also posed, due to the flow instabilities and turbulence. At the operating points selected for study, it is expected some turbulence since the Reynolds numbers point so: $10^4 < Re_{\text{inlet diameter}} < 10^5$; the flow is unstable at least near the obstructions. This turbulence model is selected because it gives accurate results [1]. In these works the turbulence model is under conditions that can be considered similar to the cases considered here. An important characteristic of this model is its near wall accuracy, which is appropriate for wall heat transfer calculations [79].

Thermal radiative heat transfer in the HTF volume is also modelled. The radiative heat transfer equation is posed [44]. The oil is considered transparent at wavelengths $8\mu\text{m}$, thus its attenuation coefficient, a , is zero. This is a representative wavelength for thermal radiation. The walls in contact with the oil are considered opaque, ideally diffusive and grey.

The insulator and steel bodies are solid bodies where heat transfer is well

Figure 3.22: Nusselt number for natural convection in the cavity receiver over the inclination angle.

modelled by conduction [6]. Perfect contact is assumed between solid walls.

Boundary Conditions

A fixed pressure is set at the flow outlet, while a prescribed speed is set at the inlet. The outlet and inlet surfaces can be identified in Fig. 2.4. The speed is given by the desired volume flow rate of HTF $\dot{V}$. Zero gradient conditions are used for the boundary conditions of the remaining fields.

The insulator surface in contact with the surrounding air is coupled with natural and radiative heat transfer modelling heat loss to ambient. Natural convection losses are given by the following expression:

$$q = h_{\text{conv,amb}}(T_{\text{wall}} - T_{\text{amb}}) \quad (3.8)$$

where $h_{\text{conv,amb}}$ is the coefficient of heat transfer. It is set as constant given in Table 3.10. This value is representative of natural convection for a general immersed body [6].

Thermal radiation losses are given by the following expression:

$$q = \epsilon_t \sigma (T_{\text{wall}}^4 - T_{\text{sky}}^4) \quad (3.9)$$

where $\epsilon_t = \alpha_t$ is the emissivity in the thermal band, and α is the absorptivity. The values are considered constant given in Table 3.9.

Figure 3.23: Streamlines for $v_{\text{wind}} = 22.5 \text{ km/h}$.

The irradiance profile obtained from a Monte-Carlo Ray Tracing simulation [36](see below), is added to the active walls as a fixed heat source. By this way the solar irradiance is taken into account in the model. The sky temperature, T_{sky} is considered to be 5 K below ambient, which is representative of a humid continental climate [19].

The turbulence intensity at the inlet and outlet of the receiver are set to 5 % [79]. This value is representative of low turbulence levels for fully developed flow in a bounded flow.

On the other hand, the temperature at the inlet of the HTF T_{inlet} is fixed to the value given in Table 3.10, which is representative of the operating temperature of the PDC.

The irradiance profile from the parabola is necessary to characterize the absorber. Solar radiation is modelled by the radiative heat transfer equation. The irradiance profile is obtained from [36]. The error for the concentrator, $\sigma_{\text{conc,error}}$, and the Direct Normal Irradiation, DNI, are given in Table 3.10. The resulting irradiance distribution in the receiver walls is shown in Fig 3.27. This profile is used as a fixed heat source as explained above.

Temperature and pressure characterization

The increase in temperature is defined as $\Delta T = T_{\text{outlet}} - T_{\text{inlet}}$. The pressure drop is computed as: $\Delta p = p_{\text{inlet}} - p_{\text{outlet}}$. In these expressions, T and p are

Figure 3.24: Streamlines for $v_{\text{wind}} = 47.5 \text{ km/h}$.

total temperature and total pressure respectively. They are computed as the area average over the considered surface.

3.4.2 Results

Simulation

The model is solved numerically by the Finite Volume Method[45]. The momentum and transport equations are solved by using the pressure-based SIMPLEC method with 1 skewness correction. Linear upwind spatial discretization is used. For the radiation modelling the Discrete Ordinates method is used[44]. Theta and phi divisions are set to 5, and 3 pixels are used.

The convergence criteria are a relative residual of 10^{-3} for momentum, continuity and turbulence quantities, and 10^{-6} for energy and radiation. In these simulations the mean outlet temperature and velocity, T_{outlet} and v_{outlet} are monitored until a steady value is reached over the solver iterations. In some cases when better initial conditions are set, some residuals may not drop below the previous limits. In that cases it is checked that they and the monitored variables are stable. The mesh has been recomputed until a reasonable mesh has been obtained (refinement in the necessary zones). Since qualitative results are sought the mesh is considered to be good enough for representing the solution and some figures of interest.

Figure 3.25: Temperature contour at the symmetry plane for $v_{wind} = 22.5 \text{ km/h}$.

Results

The fixed operating conditions are shown in Table 3.10. For $110 \leq \dot{V} \leq 1100 \text{ L h}^{-1}$, 10 values are studied. The temperature and velocity fields are shown in Figs. 3.28, 3.29, 3.30 and 3.31 for selected $\dot{V}$ values.

From Fig. 3.32 and Fig. 3.33 it is found that an optimal working operating point exists. As the speed increases the required work done by the pump is greater but the temperature difference is smaller. For low speeds the pump power is lower but the temperature is higher, giving a higher thermal loss and degrading the overall energy transfer process.

3.4.3 Conclusions

The nonuniform irradiance distribution generated from the parabola and the relatively low velocity of the oil in some areas generate a highly varying temperature distribution in the receiving walls. There are considerable peaks of temperature, but the materials should tolerate them. However, high peaks tend to degrade the materials and put some safety concerns due to the material integrity under unwanted events (system emergency stops, etc.). The peaks seem to be located in the steel, and the thermal energy seems to be redistributed inside the cavity relatively well. In order to smooth the temperature distribution, the velocity pattern of the oil should be changed. A new flow distribution is desirable which

Figure 3.26: Temperature contour at the symmetry plane for $v_{wind} = 47.5 \text{ km/h}$.

is achievable with a change in the design. New designs that maintain the main advantages of the shown receiver are advisable for future investigations. On the other hand, the model for the losses could be improved to include varying coefficients with temperature, which are held constant in this work.

Table 3.9: Physical properties of the materials used

Dimension	Property	Value
MarlothermSH	n	1.5
	a	0
Batt Insulator	ρ	20
	c_p	1030
	k	0.035
	α_t	0.5
Steel	α_t	0.5

Table 3.10: Operating conditions and parameters

$T_{\text{inlet}} = 150\text{ }^{\circ}\text{C}$
$T_{\text{amb}} = 27\text{ }^{\circ}\text{C}$
$T_{\text{sky}} = 22\text{ }^{\circ}\text{C}$
$p_{\text{amb}} = 1\text{ atm}$
$h_{\text{conv,amb}} = 6.5\text{ W m}^{-2}\text{ K}^{-1}$
$\sigma_{\text{conc,error}} = 7.28\text{ mrad}$
$\text{DNI} = 1000\text{ W m}^{-2}$

Figure 3.27: Irradiance profile at the cavity receiver walls. In the picture the receiver is oriented so that the bottom of the cavity is shown first.

Figure 3.28: Temperature and velocity distributions for $\dot{V} = 1100 \text{ L h}^{-1}$. The flow pattern does not change appreciably and the temperature peaks in the black walls increase. The temperature at the insulator increases considerably.

Figure 3.29: Temperature and velocity distributions for $\dot{V} = 800 \text{ L h}^{-1}$. The mayor temperatures are reached in the interior of the cavity while the front black face is cooler. The cold oil enters and is heated as it is advected to the outlet.

Figure 3.30: Temperature and velocity distributions for $\dot{V} = 450 \text{ L h}^{-1}$. The mayor temperatures are reached in the interior of the cavity while the front black face is cooler. The cold oil enters and is heated as it is advected to the outlet.

Figure 3.31: Temperature and velocity distributions for $\dot{V} = 110 \text{ L h}^{-1}$. The mayor temperatures are reached in the interior of the cavity while the front black face is cooler. The cold oil enters and is heated as it is advected to the outlet.

Figure 3.32: Relation between the temperature drop and flow speed.

Figure 3.33: Relation between the pressure drop and flow speed.

Chapter 4

Thermal performance of the receivers

In this chapter, the thermal performance of the PDC defined in section 2.3.1 is analysed for each one of the following three thermal receivers: the proposed cavity receiver of section 2.3.2, an equivalent flat-plate receiver, and the proposed cavity receiver with a cover.

Each system is modelled with a lumped-element thermal model in order to compute the energy flows of interest. A statistical average of the thermal efficiency over a selected typical meteorological year (TMY) is used for quantifying the thermal performance of the systems. The used TMY is for the period 2006 to 2014, containing data of DNI, wind speed, and ambient temperature for the location of the PDC as defined in section 2.3.1. The lumped-element models are based on first principles and some assumptions that will be detailed along the exposition. The models are numerically solved in a computer by implementing it directly in the modelling language Modelica®¹ [72]. The assumptions include heat losses such as reradiation, convection, and conduction. Convection includes natural and forced convection. Forced convection is included, since it can be important even for a light breeze.

4.1 Model of the parabolic dish collector

A sketch of the PDC system with the general receiver $\mathcal{R}$ is shown in Fig. 4.1. The PDC is to transform solar energy into internal thermal energy on the HTF.

¹Modelica® is a registered trademark of the Modelica Association.

Figure 4.1: Sketch of the model for the PDC with the generic thermal receiver R .

Focal distance f	2.2 m	Rim Φ	54.2°
$D_{\text{conc ap}}$	4.5 m		
$A_{\text{conc ap}}$	15.9 m ²	A_{conc}	16.9 m ²

Table 4.1: Collector dimensions.

This technology involves diverse aspects of solar engineering: solar tracking, concentrator optics, radiative heat transfer, and heat transfer by conduction and convection.

The PDC has an aperture surface with area $A_{\text{conc ap}}$, whose value is given in Table 4.1. It is formed by the concentrator and the receiver. The concentrator is a paraboloid of focal length f and rim angle Φ , with the focus denoted by $\mathbf{F}$. The parabola axis is $\mathbf{z}$ which is a symmetry axis. $A_{\text{con}} > A_{\text{conc ap}}$ is the area of the mirror. The receiver $\mathcal{R}$ is centered along the axis $\mathbf{z}$, with the aperture containing $\mathbf{F}$.

The general receiver $\mathcal{R}$ is considered to describe the common parts of the models for the three receivers. The specificities will be described in three sections for each one of them.

The receiver $\mathcal{R}$ operates at a temperature denoted by $T_{\mathcal{R}}$. The characteristic temperature of the HTF during its passage through the receiver is denoted by $T_{\mathcal{HTF}}$. It is assumed that $T_{\mathcal{R}} = T_{\mathcal{HTF}}$.

The surroundings are the ambient air of the atmosphere. The ambient temperature is denoted by T_{amb} . The air is modelled by an ideal dry gas with NASA

Glenn coefficients [26], already available in the MSL. A constant normal pressure is assumed, $p_{\text{amb}} = 1 \text{ atm}$.

Wind is modelled as a uniform flow with speed v_{wind} in a fixed side-on direction [56]. The concentrator and bars are not taken into account in the analysis.

The inclination of the sun is θ and the direct normal irradiation is E_{DNI} [81].

The rate of heat transfer to the PDC by solar radiation is $\dot{Q}_{\text{solar}}$. It includes only the radiation scoped by the concentrator aperture. The heat transfer to the HTF from the PDC (which occurs in the interior of the receiver) is $\dot{Q}_{\text{HTF}}^{\mathcal{R}}$. Energy loss to the ambient is represented by the rate of heat transfer $\dot{Q}_{\text{loss}}$.

The PDC follows a steady-flow process [12]. It has the following form of the energy balance

$$\dot{Q}_{\text{solar}} = \dot{Q}_{\text{HTF}}^{\mathcal{R}} + \dot{Q}_{\text{loss}}. \quad (4.1)$$

The value of $\dot{Q}_{\text{solar}}$ is given by the relation

$$\dot{Q}_{\text{solar}} = E_{\text{DNI}} A_{\text{conc ap}}. \quad (4.2)$$

The loss $\dot{Q}_{\text{loss}}$ is the result of the losses in the concentrator, spillage, and receiver:

$$\dot{Q}_{\text{loss}} = \dot{Q}_{\text{conc}} + \dot{Q}_{\text{spill}}^{\mathcal{R}} + \dot{Q}_{\text{loss}}^{\mathcal{R}}. \quad (4.3)$$

The rate of heat transfer due to the concentrated radiation impinging on the receiver from the concentrator is given by $\dot{Q}_{S_{\mathcal{R}}}$. The rate of heat loss in the receiver is $\dot{Q}_{\text{loss}}^{\mathcal{R}}$. The energy balance in the receiver gives the following relation

$$\dot{Q}_{S_{\mathcal{R}}} = \dot{Q}_{\text{HTF}}^{\mathcal{R}} + \dot{Q}_{\text{loss}}^{\mathcal{R}}. \quad (4.4)$$

The loss in the receiver $\dot{Q}_{\text{loss}}^{\mathcal{R}}$ is the result of the losses from reradiation, convection, and conduction:

$$\dot{Q}_{\text{loss}}^{\mathcal{R}} = \dot{Q}_{\text{rerad}}^{\mathcal{R}} + \dot{Q}_{\text{conv}}^{\mathcal{R}} + \dot{Q}_{\text{cond}}^{\mathcal{R}}. \quad (4.5)$$

4.1.1 Thermal performance of the PDC

The thermal performance is quantified by the following variables: annual average thermal efficiency, fraction of losses, and maximum temperature. The fraction of losses are for the losses: spillage, convection, reradiation, and conduction.

4.1.2 Thermal efficiency

Thermal efficiency is a figure of great practical importance in any solar power and thermal plant. The thermal efficiency of the PDC is defined as the ratio between the delivered energy to the HTF to the total energy from the solar fuel (solar energy) [81]

$$\eta_{\mathcal{R},t}(t) = \frac{\dot{Q}_{\text{HTF}}^{\mathcal{R}}(t)}{\dot{Q}_{\text{solar}}(t)}, \quad (4.6)$$

where t is the time.

Following a similar approach to that of [27], an annual average of the thermal efficiency of the PDC is used to quantify the thermal performance of the system. The annual average thermal efficiency of the PDC is denoted by $\eta_{\mathcal{R}}^2$, and computed as

$$\eta_{\mathcal{R}} = \frac{1}{T} \int_{\text{TMY}} \eta_{\mathcal{R},t}(t) dt, \quad (4.7)$$

where T is the duration of the TMY.

The data for a year of operation is provided by the selected typical meteorological year (TMY) of section 4.1.4.

4.1.3 Fractions of losses

The fractions of losses are defined now. They are very similar to the fractions of losses defined in [27].

- Concentrator: Solar radiation absorbed in the concentrator. It is represented by $\dot{Q}_{\text{conc}}$.
- Spillage: Radiation falling outside of the receiver. The rate of heat transfer by spillage is denoted by $\dot{Q}_{\text{spill}}^{\mathcal{R}}$.
- Reradiation: Thermal radiation loss and solar radiation emissions. In the thermal region the receiver exchanges radiation with the sky. The parabola is not considered. The net outgoing solar radiation to sky is taken, but not the energy gained by solar energy, which is already accounted in $\dot{Q}_{\text{solar}}$. Reradiation includes the energy gained by diffuse radiation from the atmosphere. The rate of heat transfer of reradiation is denoted by $\dot{Q}_{\text{rerad}}^{\mathcal{R}}$. The radiation from the atmosphere is the sky model described in [19]. The sky

²Fractions are expressed in percent or in per unit interchangeably throughout the text.

is considered as large black surface surrounding the receiver at a uniform temperature T_{sky} .

- Convection: Convection heat loss to the atmosphere through the receiving walls. The rate of heat loss of convection is denoted by $\dot{Q}_{\text{conv}}^{\mathcal{R}}$.
- Conduction: Heat loss through the insulator. The rate of heat loss by conduction is denoted by $\dot{Q}_{\text{cond}}^{\mathcal{R}}$. Supporting bars and other elements are neglected.

The instantaneous fraction of loss for a given loss $\mathcal{L}$ is given by the following equation

$$\beta_{\mathcal{L},t} = \frac{\dot{Q}_{\mathcal{L}}}{\dot{Q}_{\text{solar}}}. \quad (4.8)$$

As it can be observed, fraction of losses carry the same meaning than efficiency. Sometimes the following notation change is done: $\beta_{\mathcal{L},t}^{\mathcal{R}} = \beta_{\mathcal{L},t}$, and $\dot{Q}_{\mathcal{L}}^{\mathcal{R}} = \dot{Q}_{\mathcal{L}}$.

The annual averages of the fraction of losses are used to quantify the thermal performance. The annual average of the fraction of loss $\mathcal{L}$ is

$$\beta_{\mathcal{L}} = \frac{1}{T} \int_{\text{TMY}} \beta_{\mathcal{L},t}(t) dt, \quad (4.9)$$

Sometimes the following notation change is done: $\beta_{\mathcal{L}}^{\mathcal{R}} = \beta_{\mathcal{L}}$.

The considered fraction of losses are:

- concentrator β_{conc} ,
- spillage $\beta_{\text{spill}}^{\mathcal{R}}$,
- reradiation $\beta_{\text{rerad}}^{\mathcal{R}}$,
- convection $\beta_{\text{conv}}^{\mathcal{R}}$,
- conduction $\beta_{\text{cond}}^{\mathcal{R}}$, and
- loss at the receiver $\beta_{\text{loss}}^{\mathcal{R}}$.

4.1.4 Typical meteorological year

The Typical meteorological year (TMY) from 2006 to 2014 is selected [68, 29]. This period is chosen because it is the most recent one. The data are hourly time series providing the following parameters of the model: E_{DNI} , T_{amb} , and v_{wind} . The solar inclination θ is generated by a solar position algorithm, see for example [19]. The data of the TMY and solar inclination are for the location of the PDC given in section 2.3.1.

Annual averages are simplified by the same considerations of [27]. For this, two approximations are considered. Firstly, it is assumed that a reduced subset of representatives days of the TMY gives an estimation of the exact figures. Secondly, since the thermal inertia is almost constant for hour-time intervals, then a stationary model can be used to approximate the average hourly thermal efficiency for an hour by solving the model for the average values for that hour of the TMY. As a result, the efficiency is computed by solving the model for each hourly data from the three selected days. The efficiency is computed from the total energies. Total energies are the sum of the energies at each hour computed with the model.

The selected days are given in Table 4.2. A constant normal pressure is assumed. The data includes the effects of cloud blocking and wind. The three selected days are representative of the seasons: winter solstice, equinox, and summer solstice.

	Hour ^a	$E_{\text{DNI}}/\text{W}/\text{m}^2$	$T_{\text{amb}}/^{\circ}\text{C}$	$\theta/^{\circ}$	$v_{\text{wind}}/\text{km}/\text{h}$
Winter solstice ^b	8	459	11	10	26
	9	608	12	19	28
	10	429	13	26	29
	11	467	14	30	31
	12	266	14	31	33
Equinox ^c	7	546	13	12	7
	8	701	15	24	7
	9	824	16	35	10
	10	928	17	45	14
	11	995	18	52	17
	12	1022	18	56	18
Summer solstice ^d	5	48	23	1	4
	6	399	25	11	5
	7	596	27	23	6
	8	702	29	35	7
	9	770	30	46	9
	10	826	32	58	10
	11	850	33	68	11
	12	889	33	74	12

^a Time scale UTC.

^b Selected day 24/11/2007.

^c Selected day 29/3/2012.

^d Selected day 12/7/2012.

Site: Murcia, Spain (38.0204°N, 1.1709°W, altitude 98 m).

Model: Hourly data during a typical meteorological year (TMY) for the period 2006 to 2014, from [68, 29].

The solar inclination is generated by a solar position algorithm (SPA), see for example [19].

Table 4.2: Solar and weather parameters.

4.1.5 Maximum temperature of the receiver

The last parameter considered for the quantification of the thermal performance of the PDC is the maximum temperature reached by the receiver, denoted by $T_{\mathcal{R},\max}$.

4.2 Optics of the concentrator

The rate of total heat flow of solar radiation which is available for the receiver is the radiation reflected in the concentrator $\dot{Q}_{\text{reflect}}$, given as

$$\dot{Q}_{\text{reflect}} = \rho_{\text{conc}} \dot{Q}_{\text{solar}}, \quad (4.10)$$

where ρ_{conc} is the hemispherical reflectance [44].

A simple energy balance in the concentrator gives

$$\dot{Q}_{\text{solar}} = \dot{Q}_{\text{reflect}} + \dot{Q}_{\text{conc}}. \quad (4.11)$$

The fraction of loss β_{conc} is given by

$$\beta_{\text{conc}} = 1 - \rho_{\text{conc}}. \quad (4.12)$$

The concentrator has a constant reflectance of $\rho_{\text{conc}} = 0.86$ [63], thus

$$\beta_{\text{conc}} = 0.14 = 14\%. \quad (4.13)$$

4.2.1 Solar radiation in a circle at the focal plane

It is assumed that the radiation is diffuse. The flux density concentration ratio is C_f [80]. A circle $S_{\mathcal{R}}$ of radius $r(S_{\mathcal{R}})$ around focal point is considered. The rate of heat transfer due to the solar radiation impinging in $S_{\mathcal{R}}$ from the concentrator is

$$\dot{Q}_{S_{\mathcal{R}}} = E_{\text{DNI}} \int_{S_{\mathcal{R}}} C_f dS, \quad (4.14)$$

where dS is a area element of $S_{\mathcal{R}}$.

For the collector, the C_f is an axisymmetric distribution around the focal point computed in section 4.2.2, thus $C_f = C_f(r)$, where r is the radius. Thus, Eq. (4.14) gives

$$\dot{Q}_{S_{\mathcal{R}}} = 2\pi E_{\text{DNI}} \int_0^{r(S_{\mathcal{R}})} C_f(r) r dr. \quad (4.15)$$

Figure 4.2: Sunshape $S_s(\xi)$.

4.2.2 Computation of the flux concentration

The distribution C_f is computed by a ray-tracing model of the PDC like it is done in section 3.1. Here, only one experiment to identify C_f is considered. The experiment conditions are given by the Standard Test Conditions for concentrating collectors [80], which set $E_{\text{DNI},0} = 850 \text{ W/m}^2$.

The sun is characterized by an object at infinity in the z direction, having the shape of a disk with an angular radius of $\alpha_s = 4.635 \text{ mrad}$ observed from the PDC position. The angular distribution of solar radiation is given by the sunshape function of [9], which also models limb darkening. As suggested in [9], it is selected the circumsolar ratio (CSR) of $\chi = 5\%$ for being representative of the majority of atmospheric conditions and worldwide areas. The sunshape S_s is

$$S_s(\xi) = \begin{cases} \frac{\cos(0.326\xi)}{\cos(0.308\xi)} & \text{for } \xi \in [0, \alpha_s] \text{ mrad} \\ e^{\kappa\xi^\gamma}, & \\ \kappa = 0.9 \ln(13.5\chi)\chi^{-0.3}, & \\ \gamma = 2.2 \ln(0.52\chi)\chi^{0.43} - 0.1, & \text{for } \xi \in (\alpha_s, \alpha_l] \text{ mrad} \end{cases}, \quad (4.16)$$

where α_l is the aureole angular radius $\alpha_l = 43.6 \text{ mrad}$. It is graphed in Fig. 4.2.

The error function for the concentrator [80] is used to model its imperfect

Figure 4.3: Flux concentration distribution C_f at the focal plane. The distribution of C_f is axisymmetric in the z axis.

optics. A simple Gaussian distribution is used with a standard deviation of σ_B ,

$$R(\Delta\theta) \propto e^{-\frac{\Delta\theta^2}{2\sigma_B^2}}, \quad (4.17)$$

where $\Delta\theta$ is the disturbance in the reflection angle [81]. The constant of proportionality can be computed by taking into account that the hemispherical reflectance is that of the concentrator ρ_{conc} . Only the specular and slope errors are considered, hence σ_B is given by

$$\sigma_B^2 = \sigma_{\text{specular}}^2 + (2\sigma_{\text{slope}})^2. \quad (4.18)$$

The selected values are $\sigma_{\text{specular}} = 2 \text{ mrad}$ and $\sigma_{\text{slope}} = 3.5 \text{ mrad}$, which are approximately the values used in the works [51], [62, 27]. Thus, $\sigma_B = 7.28 \text{ mrad}$. A similar value is found in [81, p. 111].

Ray-tracing is used to solve for the radiative energy transport [44]. The number of rays was 10^6 which was found to reduce random noise to a sufficient level for the purposes of this work. The resulting C_f is shown in Fig. 4.3.

4.3 Thermal receivers

The three thermal receivers are:

- The open cavity receiver (OCR) which is the proposed receiver presented of section 2.3.2.
- The covered cavity receiver (CCR) is the OCR with a window glass covering the aperture. The window glass or cover is a silica glass of thickness $T = 5$ mm.
- A flat-plate receiver (FPR) that is equivalent to the OCR in that the cavity is replaced by a flat wall. The other dimensions are the same.

4.3.1 Spillage fraction, $\beta_{\text{spill}}^{\mathcal{R}}$

The interception area of the three receivers is the same, which results in that the spillage is the same for all the receivers,

$$\beta_{\text{spill}}^{\mathcal{R}} = 0.07. \quad (4.19)$$

Note that adding concentrator reflection and spillage losses, Eq. (4.13) and (4.19), respectively, the result is a loss of already 21 %. Thus, the efficiency is limited to 79 % due to the concentrator and spillage losses.

4.4 PDC configured with OCR

The PDC with the OCR is detailed in the following.

The OCR system is shown in Fig. 4.4, with dimensions given in Table 2.1. The OCR is divided in 6 subsystems: cavity wall, front wall, insulator, cavity volume, and the cavity interior or receiver. The following temperatures are considered: T_{amb} , T_{cav} , T_{front} , T_{ins} , T_{insint} , and T_{OCR} . In the insulator a temperature gradient given by T_{ins} and T_{insint} is assumed. In the other subsystems the temperature is uniform.

The receiver volume has four surfaces $W = \{\text{front, cav, insint, HTF}\}$. The insint surface is the surface in common with the insulator.

The cavity aperture surface is denoted by cap.

Figure 4.4: Cavity receiver system.

4.4.1 Impinging solar radiation

By using Eq. (4.15), the solar radiation impinging at the cap surface, denoted by $\dot{Q}_{S_{OCR,cap}}$ is given by

$$\dot{Q}_{S_{OCR,cap}} = 2\pi E_{DNI} \int_0^{D_{cav}} C_f(r) r dr, \quad (4.20)$$

where D_{cav} is the diameter of the cavity aperture.

By using the C_f distribution of section 4.2.2, the integral of Eq. (4.20) is

$$2\pi \int_0^{D_{cav}} C_f(r) r dr = 11.29. \quad (4.21)$$

For the front surface the rate of heat transfer,

$$\dot{Q}_{S_{CCR,front}} = 2\pi E_{DNI} \int_{D_{cav}}^D C_f(r) r dr, \text{ and} \quad (4.22)$$

$$2\pi \int_{D_{\text{cav}}}^D C_f(r) r dr = 1.28. \quad (4.23)$$

Thus, the rate of heat transfer due to the impinging solar radiation in cavity aperture and front surfaces are

$$\begin{aligned} \dot{Q}_{S_{\text{OCR}},\text{cap}} &= 11.29 \cdot E_{\text{DNI}}, \\ \dot{Q}_{S_{\text{OCR}},\text{front}} &= 1.28 \cdot E_{\text{DNI}}. \end{aligned} \quad (4.24)$$

The rate of heat transfer of the total radiation impinging in the receiver is $\dot{Q}_{S_{\text{OCR}}}$,

$$\dot{Q}_{S_{\text{OCR}}} = \dot{Q}_{S_{\text{CCR}},\text{cap}} + \dot{Q}_{S_{\text{CCR}},\text{front}} = 12.57 \cdot E_{\text{DNI}}. \quad (4.25)$$

4.4.2 Radiation in the cavity volume

To analyse the surface radiation transport in the cavity volume, it is assumed that the the cavity volume as an idealized enclosure consisting of N subsurfaces, over each of which the radiosity is assumed constant [44]. The enclosure is divided in $N = 2$ subsurfaces: cavity surface (*cav*) and the aperture surface (*cap*). The surfaces are also sometimes numbered as follows: 1 for *cav* and 2 for *cap*. Each surface i has a uniform temperature T_i where $i = 1, 2, \dots, N$. The cavity bottom is assumed as a flat surface.

It is assumed that solar radiation enters diffusively and collimated radiation is not present. All the surfaces are considered opaque, directionally ideal (diffuse), diffuse reflectors, and non-gray. The cavity aperture is considered as a semitransparent sheet or window, which will be helpful for the CCR analysis.

The net radiation method of [44] is applied. The surfaces are non-gray and are represented by the use of the semigray approximation where the energy balances are separated for different spectral regions. We consider two spectral regions: thermal (*th*) and solar regions (*s*).

For a spectral region λ , an energy balance gives the following relations [44]

$$\sum_{j=1}^N [\delta_{ij} - (1 - \rho_j^\lambda F_{ij}) E_{b,j}^\lambda] = \sum_{j=1}^N \left(\frac{\delta_{ij}}{\epsilon_j^\lambda} - \frac{\rho_j^\lambda}{\epsilon_j^\lambda} F_{ij} \right) q_j^\lambda, \quad i = 1, 2, \dots, N, \quad (4.26)$$

where δ_{ij} is the Kronecker delta, ρ is the reflectance, F_{ij} is the view factor, ϵ_j is the emittance, $E_{b,j}$ is the total black body emissive power, and q_j radiative heat flux [44].

In the semigray approximation, the net radiative flux for the surface i is given by the following relation [44]

$$q_i = \sum_{\lambda \in \{\text{th}, \text{s}\}} q_i^\lambda. \quad (4.27)$$

For a transparent or semitransparent wall, the apparent emittance ϵ_a and the apparent blackbody emissive power E_{ba} [44] are given by the following relations

$$\epsilon_a = \epsilon + \tau = 1 - \rho, \quad (4.28a)$$

$$\epsilon_a E_{ba} = \epsilon E_b + \tau q_{od}. \quad (4.28b)$$

The blackbody emissive power of surface j , $E_{b,j}$ is given by the relation [44]

$$E_{b,j} = \sigma T_j^4. \quad (4.29)$$

The view factors used in equation (4.26) are given by a uniform cavity [44]

$$F_{1 \rightarrow 1} = 1 - \frac{A_2}{A_1}, \quad (4.30)$$

$$F_{1 \rightarrow 2} = \frac{A_2}{A_1}, \quad (4.31)$$

$$F_{2 \rightarrow 1} = 1, \text{ and} \quad (4.32)$$

$$F_{2 \rightarrow 2} = 0, \quad (4.33)$$

where A_i denotes the area of surface i , where $i = 1, 2$.

The above model can be applied for the OCR by considering that the cavity aperture is a transparent window, for which the apparent emittance ϵ_a^λ and the apparent blackbody emissive power $E_{ba,2}^\lambda$ are [44]

$$\epsilon^\lambda = 0, \quad (4.34)$$

$$\tau^\lambda = 1, \quad (4.35)$$

The following relations between the radiative properties of a surface are used [44]

$$\alpha^\lambda + \tau^\lambda + \rho^\lambda = 1, \quad (4.36)$$

$$\alpha^\lambda = \epsilon^\lambda. \quad (4.37)$$

The net absorbed solar radiation in the cavity surface, $\dot{Q}_{\text{rad,cav}}$, is

$$\dot{Q}_{\text{rad,cav}} = -\dot{Q}_1 = -A_1 q_1. \quad (4.38)$$

The radiosity J_i^λ [44] is given by the equation

$$J_i^\lambda = \epsilon_i^\lambda E_{\text{b},i}^\lambda + \rho_i^\lambda \left(\sum_{j=1}^N J_j^\lambda F_{ij}^s H_{oi}^\lambda \right). \quad (4.39)$$

The transmitted heat flux through the cap surface, $q_{\text{tr,out},2}$, is [44]

$$q_{\text{tr,out},2}^\lambda = \tau_2^\lambda H_2^\lambda. \quad (4.40)$$

The irradiance H_i^λ is given by the expression

$$H_i^\lambda = \sum_{j=1}^N J_j^\lambda F_{i,j}. \quad (4.41)$$

The absorbed flux in the semitransparent sheet (window) $q_{\text{abs},2}$, is

$$q_{\text{abs},2}^\lambda = \alpha_2^\lambda H_2^\lambda = 0. \quad (4.42)$$

The solar paint has the properties $\epsilon_{\text{sp}}^{\text{th}} = 0.3$ and $\epsilon_{\text{sp}}^{\text{s}} = 0.9$. Since the wall is opaque, $\rho_{\text{sp}}^\lambda = 1 - \alpha_{\text{sp}}^\lambda$ [44], $\lambda \in \{\text{th}, \text{s}\}$. We also have $\alpha_{\text{sp}}^\lambda = \epsilon_{\text{sp}}^\lambda$, $\lambda \in \{\text{th}, \text{s}\}$.

4.4.3 Reradiation loss, $\dot{Q}_{\text{rerad}}^{\text{OCR}}$

The following set of surfaces is defined $W = \{\text{front}, \text{cap}\}$, where cap is the cavity aperture surface. The faces are also labelled by an index: 1 for front, 2 for cap, 3 for ins, and 4 for sky.

The surfaces front, cap, and ins exchange radiation with only the sky surface. Thus, the total energy loss by reradiation loss is the the sum of the contributions from each surface separately, giving

$$\dot{Q}_{\text{rerad}}^{\text{OCR}} = \sum_{i \in W} \dot{Q}_{\text{rerad},i}. \quad (4.43)$$

The semigray approximation is assumed [44]. The model for the sky given in [19] is used. The sky temperature is $T_{\text{sky}} = T_{\text{amb}} - 5$ which is representative of hot, moist climates.

For the thermal spectral region, the sky model is used. In the sky model it is assumed that the sky is an isothermal black enclosure at temperature T_{sky} . The radiation field within an isothermal black enclosure is isotropic [44]. Thus, the irradiance at any surface is the total black body emissive power $E_{\text{b,sky}}$. In particular for the general surface A_i , the following relation holds

$$H_i^{\text{th}} = E_{\text{b,sky}} = \sigma T_{\text{sky}}^4, \quad i \in W. \quad (4.44)$$

For the solar spectral region the irradiance is

$$H_i^{\text{s}} = \frac{\dot{Q}_{\text{SOCR},i}}{A_i}, \quad i \in W, \quad (4.45)$$

where the value $\dot{Q}_{\text{SOCR},i}$ represents the solar irradiation impinging at the surface, defined in section 4.4.1.

In a finite surface A_i , the outgoing radiative heat flux $q_{\text{out},i}^{\lambda}$ for an spectral region λ [44] is given by the following relation

$$q_{\text{out},i}^{\lambda} = \rho^{\lambda} H_i^{\lambda} + q_{\text{tr},i}^{\lambda} + q_{\text{em},i}^{\lambda}, \quad (4.46)$$

where ρ is the reflectance, H_i is the irradiance, $q_{\text{tr},i}$ represents the trasmitted radiation, and $q_{\text{em},i}$ represents the emitted radiation. The definition of reradiation gives the following relations

$$q_{\text{rerad},i}^{\text{s}} = q_{\text{out},i}^{\text{s}}, \quad \text{and} \quad (4.47)$$

$$q_{\text{rerad},i}^{\text{th}} = q_{\text{out},i}^{\text{th}} - \sigma T_{\text{sky}}^4. \quad (4.48)$$

In the above equations the following relations were used

$$q_{\text{em},i}^{\text{s}} = 0, \quad (4.49)$$

$$q_{\text{em},i}^{\text{th}} = \epsilon_i^{\text{th}} \sigma T_i^4. \quad (4.50)$$

From the above equations, the following compact expression is derived for $\dot{Q}_{\text{rerad, front}}^{\text{OCR}}$

$$\dot{Q}_{\text{rerad, front}}^{\text{OCR}} = A_{\text{front}} (1 - \rho^{\text{th}}) \sigma (T_{\text{front}}^4 - T_{\text{sky}}^4) + \rho^{\text{s}} \dot{Q}_{\text{SOCR, front}} \quad (4.51)$$

where A_{front} is the area of the cavity front, ρ is the reflectance of the solar paint, T_{sky} is the sky temperature, and $\dot{Q}_{S_{\text{OCR}},\text{front}}$ is given by equation (4.24).

For $\dot{Q}_{\text{rerad, cap}}^{\text{OCR}}$ the following compact expression is given

$$\dot{Q}_{\text{rerad, cap}}^{\text{OCR}} = \frac{A_{\text{cap}}}{1 - \rho(1 - \frac{A_{\text{cap}}}{A_{\text{cav}}})} ((1 - \rho)\sigma(T_{\text{cav}}^4 - T_{\text{sky}}^4) + \frac{\dot{Q}_{S_{\text{OCR}},\text{ap}}}{A_{\text{cap}}}) \quad (4.52)$$

where A_{cav} and A_{cap} are the areas of the cavity interior wall and the cavity aperture, respectively, and $\dot{Q}_{S_{\text{OCR}},\text{ap}}$ is given by Eq. (4.24).

4.4.4 Absorbed radiation in the front surface

For a general surface A_i , the net radiative heat flux [44] is

$$q^\lambda = q_{\text{out}}^\lambda - q_{\text{in}}^\lambda. \quad (4.53)$$

This gives the absorbed radiation in the front wall as

$$\dot{Q}_{\text{abs}} = \alpha_{\text{sp}}^{\lambda_s} \dot{Q}_{S_{\text{OCR}},\text{front}} + A_{\text{front}} \epsilon_{\text{sp}}^{\lambda_{\text{th}}} \sigma (T_{\text{front}}^4 - T_{\text{sky}}^4), \quad (4.54)$$

where A_{front} is the area of the front wall, $\alpha_{\text{sp}}^{\lambda_s}$ is the absorptance of the solar paint in the solar region, $\epsilon_{\text{sp}}^{\lambda_{\text{th}}}$ is the emittance of the solar paint in the solar region, and $\dot{Q}_{S_{\text{OCR}},\text{front}}$ is the rate of heat transfer due to the impinging solar radiation from the concentrator computed in Eq. (4.24).

4.4.5 Convection loss, $\dot{Q}_{\text{conv}}^{\text{OCR}}$

Convection is a considerable portion of the loss in thermal receivers, around 10% of total loss [27]. Different correlations can be considered from the literature but with important discrepancies between them due to different geometry and operating conditions [70].

For computing the energy loss by convection of the OCR, the following walls are considered: cavity (*cav*), and front (*front*). The $N = 2$ surfaces are also numbered with an association between numbers and names defined as: 1 for cav and 2 for front. Convection loss, $\dot{Q}_{\text{conv}}^{\text{OCR}}$, is assumed as the sum of the contributions from the front and cavity surfaces

$$\dot{Q}_{\text{conv}}^{\text{FPR}} = \sum_{i=1}^N \dot{Q}_{\text{conv},i}. \quad (4.55)$$

Here it is assumed that the effects of the interplay between the two surfaces are negligible. Thus, the rate of heat transfer for each wall can be computed independently.

In a surface A , heat transfer by convection is given by the following equation [81]

$$\dot{Q}_{\text{conv}} = \bar{h}A(T_w - T_{\text{amb}}), \quad (4.56)$$

where T_w is the temperature of the wall, $T_w - T_{\text{amb}}$ the temperature difference between wall and atmosphere under undisturbed conditions, and $\bar{h}$ in $\text{W}/\text{m}^2\text{K}$ is the coefficient of heat transfer. The bar denotes the average value over the total surface of the body under consideration [7]. It is assumed that local coefficients are equal to the average ones, $h = \bar{h}$.

The heat transfer coefficient is given from the definition of the Nusselt (Nu) number Nu_L [7],

$$\text{Nu}_L = \frac{hL}{k}, \quad (4.57)$$

where L is the characteristic length, and k is the thermal conductivity.

The value of Nu is evaluated representing geometry, motion, thermal and transport properties with exponents [6],

$$\text{Nu}_L = \text{Nu}(\text{Re}, \text{Gr}, \text{Pr}, L/D). \quad (4.58)$$

Forced convection is determined by combinations of Reynolds (Re) and Prandtl (Pr) numbers while natural convection is characterized by variations of Prandtl and Grashoff (Gr) numbers defined as [6],

$$\text{Re} = \frac{U_\infty L}{\nu}, \quad (4.59)$$

$$\text{Pr} = \frac{c_p \mu}{k}, \quad (4.60)$$

$$\text{Gr} = \frac{g\beta(T_w - T_{\text{amb}})L^3}{\nu}. \quad (4.61)$$

They are all evaluated at a mean or film temperature defined by

$$T_{\text{film}} = \frac{1}{2}(T_w + T_{\text{amb}}). \quad (4.62)$$

Sometimes the Rayleigh number is also used defined as [6],

$$\text{Ra} = \text{GrPr}. \quad (4.63)$$

Mixed or combined natural and forced convection

Mixed convection is the combined effect of natural and forced convection. It is assumed that the heat transfer coefficient of mixed convection is given by the following expression [80],

$$h = (h_{\text{nat}}^a + h_{\text{for}}^a)^{1/a}, \quad (4.64)$$

where a is a parameter being $a = 1$ for interior/cavity surfaces and $a = 3.1$ for external/flat surfaces. This expression combines the heat transfer coefficients of natural and forced convection separately, which is not always a good approximation [80]. In the literature different approaches for mixed convection are used with low accuracy [19, 81]. The uncertainty of equation (4.64) is not determined and it can be considerable.

Front surface

For the front surface, a correlation for natural convection on flat plates is used [80]. For the laminar regime, it is given by the following expression

$$\text{Nu}_{\text{nat}} = 0.68 + \frac{0.67\text{Ra}_{\text{nat}}^{1/4}}{[1 + (0.492/\text{Pr})^{9/16}]^{4/9}}. \quad (4.65)$$

where Ra_{nat} is the Rayleigh number, and Pr is the Prandtl number. This expression is applied for inclinations of up to $\theta < 60^\circ$. The Rayleigh number is modified by replacing g by $g \cos \theta$. For higher inclinations $\theta \geq 60^\circ$, the correlation for horizontal hot, downward surfaces given in [6] is used instead:

$$\text{Nu}_{\text{nat}} = 0.27\text{Ra}_{\text{nat}}^{1/4}. \quad (4.66)$$

For forced convection the following correlation on flat plates is used [80],

$$\text{Nu}_{\text{for}} = 0.037\text{Re}^{0.8}\text{Pr}^{1/3}. \quad (4.67)$$

This correlation is valid for turbulent regimes only. However it is also used for laminar regimes in this work as it should introduce only small errors.

For the computation of Re , the characteristic length is $L = D_{\text{ins}}$ and the wind speed v_{wind} is given as an input to the model.

Cavity surface

For the cavity surface, for natural convection the correlation of equation (3.6) is used. For forced convection the correlation of equation (3.7) is used.

Total loss by convection

The total rate of heat transfer by convection $\dot{Q}_{\text{conv}}^{\text{OCR}}$, is given by the expression

$$\dot{Q}_{\text{conv}}^{\text{OCR}} = h_{\text{cav}} A_{\text{cav}} (T_{\text{cav}} - T_{\text{amb}}) + h_{\text{front}} A_{\text{front}} (T_{\text{front}} - T_{\text{amb}}), \quad (4.68)$$

where h_{cav} and h_{front} are heat transfer coefficients, and A_{cav} and A_{front} are the areas of the cavity and front surfaces, respectively.

4.4.6 Conduction loss, $\dot{Q}_{\text{cond}}^{\text{OCR}}$

Conduction is the heat loss by conduction through the insulator. In the insulator wall, the rate of heat transfer by conduction is the sum of convection and thermal radiation,

$$\dot{Q}_{\text{cond}}^{\text{OCR}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{th rad}}, \quad (4.69)$$

where $\dot{Q}_{\text{conv}}$ is the rate of heat transfer by convection and $\dot{Q}_{\text{th rad}}$ is the rate of heat transfer by thermal radiation.

Convection is computed as a mixed convection problem as described for the front or cavity surfaces. Natural convection is computed by using the correlation for general immersed bodies of [6], which is reproduced here for convenience,

$$\text{Nu}_L = 3.47 + 0.51 \text{Ra}_L^{1/4}.$$

For forced convection, the correlation of [19] is used, which is reproduced here for convenience,

$$\text{Nu}_L = 0.42 \text{Re}_L^{0.6}.$$

It is assumed that the insulator surface has an absorptance of $\alpha_{\text{ins}} = 0.5$, which is representative of the embellisher of typical insulators. Thus, thermal radiation loss is given by the following expression [81]

$$\dot{Q}_{\text{th rad}} = \epsilon_{\text{ins}} A_{\text{ins}} \sigma (T_{\text{ins}}^4 - T_{\text{sky}}^4).$$

where T_{ins} is the temperature of the surface of the insulator.

In the insulator, heat is transferred by conduction which is modelled by a model for conduction in a flat wall of thickness $L = F_{\text{ins}}$. The rate of heat transfer is given by the following expression

$$\dot{Q}_{\text{cond}}^{\text{CCR}} = \frac{A_{\text{eq}} k}{L} (T_{\text{ins,int}} - T_{\text{ins}}), \quad (4.70)$$

where $A_{\text{eq}} = 1/2 A_{\text{total}}$ is an equivalent area of the insulator with a flat wall, and A_{total} is the total area of the insulator including the exterior, interior and side areas, and $T_{\text{ins,int}}$ is the temperature of the surface ins,int.

The insulator is made of stone wool with a representative thermal conductivity $k = 0.05 \text{ W/mK}$ at a temperature 200°C .

4.4.7 Model for the interior of the receiver

The receiver system is the HTF volume depicted in Fig. 4.4. The following set of surfaces is defined $W = \{\text{cav, front, ins int}\}$. The energy balance in this system gives the following relation

$$\dot{Q}_{\text{HTF}}^{\text{CCR}} = \sum_{i \in W} \dot{Q}_i, \quad (4.71)$$

where $\dot{Q}_i$ represents the heat flow rate from the surface i . It is assumed that the heat transfer is given by the relation

$$\dot{Q}_i = A_i U (T_i - T_{\text{CCR}}), \quad (4.72)$$

where U is the overall coefficient of heat transfer. This parameter depends on the design and flow of the HTF. It is assumed an constant value of $U = 10^3 \text{ W/m}^2\text{K}$ which is representative of heat exchangers with water-water mediums [61] and of some solar collectors [81].

4.4.8 Front wall

The front wall has two surfaces: front and the receiver surface. The front surface is the surface in common with the ambient, and the receiver surface is the surface in common with the receiver core. The following heat transfers are present:

- Rate of heat transfer absorbed due to radiative energy transport in the cavity surface $\dot{Q}_{\text{rad}}$.

- Rate of heat transfer due to convection loss $\dot{Q}_{\text{conv}}$ to the cavity interior. It is positive when going outside the component.
- Rate of heat transfer to the cavity interior $\dot{Q}_{\text{rec}}$. It is positive when going outside the component.

An energy balance gives the following relation

$$\dot{Q}_{\text{rad}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rec}}. \quad (4.73)$$

4.4.9 Cavity wall

The cavity has also two surfaces: cavity and receiver surface, denoted by rec. The cavity surface is the surface in common with the cavity interior, and the receiver surface is the surface in common with the receiver core. The following heat transfers are present:

- Rate of heat transfer absorbed due to radiative energy transport in the cavity surface $\dot{Q}_{\text{rad}}$.
- Rate of heat transfer due to convection loss $\dot{Q}_{\text{conv}}$ to the cavity interior. It is positive when going outside the component.
- Rate of heat transfer to the cavity interior $\dot{Q}_{\text{rec}}$. It is positive when going outside the component.

An energy balance gives the following relation

$$\dot{Q}_{\text{rad}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rec}}. \quad (4.74)$$

4.5 PDC configured with the FPR

A sketch of the FPR is shown in Fig. 4.5. The dimensions are the same of the OCR receiver, given in Table 2.1. This receiver is the OCR with a flat opaque wall at the cavity aperture. The HTF volume is extended to occupy the cavity volume. In this case, the cavity surface is also the cavity aperture surface, $A_{\text{cav}} = A_{\text{cap}}$.

Conduction losses $\dot{Q}_{\text{cond}}^{\text{FPR}}$ are computed by using the same considerations for the OCR.

4.5.2 Convection loss, $\dot{Q}_{\text{conv}}^{\text{FPR}}$

Convection loss, $\dot{Q}_{\text{conv}}^{\text{FPR}}$, is given by the following relation

$$\dot{Q}_{\text{conv}}^{\text{FPR}} = \sum_{i=1}^N \dot{Q}_{\text{conv},i} \quad (4.79)$$

The convection loss in the front surface $\dot{Q}_{\text{conv,front}}$ is computed by using the same considerations for convection for the front wall of the OCR.

The convection loss in the cav surface $\dot{Q}_{\text{conv,cav}}$ is computed by using the same correlations of the front surface. In this case, the area of heat transfer is A_{cav} and the characteristic length is $L = D_{\text{ins}}$.

4.5.3 Heat transfer in the receiver interior

In the receiver interior, $\dot{Q}_{\text{HTF}}^{\text{FPR}}$ is computed by using the same considerations for the interior of the OCR.

4.6 PDC configured with the CCR

The CCR subsystem is sketched in Fig. 4.6. The cover has an area $A_{\text{cover}} = A_{\text{cap}}$. The OCR has the following components: cover, cavity interior, cavity wall, front wall, receiver, and insulator.

The cover is made from fused silica (SiO_2) which in the solar band, has an effective transmittance of $\tau_{\text{co}}^{\lambda_s} = 0.92$, reflectance $\rho_{\text{co}}^{\lambda_s} = 0.05$, and absorptance $\alpha_{\text{co}}^{\lambda_s} = 0.03$. In the thermal band, it is an opaque surface with absorptance $\alpha_{\text{co}}^{\lambda_{\text{th}}} = 0.97$ [44, 76]. The properties are taken at the normal angle of incidence.

The cover is considered a flat wall with a thickness of $\delta = 5$ mm. It is characterized by a temperature T_{cover} .

4.6.1 Reradiation loss, $\dot{Q}_{\text{rerad}}^{\text{CCR}}$

The reradiation loss is computed like it is done for the FPR in section 4.5.2.

4.6.2 Convection loss, $\dot{Q}_{\text{conv}}^{\text{CCR}}$

Convection loss in the CCR is computed like it is done for the FPR in section 4.5.2.

Figure 4.6: Heat transfer in the CCR receiver. It is like the OCR receiver with and cover locking the cavity interior.

4.6.3 Cover

The cover has two surfaces: ambient and cavity. The ambient surface is the surface in common with the ambient, and the cavity surface is the surface in common with the cavity interior volume. The following numbering is sometimes used: 1 for the ambient and 2 for the cavity. The following heat transfer flows through the ambient surface are present:

- Heat transfer due to convection loss $q_{conv,ambient}$. It is positive when going outside the component.
- Heat transfer due to reradiation loss $q_{rerad,ambient}$. It is positive when going outside the component.
- Heat transfer due to absorbed solar radiation $\dot{Q}_{sol\ rad,abs}$.

For the cavity surface, the following heat transfer flows are present:

- Rate of heat transfer due to radiation and convection in the cavity surface $\dot{Q}_{\text{cav}}$. It is positive when going outside the component.

The heat rate is given by the considerations of section 4.6.1.

4.6.4 Cavity volume

Radiation

Radiation in the cavity volume is analysed with the same considerations for the cavity volume of the OCR receiver of section 4.4.3. In this case the surfaces are the cavity and the cover surfaces.

Convection

Convection in the cavity is modelled by an internal natural convection problem, where the following correlation is used for natural convection [6],

$$\text{Nu} = 0.364 \frac{L}{H} \text{Ra}_H^{1/4}. \quad (4.80)$$

The characteristic length is $L = H_{\text{cav}}$ and the height $H = D_{\text{cav}}$. The temperature of the cavity is the mean of the cavity wall temperature and the cover. The area of heat transfer is the area of the cover.

4.7 Simulations and results

For the three receivers considered (OCR, FPR, and CCR), the efficiency and fraction of losses have been computed by simulation of the models described above. The set of temperatures is $T_{\text{HTF}} \in \{150^\circ\text{C}, 200^\circ\text{C}, 250^\circ\text{C}\}$.

The models are simulated with the Sundials CVODE software of [14, 13]. The model is solved with a transient solver for a fixed duration. This duration has been selected in a previous analysis for which it has been observed that the results reach constant values over time that correspond to the solution of the model.

The results for the three receivers are summarized in Tables 4.3, 4.4, and 4.5, respectively. From tables Tables 4.3 and 4.4, it can be observed that energy conservation is fulfilled within 5%. The results make sense, but nevertheless, some minor numeric errors may cause this relatively small unbalance.

T_{HTF}	η_{OCR}	$\beta_{\text{OCR,rad}}$	$\beta_{\text{OCR,conv}}$	$\beta_{\text{OCR,cond}}$	T_{OCR}
150	0.64	0.02	0.07	0.01	230
200	0.61	0.03	0.09	0.01	278
250	0.58	0.03	0.11	0.01	327

Table 4.3: Results for the open cavity receiver, OCR (temperatures in °C).

T_{HTF}	η_{FPR}	$\beta_{\text{FPR,rad}}$	$\beta_{\text{FPR,conv}}$	$\beta_{\text{FPR,cond}}$	T_{FPR}
150	0.60	0.08	0.05	0.01	715
200	0.58	0.09	0.06	0.01	754
250	0.55	0.10	0.07	0.01	793

Table 4.4: Results for the flat-plate receiver, FPR (temperatures in °C).

T_{HTF}	η_{CCR}	$\beta_{\text{CCR,rad}}$	$\beta_{\text{CCR,conv}}$	$\beta_{\text{CCR,cond}}$	T_{CCR}
150	0.70	0.04	0.03	0.01	235
200	0.68	0.04	0.03	0.01	285
250	0.66	0.05	0.04	0.01	335

Table 4.5: Results for the cavity receiver with cover, CCR (temperatures in °C).

Figure 4.7: Ratio between loss factors of each receiver and the total loss factor of the flat-plane receiver for $T_{\text{HTF}} = 150^\circ\text{C}$.

Receiver($\mathcal{R}$)	$T_{\text{HTF}}/^\circ\text{C}$	$\eta_{\mathcal{R}}$	$T_{\mathcal{R}}/^\circ\text{C}$
Flat-plate (FPR)	150	0.60	715
	200	0.58	754
	250	0.55	793
Cavity (OCR)	150	0.64	230
	200	0.61	278
	250	0.58	327
Closed cavity (CCR)	150	0.70	235
	200	0.68	285
	250	0.66	335

Table 4.6: Total efficiency and maximum temperature for each configuration of the PDC.

Figure 4.8: Ratio between loss factors of each receiver, defined by (26), and the total loss factor of the flat-plane receiver, for $T_{\text{HTF}} = 200^\circ\text{C}$.

4.7.1 Comparison of receivers

A first look at Tables 4.3, 4.4, and 4.5 unsurprisingly shows that for every temperature T_{HTF} the flat-pane receiver (FPR) has the biggest losses, followed by the cavity receiver (OCR) and finally by the cavity with cover receiver (CCR). Also, although conduction losses are identical at the three receivers, reradiation and convection losses depend both on the type of receiver and the T_{HTF} .

To compare the different loss factors, plots are given in Fig. 4.7, 4.8, and 4.9, for $T_{\text{HTF}} = 150^\circ\text{C}$, $T_{\text{HTF}} = 200^\circ\text{C}$, and $T_{\text{HTF}} = 250^\circ\text{C}$, respectively. In these plots, every loss factor is normalized as a percentage of the total loss factor of the FPR (defined as the sum $\beta_{\text{FPR, rerad}} + \beta_{\text{FPR, conv}} + \beta_{\text{FPR, cond}}$, and equalized to 100). In each figure, the three receivers are compared for every type of loss factor. It turns out that, for every temperature, the biggest reradiation losses correspond to the FPR while the biggest convection losses correspond to the OCR. On the other hand, the least reradiation losses correspond to the OCR and the least convection losses correspond to the CCR. Conduction losses are similar for the three receivers.

Thus, although compared to a flat-plate receiver a cavity receiver reduces

Figure 4.9: Ratio between loss factors of each receiver, defined by (26), and the total loss factor of the flat-plate receiver, for $T_{\text{HTF}} = 250^\circ\text{C}$.

considerably the reradiation losses, it has the disadvantage of increasing convection losses. The use of a cover for the cavity gives the best receiver, the CCR, which considerably reduces the convection losses of the OCR, increasing reradiation loss in a smaller proportion.

Considering the total losses of each receiver, it results that the OCR reduces the losses of the FPR between 15 % and more than 30 % for temperatures between $T_{\text{HTF}} = 250^\circ\text{C}$ and $T_{\text{HTF}} = 150^\circ\text{C}$. Analogously, the CCR reduces the losses of the FPR between 53 % and 58 %. Thus, the best figures regarding receiver losses correspond to the CCR.

On the other hand, from Tables 4.3, 4.4, and 4.5, it is clear that the highest temperatures (715°C - 793°C) are obtained by the FPR, the lowest temperatures (230°C - 327°C) are reached by the OCR, and the CCR reaches slightly higher temperatures than the OCR (235°C - 335°C).

The increased reradiation loss in the flat-plate seems to be due to the temperature peak at its centre around 700°C - 800°C . The cavity is not very different from the flat-plate in radiative properties, but its increased area for the reception of solar radiation reduces the temperature. Convection loss increases due to the greater area. With a glass cover, the convection is reduced.

Finally, a summary of results including the HTF temperature, the efficiency of the PDC for each receiver, and the maximum receiver temperature T_{HTF} is given in Table 4.6. Note that there is a strong reduction of losses for the OCR, and especially the CCR, over the FPR, and the improvement of the overall efficiency is less relevant but still significant. This is mainly due to the fact the the concentrator losses due to reflection and spillage are dominant and makes up 26 % of the total losses.

As a final result, comparing with the overall efficiency of the FPR, the OCR accounts for a modest improvement of 5 %-7 %, but the CCR provides a significant improvement of 17 %-20 %. And, in addition, the temperatures reached by the FPR are significantly higher than the ones reached by the OCR and the CCR. By itself, this may be a rejecting factor for the FPR in practice (like in the case considered in this work where the limit temperature of the HTF is 350 °C).

Chapter 5

Generation of industrial steam

In this chapter, the PDC is used for generation of steam for a pilot pasteurization plant. Firstly, the pasteurization plant is presented. Secondly, the design of integration of the solar plant including the PDC is detailed. Thirdly, a model of the plant is developed in Modelica to show the feasibility of the solution proposed.

The potential of the PDC for providing HTF above 150 °C encourages its use for steam generation. The main outcome of this chapter is to show a simple model of integration of the PDC in an already operating pasteurization plant. The main result is the generation of a considerable quantity of steam and the study of the complexity of the integration for further analyses.

5.1 The pasteurization plant

The pasteurization plant is shown in Fig. 5.1. It is a small-scale plant in a laboratory setting used for research. Pasteurization plants are a type of thermal treatment plants.

The plant has the functional blocks shown in the diagram of Fig. 5.2. The product subsystem is where the thermal treatment of the liquid food is performed. The boilers subsystem generates steam on demand. The plant needs electrical work to function, which is done by the (electrical) grid. The electrical power is represented by P . The ambient is represented for accounting for thermal loss.

The product system is described in the diagram of Fig. 5.3. The product is liquid food, for example milk, juice, gazpacho, etc. At the input, the product is usually refrigerated. The heat exchangers HexHeater and HexCooler are for heating and cooling, respectively. The actual pasteurization occurs after the

Figure 5.1: Image of the pasteurization plant.

Figure 5.2: Diagram of the pasteurization plant.

product leaves HexHeater and while it is transiting the holding pipe. It finishes when it is cooled in HexCooler. This scheme can be used for online pasteurization processes.

For the pasteurization process, steam is required at a pressure of 5 bar and demanded with a mass flow rate $\dot{m}_{s,d} = 5 \text{ g/s} \simeq 300 \text{ L/min}$. The saturation temperature of water is 152°C at 5 bar.

The cooling system is described in Fig. 5.4. It is a simple line of coolant.

The boilers system is shown in Fig. 5.5. It consists of an electrical boiler (EB), the steam main going into port b, the feed pump, the condensates line coming from port a, and the condensates tank. The EB is a drum boiler with inserted electrical resistances. In the model, the electrical power, P , is ideally

Figure 5.3: Diagram of the product system.

Figure 5.4: Diagram of the cooling system.

controlled in the set of values $[0, P_{\max}]$, where $P_{\max} = 15 \text{ kW}$.

5.2 Integration of the solar plant

Steam can be generated with boilers in parallel and connected to the steam main with isolating devices like shown in [64]. Thus, a second boiler fuelled by HTF will work in parallel to the EB to generate steam. This boiler will be called the solar boiler.

The processes of collecting solar energy and carrying it, stored in the HTF, to the point of the solar boiler are engineered in a solar plant. The solar plant is integrated into the boilers system as shown in Fig. 5.6.

The new boilers system is shown in Fig. 5.7. The solar boiler (SB) is a

Figure 5.5: Diagram of the boilers system in the current pilot plant.

Figure 5.6: Diagram of the pasteurization plant with the solar plant for cogeneration of steam with solar energy.

drum boiler with an internal coil through which HTF passes. The two boilers are connected to the steam main with a check valve at its output. As mentioned in many sources [74, 64, 8], in a real plant a trap set is recommended. For simplicity, only a check valve will be used in the model.

5.2.1 The solar plant

The solar plant, shown in Fig. 5.8, is a simple HTF loop. It is composed of the solar field, the piping network, and the solar pump. The solar field contains the PDC and a pressure vessel. The piping network comprises the long return and heating pipes.

In this design, energy storage has not been considered because the cost of

Figure 5.7: Diagram of the boilers system. The solar boiler (SB) is in parallel with the electrical boiler (EB).

storage technologies would be added on top of the system's costs [81, 67].

The HTF is Marlotherm SH [59]. In a real plant using Marlotherm SH, it has been found convenient to pressurize the line at 2 atm to avoid the formation of bubbles of HTF vapor in the receiver. This will also be considered in the model.

The PDC is considered with the proposed cavity receiver of this thesis, studied previously. A characteristic thermal efficiency of $\eta = 70\%$ is considered to be representative.

The pressure vessel is commonly used in power plants to support high changes of volume due to thermal expansion. It is pressurized with the inert gas N_2 since it is recommended for Marlotherm SH [59]. The vessel is closed to the ambient according to the regulation [17] for heat transfer systems operating with organic media, which avoids the oxidation caused by atmospheric air [59].

The pressure vessel is placed in the line of HTF so that more thermal inertia is added, which is advantageous for rejecting perturbations from the direct solar irradiation.

5.3 Model of the boilers system

In order to assess the performance of the proposed design for the integration of the solar plant, a model of the pasteurization plant based on first principles is built.

The model is described in Modelica [72], together with the use of models from the Modelica Standard Library (MSL) [71]. This language is selected since

Figure 5.8: Sketch of the solar plant. There is a line of HTF that circulates through the PDC and the pressure vessel.

it allows to conveniently express the hierarchy and modularization present in the diagrams shown above.

The top-level model is the pasteurization plant with the Modelica diagram shown in Fig. 5.9. The steam valve is an ideal valve with linear characteristics. The demand profile is set by a constant aperture of 0.5.

5.4 Model of the boilers system

The Modelica diagram of the boilers is shown in Fig. 5.10. The steam main goes to port b of steam, called `steamPort_b`. The tee joint `tjv` connects the two boilers. Each boiler has a control valve denoted by `cvsb` and `cveb` for the EB and SB, respectively. The opening is controlled by a ramp signal that fully opens the valve at 10 s of simulation time, during 1 s. For each boiler, there are two blocks for controlling the pressure and level. The condensates are ideally returned from port a of steam, `steamPort_a`, to a reservoir at atmospheric normal pressure

Figure 5.9: Diagram of the Modelica model for the consumption of the steam.

and a backflow temperature of 150 °C. Each boiler has a feed water pump that is modelled as an ideal flow source signalled with the desired mass flow rate. The water is fed at 80 °C temperature.

5.4.1 Model of a drum boiler

Firstly, the models for the boilers are described. The electrical and solar boilers have in common the model of a drum boiler with the only difference being in how rate of heat transfer is given.

The drum boiler sketch is shown in Fig. 5.11. This model uses the same assumptions of [88, 21]. The boiler is modelled by a dynamic control volume where water exists in a state of liquid-vapour equilibrium (saturated steam) [12]. It has two flow ports, one for steam outlet and one for liquid water inlet, denoted by so and wi , respectively.

The system is divided in drum and fluid volumes. The drum volume is the iron and insulation. The mass of the drum is m_{drum} . The drum is in thermal equilibrium with the fluid. No heat loss to the ambient is considered [88].

In the fluid volume, the independent properties are the pressure p and volume of water V_w . Phase, thermal, and mechanical equilibrium is assumed [12]. The sign convention at the ports is that a quantity is positive when going into the component.

The conservation of mass in the fluid volume gives the following relation

Figure 5.10: Modelica diagram of the boilers with solar boiler and solar plant.

Figure 5.11: Sketch of the drum boiler.

between total mass and mass flow rates [12]

$$\frac{dm_{\text{fluid}}}{dt} = \dot{m}_{\text{wi}} + \dot{m}_{\text{so}}, \quad (5.1)$$

where m_{fluid} is the mass of fluid, $\dot{m}_{\text{wi}}$ and $\dot{m}_{\text{so}}$ are the mass flow rates at ports wi and so , respectively.

The total mass of the drum boiler is

$$m = m_{\text{fluid}} + m_{\text{drum}}, \quad (5.2)$$

where m_{drum} is the mass of the drum, as stated above. The mass of fluid is the sum of the mass of its constituents,

$$m_{\text{fluid}} = m_{\text{w}} + m_{\text{s}}, \quad (5.3)$$

where m_{w} is the mass of liquid water and m_{s} is the mass of steam. For each phase, the mass is given by the definition of density $m = \rho V$. The density of the medium is related to other properties by the equation of state of water, $\rho = \rho(p)$.

The total volume of the drum is constant. The following relations are given for the volumes:

$$V_{\text{fluid}} = V_{\text{w}} + V_{\text{s}}, \quad (5.4)$$

and

$$V = V_{\text{drum}} + V_{\text{fluid}}, \quad (5.5)$$

where V is the total volume of the drum boiler.

The drum is a sphere with a diameter $D = 0.5$ m and thickness is $S = 1$ cm. Thus, the total volume of the boiler is

$$V = \frac{4}{3}\pi \left(\frac{D}{2}\right)^3. \quad (5.6)$$

The fluid volume is

$$V_{\text{fluid}} = \frac{4}{3}\pi \left(\frac{D}{2} - S \right)^3. \quad (5.7)$$

Finally, the drum volume is

$$V_{\text{drum}} = V - V_{\text{fluid}}. \quad (5.8)$$

The rate of energy transfer from the fuel (electricity or HTF) is $\dot{Q}$.

The energy transport by mass into or out of the boiler through w_i and s_o is $\dot{m}h$, where h is the enthalpy and $\dot{m}$ is the mass flow rate. It has been assumed that the kinetic and potential energies of the fluid stream are negligible [12].

It is assumed that the total energy is equal to the internal energy U , i.e. the changes of other forms of energy are neglected. The energy balance applied to the drum boiler gives the following relation among the variables involved [12]

$$\frac{dU}{dt} = \dot{Q} + \dot{m}_{w_i}h_{w_i} + \dot{m}_{s_o}h_{s_o}. \quad (5.9)$$

The internal energy U is the sum of the internal energy of each one of the constituents [12],

$$U = m_w h_w + m_s h_s - pV_{\text{fluid}} + m_{\text{drum}} c_{\text{drum,iron}} T. \quad (5.10)$$

The enthalpy of saturated water and saturated steam are determined by the equation of state of the medium, $h_w = h_w(p)$ and $h_s = h_s(p)$. The selected model for water is the model `WaterIF97_ph` of MSL [71]. Note that in MSL, a saturated state is referred to as bubble point and vapour state is referred to as dew point.

A small column pressure in the drum boiler is assumed,

$$p_{w_i} - p_{s_o} - \rho_w g h = 0, \quad (5.11)$$

where p_{w_i} is the pressure at the water inlet, p_{s_o} is the pressure at the steam output, ρ_w is the density of liquid water, and h is the height of the liquid-steam interface with respect to the water inlet. It is assumed that the pressure of the steam is uniform and equal to the pressure of the boiler, $p_{s_o} = p$, where p is the pressure of the boiler.

The drum is made of iron, with representative specific heat capacity $c_{\text{iron}} = 450 \text{ kJ/kgK}$ and density $\rho_{\text{iron}} = 7840 \text{ kg/m}^3$. They are the properties of iron for room temperature [12], but are considered representative for higher temperatures.

It is convenient to use the scaled water volume as a measure of the level of actual liquid water in the boiler. The scaled water volume is defined as

$$V_{w,scaled} = \frac{V_w}{V_{fluid}}.$$

To measure the pressure, a pressure sensor is also considered.

An assertion is included to check for the validity of the model during simulation. Since the boiler cannot run out of water or steam, the level of water $V_{w,scaled}$ is forced in the interval $[0.1, 0.9]$.

5.4.2 Electrical boiler

The electrical boiler is shown in Fig. 5.12. The model of the drum boiler is used. The energy is transferred by heat that is equal to the electrical work dissipated at the resistance, given by P , i.e., $\dot{Q} = P$.

Figure 5.12: Diagram of the solar boiler. The heating coil and the boiler core systems are shown.

The Modelica diagram of the final model is shown Fig. 5.13. It is assumed that $0 \leq P \leq P_{max}$, where $P_{max} = 15$ kW. The boiler is initialized with a pressure $p_{start} = 5$ bar and a scaled volume of water $V_{w,scaled} = 0.5$.

5.4.3 Solar boiler

The solar boiler is shown in Fig. 5.14. The model of the drum boiler is used. The energy is transferred by heat governed by the temperature difference between the boiler and the HTF flowing inside the coil.

In the fluid volume, it is assumed that boiling heat transfer is present [6] with an excess temperature around 30°C . This excess temperature is where the heat transfer is maximum at 1 bar.

Figure 5.13: Modelica diagram of the electrical boiler.

Figure 5.14: Sketch of the solar boiler.

The Modelica diagram of the solar boiler is shown in Fig. 5.15.

The heat transfer is given by the following relation [6]

$$\dot{Q} = A_{hc} h_{coht,hc} (T_{hc} - T), \quad (5.12)$$

where T_{hc} is the temperature of the heating coil surface in contact with the water, and T is the temperature of the liquid water. The temperature difference $(T_{hc} - T)$ is the excess temperature. A representative value for the coefficient of heat transfer is $h_{coht,hc} = 10^4 \text{ W/m}^2\text{K}$ [6].

The heating coil is a rectilinear pipe with a length $L = 2 \text{ m}$ and a diameter $D = 2.5 \text{ cm}$, with an area of heat transfer $A_{hc} = \pi(D/2)^2 L$. It is modelled

Figure 5.15: Modelica diagram of the solar boiler.

by a dynamic pipe of the MSL. States are exposed through the ports and the discretization is set with $n = 2$. The heat transfer from the HTF to the walls is characterized by the constant coefficient of heat transfer $h_{\text{coht}} = 10^3 \text{ W/m}^2\text{K}$ [6]. It is a representative value for turbulent forced convection in closed ducts [7].

The pressure drop is given by the correlation TurbulentPipeFlow of MSL.

5.4.4 Control of the water level in the boilers

The block for the control of the level of the electrical boiler is shown in Fig. 5.16. A bumpless PID controller is used [85, 4].

The scaled volume of water in the boilers has to be maintained at around 0.5. For this, a proportional controller is used with a gain $k = 25$.

5.4.5 Automatic control of the pressure in the boilers

The controllers for the pressure in the boilers are similar to the level controllers. In this case, the pressure has to be maintained to $p_{\text{ref}} = 5 \text{ bar}$.

For the electrical boiler, a PI controller is used with gain $k = 1 \times 10^{-2}$ and integral time $T_i = 110 \text{ s}$.

The solar pump is set with a mass flow rate $\dot{m} = 0.001 \text{ kg/s}$ when $E_{\text{DNI}} = 0$,

Figure 5.16: Modelica diagram of the level controller for the electrical boiler.

and $\dot{m} = 0.4 \text{ kg/s}$ when $E_{\text{DNI}} > 0$. These values were selected to maintain a low HTF temperature and low pressure drop.

5.5 Model of the solar plant

The Modelica diagram of the solar plant is shown in Fig. 5.17.

The initialization temperature of the HTF is $T_{\text{HTF,start}} = 180^\circ\text{C}$.

5.5.1 Return and heat pipes

Return and heat pipes are modelled as rectilinear dynamic pipes with storage of mass and energy, with a length $L = 30 \text{ m}$ and a diameter $D = 2.5 \text{ cm}$.

Heat loss

Although the pipes are well insulated, due to its length some considerable losses to the ambient can be expected.

The heat loss to the ambient is modelled by the definition of the coefficient of heat transfer [6]. Under a stationary condition, the total thermal resistance from the HTF to the ambient R is given by the sum of four thermal resistances in series: thermal resistance in the HTF (R_{fluid}), thermal resistance in the wall (R_{wall}), thermal resistance in the insulation (R_{ins}), and thermal resistance in the ambient (R_{amb}), i.e.

$$R = R_{\text{fluid}} + R_{\text{wall}} + R_{\text{ins}} + R_{\text{amb}}. \quad (5.13)$$

The insulation has typically a high thermal resistance in comparison with the resistance of the wall, ambient, and fluid in series. The total resistance is approximated by the resistance through the insulation. Thus, $R = R_{\text{ins}}$. The value R_{ins} is approximated as constant independent of temperature.

The thickness of the insulation is $l = 5$ cm. The insulation is made from mineral wool with a representative thermal conductivity $k = 0.06$ W/mK. The insulation is approximated by a flat wall resulting in a resistance $R_{\text{ins}} = l/k = 0.532$ K/W. This gives a coefficient of heat transfer $h = 1/R = 1.2$ W/m²K.

Flow model

The pipes are configured with the MSL model `turbulentPipeFlow` for modelling of the wall friction, static head, and momentum flow. The `turbulentPipeFlow` model is for the quadratic turbulent regime and it behaves well in the laminar regime. It is selected because the flow is expected to be in the turbulent regime the majority of times.

In the `turbulentPipeFlow` model, the pressure drop due to friction force is related to the wall friction coefficient λ by the relation [78]

$$\Delta p = \frac{1}{2} \lambda \frac{L}{D} \rho v |v|, \quad (5.14)$$

where v the flow speed, and L and D are the length and the diameter, respectively.

It is assumed a constant value of λ . In general, the friction factor is a function of the Reynolds number and the relative roughness $\lambda = \lambda(\text{Re}, \delta/D)$ [78]. For turbulent flows, it is no longer a function of the Reynolds number, $\lambda = \lambda(\delta/D)$. The surface condition of used copper pipes is the same of galvanized iron according to [6, p. 373], which gives a representative value of the roughness $\delta = 0.15$ mm and a relative roughness $\delta/D = 0.006$. For the interested reader, it is indicated that in [6] the roughness δ is denoted by k_s .

A representative volume flow rate of the HTF is $V = 10^3$ L/h, which gives a Reynolds number of $Re = 10^5$ for a temperature $T_{HTF} = 180$ °C. Thus, the flow is considered turbulent.

5.5.2 Pressure vessel

A sketch of the pressure vessel is shown in Fig. 5.18. It is a cylinder of diameter $D = 50$ cm and height $H = 1$ m. The system consists of the fluid volume and the exterior case with a thickness of $S = 5$ mm. The fluid volume have HTF and N_2 gas.

The system modelled as a dynamic control volume with heat loss to the ambient. It is assumed a uniform pressure in the volume. The gas is ideal.

The vessel is insulated with mineral wool of thickness $S = 5$ cm. The coefficient of heat transfer is given by the assumptions done for the heat and return pipes. Thus, $h = 1.2$ W/m²K.

5.5.3 Solar pump

The solar pump is an ideally controlled pump [71]. The controlled variable is the mass flow rate. It is configured as a static control volume. Heat loss is neglected. The efficiency is $\eta = 0.8$ which is considered a representative value.

5.5.4 Simplified PDC

The parabolic dish collector (PDC) component is modelled taking as reference the PDC with the proposed cavity receiver (OCR), described in this thesis. The Modelica diagram the PDC is shown in Fig. 5.19.

The model of the PDC uses a concentrator and a receiver components. The concentrator component represents the interception and reflection of the parabola of PDC. The sun is represented by E_{DNI} and the softness of focus is represented by f . The total reflected solar radiation, $\dot{Q}_{solar}$, is given by the relation

$$\dot{Q}_{solar} = f \rho_{conc} A_{ap} E_{DNI}, \quad (5.15)$$

where $A_{ap} = 15.9$ m² is the aperture of the concentrator and $\rho_{conc} = 0.86$ is its reflectance.

The receiver component is a dynamic vessel with mass and energy storage [12, 71]. This system has a rate of heat transfer $\dot{Q}_{solar}$ representing the impinging

solar radiation from the concentrator. The intercepted radiation is represented by $\dot{Q}_{ab}$, and it is given by the efficiency $\eta_s = 0.93$ taking into account the spillage of radiation. Thus,

$$\dot{Q}_{ab} = \eta_s \dot{Q}_{solar}. \quad (5.16)$$

The efficiency $\eta_s = 0.93$ accounts for the spillage loss, i.e. $\eta_s = 1 - \beta_{spi}$, where $\beta_{spi} = 0.07$.

The thermal loss to the ambient is represented by $\dot{Q}_{loss}$ given as

$$\dot{Q}_{loss} = A_{ht} h (T_{rec} - T_{amb}), \quad (5.17)$$

where A_{ht} is the area of heat transfer, T_{rec} is the temperature of the receiver, and T_{amb} is the ambient temperature. The coefficient of heat transfer is $h = 58 \text{ W/m}^2\text{K}$.

The coefficient of heat transfer is set to obtain a thermal efficiency similar to that obtained for the PDC and the open cavity receiver (OCR). A representative value of thermal efficiency is $\eta_{th} = 70\%$.

To compute h , a stationary energy balance in the receiver is considered:

$$\dot{Q}_{HTF} = \rho_{conc} \eta_s A_{ap} E_{DNI} - A_{ht} h (T_{rec} - T_{amb}), \quad (5.18)$$

where $\dot{Q}_{HTF}$ represents the energy gained by the HTF. From this expression, for representative values of $E_{DNI} = 850 \text{ W/m}^2$, $A_{hf} = 0.15 \text{ m}^2$, $T_{rec} = 180^\circ\text{C}$, and $T_{amb} = 25^\circ\text{C}$, it is obtained $h = 58 \text{ W/m}^2\text{K}$.

The volume of the vessel is the volume of the OCR receiver $V = 12.8 \text{ L}$.

The controller component is in an open loop configuration and it simply gives the constant value $f = 1$.

5.6 Results

Two cases of the model are considered, each one considering a different day of the typical meteorological year (TMY) of section 4.1.4. For each case, the model is simulated and the following results are obtained. The mass flow rate of steam at each boiler and the total mass flow rate of steam are presented. The pressure in the tee junction is also presented. The mass flow rate of steam at the electrical boiler and solar boiler are $\dot{m}_{EB}$ and $\dot{m}_{SB}$, respectively. The total mass flow rate of steam is $\dot{m}_t$. The pressure in the tee junction is p and represents the pressure of the steam line.

The model is simulated with the solver CVODE [13, 14]. The maximum time step was set to 100 s. The minimum time step was set to 1×10^{-9} s. Absolute and relative tolerances were set to 1×10^{-6} . The maximum order was set to 5. In general, it was found that adjusting the parameters to obtain a more exact solution is preferable when the model has events (for example, due to check valves) over left then for when the model is continuous.

The simulations start at 0 h of the day and finish at 24h.

The first case is for the day 5/8/2006 of the TMY. In Fig. 5.21 the E_{DNI} is shown. $\dot{m}_{\text{EB}}$, $\dot{m}_{\text{SB}}$, and $\dot{m}_{\text{t}}$ are shown in Fig. 5.21. p is shown in Fig. 5.22.

The second case is for the day 22/1/2012 of the TMY. In Fig. 5.23 the E_{DNI} is shown. $\dot{m}_{\text{EB}}$, $\dot{m}_{\text{SB}}$, and $\dot{m}_{\text{t}}$ are shown in Fig. 5.24. p is shown in Fig. 5.25.

The behaviour of the signals $\dot{m}_{\text{EB}}$ and $\dot{m}_{\text{SB}}$ indicates that the proposed design generates steam in the solar boiler when solar energy is available. The steam demand is satisfied but the total steam mass flow rate, $\dot{m}_{\text{t}}$ has some disturbances from an ideal constant mass flow rate that in practice could be important. The disturbances are more noticeable in the pressure p . These disturbances are due to the indirect coupling of the pressure of the two boilers. A more complex control design should give a more ideal steam supply. As a result, a model has been developed that serves as a benchmark for analysis and optimization of design parameters, including the solar plant, regulator, pumps, pipes, HTF, etc.

Figure 5.17: Modelica diagram of the solar plant.

Figure 5.18: Sketch of the pressure vessel.

Figure 5.19: Modelica diagram for PDC.

Figure 5.20: Direct normal irradiance E_{DNI} for day 5/8/2006.

Figure 5.21: Mass flow rate of steam in the electrical boiler $\dot{m}_{\text{EB}}$, in the solar boiler $\dot{m}_{\text{SB}}$, and total $\dot{m}_{\text{t}}$, for day 5/8/2006.

Figure 5.22: Pressure of the steam p for day 5/8/2006.

Figure 5.23: Direct normal irradiance E_{DNI} for day 22/1/2012.

Figure 5.24: Mass flow rate of steam in the electrical boiler $\dot{m}_{EB}$, in the solar boiler $\dot{m}_{SB}$, and total $\dot{m}_t$, for day 22/1/2012.

Figure 5.25: Pressure of the steam p for day 22/1/2012.

Chapter 6

Conclusions and future works

The final conclusions are a discussion about the results of previous chapters. The results are digested and discussed from an overall perspective from the context of the preliminary objectives of this thesis. From the conclusions, some research lines are envisaged where the most promising advances are expected. These lines might serve the interested reader as a guide for future efforts in the field of parabolic dish collectors.

6.1 Conclusions

For convenience, at the beginning of each part of the conclusions a reference to its related section or chapter is included in bold text.

Chapter 3 Analysis of the proposed cavity receiver, section 3.1 Radiation performance of the cavity receiver

The results show that the radiation performance of the cavity receiver can be influenced by varying the geometry. Recall that, to model the radiative energy transport in the parabola and cavity system, a ray-tracing model was considered. In the model, a fixed E_{DNI} and an ideally oriented parabola are assumed. For comparison, a squared flat-plate receiver was also considered with a size comparable to the cavity receiver.

The effect on radiation absorption of displacing the cavity receiver along the parabolic axis was studied. It was found that there is a range of positions around the focal point for which the radiation absorbed is maximal. In particular, the range comprises approximately the positions $(-3, 3)$ cm for a maximum change of the radiation absorbed in around 1 % (see Fig. 3.4).

Regarding the comparison between the cavity and flat-plate receivers, the results show that the cavity receiver absorbs more radiation than the flat-plate receiver, with a relative difference of 2.3%. The specific effect of the dome forming the bottom of the cavity has not been analysed.

The distribution of flux density in the side wall of the cavity receiver has a considerable peak that changes with varying receiver positions (see Fig. 3.5). For positions closer to the parabola, the peak is smaller. Thus, the receiver can be moved closer to the parabola if a more uniform flux distribution is desired.

Finally, the effect on radiation absorption of varying the aperture diameter and the cavity diameter was also studied. It is found that the aperture and cavity sizes do not influence noticeably the amount of radiation absorbed. Thus, the aperture can be suppressed.

Chapter 3 Analysis of the proposed cavity receiver, section 3.2 Energy loss by convection from the cavity receiver and section 3.3 Correlations of natural and forced convection.

The energy loss by convection from the proposed cavity receiver is of great interest since in cavity receivers, convection causes an important loss of energy [81]. For the proposed size and temperature, there are no applicable correlations in the literature. An accurate correlation for forced convection has not been found either. As with other analyses, a squared flat-plate receiver was also considered for comparison.

The convection in the proposed cavity receiver was analysed with a high-fidelity model based on a typical CFD modelling approach. This approach involves a considerable effort from the modeller for the model construction, numerical resolution, and computational resources. Turbulence increases the complexity of the problem, and a turbulence model is used in order to manage it [79]. The precision and insight gained into the problem with a CFD model outweighs the required efforts.

In natural convection, the effect of the inclination θ in the convection was analysed because the changes are quite significant depending on angle. For an inclination of 40° (the sun is low) the cavity receiver has more energy loss than the flat-plate receiver. As the angle is increased from 40° to 70° (the sun is high), this is inverted; the cavity receiver now performs better than the flat-plate receiver (see Table (3.5). In terms of heat flux, the cavity receiver always performs better than the flat-plate receiver (see Table 3.6). These observations are explained by the change in the flow pattern (see Fig. 3.13 and Fig. 3.14).

In forced convection, the effect of inclination was assumed negligible. The analyses were done for a side-on direction of wind velocity [56]. This direction was

selected because it was assumed that it is predominant over other directions. In this case, the proposed cavity receiver has more energy loss by forced convection than the flat-plate receiver in all cases. This is a consequence of the wind that drags the heated air out of the cavity (see Fig. 3.15 and Fig. 3.16). However, similar to the case of natural convection, the heat flux is smaller in the cavity than in the flat-plate receiver in all cases.

Thus, it is concluded that heat loss by convection is higher in the proposed cavity receiver than in the flat-plate receiver except for some favourable conditions of the inclination angle. This is due to the increased area of losses in the cavity receiver, since the energy flow per unit of area, the heat flux, was found to be significantly smaller. This behaviour of is also pointed out in [81] for convection in cavity receivers for higher temperatures.

A strategy for suppressing convection in cavity receivers [76] is glazing the aperture with a window. For this reason, the proposed cavity receiver with a glass cover was considered in chapter 4.

In section 3.3, a CFD analysis performed to obtain natural and forced correlations for the proposed cavity receiver based on numerical experiments.

Chapter 3 Analysis of the proposed cavity receiver, section 3.4 Energy transfer in the proposed cavity receiver

A high-fidelity CFD model of the heat transfer in the interior of the proposed cavity receiver shown in Fig. 2.4 was used. The insights and valuable information extracted from the results is of key importance for understanding the experimental performance of the proposed cavity receiver.

The very high non-uniform distribution of radiative flux poses a high requirement on the heat transfer fluid (HTF) flow inside the cavity. However the relatively low speed of the HTF in some areas generates a highly varying temperature distribution in the receiving walls with important peaks that poses a serious concern about the integrity of the HTF and even the steel of the receiver walls. Even if the materials can tolerate these peaks, high temperatures tend to degrade them quickly and pose some safety concerns over the material integrity under some events like emergency stops. However, energy redistributes in other parts of the cavity relatively well due to the maze-like structure. Thus, in conclusion, the proposed cavity receiver should be improved to overcome the limitations of low speed of HTF in the areas identified in Figures 3.28, 3.30, and 3.31.

Chapter 4 Thermal performance of the receivers.

Three receivers were selected for a complete analysis with the parabolic dish collector (PDC). They are the flat-plate receiver, the proposed cavity receiver, and the proposed cavity receiver with a glass window. They are labelled as FPR,

OCR (open cavity receiver), and CCR (covered cavity receiver), respectively. For each one of them, a lumped-element thermal model of the PDC and the receiver mounted at the focus was posed. The model was simulated for a significant set of data representative of real conditions extracted from the most recent available typical meteorological year (TMY). The figures obtained are annual averages over this year which should be representative of the total thermal performance of the PDC and receivers considered.

Focusing only on the thermal performance of the receivers without considering other parts of the system, it is found that the biggest reradiation loss corresponds to the FPR while the biggest convection loss corresponds to the OCR. On the other hand, the smallest reradiation loss corresponds to the OCR and the least convection loss corresponds to the CCR. Conduction loss is similar for the three receivers.

A summary of results including only the HTF temperature, the efficiency of the PDC for each receiver, and the maximum receiver temperature T_{HTF} is given in Table 4.6. Note that there is a marked reduction of losses of the OCR, and specially the CCR, over the FPR, and the improvement of the overall efficiency is less relevant but still significant. This is mainly due the fact the the concentrator losses due to reflection and spillage are dominant and supposes a 26% of the total.

As a final result, comparing with the overall efficiency of the FPR, the OCR gives an improvement of 5%-7%, but the CCR gives a significant improvement of 17%-20%. And, in addition, the temperatures reached by the FPR are significantly higher than the ones reached by the OCR and the CCR. By itself, this may be a rejecting factor for FPR in practice since the upper limit of temperature of the HFT is 350 °C.

Chapter 5 Generation of industrial steam

The potential of the PDC for providing temperatures above 150 °C shown previously encourages its use for steam generation in a pasteurization plant. A solar plant has to be designed to use the solar energy collected from the PDC which, by itself, is not convenient for steam generation in a typical industrial plant setting. With this aim, a solar plant has been designed for a selected industrial pilot plant for pasteurization. A lumped-element model of the pilot plant with the solar plant has been built. A constant steam demand is considered which is representative of a plant operating all day. Two selected days from the TMY provide the data for the E_{DNI} representing the available solar energy (fuel) during these days.

Steam is generated from electricity by using an electrical boiler and from the

HTF by using a solar boiler. They are connected in parallel to the steam main of the pilot plant. A bumpless PID controller for the electrical boiler regulates the pressure to a constant value of 5 bar. The results show that this system is capable of using the solar energy efficiently when solar energy is available (see Figures 5.21 and 5.24). For this, a constant mass flow rate of HTF is set. The electrical boiler is coupled to the solar boiler by the pressure in the steam main. Thus, when the pressure in the solar boiler is capable of providing steam, the pressure regulator of the electrical boiler reduces the applied electrical power.

6.2 Future works

First, recall that the dimensions of the proposed receiver are taken from the work [16]. But the spillage loss was found to be relatively high ($\beta_{\text{spill}}^{\mathcal{R}} = 7\%$, see chapter 4). It is speculated that this loss can be reduced by changing the size of the receiver. On the other hand, the shape of the bottom area can also reduce multi-reflection losses (due to gray radiation) and energy loss (due to convection), which should be also studied in a future work. Thus, an optimization of the dimensions of the proposed cavity receiver for the PDC is proposed as an interesting research line to increase the thermal efficiency of the system for the three receivers considered in chapter 4.

Second, several correlations for natural and forced convection of the proposed cavity receiver has been developed. An experimental validation of these proposed correlations is proposed as a future work. Moreover, the temperature was assumed to be uniform in the walls of all the models developed in chapter 3. The validity of this assumption should be revised in an extended work.

Third, the high temperature in the flat-plate receiver is the reason why it is discouraged. Some approaches to overcome this limitation deserves some research efforts. A Fresnel lens could be used as a glass window to diffuse the radiation before it impinges in the surface of the flat-plate receiver. An alternative approach is to displace the receiver. This could be also apply to improve open and covered cavity receivers.

Fourth, the proposed cavity receiver can be positioned closer to the parabola, which can help to reduce forced convection since the maximum radiative flux can be displaced to the interior.

Fifth, regarding the geometry of the interior of the receiver, a new interesting work is envisaged about its design. The objective is to overcome the limitations of low flow speed of the HTF.

Sixth, regarding the generation of industrial steam with the PDC, a cost analysis is necessary to assess the cost-effectiveness of scaling up the plant. The construction of a cost-effective analysis model of the plant is interesting. The application Greenius [55] is useful for this purpose.

Seventh, given the importance of energy loss by forced convection in the convection loss, shielding of the receiver could help to reduce convection loss even further under windy conditions.

Eighth, and finally, the use of other HTF materials can be studied with the model of the solar plant. For example, air as HTF can help to reduce maintenance cost and pollution. A $1.2\text{ MW}_{\text{th}}$ solar power plant using air as HTF was tested in [25].

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Nomenclature

Symbols

α	Absorptance, [-]
β	Volumetric thermal expansion coefficient, [K ⁻¹]
δ_{ij}	Kronecker's delta, [-]
ϵ	Emittance, [-]
λ	Spectral range
F	Focal point
μ	Dynamic viscosity, $\nu = \mu/\rho$, [m ² /s]
ν	Kinematic viscosity, [Ns/m ²]
ω	Specific turbulence dissipation rate, [s ⁻¹]
Φ	Rim angle defined as the angle between the focal axis and the axis of the rim with the focus, [rad,°]
ρ	Density, [kg/m ³], Reflectance, [-]
σ	Stephan-Boltzmann constant, $5.6703 \times 10^{-8} \text{ W/m}^2\text{K}^4$
τ	Transmittance, [-]
Gr	Grashof number, [-]
Nu	Nusselt number, [-]
Nu	Reynolds number, [-]

Pr	Prandtl number, [-]
Ra	Rayleigh number, [-]
θ	Inclination, [rad,°]
c_p	Specific isobaric heat capacity, [J/kgK]
E_b	Hemispherical black body emissive power, $E_b = \sigma T^4$, [W/m ²]
E_{DNI}	Direct normal irradiance, [W/m ²]
f	Focal distance, [m]
F_{ij}	Average view factor from surface i to j, assuming finite surfaces and uniform radiosity, [-]
g	Standard gravity module, 9.81 m ² s ⁻¹
k	Thermal conductivity, [W/m ² K], Turbulence kinetic energy, [m ² /s ²]
q	Radiative flux from a surface into the enclosure, [W/m ²]
T	Temperature, [K]

Acronyms

BRDF	Bidirectional Reflectance Distribution Function
CCR	Covered cavity receiver
CFD	Computational Fluid Dynamics
CSR	Circumsolar ratio, [-]
DNI	Direct Normal Irradiance
FPR	Flat-plate receiver
HTF	Heat Transfer Fluid
MSL	Modelica Standard Library
OCR	Open cavity receiver

PDC	Parabolic Dish Collector
RANS	Reynolds-Averaged Navier-Stokes equations
SPP	Solar Power Plant