STUDY OF DIFFERENT SOLUTIONS FOR SOLAR/BIO MASS HYBRID ELECTRICITY GENERATION SYSTEMS

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STUDY OF DIFFERENT SOLUTIONS FOR SOLAR/BIO MASS HYBRID ELECTRICITY GENERATION SYSTEMS

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To my wonderful daughter.
“Wisdom is the daughter of experience.”

Leonardo da Vinci.
ABSTRACT

The long-term potential of renewable energy systems is undisputable since they rely on a source that will not run out. Nonetheless, they often depend on meteorological conditions leading to the instability of energy supply. Concentrated Solar Power (CSP) can be a reliable solution due to the easiness to dispatch energy, through the use of thermal energy storage (TES). Another attractive solution is renewable hybridisation, by combining two or more different renewable technologies in order to better explore their individual advantages. Within alternatives, hybridisation of CSP with biomass can be a powerful way of assuring system stability and reliability.

The primary goal of this work is to provide a comprehensive scientific approach concerning these hybrid CSP/biomass systems, addressing both technical and economic issues in order to deliver a strong scientific basis for future research, as well as to develop design tools for enhancing commercial deployment and technology dissemination. It is intended to address design guidelines and recommendations for the optimal combination of CSP and biomass, to attain the best balance between system performance and economic feasibility.

In order to achieve these goals, distinct CSP/biomass hybrid cases studies were assessed including the following technologies: linear focusing solar collectors; single (thermal oil and molten salt) and two-phase (water/steam) heat transfer fluids (HTF); TES; biomass combustion, gasification, and anaerobic digestion; and conventional Rankine cycle, either driven with steam or an organic fluid.

The analysis was carried out through the development of a transient numerical model using a commercial software (EBSILON® Professional), which was validated and improved with experimental data. The modelling methodology consisted of splitting the CSP/biomass hybrid power plant into subsystems, where detailed modelling was carried out for individual components. Quasi-dynamic simulations were achieved by the combination of dynamic components and a series of semi-equilibrium states at each timestep. A full dynamic approach was applied to the components that entail significant thermal inertia and where the highest temperature gradients are expected to occur, e.g. solar collectors. The experimental work consisted in the evaluation under real-life conditions of a biomass gasification system and a concentrating solar field (SF).

The most conventional CSP/biomass integration method consists in using a boiler to back up the SF heat production. The main advantages of this configuration are the improved
system stability and dispatchability. An increase of system efficiency was noticed (e.g. from 3.4% to 9.6% at a prototype scale), as a consequence of the better use of solar energy and the possibility to drive the power block near design conditions. From the economic point of view, hybridisation reduces the levelised cost of electricity (LCoE) mainly due to the improved system efficiency and capacity factor, and also through the joint use of power plant equipment. Therefore, it increases the feasibility of small-scale systems, usually hindered by the lower conversion efficiency (e.g. a LCoE of 0.175 €/kWh for a 1 MW hybrid plant was found). Biomass economic viability requires proximity to the feedstock, driving CSP to near consumption centres, where Combined Heat and Power (CHP) is conceivable. As a downside the initial investment in a hybrid facility is higher than in a conventional CSP plant (e.g. anaerobic digestion represents about 36% and 55% of the plant capital and operational expenditures). Hybridisation significantly reduces (although not eliminating) the need for heat storage. Thus, the added value from TES is associated to the enhancement of the plant reliability and controllability, acting as a buffer.

Hybridisation can also be used to avoid some technical issues present in CSP technologies with single-phase HTF (for example, there are considerable heat losses in the TES system and steam generator heat exchangers), and also with Direct Steam Generation (for example, direct steam reheat within the turbine and, in the case of once-through operation, the safety limit regarding the maximum temperature difference within one receiver cross-section, in the superheating section). To overcome these issues a biomass boiler can be installed within the HTF and steam cycle, improving the HTF temperature, or used as backup.

The findings of this PhD work lead to the conclusion that additional research should be carried out considering a more extensive range of (CSP, biomass and power cycle) technologies to explore their full potential.

**Keywords:** Concentrating solar power; Biomass; Hybrid plants
RESUMO

A longo prazo, o potencial dos sistemas de energia renovável é incontestável, uma vez que dependem de uma fonte inesgotável. No entanto, encontram-se frequentemente dependentes das condições climatéricas, resultando em instabilidade no fornecimento de energia. A utilização de sistemas de Concentração Solar Térmica (CSP) representa uma solução fiável, devido à facilidade em disponibilizar energia, com recurso a sistemas de armazenamento de energia térmica (TES). Outra solução atrativa é a hibridização de sistemas renováveis, através da combinação de duas ou mais fontes de energia renováveis com o intuito de explorar o melhor das suas vantagens individuais. Dentro das alternativas, a hibridização da CSP e biomassa pode representar uma solução valiosa para assegurar a estabilidade e fiabilidade do sistema.

O principal objetivo deste trabalho é fornecer uma abordagem científica abrangente sobre esses sistemas híbridos, abordando assuntos técnicos e económicos, com o intuito de fornecer uma base forte para futura investigação, bem como desenvolver ferramentas de projeto para promover a implementação comercial e a disseminação da tecnologia. Ainda tem como propósito avaliar diretrizes de projeto e recomendações quanto à melhor combinação entre a CSP e a biomassa, para atingir o melhor balanço entre o desempenho técnico e a viabilidade económica.

Para alcançar os objetivos propostos, foram avaliados vários casos de estudo de sistemas híbridos de CSP/biomassa, incluindo as seguintes tecnologias: coletores concentradores lineares; fluidos térmicos (HTF) mono (óleo térmico e sais fundidos) e bifásicos (água/vapor); TES; combustão, gaseificação e digestão anaeróbica de biomassa; e o ciclo de Rankine convencional, com vapor ou com um fluido orgânico.

O estudo foi realizado com o desenvolvimento de um modelo numérico transiente, com recurso a um programa computacional comercial (EBSILON® Professional), o qual foi validado e melhorado com resultados experimentais. A metodologia de modelação consiste em dividir a central híbrida em subsistemas, onde os componentes individuais são modelados de forma detalhada. A simulação quase-dinâmica foi conseguida pela combinação de componentes dinâmicos e uma série de estados de semi-equilíbrio a cada intervalo de tempo. A abordagem totalmente dinâmica foi aplicada aos componentes com inércia térmica significativa e onde se esperam os maiores gradientes de temperatura, como os coletores solares. O trabalho experimental consistiu na avaliação do desempenho de um sistema de gaseificação e de um campo solar em condições reais.
A metodologia mais convencional para a integração de CSP e biomassa consiste em utilizar uma caldeira para suportar a produção térmica do campo solar (SF). As principais vantagens desta configuração são a melhoria da estabilidade do sistema e da disponibilidade de energia. Observou-se um aumento da eficiência do sistema (de 3,4% para 9,6% à escala de protótipo), como consequência do melhor uso da energia solar e da possibilidade de operar o ciclo de potência em condições próximas da nominal. Do ponto de vista econômico, a hibridização reduz o custo da eletricidade (LCoE), principalmente devido ao melhoramento da eficiência e da disponibilidade, e também como resultado da partilha de equipamentos da central. Portanto, aumenta a viabilidade dos sistemas de pequena escala, usualmente desconsiderados pela reduzida eficiência (como exemplo, foi calculado um LCoE de 0,175€/kWh para uma central híbrida de 1 MW). A viabilidade econômica da biomassa implica proximidade com a matéria-prima, aproximando a CSP dos centros de consumo, onde a cogeração (CHP) é concebível. Como desvantagem, o investimento inicial de uma instalação híbrida é superior ao de uma central de CSP convencional (por exemplo, a digestão anaeróbica representa cerca de 36% e do capital investido e 55% dos custos operacionais). A hibridização permite reduzir significativamente a necessidade de armazenamento térmico. Assim, o valor acrescentado do TES está relacionado com o melhoramento da estabilidade e controlo da central, atuando como um estabilizador.

A hibridização também pode ser utilizada para evitar alguns problemas técnicos presentes em tecnologias de CSP com fluidos monofásicos. (como as consideráveis perdas de calor no TES e nos permutadores de calor do gerador de vapor), e também com geração direta de vapor (por exemplo, o reaquecimento direto do vapor na turbina e, no caso da operação em once-through, o limite de segurança relativo à diferença de temperatura máxima na secção transversal do tubo absorvedor, na secção de sobreaquecimento). Para resolver estes problemas, uma caldeira a biomassa pode ser utilizada no ciclo do HTF ou no de vapor, aumentando a temperatura do HTF, ou utilizada como backup. Os resultados deste trabalho de doutoramento, levam à conclusão da necessidade de realizar investigação adicional considerando a ampla gama de tecnologias (CSP, biomassa e ciclo de potência), de forma a explorar todo o seu potencial.

**Palavras-chave:** Concentração Solar Térmica; Biomassa; Centrais híbridas
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1 INTRODUCTION

Hitherto, the success of the electricity sector has been well-defined by ensuring security and reliability at low cost, empowered by the attractiveness of low-risk investment and secure revenues. These capabilities were attained mostly through the use of non-renewable energy sources (fossil-based fuels).

However, since the 90s, with the increase of apprehensions related to world global-warming, there was a consensus concerning the inevitability of diminishing greenhouse emissions [1]. These issues emphasised the need to decarbonise the electricity sector, responsible for about one-quarter of the greenhouse gases (GHG) emissions in the European Union (EU) and 29% in the United States of America (USA) (Figure 1-1).

To overcome such issues, the energy policy has to privilege power generation from renewable sources, leading to massive deployment and technological development of renewable energy technologies in order to shift to a more sustainable world. Nowadays energy policies try to reach a balance between security, environmental sustainability and affordability.
The potential of renewable energy systems is undeniable since they rely on a source that will not run out. Nonetheless, they often depend on meteorological conditions (e.g. solar, wind), leading to the unreliability of energy supply and consequently to grid connection issues.

Whilst it is easy to accommodate a small share of unstable renewable generation, larger shares bring new challenges to the electrical sector, enhancing instability and unreliability. These challenges were augmented with the large-scale deployment of wind and solar PV [4]. Renewable based power generation capacity increased about 800 GW between 2011 and 2016 (see Figure 1-2), of which the shares of wind energy and solar PV are 35.4% and 32%, respectively [5]. Furthermore, the forecast for the forthcoming five years shows that PV represents the most significant annual capacity addition. Most of the variable generation of renewables is compensated by the use of conventional fossil power plants for backup energy, which represents not only a non-environmental solution, but a limit to the penetration of renewables onto the grid as well. Additionally, the ambitious carbon reduction goals decided in Paris, are surely not attainable with a fossil-based backup strategy.
Figure 1-2 - Global renewables-based power and share of total capacity additions type.

Reprinted from: [5].

Electricity storage is undergoing intensive research. However, until now, no viable technology was found for optimal grid integration [6]. Moreover, pumped hydro energy storage is the only current electricity grid storage technology with substantial deployment throughout the world [7]. Whilst a feasible solution for electrical energy storage is within research, Concentrating Solar Power (CSP) can be a reliable solution due to the easiness to dispatch energy. Solar thermal energy is being presented as a resilient solution and CSP installed capacity experienced a considerable boost since 2009, mostly in Spain and United States (see Figure 1-3). Note that the results do not include the most recent data. In 2016 alone, about 110 MWel of CSP came online, and the global capacity improved to more than 4.8 GW. Additionally, new CSP projects are currently under development and commissioning in the MENA (Middle East and North Africa) region and China.

The ability to deliver dispatchable power is mostly related to the integration with thermal energy storage tanks. Nonetheless, the typically required storage size is considerable, usually representing an increase in the initial costs and a significant investment risk.
The use of bioenergy for electricity doubled from 2008 to 2015 (see Figure 1-4), due to a favourable energy policy. Nevertheless, the use of bioenergy within the countries electricity generation portfolios is limited. In 2016, 90% of all bioenergy capacity was confined to twenty-six countries. Additionality bioenergy electricity generation relies on a broad scope of biomass and wastes fuels.

An attractive solution is renewable hybridisation, that consists in the combination of different renewable technologies aiming to better explore their individual advantages.
Renewable hybrid generation systems represent a resourceful way of uninterruptedly meeting electrical energy demands.

Hybridisation of concentrating solar power with biomass can be a powerful way of assuring system stability and reliability, within the renewable concept. The possibility of both sources supplying thermal energy to drive a power generation block, extend the benefits to the sharing of plant equipment (e.g. turbine, heat exchangers, pumps), reducing installation costs. Furthermore, the easiness to store solar energy can further promote stability.

CSP standalone plants require huge values of direct normal irradiance only available in the sunbelt countries, usually in remote locations. Moreover, solar radiation is unpredictable, and unavailable during night time. On the other hand, biomass is available in different forms depending on the region, from woody biomass, to crop residues or organic waste. Although partially predictable, the seasonal variability affects the consistency of supply, storage and transport, and can introduce unique challenges. Furthermore, whilst biomass is still competitive, some of the feedstock prices have increased in recent years (see Figure 1-5).
Despite the undeniable potential of such systems, hybridisation still presents ambitious challenges, in order to guarantee that the system operates as a whole, and power reliability and stability are achieved.

1.1 Thesis structure

The thesis work is presented in eight chapters. The first chapter includes a summary of renewable energy and also CSP and biomass current status, as well as the thesis structure. Chapter 2 comprises a comprehensive review of CSP and biomass conversion technologies, as well as the state-of-the-art of CSP/biomass hybrid systems.

The research methodology is addressed in Chapter 3 where the purposed work is justified. The CSP/hybrid assessment was carried out through a developed dynamic numerical tool that permits the evaluation of different solutions, described in Chapter 4.

Chapter 5 includes the experimental work carried out for the assessment of gasification technology, and the numerical model validation. Model validation is presented in Chapter 6. Evaluation of different CSP with biomass hybridisation concepts was carried out through case studies, described in Chapter 7. The last chapter comprises the core assessment conclusions.

1.2 References


Chapter 2: State of the art

2 STATE OF THE ART

In this chapter, a comprehensive overview of CSP, biomass and CSP/biomass hybrid systems is presented. Initially, CSP technology is reviewed, and the hybridisation concept addressed. Afterwards, the advantages of CSP/biomass hybrid systems are described, and a biomass technology review introduced. The chapter ends with a wide-ranging review over CSP/biomass technologies based on available research literature and projects.

2.1 Overview of Concentrating Solar Power technology

Concentrating solar power plants use mirrors to concentrate the solar radiation, in order to produce heat for electricity generation through conventional thermodynamic processes. The concept of using solar radiation concentration is not new and has been described by Archimedes at around 200 BC. However, the technology and commercial proof was achieved only in the 1980s, with the deployment of nine parabolic trough power plants in California, with a total of 354 MW\textsubscript{el} of installed capacity, named Solar Energy Generating Systems (SEGS) [1].

Unlike other solar technologies (e.g. photovoltaics), CSP depends solely on beam (direct) solar radiation, not being able to use diffuse solar radiation. This makes it suitable for regions with high values of Direct Normal Irradiance (DNI), such as Sun Belt countries (Figure 2-1). In this region the CSP potential is vast, typically many times higher than electricity demand, increasing the interest of electricity export through high
voltage lines. For example, the concept was address under the BETTER project scope, funded by the European Commission to promote energy security by importing electricity from North-African countries [2].

![Figure 2-1 - Direct normal irradiance world map. Reprinted from: [3]](image)

There are four core CSP technologies for solar concentration: Parabolic Trough Collectors (PTC), Linear Fresnel Reflectors (LFR), solar tower with Central Receiver (CR) and Solar Dish (SD) They can be grouped in either linear (PTC and LFR) or point focusing (CR and PD) [4, 5]. Whilst linear collectors are usually endowed with a single axis tracking mechanism to follow the sun; point focus technologies require a two-axis tracking device. Each technology has specific advantages and limitations, wide-spread the interest of research and commercial deployment to all types of CSP technologies.

A CSP system is mainly constituted by three components (Figure 2-2): the solar field (SF), the solar receiver and the energy conversion system (power block). The solar field is a set of solar collectors where direct normal radiation is concentrated into a receiver. The solar receiver can be either part of the collector (e.g. PTC, LFR) or independent (e.g. CR). The generated thermal energy is directly used or collected by a heat transfer medium (e.g. thermal oil, steam, molten salt) in the receiver. This energy is then used to drive a thermal power generation system.
2.1.1 Linear focusing collectors

Parabolic trough technology is based on parabolic shaped mirrors, used to concentrate solar radiation into receivers, located at the focal line of the parabola (Figure 2-3). The receivers are usual steel tubes with a radiative coating, in order to increase the absorption of thermal energy and reduce radiation losses. Since the operating temperatures can go up to about 550°C, the absorber tubes are usually enclosed with an evacuated glass tube, in order to reduce convective losses.
For collecting the thermal energy, a heat transfer media is used. Most of the installed parabolic trough power plants rely on synthetic diathermic oil, which limits the operating temperature to about 390°C, due to oil degradation at higher temperatures and a consequent decrease in the global conversion efficiency of the plant.

In the last decade, due to intensive research, steam and molten salt have been used to overcome the temperature limitations. Moreover, steam and molten salt reduce the power plant complexity, since the generated steam can be directly used to drive a steam turbine and molten salt permits direct storage.

The collectors are set in rows or loops, usually north-south oriented to maximise the energy collected during the year. The parabolic trough solar collector (parabolic mirrors and absorber) is endowed with a single axis tracking mechanism, to track the sun from east to west during the day, to ensure a proper incidence angle of the beam radiation on the mirrors [7].

Usually, the power block is based on a typical Rankine cycle. The heat collected in the absorbers is used to generate steam to drive a steam turbine. The steam turbine efficiency is directly related to the operative temperature, and for lower temperature applications, the Organic Rankine Cycle (ORC) can be used [8]. The principle is the same as with a steam turbine, but the working media is an organic fluid with a lower boiling point. At lower temperatures, ORC systems can provide better efficiencies than steam turbines at equal temperatures. Nevertheless, the associated cost per MW_{el} is higher.

Leading countries in PTC and CSP installations are USA and Spain. In the USA the current power plants are usually deployed in the 100 MW_{el} range, based on economies of scales. Whilst CSP technology efficiency is nearly unaffected by the size, the costs significantly drop. In Spain, power plants are limited to 50 MW_{el} by the Spanish Special Regime.

The linear Fresnel reflector technology relies on a set of rows of grounded based mirrors, tilted at different angles in order to concentrate solar radiation into an elevated fixed receiver (Figure 2-4). The tracking mechanism uses a single axis as the parabolic trough. However, it is applied to each row of mirrors.
Regarding optics, the ideal reflectors for single receivers are parabolic or paraboloidal based mirrors. Nevertheless, the development of such mirror shapes is unwieldy at a larger scale, due to the complexity of the structural requirements in order to withstand loadings, along with the amplification of issues concerning operation and maintenance. To overcome such problems, LFR collectors emulate the concept of larger reflectors using a set of flat linear reflectors displaced over the ground surface, reducing the structural intricacy and enhancing maintenance operations [9]. As expected, the optical performance is lower than with PTCs. To solve such issue, a multi-tube receiver is usually used, or a second concentrator (mirror) above the static receiver. In any case, for the traditional LFR collectors the optical losses are higher, however new concepts with improved optical performance are under development. On the other hand, one of the advantages of this technology is its simplicity, and therefore the associated manufacturing and installation costs are lower.

Concerning the heat transfer media and the power block, a LFR power plant is similar to a PTC plant. Either synthetic oil, molten salt or water/steam are feasible working fluids. The generation system usually relies on a conventional steam turbine cycle. The investment costs are usually lower than with other CSP technologies due to the inherent more straightforward solar field.
2.1.2 Point focus technology

A central receiver system consists of an array of heliostats in order to reflect solar beam radiation onto an elevated single receiver, placed on a central tower (Figure 2-5). The set of mirrors are adequately displaced in the solar field to promote optical efficiency and are endowed by a two-axis track mechanism [10].

The receiver is a critical component of the system, where the concentrated solar radiation is intercepted and absorbed at high temperatures. The generated heat is collected by a thermal medium and used to drive a thermal, electrical generation system.

Concerning the heat transfer fluid, either water/steam, molten salt or synthetic oil can be used to drive a steam Rankine cycle. For higher temperature applications, gas can be used as heat transfer medium (e.g. air), and the power generation system can rely on the Brayton cycle.

![Figure 2-5 - Central receiver system. Source: Adapted from [4].](image)

The CR system complexity is high. The receiver design involves several parameters, and the array of heliostats requires a rigorous and optimised control. On the other hand, these systems can achieve higher concentration factors, and consequently higher temperatures and efficiencies, stimulating the interest and research.

The Parabolic Dish (PD) technology relies on paraboloidal dish-shaped mirrors to concentrate solar energy into a receiver (Figure 2-6), strategically placed close to the focal...
point. Usually, the receiver is associated with a Stirling engine, allowing direct electrical generation. The Stirling engine uses heated gas (e.g. hydrogen, helium) to drive the shaft. Nevertheless, the PD can be used with other conventional thermodynamic conversion cycles (e.g. Rankine, Brayton).

![Parabolic dish technology](image)

Figure 2-6 - Parabolic dish technology. Source: Adapted from [4].

The commercial deployment of Parabolic Dish technology is negligible, though they have the highest optical efficiency and concentration ratio. The sun tracking is achieved by a two-axis tracking mechanism, yet unlike CR the receiver is not fixed, reducing the cosine effect. Another advantage of this technology is that it does not require water since the cooling is achieved by the use of a radiator [11].

Contrasting with other CSP technologies, PD thermal inertia is minor reducing the ability to provide dispatchable and stable energy power. Therefore, they are usually used for distributed generation in remote locations.

It is worth to note that there are other types of solar concentrators feasible for power generation, particularly for low-temperature applications. A Compound Parabolic Concentrator (CPC) is a non-imaging concentrator since they do not produce an optical image of the sun [12]. The main advantages of these concentrators are the ability to accept radiation in a wider range, either beam or diffused, reducing the need for permanent tracking and also the reduced costs. However, concentration ratios are usually lower than the aforementioned technologies, reducing the ability to achieve high temperatures, desirable for power generation.
2.1.3 Technology maturity

The earmark between CSP technologies is noticeable on their maturity as well, perceptible by the deployment rates (Figure 2-7). In 2015, 85% of the CSP plants in operation and 50% of the plants under construction rely on parabolic trough technology. The central receiver technology is emerging due to inherent advantages related to higher operating temperatures, although they still lack maturity when compared with PTC.

The parabolic trough can be considered as a mature technology [4-6, 13], with a considerable number of manufacturers [14] and more than 20 years of operating experience with respectable results. They can be considered low-risk projects [5]. Therefore, for novelty concepts, such as hybrid power plants, financing would be easier with PTC technology.
2.1.4 Dispatchability

The oil shocks in the 1970s, along with environmental consequences from the use of fossil fuels, triggered massive research and deployment of renewable electricity generation. Over time, the penetration of renewables in the grid increased and raised some issues concerning dispatchability, security and stability [15].

Today’s electricity system operates based on dispatchable generation. The intensification of variable generation deployment in the grid requires new management
schemes. The advantages of the upswing of wind and more recently photovoltaic systems are undeniable. Nonetheless, their impact in the grid is associated with a few drawbacks, such as more substantial and volatile power flows associated with security challenges [16].

It is possible to accommodate larger capacities of transient renewable energy generation, by means of a vertically integrated monopoly system (e.g. EDP in Portugal). Usually, dispatchable energy generation systems are used to accommodate the variability of renewable sources. Nonetheless, the typical forms of dispatchable power generation rely on non-renewable sources (e.g. natural gas, coal), increasing the grid dependence on fossil fuels [17].

Nowadays, the concept of smart-grids is growing. The idea consists in adjusting the demand in order to bridge the load and generation [18]. Nevertheless, this concept requires complex adaptations in the power generation and grid systems, as well as in the larger power consumptions poles. In other words, a novel electricity system. Moreover, the increase of electrical generation, when renewable sources are favourable, significantly affects electricity prices, with a significant decline in the production costs, inducing a significant impact on the economy of the electricity market [19].

The strategies of prices and tariffs in the power system are one of the keystones drivers of renewable energy. On the other hand, specific products have been implemented to compensate demand-response providers to balance the grid.

The variability of renewables is a hindrance to their continuous and broader deployment. The current trend on building a renewable flexible dispatchable power can be enhanced on the supply side. Dispatchable renewables, such as bioenergy based and CSP plants can underwrite a flexible and predictable power system, ensuring a fast response on the supply to accommodate the demand.

2.1.5 The value of thermal storage

If a CSP plant relies solely on solar energy, electrical output will be limited to sunshine hours and contingent with daily transients of solar radiation. The system can be enhanced through the use of Thermal Energy Storage (TES) and hybridisation with fuel
combustion systems, extending the operating range of the power plant and improving both looked-for dispatchability and electrical stabilisation of the grid.

Whilst thermal power plants are feasible for energy storage, other renewable (e.g. solar PV, wind) require energy storage in the form of electricity that represents a much more complex and expensive proposal.

Dispatchability is the main advantage of CSP among other renewable resources. Indeed, the flexibility to feed the grid on demand represents the key to CSP deployment. This advantage is being hitched with the use of thermal energy storage (Figure 2-8).

![Typical CSP plant operation with storage. Reprinted from: [5].](image)

The most comprehensive thermal energy storage system consists in two-tanks of molten salt. Molten salt can be used as direct or indirect storage depending on the CSP plant technology and design, mainly on the heat transfer medium [20]. Nowadays, most of the existing and planned commercial CSP plants include a TES system. In the last two years, all the new facilities incorporated thermal energy storage [21].

Despite the attractiveness of the TES concept, it is associated with some drawbacks. One of the most critical design parameters of a CSP plant is the ratio between the actual size of the solar field and the required field size to drive the power block at nominal power, for the design DNI at solar noon at the plant location. This ratio is usually defined as solar multiple.
In a CSP plant with a solar multiple of 1, nominal power generation will only be contingent to peak sunlight hours. In order to extend nominal power generation and increase system efficiency, a higher multiple is mandatory. The solar multiple is a design parameter of a CSP plant, and for solar-only mode, the typical values vary between 1.1 and about 1.5. On the other hand, if the plant is associated with TES a higher solar multiple is desirable. It is necessary to produce extra heat during the day to charge the storage tank. The solar multiples range increase (3 to 5), for larger storage systems [5].

Another design parameter is the storage size. It is often quantified by the time that the plant can operate exclusively from storage at nominal power. Small storage tanks are used mainly to accommodate short transients in solar radiation and evening power demand. On the contrary, the larger systems are limited to 24 hours of operation per day during summer months. Nevertheless, the solar irradiance discrepancies between summer and winter times result in a non-uniform generation over the year. Moreover, designing a system for the lower radiation season is not economically feasible, since it requires higher solar multiples and thus energy waste in the summer months.

In 2015, the costs of a CSP plant with storage were in the range of 6000 to 9000 USD/kW_{el} (see Figure 2-9). Analysing the cost of a plant without storage (4000 to 6000 USD/kW_{el}) it is noticeable that the values decrease about one third. It is noteworthy that the presented costs are related to a small storage system (6 hours). Despite the favourable cost reductions in the last years and for the forthcoming decades, the storage system will continue to play an essential role in the CSP system global cost [22].

Figure 2-9 - CSP investment with and without storage cost projections. Reprinted from: [5].
Furthermore, more than 50% of the costs of a PTC power plant are associated with the solar field and thermal energy storage (see Figure 2-10). In addition, the storage tank size is sustained by the solar multiple, and consequently by an increase of the solar field size.

Notwithstanding the dispatchability and energy security improvements that arise from the use of thermal energy storage, it also results in an increase of power plant costs. The upsurge of costs represents an extra barrier to a massive deployment of small-scale CSP plants. Therefore, CSP economic feasibility is usually based on economies of scale and associated with intensive and risky upfront capital investments.

Figure 2-10 - Simplified cost breakdown structure of a 50 MW STE plant with 7.5 hours of storage. Reprinted from: [6].

2.1.6 CSP Hybridisation

Whilst the dispatchability has been widely proven through the use of TES, it still represents a costly solution [23]. Like CSP power plants with TES, the combination of CSP with other heat sources (i.e. hybrid solar power plants) allows the de-coupling of solar incident power and electricity generation.

CSP hybridisation has been achieved in different forms, mostly with fossil fuels. The most common is the use of a backup boiler to supply extra energy to drive the power
block in times of low radiation and the absence of daylight (Figure 2-11) or to boost (i.e. superheat) steam temperature.

![Figure 2-11 - CSP operation with storage and backup. Reprinted from: [6].](image)

Despite the undeniable advantages of such systems, hybridisation still presents ambitious challenges, in order to guarantee that the system operates as a whole, and power reliability and stability are achieved.

A European Solar Thermal Electricity Association (ESTELA) report from 2012, compared the state-of-the-art from parabolic trough technology, including storage and forthcoming targets for solar-only and hybrid, in 2015 (Figure 2-12). It is noticeable that CSP with energy storage entails higher investments, but nonetheless the oversized solar field improves the solar output.

The cost index is below 1, meaning that the specific investment for doubling the output is not the double, due to equipment sharing. The use of molten salt allows the additional use of the heat transfer medium for storage, increasing the simplicity of the plant and compensating offsetting the associated costs. Furthermore, the higher operating temperature leads to an increase in power block efficiency, resulting in a cost index of 0.73. Further developments are only achievable through novel heat transfer fluids.
The same report considers that hybrid CSP plants represent an attractive option for growing energy markets, where energy dispatchability and firm capacity are key targets. The cost reduction with hybridisation is highlighted as the main advantage. The lower cost is related to a more conventional plant design, higher operating temperatures, and lower cost of the heat transfer fluid (e.g. water/steam). As aforementioned, the associated costs of CSP plants are upfront increasing the investment risk, so that hybridisation can enhance the deployment at smaller scales, expanding the technology to other markets and companies.

There are different approaches regarding CSP hybridisation. Currently, most CSP plants rely on a backup boiler usually running on fossil fuels, mostly used to compensate fluctuations in solar radiation during sunlight hours, to improve the start-up time at sunrise and enable night operation.

The first commercial CSP plant, SEGS, relies on natural gas as backup energy. In summer it is used to extend operation after sunset, and during winter to supply the extra energy required during low radiation hours. The natural gas share is limited to 25% of the primary output. Spanish CSP plants followed the same concept to grant dispatchability and stable capacity. However, with stricter requirements regarding the natural gas share (12% to 15%), depending on the support system.
Another concept is the combination of CSP with coal-fired steam power plants [24]. In this approach, the solar field is placed in parallel with the coal boiler or to replace the plant steam preheaters. The most usual is the use of CSP as a booster for coal power plants. In this concept, the feed-water is preheated by the solar field as an alternative to the use of the steam extracted from the turbine. Therefore, the power output is enhanced, since more steam is expanded in the turbine and also CO₂ emissions are diminished.

Integration of CSP and combined cycle power plants is an additional solution [25]. Within this concept, the residual heat from the exhaust gas is used to drive a steam turbine. The solar field can supply extra energy in order to generate more steam, and therefore increase the generation output. This concept is frequently used, as an add-on to either existing or new fossil power plants (Figure 2-13).

It is noteworthy that the above concepts are non-renewable. Moreover, typically CSP plant projects are associated with incentives, such as feed-in tariffs in order to attract investment. Such incentives are grounded by the renewable nature of the resource and lack of maturity of CSP technology. Therefore, the use of fossil fuels as a hybrid solution
is usually regulated or even prohibited, lessening the looked-for dispatchability. As an example, the aforementioned incentives for Spain where removed for natural gas. [27]

Within the concept of a fully renewable energy system, biomass is the ideal contender for CSP hybridisation. In this concept, biomass can be used as energy backup and to increase the operating temperatures of the system, enhancing both dispatchability and system global efficiency.

With this kind of synergy, solar energy takes the primacy in the primary output, and the electrical generation system is driven as long as possible by solar energy. To overcome the problem of the intermittent nature of solar energy (e.g. cloudy day or night), biomass is used. Therefore, biomass will allow the stabilisation during the night regimes when solar radiation is absent and enhance the thermal power when daily conditions are not favourable. Whilst solar operation extension with thermal energy storage is feasible, a fully dispatchable system will be achieved only in summer months. On the other hand, biomass represents a renewable dispatchable source, with a considerable potential for enhancing concentrating solar power technology (Figure 2-14).

![Figure 2-14 - Renewable generation fit. Reprinted from: [28].](image)

To operate in a seamless symbiosis, several aspects need to be addressed. The choice of a proper biomass technology will depend on different critical issues, namely: size of
the plant, CSP and power block technology, working fluid, operating conditions (i.e. temperature and pressure) and locally available biomass resources.

2.1.7 Bioenergy

Biomass represents all non-fossil organic matter that has an inherent chemical energy content [29], and is available in different forms such as wood, crop residues or organic waste (Figure 2-15), with distinct energy densities, physical states and associated costs. There is a significant dispersion of the costs, mostly related with the easiness of use. Whilst international traded feedstocks can be used in conventional residential boilers without significant adaptations, the use of wastes require pre-processing facilities to enable the use of the feedstock energy.

![Figure 2-15 - Typical biomass feedstocks and cost (USD/GJ). Adapted from: [30].](image)

There are several processes to convert biomass into fuels or energy (Figure 2-16). These conversion processes can be subcategorised into thermal (e.g. combustion, pyrolysis and gasification) or biochemical (e.g. fermentation, anaerobic digestion (AD) and chemical conversion).
CSP hybridisation relies on the ability of both systems (solar and biomass) to generate heat, limiting the technology choices to combustion, pyrolysis, gasification and anaerobic digestion.

Combustion is the most direct and ancient process of biomass conversion into thermal energy and the most common form of bioenergy [32]. At user scale, it is still used as primary source of heat in less developed countries [33]. At a larger-scale, biomass combustion plants are a well-established technology, used either directly for heat production or indirectly for power generation (e.g. Rankine cycle). Modern combustion plants rely on different technologies: fixed bed, suspension or fluidised bed (see Figure 2-17) [32]. The technology choice is mostly associated with the feedstock characteristics, for example: density (bulk and energy), dimension, moisture content, calorific value, volatile matter.
There are two types of fixed bed combustion systems: underfeed stokers and grate furnaces. As a shared feature, combustion takes place in the grate. In grate furnaces the primary air flows through a fixed bed where drying, gasification, and charcoal combustion take place, that is where the combustible gases are produced. In the grate, the fuel gases are burnt with secondary air. Grate furnaces are suitable for biomass fuels with varying particle sizes, high moisture and ash content. Nevertheless, there are limitations regarding the fuel mixtures. Distinct types of grates exist, classified according to the fuel transport type over the grate: fixed, moving, travelling, rotation and vibrating [32, 34, 35].

In underfeed stokers, the burned biomass is pushed downwards by fresh biomass. This kind of systems is characterised by a low cost and small/medium scale capacity (up to 6 MWth). It is adequate for low ash content feedstock and with particle dimensions below 50 mm. In a suspension bed, fine particles (e.g. sawdust) are pneumatically injected and dried using suspension burners. The most significant disadvantages of the systems are the need for a constant and low fuel size and moisture content [32].

A fluidised bed combustion comprises a cylindrical vessel filled with bed material (e.g. sand, silica), which is fluidised through primary air flow from the perforated bottom plate. The air flow fluidises the bed and significantly enhances heat transfer. As a result, complete combustion is possible with a low air excess demand [36]. Lower temperatures (650°C to 900°C) than with a fixed bed (usually 100°C to 200°C higher) are required to
avoid ashes issues. Regarding the biomass feedstock, this system can operate with distinct biomass mixtures, yet with limitations concerning particle size and contaminants. Additionally, the start-up process is rather long (8 to 15 hours) and thus not suitable for small-scale power plants [32, 37].

Depending on the air (fluidisation) velocity, fluidised bed combustion systems are classified into two types, Bubbling Fluidised Bed (BFB) or Circulating Fluidized Bed (CFB). In BFB, an upwards air stream with velocities between 1 m/s and 2 m/s supports the bed particle fluidisation. Increasing the air velocity (5 to 10 m/s) permits to drag out the bed and fuel particles with the gas stream, and after that separate them in a cyclone and redirect to the reactor (circulation).

Pyrolysis can be defined as the thermal decomposition of carbonaceous matter in the absence of oxygen. The main products are solid charcoal, bio-oil and volatile gas, and their balance depends on the reaction temperature, heating rate and residence time. The volatile gas can be used for heat production. However, the quantity produced is usually too low to enable the feasibility for CSP hybridisation [35].

As an alternative to direct combustion, biomass gasification is one of the most promising technologies. Gasification can be seen as an extension of pyrolysis, where the gas, char and tar produced by pyrolysis are subjected to further reactions. The gasification process encompasses four stages: drying, pyrolysis, combustion, and reduction.

Initially, the feedstock moisture content is reduced, preferably before entering the gasifier. At temperatures between 150ºC and 400ºC and within an oxygen-depleted environment, pyrolysis occurs and results in char and tar gases. Part of the tar gases is burnt to sustain the endothermic reactions. In the end, the char content is reduced to fuel gas [38].

The primary product is a fuel gas rich in hydrogen and carbon monoxide. The gas mixture can also contain light hydrocarbons (e.g. methane), carbon dioxide, water vapour and nitrogen in different proportions, as well as particles (e.g. char, ash, soot). The gas composition is highly dependent on the reactor type and operating temperatures. Other key variables are the feedstock size and chemical composition, and the gasifier technology. The produced gas can be used for heat production through the use of a gas boiler, or for power generation via combustion engines or gas turbines.
Similarly to combustion, there are distinct gasifier technologies: fixed bed (e.g. updraft and downdraft), fluidised bed, and entrained bed (Figure 2-18) [39]. The latter was developed for coal gasification, and is usually not used with biomass due to the feed material size (< 0.1-0.4 mm) limitation [40].

Depending on the flow direction, fixed bed gasifiers can be classified into downdraft, updraft and cross flow. The simpler gasifier is the updraft type, where biomass is fed from the reactor top and moves in counter-current with the air, supplied from the bottom. One of the main advantages is the internal heat exchange that permits a lower gas temperature at the reactor outlet, and thus better overall efficiencies. On the other hand, in a downdraft configuration air and feedstocks move in the same direction. The main advantages in comparison to the updraft type is a lower tar content in the gas, however at a higher temperature and thus with an overall reduced efficiency [36, 41].

The features of fluidised bed gasification and combustion are comparable. The biomass feedstock is suspended into a hot fluidised bed (bubbling or circulating) [42]. In comparison to a fixed bed system, pyrolysis is significantly faster. The continuous process of mixing solid feedstock through the bed permits to attain uniform, yet lower temperatures, and so lower tar conversion rates [37, 38, 40, 42].

Fluidised bed gasifier systems are of two types: bubbling and circulating fluidised bed. The bubbling fluidised bed concept is more straightforward, whilst the circulating
fluidised bed more adequate for large-scale power plants. The oxidiser is a crucial variable within the gasification process [35, 37, 39]. Whilst air is the most conventional choice, oxygen results in higher lower heating value by the absence of nitrogen, though at higher costs. Another option is the use of steam [38, 40].

Regarding the biochemical processes, anaerobic digestion is a process where organic matter is decomposed through microorganisms (i.e. bacteria) in the absence of air, resulting in a gaseous fuel rich in methane (biogas). The biogas can be used to generate heat or electricity, or both [43].

In the absence of oxygen, bacteria decompose the organic matter in order to sustain energy for their metabolism, resulting in methane by-product. The process can be described through four stages: hydrolysis, acidogenesis, acetogenesis and methanogenesis (see Figure 2-19) [44].

Within hydrolysis, the acidogenic bacteria decompose the large organic polymers into amino-acids, sugars and long chain acids. Over acidogenesis, fermentative bacteria convert the hydrolysis products into methanogenic substrates, mostly acetate acids and hydrogen. Some of the acidogenesis products cannot be directly converted into methane. Those, are converted into methanogenic substracts over acetogenesis. The last stage entails the production of methane and carbon dioxide through methanogenic bacteria [45].
The bacteria activity is correlated to the digestion temperature, and can occur at distinct temperatures: psychrophilic (<25°C), mesophilic (25°C to 35°C) and thermophilic (35°C to 55°C). Higher temperatures result in lower retention times, and higher gas yield. However, temperature stability is a crucial parameter [43].

There are different types of reactors: dry or wet, batch or continuous, single-step or multi-step, and single-phase or multi-phase (see Figure 2-20) [46]. As a shared feature, the reactor choice is intrinsically related to the feedstock proprieties. In a batch dry reactor, feedstocks with a solid content over 30% are fed once, and retention times extended to full degradation. Thus, the gas production is characterised by a peak; when it ceases, part of the feedstock is renewed by fresh organic matter [46]. The main advantage is the simplicity. Through the absence of moving parts, feedstock contaminants are not an issue. On the other hand, the biogas yield is not optimised.
In a continuous wet reactor, the feedstock dry solids content should be lower than 12% (e.g. sewage slurry). This reactor type includes a stirring system to assure an appropriate mixing of the feedstock. Whilst most of the commercial systems are operated in a single-step, a two-step system can be used. The main advantage of multi-step is the possibility to recycle the liquid digestate of the second reactor [43, 46]. Another typical system is a dry continuous reactor. The process consists in continuously feeding dry matter, with a solid content between 20 to 40% [47]. It can be thoroughly mixed, yet tends to be a plug flow system [46].

The choice of the conversion technology depends on the type, availability and cost of local biomass feedstock, as well as of the purpose. Although a wide range of biomass feedstocks can be used with different technologies, the output will not be the same. Designing a boiler or a gasifier for a wide range of feedstocks is also possible, but there is the drawback of the system having to be designed for the worst-case scenario. Moreover, for hybridisation with CSP temperature is critical, which is contingent on both feedstock nature and the biomass conversion technology employed.

Biomass is a dispatchable renewable energy resource, yet some issues should be addressed to assure bioenergy sustainability, that have been discussed in detail [48, 49]. The most important is an environmental concern, related to the overuse and CO2 emissions. Biomass is a keystone in the world environmental balance and should be sustainably explored, which is conceivable trough proper management and policy regulation. Intensification of forest biomass exploration, represents a thoughtful drawback to soil fertility, and degradation to the ecosystem as well. Agricultural crops
are usually used as food, bespeaking the increase of food prices (e.g. bioethanol from corn in the US).

The energetic use of effluents or organic waste, can be addressed by anaerobic digestion and presents a feasible solution for the waste disposal issue. Notwithstanding the unquestionable value of the concept, the residues are assorted leading to distinct and unsystematic biogas compositions. The risks associated with biogas power plants are higher due to several unsuccessful cases, and companies narrow most of the know-how. A possible alternative consists in using Refuse Derived Fuel (RDF) as combustion or gasification feedstock [50, 51].

The most prominent key-driver of biomass for heat and power production is the CO₂ negligible emission. The concept has been recently questioned. Biomass and fossil fuel CO₂ emissions are similar, yet the CO₂ released during biomass conversion is balanced by the one absorbed during its growth. The disagreement is related to the whole lifecycle analysis that includes transport and soil use. Nevertheless, biomass emissions are usually lower than with fossil fuels for gasification, and if organic wastes are used the reductions are extended to methane as well (landfill emissions).

2.2 CSP/biomass hybridisation

As aforementioned, the concept of CSP hybridisation is not new. CSP hybrid systems go back to the first CSP power plants: the SGES project from the 80s with the use of backup boilers running on natural gas. The ensuing CSP plants were deployed within the same concept, with limits regarding the fossil fuel share in the primary output. The dispatchability issue was addressed by the use of TES. The economic crisis together with the urge to replace fossils fuels led to the elimination of feed-in tariffs for fossil fuel share. Furthermore, there was an increase of concerns regarding energy security associated with an accretion of energy dispatchability worth. A fully renewable alternative consists in CSP/biomass hybridisation. In this section, a comprehensive literature review of the CSP/biomass theme is presented.

In the past seven years, the interest on CSP/biomass hybrid power plants significantly increased. Several studies have been conducted considering different combinations of CSP and biomass technologies.
In 2011, HongJuan et al. [52] studied the hybridisation concept by using a PTC solar field with thermal oil to preheat the power block feedwater of a biomass power plant. The assessment was carried out through an equivalence enthalpy descending analysis method, and the purpose was to reduce the amount of steam extracted from the turbine to the preheaters. The authors concluded that hybridisation improved both technical and economic performance. A reduction in the biomass consumption was noticed, as well as on the solar field efficiency. Furthermore, the results from the analysis showed that the best results regarding performance and economy are associated with higher replacements rates.

In 2012 Moreno-Pérez and Castellote-Olmo [53], assessed the concept of PTC hybridisation with biomass, with emphasis on the Spanish market. The evaluation was carried out by a numerical model and considering the replacement of a molten salt TES system by a biomass combustion boiler driven by olive residues. The authors summarised the key advantages: increased system efficiency, lower Levelised Cost of Electricity (LCoE) and reductions on the net collector surface. Moreover, the economic viability of such plants is attained for lower values of direct normal irradiance due to lower LCoE, extending the implementation range. The immature biomass market is mentioned as a drawback to the technology deployment. The authors conclude that a smaller hybrid size power plant can be profitable, requiring lower capital investments and consequently lower financial risk.

Nixon et al., assessed through numerical simulation the feasibility of solar-biomass hybrid power plants for India [54]. The authors evaluated different generation options (e.g. CHP, tri-generation) and different power outputs (2 to 10 MWel), as well as different solar multiples. Concerning costs, the authors assert that the LCoE is higher than with conventional energy sources, yet competitive with other renewable energy technologies. Concerning payback periods, the results showed that biomass-only represented the best solution. Competitiveness of a hybrid plant requires an increase of about 1.2 to 3.2 in the feedstock cost. Additionally, the authors defend a LFR collector with a secondary CPC as the best solution.

In 2012, Coelho et al. analysed different possibilities for central receiver systems to be hybridised with biomass [55]. The authors addressed the use of a biomass combustion
boiler, and forest waste as feedstock, for the south of Portugal, and concluded that hybridisation is technically and economically viable. The hybrid power plant showed higher efficiencies than conventional CR plants, and reduced consumption when compared with biomass-only power plants. The novelty of the concept hindered the profit, resulting in higher risks.

In the same year, Peterseim et al. [56] presented an analysis of CSP and energy waste hybridisation plants, focusing on the selection of CSP technology, viable locations and integration concepts. Concerning location, the authors stressed that an annual DNI of 1500 kWh/m² was required, representing a 25% reduction from CSP-only requirements (2000 kWh/m²). Furthermore, the similarities between both plants enhanced the reduction of the LCoE.

With the aim of joint uninterruptable generation and fuel production, in 2013 Vidal and Martin [57] evaluated a novel hybrid CSP/biomass poly-generation plant, for the south of Spain, using a multi-period mixed-integer nonlinear program. Indirect gasification was found out to be the optimal solution for biomass conversion, with CR plants for fuel production. The LCoE was penalised since the cost of this concept requires double of the investment when compared to conventional fossil fuel power plants.

Another novel concept was presented by Angrisani et al. [58], consisting in Combined Heat and Power (CHP) based on a Stirling engine. Electrical and heat production relied on a biomass fluidised bed combustor and concentrated solar radiation, by the use of a Scheffler mirror to concentrate solar radiation onto the bed. One of the key advantages is operation at higher temperatures than typical CSP, increasing the system performance. The novelty of the concept and the reduced size of the proposed system lead to unfeasible investments.

Borello et al. [59], analysed a small-scale CHP based on the typical Rankine cycle. Thermal energy relied on a PTC plant hybridised with a biomass boiler. The work was carried out through transient numerical simulation. The results for a winter day showed an excellent agreement between demand and supply for heat and power, and the primary energy ratio was favourable. On the other hand, the cost of such power plant was found to be high, and the economic viability requires costs reductions, particularly in the solar field.
Peterseim et al. [60], assessed the best CSP technologies for hybridisation, with conventional and non-conventional fuels, based on a literature review. The authors used a multi-criteria decision-making tool based on data analysis through an analytical hierarchy process. The results showed LFR as the best technology for feed-water preheating and steam boost application, at temperatures below 450ºC. The second-best solution, for lower temperature application, is PTC technology. Higher temperature feasibility is limited to solar tower and parabolic dish technology. Above 450ºC, CR showed the best performance with Direct Steam Generation (DSG) instead of molten salt. For temperatures higher than 580ºC, the only solution is PD. Another conclusion was that a reliable assessment concerning the most suitable technology for a specific CSP project is contingent to the project parameters (e.g. DNI, climate condition, space constraints).

Peterseim et al. [61], used a case study to analyse the benefits of a CSP/biomass hybrid power plant in Australia. The chosen CSP and biomass technologies were a central receiver and a water tube boiler using straw as feedstock. The results showed a 43% reduction in the investment when compared with a standalone CSP plant, which can be seen as an improvement for CSP employment.

In 2014, several CSP/biomass hybrid concepts and issues were addressed. Srinivas and Reddy [62], studied the integration of CSP with an existing biomass plant. The objective was the biomass consumption reduction, as the case study plant lacked of biomass feedstock. The control of the hybrid plant was achieved by the biomass burner and solar share limited to 50% in order to avoid combustion at lower fuel rates. With the increase of the solar share (10% to 50%), the fuel plant efficiency experienced a considerable boost from 16% to 29% related to the stabilisation in the supply side. On the other hand, there was a reduction in the system efficiency during daylight, due to solar collectors’ lower efficiency when compared with the biomass boiler.

Miguel and Corona [63], studied the environmental issues of replacing natural gas with biogas or biomethane to run the backup boiler of a parabolic trough power plant in Spain. The analysis was carried based on a Life Cycle Assessment (LCA) methodology. The results showed a significant reduction in the environmental impact associated with climate change, land transformation and fossil fuel depletion. On the other hand, the use
of biomass-based gaseous fuels increased the impact of water depletion and toxicity. Even so, overall positive effects outweigh the negative ones.

Peterseim et al. [64], modelled the possibility of using biomass boilers to superheat steam, to increase system efficiency in Australia. The plant was 50 MWel with 7.5 hours of molten salt thermal storage. The results showed an increase of 10.5% in the net efficiency and a 23.5% reduction on the investment. An assessment concerning 17 hybrid configurations of CSP/biomass hybrid power plants was carried by the same authors [65]. The analysis encompassed technical, economic and environmental performances. The study was conducted by numerical simulation of distinct combinations of PTC, LFR and CR technologies with biomass grate, fluidised bed and gasification. The higher efficiencies were associated with solar tower hybridisation (32.9%), whilst LFR/biomass represented the lower investment.

Central receiver technology hybridisation with biomass has been addressed in recent studies [66-68]. Coelho et al. [66], analysed several base cases for power plants and hybrid CSP/biomass options, for the Portuguese market. The authors stressed that the central receiver system LCoEs are impaired by the power block capacity factors and thermal energy storage cost. Also, the recent increase in biomass prices jeopardised the viability of biomass power plants. Hybridisation can provide the economic turnover if bonus feed-in tariffs were granted. Wood gasification, RDF pellets, biogas from a wastewater anaerobic digestion and landfill were analysed. The combination of biogas from wastewater sludge with an atmospheric volumetric receiver, led to the lower LCoE (0.15 €/kWh) and an optimistic payback period of 13 years.

The concept of CSP/biomass hybridisation for the Spanish market has been addressed, focusing on the mature technology of parabolic trough [69, 70]. The energy payback time was found to be higher for the hybrid power plant (1.72 to 1.83 years) than the standalone solar power plant (1.44 years) [69]. Colmenar-Santos et al. [70], proposed and analysed CSP/hybridisation as an alternative to thermal energy storage. The case studies were Spanish CSP power plants without TES and the proposed biomass technology was based on biogas from waste. Some advantages and weakness were summarised. The authors concluded that the potential is considerable, yet the technology deployment can struggle due to some drawbacks. Biogas plants are deployed in lower
scale than CSP, and so hybridisation would require innovative biomass plant concepts and consequently higher risks. The Spanish legislation limited the share of other energy sources on the plant output, and biomass hybridisation was not explicitly addressed.

Whilst several studies have been conducted focusing on CSP/biomass hybrid power plants, the first commercial power plant only started to operate in December 2012 [71]. The Borges Termosolar project (Figure 2-21) is a 22.5 MW$_{el}$ hybrid CSP/biomass plant in Spain.

![Figure 2-21 - Borges Termosolar power plant. Source [72].](image)

The solar field relies on PTC technology with an aperture area of about 180 000 m$^2$. As heat transfer medium, thermal oil is used, and two biomass boilers (22 MW$_{th}$) are employed to drive a conventional steam Rankine cycle. In order to enhance the power cycle efficiency, the boilers are used to boost steam temperature up to 520°C, hardly achieved with PTC and thermal oil combination [73]. Whilst most of the CSP power plants are confined to high DNI, and clear sky regions (e.g. desert), where biomass is usually scarce, Termosolar Borges project is placed near Barcelona, further north than other Spanish CSP projects (Figure 2-22).
In 2014, the Rende power plant Cycle plant was announced. The project consists of a moderate hybridisation of an existing biomass power plant (14 MW_{el}) built-in back in 2000 with a small LFR solar field (1 MW_{el}) [74]. The system design temperature is about 300ºC and as feedstock virgin wood will be used [73].

In 2015, India’s Center for Study of Science, Technology and Policy (CSTEP) announced the construction of a 3 MW_{el} CSP/biomass power plant in the village of Barun [75]. The Scalable CSP Optimised Power Plant Engineered with Biomass Integrated Gasification (SCOPEBIG) project, is being carried under the EU-India Cooperation on Renewable Energy. The solar field relies on parabolic trough technology with direct steam generation, and a biomass gasifier is used for hybridisation. Both sources share 50% of the power generation. The project aims to address specific issues such the use of untapped and abundant biomass resources in India, as well as develop viable business models of CSP/biomass hybrid power plants for broader dissemination [76].

A novel system has been developed by AORA, named as The Tulip hybrid system (Figure 2-23). The system has a nominal power output of 100 kW_{el} achieved by a gas micro-turbine. The system scalability is assured by modularity and uses a solar tower system hybridised with a boiler, suitable for both fossil and biomass gaseous fuels (e.g. natural gas, biogas). In addition to hybridisation, the concept of small modularity is interesting since it reduces the complexity of the heliostat control. The first prototype was
installed in Israel in 2009. Additionally, a prototype was implemented in 2012, in the solar platform of Almeria (PSA).

![Figure 2-23 - AORA Tulip concept. Adapted from: [77].](image)

Despite the undeniable value of the aforementioned studies, and the manifold of solutions assessed, most of the research work lacks experimental work. Whilst commercial projects are already deployed and others under development, none addresses specific research issues.

In the framework of REELCOOP (REnewable ELectricity COOPeration), a project co-funded by the EU, two novel hybrid CSP/biomass power plant prototypes were developed since September 2013 and implemented in Morocco and Tunisia.

One of the prototypes is representative of micro-cogeneration systems (6 kW_{el}) suitable for distributed generation. The solar field uses stationary CPC solar collectors to drive a micro CHP-ORC with an expander (Figure 2-24). As backup energy, the system relies on a biomass-fired boiler [78]. This prototype was installed in Morocco (Benguerir).
The other prototype is demonstrative of a centralised generation system on a reduced scale (60 kW_{el}) [79]. The solar field uses PTC technology in order to directly generate steam (DSG) [80]. Auxiliary energy is provided by a steam boiler using biogas as fuel. Biogas is produced by anaerobic digestion of canteen organic waste remains [81]. The system layout allows either hybrid or individual operation with each thermal source (solar-only or biogas-only). Thermal latent heat storage was also foreseen to compensate short-transients from the solar field [79]. This prototype is installed in Tunisia (Tunis), as shown in Figure 2-25.

In the last two years, additionally to the work presented within this PhD thesis, new CSP/biomass hybrid concepts and issues were addressed. The most novel concepts and results are described in the following paragraphs.

One innovative project is the HYSOL one [82-85]. The purpose is to deliver dispatchable and firm energy over the entire year using a fully renewable system. The
system relies on a combination of Brayton and Rankine cycles, yet differing from the conventional Combined Cycle Gas Turbine as it allows decoupling the operation of the two power systems. Both gas and steam turbine operation are assured by the solar field and biogas. The decoupling is assured by the use of TES. Moreover, TES is charged by both SF and gas turbine exhaust. The main improvements are the cost-competitiveness, flexibility and environmental sustainability.

In 2016, Zhang et al. studied a hybrid power plant with biogas from liquid cattle manure [86]. The authors concluded that for a constant biogas yield, a bigger digester is required with a higher hydraulic retention time. A further conclusion is the possibility to control the digester temperature (oscillations lower than 0.8°C) using waste steam from the power block.

Studies concerning the viability of CSP/biomass hybridisation for specific locations were also carried out. Giglio et al. [87] evaluated the use of a DSG PTC solar field with biomass combustion for Italy, and concluded that hybridisation permits the LCoE reduction. Another concept consists in using biomass gasification to drive a combined cycle in Brazil. The main novelty is the syngas storage to accommodate demand [88]. However, the concept is only presented, and an improved assessment is required.

The use of CSP/biomass hybrid power plants for CHP and Combined Cooling Heating and Power (CCHP) was also addressed [89-91]. A common conclusion is that a hybrid system improves the overall efficiency and is more cost-effective.

2.3 Conclusions

The potential of CSP/biomass hybrid systems is enormous. The main advantage of a CSP/biomass hybrid system is dispatchability. The better use of both energy resources (solar and biomass) results in an overall efficiency enhancement. From an economic point of view, there are advantages, such as the joint use of power plant equipment [92] reducing the investment, and a possible feed-in tariff associated with a fully renewable system, and CO₂ emission reduction.

The solar share and the peak demand are intrinsically related, leading to a reduction of the levelised cost of electricity. Additionally, this cost reduction decreases both financial investment and risk. The economic performance of biomass systems requires
proximity to the feedstock sources. Thus, CSP needs to be installed near power consumption centres. Moreover, whilst the solar resource continues to be a critical parameter, the economic viability requires lower radiation values. One practical example is the Borges Termosolar project that is placed near Barcelona, northern than other Spanish CSP projects.

Notwithstanding the above benefits, CSP/biomass hybridisation presents some challenges. This synergy requires both abundant solar radiation and biomass feedstock, limiting the implementation area. Despite biomass maturity, the plant sizes are usually smaller when compared with CSP and scalability is always an extra risk. Biomass needs to be acquired at reasonable costs to assure the economic competitiveness.

Furthermore, the optimal design and economic assessment complexity are hindered by the several hybridisation possibilities. To attain a better economic performance, the optimal synergy of the plant needs to be evaluated. Innovative design guidelines and novel control schemes are required, independently of the solution adopted. Furthermore, the solar field, TES and boiler should be dimensioned to enhance the economic balance of the plant.

2.4 References


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3 RESEARCH OBJECTIVES AND METHODOLOGY

In the previous Chapter, CSP, biomass and CSP/biomass current state-of-the-art was presented, including their main features and drawbacks. Despite the numerous advantages of CSP and biomass hybridisation, insufficient studies had been carried out, mostly deprived of experimental work. Reports [1-3] refer the benefits of CSP hybridisation, as well as consider the idea as a critical challenge for technology deployment. Issues such as the boiler control, the need to evaluate the optimal design and economic convenience of the systems, and the necessity of pilot prototypes in order to sustain large-scale projects are addressed. In this Chapter, the research objectives and methodology are presented.

3.1 Objectives

In order to contribute to CSP development and widespread, enhancing energy security, it is proposed in the scope of this PhD thesis to study different solutions for solar/biomass hybrid generation systems, within the concept of both systems supplying thermal energy to drive a power generation block.

The primary goal of this work is to provide a comprehensive scientific approach concerning these hybrid systems, addressing both technical and economic issues in order
to deliver a strong scientific basis for future research, as well as develop design tools for enhancing commercial deployment and technology dissemination.

It is intended to address design guidelines and recommendations for the optimal combination of CSP and biomass, to attain the best balance between system performance and economic feasibility. It is also proposed to evaluate the project parameters that influence the design. This evaluation includes the local resources (e.g. DNI, available biomass), solar and biomass shares, power plant size, operating temperatures, and also the use (or not use) of thermal energy storage.

To address specific system enhancements, the evaluation will be carried out beyond global performance. There are several technical challenges concerning the seamless operation of solar and biomass energy. For example, the interchange between energy sources, optimal start-up and cool-down periods. It is intended to provide information and recommendations concerning the best control schemes for these hybrid power plants.

Also, solutions for the present technical and economic challenges of CSP and biomass power standalone systems (e.g. energy and time consumption during start-up, use of reheat in the Rankine cycle) are addressed as well, throughout the concept of smart hybridisation.

3.2 Research Methodology

In order to achieve the objectives proposed in this work, it was decided to assess the potential of CSP/biomass hybridisation through cases studies. The evaluation of CSP and biomass hybridisation involves the combination of distinct research themes. Therefore, it was necessary to impose limits regarding the technologies evaluated in this study.

Regarding CSP technologies, the choice was centred in line-focus collectors, i.e. PTC. The decision was based on the maturity, increasing the range of practical application of this study and enhancing market deployment.

The commercial deployment of CSP plants requires considerable investments, usually attainable by finance consortiums. The maturity of the PTC technology enhances the feasibility of marketable deployment, and the further risk is solely related to the hybridisation concept. Furthermore, most of the CSP plants in operation use PTC with
thermal oil and can be easily enhanced with a biomass boiler. This increases the range of practical application of this study. Also, it is possible to extend this work to other collectors, such as LFR. The possibility of using or not using thermal energy storage systems is also considered.

Concerning the heat transfer media, both single-phase (i.e. thermal oil and molten salts), and two-phase fluids (water/steam) are evaluated. Thermal oil was used since the first CSP plant, and the main drawback is the temperature limitation. The advantages of direct steam generation in the solar field are the upsurge of the operating temperatures and the direct interface between the solar field and power block. On the other hand, control issues regarding the steam evaporation in the solar field and the need for phase change storage are the main drawbacks. The possibility to directly store solar energy is achievable through the use of molten salts, which also allow operation at higher temperatures than thermal oil. Whilst the commercial deployment of molten salt parabolic trough power plants is at an early stage [4], the potential is enormous, and possible hybridisation enhancements can permit further developments.

Hybridisation is achieved through the use of biomass boilers. Three distinct conversion technologies are considered as possible solutions for hybridisation: direct combustion, gasification and anaerobic digestion. This set of solutions, extend the hybridisation scenario to the most common biomass conversion technologies. Furthermore, with these technologies it is possible to operate with a broad sort of feedstocks, unbounding the scope of the study. The considered power cycle is a conventional Rankine cycle, either driven with steam or an organic fluid, depending on the operative temperature.

Four main tasks composed the predefined work plan (see Figure 3-1): numerical model development; experimental testing under real-life conditions; numerical model validation; and assessment of case studies.
3.2.1 Numerical model development

The first task consisted in the development of a quasi-transient numerical model using a commercial software EBSILON® Professional, to evaluate the hybrid system thermodynamic performance. The developed model encompasses the combination of the solar field, biomass, thermal energy storage and power block systems, including all subcomponents.

In order to accurately describe the system dynamic behaviour, and thus evaluate the performance under distinct operation stages, simulations were carried out in a transient
basis. In real operation, the power plant is controlled depending on numerous external and internal variables. To describe this behaviour at different operation stages, a Pascal-based code was developed.

The result is a versatile computer tool, comprehensively described in Chapter 4, that simulates the dynamic hybrid power plant performance, throughout the thermodynamic balance of individual components for different meteorological conditions and operation schemes.

3.2.2 Experimental testing under real-life conditions

The experimental work encompasses two tasks that consist in the evaluation under real-life conditions of a concentrated solar-field and a biomass gasification system. The hybrid CSP/biomass assessment was carried out through the developed numerical model. Thus, the accuracy of the results rely on the model, which includes simplifications and empirical correlations. Additionally, the model complexity was increased by the inclusion of a two-phase fluid flow in the absorber pipes.

In order to ensure the model consistency, and thus results, a test campaign was carried out at the DISS (DIrect Solar Steam) loop in Almeria to collect operation data under real-life conditions. Experimental performance testing under a real-life context was carried out during an eight-day period, under the SFERA2 (Solar Facilities for the European Research Area) program, an EU-funded research project. The operation data were monitored. The data acquired on site included environmental variables (e.g. solar direct normal irradiance, ambient temperature), as well as system variables (e.g. mass flow, pressure, temperature).

Both recirculation and once-through (OT) operation modes were tested. For each operation mode, system variables were measured throughout operation conditions and distinct operation stages including start-up, shut-down, and steady and transient solar radiation conditions.

The second task consists in a biomass gasification system performance assessment, carried out in the Faculdade de Engenharia da Universidade do Porto (FEUP), by the use of a small gasifier system. The system is constituted by a downdraft gasifier, where woody biomass is converted into producer gas. The set of experimental tests consisted in
the evaluation of the system performance at steady conditions for different loads (i.e. flow rates), and for different operation stages, including start-up and shut-down. The system was enhanced with a set of distinct and calibrated instruments (e.g. thermocouples, pressure sensors, flow meters) at relevant locations, and operation data were monitored and recorded, throughout the use of a data logger and a LabVIEW developed code. A thorough description of the experimental work carried out, as well as results, are addressed in Chapter 5.

3.2.3 Numerical model validation

Within this task, the operation results from experimental testing with the solar-field were used to validate and tune-up the developed numerical model. Additional tests were carried out to evaluate the numerical model performance for other Heat Transfer Fluid (HTF). The results are discussed in Chapter 6.

3.2.4 System technical and economic assessment

The validated numerical model was used to study different solutions – case studies – for CSP/biomass hybridisation in Chapter 7. The assessment includes different regions, with different solar radiation and biomass resources, and distinct CSP and biomass technologies, working fluids, solar and biomass shares, power plant sizes, as well as the use of storage. The economics of hybridisation was also assessed.

3.3 References


4. Wu, C., China’s first molten salt trough CSP demonstration loop in Gansu Aksay put into operation at 11:18 am, 12th, October, in CSP Plaza. 2017
The assessment of hybrid concentrating solar power and biomass plant combinations was carried out through a numerical model. Currently (2017) there are standardised methodologies for CSP [1] yield assessment. Nevertheless, these guidelines do not encompass hybrid systems, and are furthermore solely intended for annual yield.

There are many available software and codes for CSP simulation [2, 3], with different degrees of complexity and accuracy. In the scope of this work, the commercial software EBSILON® Professional, version 12.02.02 was used. Ebsilon is a simulation software designed for power plant and thermodynamic processes analysis. The software includes a wide range of component libraries, for which steady-state continuity, momentum and energy balances are simulated, under design and off-design conditions. In addition to the general Ebsilon libraries, the EbsSolar library was used, consisting of a component library (e.g. line focusing collectors, sun) specifically designed for solar thermal power plant simulations.

The modelling methodology consists on splitting the CSP/biomass hybrid power plant into subsystems (see Figure 4-1): solar field (SF); thermal energy storage (TES); biomass system (BS); and power block (PB). Modelling also requires a further breakdown of the subsystems into individual components (e.g. solar collectors in the SF). Afterwards, simulations are carried out considering the entire system. Nevertheless, the applied methodology permits to achieve individual assessment and control of the individual subsystems and components, through an implemented operation strategy.
At the subsystem and component level, modelling is carried out by evaluating thermodynamic properties (e.g. enthalpy) and solving continuity, momentum and energy balances. Thus, component performance can be evaluated at a detailed level. Nonetheless, detailed modelling is not extended to all components, neither to all operation modes (e.g. off-design). In the case of biomass performance, empirical correlations from experimental tests (Chapter 5) and literature data are used.

Notwithstanding that Ebsilon was designed for steady-state calculations, quasi-dynamic simulations were carried out, by the combination of dynamic components and a time-series, where calculations are carried out for each timestep. For the solar collectors and headers, a fully dynamic approach was considered.

Moreover, a Pascal-based code was developed using EbsScript, to perform the required simulations and individual components calculations (e.g. convective heat transfer coefficient) at each timestep, and also to control the hybrid system operation.

In the following sections, the methodology for modelling subsystems’ components and the related assumptions are presented and analysed. Later, a general overview of the operation strategy is described, and a simplified economic model defined.
4.1 Meteorological data

The performance of all hybrid power plant subsystems depends on meteorological data. Whilst TES, BS and PB simulations only require the knowledge of ambient temperature and humidity, the solar field assessment also requires knowledge of the normal incident irradiance at the collectors.

In this study, meteorological data for distinct locations and with different timesteps were used. Both specific hybridisation enhancements and annual performance were assessed using either one representative Typical Meteorological Year (TMY) or locally measured data. The TMY data was obtained from Meteonorm software [4].

Within the Ebsilon environment, the Sun component is used as an interface between meteorological data and other subsystems. Required inputs are: incident beam irradiance ($I_b$); ambient temperature ($T_{amb}$); wind speed ($vel_{wind}$) and wind direction ($wind_{direction}$); time reference system, i.e. local time ($LT$) or solar time ($ST$); Time Zone ($TZ$); date and time; and geographic coordinates, i.e. latitude ($\phi$) and longitude ($\lambda$).

The Sun relative position changes over the year, and thus depends on the date, time and location. There are numerous algorithms to calculate the sun position, with different uncertainties in solar zenith and azimuth angle calculations [5]. In this study, sun position and solar collector incident angles are calculated according to DIN 5034 [6]. Sun position at any place on earth, and at any time, can be described by the elevation ($\gamma_S$) and azimuth angles ($\alpha_S$) (see Figure 4-2).

![Figure 4-2 - Elevation ($\gamma_S$) and azimuth angles ($\alpha_S$) for sun position according to DIN5034.](image)

Adapted from [7].
Whilst $\gamma_S$ is contingent to the latitude ($\varphi$) and hour angle ($\omega$) as shown in equation 4-1 [7], $\alpha_S$ depends on the $\gamma_S$, latitude ($\varphi$), solar declination angle ($\delta$) and solar time ($ST$), i.e. equation 4-2 [7].

\[
\gamma_S = \arcsin(\cos \omega \cos \varphi \cos \delta + \sin \varphi \sin \delta)
\] (4-1)

\[
\alpha_S = \begin{cases} 
180^\circ - \arccos \frac{\sin \gamma_S \sin \varphi - \sin \delta}{\cos \gamma_S \cos \varphi} & \text{for } ST \leq 12:00 \\
180^\circ + \arccos \frac{\sin \gamma_S \sin \varphi - \sin \delta}{\cos \gamma_S \cos \varphi} & \text{for } ST > 12:00
\end{cases}
\] (4-2)

If $LT$ is used as time reference system, $ST$ can be calculated by equation 4-3 [7], where $MLT$ and $EoT$ stand for Mean Local Time and Equation of Time, respectively. $MLT$ is calculated through equation 4-4 [7], using $LT$, $\lambda$ and $TZ$; e.g. for Greenwich Mean Time (GMT) $TZ$ is 0.

\[
ST = MLT + EOT
\] (4-3)

\[
MLT = LT - \frac{15^\circ TZ - \lambda}{15^\circ}
\] (4-4)

The $EoT$ (4-5) [7] depends on the day angle ($J'$), which can be evaluated through equation 4-6 [7], where $doy$ stand for the day of the year and $n_{doy}$ for the number of days of the year (i.e. 365 for a regular year and 366 for a leap year).

\[
EOT = 0.0066 + 7.3525 \cos(J' + 85.9^\circ) + 9.9359 \cos(2J' + 108.9^\circ) + 0.3387 \cos(3J' + 105.2^\circ)
\] (4-5)

\[
J' = 360^\circ \frac{doy}{n_{doy}}
\] (4-6)

The sun position over the year significantly depends on the declination angle ($\delta$), i.e. the angular position of the sun at solar noon concerning the plan of the equator, which changes over the year from -23.45º to 23.45º [8]. The solar declination angle is calculated from $J'$ and the hour angle ($\omega$) [7].
\[ \delta = 0.3948 - 23.2559 \cos(J' + 9.1^\circ) - 0.3915 \cos(2J' + 5.4^\circ) - 0.1764 \cos(3J' + 26^\circ) \]  
\[ \omega = (12:00 - ST)15^\circ \]

### 4.2 Solar field

The solar field subsystem comprises the solar collectors, piping, and distributing and collecting headers. Also, for direct steam generation simulation, under the recirculation operation concept, the solar field entails a steam drum.

#### 4.2.1 Solar collectors

For modelling the solar collectors, the line focusing solar collector component is used, where an energy balance between \( I_b \) and the heat transfer fluid is calculated. The radiation incidence angle \( (\theta_i) \) (see Figure 4-3) is calculated from the sun position (i.e. \( \alpha_S \) and \( \gamma_S \)); collector orientation \( (\gamma_{coll}) \), i.e. collector azimuth angle (0º at north and positive to eastern direction); and collector tilt angle \( (\alpha_{tilt}) \), i.e. inclination between the collector axis and horizontal plane. For Parabolic Trough Collectors (PTC) [9],

\[ \cos \theta_i = \sqrt{(1 - \cos(\alpha_S - \alpha_{tilt}) - \cos(\alpha_{tilt}) \cos(\alpha_S)(1 - \cos(\gamma_S - \gamma_{coll}))^2} \]  
\[ (4-9) \]

For linear Fresnel reflector collectors, an additional transversal incidence angle is calculated [9],

\[ \tan \theta_{i, tr} = \frac{\cos(\alpha_S) \sin(\gamma_S - \gamma_{coll})}{\sin(\alpha_S - \alpha_{tilt}) + \sin(\alpha_{tilt}) \cos(\alpha_S)(1 - \cos(\gamma_S - \gamma_{coll}))} \]  
\[ (4-10) \]
The available energy within each collector ($Q_{\text{solar}}$) is defined as a function of $I_b$ at the net collector aperture area ($A_{\text{net, coll}}$), considering the nominal collector optical efficiency ($\eta_{\text{opt,0}}$), incident angle modifier ($\eta_{\text{IAM}}$), collector focus state ($f$), shading ($\eta_{\text{shad}}$) and end losses ($\eta_{\text{end}}$), and mirror cleanliness ($\eta_{\text{clean}}$) [1],

$$Q_{\text{solar}} = I_b A_{\text{net, coll}} \eta_{\text{opt,0}} \eta_{\text{IAM}} f \eta_{\text{shad}} \eta_{\text{end}} \eta_{\text{clean}} \quad (4-11)$$

The $\eta_{\text{opt,0}}$ describes the collector optical efficiency under the assumptions of no heat losses (i.e. same ambient and HTF temperature), incidence angle of 0º, clean collector mirror and without shading and deformations. Due to specific optics, in the case of a LFR, peak optical efficiency can be attained at non-perpendicular irradiance.

The peak optical efficiency is measured for a specific aperture area, which is calculated through the product of nominal collector width ($w_{\text{coll}}$) and length ($l_{\text{coll}}$) and the area ratio ($A_{\text{ratio}}$), i.e. the ratio between active and gross reflective collector area [10],

$$A_{\text{net}} = l_{\text{coll}} \times w_{\text{coll}} \times A_{\text{ratio}} \quad (4-12)$$

It is noteworthy that if $\eta_{\text{opt,0}}$ is related to the gross area, $A_{\text{ratio}}$ is set to 1. Also, for LFR collectors, $w_{\text{coll}}$ stands for the system width, and consequently the $A_{\text{ratio}}$ is used to define the net aperture area of the LFR collector.
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The $\eta_{IAM}$ accounts for the optical losses and solar radiation spillage for distinct incident angles. For a LFR, $\eta_{IAM}$ is calculated from the longitudinal ($\eta_{IAM,l}$) and transversal ($\eta_{IAM,t}$) incidence angle modifier (equation 4-13). In contrast, for a PTC equation 4-13 [11] is simplified as only the longitudinal term is required.

$$\eta_{IAM,LFR} = \eta_{IAM,l}(\theta_i) \times \eta_{IAM,tr}(\theta_i)$$  \hspace{1cm} (4-13)

The $\eta_{IAM,l}$ or $\eta_{IAM,t}$ can be found in the literature with different formats, and thus in this study two approaches were used to calculate $\eta_{IAM}$ depending on the reference data. One consists on the use of table-based values, i.e. incident angle modifier as a function of incidence angle tabulated values. In that case, calculations are carried out using a second order interpolation.

The other method consists on using a polynomial equation to describe $\eta_{IAM}$ as a function of $\theta_i$. For a PTC, equation 4-14 [10] is used, whilst for a LFR both equations 4-15 and 4-16 [10] are used for $\eta_{IAM,l}$ and $\eta_{IAM,t}$ correlations, respectively.

$$\eta_{IAM,l,PTC} = \left(1 - \eta_{IAM,l,A} + \eta_{IAM,l,A} \cos(\theta_i)\right)\left(\eta_{IAM,l,cos} \cos(\theta_i)\right) + \sum_{z=0}^{5} \eta_{IAM,l,z} \theta_i^z$$ \hspace{1cm} (4-14)

$$\eta_{IAM,l,LFR} = \sum_{z=0}^{5} \eta_{IAM,l,z} \theta_i^z$$ \hspace{1cm} (4-15)

$$\eta_{IAM,t,LFR} = \sum_{z=0}^{5} \eta_{IAM,t,z} \theta_i^n$$ \hspace{1cm} (4-16)

When the sun height is low (e.g. sun appearing/disappearing on the horizon), shading between collector rows is likely to occur. This effect is evaluated through the collector row to row distance ($w_{p,row}$) and $w_{coll}$, as well as the tracking angle ($\rho_{track}$) [14],

$$\eta_{shad} = 1 - \min\left[1, \max\left(0, 1 - w_{p,row} \times \frac{\cos(\rho_{track})}{w_{coll}}\right)\right]$$ \hspace{1cm} (4-17)

where the tracking angle is calculated by,

$$\tan(\rho_{track}) = \frac{\cos(\alpha_S) \sin(\gamma_S - \gamma_{coll})}{\sin(\alpha_S - \alpha_{tilt}) + \sin(\alpha_{tilt}) \cos(\alpha_S) \times (1 - \cos(\gamma_S - \gamma_{coll}))}$$ \hspace{1cm} (4-18)
It is noticeable that the track angle is identical to the transversal angle of a LFR (see Figure 4-3) and used to distinguish between the incidence angle and the actual collector track state.

Occasionally, reflected incident radiation does not hit the collector receiver tube (end losses). Also, the reflected radiation can hit the receiver tube of the next collector (end gains). These losses/gains depend on incidence angle, solar collector length and the average distance between the reflector and receiver (lf), and also on the solar field layout, specifically on the collector row length (ls,row) [1, 10],

\[ \eta_{end} = 1 - \min \left( 1, \frac{lf}{l_{coll}} \tan(\theta l) \right) + \max \left( 0, \min \left( 1, \frac{lf}{l_{coll}} \tan(\theta l) \right) - \frac{ls,row}{l_{coll}} \right) \] (4-19)

For a LFR, lf is equal to the focal length (f_length), i.e. distance between the mirror surface and the receiver. On the other hand, for a PTC is calculated from [1],

\[ l_f = f_{\text{length}} \left( 1 + \frac{w_{coll}^2}{48 f_{\text{length}}} \right) \] (4-20)

One consequence of outdoor installation is soiling. Losses as a result of the collector non-ideal cleanliness are assessed by considering an effective annual average mirror cleanliness efficiency (\(\eta_{\text{clean}}\)). The collector focus state (f), permits to change individual collector focus status based on operation strategy. Depending on the solar multiple and available radiation, a surplus of energy may occur. To avoid this issue, solar collectors are turned out from the ideal tracking angle, thus reducing the thermal energy output.

The effective heat received by the HTF (\(\dot{Q}_{\text{eff,HTF}}\)) is defined through the energy balance between the available solar energy (\(\dot{Q}_{\text{solar}}\)) and thermal losses (\(\dot{Q}_{\text{loss,coll}}\)),

\[ \dot{Q}_{\text{eff,HTF}} = \dot{Q}_{\text{solar}} - \dot{Q}_{\text{loss,coll}} \] (4-21)

The heat losses at the collectors’ receivers (equation 4-22) are related to the temperature difference between the heat transfer fluid and the ambient air (a_z terms), radiant heat flux within the absorber tubes (b_z terms), and fluid temperature (c_z and d_z terms [10]),
\[ \dot{Q}_{loss,\text{coll}} = l_\text{coll} \left( \sum_{z=0}^{Z=4} a_z (T_{HFT} - T_{amb})^z + \sum_{z=0}^{Z=4} b_z l_b \frac{\eta_{opt}}{\eta_{opt,0}} (T_{HFT} - T_{amb})^z + \sum_{z=1}^{Z=0} c_z T_{HFT} + \sum_{z=0}^{Z=0} d_z l_b \frac{\eta_{opt}}{\eta_{opt,0}} T_{HFT} \right) \] (4-22)

where the optical efficiency \( (\eta_{opt}) \) is assessed with [10],

\[ \eta_{opt} = \eta_{IAM} \times f \times \eta_{shad} \times \eta_{end} \times \eta_{clean} \] (4-23)

To improve the model heat loss accuracy, the calculation is carried out for distinct sections. The collector absorber is split into a number of sections along its length and temperature continuity assumed in the boundary surfaces. The heat flux in each segment is assumed uniform and normal to the surfaces; the section temperature is calculated through the arithmetic mean between the temperatures at the control volume boundaries (see Figure 4-4).

4.2.2 Pressure drop model

The HTF flow irreversibility over the receivers, pipes, and headers results in a pressure drop. These losses influence the selection of pumps, as well as pipe sizes. Furthermore, the accuracy of pressure loss calculations becomes an issue when a two-
phase HTF is used. Under the DSG concept water evaporation occurs within the collectors, and thus the pressure drop over the loop influences both the beginning and ending of the evaporation zones.

The two-phase flow is characterised by the discontinuity between the liquid and gas flow, and thus, thermodynamic proprieties may change abruptly. Consequently, modelling is a challenging task, and different approaches can be used: two-fluid, drift, homogeneous and heterogeneous models. The two-fluid and drift models encompass a complex flow model, where continuity, momentum and energy equations are solved for both phases. Consequently, they are confined to complex software developed for nuclear industry, such as RELAP, RETRAN, ATHLET and CATHARE [12, 13].

Hence, in this study, pressure losses ($\Delta P_{\text{coll}}$) were assessed considering both single and two-phase flows. The model entails the calculation of pressure losses considering a smooth pipe ($\Delta P_{\text{coll,S}}$) and rough pipe ($\Delta P_{\text{coll,R}}$). Afterwards, the worst case is considered [10],

$$\Delta P_{\text{coll}} = \max(\Delta P_{\text{coll,S}}, \Delta P_{\text{coll,R}}) \quad (4-24)$$

The smooth pipe calculations rely on the Friedel two-phase flow pressure drop [12]. The model consists in the calculation of the pressure drop ($\Delta P_{\text{coll,S}}$) within the absorber tube as if it was a single-phase flow, for both liquid ($\Delta P_{\text{coll,L}}$) and vapour ($\Delta P_{\text{coll,G}}$) [12]. Subsequently, a two-phase multiplier factor ($\phi_L$) is used, calculated from the Weber ($W_{EL}$) and Froude ($F_{FL}$) numbers for the liquid-phase flow, as well as the steam quality ($x$) [12].

In the following equations: $\dot{m}$ stands for the mass flow rate, $\rho_{L,G}$ for the fluid density as if it was saturated liquid or vapor, respectively, $D_{\text{int}}$ for the absorber tube internal diameter, $\mu_{L,G}$ for the fluid dynamic viscosity as if it was saturated liquid or vapor, respectively, and $\sigma$ for the surface tension.

$$\Delta P_{\text{coll,S}} = \Delta P_{\text{coll,L}} \phi_L^2 \quad (4-25)$$

$$\Delta P_{\text{coll,L}} = \zeta_L \times \frac{8\dot{m}^2}{\rho_L \pi^2 D_{\text{int}} g} \quad (4-26)$$
\[ \phi_L^2 = (1 - x)^2 + x^2 \left( \frac{\rho_L \zeta_G}{\rho_G \zeta_L} \right) + 3.43x^{0.69}(1 - x)^{0.24} \left( \frac{\rho_L}{\rho_G} \right)^{0.8} \left( \frac{\mu_G}{\mu_L} \right)^{0.22} \quad (4-27) \]

\[ \frac{\mu_G}{\mu_L}^{0.89} Fr_L^{-0.047} We_L^{-0.033} \]

\[ We_L = \frac{16 \dot{m}^2}{\sigma \rho_L \pi^2 D_{int}^3} \quad (4-28) \]

\[ Fr_L = \frac{16 \dot{m}^2}{g \rho_L \pi^2 D_{int}^5} \quad (4-29) \]

The single-phase pressure drop coefficients \((\zeta_{L,G})\) \(^{12}\) are dependent and a function of the Reynolds number \((Re_{L,G})\) \(^{14}\),

\[ Re_{L,G} = \frac{4 \dot{m}}{\pi D_{int} \mu_{L,G}} \quad (4-30) \]

\[ \zeta_{L,G} = \begin{cases} \frac{64}{Re_{L,G}} & \text{for } Re_{L,G} \leq 1055 \\ 0.86859 \ln \left( \frac{Re_{L,G}}{1.964 \ln(Re_{L,G}) - 3.825} \right)^{-2} & \text{for } Re_{L,G} > 1055 \end{cases} \quad (4-31) \]

On the other hand, the aforementioned model discards the pipe roughness. Thus, a correlation between single-phase and two-phase flow pressure drop in rough pipes is used. Initially, the pressure drop is calculated considering single-phase flow, for liquid \((\Delta P_{coll,R,L})\) and gas \((\Delta P_{coll,R,G})\) \(^{14}\),

\[ \Delta P_{coll,R,L,G} = \zeta_{R,L,G} \times \frac{8 \dot{m}^2}{\rho_L \pi^2 D_{int}^5} \quad (4-32) \]

The pressure drop coefficient \((\zeta_{L,G})\) is a function of the internal pipe diameter \((D_{int})\), pipe roughness \((\varepsilon)\), as well as the Reynolds number \((Re)\). In order to calculate \(\zeta\) directly, the Swamee–Jain equation \(^{15}\) is used,

\[ \zeta_{R,L,G} = 0.25 \ln \left( \frac{\varepsilon}{3.7 D_{int}} + \frac{5.74}{Re_{L,G}} \right)^{-2} \quad (4-33) \]
The same methodology, of splitting the absorber tube into sections, used for heat loss calculation is applied to the pressure drop. Therefore, thermodynamic properties (e.g. enthalpy, pressure, steam quality) are evaluated at each section centre (see Figure 4-4).

4.2.3 Dynamic Behaviour

The solar energy uncertain nature results in significant transient effects within the solar field. Ebsilon was designed for steady-state simulations. However, a quasi-dynamic model to describe the power plant behaviour was developed in this work, by using a time-series function where calculations are carried out for each timestep.

The solar field entails a significant amount of HTF and steel which result in thermal inertia. Therefore, the detailed dynamic behaviour of the solar collectors and headers was considered, using an indirect storage (IS) component. A code developed within EbsScript is used to calculate the required variables (e.g. convective heat transfer coefficient) at each timestep.

The IS enables transient heat exchange calculation between the working fluid (i.e. water/steam) and the collector absorber tube. The Fourier heat transfer differential equation is discretised in a two-dimensional spatial domain using a finite volume method (see Figure 4-5) and in time by an iterative Crank-Nicholson method [16] (equation 4-34), considering the pipe density ($\rho_{\text{pipe}}$), specific heat ($C_{p_{\text{pipe}}}$), and conductivity ($\lambda_{\text{pipe}}$),

$$\rho_{\text{pipe}}C_{p_{\text{pipe}}} \frac{\partial T_{\text{pipe}}}{\partial t} = \lambda_{\text{pipe}} \left( \frac{\partial^2 T_{\text{pipe}}}{\partial x^2} + \frac{\partial^2 T_{\text{pipe}}}{\partial y^2} \right) \quad (4-34)$$
The convective heat flux ($\dot{Q}_{HTF,pipe}$) between the heat transfer fluid and the pipe walls is calculated using Newton’s cooling law [17], for each control volume surface area ($A_{surf,CV}$),

$$\dot{Q}_{HTF,pipe} = \alpha_{int} A_{surf,CV} (T_{HTF} - T_{pipe,ext})$$  \hspace{1cm} (4-35)

The dynamic behaviour of the HTF over the absorber tube, is modelled through the energy and mass balance at each element volume, i.e. the energy ($h_{in}$ and $h_{out}$) transported by the mass flow rate ($\dot{m}$), $\dot{Q}_{HTF,pipe}$, fluid mass ($m_{HTF}$) and internal energy change within the pipe ($du_{HTF}$) [10],

$$\dot{m}_{HTF}(h_{in} - h_{out}) - \dot{Q}_{HTF,pipe} = m_{HTF} \frac{du_{HTF}}{dt}$$  \hspace{1cm} (4-36)

A higher number of elements in the mesh significantly improves simulation accuracy, however at the expense of an increased computational effort. Therefore, mesh size was defined based on a sensitivity analysis and considering that the temperature gradient is much higher in the axial direction. The timestep ($\Delta t$) is defined considering the Courant–Friedrichs–Lewy stability condition [18], for each element size ($\Delta x$),
\[
\frac{4m}{\pi \rho_{HTF} D_{int}^2 \Delta x} \Delta t < 1 \quad (4-37)
\]

The IS component requires the convective heat transfer coefficient \( \alpha_{int} \) as input, which changes with time. As the convective heat transfer coefficient (i.e. Nusselt number) is calculated through empirical correlations, a developed internal algorithm is used to calculate \( \alpha_{int} \) at each iteration. For single phase flow \( \alpha_{int} \) was estimated by the Dittus-Boelter equation [17], using the Reynolds (\( Re \)) and Prandtl (\( Pr \)) numbers, and the fluid thermal conductivity (\( \lambda_{HTF} \)),

\[
\alpha_{int} = \frac{\lambda_{HTF}}{D_{int}} 0.023Re^{0.8}Pr^{0.4} \quad (4-38)
\]

Whilst modelling and simulating CSP with single phase fluid (e.g. thermal oil) is relatively simple, two-phase heat transfer and fluid flow in long horizontal pipes increases modelling challenges and complexity.

In the case of two-phase flow, heat transfer coefficient assessment is a challenge. Flow boiling inside horizontal tubular channels results in different flow patterns, characterised by asymmetric distributions of liquid and vapour phases (see Figure 4-6). Gravity and different densities between the two-phases influence the flow pattern, and can jeopardise absorber lifetime when the liquid phase sets at the lower part of the tube and thus the upper part is not adequately wetted.

![Figure 4-6](image)

**Figure 4-6 – Two-phase flow patterns in horizontal tubes. Reprinted from [19].**
Many modelling methods exist for DSG simulation, with distinct degrees of complexity and accuracy [13], and many correlations. The problem complexity is enhanced by combining heat transfer and fluid flow. Despite the topic unquestionable interest, it is beyond the scope of this thesis. For more details, the reader is addressed to [19, 20].

In this study, a simplified model was implemented based on Stephan [20]. First, it is necessary to check if the stratified flow is present, through evaluation of the Froude number (equation 4-29). If the Froude number is lower than 0.04, then the flow is considered stratified, and the convective heat transfer coefficient \( \alpha_{2,\text{phase}} \) evaluated by Shah’s correlation [21],

\[
\frac{\alpha_{2,\text{phase}}}{\alpha_L} = 3.9 F_r^{0.24} \left( \frac{x}{1-x} \right)^{0.64} \left( \frac{\rho_L}{\rho_G} \right)^{0.4} \tag{4-39}
\]

where for the liquid heat transfer coefficient \( \alpha_L \), McAdams equation [22] is used,

\[
\alpha_L = 0.023 \left( \frac{A_L}{D_{\text{int}}} \right) \left( \frac{\dot{m}(1-x)D_{\text{int}}}{\mu_L} \right)^{0.8} Pr^{0.4} \tag{4-40}
\]

Otherwise, annular flow is assumed (i.e. the tube is entirely wetted) and the Chen correlation [23] for vertical tubes is used (equation 4-41). The correlation consists in adding contributes from both nucleate boiling \( \alpha_{nb} \) (i.e. bubble formation) and convective terms \( \alpha_{cb} \),

\[
\alpha_{2,\text{phase}} = \alpha_{nb} + \alpha_{cb} \tag{4-41}
\]

Chen proposed the calculation of the nucleate boiling term, from the free convection \( \alpha_{nb,fc} \) nucleate boiling. Forced convection in comparison to free convection results in a steeper temperature gradient at the boundary layer, and hence more heat is removed from the pipe wall, and the bubble formation is partially suppressed. Accordingly, a suppression factor \( S \leq 1 \) is used [23] (equation 4-42).

\[
\alpha_{nb} = S \alpha_{nb,fc} \tag{4-42}
\]
Concerning the convective term, it is related to the heat transfer for single-phase liquid. Chen noted that heat transfer is augmented, from both increased liquid velocity because of vapour formation, and also from the presence of vapour bubbles. This heat transfer improvement is accounted through an enhancement factor, $F (\geq 1)$ [23],

$$\alpha_{cb} = F\alpha_L$$

(4-43)

The single-phase heat transfer coefficient is calculated through the Dittus-Boelter correlation (equation 4-38), and nucleate boiling free convection heat transfer coefficient from the Forster-Zuber correlation [24],

$$\alpha_{nb,fc} = 0.00122 \left( \frac{\lambda_{L}^{0.79} c_{pL}^{0.45} \rho_{L}^{0.49}}{\sigma^{0.5} \mu_{L}^{0.29} \Delta h_{LG}^{0.24} \rho_{G}^{0.22}} \right) \Delta T_{sat}^{0.24} \Delta P_{sat}^{0.24}$$

(4-44)

where $\lambda_L$ is the liquid conductivity, $\Delta h_{LG}$ is the heat of vaporisation, $\Delta T_{sat}$ is the difference between the fluid ($T_{sat,HTF}$) and pipe wall ($T_{pipe,ext}$) saturation temperatures, and $\Delta P_{sat}$ is the difference between the saturation pressures ($P_{sat}$) at the fluid and pipe wall temperatures [24],

$$\Delta T_{sat} = T_{sat,HTF} - T_{pipe,ext}$$

(4-45)

$$\Delta P_{sat} = P_{sat}(T_{HTF}) - P_{sat}(T_{pipe,ext})$$

(4-46)

The two-phase enhancement factor ($F$) of Chen is a function of the Martinelli parameter ($X_{tt}$) [23],

$$F = \left( \frac{1}{X_{tt}} + 0.213 \right)^{0.736}$$

(4-47)

which is assessed by the Lockhart-Martinelli correlation [12],

$$X_{tt} = \left( \frac{1-x}{x} \right)^{0.875} \left( \frac{\rho_{L}}{\rho_{G}} \right)^{0.5} \left( \frac{\mu_{L}}{\mu_{G}} \right)^{0.125}$$

(4-48)
Later, the suppression term \( S \) is calculated as a function of the two-phase Reynolds number \( \text{Re}_{2,\text{phase}} \) [23],

\[
S = \frac{1}{1 + 2.53 \times 10^{-6} \text{Re}_{2,\text{phase}}^{1.17}}
\]  

(4-49)

Where the \( \text{Re}_{2,\text{phase}} \) number is as an enhancement of the liquid Reynolds number by using the two-phase enhancement factor \( F \) [23],

\[
\text{Re}_{2,\text{phase}} = \text{Re}_L F^{1.25}
\]  

(4-50)

It is noteworthy that the two-phase model equations include unknown variables (e.g. \( T_{\text{pipe,ext}} \)), and thus an iterative calculation is required.

As aforementioned, both collector and IS components are split into sections, where pressure and heat transfer calculations are carried out. Nevertheless, the typical linear focusing collectors’ length exceeds 100 meters, which jeopardises the calculations, especially for the two-phase flow heat transfer and pressure drop. In fact, as the collectors’ model involves first steady-state and subsequently transient calculations, thermodynamic properties will significantly change between the collector inlet and outlet (steady), which will hinder transient calculation accuracy. In order to enhance the numerical model performance, each collector is further divided into sections, which include a solar collector (SC) and an indirect storage component (see Figure 4-7).

![Figure 4-7 – One collector model with ten sections used for DISS loop assessment.](image)

Whilst increasing the number of sections enhances model performance, it will hinder both modelling and calculation effort. Therefore, the number of sections was defined based on a compromise between accuracy of simulation results and computation/modelling effort. A sensitivity analysis for the number of sections is carried out in Chapter 6.
4.2.4 Piping

The connection of solar collectors in series requires pipes and fittings and therefore flow irreversibilities will lead to pressure and heat losses. A piping component was used to model these losses.

Pressure drop is modelled using a similar methodology as for solar collectors (see section 4.2.2). The only difference is related to an additional geodetic height difference. For example, the collectors interconnections in the DISS test facility [25] (see Figure 4-8) differ in height. Consequently, the pressure model encompasses an additional component to assess pressure difference ($\Delta P_{\text{pipe,geod}}$) due the geodetic height difference $\Delta z$ [14],

$$\Delta P_{\text{pipe,geod}} = \rho_{\text{HTF}} g \Delta z$$  \hspace{1cm} (4-51)

![Figure 4-8 - Interconnections between solar collectors along of the DISS test facility. Reprinted from [26].](image)

Heat losses ($\dot{Q}_{\text{loss,pipe}}$) between the HTF and ambient are assessed by internal and external convection and conduction at insulation, considering a constant heat flux (i.e. using the HTF average temperature) [17],

$$\dot{Q}_{\text{loss,pipe}} = \frac{1}{a_{\text{int}} \pi D_{\text{int}}^2} \frac{1}{2 n L_{\text{pipe}} k_{\text{ins}} \ln \left( \frac{D_{\text{ext},\text{ins}}}{D_{\text{int},\text{ins}}} \right)} \frac{1}{a_{\text{ext}} \pi D_{\text{ext},\text{ins}}^2} \frac{1}{T_{\text{HTF}} - T_{\text{amb}}}$$  \hspace{1cm} (4-52)

It is noticeable that equation 4-52 neglects pipe conduction, i.e. pipe internal and external temperatures are assumed to be the same. This simplification is based on the high thermal conductivity and small thickness of the steel pipe.
For the internal convective heat transfer coefficient ($\alpha_{int}$) the same methodology as the one used for solar collectors (see section 4.2.3) was applied. On the other hand, for the external convective coefficient assessment, crosswind over the pipe was assumed and the Churchill-Bernstein correlation [17] used,

$$\alpha_{ext} = \frac{\lambda_{air}}{D_{int}} \left( 0.3 + \frac{0.62Re_D^{1/2}Pr^{1/3}}{1 + (0.4/Pr)^{2/3}} \left( 1 + \left( \frac{Re_D}{282000} \right)^{5/8} \right)^{4/5} \right)$$  \hspace{1cm} (4-53)

Collectors’ interconnecting pipes have a rather small length when compared with the solar field. Thus, to reduce computational effort, the dynamic behaviour of the pipes was neglected.

### 4.2.5 Distributing and collecting headers

The solar field is constituted by many collectors’ loops. In order to reduce computational effort, a one-loop simplification was used. This methodology consists of a detailed simulation of a representative loop and considering identical behaviour for similar loops. This approach is possible by the use of distributing and collecting headers components. These components enable consideration of the fluid stream distribution and collection, without neglecting mass flow, heat and momentum balances. Whilst mass flow rate within the loops is considered identical, enthalpy and pressure are different due to heat and pressure losses along the header (see Figure 4-9).

Pressure and heat losses at the header were assessed by an identical code as the one used for piping, with single and two-phase flow consideration. Nevertheless, the model requires the definition of the header diameter in each section, either considering it constant or through a constant velocity approach, as shown in Figure 4-9.
It is also necessary to identify the location of the representative loop and the path where the highest-pressure drop occurs, which depend on the solar field configuration, i.e. on the flow arrangement. Calculations are carried out for each section, with a fluid temperature ($T_{HTF}$) determined at the section centre. The headers entail a considerable mass of steel and HTF, and so an indirect storage component was used for each header.

### 4.2.6 Steam drum

In the case of DSG and with the recirculation concept, the solar field encompasses a steam drum where water and steam streams are separated. Steam drum modelling consists first in the separation of the water and steam components. Afterwards, two storage tank components are used for water and steam (see Figure 4-10). Modelling of those tanks is identical to thermal energy storage tanks, where transient effects are accounted by solving at each timestep, mass, energy and momentum balances (see section 4.3.1).

![Figure 4-10 – Schematic of the steam drum model.](image)
4.3 Thermal energy storage

The TES system is mainly constituted by the storage tank(s), piping connections and, in case of different fluids at the SF and TES, Heat Exchangers (HEXs) are required for the interface between the SF and TES subsystems. Piping modelling has been described in section 4.2.4.

4.3.1 Storage Tanks

Within the storage tank, transient effects are accounted by solving at each timestep, mass, energy and momentum balances. Concerning the mass balance, the mass at the end of the timestep \( m_{t+\Delta t} \) is equal to the mass at the beginning \( m_t \) of the timestep plus the balance between mass flow rate at the storage inlet \( \dot{m}_{\text{in}} \) and outlet \( \dot{m}_{\text{out}} \) during the timestep \( \Delta t \),

\[
m_{t+\Delta t} = m_t + \dot{m}_{\text{in}} \Delta t - \dot{m}_{\text{out}} \Delta t \tag{4-54}
\]

The thermal energy in the storage tank at the end of the timestep \( Q_{\text{sto},t+\Delta t} \) entails the energy at the beginning of the timestep \( Q_{\text{sto},t} \) and the energy balance between charging \( \dot{Q}_{\text{sto,in}} \), discharging \( \dot{Q}_{\text{sto,out}} \) and losses \( \dot{Q}_{\text{sto,loss}} \) over the timestep,

\[
Q_{\text{sto},t+\Delta t} = Q_{\text{sto},t} + (\dot{Q}_{\text{sto,in}} - \dot{Q}_{\text{sto,out}} - \dot{Q}_{\text{sto,loss}}) \Delta t \tag{4-55}
\]

Both energy fluxes from charging and discharging are calculated by the product of the mass flow rate and enthalpy. On the other hand, heat losses depend on the temperature difference between the HTF and ambient, internal and external convection, and conduction at the insulation. Thus, a detailed heat loss models would increase modelling and simulation complexity and would require a detailed design of the storage tank. Alternatively, specific heat losses \( q_{\text{sto,loss}} \) are estimated at design conditions,

\[
\dot{q}_{\text{sto,loss}} = \frac{Q_{\text{sto,loss}}}{m_{\text{HTF}}(T_{\text{sto}} - T_{\text{amb}})} = \frac{1}{2\pi l_{\text{sto,ins}} n (\rho_{\text{ext,ins}} \nu_{\text{int,ins}})^{0.5} a_{\text{ext}} d_{\text{ext,ins}}} \tag{4-56}
\]
Where the external convective heat transfer coefficient is calculated by the Churchill-Bernstein correlation (equation 4-53). Afterwards, $\dot{q}_{sto,loss}$ is used for calculations at off-design operation, i.e. for distinct storage HTF mass ($m_{HTF}$) and temperatures,

$$
\dot{Q}_{sto,loss} = m_{HTF}\dot{q}_{sto,loss}(0.5(T_{sto,t+\Delta t} + T_{sto,t}) - T_{amb})
$$

(4-57)

Outlet thermodynamic proprieties depend on the proprieties inside the tank, i.e. enthalpy of the storage tank outlet is calculated by the average of enthalpy at the beginning and end of the timestep,

$$
h_{sto,out} = 0.5(h_{sto,t} + h_{sto,t+\Delta t})
$$

(4-58)

where enthalpy at the end of the timestep is calculated by,

$$
h_{t+\Delta t} = \frac{m_{in}h_{t} + (m_{in}h_{in} - \dot{q}_{sto,loss})\Delta t}{m_{t} + m_{in}\Delta t}
$$

(4-59)

The above equations are implicit, and thus an iterative process is required, estimating the enthalpy at the end of the timestep. A relative carriable change lower than $10^{-7}$ was set as stop criteria. To reduce model complexity, pressure drop within the storage tanks is not modelled, yet defined as an input for design conditions and evaluated at off-design condition.

4.3.2 Heat Exchanger

As above-mentioned, HEXs are necessary if different HTF fluids are used in the SF and TES. Modelling the heat exchanger consists in specifying three of the four hot and cold temperatures at the design conditions, and also the pressure drop in the cold and hot sides. Temperatures are defined directly or indirectly, by setting the lower or upper terminal temperature difference. Afterwards, an iterative process is used to assess both the amount of heat transferred and the nominal value of the product of the overall heat transfer coefficient and heat exchanger surface area ($UA)N$. The heat transfer ($Q_{HEX}$) in the HEX is,
\[ \dot{Q}_{HEX} = (UA)_N \Delta T_{lm} \quad (4-60) \]

Where the mean logarithm temperature difference \((\Delta T_{lm})\) is evaluated from cold and hot side temperatures [17],

\[ \Delta T_{lm} = \left( \frac{T_{out,hot}-T_{inc,cold}}{ln\left(\frac{T_{out,hot}-T_{inc,cold}}{T_{in,hot}-T_{out,cold}}\right)} \right) \quad (4-61) \]

and \((UA)_N\) can be estimated from equations 4-62 and 4-63 [10],

\[ (UA)_N = \frac{\dot{m}_{hot}(h_{in,hot}-h_{out,hot})}{\Delta T_{lm}} \quad (4-62) \]

\[ (UA)_N = \frac{\dot{m}_{cold}(h_{out,cold}-h_{in,cold})}{\Delta T_{lm}} \quad (4-63) \]

At off-design conditions, the overall heat transfer coefficient \(U\) is corrected using standard characteristic curves (see Figure 4-11) and a second-order interpolation [10]. In part-load operation, the product of the overall heat transfer coefficient and heat exchanger surface area is calculated by the ratio between the actual and nominal mass flow rate. Ebsilon contains an extensive database of characteristic curves for distinct HEX types. Whilst detailed modelling of the HEX is possible, it would increase both computational and modelling effort and also would require specific information concerning the HEX used.
CSP/biomass hybridisation can be achieved by the combination of different biomass technologies and also by distinct degrees of integration, i.e. at different zones of the power plant. BS model was developed with the purpose to permit application for a wide range of technologies. Accordingly, the hybrid section includes a boiler, and indirect storage component for dynamic modelling (e.g. start-up) and piping for additional pressure and heat losses.

4.4.1 Boiler

The boiler is composed of a combustion chamber, gas and air compressor, and a set of HEXs. The combustion mass balance includes air, fuel, ashes and flue gas,

\[ \dot{m}_{\text{fuel}} + \dot{m}_{\text{air}} = \dot{m}_{\text{fue.gas}} + \dot{m}_{\text{ash}} \]  

(4-64)

The combustion chemical reaction is modelled considering fuel chemical composition, and excess of air to stoichiometric combustion (input). As output, the
detailed species compositions leaving the combustion chamber are retrieved. For more
details regarding combustion, the reader is addressed to [27]. Radiation losses within the
chamber are calculated according to EN 12952 standard [28]. Flue gas enthalpy requires
flue gas temperature input, for design conditions.

4.4.2 Heat exchanger

Afterwards, the flue gas exchanges heat with the HTF in one or more heat exchangers.
In the case of a single-phase HTF (e.g. thermal oil), boiler modelling requires only one
heat exchanger. On the other hand, if a steam boiler is modelled (see Figure 4-12), at least
three heat exchangers are used: preheater (PH), evaporator (EVAP) and superheater (SH).
Additional heat exchangers are used if the flue gas heat is used to improve steam wetness
within the power block, i.e. a reheater (RH), and to preheat the air used for the combustion
chamber, i.e. economiser (ECON). Another ECON can be found to recover produced gas
high-temperature heat.

![Figure 4-12 – Schematic of the biomass steam boiler model.](image-url)
The biomass conversion system was modelled using empirical correlations from the literature and experimental work (see Chapter 5), and implemented within EbsScript through a Pascal code. Produced gas temperature and flow rate are calculated, and values transmitted to the combustion chamber gas line. Gas storage is considered through the use of a gasometer, modelled as a storage tank (see section 4.3.1).

The HEXs are modelled according to section 4.3.2. The boiler efficiency is calculated by the ratio between the energy transferred to HTF and fuel energy used, i.e. product of the mass flow rate and Lower Heating Value \((LHV_{fuel})\) of the fuel,

\[
\eta_{boiler} = \frac{\dot{m}_{HTF}(h_{HTF, out} - h_{HTF, in})}{\dot{m}_{fuel} LHV_{fuel}} \tag{4-65}
\]

Operation at part-load is modelled using a characteristic line to calculate the flue gas outlet temperature, and thus enthalpy. The characteristic line was defined using literature data, considering boiler efficiency at part-load operation.

4.4.3 Compressor

Compressors assure gas and air flow to the combustion chamber. A simplified methodology was used for modelling the compressors. A nominal isentropic efficiency is set as an input, and gas outlet enthalpy calculated through equation 4-66. A characteristic curve is used to define isentropic efficiency at part load conditions.

\[
\eta_{is,comp} = \frac{h_{out,is} - h_{in}}{h_{out} - h_{in}} \tag{4-66}
\]

The compressor work \((W_{comp})\) is assessed using the first law of thermodynamics for steady-state and neglecting heat loss,

\[
W_{comp} = \dot{m}_{air,gas} (h_{out} - h_{in}) \tag{4-67}
\]

Also, the compressors are coupled to a motor to evaluate parasitic consumption, considering electrical \((\eta_{elec})\) and mechanical \((\eta_{mech})\) efficiencies,
\[ \dot{W}_{e\text{l},c\text{ompressor}} = \frac{\dot{W}_{c\text{ompressor}}}{\eta_{\text{mech}}\eta_{\text{e\text{l}}} (4-68)} \]

4.5 Power block

For linear focusing CSP systems, the power block choice is limited to the conventional Rankine cycle, or for a lower power to the organic Rankine cycle. In each case, within Ebsilon the power block was modelled by the combination of turbine stages, a generator, HEXs, deaerator and piping. The only exception regards REELCOOP prototype 3, where the regenerative Organic Rankine Cycle was simulated using EES software [29], since property data of the organic fluid (Solvay SES36) were not available, neither in EES nor in Ebsilon databases. The data values were introduced into tables from which the thermodynamic properties (e.g. enthalpy, entropy) are calculated by linear interpolation.

A typical steam turbine cycle includes many HEXs, either for the steam generator (PH, EVAP and SH) or to enhance the overall performance of the turbine, e.g. using economisers for recovering waste heat. In any case, the HEXs and piping were modelled according to section 4.3.2 and 4.2.4, respectively.

It is noteworthy that detailed information concerning steam turbines is scarce and mostly confined to the manufacturer. Additionally, steam turbines are tailored made for specific plants. In this study, the turbine was modelled considering literature data, as well as using conventional design rules. For example, for a 1 MWel steam turbine, personal communication with Siemens allowed to define the maximum enthalpy change within turbine stages of 500 kJ/kg, an isentropic efficiency lower than 60%, and a pressure limit to 40/45 bar for economic viability. In case of limited literature data, design rules from [30] were used.

4.5.1 Turbine stages

In general, a turbine stage is defined by the inlet pressure and isentropic efficiency, whilst the pressure at the outlet is defined either by the next turbine stage or from the condensation pressure (i.e. at the last turbine stage). Mechanical energy at the output shaft is calculated also considering mechanical and wetness efficiencies and energy losses,
\[ W_{\text{turb}} = m_{\text{in}} (h_{\text{in}} - h_{\text{out}}) \eta_{\text{wetness}} \eta_{\text{mech}} - \dot{Q}_{\text{turb,loss}} \] (4-69)

At off-design operation, inlet pressure is calculated by the modified Stodola law of the ellipse [31],

\[ P_{\text{in}} = \sqrt{P_{\text{out}}^2 + \left( \frac{m_{\text{in}}}{m_{\text{in},N}} \right)^2 \left( \frac{T_{\text{in}} + 273.15}{T_{\text{in},N} + 273.15} \right) \left( P_{\text{in},N}^2 - P_{\text{out},N}^2 \right) } \] (4-70)

Thermodynamic properties at the turbine outlet are evaluated through isentropic efficiency. Off-design isentropic efficiency is calculated using second order interpolation of the characteristic line of \( \eta_{\text{is}} \) as a function of the ratio between actual and nominal mass flow rate,

\[ \eta_{\text{is,turb}} = \frac{h_{\text{in}} - h_{\text{out}}}{h_{\text{in}} - h_{\text{out},\text{is}}} \] (4-71)

For turbine stages with a slightly lower degree of superheating at the turbine inlet, expansion can result in saturated steam, and therefore efficiency should take into account steam wetness. According to Baumann, turbine efficiency reduction can be accounted by the product between the average steam wetness and the Baumann factor \( (a) \), which was set to 1 according to [32],

\[ \eta_{\text{wetness}} = 1 - a \frac{x_{\text{in}} + x_{\text{out}}}{2} \] (4-72)

4.5.2 Generator

Within the generator, mechanical work from the turbine is converted into power. Ebsilon includes an extended database of generator efficiencies, at design and off-design, for distinct power values. So, it is possible to calculate the effective turbine power at any timestep,

\[ \eta_{\text{gen}} = \frac{\text{power}}{W_{\text{turb}}} \] (4-73)
4.5.3 Dynamic Behaviour

During the start-up or ramp-up of the power block, thermal energy is dumped, i.e. there is no power output. Also, cold, warm and hot start-ups are expected to occur in a hybrid CSP/biomass power plant, and thus dependent on PB temperatures and so on the cool-down behaviour.

Detailed modelling of the PB during the start-up would require experimental tests and assessment of the water and steel mass within the PB. In this study, it was not possible to carry on any of these.

Therefore, PB operation during start-up, cool-down and stand-by are modelled based on manufacturer or literature data and implemented by code using EbsScript. The energy required for the cold start-up \(Q_{PB,\text{st-up}}\) at nominal conditions is calculated by

\[
Q_{PB,\text{st-up}} = \dot{Q}_{PB,\text{inp}}\Delta t_{PB,\text{st-up}}
\]  

(4-74)

The heat required, \(\dot{Q}_{PB,\text{inp}}\), depends on the energy source (i.e. SF, BS and TES). On the other hand, the energy required to start-up the power block \(Q_{PB,\text{st-up}}\) is defined from literature [11, 33-37], either directly or using the time required for a cold start-up \(\Delta t_{PB,\text{st-up}}\).

Off-design conditions (e.g. warm start-up), depends on the standby/cool-down behaviour of the system, i.e. on the energy available at the PB \(Q_{PB}\), the \(\dot{Q}_{PB,\text{inp}}\), standby or cool-down time \((\Delta t_{PB,\text{CD or SB}})\) and PB losses\((\dot{Q}_{PB,\text{loss}})\),

\[
Q_{PB,\text{st-up,OD}} = Q_{PB} + \dot{Q}_{PB,\text{inp}}\Delta t_{PB,\text{CD or SB}} - \dot{Q}_{PB,\text{loss}}\Delta t_{PB,\text{CD or SB}}
\]  

(4-75)

It is noteworthy that in the case of cool-down (instead of standby) equation 4-75 is simplified, by eliminating the \(\dot{Q}_{PB,\text{inp}}\) term.

4.5.4 Deaerator

Within the model, a deaerator is used to heat up condensate and make-up water to a specified temperature, using steam from a turbine stage extraction. Also, it is possible to consider steam losses to be balanced by the make-up water, and pressure losses within
the deaerator. The mass balance is solved at each time-step, considering mass conservation,

\[ \dot{m}_{\text{extr}} + \dot{m}_{\text{cond}} + \dot{m}_{\text{m-up}} = \dot{m}_{\text{vap,loss}} + \dot{m}_{\text{fw}} \tag{4-76} \]

The turbine extraction mass flow rate (\(\dot{m}_{\text{extr}}\)) is calculated using the first law of thermodynamics for an open and steady system and neglecting heat losses,

\[ \dot{m}_{\text{extr}} h_{\text{extr}} = \dot{m}_{\text{fw}} h_{\text{fw}} - \dot{m}_{\text{cond}} h_{\text{cond}} - \dot{m}_{\text{m-up}} h_{\text{m-up}} + \dot{m}_{\text{vap,loss}} h_{\text{vap,loss}} \tag{4-77} \]

Where the subscripts fw, cond, m-up and vap,loss stand for feed-water, condensate, make-up water and vapour losses, respectively.

### 4.6 Fluids

Many fluids were used in this work. Water/steam thermodynamic properties were calculated according to IAPWS-97 [38]. The ideal gas formulation was considered, and chemical characterisation of gases was carried out on the molar basis with proprieties evaluated according to FBDR [39]. Ebsilon database includes thermal oils and molten salts. In any case, modelling an HTF entails the definitions of density (\(\rho\)), specific heat (\(cp\)), thermal conductivity (\(\lambda\)), as a function of fluid temperature (\(T_{\text{HTF}}\)), using a fifth order polynomial equation,

\[ \psi_{HTF} = \sum_{z=0}^{5} \psi_z T^z \tag{4-78} \]

For the kinematic viscosity (\(\nu\)) [40],

\[ \nu = \exp\left( \frac{\nu_0}{T + \nu_1} + \nu_2 \right) \tag{4-79} \]

### 4.7 Pumps

Pumps are used to assure fluid circulation within the installation. For that reason, it is difficult to define the specific subsystem. Nevertheless, the use of pumps in a model is essential to account for parasitic consumption. A similar methodology as the one used for
modelling the compressor was applied (section 4.4.3). Outlet thermodynamic properties are evaluated through the isentropic efficiency, equations 4-80 and 4-81 for water and HTF, respectively.

\[
\eta_{is,pump,H2O} = \frac{h_{out,\text{is}}-h_{in}}{h_{out}-h_{in}} \tag{4-80}
\]

\[
\eta_{is,pump,HTF} = \frac{v_{HTF}(P_{out}-P_{in})}{c_{HTF}(T_{out}-T_{in})} \tag{4-81}
\]

A typical characteristic curve [10] is used to define isentropic efficiency at part load conditions. Also, the pumps are coupled to a motor (see section 4.4.3) to assess parasitic consumption, and so equations 4-82 and 4-83 are used to evaluate the pump work and electrical consumption, respectively.

\[
\dot{W}_{pump} = \dot{m}_{\text{fluid}} (h_{out} - h_{in}) \tag{4-82}
\]

\[
\dot{W}_{\text{elect, pump}} = \frac{\dot{W}_{pump}}{\eta_{\text{mech, elect}}} \tag{4-83}
\]

4.8 Operation strategy

Operation strategy entails control of the different subsystems and components, at distinct operation stages (e.g. start-up) and is carried out by a developed code using EbsScript tool. Distinct operation strategies are used depending on the assessment purpose. Nevertheless, the general operation strategy can be summarised (see Figure 4-13) for a cyclic operation:

- the load curve defines system operation strategy;
- operation strategy is assessed at the key controller, where the code is implemented;
- after load curve analysis and thus demand definition, the subsystem (SF, TES, BS, PB) states are evaluated;
• subsystem states in conjunction with meteorological data and operation strategy allow the definition of individual components control;

• within control strategy, as general rules: solar field has primacy over biomass; biomass is always available, and the biomass boiler can operate between the minimum and maximum limits;

• afterwards simulation for the specified timestep is carried out;

Figure 4-13 – Schematic of the general simulation operation control.

4.9 Economic assessment

The economic performance of a power plant can be evaluated through the use of different methods. One straightforward way to summarise and compare the overall attractiveness of case studies is to evaluate their levelised cost of electricity,
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\[ LCoE = \frac{CAPEX \times CRF + OPEX}{E} \quad (4-84) \]

The \( LCoE \) permits the assessment of annual energy generation \((E)\) on the basis of weighted average costs [41]. The CAPital EXpenditure \((CAPEX)\) includes the overnight and financing costs. On the other hand, the OPeration EXpediture \((OPEX)\) includes both fixed and variable power plant operating costs. The \( CAPEX \) is depreciated over the lifetime \((n_{\text{years}})\), and thus annuities are assessed by the Capital Recovery Factor \((CRF)\), considering the interest or return rate \((i)\),

\[ CRF = \frac{i(1+i)^{n_{\text{years}}}}{(1+i)^{n_{\text{years}}}-1} \quad (4-85) \]

In the case of combined heat and power the levelised cost of electricity \((LCoE_{CHP})\) should be reduced by the heat revenues [42],

\[ LCoE_{CHP} = \frac{CAPEX \times CRF + OPEX}{E} - \frac{H \times H_{\text{price}}}{E} \quad (4-86) \]

where \( H \) stands for the annual heat produced and \( H_{\text{price}} \) for the heat price.

4.10 Conclusions

In this chapter, a numerical model developed for CSP/biomass hybrid systems was presented, including the modelling methodology and assumptions. Modelling was carried out using a commercial software: EBSILON® Professional.

The modelling methodology consists of splitting the CSP/biomass hybrid power plant into subsystems: solar field, thermal energy storage, biomass system, and power block. Within the subsystems, detailed modelling was carried out for individual components (e.g. solar collectors in the SF). Whilst simulations encompass the entire system performance, this methodology allows assessment and control of individual subsystems and components, through an implemented operation strategy.

At the subsystem component level, modelling is carried out by evaluating thermodynamic properties (e.g. enthalpy) and solving continuity, momentum and energy
balances. Thus, component performance can be evaluated at a detailed level. Nonetheless, detailed modelling is not applied to all components and operation modes (e.g. off-design).

Notwithstanding that Ebsilon was designed for steady-state calculation, quasi-dynamic simulations were carried out, by the combination of dynamic components and a time-series, where calculations are carried out for each timestep. A full dynamic approach was applied to the components that entail a significant quantity of HTF and steel (i.e. thermal inertia) and where the highest temperature gradients are expected to occur, as the solar collectors.

Operation strategy and thus subsystem and components control at each timestep were achieved by a developed Pascal based code. The developed numerical model entails implicit equations, and empirical correlations (e.g. convective heat transfer coefficients); thus, additional programming was required to solve the equations. The economic performance of the CSP/biomass hybrid system is assessed through the levelised cost of electricity, either for power generation or CHP.

The BS is not modelled in detail. Literature data and empirical correlations are used to model the biomass conversion technologies. Also, as above mentioned, the numerical model entails assumptions and empirical correlations. Therefore, to enhance and verify the numerical model performance, experimental tests were carried out (Chapter 5). The tests include the assessment: of a gasification system to obtain the necessary empirical correlations; and a DSG loop, where tests were carried out to verify the CSP numerical model performance (Chapter 6).

4.11 References


5 EXPERIMENTAL WORK

Whilst the CSP/biomass hybrid assessment has been carried out by numerical simulation, the numerical model (Chapter 4) was upgraded, fine-tuned and validated (Chapter 6) through experimental work. The experimental work encompassed two tasks, consisting in the evaluation under real-life conditions of a biomass gasification system and a concentrating solar-field.

Gasification involves solid biomass conversion into a gaseous fuel, by a convoy of chemical reactions that occur at elevated temperatures, and therefore system thermal inertia and waste heat need to be assessed. Also, hybridisation implies a gasifier operation at distinct demand (i.e. part-load) and increasing the number of start-up and cool-down cycles. Although gasification experimental [1-4] and modelling [5-7] studies have been carried out, results were mostly focused on the gasification chemical process itself, discarding operation at different loads and useful heat from the producer gas. Furthermore, CSP/gasification hybrid assessments [8-11] are scarce in experimental tests, and the output was evaluated on a global basis, i.e. specific hybridisation enhancements were neglected.

One can argue that experimental work should include other biomass technologies such as combustion and anaerobic digestion, as well as distinct biomass residues and also different CSP heat transfer fluids (e.g. oil, salts). With the exception of molten salts, these experimental tests were foreseen in the scope of REELCOOP project [12]. Furthermore, biomass combustion is a mature technology that has been studied by many [13-16].
Regarding AD, chemical reactions take place at low temperatures, and thus the most critical output for hybrid operation is AD yield and biogas composition [17, 18]. Therefore, for modelling both combustion and AD literature data were used. Regarding CSP, modelling and simulation with a single-phase fluid (e.g. thermal oil) is relatively straightforward [19]. On the other hand, two-phase heat transfer and fluid flow in long horizontal pipes increase the modelling challenges and complexity. Therefore, the numeric model performance was only assessed for direct steam generation, based on experimental tests carried out at DISS test facility at Plataforma Solar de Almeria (Spain).

### 5.1 Biomass gasification assessment

The performance assessment of a biomass gasification system was carried out at the Faculdade de Engenharia da Universidade do Porto (FEUP), using a small-scale gasifier system (see Figure 5-1). The tests consisted on the evaluation of the system performance for steady-state conditions at different loads, and for distinct operation stages, including start-up and shut-down. Test operation data were monitored and recorded, through the use of a data logger and a LabVIEW developed code.

![Figure 5-1 – Biomass gasification system.](image)
The biomass gasification system is a commercial unit named PP20 Power Pallet manufactured by All Power Labs – Carbon Negative Power & Products (APL) [20]. The system is mainly constituted by a downdraft gasifier and a combustion engine coupled to a power generator. In other words, the system converts woody biomass into a combustible gas (producer gas), which is used to drive a conventional spark-ignition combustion engine/power generator set.

In the following sections, the gasification system is presented, including the installed instrumentation and datalogger system, as well as the system control. Subsequently, the used biomass feedstock (i.e. woodchips) is characterised concerning size, moisture content, density and lower heating value. The prime variables and assumptions for the performance assessment are described.

In order to assure a hybrid seamless operation and considering the variable nature of solar radiation, the biomass system will operate either at nominal load (e.g. night) or part load (e.g. low solar radiation). Furthermore, numerous start-up and cool-down cycles are foreseen to backup solar energy (e.g. clouds). Thus, the results and discussion section include the most significant tests for:

- steady operation at different loads;
- cold and hot start-up;
- cool-down.

At the end, critical issues during operation are presented. In the author’s opinion, this information is crucial for gasification researchers.

5.1.1 Gasification system

It is possible to divide the system processes into the flow: of solids, producer gas and flue gas (see Figure 5-2). The gasifier has a downdraft configuration reactor, where the solids flow downwards from the hopper to the reactor by gravity. Initially, solid biomass flows from the hopper to the drying bucket (DB), where its moisture content is decreased. Within the gasifier, biomass is further dried and submitted to a convoy of reactions (i.e. pyrolysis, combustion, tar cracking and reduction). For more details, the reader is addressed to chapter 2.
In addition to the producer gas, conversion output results in ashes from charcoal reduction, which are automatically purged to the gas vessel, to assure gas flow. Producer gas leaves the reactor at very high temperatures (form 500°C to 700°C) and contains suspended particles (e.g. charcoal dust), and thus conditioning is required. Part of the heat is used to preheat the intake air within gasifier air lines (improving both combustion and tar cracking efficiency) and suspend particles are separated at the gas cyclone to avoid fouling.

![Figure 5.2 - Gasification flow of solids, producer gas and flue gas.](image)

Afterwards, the gas flows through the drying bucket, which is recuperator type heat exchanger where biomass moisture content is minimised using heat from the producer gas. To avoid filter media pyrolysis the producer gas is further cooled down using corrugated dissipation pipes, as shown in Figure 5-3-a. To protect the engine from tar, residual tar gases are condensed at the filter media, which is constituted by sifted biomass of different sizes (see Figure 5-3-b). The producer gas is pulled out from the reactor through a vacuum system, either with a set of gas blowers and flare (i.e. during start-up) or by the combustion engine.
Before entering the engine, a lean mixture of gas and air (additional 5% of air compared to the stoichiometric air-fuel ratio) is established and condensates collected. The gas/air mixture combustion takes place in a spark-ignition engine, with 3000 cm$^3$ and four cylinders. The main differences compared to a conventional automobile engine are: fuel/air mixture and spark timing to compensate the slow flame propagation of producer gas. The engine's exhaust gas temperature is in the range of 400˚to 600˚C, and the heat is recovered in the reactor to drive the pyrolysis reaction.

5.1.2 Instrumentation, data logging and automation

In order to monitor all the relevant operation data and to control, the system was equipped with 18 K-type thermocouple probes placed at suitable locations, two orifice plate flow meters for measuring the syngas and air volumetric flow rate, and four differential pressure sensors (see Figure 5-4). Also, the gas chemical composition was analysed through the use of a gas analyser.
Inside the reactor, nine thermocouples were installed. Three thermocouples are used for measuring temperatures in the reaction and reduction zones (see Figure 5-5-a), and one to prevent overheating. The other six are coupled in a 6-point thermocouple probe, which is installed at the top of the reactor (Figure 5-5-b). A ceramic sheath was used to protect the probe from the reactor high temperatures. This probe permits the assessment of the temperature profile inside the reactor within the pyrolysis zone.

![Figure 5-5 – Thermocouples location within the gasifier.](image)

Other thermocouples were installed in relevant locations in the experimental apparatus, i.e. to measure the gas temperature at: reactor outlet, inlet and outlet of the drying bucket and filter, engine exhaust gas temperature at engine outlet and after heat exchange in the reactor. Additionally, one thermocouple is used to monitor the engine coolant temperature and another for ambient temperature measurement. Installation of the different sensors required the design and manufacture of several fittings (see Figure 5-6).

![Figure 5-6 – Custom fittings for instrumentation.](image)
The air and gas orifice plate flow meters were installed at the reactor air inlet valve and after the filter, respectively (Figure 5-7). Concerning the pressure sensors, two are installed inside the reactor to measure both combustion and gas outlet pressure, and one at the top of the filter. The other pressure sensor is used to monitor the engine oil pressure. These sensors are mostly used for controlling and monitoring the operation (e.g. flow restriction).

The producer gas chemical composition is measured using a gas analyser (Figure 5-8-a). Gas is extracted at the filter outlet using a vacuum pump and further filtered through two water scrubbers, a charcoal filter, and a tar 0.3µm filter element (see Figure 5-8). The circuit ends with the gas being rerouted to the system.
5.1.3 Instrumentation calibration and precision

Instrumentation precision was evaluated, and calibration was carried out for the flowmeters. The K-type thermocouple probes were acquired from Omega, and have a maximum error of 2.2°C [21]. Thermocouple probes precision and linearity were evaluated for the 0°C to 140°C temperature range. This assessment was accomplished using as standards: a temperature calibrator [22], an isothermal water bath [23] and an ice bath. Results showed a maximum error of 0.2°C. Nevertheless, it was not possible to verify the linearity and precision for higher temperatures (above 140°C), and therefore the manufacturer error was considered.

Both orifice plate flow meters are custom made and were not designed according to typical standards, and thus calibration was required. Calibration was carried out using an orifice plate as standard and a Pitot tube for minimising measurement errors. The calibration fluid was air, and the test was carried for air and gas measurement range (see Figure 5-9).
The orifice plate dimensions were measured, specifically pipe \((D_{\text{pipe}})\) and hole \((D_{\text{hole}})\) diameters. The ratio between pipe and hole diameters \((\beta)\) hole area \((A_{\text{hole}})\) was calculated, using equations 5-1 and 5-2, respectively.

\[
\beta = \frac{D_{\text{hole}}}{D_{\text{pipe}}} \quad (5-1)
\]

\[
A_{\text{hole}} = \frac{\pi D_{\text{hole}}^2}{4} \quad (5-2)
\]

The calibration procedure consisted on setting up and measuring the flow rate by using the calibrated orifice plate, and verifying it by using a Pitot tube. Afterwards, the pressure drop \((\Delta P_{\text{orifice}})\) within the orifice plate hole is measured. The procedure was repeated for different flow rates within the measurement range.

The volumetric flow rate \(\dot{V}\) is calculated based on the continuity and Bernoulli equations, and considering friction in the duct flow, through equation 5-3 [24]:

\[
\dot{V} = C_d A_{\text{orifice}} \sqrt{\frac{2\Delta P_{\text{orifice}}}{\rho(1-\beta^4)}} \quad (5-3)
\]

The discharge coefficient \(C_d\) accounts for discrepancies and is a function of \(\beta\) and Reynolds number \((Re_D)\). According to ISO 5167-1:2003 [25], \(C_d\) should be calculated through equation 5-4:
\[ C_d = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^8 + 91.71\beta^{2.5}Re_D^{-0.75} + \frac{0.095}{1-\beta^4}F_1 - 0.0337\beta^3 F_2 \] (5-4)

The correlation factors \( F_1 \) and \( F_2 \) depend on the tap position. As the used orifice plates do not follow a standard design, the correlation factors were calculated to minimise the sum of squared errors of the prediction. Diameter ratio, orifice area and calculated correlation factors are shown in Table 5-1. The maximum measured errors were 1.5% and 1.2% of the measurement range for the gas and air flow meters, respectively.

<table>
<thead>
<tr>
<th>Orifice Plate</th>
<th>( \beta )</th>
<th>( A_{\text{Orifice}} \text{ [m}^2\text{]} )</th>
<th>( F_1 )</th>
<th>( F_2 )</th>
<th>Maximum Error [ % of measurement range ]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>0.516</td>
<td>8.88E-5</td>
<td>89.93</td>
<td>-60.59</td>
<td>1.2%</td>
</tr>
<tr>
<td>Gas</td>
<td>0.538</td>
<td>3.79E-4</td>
<td>9.327</td>
<td>0</td>
<td>1.5%</td>
</tr>
</tbody>
</table>

Monolithic silicon pressure sensors, specifically piezoresistive transducers were used for pressure measurements. For the combustion, reactor and filter outlet gas pressure measurement, model MPXV7007 from NXP Semiconductors N.V. manufacturer was used. For differential pressure measurement at the orifice plate flow meters, the same model was used for gas and MPXV7002 for air. The pressure sensor range and accuracy of both models are shown in Table 5-2.

<table>
<thead>
<tr>
<th>Model</th>
<th>Range [kPa]</th>
<th>Accuracy [% of full scale span]</th>
</tr>
</thead>
<tbody>
<tr>
<td>MPXV7002</td>
<td>-2 to 2</td>
<td>±2.5%</td>
</tr>
<tr>
<td>MPXV7007</td>
<td>-7 to 7</td>
<td>±5%</td>
</tr>
</tbody>
</table>

The gas analyser, model 3100P from Wuhan cubic manufacturer, is designed specifically for syngas analysis, allowing measurements of CO\(_2\), CO, H\(_2\), O\(_2\), CH\(_4\) and other hydrocarbons C\(_n\)H\(_m\) contents (see Table 5-3).
Table 5.3 – Gas analyser 3100P technical specification [28].

<table>
<thead>
<tr>
<th>Components</th>
<th>Analytic method</th>
<th>Range</th>
<th>Resolution</th>
<th>Precision</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO₂</td>
<td>NDIR</td>
<td>20%</td>
<td>0.01%</td>
<td>≤2%</td>
</tr>
<tr>
<td>CO</td>
<td>NDIR</td>
<td>30%</td>
<td>0.01%</td>
<td>≤2%</td>
</tr>
<tr>
<td>H₂</td>
<td>TCD</td>
<td>30%</td>
<td>0.01%</td>
<td>≤2%</td>
</tr>
<tr>
<td>O₂</td>
<td>ECD</td>
<td>5%</td>
<td>0.01%</td>
<td>≤2%</td>
</tr>
<tr>
<td>CH₄</td>
<td>NDIR</td>
<td>10%</td>
<td>0.01%</td>
<td>≤2%</td>
</tr>
<tr>
<td>C₇H₸</td>
<td>NDIR</td>
<td>10%</td>
<td>0.01%</td>
<td>≤2%</td>
</tr>
</tbody>
</table>

The calibration process is specific for each gas component and is only required if the gas measurement error exceeds technical specifications. Thus, a gas sample with typical syngas composition was prepared in the laboratory and measured using the gas analyser. Results are presented in Table 5.4 and show measurement errors inferior to the manufacturer’s precision, and thus calibration was not required.

Table 5.4 – Gas analyser measurement error.

<table>
<thead>
<tr>
<th>Gas component</th>
<th>Sample Concentration [%]</th>
<th>Measured Concentration [%]</th>
<th>Error absolute [%]</th>
<th>Relative error [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH₄</td>
<td>5.00</td>
<td>4.98</td>
<td>0.02</td>
<td>0.4</td>
</tr>
<tr>
<td>CO</td>
<td>20.00</td>
<td>20.00</td>
<td>0.08</td>
<td>0.0</td>
</tr>
<tr>
<td>C₇H₸</td>
<td>3.00</td>
<td>3.06</td>
<td>0.06</td>
<td>2.0</td>
</tr>
<tr>
<td>CO₂</td>
<td>15.00</td>
<td>15.09</td>
<td>0.09</td>
<td>0.6</td>
</tr>
<tr>
<td>H₂</td>
<td>20.00</td>
<td>20.02</td>
<td>20.02</td>
<td>0.1</td>
</tr>
</tbody>
</table>

5.1.4 Automation and data logging

The gasifier was ordered with a control unit, that serves for both system operation automation and data logging. The data received from the several sensors on the gasification system is used as input for subsystems operation command, e.g. grate shaking, to maintain the flow of feedstock and to purge small char particles from the gasifier bed, to control the feedstock feeder, to adjust the air/fuel mixture for the engine to ensure complete combustion of the producer gas, to trigger an alarm.

The control unit is driven by an open-source Arduino software code, and thus it can be reprogramed for research, development and customisations. Also, the installation of new sensors required the use of an extra datalogger, an Agilent 34970A unit. A digital
interface and a communication protocol were developed, using LabView software to monitor and record relevant data from both dataloggers and gas analyser, at a 5-second timestep.

5.1.5 Biomass Feedstock

The feedstock is a crucial variable in the system performance, to assure a desirable producer gas yield, operation reliability and reduced maintenance. To avoid typical issues of downdraft gasifiers, there are three general feedstock requirements: particle size (to maintain gas and solids flow dynamics), moisture content (to sustain high temperatures), and ash content (to prevent clinker formations). As this study aims the evaluation of the gasifier performance, dry pine woodchips were used, as recommended by the manufacturer. Woodchip gasification is a relatively mature technology, however there are various occurrences of system failures [29].

According to the manufacturer, particle sizes should ideally be between 1 and 2.5 cm. Oversized chips cannot exceed 4 cm, and fines (under 1 cm) are limited to 10% of the feedstock. Regarding moisture content, it should be inferior to 30%, and ideally below 15% to enhance a fast start-up. A higher moisture content will require a significant amount of energy to dry it up and consequently decrease system performance.

In order to assure particle size requirements (i.e. feedstock size homogeneity), the woodchips were sifted using three metal cloth meshes, with 2.54, 1.27 and 0.3175 cm (see Figure 5-10). This task was arduous and indicates the absence of a feedstock market for wood biomass gasification, consequently hindering dissemination of this technology.
The woodchips were mixed, and a representative sample collected to evaluate particle characteristic-length and moisture content. The length of each woodchip was measured using a calliper rule. The results were analysed through descriptive statistics, and a Shapiro-Wilk normality test was also carried out. The mean value of the characteristic length is 2.218 cm, with a standard deviation of 0.408 (Figure 5-11). Also, the sample characteristic length follows a normal distribution.

Concerning the moisture content, initially an electric moisture meter was used. However, the low moisture content of the wood chips along with the low resolution of the wood moisture meter jeopardised the accuracy of the results. Therefore, as an alternative, the moisture content of the woodchip samples was assessed by the ASTM D4442 [30], Method B—Secondary Oven-Drying Method. This method requires an oven...
capable of maintaining 103°C ± 2°C and scales with a sensitivity of at least 0.1% of the sample weight. The procedure starts with weighing the wet biomass sample in the scales and then placing it in the oven. At each 4-hour interval, the sample is removed from the oven and reweighted. The procedure ends when there is no meaningful change in the sample weight. For this assessment, three samples of 100 grams were prepared, and the procedure was carried out using a scale with a sensitivity of 0.01 grams and a 29 dm³ convective oven with a microprocessor-based temperature control. Moisture content was evaluated for both wet (equation 5-5) and dry basis (equation 5-6). Results showed an average moisture content ($\omega$) of 9.2% and 8.4%, in the dry and wet basis, respectively (see Table 5-5).

$$\omega_{wb} = \frac{m_{H_2O}}{m_{samp,w}} \times 100 \% \tag{5-5}$$

$$\omega_{db} = \frac{m_{H_2O}}{m_{samp,d}} \times 100 \% \tag{5-6}$$

<table>
<thead>
<tr>
<th>$m_{samp,w}$ [g]</th>
<th>$m_{samp,d}$ [g]</th>
<th>$W_{wb}$ [%]</th>
<th>$W_{db}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>91.5</td>
<td>8.5</td>
<td>9.3</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>91.7</td>
<td>8.3</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>91.6</td>
<td>8.4</td>
</tr>
</tbody>
</table>

A similar process was set for calculating the biomass basic density (equation 5-7) and the mass density (equation 5-8), differing on the use of the dry or wet mass of the sample ($m_{samp,d}$ and $m_{samp}$ respectively).

$$\rho_{basic} = \frac{m_{samp,d}}{V_{samp}} \tag{5-7}$$

$$\rho_{mass} = \frac{m_{samp}}{V_{samp}} \tag{5-8}$$

As woodchips have irregular shapes, a sample of 100 cylindrical shape pieces was prepared, and thus the volume ($V_{samp}$) calculated by equation 5-9. The diameter of the sample ($D_{samp}$) is constant, and only the thickness needs to be measured ($e_{samp}$), using a
calliper rule. Results showed an average basic density of 209.8 kg/m$^3$ and mass density of 220.2 kg/m$^3$.

\[ V_{\text{samp}} = \frac{\pi D_{\text{samp}}^2}{4} e_{\text{samp}} \]  

(5-9)

There are standards to determine solid biofuels heating value (e.g. CEN/TS 14918:2009). Nevertheless, in this study, the biomass source was pine, specifically Pinus pinaster, an abundant species in Portugal, for which properties had been evaluated by many [31-33]. Therefore, an average value for the LHV on a dry basis of 20.05 MJ/kg was used from Phyllis2 database [34]. The lower heating value considering the biomass moisture content ($\omega_{wb}$) was calculated using equation 5-10 [35]:

\[ LHV = LHV_{\text{dry}} \left(1 - \frac{\omega_{wb}}{100}\right) - 2.443 \frac{\omega_{wb}}{100} \]  

(5-10)

5.1.6 System performance evaluation

The primary variables for gasifier performance evaluation were: biomass consumption; air and gas volumetric flow rate; operating temperatures; gas composition; and LHV.

Evaluation of the biomass consumption was carried out considering an average value, i.e. measuring the volume variation in the feedstock hopper. The biomass depth variation ($\Delta z$) inside the hopper was measured at the beginning and end of the test, using a metallic ruler, as shown in Figure 5-12. Afterwards, biomass consumption was calculated by equation 5-11, where $D_{\text{hopper}}$ is the hopper diameter, and $\Delta t$ stands for the total time of the experiment.

\[ m_{\text{biom}} = \rho_{\text{biom}} \frac{\pi D_{\text{hopper}}^2}{4} \frac{\Delta z}{\Delta t} \]  

(5-11)
Because of the space between woodchips, it was necessary to evaluate biomass bulk density $\rho_{\text{biom}}^*$. A conic metal bucket was used, and biomass bulk density was measured by filling the bucket with woodchips (without compacting) and weighting. Five measurements were taken, and the average bulk density was about $254.3 \text{ kg/m}^3$. This value is about 25% higher than literature values [36], which is justified by the sifting process.

Concerning biomass consumption, better-quality results would be achieved if intermediate biomass consumption measurements were taken. I.e., it would be possible to quantify biomass consumption during distinct operation stages (e.g. start-up). This solution was attempted. Nevertheless, it was found that opening the hopper whilst the gasifier is running represents a risky procedure. Removing the hopper cover results in excess of air entering the gasifier which can be observed at the flare, i.e. combustion takes place outside the flare. It would be possible to overcome this issue by tuning the air/gas flow rate, yet it would require two persons working. An additional problem is related to unbalanced operation pressure at the reactor, which results in reverse gas flow. As the producer gas composition comprises carbon monoxide, it represents a considerable health hazard for the operator. In this assessment, to differentiate biomass consumption at distinct operation stages, the procedure consisted of evaluating these operation stages individually. Each test included a start-up, and system start-up was tested sufficient times to evaluate the consumption at that stage. Afterwards, the start-up time was subtracted from the total test consumption.
On the other hand, the predefined time-step (5 seconds) was used for the measurement of other relevant variables, as well as for the required calculations. The EES software was used, and a routine developed to automatize the calculation process.

Inside the reactor, primary variables are the temperature, especially the reduction and restriction temperature. Higher temperatures result in enhanced gas yield, i.e. improved conversion efficiency. Gasifier performance as a whole was evaluated through cold gas ($\eta_{CG}$) and hot gas ($\eta_{HG}$) efficiencies, by equations 5-12 and 5-13, respectively [37]:

$$\eta_{CG} = \frac{\dot{m}_{gas}LHV_{gas}}{\dot{m}_{biom}LHV_{biom}} \quad (5-12)$$

$$\eta_{HG} = \frac{\dot{m}_{gas}LHV_{gas}+\dot{m}_{gas}c_{p_{gas}}\Delta T-\dot{m}_{gas}c_{p_{gas}}\Delta T_{DB}}{\dot{m}_{biom}LHV_{biom}} \quad (5-13)$$

Whilst cold gas efficiency is defined by the ratio between energy output (producer gas) and the energy input (biomass), i.e. only considering chemical conversion, the hot gas efficiency includes the sensible heat of the gas. Accordingly, if the producer gas is used to drive an engine, the $\eta_{CG}$ is commonly used, and if the gas is used for heat (e.g. combustion in gas burners), the $\eta_{HG}$ is recommended. Note that in equation 5-13, the heat used to dry up biomass is subtracted, as is not dumped. Also, using the engine changes the energy balance within the gasifier, with pyrolysis being partly sustained by the engine exhaust flow rate.

Mass flow rates were calculated using density and volumetric flow rate. Volumetric air and gas flow rates were calculated as explained in section 5.1.3. Air density was evaluated considering a real gas and using ambient temperature and pressure. Whilst ambient temperature was measured at every 5 seconds, the pressure was calculated by the average value of measurements taken at the beginning and end of the experimental test.

Gas composition was measured allowing the calculation of the producer gas density (equation 5-14), Lower heating value (equation 5-15) and specific heat (equation 5-16). The syngas LHV was calculated on the dry basis, where $Y_{(CO,CH_4)}$ and LHV$_{(CO,CH_4)}$ are the molar fractions and the lower heating value of the different gas constituents, respectively. Distinct component properties ($\rho$, LHV and $c_p$) were calculated through EES properties database, using measured gas temperature and pressure. The producer gas energy flux
(\(\dot{Q}_{\text{producer, gas}}\)) was calculated by the use of equation 5-17. Other higher hydrocarbons are present in the syngas composition; however, the gas analyser does not permit to assess their detailed composition. Also, their molar fraction is relatively low when compared to the other gas constituents. Thus, the contribution of these higher hydrocarbons was neglected.

\[
\rho_{\text{producer, gas}} = \sum Y_Z \rho_Z, \text{with } z = CO, CO_2, CH_4, H_2, O_2 \tag{5-14}
\]

\[
LHV_{\text{producer, gas}} = \sum Y_Z LHV_Z, \text{with } z = CO, CH_4, H_2 \tag{5-15}
\]

\[
C_p_{\text{producer, gas}} = \sum Y_Z C_p_Z, \text{with } z = CO, CO_2, CH_4, H_2, O_2 \tag{5-16}
\]

\[
\dot{Q}_{\text{syngas}} = \dot{m}_{\text{syngas}} \times LHV_{\text{syngas}} \tag{5-17}
\]

One of the most important variables used to evaluate gasifier performance is the equivalence ratio (ER), defined as the ratio between measured and stoichiometric air/fuel ratio (equation 5-18). ER controls the biomass consumption ratio, as well as allowing to tune up the H2 and CO gas content [38].

\[
ER = \frac{(\text{Air}/\text{Fuel})_{\text{measured}}}{(\text{Air}/\text{Fuel})_{\text{stoichiometric}}} \tag{5-18}
\]

The amount of air required for stoichiometric combustion of biomass is evaluated using equation 5-19 [39]. Biomass carbon, hydrogen, oxygen and sulphur molar contents from [34] were used, and F is the actual to theoretical air factor, in this case, 1. The calculated stoichiometric air to fuel ratio is about 5.72 Nm\(^3\)/kg, meaning that about 1 kg of wood chips require 5.72 cubic meters of air for combustion.

\[
Y_C C + Y_{H_2} H_2 + Y_{O_2} O_2 + Y_{N_2} N_2 + Y_S S + F Y_{air} O_2 + 3.76 F Y_{air} N_2 \rightarrow Y_C CO_2 +
Y_{H_2} H_2 O + Y_S SO_2 + (3.76 Y_{air} + Y_{N_2}) N_2 + (F - 1) O_2 \tag{5-19}
\]
5.1.7 Results and discussion

In this section, experimental tests are described, and results presented and discussed. It was foreseen to test the gasification system using the IC engine. However, the IC engine failed during the test period as described in section 5.1.12. Therefore, tests were carried out using fan blowers to drive the gasifier.

Before each experimental test, a leakage test was carried out. The system operates at vacuum pressures, and air leaks represent hazards for the system and user, e.g. internal fires. The leakage test consisted of sealing the gasifier air intake and using the gas blowers to set the system to vacuum pressure. Afterwards, the time to reach atmospheric pressure was measured, and if it took more than 60 seconds, the system was considered sealed. Test results showed values above 120 seconds, and thus the system has no significant leaks.

5.1.8 Steady State – Distinct loads

The first task was to evaluate the gasifier operation at steady state conditions, to characterise and establish a baseline operation performance. The system was tested four times for 5 to 7 hours, at maximum flow rate and under steady conditions. Operation results for 2 hours of one representative test are shown.

Figure 5-13 shows test reduction, restriction and gas reactor outlet temperatures. Reduction and restriction temperatures average values are about 729°C and 915°C, respectively, and thus within the expected temperature range, 650°C to 900°C for reduction, and 800°C to 1200°C for restriction [40]. Temperatures are relatively stable during the test, and oscillations are a consequence of operation without engine, and gasifier subcomponents intermittent and automatic operation, e.g. fuel and ash augers, grate shaker. The gasification temperatures are not subjected to control, but influenced by the feedstock and reactor design, as well as system operation. Concerning the gas outlet temperature, the average value is about 550°C, and fluctuations are less noticeable. It is noteworthy that the gas outlet temperature can be enhanced by previously heating the oxidiser (i.e. air).
Producer gas molar composition (see Figure 5-14) comprises: 24.78% of carbon monoxide, 18.72% of hydrogen, 11.1% of carbon dioxide and 2.1% of methane. An oxygen average molar fraction of 0.14% was measured, showing that oxidation is not complete, and also producer gas entails 3.8% of undesirable heavier hydrocarbons (C_nH_m), showing an incomplete gasification process. The average producer gas lower heating value is about 5.51 MJ/kg.

Figure 5-13 - Reduction, restriction and gas reactor outlet temperatures for 2 hours at steady state conditions.

Figure 5-14 – Producer gas molar composition and LHV for 2 hours at steady state conditions.
As aforementioned, the gasifier does not operate under steady state. Thus oscillations (about ±10% of average values) in the gas and air mass flow rates occur (see Figure 5-15), as the flow rate depends on the pressure drop along the system (e.g. reactor, pipes). To minimise this issue, the engine must be used or a blower able to sustain a higher pressure drop (e.g. boiler gas blower). Gas and air mass flow rate average values are 13.6 kg/s and 21.35 kg/s, respectively. In other words, 1 kg of producer gas requires about 0.64 kg of air. As the lower heating value of producer gas is stable during the test, fluctuations in the producer gas heat rate are related to the mass flow rate. An average gas energy flux of about 31.64 kW_th was measured.

![Figure 5-15 – Producer gas and air mass flow rate, and gas heat rate for 2 hours at steady state conditions.](image)

The equivalence ratio (see Figure 5-16) follows a similar profile to air mass flow rate, as a consequence of using the test average biomass consumption. An average ER of 0.26 was calculated, which is within the expected range (0.2 to 0.35) [41]. Results show an average cold gas efficiency of about and 69.4%, which is about 18% lower than the manufacturer announced value.
The temperature profile in the reactor is shown in Figure 5-17. It is visible that drying and pyrolysis (200°C to 500°C) occur in the first half of the reactor. At lower depths, temperatures drastically increase, and the highest values stand in the combustion zone. Subsequently, temperatures decrease, and the reduction reaction takes place at the reactor bottom.

Hybrid operation implies variable operation of the gasifier which will affect gasification performance. Thus, the gasifier behaviour was assessed for distinct loads, i.e.
controlling the gas flow rate using a gas fan potentiometer. Whilst repeatability was attempted, a suitable relation between potentiometer position and gas flow rate was not found. Thus, the set of tests entailed after gasifier start-up, the adjustment of the gas fan potentiometer to attain a desirable flow rate. Subsequently, no further adjustments were made in the potentiometer. As gas flow rate oscillates during the tests, the load is defined as the ratio between test average and nominal (i.e. maximum) mass flow rate, as shown in equation 5-20:

$$\text{Load} [\%] = \frac{\dot{m}_{\text{avg, test}}}{\dot{m}_{\text{avg, N}}}$$  \hspace{1cm} (5-20)

The assessment was carried out for loads between 100% and about 20%, with a 10% step. Average, maximum and minimum mass flow rates for partial load tests are shown in Figure 5-18. For each load, results include a set of three tests with at least three hours of steady state.

![Figure 5-18 – Gas mass flow rate at distinct loads.](image)

Results for restriction and reduction temperature are shown in Figure 5-19-a) and b), respectively. At lower mass flow rates, chemical reactions occur at lower temperatures. Nonetheless, until 50% load, there is no significant decrease in the reduction and
restriction temperature, and also temperatures are above the manufacturer lower threshold.

Whilst at 30% and 20% loads restriction average temperatures are above the lower limit (i.e. 800°C), results show that the minimum temperatures go down to about 760°C. Regarding the reduction temperature, the lower limit (i.e. 650°C) is surpassed by minimum temperatures at 80% load, and even by average temperatures for loads between 20% and 40%.

These limits do not stand for significant operation risks, and are mostly indicative of the gasification behaviour and used as a threshold to start up the engine. However, as chemical reactions occur at lower temperatures, this will result in an immediate chemical conversion efficiency drop, and therefore the content of undesirable products (e.g. tar) will increase.
The effect of load on the gas lower heating value is presented in Figure 5-20. There is a slight increase of the LHV by reducing the gas flow rate. The highest average value (5.97 MJ/kg) was measured at 64% load. The lower values of the restriction temperatures at 20% and 30% load result in average gas LHV below 5.5 MJ/kg. Nevertheless, at any test, the LHV is reasonable to drive either an engine or a boiler.
Gasifier performance at distinct loads can be observed through the equivalence ratio and cold gas efficiency (see figure 5-21). The average ER varies from about 0.250 to 0.265 between 100% and 60% load, except for 80% load where a moderately lower average ER was measured. The lower air to biomass ratio (i.e. ER) at 80% load, results in an enhancement of cold gas efficiency to 73.7%, representative of the finest tested gasifier operation conditions.

From 50% to 20% load, ER significantly increases, and consequently cold gas efficiency drops. Lower gas mass flow rates, result in lower chemical reaction temperatures (Figure 5-19), and thus lower efficiency. As for the same amount of biomass, more air is used, combustion becomes more dominant [42], and therefore both carbon monoxide and hydrogen contents at the producer gas are lessened [43], which results in a lower LHV (Figure 5-20). Notwithstanding the significant cold gas efficiency drop, the partial load operation is feasible yet at lower conversion efficiency.
Figure 5-21 – a) ER and b) cold gas efficiency at distinct loads.

5.1.9 Useful heat from gas

The producer gas leaves the gasifier at high temperature, and gas conditioning requires lower temperatures. Therefore, there is a significant amount of heat at a high temperature that can be recovered, and which can be evaluated through hot gas efficiency, i.e. considering the sensible heat of the gas.

Results show that hot gas efficiency varies between 89.7% at 80% load and 76.8% at 30% load (see Figure 5-22). Similar cold and hot gas efficiency profiles are expected, as both include the heat of the gas and biomass (equations 5-12 and 5-13). Nevertheless, the
difference between both decreases with load, because of the producer gas sensible heat reduction, i.e. producer gas temperature in the reactor outlet.

![Figure 5-22 – Hot and cold gas efficiency at distinct load.](image)

The ratio between the useful heat ($\dot{Q}_{usef}$) and producer gas energy flux ($\dot{Q}_{producer,gas}$) is about 22.5% at 100% load, and slightly decrease with load to about 20%. On the other hand, the decrease of useful heat with biomass energy flux is more significant, with a reduction from 17.5% at 100% load to 14% at 30% load. This fact is a consequence of the cold gas efficiency decrease at lower loads.

![Figure 5-23 – Ratio between gas useful heat and gas and biomass heat rate at distinct loads.](image)
The producer gas sensible heat \( (\dot{Q}_{\text{gas,total}}) \) encompasses gas heat loss/exchange between: reactor outlet and cyclone \( (\dot{Q}_{\text{Cyc}}) \); cyclone and drying bucket inlet \( (\dot{Q}_{\text{Cyc-DB}}) \); drying bucket \( (\dot{Q}_{\text{DB}}) \) inlet an outlet; drying bucket outlet and filter \( (\dot{Q}_{\text{DB-Filter}}) \).

The ratio between these heat rates and producer gas sensible heat at distinct loads are shown in Figure 5-24. At maximum flow rate, about 80% of high-temperature heat is lost to the environment, and so it is possible to recover it. Furthermore, 14% is used in the drying bucket and a smaller amount of 6% of heat at a lower temperature (<82°C) is dissipated to the environment.

As load decreases, both mass flow rate and temperature at the reactor outlet decrease as well. Consequently, the amount of the ratio of the heat dissipated between the reactor and cyclone outlet increases to a maximum of 66% at 30% load. On the other hand, the ratio of heat dissipated between the cyclone outlet and drying bucket inlet decreases. Nevertheless, the sum of both ratios increases, i.e. the ratio of high-temperature heat increases.

Figure 5-24 – Ratio between \( \dot{Q}_{\text{Cyc}} \), \( \dot{Q}_{\text{Cyc-DB}} \), \( \dot{Q}_{\text{DB}} \), \( \dot{Q}_{\text{DB-Filter}} \) and gas sensible heat at distinct loads.
5.1.10 Start-up

Start-up is part of a power plant operation routine. Furthermore, the number of start-up cycles is expected to increase to accommodate intermittent solar radiation behaviour. To develop a proper hybrid control scheme, it is essential to evaluate both cold and hot start-ups of the gasification system.

Start-up assessment encompasses the time required to attain desirable operation temperatures, i.e. restriction and reduction temperatures higher than 800°C and 650°C, respectively. Restriction temperature increases from ambient temperature to 800°C in about 6 minutes (see Figure 5-25), significantly quicker than reduction temperature. It takes about 12 minutes to attain a reduction temperature of 650°C. Furthermore, the lag between reduction and restriction temperature during the start-up increases with time during start-up. This behaviour is a consequence of the reduction reaction being sustained by restriction.

![Figure 5-25 – Restriction and reduction temperatures during cold start-up.](image)

It is interesting to see the similar start-up behaviour between the molar fraction of carbon monoxide and hydrogen and restriction and reduction temperatures, respectively (see Figure 5-26). As restriction temperature increases, biomass pyrolysis starts at the top of the reactor and results in charcoal and tar gas. The tar gases descend to the combustion and reduction zone, where thermal cracking occurs, which results in higher molar
fractions of carbon monoxide (about 56% for CO and 1.7% for H2 if stochiometric [44]). Additionally, tar is subjected to combustion in the reactor resulting in additional carbon monoxide.

The descendent gas flux assures that combustion heat is used to drive reduction. Nonetheless, the thermal inertia of the system (i.e. gasifier and biomass) delay the heating up process (see Figure 5-27). Concerning gas LHV, stabilisation is achieved at about the same time as restriction temperature reaches 800ºC.

Figure 5-26 – Producer gas LHV and CO and H2 molecular fractions during cold start-up.
For modelling purposes, it is ideal to describe gasification behaviour during start-up through an equation. Hybrid operation results in start-ups at distinct temperatures, and thus dimensionless temperatures for restriction and reduction were defined, as the ratio between the difference of actual and initial temperature and the difference between final and initial temperature (equation 5-21). An identical procedure was used to define dimensionless time (equation 5-22). Afterwards, a regression analysis was used.

\[
T^*_{\text{st-up}} = \frac{T_{\text{meas}} - T_{\text{init}}}{T_{\text{max}} - T_{\text{init}}} \quad (5-21)
\]

\[
t^*_{\text{st-up}} = \frac{t_{\text{meas}} - t_{\text{init}}}{t_{\text{max}} - t_{\text{init}}} \quad (5-22)
\]

Both restriction and reduction temperature behaviour during start-up are described through a polynomial fifth order equation (Figure 5-28). One can argue that both dimensionless temperatures should be related to the gasifier. Nevertheless, such approach would require a detailed knowledge of the gasifier design.
As abovementioned, hybrid operation requires start-up at distinct temperatures (i.e. hot start-up). Also, a hot start-up will not require external energy sources (e.g. propane gas torch). Tests were carried to evaluate start-up whilst the reactor is still hot. It was found that the lower restriction temperature to start-up the system without external energy requirement is about 300ºC (see Figure 5-29). At lower restriction temperatures, the cold gas flux further decreases temperature inside the reactor, jeopardising self-starting.

Furthermore, the empirical correlations for restriction and reduction temperatures were tested, and results show an excellent agreement between measured and calculated
restriction temperatures at each instant. Regarding reduction temperature, whilst both initial and final temperatures during start-up are well-estimated through the correlation, during the heating up process temperature is underestimated. Nonetheless, for modelling purposes only temperatures at the beginning and end of the start-up, as well as the time required are relevant.

![Figure 5-29 – Restriction and reduction temperatures during hot start-up at 300ºC.](image)

5.1.11 Cool-down

Cool-down behaviour assessment aims to characterise thermal inertia of the system, to account for the time that the system takes to cool-down, and also to understand how long the gasifier can be set at stand-by conditions, i.e. able to start-up without external energy sources. The main variables are restriction and reduction temperatures within the reactor (see Figure 5-30). Both reduction and restriction temperature follow an exponential curve type. Furthermore, both tend to the same value with time, as a consequence of reduction reaction dependence on restriction.
Following a similar process as the one defined for start-up, dimensionless temperatures and time were used. The main difference is related to the dimensionless temperature. During cool-down, reactor heat is lost to the environment, and thus the ambient temperature is relevant. The dimensionless temperature was defined as the ratio between the difference of actual temperature and ambient temperature and the difference between initial and ambient temperature (equation 5-23). Regression analyses for restriction and reduction temperatures are shown in Figure 5-22 a) and b), respectively.

\[ T^{*}_{\text{CD}} = \frac{T_{\text{meas}} - T_{\text{amb}}}{T_{\text{init}} - T_{\text{amb}}} \]  

(5-23)
5.1.12 Experimental issues

Whilst the value of experimental tests is undeniable, they are usually associated with a set of drawbacks that hinder experimental practice. In this section, the most serious issues and solutions are presented. For a more general information concerning small-scale gasification, the reader is referred to [3].

The main issue with the system was the IC engine that only worked for a couple of hours. Afterwards, the engine did not crank, although the gas energy quality was above average (see page 115). Therefore, the engine was checked. The check-up included
external valves (e.g. air/gas mixture valve), spark plugs and a compression test. It was found that the compression ratio was faulty in two cylinders, because of stuck engine valves. The solution implied to rebuild the engine cylinder head (see Figure 5-32). The process is time-consuming, especially without the proper tools and experience.

![Figure 5-32 – IC engine disassembly.](image)

After repairing the system was retested, once more without success. The reason for this failure was that the engine starter brushes were damaged (Figure 5-33). As initially the engine valves were stuck, the engine starter was subjected to several start-ups attempts at a rather high effort to move the engine. Consequently, the starter overheated, and brushes were damaged. Whilst replacing the starter brushes is not a tough task, to identify the cause and find the suitable brushes is. The engine starter was rebuilt. However, it did not solve the engine issues.
A newer engine check-up was carried out, and the compression rate was still lower than expected. To avoid opening the engine again, a less intrusive solution was tested. It consists in purging the engine by running it with liquid fuel to clean up possible condensate or tar residuals that are fouling engine components.

The procedure consists of connecting a purge unit to the engine (Figure 5-34-a), which requires several adaptations (e.g. remove oil sensor, electrical connections). Subsequently, the engine was started, as well as the fuel pump. The primary challenge is the manual setup of the air/fuel mixture (Figure 5-34-b). Preliminary tests resulted in a successful engine start-up, nevertheless unable to sustain operation for extended periods. During start-up, the engine temperature increases, and so it is necessary to reduce the amount of fuel. In this test, the proper balance of fuel and air was not attained, a thus a small explosion at the intake occurs, i.e. backfire.
Another consequence of running the system without the engine is the battery discharge. The system was designed to operate independently of the grid, and so the battery is used during start-up of the gasifier and engine, and afterwards recharged using the engine alternator. To solve this issue, individual component consumptions were evaluated and a 1kW 12-volt power supply installed. The electrical circuit was adapted to eliminate battery requirements. Currently, the battery is only used to crank the engine.

Notwithstanding that the assessment of combustion engine is not intended in this thesis, neither a crucial component to assess the gasification performance, the system was designed to operate with the engine. Moreover, operation at design condition and possible improved operation efficiency can only be attained using the engine-generator set. Furthermore, the generator can be used to power a variable resistance equipment (e.g. electric heater), and hence the gasifier load can be easily controlled by adjusting the power consumption of the equipment.

5.2 Solar Field Performance Assessment

In order to validate and tune-up the CSP numerical model (Chapter 4), reliable operation data were required. Consequently, a set of experimental tests in a DSG test facility was necessary. Correspondingly, the test facility should be adequately instrumented, including a wide range of sensors and transducers (e.g. flow meters, pressure and temperature sensors) installed throughout the facility. The only test facility that encompasses the aforementioned requisites is the DISS experimental plant in Plataforma Solar de Almería (PSA).

The DISS test facility was the first life-size experimental facility where DSG processes can be studied under real solar conditions (Figure 5-35). It was installed in the largest concentrating solar technology research, development and test centre in Europe, PSA. The test facility has been subjected to various updates during the years of operation [19, 45, 46]. Furthermore, the success of this work was dependent on the abundant availability of solar radiation, which was expected due to the prime location of PSA, in the desert of Tabernas (South of Spain).
The solar field performance evaluation under real-life context was carried out during an eight days period, under SFERA2 (Solar Facilities for the European Research Area) program, an EU-funded research project.

Experimental tests were carried out for both recirculation and once-through operation modes. The operation data were monitored and recorded, with a 5-second timestep. The data acquired on site include environmental variables (e.g. solar direct normal irradiance, ambient temperature), as well as the system variables (e.g. mass flow rate, pressure, temperature).

For each operation mode, system variables were measured throughout a one-day cycle at different inlet and outlet conditions (i.e. pressure and temperature). Tests were carried out to evaluate distinct operation stages including start-up, shut-down and steady conditions (design and off-design). In general, the experimental work was carried out according to the predefined control scheme of the DISS plant. However, some operation stages required manual intervention. Despite test campaign success, experimental issues occurred, e.g. a faulty collector 4 due to a communication error, and defective collector 6 sensors, which resulted in an additional model validation challenge. The following sections include a summary and presentation of the most relevant tests carried out, under OT and recirculation operation modes.

An additional output from this test campaign was the hands-on experience. The opportunity to co-operate the DISS loop proved to be a valuable practice and enrichment
experience, at both professional and personal level. Also, it was possible to understand operation difficulties and challenges, e.g. steam hammer, absorber tube overheating, the necessity to foresee the coming time as consequence of the system response lag time. New ideas to overcome such issues, such as the use of gasification heat surplus [48] are a yield from this test experience.

5.2.1 DISS layout and P&ID

A crucial work task was an onsite survey of the DISS facility. This task involved a thorough understanding of the operation and components of the DISS facility. A survey was conducted, through an assessment of both test facility and P&ID documentation, in order to collect all the relevant information concerning the main system components (e.g. solar collectors, pumps, steam-drum) and subcomponents (e.g. valves, fittings, insulation). The DISS facility is implemented at a commercial scale which outcomes in an extension of hundreds of meters of pipes and include a wide range of components. Furthermore, the many updates result in several documents. This survey also included facility instrumentation, where instrument type, location and error were assessed.

A schematic layout of the DISS test facility is shown in Figure 5-36. It is constituted by 16 parabolic trough collectors: 11 LS-3, 2 Eurotrough ET-100 and 3 Solarlite 4600+. Whilst the nominal length of collectors 1 to 8 and collector 11 is 50 meters, both collectors 9 and 10 have 25 meters. Collectors 0A, 0B, 1A, 1B and 12 have a nominal length of 100 meters. Concerning the nominal aperture area, LS-3 collectors have 276.5 m² (50-meter nominal length) and 138.3 m² (25-meter nominal length), whilst the ET-100 collectors have 553 m² and the SL4600+ 441.6 m² [49]. All the collectors were equipped with SCHOTT PTR-70-DSG receivers. In the scope of this study, tests were conducted without collector 12, as it was not necessary to attain desirable outlet conditions, and collector 4 was defocused due to a communication error.

At once-through operation mode, cold water from the Balance of Plant (BOP) enters the loop into collector 0A and is preheated, evaporated and superheated in the following collectors. Whilst the test facility includes several water injection lines at appropriate locations to either control loop outlet temperature or end of evaporation, only one has been used in this study, before collector 11 (see Figure 5-36). At the recirculation
operation concept, water is preheated and partially evaporated from collectors 0A to 8. Afterwards, water and steam content are separated at the steam drum. The water content is then recirculated, and the saturated steam is superheated in the following collectors (i.e. 9 to 12).

![Diagram](image)

**Figure 5-36 – DISS test facility layout.**

### 5.2.2 Once-through

One of the tests carried out under once-through operation mode, consists on attaining superheated steam by reducing step-by-step the mass flow rate, as shown in Figure 5-37. This test was carried out using collectors from 1A (inlet) to 11 (outlet) and without water injection. Initially, water is preheated and partially evaporated for a loop outlet pressure of about 32 bar. Afterwards, the mass flow rate is reduced and kept steady to assess the system response. At 13:48 the mass flow rate is set to 0.9 kg/s and outcomes in superheated steam after 13 minutes. A further mass flow rate reduction at 13:55, results in a higher superheating degree, i.e. 370°C after 20 minutes. These results are crucial for the numerical model tune-up, as shown in Chapter 6.

Figure 5-37 b) shows the pressure and temperature over the loop at 15:30. At this time, loop outlet conditions are superheated steam at 370°C. Within collector 1A, preheating takes place with temperature increasing from about 160°C to 250°C. Most of the loop extension is used for steam generation (i.e. collector 1B inlet to collector 6 outlet) where temperature variation is insignificant and a consequence of flow irreversibilities (i.e. pressure drop). Evaporation ending in collector 6 outlet is well noticed by the divergence between saturation and the measured temperature. It is noteworthy that in
Figure 5-37 b), the loop length is solely related to the collectors. Nonetheless, there are pipe connections amid the collectors that influence results. Between collector 8 outlet and collector 9 inlet, there is a significant distance, and thus outlet temperature and pressure of collector 8 (800 meters) are hindered by this distance. Afterwards steam is superheated, attaining 369ºC at collector 11 outlet.

![Figure 5-37](image)

**Figure 5-37** – Once-through operation mode at 30 bar: a) daily cycle overall results; b) pressure and temperature along the loop at 15:30.

A similar test was carried out for a steam outlet pressure of 60 bar, which required the use of collectors 0A and 0B (Figure 5-38). In this test temperature at the outlet is controlled by cold water injection at collector 11 inlet.
At 12:28 solar radiation was enough to sustain superheated steam at 350ºC in the loop outlet. At about 13:42 the flow rate was further reduced to 0.72 kg/s and 360ºC attained at 14:02. Nevertheless, direct normal irradiance was lower and less stable than in the previous test. Consequently, loop outlet temperature oscillations are higher. This issue could be avoided by using additional injection lines, i.e. control end of evaporation. Nonetheless, this implies additional variables in the test results and thus would enhance the model validation complexity.

Some of the outlet temperature variations are related to the operation, e.g. at 11:13 loop outlet temperature was about 360ºC; however, it was required to level up the water in the feedwater tank, and consequently, outlet conditions went down to saturated steam. Whilst at 11:47 outlet temperature started to increase, an error changed collector 0B from focused to stowed position for about 13 minutes, and therefore at 12:00 the loop outlet temperature started to decrease.

After 13:46 DNI started to decrease and a first small transient occurred at 14:15. Later, at 14:25 a set of clouds resulted in a significant radiation drop (down to 0). This set of transients extended until 14:43. During the transient mass flow rate was kept steady and minor changes were related to the system response itself. Steam superheating only restarted at 14:57 and about 12 minutes later the loop outlet temperature was 328ºC. Real-life conditions involve perturbations in the energy source, as transients from clouds. Prediction of the system behaviour under such conditions represents a challenging task. It is interesting, to verify how the model performs under these circumstances (Chapter 6).

This test was carried out with lower solar radiation and also at higher pressure, and thus temperatures. Consequently, heat losses are higher. E.g. at 14:10 (Figure 5-38-b) water preheating occurred in the loop first 300 meters (i.e. collector 0A inlet to collector 1A outlet), and superheating started at collector 7. Loop outlet temperature was attained through cold water injection at collector 10 outlet (850 meters), which can be noticed by a significant temperature drop. This considerable temperature decrease was not intended, and could be avoided by an improved control scheme (e.g. using water injection to control evaporation end). Nonetheless, for modelling validation purposes it was essential to minimise injection valve position changes. Otherwise, it would enhance validation complexity.
Figure 5.38 - Once-through operation mode at 60 bar: a) daily cycle overall results; b) pressure and temperature along the loop at 14:10.

5.2.3 Recirculation operation mode

One daily cycle under recirculation mode is presented in Figure 5-39 a). Outlet conditions were set to 60 bar and 400ºC, attained through automatic mass flow rate control and water injection. Start-up was carried out under manual mode for safety reasons, and at about 10:30 automatic feedwater mass flow rate control was settled. To increase loop outlet temperature, it was necessary to reduce the amount of steam in the superheating section. For this reason, it was necessary to progressively defocus five collectors in the evaporation section from 10:30 to 12:20.
Loop outlet temperature stabilisation was attained between 12:29 and 14:41, at about 412°C. During this period about 2/3 of the loop inlet mass flow rate was recirculated, i.e. steam quality around 0.33. Furthermore, steady conditions are noticeable by the same feedwater and recirculation mass flow rate profile (see Figure 5-39 b). The mass flow rate oscillations are related to the testing facility automatic control scheme, i.e. feed water and recirculation pumps depend on the pressure drop at control valves and level at the steam drum. For more details, the reader is addressed to [50].

Figure 5-39 - Recirculation operation mode at 60 bar and 400°C: a) DNI, loop inlet and outlet temperatures, and loop outlet pressure; b) feedwater and recirculation mass flow rate, and steam drum level.
The recirculation concept results in an additional pressure drop at the steam drum, i.e. after collector 8 (at 800 meters) as shown in Figure 5-39 c). The saturated steam from the drum entered in collector 9 and was superheated from 280ºC to 410ºC.

![Graph showing pressure and temperature along the loop at 13:00.]

**Figure 5-40 - Recirculation operation mode at 60 bar and 400ºC: pressure and temperature along the loop at 13:00.**

A similar test was carried out (Figure 5-41) for 350ºC at loop outlet. The main difference was the occurrence of solar radiation transients. Loop outlet temperature stabilisation was only achieved when the superheating pressure stabilised (i.e. 11:50). During the start-up, water injection was used to control the loop outlet temperature, and at 11:15 temperature dropped to saturation due to faulty control, i.e. excess of water. Whilst at 12:10 automatic loop level control was set, a significant set of clouds hindered stabilisation. After the first set of solar radiation transients, the outlet temperature raised to 400ºC. The second set of clouds delayed stabilisation until 13:50.
5.3 Conclusions

In this chapter, experimental work carried out in the scope of this PhD thesis was presented. The experimental work encompassed two tasks: evaluation under real-life conditions of a biomass gasification system and of a concentrating solar field.

Gasifier performance assessment was carried out using a small-scale gasifier system. A set of experimental tests were carried out, which included evaluation of the system performance at steady conditions for different loads (i.e. flow rates), and for distinct
operation stages including start-up and shut-down. The system was enhanced with a set of distinct and calibrated instruments (e.g. thermocouples, pressure sensors, flow meters) at relevant locations, and operation data were monitored and recorded, throughout the use of a data logger and a LabVIEW developed code.

Gasifier performance at steady state conditions showed relatively stable reduction and restriction temperatures, with average values of about 729°C and 915°C, respectively. These high temperatures assure a producer gas LHV of about 5.5 MJ/kg, with a CO and H2 average molar fraction of about 25% and 19%, respectively.

An average equivalence ratio of 0.26 and cold gas efficiency of about 69% was calculated for a gas energy flux of about 32 kWth. Decreasing the gas flow rate, resulted in a decrease of both reduction and restriction temperatures. Nonetheless, the gas LHV increased until 62.5% where a maximum of 6 MJ/kg was measured and afterwards decreased to 5.25MJ/kg.

Concerning the equivalence ratio, it decreased with load, attaining a minimum value at 80%, where the best cold gas efficiency results were found (74%). Subsequently, ER increased, and cold gas efficiency dropped to a minimum average value of about 56%.

The gas left the reactor at rather high temperatures which resulted in a hot gas efficiency of about 87%. A similar behaviour between hot and cold gas efficiency was found. Nevertheless, as the temperatures inside the reactor decrease, the producer gas outlet temperature decreases as well. Therefore, the gas sensible heat potential is lower. Most of the available heat (about 80%) is wasted to the environment at a rather high temperature, and so with good potential to be recovered.

During start-up, the restriction temperature threshold (800°C) was attained in about 6 minutes, whilst the reduction temperature evolution was slower (about 12 minutes to attain 650°C). This result is a consequence of the reduction chemical reaction being supported by restriction. Carbon monoxide and hydrogen molar content profile during start-up are similar to restriction and reduction temperatures, respectively. Initially, tar gases thermal cracking results in higher CO contents. As soon as reduction inertia is surpassed, the H2 content starts to increase. Regarding LHV, stabilisation is achieved at about the same time as restriction temperature threshold. Gasifier auto start-up (i.e. 145
without external energy source) requires a minimum threshold of 300ºC for restriction temperature.

Start-up restriction and reduction temperature behaviour were modelled through a 5th order polynomial equation and tested in a hot start-up. Results showed a good agreement between initial and final temperatures, but nevertheless, during the heating up process, the estimated and measured reduction temperatures differed. Cool-down restriction and reduction temperatures follow an exponential curve and were modelled accordingly.

Whilst modelling and simulating CSP with a single-phase fluid (e.g. thermal oil) is relatively simple, two-phase heat transfer and fluid flow in long horizontal pipes increase the modelling challenges and complexity. Therefore, a set of experimental tests at DISS test facility were carried out. The solar field performance evaluation under real-life was carried out for both recirculation and once-through operation modes. The operation data (i.e. environmental and system variables) were monitored and recorded, with a 5-second timestep.

An onsite survey of the DISS facility was successfully carried out, encompassing collection of all the relevant information concerning the system components and instrumentation.

The once-through operation was evaluated for outlet pressures of 30 and 60 bar. One of the tests consisted in attaining superheated steam by gradually reducing the feedwater mass flow rate, which permitted evaluation of the system response time to mass flow rate changes. Comparing pressure and temperature within the loop, it was possible to verify that at 30 bar most of the loop extension was used for evaporation. On the other hand, for a higher loop outlet pressure (i.e. 60 bar) and thus temperature, heat and pressure losses are higher and consequently the preheating section increases.

Loop outlet temperature control, through cold water injection before the last loop collector, was evaluated. Only one injection line was used to reduce the number of variables for the numerical model (i.e. decrease validation complexity). However, such option resulted in a non-intended significant temperature drop between the last two collectors.

The recirculation mode was evaluated at an outlet condition of 60 bar and 350ºC or 400ºC. To attain these temperatures, it was necessary to balance the mass flow rate of
saturated steam entering the superheating section. This control was attained by defocusing collectors in the evaporation section. After start-up, automatic control of the feedwater flow rate and steam drum level was set, which resulted in oscillations in the flow rate. Concerning pressure and temperature evolution along the loop, recirculation resulted in an additional pressure drop at the steam drum. Nevertheless, this is one of the main advantages of recirculation operation mode over once-through, i.e., a fixed end of the evaporation zone. It is important to mention that it is possible to define the end of evaporation under once-through operation, but it will require the use of additional injection lines.

For both operation modes, it was possible to test the system responder under real-life conditions such as perturbations in the energy source, i.e. solar radiation transients from clouds. Prediction of the system behaviour under such conditions represents a challenging task. It is interesting to verify how the model performs under these circumstances (Chapter 6). An additional output from the test campaign was the hands-on experience. The opportunity to co-operate the DISS loop proved to be a valuable practice and enrichment experience, at both professional and personal level. It was also possible to understand operation difficulties and challenges, e.g. steam hammer, absorber tube overheating, system response time. New ideas to overcome such issues, such as the use of gasification heat surplus [48] are a yield from this testing experience.

In summary, a successful test campaign was carried out, and adequate data were collected for the numerical model tuning and validation.

5.4 References


3. Rollinson, A. N. and Williams, O., Experiments on torrefied wood pellet: study by gasification and characterization for waste biomass to energy applications. Royal Society Open Science, 2016. 3(5)


Chapter 5: Experimental work


6 NUMERICAL MODEL VALIDATION

The hybrid CSP/Biomass assessment is carried out through the numerical model presented in Chapter 4. Thus, the accuracy of the results rely mostly on the numerical model, which includes simplifications and empirical correlations. Model complexity was increased by the inclusion of a two-phase fluid flow in the absorber pipes (DSG).

In order to ensure the model reliability, a test campaign was carried out at the DISS loop in Almeria (see Chapter 5) to collect operation data under real-life conditions. In this chapter, the numerical model performance is assessed against the experimental test results.

Initially, the model inputs are presented, including solar collectors and absorbers, as well as piping specifications. The numerical model accuracy was enhanced by splitting the collectors into a number of sections, however at the expense of computation time. To define the best compromise between the number of sections and simulation time, a sensitivity analysis was carried out, which consists on evaluating the model performance and the time needed to calculate the loop outlet temperature.

Afterwards, the model was tuned using experimental data from a test at 30 bar with once-through operation. The test entails the mass flow rate reduction within the loop step-by-step, until reaching superheating. This tuning-up process consists in assessing the loop
outlet temperature and pressure from both experimental and model results, by revising the following variables: overall collector efficiency through the cleanliness efficiency and the pipe roughness within the collectors’ connections. It is noteworthy that the use of the cleanliness efficiency is a numerical artifice, as it is not expected that adjacent collectors in the same loop have significant differences regarding their cleanliness.

Subsequently, the developed and tuned numerical model is used to evaluate the loop outlet temperature and pressure for both operation modes (i.e. once-through and recirculation). For each case, the assessment includes operation at distinct stages: start-up, cool-down, steady and transient solar radiation, and a comparison between the experimental and numerical results is carried out.

Although the comparison between the numerical model and experimental test results is confined to DSG, evaluation of the model performance for thermal oil and molten salts loops was carried out. This assessment entails a sensitivity analysis of the model results for different collector sections.

6.1 Model inputs

Initially, the DISS loop was modelled. This task included the setting-up of the technical specifications of the collectors, receiver pipes and piping connections within the loop. The DISS loop has been subjected to several updates over time, and so includes different collectors. The modified LS-3 (Luz System 3) collector was installed under the DISS project framework [1]; afterwards, in the scope of the INDITEP project [2], two additional Eurotrough ET-100 collectors were added; later, and within the DUKE project [3] the loop was extended by 300 meters, through the addition of three Solarlite SL4600+ collectors. A summary of the loop collectors’ technical specifications is presented in Table 6-1.

Additionally, under the DUKE project, the absorber tubes of all collectors were updated with SCHOTT PTR-70-DSG receivers. Feldhoff et al. [3] analysed the receiver tubes performance at the ThermoRec test bed and defined the length-specific heat loss regression (equation 6-1). The steel receiver tubes have an external diameter of 70 mm, a wall thickness of 5.6 mm, and a length of 4.06 m [4].
\[
\dot{Q}_{loss,coll} = I_{coll} (0.16155 T_{HTF} + 6.4407 \times 10^{-9} \times T_{HTF}^4)
\]  

(6-1)

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<th>Length [m]</th>
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<th>Nominal aperture area [m²]</th>
<th>Optical efficiency [%]</th>
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</tr>
<tr>
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<td>5.76</td>
<td>50</td>
<td>48</td>
<td>276.5</td>
<td>66.7%</td>
</tr>
</tbody>
</table>

The loop includes many pipe interconnections between the collectors, which vary over the loop and result in a significant steel and water/steam mass. For the connections between collectors 1 and 9, data from [6] were used. For the remaining loop, data from the P&ID collected during the test campaign were utilised.

6.2 Number of sections - sensitivity analysis

The solar field numerical model consists of a line focusing solar collector where an energy balance between incident beam radiation and the heat transfer fluid is calculated, and an indirect storage component is added to account for steel and water thermal inertia. The linear-focus solar collector’s length can exceed hundreds of meters, e.g. Flabeg’s Ultimate Trough [7] is about 250 m long. Moreover, the tendency regarding collectors’ size is to increase it, as it promotes cost reductions [8].

Concerning the DISS loop, the most recent collectors have a gross length of 100 meters. Thus, thermodynamic proprieties can change significantly over the collector
length, which will affect the numerical model precision. To improve the calculation performance, the collectors are split into a number of sections, and each section includes a solar collector and an indirect storage component.

On the other hand, both modelling and computational efforts increase. Regarding modelling, it is necessary to implement the algorithm to calculate the internal convective two-phase flow heat transfer coefficient for each IS component. Additionally, for each time-step, variables, thermodynamic properties, and also continuity, momentum and energy equations are solved, increasing the calculation time.

Accordingly, the number of collectors’ sections was evaluated by comparing the calculated loop outlet temperature with the measured one. As expected, the average relative error diminishes with the increase in the number of sections (see Figure 6-1), from 1.95% to 0.49%. Nevertheless, the computational time increases: if the collector is divided into fourteen sections, simulating one-hour operation would require about 13 hours of computation effort, although for one section only 15 minutes are needed.

![Figure 6-1](image)

**Figure 6-1 – Comparison of the loop outlet temperature average error and relative computational time for a distinct number of collector sections.**

The average relative error is not high, even for one section (i.e. 1.95%). For a loop outlet temperature of 400ºC, the absolute average error would be about ±7.8ºC, with an quick simulation time. Even so, the model will be used to evaluate the loop performance under hybrid operation, and so local errors are also relevant.
The most ambitious modelling task was the two-phase fluid flow and heat transfer, and thus evaporation tends to be the most likely zone for calculation errors. A comparison of the collector 7 outlet temperature under evaporation, and when superheating starts is shown in Figure 6-2, for one, ten and twenty collector sections.

Until about 15:00, the collector outlet temperature is constant with a slight reduction at about 14:00. Over this period, all the models overestimate the outlet temperature. Nevertheless, the maximum temperature difference measured was about 3.8ºC for one section. As expected, the temperature difference is attenuated by increasing the number of sections.

At about 15:00 steam superheating starts in the collector. Even though the calculated outlet temperature with the 10 and 20 sections model can predict well the measured temperature profile, the one section model fails: first superheating starts about 5 minutes later, and afterwards temperature rises about 32ºC instantly. The highest temperature difference over this period is equal to 23ºC.

As a consequence of the above analysis, and based on a compromise between model performance and modelling and computation effort, the collectors will be divided into ten sections for assessing specific hybridisation enhancements. Despite the compromise, about 52 hours are necessary for simulating 12-hours of operation. On the other hand, the same method would not permit a one-year assessment within a reasonable time. Therefore, for an annual analysis, only one section is used, and solar-field transient calculations are confined to solar radiation times, i.e. night operation is assessed on a steady state basis. It is noteworthy that the computation time (see Figure 6-1) is related to the DISS loop simulation which entails fifteen collectors. Therefore, for loops with a lower number of collectors, calculation performance will be enhanced. Furthermore, the simulation time-step is identical to the experimental data time-step (five seconds); on the other hand annual calculations are typically carried out on an hourly basis, or for better precision using a fifteen-minute time-step [9], and thus at a lower computational effort.
Figure 6-2 – Comparison of collector 7 outlet temperature under evaporation and superheating for one, ten and twenty sections.

6.3 Model tuning

The model was tuned using experimental data from the test campaign at DISS, specifically for operation under once-through mode at 30 bar. This tuning-up process consists on the evaluation of the loop outlet temperature and pressure from both experimental and model results. The revised variables were the overall collector efficiency through the cleanliness efficiency (see Table 6-2) and the pipe roughness within the collectors’ connections.
Table 6-2 – Collectors cleanliness efficiency [%] after tuning-up.

<table>
<thead>
<tr>
<th></th>
<th>0A</th>
<th>0B</th>
<th>1A</th>
<th>1B</th>
<th>1</th>
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<td>9A</td>
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</tbody>
</table>

The experimental test entails the mass flow rate reduction within the loop step-by-step, until reaching superheating. A comparison between the definitive version of the model and experimental results is shown in Figure 6-3. It is possible to see a similar behaviour from the model response to a mass flow rate reduction at about 14:00. Afterwards, a rather low superheating degree (about 40ºC) results in oscillations at the loop outlet temperature. Despite the model ability to reproduce this finding, the fluctuation pattern is different, and thus minor discrepancies between the measured and simulated outlet temperature are noticed. Further mass flow rate reduction results in a higher degree of superheating. Once more, the model predicts well the start of the temperature increase, as well as the temperature profile during heating up. Nonetheless, it tends to slightly overpredict the loop outlet temperature (i.e. about 5ºC). This difference can be related to the absorber heat loss regression (equation 6-1), that underestimates heat losses at higher temperatures.

Regarding the loop outlet pressure, there is an excellent agreement between the model and test results until 12:43. Afterwards, the mass flow rate reduction results in a hardly noticeable loop outlet pressure increase for about 10 minutes, followed by a pressure drop. Further mass flow reduction results in an additional loop outlet pressure drop. Over this period, model results diverge from the measured values. Initially, the pressure increases and a maximum difference of about 0.7 bar is noticed. Subsequently, the solar field outlet pressure begins to decrease. Once superheating starts, there is a good agreement between model and experimental pressure profiles. However, the model overpredicts pressure.

These discrepancies show that as the steam quality in the collector increases, the pressure model error increases as well. Furthermore, the calculation error influences the above mentioned temperature difference, that is, the calculated pressure drop over the loop is lower than the measured one, and thus evaporation occurs at a higher temperature. Hence, superheating also starts at a higher temperature.
Although the above-described results show a considerable agreement between model and experimental results, it is noteworthy that the model was tuned-up using the same results. So, it is crucial to evaluate how the numeric model performs for other input data. In the following sections model and experimental results are compared under once-through operation for 30 and 60 bar at distinct operation stages: start-up, cool-down, steady and unsteady solar radiation.

6.4 Once-through

Figure 6-3 - Experimental vs Model– Once-through - Superheating at 30 bar.
6.4.1 Start-up

During start-up, the cold-water temperature increases until evaporation temperature. The process consists of progressively focusing the collectors at the same time as more water is fed to the loop. A gradual heating up reduces the probability of technical issues, e.g. steam hammer. Regarding the loop outlet pressure, it increases as well, until reaching the predefined value. As a result of the low temperature of the whole loop (i.e. HTF and metal), thermal inertia plays a significant role.

Concerning modelling the most complex task is to set up the collector focusing process. The complexity is related to the lag between the operator’s instruction and the practical collector focusing. Thus, even when the focusing instruction is set, the collector can be entirely or partially focused. Within the model, the focusing status was defined for each collector using the collector angle (i.e. position) from measured data.

During start-up, the numerical model shows an excellent agreement with experimental data (see Figure 6-4). Over the heating up process, the mass flow rate increases in a non-linear manner, and some swift upsurges are noticed. This output is a consequence of the operator commands, and mostly related to the BOP control (e.g. control of the drums water level). Even so, the effect of the mass flow rate oscillations in the loop outlet temperature and pressure is well described by the numerical calculations.

After evaporation stabilisation (i.e. at about 10:15), more water is fed into the solar field at the same time as solar radiation increases. Minor mass flow rate changes (e.g. 10:49) affect the loop outlet temperature and pressure during a specific time. The model can describe well this behaviour. Still, it is possible to notice that after 10:50 the calculated outlet pressure is higher than the measured one, which will also influence the loop outlet temperature. This behaviour is like the one observed in section 6.3 and is related to the model performance at a higher steam content and temperature.
6.4.2 Cool-down

During cool-down, the temperature within the loop is reduced by gradually defocusing the collectors and reducing mass flow rate. As the collectors are not defocused all at once, neither instantaneously, it was necessary to define the defocusing process of each collector individually, using the collector rotation angle. This process was carried out by an identical method as the one used for the focusing process (see section 6.4.1). Simulation and measured results of the loop outlet temperature and pressure over a cool-down period at 30 bar are shown in Figure 6-5.
Concerning loop outlet temperature, the model overestimates the value by about 5°C before the cool-down process starts. Subsequently, as the mass flow rate is reduced, the temperature difference between simulation and experimental values decreases. Furthermore, during this period there are times when the model calculated temperature is lower than the measured. In any case, the model is able to describe the temperature behaviour through cool-down. Regarding the loop outlet pressure, an analogue behaviour is found. The model overestimates before the mass flow rate reduction, and subsequently discrepancies are lower.

Figure 6-5 - Experimental vs Model – Once-through – Cool-down at 30 bar.
6.4.3 Constant solar radiation

In general, the numerical model can accurately predict the solar field behaviour over start-up and cool-down. However, during evaporation, as the steam quality at the loop outlet increases, the pressure model diverges from experimental measurements, increasing the loop outlet temperature.

At higher pressures (i.e. 60 bar) and under stable solar radiation, the simulated outlet pressure is about 0.5 bar higher, which results in a good agreement regarding the loop outlet temperature (see Figure 6-6). During this test, water is being injected into the loop last collector inlet. As the injection control is manual, the loop outlet temperature decreases about 30ºC between 12:30 and 13:45.

Over this period, the simulated outlet temperature reduction is only 15ºC until stabilisation at 13:22. This divergence is an outcome of the inadequate injection modelling. In the injection line, it is necessary to define mass flow rate, pressure and temperature, variables that were not measured; experimental results are confined to the valve opening position and pressure in the injection pipe.

At 13:40, the mass flow rate is further reduced to increase the loop outlet temperature, which results in a numerical model deviation from experimental results. This discrepancy is justified by the different temperatures when the mass flow rate is reduced, as well as by the distinct regulation of the injection line.
6.4.4 Transient solar radiation

Real-life conditions involve perturbations in the energy source, such as short-transients from clouds, or in the plant itself (e.g. dust), which were expected to occur during the test campaign. Prediction of the system behaviour under such conditions is a challenging task.

Figure 6-7 shows a comparison between experimental data and simulation results (i.e. loop outlet pressure and temperature) for once-through operation mode at 60 bar, under a significant radiation transient. At about 14:25 a set of clouds results in a considerable drop in DNI from about 830 W/m² to 0 W/m². This set of solar radiation transient continues for about 20 minutes. At this stage system control was ineffective and as a result
the loop outlet temperature and pressure decrease, from superheated to saturated steam. The numerical model response to the transient is similar to the measured data, with a considerable agreement until 14:45.

After the set of radiation transients, the solar field starts to heat up again, and inconsistencies between measured data and simulation results tend to be more significant. The model predicts an early end of the evaporation stage, and consequently superheating starts earlier. This outcome is mostly related to the discrepancy between measured and simulated pressure drop over the loop. The differences are further augmented during superheating stage. Whilst the model predicts an earlier start of steam superheating, all other inputs (DNI, loop and injection mass flow rate, inlet pressure) are kept identical to the measured data. Therefore, the model overestimates the loop outlet temperature by about 13ºC.

Figure 6-7 - Experimental vs Model– Once-through – Transient solar radiation at 60 bar.
Moreover, a thorough result analysis shows that during this period the model predicts that evaporation ends in a different collector, compared to measured data (see Figure 6-8): although evaporation ends at collector 6 outlet (700 m), the model foresees that superheating starts at collector 6 inlet (650 m).

![Figure 6-8 – Temperature over the loop at 15:00.](image)

6.5 Recirculation

Operation under the recirculation concept implies the additional modelling of the steam drum. On the other hand, the end of evaporation is well defined, and superheating is confined to collectors 9, 10 and 11. In comparison with once-through, an added input is required: the recirculation mass flow rate. The assessment of the numerical model and experimental results was carried out for a loop outlet pressure of 60 bar and operation during start-up, steady and transient solar radiation, and cool-down.

6.5.1 Start-up

The start-up procedure is similar to the one described for once-through operation (see section 6.4.1). At the beginning of the day, the steam drum is full. Therefore, both recirculation and feedwater pumps are used to set up the level. Both mass flow rates are controlled through the use of valves. The control concept consists on limiting the recirculation pump outlet pressure, at the same time as assuring a minimum mass flow rate of 0.55 kg/s.

Experimental and modelling loop outlet temperature and pressure are shown in Figure 6-9. At about 09:22 the recirculation mass flow rate is lower than 0.55kg/s, which results
in the collectors stowed from automatic control. The mass flow rate reduction is a consequence of the pressure at the steam drum, i.e. as the pressure in the drum increases, the recirculation pump outlet pressure increases as well. For safety reasons, the difference between the recirculation pump outlet pressure and the loop inlet pressure is limited. If the limit is surpassed, the collectors are automatically moved to stow position and the recirculation mass flow rate reduced. Nonetheless, at about 09:33 the system starts to recover, and the recirculation mass flow rate starts to increase again, at the same time as the collectors within the evaporation section are focused.

![Image of graphs showing experimental vs model recirculation start-up at 60 bar]

Figure 6-9 - Experimental vs Model – Recirculation – Start-up at 60 bar.

At 09:45, the recirculation flow rate is kept constant and the feedwater flow rate reduced. Simultaneously, the collectors within the superheating section (i.e. collectors 9, 10 and 11) are focused. The outcome is noticed in the loop outlet temperature at 09:47,
as it begins to increase more significantly. Over this period the model overpredicts both loop outlet pressure and temperature, with a maximum difference of 2.6 bar and 10ºC, respectively.

6.5.2 Constant solar radiation

A test carried out under almost constant solar radiation consists in increasing the steam outlet temperature by reducing the feedwater mass flow rate (see Figure 6-10), whilst keeping the recirculation mass flow rate constant. The lower feedwater flow rate affects the recirculation rate and so the steam quality at the drum. Consequently, during this period some of the collectors within the evaporation zone were defocused as needed. As a result, less steam is superheated, and the outlet temperature is higher.

Figure 6-10 - Experimental vs. Model – Recirculation – Steady solar radiation at 60 bar.
Comparing the experimental and numerical results, it is noticeable that in general, the numerical model overestimates the loop outlet pressure and temperature. On the other hand, the model is able to describe the effect of the feedwater flow rate reduction and the collector defocusing process.

6.5.3 Transient solar radiation

One of the numeric modelling most challenging tasks is to accurately describe the solar field behaviour under transient solar radiation. Until 12:23 the solar radiation is almost steady (see Figure 6-11), and the difference between measured and simulated loop outlet temperature is mostly related to faulty water injection modelling. Also, during this period the pressure drop over the loop is underestimated.

Afterwards, a set of clouds reduces solar radiation to values lower than 100 W/m². This set of radiation transients extends until 12:40. Firstly, both feedwater and recirculation mass flow rates are augmented to prevent absorber dry out. Subsequently, the recirculation flow rate is reduced to the start value. During the transient period, initially the loop outlet pressure falls by about 7.5 bar, with a good agreement from the model. Afterwards, the pressure recovers to the initial state. Regarding the loop outlet temperature, it follows a similar behaviour with an initial reduction (about 28°C) followed by a recovery period.

The model can describe the loop outlet temperature accurately during the reduction and recovery period. Nevertheless, there is a mismatch when the recovering period ends, which could be related to the injection modelling. Subsequently, the excess of the water fed into the loop during the transient period results in a reduction of the loop outlet temperature until steam saturation, which continues for about 10 minutes. When the loop temperature starts to increase again, another set of clouds hinders the readjustment. Thus, more water is fed into the loop and stabilisation is delayed until 14:50.
6.5.4 Cool-down

The cool-down process under recirculation operation entails first the defocusing the collectors in the superheating section. Accordingly, the loop outlet temperature and pressure are reduced. Afterwards, feedwater and recirculation mass flow rates are augmented, and as the loop outlet pressure decreases, the collectors in the evaporation section are gradually defocused (see Figure 6-12).

As soon as all the collectors within the loop are defocused, the recirculation flow rate is reduced, and cold feedwater supplied into the loop. Despite the initial overestimation, throughout the cool-down course model results (i.e. loop outlet pressure and temperature) are well matched to experimental ones.
The test campaign only encompassed tests with water/steam as heat transfer fluid. However, the model will be used to evaluate CSP performance with other HTFs (i.e. thermal oil and molten salts). These HTFs are single-phase fluids, and accordingly, the model complexity associated with the two-phase fluid flow heat transfer does not exist. Therefore, the calculation sensitivity to the number of sections (see section 6.2) should be lower, and so modelling and calculation effort reduced.

A similar methodology as the one used for the sensitivity analysis of the number of collector sections was used to verify and define the adequate model for thermal oil and
molten salts. The main difference is related to the absence of experimental results, i.e. the model results for distinct sections are compared against each other.

This assessment requires the definition of a solar field loop for thermal oil and molten salts. In both cases, the loop consists of four Eurotrough ET-150 collectors (see Table 6-3) with a SCHOOT PTR70 absorber tube. Connections between the collectors are disregarded. For the absorber heat losses, data from [10] were used to define equation 4-22 coefficients. It is worth to mention that the data used are related to an older SCHOOT PTR70 generation, which was not designed for molten salts. SCHOOT PTR70 has been subjected to developments over time, and only at the fourth-generation was intended for the use of molten salt heat transfer fluid. Nevertheless, the main changes are a new steel grade that permits 550ºC and a novel absorber coating [11]. According to the author’s knowledge, there is no published information concerning the performance of the fourth-generation receiver, and so older data were used for this assessment [12].

Table 6-3- Eurotrough ET-150 technical specifications. Adapted from [12].

<table>
<thead>
<tr>
<th>Aperture width [m]</th>
<th>Length [m]</th>
<th>Nominal Length [m]</th>
<th>Nominal aperture area [m²]</th>
<th>Optical efficiency [%]</th>
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<td>150</td>
<td>142</td>
<td>817.4</td>
<td>75%</td>
</tr>
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</table>

Regarding the HTFs, Therminol VP-1 [13] and the eutectic mixture of 60% sodium nitrate (NaNO₃) and 40% potassium nitrate (KNO₃) [14], like the one used in the Archimede Solar Energy molten salt demo solar plant [15], were considered.

Regarding the loop, operation temperatures and mass flow rates are different (see Figure 6-13). For thermal oil, at design conditions, a mass flow rate of 7.2 kg/s was defined which results in a loop outlet temperature of about 395ºC, below the technical operational limit (400ºC). Concerning molten salts, a lower flow rate (4.62 kg/s) is required to attain about 550ºC at the loop outlet.
In both cases, the maximum errors are expected to occur at the loop outlet, where the temperatures are higher and consequently heat losses more significant. Furthermore, the results are affected by calculation errors for other collectors, i.e. propagation errors. Concerning the pressure drop, it will only influence the pumps parasitic consumption, as both fluids are considered incompressible.

The assessment consists on evaluating the loop outlet temperature response for a mass flow rate reduction under quasi-steady solar radiation. For thermal oil, a comparison between one and ten sections is shown in Figure 6-14. The agreement between both results is excellent with a most significant temperature difference of 0.26°C. Consequently, for thermal oil it is possible to precisely model the solar collectors with just one section. Thus, less than six minutes are required to simulate one-hour operation using a five seconds time-step.

Concerning molten salts, an identical comparison was carried out, as shown in Figure 6-15. Nonetheless, the differences between the loop outlet temperature for one and ten
sections are higher than the ones verified for thermal oil. The maximum discrepancy is about 1.4°C and occurs during the heating-up phase (highlighted in Figure 6-15).

![Figure 6-15 – Molten salts loop outlet temperature for one and ten collector sections.](image)

In order to evaluate the loop outlet temperature sensitivity to the number of sections in each collector, similar tests were carried out for two and four sections. A comparison between the maximum and average temperature difference related to the ten sections case, as well as simulation relative time, are shown in Figure 6-16.

![Figure 6-16 - Comparison of the loop outlet temperature maximum and average difference and relative computational time for a distinct number of collectors’ sections.](image)
By adding one section, the maximum difference is reduced by more than 50%, though the computation time increases about 81%. In the case of four sections, the highest temperature difference is reduced in about 85% from the one section case, yet with a significant increase in the computation time (350%). Nevertheless, the most significant difference is rather low, considering the model simplifications and empirical correlations. Moreover, the average temperature difference is below 1ºC. So, for annual simulations, the collectors will entail only one section, and for assessing specific hybridisation enhancements, four sections will be considered.

6.7 Conclusions

In this chapter, the developed numerical model (Chapter 4) performance was assessed against experimental results from a test campaign carried out at the DISS loop (Chapter 5). This evaluation is justified since the hybrid CSP/biomass assessment hinges on the numerical model which comprises assumptions and empirical correlations.

The collector field numerical model consists of a line focusing solar collector and an indirect storage component. As the linear-focus solar collector’s length can exceed hundreds of meters, properties can change significantly within the collector. So, a method of splitting the collectors into a number of sections was applied. A sensitivity analysis was carried out to evaluate the effect of the number of sections on the loop outlet temperature, by the calculation of the average and maximum loop outlet temperature errors, as well as the computational time.

Results showed that a higher number of sections improves the model performance at the expense of simulation time. Thus, for DSG specific hybridisation enhancements each collector will be split into ten sections, whilst for annual simulations only one section will be considered. The average error for one and ten sections are below 2% and 1.3%, respectively. On the other hand, the highest temperature difference verified was of about 23ºC for one section. Regarding the simulation time, about 52 hours are necessary to simulate 12 hours of operation, with the ten sections model. Nevertheless, this outcome is associated with a reduced time-step (five-seconds) and with the DISS loop which encompasses 15 collectors.
Although the comparison between the numerical model and experimental test results is confined to DSG, a sensitivity analysis concerning the number of sections was carried out for a thermal oil and a molten salts loop. In the case of thermal oil, no significant differences (maximum 0.26ºC) were noticed between one and ten sections.

On the other hand, for molten salt, differences are higher (1.4ºC) during the heating-up phase. Therefore, for molten salts, four sections are used to describe the hybrid system behaviour for specific enhancements and one section for annual simulations.

The numerical model performance was evaluated against experimental tests, under once-through and recirculation operation for pressures of 30 and 60 bar, and at distinct operation stages: start-up, cool-down, steady and unsteady solar radiation.

In general, it is possible to conclude that the model can predict the solar field behaviour with high accuracy. Nevertheless, sporadically the model incorrectly foresees the loop outlet results, and it was found that it tends to overpredict the loop outlet temperatures, especially at higher temperatures (i.e. pressures). This outcome can be related to the two-phase fluid flow heat transfer model, model inputs and simplifications, injection modelling, measurement errors, and not considering the BOP in the model.

The two-phase fluid flow heat transfer model comprises empirical correlations for the pressure drop and heat transfer coefficient calculation. Thus, they are limited to ideal operation conditions, e.g. uniform evaporation. Also, during evaporation pressure and temperature are bounded, and so pressure drop calculation errors affect the loop outlet temperature. The model inputs include the receiver tube regression, which can result in systematic errors, as underpredicting the heat losses at higher temperatures [3]. Furthermore, the solar collector numerical model disregards some optical parameters, e.g. collector acceptance angle, which may affect the output. Furthermore, although the pipe insulation and geometry have been considered according to [6] and the DISS loop P&ID, the model does not entail irregularities, e.g. pressure losses at pipe bending or lower insulation thickness in specific zones.

Some of the differences are related to the unsuccessful injection modelling. The measured data are not enough to accurately model the water injection. It was a wrong decision to run the DISS loop with injection, as the tests were intended for numerical model validation. Additionally, it is possible that part of the deviations concerns
experimental measurement errors. Another likely source is the effect of not considering, within the model, the HTF and steel mass in the BOP of the DISS power plant, which extends over hundreds of meters and includes many components, e.g. drums.

Different solutions could be applied to solve these discrepancies. One is to tune up the model for the DISS plant, but that would result in a non-replicable model for other solar loops. Another solution would be the fine-tuning of the model at the expense of both modelling and computational effort. It is noteworthy that despite model simplifications, it results in a significant computation and modelling effort.

Regarding the modelling and simulation time, the software code can be improved. For example, for the time-series calculations, it should be possible to perform the simulations in the background and thus reduce the computation graphic effort. Regarding modelling, the software misses some features in the programming area, such as generic pointers that could be useful to model and replicate a collector section (i.e. line focusing solar collector and an indirect storage component).

Despite the discrepancies above mentioned, and the modelling and computation effort, the numerical model performance can be generally considered as very good. It can accurately predict the loop outlet temperature, and pressure response to mass flow rate changes in the solar field, and solar radiation transients, which is crucial for assessing specific hybridisation enhancements.

6.8 References

1. Eck, M., et al., Direct Steam Generation in parabolic troughs at 500°C - first results of the REAL-DISS project. 2011.


7 ASSESSMENT OF CSP/BIOMASS HYBRID PLANTS – CASE STUDIES

In this chapter, hybridisation of concentrating solar power with biomass is assessed, through case studies. To begin with, the REELCOOP prototype 3 is presented and evaluated. It contains a parabolic trough solar field with DSG and an anaerobic digestion system, driving an ORC power cycle. The study is extended using a scaled-up and enhancement of the REELCOOP prototype, where the possibility of CHP and the economic performance are evaluated.

Subsequently, specific advantages of hybridisation for a water/steam HTF solar field are addressed. This includes both operation under once-through and recirculation concepts, considering two types of collectors: Parabolic through and linear Fresnel reflector collectors. Regarding biomass conversion, both anaerobic digestion and gasification are considered.

Likewise, specific hybridisation enhancements are assessed for a single-phase heat transfer fluid (i.e. thermal oil and molten salts) solar field. In this case, hybridisation is attained within the HTF and steam cycles by combustion and gasification.
Note that the case studies are limited to a moderately lower electrical output (maximum 5 MWel). It is well known that CSP plants are designed in the range of 100 MWel taking advantage of the economy of scales. On the other hand, the hybridisation novelty enhances the financial risk, and can thus be better suited for a lower scale. Also, the economic viability of biomass plants demands a location in proximity to feedstock sources. By increasing the proximity between the CSP/biomass hybrid power plants and feedstock sources, a lower power demand is expected. For CSP this could result in a higher deployment rate and hence lower costs. Furthermore, in case of a scale-ups, the assessment results will be improved both technically and economically.

7.1 Dispatchability, stabilisation, efficiency

In this section, a CSP/biomass hybrid mini power plant is presented and assessed. Hybridisation is attained by placing the solar field and boiler in parallel. Annual simulations were carried out for solar-only and hybrid modes. Distinct operation ranges and boiler sizes were analysed. Simulation results are presented, such as solar field annual generated heat and efficiency, boiler efficiency and biogas consumption, annual generated electrical energy and ORC efficiency, dumped heat, solar and biomass shares, and global system efficiency. Hourly results are presented for standard days, with and without hybridisation, showing the advantages of hybridisation.

7.1.1 The hybrid mini power plant

Within the REELCOOP project framework, co-funded by the EU [1], a mini CSP/biomass power plant prototype was installed in Tunis. The prototype consists of a solar field and a biogas steam boiler to drive an organic Rankine cycle.

The solar field relies on parabolic trough collector technology and is constituted by three parallel loops of four PTMx/hp-36 collectors developed by Soltigua [2], with a net collecting surface of 984 m². Direct steam generation is achieved in the solar field, and the recirculation concept was adopted. The solar field is supplied with subcooled water, and partial evaporation takes place within the solar collectors. The water/steam mixture is then separated in a steam drum, and therefore only saturated steam leaves the solar
field. The leftover water is then recirculated. Complete evaporation enhances control complexity, implying unnecessary risks over the solar collector absorbers [3].

A biogas steam boiler provides auxiliary energy. The biogas is produced by anaerobic digestion of canteen organic waste remains, showing a potential solution for the problem that waste disposal represents [1]. The system layout (Figure 7-1) consists in placing the solar field and the biomass boiler in parallel, allowing thermal generation to be achieved either from solar energy or biomass (i.e. solar-only or biogas-only), or the combination of both.

In order to reduce thermal energy waste as well as biogas consumption, and to compensate short transients from solar power, a storage tank was foreseen in the project. Since the storage tank will be charged with saturated steam from the solar field, an isothermal latent thermal energy storage concept has been adopted. Whilst typical thermal energy storage systems concern sensible heat storage with temperature change, the latent heat solution uses Phase Change Materials (PCM). Nevertheless, PCMs for low temperatures are scarce and present stability issues [4].

The hybrid mini power plant (Figure 7-1) has a nominal electrical output of 60 kW and relies on a regenerative ORC as power generation system, developed by Zuccato Energia [5]. The turbine/generator block was adapted to assure operation at partial load, in order to compensate solar energy fluctuations, with a nominal gross efficiency reaching 13.3%. Saturated steam at 170°C will drive the ORC, which allows the power circuit to also operate with available waste heat.
Figure 7-1 - REELCOOP CSP/biomass mini power plant. Reprinted from [3].

7.1.2 Simulation assumptions

The simplified system simulation scheme is presented in Figure 7-2. The solar field model includes the sun, parabolic trough collectors, distributing and collecting headers, and recirculating and feed water pumps. The collectors and piping were modelled according to the manufacturer information and prototype P&ID. The reader can refer to Chapter 4 for more details concerning the numerical model.

The model requires as input meteorological data which were obtained using Meteonorm software for the prototype site location (Tunis, Tunisia), on an hourly basis for a typical meteorological year. The annual solar radiation is about 1799 kWh/m².
Hybridisation was modelled through a heat injection component, representative of the boiler. This component acts as an ideal heat exchanger, promoting the interface between the boiler output and the mass flow rate of water/steam. In order to control the boiler output, a code was created using EbScript to impose operating limits according to the manufacturer data, as well as to calculate the required heat flow. The boiler output is controlled through the water/steam mass flow rate, acting as an auxiliary heater of the solar field.

For simulation purposes, the Viessmann VITOMAX 200-HS boiler model [6] with economiser was used. Two different boiler sizes were the object of analysis, with nominal heat outputs of 380 kWth (0.5 ton/hr of saturated steam) and 530 kWth (0.7 ton/hr of saturated steam), and both with a minimum thermal output of 100 kWth. In order to account for the transient behaviour of the boiler, an IS component was used. As inputs, the water volume of the boiler was used, and the mass of steel estimated, considering that the boiler takes half an hour from cold start to design conditions. For estimating biogas consumption, an additional computer model was created using Ebsilon, mainly constituted by a combustion chamber and two heat exchangers (steam evaporator and economiser), as shown in Figure 7-3.
The model uses the water/steam mass flow rate, as well as the inlet and outlet enthalpies to calculate the required boiler output. These inputs were obtained from the thermal generation system model results.

The boiler efficiency, combustion heat output and flue gas temperature are determined as a function of the ratio between the boiler output and the rated output, considering the manufacturer data (see Figure 7-4). In the combustion chamber, the combustion of the mix of air and biogas is modelled, considering a 3% oxygen excess in the flue gas. As output, it retrieves the biogas mass flow rate. Biogas consumption was estimated considering a yearly average lower heating value (19.27 MJ/m$^3$).
7.1.3 Biogas from food waste

The boiler runs on biogas obtained through anaerobic digestion of local organic waste (university canteen food waste). The use of local organic waste is justified by the associated low cost and forward-looking environmental consequences related to waste disposal.

A simple scheme of a biogas plant for prototype 3 hybridisation is represented in Figure 7-5. The process starts with feeding the digester with organic matter. In the absence of air, a biochemical process (anaerobic digestion) starts, from which two products are obtained: biogas and digestate. Since AD is a biochemical process, biogas production is rarely constant, due to different feedstock properties and operating conditions. Therefore, a buffer (gasometer) is required for biogas storage to compensate the fluctuations, ensuring constant flow at the boiler. In the last stage, biogas is converted into heat by using a boiler.
An analysis of locally available biomass resources was carried out (Table 7-1). Note that the presented values are estimates since they can vary day by day due to changes in canteen organic waste. Regarding the available residues, the canteen provides a daily average of 350 kg of organic waste. Since the canteen will operate at least 200 days a year, about 70 tons of organic waste are available per year.

**Table 7-1 - Organic Waste Availability and properties.**

<table>
<thead>
<tr>
<th>Daily Organic Waste Quantity - Mass</th>
<th>350 kg/day</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density of Organic Waste [OW]</td>
<td>0.7 tons/m³</td>
</tr>
<tr>
<td>Daily Organic Waste Quantity - Volume</td>
<td>0.5 m³/day</td>
</tr>
<tr>
<td>Total Solids Fraction [%TS]</td>
<td>21%</td>
</tr>
<tr>
<td>Volatile Solids Fraction [%VS]</td>
<td>63% %TS</td>
</tr>
<tr>
<td>Biogas Yield</td>
<td>0.615 m³/kg of VS</td>
</tr>
</tbody>
</table>

Considering the available organic waste, as well as the previously presented properties (Table 7-1), it is possible to estimate the daily biogas production, through the specific gas yield and the daily input of volatile solids (VS) in the substrate,

\[
\dot{V}_{biogas} = \dot{m}_{VS} \times Biogas\_yield
\]  

\[\text{( 7-1 )}\]
With the available canteen residues, it is possible to produce roughly 30 m$^3$ of biogas per day. A study conducted in the REELCOOP project shows that it is possible to increase daily biogas production to about 60 m$^3$/day, using five digesters in parallel.

As an auxiliary system, the biogas system should be able to supply enough energy to assure the proper operation of the ORC. Two different working conditions (nominal and minimum power) were analysed in order to check the system viability. The ORC at nominal power requires 450 kW$_{th}$ of thermal energy to produce 66.3 kW$_{el}$ of gross electrical power, with 15% efficiency. At minimum power the ORC can operate at 38% of nominal power (25 kW$_{el}$ gross power), with an efficiency of 11%, requiring 220 kW$_{th}$ of thermal energy.

Table 7-2 shows the biogas flow rate needed to achieve the ORC requirements, considering a boiler efficiency of 85% and a biogas lower heating value of 5.5 kWh per cubic meter.

<table>
<thead>
<tr>
<th>Description</th>
<th>Thermal Output [kW]</th>
<th>Boiler Efficiency [%]</th>
<th>Thermal Input [kW]</th>
<th>LHV biogas [kWh/m$^3$]</th>
<th>$V_{biogas}$[m$^3$/h]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal Power</td>
<td>450</td>
<td>85%</td>
<td>530</td>
<td>5.5</td>
<td>97</td>
</tr>
<tr>
<td>Minimum Power</td>
<td>220</td>
<td>85%</td>
<td>258</td>
<td>5.5</td>
<td>47</td>
</tr>
</tbody>
</table>

The amounts of gas needed to run the ORC, for different running times and loads, are presented in Figure 2. The system needs about 281 m$^3$ to 561 m$^3$ of biogas to run the ORC at minimum power, during 6 to 12 hours respectively. The ratio between biogas production and boiler demand is highly unbalanced. However, it is possible to overcome this issue by adding local existing feedstock sources. For this assessment, it was considered that enough residues were available.
7.1.4 Organic Rankine Cycle

The regenerative Organic Rankine Cycle was simulated using EES, since property data of the organic fluid (Solvay SES36) were not available neither in EES nor in Ebsilon databases. The data values were introduced into tables from which the thermodynamic properties (e.g. enthalpy, entropy) are calculated by linear interpolation.

It was assumed that the ORC operates at steady-state, with the thermal inertia of the power block neglected. Furthermore, the evaporation and condensation temperatures are considered constant, as well as the pump efficiency and the temperature difference in the regenerator.

As input, the code requires the saturated steam mass flow rate obtained from the thermal generation system simulation results. The steam mass flow rate is used to calculate the isentropic turbine efficiency, by means of a second order equation based on manufacturer data. Within the ORC model, calculations for individual components (i.e. turbine, pump and heat exchangers) are achieved by applying the first law of thermodynamics. Irreversibilities are considered at the turbine and pump. As main output results, the code provides the gross and net electrical power, organic fluid mass flow rate, as well as the parasitic consumption and the condenser thermal requirements. Moreover, the developed model can be used as a standalone program, with a visual interface where
the ORC efficiency and output power, in addition to relevant cycle thermodynamic properties, are estimated (see Figure 7-7).

7.1.5 Operation Modes

To optimise the system operation and to enhance similarity with the real system operation, the thermal generation system model is controlled, and also distinct operation profiles are considered. Ebsilon contains a wide-range of components suitable for control purposes, such as property value transmitters, calculators and simple linear controllers. Nevertheless, more complex control tasks, such as interchange between operation profiles, require the development and implementation of a code. Therefore, a computational code was developed, using a Pascal-based language and implemented in the numerical model using the EbScript tool.

7.1.6 Solar-only operation mode

The system steam production is controlled through mass flow balance at the steam drum. The same concept was used in the simulation model. During system operation, the recirculating mass flow rate is kept constant at about 0.5 kg/s. In real conditions, the feed
water pump will operate when the water level in the steam drum drops below a predefined level. Since the simulations were carried out on an hourly basis, it was assumed that the feed water mass flow rate should balance the water removed from the recirculating pump, and so is intrinsically related to the steam quality after the distributing header.

If the saturated steam mass flow rate that leaves the solar field exceeds the maximum requirements of the ORC, it is then separated using a simple splitter (see Figure 7-2). In real operation, the solar collectors should change their state to partial defocused to control the steam production. In the simulation model, the excess of energy is accounted as dumped, providing an input for a PCM storage tank design. The last separation occurs before the ORC, through a water-steam drainer. This component is mostly used in the warm-up and cool-down profiles, to establish a more realistic thermodynamic balance.

To obtain a more accurate approach in the simulation model, distinct operating profiles were created. A code was developed using EbScript to allow the automatic interchange between the profiles using dynamic variables, e.g. direct normal irradiance, hour of the day, mass flow rates, and so forth. Four profiles constitute the solar-only operating mode: warm-up, operating, cool-down and stop.

At the beginning of the day if DNI exceeds 200 W/m² the collectors change their state to focus, and the recirculating pump is activated. This represents the warm-up profile. The operation profile starts with solar field steam production and operates as described before.

At the end of the day, if DNI is lower than 200 W/m², the collectors change their state to defocus, and water circulates until the system cools down. At night the system is off, with solar field collectors defocused and the pumps shut-down. Even during the night, the thermal inertia of the system, as well as the heat losses to the ambient, were considered.

7.1.7 Hybrid operation mode

The hybrid operation analysis was carried out based on the assumptions of the system running 12 or 24 hours daily, at ORC minimum and nominal power. Concerning the control, the hybrid mode comprehends four and two operating profiles for the 12 and 24-
hour regimes, respectively. The 12-hour operating regime differs from the solar-only, on the warm-up and operating profiles.

The warm-up begins at 7:00 with the start of the boiler to warm-up the ORC and the feed water pipe of the solar field. If DNI exceeds 200 W/m², the recirculating pump is activated, and the solar collectors focused. During the operation regime (08:00 to 20:00) the hybrid mode is activated, with the boiler compensating the requirements (nominal or minimum) to drive the ORC turbine. This control is achieved by saturated steam flow rate balance. In preliminary simulation results, it was noticed that during summer the system could start earlier (at 7:00), and the 12-hour operation regime acted as a constraint to the solar field. To overcome this issue, at the beginning of the day if DNI is above 200 W/m² the collectors are focused earlier.

The 24-hour regime just encompasses two operation modes: hybrid and boiler-only. During the daylight period the system operates in hybrid mode, and at night the collectors are defocused, and the solar field is cooled down. After that, the system relies solely on the boiler.

For both cases, the minimum boiler power of 100 kWth was considered, to assure electricity generation stability, during the predefined operating period. Otherwise, the boiler would be submitted to consecutive start-ups and shut-downs, and shortages in the electrical generation would be expected, due to solar radiation transients.

7.1.8 Results

If the system relies solely on solar energy the annual heat generated is about 663 MWhth (Table 7-3). Almost one-quarter of the heat is dumped, mostly related to energy dearth. The dumping rate results are divided into two items (excess and scarcity), representing the heat dumped due to the excess of energy or due to insufficient energy to drive the ORC turbine, respectively.

The annual power generated is 61 MWhel with an ORC average annual efficiency of 9.2% and an annual running time of 1420 hours. The system efficiency is 3.4% hindered by the excessive dumping rates.
Table 7-3 - Solar-only simulation annual results.

<p>| | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Direct Normal Irradiance - DNI</td>
<td>1799.4</td>
<td>kWh/(m²·y)</td>
<td></td>
</tr>
<tr>
<td>Annual Heat Generated SF</td>
<td>663</td>
<td>MWh</td>
<td></td>
</tr>
<tr>
<td>Specific Thermal Field Output</td>
<td>674</td>
<td>kWh/m²</td>
<td></td>
</tr>
<tr>
<td>Mean Annual SF Efficiency</td>
<td>37.4%</td>
<td>%</td>
<td></td>
</tr>
<tr>
<td>Dumping Rate - Excess</td>
<td>7.9%</td>
<td>%</td>
<td></td>
</tr>
<tr>
<td>Dumping Rate - Scarcity</td>
<td>16.3%</td>
<td>%</td>
<td></td>
</tr>
<tr>
<td>Annual Useful Heat - ORC</td>
<td>503</td>
<td>MWh</td>
<td></td>
</tr>
<tr>
<td>Annual Power Generated</td>
<td>61</td>
<td>MWh</td>
<td></td>
</tr>
<tr>
<td>Mean Annual ORC Efficiency</td>
<td>9.2%</td>
<td>%</td>
<td></td>
</tr>
<tr>
<td>ORC - Number of hours running</td>
<td>1420</td>
<td>hr</td>
<td></td>
</tr>
<tr>
<td>Annual Dissipated Heat - Condenser</td>
<td>437</td>
<td>MWh</td>
<td></td>
</tr>
<tr>
<td>Maximum Heat Dissipated - Condenser</td>
<td>387</td>
<td>kWh</td>
<td></td>
</tr>
<tr>
<td>Mean Annual System Efficiency</td>
<td>3.4%</td>
<td>%</td>
<td></td>
</tr>
</tbody>
</table>

Hybridisation improved the solar field output by 3% (Table 7-4). This result is related to the system start-up, since the SF feed water is already warmer, and consequently less solar energy is required to achieve steam generation. Furthermore, this improvement is extended to the solar field annual efficiency. In worth to note, that the rather low operating temperature result in a lower energy requirement to heat-up the SF. Consequently, the improvement of the SF output, and thus efficiency, are more significant in case of higher temperatures.

The second improvement of hybridisation was the extinction of dumping rates associated with scarcity of energy. The scarcity of energy was surpassed with hybridisation, with the fulfilment of the ORC minimum thermal power requirements. Moreover, ORC operation near nominal conditions enhanced the ORC efficiency. On the other hand, the excess of energy increased. This fact is related to minimum operating conditions of the boiler (100 kWth), leading to energy waste predominantly in the summer months when solar radiation is highly available.

The excess of energy can be reduced by implementing a storage tank in the system. The benefits extend beyond the ability to store the excess of solar field thermal energy. If the storage can provide more than 30 minutes of thermal energy requirements to drive the ORC turbine, the need of having the boiler in permanent operation is eliminated. In other words, it can act as system buffer to compensate thermal output fluctuations from the
solar field and boiler, reducing the amount of wasted biogas. Despite the discontinuous operation of the boiler, due to the solar irradiance transients, the average biogas boiler efficiency is still high (about 93%) for all cases.

Table 7-4 - Hybrid simulation annual results.

<table>
<thead>
<tr>
<th></th>
<th>Hybrid - 24 hours operation</th>
<th>Hybrid - 12 hours operation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>530 kW&lt;sub&gt;th&lt;/sub&gt; Boiler</td>
<td>380 kW&lt;sub&gt;th&lt;/sub&gt; Boiler</td>
</tr>
<tr>
<td>Annual Heat Generated SF</td>
<td>683</td>
<td>MWh&lt;sub&gt;th&lt;/sub&gt;</td>
</tr>
<tr>
<td>Specific Thermal Field Output</td>
<td>694</td>
<td>kWh&lt;sub&gt;th&lt;/sub&gt;/m&lt;sup&gt;2&lt;/sup&gt;</td>
</tr>
<tr>
<td>Mean Annual SF Efficiency</td>
<td>38.6%</td>
<td>%</td>
</tr>
<tr>
<td>Annual Heat Generated - Boiler</td>
<td>3359</td>
<td>2546</td>
</tr>
<tr>
<td>Annual Combustion Heat - Boiler</td>
<td>3619</td>
<td>2746</td>
</tr>
<tr>
<td>Mean Annual Boiler Efficiency</td>
<td>92.8%</td>
<td>92.7%</td>
</tr>
<tr>
<td>Annual Biogas Consumption</td>
<td>655</td>
<td>497</td>
</tr>
<tr>
<td>Average Biogas Consumption</td>
<td>1795</td>
<td>1362</td>
</tr>
<tr>
<td>Solar Share</td>
<td>17%</td>
<td>21%</td>
</tr>
<tr>
<td>Dumping Rate - Excess</td>
<td>2.5%</td>
<td>3.1%</td>
</tr>
<tr>
<td>Dumping Rate - Scarcity</td>
<td>0.0%</td>
<td>0.0%</td>
</tr>
<tr>
<td>Annual Useful Heat - ORC</td>
<td>3941</td>
<td>3128</td>
</tr>
<tr>
<td>Annual Power Generated</td>
<td>515</td>
<td>380</td>
</tr>
<tr>
<td>Mean Annual ORC Efficiency</td>
<td>12.7%</td>
<td>11.8%</td>
</tr>
<tr>
<td>ORC - Number of hours running</td>
<td>8760</td>
<td>8760</td>
</tr>
<tr>
<td>Annual Dissipated Heat - Condenser</td>
<td>3389</td>
<td>2720</td>
</tr>
<tr>
<td>Maximum Heat Dissipated - Condenser</td>
<td>387</td>
<td>387</td>
</tr>
<tr>
<td>Mean Annual System Efficiency</td>
<td>9.6%</td>
<td>8.4%</td>
</tr>
</tbody>
</table>
System annual efficiency experienced a considerable boost with hybridisation, from 3.4% to almost 10%, due to the high-efficiency boiler, along with improved efficiencies of the SF and ORC (Figure 7-8). In the REELCOOP framework, a 10% system efficiency was proposed as a target, which is nearly attainable with hybridisation.

One of the main advantages of hybridisation is the stability of the system, to promote dispatchability. This benefit can be observed in the simulation for the 21st of December (Figure 7-9), where the generator is operating 24 hours at nominal power with the 530 kWth boiler.
Figure 7-9 - Winter solstice 24-hour hybrid operation with a 530 kWth boiler.

On the 21st of December, the boiler is supplying 450 kWhth until 9:00, fulfilling the ORC requirements during night operation. At this time the DNI is above 200 W/m², and the collectors change their sate to focus. Steam production from the solar field starts at 10:00, with a thermal output of about 50 kWth. When compared with solar-only, the steam generation starts one hour later and with half of the hybrid production. During the day the solar field is unable to supply the minimum conditions to drive the ORC turbine. Nevertheless, it contributes to a reduction in the amount of required boiler energy and biogas consumption.

On the Summer solstice (21st June), the generation stability was attained even with the smaller boiler (Figure 7-10). However, the heat dumped due to excess of energy increased. In such days, the boiler should be used only during start-up and shut-down, reducing the amount of biogas consumption and heat dumped. The boiler starts to operate at 07:00, at minimum power in order to warm-up the SF and ORC. From 8:00 to 20:00 the nominal electricity generation is achieved mostly from solar energy.
Either Summer or Winter solstice results showed that electricity stabilisation can be achieved during the daylight period through hybridisation, moreover with a significant solar share in the summer (75%) and less relevant (13%) in the winter. This synergy is noteworthy, since during the daylight period electricity prices are usually higher, related to peaks in network consumption.

The estimated biogas production of 60 m$^3$/day is far below the consumption requirements. On an annual basis, the excess of energy related to the boiler minimum operating conditions represents 738 hours of operation and 14 dam$^3$ per year. If we consider as an example the case of 12 hours of operation at minimum power, this represents 9% of the annual biogas consumption and 16% of the running hours. This denotes a minor contribution to CSP/biomass hybridisation. Despite hybridisation allowing to relocate a power plant near urban centres where organic wastes are more abundant, the absence of a well-established biomass market represents a drawback to the dissemination of this alternative source [8].

**7.1.9 Dumped heat**

As aforementioned, either solar-only or hybrid operation results showed an excess of thermal energy, mostly in the summer months when solar radiation is abundant. The
energy waste can be overcome with a storage tank. To define the ideal storage capacity, the daily average values of the heat dumped due to energy excess were analysed for the solar-only mode (Figure 7-11). The analysis was not extended to the hybrid mode since it enhances the dumped heat and consequently the storage capacity. Furthermore, this overestimation can easily be eliminated with a small storage capacity. The maximum value for the average daily dumped heat is 332 kWh (June), and the minimum is 10 kWh (December). It is worth to note that the maximum daily value of the excess heat does not allow to drive the ORC at nominal power for one hour.

![Figure 7-11 - Heat dumped in the solar-only operation mode.](image)

### 7.2 Combined Heat and Power and Economic Performance

The REELCOOP hybrid mini power plant has a nominal electrical output of 60 kW, and thus the economic assessment of the prototype is hindered by the small system scale. Moreover, it would not illustrate the real value of the REELCOOP concept. Therefore, in this section, the hybrid concept is assessed as a scaled up and enhanced REELCOOP system. Also, the option to produce useful heat is evaluated.

#### 7.2.1 Design considerations

From the experience of the REELCOOP project, four main variables were well-defined for the hybrid power plant design: power block, scale, operation conditions and the usability of heat.
One of the main advantages of generating steam within the SF is the possibility to use it directly on the PB, eliminating the heat exchanger between the SF and the PB. Thus, the PB should rely either on a conventional Rankine cycle or on a steam engine.

Up to a scale of 100 kW\textsubscript{el}, the Rankine cycle market is scarce, and steam turbine isentropic efficiencies are significantly low. Accordingly, to have a PB with an acceptable efficiency, and a reasonable levelised cost of electricity, it is necessary to scale up the REELCOOP prototype. It is well known that CSP benefits from economies of scale. Centralised generation is usually accomplished with larger capacities, in the MW power range.

Various steam engines and turbine models were analysed and compared, within the range of 100 kW\textsubscript{el} to 2000 kW\textsubscript{el}. In order to keep up the balance between PB efficiency and system capital expense, a power output of 1 MW\textsubscript{el} was defined. It is noteworthy that at this scale steam turbines are mostly used for waste heat recovery, and isentropic efficiencies continue to be significantly lower than the ones used in conventional steam power plants. This issue is not vital when heat surplus drives the turbines; however, heat from solar and biomass is costly. On the other hand, the technology is proven and operation reliable.

Steam turbine operation conditions (pressure and temperature) significantly influence the PB efficiency. Whilst it is possible to enhance PB efficiency by generating steam within the SF and use it in a turbine over 100 bar, it would represent numerous challenges. At a 1 MW\textsubscript{el} scale, most of the commercial turbines operate under 45 bar, as over this threshold costs significantly increase.

On the SF side, it is necessary to account for the pressure drop within the collector absorber tubes and pipes, which implies even higher pressures at the SF inlet. Furthermore, superheated steam is essential to assure a low wetness at the last turbine stage outlet. Though, higher operating temperatures within the SF imply higher thermal losses. PB steam inlet conditions were defined for 40 bar and 350°C.

As abovementioned, steam turbines at this scale are still characterised by low isentropic efficiency, which results in a significant amount of wasted heat. To improve the economic balance of the plant, the use of the waste heat is addressed, i.e. combined
heat and power. Consequently, it was decided to assess two case studies, one where the hybrid power plant is used solely for power generation and another with CHP.

7.2.2 Power block

Both turbine/generator sets are based on the SST-110 model from Siemens. This specific model is a dual casing turbine on one gearbox, with the possibility of being used as backpressure or condensing unit, with or without extraction. Other relevant characteristics are quick-start without pre-heating and commercial use in cogeneration plants.

The power blocks were modelled using the EBSILON® Professional commercial software considering basic project rules. A 60% design isentropic efficiency was defined for the steam turbines, according to a personal communication from the manufacturer. Case 1 (C1) results in a Rankine cycle efficiency of about 22%, sustained by a steam mass flow rate of about 1.77 kg/s (see Figure 7-12).

For case 2 (C2), two hypotheses were assessed, backpressure or condensing units with a middle extraction. Whilst backpressure results in more significant CHP efficiencies, it decreases electrical conversion efficiency. As a consequence, a higher steam flow rate is required, which infers a larger SF and boiler, and a consequent increase in the system capital expense.

![Figure 7-12 – Case 1 and 2 power blocks.](image-url)
On the other hand, the use of a condensing turbine with intermediate extraction permits an adequate balance of the power to heat ratio, accomplished with a minor decrease in the electrical conversion efficiency. C2 (Figure 7-12) PB has an electrical conversion efficiency of 19% driven by a steam mass flow rate of about 1.95 kg/s. Extraction pressure was set to 1 bar (100ºC) allowing the use of steam for conventional domestic hot water and also to drive a single stage absorption chiller. The nominal power to heat ratio is about 51%. Both C1 and C2 key specifications are presented in Table 7-5.

<table>
<thead>
<tr>
<th></th>
<th>Case 1</th>
<th>Case 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>PB Nominal Power [kW,el]</td>
<td>1000</td>
<td>1000</td>
</tr>
<tr>
<td>PB Nominal Electrical Efficiency [%]</td>
<td>22.32%</td>
<td>19.15%</td>
</tr>
<tr>
<td>PB Nominal Heat Output [kW,th]</td>
<td>0</td>
<td>1960</td>
</tr>
<tr>
<td>Nominal Power to Heat ratio [%]</td>
<td>0</td>
<td>51%</td>
</tr>
</tbody>
</table>

### 7.2.3 Solar Field

The SF was dimensioned and optimised using a free software: Greenius. In contrast to the original REELCOOP prototype with a parabolic trough collector by Soltigua, a generic parabolic trough collector with a larger aperture width of 4.6 m and a vacuum receiver was considered, in order to reach outlet temperatures of 350ºC with high efficiencies. The optical efficiency of the collector is defined to 77%, with reference to the Thai One CSP power plant [9]. As in the REELCOOP prototype, the recirculation concept was adopted over the once-through one, to assure both operation stability and controllability under solar radiation transients. It is noteworthy that the operation of a CSP plant at this scale should be automatic, to reduce human resource needs, and thus costs. Within this concept, water is preheated and partially evaporated in the evaporator section. Subsequently, at the steam drum the water content is separated and recirculated, whilst the steam content is superheated in the superheating (SH) section, and subsequently used to drive the steam turbine.

The Case 1 SF is constituted by four loops of four collectors in the evaporator section, and one loop of three collectors in the superheating (SH) section (see Figure 7-13), with a total effective mirror area of about 10000 m². The recirculation rate was set to 3. At
design conditions Case 1 SF can provide a thermal output of 5761 kW\textsubscript{th}, i.e. a solar multiple of about 1.3.

On the other hand, to achieve 1MW\textsubscript{el} Case 2 requires a bigger SF, achieved through an extra loop in the EVAP section and one additional collector in the SH section, increasing the effective mirror area to about 12700 m\textsuperscript{2}. The nominal thermal output is 7276 kW\textsubscript{th} at design conditions. A summary of the SF specifications and outputs at design conditions is presented in Table 7-6.

<table>
<thead>
<tr>
<th></th>
<th>Case 1</th>
<th>Case 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Collector Type</td>
<td>Parabolic Trough, Vacuum Receiver</td>
<td>Parabolic Trough, Vacuum Receiver</td>
</tr>
<tr>
<td>Collector effective area [m\textsuperscript{2}]</td>
<td>529</td>
<td>529</td>
</tr>
<tr>
<td>Reference DNI [W/m\textsuperscript{2}]</td>
<td>800</td>
<td>800</td>
</tr>
<tr>
<td>Evaporator N\textsuperscript{o} of loops</td>
<td>4</td>
<td>5</td>
</tr>
<tr>
<td>Evaporator N\textsuperscript{o} of collectors per loop</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Evaporator Aperture [m\textsuperscript{2}]</td>
<td>8464</td>
<td>10580</td>
</tr>
<tr>
<td>SH N\textsuperscript{o} of loops</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>SH N\textsuperscript{o} of collectors per loop</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td>SH Aperture [m\textsuperscript{2}]</td>
<td>1587</td>
<td>2116</td>
</tr>
<tr>
<td>SF Nominal Thermal Output [kW\textsubscript{th}]</td>
<td>5761</td>
<td>7276</td>
</tr>
<tr>
<td>Solar Multiple</td>
<td>1.3</td>
<td>1.4</td>
</tr>
<tr>
<td>SF Inlet Temperature [°C]</td>
<td>135</td>
<td>100</td>
</tr>
<tr>
<td>SF Outlet Temperature [°C]</td>
<td>350</td>
<td>350</td>
</tr>
</tbody>
</table>
7.2.4 Hybridisation

A significant advantage of CSP plants is the ability to generate power during peak demand (e.g. late afternoon) using TES tanks. Whilst DSG presents several advantages over other CSP fluid technologies, storing energy is a challenging task. Steam latent heat storage requires the use of phase change materials, also addressed under the REELCOOP project framework [4].

In the REELCOOP prototype, another solution is being tested, that consists of using a steam boiler driven by biogas, to backup SF thermal production and to extend power generation without solar radiation. Therefore, in this assessment, a steam boiler driven by biogas with a nominal efficiency of 85% was considered. The boiler is placed in parallel with the SF (Figure 7-13), to either backup SF operation (e.g. winter times) or to work individually (e.g. at night). Like that, it is always possible to either drive the PB at nominal power or to accommodate demand.

As in REELCOOP, biogas is produced from anaerobic digestion of food waste (FW). The main advantage of FW is that it is a surplus, and therefore inexpensive. On the other hand, the system design is complex due to the potential variety of the biomass. An annual average biogas LHV 24.34 MJ/m$^3$ was defined, and the nominal boiler output is 5 and 6 MWth, for C1 and C2, respectively.

7.2.5 Simulation considerations

Annual simulations were carried out in Greenius. The software is a powerful simulation tool for calculation and analysis of renewable power projects. Additionally, it permits the use of water/steam as heat transfer fluid and is therefore suitable for DSG simulations, with calculations being carried out in a few seconds. Likewise, it is possible to simulate hybrid power plants [10].

The selection of Greenius software (instead of Ebsilon) is justified by the improved calculation performance with an acceptable annual results accuracy. For example, a comparison between Ebsilon and Greenius was carried out for the REELCOOP prototype solar field (Section 7.1.1). Results show an acceptable agreement between both software (see Table 7-7).
First, simulations were carried out for the PB at full and part load conditions, using EBSILON. Afterwards, simulation results were used as input for Greenius using a parametric table.

The project site was set to Tunis location (36.83ºN and 10.23ºE), and TMY weather data obtained from Meteonorm software were used. The annual sum of direct normal irradiance is 1922 kWh/m². A simple load curve and operation strategy was defined, that consists on full load capacity from 6 to 22 h. Whilst this is not accurate, it is demonstrative of a possible load demand for a 1MWel scale power plant, with consumption starting in the early morning and ending in the late afternoon.

7.2.6 Results

The main annual simulation results for the two case studies are presented in Table 7-8.

Table 7-8. In both cases, nominal load operation was assured for the predefined 5840 hours. The solar share varies from 27.5% to 28.8%, for C1 and C2 respectively. The low solar energy share results from the predefined 16 h of full load operation strategy, and the reduced solar multiple necessary to cut heat dumping rate during summer periods, as storage is absent. The dumping rate for C1 is about 3% and slightly higher for C2 (4.5%). As C2 is designed for CHP, this heat could be useful, but this hypothesis was not addressed in this study.

C1 annual heat produced at the SF is about 7750 MWhth, which results in a specific thermal field output of 771 kWhth/m². The larger SF in C2 increases heat production by about 27% (9817 MWhth). The average annual SF efficiency is about 40% for both cases, with a slightly better result for C2 due to the lower SF average temperature.

One advantage of CSP/biomass hybridisation is the possibility to operate at nominal load even in low radiation periods. Consequently, average annual PB efficiencies (21.3%
for C1 and 18.0% for C2) are close to design values. Mean annual system electrical efficiencies are 13.2% and 11.3%, for C1 and C2, respectively. On the other hand, the CHP efficiency for C2 is 33.8% with an annual average power to heat ratio of 50%.

Biogas daily consumption was assessed to estimate the anaerobic digestion system size. This result was required for the energy cost assessment (Section 7.2.7). C2 daily average biogas consumption is about 11000 m$^3$, with an annual consumption of 4 km$^3$. C1 biogas consumption is about 14% lower, with a daily average of 9500 m$^3$ (Table 7-8).

### Table 7-8 – Case 1 and 2 main results.

<table>
<thead>
<tr>
<th></th>
<th>Case 1</th>
<th>Case 2</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>DNI [kWh/(m$^2$.y)]</strong></td>
<td>1922</td>
<td></td>
</tr>
<tr>
<td><strong>Load Curve</strong></td>
<td>6h-22h 1MW$_{el}$</td>
<td></td>
</tr>
<tr>
<td><strong>Annual Heat Generated SF [MWh$_{th}$]</strong></td>
<td>7750</td>
<td>9817</td>
</tr>
<tr>
<td><strong>Specific Thermal Field Output [kWh$_{th}$/m$^2$]</strong></td>
<td>771</td>
<td>773</td>
</tr>
<tr>
<td><strong>Mean Annual SF Efficiency [%]</strong></td>
<td>40.1%</td>
<td>40.2%</td>
</tr>
<tr>
<td><strong>Annual Heat Generated – Boiler [MWh$_{th}$]</strong></td>
<td>19840</td>
<td>23100</td>
</tr>
<tr>
<td><strong>Mean Annual Boiler Efficiency [%]</strong></td>
<td>85%</td>
<td>85%</td>
</tr>
<tr>
<td><strong>Annual Biogas Consumption [km$^3$]</strong></td>
<td>3.45</td>
<td>4.02</td>
</tr>
<tr>
<td><strong>Average Biogas Consumption [m$^3$/day]</strong></td>
<td>9500</td>
<td>11000</td>
</tr>
<tr>
<td><strong>Solar Share [%]</strong></td>
<td>27.5%</td>
<td>28.8%</td>
</tr>
<tr>
<td><strong>SF Dumped Heat [MWh$_{th}$]</strong></td>
<td>232</td>
<td>444</td>
</tr>
<tr>
<td><strong>Annual Useful Heat from SF and Boiler [MWh$_{th}$]</strong></td>
<td>27400</td>
<td>32500</td>
</tr>
<tr>
<td><strong>Annual Power Generated [MWh$_{el}$]</strong></td>
<td>5840</td>
<td>5840</td>
</tr>
<tr>
<td><strong>Mean Annual PB Efficiency [%]</strong></td>
<td>21.3%</td>
<td>18.0%</td>
</tr>
<tr>
<td><strong>Mean Annual System Electrical Efficiency [%]</strong></td>
<td>13.7%</td>
<td>11.3%</td>
</tr>
<tr>
<td><strong>Annual Heat Output [MWh$_{th}$]</strong></td>
<td>-</td>
<td>11600</td>
</tr>
<tr>
<td><strong>Mean Annual Power to Heat ratio [%]</strong></td>
<td>-</td>
<td>50.3%</td>
</tr>
<tr>
<td><strong>Mean Annual System Efficiency [%]</strong></td>
<td>13.7%</td>
<td>33.8%</td>
</tr>
</tbody>
</table>

Typical operation for summer and winter days are shown in Figure 7-14 for C1. In summer, the turbine is driven solely by biomass at early morning and night, i.e. when solar radiation is not sufficient to run the SF. On the other hand, from 9 h to 17 h, power generation is sustained exclusively by the SF. Therefore, in a typical summer day, only two boiler start-ups can occur. In a typical winter day, boiler operation extends to sixteen hours. Nevertheless, from 11 h to 17 h the boiler is driven at partial load.
C2 operation in typical summer and winter days is similar to C1 (see Figure 7-15). The main difference is related to heat production, also continuous. A lower heat demand is expected to occur during summer, and therefore the heat can be used to drive an absorption chiller for cooling. Considering a chiller Coefficient of Performance (COP) of 0.7, it would be possible to produce about 22400 KWh of cooling per day.
7.2.7 Energy costs

For assessing the system capital and operation costs, specific system costs were defined based on manufacturer information and energy cost reports. For the SF, a value of 400 €/m² was defined, which is higher than in conventional CSP plants due to the small scale of this case study. Turbine and boiler specific costs were defined as 800 €/kWₑ and 8 €/kWₜ, respectively. For the anaerobic digestion system, specific costs were defined according to the average values from the IRENA report on biomass costs for power generation [11]. Considering that biogas will be used to drive a steam boiler, instead of a combustion engine, prime mover and electrical costs were subtracted. Also, a capacity factor of 70% was considered. Other costs include project development, insurance during

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Figure 7-15 - Case 2 typical daily operation for summer (top) and winter (bottom).
construction, supervision and start-up. Project contingencies were estimated considering 5% of the aforementioned costs. Cost structures for C1 and C2 are presented in Figure 7-16. SF and AD system represent about 80% of the CAPEX for both cases, of which 55% are related to the SF.

Operational expenditure, include operation and maintenance, replacements and equipment insurance. As expected, AD operation has the highest share (see Figure 7-17), over 55%. Food waste pre-treatment and on-site handling and processing increase process complexity and thus costs. No costs were considered for food waste residues: First, even in countries where subsidies exist (e.g. tipping fees), the values applied for FW are quite low. Second, Tunisia faces thoughtful issues concerning waste management, mostly subsidised by the state [12]. The hybrid power plant is an option to overcome this issue, and consequently, it was assumed that FW costs would be marginal.
Table 7-9 includes a summary of equipment specific costs, as well as the associated CAPEX.

<table>
<thead>
<tr>
<th></th>
<th>Case 1</th>
<th>Case 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spec. SF Cost [€/m²]</td>
<td>400.00</td>
<td>400.00</td>
</tr>
<tr>
<td>Spec. PB Cost [€/kWₜₐ]</td>
<td>800.00</td>
<td>800.00</td>
</tr>
<tr>
<td>Spec. Boiler Cost [€/kWₜₐ]</td>
<td>8.00</td>
<td>8.00</td>
</tr>
<tr>
<td>Spec. AD Cost [€/kWₜ₈₈]</td>
<td>678.00</td>
<td>678.00</td>
</tr>
<tr>
<td>CAPEX SF [€]</td>
<td>4,020,400.00</td>
<td>5,078,400.00</td>
</tr>
<tr>
<td>CAPEX PB [€]</td>
<td>800,000.00</td>
<td>800,000.00</td>
</tr>
<tr>
<td>CAPEX Boiler [€]</td>
<td>39,200.00</td>
<td>44,000.00</td>
</tr>
<tr>
<td>CAPEX AD [€]</td>
<td>3,388,000.00</td>
<td>4,066,000.00</td>
</tr>
<tr>
<td>Total CAPEX (incl. Contingencies) [€]</td>
<td>9,477,115.00</td>
<td>11,259,217.00</td>
</tr>
<tr>
<td>Total Annual OPEX [€]</td>
<td>283,339.00</td>
<td>331,978.00</td>
</tr>
</tbody>
</table>

One simple and appropriate way to summarise and compare the overall attractiveness of both case studies is the levelised cost of electricity. In case 2, to account for the CHP value, heat revenues were subtracted from generation costs (equation 4-86).
As the energy market does not include the price for heat, the average cost of natural gas (about 41 €/MWh) in Tunisia [13], divided by a typical boiler efficiency (90%), was assumed for H_{price}.

C1 results in a LCoE of 175.4 €/MWh_{el} (Table 7-10), which is favourable considering the small scale of the power plant. This result is a consequence of hybrid configurations and contingent to the case study assumptions. Whilst the AD system significantly increases the CAPEX, it improves the capacity factor. Regardless of the PB low nominal efficiency, the annual operation is carried out near design condition, eliminating even lower efficiencies from partial load operation. Moreover, both SF and AD system share the same PB, diluting the cost.

For C2, despite the higher CAPEX and lower PB efficiency, the LCoE_{CHP} is 126.3 €/MWh_{el}. This value assumes that it is possible to sell all the produced heat at natural gas price. The author is aware that for selling the heat it is necessary to be near a consumption (population) centre. First, the small scale of the case study places it in the decentralised generation market. Second, using FW as fuel implies that the plant is placed nearby the consumers.

It is noteworthy that Tunisia massively subsidises natural gas suppliers [13]. If those subsidies were accounted for, the heat revenues would be twice as high, and the LCoE_{CHP} would be even lower. Additionally, the level of temperature of the heat allows its use for cooling purposes, and thus improves the LCoE_{CHP}.

Notwithstanding the attractiveness of the LCoE results, at least when compared with EU average electricity costs, Tunisia average electricity production cost is about 95 €/MWh_{el} [13]. Furthermore, the average electricity retail price is about 48.7€/MWh_{el}, due to substantial direct and indirect subsidies. On the other hand, power consumption is considerably increasing every year (5%), and currently, Tunisia is a net importer of energy. Thus, an intensification of the energy costs is expected in the coming years. The situation is similar in other MENA countries.
Table 7-10 - Levelised cost of energy.

<table>
<thead>
<tr>
<th></th>
<th>Case 1</th>
<th>Case 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Life time [years] – Annuities</td>
<td>25</td>
<td>25</td>
</tr>
<tr>
<td>Interest rate [%]</td>
<td>6.0%</td>
<td>6.0%</td>
</tr>
<tr>
<td>CRF [%]</td>
<td>7.82%</td>
<td>7.82%</td>
</tr>
<tr>
<td>LCoE or LCoE_{CHP} [€/MWh]</td>
<td>175.4</td>
<td>126.3</td>
</tr>
</tbody>
</table>

7.3 Once-through operation

In sections 7.1 and 7.2, hybridisation of CSP with water/steam has been assessed on the consideration of placing the biomass boiler in parallel to the solar field and thus used to backup thermal energy in times of low radiation. Nevertheless, hybridisation can also be used to address specific technical issues.

In this section, the operation of a 5MW_{th} hybrid CSP/biomass gasification plant is assessed, under once-through operation. The hybrid power plant performance evaluation includes operation during start-up, and also under transient and constant solar radiation. For each case, hybridisation enhancements are evaluated.

7.3.1 The Hybrid Power Plant

The hybrid power plant simplified layout is presented in Figure 7-18. The solar field relies on parabolic trough technology, and DSG is attained within the solar field under a once-through operation mode. The solar field is constituted by five loops of 10 Eurotrough-type collectors, with a nominal length of 100 m and a width aperture of 5.76 m. Modified Schott PTR-70-DSG absorber tubes are used (with a wall thickness of 5.6 mm). In this study, the hybridisation purpose is to backup solar thermal generation and to boost the outlet steam temperature. Hybridisation is achieved using six downdraft biomass gasifiers driven by woodchips, each one with a 1 MW_{th} capacity. Modularity justifies this solution, and thus enhances part load operation, as well as reducing possible technical issues concerning larger gasifiers.
Gasification results in a flammable gas (producer gas) at above 650°C. An average LHV of 5.4 MJ/kg was used, calculated from experimental chemical gas composition measurements. Gas conditioning is advisable and requires lower temperatures. Therefore, a heat exchanger (HEX1) is proposed to boost solar field outlet steam temperature. Whilst DSG operation under OT mode presents advantages over the conventional recirculation mode, specific issues need to be assessed. For example, there is a safety limit for the temperature gradient within one receiver cross-section in the superheating section. To overcome this issue, a minimum flow rate within the absorber is imposed, which hinders solar field outlet temperature under low radiation periods. Thus, the use of the HEX1 to boost steam temperature can be a vital enhancement for OT operation. Afterwards, producer gas is burned in a conventional steam boiler, used to backup solar field thermal generation.

At nominal conditions (DNI of 850 W/m²) the solar filed heat production is about 16.7 MWth. Turbine nominal conditions require 17 MWth as input. This rather low solar multiple (about 1) is justified to reduce the amount of heat dumped, as storage is absent (Section 7.1). On the other hand, at nominal conditions, the steam boiler delivers 5.59 MWth and the HEX1 an additional 0.29 MWth. Therefore, the thermal generation system (solar field and gasifier/boiler set) is able to supply about 133% of the power block.
thermal energy demand. To accommodate short and unforeseen solar radiation transients, storage is recommended for a CSP/biomass plant (Section 7.1) in this study, a gas storage tank is used to act as system buffer.

The power block is based on the SST-111 Siemens compact steam turbine. A nominal power of 5 MWel is attained with three expansion stages (see Figure 7-19), resulting in a net power cycle efficiency of 29.1%.

![Figure 7-19 – 5 MWel Power block simplified model layout.](image)

7.3.2 Results

In this section, the hybrid power plant is assessed under distinct operation conditions: start-up, transient and constant solar radiation. For each case, solar field and system performance are evaluated.

Regarding modelling, simulations were carried out using meteorological data from the DISS test campaign in Almeria, with a time-step of 5 seconds (see Chapter 5). In order to reduce the computation effort, only one loop of collectors is simulated, assuming that
the behaviour of other loops is identical. The quasi-transient modelling method is extended to solar field piping (collector connections and headers). Gasifier thermal inertia, as well as start-up and cool-down behaviour, was modelled by a combination of an IS component and an algorithm, both developed and validated using experimental data. A simple control strategy was defined and consists in setting up a constant pressure at the loop outlet, and control outlet temperature through loop and injection mass flow rates. Limits were imposed on the loop and injection mass flow rates, as well as on the variable changes between time steps. For more details, the reader is addressed to Chapter 4.

7.3.3 Power plant start-up

Plant daily routine usually includes a start-up. Hybridisation can improve this routine, by enhancing a fast start-up as well as, reducing the number of start-ups during the plant lifespan.

Figure 7-20 shows simulation results for the hybrid plant operation under solar field start-up, for one hour. From 09:00 to 09:20 the solar field outlet temperature is about 295.6°C (i.e. under phase change since the steam pressure at the outlet of the solar field is 80 bar), and the HEX1 used to support evaporation. This result is a consequence of the imposed minimum flow rate of 3.5 kg/s (0.7 kg/s within one loop). Nevertheless, hybridisation assured electricity generation during this period.

It is possible to notice that full evaporation at HEX1 is only achieved at 09:03 when the turbine steam flow rate is about 5.76 kg/s. From 09:21 to 09:46 steam temperature at collector loop outlet increases to the design temperature of 400°C. Afterwards, SF steam mass flow rate rises about 31% until 11:00, with the turbine driven at about 98%, near nominal conditions. A fast start-up is achieved since the SF and power block are already warmer. It is noteworthy that with a larger solar field a similar or even faster start-up could be attained. Nevertheless, with this hybrid solution and as storage is inexistent, there is no real benefit in an over-sized solar field, which would result in excessive dumping rates and operation with defocused collectors.
7.3.4 Operation under Solar Radiation Transient

The transient nature of solar radiation, as well as the technical issues associated with a two-phase flow heat transfer, under once-through operation mode, enhance grid stabilisation risk, as well as thermal and mechanical stresses at main plant components (e.g. solar collectors, turbine). Compared to single phase CSP heat transfer fluids (e.g. thermal oil) thermal energy storage design is a challenging task. Comparing once-through with recirculation operation modes, the absence of a steam drum reduces system controllability and stability.

Simulation results for hybrid operation during 1 hour with a significant solar radiation transient are shown in Figure 7-21. Until about 12:23, solar radiation increases to about 950 W/m², and thus the solar field steam mass flow rate increases as well. If the turbine is driven exclusively by steam from the solar field, power oscillates between 77.6% (3.16 MW_{el}) and 84% (4.23 MW_{el}) of nominal power. On the other hand, hybridisation enhances dispatchability, plant capacity factor and power stabilisation through the use of the steam boiler to back up the solar steam mass flow rate, and thus to drive the turbine at nominal conditions. Nevertheless, solar radiation increases during this period and consequently the solar field thermal output, as well as the steam flow rate. Therefore, electricity generation stabilisation is attained by controlling the gas flow rate at HEX 1.
Between 12:23 and 12:40, a set of clouds results in a significant (953W/m\(^2\) to 162 W/m\(^2\)) and sudden (about 2 minutes) drop of solar radiation, followed by a set of radiation transients. If the system relies solely on solar energy, the turbine steam inlet temperature decreases at about 22.5ºC/min rate. Afterwards, the steam flow rate is not enough to drive the power block at minimum power, with the turbine set to standby conditions for about 26 minutes. This result is a consequence of the combination of the solar radiation transient and the minimum mass flow rate set within the loop.

Whilst hybridisation is not able to stabilise electricity generation during the transient, several advantages are observed. Concerning power, results show a minimum value of 3.5 MW\(_{el}\) and the absence of power shortages during the entire solar radiation transient. Furthermore, recovery from part load operation is faster and nominal power is attained at 12:47. It is noteworthy that other control strategies can result in better plant performance.

HEX1 increases the solar field steam temperature, attenuating the solar field outlet temperature reduction. Additionally, the boiler is delivering steam at 400ºC, and thus the turbine inlet temperature decreases at a rate of about 18ºC/min (20% lower than with solar only), reducing the thermal stress at the turbine.

![Diagram](Image)

**Figure 7-21 - Hybrid power plant operation during 1 hour with high radiation transient.**
7.3.5 Operation under Constant Solar Radiation

Figure 7-22 shows simulation results for the hybrid power plant under steady and abundant solar radiation, during a 5 hour period. Despite the relatively low solar multiple (about 1) and low biomass share (about 33% of the power block nominal thermal demand), it is possible to attain nominal and stable power generation.

In this period, both radiation and the solar field thermal output are higher than design values. Thus, the steam turbine is driven solely by the solar field most of the time and the gasification backup system can be shut down, increasing biomass savings.

Nevertheless, between 10:00 and 10:47 the plant is operated in hybrid mode. In this case, hybridisation is attained through the use of HEX 1 to boost and control solar field outlet temperature slightly. The steam boiler is kept off, as it will result in a surplus of heat. This strategy improves the power output by about 1.4% from the solar only case, and electricity generation stabilisation for more extended periods.

Since the backup system (gasifier and boiler) is down, it would be interesting to evaluate the plant performance under a significant radiation transient.

Figure 7-22 - Hybrid power plant near-steady operation during 5 hours with high radiation.
7.4 Turbine reheat

Most of the CSP plants rely on a conventional Rankine cycle. Although the advantages of steam reheat within turbine stages has been widely proven, there are technical challenges for reheat in the DSG configuration [14]. Usually, an indirect reheat configuration is adopted, as direct reheat is hindered by the distance between the turbine and the solar field, ensuing excessive pressure losses [15]. Therefore, the use of reheat is mostly justified by the increase of steam quality in the last turbine stage, and not explicitly associated with system efficiency enhancement. However, CSP hybridisation with a steam generator driven by biomass eliminates this constraint.

In this section, a 5 MW_{el} hybrid renewable electricity generation system is assessed. The system relies on a combination of solar and biomass sources to drive a 5 MW_{el} steam turbine identical to the one assessed in section 7.3. System performance is evaluated through numerical simulation using Ebsilon professional software. Annual simulations were carried out on an hour-by-hour basis, for solar-only and hybrid modes. The use of direct reheat within the turbine is assessed.

7.4.1 CSP/biomass Power Plant

The solar field relies on linear Fresnel reflector technology and is constituted by six loops of 12 collectors, with a net aperture area of 513.6 m^2 per collector. The collectors’ axes are oriented North-South, and the solar multiple is 1, in order to decrease the amount of heat dumped.

Direct steam generation is achieved within the loops. To guaranty system controllability and stability, the recirculation concept was adopted [16]. Thus, each loop is subdivided into two sections (see Figure 7-23): evaporation of water (8 collectors) and superheating of steam (4 collectors). Within this concept, the cold-water flows throughout the first eight collectors, where it is partially evaporated. Afterwards, the water/steam mixture is separated in a steam drum. The saturated steam is superheated to the design temperature in the last four collectors. The remaining water in the steam drum is then recirculated.
In view of operation differences within the loop sections, two collector models were chosen from the Novatec Solar manufacturer [17]: Nova-1 and Supernova models for the evaporation and superheating section, respectively. The models differ on the Supernova enhancement with evacuated tubes, an aimed characteristic for the superheating section. The loop layout was designed to enhance system performance, through a length reduction between loop outlet and power block inlet, and therefore decreasing heat and pressure losses. Moreover, this layout results in a compact and modular SF.

The mass flow rate within the solar field loop was defined based on a compromise between avoiding water stratification in the absorber tubes and a feasible pressure drop [18]. At nominal conditions the mass flow rate and the recirculation rate within one loop are 1.3 and 0.5 kg/s, resulting in a steam quality at the steam drum of 0.75. The steam outlet pressure and temperature were defined as 80 bar and 400°C, controlled by water injection before the last loop collector.

Figure 7-23 – One loop simplified model layout.
Power plant hybridisation is attained through the use of a steam generator, used for enhancing and backing up solar field heat generation, in order to increase both dispatchability and global system efficiency. The steam generator has a nominal efficiency of 91% and is driven by biogas resultant from anaerobic digestion of organic waste, similarly to the one investigated in the REELCOOP project [3].

Concerning the steam generator model, it consists of a combustion chamber, where the biogas/air mixture combustion is modelled. The resulting hot flue gas exchanges heat in four or five heat exchangers (see Figure 7-24): a superheater; re heater (optional); an evaporator; a preheater; and an economiser.

![Steam generator simplified model layout.](image)

The power block is based on the SST-111 Siemens compact steam turbine. Similarly, to the one used in Section 7.3, the 5 MWel power is attained with three expansion stages
Nevertheless, the use of direct reheat is assessed, resulting in a net power cycle efficiency improvement to 29.4%. In this case, the steam between the first and second turbine stages is directly reheated in the biomass steam generator (see Figure 7-24).

The system was designed for the Algarve region in the South of Portugal. Abundant solar radiation characterises the region, with an annual value of direct normal irradiance of about 2344 kWh/(m²·year). Moreover, the power demand in the region drastically changes along the year, due to the significant tourism influx in the Summer months. Therefore, the region would benefit from a dispatchable and flexible renewable power generation system. Furthermore, demand and availability of energy resources are linked, since solar radiation is abundant in summer months and organic waste increases with human activity.

Figure 7-25 - 5 MWel Power block with reheat simplified model layout.
Three case studies were analysed: solar only, and hybrid without and with reheat. Within the solar only model, the system generation is attained exclusively through solar radiation. The hybrid models encompass the assumption of the system operating 16 hours per day at full power during the whole year. For each case, the nominal system power (5 MW<sub>e</sub>) was kept constant.

### 7.4.2 Results

The annual simulation results for the three case studies are presented in Table 7-11. If the system operates exclusively on solar energy, the annual heat generated in the solar field is 27594 MWh<sub>th</sub>. The turbine operation is confined to 2270 hours, with an annual generated power of 6403 MWh, mostly related to summer months.

On the other hand, hybridisation improved the annual heat generated in the solar field to 27979 MWh<sub>th</sub>. The rather low solar multiple is justified by the absence of storage, and the defined 16-hour operation range in the hybrid mode resulted in a significant reduction of the solar share to about 25%. Nevertheless, energy dispatchability and generation stability were attained, with a net annual generation of about 31.7 GWh<sub>e</sub>.

<table>
<thead>
<tr>
<th>Table 7-11 - Annual simulation results.</th>
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<tr>
<td><strong>Hybrid</strong></td>
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<tr>
<td>Direct Normal Irradiance – DNI</td>
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<tr>
<td>Annual Heat Generated – SF</td>
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<tr>
<td>Mean Annual SF Efficiency</td>
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<tr>
<td>Annual Heat Generated – Steam Generator</td>
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<tr>
<td>Steam Generator - Reheat Share</td>
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<tr>
<td>Mean Annual Steam Generator Efficiency</td>
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<tr>
<td>Annual Biogas Consumption</td>
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<tr>
<td>Annual Useful Heat – Power Block</td>
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<tr>
<td>Annual Power Generated</td>
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<tr>
<td>Annual Power Block Efficiency</td>
</tr>
<tr>
<td>Solar Share</td>
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<tr>
<td>Running hours</td>
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<tr>
<td>Mean Annual System Efficiency</td>
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Regardless of the power plant small scale, hybridisation results in a significant improvement of the mean annual system efficiency, from 7% to 18% (see Figure 7-26). This enhancement is associated with the increase of the solar field (29.7% to 32.3%) and power block (23.5% to 29.1%) mean annual efficiencies, as well as through the use of a high-efficiency steam generator. Note that the generator efficiency increases as well with reheat.

The solar field efficiency enhancement is related to the improved conversion of solar radiation at the beginning of the day. In a solar-only power plant, early system start-up can be easily accomplished by using a biomass boiler. This development leads to an increased power block efficiency, related to the operation conditions of the turbine, since the system can be driven close to design conditions.

Figure 7-26 - Annual mean efficiencies for distinct operation cases.

The implementation of direct reheat within the turbine stages leads to a 2.5% (relative) mean annual system efficiency improvement, from 17.6% to 18.1%, as a consequence of the steam generator and power block efficiency enhancement. In this example, the turbine was designed for solar-only operation, and therefore the reheat
improvement can be more significant if the turbine design is optimised. Also, the small scale of the turbine and the reduced number of stages hinder the effect of reheat.

As result of the power block efficiency increase (28.7% to 29.1%), a lower steam mass flow rate is required to attain nominal operation conditions, and therefore, as the solar field size was kept constant, the solar multiple increases. In other words, the design conditions could be attained with a solar field of five loops. Consequently, there was a marginal increase in the solar share (25.0% to 25.5%), considering that hybrid operation was modelled so that solar energy takes priority in the electric generation portfolio. Furthermore, a 5% reduction in the biogas consumption was noticed.

In addition, a desirable low steam wetness after the last turbine stage is assured, reducing the turbine mechanical stress (see Figure 7-27). Nevertheless, neither solar only nor hybrid operation modes result in a steam content below the manufacturer tolerated values (80% to 90%). The higher steam content after the low-pressure turbine section highlights the necessity to optimise the turbine operation, to improve the power block efficiency.

![Figure 7-27 - Steam quality after the third turbine stage – 13th June.](image)
7.5 Single phase heat transfer fluid - Boost HTF temperature and backup steam cycle

In the author’s opinion, concerning linear CSP working fluids, steam is the most suitable for hybridisation by the absence of storage. Nevertheless, either thermal oil and molten salts can benefit from biomass hybridisation. In this section, CSP/biomass hybridisation enhancements are assessed for single-phase heat transfer fluids.

This evaluation entails the analysis of placing a biomass boiler within the HTF cycle and the steam cycle. In the first, biomass is used to boost the solar field and storage outlet temperature. The inherent advantage is that part load operation is only conditioned by a lower HTF mass flow rate, and thus temperatures within the PB are near design values. This concept is being used in the Termosolar Borges power plant [19]. In the HTF cycle, no backup is addressed, as the TES system permits to extend operation without solar radiation. On the other hand, within the steam cycle, a biomass boiler is used to back up steam generation. Besides the advantages, specific design considerations are evaluated for each case.

7.5.1 The solar field

The solar field relies on parabolic trough technology, with 13 loops of 4 Eurotrough ET-150 collectors (see Figure 7-28). At design conditions (800 W/m²) the solar field is able to increase oil temperatures from 293°C and leaves at 391°C. The nominal solar field thermal power is about 22.8 MWth, which represents a solar multiple of about 1.3. The additional energy is stored in a conventional storage system, which consists of two molten salt storage tanks. For a hybrid system, the advantages of TES extend beyond the decoupling of electrical and thermal demand. It can also be used as a buffer over the boiler start-up (see section 7.1).
Concerning hybridisation, within the thermal oil system, a combustion boiler driven by refuse derived fuels feedstock is used to boost the HTF temperature. The use of RDF permits to address the issue of non-recyclable materials and thus reduces landfill emissions. An additional advantage is the proximity of the consumption centre and feedstock. For the RDF properties, data from [20] were used.

The advantages of enhancing the HTF temperature are mostly related to part load operation, for example during the solar field start-up (see Section 7.5.3) and also in the absence of solar energy (see Section 7.5.4).

### 7.5.2 Power block

A power block similar to the one used in Section 7.3 was considered, i.e. with 5 MW\textsubscript{el} attained in three stages without reheat. Nonetheless, the steam generator entails a SH, EVAP and PH HEXs between the solar field and the power block, and result in a lower steam temperature (378ºC). Consequently, the steam pressure was reduced from 80 to 70 bar to assure a steam quality of about 0.87 at the turbine outlet.
Hybridisation is attained by a syngas boiler to back up steam production. As gasifier feedstock, RDF is also considered. The concept of RDF gasification is being used at commercial scale, for example in the Lahti Energia plant in Finland [21].

The reason for the adoption of gasification technology over combustion is related to the boiler capacity and parasitic consumption. Gasification requires lower amounts of air and flue gas. Therefore, the result is a lower parasitic consumption in the air and gas blowers. Furthermore, the size of the downstream components is reduced. Both advantages are noteworthy as result of the required boiler size to drive the power block at nominal conditions.

![Figure 7-29 – Thermal oil power block layout.](image)

### 7.5.3 Start-up of the solar field

In this section, the hybrid power plant is assessed during the solar field start-up. It is assumed at the beginning of the simulation that the power block is being driven by the TES energy, i.e., a warm start-up of the SF is evaluated. Also, hybridisation is being limited to the HFT temperature enhancement without considering back up in the steam cycle.
Initially, the solar field is in recirculation mode, noticed by the absence of mass flow rate from the SF (see Figure 7-30) or heat output (see Figure 7-31). On the other hand, storage is under the discharging process allowing to drive the power block.

If solar energy exclusively drives the power cycle, the thermal oil outlet temperature is limited to 365ºC over this period, and power generation reduced to 80% of the nominal load. Also, solar energy is used to heat up the SF instead of electrical generation.

Through the additional heat from the biomass boiler, the HTF temperature is boosted to 391ºC, and consequently, the power block is driven near design conditions. Furthermore, the additional energy from biomass permit that part load is only noticed in the steam generator by a lower HTF flow rate.

As soon as the solar field outlet temperature rises to 365ºC, part of the recirculation mass flow rate is mixed with the TES flow rate and redirected to the steam generator. The mixture between the SF and TES HTF is also boosted to 391ºC. Nevertheless, a lower amount of energy from the boiler is required. Subsequently, the HTF flow rate from the solar field increases at the same time as HTF flow rate from TES decreases. Although the
TES is operated over two and a half hours, the amount of energy from TES discharging is low (equivalent to about 0.5 hours of full load).

Note that it is possible to achieve the same output without hybridisation, though at the cost of a larger solar field. To increase the amount of energy of the SF more loops are required. Consequently, a bigger TES system is also necessary to reduce the amount of heat dumped. Additionally, to improve the loop outlet temperate it is necessary to add a new collector to each loop, enlarging the SF size by 25%. Alternatively, using collectors with a large aperture area, like the Flabeg’s Ultimate Trough [22].

![Diagram](image.png)

Figure 7-31 – Heat and power - Solar Field start-up.

7.5.4 Cool-down

At the end of the day, solar radiation decreases and so the solar field output. Consequently, the stored energy is used to drive the power block. Initially, the solar field mass flow rate decreases to sustain 391°C at the outlet (see Figure 7-32). Afterwards, the minimum mass flow rate within the loop hinders the loop outlet temperature, and consequently at about 17:16 the solar field is set to recirculation mode with the collectors defocused. Over this period, the mass flow rate from TES increases to sustain power generation.
If the system relies solely on solar energy, during storage discharge operation, the HTF temperature is considerably lower (365°C) than nominal condition (391°C). This results in a reduction of the steam temperature, and so the electrical generation is confined to 80% of the nominal load (see Figure 7-33).

On the other hand, boosting the loop outlet temperature permits to operate the power block close to design conditions. TES discharge process continues until about 20:35, when the storage reaches the lower level. Note that it is possible to further extend operation through the use of larger solar field and storage tanks.

In this case study, hybridisation within the steam cycle is used to enhance dispatchability. There are advantages of placing the backup boiler within the steam cycle, instead of within the HTF cycle. One is the maturity of biomass steam boiler technology in comparison to thermal oil. Another is related to the steam generator HEXs between the HTF and steam cycle. If the boiler was placed on the HTF side, about 17.8 MWth would be required to drive the power block at nominal conditions. On the other hand, within the steam cycle the thermal demand is reduced, and thus a better system efficiency is attained.
7.6 Single phase heat transfer fluid - Boost steam temperature

An additional improvement of biomass hybridisation for single-phase heat transfer fluids consists in improving the steam parameters (both steam and pressure at the turbine inlet) within the power block, and thus enhance the power block efficiency.

In this section, the solar field from Section 7.5 is considered. I.e. thirteen loops of four ET-150 collectors and biomass boiler driven on RDF feedstock to boost the HTF fluid temperature.

7.6.1 Power block

The main difference regards hybridisation in the steam cycle. A syngas boiler is used to back up steam production. However, an additional HEX (see Figure 7-34) permits to increase the steam temperature to 540ºC. Therefore, the turbine can also be driven at higher pressure (90 bar) without jeopardising steam quality at the turbine outlet (0.93). Additionally, the efficiency is enhanced in about 8.7%. This add-on is similar to the one used in the SHAMS 1 power plant [23], yet in this case study without consuming fossil fuels (i.e. natural gas).
7.6.2 Constant solar radiation

To assess the advantages of enhancing the steam parameters, the system performance is evaluated under near-steady and abundant solar radiation (see Figure 7-35). Beam radiation is above the design conditions (800 W/m²), and thus results in excess of energy, which is stored. Over this period, hybridisation within the HTF cycle is not required, as the solar field can keep a steady outlet temperate of 391°C.

![Figure 7-34 - Thermal oil power block layout – boost steam temperature.](image)

![Figure 7-35 - DNI, temperatures and mass flow rates - Solar Field under near-steady and abundant solar radiation.](image)
The main advantage is noticed by comparing the energy to storage in solar-only and hybrid operation modes (see Figure 7-36). If the system operates under solar-only, the amount of stored energy is about 73%, when compared to hybrid operation. This improvement is a consequence of the enhanced power block efficiency and of the additional energy in the steam cycle. Also, it shows that the solar field can be reduced, or the storage size increased to accommodate the additional thermal energy. This is an essential result concerning costs, as the SF and TES account for more than 50% of a CSP plant cost structure [24].

Another advantage is related to the possibility to operate a solar field and TES system at lower temperatures, without jeopardising the power block efficiency. When compared with DSG (see Section 7.3) the solar field outlet temperature is lower, and thus thermal losses are reduced and the efficiency enhanced. However, the steam pressure and thus system efficiency are augmented.

Further benefits are related to the parasitic consumption. The power block and solar efficiency increases and so the mass flow rate of both HTF and steam are reduced. In the case of the HTF, the nominal mass flow rate within the solar field was reduced to about 78.5%.

![Figure 7-36 - Heat and power - Solar Field under near-steady and abundant solar radiation.](image-url)
7.7 Molten Salts

Whilst the results presented in the single-phase heat transfer fluid assessment are exclusively associated with thermal oil as HTF, the same concept was used for a solar field with molten salts. As result, the same advantages were noticed, and therefore in this section only the most relevant differences are highlighted.

7.7.1 Solar field

Within the solar field, the main differences are the loop temperatures and mass flow rate, and also the use of direct storage (see Figure 7-37). For the assessment, the eutectic mixture of 60% sodium nitrate (NaNO3) and 40% of potassium nitrate (KNO3) [25] was considered, similar to the one used in the Archimede Solar Energy molten salt demo solar plant [26].

With the same solar field configuration (see Section 7.5.1) it is possible to attain a nominal thermal power of 21.6 MWth at design conditions (800 W/m²). On the other hand, the mass flow rate is about 56 kg/s, about 60% lower when compared with thermal oil. Note that when compared to the hybrid system presented in Section 7.6, the mass flow rates are about the same. Additionally, viscosity and operational pressures are lower, and hence the pump parasitic consumption is reduced.

The loop outlet temperature was defined for 545°C (about 39% higher than with thermal oil). Whilst the solar field nominal thermal power is marginally lower than for thermal oil, the solar multiple is higher (about 1.36) due to the power block higher efficiency.

A storage system with two tanks of molten salts was considered. However, the solar field mass flow rate is directly stored in the hot tank. Afterwards, the molten salts are discharged from the hot tank and redirected to the steam generator. The cooled HTF is then stored in the cold tank, which is again heated up in the solar field. The direct storage concept permits to reduce the number of HEX and so energy losses. Furthermore, it acts as a direct buffer between charging (solar field) and discharging (power block).

Regarding hybridisation, in the molten salts cycle, a biomass boiler is used to heat up the HTF to 545°C, eliminating the temperature drop from thermal losses at the hot tank.
In this case, wood pellets were considered as feedstock for the combustion boiler. The reason for this update is the limit concerning the maximum combustion temperature (430ºC) with RDF [27]. Above this threshold, high-temperature corrosion and ash fusion problems are expected.

7.7.2 Power block

In the power block (see Figure 7-38), the main differences are related to the steam parameters. At the steam generator outlet, a temperature of 535ºC was considered, about 44% higher than with thermal oil. The steam temperature is enhanced by 5ºC, and thus the energy required for the SH2 is about 3% in comparison to thermal oil. Furthermore, even in the case when the HTF booster is off the energy required for SH2 is only 0.210 MWth. The additional cost and complexity of adding another HEX in the steam loop must be considered, to properly evaluate the advantages of boosting steam temperature in case of a molten salts loop. On the other hand, this add-on can be useful for a power block with higher steam parameters [28]. This fact does not affect the backup boiler added.
value. Whilst molten salts tanks permit to decouple generation from the solar resource, the cost is in the range of 30$ to 40$ per kWh and augmented for small-scale systems [29]. Therefore, the biomass cost for heat can be competitive especially if the feedstock is waste [11, 30].

Figure 7-38 – Molten salts power block layout.

7.8 Conclusions

In this Chapter, hybridisation of concentrating solar power with biomass was assessed, through case studies. Concerning CSP, the assessment included linear focus collectors (PTC and LFR) and both two-phase HTF (i.e. water/steam) and single-phase HTF (i.e. thermal oil and molten salts). Regarding biomass, combustion, anaerobic digestion and gasification were evaluated for distinct feedstocks (e.g. food waste, RDF).

In the author’s opinion, within linear CSP working fluids, steam is the most suitable for hybridisation by the absence of an available and economically feasible latent heat storage technology. Nonetheless, new latent heat TES concepts are under development. A comparison at a prototype scale (i.e. 60 kWel) of DSG operation under solar-only and hybrid operation with biogas from food waste AD, shows that if power generation is exclusively dependent on the solar field, electrical generation is mostly confined to sunnier months and negligible in the winter. The annual heat dumping rate ascends to
almost one quarter, mostly related to the inability of the solar field to supply the minimum thermal energy to drive the ORC turbine.

System hybridisation proved to stabilise the system regarding electrical power generation during the whole year. Additionally, the downside of the dumped heat, due to the scarcity of thermal energy, was surpassed. Hybridisation improvements where extended as well to the SF and ORC. The SF output increased marginally (3% relative) since the system is already warmer in the morning, and solar radiation is used exclusively for steam generation. The ORC efficiency increase is in the range of 15% to 38% (relative) and was achieved by a stable operation near turbine design conditions.

On the other hand, there was an increase of the system energy excess due to the downsides of boiler start-up time (about 30 minutes) and minimum operational heat input (100 kWth). The simulated operation profiles were created on the basis of energy dispatchability and demand response-ability. In addition to that, boiler operation was extended to the system operation range to compensate possible short transients from solar power. This issue can be overcome with the implementation of a storage tank. From simulation results, the maximum average daily dumped heat ascends to 332 kWth. This value is quite small when compared with the system size. For example, it does not allow to run the system for one hour at nominal power. This proves that the solar field design is appropriate, and hybridisation significantly reduces (although not eliminating) the need for storage. Note that for all simulations the solar share is rather low, with a maximum of 44% for the case when the boiler is used to assure 12 hours of continuous operation at minimum power.

The improvements in the SF and ORC efficiencies, along with extended operation ranges and a highly efficient boiler, lead to a considerable boost in the system efficiency, which increased from 3.4% to 9.6% for the small system scale (60 kWel) of the REELCOOP hybrid mini power plant. This small-scale hindered the economic assessment, not illustrating the real value of the hybridisation concept. Therefore, the hybrid concept was assessed as a scale up (1 MWel) and enhancement of the REELCOOP prototype.
In addition, the option to produce useful heat was evaluated, through two case studies which diverged on their purpose: one for power generation (C1) and the other for combined heat and power (C2).

Simulations were carried out under the assumption of 16 hours of continuous load demand per day (6 to 22 h). C1 power block higher efficiency resulted in an annual average system efficiency of about 13.7%, with a solar share of 27%. Despite C2 lower average electrical conversion efficiency (11.3%), the utilisation rate was 33.8%, with an average power to heat ratio of 50%.

Concerning costs, the SF and AD costs were about 80% of the total CAPEX. C2 CAPEX (11.3 M€) was about 19% higher than C1 (9.5 M€). On the other hand, AD share was above 55% in the OPEX cost structure, justified by the food waste AD system operation complexity.

The LCoE obtained for C1 was 175.4 €/MWhel, which is very attractive considering the small scale of the power plant (1 MWel). This result is a consequence of hybrid operation, enhancing power block and system efficiency and also reducing cost by equipment sharing (e.g. PB). Results showed even better values for C2, with a LCoECHP of 126.3 €/MWhel.

On the other hand, the assessment was carried out for Tunisia, where the energy market is heavily subsidised, and average electricity production costs and retail prices are 95€/MWhel and 48.7€/MWhel, respectively. With energy consumption increasing about 5% each year and as a net energy importer, it is expected that costs will increase in the coming years. A comparable situation occurs in most other MENA countries. Also, no subsidies were used for the case studies, which would significantly improve the economic assessment.

The advantages of CSP/biomass hybridisation extend beyond the solar field back up by adding a boiler in parallel to the SF. To evaluate specific hybridisation enhancements, a 5 MWel hybrid CSP/biomass power plant was presented, modelled and simulated. The solar field relied on parabolic trough technology with DSG under once-through operation. Biomass hybridisation was attained through a woodchip gasification system, to drive a backup gas steam boiler or to boost solar field outlet temperature using an additional heat exchanger, HEX1.
The hybrid plant performance was assessed for three different operation conditions: start-up, near-steady and transient. During start-up, the HEX1 was used to support evaporation and subsequently to boost solar field outlet temperature. A minimum flow rate of 0.7 kg/s within one loop was imposed in order assure the safety limit for the temperature gradient within one receiver cross-section in the superheating section. The excess of water hindered the solar field outlet temperature during part of the start-up. Nevertheless, hybridisation assured electricity generation during this period from the added steam from the steam boiler, and also improved the solar field start-up time through the use of HEX1.

Whilst hybridisation was not able to stabilise electricity generation during the transient, several advantages were noticed. Concerning power, results showed a minimum of 70% of nominal power and the absence of power shortage (as it occurs if the system relies solely on solar energy) during the entire solar radiation transient. Furthermore, recovery from part load operation was faster, and the turbine inlet temperature drop rate was 20% lower than for the solar-only case, reducing the turbine thermal stress. Notwithstanding the relatively low solar multiple (about 1) and low biomass share (about 33% of the power block nominal thermal demand), it is possible to attain nominal and stable power generation, under steady and abundant solar radiation.

Most of the CSP plants rely on a conventional Rankine cycle. Although the advantages of steam reheat within turbine stages has been widely proven, there are technical challenges for reheat in the DSG configuration. Usually, an indirect reheat configuration is adopted, as direct reheat is hindered by the distance between the turbine and the solar field, ensuing excessive pressure losses. Therefore, the use of reheat is mostly justified by the increase of steam quality in the last turbine stage, and not explicitly associated with system efficiency enhancement. However, CSP hybridisation with a steam generator driven by biomass eliminates this constraint.

To evaluate the possibility to use direct reheat, a 5 MW\textsubscript{el} CSP/biomass power plant located in the Algarve region, in Portugal, was assessed. The solar field relied on linear Fresnel reflector technology, and the recirculation concept was adopted. Power plant hybridisation was achieved through the use of a steam generator, used for enhancing and backing up solar field heat generation, in order to increase both dispatchability and global
system efficiency. The steam generator had a nominal efficiency of 91% and was driven by biogas resultant from anaerobic digestion of organic waste. Abundant solar radiation characterises the Algarve region with an annual value of direct normal irradiance of about 2344 kWh/(m²·year). Moreover, the power demand drastically changes along the year, due to the significant tourism influx in the Summer months. Therefore, the region would benefit from a dispatchable and flexible renewable power generation system. Furthermore, demand and availability of energy resources are linked, since solar radiation is abundant in summer months and organic waste increases with human activity.

Three distinct cases (plants) were analysed: solar only, hybrid without and with reheat. If the system depended solely on solar energy, annual electrical generation would be confined to 2063 hours/year and mostly during summer months. Hybridisation resulted in an enhancement of system dispatchability and generation stability. Furthermore, hybridisation improved the annual solar field (29.7% to 32.3%) and power block (23.5% to 29.1%) efficiencies, and thus the system annual efficiency (7% to 18%).

The use of direct reheat in the turbine is usually hindered in a CSP system. This drawback was eliminated with a biomass steam generator. The results showed an enhancement of the system efficiency from 17.6% to 18.0%, associated with an efficiency improvement of the steam generator, as well as the power block. Nevertheless, it was noticed that this enhancement could be more significant if the turbine design was optimised. Also, a lower steam wetness in the low-pressure turbine stage was assured, reducing the turbine mechanical stress.

Single-phase CSP heat transfer fluids (i.e. thermal oil and molten salts) can also benefit from biomass hybridisation. The assessment entailed the use a biomass boiler within the HTF cycle and the steam cycle. Regarding the HTF cycle, a biomass combustion boiler was used to boost the HTF outlet temperature. No backup was addressed, as the TES system permits to extend operation without solar radiation.

On the other hand, within the steam cycle, a biomass syngas boiler was used to back up steam generation and also to improve the live-steam temperatures and pressure, and consequently the power block efficiency. Regarding feedstock, RDF and wood pellets were considered for thermal oil and molten salts, respectively. The reason for the use of different feedstocks is the limit concerning the maximum combustion temperature of
RDF (430°C), which does not allow the use with molten salts. Above this threshold, high-temperature corrosion and ash fusion problems are expected.

The HTF boost was evaluated during a warm solar field start-up. The additional heat from the biomass boiler permitted to boost the HTF to 391°C and consequently the power block may be driven near design conditions. In comparison to solar-only mode, the temperature loss from the storage process reduced power generation to 80% of the nominal load.

At the end of the day, solar radiation decreases and so the solar field output. Over this period, the TES was discharged to sustain power generation. If the system relies solely on solar energy, during storage discharge operation, the HTF temperature would be considerably lower (365°C) than nominal conditions (391°C). This resulted in a reduction of the steam temperature, and so of the electrical generation to about 80% of the nominal load. On the other hand, boosting the loop outlet temperature permitted to operate the power block close to design conditions. When the storage reached the lower limit, hybridisation within the steam cycle was used to enhance dispatchability. There are advantages in placing the backup boiler within the steam cycle, instead of within the HTF cycle. One is the maturity of biomass steam boilers in comparison to thermal oil. Another is related to the steam generator HEXs between the HTF and steam cycle. In the steam cycle the thermal demand is reduced, and thus a better system efficiency is attained.

A further hybridisation option consists in using biomass to boost the steam temperature. Through boosting the steam temperature from 378°C to 540°C, it was possible to drive the turbine at a higher pressure (90 bar) without jeopardising steam quality at the turbine outlet. Additionally, the power block efficiency was enhanced by 8.7%. The evaluation was carried out for steady and abundant solar radiation, and results showed that for solar-only the amount of energy stored was about 73% when compared to hybrid operation. This is a consequence of the enhanced power block efficiency and additional energy in the power block.

Regarding design considerations, the solar field can be reduced, or the storage size increased to accommodate the additional thermal energy. This is a crucial result concerning costs, as the SF and TES account for more than 50% of a CSP plant cost structure. Another advantage is related to the possibility to operate a solar field and TES...
system at lower temperatures, without jeopardising the power block efficiency. When compared with DSG the solar field outlet temperature is lower, and thus thermal losses are reduced and the efficiency enhanced. However, the steam pressure and thus the power block efficiency are augmented. Further benefits are related to parasitic consumptions. The power block and solar efficiency increases and so the mass flow rate of both HTF and steam are reduced. In the case of the HTF, the nominal mass flow rate within the solar field was reduced to about 78.5%.

In the case of molten salts, the main differences were the loop temperatures and mass flow rate, and also the use of direct storage. The loop mass flow rate was about 60% lower when compared with thermal oil. The loop outlet temperature was defined for 545°C (about 39% higher than with thermal oil). Whilst the solar field nominal thermal power was marginally lower than for thermal oil, the solar multiple was higher (about 1.36) due to the power block higher efficiency. Regarding hybridisation, in the molten salts cycle, a biomass combustion boiler was used to heat up the molten salts to 545°C, eliminating the temperature drop from thermal losses at the hot tank. In the power block, the main differences were related to the steam parameters.

At the steam generator outlet, a temperature of 535°C was considered, about 44% higher than with thermal oil. The steam temperature was further increased (by 5°C) through the use of biomass, and thus the energy required was about 3% in comparison to thermal oil. The additional cost and complexity of adding another HEX in the steam loop must be considered, to properly evaluate the advantages of boosting steam temperature in case of a molten salts loop. This fact does not affect the backup boiler added value.

7.9 References

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8 CONCLUSIONS AND FUTURE WORK

8.1 Conclusions

The long-term potential of renewable energy systems is undisputable since they rely on a source that will not run out. Nevertheless, they strongly depend on meteorological conditions (e.g. solar, wind), leading to uncertainty on the instantaneous energy supply and consequently to grid connection issues. Concentrating solar power can be a reliable solution due to the easiness to dispatch energy, through the use of thermal energy storage. Another attractive concept is renewable hybridisation. It consists in the strategic combination of different renewable technologies in the power generation portfolio by taking advantage of each technology. Within alternatives, hybridisation of CSP with biomass can be a powerful way of assuring system stability and reliability.

In this study, hybridisation of concentrating solar power and biomass was assessed through case studies. The case studies entail distinct possibilities of hybrid integration and purpose. Concerning CSP the technology was limited to linear focus systems (i.e. parabolic trough collectors and linear Fresnel reflectors) by reason of the technology maturity. Also, both two-phase (water/steam) and single-phase (i.e. thermal oil and
molten salts) heat transfer fluids were evaluated, as well as the use of storage. On the biomass side, combustion, gasification and anaerobic digestion were addressed with different feedstock sources. Regarding the power block, either the conventional steam Rankine cycle or the Organic Rankine Cycle were considered. The assessment was carried out through the development of a quasi-transient numerical model, validated and improved with experimental data. The experimental work consisted in the evaluation under real-life conditions of a biomass gasification system and a concentrating solar-field.

8.1.1 Hybridisation to back up steam production

The most conventional CSP/biomass integration method consists in using a boiler to back up the solar field. For a SF operating with water/steam and in a parallel configuration with the biomass boiler, two steam generators can operate simultaneously or individually, to drive a power generation cycle. The main advantages of this layout are the improved system stability and dispatchability.

The assessment comprised different load curves scenarios, for example, twenty-four hours at full load or minimum power. In each case, results showed that hybridisation denotes a powerful way of assuring system stability and reliability, through the extension of the operating range even in the absence of a thermal energy storage system. Furthermore, electrical grid stabilisation is promoted through the system flexibility allowing to accommodate fluctuations on the demand-side.

On sunnier days, the boiler should be used only before and during the solar field start-up, and also over and after the shutting-down period. As that, solar energy is used to drive the power block over the daylight period and the boiler to enhance the SF start-up and extend operation. On the other hand, low radiation days (e.g. winter season) require more energy from the biomass boiler for power generation. In such days, and for a lower solar multiple, solar energy is unable to supply the minimum conditions to drive the turbine. Whilst biomass can be used to sustain power generation over the whole day, hybridisation permits to reduce the feedstock consumption.
Note that all boilers have a minimum operation capacity and a certain amount of time is required for start-up. Therefore, if the system is designed on the basis of energy dispatchability and demand response-ability, an excess of energy will occur. To overcome this issue, improved control strategies are required, or boilers with the ability to operate under standby conditions for longer times. Nevertheless, it is noteworthy that the advantages of operating the boiler during all day are extended to the decrease of the power block number of start-ups and shut-downs, and therefore to a reduced mechanical stress of the equipment.

The solar multiple plays an essential role in the CSP/biomass plant design, especially for a system without storage. Whilst a higher solar multiple permits to overcome the scarcity of solar energy in low radiation times, it will result in higher amounts of energy dumped during times of abundant solar energy. Furthermore, it influences solar and biomass shares and thus it is recommended to consider a compromise between the heat dumped and envisioned CSP and biomass shares, for the solar field design.

From a technical viewpoint, efficiencies from the SF and PB increase. For example, at a prototype scale (60 kWel) the solar field and power block efficiencies were augmented by 3% and 38%, respectively. Note that the slightly low SF operating temperature, limit the improvement of the SF efficiency.

The solar field efficiency enhancement is related to the improved conversion of solar radiation at the beginning of the day. In a solar-only power plant, early system heat up can be easily accomplished by using a biomass boiler. This leads to an increased power block efficiency, related to the operation conditions of the turbine, since the system can be driven close to design conditions. Also, commercial steam boilers are characterised by higher efficiencies and ability to operate a part load. Consequently, the global system efficiency is augmented. For example, for the prototype characterised by a solar multiple of 1, and under the assumption of twenty-four hours of daily operation, the system efficiency was augmented by about 282%, yet at the expense of solar share (17%).

From the economic point of view, there are several advantages even for small-scale power plants characterised by a lower conversion efficiency. The downside of the hybrid
system is that unavoidably the initial investment is higher than in a conventional CSP plant. The solar field and biomass system account for almost 80% of the capital expenditure, of which 55% are related to the SF. Regarding operational expenditure, anaerobic digestion has the highest share (over 55%). This value is related to food waste anaerobic digestion, which entails a complex pre-treatment and on-site handling, and thus higher costs. In the case of simpler biomass conversation technologies or feedstocks, the costs are expected to decrease. Results show a Levelised cost of electricity of 0.175 €/kWhel, which is favourable considering the small scale of the power plant (1MWel). This result is a consequence of hybrid configurations. Whilst the AD system significantly increases the CAPEX, it improves the capacity factor. Regardless of the PB low nominal efficiency, the annual operation is carried out near design condition, eliminating even lower efficiencies from partial load operation. Moreover, the joint use of power plant equipment (e.g. both SF and AD system share the same PB) enable the cost dilution.

The hybrid system requires both abundant solar radiation and feedstock. Therefore, CSP/biomass hybridisation allows CSP migration from desert areas to load centres, closer to power consuming centres where biomass feedstock is more available. Consequently, the use of the power block waste heat is possible, i.e. a combined heat and power system. The additional use of heat plays an important role, particularly at a small-scale where turbines are characterised by a lower isentropic efficiency. Despite the increase of capital expenditure and reduced power block efficiency, the levelised cost of electricity for a CHP system is lowered to 126.3 €/MWhel through the added heat revenues.

The attractiveness of the costs can be further improved through a subsidised feed-in tariff, which was not considered in this study. The ability to provide dispatchable power through a fully renewable system can endorse a subsidised tariff. Furthermore, the case study was assessed for Tunisia, which is characterised by massive subsidies to natural gas suppliers. If those subsidies were accounted for, the heat revenues would be twice as high, and the levelised cost of electricity for a CHP would be even lower. Additional reductions may result from the level of temperature of the heat, which may allow its use for cooling purposes. Furthermore, Tunisia faces issues concerning waste management, mostly subsidised by the state, for which AD is a promising solution.
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The added value of this hybrid configuration is more significant for a solar field operating with water/steam, as a consequence of the absence of a reliable latent heat thermal energy storage.

Nevertheless, single-phase heat transfer fluids can also benefit from a biomass backup system. The above advantages are also noticed. The main differences are related to design considerations, i.e. the boiler location and the added value from thermal energy storage. In this study, the backup boiler was placed within the steam cycle, instead of the HTF loop. The inherent advantages are the steam boiler technology maturity and the lower demand for heat.

Regarding the use of a TES system, technical and economic advantages should be assessed for each case. Whilst the hybrid system does not require storage tanks to extend operation during low radiation times, several benefits are noticed. Storage can improve the plant reliability and controllability by acting as a buffer during the boiler start-up. Additionally, it can improve the utilisation rate of the solar energy. On the other hand, thermal storage increases the project capital expenditures, and so the overall economic balance needs to be evaluated.

Regarding the use of biomass to back up the solar energy, anaerobic digestion and gasification were evaluated. Both conversion technologies permit to augment the hybrid plant capacity factor and operation stability. In the case of anaerobic digestion, the solar share is hindered. Although biogas production is not entirely constant, the chemical conversion system operates permanently. Therefore, a gasometer is required to store the gas when solar energy drives the system, and after that, the gas is used to improve the plant capacity factor. On the consideration that the anaerobic digestion system can supply enough energy to drive the power block at nominal conditions, the quantity of gas to be stored is equivalent to the number of nominal hours when the boiler is off. Consequently, the gasometer needs to be designed for sunnier times. To avoid an oversized system and biogas waste, discharge and thus power generation should occur every day.

On the other hand, gasification permits a more flexible operation. Start-up and shutting-down can be controlled merely by a gas blower. The same advantage can arise
from combustion technology. However, gasification requires a lower amount of air and thus results in a lower amount of syngas and a reduced size of the downstream equipment. Both are essential features considering the biomass backup system size. The downsides are the maintenance and technical issues.

Regarding feedstock, the primary rules are the plant location proximity, in order to reduce both costs and logistics, and the feedstock availability, considering a seasonal variation. Although practically predictable, the seasonal variability affects the consistency of supply, whilst storage and transport can significantly increase the cost.

Although the use of waste (e.g. food waste) as feedstock is characterised by a low cost and permits to address waste disposal issue, the absence of a waste feedstock market can hinder the use. Alternatively, commodity fuels (e.g. wood pellets) can be used, yet at the expense of a higher feedstock cost. Additionally, to the above-mentioned variables, the boiler steam parameters also conditionate the feedstock. For example, the use of RDF is limited to a steam temperature of about 430ºC and thus is not suitable for improved steam cycles.

8.1.2 Hybridisation to boost HTF and steam temperatures

CSP/biomass hybridisation is not confined to the use of a backup boiler. By enhancing the HTF temperature and further superheat steam, it is possible to improve the overall system efficiency and address specific technical issues.

Whilst direct steam generation operation under the Once-Through mode presents advantages over the conventional recirculation mode, specific issues need to be assessed. For example, there is a safety limit for the temperature difference within one receiver cross-section in the superheating section. To overcome this issue, a minimum flow rate within the absorber is imposed, which hinders the SF outlet temperature under low radiation periods.

A hybrid tested solution consists of using a set of gasifiers to backup solar thermal generation and boost the outlet steam temperature. In this case, the producer gas is burned in a steam boiler to back up the solar field. Nevertheless, gas conditioning is advisable
and requires lower temperatures. Therefore, a superheater heat exchanger is proposed to reduce the flammable gas temperature (about 650ºC) and boost the SF outlet steam temperature. To further improve the system controllability and thus reliability, a producer gas storage tank is used to act as a buffer between the superheater and boiler demand.

During the SF start-up, the SF temperature is initially limited to the saturation temperature (under phase change) as a consequence of the minimum loop mass flow rate. Over this period, the additional heat exchanger is used to support evaporation, and afterwards to superheat the steam temperature, whilst the backup boiler assured power generation. An additional advantage is the possibility to stabilise electricity generation by controlling the producer gas flow rate in the heat exchanger, through the use of a gas storage tank.

Over a significant solar radiation transient, the hybrid plant is not able to stabilise electricity generation, but several advantages are observed. Concerning power, results show a minimum value of about 70% of the nominal load and the absence of power shortages during the entire solar radiation transient, as it would occur in the case of solar-only. Furthermore, a faster recovery was noticed. The superheater permitted to attenuate the solar field outlet temperature reduction, and together with the backup boiler to reduce the turbine inlet temperature decrease rate to about 18ºC/min (20% lower than with solar-only), reducing the thermal stress at the turbine.

Most of the CSP plants rely on a conventional Rankine cycle. Although the advantages of steam reheat within turbine stages had been widely proven, there are technical challenges for reheat in the DSG configuration. Usually, an indirect reheat configuration is adopted, as direct reheat is hindered by the distance between the turbine and the solar field, ensuing excessive pressure losses. However, CSP hybridisation with a steam generator driven by biomass eliminates this constraint, by including an additional superheater HEX in the biomass steam generator. Results show that a lower steam wetness is assured in the low-pressure turbine outlet, reducing the mechanical stress. A further advantage is improved annual system generation by about 2.5%. Note that these advantages can be enhanced if the turbine design is optimised.
For single-phase heat transfer fluids, two enhancements were evaluated and consist of boosting the HTF temperature and improve the steam parameters within the power block.

In the case of a solar field using thermal oil, thermal losses arise from the TES charging and discharging process. Consequently, if TES is used the HTF temperature is significantly lower, and so the PB efficiency and electrical output are reduced. Hybridisation permits to boost the HTF temperature to design value and consequently the power block is driven near nominal conditions. Furthermore, if demand requires operation at part load, it will only be noticed by a lower HTF mass flow rate. On the other hand, if molten salts are used within the SF, the direct storage concept is adopted, and temperature losses are lower. Nonetheless, the use of a biomass boiler compensates the temperature drop in the hot tank.

The use of an additional superheater within the steam cycle results in improvements for a solar field with thermal oil. The power block efficiency was enhanced in about 8.7%, by operating at higher temperature and thus pressure. Furthermore, the solar energy demand is lower and accordingly it is possible to reduce the solar field size and costs. Another advantage is related to the possibility to operate a solar field and TES system at lower temperatures, without jeopardising the power block efficiency.

In the case of molten salts, the higher temperature of the HTF already breaks down the power block steam temperature constraint, and consequently, the advantages of increasing the steam temperature are lower. So, the additional cost and complexity of adding another HEX to the steam loop must be considered. However, the concept can be useful for advanced steam cycles.

In this PhD thesis a number of concepts and research tasks were carried out, which involved novelty in methods, prepositions and findings. The most relevant are described as follows. The developed CSP/biomass quasi-transient numerical model is entirely new. Further originality arises from the methodology of splitting the solar collectors into sections and from the validation of the numerical model with DSG under transient radiation. Regarding experimental work, the assessment of a gasifier for CSP hybridisation is novel, including the part load operation, start-up and cool-down stages,
as well as the recovery of waste heat. In this work, CSP/biomass hybridisation was assessed beyond global performance, allowing to evaluate the specific system enhancements. Novel concepts to overcome CSP DSG operation issues were proposed and assessed: direct reheat using a biomass steam generator, and producer gas waste heat recovery to superheat steam in solar once-through boilers.

The findings of this PhD work lead to the conclusion that additional research should be carried out considering a more extensive range of (CSP, biomass and power cycle) technologies to explore their full potential.

8.2 Future Work

This study encompassed several case studies with a wide range of CSP/biomass hybrid options, and it can be used as a base for new assessments. The findings of this PhD work lead to the conclusion that additional research should be carried out considering a more extensive range of (CSP, biomass and power cycle) technologies to explore their full potential, for which enhancements, challenges and unexplored areas are presented in the following paragraphs.

Regarding concentrating solar power, there is an ongoing research and development. The development of novel collectors with a more significant net aperture area, highly reflective mirrors, and anti-soiling coating permit operation at a higher efficiency and thus higher temperatures. Furthermore, the use of pressurised gases (e.g. CO₂) and enhanced thermal fluids (nanofluids) outcome in new challenges. Also, the central receiver technology deployment rate and research are dramatically increasing. Whilst, in theory, it is possible to achieve temperatures of 1000°C with central receiver systems, there are still significant technological barriers. Hybridisation can be used to overcome these constraints. Concerning thermal energy storage, new approaches are being tested to overcome the latent heat (e.g. corrosion) and thermochemical storage concept challenges. It would be interesting to extend and update this study for the emerging technologies. Additionally, the use of solid sensible heat storage is an attractive option for high-temperature operation at low cost, which can bring benefits to a hybrid power plant. The
production of heat has been addressed in this study on a CHP basis. Nevertheless, the use of CSP for process heat can also benefit from biomass hybridisation.

Another motivating approach is to extend the benefit of hybridisation to bioenergy issues. For example, gasification requires thermal energy for the start-up which can be supported by solar energy. Further use, entails the solar chemical concept. Within this concept, solar energy is used to provide electricity and heat to drive the chemical reactions for the production of (bio)fuels (e.g. hydrogen) production.

Regarding the CSP/biomass benefit for the power block, four main features can be analysed: improvement of the conventional Rankine cycle; advanced Rankine cycles; Brayton cycle; and combined cycle. Within the conventional Rankine steam cycle, further improvements can be achieved by increasing feedwater preheating, or through the use of the boiler blowdown heat. The use of a supercritical Rankine cycle can offer improved conversion efficiencies, but is nevertheless currently limited to power outputs beyond 500 MWel. Hybridisation of CSP/biomass can simplify the use of a Brayton cycle, using the concept of a solar-hybrid gas turbine. This alternative offers a high potential through the improved conversion efficiency. Hybridisation can also be used to drive (supercritical) carbon dioxide cycles. The integrated Solar Combined Cycle concept is not new, yet mostly associated with fossil fuels. By replacing the fossil fuel system with biomass, it is possible to provide dispatchable and full renewable power generation at higher efficiency than conventional Rankine cycles.

It is worth to note, that despite the above topics being individually addressed, a multi-criteria analysis is required for CSP/biomass hybrid systems, including the development, modifications and innovations on plant control schemes.

This synergy requires both abundant solar radiation and biomass feedstock, and therefore it would be motivating to develop a database of possible locations including information about solar and biomass resources. Through the use of these results, it is also possible to develop a better performing numerical model to evaluate the optimal design and economic suitability of the application, through the global plant cost.
Nowadays, renewable energy technologies compete against each other on a commercial basis. The same concept should not be accepted in research. In the author’s opinion, the distinct technologies should be combined to attain the best of each other. In a broader scope, the development of smart renewable centres is a looked-for concept. Under this point of view, CSP/biomass hybrid systems can be combined with other renewable technologies to provide a more reliable grid.