

CRANFIELD UNIVERSITY

ALI AL RAMADHN

Annual Performance of a Novel Configuration for an Integrated Solar
Combined Cycle Utilising Municipal Solid Waste

SCHOOL OF WATER, ENERGY AND ENVIRONMENT
Energy & Power

PhD
Academic Year: 2021 - 2022

Supervisor: Dr. Kumar Patchigolla
Associate Supervisor: Prof. Chris Sansom
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This thesis is submitted in partial fulfilment of the requirements for
the degree of PhD

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ABSTRACT

Climate change has been a major incentive for the global power generation industry to move towards the implementation of sustainable renewable energy technologies in order to reduce the emissions of greenhouse gases, especially carbon dioxide emissions. Concentrated solar power (CSP) has established itself as one of the common renewable energy technologies for large scale power generation. A further attractive feature of this solar technology is its hybrid operation in the form of integrated solar combined cycle (ISCC) which facilitates control and ensures that the power plant is available to meet demand whenever it occurs. ISCC commonly uses natural gas to operate the combined cycle but this CSP hybrid system also has the potential to limit its use of this fossil fuel with a more environmentally friendly fuel, namely the produced syngas from solid feedstock gasification which can be accomplished by further integration of the gasification reactor with ISCC. The organic fraction of municipal solid waste (MSW) was selected for this application, both to replace natural gas as well as for its value as a waste management method. In the present work, the thesis studies and contrasts four configurations of ISCC based on two factors, the type of fuel and the level of solar thermal contribution. One configuration represents the conventional form by using natural gas (ISCC 1) while another configuration uses municipal solid waste (ISCC 2) and in both cases, the solar field generates high -pressure saturated steam using parabolic trough with thermal oil. The last two configurations are related to the research proposal for ISCC which states that this hybrid system runs on municipal solid waste and utilises enhanced solar thermal contribution. This enhanced thermal power from the solar field is used to generate high-pressure superheated steam using parabolic trough with molten salt (ISCC 3) or solar power tower with molten salt (ISCC 4). In all cases, the fuel runs the combined cycle, and the solar field operates in parallel to provide extra steam for the hybrid system. However, the use of gasification in ISCC 2, ISCC 3 and ISCC 4 generates extra steam for the hybrid system through syngas cooling system which is attached to the low-pressure section of the steam turbine cycle. In this work, models are developed to investigate the differences between the various configurations in terms of technical and economic performances using

Spain and Saudi Arabia as case studies. The results indicate that the use of a solar power tower in the proposed concept, ISCC 4, gave the highest electricity production at 646 GWh with a solar share of 12.80% under Spanish weather and 644 GWh with a solar share of 15.24% under the Saudi Arabian weather. Furthermore, ISCC 4 offered the lowest levelised cost of electricity at 28.45 \$/MWh and 28.62 \$/MWh for Saudi Arabia and Spain, respectively, when the novel concept (ISCC 3 and ISCC 4) is compared to the conventional concept (ISCC 1). The main thesis contribution was to reveal the impact of coupling the ISCC using enhanced solar thermal power with municipal solid waste gasification and its potential as Waste-to-Energy plant. Based on the study presented outcomes, The proposed concept of integrated solar combined cycle (ISCC 3 and ISCC 4) demonstrated its practicality against conventional concept (ISCC 1) due to achieving higher performance outcomes with lower costs. The outcomes of ISCC 2 in both countries presented slightly lower LCOE values than the novel concept indicating that the replacement of fuel alone did not show a significant impact against the novel concept in terms electricity production cost.

Keywords: Concentrated Solar Power, Integrated Solar Combined Cycle, Municipal Solid Waste, Gasification, CSP Hybrid System, Hybridisation, Waste-to-Energy, Renewable Power Generation, Green Electricity.

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LIST OF ABBREVIATIONS

AF	Auxiliary Firing
AP	Air Preheater
Aspen Plus	Advanced System for Process Engineering Software
b	Boiler
CATT2	Computer-Aided Thermodynamic Tables 2 Software
CC	Combined Cycle
CCGT	Combined Cycle Gas Turbine
CCS	Carbon Capture and Storage
CD/COND	Condenser
CFD	Computational Fluid Dynamics Software
CP	Critical Point
CRF	Capital Recovery Factor
CSP	Concentrated Solar Power
DA/DEAR	Deaerator
DEMs	Discrete Element Methods Software
DLR	German Aerospace Centre
DNI	Direct Normal Irradiance
DSG	Direct Steam Generation
EC/ECO/ECON	Economiser
ER	Equivalence Ratio
ET	EuroTrough
EV/EVA	Evaporator
FP	Feedwater Pump
FS	Fuel Share
FW	Feedwater
G	Generator
GT	Gas Turbine
GTC	Gas Turbine Cycle
HP	High Pressure
HPT	High Pressure Turbine
HRSG	Heat Recovery Steam Generator
HTF	Heat Transfer Fluid

HX	Heat Exchanger
IP	Intermediate Pressure
IPT	Intermediate Pressure Turbine
ISCC	Integrated Solar Combined Cycle
LCOE	Levelised Cost of Electricity
LF	Linear Fresnel
LHV	Lower Heating Value
LMTD	Logarithmic Mean Temperature Difference
LP	Low Pressure
LPT	Low Pressure Turbine
MBT	Mechanical Biological Treatment
MS	Molten Salt
MSW	Municipal Solid Waste
O&M	Operations and Maintenance
OFWH	Open Feedwater Heater
PR-BM	Peng-Robinson with Boston-Mathias Equation of State
PT/PTC	Parabolic Trough Collector
PVGIS	Photovoltaic Geographical Information System
R	Reaction
RH	Reheater
R-SRC	Regenerative Steam Rankine cycle
S	Stream in a Water/Steam Cycle
SAM	System Advisor Model
SEGS	Solar Electric Generating System
SGCC	Solar Gasification Combined Cycle
SH	Superheater
SHCC	Solar Hybrid Combined Cycle
SPT	Solar Power Tower
SS	Solar Share
SSG	Solar Steam Generator
ST	Steam Turbine
STC	Steam Turbine Cycle
SWOT	Strengths, Weaknesses, Opportunities and Threats

TET	Gas Turbine Exit Temperature
TIT	Gas Turbine Inlet Temperature
TO	Thermal Oil
TRNSYS	Transient System Simulation Software
UNEP	United Nations Environment Programme
WtE	Waste-to-Energy

LIST OF SYMBOLS

A_{ap}	Solar Field Aperture Area
c_p	Specific Heat at Constant Pressure
$cold$	Cold Fluid Parameter in a Heat Exchanger
f	Pressure Fraction
F	Purchase Cost of Fuel
h	Specific Enthalpy
hot	Hot Fluid Parameter in a Heat Exchanger
I_b	Beam Radiation
i_r	Interest Rate
J	Incident Modifier
K	Radiation Loss Factor
k	Specific Heat Ratio of Air
$\dot{m}$	Mass Flow Rate
n	Number of Operational Years
η	Efficiency
P	Pressure
P_i	Pressure at State i
$\dot{Q}$	Thermal Power
r_p	Pressure Ratio
s	Specific Entropy
sat	Saturated Steam Parameter in Heat Exchanger
T	Temperature
v	Specific Volume
$\dot{W}$	Power Output
y	Number of measured HTF temperatures
Z	Cost

1 Introduction

1.1 Climate Change

Climate change on Earth refers to a collection of notable changes in the weather and the climate in general, which are expected to lead to significant consequences in our lives. This is because climate change involves a variety of global climate events, such as increase of temperature, heat waves, storms and floods and therefore could represent the greatest health threat to humanity through negatively influencing the social and environmental determinants of health, such as the availability of safe drinking water, clean air and secure and sufficient shelter with estimated damage costs of 2 to 4 billion US dollars per year by 2030 [1]. In addition to that, the World Health Organisation has predicted annual 250,000 extra deaths may occur as the result of the substantially increased numbers of patients with malaria, malnutrition, heat stress and diarrhoea due to climate change, which will strongly affect places with weak health infrastructure renders them incapable of coping effectively without assistance in their preparations and responses [1]. While the issue of climate change might be perceived as a new phenomenon due to the recent worldwide attention and media coverage, this is far from the case; scientists have found that since the formation of Earth, its climate has been changing naturally for around 4.5 billion years, as a result of natural events such as volcanic eruptions, shifts in Earth's crust, and changes in the planet's orbit [2]. Indeed, the last 650,000 years alone have seen seven natural climate cycles related to ice ages up until 11,700 years ago, marked the start of the modern climate [3]. On the other hand, modern climate change is novel in that it has been significantly anthropogenic, especially since the Industrial Revolution in the 1800s.

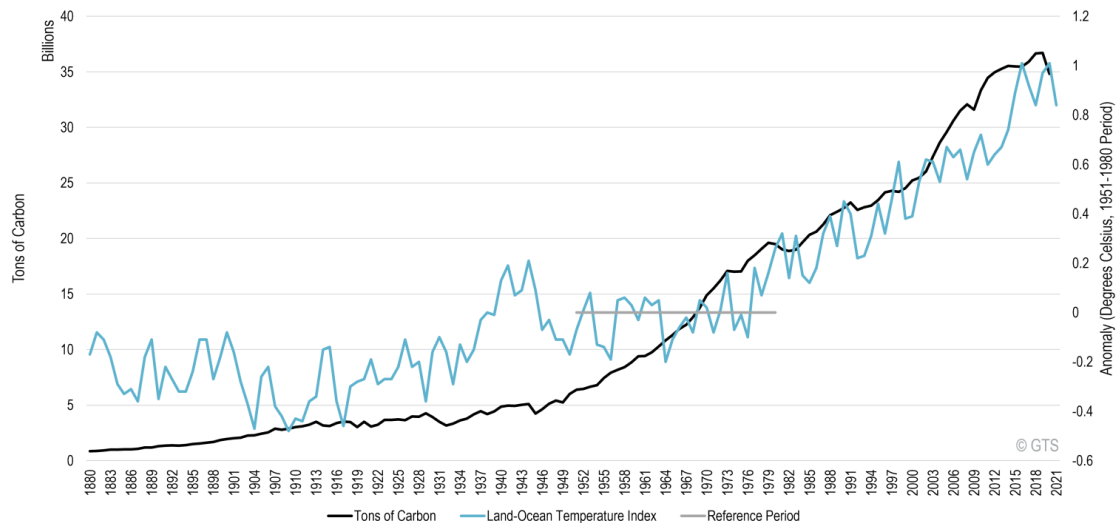


Figure 1-1 World CO₂ emissions and average temperature, 1880 - 2021 [4] (The graph also considers the deviation in degrees Celsius from the 1951-1980)

The Industrial Revolution was a major historical milestone, one that substantially altered both the consumption methods of energy resources and the level of consumption, which has led to enormous use of fossil fuels throughout the years. As a result, the carbon dioxide released from burned fossil fuels has substantially influenced the current climate by producing a sharp rise in greenhouse gases in the atmosphere beyond the natural amounts, as illustrated in Figure 1-1. Furthermore, it has been observed that from the preindustrial times until the 21st century, the overall rise of greenhouse gases by human activities for carbon dioxide, methane and nitrous oxide increased by over 40%, more than 150%, around 20%, respectively, leading the planet to transition towards a warmer climate [5]. Due to these effects, the use of fossil fuels, particularly in power generation which had the biggest carbon dioxide emissions by sector in 2021, releasing around 14.6 billion tonnes of carbon dioxide that accounted for 46% of the global rise in emissions [6]. For this reason, power generation sector has been definitively identified as one of the major contributors to climate change. As such, intensive research and development to reduce emissions from the electricity production industry is seen as a crucial factor in accomplishing a sustainable future.

As a result of this urgency, global negotiations on this matter began in 1972 with the first international environmental summit in Stockholm, Sweden, which was the primary driver behind the creation of the United Nations Environment Programme (UNEP), which seeks to promote sustainability and preserve the natural environment [7]. Since then, several conferences and summits on climate change have been conducted in locations such as Geneva (Switzerland) in 1979, Toronto (Canada) in 1988, Geneva (Switzerland) in 1990, Kyoto (Japan) in 1995, Doha (Qatar) in 2012 and finally Paris (France) in 2015, producing the accords known as the Paris Agreement [7]. The Paris conference was a historic worldwide agreement which required each nation to establish their own national long term plan with the aim to substantially achieve the agreed global targets based on a variety of key aspects [8]. One of these key aspects is to strengthen the global response to climate change for the purpose of keeping the rise of global temperature well below 2 degrees Celsius and further seeking to limit this increase to 1.5 degrees Celsius. Another key aspect is to reach the peak of greenhouse gas emissions as soon as possible in order to attain climate neutrality and therefore quickly reach the temperature goal. Moreover, the commitments of all parties to the agreement should be measured every five years in order to assess the nations' progress, share information with the other nations and encourage the developing nations in the adoption of cleaner, more climate-resilient technologies through provision of the necessary financial support. As a result of these environmental conferences, the issue of climate change now has been acknowledged not only by the majority of scientists around the world but also by governments, companies (even large fossil fuel companies) and individuals. However, conventional technologies are still strongly influential in the current stage of electricity production as shown in Figure 1-2 and a substantial shift toward renewable energies therefore must be accomplished as soon as possible.

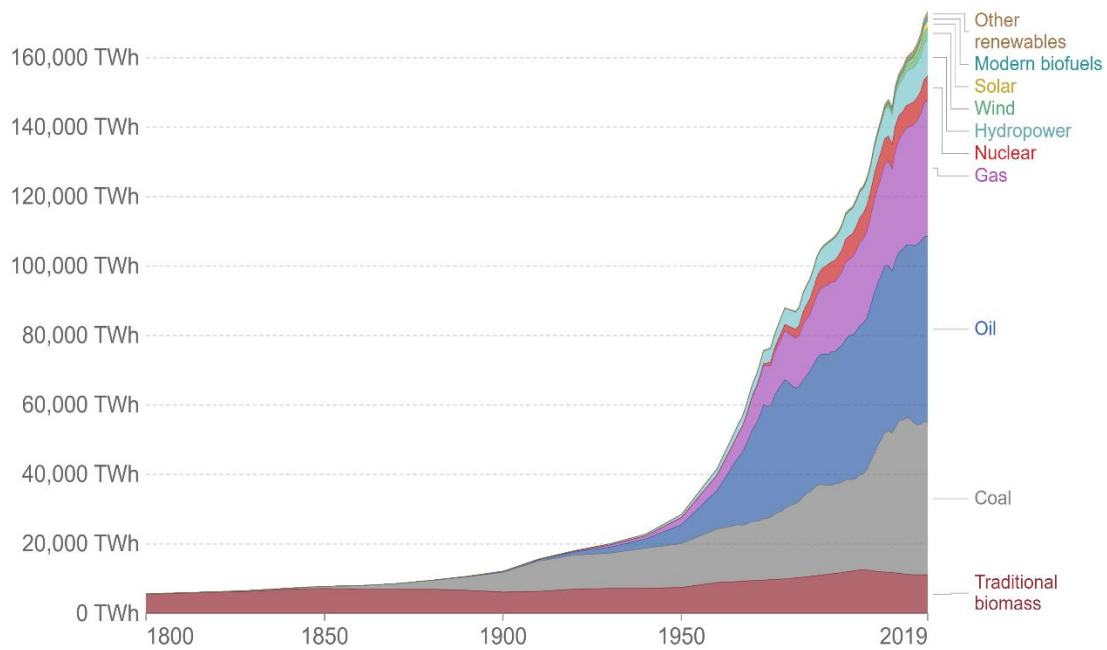


Figure 1-2 Energy consumption worldwide by source [9]

According to International Energy Agency, renewable technologies are on the rise in all world regions and have become competitive with conventional energy sources, with estimated global renewable capacity of 4800 GW by 2026, equivalent to the current combined capacities of fossil fuels and nuclear around the world, despite the rising costs of key materials to construct these technologies [10]. Furthermore, growth in renewables can be accelerated when governments address national key barriers, including permits and grid integration challenges, social acceptance of projects, insufficient remuneration, inconsistent policy approaches and high financing costs [10]. Among those renewable resources that have the potential to address these concerns while substantially achieving beneficial targets for the environment and offering a high share of power generation, is concentrated solar power (CSP). This technology is predicted to reach a global electricity contribution of 7% by the year 2030 and 25% by the year 2050 [11] in addition to creating a significant number of jobs in the power generation market [12].

1.2 Role of Concentrated Solar Power

It is widely believed that solar energy alone can outperform other renewable energy sources in satisfying global energy demand while at the same time maintaining the desirable aspects with regards to the environment and safety [13]. This belief is largely based on the fact that the Sun is the most abundant source of energy for Earth, with the solar power rate having been estimated to reach 120 petawatts, meaning the amount of solar energy reaching the surface of the planet in a single day can fulfil global energy demands for more than 20 years [14]. For this reason, the development and innovation of solar energy technologies, including CSP, has become a major target for a variety of institutions in academia and industry. This technology, which is sometimes referred to as a solar thermal system, uses curved or flat mirrors that are placed on the ground to harness solar radiation through the concentration of sunlight onto a small area in which radiation is converted into heat input to a thermodynamic cycle, often the steam turbine cycle. Some of the main characteristics of CSP are its expected high efficiencies, due to the use of high temperature input with thermodynamic cycles; its sole dependency on the direct component of received solar radiation, such that the diffused and reflected components are lost; the need for high values of direct normal irradiance (DNI); and the suitability of the technology for large-scale power plants due to the high capital cost [15]. Since this solar plant technology is strongly dependant on DNI, the practicality of CSP varies significantly around the world based on radiation intensity, as illustrated in Figure 1-3. Hence, regions with high DNI are present on almost each continent, but only in specific regions with little cloud cover, which are often in arid, desert regions such as Australia, Western Asia, North Africa, North American Southwest and Southern Europe. Furthermore, CSP technology has already been commercially implemented around the world, with a total capacity of 9162 MW, composed of 6246 MW of operational plants, 1424 MW under construction and 1492 MW under development, as shown in Figure 1-4.

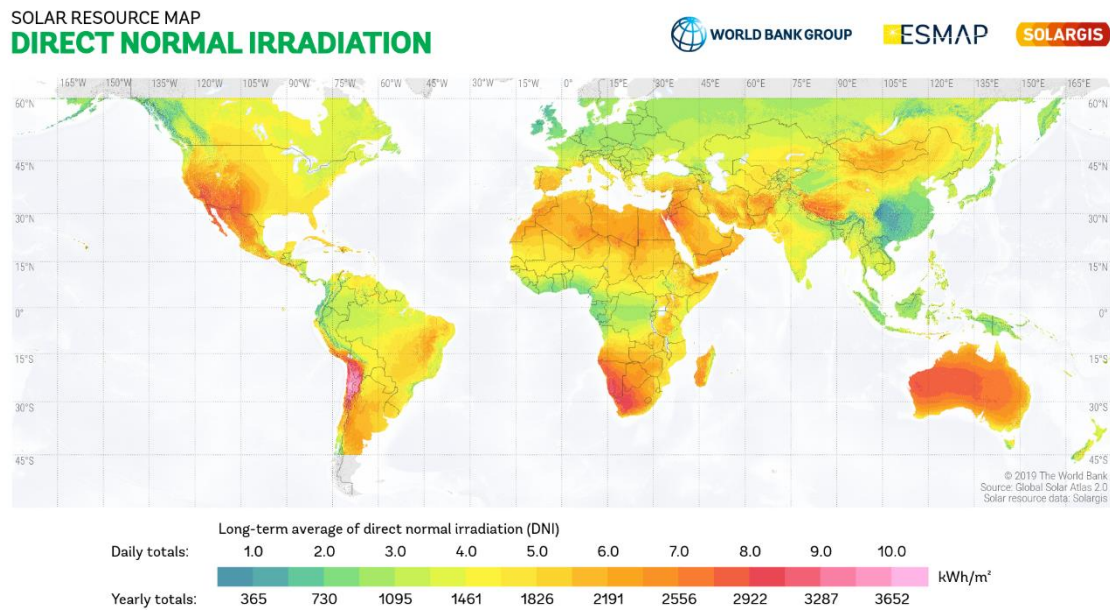


Figure 1-3 Worldwide DNI map [16]

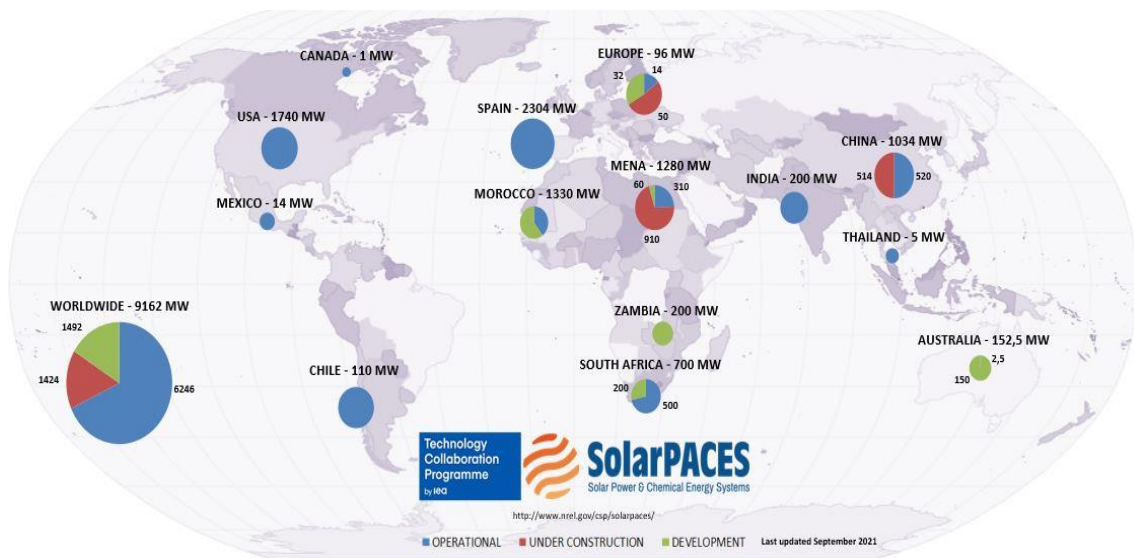


Figure 1-4 CSP projects worldwide map [17]

Another characteristic of the CSP industry is the ability to offer energy dispatchability which keeps the solar plant operating at night or during cloudy weather, not only through thermal energy storage but also through hybridisation with other technologies, which can be from renewable or fossil fuel resources. However, the former dispatchability method tends to increase the overall cost of the CSP plant, and as such, the latter method has gained much attention in recent years [18]. The basic concept of a hybrid power plant is the combination of two

or more distinct sources of electricity production in one power plant in order to compensate for the weaknesses of one source by taking advantage of the strengths of the other [19]. The attractiveness of hybridisation concept usually attributed to several benefits, such as a reduction in capital costs by sharing equipment between several energy sources, and an increase in reliability [20]. Furthermore, the operation of a hybrid system can be carried out in both off-grid (standalone) and on-grid modes [21]; as such, power plants that incorporate at least one renewable technology are very attractive in remote localities, where it is usually economically unfeasible to expand the central electricity network to those areas [22]; more importantly, such hybrid systems can help protect the local environment from being disturbed by urbanisation. It is worth noting that the earliest studies on the concept of hybridisation were conducted during the 1980s, but global attention began in earnest after 2005, with significant growth in research and development over the years [23]. Hence, there have been various investigations on how different renewable energy technologies perform when integrated with other energy resources. In the case of CSP hybridisation, there are several hybrid systems examples which include coupling concentrated solar power with biomass, geothermal, wind, coal and natural gas in various integration concepts [24], the features of which are summarised in Table 1-1.

Table 1-1 General features of CSP hybrid systems with other energy sources

	Configuration	Advantages [24]	Disadvantages [24]
Coal	<p>Solar steam generation in parallel to coal boiler [25]: Solar power tower using molten salt at 571 °C.</p> <p>Feedwater preheating for coal boiler [26]: Parabolic trough using thermal oil at temperature of 260 °C.</p> <p>Air preheating for coal boiler [27]: Solar power tower using air with ceramic particles at 540 °C.</p>	<p>Uses common, low-cost linear concentrator technologies, can retrofit existing coal plants, low cost and abundant fuel, many possible injection points for solar at different temperatures, coal provides dispatchability, capital cost savings from shared equipment.</p>	<p>High-emission fuel, equipment age mismatch may be an issue for retrofitting, efficiency limited by Rankine cycle power block, high-temperature gasification needs technological development.</p>

	Configuration	Advantages [24]	Disadvantages [24]
Natural Gas	<p>Inlet air heating for gas turbine [28]: Parabolic dish using air at 900 °C.</p> <p>Steam-injected gas turbine [29]: linear Fresnel using water at 238 °C.</p> <p>Solar with combined cycle [30]: parabolic trough using thermal oil at 393 °C.</p>	<p>Low-cost and abundant fuel, low-emission fuel, many possible configurations, many possible injection points for solar at different temperatures, efficient ISCC operation possible, provides dispatchability, capital cost savings from shared equipment.</p>	<p>High-temperature solar technology still needs development, efficient ISCC has relatively low solar share.</p>
Biofuels (from organic solid feedstock)	<p>Solar steam generation in parallel to boiler [31]: Solar power tower using molten salt at 540 °C.</p> <p>Solar contribution in processing feedstock [32]: solar power tower using air at temperature of 877 °C.</p>	<p>Potential for purely renewable power generation, many possible configurations, many possible injection points for solar at different temperatures, biofuel provides dispatchability, capital cost savings from shared equipment.</p>	<p>Limited biofuel availability may require other fuel sources, high-temperature gasification needs technological development.</p>
Geothermal	<p>Solar steam generation in series after geothermal heat exchanger [33]: Parabolic trough using thermal oil at temperature of 170 °C - 200 °C.</p>	<p>Potential for purely renewable power generation, geothermal provides some dispatchability, utilizes low-temperature and low-cost solar collectors, opportunities for multi-generation to improve efficiency, capital cost savings from shared equipment.</p>	<p>Limited to low temperature, inefficient power cycles, multi-generation systems require co-location of solar, geothermal, and multiple heat sinks, geothermal brine can cause corrosion and scaling issues on solar equipment.</p>
Wind	<p>CSP and wind hybridisation at grid level [34]: Parabolic trough using thermal oil at 380 °C.</p>	<p>Potential for purely renewable power generation, different energy availability profiles provide some reliability.</p>	<p>Energy sources not dispatchable, little synergy at equipment level as hybridisation is generally only at grid level, no capital cost savings from shared equipment.</p>

1.3 Integrated Solar Combined Cycle

Integrated solar combined cycle (ISCC) is a term that can describe any hybridisation between concentrated solar power and a combined cycle, which can be configured in a variety of ways depending on the application. The most common concept of ISCCs tends to use the solar steam generator of the solar field to generate steam in parallel with the steam heat recovery steam generator (HRSG) of the combined cycle, to either boost electricity production or reduce fuel consumption in the power block [35]. In commercial power plants in Table 1-2., the solar contribution typically uses a parabolic trough with thermal oil to extract part of the preheated water from using combusted natural gas exhaust in the gas turbine, to be further heated in the solar field, turned into steam, then returned to the HRSG to be superheated by the exhaust gases of the gas turbine, as illustrated in Figure 1-5. This use of solar field increases the overall amount of produced steam. Hence, the steam turbine needs to be oversized in order to handle this rise of steam mass flow rate leading the hybrid system to have higher power generation capacity. The first proposal of this type of integration of solar thermal technology with modern combined-cycle power plants can be traced back to the early 1990s at Luz Solar International in California, United States [36]. Later, the interest in ISCC that arose at the beginning of this century caused the Global Environment Facility agency to grant the construction of some plants to developing countries [37]. This hybrid system is currently considered to more effectively utilise the environmental benefits of solar energy technology, while at the same time taking advantage of the high-efficiency combined cycle, which generally has an efficiency of more than 50% [20].

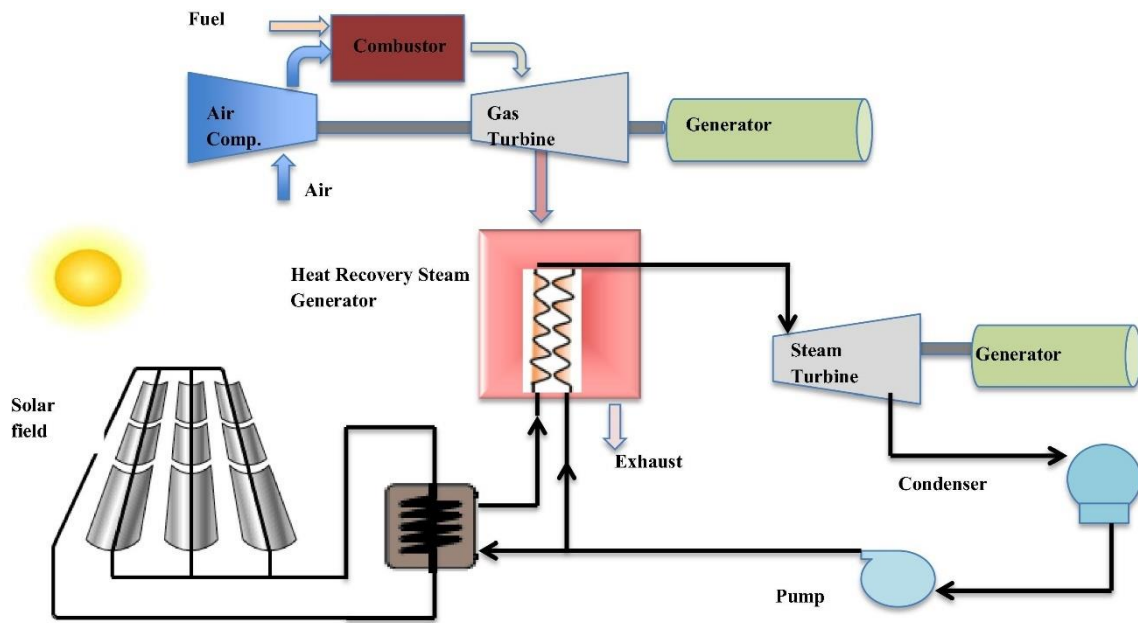


Figure 1-5 An example diagram of an ISCC plant [38]

In fact, integrated solar combined cycle systems have several advantages when compared to a standalone CSP. For example, these systems can provide an improved conversion efficiency of solar radiation into electricity [39], they can offer avoidance of inefficiencies induced by daily shutdown and start-up, they tend to have a reduced incremental cost for a larger steam turbine, they show overall better design performance throughout the year [40–42], their solar input can be seen as a balancing factor to the remarkably decreased power output and efficiency of combined cycles during the hottest days of the year [43] and lastly, the sharing of equipment and the exclusion of thermal energy storage can significantly drop costs in ISCC systems by 35–40% against solar only plant [38]. Due to the expected benefits of coupling CSP with combined cycles, governmental organisations and energy companies have been motivated to seriously invest on this hybridisation concept [39]. Based on the growing interest in ISCC research and development, it is estimated that this CSP hybrid technology will be a dominant source within the power generation industry in the near future [44]. There currently are several integrated solar combined cycle power plants with different plant capacities in different parts of the world. For instance, Morocco has invested in an ISCC power plant named Aïn Beni Mathar with a solar electricity share of around 6%, situated in the eastern part of the

country, about 88 km south of the city of Oujda [36]; Egypt has constructed the Kuraymat ISCC, with a 12.5% solar contribution, located 90 km south of Cairo and about 2.5 km from the eastern shore of the Nile [36]; the Hassi R'mel plant, located in northern Algeria [45], has around 12% solar electrical share; and lastly, the Martin Next Generation Solar Energy Center in the United States, located in western Martin County, Florida [36], has the largest total capacity of all ISCCs but a very low solar electricity contribution of around 2%.

Table 1-2 Some global natural gas-based ISCC plants [46–54]

Project Name	Location	Capacity (MW_e)	Solar Elec. (MW_e)	Construction year	Status
Ain Beni Mathar	Morocco	470	30	2008	Operational since 2011
Taiba	Saudi Arabia	3780	180	-	Planning
Waad Alshamal	Saudi Arabia	1390	50	2018	Operational
ISCC Duba 1	Saudi Arabia	600	43	2016	Under construction
ISCC Hassi R'mel	Algeria	150	20	2007	Operational since 2011
ISCC Kuraymat	Egypt	140	20	2008	Operational since 2011
Martin Next Generation	United States	3780	75	2009	Operational since 2010
Victorville 2	United States	563	50	2013	Under development
Al Abdaliyah	Kuwait	280	60	-	Under development
Archimede	Italy	765	5	2008	Operational since 2010
Ningxia	China	-	92	2013	On hold

Although the integrated solar combined cycle is one of the most attractive applications of concentrated solar power technology, due to the features mentioned earlier, its primary disadvantage is its small renewable contribution when compared to other CSP hybrid systems or to CSP standalone systems that offer either a higher renewable share or completely green electricity. One potential solution to this issue is exploring the potential of the solar field for improvement in overall performance in addition to enhancement of the solar share of the hybrid system. This approach was investigated through a variety of methods, most if not all of which can be implemented either individually or together, such as studying the impact of solar field type, the solar working fluid type and its operating conditions, the production of steam with or without a solar steam generator or the placement of solar heat integration points [55–60]. Another possible solution involves the CSP hybridisation with biofuel technologies while maintaining the overall operation concept of ISCC which requires using fuel as the main thermal input and driver of the hybrid system. Furthermore, this approach clearly suggests the replacement of the fossil fuel, commonly natural gas, in the integrated solar combined cycle with a fuel that is more environmentally friendly, especially the produced gas from processing solid feedstock such as organic urban waste or municipal solid waste (MSW). The motive for considering MSW as fuel instead of other green solid feedstock is the global availability of this resource, which renders supply issues less likely when compared to feedstock from agriculture or forests. That said, the enormous availability of this resource is also a disadvantage, insofar as it requires implementing green disposal strategies.

1.4 Municipal Solid Waste for ISCC

1.4.1 Overview of MSW

Waste, trash, garbage or municipal solid waste are some of the many names for a natural result of the consumption of various goods by humans. Methods of waste disposal have taken many forms throughout history. Some of the notable events that occurred in ancient times include the development of the first recorded waste disposal method in Greece in 3000 B.C., which involved using a

landfill with large holes to dump waste. Another major milestone was the development and use of simple composting and recycling methods in China in 2000 B.C. In addition, Greece developed a law which required any waste disposal to be at least one mile from the city in order to prevent illness and preserve the beauty of the city [61]. In modern times, the management of municipal solid waste consists of waste generation control, collection, transfer, storage, processing and disposal in accordance with factors such as public health, engineering, and the economic environment [62]. However, the significant growth in the amount of generated waste in recent years has shown that traditional waste disposal methods have become less effective in satisfying the previous essential factors of waste management. This is related to increased global waste generation, which, according to the World Bank, had reached 2.01 billion tonnes per year in 2016. This waste consists of food and green waste, glass, metal, paper and cardboard, rubber and leather, plastic and wood, as illustrated in Figure 1-6. In addition, waste generation is expected to grow substantially by 2050, especially in sub-Saharan Africa and South Asia, as seen in Figure 1-7 [63].

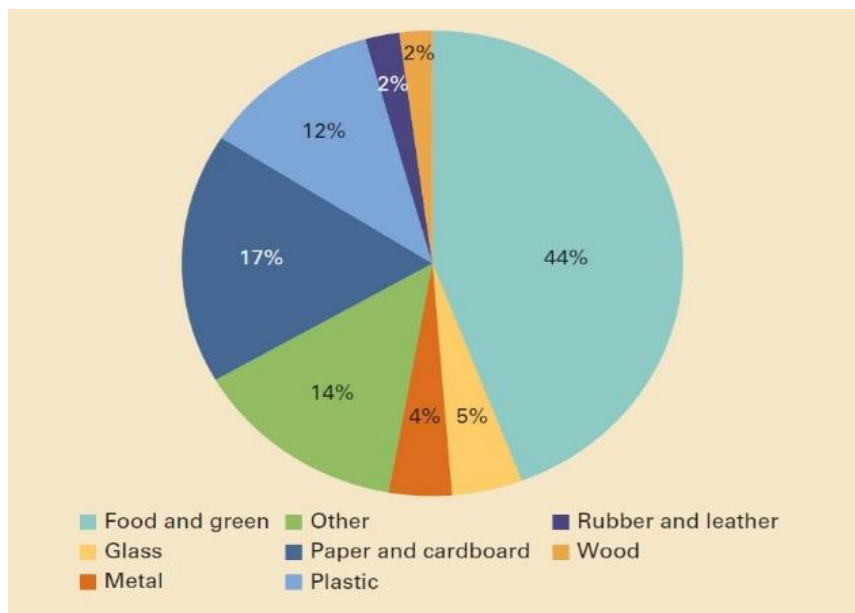


Figure 1-6 Global Waste Composition [63]

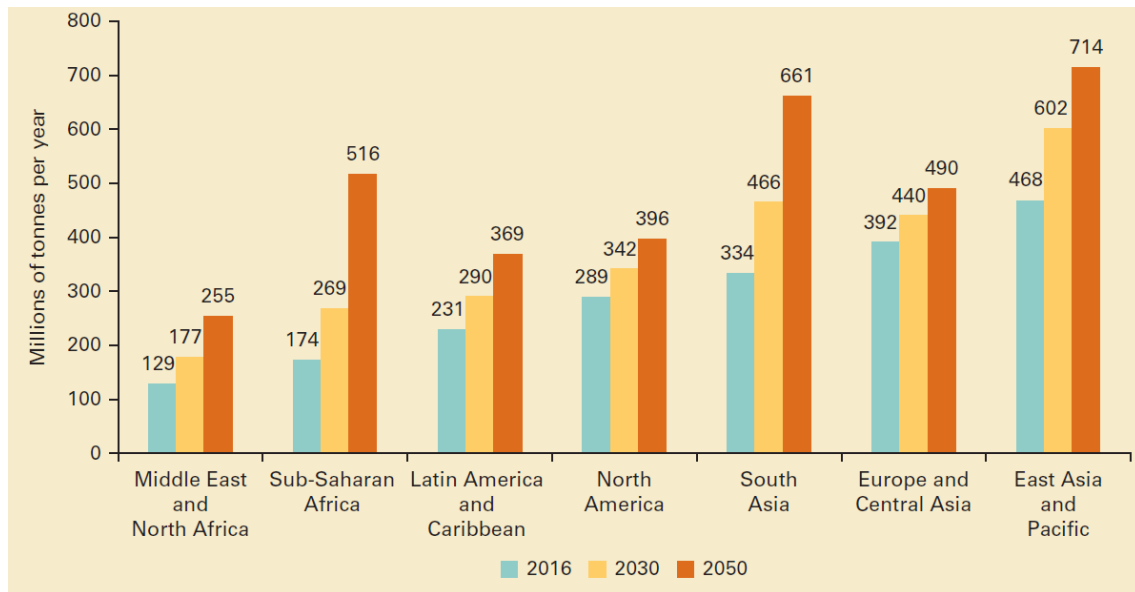


Figure 1-7 Global waste generation by region [63]

One cause of this sharp rise is the remarkable change in the concept of consumption across the years, which has involved providing the market with more options for the same product, but which generally tend to have shorter lifespans than their predecessors [64]. Another factor is related to the large migration from rural areas to urban areas, which has been influential in the level of generated waste, since urban culture tends to transform communities' lifestyles, social relationships and social structures, including the level of consumption of goods [65]. Based on the current global increase in waste production, more sustainable disposal methods are crucial at the national and global levels in order to eliminate the harm caused by landfills. It is worth noting that landfilling has been one of the most common disposal options because waste management authorities have been encouraged by certain benefits. Concretely, landfilling has the lowest costs among all the available disposal methods, it requires no skilled labour to operate, and it has the potential for landfill gas production, which when captured properly, can offer a great resource for power and heat applications [66]. On the other hand, landfilling leads to some crucial environmental and economic issues: significant emissions of greenhouse gases when landfill gas is not properly captured, substantial contamination of soil and groundwater from burying various materials and notable odour problems at the site, in addition to the need to designate a large tract of land in order to bury an enormous and continuous

supply of waste [66]. It is worth highlighting that collected municipal solid waste is not homogeneous, as shown in Figure 1-6, and its greatest portion is typically from organic matter. As such, the high percentage of organic matter in this feedstock makes it suitable for gasification technology.

1.4.2 Gasification Technology for MSW Management

Gasification technology belongs to the class of thermochemical treatment processes, which involve the conversion of solid matter into a gaseous fuel with useable heating value, thus making the gas produced suitable for combustion in order to release energy [67]. As mentioned earlier, the process of gasification converts any solid carbonaceous material into gas, which is known as synthesis gas (syngas) and is composed of H_2 , CO, CO_2 and CH_4 . This results from the use of a gasification medium or gasifying agent, such as air, steam or O_2 , in a controlled amount [68]. This feature of gasification technology has made it attractive within the Waste-to Energy (WtE) industry, since this industry is known for its ability to use waste in thermodynamic cycles for electricity production. In addition, the technology enjoys a wide range of advantages as a MSW disposal method, such as the preservation of landfill space through a notable reduction in waste mass and volume, by about 70–80% and 80–90%, respectively [69]. Gasification technology also leads huge savings in land costs since this type of WtE plant requires less space than a landfill to dispose of waste, around 100,000 m^2 for the former versus 3000,000 m^2 for the latter [70]. Furthermore, it allows for the elimination of organic contaminants within the feedstock [71] and the reduction of greenhouse gas emissions as a result of the organic waste experiencing anaerobic decomposition [70]. These are some of the general benefits for Waste-to-Energy plants versus landfilling; however, gasification technology has its own unique features and benefits. One of these features in the WtE industry is the flexibility of operation, with various energy conversion devices and each type offering different performance level for this integration, as illustrated in Table 1-3. Gasification technology as a power generation tool has the flexibility under different thermodynamic cycles technologies, with the expected benefits and considerations depending on the type selected for a

specific application. Based on the aforementioned ISCC renewable share argument, the use of gasification is clearly more suitable for implementation as the driver of the gas turbine.

Table 1-3 Comparison between gasification WtE plants based on energy conversion device [72]

	Gas engine	Steam turbine	Gas turbine
Net efficiency	14 - 26%	15 - 24%	20 - 30%
Advantages	Relatively inexpensive with high electrical efficiency, reliable and durable in addition to slight impact on efficiency under partial load.	The turbine components are not exposed to combustion products, has high electrical efficiency, long maintenance intervals, high availability in addition to high specific work. inexpensive with high electrical efficiency, reliable and durable in addition to slight impact on efficiency under partial load.	High electrical efficiency with long maintenance intervals. Compact assembly, high availability in addition to offering high exhaust temperatures.
Disadvantages	Combustion products are directly affecting the engine components material, has expensive and short maintenance intervals in addition to low availability	The impact of partial load on efficiency is significant. Expensive configuration. The plant needs to be extremely large in order to satisfy the space requirements for the heat recovery steam generator and the condenser.	The components of turbine are exposed to combustion products, moderately expensive in addition to experiencing significant efficiency reduction due partial load.
Syngas cleaning level	Tar: $100 \frac{mg}{m_N^3}$, Dust: $50 \frac{mg}{m_N^3}$, Alkalies: 0.1 ppm, wb, Heavy metals: 0.1 ppm, wb, H ₂ S: 20 ppm, vb	Tar: not limited, Dust: not limited, Alkalies: not limited, Heavy metals: not limited, H ₂ S: not limited	Tar: $10 \frac{mg}{m_N^3}$, Dust: 5 $\frac{mg}{m_N^3}$, Alkalies: 0.1 ppm, wb, Heavy metals: 0.1 ppm, wb, H ₂ S: 20 ppm, vb

1.5 Research Motivation and Scope

Based on what presented in earlier sections, the global response to climate change has been a major driving force to the growing market of renewable energy technologies in a variety of different areas. This involves two important areas, the power generation sector and waste management sector. The use of integrated solar combined cycle presents an interesting approach to utilise renewable technologies from both sectors for a higher reduction in greenhouse gas emissions. For this reason, this research centres on the impact of coupling municipal solid waste gasification with ISCC which includes exploring three main parts. Firstly, the various operating options associated with solar field in integrated solar combined cycle. Secondly, the current and potential integration methods between the gasifier and ISCC production. Finally, the available concepts to estimate and evaluate the outcomes of the gasification process.

1.6 Thesis Structure

The thesis structure is given as follows,

Chapter 1: The motivation and scope of this PhD thesis is reached through a broad overview of climate change causes, especially, the excessive use of fossil fuels and landfilling of waste. Moreover, the potential of renewable technologies to reduce greenhouse gas emissions has also been outlined. This includes highlighting the expected benefits from coupling municipal solid waste gasification with integrated solar combined cycle.

Chapter 2: The state-of-the-art review of ISCC and gasification process related to this PhD thesis is illustrated. This leads to obtaining the topic key knowledge gaps and establishing the research novelty, aim and objectives.

Chapter 3: This chapter describes the overall methodology to achieve the study objectives in order to reach its aim. For the research assessments, four configurations of the studied integrated solar combined cycle are proposed which include two configurations representing the thesis novel concept. Moreover, this chapter also contains the selection of potential locations from two different countries, Spain and Saudi Arabia, to conduct this investigation.

Chapter 4: This chapter explains the characteristics of each major component in the studied hybrid system. The model development of solar field, combined cycle and gasifier and their validation is also presented. Finally, the optimum operating parameters for the solar field and gasification process are obtained through sensitivity analyses.

Chapter 5: This chapter focuses on how the four configurations of studied hybrid system performs while using the components optimal operating parameters. The evaluation is conducted under design conditions and off-design conditions to highlight the effects of the varying Spanish and Saudi Arabian weathers annually.

Chapter 6: The economic performance of the studied ISCCs is presented in this chapter. Common cost functions are applied for each component in order to calculate the levelised cost of electricity.

Chapter 7: The chapter offers an overall discussion on the obtained annual outcomes. This involves explaining how the technical and economic outcomes are influenced based on the fuel type, heat transfer fluid type and solar field type.

Chapter 8: In this chapter, conclusions and recommendations for future work are provided.

2 Literature Review

The purpose of this literature review is to achieve an adequate understanding of the thesis subject. This includes focusing on previous studies that proposed methods of coupling the solar field with an integrated solar combined cycle and coupling the gasifier with ISCC, as well as methods of evaluating operating parameters, conditions and outcomes for both solar field and gasification technologies. This chapter is divided into three parts. First, it will cover previous studies associated with the conventional integrated solar combined cycle, which runs on natural gas, but with different configurations based on solar field modifications. Then, the research will examine the state of the art of coupling gasification technology with ISCC, with a focus on how the integration points of the gasifier influence the overall performance. Finally, the chapter will consider a survey of available studies on the estimation of gasification process outcomes, especially for municipal solid waste feedstock. Through this review, the gaps in the current state of knowledge will be revealed, thus permitting the identification of all required parameters that may affect the performance of the hybrid system in this thesis. A summary of the findings of this review, along with an explanation of certain decisions, is presented at the end of the chapter.

2.1 ISCC Research and Development

Most integrated solar combined cycles, whether already built or under construction, have a solar field that operates in parallel with the heat recovery steam generator of the combined cycle, which in most cases employs a parabolic trough solar field to produce saturated steam through the use of synthetic oil at temperatures around 390 °C [73]. However, the literature offers a variety of potential ISCC configurations which have been evaluated with respect to different criteria, such as their type of working fluids and solar collectors. One such study was conducted by Manente et al. [58], who evaluated six ISCC layouts that used a solar power tower, a parabolic trough and a linear Fresnel reflector coupled with a triple-pressure natural gas combined cycle as illustrated in Figure 2-1 with capacity of 390 MW_e creating six ISCC layouts as shown in Figure 2-2 and Figure 2-3. The study modelled the natural gas combined cycle using the Thermoflex

software. For the gas turbine sub-model, the study used a Siemens SGT5-4000F with electrical power output of 254.25 MW_e and an LHV based efficiency of 37.5%. For the high-pressure section of HRSG, there were three economiser stages, an evaporator and two superheater stages, the last of which sent the steam into the high-pressure steam turbine at a temperature of 540 °C and a pressure of 101 bar. Moreover, the intermediate-pressure section consisted of an economiser, an evaporator and a superheater, which produced steam at a temperature of 305 °C and pressure of 16 bar. The steam then was mixed with the high-pressure turbine outlet steam and directed into reheaters that enhanced the temperature of the mixture up to 540 °C, after which the mixture was introduced to the intermediate-pressure turbine. The low-pressure section then applied an evaporator and superheater, which produced steam with a temperature of 300 °C and pressure of 4 bar, which was then mixed with the intermediate pressure outlet before entering the low-pressure turbine.

The assumed power outputs from the different turbines were 31.68 MW_e, 29.33 MW_e and 76.39 MW_e for the high-pressure, intermediate-pressure and low-pressure turbines, respectively. The isentropic efficiencies of the steam turbines were calculated by Thermoflex based on the inlet turbine temperature and pressure, the outlet turbine pressure and the volumetric flow rate, which was found to be 86.5% for the high-pressure turbine, 90.9% for the intermediate-pressure turbine and 93.3% for the low-pressure turbine. The authors highlighted that these values were in agreement with reference study from the literature. The calculated efficiency for the intermediate-pressure turbine by Thermoflex was slightly higher, with a deviation of less than 1%; the only clear deviation was for the low-pressure turbine efficiency, which the software overestimated by 2% points.

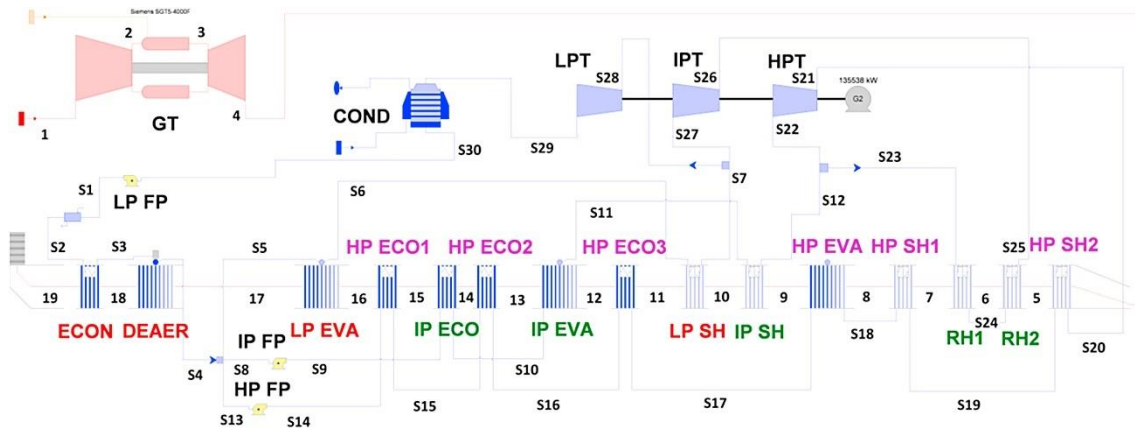


Figure 2-1 Natural gas combined cycle power plant layout by Manente et al. [58]

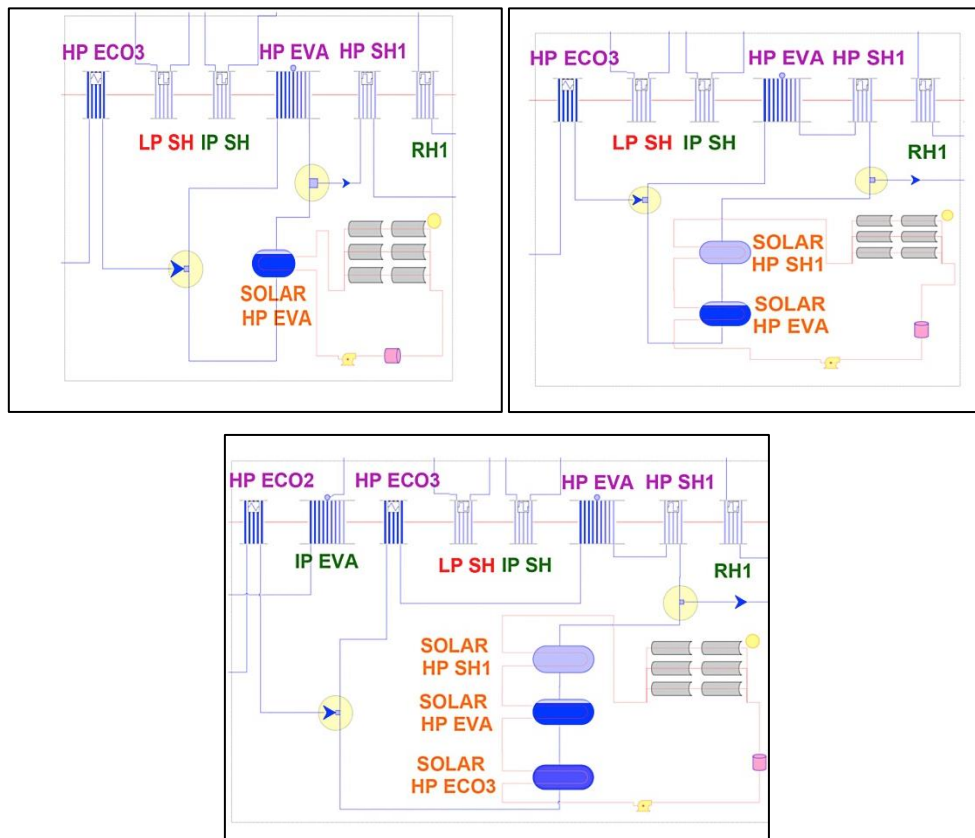


Figure 2-2 ISCC with PT layouts (top left: EVA; top right: EVA-SH; bottom centre: ECO-EVA-SH) by Manente et al. [58]

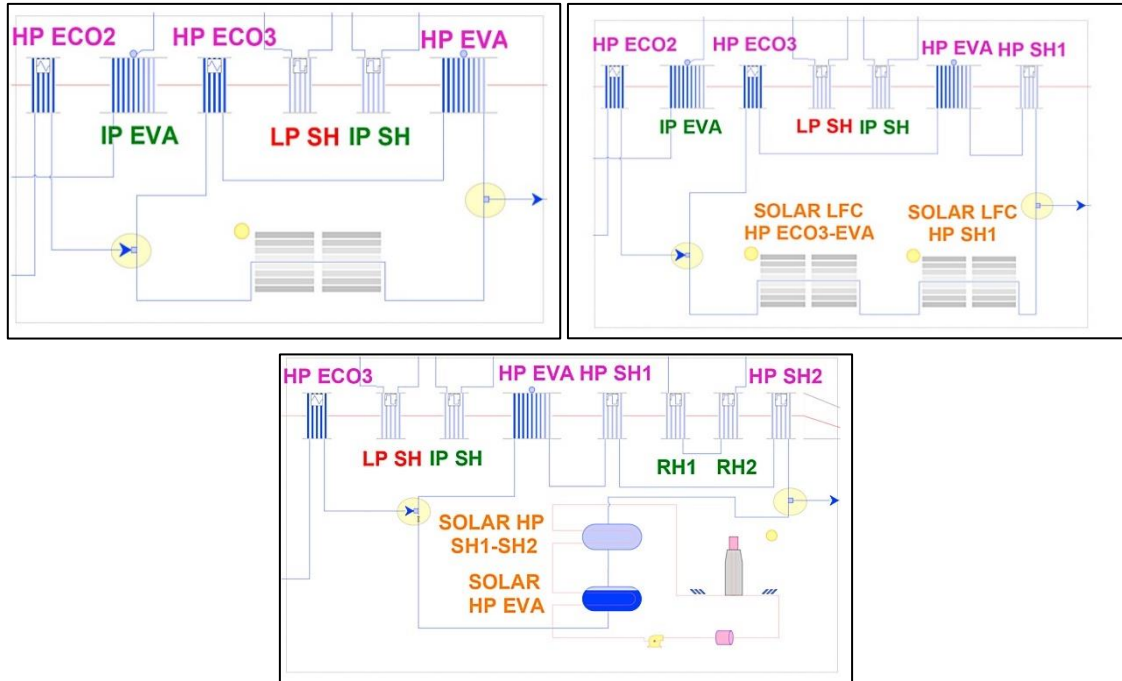


Figure 2-3 ISCC with LF (top left: EVA; top right: ECO-EVA-SH) and ISCC with SPT (bottom: EVA-SH) by Manente et al. [58]

For the solar field, the authors utilised the built-in models in Thermoflex/Peace based on a direct normal irradiance of 857 W/m^2 to evaluate the optical and thermal performance of the parabolic trough, solar power tower and linear Fresnel reflector. In the parabolic trough, the assumed geometric and optical parameters were close to the typical data of Euro Trough (ET) collectors but slightly lower in terms of peak optical efficiency, which was 75% versus 76.5% for ET 100 and 80% for ET 150. There were two types implemented in the parabolic trough: thermal oil operating at a maximum temperature of $393 \text{ }^\circ\text{C}$, analogous to conditions for the working fluid of the Andasol power plant in Spain, and molten salts, which can achieve much higher operating temperatures. In the solar power tower, the molten salt was heated to operating temperature of $565 \text{ }^\circ\text{C}$, analogous to the operating conditions of the Gemasolar power plant in Spain. The implemented receiver received concentrated solar radiation from the heliostats with an average heat flux of 675 kW/m^2 , which is a common value for receivers in solar power tower plants. For the linear Fresnel reflector, the study assumed peak optical efficiency of 67% and 64.7% for the preheating–evaporation section

and superheating section, respectively. It was observed that these values were slightly below those given in the reference linear Fresnel study in the literature.

The six layouts of the integrated solar combined cycle, as illustrated in Figure 2-2 and Figure 2-3, were represented as ISCC 1 to ISCC 6, operated using a solar field that added 50 MW_e to the overall power output and ambient temperature of 15 °C, in addition to the DNI value mentioned earlier. In ISCC 1, the high-pressure preheated water leaving the economiser HP ECO 3 was split into two streams: one went to the HRSG evaporator HP EVA and the other went to the parabolic trough solar field to be evaporated through the solar evaporator SOLAR HP EVA. The two streams of saturated steam were then mixed and superheated in the HRSG through superheater HP SH 1. In the ISCC 2 configuration, a fraction of the high-pressure preheated water leaving the economiser HP ECO 3 went to the parabolic trough solar field to be evaporated and superheated to reach 454 °C through the solar steam generators SOLAR HP EVA and SOLAR HP SH 1, respectively; the two streams of steam were then mixed in HRSG superheater HP SH 1 and further superheated to reach 540 °C. For the layout of ISCC 3, part of the high-pressure feed water went through preheating, evaporation and superheating up to 454 °C in the parabolic trough solar field economiser SOLAR HP ECO 3, evaporator SOLAR HP EVA and superheater SOLAR HP SH 1; the resulting solar superheated stream was then mixed with the boiler superheated steam in HRSG superheater HP SH 1, with the mixture reaching a temperature of 540 °C.

For configuration ISCC 4, a fraction of the high-pressure preheated water leaving the economiser HP ECO 3 went to the solar tower solar field to be evaporated and superheated up to 540 °C through the evaporator SOLAR HP EVA and superheater SOLAR HP SH 1–SH 2; after the HRSG superheaters, the two streams of superheated steam were mixed and sent to the high-pressure turbine. In configuration ISCC 5, the linear Fresnel solar field extracted a fraction of the high-pressure preheated water leaving the economiser HP ECO 2 at 207 °C to be evaporated in the solar field, which had no solar steam generator, after which the two streams of saturated steam were mixed and superheated in HRSG.

Lastly, in the ISCC 6 layout, the linear Fresnel solar field extracted a fraction of the high-pressure preheated water leaving the economiser HP ECO 2 to be further preheated, then evaporated and superheated in the solar field, after which the two streams of superheated steam were mixed and reached a temperature of 540 °C in the HRSG. Based on the study calculations, the authors concluded that the best concentrated solar power to be coupled with a 390 MW_e triple-pressure combined cycle and to provide an additional 50 MW_e of electricity was a parabolic trough with thermal oil as heat transfer fluid, indicating that ISCC 1 offered the best outcomes.

In another study, Hosseini et al. [74] investigated the performance of integrated solar combined cycles under the Iranian climate. In this study, six thermal power plants with different layouts and capacity were evaluated and compared with each other based on technical and economical characteristics: two simple gas turbines with 123.4 MW, a combined cycle power plant with 2 × 123.4 MW gas turbines and a 123.4 MW steam turbine, an integrated solar combined cycle with a 33 MW_e solar field (ISCC-33), an integrated solar combined cycle with a 67 MW_e solar field (ISCC-67), an integrated solar combined cycle with a 67 MW_e solar field and an auxiliary firing system (ISCC67-AF) and a solar electric generating system (SEGS) with a 67 MW steam turbine and a 67 MW solar field. In order to specify the solar field capacity, the steam turbine was assumed to operate at least at 50% capacity during non-solar hours, and therefore the maximum capacity considered for the solar field was 100 MW. However, the solar field cannot exceed a capacity of 100 MW for SEGS and ISCC due to the issue of rising pressure loss in the solar field, and as a result the study assumed two capacity values, 33 MW_e and 67 MW_e.

Based on an annual simulation with a month for power plant overhauls, the gross annual energy produced in the different cases was 1732 GWh (7017 hours), 2730 GWh (7343 hours), 2799 GWh (6950 hours), 2870 GWh (6453 hours), 2958 GWh (6658 hours) and 140 GWh (2092 hours) for the gas turbine, combined cycle gas turbine, ISCC – 33, ISCC – 67, ISCC 67 – AF and SEGS, respectively. The consumption of fuel in conventional power plants was found to be higher than in

the others; the gas turbine had the highest value followed by the combined cycle, while the integrated solar combined cycle systems offered lower values, with ISCC-67 providing the lowest fuel consumption at about 16% less than the combined cycle. The ISCC-67 and SEGS had the same solar field capacity, but ISCC-67 had higher efficiency, which made its fuel consumption lower. In terms of CO₂ emissions, the maximum production occurred in the gas turbine unit and was lower in SEGS, followed by the conventional combined cycle, ISCC-33, ISCC-67, respectively. When the cost of environmental effects was considered, the study results indicated that the ISCC-67 had the lowest levelised cost of electricity, which was about 10% and 33% less than the conventional combined cycle and gas turbine, respectively.

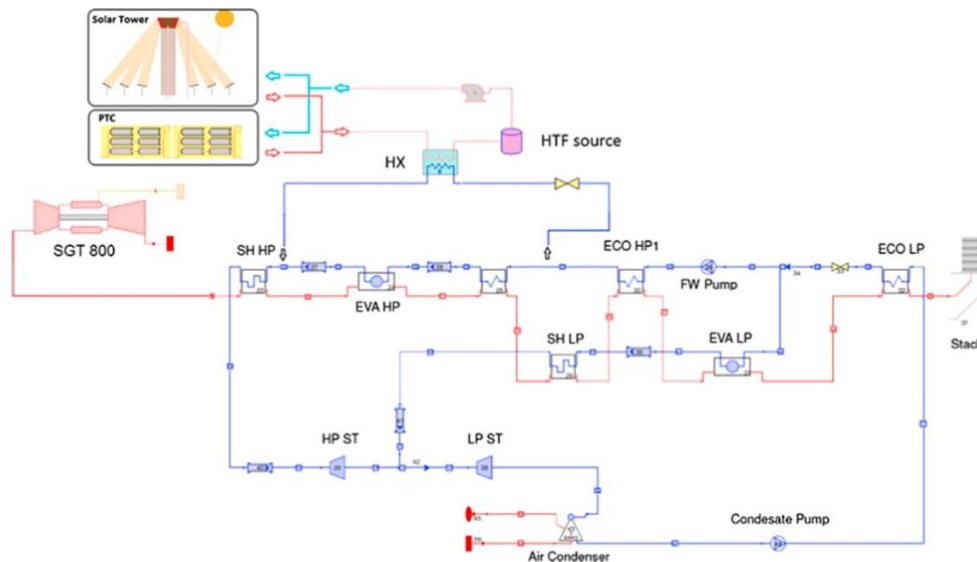


Figure 2-4 ISCC configurations based on parabolic trough and solar power tower by Franchini et al. [75]

In another study, the effect of concentrated solar power type on the integrated solar combined cycle was observed by Franchini et al. [75], who assessed the thermal performance of this hybrid system with a gas turbine SGT-800 (Siemens GT model 800), two solar fields, and a parabolic trough against a solar power tower as illustrated in Figure 2-4. For both configurations, the main assumptions included their occupying the same solar field aperture area, coupled with a 67.2 MW_e dual pressure combined cycle and boosting the net power output of ISCC by around 89 MW using the solar field thermal output of approximately 66 MW_{th}.

This was accomplished by operating the system in parallel with the heat recovery steam generator to extract a fraction of the high-pressure saturated water from the economiser to produce high-pressure saturated steam, which was placed back in the boiler. The solar fields had the same heat transfer fluid, thermal oil operating between 292 °C and 392 °C.

The modelling and simulation procedure to analyse the effect of CSP technology on ISCCs was achieved by combining in-house computer code with two commercial software packages, TRNSYS and Thermoflex, to predict yearly plant performance under actual operating conditions. TRNSYS was used to estimate the temperature and flow rate of the heat transfer fluid based on climate conditions, while Thermoflex used these HTF values as input to simulations. Heat transfer fluid circulation was considered only during the day and thermal losses at night were neglected. Moreover, the parabolic trough overall efficiency for actual operating conditions was estimated using

$$\eta_{parabolic\ trough} = J * [A + B * (\Delta T)] + C \frac{\Delta T}{I_b} + D \frac{\Delta T^2}{I_b} \quad (2-1)$$

where J is the incident modifier and influenced by the non-perpendicularity of the solar radiation angle of incidence, A is a coefficient representing the optical efficiency, B , C and D describe the coefficients for the loss of heat collector element, with ΔT representing the temperature difference between the heat transfer fluid and the ambient temperature (K) and I_b is the actual beam radiation (W/m²). For heliostat solar field modelling, an efficiency matrix was implemented, and its values were compared against experimental data by the German Aerospace Centre (DLR). Hence, the heliostat field efficiency was considered a function of the solar zenith and azimuth angles and a tower with a receiver was modelled as a simple black body absorber with a constant optical efficiency of 90%.

This was done by first considering simulations of representative days in order to observe seasonal performance differences. The results showed that when a parabolic trough was implemented in an ISCC, the hybrid system achieved greater interception of solar radiation during summer months than an ISCC with

a solar power tower; on the other hand, the performance outcomes were reversed during winter, which showed a drastic drop in the performance of ISCCs with a parabolic trough. The second step was to conduct annual simulations, which presented results indicating better performance with almost constant interception throughout the year by ISCCs with a solar power tower, which achieved annual solar-to-electric efficiency of 21.8%, an annual power output of 610.47 GWh and a solar fraction of 9% versus parabolic trough values of 18%, 603.41 GWh and 7%, respectively. The authors concluded that these outcomes were attributed to the higher collection and conversion efficiencies of solar power towers when compared to parabolic troughs. Systems with solar power towers provided the highest solar electricity production and solar-to-electric efficiency, and therefore provided higher solar stability within the hybrid system.

In a study by Turchi and Ma [76], the authors investigated the integrated solar combined cycle with several proposed configurations. The approach suggested in this study was to use conventional gas turbines with CSP facilities and exhaust gas with temperatures ranging from 420 °C to 600 °C, which was used to heat the heat transfer fluid. In the first configuration, gas turbines were coupled with parabolic troughs that used thermal oil as a heat transfer fluid with an operating hot temperature of 390 °C. The second concept involved coupling the gas turbines with parabolic troughs that used molten salts as a heat transfer fluid with an operating hot temperature of 450 °C, in addition to including thermal energy storage. Based on the results of the simulation, it was found that coupling a single 40 MW gas turbine with a 100 MW parabolic-trough thermal-oil-based plant offered lower cost and more efficiency than using two separate power plants. In the general, the solar field size of a solar only plant tends to be larger than required size to run the steam turbine in order to achieve more nominal conditions on the power block throughout the day [77]. However, this oversized solar field collects significant amount of solar heat during summer which cannot be utilised for the selected turbine capacity and therefore, this extra solar heat is dumped or wasted. Hence, Turchi and Ma [76] use of gas turbine showed that the thermal power input can be shared by the solar heat and turbine exhaust and therefore this helped in reducing the size of solar field. By doing so, the solar dumping was

reduced, the efficiency of solar conversion into electricity was improved and total costs were dropped in addition to offering a significant annual increase in power generation. For the second design, the use of a gas turbine reduced the size of the solar field and energy storage, leading to capital cost savings of more than \$100 million even after accounting for the costs of the gas turbine system.

Amelio et al. [78] conducted a performance evaluation of a novel solar system for integrated solar combined cycles. In this plant, air was extracted from the ambient environment through the compressor of the gas turbine and then was sent to a linear parabolic collector field, which preheated this air up to a maximum temperature of 580 °C under design conditions. The heated air then was introduced to the combustion chamber and mixed with natural gas in order to approach a temperature of 1200 °C to rotate the turbine of the Brayton cycle. The exhaust gases then were directed into the heat recovery steam generator in order to run the steam turbine using superheated steam, which concluded by recovering all the waste heat from the gas mixture, which was then discharged into the atmosphere at a temperature of about 110 °C. Moreover, the expected problems related to the use of heat transfer fluids such thermal oils and molten salts were avoided in this proposed configuration because the plant, particularly the solar field, had much simpler design since there was no need for a solar steam heat exchanger for transferring heat from oil to water or from molten salts to water. The modelling of the solar field involved orienting the collectors along the north-south direction to rotate them around this axis throughout the day to follow the Sun under the assumption that the system was in Almeria in Spain. This was due to the very high solar radiation in the area, which can reach about 2300 kWh/m² annual direct normal irradiance.

In this study, the direct normal irradiance value was equal to 850 W/m², which was related to other design and weather conditions: it corresponded to a time of 12:00 on June 21, an air temperature of 25 °C, a collector efficiency of 61.9% and an efficiency of 37% for the gas turbine. Due to varying ambient air temperature, the air density and turbine pressure ratio also varied, which affected the gas flow rate. All these changes were considered in the calculations by THERMOFLEX.

For ambient temperature increases in the range 0 °C–50 °C, there was a drop of 27% in the power output of the gas turbine, a drop of 4% in the power output of the steam turbine and a drop of 26% in the total power. For an annual simulation, the values pertained only to the daylight hours (3252 hours), which produced 1,192.5 GWh net electrical energy with an average yearly efficiency of 60.9%, which was higher than the reference traditional combined cycle plant without solar integration (51.4%). The amount of natural gas consumed throughout the year was found to be 152,236 tonnes, while the natural gas consumption for a traditional combined plant was 180,397 tonnes; therefore, the implementation of solar energy with a combined cycle offered a 15.5% annual saving of natural gas. It is worth noting that the authors in this study did not include the expected air pressure drop in the solar field as highlighted in Table 2-1. Hence, the outcomes this paper can be considered to have low accuracy.

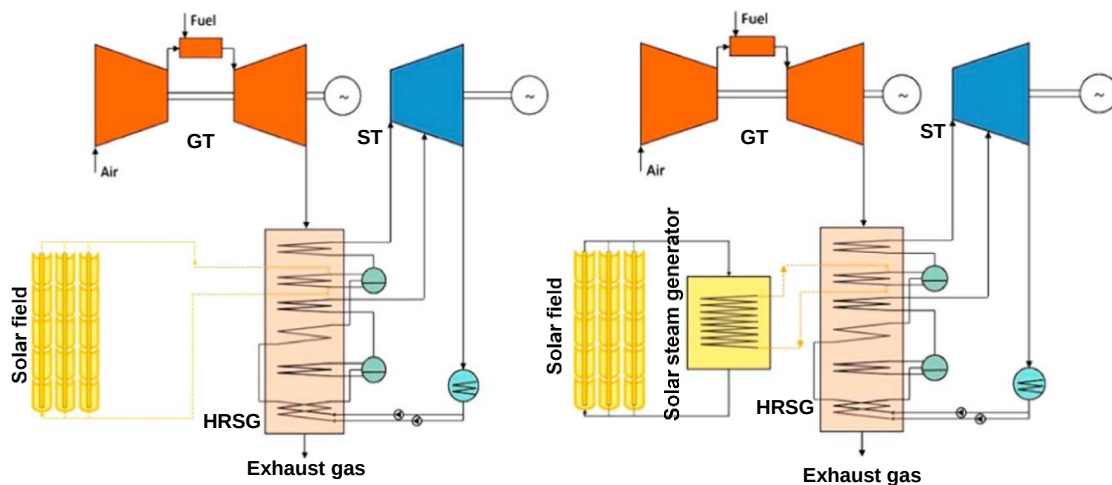


Figure 2-5 ISCC-parabolic trough evaporation configurations (DSG (left) vs HTF (right)) Rovira et al. [57]

With the work by Rovira et al. [57], the impact of solar field type on integrated solar combined cycle was not considered. Their focus was rather on how the hybrid system operated when a parabolic trough used two working fluid concepts: direct steam generation (DSG), involving water with no solar steam generator, versus heat transfer fluid (HTF), involving thermal oil with a solar steam generator, as shown in Figure 2-5. The authors examined four possible

configurations for DSG and HTF, all of which extract a fraction of water from the HRSG but differ based on solar integration points, with the steam cycle involving either evaporation, preheating and evaporation, evaporation and superheating or preheating, evaporation and preheating. In order to identify which configurations can achieve the best outcomes, three types of assessment criteria were employed to compare the configurations studied. In the first, the performances of various layouts were investigated by assuming that the solar field thermal output in all cases was fixed at 50 MW_{th}. In the second, the outcomes of these cases were observed based on the assumption that all configurations had the same solar field size or aperture area rather than constant solar thermal output. The final comparison involved fixing the size of both solar field and steam generators for all configurations.

When the net solar thermal power was fixed, the performance results illustrated some similarities between a group of layouts, namely DSG evaporation and evaporation and superheating configurations. These similarities had their counterparts in HTF concepts such as power production, maximum net incremental solar efficiency and minimum heat rate, which was attributed to the similarity of global energy balance and therefore offered a constant production of high-pressure steam. In the standardisation of solar field size, the authors found that the DSG evaporation configuration offered the best maximum power output, maximum net incremental solar efficiency and minimum heat, followed by HTF evaporation, HTF evaporation superheating and DSG evaporation superheating. For the final assessment criteria, both the solar field and solar steam generator had fixed sizes, which showed similar performance between DSG evaporation and DSG evaporation superheating; however, the simplicity of the former configuration made it a more attractive choice. The authors concluded that configurations which used solar energy to evaporate the high-pressure steam provided better performance than those with preheating. Furthermore, the HTF concept was penalised by having higher irreversibility because of the need to use an additional steam generator and therefore it was suggested that DSG configurations offer the best performance results.

Horn et al. [79] investigated the use of an integrated solar combined cycle system in Egypt from a technical and economic viewpoint, with the support from the Global Environment Facility, for two solar systems or configurations: a parabolic trough collector field and a volumetric air receiver tower. The steps to determine the incremental costs and environmental benefits included the identification of the baseline, in this case a traditional combined cycle power plant; the identification of the alternative proposed for the sake of achieving global environmental benefits, in the present case an integrated solar combined cycle; the estimation of the baseline cost; the prediction of the alternative system cost; the deduction of the baseline cost from the alternative cost in order to determine the incremental cost; and the estimation of the environmental benefits from using the alternative (the integrated solar combined cycle). The cost estimates were throughout the entire lifetime of both plants and comprised investment costs, fuel costs and operation and maintenance costs, which were used without considering inflation. For the incremental cost, the study assumed that both types of plants offered equivalent services with respects to capacity and energy for the sake of comparison, and therefore the conventional combined cycle plant was assumed to have the same plant size and capacity as the integrated solar combined cycle. The incremental cost was calculated as the difference between the costs of the integrated solar combined cycle plant and the reference combined cycle plant over the lifetime of the plants.

Regarding the environmental benefits of the alternative system, both the reference and ISCC systems were operated using natural gas as fuel, which led to CO₂ emissions. However, the integrated solar combined cycle required less natural gas than the conventional combined cycle because of its share of solar generation and therefore the operation of the ISCC lowered total CO₂ emissions. The authors assumed a discount rate of 10% over 25 years of service which resulted in a levelised cost of electricity for an integrated solar combined cycle with a parabolic trough of 3.08 US\$/kWh and 3.07 US\$/kWh for the ISCC with an air tower. In terms of capacity cost, the highest parameter was the levelised cost of electricity (2.04 US\$/kWh) for both ISCC types, whereas the cost of energy was only 1.03 US\$/kWh, with fuel costs representing more than 80% of the costs

of energy, with the remaining costs attributed to variable operation and maintenance costs. The parabolic trough configuration used a solar share of 9%, which resulted in 9.5 US\$/kWh, while the air tower configuration offered 10.19 US\$/kWh for a solar share of 8.2%. A sensitivity analysis was conducted to evaluate the impact of varying key parameters on the economic indicators of the integrated solar combined cycle.

It was found that the incremental cost was mainly affected by the cost of investment, as well as being partially influenced by the price of fuel. The effect of discount rates on the incremental cost was minimal, despite strongly influencing the costs and cost savings. Although the price of fuel had little impact on the incremental cost, when high changes in the fuel price were considered, such as doubling the price, the impact reached as much as 10% of the incremental cost. It was concluded that the estimated incremental cost of solar energy for both project types was within the limit of USD 50 million set by the World Bank. Hence, the project offered an attractive renewable option with environmental and economic benefits for power generation in Egypt.

In a study conducted by Elhaj et al. [80], Visual Basic 6.0 was used to develop a mathematical model to estimate the benefits of modifying an existing gas turbine, located in the city of Misurata in Libya, into an integrated solar combined cycle. The authors implemented the meteorological data of the city from a database with the DesalSolar 1.0 software, while the steam properties were obtained using the Computer Aided Thermodynamics software (CATT2). The authors analysed the impact of the solar concentration ratio and the solar steam generator effectiveness on the performance of the proposed ISCC, finding that modifying a simple gas turbine into an ISCC offered many advantages in terms of electricity production and thermal efficiency. Another ISCC investigation, conducted by Elhaj et al. [81], evaluated the impact of the gas turbine, the solar field, and the steam turbine on the performance of an integrated solar combined cycle. The focus of the authors was on the effect of the solar contribution and the efficiency, in addition to the primary design parameters of these main components featuring the hybrid power plant. Based on the outcome of the simulations, it was found

that the factor with the greatest impact on plant performance was the ratio of the mass flow rate of the steam to that of the heat transfer fluid. Furthermore, the performance assessment indicated that the solar field, gas turbine and steam turbine were able to achieve 62%, 40% and 32% efficiency, respectively, resulting in an overall plant efficiency of 29%.

An assessment of ISCC Southern Greece was conducted by Bakos and Parsa [82], focusing on the economic performance of an integrated solar combined cycle, in addition to investigating the impact of solar field size on overall costs. The ISCC under consideration was composed of a 15 MW steam turbine, a 35 MW gas turbine and a parabolic trough solar field. The solar field was assumed to be between 30,000 m² and 180,000 m² for the six fields considered in this assessment. The software TRNSYS-STECC was implemented to model the hourly consumption of fuel and the contribution of the solar input in two operation modes, the boosting mode and fuel-saving mode. It was found that the fuel saving mode was able to provide a lower levelised cost of electricity than the boosting mode. Moreover, a higher electricity production cost was observed when the solar field size was increased. The authors concluded that when the Greek exported electricity price was increased from 0.05026 €/kWh to 0.060 €/kWh, the ISCC could be cost effective even with larger solar field sizes.

Another investigation related to the concept of working fluid in integrated solar combined cycles was carried out by Cau et al. [60], who suggested and assessed the replacement of the conventional thermal oil with carbon dioxide as a heat transfer fluid in order to increase the solar field operating temperature and thus improve the solar energy conversion efficiency, as seen in Figure 2-6 which shows two different solar steam generator (SSG) configurations. In the first configuration (SSG-1), the solar steam generator's task was to preheat, evaporate and superheat the high-pressure stream of solar water/high-pressure steam. For the second configuration (SSG-2), the solar steam generator's role was to vaporise the stream of solar water/steam in high-pressure and intermediate-pressure sections. The preheating of water and superheating of steam for the SSG-2 configuration was achieved through the conventional heat

recovery steam generator of the steam cycle, which was based on a triple-pressure level heat recovery steam generator with steam reheating. ISCC with SSG-1 vs ISCC with SSG-2.

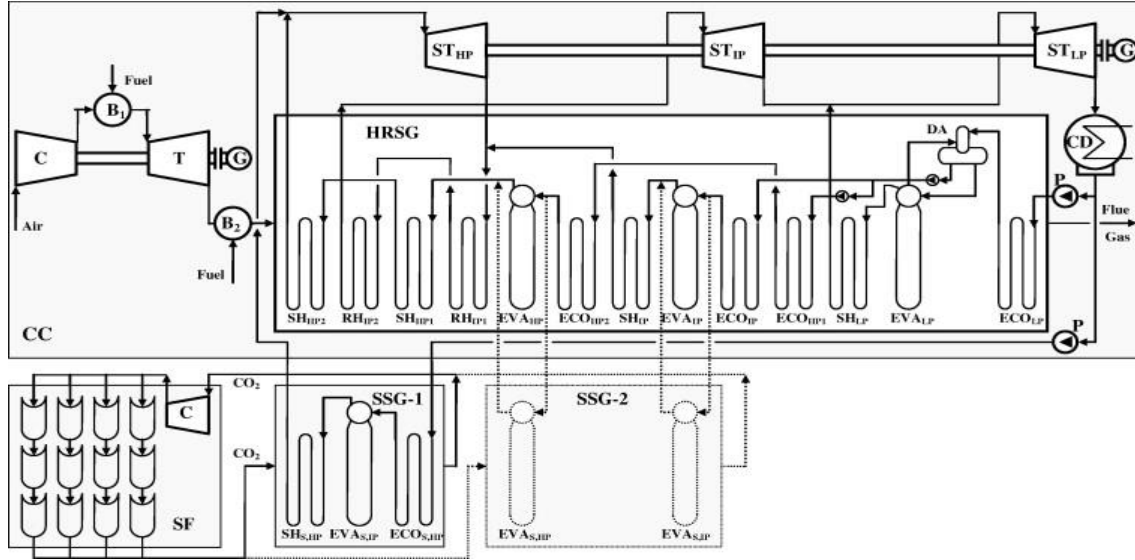


Figure 2-6 ISCC with CO₂ as heat transfer fluid configurations (C: Compressor; B: Combustion chamber; G: Generator; SF: Solar field; CD: Condenser; DA: Deaerator; P: Pump; ST: Steam turbine; CC: Combined cycle) by Cau et al. [60]

For the modelling and simulation of the proposed configurations, the GateCycle™ simulation software (version 5.40) was employed. Since this software did not have a solar collector library, the authors developed a dedicated model for a parabolic trough which simulated the solar field performance based on direct normal irradiance, the technical specifications of the parabolic trough and the composition, thermodynamic properties, temperature and pressure of the heat transfer fluid. For the combined cycle, the study assumed the use of a GEMS531 (FA) gas turbine coupled with a triple pressure level HRSG, which was able to provide a capacity of 252.6 MW with a design efficiency of 54.2% to the combined cycle at ISO conditions. The performance assessment was carried out with the solar field options evaluated at three solar steam generator inlet temperatures: 450 °C, 500 °C and 550 °C. Moreover, the study observed the effects of varying the direct normal irradiance in both SSG-1 and SSG-2 configurations, considering irradiances of 800 W/m², 850 W/m² and 900 W/m². It was found that the SSG-2 concept needed slightly more solar field area than the SSG-1 in order to achieve

the same CO₂ thermal power. For a DNI of 900 W/m² with a maximum CO₂ temperature of 550 °C, the calculations showed 75 MW_{th} of CO₂ thermal power was obtained when the solar field aperture area was 120,830 m² for the SSG-1 configuration and 121,530 m² for the SSG-2 configuration.

The net solar conversion efficiency was not dependant on the CO₂ thermal power, but rather depended only on the configuration of the solar steam generator. For a maximum CO₂ temperature of 550 °C, the net solar conversion efficiency in the proposed ISCC was 23.2% for the SSG-1 configuration with a design point DNI of 800 W/m² and 24.8% for the SSG-2 configuration with a design point DNI of 900 W/m². Moreover, it should be noted that for both SSG configurations considered, the values of the ISCC efficiency were slightly higher than in conventional standalone concentrated solar power systems (parabolic troughs with thermal oil) which have a net efficiency ranging from 20% to 23% but were close to the values for direct steam generation solar plants, with an expected efficiency range of 22% to 27%. In terms of the annual simulations, this study evaluated the annual performance of the proposed configurations against the 8760 hourly values of DNI calculated in Cagliari, Sardinia, Italy in 2005. For both SSG configurations, the annual operating hours of the solar field depended on the availability of annual DNI, which varied over the course of the year on a nearly hourly basis, since there was no thermal energy storage, which indicated that the maximum solar contribution was obtained in summer during daylight alone, since the minimum assumed DNI to operate the solar field of 200 W/m².

The solar field energy contribution was calculated as the difference between the energy from integrated solar combined cycle and the energy from the reference combined cycle gas turbine. During the winter months, measured from November to February, the solar radiation contribution to the net energy production of the studied hybrid system was lower than 1%, whereas this contribution exceeded 2–3% during the summer months, with maximum values of 3.4% for SSG-1 and 3.6% for SSG-2 measured in July. The authors concluded that the maximum designed CO₂ thermal input for SSG-2 was limited to 75 MW due to its dependency on the thermal energy of HRSG, whereas the partial connection of

SSG-1 with the boiler allowed it to have a maximum thermal energy input of 150 MW. The simulations also considered three direct normal irradiances, 800 W/m², 850 W/m² and 900 W/m². It was found that the solar energy conversion efficiency increased with higher CO₂ temperatures and vice versa. Furthermore, the analysis highlighted that using CO₂ for evaporation and superheating showed better performance than the other route, with a difference in efficiency of about 0.5%. Furthermore, the integration of solar energy with the combined cycle increased the levelised cost of electricity from 6.79 c€/kWh (combined cycle gas turbine) to 6.91 c€/kWh (integrated solar combined cycle) due to high costs related to solar field installation. It is worth mentioning that this study did not take into account the impact of CO₂ features within the solar field, as shown in Table 2-1, especially the high pressure drop and therefore the actual overall efficiency would be lower than obtained value by the authors.

In a study by Li and Yang [40], the authors introduced an ISCC with two solar fields based on direct steam generation instead of the more conventional single-stage solar input, as illustrated in Figure 2-7. This novel two-stage ISCC plant was composed of two solar fields; a gas turbine; a dual-pressure, single reheat steam turbine; and a heat recovery steam generator. From the deaerator, the liquid water was pumped to the HRSG to be preheated to the saturation level in the economisers, and then a fraction of this stream was evaporated in the direct steam generation solar fields at two different pressure levels. It was then returned to the HRSG to be superheated by the gas turbine exhaust gas before heading to the steam turbine. Moreover, the steam was returned to the boiler to be reheated after the high-pressure turbine stage. To model this configuration, the Aspen Plus software was used based on the mass balance, energy balance, and species balance of the components, with 0.01% taken as the relative convergence error tolerance. In addition, the thermodynamic models RK-SOAVE and STEAM-TA were applied for the thermal property calculations. Furthermore, General Electric 7FA was used for the gas turbine, with a 16.6 pressure ratio and 1300 °C and 595 °C for the turbine inlet and exhaust temperatures, respectively. The HRSG was assumed to provide high-pressure steam with a pressure of 80

bar, a temperature of 560 °C, a condensation pressure of 0.06 bar, and a design stack temperature of 80 °C.

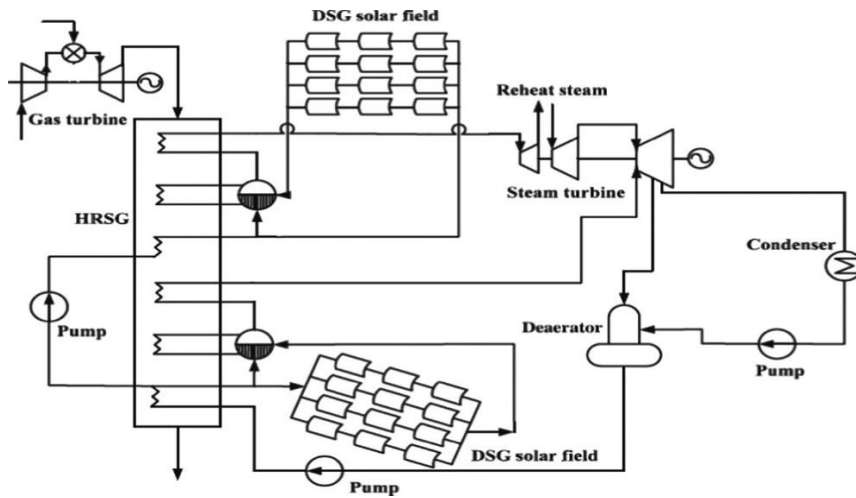


Figure 2-7 ISCC layout with two-stage solar input by Li and Yang [40]

In this study, the gas turbine was assumed to operate at full load at all times, but the steam turbine operated depending on the varying steam mass flow rate as result of the exhaust gas conditions and the solar field contribution. For the solar blocks, the solar field was assumed to be a Euro Trough ET 150 parabolic trough operating with the direct steam generation method. Furthermore, the selected location for this study was Yulin, in the Shaanxi province in China, which has an elevation of 1058 m, and the design point for the solar field was adopted based on weather data at noon on June 21st. To evaluate the system, the study aimed to obtain certain parameters, namely the thermal efficiency of the ISCC, exergy efficiency of ISCC, fuel-based net electricity efficiency, steam cycle thermal efficiency, solar share, solar-to-electricity efficiency, and fuel saving ratio. With an additional steam flow rate into the bottoming steam cycle due to the solar field contribution, the steam working parameters, pressure and temperature, for the low-pressure steam and pressure of the reheat steam, were optimised based on the assumption that the parameter values of the gas turbine exhaust gas and temperature of the stack remained constant, while the working parameters of the high-pressure steam remained fixed at a pressure of 80 bar and a temperature of 560 °C. By contrast, the working parameters of the low-pressure steam varied within the range 4 bar–8 bar for pressure and 290 °C –330 °C for temperature,

while the reheat steam working parameters varied within the range 14 bar–22 bar for pressure and 550 °C for temperature.

In this simulation, the optimum low-pressure steam pressure input that achieved maximum thermal efficiency was 5 bar, while the pressure of the reheating stage was assumed to be 16 bar. Moreover, the assumed temperature of the low-pressure steam was 320 °C, which achieved a system thermal efficiency of 53.4% and a system net solar-to-electricity efficiency of 29.8%. The authors observed that increasing the temperature of the low-pressure steam to more than 320 °C resulted in a substantial drop in system performance. This was attributed to the limited sensible heat transfer to feedwater when the gas turbine exhaust heat was fixed; therefore, the rise of steam temperature resulted in a reduction of the steam mass flow rate and net electricity output for the purpose of maintaining the minimal temperature difference in the low-pressure superheater. One of the observed outcomes in the novel two-stage ISCC system included its contribution of 27.8% of the total thermal energy input, which significantly increased the net power output and reduced gas consumption by 23.6% when compared to a reference conventional combined cycle plant with similar power output. Furthermore, this novel integrated cycle accomplished a 29.8% net solar-to-electricity conversion efficiency using solar heat input temperature less than 300 °C, which exceeded that of the one-stage ISCC power plant by about 1.2 percentage points. The steam cycle thermal efficiency experienced an improvement due to solar integration of 1.1 percentage points and 2.72 percentage points more than that of the one-stage ISCC and conventional combined cycle systems, respectively.

From the second law perspective, the proposed hybrid system reached higher values for exergy efficiency by about 2.5 percentage points and 6.6 percentage points for the one-stage ISCC and conventional combined cycle systems, respectively. A further evaluation was carried out for two representative days which differed in weather conditions. For simplification, the authors focused on varying the direct normal irradiance while fixing the ambient temperature at 25 °C and ignoring the impact of wind speed. These two typical days represented

summer, with high solar radiation (June 6th) and winter, with weak solar radiation (December 26th). The time interval of the simulation was set to one hour, since it was seen that the two-stage integrated solar combined cycle was able to operate with solar heat input for about 12 hours on June 6th, while the operation hours dropped significantly to only about 4 hours on December 26th, due to the availability level of the solar radiation. Furthermore, the system was able to operate near design conditions for long periods throughout the summer day and therefore resulted in a low partial load efficiency penalty. On the other hand, the weak solar radiation in winter provided a low solar heat contribution, which resulted in a substantial drop in net power output along with poor steam turbine off-design performance. Even though solar energy was available for more than 10 h per day during the winter, the two-stage integrated solar combined system only managed to harness useful solar radiation to operate for about 4 hours. Furthermore, the steam capacity of the proposed two-stage hybrid system was greater than that of the steam turbine in the reference conventional combined cycle in order to facilitate the extra solar energy input, which resulted in its achieving a lower net power output at night than the combined cycle plant, since the oversized turbine operated under far off-design conditions in comparison to the reference turbine.

The authors concluded that the steam turbine cycle achieved optimal conditions at 5 bar and 320 °C and 16 bar and 560 °C, for low-pressure and reheating steam, respectively, which led to the enhancement of system performance; as such, the two-stage ISCC plant offered better outcomes than a conventional one stage ISCC plant. Moreover, it was observed that there was a slight rise in the net solar-to-electricity and steam cycle thermal efficiencies, by about 1.2 and 1.1 percentage points, respectively. Under nominal conditions, the novel two-stage ISCC achieved a net electricity efficiency as high as 74%, which was 30.9% higher than the reference combined cycle value. Furthermore, the novel system accomplished 23.6% fuel savings with a solar share of 27.8% at solar heat input around 300 °C, which was found to be higher than for a standalone solar plant with the same temperature level. Finally, the annual electricity production of the

proposed configuration reached 3826.88 GWh based on a capacity factor of 73.6% and 6447 hours of full load operation.

In the study conducted by Montes et al. [41], the annual operation of an ISCC system was compared to a conventional combined-cycle gas turbine plant in order to determine the effect of solar thermal power on gas turbine performance during days with high solar radiation and high ambient temperature. To accomplish that, two sites were considered for annual simulations, Almería in Spain and Las Vegas in the United States. The suggested configuration of the studied ISCC involved coupling the solar field based on direct steam generation to the high-pressure level of the heat recovery steam generator. Moreover, the task of the direct steam generation solar field was to extract feedwater from the deaerator for the purposes of preheating and evaporating and then send this wet steam to the high-pressure drums within the Rankine cycle, to then be superheated in HRSG using exhaust gases before introducing it to the steam turbine. The solar field had a fixed size of 50 MW_{th} for thermal power production without the deployment of thermal energy storage. The combined cycle gas turbine was composed of a 2 × 1 configuration: two gas turbines with 72.5 MW_e for each turbine, a steam turbine with 75 MW_e, two heat recovery steam generators for steam production with a high-pressure of 90 bar and with a low-pressure at 8 bar.

The modelling of the hybrid system was achieved through the well-known TRNSYS software and indicated that the higher temperatures of Las Vegas influenced the combined cycle gas turbine, causing it to provide lower performance than the one in Almería, but the introduction of a direct steam generation solar field presented the opposite performance outcomes. Moreover, the results of an economic assessment showed that the hybrid system plant in Las Vegas provided a lower levelised cost of electricity than its counterpart plant in Almería. It was found that the economic outcome of the Las Vegas plant (80.52 €/MWh) was also lower than that achieved by the reference CCGT (81.29 €/MWh). However, the hybrid system did not perform identically in Almería because it provided a higher levelised cost of electricity (80.55 €/MWh) than that

corresponding to the reference CCGT plant (80.35 €/MWh). It was concluded that the integration between solar power and natural gas must be evaluated within a future scenario with very high gas prices in order to highlight the potential boost due to the solar unit.

Another two-stage approach, proposed by Li et al. [83], was similar to a previous study [40] but differed in having an integrated solar combined cycle which utilised concentration and non-concentration solar collectors. Some collectors were parabolic troughs with direct steam generation and the other collectors were evacuated tube collectors. The parabolic trough was assigned to generate saturated steam at high temperature and pressure, while evacuated tube collectors generated saturated steam at low temperature and pressure. This study assumed a combined cycle gas turbine and used the main technical parameters and collector loop inputs from the reference study [40], EuroTrough collectors for the DSG solar field and a 800 W/m² design point for direct normal irradiance. The authors compared the performances of the proposed cascade layout against one-stage ISCC and combined cycle plants, which illustrated that the proposed configuration produced higher solar heat and a significant rise in net power output compared to the other configurations, with a thermal efficiency of 53.4%. Furthermore, the suggested cascade system offered a 3.4% higher fossil fuel saving ratio than the one-stage ISCC based on the same electricity. From an economic perspective, the levelised cost of electricity and payback period were used to evaluate the proposed cascade ISCC configuration. By analysing parameters and data according to several references, the total cost of the plant was USD 482.9 million, with a specific investment of USD 816/kW, levelised cost of electricity of USD 0.06/kWh and payback period of 5.5 years. Hence, the authors concluded that the novel cascade ISCC is more economically beneficial than the common ISCC system.

Gunasekaran et al. [84] investigated four configurations of integrated solar combined cycles equipped with carbon capture and storage (CCS) technology. In the four integration schemes, the solar field was integrated in different fashions, applying either vaporisation, preheating, heating of intermediate-pressure steam,

or heating of low-pressure steam from the Rankine cycle. In the vaporisation and preheating layouts, the water/steam used was generated for the high-pressure stages of the Rankine cycle. This high-pressure steam was added at higher temperatures in comparison to the intermediate-pressure and low-pressure stages, which increased the efficiency of the hybrid power.

The inlet temperature of the intermediate-pressure turbine was 600 K, which was slightly greater than the boiling point of water at 100 bar (585 K). Hence, the heating of the intermediate-pressure turbine inlet stream added solar thermal energy in the temperature range from 600 K to 660 K, which was larger than the temperature produced by adding solar thermal energy to the vaporisation scheme (585 K), in addition to the preheating scheme (340 K to 560 K). The thermal oil Therminol-VP-1 was applied as the heat transfer fluid for all integration concepts except for preheating. In the vaporisation and heating of the intermediate and low-pressure turbine inlet streams, the solar thermal energy harnessed by the parabolic trough was used with the heated thermal oil to heat the working fluid of the Rankine cycle in a heat exchanger, while in the preheating scheme, the solar energy was directly used to heat high-pressure water after it was pressurised by the pump, and before it was fed to the drum of the boiler. The study applied ASPEN CUSTOM for modelling and simulation of the solar system, while ASPEN PLUS and JACOBIAN were used to model the power plant. The authors found that the configuration that used solar energy for either evaporating or heating intermediate-pressure water/steam managed to provide higher production of electricity than the other selected configurations in this investigation. Furthermore, the authors compared solar thermal power plants without a carbon capture system, such as solar hybrid gas turbines, to the studied configuration, finding that the integrated solar combined cycle with carbon capture, which involved integration of vaporisation, offered better outcomes than most other solar hybrid systems available in the literature.

In work by Calise et al. [43], the effect of solar heat temperature within an integrated solar combined cycle power plant, as shown in Figure 2-8, was investigated with a particular focus on the combined cycle. Theoretically, the

efficiency of the combined cycle significantly increases when the solar heat temperature is at its highest value, but this would negatively affect the solar collector efficiency. Conversely, the efficiency of the solar collector tends to improve at lower solar heat temperatures. For this reason, the authors addressed this issue by suggesting a more realistic and stable approach based on coupling the solar field with the low-temperature side of the Rankine cycle. To assess this configuration, a detailed dynamic model of a triple-pressure combined cycle was developed in order to achieve real-time operation throughout the year depending on environmental conditions, mainly radiation and temperature. Moreover, the model had to evaluate other time-dependent variables in the system such as node pressures, node temperatures, heat flows, power, and mass flow rates, in addition to conducting calculations related to energy and economic balances for the purpose of implementing thermo-economic optimisations in the system.

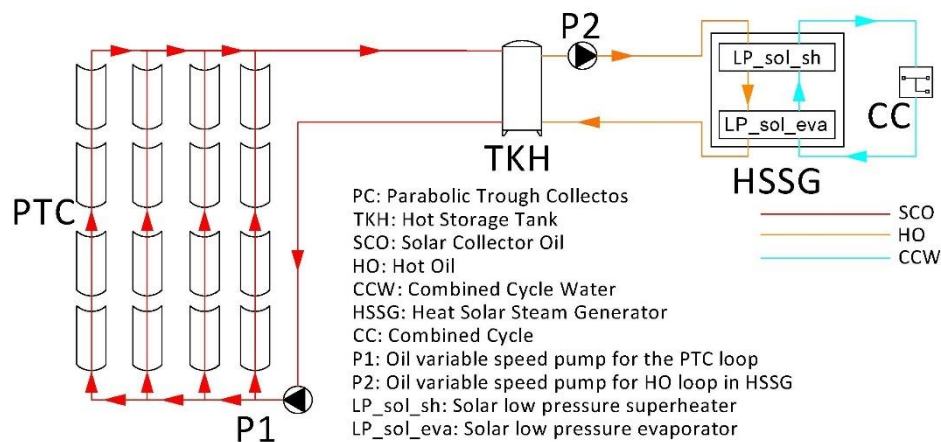


Figure 2-8 ISCC layout by Calise et al. [43]

The ISCC system was modelled in the TRNSYS software which was implemented under weather conditions in Almería (Spain) with a range of net power outputs and aperture areas to identify the optimal values. By observing the dynamic model simulations, it was found that the system studied offered a 2% solar fraction based on a solar field area of 80 000 m², along with an average solar collector efficiency of about 48% and an approximately 1% increase in the overall system electrical efficiency with respect to a conventional combined cycle. These results indicated that this hybridisation was very beneficial from a technical perspective, although the system experienced a decrease in performance in the

summer months. This is expected due to the changes in density of gas turbine inlet air leading to lower mass flow rate at high ambient temperatures [85]. Lastly, the authors highlighted that the thermo-economic model showed that the profitability of the hybrid system, in addition to the price of electricity generated via solar energy, was extremely affected by solar field capacity, and therefore a significant public funding such as feed-in tariff might be required.

El-Sayed [86] proposed applying an integrated solar combined cycle with direct steam generation at Kuraymat in Egypt. This proposed power plant consists of a 65 MW steam turbine, a 30 MW gas turbine and a solar field of 190,000 m², which was responsible for producing 90 MW of solar thermal energy at the design point direct normal irradiance of 720 W/m², in addition to operating in both power-boosting and fuel-saving modes. The selected location of the proposed plant at Kuraymat was on east shore of the River Nile, which was characterised by an annual direct normal radiation of 2400 kWh/m². Furthermore, the available land was flat desert with soil mixture of sand, soft rocks and gravel. Meteorological data provided the annual variation of solar radiation, with an average daily direct radiation of 4 kWh/m² and 9 kWh/m² in December and June, respectively. The proposed solar farm system comprised a large-aperture area of mirrors which had a parabolic shape and an operating angle of 80°, which would track the Sun's orbit by being installed in a north/south orientation. The input feedwater was pumped at a pressure of 1 bar and temperature 100 °C, then enhanced and evaporated to produce a steam output with a pressure of 80 bar and a temperature of 300°C.

Based on the results of the analysis, the cost-benefit ratio of the integrated solar combined cycle was found to be between 1.25 and 1.35, due to the mode of operation. The author concluded that the power-boosting mode showed more economical outcomes than the fuel-saving mode and that the sensitivity analysis indicated that using direct steam generation in ISCC could become feasible from an economic perspective when the solar field cost dropped and the processing of natural gas increased. The author concluded that international cooperation should be established to evaluate the potential of electrical energy transmission

from countries with high solar energy to countries with high demand and limited solar radiation. Hence, this approach of trading electricity promotes clean energy transfer between different countries.

Livshits and Kribus [29] proposed a new concept utilising the solar radiation harnessed by a parabolic trough solar field to improve the thermodynamic performance of a gas turbine with steam injection. The proposed concept included a condenser for recovering and recycling the injected water, which reduced water consumption. A thermodynamic simulation of the studied system was conducted using the Honeywell UniSim process simulation software with standard library component models, namely the water pump, the compressor and turbine models with an isentropic efficiency, thus illustrating the internal losses with realistic performance of each component. A pressure drop and minimum temperature approach was used for the heat exchanger to represent friction losses and the finite heat transfer area. The inlet air into the compressor was fixed at typical ambient conditions of 1 bar pressure, 25 °C temperature and 30% relative humidity.

In the modelling and simulation, four commercial gas turbines were considered: the GE LM2000 at 17.56 MW_e, the Solar Centaur 40-T4700 at 3.51 MW_e, the GE LM6000 PC at 43.57 MW_e and the Mitsubishi 701G at 33.4 MW_e. The overall conversion efficiency, the solar-to-electricity efficiency, the solar fraction as a function of the steam-to-air ratio, the incremental efficiency and the specific consumption of water were evaluated. The study examined a wide range of operating parameters and a wide range of steam-to-air mass flow ratios for the proposed system, which was compared to a conventional cycle with similar operational parameters. The simulation results indicated that an overall conversion efficiency of 40%–55% was feasible and that a solar fraction up to 50% could be achieved. Moreover, the incremental efficiency had a range of 22%–37%, which corresponded to the solar-to-electricity efficiency range of about 15%–24%, showing that the proposed system performed similarly to and even exceeded typical solar power plants.

It is worth noting that typical synthetic oil alternatives have certain features which can strongly affect the power plant and therefore should be known as seen in Table 2-1. With regards to solar collectors' characteristics, Section 4.3 highlights the differences between parabolic trough and solar power tower.

Table 2-1 Main features of other HTF when compared to thermal oil [87]

Fluid	Advantages	Disadvantages
Molten Salt	Higher operating temperature, and no pollution or fire hazards.	More complex design and higher electricity consumption.
Water/steam	Higher operating temperature, simple plant design and no pollution or fire hazards.	More complex solar field control and higher pressure in the solar field
Gas	Higher operating temperature, and no pollution or fire hazards.	Higher pressure in the solar field, more complex solar field control and poor heat transfer in the receiver tubes.

2.2 Simulation of Gasification Process

Some of the reasons behind the sharp rise in municipal solid waste (MSW) generation in the last years are population growth, changes in consumption habits, urbanisation, rapid industrialisation and economic growth [88]. The use of landfills for waste disposal leads to CH₄ generation from the degradation of organic contents, which presents serious environmental risks [89]. To resolve this issue, the use of Waste-to-Energy concepts such as gasification technology has been encouraged. Although most reported gasification literature has primarily focused on homogeneous feedstocks such as coal and wood, municipal solid waste recently has been considered as a potential option for energy recovery [90].

For instance, Panepinto et al. [91] investigated a hundred plants around the world which utilised a gasification process for MSW treatment. The authors aim from this study was to compare MSW gasification against conventional treatment methods, namely incineration. It was observed that gasification showed competitive technological features relative to incineration. The authors found that direct combustion of MSW offered higher-power production generally, but that for specific power plants such as those employing a combined cycle gas turbine,

MSW gasification performed better. The work of Medina Jimenez et al. [92] aimed to assess electricity generation potential in Brazil via MSW gasification by comparing three types of commercial gasification technologies. It was found that one technology achieved 30% electricity generation efficiency with a USD 65.22 MWh⁻¹ cost of produced electricity after the gate fee (the charge on waste disposers to accept their waste in treatment facilities [93]) for the city of Santo André was applied. The authors concluded that for Brazilian conditions, municipal solid waste gasification can compete with other renewable energy technologies, allowing to not only generate low-cost green electricity but also play a role in disposal of MSW. The main challenges of this technology were explored by Indrawan et al. [94] in their study, which focused on recent power generation developments using pure syngas generated from the gasification of MSW at a distributed power generation scale. The authors highlighted that MSW generally has a higher sulphur content and that therefore gasification of MSW is more challenging than that for homogenous feedstock; as a result, higher attention to the operating conditions is needed in order to reduce slag build-up on the bottom of the reactor and to offer a syngas composition with acceptable fuel contaminates for power engines such as gas turbines. Hence, the prediction of the gasification of produced syngas is an essential task for this research. There are a variety of routes to achieve this task, as seen in the literature.

2.2.1 Equilibrium Modelling

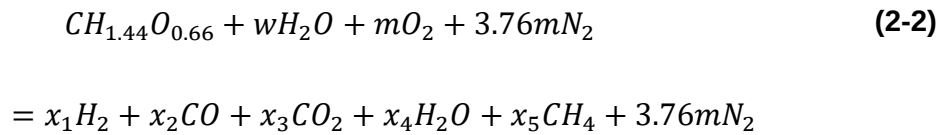
One such route for the characterisation of the gasification process is thermodynamic equilibrium modelling, whose major feature is its independence from the gasifier design, achieved by only focusing on the effect of the fuel and the process on the yields of the desired products [95]. This type of tool is usually divided into two modelling categories, stoichiometric models and the non-stoichiometric models, in which the former category is based on equilibrium constants, while the latter involves minimisation of the Gibbs free energy [96]. In the equilibrium constant approach, not all chemical reactions are included in the calculations; rather, the model only considers the most important reactions for the prediction of the syngas composition, which is expected to produce errors in the

model. This issue does not occur in the case of non-stoichiometric modelling, which does not require identification and consideration of chemical reactions [96]. In general, this did not obstruct the use of equilibrium models for predicting gasification process outcomes in various studies and the literature in fact offers several examples of reasonable simulation results achieved using this type of modelling. For example, Chern et al. [97] developed an equilibrium model to assess the accuracy of estimating the performance of a downdraft gasifier which utilised woody material as feedstock and air as a gasifying agent in addition to varying the values of the operating parameters, with the air/feed ratios ranging from 1.1 to 2.1 and the moisture/feed ratios ranging from 0.05 to 0.3. Moreover, the effects of some major parameters such as the syngas exit temperature, the amount or concentration of char, and the syngas composition were assessed by comparing them to experimental data from a commercial-scale gasifier. Finally, the model simulation results were in general agreement with the experimental data, indicating that the performance of the wood gasifier can be reasonably predicted by the equilibrium model.

Another equilibrium model was proposed by Melgar et al. [98]. In this study, the model was used to accurately estimate the impact of the air/fuel ratio and the moisture content in a feedstock moisture on the composition of the syngas produced, along with the heating value, engine fuel quality and overall gasifying efficiency. The gasifier temperature was assumed to be the main driving parameter for the entire gasification process. Since the temperature was taken as the main function variable for the equilibrium constant, this variable directly influenced the thermal state of the gasification and affected the composition of the syngas produced. Hence, the authors used the energy equation to calculate the gasifier temperature and obtained the related fuel/air ratio and moisture content. In another approach, Ratnadhariya and Channiwala [99] suggested a three-zone equilibrium and kinetic free model for a gasifier, which divided each zone according to the gasification stages. Drying and pyrolysis were combined in the first zone of the model, while oxidation and reduction were modelled in the second and third zones, respectively. In the drying and pyrolysis zone, the species considered were solid carbon, carbon monoxide, carbon dioxide,

methane, C₂H₂, hydrogen and water. It was assumed that 80% of the available oxygen produced steam, while the rest was used with carbon to produce carbon monoxide and carbon dioxide in the decomposition stage. In the oxidation zone model, it was assumed that all the hydrogen coming from the decomposition zone was combusted to produce water and that for the char (solid carbonaceous residue) oxidation, carbon monoxide and carbon dioxide were released. Furthermore, it was assumed that carbon monoxide, carbon dioxide methane and C₂H₂ went through the reduction zone without interacting with oxygen. In the reduction zone, water-gas shift and Boudouard reactions were incorporated.

Another study related to equilibrium modelling was conducted by Zainal et al. [100], who proposed a downward gasifier model to predict the produced syngas composition and the calorific value of the woody material represented, by CH_{1.44}O_{0.66}. In their model, the authors assumed thermodynamic equilibrium for all reactions, including a global gasification reaction that was written as shown in Equation (2-2):



Here w is the moisture content per kmol of wood, m , the oxygen content per kmol of wood, and x_1, x_2, x_3, x_4 and x_5 , the coefficients of constituents of the products. The assumed gasification reaction, energy and elemental balances were solved using the Newton–Raphson method, which showed that the estimated values matched with the experimental values from the literature. After that, the impact of varying the parameters was explored. For the effect of feedstock moisture content on the composition of the produced syngas gas at 800 °C, it was observed that the amount of inert nitrogen remained almost constant with changing moisture, which was also the case for methane, but at a very low concentration. Furthermore, the hydrogen concentration increased continuously from 20% to 25% for a moisture content rise from 0% to 40%, while carbon dioxide increased from 5% to 15%. On the other hand, the concentration of carbon monoxide followed the opposite trend, falling from about 28% to about 15% for the same

variation in moisture content. The authors also tested the impact of moisture content for different feedstock types, such as paddy husk, paper and municipal solid waste. The simulation showed that the hydrogen content of the produced syngas increased linearly with increasing moisture content for all types of feedstocks. Moreover, the highest hydrogen content was obtained for paper since this raw material consisted only of hydrogen, carbon and oxygen. Conversely, the concentration of carbon monoxide dropped in all cases with increasing moisture content and the highest value was achieved by paddy husk. The methane concentration increased linearly with moisture content, achieving its highest point for paper, but remained quite low in comparison to carbon monoxide and hydrogen. Finally, the authors evaluated the calorific value as a function of moisture content and gasification temperature for all raw materials, which in both cases lead to the decrease of syngas calorific value by increasing the two parameters.

In another study, based on work by Zainal et al. [100], the authors Jarungthammachote and Dutta [101] proposed including nitrogen in the feedstock chemical formula to improve the results of the simulation. For example, the results of this modified model were in better agreement with the experimental values than was the unmodified model (the Zainal et al. [100] model). These results were based on the gasification performance impact of the moisture content of municipal solid waste from eleven provinces in Thailand. The results indicated that increasing the moisture content from 0% to 40% gradually increased the concentration of hydrogen from 16.04% to 20.11%, while carbon monoxide presented an opposite effect, with values dropping from 25.01% to 12.01%. In the case of methane, there was a very low percentage in the produced syngas, but methane still showed an increase from 0.134% to 1.86%, while carbon dioxide increased from 6.00% to 15.74%. In addition to that, it was found that the rise of moisture content from 0% to 40% resulted in a decrease in the gasifier temperature from 1374 K to 1064.3 K, which accompanied a decrease in the calorific from 4.93 MJ/Nm³ to 4.42 MJ/Nm³ for the same moisture range.

In another study, Vaezi et al. [102] proposed a modified version of the thermodynamic equilibrium model reported by Jarungthammachote and Dutta [101], which was in close agreement with experimental results. The authors assumed a high residence time for the reactants to reach chemical equilibrium, the gasification reactor was considered adiabatic, all involved gases were taken to behave ideally, all the carbon content in feedstock was gasified and therefore any char formation was negligible, and ash was considered inert and therefore did not participate in any chemical reaction. Finally, the syngas composition in the simulation did not include tar, but rather only hydrogen, carbon dioxide, carbon monoxide, steam, methane and inert nitrogen. To validate this model by comparing the obtained syngas compositions to those of the experiments, four simulations were carried out, which differed in feedstock properties and experimental operating conditions. It was found that the model developed agreed well with the results of the experiments in terms of the syngas composition and its calorific value. Moreover, the authors studied 55 different materials to obtain the minimum and maximum amounts of hydrogen, carbon and oxygen, along with their effects on syngas characteristics such as calorific value, outlet gas temperature, and gasification efficiency. It was reported that the calorific value was more strongly influenced by the carbon/hydrogen ratio than by the oxygen content.

Lastly, tar content was considered in the equilibrium model used by Barman et al. [103] in order to obtain more realistic results. Nevertheless, the authors argued that the complexity of tar formation made it difficult to estimate the tar content, and therefore an established 4.5% mass percentage content from the literature was employed in their modelling of the gasification process. In this study, the modelling results for the fixed-bed downdraft gasifier were compared both with reported reference models in the literature and with experimental results. The performance analysis of the gasifier was carried out by calculating the variation in the gas composition and gross calorific value based on the air-to-fuel ratio, in addition to variation in the amount of gas as a result of the moisture content of the feedstock. It was observed that an increase in the air-to-fuel ratio led to increased nitrogen and carbon dioxide concentrations, whereas the

concentrations of carbon monoxide, hydrogen and methane were reduced. It was also noted that due to the rise in the air-to-fuel ratio, the gas produced had a lower gross calorific value. This occurred due to the reduction in combustible gases. The calculations also indicated that the gas flow rate increased with increasing moisture content, possibly because the moisture took part in the gasification reactions. Overall, the study concluded that the results of the model developed were similar to the reference model as well as to the experimental investigations.

2.2.2 Software Modelling

Another popular method used to study the gasification process and its outcomes is the use of models built in commercial software. One of these studies, carried out by Nikoo and Mahinpey [104], proposed a model for a fluidised bed gasifier in Aspen Plus using Gibbs equilibrium and FORTRAN codes to address both reaction kinetic modelling and hydrodynamic parameters, as illustrated in Figure 2-9. The reactor block RYIELD was useful when the yield distribution data were available, but the stoichiometry and kinetics were unknown. For RGIBBS, the reactor modelled the single-phase chemical equilibrium, which was useful when the temperature and pressure were known but the stoichiometry was unknown. Furthermore, the role of the RCSTR reactor block was to model the continuous stirred-tank reactor, which was useful when the reaction kinetics were known and when solids participated in the reactions. Some of the kinetic assumptions used for this gasifier model were an isothermal steady state process; that CO, H₂, CO₂, CH₄ and H₂O were the only volatile products; that the particles had uniform size and spherical shape with constant average diameter; and that the char consisted only of carbon and ash. The hydrodynamic assumptions were as follows: bed and freeboard were the two regions of the fluidised bed gasifier, the gas volumetric flow rate increased with height, and there was perfect mixing of solid particles and bed material. To validate this model, the simulation results were compared to pine gasification in a lab-scale fluidised bed gasifier.

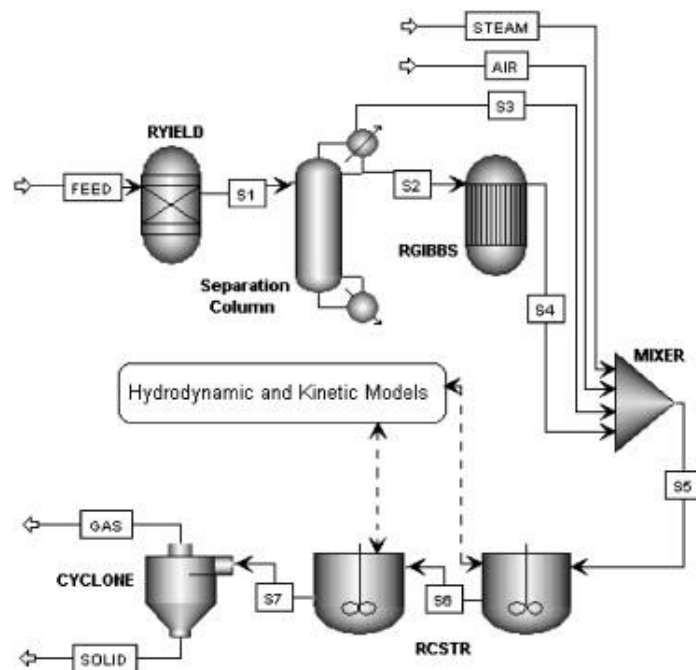


Figure 2-9 Fluidised bed gasifier model suggested by Nikoo and Mahinpey [104]

Hence, the comparison parameters were the effects of the temperature, equivalence ratio, steam-to-biomass ratio and particle size on both the composition of produced gas and carbon conversion efficiency. For a temperature range from 700 °C to 900 °C, the simulation of hydrogen and carbon monoxide content at temperatures beyond 800 °C was in agreement with the corresponding values for the experimental gasifier, while the carbon dioxide production values were below the experimental values over the entire range. However, the methane content simulation showed a good accuracy for produced methane within the temperature range under consideration. Hence, it was found that higher temperatures improved the gasification process due to the increase in both carbon conversion efficiency and hydrogen generation. Furthermore, it was observed that increasing the temperature reduced the methane and carbon monoxide percentages. With respect to the equivalence ratio, the results showed that the higher its value, the higher the carbon conversion efficiency and carbon dioxide fraction and the lower the content of carbon monoxide, hydrogen and methane. In terms of the steam-to-biomass ratio, the production of hydrogen and carbon monoxide increased when this ratio was increased, but carbon dioxide and carbon conversion efficiency decreased. Lastly, the impact of average

particle size was negligible, since it did not show a clear effect on the composition of the produced syngas.

In another approach, Ku et al. [105] studied the gasification process by using computational fluid dynamics (CFD) and discrete element methods (DEMs) to develop a model of a fluidised bed gasifier. In this model, particles were tracked and individually associated with a range of thermochemical and physical properties, such as composition, density and size, in order to obtain detailed information throughout the lifetime of the particle. Moreover, the study considered other parameters such as hydrodynamics, particle collisions and shrinkage, turbulence, radiation, heat and mass transfer. The impact of different operating conditions on the gasification performance were also assessed which were the gasifier temperature, the steam-to-biomass mass ratio and the biomass injection point for the gasifier (Feeding points generally should not be too close to the expanded bed surface of the fluidised bed gasifier in order to avoid, excessive volatiles combustion of the in the fluidizing air produced can occur [67]). Based on the results of the simulations, high values of the gasifier temperature increased the production of hydrogen and carbon monoxide. For the steam-to-biomass ratio, higher values increased hydrogen and carbon dioxide concentrations but reduced the concentration of carbon monoxide. The authors also found that as the biomass injection point height increased, carbon conversion decreased.

In a study by Ramzan et al. [106], the authors used the Aspen Plus software to separately predict the gasification process of three different feedstocks – food waste, municipal solid waste and poultry waste – under varying operating conditions. Moreover, the gasification was modelled in three steps: drying of feedstock before it entered the gasifier, decomposition of the feedstock into char and volatile components using FORTRAN statements for the yields, and finally minimizing the Gibbs free energy for the partial oxidation and gasification reactions, as illustrated in Figure 2-10. A RSTOIC reactor block was implemented to reduce the moisture content in the feedstock, then a RYIELD block was used to decompose the feedstock by converting non-conventional material into conventional elements (H, C, O, S, N, etc). A RGIBBS reactor block handled

three-phase equilibrium, as well as calculating the syngas composition. Lastly, two SEP blocks were used, one to separate moisture from the dried feedstock and the other to separate the gases produced from the ash. The model was validated against a lab-scale gasifier, showing overall similarities between the two results. To run the simulations, the authors made certain assumptions: the model used was an isothermal gasifier model with steady and kinetic free state, H_2S and NH_3 were the only contaminant gases in the produced syngas, and tars with other heavy products were ignored to reduce the complexity of the hydrodynamics.

Furthermore, the Peng-Robinson equation of state with Boston-Mathias alpha function (PR-BM) was applied to the calculations of the properties of the components of the gasification process. A model sensitivity analysis was conducted by varying the gasifier temperature, moisture content, equivalence ratio and steam-to-biomass ratio in order to observe these parameter's impacts on the syngas composition, syngas heating value, hydrogen fraction and cold gas efficiency. It was observed that in this model, the gasifier performance improved with increasing temperature due to the increase in carbon monoxide and hydrogen concentrations in syngas, which ultimately offered higher cold gas efficiency and heating value. Increases in the equivalence ratio, on the other hand, decreased the concentrations of carbon monoxide and hydrogen and therefore reduced the cold gas efficiency. It was observed that the feedstock moisture content strongly affected the heating value of the syngas, which reduced the gasifier performance in the presence of high moisture. Among the three wastes considered, food waste exhibited the highest concentrations of carbon monoxide and hydrogen, cold gas efficiency and heating value.

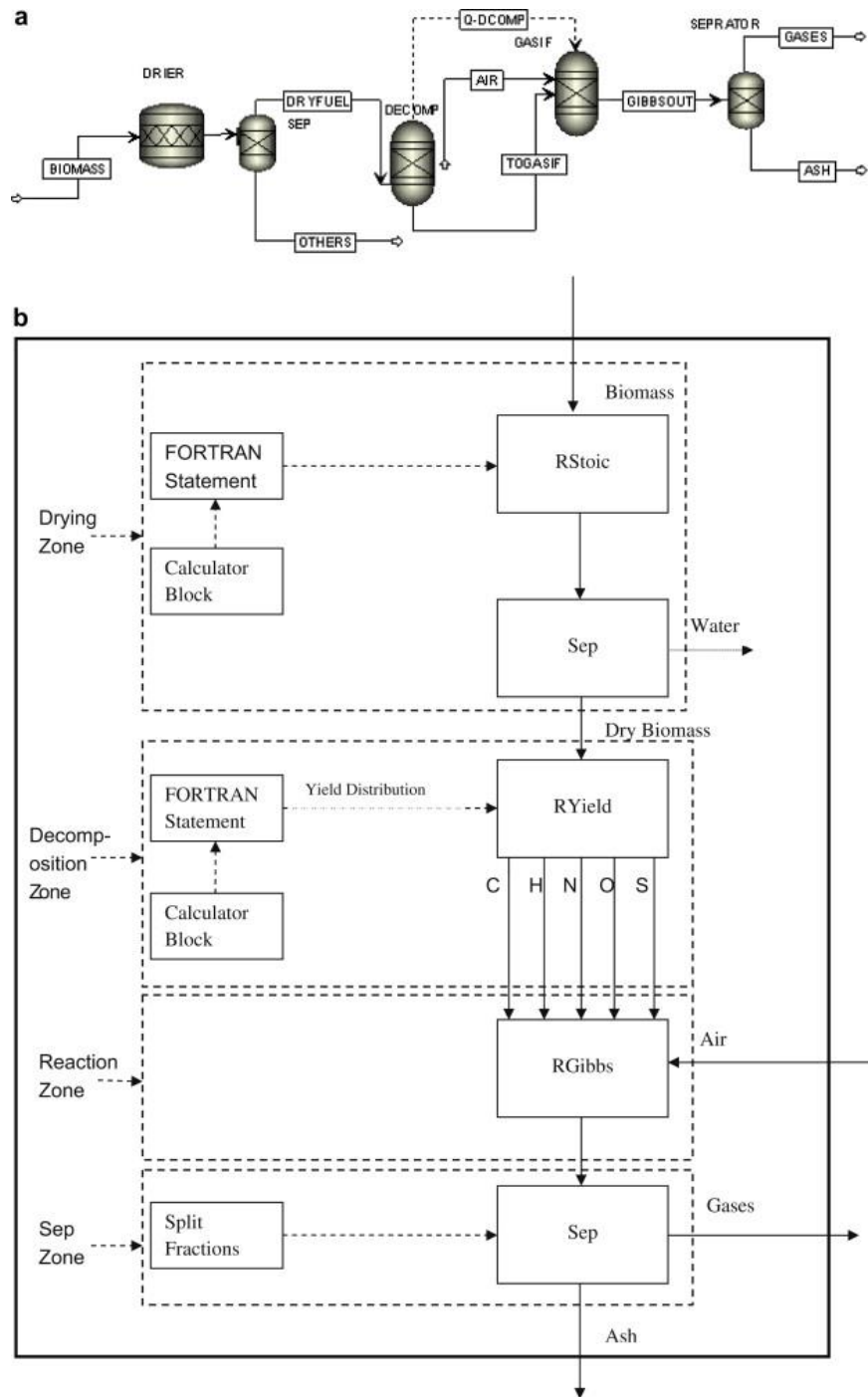


Figure 2-10 Gasifier model (a) Model (b) simulation calculation procedure [106]

In work conducted by Janajreh et al. [107], the authors investigated the gasification conversion efficiency of a commercial downdraft gasifier with capacity for 20 kW that received wood chips of medium size, as well as observing the temperature of ten different locations throughout the gasifier. They found that the reactivity was affected by the heating value and amount of moisture in the

feedstock. A CFD model for the studied gasifier was developed and the temperatures along the reactor axial direction near the air nozzle outlet were calculated and compared to experimental results. It was found that the average temperature of the CFD model was higher than the measured experimental value. The efficiency of gasification was assessed through the cold gas efficiency, which was observed to be 19% less than that calculated for the ideal case. The equilibrium model did not capture the physics and chemistry inside the downdraft gasifier when compared to other types of gasifiers, due to the complexity of the flow inside the downdraft gasifier.

Kuo et al. [108] developed an Aspen-Plus-based model for the purpose of evaluating the gasification of raw bamboo, torrefied bamboo at a temperature of 250 °C and torrefied bamboo at a temperature of 300 °C in a downdraft fixed-bed gasifier using thermodynamic analysis. The following assumptions were employed in this study to simplify the gasification simulations: the gasification processes were taken to be isothermal and to occur in a steady state; the system was considered to operate in the thermodynamic equilibrium state, which meant that the residence time of reactants was sufficiently long for the reactions in the gasifier to reach chemical equilibrium; the feedstock was introduced at normal conditions, i.e. a temperature of 25 °C and pressure of 1 atm; the produced syngas was composed of a mixture of water, nitrogen, hydrogen, carbon monoxide, carbon dioxide, and methane, with all the gases behaving according to the ideal gas law; the sulphur content in the feedstocks and the formation of air pollutants were neglected; and, lastly, the char was assumed to be composed only of solid carbon and ash and any tar formation was disregarded. The gasification model was then simulated based on minimisation of the Gibbs free energy, in addition to applying mass and energy balances based on the feedstock being primarily composed of carbon, hydrogen, oxygen and nitrogen. The property models and parameters were assigned with Aspen Plus version 7.3.

The MCINCPD stream class was used for feedstock and ash properties, since these properties were not available in the database of the standard Aspen Plus component. Furthermore, the study applied the Peng–Robinson equation of state

to estimate the physical properties for the gasified feedstock. The HCOALGEN model was used for the calculations of the enthalpies of nonconventional components. For the prediction of the feedstock density, the study applied the DCOALIGT model. The model blocks were as follows: RSTOIC for drying the feedstock, FLASH2 for calculating the vapour–liquid equilibrium, RYIELD for decomposing the feedstock based on the proximate and ultimate analyses, RGIBBS for the gasification and combustion of the decomposed feedstock, Heater for either supplying heat to generate steam or cooling the gas produced, SSPLIT for separating ash from the gas produced and Mixer for blending air and steam into a single stream. It was observed that the gasification process produced a better syngas yield when the temperature of torrefaction was increased. Torrefied bamboo at 300 °C had a higher heating value than raw bamboo or torrefied bamboo at 250 °C, which as a result caused torrefaction at 300 °C to have the lowest cold gas efficiency. For raw bamboo and torrefied bamboo at 250 °C, the decrease in the modified equivalence ratio and the increase in the steam supply ratio, corresponded to an increase in carbon monoxide and hydrogen, which enhanced the cold gas efficiency. It was observed that the highest carbon content was seen in torrefied bamboo at 300 °C, which as a result also had the lowest carbon conversion among the three fuels. In terms of cold gas efficiency and carbon conversion, the results of this study indicated that torrefied bamboo at 250 °C was more feasible than the other two feedstocks for the gasification process. However, the authors concluded that the use of feedstock torrefaction implied additional equipment and operating costs, such as energy demand for feedstock torrefaction, torrefied cooling, and production of tars. On the other hand, the two torrefied feedstock materials were considered in this investigation due to the limited presence of experimental data in the literature. Hence, the authors proposed further investigation of the evaluation of the capital and operating cost of the feedstock torrefaction, as well as the specification of the optimum temperature of torrefaction.

2.3 Hybridisation of Gasification Technology with ISCC

The coupling of a gasifier to convert renewable solid feedstock into syngas in an integrated solar combined cycle system was considered in several investigations in the literature. One such study was conducted by Tanaka et al. [109] who proposed a hybrid system which consists of a concentrated solar power system, a combined cycle and a gasifier, along with an air separation unit and molten-salt energy storage as seen in Figure 2-11. The aim of this study was to analyse the thermodynamic performance of this proposed system in order to produce an accurate evaluation of the bubbling fluidised bed gasifier and identify the optimal integration strategy for the heat-exchanger network. The authors assumed that a solar power tower was used for solar energy input, with the Gemasolar power plant in Spain as its reference, with the system at 100 MW_{th} and with cold and hot temperatures of 290 °C and 565 °C for the salts. For the gasification reactor, two types of gasifying agents were used separately, one of which was a mixture of H₂O and O₂ and the other of which was an enriched air mixture (mixture of air and O₂) with a maximum oxygen fraction of 0.4. The temperature of the gasifier bed was fixed at 800 °C and the lowest temperature of the syngas before it reached the clean-up stage was set to 350 °C in order to avoid tar condensation.

Furthermore, the authors chose the design parameters of the gas turbine to have a pressure ratio of 14, a turbine inlet temperature of 1183 °C, isentropic efficiencies of 0.76 and 0.9 for the compressor and turbine, and a pressure drop of 2% in the combustion chamber with 99.5% combustor efficiency, 99% mechanical efficiency and 97.2% generator efficacy. However, they also highlighted that the use of syngas from the gasified biomass caused the gas turbine to operate under its design conditions and therefore air bleeding method was applied. This involved extracting a fraction of the compressed air as a stream in order to maintain the turbine inlet temperature. To model the gasification process, the authors applied a semi-kinetic model of the gasifier for the purpose of estimating the methane concentration in the produced syngas and the char conversion, as well as to observe the effect of geometry while keeping the computational time low. The gasifier model included three major sections: pyrolysis and volatile combustion, continuously stirred tank reactor and plug flow

reactor. In this model, the plug flow reactor was utilised for the freeboard, the production of tar was taken as negligible for simplicity, the calculation of the char residence time was achieved by mass balance in order to keep a constant bed volume and the pore diffusion impact was taken to be the char gasification rate. The pyrolysis and volatile combustion stage were considered to end instantly when the feedstock was fed into the gasifier since their reaction times were lower than for the char gasification. A pilot-scale gasifier was considered to validate the gasifier model of this study.

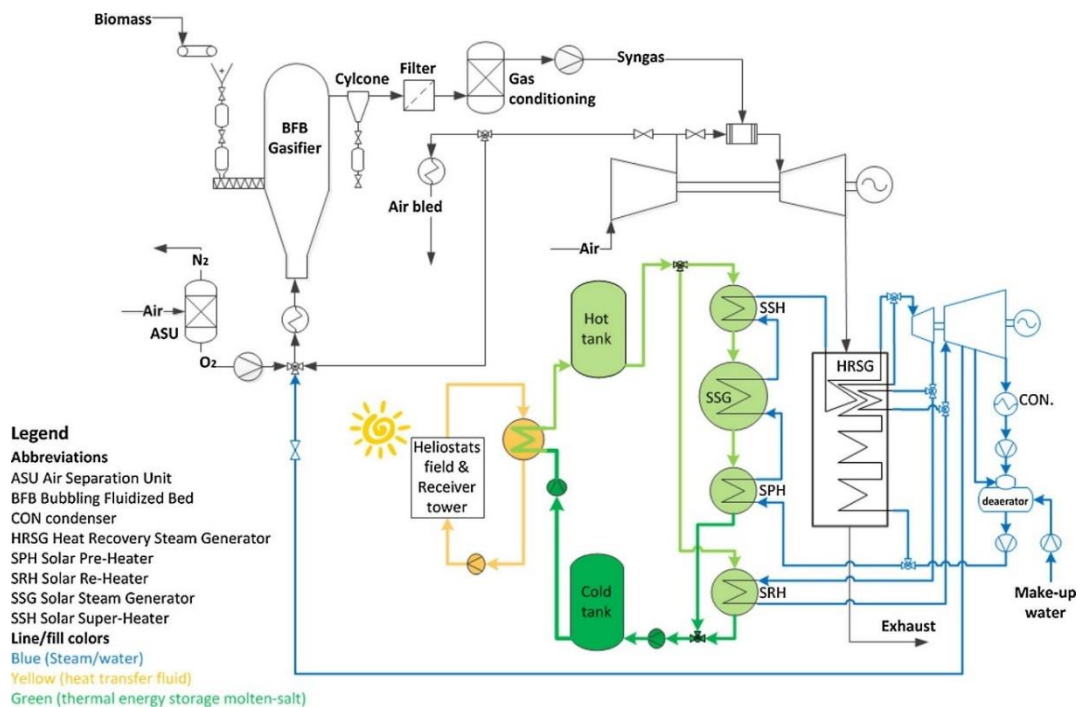


Figure 2-11 A hybrid ISCC-Gasification by Tanaka et al. [109]

For the process integration, the authors used the HEATSEP method in order to allow the configuration to be optimised without considering the details of the heat transfer sections through the selection of optimisation decision variables to maximise the net electricity. These variables were the maximum pressure and maximum temperature of the Rankine cycle steam, along with the steam mass flow rate in this cycle, which were performed through the sequential quadratic programming algorithm. The validation of the semi-kinetic gasifier model against a pilot scale gasifier involved scaling it up since the experimental reactor had a capacity of 28.6 kW–68.3 kW based on the feedstock lower heating value, while

the assumed model operated at 70 MW, and therefore the feeding rate increased from 10 kg/h–21.6 kg/h to 1.5 tonnes per hour which led the authors to consider the effect of geometry in this study. The simulation results were in good agreement with the experimental data from the literature. Hence, the impact of the gasifier on the hybrid system was observed through changing its gasifying agent. The enriched air increased the net power output and efficiency more than the $\text{H}_2\text{O}-\text{O}_2$ mixture mainly due to the difference in the needed energy to prepare the gasifying agent; however, the fraction of oxygen in the mixtures caused different outcomes. The increase of the oxygen fraction from 21 vol.% to 40 vol.% in the enriched air mixture slightly reduced the overall efficiency from 36.7% to 35.9%, while in the $\text{H}_2\text{O}-\text{O}_2$ mixture, the overall efficiency slightly increased from 32.7% to 33.7%.

This was attributed to the fact that production of oxygen for enriched air required more energy, but the energy demands of steam production in the $\text{H}_2\text{O} - \text{O}_2$ mixture was higher than for oxygen production. Moreover, preheating the gasifying agent in a range of 300 °C to 600 °C showed an overall efficiency improvement in the gasification process with increasing temperature. The impact of coupling the gasifier, solar field with energy storage and combined cycle together was evaluated based on the expected improvement against the standalone CSP and standalone gasifier-combined cycle. The authors assessed the hybrid system by varying the solar thermal input while keeping the feedstock flow rate constant, which indicated that increasing solar heat resulted in a moderate improvement in the overall efficiency. Moreover, the electricity output of the hybrid system achieved higher values than the sum of the outputs of the standalone CSP and standalone gasifier-combined cycle.

In Bet Sarkis and Zare's [68] study, the authors further explored the thermodynamic performance of two novel configurations of an integrated solar combined cycle coupled with a gasifier, as well as evaluating their costs against a gasifier-combined cycle. Both configurations were based on the standard gasifier-combined cycle configuration, consisting of a gasifier that was fed feedstock with ambient air to produce syngas to run the gas turbine by feeding it

to the combustion chamber. The combustion products left at high temperature to pass through the air preheater, which enhanced the temperature of the compressed air. Moreover, the bottoming cycle was a regenerative steam Rankine cycle, which included an open-feed water heater to operate as a feed heater as well as a deaerator. One of the proposed configurations (the HTF concept) used a parabolic trough with thermal oil for the solar field in order to preheat the feedwater before the water entered the open feed water heater, as shown in Figure 2-12. The other configuration (the DSG concept) involved the use of direct steam generation in the parabolic trough solar field for the purpose of directing additional steam into a low-pressure steam turbine, as illustrated in Figure 2-13. For the modelling and simulation of these proposed systems, the authors assumed the following: the hybrid system operated under steady-state conditions, there were no changes in the potential and kinetic energies and exergies, the combustion products and air behaved as ideal gas mixtures, all the gasification reactions in the gasifier achieved thermodynamic balance and there was no drop in pressure through the pipelines and heat exchangers. Furthermore, the mechanical efficiencies of the turbines, compressor and pumps were neglected, and the syngas generated was taken to exit the reactor at the gasification temperature. Hence, the different configurations were modelled in the Engineering Equation Solver software by considering each component as a control volume and applying conservation of mass and the first and second laws of thermodynamics.



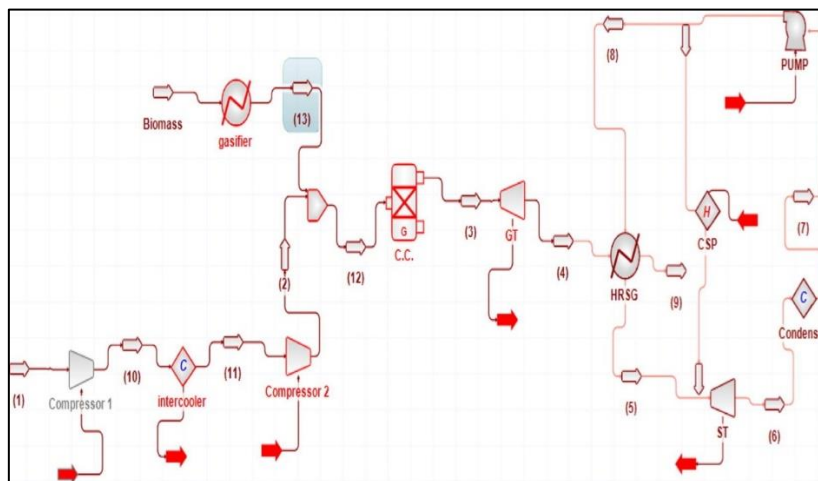
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well with those reported in the literature, which had 21.06% hydrogen, 19.61% carbon monoxide, 0.64% methane, 12.01% carbon dioxide and 46.68% nitrogen. The gas turbine and steam turbine cycles models were also found to achieve acceptable agreement with their counterparts in the reference studies. Using the validated models, the study focused on the performance of the different layouts considered in terms of net power output, energy and exergy efficiencies, solar share, levelised cost of electricity and carbon dioxide emissions.

It was observed that the energy and exergy efficiencies of the gasifier-combined cycle configuration were higher than for the two different hybrid system configurations. However, this system produced the lowest net power output and had the highest CO₂ emissions and levelised cost of electricity. With regard to the proposed configurations by this study, the configuration with direct steam generation showed could provide higher power output and solar share than both the heat transfer fluid configuration and the gasifier-combined cycle configuration for the same feedstock flow rate. Moreover, the layout of the hybrid system using the DSG concept also produced lower CO₂ emissions than the other two configurations. In terms of costs, the DSG layout provided a lower levelised cost of electricity than the other layouts. This indicated that using the solar field in the DSG configuration lowered the overall levelised cost of electricity and compensated for the additional costs from the solar field by offering additional power output when compared to the other considered configurations. However, the HTF configuration offered a levelised cost of electricity which did not strongly compensate for the additional solar field costs, making it the most expensive system among the three systems studied. As a result, the authors of this study concluded that the use of direct steam generation in the proposed configuration was best for a hybrid system of gasification with an integrated solar combined cycle.

Another study which explored and evaluated the various ways to combine integrated solar combined cycles with gasification technology was carried out by Milani et al. [110]. The authors proposed three different options for hybridisation, which were assessed under Brazilian weather conditions, as seen in Figure 2-14.

Two of these configurations differed in how the solar field operated within the hybrid system: either in series with the heat recovery steam generator as a preheater or in parallel with the HRSG. However, the third configuration moved the Rankine cycle from a bottoming cycle to a topping cycle that ran on solar input, and in which turbine outlet steam was applied as a gasifying agent in a layout designated as ‘steam extraction’. The study modelled these configurations in several steps. The first was to model the components of the different hybrid system layouts in the AspenTech Hysys software. The second was to couple components such as CSPs and gasifiers to create the configurations in the same software. The third was to run the simulation and identify the required solar thermal input. The fourth step was using the System Advisor Model (SAM) to provide the necessary solar heat demand and obtain the design parameters of the CSP, such as solar field area. The fifth step involved the inclusion of syngas storage, specifying the needed size based on an iterative process. The sixth and final step was as to end the iterative process when the capacity of the steam turbine and gas turbine, the solar field area, the solar multiple, the number of loops and the heat demand for the solar field had been obtained.



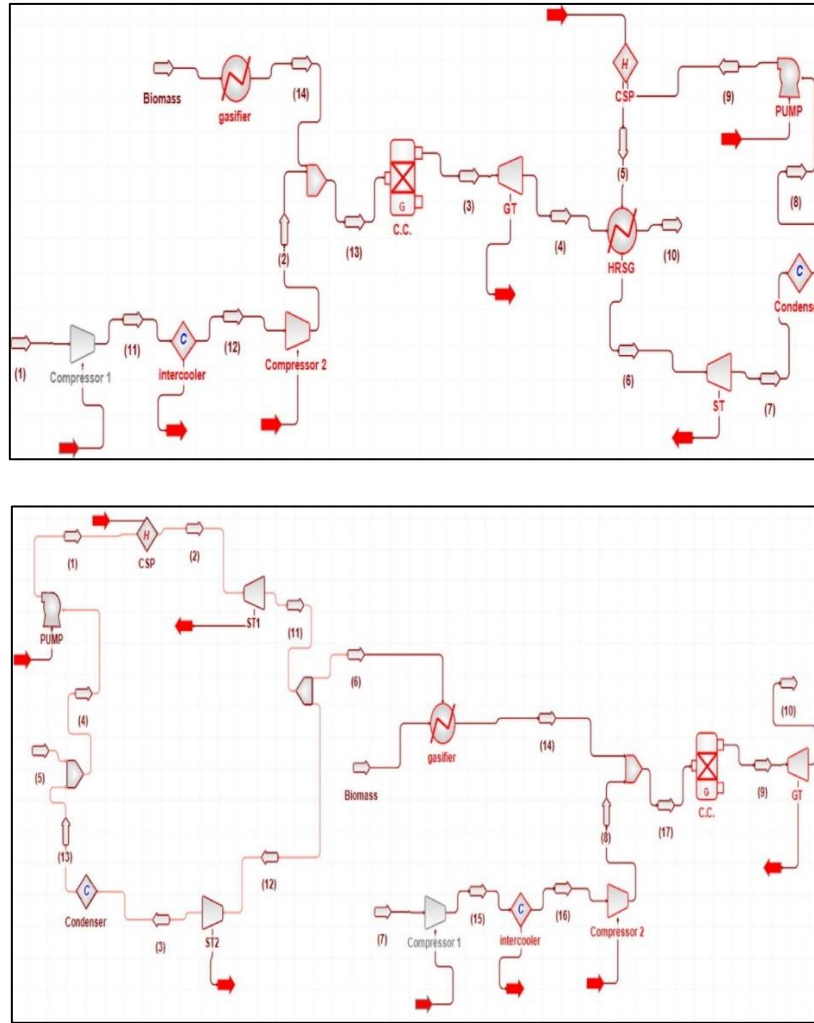


Figure 2-14 ISCCs (Top: Parallel, Middle: Series, Bottom: Steam Extraction) [110]

In the Series configuration, the steam from the Rankine cycle was exposed to solar heat after being pressurised, then directed to the heat recovery steam generator. In the parallel configuration, the steam from the Rankine cycle was exposed to solar heat only when the gas turbine was not running, and therefore when there was no available heat input from the boiler. In the steam extraction, the steam from the Rankine cycle was heated through solar heat and part of it was extracted to operate as a gasifying agent to meet the gasification process requirements, which in turn operated the Brayton cycle. Moreover, the different roles of concentrated solar power were modelled by setting the steam turbine capacity and solar multiple. The authors assumed a fixed solar multiple of 1.3, which was the cheapest option from earlier studies, and varied the steam turbine capacity. The solar field thermal output was equal to 7.69 MW_{th}, 61.76 MW_{th} and

63.7 MW_{th} for the Series, Parallel and Steam Extraction configurations, respectively, which resulted in gross turbine outputs of 2.6 MW_e, 21 MW_e and 22 MW_e. In AspenTech HYSYS, the concentrated solar power plant was modelled as a heater. The results of the simulations showed that among the three proposed configurations, the series configurations had the highest levelised cost of electricity due to the use of 96.82% of the total produced syngas from the gasification process to provide steam for the Rankine cycle. It was observed that the solar share of the parallel configuration was smaller than in the other cases, but its operation conditions permitted the storage of syngas, which as a result reduced the consumption of feedstock fuel. The authors concluded that each of the proposed power plants managed the stored syngas differently, with the goal to keep the gasifier operating at full load; however, the energy demands of the gasifier and their influence was not in the scope of the study work, so the authors recommended that they be considered in future studies.

In another approach to coupling the gasifier to the integrated solar combined cycle, Liu et al. [32] proposed two possible configuration schemes which used the high-temperature solar power tower to heat either the gasification process or the combined cycle, as illustrated in Figure 2-15. In the first configuration, option 1, the solar thermal input was used to heat the gasifier for the purpose of producing syngas and then used to operate the combined cycle. This was designated as the solar gasification combined cycle (SGCC). In the second configuration, option 2, the solar thermal input from the solar power tower was used to heat the compressed air from the gas turbine and then sent back to the combustion chamber; this option was referred to as the solar hybrid combined cycle (SHCC). In order to compare these proposed configurations, solar concentration was used in both cases, while the feedstock rate compensated for solar energy variations. Furthermore, the gasifier in SGCC used steam as a gasifying agent, while SHCC used oxygen which was obtained by an air separator unit for the gasification process of the collected feedstock, cotton stalk from Xinjiang in western China.

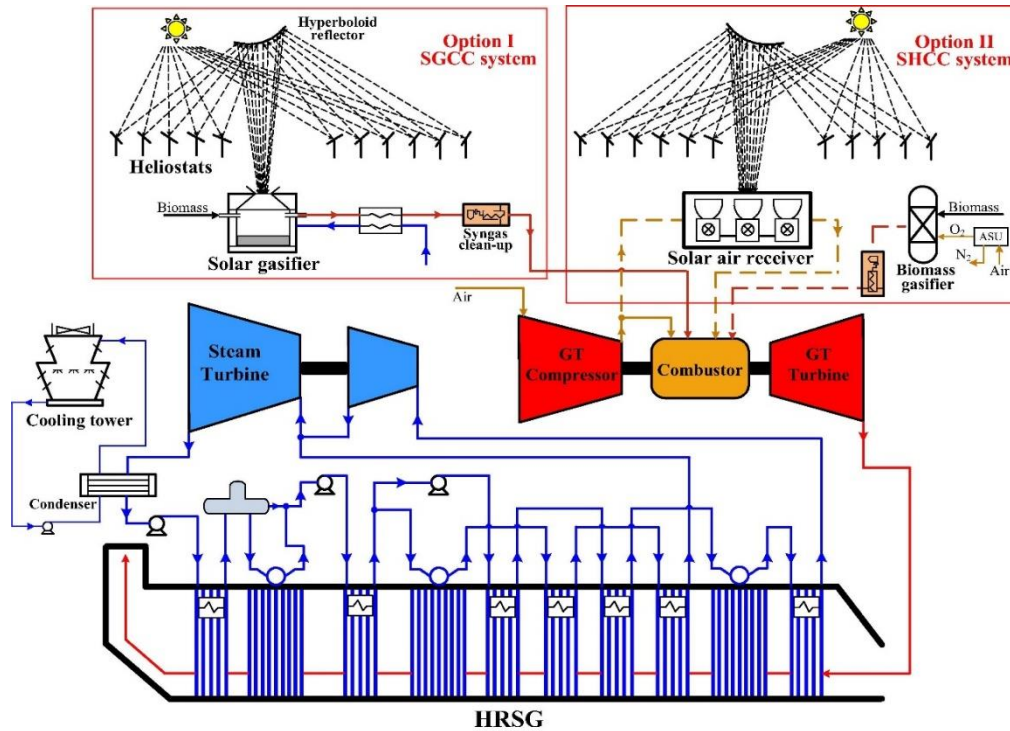


Figure 2-15 ISCC-Gasifier (Option 1:solar gasification; Option 2 air heating) [32]

The results of the calculations of the proposed model were obtained under on-design and off-design thermodynamic analysis. The first finding was that hybridisation was a valuable method to improve the conversion efficacy of solar energy. It was found that converting and storing solar thermal energy in a chemical fuel was feasible. This was achieved by solar feedstock gasification and attained a net solar share of 19.01% and net solar-to-fuel efficiency of 61.23% at the specific gasification temperature of 1150 K. Second, the SGCC system presented better performance with its annual system solar-to-electric efficiency and overall energy efficiency reaching 18.49% and 29.36% compared with the SHCC system of 15.13% and 28.03% and respectively. Third, the SHCC showed efficient utilisation of the solar energy by directly introducing it to the topping Brayton power cycle, which had a concise and simple system configuration that can be highly suitable for practical applications such as the drop of fuel consumption by heating the compressed air, reduction of energy loss during the intermediate energy conversion process. Lastly, the authors concluded that the comparison results offered a promising approach to utilising hybridisation

between gasification and integrated solar combined cycles due to the abundant renewable energy resources in western China.

2.4 Knowledge Gap

In this chapter, an extensive literature review was carried out. The focus was to identify key knowledge gaps around increasing the renewable share of ISCC through MSW gasification. These gaps are therefore the basis of the research whose main aim and objectives are detailed in the next section. Since hybridisation between ISCC and MSW gasification is quite rare, the research knowledge contribution has been directed toward the following areas:

- The evaluation of how a heterogenous feedstock, such as municipal solid waste, can affect the performance of the integrated solar combined cycle of the CSP hybrid system, since its lower heating value (LHV) ranges from 7 MJ/kg to 17 MJ/kg [111] depending on the collected composition, whereas the LHV of natural gas has a range from 41.91 MJ/kg to 49.81 MJ/kg [112].
- The highlighting of the potential of the solar field in adjusting the operating conditions and type of solar field for the purpose of improving the solar thermal contribution within the integrated solar combined cycle in order to address the effects of the notable characteristics of produced syngas from municipal solid waste gasification on the overall system.
- The determination of whether it is practical to increase the renewable share in the integrated solar combined cycle by converting it into a type of Waste-to-Energy power plant.

2.5 Aim and Objectives

Based on the knowledge gap mentioned above, this research has the following aim: to investigate the annual performance of a novel integrated solar combined cycle that uses municipal solid waste and an enhanced solar thermal contribution in order to evaluate the capability of this ISCC to function as a 100% renewable power plant. To achieve this aim, the following objectives have been established:

1. To propose a possible configuration and select suitable locations for two case studies that satisfy the novel ISCC conditions by specifying the integration

points of the renewable components and their main operating parameters within the hybrid system.

2. To analyse the impact of the suggested solar integration points on ISCC in order to evaluate the capability and overall benefits of using an enhanced solar thermal contribution as a booster for the operating conditions of the generated steam.
3. To assess the influence of MSW gasification on the hybrid system to highlight the main technical parameters affected by replacing natural gas.
4. To evaluate the annual performance of a novel ISCC by considering both its technical and economic aspects, for the purpose of illustrating its potential benefits for the power generation market.

3 Overall Methodology

3.1 Introduction

This chapter describes the research methodology to accomplish the objectives, thus ultimately fulfilling the overall aim, through the following methods:

- *Literature review and data collection:* an extensive literature review was conducted using published resources such as textbooks, journals and conferences. This method allows the research to explore the current status of the conventional integrated solar combined cycle and possible paths for improvement, especially in terms of the solar field. The other aspect of this method is surveying the previous studies on the available options for coupling the gasifier to the ISCC as well as the prediction of the outcomes of the gasification process. In both cases, the research needed to specify the likely parameters to fulfil the expected operating conditions of the proposed configuration, which in the case of the solar field included the type of collector, type of working fluid and temperature range, and for the gasifier, involved the type of reactor, type of gasifying agent and temperature range. Based on the various parameters considered, the research used four ISCC scenarios, comparing two possible configurations of the proposed concept and two configurations, one representing the typical ISCC and the other using MSW without any changes in the solar field. The other important consideration was to choose the potential locations for implementing this novel integrated solar combined cycle, which in this research primarily regions within Spain and Saudi Arabia.
- *Modelling and validation:* the modelling and validation of the proposed ISCC, in addition to that of the other configurations studied, was carried out using the System Advisor Model software for solar field parameters and then transferring these values into the Thermoflex software to observe the hybrid system outcomes throughout the year under Spanish and Saudi Arabian weather conditions.
- *Sensitivity analysis:* a sensitivity analysis was conducted with respect to the variation of the design parameters and operating conditions in the solar field

and gasifier in order to select the optimum input values for the novel hybrid system under design conditions.

- *Performance comparison*: the study calculated the annual performance of all the studied configuration or scenarios and compared them in order to observe how the proposed ISCC differ from the conventional configuration.

Figure 3-1 summarises a workflow structure that describes the overall methodology in terms of the objectives, which are divided into three major parts, the details of the novel configuration and potential locations (objective 1), the assessment of components based on assumed design and operating parameters (objectives 2 & 3) and the evaluation of the suggested hybrid system while considering the input variation over the course of the year (objective 4).

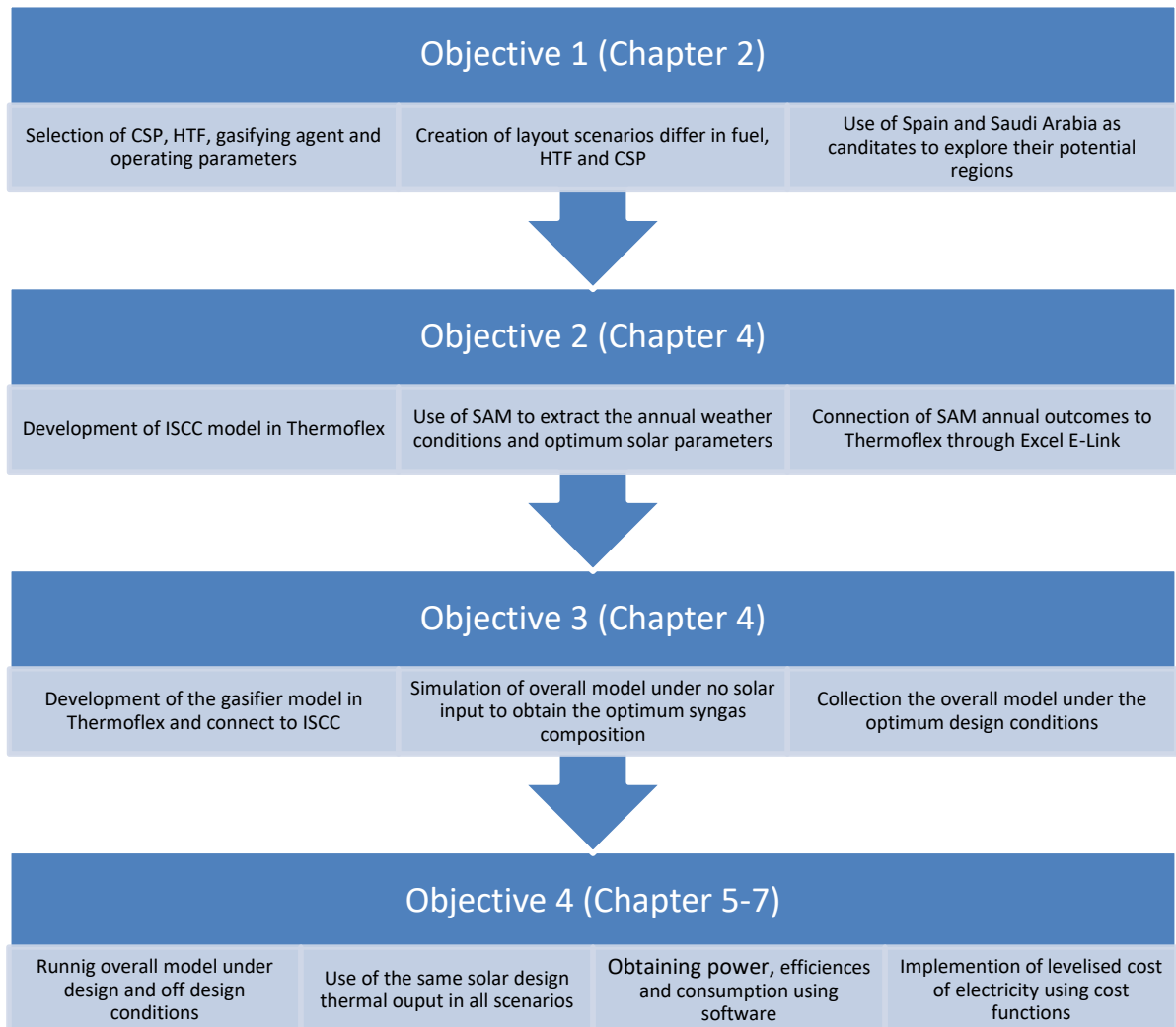


Figure 3-1 Overall summarised methodology flow chart

3.2 Suggested Configurations

As previously mentioned, there are two essential research questions to consider with regard to the proposed configuration for integrated solar combined cycles. First, the proposed novel hybrid system could involve using the generated syngas from the gasification of municipal solid waste, and such the impact of changing the fuel needs to be highlighted. Secondly, the implementation of a fuel with lower heating value than natural gas can change the operating conditions within the hybrid system, thus affecting the overall performance; therefore, the solar field contribution can be used to resolve some of these issues. Several possible configurations can be chosen for this hybridisation between ISCC and gasification technology. In this thesis, the choice of configuration depends on the integration points of the solar field and gasifier with the combined cycle and their level of heat input. Moreover, the integration points within this study are focused on maintaining the solar field thermal input for the Rankine cycle and connecting the gasification with the Brayton cycle for fuel supply. In order to evaluate the performance of the proposed ISCC concept, the research uses four scenarios in a comparative analysis based on the fuel, CSP type and integration point.

- **ISCC 1:** In this configuration, the integrated solar combined cycle operates in its conventional mode, which runs the power block using natural gas and the parabolic trough with thermal oil. In parallel to the steam generated by the gas turbine exhaust, the solar field adds saturated steam to the heat recovery steam generator by extracting a fraction of the water already preheated in this boiler.
- **ISCC 2:** In this configuration, the integrated solar combined cycle differs from its conventional mode by adding a gasifier to replace the natural gas with syngas from municipal solid waste gasification. The solar field uses a parabolic trough with thermal oil to generate steam in similar manner to ISCC 1. The purpose of this change is to highlight the effect of fuel replacement on this CSP hybrid system.
- **ISCC 3:** In this configuration, the integrated solar combined cycle differs from its conventional mode by adding a gasifier to replace the natural gas with municipal solid waste, in addition to changing the solar field design and

operating conditions. The parabolic trough solar field uses molten salt to add superheated steam directly to the steam turbine. The purpose of this change is to highlight the effect of both fuel replacement and solar thermal modification on this CSP hybrid system.

- **ISCC 4:** In this configuration, the integrated solar combined cycle differs from its conventional mode by adding a gasifier to replace the natural gas with municipal solid waste, in addition to changing the solar field design and operating conditions, similarly to ISCC 3. However, a solar power tower with molten salt is used to add superheated steam directly to the steam turbine. The purpose of this change is to highlight the effect of fuel replacement, solar thermal modification and CSP type on the integrated solar combined cycle.

3.3 Potential Locations

Since CSP relies on a high value of the direct normal irradiance, which varies from one location to another, as illustrated in Figure 1-3, the overall system performance is notably influenced by this variation. As a result, concentrated solar power plants are generally constructed in arid or semi-arid areas such as in northern and southern Africa, southern Europe, western Asia, the south-western United States, Mexico, Peru, Chile, north-western India, western China and Australia in order to achieve the best performance outcomes [113]. Due to the intermittent nature of solar energy, which allows it to operate only during daylight, and which also causes power generation to fluctuate depending on the hour and level of clear sky, hybridisation is one of the applicable solutions. Furthermore, hybridisation can be even more favourable than having thermal storage, since hybrid plants not only resolve issues caused by solar energy variations but also expand the suitable locations for CSP by reducing the required direct normal irradiance from more than 2000 kWh/m²/year to around 1700 kWh/m²/year, therefore bringing CSP closer to load centres rather than locating the power plant deep in arid or semi-arid areas to obtain very high DNI [114]. Since the use of municipal solid waste or urban waste is an essential component of the novel integrated solar combined cycle, the hybrid system should be located near urban areas to maintain sufficient waste supply. Among sunbelt countries, there are

many interesting candidates that not only can fulfil the solar requirements but can also satisfy the MSW requirements due to their high waste generation, and in some cases, due to low use of green methods for waste disposal. However, most of these countries are large enough to have regions which differ from each other in terms of received direct normal irradiance and the amount of available waste. In this research, potential locations will be selected in order to study the effect of novel hybrid system on two interesting places in terms of CSP status, Spain as a leading country in this technology and Saudi Arabia with their big plans to implement CSP technology. Based on these countries, the research will explore their regional conditions in order to highlight practical locations for implementing the proposed hybrid system.

3.3.1 Spain

To comply with the Paris Agreement, the European Union has required its member states to develop and implement their own national plans to reduce EU GHG emissions by at least 55% by 2030 and to continue this progress to become the world's first climate-neutral continent by 2050 [115]. Hence, when the Spanish government submitted its national plan for 2021–2030 to the European Commission, stating that Spain will achieve a 23% reduction in greenhouse gas emissions relative to 1990, as well as boosting the share of renewable energy in electricity generation to 74% by the end of this decade, one of these renewable energies included utilising CSP, with production planned to be increased from 2.3 GW to 7.3 GW [116]. Since Spain pioneered feed-in-tariff funding (government support to make solar thermal power attractive and profitable for investors) in 2007, concentrated solar power projects have experienced a significant rise to achieve 50 plants with a total capacity (standalone and hybrid) of 2.3 GW in 2013 [117]. As a result, this huge installed capacity has significant potential impact on Spain, which has been explored by several authors. Corona et al. [118] study has evaluated concentrated solar power based on its socio-economic and environmental impacts on Spain. The authors used multiregional input-output analysis in their assessment, which was based on additional economic activity, value added, climate change, employment creation, acidification, photochemical

oxidant formation and primary energy consumption. The authors observed that producing energy with CSP and selling it to the Spanish market presents an overall positive effect on the economy, employment and environment. In a more techno-economic assessment, Parent et al. [119] investigated the potential of concentrated solar power, especially parabolic troughs, to achieve grid parity in Spain without any subsidies by using System Advisor Model in their modelling and simulation. The simulation results showed that a 50 MW parabolic trough with 4h energy storage as well as a Power Purchase Agreement price of €0.20 per kWh offered the most viable option for a CSP project in Spain with no subsidy condition. It was concluded that the CSP industry still has not reached a fully self-sustained level without incentives.

In another study, the hybridisation of CSP with biomass in Spain was explored by Hussain et al. [120], whose analysis involved three different configurations: standalone biomass, hybrid CSP-biomass and hybrid CSP-biomass with energy storage. The simulation results showed a significant 26% drop in fuel costs for the hybrid CSP-biomass system when compared to standalone biomass. Furthermore, the levelised cost of electricity of standalone biomass is lower than for hybrid CSP-biomass due to the initial cost of the solar field. It was also found that the fuel expenditure of hybrid CSP-biomass with energy storage offered only a 3.8% drop relative to hybrid CSP-biomass without energy storage. Spain planned to reduce emissions from the latter system through penalisation of landfills and encouraging the use of other routes such as reuse, recycling and energy recovery [116]. For this reason, there have been a variety of studies to explore the Spanish MSW management status. Fernández-González et al. [121] assessed municipal solid waste treatment via Waste-to-Energy technologies, including gasification processes, in 13 municipalities in southern Spain based on economic and environmental costs. It was found that when revenue from electricity production was included, gasification technology offered the highest drop in overall cost, at 50%. With respect to environmental effects, Waste-to-Energy technologies in general reduce emissions to the atmosphere due to reducing landfilling; in this particular case, gasification technology led to a 4% drop in emissions.

In another MSW study, Uche-Soria and Rodriguez-Monroy [122] investigated how municipal solid waste management could help the environment of isolated locations such as the Canary Islands, in addition to providing heat and electricity. Based on predicted annual MSW generation, characterisation of samples, selection of thermal treatment and evaluation of the energy produced, the authors observed encouraging results. It was found that the greenhouse gases emissions were reduced by half using treatment technologies, including gasification, to replace landfilling. In another work, Aracil et al. [123] also explored the potential of MSW treatment methods for the production of electricity, with a notable focus on incineration and gasification versus landfilling. The gasifier involved three options: a fluidised bed gasifier with either an internal combustion engine or with organic Rankine cycle and a grate gasifier with a steam Rankine cycle. The authors carried out this study in Spanish locations which were heavily dependent on mechanical biological treatment (MBT) and where therefore more than 70% of refuse was sent to landfills. It was observed that incineration and gasification offered better outcomes, which could help Spain to fulfil EU environmental regulations. All gasification layouts produced less emissions than incineration, as well as having more potential to integrate with MBT plants. Furthermore, it was found that for a given gate fee, the fluidised bed gasifier with internal combustion engine had the highest profitability due to higher electrical efficiency. Nevertheless, the fluidised bed gasifier with organic Rankine cycle provided better results in the short term due to its high technical reliability and low gate fee in landfill countries.

Based on targets for Spain, the likelihood of benefitting from an ISCC configuration like the one proposed in this research is very high. Hence, it is important to identify which parts of Spain, listed in Table 3-1 can satisfy the thermal input requirements, as well as those that potentially can benefit the most of this technology. A range of DNI is used in the table above to show that each region has different conditions depending on location. Based on solar conditions, all regions can satisfy the minimum CSP requirement in a hybrid system except Basque Country, Asturias and Cantabria, which are below 1700 kWh/m²/year.

The second aspect to consider is maintaining the continuous operation of the Brayton Cycle in the proposed ISCC by supplying the cycle with gasified MSW.

Table 3-1 Main parameters data of Spanish regions

Region	DNI (kWh/m ² /Year) [124]	Population in 2018 [125]	MSW Generation in 2018 (1000 tonnes)[126]	Electricity Consumption (GWh) [127]
Andalucía	1752–2278	8,410,095	4,427	40,237
Murcia	1778–2216	1,475,569	730	9,613
Extremadura	1821–2158	1,070,454	509	5,055
Castile La Mancha	1759–2132	2,032,595	967	11,748
Valencia	1613–2062	4,946,234	2,545	27,500
Madrid	1690–2099	6,549,519	2,543	28,624
Aragon	1296–2033	1,313,135	651	10,731
Catalonia	1215–1982	7,488,717	3,471	47,323
Navarra	938–1825	643,866	294	5,126
La Rioja	1201–1737	312,884	134	1,713
Basque Country	913–1570	2,170,868	1,041	16,711
Austrias	792–1471	1,027,624	501	10,639
Galicia	927–1818	2,703,149	1,138	19,913
Cantabria	840–1606	581,294	329	4,275
Castile and León	1201–2062	2,418,555	1,086	14,288
Canary Islands	1146–2825	2,177,048	1,331	8,840
Balearic Islands	1358–1789	1,166,923	959	6,052
Ceuta	1478–1756	85,209	41	207
Melilla	1478–1756	84,708	41	213

From Table 3-1, we see that there are thousands of tonnes produced each year and therefore the waste availability criteria can be satisfied in most of the listed places. Since the research aims to identify three potential regions, the three most populous regions – Andalucía, Catalonia and Madrid – are considered the best options to maintain a sufficient supply of waste. Furthermore, these densely populated regions not only generate the highest amount of waste, but also have the highest electricity consumption, and as such utilising the proposed system reduces their emissions due to power generation. Due to its geographic features, Andalucía is considered to be the richest autonomous community with regard to renewable resources such as solar, wind and biomass. This region not only has 43% of all installed CSP megawatts in the whole country but also possesses a

total of 6104 MW from renewable energy power plants which provides 39% of electricity consumption locally [128]. More importantly, Andalucía is believed to have the potential for renewable plants, mostly solar, in excess of 310 GW, and with the steady progress of renewable projects, the Andalusian leadership are seeking a 45% contribution to national renewable power [128]. For this reason, the implementation of the novel ISCC, which utilises solar energy and waste to produce green electricity, in Andalucía agrees with regional and national targets. Similarly, Catalonia has planned its own green goals which aim to achieve 50% green electricity by 2030, in addition to a 40% reduction in greenhouse gas emissions compared to 1990 [129,130]. Another interesting feature of Catalonia is its expertise with CSP hybrid power that runs on solid feedstock, since the first CSP hybrid with biomass plant was constructed in this region [131]. The last potential region to consider is Madrid, partly since it can provide enough thermal input to run the proposed system, but more importantly because it would boost the region green electricity, since it has one of the lowest renewable shares of the nation, with an installed capacity and generation of 0.4% and 1.1%, respectively [130].

3.3.2 Saudi Arabia

When Saudi Arabia started exporting oil in 1939, which was just a year after its discovery, the country became the highest producer and exporter around the world within a few years, which significantly increased its economic growth [132,133]. As a result, Saudi Arabia has experienced rapid development to meet its growing energy demands, which grew at an alarming rate in comparison to other countries and were predicted to reach 120 GW of electricity consumption by 2032 [134–136]. For this reason, the decision makers in the Arabian kingdom have submitted their ambitious climate change plan to the United Nations Framework Convention on Climate Change, which aims to achieve a 130-million-tonne annual reduction in CO₂ by 2030 [137]. To achieve this goal, Saudi Arabia authorised a huge investment in renewable energy technologies in power generation, including concentrated solar power, with an expected target of 2.7 GW in 2030 and later plans to continue increasing its capacity to reach 25 GW in

2040 [138,139]. With respect to renewable power generation in Saudi Arabia, Al Garni et al. [140] have proposed a multicriteria decision making approach according to an analytical hierarchy process in order to evaluate and rank five renewable resource technologies: biomass, wind, geothermal, solar photovoltaic and concentrated solar power. The authors incorporated technical, socio-political, environmental and economic criteria into the study. It was observed that the most promising renewables are solar technologies, with concentrated solar power ranked second, since the first achieved the highest values in all sub-criteria, such as resource availability, efficiency, technology maturity, land requirement, emissions level, job creation, energy-leading position, capital costs and operational and maintenance costs. The authors recommended that Saudi Arabia invest more in solar energy and implement an energy mix policy to preserve its finite resources for the future and to support strong economic and industrial growth.

For the role of concentrated solar power in the Arabian Kingdom, Kassem et al. [141] have conducted a techno-economic analysis of the different types of concentrated solar power based on their strengths, weaknesses, opportunities and threats (SWOT) under Saudi Arabia's energy requirements and weather conditions. The authors assumed six types of power plant scenarios to evaluate performance and costs: parabolic trough (100MW with no storage, 170 MW with 3h of storage and 200 MW with 6h storage), solar power tower (110 MW with 10h storage and 200 MW with 12h storage) and linear Fresnel (125 MW with no storage). Based on SWOT, the parabolic trough is the most mature type and therefore can play a key role in the early stages of CSP integration in Saudi Arabia. Based on the simulation results, the highest-capacity factors per initial cost and annual energy were obtained by the solar power tower due to its ability to incorporate high levels of energy storage. However, it was observed that linear Fresnel systems, still in their infancy for large-scale operations, achieved the lowest levelised cost of electricity in comparison to other configurations. The authors concluded that the variety of these CSP layouts and their simulation results in this study will be useful to help decision makers to focus on practical scenarios in planning CSP projects.

As a renewable resource, municipal solid waste offers a great potential in Saudi Arabia since 15.3 million tonnes of MSW was generated during 2014, which was estimated to double by 2033 [66,142]. This can be traced to the drastic growth in Saudi Arabia's population, which increased from 7 million in 1975 to 27 million in 2010 and then continued growing steadily to reach 35 million in 2020 [66,143]. Furthermore, the majority of the waste collected is taken to landfills that are already classed as mature and therefore reaching the end of their useful years [66,144]. Ouda et al. [66] reviewed the potential of Waste-to-Energy technologies around the world as a power generation and waste disposal method, using Saudi Arabia as a case study. It was observed that these technologies could make a substantial contribution to renewable energy production in the Arabian Kingdom as well as reducing the issues of landfilling and its environmental impacts.

In a thorough study, Hadidi and Omer [145] analysed the status of municipal solid waste management in Saudi Arabia and proposed a financial model for Waste-to-Energy technologies to evaluate their investment viability in resolving waste disposal issues while meeting the forecasted energy demand. The technologies considered were gasification and anaerobic digestion. It was found that anaerobic digestion provided a higher net present value, but gasification offered a higher capacity to handle waste and power generation. The authors concluded that gasification plants are more suitable for highly populated areas, whereas anaerobic digestion is more appealing in smaller populated areas. Based on the previous discussion, Saudi Arabia has shown great potential for various renewables including concentrated solar power and municipal solid waste. Hence, it is important to identify which parts of Saudi Arabia, listed in Table 3-2, can satisfy the thermal input requirements and can potentially benefit the most from this technology. From Table 3-2, all regions have certain locations which can satisfy the minimum CSP requirement in a hybrid system. With regards to the Brayton Cycle fuel supply, the three most-populous regions in Saudi Arabia are Makkah, Riyadh and Eastern Province. In the case of Makkah, the region not only has the highest emissions per capita due having the highest population in the Arabian kingdom, but it is also the most visited region due to the presence of the

Islamic holy city of Mecca. The General Authority for Statistics estimated that there were more than 21 million visitors to Makkah in 2019 [146,147].

Table 3-2 Main parameters data of Saudi Arabian regions

Region	DNI (kWh/m ² /Year) [124]	Population in 2017 [148]	MSW Generation in 2017 (1000 tonnes) [149]	Electricity Consumption (GWh) [150]
Tabouk	2336–2986	910,030	678	7,621
Makkah	1628–2477	8,557,766	6,372	71,663
Madinah	2143–2821	2,132,679	1,588	17,859
Ha'il	2015–2551	699,774	521	6,145
Qassim	2022–2285	1,423,935	1,060	12,504
Riyadh	1989–2252	8,216,284	6,118	72,147
Eastern Province	1774–2135	4,900,325	3,649	69,643
Northern Borders	1924 – 2559	365,231	272	5,191
Al-Jouf	2161–2920	508,475	379	7,226
Aseer	1544–2519	2,211,875	1,647	13,004
Al Baha	1500–2362	476,172	355	2,799
Jazan	1383–2059	1,567,547	1,167	9,216
Najran	2055–2515	582,243	434	3,423

For this reason, Makkah has an excellent opportunity to implement projects like the novel ISCC described in this research because of the strong potential to reduce the region's enormous total greenhouse gas emissions through green power generation and waste disposal. In a similar manner, the region of Riyadh contains the capital city, Riyadh, which is the largest city in the country and one of the fastest-growing cities in the world and, as in any large city, pollution issues have been highly impactful. As such, the country's government has developed a plan for the city to invest in a variety of green projects related to climate change, such as monitoring power plant emissions and raising awareness to reduce landfilling and increase recycling [151]. Hence, using the novel ISCC in Riyadh region could play a significant role in improving air quality and pollution issues, especially for the city of Riyadh. The last province to consider is Eastern Province, which is the largest province in the country but has around half the population of Makkah and Riyadh. However, one potential advantage of utilising the hybrid system in Eastern Province is that this would present a range of possible environmental and economic opportunities beyond the province borders, since Saudi Arabia has already shown interest in projects that link with other countries

[152]. A possible investment link for the Eastern Province is through MSW supply with the Gulf states, therefore boosting waste resources while lowering landfilling locally and in neighbouring countries.

3.1 Implemented waste

Based on the definition of gasification technology given earlier, the composition of the collected municipal solid waste is one of the most influential parameters in gasification. The heterogenous nature of municipal solid waste means that there is an expected level of fluctuation in its physical and chemical composition depending on the various materials within the overall feedstock which leads to different capabilities in power generation. Hence, the assessment of the energy input to the system can lead to obtaining several optimum operating conditions based on the feedstock examined, which can differ from location to another and from each collection hour to the next. In addition, these variations in collected waste feedstock, when applied for the same nominal power block capacity, can lead to notable changes in the fuel consumption needed to produce electricity as a result of fluctuations in the heating value. With two locations at different countries, the composition of municipal solid waste applied in this research should represent typical feedstock data in Spain and Saudi Arabia with nearly fixed physical and chemical characteristics in order to implement assessments, including the annual performance. However, as of this research, there is very limited or rare data in the literature regarding proximate and ultimate analyses of municipal solid waste in Al Ahsa, Saudi Arabia. For this reason, the composition of waste data from the Spanish city, Sevilla, can be assumed for both locations to assess the general impact of municipal solid waste on an integrated solar combined cycle at optimum gasification operating conditions.

3.2 Conclusions to Overall Methodology

This chapter presented the main steps and methods of the overall methodology to satisfy the thesis aim and objectives. The inclusion of other configurations as conventional concepts was proposed in this investigation in order to conduct a performance comparison between novel and conventional concepts. Based on

their known capabilities for model development and simulation, System Advisor Model and Thermoflex were suggested for extracting solar field parameters and overall simulations, respectively. To accomplish that, the research considered Saudi Arabia and Spain as case studies which required identifying potential locations to operate and benefit from the novel concept. Two cities were chosen for this task, Al Ahsa in eastern Saudi Arabia and Sevilla in southern Spain.

4 Component Performance

4.1 Introduction

This chapter is focused on the performance evaluation of the renewable components of system studied, municipal solid waste gasification and solar fields, in order to obtain their optimum design and operating parameters. First, the major features of the hybrid system components were presented to obtain an overall understanding of the system capabilities. Secondly, the modelling and validation process for each component was described. Finally, a sensitivity analysis was conducted using simulations for the variables corresponding to several components.

4.2 Description of the Combined Cycle Gas Turbine

4.2.1 Gas Turbine Cycle

The technology of the gas turbine cycle (GTC) is composed of two distinctive parts: natural gas, which is its most common fuel and the turbine itself, both of which followed interesting routes throughout history until they were first used together in modern power generation. Natural gas was known for millennia since the gas leaked from reservoirs near the surface of the earth and ignited, which had a significant symbolic importance for religions in Ancient Greece, Persia and India but the earliest recorded use of natural gas as an energy source, for heating purposes, was by Chinese craftsmen and the Native Americans of North America, especially around Lake Erie [153]. Centuries after that, the commercial utilisation of natural gas for power generation began with the use of this fuel in boilers to raise steam for steam turbines, since natural gas provided a cheap and reliable means of electricity production in oil-producing countries. However, the first combination of natural gas and gas turbines in power generation appeared during the 1930s and 1940s, with the technology becoming commercially established during the 1950s [153]. The Brayton cycle or simple gas turbine cycle has been applied for gas turbine plants due to its ability to manage large volume flow and offers high efficiency close to an ideal gas cycle [154]. Moreover, the Brayton cycle found in this thesis is for the open gas turbine cycle which involves

extracting from and releasing to the atmosphere through four essential processes or states, as illustrated in Figure 4-1 and Figure 4-2. The first process is isentropic compression between the compressor inlet and outlet, state 1 – 2, which involves ambient air experiencing an isentropic rise in pressure and temperature from P_1 , T_1 to P_2 and T_2 . Next is isobaric heat addition between the combustion chamber inlet and outlet, state 2 – 3, where the compressed air is fed to the combustor and mixes with the fuel at constant pressure, raising the temperature from T_2 to T_3 . The next process is isentropic expansion between the turbine inlet and outlet, state 3 – 4, where the burned mixture of fuel and air releases high-temperature combustion gases that expand through the turbine, thus lowering the pressure and temperature from P_3 , T_3 to P_4 and T_4 . The final process, which completes the cycle, is isobaric heat rejection between the turbine outlet and atmosphere, state 4 – 1, which basically refers to the exhaust gases leaving the open cycle gas turbine. The compressor in a modern gas turbine, with a common efficiency of 87%, typically has 10 to 16 sets of blades and vanes, known as compressor stages, where air passing through is partially compressed until achieving the target pressure at the compressor exit based on the manufacturer compression or pressure ratio (typically between 15:1 and 30:1), which also increases the air temperature [153].

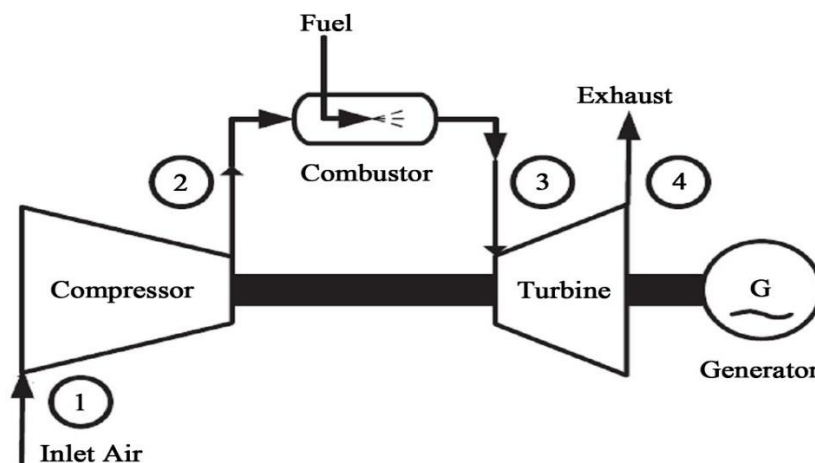


Figure 4-1 Schematic diagram of simple gas turbine cycle [155]

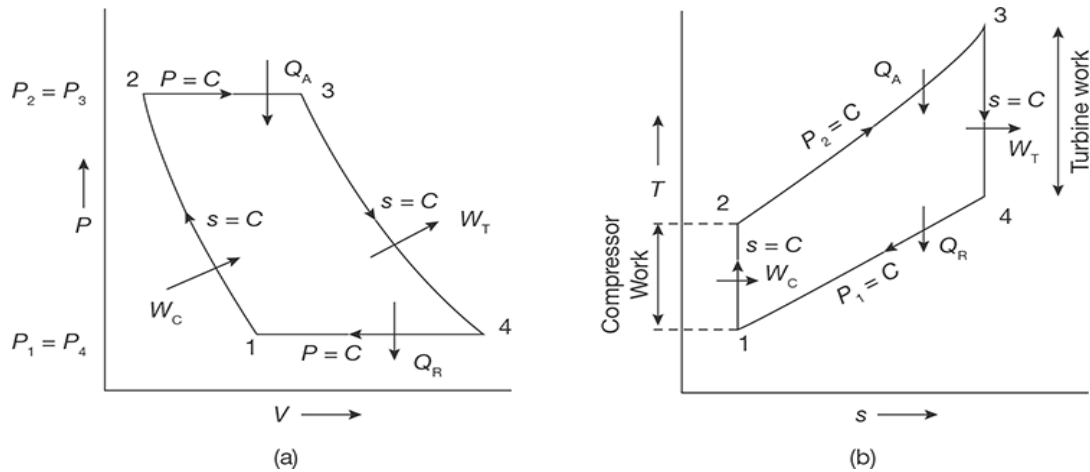


Figure 4-2 Gas turbine state diagrams (a) P-V (b) T-s [156]

For the combustion chamber, the most common method is to mix the compressed air and fuel through the nozzles before they are injected into the combustor and ignited, releasing heat that raises the combustion zone temperature to over 1900 °C, significantly higher than most materials' capabilities, which requires extracting some of the compressor air to cool the combustion chamber walls, after which the combusted fluid velocity is increased before entering the turbine by passing through a convergent duct, thus converting its static pressure into dynamic pressure [153]. In the turbine, there are stages similar to the compressor, but that perform the opposite task by expanding the fluid in each stage to rotate the turbine and generate power with an efficiency dependent on the inlet gas temperature and the temperature of the gas leaving the final stage of the gas turbine, which in a standalone or simple form is advised to be as cool as possible for maximum efficiency, but in the combined cycle power form, should not be cooled strongly in order to exploit the stream heat in steam generation [153].

Generally, the energy evaluation of the simple gas turbine cycle requires several assumptions, such as treating the gas mixture exiting the combustion chamber as air in addition to considering air as an ideal gas, a constant mass flow rate throughout the cycle, negligible changes in the working fluid kinetic energy at the entrance and exit, and finally no heat and pressure losses in the combustion chamber and ducts [154]. Based on these assumptions, the energy flow for each state in the gas turbine unit, which includes power consumed by the compressor

$\dot{W}_{Compressor}$ (MW), thermal power input by the combustion chamber $\dot{Q}_{Combustion}$ (MW), power produced by the turbine $\dot{W}_{GT}$ (MW) and thermal power wasted by the turbine exhaust $\dot{Q}_{Exhaust}$ (MW), can be described through the following equations:

$$\dot{W}_{Compressor} = \dot{m}_{air} * (h_2 - h_1) \quad (4-1)$$

$$\dot{Q}_{Combustion} = (\dot{m}_{air} + \dot{m}_{fuel}) * (h_3) - \dot{m}_{air} * (h_2) \quad (4-2)$$

$$\dot{W}_{GT} = (\dot{m}_{air} + \dot{m}_{fuel}) * (h_4 - h_3) \quad (4-3)$$

$$\dot{Q}_{Exhaust} = (\dot{m}_{air} + \dot{m}_{fuel}) * (h_4 - h_1) \quad (4-4)$$

Since stages 1-2 and 3-4 are isentropic, $P_1 = P_4$ and $P_2 = P_3$ and therefore:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = \frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}} \quad (4-5)$$

Where T_1 to T_4 , P_1 to P_4 and h_1 to h_4 are the temperature (K), pressure (MPa) and enthalpies (kJ/kg) at each stage in the gas turbine cycle, respectively, and k is the specific heat ratio of the air, $\dot{m}_{air}$ is the mass flow rate of air entering the compressor (kg/s) and $\dot{m}_{fuel}$ is the mass flow rate of fuel injected into the combustion chamber (kg/s). By knowing the heat transfer or work of each stage, the thermal efficiency of the gas turbine can be expressed through the following equations, with r_p representing the pressure ratio:

$$\eta_{GTC} = \frac{\dot{W}_{GTC,net}}{\dot{Q}_{GTC,in}} = \frac{\dot{W}_{GT} - \dot{W}_{Compressor}}{\dot{Q}_{Combustion}} = 1 - \frac{T_1 \left(\frac{T_4}{T_1} - 1\right)}{T_2 \left(\frac{T_3}{T_2} - 1\right)} = 1 - \frac{1}{r_p^{\frac{k-1}{k}}} \quad (4-6)$$

It was found that a high-efficiency open cycle gas turbine can reach an efficiency up to 46%, which is the result of improvement in turbine material to withstand higher inlet turbine temperature over the years, from 400°C in 1903 up to 1600°C in 2011 [153]. Furthermore, an increase of 8 °C in the turbine inlet temperature can improve electricity production by 1.5% to 2.0% in addition to producing an efficiency rise of 0.3% to 0.6% [153]. This progress is attributed to the natural gas

combustion properties of this type of prime mover. However, gas turbines have lax restrictions on the properties of the fuels that can be employed and therefore allow for a wide range of fuels such as syngas [94], which is the fuel considered in this research. Nevertheless, the combustion properties of syngas are lower than those of natural gas, which leads to a reduction in the turbine inlet temperature and therefore provides reduced efficiency and power outputs. Furthermore, gas turbine fuel limitations includes a lower heating value greater than 4 MJ/Nm³, alkali metals levels less than 0.2–1 ppm, tars less than 5 mg/Nm³, sulphur (hydrogen sulphide and carbonyl sulphide) less than 20 µL/L and nitrogen (hydrogen cyanide and ammonia) less than 50 µL/L, in order to avoid premature ageing of the gas turbine [94]. Hence, syngas must be clean up in to avoid any issues with regards blocking, corrosion and erosion of the downstream equipment which can be accomplished through a variety of methods such as using of water and other solvents at low temperatures (below 250 °C) or using filters at high-temperatures (above 400 °C) in order to separate impurities [67].

Like all turbomachinery, gas turbines experience a loss of performance with time. Part of this performance degradation, typically the part associated with compressor fouling, is recoverable and can be partially fixed by water washing or more fully resolved through mechanical cleaning of compressor blades and vanes [35]. Mechanical cleaning requires the unit to be opened, resulting in a loss in the plant capacity factor in addition to that, gas turbines also undergo non-recoverable losses, due mostly to increased clearances in the turbine and compressor sections as well as changes in airfoil contours and surface finish, which can only be fixed through replacement of affected parts at required inspection intervals [35].

4.2.2 Steam Turbine Cycle

The concept of the steam turbine cycle (STC) is considered to be a cross between a windmill and hydropower turbine designed to extract energy from a moving fluid, namely water in its steam phase [157]. Moreover, the steam turbine has a long history starting with the development 2000 years ago by Hero of Alexandria of a simple closed spherical vessel on bearings that discharged steam through a pipe

or multiple pipes, with the steam produced by a boiler, which represents a reaction machine. Centuries later, Giovanni de Branca designed the impulse steam turbine in Italy in 1629 [158]. However, the interest in steam turbines as a form of power generation occurred in the early nineteenth century with William Avery in 1831 (United States), Robert Wilson in 1848 (Scotland), Osborne Reynolds in 1875 (England) and later Charles Parsons in 1884 (England) [158]. Since that time, steam turbines have been essential in the power generation industry, with offerings ranging from single stage turbines to multi-stage turbines with 30 or more stages, with a corresponding capacity range from hundreds of kW to around 1900 MW [159]. Steam turbines operate based on the dynamic action of steam when its pressure drops as it moves through a passage and its heat energy is converted into high-velocity kinetic energy to move blades (based on two modes, impulse and reaction, as shown in Figure 4-3) and rotate the turbine shaft [160]. The different stages of turbine operate with the stream volume expansion and pressure drop, leading to an increase in the blade size with each stage. Currently, steam turbines can be employed in several categories of power generation plants, including thermal or fuel-based, nuclear and renewable plants, and as a result is widely used around the world, with estimated steam turbine contribution in electricity generation reaching around 12 trillion kWh in 2007, 15 trillion kWh in 2020, 16 trillion kWh in 2030 and 17 trillion kWh in 2040 out of a world total electricity generation of 19 trillion kWh, 25 trillion kWh, 30 trillion kWh and 37 trillion kWh, respectively [159].

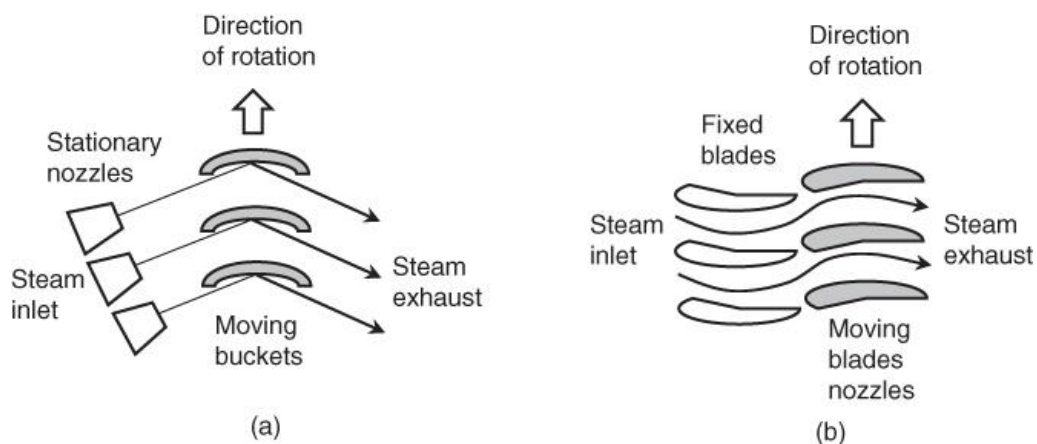


Figure 4-3 Steam turbines: (a) Impulse turbine (b) reaction turbine [161]

Furthermore, the steam turbine is more advantageous than other prime movers since it requires less floor space, less operator attention and lower lubricating oil consumption, as well as having high reliability, low maintenance costs, and the ability to operate at very high steam temperature and expand to lower exhaust pressure [162]. The Rankine cycle, considered as the standard cycle for steam power plants, includes a turbine, pump, boiler and condenser [154]. Furthermore, the Rankine cycle found in this thesis is a simple closed cycle which uses water as a working fluid to run the steam turbine after passing it through the boiler, through four essential processes or states using saturated or superheated steam as illustrated in Figure 4-4 and Figure 4-5. In this diagram, 1 – 2 or 1' – 2' represents the steam isentropic expansion through the turbine, causing it to rotate and generate power, which leads to reducing the pressure and temperature of the fluid from P_1, P_1', T_1, T_1' to P_2, P_2', T_2, T_2' . State 2 – 3 or 2' – 3 is at the condenser, where the turbine outlet saturated steam is converted into saturated liquid under isobaric heat rejection conditions. In state 3 – 4, the saturated liquid undergoes isentropic compression in the pump to reach the level of the boiler pressure, which to increase the temperature slightly from P_3, T_3 to P_4, T_4 . Next, state 4 – 1 or 4 – 1' is where the compressed water at P_4 and T_4 is fed into the boiler under isobaric heat addition in order to be converted into saturated steam at P_1, T_1 or superheated steam at P_1', T_1' . The turbines of the steam turbine unit are composed of one or more stages that may receive the steam, thus converting the available fluid kinetic energy to mechanical work by rotating the turbine blades. The steam can be generated from the boiler at different pressures; it first starts at the high-pressure (HP) stage then followed by intermediate-pressure (IP) and low-pressure (LP) stages, in which the energy available in the stream reaches its lowest value at the latest stage [163].

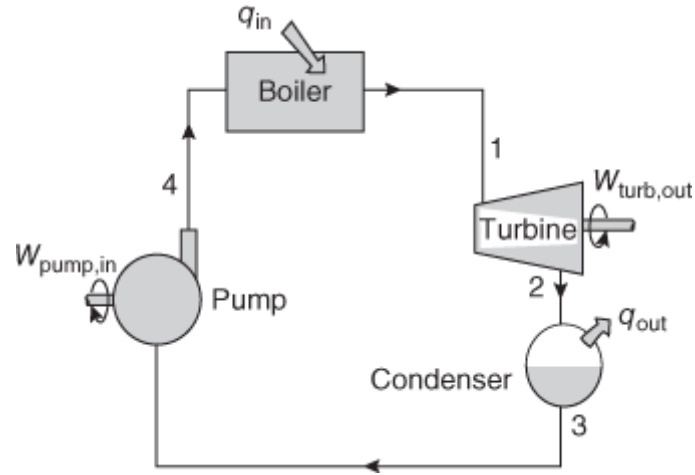


Figure 4-4 Schematic diagram of simple steam turbine cycle [164]

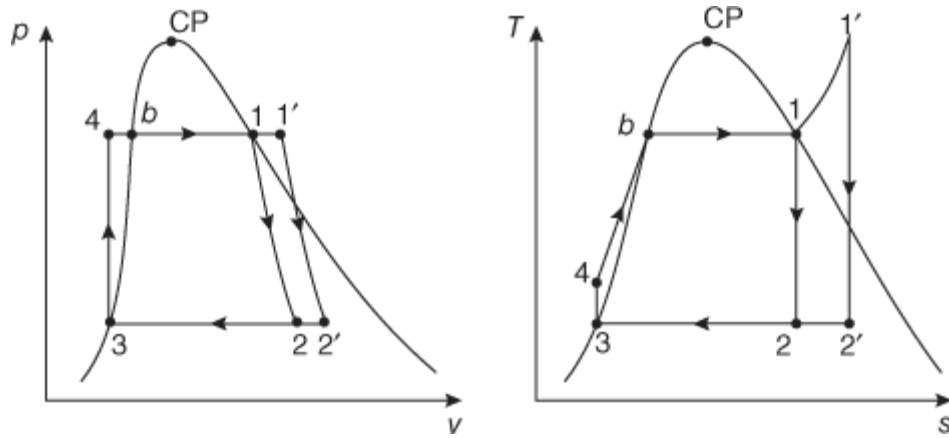


Figure 4-5 Rankine cycle P-v (left) and T-s (right) diagrams for saturated steam and superheated steam cycles (b: boiler and CP: critical point) [164]

Furthermore, the stream supplied must maintain a certain level of purity, by measuring the steam contamination regularly and blow down the boiler when it is too high [165], in order to avoid the impact of elements or compounds such as silica [166], which tends to affect the material of steam turbine by increasing nozzle erosion, blade surface roughness and blade corrosion, for example [167]. The energy flow for each stage in the steam turbine unit, including power consumed by the pump $\dot{W}_{pump}$ (MW), thermal power input by the boiler $\dot{Q}_{Boiler}$ (MW), work produced by the turbine $\dot{W}_{ST}$ (MW) and waste heat from the condenser $\dot{Q}_{COND}$ (MW), can be expressed through the following equations:

$$\dot{W}_{pump} = \dot{m}_{water} * (h_2 - h_1) = \dot{m}_{water} * v * (P_2 - P_1) \quad (4-7)$$

$$\dot{Q}_{Boiler} = \dot{m}_{water} * (h_3 - h_2) \quad (4-8)$$

$$\dot{W}_{ST} = \dot{m}_{steam} * (h_3 - h_4) \quad (4-9)$$

$$\dot{Q}_{COND} = \dot{m}_{steam} * (h_4 - h_1) \quad (4-10)$$

Here h_1 to h_4 are the enthalpies (kJ/kg) at each stage in the steam turbine cycle, v is the specific volume of the water (m³/kg), P_1 and P_2 (MPa) are water pressure at the pump inlet and outlet and lastly $\dot{m}_{water}$ and $\dot{m}_{steam}$ (kg/s) are the mass flow rate of liquid water and vapour water, respectively. By knowing the heat transfer or work at each stage, the thermal efficiency of the steam turbine (η_{STC}) can be expressed with the following equations:

$$\eta_{STC} = \frac{\dot{W}_{STC,net}}{\dot{Q}_{STC,in}} = \frac{\dot{W}_{ST} - \dot{W}_{Pump}}{\dot{Q}_{Boiler}} = 1 - \frac{\dot{Q}_{Condensor}}{\dot{Q}_{Boiler}} \quad (4-11)$$

The steam turbine can operate under high-temperature and high-pressure inlet steam conditions to improve the efficiency of the cycle, which ranges from 593 °C to 600 °C for the temperature [159] and from subcritical 100 bar to supercritical 300 bar for the pressure [168]. In combined cycle gas turbine applications, steam turbines had relatively low live steam parameters (inlet steam parameters such as temperature and pressure) in the past but with the rise in gas turbine exhaust temperatures throughout the years, the steam turbines of CCGT have reached the level of those at conventional steam power plants, as shown in Table 4-1

Table 4-1 Changes in steam turbine in CCGT due to gas turbine advances [169]

Changed parameter	Traditional	Modern
Gas turbine exhaust temperature	450 °C - 550 °C	550 °C - 650 °C
Steam turbine inlet temperature	420 °C - 520 °C	520 °C - 600 °C
Number of steam pressure stages or levels	1 or 2	2 or 3
Steam turbine inlet pressure	30 bar – 100 bar	100 bar – 170 bar
Number of gas turbines per steam turbine	1 to 5	1 or 2

It is worth noting that the blades of the steam turbine are exposed to high-pressure and high-temperature conditions based on the steam as a working fluid. At the high-pressure stages, the steam is expected to be completely dry or in other words, all water is in the gaseous state (dryness fraction of 1 [170]). For the intermediate-pressure stages, it is expected to receive wet steam, which means the presence of water liquid particles within the steam (dryness fraction less than 1 [170]), as such, the applied material of the blades should resist corrosion and erosion. Furthermore, high centrifugal stresses also affect the condition of the blades, especially when the steam speed is high and its dryness fraction is lower than 0.9 [160]. In order to prevent erosion, the following methods can be adopted: raising the temperature of the steam at the turbine inlet to avoid the dryness fraction dropping below 0.9 at the exit, utilising a reheat cycle to keep the dryness fraction at the exit within limits, providing the turbine with drainage belts to drain the water droplets on the outer periphery and implementing a shield made of a hard material, such as stellite inserts [171], on the leading edge of the turbine to prolong blade life [160].

4.2.3 Heat Recovery Steam Generator

There are a large number of variable boiler designs for steam generation which are classified either by the arrangement of their pressure parts (e.g., water tube boilers and large volume boilers), their firing concept (e.g., grate boilers, fluidised bed boilers) or their application (e.g. utility boilers, recovery boilers and heat recovery steam generators) [172]. The term 'heat recovery steam generator' is used in both process and power industries to refer to devices that generate steam by recovering waste heat from the exhaust gas of gas turbines, diesel engines, effluents from chemical plants and refineries, kilns and furnaces, incineration systems or any hot gas generators; however, for unknown reasons, the term has been strongly associated with waste heat boilers connected to gas turbines [173]. In any case, the purpose of this unit is to receive the hot exhaust gas from the gas turbine and transfer its thermal energy to the steam cycle until the flue gas temperature is low enough that the gas may be returned to the environment through the stack. This exhaust gas can have various operating conditions

depending on the gas turbine; for example, the unit can produce a flow rate of 1.8 kg/s for a 180 kW turbine up to 500 kg/s for a 143 MW turbine, in addition to having a temperature range of 482 °C to 593 °C [35]. Furthermore, the flexibility of the heat recovery steam generator design, which allows its size to be changed for large or small gas turbines, has become an essential cause of the success of the combined cycle design [174]. Moreover, this boiler is also compatible with varying steam turbine pressure levels; that is, it can operate under single-pressure up to triple-pressure turbines. However, the use of single-pressure HRSG limits the heat recovery process due to their inability to lower the temperature of the exhaust gas below the temperature of saturated steam and as a result, the addition of pressure levels has been considered to effectively capture heat waste [35,174]. In Figure 4-6, a dual-pressure HRSG is presented where each pressure level consists of an economiser, an evaporator and a superheater, where the exhaust gas flows through them starting from superheater, followed by the evaporator, and then the economiser, in counter-flow to the feedwater. In this process, the economiser role is heat to the liquid water to reach near saturation liquid then converted into its saturated steam (with certain level of moisture) in the evaporator and finally this steam is exposed a very high temperature conditions to reach superheated steam (all traces of moisture are removed). This heat transfer creates a temperature profile for each stage in the boiler, as illustrated in Figure 4-7, which are called the pinch point and approach temperatures. The former is described as the difference between the temperature of the exhaust gas and the temperature of the saturated steam in the evaporator section, while the latter is defined as the difference between the temperature of the saturated steam and the preheated water at the evaporator [35]. The pinch temperature can be selected based on the evaporator pressure or via the adjustment of the heat transfer area of the HRSG design, which tends to play a role in the effectiveness of the evaporator and the transfer of heat between hot and cold streams when its value is reduced, while conversely, higher values of the approach temperature enhance heat recovery in the HRSG but thermodynamic and economical perspectives must be considered to produce an optimum design [175].

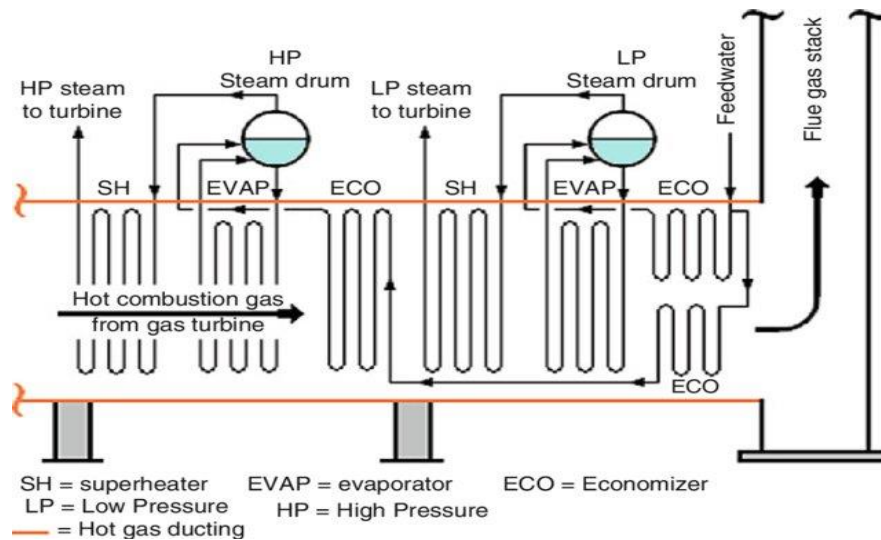


Figure 4-6 A diagram of a dual-pressure heat recovery steam generator [176]

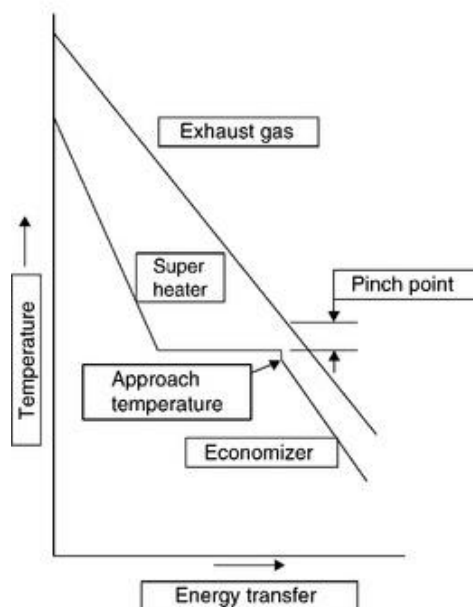


Figure 4-7 A diagram of energy flow in HRSG [35]

The feedwater goes through three stages in HRSG, in counter-flow to the exhaust gas: first, the economiser section, where the water is heated, leading to the rise of the water temperature to approach the saturation temperature of the evaporator section; then the evaporator section, where the water is boiled until it is converted into saturated steam; and lastly the superheater, where the steam temperature is increased to become superheated steam. The main effect of first preheating the feedwater in the economiser is that the exhaust gas leaves the

HRSG at a substantially lower temperature, which indicates better heat recovery, in addition to enhancing the steam flow by around 75% [174]. Other parameters to consider in HRSG include the amount of surface area for the heat transfer, which if not adequately selected, could lead to the loss of exhaust gas thermal energy; the surface area sequencing of the economiser, evaporator and superheater arrangement in the different pressure sections; and the stack temperature, which at low values, offers greater thermal energy recovery [174]. Furthermore, typical operating temperatures require the exhaust gas to exit the evaporator section with temperatures higher than water saturation; in other words, at a pinch temperature between 14 °C and 20 °C [174]. With knowledge of the stages of the heat recovery steam generator, the steam flow rate within this component can be expressed through the following equation [35]:

$$\dot{m}_{steam} = \frac{\dot{m}_{exhaust} * c_p * (T_2 - T_1) * L}{(h_2 - h_1)} \quad (4-12)$$

where $\dot{m}_{steam}$ is the mass flow rate of the steam (kg/h), $\dot{m}_{exhaust}$ is the mass flow rate of the exhaust gas (kg/h), c_p is the specific heat at constant pressure, T_1 is the temperature of the stack exhaust (°C), T_2 is the temperature of the exhaust gas exiting the gas turbine (°C), h_1 is the enthalpy of the water at the evaporator inlet (kJ/kg), h_2 is the enthalpy of the steam leaving the HRSG, the heat at a constant steam pressure flow rate (kJ/kg) and lastly L is the radiation loss factor of approximately 0.985. The amount of steam generated in HRSG can also be predicted through energy balance based on the pinch temperature (T_{pinch}), which is estimated as the difference between the gas outlet temperature at the evaporator ($T_{gas,out}$) and the saturation temperature of the water-steam mixture within the evaporator (T_{sat}), as illustrated in Equation (4-13):

$$T_{pinch} = T_{gas,out} - T_{sat} \quad (4-13)$$

The amount of heat released from the gas turbine and applied in steam generation in the evaporator ($\dot{Q}_{released}$) is the result of the mass flow rate of the exhaust gas ($\dot{m}_{gas}$), the heat capacity of the gas (C_p) and the temperature

difference in the evaporator for the gas inlet temperature ($T_{gas,in}$), as well as the saturation and pinch temperatures as shown in Equation (4-14):

$$\dot{Q}_{released} = \dot{m}_{gas} C_p (T_{gas,in} - T_{sat} - T_{pinch}) \quad (4-14)$$

By considering radiation losses through the casing, the heat absorbed ($\dot{Q}_{absorbed}$) is the product of the released heat and the heat collection efficiency (η_{EV}), which is usually in a range of 99% to 99.5%, as presented in Equation (4-15):

$$\dot{Q}_{absorbed} = \eta_{EV} \times \dot{Q}_{released} \quad (4-15)$$

The energy required to produce one unit mass of steam involves the specific enthalpy of the steam leaving the evaporator ($h_{s,out}$), the specific enthalpy ($h_{w,in}$) of the water entering the evaporator, the specific enthalpy of the water at the evaporator saturation temperature ($h_{w,sat}$) and the flow rate of water discharge as a fraction of overall steam flow, as shown in Equation (4-16):

$$\Delta h_s = h_{s,out} - h_{s,in} + BD (h_{w,sat} - h_{w,in}) \quad (4-16)$$

Hence, the mass flow rate of the steam generated from the evaporator can be estimated using Equation (4-17):

$$\dot{m}_s = \frac{\dot{Q}_{absorbed}}{\Delta h_s} \quad (4-17)$$

A designer of a heat recovery steam generator needs to consider two design perspectives, thermal and mechanical [174]. The thermal design involves increasing the amount of recovered heat and reducing the stack temperature while overcoming three main challenges: a large HRSG face area is required to withstand a large amount of gas flow, a large HRSG heating surface is needed to maintain the small temperature difference between the gas turbine exhaust and fluid and lastly the face area value of the HRSG is affected by the exhaust gas pressure. For the mechanical design, the designer needs to consider that the exhaust gas leaving the gas turbine is extremely turbulent, which can affect the structure and casing of the HRSG; the pressure parts (e.g., superheaters, evaporators, economisers, steam drums, reheaters, and piping) must be

designed based on a boiler code; and lastly, the designer needs to include the effects of tube vibration and acoustic resonance. In order to keep the HRSG operating effectively, there are certain failure causes that need to be inspected and repaired, namely thermal fatigue of superheaters and reheaters, flow-accelerated corrosion and under-deposit corrosion in high-pressure evaporator tubes [177]. This is usually accomplished through hot and cold inspections, of which the former inspection involves evaluating the HRSG from the outside during operation or after recent shutdown and the unit is still hot, while the latter inspection is performed when the HRSG is shut down and cooled off, to assess variable parts such as the connection point between the turbine exhaust and the HRSG inlet duct, as well as the presence of hot spots on the casing of the HRSG [174].

4.2.4 The Combined Cycle

This concept, as stated earlier, involves using the gas turbine to generate power and making use of the thermal energy of the exhaust gas in steam production through the heat recovery steam generator in order to run the steam turbine, thereby increasing the overall electricity production in the power plant as illustrated in Figure 4-8. Furthermore, the distribution of energy within the system varies for each component, as shown in Figure 4-9, which indicates that a gas turbine tends to convert around 40% of the energy to power, while the remaining 60% of the energy is received by the heat recovery steam generator and then implemented to run the steam turbine, which can convert about 20% of the energy into power [35]. Moreover, this concept can combine gas and steam turbines in a single shaft with a large generator or have each in separate shafts with smaller generators. The main benefit of the former is the use of one fewer of each of the transformer, generator, and electrical subsystems, but it requires adding multiple gas turbines to satisfy the growing plant size, which affects the advantage of having one generator/transformer fewer [35]. The main disadvantage of a single-shaft layout is the need to increase the number of steam turbines, condensers and condensate systems in order to match the number of gas turbines used in the power plant, while the multiple-shaft layout needs only one steam turbine and

condenser, in addition to having a heat sink capable of handling three gas turbines [35]. Moreover, using one larger steam turbine and heat sink offers lower costs as a result of economies of scale and allows the plant to use a higher-pressure and more efficient steam cycle; for this reason, the associated number of gas turbines and the overall plant size required have a significant impact on which of the two is the more economical option, single shaft or multiple shaft, for the combined cycle plant. Historically, the first commercial application of the combined cycle concept was in the United States in 1949, where a gas turbine was added to an established fossil-fuel-fired steam power station to heat the boiler's feed water, boosting the steam plant output. In later years, during the 1950s and 1960s, combined cycle systems applied the exhaust gas from a gas turbine as the combustion air for the conventional boiler of the steam turbine which improved the overall efficiency by few points [153]. During the 1970s and 1980s, the concept of combined cycle, as shown in Figure 4-8, became common especially near the end of the 1980s, when the major manufacturers explored the idea of designing both gas and steam turbines to specifically be implemented together, leading to the modern generation of notably high-efficiency combined cycle power plants [153].

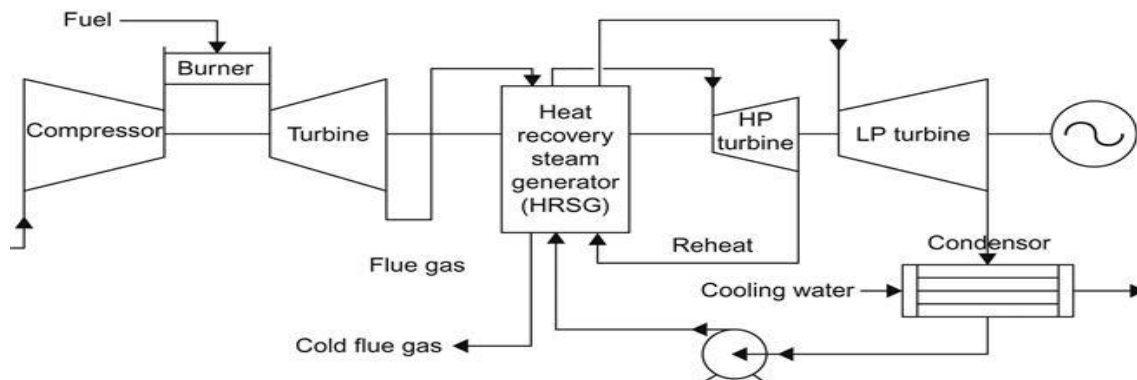


Figure 4-8 Schematic diagram of the combined cycle gas turbine plant [153]

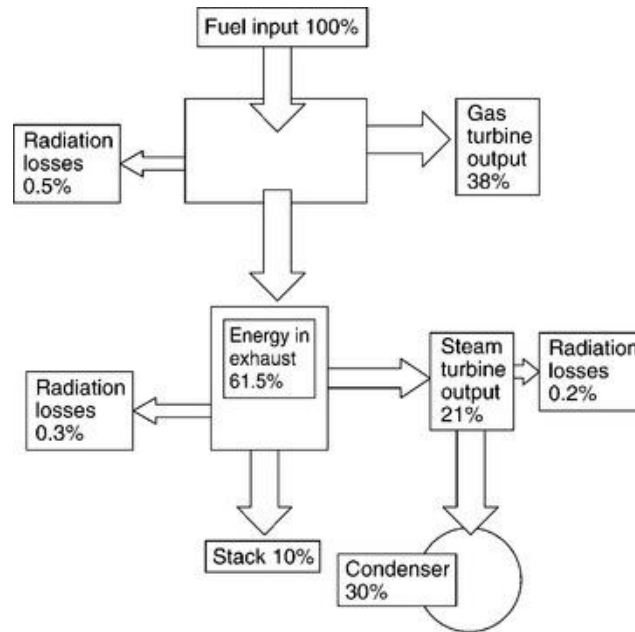


Figure 4-9 Energy flow diagram of the combined cycle gas turbine plant [35]

Since that time, combined cycle gas turbine plants have continued to improve, and therefore their efficiency has continued to increase over the years, which can partly be attributed to the compact design of this power plant concept, which reduces heat loss, but mostly has been related to the improvement of the turbine inlet temperature, which has reached values around 1600 °C, offering greater thermodynamic efficiency. Later, the efficiency has reached 65% by reaching a feasible turbine inlet temperature of 1700 °C [178]. Originally, large CCGT plants that operated mostly at full load were used for base load power generation, but due to the volatility of gas prices between the late 1990s and 2000s, it was found that this was economically unviable, and as a result, power plants frequently operate at much lower than 100% capacity factor. For example, the United States typically has been using 40% or less of its CCGT capacity factors in most recent years [157]. However, manufacturers have been modifying combined cycle plants to maintain good efficiency and emissions performance while operating at less than 100% load, which means the ability to quickly start up and to change output, as well keeping system parts warm, by either heating them or by maintaining power generation at very low load when electricity demand is low in order to return to full output quickly [157].

Currently, the combined cycle gas turbine plant is considered the most efficient power generation plant in the world, with an efficiency that can reach 60.75% [153] and flexible sizes (as low as 60 MW and as high as 1500 MW), in addition to other features such as having very low emissions of carbon dioxide, unburned hydrocarbons, carbon monoxide, oxides of nitrogen; a compact footprint when compared to other fossil fuel plants; and a short construction time, with a 200 MW plant possible to complete in less than two years [35]. The performance of the combined cycle is affected by gas turbine performance variations based on ambient conditions. For example, a rise in ambient temperature leads to a lower pressure ratio but a higher exhaust gas temperature, which results in a decrease in the gas turbine efficiency and output [179]. However, an increase in ambient temperature has a different impact on the performance of the combined cycle, because increasing the exhaust gas temperature tends to increase power output and the efficiency of the steam cycle [179]. Nevertheless, the overall thermal efficiency of CCGT tends to decrease with increasing ambient temperature due to the rise in the work consumption of the compressor and overall thermal losses [180]. As with all turbomachinery, the performance of gas turbines decreases with time, but can be partially recovered through water washing or mechanical cleaning of compressor blades and vanes [35].

With steam turbines, the main impact on the combined cycle includes three design criteria [181]. First, the number of pressure levels can be increased, which leads to having a smaller average temperature difference between the flue gas and the water/steam stream. Second, the type of steam turbine can play a significant role in performance, depending on the application, such as using back-pressure turbines for heat applications but condensing steam turbines for power generation. Finally, the use of steam reheating in the heat recovery steam generator after steam exits the high-pressure steam turbine in order to be sent to intermediate and low-pressure steam turbines with higher temperatures and higher quality in the low-pressure turbine exit steam. The current research and development route for combined cycle gas turbines involves boosting the turbine inlet temperature, increasing the compressor pressure ratio, and designing

advanced materials for both the combustion chamber and the turbine to withstand the hot environment created by the higher temperatures [35].

4.3 Description of Solar Field

4.3.1 Parabolic Trough

The first recorded document describing the optical properties of a parabolic trough collector, written by the Greek mathematician Diocles in the second century B.C., explained the reflection of solar rays toward a common point in a parabolic mirror. However, this concept was not constructed until 2000 years later, in the late 19th century, when the Swedish engineer John Ericsson designed and built a parabolic trough collector to produce saturated steam for a 373 W steam engine, but the technical details were kept secret even after his death in 1889 [182]. However, in 1913, the American engineer Frank Shuman, with the financial support of British investors and the technical advice of British professor C.V. Boys, designed and built a parabolic trough to produce saturated steam to run a steam engine for farming irrigation in Egypt. The first instance of converting solar energy into mechanical power was in 1935 by C.G. Abbot [182]. In the 1980s, the parabolic trough achieved commercial status for power generation in California in the United States, with nine power plants [183] that remained the only commercial CSP plants worldwide until 2007, due to Spain's introduction of a feed-in-tariff for CSP projects [184].

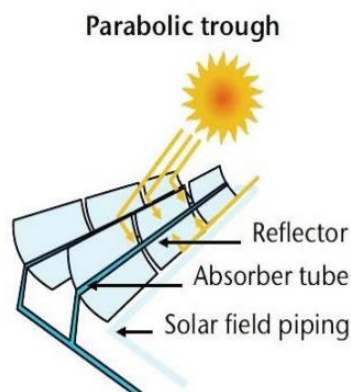


Figure 4-10 Parabolic Trough [185]

A parabolic trough is essentially a very large curved mirror designed to reflect the solar radiation onto a receiver, as shown in Figure 4-10. Hence, a parabolic trough solar field contains hundreds of mirrors positioned in parallel rows and aligned on a north-south axis, in order to maintain a continuous focus of solar radiation on the receivers by tracking the sun throughout the day from east to west [12]. The receiver is a tube, with specific coatings to maximise heat absorption and minimise heat losses, containing a flowing heat transfer fluid (HTF), which absorbs thermal energy from the concentrated sunlight; the hot heat transfer fluid then flows into a heat exchanger, where the HTF heat is transmitted to water for steam generation [35]. Compared to other concentrated solar power technologies, the parabolic trough is the most widely used worldwide [12]. A single parabolic trough may have an overall length of about 150 m, composed of sections that are generally 5–20 m in length, in addition to 5–8 m in width and 2 m in depth, and must be able to follow the Sun across the sky by rotating about an axis along its length in a north–south alignment [183].

The current commercial status of PT-based power plants indicates that 90% of CSP plants implement parabolic troughs for solar fields [186]. Moreover, the parabolic trough led the market in the growth of CSP-based power generation in the year 2018, with a share of 70% [187]. Table 4-2 lists some examples of commercial parabolic-trough-based CSP plants. Currently, the total capacity of parabolic trough projects is around 4950 MW produced by 87 power plants, with Spain having the highest installed capacity, followed by the United States [186].

Table 4-2 Examples of parabolic trough based CSP projects [48,186]

Project name	Location	Status	Heat transfer fluid	Solar Output (MW)
Andasol-1	Spain	Operational since Nov 2008	Dowtherm A at 293 °C – 393 °C	50
Arcosol 50	Spain	Operational since Dec 2011	Diphenyl/Diphenyl Oxide at 293 °C – 393 °C	49.9
Arenales	Spain	Operational since Nov 2013	Diphyl at 293 °C – 393 °C	50
Colorado Solar Project	United States	Operational since 2010	Xceltherm 600 at 190 °C – 300 °C	2
Holaniku	United States	Operational since Dec 2010	Xceltherm 600 at 93 °C – 176 °C	2
Bokpoort	South Africa	Operational since March 2016	Dowtherm A at 293 °C – 393 °C	55

Ilanga I	South Africa	Operational since Nov 2018	Thermal oil at 293 °C – 393 °C	100
Kathu Solar Park	South Africa	Operational since Jan 2019	Thermal oil at 293 °C – 393 °C	100
Delingha	China	Operational since Oct 2018	Thermal oil at 293 °C – 393 °C	50
Gansu Akesai	China	Under Construction	Molten salt	50
Godawari Solar Project	India	Operational since June 2013	Dowtherm A at 293 °C – 393 °C	50
Airlight Energy Ait-Baha	Morocco	Operational since June 2014	Air at 270 °C – 570 °C	3
NOOR I	Morocco	Operational since Dec 2015	Dowtherm A at 293 °C – 393 °C	160
NOOR II	Morocco	Operational since Jan 2010	Thermal oil at 293 °C – 393 °C	200
DEWA	United Arab Emirates	Under Construction	Thermal oil	600
Shams 1	United Arab Emirates	Operational since March 2013	Therminol VP-1 at 300 °C – 400 °C	100
Archimede	Italy	Operational since July 2010	Molten salt at 290 °C – 560 °C	5
Thai Solar Energy 1	Thailand	Operational since Jan 2012	Water at 201 °C – 340 °C	5

Regarding the maintenance and operation of parabolic troughs, there are two essential activities: periodic measurement of mirror reflectivity and washing. It has been observed that the reflectivity of a mirror can directly influence the amount of useful thermal energy harnessed by the solar collectors; for example, a 10% higher reflectivity leads to a 10% rise in useful thermal energy generated [182]. Furthermore, ambient air carries dust that tends to cover the surface of the mirrors, reducing the mirror reflectivity; as a result, mirrors must be washed regularly, depending on the geographical location and season [182].

4.3.2 Solar Power Tower

With the solar power tower, one of the first recorded works was conducted in Russia in 1957 by the Russian Professor Victor Baum, who proposed the concept of ground-based mirrors (heliostats) separated from the receiver would focus sunlight on a single receiver [188]. Furthermore, Baum found that the mirrors on the polar side of the receiver were exposed to an average angle of incidence closer to zero, which showed better effectiveness; therefore, Baum suggested mounting heliostats on a train of rail cars moving on a circular rail route to maintain the focus on the polar side. However, he did not receive funding for such a project, even though the university gave him permission for construction of the

device. Another early work on the concept of solar power tower was carried out in Italy in 1968 by Professor Giovanni Francia, who implemented a simple clockwork drive to automatically redirect the solar beam to the receiver to produce very high-temperature steam with little supervision [182]. In 1972, the solar power tower or central receiver concept was reinvented by Hildebrand et al., who proposed tall tower surrounded by a field of heliostats with a hemispherical or cylindrical receiver on top [189].

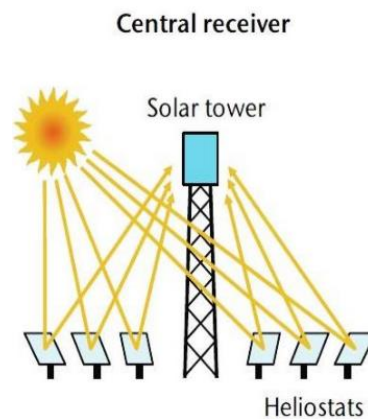


Figure 4-11 Solar Power Tower [185]

The principle of the solar power tower is the same as that of the parabolic trough, which involves focusing the sunlight onto a solar receiver to raise the temperature of heat transfer fluid and then using the heated fluid in steam generation to generate electricity. However, the main difference lies in replacing the linear receiver with a single-point receiver placed at the top of the central tower to capture all the thermal energy provided by the heliostats around it [183]. A solar power tower is a system with large flat mirrors that focus sunlight to the top of the tower, which can be centrally located at the solar field as presented in Figure 4-11[190]. Because the receiver is generally made of metals or ceramics, SPTs can maintain stable operation at very high working temperatures [12]. The flat mirrors, also known as heliostats, follow the sun movement in a two-axis tracking pattern during the day to heat the working fluid and then transfer the energy to a Rankine cycle [191]. Among concentrated solar power technologies, solar power towers are one of the fastest growing solar thermal projects [12].

Table 4-3 Examples of solar tower based CSP projects [48,192]

Project name	Status	Location	Production start date	Heat transfer fluid	Turbine capacity (MW)
<i>Sundrop</i>	Operational	Australia	2016	Water	1.5
<i>Cerro Dominador</i>	Operational	Chile	2021	Molten salt	110
<i>Shouhang Dunhuang II</i>	Operational	China	2018	Molten salt at 290 °C – 565 °C	100
SUPCON	Operational	China	2013	Water at 310 °C – 565 °C	10
<i>Jülich</i>	Operational (pilot plant)	Germany	2008	Air	1.5
<i>NOOR III</i>	Operational	Morocco	2018	Molten salt	150
GEMASOLAR	Operational	Spain	2011	Molten salt at 290 °C – 565 °C	20
<i>Khi Solar One</i>	Operational	South Africa	2016	Water	50
<i>Greenway Mersin</i>	Operational (pilot plant)	Turkey	2012	Water	1.4

4.3.3 Features and Performance

Although the two concentrated solar power technologies mentioned above follow the same concept of converting solar energy into thermal energy, each has unique features, which are summarised in Table 4-4. Both parabolic troughs and solar power towers have different capabilities for solar energy conversion, with the latter having higher annual solar-to-electricity efficiency while the former tends to require less land in order to produce the same given gross power output. Both technologies can generally achieve very high operating temperatures, depending on the properties of the applied heat transfer fluid, for the purpose of converting the solar field feedwater into either saturated steam or superheated steam. Currently, the parabolic trough is the most mature concentrated solar power type, with lower development risk than the solar power tower; however, in terms of technology outlook, solar power towers recently have shown more promising improvements than parabolic troughs.

Table 4-4 Major features of CSP technologies: PT and SPT

Features	Parabolic Trough	Solar Power Tower
Annual solar-to-electric efficiency (%) [12]	15	20 - 35
Optical efficiency [12]	medium	medium
Heat transfer fluid [12]	Synthetic oil, water/steam, molten salt, air	Water/steam, molten salt, air
Operating temperature range (°C) [12]	290 - 550	250 - 650
Steam conditions (°C)/bar [12]	380 to 540/100	540/100 to 160
Land use (km ² /MW) [193]	0.025	0.036
Development status [12]	Most proven	Mature
Technology development risk [12]	Low	Medium
Outlook for improvement [12] [113]	Limited in terms of options. The focus is to increase performance and lower costs by replacing components such as mirror material and working fluid type	Very significant in terms of applications. The focus is on utilising the high working temperatures not only for Rankine cycle but also for Brayton cycle (e.g., heating compressed air)

With any performance evaluation, the boundaries of the evaluation need to be defined. In the case of the CSP system, this is illustrated in Figure 4-12. The major thermal parameters that can be evaluated within these boundaries are the thermal power output, outlet temperature and thermal efficiency. First, the thermal power output is the amount of thermal power generated from the solar field and harnessed by the heat transfer fluid, as calculated in Equation (4-18) [184], where $\dot{Q}_{CSP}$ is the thermal power output (kW), $\dot{m}_{HTF}$ is the mass flow rate of the heat transfer fluid (kg/s), $h_{HTF,out}$ is the enthalpy of the hot HTF exiting the solar energy system (kJ/kg), and $h_{HTF,in}$ is the enthalpy of the cold HTF entering the solar energy system (kJ/kg):

$$\dot{Q}_{CSP} = \dot{m}_{HTF}(h_{HTF,out} - h_{HTF,in}) \quad (4-18)$$

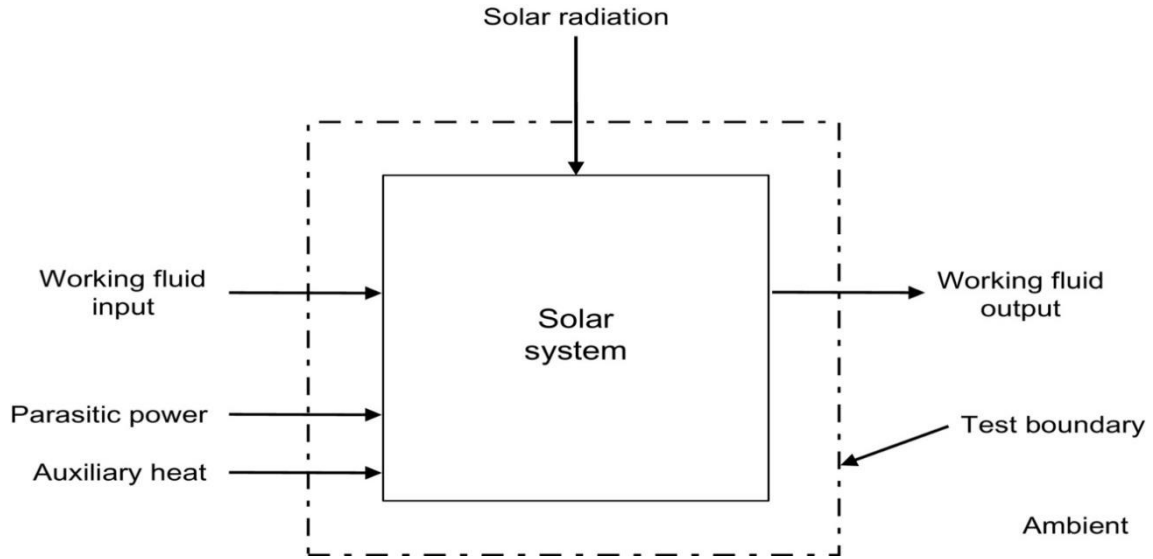


Figure 4-12 Simple schematic of the CSP system test boundary [184]

The outlet temperature is related to the hot temperature of the heat transfer fluid exiting the solar energy system and therefore has a direct effect on the power cycle, which can be estimated using instruments and average values, as illustrated in Equation (4-19) [184], where y is the number of measured HTF temperatures and $T_{HTF(out)}$ is the average HTF hot temperature (K):

$$T_{HTF(out)} = \frac{1}{y} \sum_{i=1}^y T_{HTF(out),i} \quad (4-19)$$

The last common parameter of the thermal evaluation is the CSP efficiency, which is related to the performance solar energy system in converting solar radiation into useful thermal power [184].

4.4 Description of Gasification Technology

4.4.1 Background

The investigation of the concept of gasification with solid feedstock can be traced back to the year 1659 in England, when Thomas Shirley used it to provide town gas for street lighting [67]. Until 1940, lighting and heating for streets and homes

remained the main purpose of gasification, which helped the Industrial Revolution by extending factories' working hours, especially on short winter days, which continued even though the need for gasification declined with the discovery of natural gas [67]. However, gasification experienced another milestone between 1940 and 1975, when the process was used to run internal combustion engine and chemical synthesis into oil due to oil shortages during the Second World War, resulting in the construction of a million small gasifiers for transportation uses [67]. With the end of the war and the turn toward oil-rich West Asian countries (the Middle East), the gasification application to transportation and chemical production was eliminated, but the gasification process continued in other fields, such as syngas production from natural gas and naphtha through steam reforming [67]. Gasification had another milestone due to the 1973 oil embargo on the United States and other western countries, which strongly impelled the development of gasification and other technologies to reduce these countries' dependence on imported oil. Even after the embargo had ended and natural gas prices had dropped, some governments continued improving gasification, which led to their supporting the use of the gasifier in large scale environmental and power generation projects, such as integrated gasification combined cycles [67]. Since the year 2000, environmental concerns, in conjunction with political instability in some oil-producing countries, have given another development push to gasifiers, rendering the gasification process a renewable technology by replacing coal with carbon-neutral feedstock [67].

4.4.2 The Chemistry of Gasification

As shown in Table 4-5, the gasification process consists of a variety of chemical reactions, that generate syngas, a mixture of gases with concentrations strongly dependant on the type of feedstock, chemical composition of the feedstock, gasifying agent and operating conditions used in the gasifier [194–197]. Furthermore, the kinetic rates and conversion extents of the chemical reaction occurring within the gasifier are influenced by the temperature, pressure, gasifier design and gas composition [196–199].

Table 4-5 Common gasification reactions [67]

Reaction Type	Reaction
Carbon reactions	
R1 (Boudouard)	$C + CO_2 \leftrightarrow 2CO + 172 \text{ kJ/mol}$
R2 (water-gas or steam)	$C + H_2O \leftrightarrow CO + H_2 + 131 \text{ kJ/mol}$
R3 (hydrogasification)	$C + 2H_2 \leftrightarrow CH_4 - 74.8 \text{ kJ/mol}$
R4	$C + 0.5O_2 \rightarrow CO - 111 \text{ kJ/mol}$
Oxidation reactions	
R5	$C + O_2 \rightarrow CO_2 - 394 \text{ kJ/mol}$
R6	$CO + 0.5O_2 \rightarrow CO_2 - 284 \text{ kJ/mol}$
R7	$CH_4 + 2O_2 \leftrightarrow CO_2 + 2H_2O - 803 \text{ kJ/mol}$
R8	$H_2 + 0.5O_2 \rightarrow H_2O - 242 \text{ kJ/mol}$
Shift reactions	
R9	$CO + H_2O \leftrightarrow CO_2 + H_2 - 41.2 \text{ kJ/mol}$
Methanation reactions	
R10	$2CO + 2H_2 \rightarrow CH_4 + CO_2 - 247 \text{ kJ/mol}$
R11	$CO + 3H_2 \leftrightarrow CH_4 + H_2O - 206 \text{ kJ/mol}$
R14	$CO_2 + 4H_2 \rightarrow CH_4 + 2H_2O - 165 \text{ kJ/mol}$
Steam reforming reactions	
R12	$CH_4 + H_2O \leftrightarrow CO + 3H_2 + 206 \text{ kJ/mol}$
R13	$CH_4 + 0.5O_2 \rightarrow CO + 2H_2 - 36 \text{ kJ/mol}$

Furthermore, the gasification process is often considered to have four stages or zones, in which the feedstock undergoes a certain process before being converted into syngas: drying, pyrolysis (decomposition), oxidation and gasification, as illustrated in Figure 4-13 [200]. As can be seen from the schematic diagram, the feedstock is fed into the gasifier, first passing through the drying zone in a temperature range of around 100 °C–135 °C in order to evaporate

moisture in the input feedstock. After that, the dried feedstock is decomposed in the pyrolysis zone under temperatures of 500 °C–550 °C into volatile gas and char, which is then reacted with oxygen in the oxidation zone under temperatures of around 800°C – 1000°C. In the last stage, the gasification zone transforms carbonaceous materials into a mixture of gases under temperatures of 800 °C – 1000 °C.

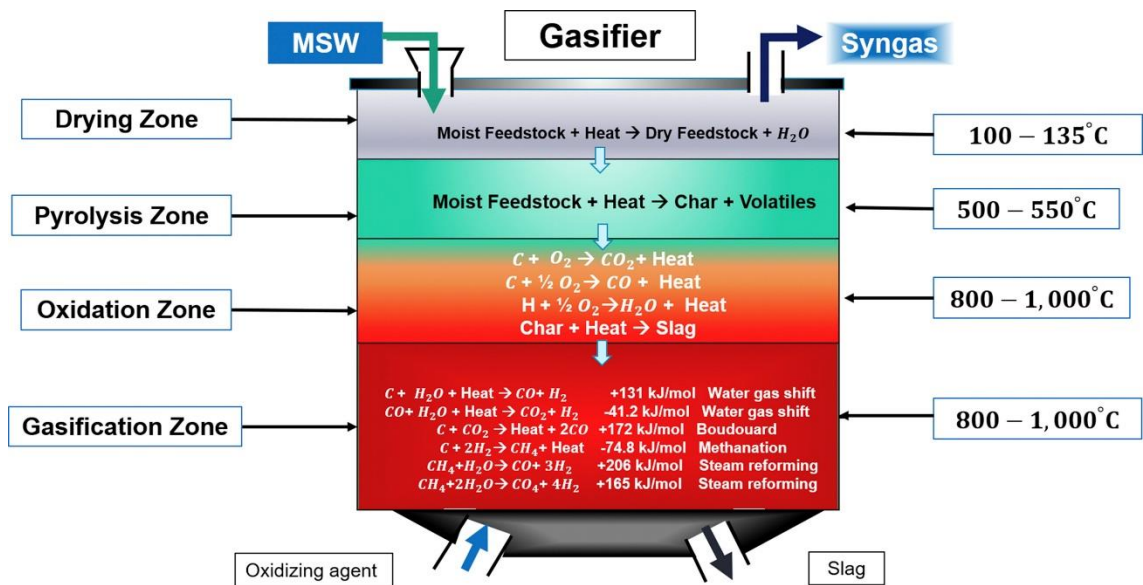


Figure 4-13 The waste gasification steps and reactions [200]

4.4.3 Features and Performance

Although all gasification technologies share the same concept of converting solids into gas, each type has its own design and operating conditions, which offer different outcomes in terms of the quality and quantity of the syngas. This is achieved by utilising the gasifying agent in a controlled amount, generally less than that required for stoichiometric combustion under atmospheric pressure and temperatures between 550 °C and 1600 °C, and which can be used with different types of reactors or gasifiers, as illustrated in Table 4-6 [72].

Table 4-6 Main features of common gasifiers [72]

Gasifier type	Particle size, bulk density and moisture	Temperature profile	Process flexibility, scale-up problems and costs.
Downdraft fixed bed	Waste particles can have a diameter up to 100 mm. The bulk density is larger than 500 kg/m ³ for downdraft and larger than 400 kg/m ³ for updraft. Moisture must be less than 20% and 50% for downdraft and updraft, respectively	Very large high-temperature gradients can occur.	Very limited flexibility, because any change of variables requires a new design. Can be scaled while carefully considering temperature control. Simple reactor with few investment costs.
Updraft fixed bed		Large temperature gradients.	
Bubbling fluidised bed	Waste particle must not exceed 150 mm and bed particles range between 0.08 and 3 mm. The bulk density is higher than 100 kg/m ³ . Moisture must be less than 55%.	Temperature is almost constant in the vertical direction with very low variation in the radial direction. The temperature ranges between 550 °C – 1000 °C.	Excellent flexibility for different solid wastes and operating temperature. Potential problems must be considered and usually a pilot plant is needed. Moderate due to having a small size in addition to low maintenance costs.
Circulating fluidised bed	The maximum diameter limit is 100 mm with bed particle diameter between 0.05 mm and 0.5 mm. The bulk density is greater than 100 kg/m ³ . Moisture must be less than 55%.	Small temperature gradients with a temperature range of 900 °C –1000 °C.	High-quality and varied gasifying agents can be fed at various heights. Some large-scale projects already have been planned. Capital cost tends to be higher than bubbling fluidised bed. In general, this gasifier is convenient for large scale plants.
Entrained flow	Fine fuel particles smaller than 1 mm are added to water before feeding it into gasifier. The bulk density is greater than 400 kg/m ³ . Moisture must be less than 15%.	The temperature range is 1200 °C –1500 °C.	Very limited. Size and energy content of the waste is in a narrow range and commonly requires pre-treatment. The long design allows for very large reactors. Very high capital and operation costs.

The evaluation of gasification involves cold gas efficiency, hot gas efficiency or net gasification efficiency, based on desired parameters such as syngas quality and the flow rate of syngas and feedstock. Cold gas efficiency is defined as the potential energy output over the energy input, as shown in Equation (4-20) [67], where η_{cg} is the cold gas efficiency; m_{syngas} and $m_{solid\ fuel}$ are the mass of

syngas and solid fuel, respectively (kg); and $\dot{Q}_{syngas}$ and LHV_f are the lower heating values for the syngas and solid fuel, respectively (MJ/kg):

$$\eta_{cg} = \frac{\dot{Q}_{syngas} * m_{syngas}}{LHV_{solid\ fuel} * m_{solid\ fuel}} \quad (4-20)$$

Hot gas efficiency is related to the time when the produced gas does not undergo the cooling process in order to achieve better use of energy, which can be calculated as shown in Equation (4-21) [67]. where η_{hg} is the hot gas efficiency, $T_{solid\ fuel}$ and T_{syngas} are the temperature of the solid fuel entering the gasifier and the syngas exiting the gasifier, respectively (K); and C_p is the specific heat capacity of the gas produced (MJ/kg.K):

$$\eta_{hg} = \frac{\dot{Q}_{syngas} * m_{syngas} + C_p * m_{syngas} (T_{syngas} - T_{solid\ fuel})}{LHV_{solid\ fuel} * m_{solid\ fuel}} \quad (4-21)$$

Finally, the net gasification efficiency considers the energy content provided by the gasification medium, energy credit from oil and tar formation and recovery of the heat of vaporisation within the produced gas, which can be expressed in Equation (4-22)

$$\eta_{net-Gasifier} = \frac{\dot{Q}_{syngas}}{\dot{Q}_{solid\ fuel}} \quad (4-22)$$

The quality of syngas is commonly assessed by LHV_{syngas} , which can be estimated using Equation (4-23) [201]. Furthermore, the concentrations of hydrogen and carbon dioxide need to be increased in order to obtain higher syngas quality, since methane concentration is generally low in gas produced by gasification [202].

$$LHV_{syngas} = (25.7 * H_2\% + 30 * CO\% + 85.4 * CH_4\%) * 4.2 \quad (4-23)$$

The volume flow rate of the produced gas V_{syngas} (N.m³/s) can be estimated through Equation (4-24), which includes the thermal output the gas can release, $\dot{Q}_{syngas}$ (MW_{th}) and LHV_{syngas} (MJ/N m³) [67]:

$$V_{syngas} = \frac{\dot{Q}_{syngas}}{LHV_{syngas}} \quad (4-24)$$

For the feedstock flow rate $\dot{m}_{solid\ fuel}$, it can be estimated through Equation (4-25) which includes the amount of thermal output $\dot{Q}_{syngas}$ the gas can release, $LHV_{solid\ fuel}$ and the gasifier efficiency $\eta_{net-Gasifier}$ [67]

$$\dot{m}_{solid\ fuel} = \frac{\dot{Q}_{syngas}}{LHV_{solid\ fuel} * \eta_{net-Gasifier}} \quad (4-25)$$

4.5 Models Development and Validation

The conventional layout of an integrated solar combined cycle plant, as seen earlier, consists of the concentrated solar power technology (solar field and solar steam generator) and the combined cycle gas turbine (gas turbine, steam turbine, condenser, feed water pumps and the heat recovery steam generator). As a base load provider, the combined cycle operates via the gas turbine generating electricity by rotating the turbine blades by mixing fuel with pressurised air in the combustion chamber, then uses the exhaust gas in the preheating, evaporating and superheating of the water/steam stream in the HRSG to drive the steam turbine. As a peak load provider, the solar field harnesses solar heat through the heat transfer fluid flowing in the absorber tubes, to be used within the solar steam generator to heat the extracted fraction of preheated water from the HRSG economiser to produce solar steam. The total steam produced then flows into the inlet of the steam turbine to generate electricity. Both ISCC 1 and ISCC 2 follow the same power generation route in general but differ in the fuel type and in the number of components.

4.5.1 Modelling of Combined Cycle Gas Turbine

The modelling of the combined cycle gas turbine can be accomplished using the Thermoflex software as shown in Appendix C, which tracks energy and mass balances, as described earlier in Section 4.2. Furthermore, this CCGT model can be influenced by a variety of assumptions related to the interlinked parts: gas turbine, HRSG and steam turbine. For example, the gas turbine design or brand

plays an important role in producing electricity as a unit and providing thermal power for the steam turbine unit while generally maintaining an acceptable efficiency. Hence, the first step was to select a commercial gas turbine design from the software library – the SGT-800 by Siemens – due this turbine having a low lifecycle cost, proven reliability, excellent performance and availability and fuel flexibility [203], in order to offer more realistic simulations. The other assumption involved the pressure levels of the steam cycle, which meant specifying the number of pressure inlets of steam into the steam turbine generated by the HRSG. In the heat recovery steam generator, the increase in steam generation from the transition from a single pressure to various pressures improves the combined cycle efficiency [204]. However, there might be a limited benefit for the HRSG when changing from dual-pressure to triple-pressure, which was observed to add extra costs for minimal performance improvement, such as a 0.2% and 0.7 MW rise in net efficiency and net output, respectively [169]. Hence, this research utilises a dual HRSG, which requires two superheaters, evaporators, economisers, steam turbine stages and pumps in order to generate the two streams of steam at high and low-pressure. The last assumption was the type of condensation implemented in the model. It was found that an air-cooled condenser can result in an overall plant efficiency drop by 1%–5% when compared to a water-cooled condenser; however, its main advantage is reducing water usage, especially given that CSP plants tends to be located in places with water resource shortages [205], and therefore the research selected an air-cooled condenser in the model assumptions. It is worth mentioning that the software library does not offer a commercial list of steam turbine designs, but the research assumed the SST-600 from Siemens, which is known for having outstanding reliability, efficiency and fast start up. In addition, its high power output potential and steam inlet temperature, which can reach up to 200 MW and 565 °C, respectively [206], can satisfy the operating conditions within the modelled combined cycle gas turbine. In order to evaluate the accuracy of the CCGT model studied, the simulation results were compared to the reference study conducted by Franchini et al. [75], as shown in Table 4-8 and based on the assumptions mentioned earlier and the input data in Table 4-7. In terms of power

outputs, the gas and steam turbine from the model offered very close values with deviations of only -0.86% and -0.72%, respectively.

Table 4-7 Main input parameters of CCGT at ISO conditions

Design Parameter	Value
GT pressure ratio	19.9
GT power output (MW)	46.4
HP-LP pressures (bar)	69–7.7
HP-LP superheater temperature (°C)	522–218
Approach point temperature (°C)	26
HP-LP pinch point temperature (°C)	9–8
HP-LP turbine efficiency (%)	87.5–83
Condenser pressure (bar)	0.06

Table 4-8 Comparison of CCGT performances (Reference vs Model)

Design Parameter	Reference [75]	This Study	Difference (%)
GT power output (MW)	46.4	46	-0.86
GT exhaust temperature (°C)	548	550	0.36
GT exhaust flow (kg/s)	131.5	130	-1.14
ST power output (MW)	20.8	20.65	-0.72
CCGT gross output (MW)	67.2	66.7	-0.82
CCGT net output (MW)	66.5	65.03	-2.21
Natural gas consumption (MW _{th})	124.4	124.36	-0.03
CCGT net thermal efficiency (%)	53.5	52.3	-2.24

The same can be said regarding the amount of fuel used in the combustion chamber, resulting in the exhaust gas flow maintaining a very high exhaust gas temperature with a deviation of 0.36% from the reference value. These small differences can be attributed to the selected version of SGT-800, since the software library has several variants that differ slightly in their fixed design operating conditions. In terms of the combined cycle net values, there is a somewhat higher deviation of -2.21% and -2.24% for net output and net

efficiency, respectively. Since these values are for the net results, this is of course related to the amount of energy lost to operate components within the CCGT, highlighting that the assumed model loses more energy from the total gross outcomes. However, the simulation values are still in a good agreement with the reference study.

4.5.2 Modelling of Solar field

The modelling of the solar field first conducted in System Advisor Model as shown in Appendix B to extract the annual weather and solar data. Then solar fields are modelled in the Thermoflex software by involve using four major blocks: the CSP technology (parabolic trough or solar power tower), representing both mirrors and receiver as one block, the solar steam generator, the HTF pump and the tank. This solar field model is simulated based on energy and mass balances, as described earlier in Section 4.3. Furthermore, this model can be influenced by a variety of assumptions related to the geometrical and operational assumptions of the solar field. For example, the design of the parabolic trough mirror collector can affect the conversion process of solar radiation into thermal power to later be used for steam generation. Hence, the first step was to select a commercial design from the software library, namely the EuroTrough-150 collectors, due to several advantages, such as its having less shading due to improved absorber support, lower operation and maintenance requirements and a simplified manufacturing that reduces the costs of on-site assembly [207,208], in order to offer more realistic simulations. This represented the first assumption. The second assumption was the use of main geometrical data from the Gemasolar power plant [209] for the solar power tower input which is a commercial plant in Spain. The final assumption was the choice of heat transfer fluid temperature, where the two fluids are heated to 392 °C and 550 °C for Therminol VP-1 and Solar Salt, respectively, based on the temperature range of the reference study [75]. In order to evaluate the accuracy of the solar field model studied, the simulation results were compared to solar field values from the reference study conducted by Franchini et al. [75], as shown in Table 4-10, based on the assumptions mentioned earlier and the input data in Table 4-9 at ISO conditions.

All the considered CSP cases receive high-pressure preheated water from the HRSG and differ in the solar steam condition and injection locations.

Table 4-9 Main geometrical input parameters of the solar field [209,210]

Design Parameter	Value
PT Aperture width (m)	5.76
PT Focal length (m)	1.71
PT Length per assembly (m)	150
PT Mirror area per drive (m ²)	817
PT Receiver diameter (m)	0.07
SPT Heliostat width (m)	10.9
SPT Heliostat height (m)	10.9
SPT Max Heliostat distance to tower ratio	8
SPT Receiver height (m)	14.22
SPT Receiver diameter (m)	8.89

Table 4-10 Comparison of solar field performances (Reference vs Model)

Design Parameter	Reference [75]	This Study	Difference (%)
PT with Thermal Oil			
Solar thermal output (MW _{th})	65.5	65.5	0.00
Solar steam flow (kg/s)	28.5	28.4	-0.35
Solar gross electricity (MW _e)	25.1	24.4	-2.79
PT with Molten Salt			
Solar thermal output (MW _{th})	56.6	56.6	0.00
Solar steam flow (kg/s)	20.13	19.7	-2.14
Solar gross electricity (MW _e)	23.7	22.4	-5.49
SPT with Molten Salt			
Solar thermal output (MW _{th})	60.9	60.9	0.00
Solar steam flow (kg/s)	21.62	20.49	-5.23
Solar gross electricity (MW _e)	25.5	24.1	-5.49

The parabolic trough with thermal oil returns the saturated steam to the boiler for superheating while the parabolic trough and solar power tower use molten salt to generate solar superheated steam and then inject the steam directly to the

turbine. In both steam conditions, the solar field steam is mixed with the HRSG-produced steam which increases the overall stream going into the steam turbine as the power output rises. In terms of thermal power outputs, the model values were aimed to be the same as the reference values by fixing the inlet and outlet temperatures of the HTF while adjusting the HTF flow in the software and then observing the effects on other parameters. The solar-generated steam and expected solar gross power were affected, with model values offering much smaller outcomes than reference values. These deviations can be attributed the geometrical assumptions of the model versus the ones used by Franchini et al. [75]. However, the overall simulation values are still in good agreement with the reference study.

4.5.3 Modelling of Municipal Solid Waste Gasification

The modelling of the gasification process of municipal solid waste can be performed in the Thermoflex software as shown in Appendix C using a simple gasifier block with a heat exchanger and cleaning-up blocks in order to cool and remove contaminants from the produced syngas, respectively, following the chemical reactions, energy and mass balances as described earlier in Section 4.4. Moreover, this gasifier model is dependent on some assumptions related to operating conditions and expected reaction elements and products. For example, the operating conditions depend on the gasifier design; by assuming that the gasifier block represented a fluidised bed gasifier, the gasification temperature had a range from 550 °C to 1000 °C. The reason for considering the fluidised bed gasifier to convert MSW into syngas was because this gasifier has several advantages, such as the ability to maintain constant or uniform temperature within the reactor, flexibility to a wide range of feedstock and low maintenance cost [72]. In this reactor, the fluidisation concept makes both the fuel and inert bed material act as a fluid, thus leading to an efficient mixing of all feedstock particles; in addition, the gasification process operates under nearly isothermal conditions, meaning that it offers a carbon conversion efficiency up to 95% [211]. For the gasification process, the gasifier used in this study receives a MSW feedstock with composition shown in Table 4-11, and compared to the outcomes of the

reference work by Campoy et al. [212], which uses a pilot plant gasifier. This study model for the gasifier is assumed to be steady state and isothermal, while the volatile products mainly consist of H₂, CO, CO₂, CH₄, and H₂O, with all sulphur going to produce H₂S, all chlorine going to HCl, and the char containing only carbon and ash. In order to evaluate the accuracy of the studied gasifier model, the simulation results were compared to the reference study conducted by Campoy et al. [212], as shown in Table 4-11. Based on the assumptions mentioned earlier, the composition of syngas was predicted using Thermoflex simulation. For hydrogen concentration, the model offered a value with a deviation 4.05% while it was 1.36% for carbon monoxide. With regard to the carbon dioxide and methane concentrations, their values deviated from reference study by -0.23% and -2.78%, respectively. It can be seen that each element had different level of deviation, but the overall model outcomes were still in good agreement with the reference study.

Table 4-11 Main input parameters of MSW gasifier at ISO conditions

Design Parameter	Value
MSW proximate analysis (%)	
Moisture	14.30
Ash	44.6
Volatile Matter	47
Fixed Carbon	8.5
MSW ultimate analysis (%)	
C	21.98
H	2.47
N	1.38
S	0.41
O	21.27
Gasifier operating conditions	
Gasifier temperature (°C)	770
Gasifier pressure (bar)	1.013
Gasifying agent flow (Nm ³ /h)	12
MSW flow (kg/h)	17.8

Table 4-12 Comparison of MSW gasification performances (Reference vs Model)

Syngas Composition (%)	Reference [212]	Study model	Difference (%)
H ₂	10.2	14.25	4.05
CO	12.4	13.76	1.36
CO ₂	13.6	13.37	-0.23
CH ₄	2.8	0.016	-2.78

4.6 Sensitivity Analysis

The general purpose of sensitivity analysis is to obtain the variations in an output parameter of a system as a result of important applied input variables. In this section, the sensitivity analysis was carried out for the solar field and gasifier using simulations of the validated sub-models from the previous sections. Furthermore, the sensitivity analysis was very essential in assessment of those renewable components because it was used to indicate the key variables that affected solar field and gasifier performances, which were essential components in running the power block of the hybrid system studied, and as a result could also influence their contributions within the studied hybrid system. This evaluation of components helped to find the optimal design and operating conditions for the thermal inputs into the overall hybrid system.

4.6.1 Solar Field Operating Conditions

There are various parameters that play a significant role in the contribution and overall performance of the solar field. In this analysis, the evaluation was carried out using the weather conditions of an average summer day in the month of June in order to obtain approximate solar field design values, based on the selected regions in Spain and Saudi Arabia. The varying weather data were collected from the Photovoltaic Geographical Information System (PVGIS) by the European Commission, which provides typical meteorological year data of the main variables, including ambient temperature, ambient pressure, relative humidity, direct normal irradiance, global irradiance on the horizontal plane, diffuse irradiance on the horizontal plane, surface infrared irradiance on the horizontal plane and wind speed [213]. As was described earlier, the collection of solar heat

forms the basis of concentrated solar power technology. For this reason, the primary assessment aim in this section was to observe the effect of the design parameters under consideration on the solar field operation and contribution based on understanding from Section 4.3:

- The design point direct normal irradiance, highlighting the expected amount of received radiation per unit area in the examined regions.
- The hot temperature of the heat transfer fluid, presenting the predicted level of heat transfer (as shown in Figure Apx C- 8).

The simulations for these parameters were implemented through the System Advisor Model based on the assumption that the solar field of the integrated solar combined cycle could be modelled as a standalone concentrated solar power plant in the software. The motivation to use this method was to carry out the sensitivity analysis under approximately real weather data from the chosen representative days, based on the values illustrated in Table 4-13.

Table 4-13 Main input parameters in SAM software for the different ISCCs

SAM System Parameter	Parabolic Trough		Solar Power Tower
Cycle turbine gross power (MW _e)	23.5	23.5	23.5
Cycle thermal power (MW _{th})	66	66	66
Solar Multiple	1	1	1
Design point DNI (W/m ²)	700 - 1000	700 - 1000	700 - 1000
Type of HTF	Therminol VP-1	Hitec Solar Salt	Hitec Solar Salt
HTF cold temperature (°C)	293	300	300
HTF hot temperature (°C)	392	550 - 580	550 - 580
Max HTF temperature (°C)	400	593	593
Min HTF temperature (°C)	12	238	238

4.6.1.1 Effect of Design DNI

Based on the definition of concentrated solar power technology mentioned earlier, the direct normal irradiance is the most important parameter in this solar energy concept. However, the fluctuating nature of this energy input throughout the day, based on the location examined, made it extremely necessary to find the

optimal design point DNI. In addition to that, the variations in design DNI for the same nominal power block capacity basically led to changing the area of the solar field. In other words, the use of high design DNI implies the need for lower area in the solar power plant and vice versa. By having the same solar field thermal power output, the effect of the assumed variable was observed for the solar power tower with molten salt, parabolic trough with molten salt and parabolic trough with thermal oil using the weather data of the two locations considered. In Figure 4-14, the different solar fields show that the amount of solar radiation converted into thermal power at different hours of the day under consideration was strongly influenced by the applied design point of the direct normal irradiance. The estimated values varied from few megawatts to approximately 48 MW, 44 MW and 40 MW when $\text{DNI} = 1000 \text{ W/m}^2$ for the solar power tower with molten salt, parabolic trough with molten salt and parabolic trough with thermal oil, respectively. On the other hand, the maximum values of the thermal solar power of these different solar fields when $\text{DNI} = 700 \text{ W/m}^2$ jumped from 48 MW, 44 MW and 40 MW to approximately 62 MW, 59 MW and 56 MW.

As with the Spanish case, the amount of solar thermal power produced for the three solar fields based on the design point DNI was also examined for the day considered and for Saudi Arabian conditions, as illustrated in Figure 4-15. The simulations highlighted that when $\text{DNI} = 1000 \text{ W/m}^2$, the solar thermal power could reach almost 49 MW, 48 MW and 44 MW for the solar power tower with molten salt, parabolic trough with molten salt and parabolic trough with thermal oil, respectively. The same effect of DNI reduction was also observed in the Saudi Arabian area, which enhanced the expected maximum values of the thermal solar power of these three solar fields from 49 MW, 48 MW and 44 MW to around 68 MW, 66 MW and 61 MW. The reason for this was related to how the reduction in solar field size due to higher design point DNI, for example 1000 W/m^2 , has resulted in implementing a smaller harnessing area in locations which typically received below 800 W/m^2 of direct normal irradiance, leading to the production of low solar thermal power when compared to solar fields with lower design point DNI. Consequently, the rise in the produced thermal power from the solar field in an integrated solar combined cycle setup can strongly boost the amount of

generated steam in the HRSG, leading to a drop in stack temperature, which in turn can improve the efficiency of the HRSG.

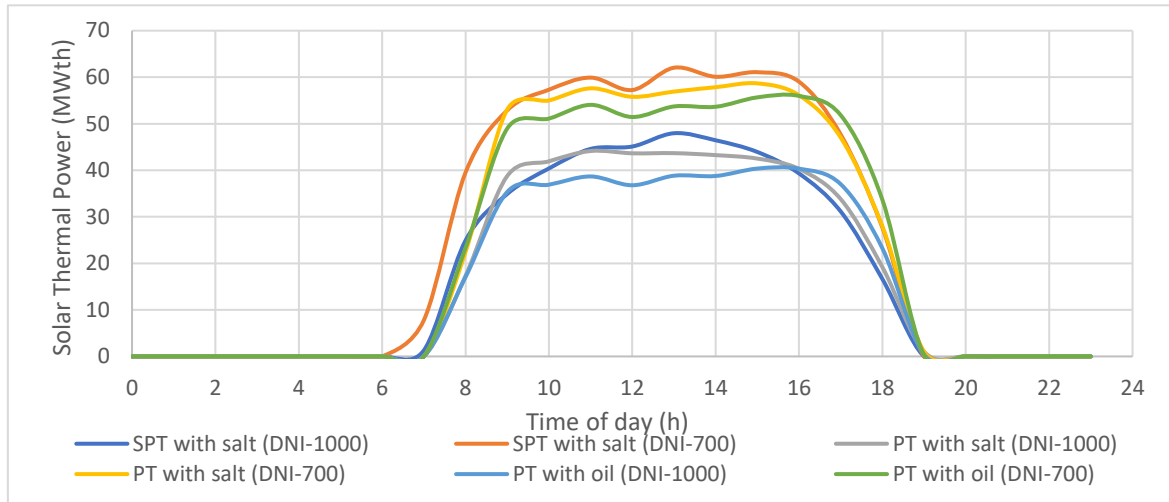


Figure 4-14 Effect of design point DNI on the solar fields for Spanish summer

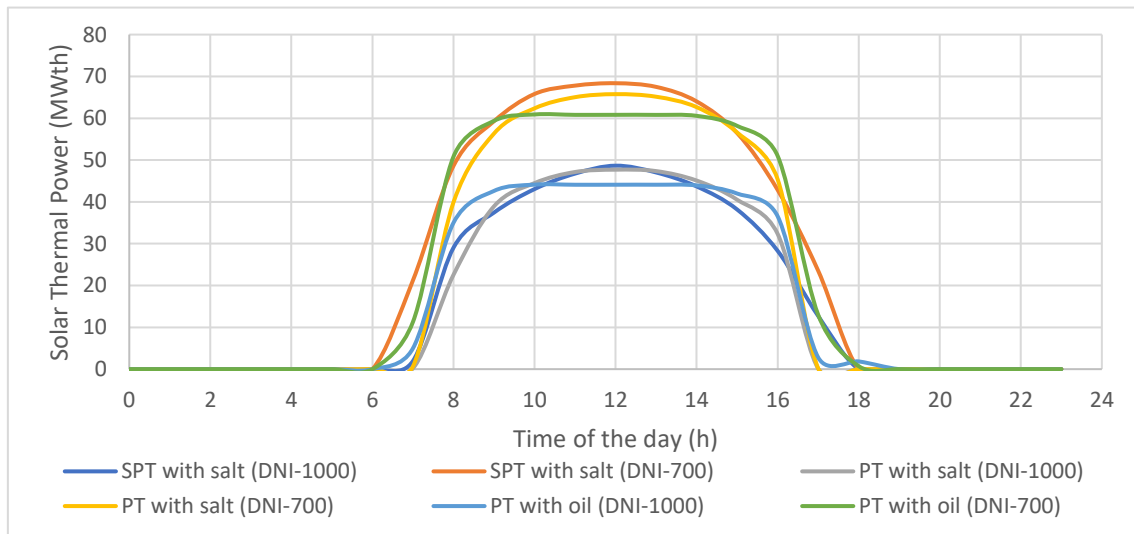


Figure 4-15 Effect of design point DNI on the solar fields for Saudi Arabian summer

4.6.1.2 Effect of HTF Temperature

The other important parameter in concentrated solar power technology is the heat transfer fluid, which absorbs and delivers the heat from focused sunlight radiation to the power block stage in order to generate steam. Since all studied solar field cases for the integrated solar combined systems considered in this research differ in the injection point of the solar-generated steam, the heat transfer fluid temperature range not only plays a significant role in the amount of solar thermal

power, as shown in Equation (4-18), but also in the level of steam production used to offer saturated or superheated steam. With thermal oil, the temperature range usually implies operating around its maximum hot temperature. Nevertheless, the temperature range of molten salt, which is the focus of this section, can be larger since the common applied hot temperature is far below the maximum limit of this fluid. For this reason, variations in the temperature of this heat transfer fluid were examined for the same design solar thermal power in order to observe the effect of the assumed variable on the solar power tower and parabolic trough using the weather data from the two considered locations in the two countries.

Table 4-14 shows that for the two solar fields using molten salt under the weather conditions of Spain and Saudi Arabia, the total amount of solar radiation converted into thermal power over the considered day was slightly affected by the applied design hot temperature. The simulations of the solar power tower under Spanish conditions varied temperatures from 550 °C to 580 °C, which caused the thermal power to drop from approximately 380 MW to approximately 377 MW. The same trend was observed with the parabolic trough, leading to thermal power reduction from around 410 MW to 400 MW as a result of increasing the hot temperature of the molten salt. Analogously to the Spanish case, the total amount of solar thermal power produced by the solar power tower and the parabolic trough based on the design HTF temperature was also examined for the day under consideration using Saudi Arabian weather conditions. The simulations illustrated that the higher the hot temperature of molten salt, the lower the solar thermal power it offered, with 357 MW to 355 MW for a solar power tower at 550 °C to 580 °C. The same effect of thermal power reduction with increasing temperature was also observed in the parabolic trough (377 MW to 388 MW). This impact of the heat transfer fluid can be attributed to the heat transfer concept shown in Equation (4-18), which indicated that higher values of the hot temperature would lead to higher thermal power. However, having a higher molten salt temperature could not maintain a higher thermal power because most of the harnessed solar radiation was used to boost the temperature of this heat transfer fluid but for a smaller fluid mass flow rate. Consequently,

there is a clear effect of molten salt temperature on the produced thermal power from the solar field in an integrated solar combined cycle setup, which in turn affects the amount and quality of the solar generated steam for the steam turbine, influencing the power output and overall stream used in the turbine.

Table 4-14 Effect of molten salt outlet temperature on the solar thermal power

Molten Salt Outlet Temperature (°C)	Solar Thermal Power (MW _{th})			
	Spain		Saudi Arabia	
	Solar Power Tower	Parabolic Trough	Solar Power Tower	Parabolic Trough
580	377.45	399.92	352.92	377.37
570	378.45	402.28	354.49	379.74
560	379.47	406.34	355.76	383.95
550	380.47	409.53	357.26	387.52

4.6.2 Gasification Operating Conditions

In gasification technology, there are several factors which can greatly influence the level of thermal contribution from the generated syngas and the overall performance of this conversion process on the applied solid feedstock. In this analysis, the assessment was conducted based on the use of proximate and ultimate analyses of the selected municipal solid waste in the simulations in order to observe the effect of varying the parameters under consideration on the gasification process operation and contribution, as explained in Section 4.4. Concretely, the following parameters were considered:

- The level of moisture in the implemented MSW composition
- The amount of air entering the reactor
- The applied temperature of the gasifier

The simulations for the gasification technology in the hybrid system were conducted in this section to obtain the optimum syngas composition to replace natural gas in order to run the power block. Since there is a significant difference between the two fuels in terms of their energy content, the research targets the optimum syngas composition that offers the same power output, with less feedstock consumption, when used in a conventional combined cycle gas turbine with no gasifier attached to the system. The motivation to use this method was to carry out the sensitivity analysis and find the optimum operating conditions based on the ranges of variation as illustrated in Table 4-15.

Table 4-15 Main variable and fixed operating parameters in Thermoflex software for the gasification

Parameter	Value
CCGT net power (MW _e)	66.5
Gasifier temperature (°C)	600 - 1000
Air/fuel ratio	0.5 – 0.8
Moisture content (%)	10 – 14
Ambient temperature (°C)	25
Ambient pressure (bar)	1.013

4.6.2.1 Effect of Moisture Content in MSW

The contents of gases within the syngas have a strong influence on this fuel capability and status in power generation. With higher hydrogen and carbon monoxide contents, the syngas consequently has a higher energy content whereas having higher methane and carbon dioxide contents follow an opposite trend [214]. Based on the moisture content characteristics, a substantial percentage of the deliverable energy from a gasification plant tends to decrease by offering less heating value as a fuel, due to the fact that the amount of energy implemented to remove or evaporate this moisture is not recovered [67]. The level of moisture can fluctuate depending on the nature of the received feedstock, but more importantly, this parameter is controllable, leading the feedstock to be dried

through drying process inside a system or leaving it outside for some time to dry naturally. Moreover, there is an effect of moisture content on the quality of the syngas produced, under the assumption that municipal solid waste was dried naturally until the percentage of moisture reached one of the values in the considered range. The motive was to observe the impact of each moisture value on the lower heating value of the gasification product for the purpose of selecting the feedstock composition with optimum outcomes for running the studied integrated solar combined cycle.

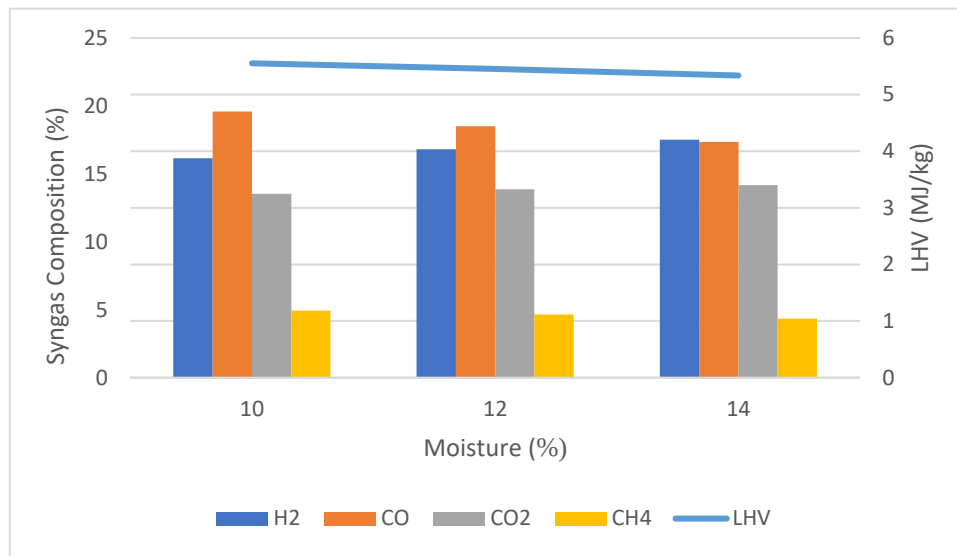


Figure 4-16 Effect of MSW moisture content on the composition of syngas

The moisture content of the municipal solid waste was varied from 10% to 14.3% while fixing the values of the other parameters in the gasifier, as illustrated in Figure 4-16. When observing the composition of produced MSW syngas, we see that there was a notable rise in the concentration of carbon monoxide from around 17% to 20%, which was achieved by the decrease in moisture content to 10%. Moreover, the concentration of methane was also increased through further evaporation of feedstock moisture, but the rise was less than 1%, from 4.3% to 4.9%. Conversely, the concentration of hydrogen experienced a clear reduction from approximately 18% to 16% as a result of the decrease in the moisture content within municipal solid waste. Similarly, there was a slight reduction in the concentration of carbon dioxide from 14.1% to 13.5% following the decrease of moisture content from 14.3% to 10%. The impact of the moisture content on the

lower heating value of the produced syngas was also highlighted in Figure 4-16. The simulations indicate that the lower heating value was improved slightly from around 5 MJ/kg to 6 MJ/kg, with lower values of moisture content leading to a rise in the carbon monoxide and methane concentrations.

4.6.2.2 Effect of Air/Fuel Ratio

In a combustion chamber, the amount of air to apply is determined by the relationship between the stoichiometric amount of air and its excess air coefficient while in the gasification process. The amount of air supply is indicated by what is known as the equivalence ratio (ER), highlighting the relationship between the stoichiometric air and the actual air, which is basically a fraction of the stoichiometric air, as presented in Equation (4-26) [67]:

$$ER (< 1)_{gasification} = \frac{actual\ air}{stoichiometric\ air} = ER (> 1)_{combustion} \quad (4-26)$$

From the equation, the equivalence ratio must be less than 1, since the actual air used in gasification is always less than stoichiometric air. However, the value of this ratio should be chosen wisely, because an excessively low equivalence ratio value may lead to several problems, including incomplete gasification, char formation beyond acceptable limits and a drop in the heating value of the gas produced [67]. On the other hand, very high values of the equivalence ratio can result in excessive generation of products from complete combustion, such as carbon dioxide and steam, at the expense of targeted products, such as carbon monoxide and hydrogen, leading to an overall decrease in the heating value of the produced syngas; for these reasons, the equivalence ratio tends to be kept within the range of 0.20–0.30 to achieve practical gasification outcomes [67].

By using the recommended range of the equivalence ratio, the actual air values for gasification were calculated in order to be used as air/fuel ratios in the Thermoflex simulations. Hence, the air/fuel ratio was varied from 0.5 to 0.8 while keeping the values of the other parameters in the gasifier fixed, as illustrated in Figure 4-17. By viewing the fluctuation in the composition of the produced MSW syngas, it can be seen that there was a notable decrease in the concentration of carbon monoxide and methane, from around 20% to 15% and 8% to 5%,

respectively, which were obtained by increasing the air/fuel ratio to 0.8. Moreover, the concentration of hydrogen also experienced a slight drop, from 14.6% to 12.5%, as a result of increasing the amount of air entering the gasifier. Conversely, the concentration of carbon dioxide experienced a slight increase from 15% to 15.01% when the air/fuel ratio increased from 0.5 to 0.8 within the gasification process for municipal solid waste. In addition to that, the impact of varying the air-to-fuel ratio on the lower heating value that the produced syngas could provide was also illustrated in Figure 4-17. Based on the simulation outcomes, the lower heating value of the MSW syngas notably decreased from around 6 MJ/kg to 4 MJ/kg with higher values of the air/fuel ratio, which caused the drop in the concentrations of carbon monoxide, methane and hydrogen.

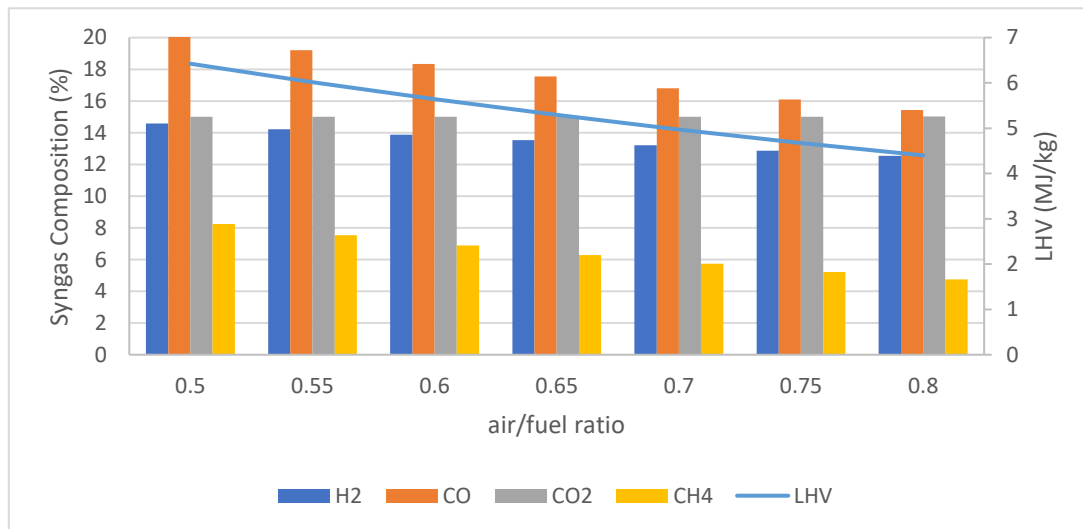


Figure 4-17 Effect of air/fuel ratio on the composition of produced MSW syngas

4.6.2.3 Effect of Gasification Temperature

The choice of gasification temperature is an important process choice, since all the chemical reactions within the gasifier need must occur under a certain hot setting level. This temperature level in the reactor can be controlled depending on the target product conditions. Moreover, the effect of gasifier temperature on the quality of produced syngas can be evaluated through observation of the composition and the lower heating value of the gasification product, for the purpose of selecting the syngas composition with optimum outcomes for running the studied integrated solar combined cycle.

The gasifier temperature was varied from 600 °C to 1000 °C while keeping the values of the other parameters in the gasification process fixed, as illustrated in Figure 4-18. When observing the composition of produced MSW syngas, we see there was a significant increase in the concentration of carbon monoxide and hydrogen from around 4% to 25% and 4% to 20%, respectively, which was achieved by increasing the gasification temperature to 1000 °C. Conversely, the concentration of carbon dioxide was greatly reduced from approximately 26% to 8% by the rise of temperature. Similarly, the concentration of methane experienced a clear reduction from around 13% to 0.6% as result of enhancing the reactor temperature for the moisture content within municipal solid waste. The impact of the gasification temperature on the lower heating value of the produced syngas was also highlighted in Figure 4-18. The simulation outcomes indicated that the lower heating value was slightly improved from 4.4 MJ/kg to 5.5 MJ/kg with higher values of the applied temperature, which led to the significant rise in the carbon monoxide and hydrogen concentrations compensating the drop of methane concentration.

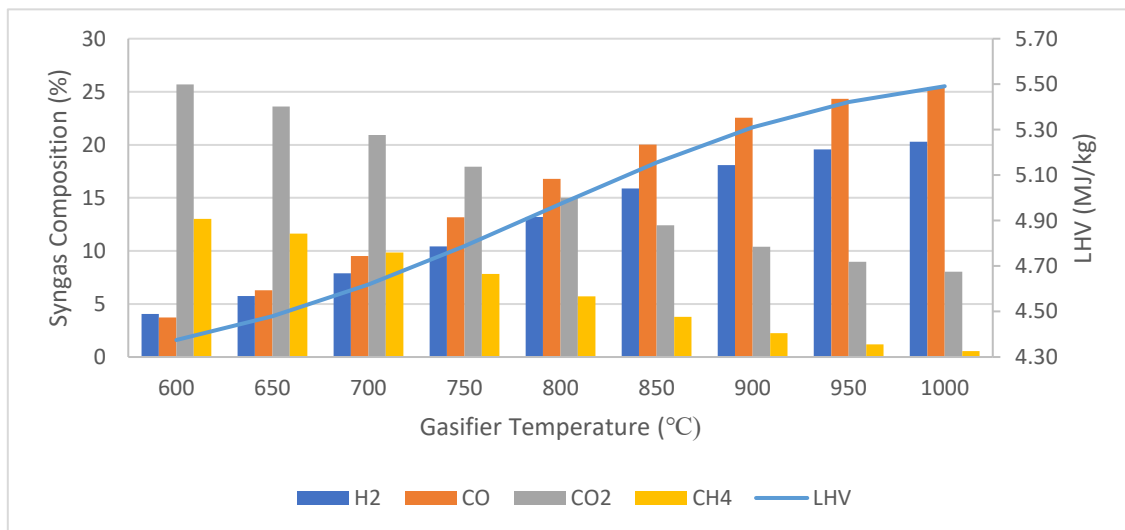


Figure 4-18 Effect of temperature on the composition of produced MSW syngas

4.6.3 Conclusions to Component Performance

At first glance, it seemed theoretically possible that a rise in the design point DNI, solar multiple and heat transfer fluid temperature could achieve much better results for the solar fields. However, the lack of energy storage and the

intermittent nature of solar radiation illustrated a clear effect on the outcomes of the solar field, especially the useful thermal power and efficiency operating under the weather conditions of the summer day under consideration. Based on simulation results for the different parameters, the optimum variables were a design point DNI of 700 W/m^2 , a solar multiple of 1 and a heat transfer fluid outlet temperature of 550°C for molten salt. With respect to the gasifier parameters, there are many possible optimal operating values to utilise for the gasification of municipal solid waste, depending on the goal the syngas composition should accomplish. Since this thesis assumed that the target of the syngas composition was to achieve approximately the same net power output as natural gas when running the power block of the integrated solar combined cycle under study, it was found that a moisture content of 10%, a gasifier temperature of 900°C and an air/fuel ratio of 0.6 gave a MSW syngas composition with a lower heating value of 6 MJ/kg , which adequately met the gasification target. Finally, the optimum values for the solar field and gasifier will further be assessed by implementing them for the studied integrated solar combined cycle under Saudi Arabian and Spanish weather conditions throughout the year.

5 Performance of the hybrid system

5.1 Introduction

The previous chapter described the components of the integrated solar combined cycle studied and their performance in terms of optimum design and operating parameters. Hence, this chapter attempts to implement the component values obtained in the studied scenarios for the integrated solar combined cycle in order to assess the impact of gasified municipal solid waste and solar field thermal power under varying weather data in Spain and Saudi Arabia throughout the year. First, the major features of the hybrid system configurations studied are presented to obtain an overall understanding of each scenario. Second, the integration method between the solar field, gasifier and combined cycle, carried out through the Thermoflex software, is described. Finally, annual simulations of the hybrid system were conducted to evaluate system performance based on the optimum component parameters.

5.2 Studied Configurations

As stated in the methodology, this research utilised four integrated solar combined cycle scenarios, representing the novel configuration versus the conventional one and these models can be seen in Appendix A. In all cases, the combined cycle gas turbine implemented had the same design parameters as illustrated in Table 4-7. While the gas turbine remained the same, the steam turbine slightly changed among the different configurations, depending on the effect of the type of solar and fuel contributions. The details of the four layouts considered are illustrated as follows:

ISCC 1: This configuration, as shown in Figure 5-1, was fuelled by natural gas and used the solar steam generator to extract a fraction of the preheated water from the high-pressure level economiser of the heat recovery steam generator in order to be evaporated and returned to the high-pressure drum of the HRSG as saturated steam using the parabolic trough with thermal oil in the temperature range of 292 °C to 392 °C. The solar steam was then mixed with the other boiler's saturated steam before entering the superheater.

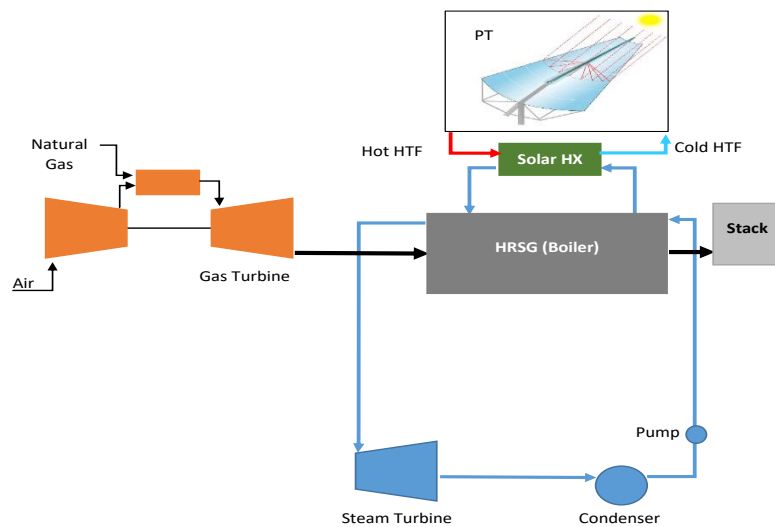


Figure 5-1 ISCC 1 configuration

ISCC 2: This configuration, as shown in Figure 5-2, was fuelled by municipal solid waste through a gasification process and used the solar steam generator to extract a fraction of the preheated water from the high-pressure level economiser of the heat recovery steam generator, to be evaporated and returned to the high-pressure drum of the HRSG as saturated steam using the parabolic trough with thermal oil in the temperature range of 292 °C to 392 °C. The solar steam was then mixed with the saturated steam from the other boiler before entering the superheater.

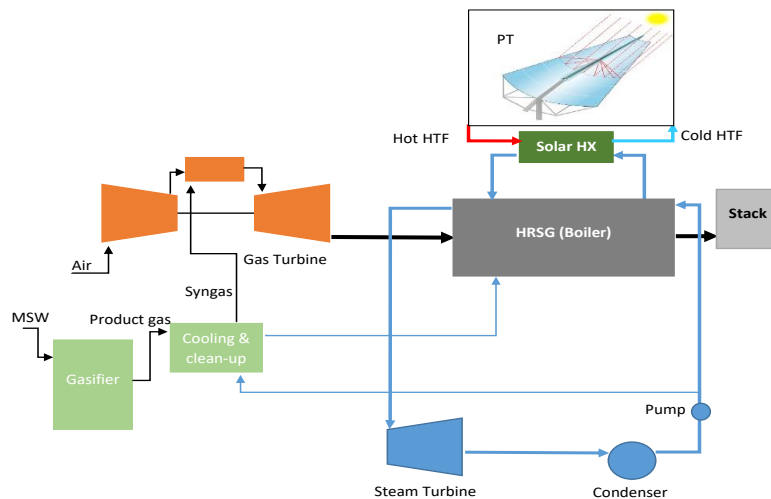


Figure 5-2 ISCC 2 configuration

ISCC 3: This configuration, as shown in Figure 5-3, was fuelled by municipal solid waste through the gasification process and used the solar steam generator to extract a fraction of the preheated water from the high-pressure level economiser of the heat recovery steam generator, to be evaporated, superheated and then mixed with high-pressure superheated steam from the HRSG using the parabolic trough with molten salts in the temperature range from 300 °C to 550 °C. This solar steam is expected to influence the temperature of the steam mixture conditions by either maintaining or enhancing the temperature at the steam turbine inlet.

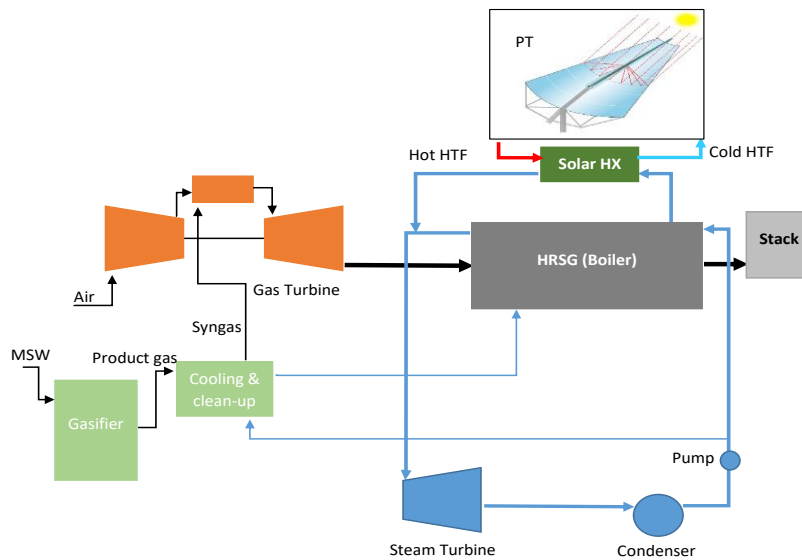


Figure 5-3 ISCC 3 configuration

ISCC 4: This configuration, as shown in Figure 5-4, was fuelled by municipal solid waste through the gasification process and used the solar steam generator to extract a fraction of the preheated water from the high-pressure level economiser of the heat recovery steam generator in order to be evaporated, superheated and then mixed with high-pressure superheated steam from the HRSG using the solar power tower with molten salts in the temperature range from 300 °C to 550 °C. The solar steam is expected to influence the temperature of the steam mixture conditions by either maintaining or enhancing the temperature at the steam turbine inlet.

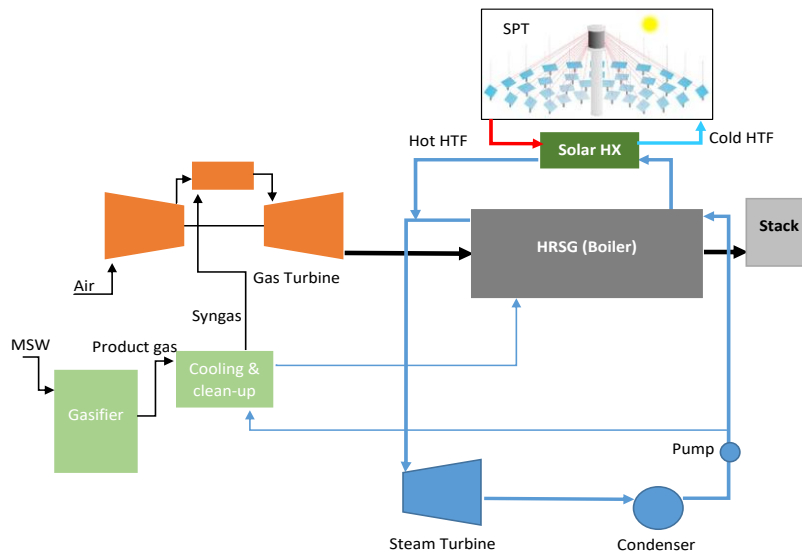


Figure 5-4 ISCC 4 configuration

5.3 Model Development and Validation

Based on the modelling of each component performed in the previous chapter, the overall model was created by assembling these sub-models in various ways to represent the studied configurations of the integrated solar combined cycle. The validation of the hybrid system was already established by validating the sub-models of the components, as shown in Section 4.5. To illustrate the model of the integrated solar combined cycle in the software environment, Figure 5-5 is presented, which based on the solar field type, represents either ISCC 3 and ISCC 4, which are two forms of the proposed novel configuration given in this research. This software was chosen for the modelling of the hybrid system due to its capability to model and simulate various types of energy resources, including concentrated solar power technology and municipal solid waste gasification under design and off-design conditions, which is necessary for the annual performance estimations in this research. The two forms of the proposed hybrid system plant consist of a solar field, which is either a parabolic trough, for ISCC 3, or a solar power tower, for ISCC 4; molten salt as a heat transfer fluid; a gas turbine; and a dual pressure heat recovery steam generator with a steam turbine. The other two configurations, ISCC 1 and ISCC 2, have the same overall

model blocks but differ both in their solar produced steam and the use of the MSW gasification process.

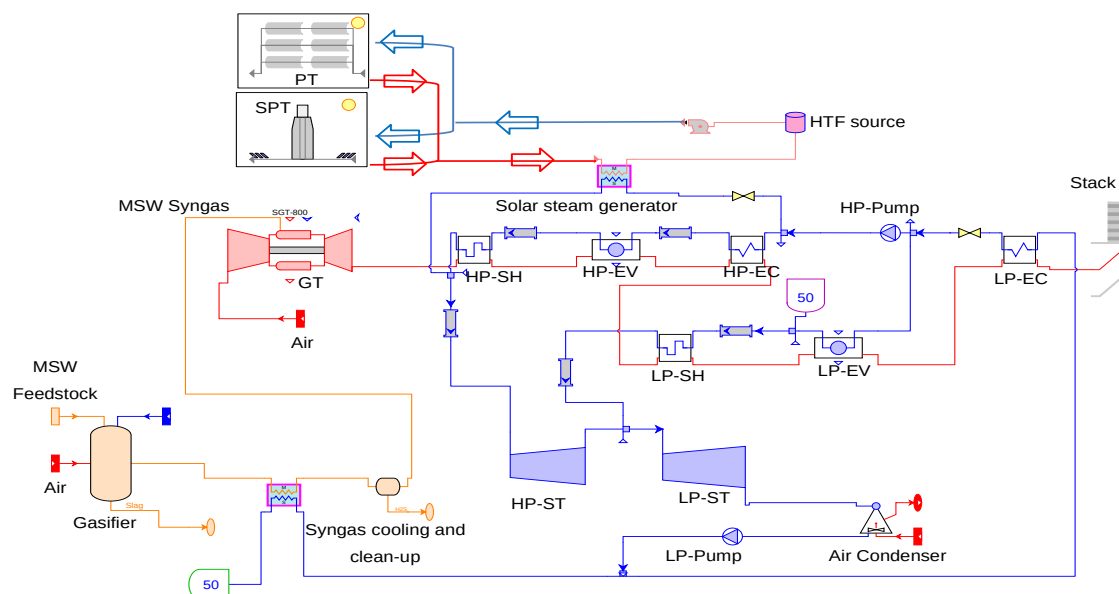


Figure 5-5 Model of the proposed ISCC (ISCC 3 & ISCC 4) in Thermoflex

5.4 Performance Evaluation Indicators

The performance evaluation of the integrated solar combined cycle for both configuration conditions under consideration was carried out. The motive was to determine whether the novel integrated solar combined cycle can achieve the required technical features and outcomes, which are normally obtained using this CSP hybrid system at its conventional form. In order to properly assess how the various studied configurations reacted to changes in ambient conditions throughout the year, the performance evaluation covered several assessment factors or indicators. Since modifications to the solar field and fuel type were considered in this research, these performance indicators covered both system and component characteristics, which illustrated the typical overall outcomes, in addition to highlighting the changes in operating conditions within the system.

5.4.1 System and Component Electrical Output

One of the most important steps in the assessment of any power generation system is estimating the amount of electricity offered. The motive for considering this parameter is its potential to highlight the capability of a system to potentially

operate as an acceptable power plant. For this reason, the research included the level of electricity generation as an indicator to evaluate the proposed configuration of the integrated solar combined cycle to illustrate its potential as a power plant, especially when compared to its current configuration. For this type of hybrid system, it was also necessary to find the solar fraction of total produced electricity due to the differences between the solar fields in the studied configurations. The equations used for the calculation of total electricity and its solar fraction are as follows:

$$\dot{W}_{CCGT} = \dot{W}_{GT} + \dot{W}_{ST(GT\ Exhaust)} \quad (5-1)$$

$$\dot{W}_{ISCC} = \dot{W}_{GT} + \dot{W}_{ST(GT\ Exhaust)} + \dot{W}_{ST(CSP)} \quad (5-2)$$

$$\dot{W}_{ST(CSP)} = \dot{W}_{ISCC} - \dot{W}_{CCGT} \quad (5-3)$$

Here Equation (5-1) is used to represent the total amount of net power produced by the combined cycle gas turbine $\dot{W}_{CCGT}$ (MW), or the combined cycle of the ISCC when there is no sunlight input (cloudy or at night), through the net power combination of gas turbine $\dot{W}_{GT}$ (MW) and steam turbine $\dot{W}_{ST}$ (MW). In Equation (5-2), the total produced net power from the integrated solar combined cycle $\dot{W}_{ISCC}$ (MW) includes the solar thermal contribution by using the increased net electrical generation from the steam turbine, $\dot{W}_{ST(CSP)}$ (MW), which is the steam turbine that receives the steam flow generated by the gas turbine exhaust through the heat recovery steam generator and the solar field. To specify the amount of solar-generated electricity $\dot{W}_{ST(CSP)}$ (MW), Equation (5-3) simply subtracts the total generated electricity generated without sunlight from that generated with solar input.

5.4.2 System and Component Thermal Power

Another important step in the assessment of a power generation system such as an integrated solar combined cycle is estimating the amount of energy input. The motive for considering this parameter is not only to emphasise the total energy consumption that must be provided for proper operation, but also to specify the amount of solar contribution in this energy input. For this type of hybrid system, it

was also necessary to present the solar fraction of the total consumed thermal power due to the differences between the solar field operating conditions in the studied configurations. The equations used for the calculation of total thermal energy and its solar share are as follows:

$$SS = \frac{\dot{Q}_{CSP}}{\dot{Q}_{Fuel} + \dot{Q}_{CSP}} * 100 \quad (5-4)$$

$$FS = \frac{\dot{Q}_{Fuel}}{\dot{Q}_{Fuel} + \dot{Q}_{CSP}} * 100 \quad (5-5)$$

$$\dot{Q}_{CSP} = \dot{m}_{HTF} * c_p * (T_{HTF(hot)} - T_{HTF(cold)}) \quad (5-6)$$

$$\dot{Q}_{Fuel} = \dot{m}_{Fuel} * LHV_{Fuel} \quad (5-7)$$

Here Equation (5-4) and Equation (5-5) represent the solar share (SS) (%) and fuel share (FS) (%) as fractions of the total amount of thermal power, which is produced by the combination of thermal power provided by the solar field conversion of sunlight $\dot{Q}_{CSP}$ (MW) and the generated thermal heat as a result of fuel combustion $\dot{Q}_{Fuel}$ (MW). In Equation (5-6), the amount of solar thermal power can be estimated using the operating conditions of the working fluid in the solar field, where $\dot{m}_{HTF}$ (kg/s), c_p (kJ/kg °C), $T_{HTF(hot)}$ (°C) and $T_{HTF(cold)}$ (°C) are the heat transfer fluid mass flow rate, specific heat capacity, and the temperature of the hot and cold streams, respectively. In order to estimate the amount of thermal power obtained from fuel combustion, it is common to use Equation (5-7), which specifies the fuel mass flow rate $\dot{m}_{Fuel}$ (kg/s) and the lower heating value LHV_{Fuel} (kJ/kg) for either natural gas or municipal solid waste.

5.4.3 System and Component Efficiencies

A further crucial parameter to consider when assessing any power generation system is the quality of the applied system and its components in utilising the available energy input while fulfilling the necessary energy requirements, in other words, the efficiency. The motive for implementing this parameter in the evaluation is its potential to highlight the obtained useful outcomes that the system can provide after receiving the energy input. Hence, this research

illustrates how the efficiency of the integrated solar combined cycle differs as a result of changes in solar and fuel energy inputs. In addition to that, the efficiency of the solar field was also presented, due to the modifications in the operating conditions and technology type. The equations used for the calculation of the different efficiencies are as follows:

$$\eta_{th-CCGT} = \frac{\dot{W}_{CCGT}}{\dot{Q}_{Fuel}} * 100 \quad (5-8)$$

$$\eta_{th-ISCC} = \frac{\dot{W}_{ISCC}}{\dot{Q}_{Fuel} + \dot{Q}_{CSP}} * 100 \quad (5-9)$$

$$\eta_{sol-rad-el} = \frac{\dot{W}_{CSP}}{DNI * A_{ap}} * 100 \quad (5-10)$$

$$\eta_{sol-th-el} = \frac{\dot{W}_{CSP}}{\dot{Q}_{CSP}} * 100 \quad (5-11)$$

$$\eta_{rad-th} = \frac{\dot{Q}_{CSP}}{DNI * A_{ap}} * 100 \quad (5-12)$$

Here Equation (5-8) represents the expected thermal efficiency of the combined cycle gas turbine or the combined cycle of the ISCC when there is no sunlight input (cloudy days or at night), η_{CCGT} (%), based on the amount of thermal input by fuel combustion and the resulting net power output from the power block. In Equation (5-9), the expected thermal efficiency of the integrated solar combined cycle η_{ISCC} (%) includes the effect of the solar contribution in terms of the thermal input and the rise in overall net electricity generation. To generally illustrate the amount of available solar radiation at a specified location and aperture area, $DNI * A_{ap}$ (MW), that has been converted into net solar electricity, the assessment can implement what is known as ‘solar radiation to electricity’, shown in Equation (5-10). Furthermore, Equation (5-11) focusses on highlighting the amount of thermal power from the solar field that has been transferred into net power output through the solar thermal to electricity efficiency $\eta_{sol-th-el}$ (%). Lastly, Equation (5-12) basically evaluates the quality of the implemented solar field in converting radiation into useful thermal power through what is known as the thermal efficiency of the solar field, η_{rad-th} (%).

5.4.4 Inlet Temperature of Steam Turbine

A final important parameter in the assessment of the power generation system considered is the impact of utilising a fuel with less heating value than natural gas, in this case municipal solid waste. The motive for considering this parameter is not only to observe the effect of operating the steam turbine at lower temperatures but also to specify the impact of solar steam on the inlet stream of the steam turbine. Since the temperature of the steam turbine $T_{ST(in)}$ is basically a case of mixing fluid between two streams with different temperatures, the calculations can be accomplished based on the following equations [215]:

$$m_{Mix} = m_1 + m_2 + \dots + m_n \quad (5-13)$$

$$T_{Mix} = \frac{m_1 * c_1 * T_1 + m_2 * c_2 * T_2 + \dots + m_n * c_n * T_n}{m_1 * c_1 + m_2 * c_2 + \dots + m_n * c_n} \quad (5-14)$$

Here Equation (5-13) represents the total mass of a mixture m_{Mix} (kg) composed of several masses of fluids, $m_1 + m_2 + \dots + m_n$, which in this case is due to the solar-produced steam mixing with the steam generated by the municipal solid waste. For the temperature of the mixture T_{Mix} (°C) or at the inlet of steam turbine, we can use Equation (5-14) based on the temperatures of the fluids $T_1 + T_2 + \dots + T_n$ (°C) and the specific heats of the fluids $c_1 + c_2 + \dots + c_n$ (kJ/kg °C).

5.5 The Meteorological Data

By using the System Advisor Model, the meteorological data from the European Commission PVGIS database were extracted and implemented in the performance evaluation. There are a variety of ambient weather conditions which can influence the performance of the integrated solar combined cycle, but the two most important parameters were the direct normal irradiance (W/m²) and ambient temperature (°C). For the direct normal irradiance, Figure 5-6 illustrates this solar radiation throughout the year, with a maximum value of approximately 1000 W/m² in some hours and total of around 2162 kWh/m² yearly in the Spanish city, Sevilla. Maintaining the design point DNI at 700 W/m² for the different solar fields was suitable for harnessing the radiation with these annual fluctuations beyond 700 W/m² on this Spanish city, Sevilla. In Figure 5-7, the direct normal irradiance is

shown under the weather conditions of Saudi Arabia with hourly variations that can reach maximum value of approximately 1000 W/m^2 and a yearly total of around 2345 kWh/m^2 . As with the Spanish conditions, it can be emphasised that using the design point DNI at 700 W/m^2 when sizing the considered solar fields seemed reasonable for better collection of the varying radiation throughout the year for this Saudi Arabian city, Al Ahsa.

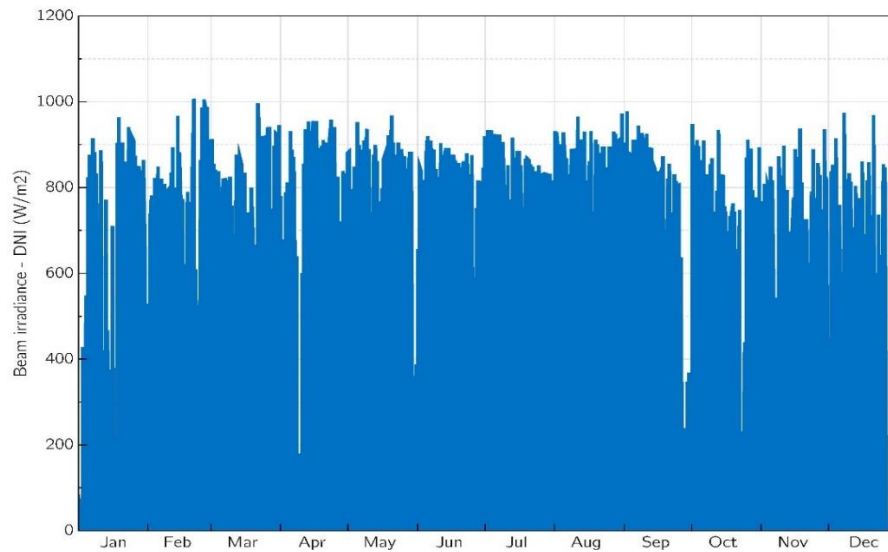


Figure 5-6 The variations of direct normal irradiance during a typical year under Spanish weather conditions (extracted from SAM)

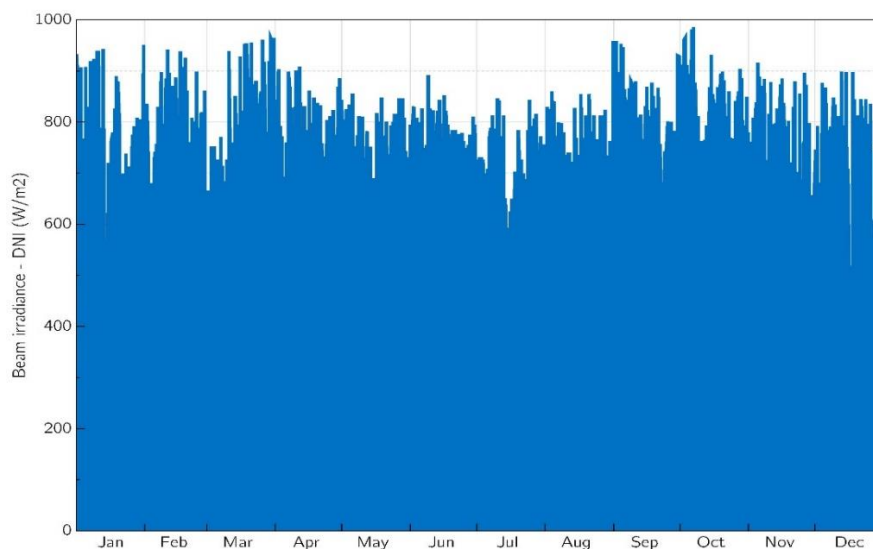


Figure 5-7 The variations of direct normal irradiance during a typical year under Saudi Arabian weather conditions (extracted from SAM)

With respect to the ambient temperature, Figure 5-8 illustrates the variations of this weather parameter throughout the year with a maximum value of approximately 40 °C typically in summer and an annual average of around 19 °C. In Figure 5-9, the ambient temperature is shown under the weather conditions of Saudi Arabia, with hourly variations that can reach a maximum value of approximately 47 °C during summer and an annual average of 26 °C. Based on these figures, the integrated solar combined cycle will experience greater impact during summer in terms of gas turbine performance, due to the noticeable drop of inlet air density. This effect of higher values of the ambient temperature is expected to last longer in Saudi Arabia than in Spain, since the former country is exposed to ambient temperatures higher than 40 °C from May until September, while the latter's ambient temperatures tend to be around or below 35 °C during the same season.



Figure 5-8 Annual variations of ambient temperature in Sevilla, Spain

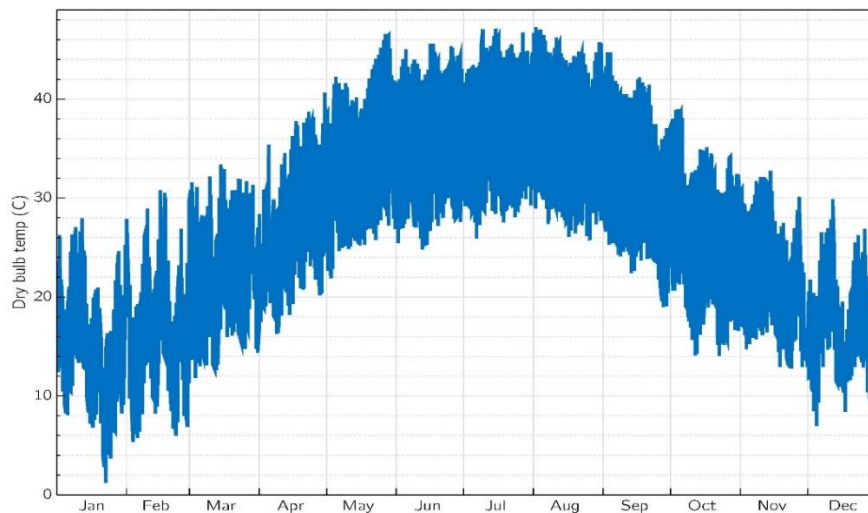


Figure 5-9 Annual variations of ambient temperature in Al Ahsa, Saudi Arabia

5.6 Main Assumptions

Based on the optimum design conditions for the solar fields and gasification of municipal solid waste, which were selected in the previous chapter, the off-design conditions for the different integrated solar combined cycles were evaluated throughout the year. This was accomplished by considering variation in the direct normal irradiance and ambient temperature for each hour of the year and calculating the performance of 8760 operating cases or points for the integrated solar combined cycle layouts studied. In order to run and evaluate these simulations, the following assumptions were implemented for this CSP hybrid system:

- The amount of municipal solid waste at the site was taken to be abundant, and therefore it was assumed that there were no feedstock supply issues and the power plant ran continuously on baseload using the produced syngas.
- The solar field was expected to provide additional thermal power only during peak hours, except during the night or on cloudy days.
- The integrated solar combined cycle in all configurations was considered to operate at full load and shutdowns or maintenance were not considered.
- The design thermal power for the solar field was chosen to be 66 MW_{th} for all studied layouts based on the solar thermal power in reference study [75].

- The steam generated by ISCC 1 and ISCC 2 was at a saturated level with temperatures around 285 °C, while ISCC 3 and ISCC 4 produced a superheated steam at 530 °C.
- The city of Sevilla in Spain and the city of Al Ahsa in Saudi Arabia were the specified locations for the weather conditions inputs in the simulations.
- The implemented municipal solid waste composition was dry, composed of organic matter only and was the same for ISCC 2, ISCC 3 and ISCC 4.
- In the Thermoflex software, the flow of the obtained heat transfer fluid from SAM software was used as a solar input in simulations to represent the effect of weather variation on the solar field for each hour throughout the year.

5.7 Results of the Hybrid System Simulations

5.7.1 Design Conditions

In this step, all configurations were considered to operate for one hour under ISO conditions to observe the effect of the parameters of the selected optimum components on the typical operating conditions and outcomes for the hybrid system. The results of these design conditions showed a difference in performance outcomes caused by the use of municipal solid waste gasification and the enhancement of solar steam conditions, as presented in Table 5-1.

Table 5-1 Performance of ISCC configurations at design conditions (one hour)

Parameter	ISCC 1	ISCC 2	ISCC 3	ISCC 4
$\dot{W}_{ISCC}$ (MW)	88.3	90.25	92.7	92.5
$\dot{W}_{CSP}$ (MW)	23.3	21.95	24.4	24.2
$\dot{Q}_{Fuel}$ (MW _{th})	124.5	114.5		
SS (%)	34.65	36.62		
$\eta_{th-ISCC}$ (%)	46.35	50	51.36	51.25
T_{STin} (°C)	524	514	524	524

By comparing configurations ISCC 1, ISCC 2, ISCC 3 and ISCC 4 using Table 5-1, it can be argued that the implementation of gasification technology in ISCC 2, ISCC 3 and ISCC 4 offered an increase of around 2% to 5% in the net produced electricity when compared to the ISCC 1, due to the additional steam generated while cooling the produced MSW syngas, and therefore resulted in a lower required consumption of municipal solid waste, as described in Equation (5-7). A similar argument could be made for the solar fields that used molten salt, which generated superheated steam, and which were able to offer an increase in the net produced electricity by around 2.5% for ISCC 3 and 3% for ISCC 4 when compared to ISCC 2, which used thermal oil to generate saturated steam. This was attributed to the improved stream conditions at the inlet steam turbine, which managed to keep steam mixture operating at the design temperature, as described in Equation (5-14). Finally, the comparison between ISCC 3 and ISCC 4 indicated that the former was capable of reaching higher outcomes at the operating conditions considered. This was related to the parabolic trough's tendency to have higher optical efficiency than the solar power tower during summer, which is the typical design season for concentrated solar power technology, and therefore a larger portion of the incident solar radiation can be harnessed [75]. In addition to that, the ISCC 3 configuration offered the best thermal efficiency for an integrated solar combined cycle in the hot months. Having said that, a proper assessment required observing how each of the studied configurations performed under off-design conditions, which included the impact of varying weather conditions throughout the year, in order to have more realistic outcomes for the comparative analysis.

5.7.2 Off-Design Conditions

5.7.2.1 Effect on power outputs

Based on the concept of the integrated solar combined cycle, as mentioned earlier, the fluctuations in weather conditions, such as the direct normal irradiance and the ambient temperature, as illustrated in Figure 5-6 through Figure 5-9, was expected to have a significant impact on the power generation process of this hybrid system power plant. For this reason, it was necessary to indicate which

configuration was able to maintain high electricity production under the varying seasons conditions of both Spain and Saudi Arabia.

Table 5-2 Effect of Spanish weather on the total monthly net power output

	ISCC 1 (GWh)	ISCC 2 (GWh)	ISCC 3 (GWh)	ISCC 4 (GWh)	Monthly DNI (kWh/m ²)	Average Ambient Temperature (°C)
January	49.50	51.94	51.78	53.27	117	11
February	45.37	47.69	47.44	48.87	128	13
March	51.80	54.54	54.38	55.60	190	14
April	50.34	53.26	53.37	53.97	187	16
May	51.51	55.59	55.85	56.16	207	22
June	47.79	52.86	53.31	53.62	243	26
July	49.72	55.33	56.05	56.22	271	28
August	49.27	54.47	54.98	55.30	255	26
September	48.22	52.77	52.79	53.75	188	24
October	49.82	53.41	53.13	54.68	159	20
November	47.78	50.37	50.10	51.97	116	15
December	49.12	51.58	51.32	53.00	105	12

In Table 5-2, the differences in the net power output between the studied cases of the integrated solar combined cycle were presented on a monthly basis in order to observe the fluctuations of outcomes during the year in Spanish weather. The simulation results for all cases were approximately between 45 GWh and 59 GWh with the highest values occurring during the warm and hot seasons. For the conventional layout, ISCC 1 managed to achieve a maximum production of 51.80 GWh and an average of 49.19 GWh per month. In the other layouts that implemented municipal solid waste, the produced electricity reached higher values than the conventional configuration. With ISCC 2, it managed to reach a maximum of 55.59 GWh, in May in addition to an average of around 54 GWh.

However, ISCC 3, using higher solar thermal conditions, was able to reach 56.05 GWh in the same month, while the overall average was 52.87 GWh. Nevertheless, it is noted that among the four configurations, ISCC 4 offered the highest maximum and average values, around 56.22 GWh and 53.87 GWh, respectively. As with the Spanish case, the amount of produced electricity in Saudi Arabia followed similar trends with varying solar radiation and ambient temperature, as illustrated in Table 5-3. The estimated outcomes of the studied configurations offered monthly averages of around 47.67 GWh, 52.67 GWh, 52.69 GWh and 53.66 GWh for ISCC 1, ISCC 2, ISCC 3 and ISCC 4, respectively. In terms of maximum values, the conventional layout ISCC 1 reached 51.16 GWh in March, while ISCC 2, ISCC 3 and ISCC 4 managed to go beyond that, with values of 54.84 GWh, 54.84 GWh and 55.81 GWh.

Table 5-3 Effect of Saudi Arabian weather on the total monthly net power output

	ISCC 1 (GWh)	ISCC 2 (GWh)	ISCC 3 (GWh)	ISCC 4 (GWh)	Monthly DNI (kWh/m²)	Average Ambient Temperature (°C)
January	51.16	53.93	53.64	55.81	187	14
February	46.58	49.25	48.99	50.34	165	16
March	50.65	54.84	54.84	55.77	189	22
April	45.44	50.48	50.63	51.03	167	27
May	47.73	54.39	54.63	54.81	217	33
June	45.72	52.39	52.74	52.94	230	35
July	45.82	52.80	52.85	53.15	213	38
August	46.36	53.39	53.66	53.97	223	36
September	46.10	52.24	52.51	53.20	220	32
October	48.40	53.37	53.46	54.69	224	26
November	48.02	51.94	51.61	53.47	162	22
December	50.06	53.05	52.70	54.78	149	17

From the net power output values of the integrated solar combined cycle in each case, the solar contribution to the overall electricity production can be highlighted. In Figure 5-10, the monthly differences in the solar net power output were presented over the course of the year under Spanish weather conditions. Based on the simulations, the values of all cases ranged from approximately 0.86 GWh to 8 GWh with highest values occurring during the warm and hot seasons. The conventional layouts, ISCC 1 and ISCC 2, which had similar solar field conditions, parabolic trough and thermal oil, were able to achieve a maximum outcome of 6.72 GWh during July and an overall average of 4 GWh per month. On the other hand, ISCC 3, which used a parabolic trough with molten salt, reached solar electricity values not far from those of ISCC 1 and ISCC 2, 6 GWh and 3 GWh for the maximum and annual average, respectively. Finally, the impact of the solar power tower with molten salt in solar electricity production for ISCC 4 was noticeably higher than all the cases in every month, with a maximum value of around 8 GWh in July and an annual average of approximately 5 GWh. As with the Spanish case, the amount of solar-produced electricity in Saudi Arabia followed similar trends during variations in weather conditions over each month as illustrated in Figure 5-11. The estimated outcomes of the studied configurations offered a monthly average of approximately 5 GWh, 4 GWh, 6 GWh for both ISCC 1 and ISCC 2, ISCC 3 and ISCC 4, respectively. In terms of maximum values, the conventional layouts, ISCC 1 and ISCC 2, reached 6.3 GWh, while ISCC 3 and ISCC4 managed to go beyond that, with solar electricity values of around 6.6 GWh and 6.9 GWh, respectively.

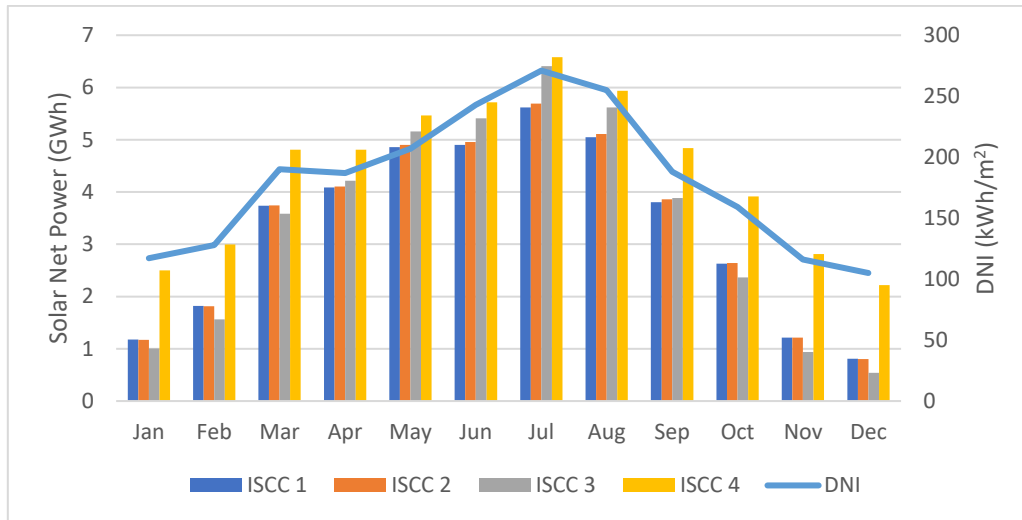


Figure 5-10 Effect of Spanish weather on the solar net power output

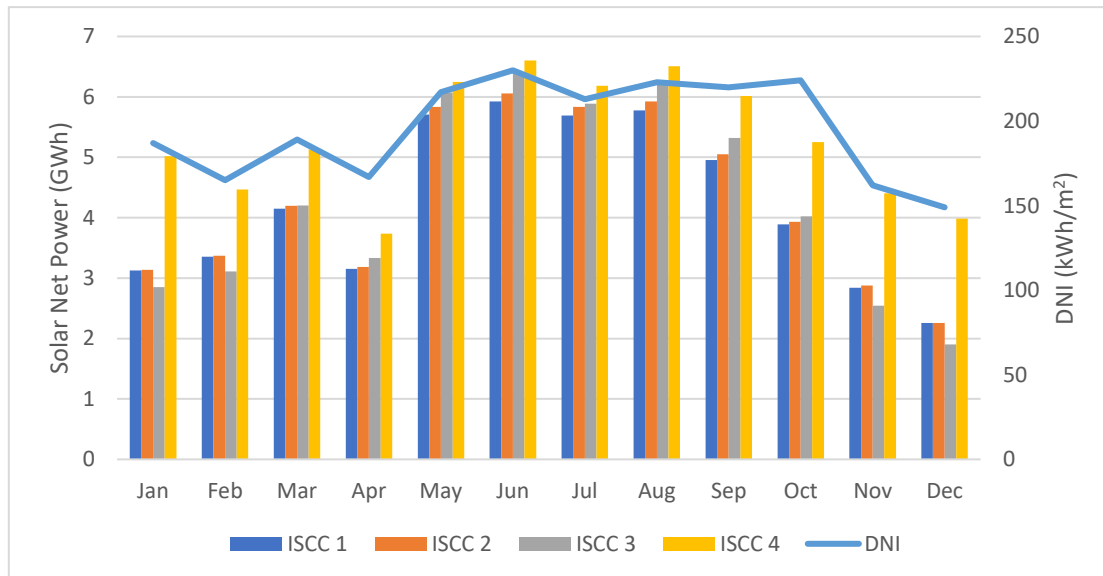


Figure 5-11 Effect of Saudi Arabian weather on the solar net power output

5.7.2.2 Effect on thermal powers

It has been highlighted how the power generation process of the integrated solar combined cycle was strongly influenced by weather conditions. The other influencing factor for power generation was the amount of available thermal power input to the system, which also changed depending on the variable weather conditions throughout the year. In Table 5-4, the monthly differences in the total thermal power input between the studied cases of the integrated solar combined cycle fluctuate based on the effect of weather in the Spanish city. The

simulations for all configurations were approximately between 80 GWh and 104 GWh, with the highest values occurring during the warm and hot seasons. For the conventional layout, ISCC 1 consumed the highest thermal power in order to produce electricity with a maximum value of around 104 GWh in August and an average of 98.2 GWh per month. In the other layouts, which implemented municipal solid waste, the total applied thermal power mostly ranged through lower values than the conventional configuration. With ISCC 2, it managed to reach a maximum of 102 GWh in July, in addition to an expected average of 93.2 GWh, while ISCC 3, which used higher solar thermal conditions was able to reach 103 GWh in the same month; however, the overall average was slightly less, at 92.9 GWh. Nevertheless, it can be noted that among the MSW gasification configurations, ISCC 4 consumed the closest thermal power to ISCC 1, with highest maximum and average values of around 104 GWh and 96 GWh, respectively. As with the Spanish case, the amount of applied thermal power for Saudi Arabia changed with varying solar radiation and ambient temperature, as illustrated in Table 5-5. The estimated outcomes for the studied configurations had a monthly average of around 96.5 GWh, 94.1 GWh, 93.2 GWh and 96.7 GWh for ISCC 1, ISCC 2, ISCC 3 and ISCC 4, respectively. In terms of the maximum values, the conventional layout ISCC 1 reached 101.7 GWh in March, while ISCC 2, ISCC 3 and ISCC4 had values of 98.2 GWh, 97.3 GWh and 101.9 GWh.

Table 5-4 Effect of Spanish weather on the monthly thermal power input

	ISCC 1 (GWh)	ISCC 2 (GWh)	ISCC 3 (GWh)	ISCC 4 (GWh)	Monthly DNI (kWh/m ²)	Average Ambient Temperature (°C)
January	95.78	88.27	87.78	91.64	117	11
February	88.25	81.71	80.97	84.76	128	13
March	102.25	95.29	94.37	97.96	190	14
April	99.67	93.43	93.13	95.18	187	16
May	103.26	98.72	98.73	99.94	207	22
June	100.10	98.21	98.68	100.01	243	26
July	103.44	101.99	103.01	103.92	271	28

August	103.84	101.89	102.52	103.90	255	26
September	96.57	93.27	92.83	95.65	188	24
October	97.55	92.18	91.22	95.49	159	20
November	92.31	85.53	84.76	89.65	116	15
December	94.67	87.21	86.50	90.89	105	12

Table 5-5 Effect of Saudi Arabian weather on the monthly thermal power input

	ISCC 1 (GWh)	ISCC 2 (GWh)	ISCC 3 (GWh)	ISCC 4 (GWh)	Monthly DNI (kWh/m²)	Average Ambient Temperature (°C)
January	100.39	93.46	92.11	98.72	187	14
February	92.04	86.12	84.59	88.93	165	16
March	101.27	96.94	95.97	99.21	189	22
April	95.11	93.51	92.83	94.90	167	27
May	97.51	97.88	97.34	98.81	217	33
June	93.99	94.85	94.36	95.94	230	35
July	94.71	96.07	94.97	96.73	213	38
August	95.19	96.38	95.95	97.73	223	36
September	94.95	94.88	94.33	97.05	220	32
October	100.54	98.17	97.14	101.85	224	26
November	94.59	90.16	88.70	94.31	162	22
December	97.60	91.16	89.69	95.72	149	17

From the values of thermal power used for the integrated solar combined cycle in each case, the solar contribution to the total thermal power can be presented. In Figure 5-12, the monthly differences in the solar thermal power from the solar fields were illustrated throughout the year under Spanish weather conditions. Based on the simulations, the values for all cases ranged from approximately 1.3 GWh to 19.4 GWh, with the highest values occurring during the warm and hot

seasons. The conventional layouts, ISCC 1 and ISCC 2, which had similar solar field conditions, a parabolic trough and thermal oil, were able to harness a maximum production of 17.4 GWh during July and an overall average of 9.7 GWh per month. On the other hand, ISCC 3, which used a parabolic trough with molten salt, reached solar thermal values not far from ISCC 1 and ISCC 2, 18.5 GWh and 9.4 GWh, for the maximum and annual average, respectively. Finally, the impact of the solar power tower with molten salt in solar thermal generation for ISCC 4 was noticeably higher than in all the other cases in every month, with a maximum value of around 19.4 GWh in July and an annual average of approximately 12.3 GWh. As with the Spanish case, the amount of solar thermal production in Saudi Arabia was affected by the variations of the weather conditions in each month, as illustrated in Figure 5-13. The estimated outcomes of the studied configurations managed to reach a monthly average of approximately 12.2 GWh, 11.2 GWh, 14.7 GWh for both ISCC 1 and ISCC 2, and ISCC 3 and ISCC 4, respectively. In terms of maximum values, the conventional layouts, ISCC 1 and ISCC 2, reached 16.5 GWh, while ISCC 3 and ISCC 4 reached values of around 16 GWh and 17.6 GWh, respectively.

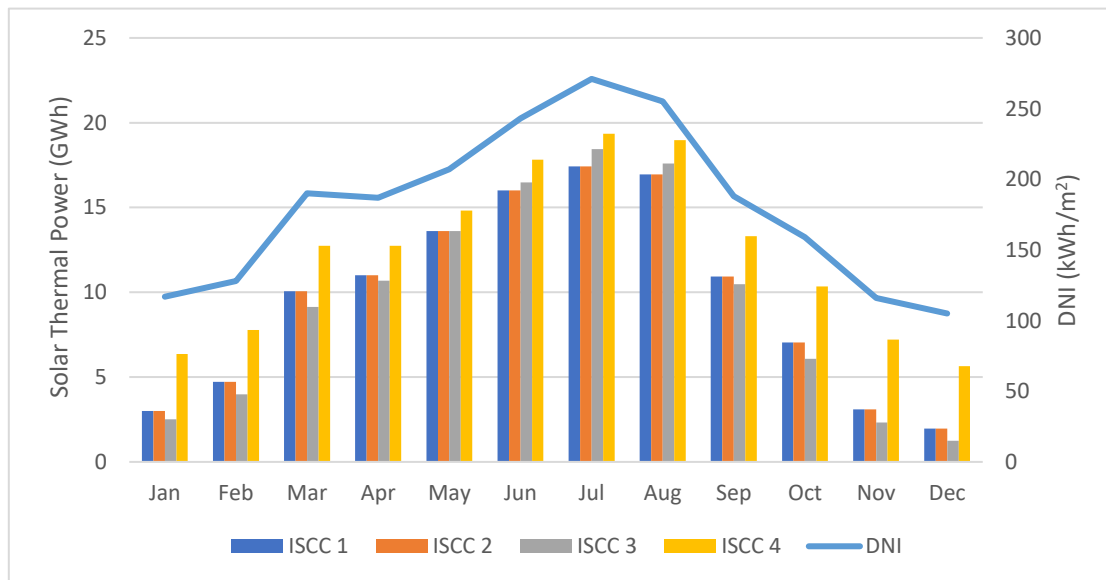


Figure 5-12 Effect of Spanish weather on the solar thermal power input

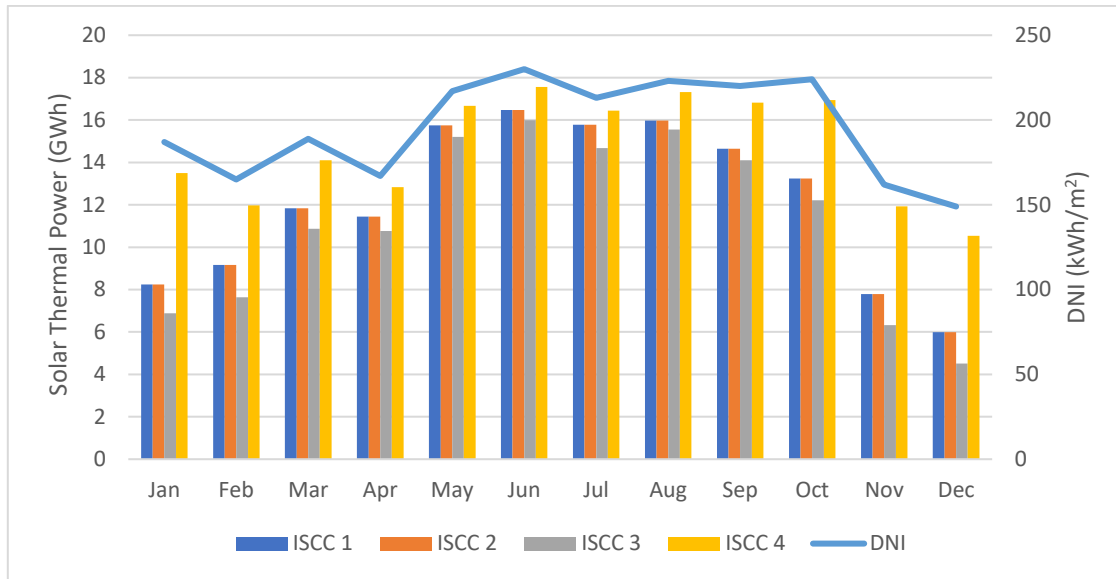


Figure 5-13 Effect of Saudi Arabian weather on the solar thermal power input

5.7.2.3 Effect on efficiencies

In order to assess the way integrated solar combined cycle handled the available energy input for electricity production, the thermal efficiency was considered a major indicator. In Table 5-6, the differences between the studied cases in terms of the integrated solar combined cycle thermal efficiency were presented on a monthly basis to observe the fluctuations in the outcomes over the year in Spanish weather. All cases were approximately between 50% and 60%, with the highest values occurring during the cold seasons. For the conventional layout, ISCC 1 had the lowest thermal efficiency range, with a maximum value of 52% and an average of 51 GWh per month. In the other layouts, which implemented municipal solid waste, the thermal efficiency ranged at higher values than the conventional configuration. With ISCC 2, it managed to reach a maximum of 59.5% in addition to an average of around 57.59%. However, ISCC 3, using higher solar thermal conditions, was able to reach 59.65%, but the overall average was 58%. For ISCC 4, the highest maximum and average values were around 58.76% and 57.55%, respectively.

Table 5-6 Effect of Spanish weather on the monthly thermal efficiency

	ISCC 1 (%)	ISCC 2 (%)	ISCC 3 (%)	ISCC 4 (%)	Monthly DNI (kWh/m²)	Average Ambient Temperature (°C)
January	51.68	58.85	58.99	58.13	117	11
February	51.40	58.36	58.58	57.66	128	13
March	50.65	57.23	57.62	56.76	190	14
April	50.51	57.01	57.31	56.70	187	16
May	49.89	56.31	56.57	56.19	207	22
June	47.74	53.82	54.02	53.61	243	26
July	48.06	54.25	54.41	54.10	271	28
August	47.45	53.47	53.63	53.22	255	26
September	49.94	56.57	56.87	56.20	188	24
October	51.07	57.94	58.25	57.27	159	20
November	51.76	58.90	59.10	57.97	116	15
December	51.89	59.14	59.33	58.31	105	12

As with the Spanish case, the level of thermal efficiency in Saudi Arabia followed similar trends with varying solar radiation and ambient temperature, as illustrated in Table 5-7 the estimated outcomes of the studied configurations offered a monthly average value of around 50.4%, 57.4%, 57.9% and 56.95% for ISCC 1, ISCC 2, ISCC 3 and ISCC 4, respectively. In terms of maximum values, the conventional layout ISCC 1 reached 51.6%, while ISCC 2, ISCC 3 and ISCC4 managed to exceed that with values of 58.7%, 59.2% and 57.8%. Another indication of how well the input energy was used was the capability of the solar field to convert solar radiation into electricity in each case of integrated solar combined cycle. In Figure 5-14, the monthly differences in the solar-to-electricity efficiency under Spanish weather conditions over the course of the year were presented, showing that all cases ranged from approximately 4% to 20%, with the highest values obtained during the warm and hot seasons.

Table 5-7 Effect of Saudi Arabian weather on the monthly thermal efficiency

	ISCC 1 (%)	ISCC 2 (%)	ISCC 3 (%)	ISCC 4 (%)	Monthly DNI (kWh/m²)	Average Ambient Temperature (°C)
January	50.97	57.70	58.24	56.53	187	14
February	50.60	57.18	57.91	56.61	165	16
March	50.01	56.56	57.14	56.21	189	22
April	47.77	53.99	54.54	53.77	167	27
May	48.95	55.57	56.13	55.47	217	33
June	48.64	55.24	55.89	55.18	230	35
July	48.38	54.96	55.65	54.95	213	38
August	48.71	55.39	55.93	55.23	223	36
September	48.55	55.06	55.67	54.82	220	32
October	48.15	54.37	55.03	53.69	224	26
November	50.77	57.61	58.19	56.70	162	22
December	51.29	58.20	58.75	57.23	149	17

In the conventional layout, ISCC 1, and ISCC 2, which had similar solar field conditions, a parabolic trough and thermal oil, were able to achieve a maximum outcome of 18.98% during June and an overall average of around 14.3% per month. On the other hand, ISCC 3, which used a parabolic trough with molten salt, reached solar-to-electricity efficiency values not far from ISCC 1 and ISCC 2, 20.3% and 14.1% for maximum and annual average, respectively. Finally, the impact of the solar power tower with molten salt on solar radiation conversion into electricity for ISCC 4 was noticeably higher than in all other cases, with a maximum value of around 18% and a monthly average of approximately 17%. As with the Spanish case, the level of solar-to-electricity efficiency in Saudi Arabia was affected by varying weather conditions in each month, as illustrated in Figure 5-15. The estimated outcomes of the studied configurations offered a monthly average of approximately 18.42%, 18.1%, 17.8% for both ISCC 1 and ISCC 2,

and ISCC 3 and ISCC 4, respectively. In terms of the maximum values, the conventional layouts, ISCC 1 and ISCC 2, reached 21.8% while ISCC 3 and ISCC 4 managed to attain values of around 22.3% and 18.4%, respectively.

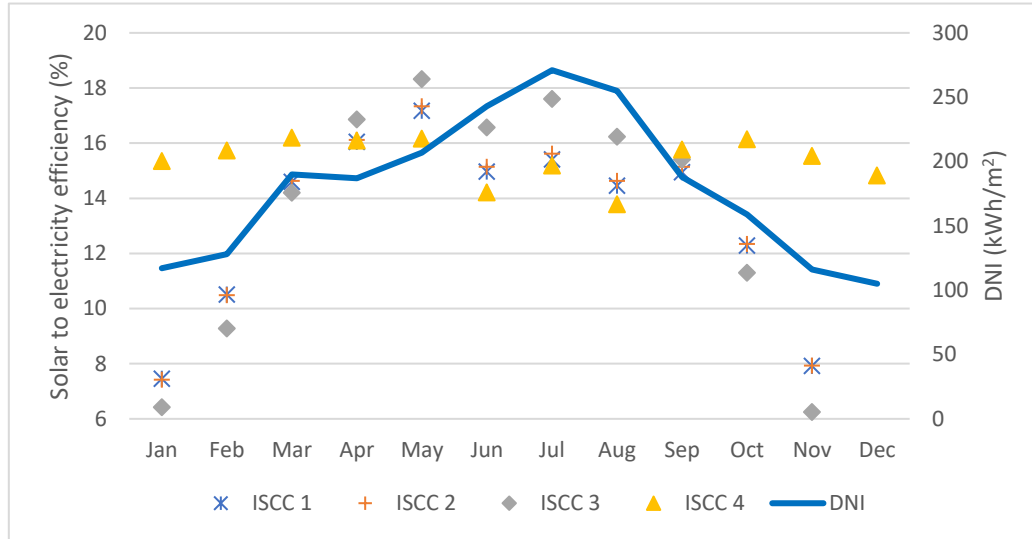


Figure 5-14 Spanish weather impact on the monthly solar to electricity efficiency

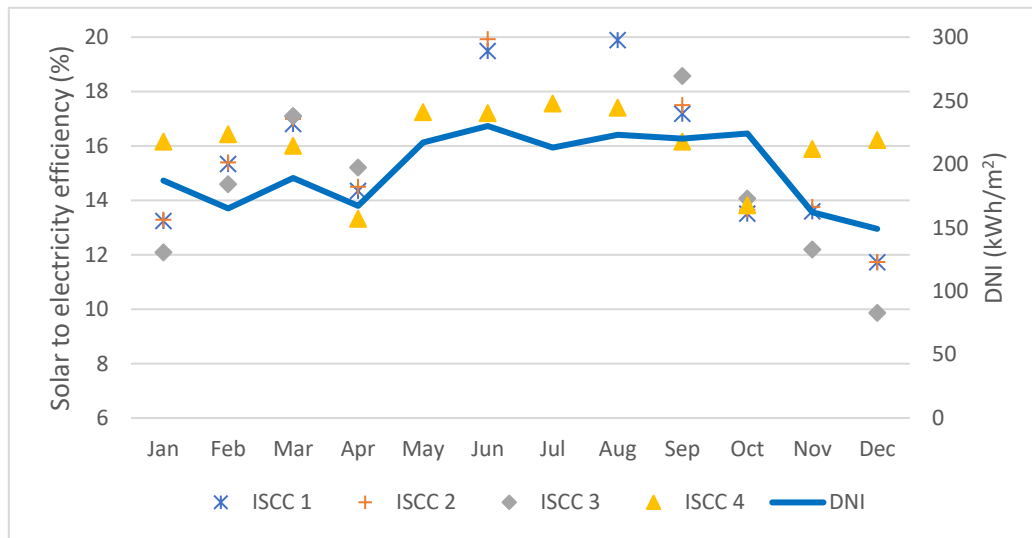


Figure 5-15 Saudi Arabian weather impact on monthly solar to electricity efficiency

5.7.2.4 Effect on inlet temperature of steam turbine

It has been highlighted that the inlet temperature of the steam entering the steam turbine also fluctuates during the operation of the integrated solar combined cycle using municipal solid waste under varying weather conditions. In Figure 5-16, the monthly differences between the studied cases fluctuate based on the effect of

Spanish city weather. From the results, we see that the use of an enhanced solar thermal contribution by adding more superheated steam into the hybrid system maintains high temperatures at the inlet of the steam turbine. ISCC 3 and ISCC 4 had an average temperature range between 515 °C and 520 °C, while ISCC 2 outcomes were below this range and dropped further in summer to around 505 °C. As with the Spanish case, the average temperature under Saudi Arabian weather conditions changed with varying solar radiation and ambient temperature, as illustrated in Figure 5-17. The estimated outcomes of the studied configurations showed that using a solar power tower and parabolic trough with molten salt helped to maintain high operating temperatures for the steam turbine. The superheated solar steam managed to boost the temperature overall stream to the range of 516 °C to 520 °C for ISCC 3 and 518 °C to 520 °C for ISCC 4, while the ISCC 2 outcomes had a lower range, namely 505 °C to 514 °C.

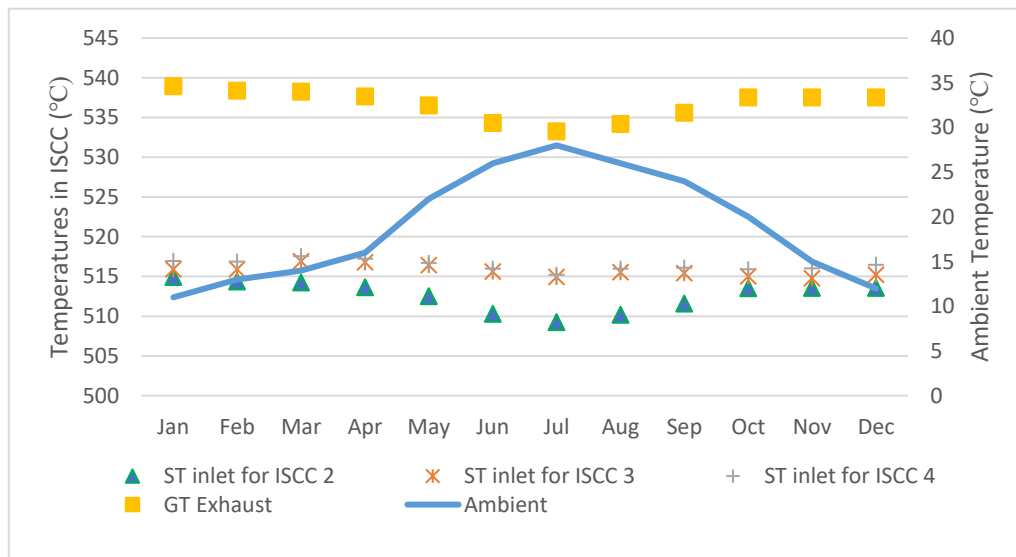


Figure 5-16 Spanish weather impact on ST inlet temperature

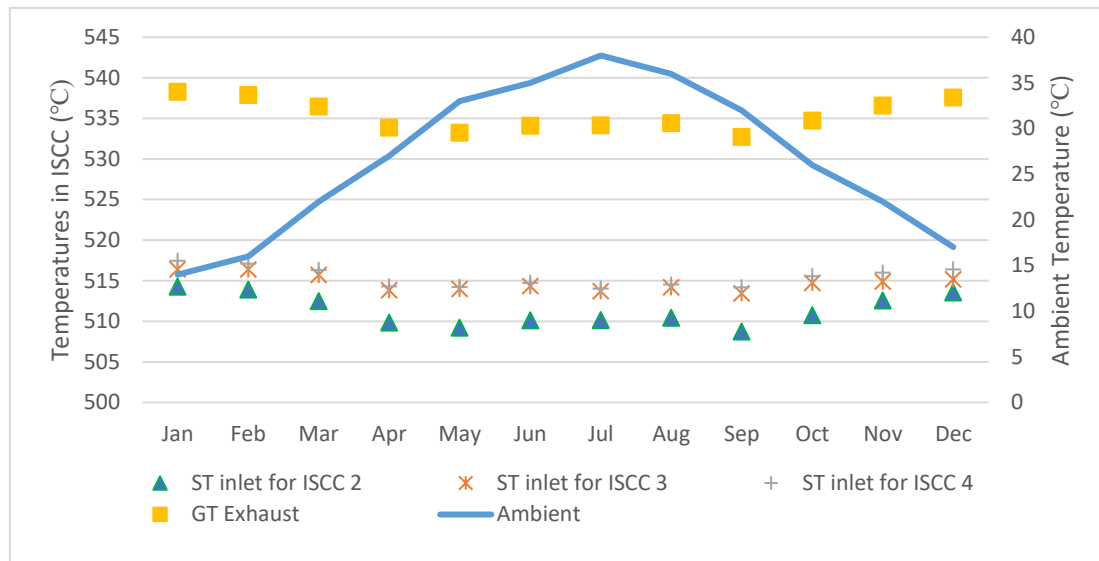


Figure 5-17 Saudi Arabian weather impact on ST inlet temperature

5.7.2.5 Total annual outcomes

For this step, all simulations related to the 8760 hours of variable weather conditions throughout the year were summed in order to observe the total impact on overall system performance in each case. As with design conditions, the ST inlet temperature was the first indication of the effect of fuel type and solar steam. The research considered an average temperature to represent the steam turbine inlet temperature during the solar hours of each month, which is shown in Table 5-8 and Table 5-9. ISCC 1 and ISCC 2 both depend only on fuel to maintain the design conditions of the steam turbine inlet. With ISCC 1, the use of natural gas can on average maintain a sufficient GT exhaust condition throughout the year to keep the steam turbine inlet around design conditions. When syngas is the main fuel to the gas turbine, as with ISCC 2, a lower inlet temperature into the steam turbine is expected due to low LHV, but the effect of ambient temperature can also be highlighted. However, the solar steam contribution in ISCC 3 and ISCC 4 can push the ST inlet closer to design conditions, in addition to reducing the effect of ambient temperature on the gas turbine.

The average inlet temperature over the whole year and the other indications used to observe the impact of the different configurations are also gathered in Table 5-8 and Table 5-9. With respect to power output, the gasifier cooling system was one of the reasons for the generation of more steam in the Rankine Cycle and managed to increase electricity production by more than 40 GWh annually when compared to ISCC 1. The other reason were the superheated solar steams from ISCC 3 and ISCC 4, which improved the conditions of the ST inlet stream in ISCC 2 and therefore boosted power generation annually by around 1 GWh in ISCC 3 and 13 GWh in ISCC 4. For solar thermal input, all configurations were designed to provide 66 MW_{th} solar energy, but the type of CSP technology nonetheless played a significant role in harnessing solar energy, which was observed in the power output values shown above. The use of a solar power tower with molten salt in ISCC 4 provided higher thermal energy input for the Rankine Cycle, which exceeded the parabolic trough values in other configurations, both with thermal oil and molten salt. This is because a significant amount of irradiation is concentrated at one receiver and therefore minimises heat losses in solar power tower, which resulted in converting more of the collected solar radiation into thermal energy than parabolic trough. In the case of parabolic trough layouts, the differences are related to the heat transfer fluid type and its temperature, which therefore reduces or increases the solar thermal input to the steam cycle.

Table 5-8 Annual outcomes for ISCC layouts based on Spanish weather

Annual energy	ISCC 1	ISCC 2	ISCC 3	ISCC 4
$\dot{W}_{ISCC}$ (GWh)	590.24	633.82	634.49	646.41
$\dot{W}_{ST(CSP)}$ (GWh)	39.69	40.01	40.69	52.61
$DNI * A$ (GWh)	292.56	292.56	288.92	343.36
$\dot{Q}_{CSP}$ (GWh)	115.79	115.79	112.59	147.10
$\dot{Q}_{Fuel}$ (GWh)	1061.91	1001.91	1001.91	1001.91
SS (%)	9.83	10.36	10.10	12.80
η_{ISCC} (%)	50.12	56.71	56.93	56.26

$\eta_{sol-rad-el}$ (%)	13.57	13.68	14.08	15.32
$\eta_{sol-th-el}$ (%)	34.28	34.56	36.14	35.76
η_{rad-th} (%)	39.58	39.58	38.97	42.84
$T_{ST(in)-Avg}$ (°C)	524	511	517	519

Table 5-9 Annual energy for ISCC layouts based on Saudi Arabia

Annual energy	ISCC 1	ISCC 2	ISCC 3	ISCC 4
$\dot{W}_{ISCC}$ (GWh)	572.04	632.06	632.26	643.96
$\dot{W}_{ST(CSP)}$ (GWh)	50.83	51.65	51.85	63.54
$DNI * A$ (GWh)	322.131	322.131	322.131	408.537
$\dot{Q}_{CSP}$ (GWh)	146.310	146.310	134.730	176.626
$\dot{Q}_{Fuel}$ (GWh)	1030.126	982.288	982.288	982.288
SS (%)	12.44	12.96	12.06	15.24
η_{ISCC} (%)	48.62	56.00	56.60	55.57
$\eta_{sol-rad-el}$ (%)	15.78	16.03	16.09	15.55
$\eta_{sol-th-el}$ (%)	34.74	35.30	38.48	35.98
η_{rad-th} (%)	45.42	45.42	41.82	43.23
$T_{ST(in)-Avg}$ (°C)	524	513	517	519

5.7.3 Conclusions to Hybrid System Performance

The annual performance evaluation of integrated solar combined cycle was conducted in this chapter. The motivation was to compare the novel concept to the other studied configurations which differed in the type of fuel, heat transfer fluid and solar field. This was accomplished by observing the weather impact from two locations, Al Ahsa in Saudi Arabia and Sevilla in Spain, on major technical parameters. Under Spanish weather, the novel concept reached highest annual total power output with 634.49 GWh for ISCC 3 and 646.41 GWh for ISCC 4 which had an average of 52.87 GWh and 53.87 GWh per month, respectively. On the other hand, ISCC 1 managed to reach 590.24 GWh with an average of

49.19 GWh per month while ISCC 2 produced 633.82 GWh with an average of around 54 GWh. Under Saudi Arabian weather, the total power output in a year was 632.26 GWh for ISCC 3 and 643.96 GWh for ISCC 4 which on average had 52.69 GWh and 53.66 GWh per month, respectively. For the other configurations, ISCC 1 managed to reach 572.04 GWh with average of 47.67 GWh per month while ISCC 2 produced 632.06 GWh with an average of around 52.67 GWh. The results of novel concept demonstrated how the gasification technology noticeably increased the power output when compared to ISCC 1. Finally, these outcomes also highlighted that increasing the temperature of steam mixture at turbine inlet improved the overall performance when compared to ISCC 2, especially, the solar contribution in electricity production.

6 Economic Performance

6.1 Introduction

The previous chapter evaluated overall annual performance in terms of the technical parameters of the hybrid system. In this chapter, the focus is to examine the economic performance of integrated solar combined cycle operating in different forms, which represent the conventional and proposed configurations. The purpose is to determine whether the hybrid system suggested by this research has an economic advantage over the typical ISCC with natural gas. According to Adrian et al. [216], one of the most influential factors to consider when choosing one thermal system design option over another is the cost of the system's final products. In fact, the cost of electricity is considered here as the main final product to analyse, since the system is a power generation technology with various design options. As in any project, the final cost of an integrated solar combined cycle is affected by its input costs, such as the capital costs of the hybrid system components, working fluid, fuel, in addition to the operating and maintenance cost.

6.2 Total Capital Cost

One of the key economic indicators was the estimation of the total investment or capital cost of the integrated solar combined cycle in each form studied. The reason for considering this indicator was related to its importance in highlighting the level of investment when constructing any power plant project. Hence, the total capital cost of this hybrid system can be calculated using the purchase cost for each component within the hybrid system. This can be accomplished using a price list from the vendors for each major piece of equipment or using cost estimations. The former tends to be more accurate but also more complex since it is based on the current market prices while the latter is more suited for overall indication of system cost, which can be tailored to the research conditions. That said, to obtain the required costs with regards to the various components in the studied hybrid system, we used cost functions, which are available in the literature. The motive for using these functions was that they related the technical

design parameters of a system to the expenses associated with its purchase and installation. The following sections highlight the major components considered and the implemented cost functions to estimate the investment for the studied integrated solar combined cycle. The component costs of the integrated solar combined cycle studied were predicted through the use of a variety of cost functions and specific costs for the Brayton and Rankine Cycles, gasifier, fuel (natural gas and municipal solid waste) and solar field (parabolic trough, solar power tower and type of heat transfer fluid). [217–221]

6.2.1 Cost Functions for the Gas Turbine Cycle

Starting with the Brayton cycle or gas turbine cycle, the applied cost functions can be focused on predicting the components costs which are the air compressor $Z_{Compressor}(\text{\$})$, combustion chamber $Z_{Combustion\ Chamber}(\text{\$})$, gas turbine $Z_{GT}(\text{\$})$ in order to obtain the total cycle cost of $Z_{GTC}(\text{\$})$ and the operational and maintenance cost of the gas turbine cycle $Z_{O\&M(GTC)}(\text{\$})$, which are shown in the following equations [68,74,217]:

$$Z_{Compressor} = \frac{75 * \dot{m}_{air}}{0.9 - \eta_{Compressor}} * \left(\frac{P_2}{P_1}\right) * \ln\left(\frac{P_2}{P_1}\right) \quad (6-1)$$

$$Z_{Combustion\ Chamber} = \left(\frac{48.64 * \dot{m}_{gas}}{0.995 - \frac{P_3}{P_2}}\right) * (1 + \exp(0.018 * T_3 + 26.4)) \quad (6-2)$$

$$Z_{GT} = \frac{1536 * \dot{m}_{gas}}{0.92 - \eta_{GT}} * \ln\left(\frac{P_4}{P_3}\right) * (1 + \exp(0.036 * T_3 - 54.4)) \quad (6-3)$$

$$Z_{GTC} = Z_{Compressor} + Z_{Combustion\ Chamber} + Z_{GT} \quad (6-4)$$

$$Z_{O\&M(GTC)} = 0.05 * Z_{GTC} \quad (6-5)$$

where $\dot{m}_{air}$ is the amount of air entering the compressor (kg/s), $\dot{m}_{gas}$ is the mass flow rate of the air and burned fuel mixture (kg/s), P_1 – P_4 and T_1 – T_4 are the pressure (bar) and temperature (K) at each stage in the cycle, respectively,

$\eta_{Compressor}$ is the isentropic compressor efficiency, η_{GT} is the isentropic gas turbine efficiency and Z_{GTC} (\$) is the total capital cost of the gas turbine cycle.

6.2.2 Cost Functions for the Steam Turbine Cycle

For the Rankine or steam turbine cycle, the cost functions for components included the cost of the heat recovery steam generator Z_{HRSG} (\$), steam turbine Z_{ST} (\$), condenser Z_{COND} (\$), pump Z_{Pump} (\$) leading to total cycle cost Z_{STC} (\$) and operational costs and maintenance of the steam turbine cycle $Z_{O\&M(STC)}$ (\$), as illustrated below [68,74,217,222]. For the heat recovery steam generator:

$$Z_{HRSG} = 4131.8 * \Sigma \left(f_{P_i} * f_{steam_i} * f_{gas_i} * \left(\frac{\dot{Q}_{HRSG_i}}{LMTD_{HRSG_i}} \right)^{0.8} \right) + 13380 \quad (6-6)$$

$$* \Sigma (f_{P_i} * \dot{m}_{steam_i}) + 1489.7 * \dot{m}_{gas}^{1.2}$$

$$f_{P_i} = 0.0971 * \frac{P_i}{30bar} + 0.9029 \quad (6-7)$$

$$f_{steam_i} = 1 + \exp \left(\frac{T_{steam_i} - 830K}{500K} \right) \quad (6-8)$$

$$f_{gas_i} = 1 + \exp \left(\frac{T_{gas_i} - 990K}{500K} \right) \quad (6-9)$$

$$LMTD = \frac{((T_{hot_{in}} - T_{cold_{out}}) - (T_{hot_{out}} - T_{cold_{in}}))}{\ln \frac{(T_{hot_{in}} - T_{cold_{out}})}{(T_{hot_{out}} - T_{cold_{in}})}} \quad (6-10)$$

$$Z_{ST} = 3880.5 * \dot{W}_{ST}^{0.7} * \left[1 + 5 * \exp \left(\frac{T_{ST(in)} * 866}{10.42} \right) \right] * \left[1 + \left(\frac{0.05}{1 - \eta_{ST}} \right)^3 \right] \quad (6-11)$$

$$Z_{COND} = 280.74 * \left(\frac{\dot{Q}_{COND}}{2.2 * LMTD_{COND}} \right) * 746 * \dot{m}_{COND} \quad (6-12)$$

$$Z_{Pump} = 708.48 * \dot{W}_{Pump}^{0.7} * \left(1 + \frac{0.2}{1 - \eta_{Pump}} \right) \quad (6-13)$$

$$Z_{STC} = Z_{HRSG} + Z_{ST} + Z_{COND} + Z_{Pump} \quad (6-14)$$

$$Z_{O\&M(STC)} = 0.02 * Z_{STC} \quad (6-15)$$

Where f_{P_i} , f_{steam_i} , f_{gas_i} , $\dot{Q}_{HRSG_i}$ and $LMTD_{HRSG_i}$ are the pressure fraction, steam temperature fraction, gas temperature fraction, heat transfer or thermal power (kW) and logarithmic mean temperature difference (K), respectively, of each stage in HRSG heat exchangers (superheaters, evaporators and economisers). $T_{hot_{in}}$ is the temperature of hot fluid entering the heat exchanger (K) which in this case is the gas turbine exhaust within HRSG and $T_{hot_{out}}$ is the temperature of cooled exhaust (K). Furthermore, $T_{cold_{in}}$ is the temperature of cold fluid entering the heat exchanger (K) which is in this case the feedwater in HRSG and $T_{cold_{out}}$ is the heated steam (K) exiting this component in a counter-current flow to gas turbine exhaust.

For $\dot{m}_{COND}$, it is the amount of the fluid used in the condenser to condense the steam to water (kg/s), $T_{ST(in)}$ is the temperature inlet of the steam turbine (K), and $LMTD$ is the log-mean temperature difference. $\dot{W}_{ST}$ and $\dot{W}_{pump}$ are the power produced or consumed from the steam turbine or pump (kW), $\dot{Q}_{HRSG}$ and $\dot{Q}_{COND}$ on the other hand are the thermal power generated or consumed by the heat recovery steam generator and the condenser. Finally, η_{ST} is the steam turbine isentropic efficiency (%) and Z_{STC} the total capital cost of the steam turbine cycle (\$).

6.2.3 Cost Functions for Fuel and Renewable Energy Components

Since the research has evaluated the solar field as a black-box model, the main components to consider in the solar cost analysis are the cost of the required aperture area $Z_{Aap (PT-TO)}$, $Z_{Aap (PT-MS)}$ and $Z_{Aap (SPT-MS)}$ (\$) and cost of the applied heat transfer fluid $Z_{HTF(TO)}$ and $Z_{HTF(MS)}$ (\$) in order to obtain the CSP capital cost Z_{CSP} for each studied solar field and then estimate the operation and maintenance $Z_{O\&M(PT)}$ and $Z_{O\&M(SPT)}$ (\$), which were calculated through the following cost functions [223–230]

$$Z_{Aap (PT-TO)} = 150 * A \quad (6-16)$$

$$Z_{A_{ap}(PT-MS)} = 180 * A \quad (6-17)$$

$$Z_{A_{ap}(SPT-MS)} = 140 * A \quad (6-18)$$

$$Z_{HTF(TO)} = 2.2 * \dot{m}_{HTF(TO)} \quad (6-19)$$

$$Z_{HTF(MS)} = 1.3 * \dot{m}_{HTF(MS)} \quad (6-20)$$

$$Z_{CSP} = Z_{A_{ap}} + Z_{HTF} \quad (6-21)$$

$$Z_{O\&M(PT)} = 0.022 * Z_{CSP(PT)} \quad (6-22)$$

$$Z_{O\&M(SPT)} = 0.023 * Z_{CSP(SPT)} \quad (6-23)$$

where the A is the solar field aperture area (m^2), $\dot{m}_{HTF}$ is the required mass flow rate of heat transfer fluid to produce thermal power (kg/s) and $Z_{CSP(PT)}$ and $Z_{CSP(SPT)}$ are the total capital cost of the parabolic trough and solar power tower, respectively. With regards to the gasifier costs, the gasification reactor $Z_{Gasifier}$ (\$), syngas cleaning system $Z_{Syngas\ Cleaning}$ (\$) and the operation and maintenance $Z_{O\&M(Gasifier\ Unit)}$ (\$) were the main parameters considered for the gasification technology and are given by the following equations [68,231–233]:

$$Z_{Gasifier} = 1600 * (\dot{m}_{dry\ feedstock})^{0.67} \quad (6-24)$$

$$Z_{Syngas\ Cleaning} = 0.05 * Z_{Gasifier} \quad (6-25)$$

$$Z_{Gasifier\ Unit} = Z_{Gasifier} + Z_{Syngas\ Cleaning} \quad (6-26)$$

$$Z_{O\&M(Gasifier\ Unit)} = 0.03 * Z_{Gasifier\ Unit} \quad (6-27)$$

Here $\dot{m}_{dry\ feedstock}$ (kg/h) is the amount of feedstock used in the gasifier to produce syngas and run the gas turbine and $Z_{Gasifier\ Unit}$ (\$) is the total capital cost of the gasification technology. With respect to the fuel costs, the municipal solid waste and natural gas purchase cost was the main parameter considered in the cost calculations. Hence, other parameters such as transportation were not included, since the research assumption was based on sufficient fuel availability

at the power plant. In the case of fuel cost, it was obtained based on a purchase cost which was assumed to be 7.6 (\$/GJ) for natural gas [234] and 3 (\$/GJ) for municipal solid waste using the following equation:

$$Z_{fuel} = F * \dot{Q}_{fuel} \quad (6-28)$$

Here Z_{fuel} is the annual cost of either natural gas or municipal solid waste (\$), F is the purchase cost of fuel (\$/kWh), $\dot{Q}_{fuel}$ is the fuel annual thermal power (kWh).

6.3 Levelised Cost of Electricity

Hence, the economic analysis in this chapter is based on obtaining the levelised cost of electricity (LCOE), since this is the most common economic indicator when comparing different power generation technologies [235]. The LCOE can be calculated by Equation (6-29) [68]:

$$LCOE = \frac{CRF * Z_{Total\ Capital} + Z_{Total\ O\&M} + Z_{fuel}}{\dot{W}_{Annual-Net}} \quad (6-29)$$

$Z_{Total\ Capital}$ is the total capital cost of the plant (\$) which sums all capital costs of the components, CRF is the capital recovery factor, $Z_{Total\ O\&M}$ represents annual operating and maintenance costs (\$) (a portion of the capital cost), and $\dot{W}_{Annual-Net}$ is the annual net power generation by the power plant (MWh). For the capital recovery factor, the following equation is used, with i_r for interest rate and n for number of operational years:

$$CRF = \frac{i_r * (1+i_r)^n}{(1+i_r)^n - 1} \quad (6-30)$$

In order to determine total capital or investment cost, in addition to the operation and maintenance cost of the studied system, it is necessary to have approximate values of the component costs in this power plant. This can be achieved using either cost functions or specific costs from the literature, which have been expressed here in US dollars (\$).

6.4 Results

Based on the cost functions of the various components mentioned earlier, the total capital cost of the integrated solar combined cycle for each case was calculated as shown in Table 6-1. This was accomplished using a variety of assumed parameters as well as technical outcomes obtained for the different components in the hybrid system, which were influenced by the weather conditions at Spain and Saudi Arabia. The details of these calculations are shown in Appendix D.

Table 6-1 Costs related to ISCC configurations

Costs (\$)	ISCC 1	ISCC 2	ISCC 3	ISCC 4
Compressor	5,315,760	5,315,760	5,315,760	5,315,760
Combustion chamber	317,612	317,612	317,612	317,612
GT Turbine	9,131,494	9,131,494	9,131,494	9,131,494
HRSO	5,232,833	5,232,833	5,232,833	5,232,833
ST Turbine (Spain)	5,312,292	5,407,293	5,445,315	5,643,563
ST Turbine (Saudi Arabia)	5,443,784	5,548,498	5,573,025	5,775,811
Condenser	1,011,865	1,011,865	1,011,865	1,011,865
HP pump	67,782	67,782	67,782	67,782
LP pump	20,962	20,962	20,962	20,962
Aperture area	20,601,000	20,601,000	24,721,200	24,385,060
Heat transfer fluid	2,129,688	2,129,688	815,256	820,872
Gasifier	-	2,785,358	2,785,358	2,785,358
Syngas cleaning unit	-	139,268	139,268	139,268
$Z_{fuel(Spain)}$	29,053,774	10,820,637	10,820,637	10,820,637
$Z_{fuel(Saudi Arabia)}$	28,184,247	10,619,285	10,619,285	10,619,285
$Z_{Total Capital(Spain)}$	49,141,288	52,160,915	55,004,705	54,872,429
$Z_{Total O\&M(Spain)}$	1,471,233	1,560,872	1,623,359	1,645,258
$Z_{Total Capital(Saudi Arabia)}$	49,272,780	52,302,120	55,132,415	55,004,677
$Z_{Total O\&M(Saudi Arabia)}$	1,473,863	1,563,696	1,625,913	1,647,903
$LCOE_{(Spain)} (\\$/MWh)$	60.87	28.59	29.15	28.62
$LCOE_{(Saudi Arabia)} (\\$/MWh)$	61.32	28.38	28.96	28.45

With regard to the steam turbine cycle, the research used a similar assumption to the gas turbine cycle by keeping most of the component operating conditions based on the conventional ISCC configurations in both countries, except for the steam turbine. The motive was to highlight potential capacity and inlet temperature for this component, since it is strongly influenced by the solar intensity of the countries considered, in addition to steam generation enhancement by the syngas cleaning system. Hence, the average of the annual power output and its temperature from the steam turbine model simulation were selected in order to choose a potential size and the corresponding expected inlet conditions for each ISCC configuration studied. Based on that, the cost function calculations of the steam turbine cycle were accomplished, and it can be seen that the higher production of steam under Saudi Arabian weather conditions increased the capacity and costs of the STC turbine component. Hence, the capital cost of the steam turbine cycle to reach \$11.78 million for ISCC 1, \$11.88 million for ISCC 2, \$11.91 million for ISCC 3 and \$12.11 million for ISCC 4, while the Spanish outcomes were \$11.65 million, \$11.74 million, \$11.78 million and \$11.98 million, in the same order. The major input variables were the $LMTD_{HRSG}$ and $LMTD_{COND}$, air condenser flow rate, the amount of generated steam in the HRSG, exhaust gas flow rate in the HRSG, heat used to generate steam, heat loss from condensation and isentropic efficiency of the turbine. It is worth noting that $LMTD_{HRSG}$ was based on the inlet and outlet temperatures of two superheaters, two evaporators and two economisers, within the boiler model simulation. For the HP steam, the temperature profile rose from 166 °C, 280 °C, 306 °C to 524 °C for the economiser inlet, economiser outlet, evaporator outlet and superheater outlet, respectively. The LP steam followed similar a curve to HP steam, with 56 °C at the economiser inlet, 164 °C at the economiser outlet, 190 °C at the evaporator and finally 238 °C at the superheater. The exhaust gas temperature profile started with 550 °C at the HP superheater inlet, which dropped to 491 °C at the HP superheater outlet, 338 °C at the HP evaporator outlet and 288 °C at the HP economiser outlet, then reached 280 °C at the LP superheater outlet, 221 °C at the LP evaporator outlet until finally exiting the economiser at 163 °C.

In the case of the fuel and renewable energy components, some of the input values were similar between the two countries but differed in the configuration type, while other inputs were different in both locations and the studied configuration. With the solar field, the design thermal power for the parabolic trough using oil, the parabolic trough using molten salt and the solar power tower using molten salt were the same for Saudi Arabia and Spain. The motive for these shared design inputs was to highlight the performance of the solar field types at different locations and weather conditions using the model simulation data. Based on this, the cost functions calculations were carried out and by adding all solar costs, the lowest CSP capital costs were obtained by ISCC 1 and ISCC 2, with approximately \$22.73 million, followed by ISCC 4 with \$25.21 million and then ISCC 3 with \$25.54 million. For the gasification technology, the applied cost function was mainly affected by one major variable, the amount of waste consumption in the power generation. Taking municipal solid waste consumption at design conditions within the hybrid system, which was essential in sizing the gasification reactor, meant that both countries had the same reactor size. Hence, the gasifier unit which includes both the reactor, and its syngas cleaning system needed an investment of around \$2.92 million.

Lastly, the fuel cost was obtained by using annual fuel consumption from Table 5-8 and Table 5-9, which showed that the ambient temperature of Saudi Arabia had more impact on the fuel thermal input than in Spain due to greater drop of inlet air density, thus reducing the air mass flow into the turbine.

By combining all the individual costs of the various components, the total capital and total operation and maintenance costs of the integrated solar combined cycle in each case were estimated, which helped in the calculation of levelised cost of electricity. These calculations were conducted based on the assumption that the interest rate was 10%, the number of years of operation was 25, and the capital recovery factor was 0.11. In Table 6-1, the outcomes of the different configurations followed the same trend in both Saudi Arabia and Spain. The conventional configuration of the hybrid system, ISCC 1, had the lowest total capital cost for its components but also required the highest cost to produce

electricity. This very high LCOE cost was attributed to the cost of natural gas in comparison to municipal solid waste. For this reason, the hybrid system concept proposed in this research, and more specifically the ISCC 4 configuration, offered the best outcomes in terms of the levelised cost of electricity when compared only to the conventional configuration, ISCC 1. However, the lowest LCOE was obtained by replacing natural gas with municipal solid waste while maintaining solar field role to generate saturated steam using parabolic trough and thermal oil.

6.5 Conclusions to Economic Performance

The economic performances of the studied ISCCs were evaluated in this chapter. The motivation was to compare the levelised cost of electricity of the novel concept to the other studied configurations. This included the use of components cost functions in order to estimate the overall capital cost for the hybrid system. Based on the technical performance variations caused by the applied weather data, Al Ahsa in Saudi Arabia and Sevilla in Spain, each location showed different economic outcomes on major technical parameters. The novel concept reached capital cost of \$55 million for ISCC 3 and \$54.9 million for ISCC 4 under Spanish weather. In the other configurations, it was \$49.1 million for ISCC 1 and \$52.2 million for ISCC 2. The use of gasification technology, different CSP and HTF types in addition to enhanced solar thermal power increased the size of steam turbine have led the novel concept to have higher capital cost than ISCC 1. Furthermore, the use of parabolic trough with molten salt in the novel concept, ISCC 3, demonstrated the need for larger investment cost than ISCC 2 even though the cost of thermal oil is higher than molten salt. This was attributed to the additional costs to maintain the flow of molten salt in the solar field under its recommended operating conditions.

Similar trend was observed under Saudi Arabian weather showing the MSW based configurations (ISCC2, ISCC3 and ISCC4) having higher capital costs than conventional natural gas-based configuration. ISCC2 offered \$52.3 million, ISCC 3 reached \$55.13 million and ISCC 4 required \$55 million while ISCC 1 capital cost was \$49.3 million. These results of capital cost served as one of the

variables in the levelised cost of electricity function which considered the amount of power generation and fuel cost. Since the main focus is comparing the novel concept to conventional concept, it was seen that ISCC3 and ISCC4 both offered better LCOE in both Spain and Saudi Arabia than ISCC 1 due to the high cost of natural gas. Finally, the overall economic evaluation indicated ISCC 2 provided slightly lower levelised costs of electricity than the novel concept in this thesis due to the use of mature parabolic trough solar field with thermal oil.

7 Overall Discussion

7.1 Introduction

The previous chapters were focused on obtaining accurate estimations of the overall annual performance, technically and economically, of the integrated solar combined cycle, based on various defining conditions, which involved this hybrid system at its conventional form with natural gas, its renewable form with municipal solid waste and the renewable form proposed in this thesis, with enhanced thermal power from the solar field. These simulations and calculations provided necessary information about the behaviour of the hybrid system under fluctuating solar radiation and weather conditions in two different locations, Spain and Saudi Arabia. The use of municipal solid waste gasification and enhanced solar thermal power noticeably altered two aspects in this hybrid system, the level of steam generation and the expenditure related to fuel replacement and solar field modification calculations, thus highlighting the impact of off-design as well as design operating conditions. The simulation cases considered have provided a great deal of information about the behaviour of the different hybrid system configurations, thus meeting all stated aims and objectives. In this study, a general indication was presented of the potential benefits of deploying the proposed integrated solar combined cycle in the two desired locations within the scope of the research. Nevertheless, the study primarily focused on the technical and economic perspectives in order to illustrate the features of the novel hybrid system relative to its conventional layout. In this chapter, the main results of the outcomes of the previous chapters will be discussed, based on their relation to the research questions, aim and objectives in order to decide whether the proposed configuration is beneficial.

7.2 The Performance of the ISCC Components

In Chapter 4, the performance of the solar field and municipal solid waste gasification were investigated to find their optimum values as individual components, in order to later assess the impact on the integrated solar combined cycle studied, which led to the first steps in achieving Objective 2 and Objective

3 of this thesis in Section 2.5. With respect to the solar field, the three major cases, a parabolic trough with thermal oil, a parabolic trough with molten salt and a solar power tower with molten salt, were affected differently by the variable design parameters, namely the direct normal irradiance and the hot temperature of heat transfer fluid. Furthermore, the amount of solar thermal power produced by the solar field was the main criterion to evaluate and select the optimal values for these two parameters.

For the assumed range of direct normal irradiance (700 W/m^2 to 1000 W/m^2), as presented in Section 4.6.1.1, it was established that having higher design point values in general was attractive because it would provide the required thermal power with a very small aperture area. Nevertheless, these high design point values illustrated a noticeable weakness in terms of the thermal contribution offered when the solar field incorporated the expected solar radiation at the studied locations in Spain and Saudi Arabia, which received a smaller amount of direct normal irradiance on typical summer days. As such, lower design point DNIs were preferable for these conditions. However, implied a need for larger aperture areas for the studied solar fields, leading to a significant rise in the overall cost due to purchasing and using more land and mirrors. However, the main advantage of these extra requirements was the obvious improvement in solar field performance. Applying lower design point values permitted increasing the solar thermal contribution to the hybrid system considered in this research by around 38.67%, 32.94% and 29.36% for the parabolic trough with thermal oil (ISCC 1 and ISCC 2), parabolic trough with molten salt (ISCC 3) and solar power tower (ISCC 4), respectively, under Spanish summer conditions, while in Saudi Arabia, the solar thermal contribution was enhanced by approximately 38.05%, 37.86% and 40.53%, in the same order. Therefore, the expected impact from this thermal enhancement was a better steam generation process and a concomitant highly useful rise in electricity production from the solar energy.

For the assumed range of working fluid outlet temperatures ($550 \text{ }^\circ\text{C}$ to $580 \text{ }^\circ\text{C}$) for the molten salt as shown in Section 4.6.1.2, it was demonstrated that utilising higher temperature values in general was quite attractive because it would offer

the required thermal power with a stronger capability to boost the steam generation to larger superheating levels. However, reaching higher temperature values presented a clear penalty with respect to the total amount of produced thermal power when the solar field operated under the intensity of solar radiation and weather conditions present during summer in Spain and Saudi Arabia. For this reason, applying lower values of the molten salt outlet temperature was still capable to maintaining the quality of the solar superheated steam without losing the crucial thermal contribution, to boost the overall stream and power output in the hybrid system of this research. For example, the sensitivity analysis showed that a heat transfer fluid at 550 °C offered a slight rise in total daily thermal power by around 0.8% and 2.4% for the solar power tower with molten salt (ISCC 4) and the parabolic trough with molten salt (ISCC 3), respectively, under Spanish summer conditions, while in Saudi Arabia, the solar thermal contribution was moderately increased by around 1.23% and 2.69%.

With regards to the gasification process for municipal solid waste, all three major cases, which differed in the solar field design and operating parameters, implemented the same input feedstock and accompanying variable parameters. For the assumed conditions of the physical and chemical composition of the collected municipal solid waste, as highlighted in Section 4.6.2, it was established that the effects of varying feedstock composition were neglected, and the composition was therefore assumed to be the same in all cases in Spain and Saudi Arabia for both design and off-design conditions. Nevertheless, this assumption helped demonstrate the impact of municipal solid waste versus natural gas under what can be described as ideal conditions of fixing the feedstock composition. Hence, the selected optimum operating parameters were fixed instead of selecting unlimited optimum values for each collected waste. It is worth noting that the applied LHV of 6 MJ/kg can be seen as representing the feedstock energy content at one of the weakest cases since the typical range can reach 17 MJ/kg [111]. The variables considered were the moisture content in the feedstock composition, the amount of air applied in gasification, and the temperature of the gasifier. Moreover, the selection of the optimal gasification

values was based on the lower heating value of the produced syngas, in addition to its composition.

For the assumed range of moisture contents (10% to 14.3%) as presented in Section 4.6.2.1, it was established that further drying of the collected municipal solid waste generally was quite attractive because it would improve the quality of the produced syngas by having higher lower heating values. Applying lower values of the moisture content increased the fuel energy content by around 20% while maintaining lower concentrations of carbon dioxide and methane in the syngas. Although further moisture content reduction could have been beneficial, it might require longer hours of natural drying or the need for a drying reactor for municipal solid waste preparation before entering the gasifier. With respect to the amount of air entering the gasifier, it was described as an air-to-fuel ratio (0.5 to 0.8) as illustrated in Section 4.6.2.2. The outcomes showed how utilising lower amounts of air in the gasification process for the assumed composition of the municipal solid waste was able to boost the quality of the produced syngas with noticeable effect on the concentrations of carbon dioxide and methane in the syngas. Finally, the variation of the temperature (600 °C to 1000 °C) as presented in Section 4.6.2.3 gave the clearest changes in the produced syngas composition. It was established that operating the gasification process at higher temperatures offered a better lower heating value with a reasonable syngas composition. It is worth noting that the focus on each gas concentration within the product of municipal solid waste gasification under the selected optimum parameters was strongly related to the quality of the produced syngas in terms of its impact on the environment as well as its capability to replace the natural gas based on this research assumptions.

7.3 The Performance of the Studied Hybrid System

In Chapters 5 and 6, the technical and economic performances of the different studied configurations of the integrated solar combined cycle were investigated under the varying weather conditions of Spain and Saudi Arabia based on the selected optimum parameters of the solar field and municipal solid waste gasification. These two chapters completed the necessary steps in achieving

objective 2 and objective 3 as well as objective 4 of this thesis in Section 2.5. With respects to the design technical performance, there were obvious differences between the studied configurations, ISCC 1, ISCC 2, ISCC 3 and ISCC 4, based on the input operating conditions. At the design conditions and off-design conditions, where all cases of the hybrid system operated for one hour and 8760 hours, respectively, the simulation results indicated several differences worth noting. With regard to the implemented fuel, it was found that the results were strongly in favour of the cases which used municipal solid waste gasification (ISCC 2, ISCC 3 and ISCC 4), in terms of the level of generated power offered due to the inclusion of a syngas cooling stage in the generation process of the low-pressure steam. Using this type of integration between the gasification technology and the steam turbine cycle can be considered very useful in reducing feedstock consumption, since there was no need to add more thermal input through the gas turbine cycle to accomplish those high electricity outcomes. For this reason, in addition to MSW having lower purchase cost than natural gas in general, it was observed that this gasifier integration feature in ISCC 2, ISCC 3 and ISCC 4 can be considered a major cause of the significant decrease in the annual fuel cost when compared to ISCC 1. In the municipal solid waste-based configurations, the annual fuel cost was \$10.8 million for Spain and \$10.6 million for Saudi Arabia while in the natural gas-based configuration, the annual fuel cost was \$29 million for Spain and \$28 million for Saudi Arabia.

With regard to the solar field conditions, it was observed that the results were attractive in terms of the power generation boost the cases where molten salt was implemented (ISCC 3 and ISCC 4) when compared to the waste-based configuration (ISCC 2) that applied thermal oil. This was accomplished, as stated before, using solar superheated steam generation with molten salt which harnessed solar radiation at higher temperatures than thermal oil. By doing so, this solar superheated steam was mixed with HRSG superheated steam causing overall steam temperature to rise at the steam turbine inlet. This clearly makes the use of solar superheated steam very desirable when the level of electricity production is the only technical indicator. However, these high solar thermal powers did not always show better outcomes in other technical aspects. Having

said that, it was noted that thermal oil-based configurations (ISCC 1 and ISCC 2) had better performance in the solar field conversion capability from solar radiation into thermal for both Spain and Saudi Arabia. Furthermore, the economic aspect was also crucial in deciding whether a configuration was desirable or not. From a component capital cost perspective, the natural gas-based configuration (ISCC 1) was the most attractive option. This due to no additional costs from the gasification technology, larger steam turbine size or solar field extra components costs. Nevertheless, the consideration of the electricity production cost, the levelised cost of electricity, was able to demonstrate how each configuration would perform over the years. As was seen earlier, the use of novel concept (ISCC3 and ISCC4) offered the better levelised cost of electricity values than the conventional concept (ISCC1). In Spain, ISCC 3 was able to provide LCOE of 29.15 \$/MWh and ISCC 4 offered 28.62 \$/MWh while ISCC1 value reached 60.87 \$/MWh. In Saudi Arabia, LCOE obtained by ISCC 3 was 28.96 \$/MWh, by ISCC4 was 28.45 \$/MWh and by ISCC 1 was 61.32 \$/MWh. For ISCC 2 in both countries, the LCOE values were slightly lower than the novel concept indicating that the replacement of fuel alone did not show a significant impact against the novel concept in terms of electricity production cost.

8 Conclusions and Recommendations

This concluding chapter provides a summary of the whole thesis and evaluates the research outcomes, responding to the original aim and objectives outlined in Section 2.5. In addition, the original contributions to knowledge that this thesis provides to the fields of renewable and solar thermal energy, are highlighted with reference to earlier studies. The extent to which the overall thesis aim has been achieved is assessed and final remarks are made on the recommendations for further work.

8.1 Conclusions

In this study, a novel concept was proposed for the configuration of an integrated solar combined cycle system. Hence, the research aimed to investigate the annual performance of this novel ISCC that uses municipal solid waste and an enhanced solar thermal contribution in order to evaluate the capability of this hybrid system to function as a 100% renewable power plant. In order to achieve this aim, the following objectives have been established:

1. To propose a possible configuration and select suitable locations for two case studies that satisfy the novel ISCC conditions by specifying the integration points of the renewable components and their main operating parameters within the hybrid system.
2. To analyse the impact of the suggested solar integration points on ISCC in order to evaluate the capability and overall benefits of using an enhanced solar thermal contribution as a booster for the operating conditions of the generated steam.
3. To assess the influence of MSW gasification on the hybrid system to highlight the main technical parameters affected by replacing natural gas.
4. To evaluate the annual performance of a novel ISCC by considering both its technical and economic aspects, for the purpose of illustrating its potential benefits for the power generation market. This was accomplished through a comparative analysis between the proposed configuration and other

configurations in order to decide whether the novel system was practical based on technical and economic assessments.

The research overall methodology included using different configurations under optimum conditions for its components. These ISCC configurations differ in one or more of the following features: type of fuel, type of heat transfer fluid or type of solar field. There were four configurations in this research, termed ISCC 1, ISCC 2, ISCC 3 and ISCC 4, of which the last two configurations represented the research novel concept. Due to the crucial impact of weather conditions on this hybrid systems, Spain and Saudi Arabia were considered as two case studies to observe whether the outcomes of the proposed configuration at two different locations were still attractive in comparison to the other studied configurations

The ISCC 1 configuration was fuelled by natural gas and used a solar steam generator to extract a fraction of the preheated water from the high-pressure level economiser of the heat recovery steam generator, to then be evaporated and returned to the high-pressure drum of the heat recovery steam generator as saturated steam using a parabolic trough with thermal oil in the temperature range of 292 °C to 392 °C. The solar steam was then injected into the HRSG to mix with saturated steam generated in the boiler to send the saturated steam mixture to the HRSG superheater. In this hybrid system, the natural gas offered thermal input near design operating conditions, letting the exhaust gas keep steam at high temperature at 524 °C for the steam turbine inlet, which then resulted in annual electricity production up to 590.24 GWh with solar share of 9.83% under Spanish weather while it was 572.04 GWh with a solar share of 12.44% under Saudi Arabian conditions. Moreover, the accomplished levelised costs of electricity based on the technical outcomes of this configuration, were 60.87 \$/MWh and 61.32 \$/MWh for Spain and Saudi Arabia, respectively, which are within the reported range in literature for integrated solar combined cycle, 20 \$/MWh to 180 \$/MWh [46].

The ISCC 2 configuration was fuelled by municipal solid waste through the gasification process and used the solar steam generator to extract a fraction of the preheated water from the high-pressure level economiser of the heat recovery

steam generator, to later be evaporated and returned to the high-pressure drum of the heat recovery steam generator as saturated steam using a parabolic trough with thermal oil in the temperature range of 292 °C to 392 °C. The solar steam was then mixed with the saturated steam with the boiler's saturated steam before entering the superheater. In this hybrid system, the municipal solid waste was not able to fully offer thermal input near design operating conditions, especially the exhaust gas which had lower temperature; therefore, the steam temperature at the steam turbine inlet dropped as did this turbine's power output. However, the water used in the gasification technology to cool the syngas was then sent back to the Rankine cycle as additional steam, which resulted in annual electricity production reaching 633.82 GWh with a solar share of 10.36% under Spanish weather while it was 632.06 GWh with a solar share of 12.96% under the Saudi Arabian conditions. Furthermore, the levelised cost of electricity achieved, based on this configuration's technical outcomes, was 28.59 \$/MWh for Spain and 28.38 \$/MWh for Saudi Arabia.

The ISCC 3 configuration was fuelled by municipal solid waste through a gasification process and used the solar steam generator to extract a fraction of the preheated water from the high-pressure level economiser of the heat recovery steam generator, to be evaporated, superheated. After that, the solar steam was mixed with high-pressure superheated steam from the heat recovery steam generator using a parabolic trough with molten salts in the temperature range of 300 °C to 550 °C. The solar steam is expected to influence the temperature of the steam mixture conditions by either maintaining or enhancing the temperature at the steam turbine inlet. In this hybrid system, the municipal solid waste was not able to totally offer thermal input near design operating conditions, especially the exhaust gas, which had a lower temperature; therefore, stream temperature at the steam turbine inlet dropped as did this turbine's power output. However, the injection of solar superheated steam at the turbine inlet helped boost the steam mixture temperature and quality in addition to the water used in gasification technology to cool down the syngas, which was then sent back to the Rankine cycle, resulting in annual electricity production reaching 634.49 GWh with a solar share of 10.10% under Spanish weather and 632.26 GWh with solar share of

12.06% under Saudi Arabian conditions. Furthermore, the achieved levelised cost of electricity based on this configuration's technical outcomes was 29.15 \$/MWh and 28.96 \$/MWh for Spain and Saudi Arabia, respectively.

The ISCC 4 configuration was fuelled by municipal solid waste through a gasification process and used the solar steam generator to extract a fraction of the preheated water from the high-pressure level economiser of the heat recovery steam generator, to be evaporated, superheated. After that, the solar steam was mixed with high-pressure superheated steam from the heat recovery steam generator using a solar power tower with molten salts in the temperature range of 300 °C to 550 °C. The solar steam is expected to influence the temperature of the steam mixture conditions by either maintaining or enhancing the temperature at the steam turbine inlet. In this hybrid system, the municipal solid waste was not able to totally offer thermal input near design operating conditions, especially the exhaust gas, which had lower temperature; therefore, steam temperature at the steam turbine inlet dropped as did this turbine's power output. However, the injection of solar superheated steam at the turbine inlet helped boost the steam mixture temperature and quality; in addition, the water used in gasification technology to cool down the syngas, which was then sent back to the Rankine cycle, resulting in annual electricity production reaching 646.41 GWh with a solar share of 12.80% under Spanish weather and 643.96 GWh with a solar share of 15.24% under Saudi Arabian conditions. Furthermore, the levelised cost of electricity achieved based on this configuration's technical outcomes were 28.62 \$/MWh and 28.45 \$/MWh for Spain and Saudi Arabia, respectively.

After achieving the aim and objectives, it can be seen that the research questions with regards to knowledge gap in Section 2.4 can be answered:

Research Question 1: *How will the syngas from municipal solid waste gasification affect the integrated solar combined cycle as a low energy content fuel replacement for natural gas?*

The impact of changing the natural gas with municipal solid waste in these studied configurations through gasification technology integration with both gas and steam turbine cycles was very clear. First, the annual total power output

under Spanish weather reached 590 GWh for the natural gas-based configuration (ISCC 1) while the waste-based configurations exceeded this value by around 43.6 GWh for ISCC 2, 44.3 GWh for ISCC 3 and 56.2 GWh for ISCC 4. In the Saudi Arabian case, ISCC 2, ISCC 3 and ISCC 4 provided 41.8 GWh, 42 GWh and 53.7 GWh, respectively, more total electricity than ISCC 1 which generated 572 GWh annually. Furthermore, the natural gas-based configuration required annual fuel thermal input of 1062 GWh under Spanish weather and 1030 GWh under Saudi Arabian weather but the studied waste-based configurations consumed less than these values by around 60 GWh for Spain and 48 GWh Saudi Arabia, respectively. In addition to that, the drop of fuel thermal input from waste-based configurations meant lower feedstock consumption was needed. The fuel cost of the ISCC 2, ISCC 3 and ISCC 4 was already expected to be less than the fuel cost of ISCC 1 by around \$18 million and \$19 million under Saudi Arabian and Spanish weathers, respectively due to the high cost of consumed natural gas. Moreover, the reduction in feedstock consumption through gasification connection to the steam cycle in ISCC has further dropped the cost of fuel in the waste-based configurations from \$11.1 million to \$10.6 million for Saudi Arabia and from \$11.5 million to \$10.8 million for Spain.

Research Question 2: *Can the solar field operating conditions be adjusted to enhance the solar thermal contribution within the integrated solar combined cycle to resolve the effects of using municipal solid waste gasification on the hybrid system?*

The role of solar field can be changed from generating saturated steam to superheated steam through replacing thermal oil with molten salt. This change demonstrated noticeable outcomes for the proposed configuration concept (ISCC 3 and ISCC 4). One of these outcomes included the improvement of annual total power outputs as described earlier. This was the result of the of increased average temperature of steam entering the turbine, around 517 °C for ISCC 3 and 519 for ISCC 4 °C in both Saudi Arabia and Spain leading to a rise in the solar produced electricity and therefore higher total power output which was closer to operating conditions of conventional integrated solar combined cycle, 524 °C.

Another reason related to solar power outputs was the result of the applied type of solar field. The use of solar power tower with molten salt in the proposed configuration concept, ISCC 4, was able to boost the solar electricity approximately 12 GWh higher than ISCC 3 under the Spanish and Saudi Arabian weathers.

Research Question 3: *Can the combination between municipal solid waste and adjusted solar field in the integrated solar combined cycle offer a practical type of Waste-to-Energy power plant?*

The proposed configuration concept of integrated solar combined cycle (ISCC 3 and ISCC 4) demonstrated that it can be a practical option as a Waste-to-Energy plant. This was due to achieving both high performance outcomes and low electricity costs when compared to the other configurations (ISCC 1 and ISCC 2). In terms of the technical performance, the effect of the proposed concept on major parameters was presented earlier in answers to **Research Question 1** and **Research Question 2**. In terms of the economic performance, the proposed configuration concept required higher capital cost than the conventional concept but it offered a significant drop in LCOE. by around 30 \$/MWh. Between the novel concept configurations, ISCC 4 offered lower costs than ISCC 3 by around \$130,000 in capital cost and 0.5 \$/MWh in electricity production cost for both Saudi Arabia and Spain.

Finally, this research demonstrated the novel concept offered the best technical and economic performances even at two different locations. However, there was notable differences between the outcomes from these considered locations, Al Ahsa in Saudi Arabia and Sevilla in Spain. Due to the higher ambient temperature and amount of radiation in the Saudi Arabian city, the integrated solar combined cycle was able to produce more solar thermal power than its counterpart in the Spanish city. This smaller production of solar thermal power in Spain was balanced by fuel contribution to the hybrid system leading to achieve higher total electricity production than ISCC in Saudi Arabia. Nevertheless, the two locations managed to offer close levelised costs of electricity with slightly lower value obtained by ISCC in Saudi Arabia. Hence, these outcomes highlighted that the

novel concept can be more beneficial technically in Sevilla city but from economic perspective, Al Ahsa city can be better.

8.2 Recommendations

The aim and objectives of this thesis have been successfully achieved but this is merely a step along the road in the field of integrated solar combined cycles. There are several quite interesting areas that have not been studied because of both time constraints and some areas having been outside the scope of this thesis. Nevertheless, these following areas are worth exploring in future work:

1. It was found that the implemented fuel was strongly in favour of the cases which used municipal solid waste gasification, producing much more power output even when there was no solar input. However, the research did not consider the possible improvement methods which could have been utilised in a conventional natural-gas-based layout to enhance the power output in the gas turbine, such as inlet air cooling through fogging (to spray inlet airflow with small water droplets to cool the air by evaporation), chilling (using a large unit to maintain the compressor inlet air temperature at a specified target temperature) [236] or fuel heating (heating the fuel to higher temperatures before entering the combustion chamber) [237].
2. The applied composition of municipal solid waste was maintained constant in the annual simulation for both countries due to lack of data. The collection and assessment of the varying MSW compositions in a region to represent the monthly feedstock data can be used to expand the knowledge in the impact of annual waste on ISCC power generation
3. It has been seen that the solar field was assessed as a black-box model throughout these thesis calculations by only focusing on the aperture area and heat transfer fluid. There are many factors that can be considered in the evaluation process of the solar field, such as the type of parabolic trough collector and the impact of the dimensions of heliostat on the solar conversion performance. In addition to that, these factors can play a significant role in the comparative cost analysis between the solar fields as well as the hybrid system configurations.

4. It was observed that the novel concept showed better economic outcomes than other configurations. However, these assessments did not include several influential costs such as feedstock supply costs. For this reason, a detailed economic analysis of the proposed ISCC for the three potential cities in both Spanish and Saudi Arabian with more economic factors in the assessments is recommended.

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APPENDICES

Appendix A ISCC Configurations in Thermoflex

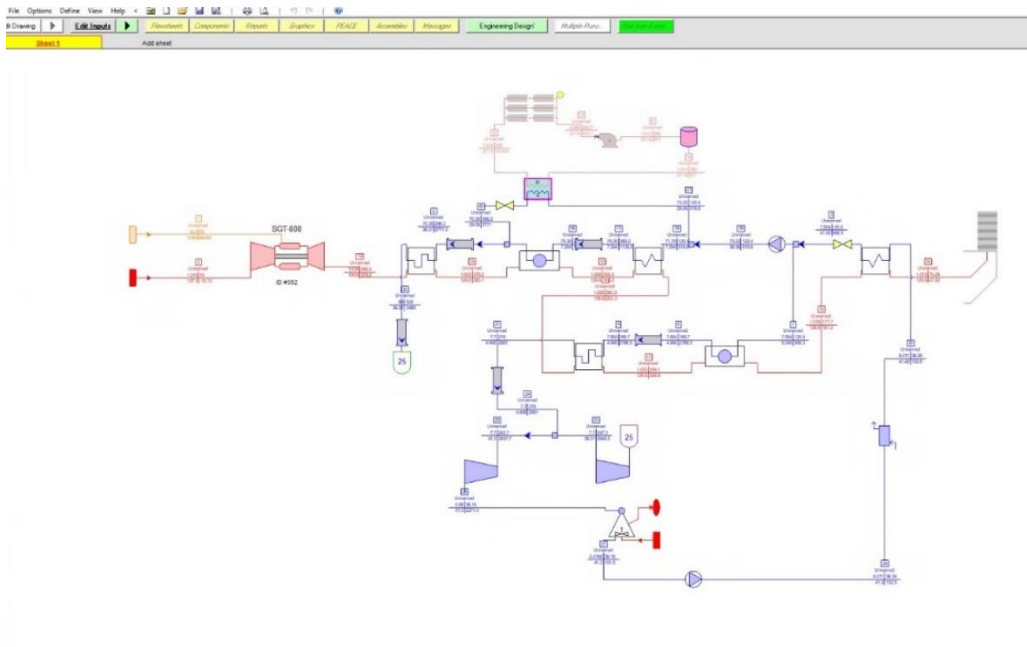


Figure Apx A- 1: ISCC 1 Configuration

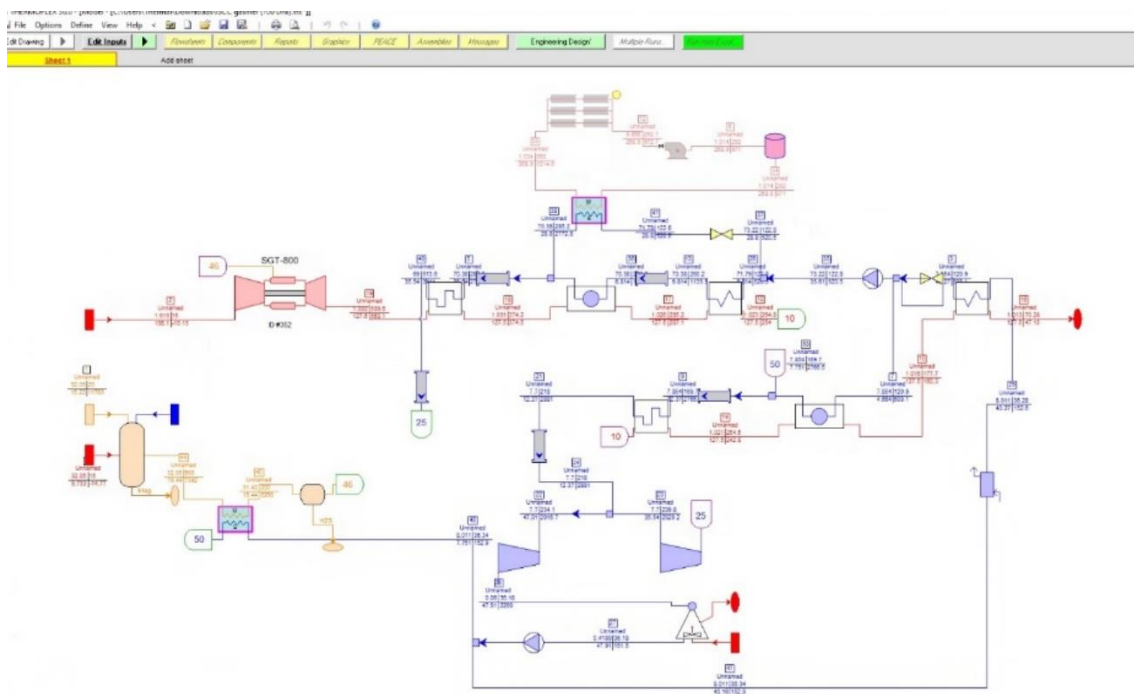


Figure Apx A- 2: ISCC 2 Configuration

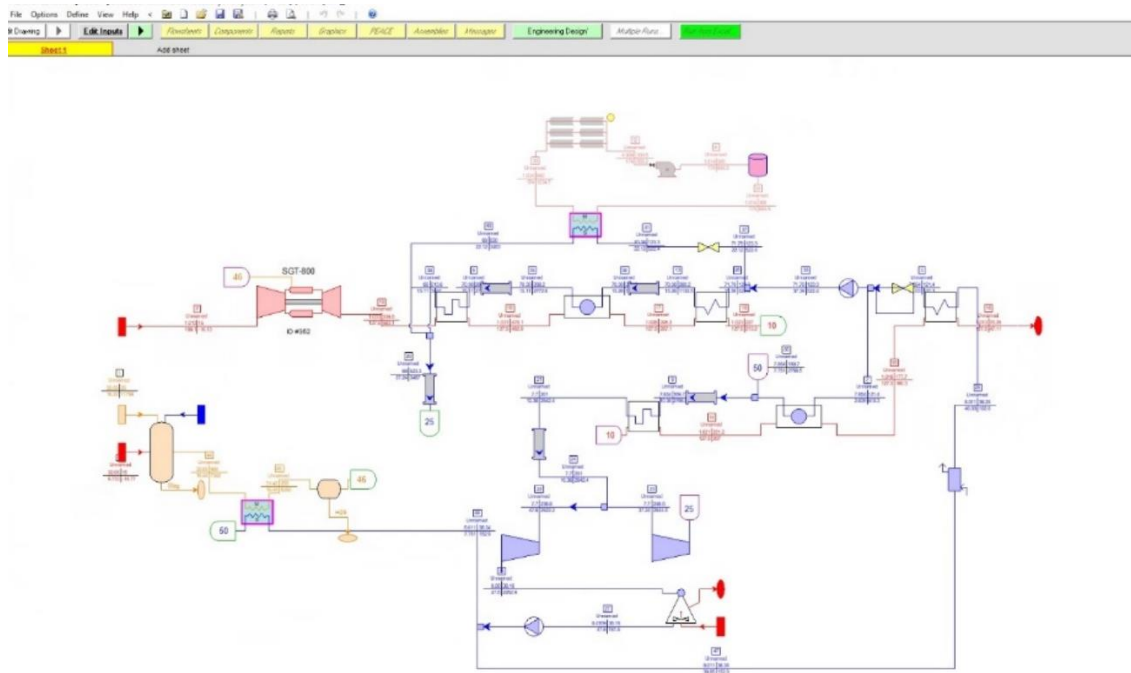


Figure Apx A- 3: ISCC 3 Configuration

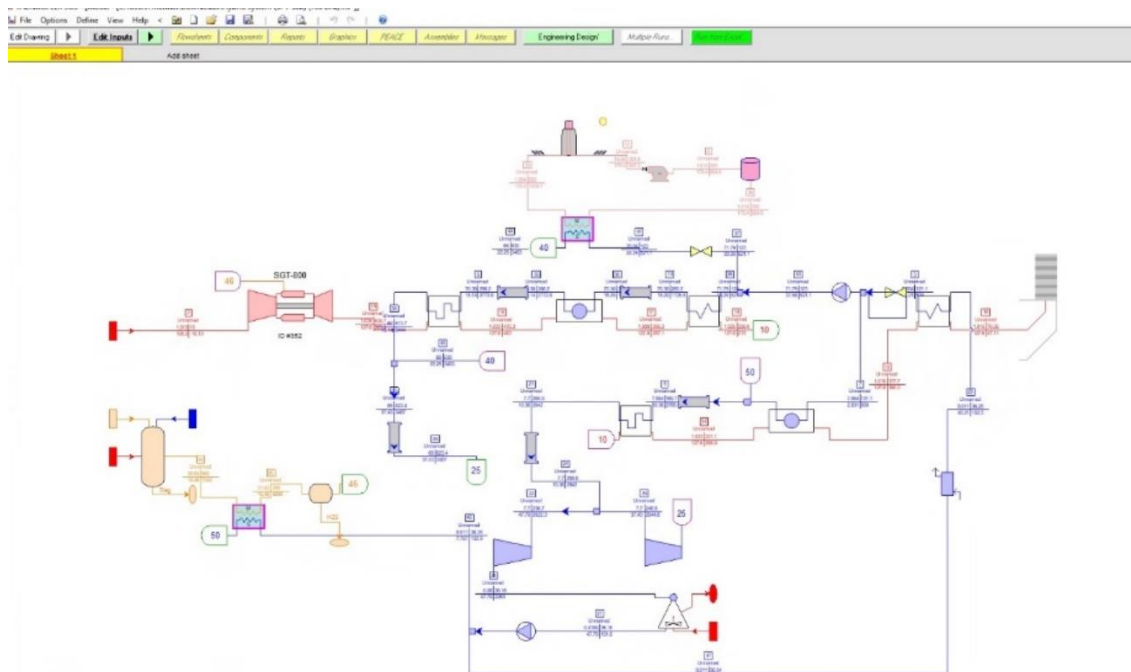


Figure Apx A- 4: ISCC 4 Configuration

Appendix B CSP Main Inputs in System Advisor Model

File

+

Add

700

Trough (phys), No financial

Location and Resource

System Design

Solar Field

Collectors (SCAs)

Receivers (HCEs)

Power Cycle

Thermal Storage

System Control

Grid Limits

Design Point Parameters

The design point parameters determine the nominal ratings of each part of the parabolic trough system. After specifying the design point parameters here, you can specify details of each component of the system on the Trough Field, Collectors (SCAs), Receivers (HCEs), Power Cycle, and Thermal Storage input pages.

Solar Field

Option 1:

Solar multiple

1

Option 2:

Field aperture

100,000.000 m²

Design point DNI

700 W/m²

Field thermal power

66 MWt

Loop inlet HTF temperature

300 °C

Loop outlet HTF temperature

550 °C

Number of loops

21

Power Cycle

Design turbine gross output

23.5 MWe

Estimated gross to net conversion factor

0.9

Estimated net output at design (nameplate)

21 MWe

Cycle thermal efficiency

0.356

Cycle thermal power

66 MWt

Thermal Energy Storage

Hours of storage at design point

0 hours

Figure Apx B- 1: Parabolic Trough Main Inputs Window

File

+

Add

700

Tower (salt), Single owner

Location and Resource

System Design

Heliostat Field

Tower and Receiver

Power Cycle

Thermal Storage

System Control

Grid Limits

Lifetime and Degradation

System Costs

Design Point Parameters

The design point parameters determine the nominal ratings of each part of the power tower system. After specifying the design point parameters here, you can specify details of each component of the system on the Heliostat Field, Tower and Receiver, Thermal Storage, and Power Cycle input pages.

Heliostat Field

Design point DNI

700 W/m²

Solar multiple

1

Receiver thermal power

66 MWt

Tower and Receiver

HTF hot temperature

550 °C

HTF cold temperature

300 °C

Thermal Storage

Full load hours of storage

0 hours

Solar field hours of storage

0 hours

Power Cycle

Design turbine gross output

23.5 MWe

Estimated gross to net conversion factor

0.9

Estimated net output at design (nameplate)

21 MWe

Cycle thermal efficiency

0.356

Cycle thermal power

66 MWt

Figure Apx B- 2: Solar Power Tower Main Inputs Window

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Appendix C ISCC Main Inputs in Thermoflex

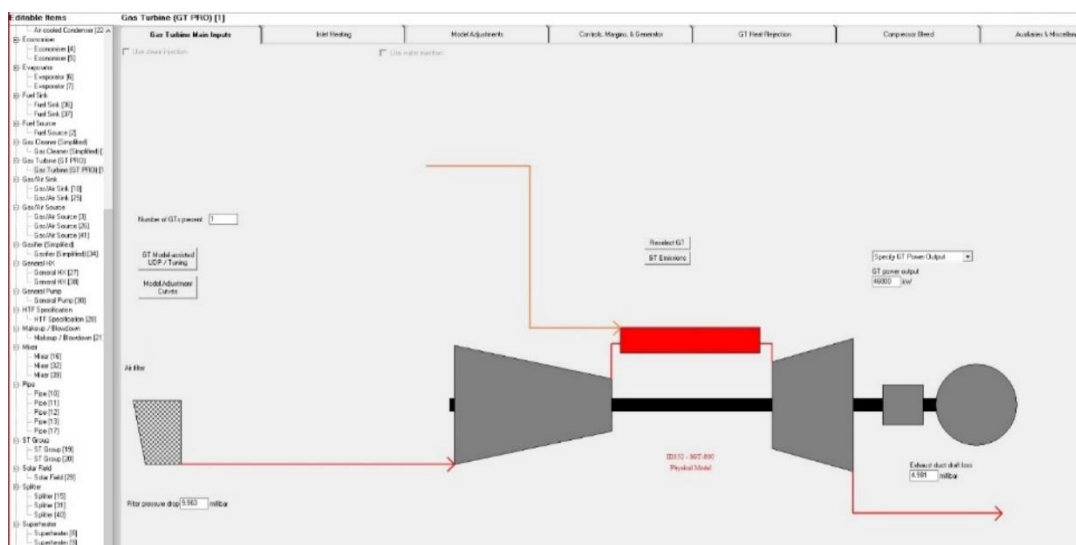


Figure Apx C- 1: Gas Turbine Input Window

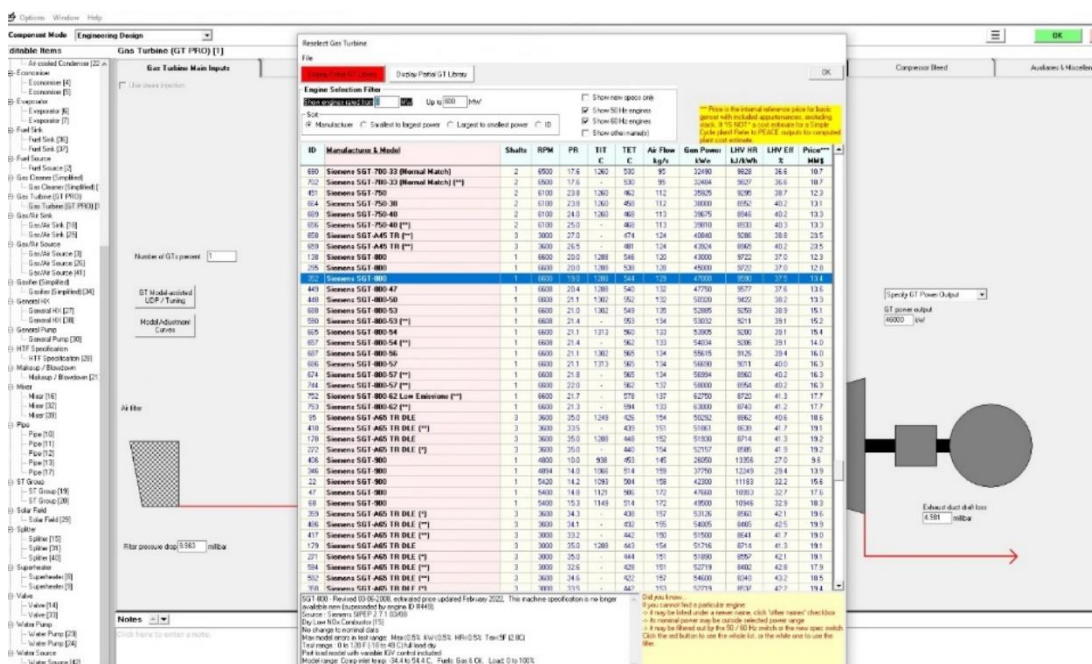


Figure Apx C- 2: Gas Turbine Type Window (Reselect GT Option)

Options Window Help

Component Mode **Modeling**

Editable Items

- Concentrator
 - Concentrator [4]
 - Concentrator [5]
- Exchanger
 - Exchanger [3]
 - Exchanger [7]
- Fuel Sink
 - Fuel Sink [36]
 - Fuel Sink [37]
- Fuel Source
 - Fuel Source [2]
- Gas Cleaner (Simplified)
 - Gas Cleaner (Simplified) [1]
- Gas Tubule (ST PFD)
 - Gas Tubule (ST PFD) [1]
- Gas/Ra Sink
 - Gas/Ra Sink [25]
- Gas/Ra Source
 - Gas/Ra Source [3]
 - Gas/Ra Source [25]
 - Gas/Ra Source [24]
- Gasifier (Simplified)
 - Gasifier (Simplified) [34]
- General HV
 - General HV [27]
 - General HV [38]
- General Pump
 - General Pump [30]
- HTF Specification
 - HTF Specification [28]
- Makrup / Blowdown
 - Makrup / Blowdown [21]
- Mixer
 - Mixer [16]
 - Mixer [32]
 - Mixer [39]
- Pipe
 - Pipe [10]
 - Pipe [11]
 - Pipe [12]
 - Pipe [13]
 - Pipe [17]
- ST Group
 - ST Group [19]
 - ST Group [20]
- Solar Tower
 - Solar Tower [29]
- Splitter
 - Splitter [15]
 - Splitter [21]
 - Splitter [42]
- Stack
 - Stack [18]
- Superheater
 - Superheater [3]
 - Superheater [3]
- Valve
 - Valve [14]
 - Valve [17]
- Water Pump
 - Water Pump [23]
 - Water Pump [24]
- Water Source
 - Water Source [42]

HTF Specification [28]

Pressure: 1.014 bar
 Mass flow: 4.536 kg/h
 Flow multiplier value (= outlet flow/inlet flow): 1
 Flow divider value (= inlet flow/outlet flow): 1
 Function: Enforce continuity
 Flow priority defined as:
☒ Priority class: Very weak
☐ Priority index: 1363 [Select Priority index](#)

Energy accounting

Energy input mode: gross
 Heat input price: 0 USD/USD
 Energy output mode: gross
 Heat output price: 0 USD/USD

Fluid Definition

Heat transfer fluid set by: Component
☒ Pick from Fluid Library
☐ Use defined fluid properties

Fluid Library

Heat Transfer Fluids

- DDU Chemical Company
 - DOL THERM 800 LIQUID
 - DDU HTF 500
 - DDU THERM 500 (Ethylen Glycol)
 - DDU THERM A LIQUID
 - DDU THERM G LIQUID
 - DDU THERM R LIQUID
 - DDU THERM T (Propylene Glycol)
 - DDU THERM T HD (Propylene Glycol)
- DUPATH THERM
 - DUPATH THERM 600
 - DUPATH THERM 630
 - DUPATH THERM 450
 - DUPATH THERM S
 - DUPATH THERM LT
 - DUPATH THERM LT
- Paratherm Corporation
 - Paratherm NF
 - Paratherm HE
 - Paratherm NR
 - Paratherm MG
 - Paratherm LR
- Schuta
 - THERMINOL LT
 - THERMINOL D 12
 - THERMINOL VP
 - THERMINOL 55
 - THERMINOL 59
 - THERMINOL 62
 - THERMINOL 66
 - THERMINOL 72
 - THERMINOL 75
 - THERMINOL VP-1
- EconHeli
 - MOBIL THERM 42
- Molten Salt
 - Molten Salt 60% NaNO₂ - 40% KNO₃ by wt
 - HITEC-XL 42% Ca(NO₃)₂ - 15% NaNO₂ - 43% KNO₃ by wt
 - Wackus Chemie AG
 - HELIOX SA
 - Other

CURRENT SELECTION

Molten Salt 60% NaNO₂ - 40% KNO₃ by wt

Minimum temperature 280 C
 Maximum temperature 593 C

Properties at specified condition:
 Vapor pressure 0 bar
 Density 1696.1 kg/m³
 Specific heat 1.571 kJ/kg°C
 Dynamic viscosity 0.0033 kg/m.s
 Thermal conductivity 0.5033 W/m.K

Notes

[Click here to enter a note.](#)

Figure Apx C- 5: Heat Transfer Fluid Input Window

Options Window Help

Component Mode **Engineering Design**

Editable Items

- Air-cooled Condenser (22)
 - Concentrator [4]
 - Concentrator [5]
- Exchanger
 - Exchanger [3]
 - Exchanger [7]
- Fuel Sink
 - Fuel Sink [36]
 - Fuel Sink [37]
- Fuel Source
 - Fuel Source [2]
- Gas Cleaner (Simplified)
 - Gas Cleaner (Simplified) [1]
- Gas Tubule (ST PFD)
 - Gas Tubule (ST PFD) [1]
- Gas/Ra Sink
 - Gas/Ra Sink [25]
- Gas/Ra Source
 - Gas/Ra Source [3]
 - Gas/Ra Source [25]
 - Gas/Ra Source [24]
- Gasifier (Simplified)
 - Gasifier (Simplified) [34]
- General HV
 - General HV [27]
 - General HV [38]
- General Pump
 - General Pump [30]
- HTF Specification
 - HTF Specification [28]
- Makrup / Blowdown
 - Makrup / Blowdown [21]
- Mixer
 - Mixer [16]
 - Mixer [32]
 - Mixer [39]
- Pipe
 - Pipe [10]
 - Pipe [11]
 - Pipe [12]
 - Pipe [13]
 - Pipe [17]
- ST Group
 - ST Group [19]
 - ST Group [20]
- Solar Field
 - Solar Field [28]
- Splitter
 - Splitter [15]
 - Splitter [21]
 - Splitter [42]
- Superheater
 - Superheater [3]
 - Superheater [3]
- Valve
 - Valve [14]
 - Valve [17]
- Water Pump
 - Water Pump [23]
 - Water Pump [24]
- Water Source
 - Water Source [42]

Solar Field [28]

ED Main Inputs

Desired heat flux: 174 kg/h
 Desired solar multiple: 1
 Design point collector normal optical efficiency: 0.711 %
 Desired inlet temperature: 550 °C
 Exit header pressure drop: 1.551 bar
 Exit header heat loss: 3.468 MJ/kg

Collector Pressure Drop Calculation (Excluding Headers)

☒ Use hardware determined pressure drop
☐ Use defined performance

Desired mass flow in receiver tubes: 1743.9 kg/h²s
 Maximum collector loophead (including headers) pressure drop: 30 %

Flow Priority

☒ Strong
☐ Weak

Notes

[Click here to enter a note.](#)

Figure Apx C- 6: Parabolic Trough Input Window

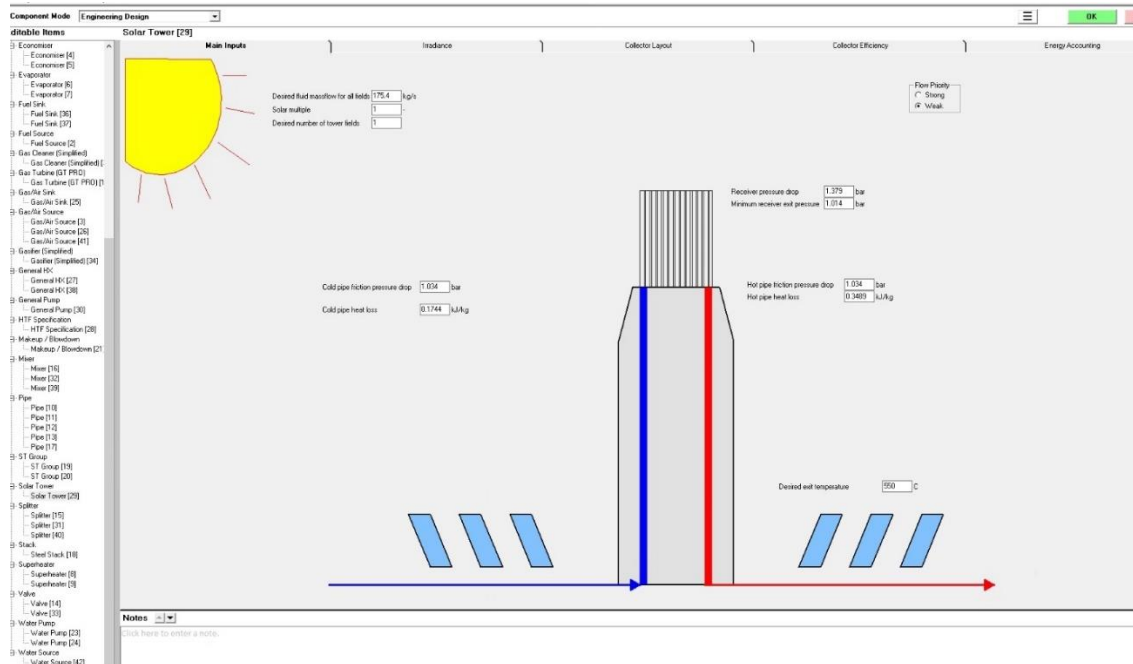


Figure Apx C- 7: Solar Power Tower Input Window

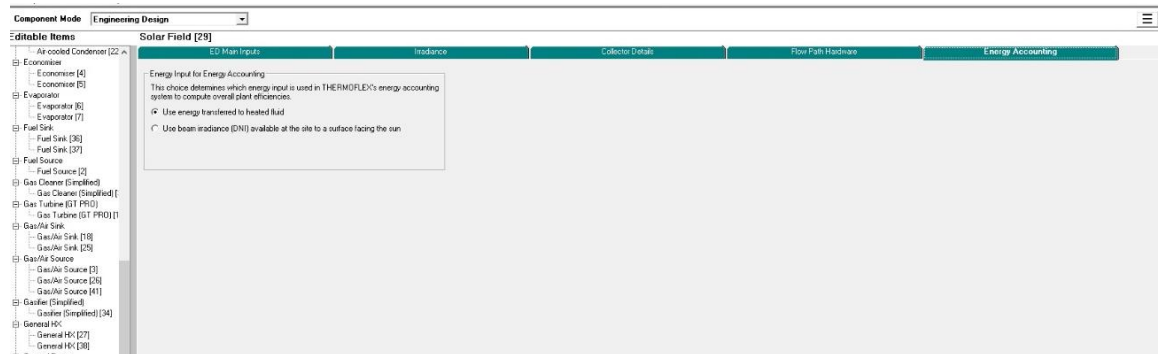


Figure Apx C- 8: HTF Option as Main Thermal Input From Energy Accounting
Solar Field Input Window

Appendix D Annual Cost Calculations

D.1 Gas Turbine Cycle

D.1.1 Compressor

Table Apx D- 1 Compressor Input Values

Input Parameter	Value
$\dot{m}_{air}$	128 kg/s
P_1	1.013 Bar
P_2	19.1 Bar
$\eta_{Compressor}$	80%

The capital cost of the compressor in GTC can calculated using Equation (6-1):

$$Z_{Compressor} = \frac{75 * \dot{m}_{air}}{0.9 - \eta_{Compressor}} * \left(\frac{P_2}{P_1}\right) * \ln\left(\frac{P_2}{P_1}\right)$$

$$Z_{Compressor} = \frac{75 * 128}{0.9 - 0.8} * \left(\frac{19.1}{1.013}\right) * \ln\left(\frac{19.1}{1.013}\right)$$

$$Z_{Compressor} = \$ 5,315,760$$

D.1.2 Combustion Chamber and Gas Turbine

Table Apx D- 2 Combustion Chamber and Gas Turbine Input Values

Input Parameter	Value
$\dot{m}_{gas}$	130 kg/s
P_2	19.1 Bar
P_3	18.34 Bar
P_4	1.03 Bar
T_3	1450.5 K
η_{GT}	85%

The capital cost of the combustion chamber in GTC can be calculated using Equation (6-2):

$$Z_{Combustion\ Chamber} = \left(\frac{48.64 * \dot{m}_{gas}}{0.995 - \frac{P_3}{P_2}} \right) * (1 + \exp(0.018 * T_3 + 26.4))$$

$$Z_{Combustion\ Chamber} = \left(\frac{48.64 * 130}{0.995 - \frac{18.34}{19.1}} \right) * (1 + \exp(0.018 * 1450.5 + 26.4))$$

$$Z_{Combustion\ Chamber} = \$ 317,612$$

And for the capital cost of the turbine in GTC, it can be calculated using Equation (6-3):

$$Z_{GT} = \left(\frac{1536 * \dot{m}_{gas}}{0.92 - \eta_{GT}} \right) * \ln\left(\frac{P_4}{P_3}\right) * (1 + \exp(0.036 * T_3 - 54.4))$$

$$Z_{GT} = \left(\frac{1536 * 130}{0.92 - 0.85} \right) * \ln\left(\frac{1.03}{18.34}\right) * (1 + \exp(0.036 * 1450.5 - 54.4))$$

$$Z_{GT} = \$ 9,131,494$$

D.1.3 Capital and O&M Costs for GTC

Table Apx D- 3 Capital Costs of GTC Components

Component	Value (\$)
Compressor	5,315,760
Combustion Chamber	317,612
Gas Turbine	9,131,494

Finally, the capital cost of the GTC and its O&M costs can be calculated using Equation (6-4) and Equation (6-5):

$$Z_{GTC} = Z_{Compressor} + Z_{Combustion\ Chamber} + Z_{GT}$$

$$Z_{GTC} = \$ 14,764,866$$

$$Z_{O\&M(GTC)} = 0.05 * Z_{GTC}$$

$$Z_{O\&M(GTC)} = \$ 738,243$$

D.2 Steam Turbine Cycle

D.2.1 Heat Recovery Steam Generator

Table Apx D- 4 HRSG Input Values

HX	$LMTD_i$ (K)	$\dot{Q}_i$ (kW)	$\dot{m}_{steam_i}$ (kg/s)	f_{P_i}	f_{steam_i}	f_{gas_i}
HP-SH	74.34	8952	14	1.13	1.94	1.58
HP-EV	52.89	22624	14	1.13	1.58	1.43
HP-EC	61.75	7220	14	1.13	1.55	1.39
LP-SH	44.29	1150.9	10	0.93	1.51	1.38
LP-EV	27.89	8460	4	0.93	1.46	1.34
LP-EC	55.22	8106	18	0.93	1.44	1.30

The estimation of HRSG capital cost can be conducted with Equation (6-6) after calculating its input parameters from Equation (6-7), Equation (6-8), Equation (6-9), and Equation (6-10):

$$Z_{HRSG} = 4131.8 * \Sigma \left(f_{P_i} * f_{steam_i} * f_{gas_i} * \left(\frac{\dot{Q}_{HRSG_i}}{LMTD_{HRSG_i}} \right)^{0.8} \right) + 13380 * \Sigma (f_{P_i} * \dot{m}_{steam_i}) + 1489.7 * \dot{m}_{gas}^{1.2}$$

$$Z_{HRSG} = 4131.8 * (889.63) + 13380 * (76.99) + 1489.7 * (130)^{1.2}$$

$$Z_{HRSG} = \$ 5,232,833$$

D.2.2 Pumps and Condenser

Table Apx D- 5 Input Values for Pumps and Condenser

Input Parameter	Value
$\dot{W}_{HP-Pump}$	188 kW
$\dot{W}_{LP-Pump}$	36 kW
$\eta_{HP-Pump}$	85%
$\dot{Q}_{COND}$	50000 kW
$\dot{m}_{COND}$	1198 kg/s
$T_{COND (in)}$	298.15 K
$T_{COND (out)}$	339.89 K
$T_{steam (COND)}$	491.15 K
$T_{water (COND)}$	309.15 K
$\dot{W}_{HP-Pump}$	188 kW

For the HP and LP pumps cost calculations, Equation (6-13) can be applied:

$$Z_{HP-Pump} = 708.48 * \dot{W}_{HP-Pump}^{0.7} * \left(1 + \frac{0.2}{1 - \eta_{HP-Pump}} \right)$$

$$Z_{HP-Pump} = 708.48 * (188)^{0.7} * \left(1 + \frac{0.2}{1 - 0.85} \right)$$

$$\mathbf{Z_{HP-Pump} = \$ 67,782}$$

$$Z_{LP-Pump} = 708.48 * \dot{W}_{LP-Pump}^{0.7} * \left(1 + \frac{0.2}{1 - \eta_{LP-Pump}} \right)$$

$$Z_{LP-Pump} = 708.48 * (36)^{0.7} * \left(1 + \frac{0.2}{1 - 0.85} \right)$$

$$\mathbf{Z_{LP-Pump} = \$ 20,962}$$

And for the condenser, its capital cost can be estimated after obtaining the value of $LMTD_{COND}$ through Equation (6-10):

$$LMTD_{COND} = \frac{((T_{steam} - T_{COND_{out}}) - (T_{water} - T_{COND_{in}}))}{\ln \frac{(T_{steam} - T_{COND_{out}})}{(T_{water} - T_{COND_{in}})}}$$

$$LMTD_{COND} = \frac{((491.15 - 339.89) - (309.15 - 298.15))}{\ln \frac{(491.15 - 339.89)}{(309.15 - 298.15)}}$$

$$LMTD_{COND} = 54 \text{ K}$$

Therefore, the capital cost of the condenser in STC is estimated using Equation (6-12):

$$Z_{COND} = 280.74 * \left(\frac{\dot{Q}_{COND}}{2.2 * LMTD_{COND}} \right) * 746 * \dot{m}_{COND}$$

$$Z_{COND} = 280.74 * \left(\frac{50000}{2.2 * 54} \right) * 746 * 1198$$

$$Z_{COND} = \$ 1,011,865$$

D.2.3 Steam Turbine

Table Apx D- 6 Steam Turbine Input Values

Input Parameter	Value for Spain	Value for Saudi Arabia
$\dot{W}_{ST-ISCC1}$	25656 kW	26568 kW
$\dot{W}_{ST-ISCC2}$	29403 kW	30506 kW
$\dot{W}_{ST-ISCC3}$	29401 kW	30391 kW
$\dot{W}_{ST-ISCC4}$	30942 kW	31983 kW
$T_{ST-ISCC1}$	823.15 K	823.15 K
$T_{ST-ISCC2}$	777 K	777 K

$\dot{W}_{ST-ISCC3}$	799 K	799 K
$\dot{W}_{ST-ISCC4}$	799 K	799 K
η_{ST}	85%	85%

The capital cost estimations of the STC steam turbine as shown in Table Apx D-7 takes into account the size variations based on the configuration and selected location based on Equation (6-11):

$$Z_{ST} = 3880.5 * \dot{W}_{ST}^{0.7} * \left[1 + 5 * \exp\left(\frac{T_{ST(in)} * 866}{10.42}\right) \right] * \left[1 + \left(\frac{0.05}{1 - \eta_{ST}}\right)^3 \right]$$

Table Apx D- 7 Capital cost of Steam Turbine

Input Parameter	Value for Spain (\$)	Value for Saudi Arabia (\$)
$Z_{ST-ISCC1}$	5,312,292	5,443,784
$Z_{ST-ISCC2}$	5,407,293	5,548,498
$Z_{ST-ISCC3}$	5,445,315	5,573,025
$Z_{ST-ISCC4}$	5,643,563	5,775,811

D.2.4 Capital and O&M Costs for STC

Finally, the capital cost of the STC and its O&M costs can be calculated using Equation (6-14) and Equation (6-15):

$$Z_{STC} = Z_{HRSG} + Z_{ST} + Z_{COND} + Z_{Pump}$$

$$Z_{O\&M(STC)} = 0.02 * Z_{STC}$$

Table Apx D- 8 Capital and O&M Costs for STC in the Different Cases

STC	Value for Spain (\$)	Value for Saudi Arabia (\$)
$Z_{STC-ISCC1}$	11,645,734	11,777,226

$Z_{STC-ISCC2}$	11,740,735	11,881,940
$Z_{STC-ISCC3}$	11,778,757	11,906,467
$Z_{STC-ISCC4}$	11,977,005	12,109,253
$Z_{O\&M(STC-ISCC1)}$	232,915	235,545
$Z_{O\&M(STC-ISCC2)}$	234,815	237,639
$Z_{O\&M(STC-ISCC3)}$	235,575	238,129
$Z_{O\&M(STC-ISCC4)}$	239,540	242,185

D.3 Fuel and Gasification Technology

Table Apx D- 9 Input Values for Gasifier and Fuel in the Different Cases

Parameter	Value for Spain	Value for Saudi Arabia
$\dot{m}_{dry\ feedstock}$	68700 kg/h	68700 kg/h
$\dot{Q}_{fuel-ISCC1}$	1062 GWh	1030 GWh
$\dot{Q}_{fuel-ISCC2}$	1002 GWh	983 GWh
$\dot{Q}_{fuel-ISCC3}$	1002 GWh	983 GWh
$\dot{Q}_{fuel-ISCC4}$	1002 GWh	983 GWh

The calculations of capital and O&M costs for the gasification technology or gasifier unit in addition to the fuel are conducted using Equation (6-24), Equation (6-25), Equation (6-26), Equation (6-27) and Equation (6-28):

$$Z_{Gasifier} = 1600 * (\dot{m}_{dry\ feedstock})^{0.67}$$

$$Z_{Syngas\ Cleaning} = 0.05 * Z_{Gasifier}$$

$$Z_{Gasifier\ Unit} = Z_{Gasifier} + Z_{Syngas\ Cleaning}$$

$$Z_{O\&M(Gasifier\ Unit)} = 0.03 * Z_{Gasifier\ Unit}$$

$$Z_{fuel} = F * \dot{Q}_{fuel}$$

Table Apx D- 10 Capital and O&M Costs for Gasifier and Fuel in the Different Cases

Cost	Value for Spain (\$)	Value for Saudi Arabia (\$)
$Z_{Gasifier}$	\$ 2,785,358	\$ 2,785,358
$Z_{Syngas\ Cleaning}$	\$ 139,268	\$ 139,268
$Z_{Gasifier\ Unit}$	\$ 2,924,626	\$ 2,924,626
$Z_{O\&M(Gasifier\ Unit)}$	\$ 87,739	\$ 87,739
$Z_{fuel-ISCC1}$	\$ 29,053,774	\$ 28,184,247
$Z_{fuel-ISCC2}$	\$ 10,820,637	\$ 10,619,285
$Z_{fuel-ISCC3}$	\$ 10,820,637	\$ 10,619,285
$Z_{fuel-ISCC4}$	\$ 10,820,637	\$ 10,619,285

D.4 Solar Field Unit

Table Apx D- 11 Input Values for Gasifier and Fuel in the Different Cases

Parameter	Value for Spain	Value for Saudi Arabia
$A_{ap\ (PT)}$	137,340 m ²	137,340 m ²
$A_{ap\ (ST)}$	174,179 m ²	174,179 m ²
$\dot{m}_{HTF(PT-TO)}$	269 kg/s	269 kg/s
$\dot{m}_{HTF(PT-MS)}$	174 kg/s	174 kg/s
$\dot{m}_{HTF(SPT-MS)}$	175 kg/s	175 kg/s

The calculations of capital and O&M costs for the solar field unit are calculated after obtaining the components costs using Equation (6-16) – Equation (6-20):

$$Z_{Aap (PT-TO)} = 150 * A$$

$$Z_{Aap (PT-MS)} = 180 * A$$

$$Z_{Aap (SPT-MS)} = 140 * A$$

$$Z_{HTF(TO)} = 2.2 * \dot{m}_{HTF(TO)}$$

$$Z_{HTF(MS)} = 1.3 * \dot{m}_{HTF(MS)}$$

Table Apx D- 12 Capital Cost of Solar Field Components

Component	Value for Spain (\$)	Value for Saudi Arabia (\$)
$Z_{Aap (PT-TO)}$	20,601,000	20,601,000
$Z_{Aap (PT-MS)}$	24,721,200	24,721,200
$Z_{Aap (SPT-MS)}$	24,385,060	24,385,060
$Z_{HTF(PT-TO)}$	2,129,688	2,129,688
$Z_{HTF(PT-MS)}$	815,256	815,256
$Z_{HTF(SPT-MS)}$	820,872	820,872

Finally, the overall costs of capital cost of the solar field unit were estimated through Equation (6-21), Equation (6-22) and Equation (6-23):

$$Z_{CSP} = Z_{Aap} + Z_{HTF}$$

$$Z_{O\&M(PT)} = 0.022 * Z_{CSP(PT)}$$

$$Z_{O\&M(SPT)} = 0.023 * Z_{CSP(SPT)}$$

Table Apx D- 13 Capital and O&M Costs of CSP Unit in the Different Configurations

Cost	Value for Spain (\$)	Value for Saudi Arabia (\$)
$Z_{CSP-ISCC1}$	22,730,688	22,730,688
$Z_{CSP-ISCC2}$	22,730,688	22,730,688
$Z_{CSP-ISCC3}$	25,536,456	25,536,456
$Z_{CSP-ISCC4}$	25,205,932	25,205,932
$Z_{O\&M(CSP-ISCC1)}$	500,075	500,075
$Z_{O\&M(CSP-ISCC2)}$	500,075	500,075
$Z_{O\&M(CSP-ISCC3)}$	561,802	561,802
$Z_{O\&M(CSP-ISCC4)}$	579,736	579,736