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**A Simulink/Simscape dynamic modelling study on shifting the
power output profile of a direct steam generation solar power
system with steam/water storage**

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Abstract

Solar thermal power plants convert thermal energy from direct solar radiation into mechanical work and electricity via a thermodynamic cycle. This method produces renewable electricity. A direct steam generation technology (DSG), which generates steam directly in the solar field's absorber tubes and feeds it directly to the turbine or thermal storage, is one of the many options in solar thermal power plants. It has a number of intriguing advantages and is a promising technology. However, one of the major problems with the use of the direct steam generation technology is that it is affected by the fluctuation of solar radiation, which results in the production of fluctuating power output and no economic competition for long-term storage. The design of the control system is complicated by the steam generation system's challenging dynamic behaviour. It is primarily caused by the coexistence of two-phase flow in the absorber tubes and the natural transitory feature of solar radiation.

In solar thermal power plants, thermal storage is a critical requirement. There are different types of heat storage systems in solar plants such as molten salts, concretes, phase change materials and steam accumulators. The steam accumulator has been adopted in Planta Solar 10 project. Steam accumulator is a viable option for decreasing the influence of changing irradiance on the power generation of solar thermal systems since they have fast reaction time and high discharge rates due to the rapid evaporation and condensation of water/steam under non-equilibrium conditions. This thesis aims to study the dynamic behaviour of a novel direct steam generation solar power system integrated with a steam/water accumulator for shifting its power output profile to match the electricity demand profile in a Libyan hospital as a case study.

Another issue in DSG solar power systems is that expansion in turbine might happen in liquid-vapour two-phase region. Use of a cascade Rankine cycle can be a solution, in which an organic Rankine cycle (ORC) is a bottom cycle, while a steam Rankine cycle is a top cycle with a higher condensation temperature to mitigate the issue of wet expansion. The bottom cycle can also run separately using low-temperature residue steam/water in the accumulator to offer a further capacity in shifting power output profile. Therefore, the design of a cascade steam-organic Rankine cycle integrated with steam\water accumulator is studied from operating and

thermodynamic perspectives, aiming to fully unlock the potential of such advanced high-efficiency cogeneration systems.

This thesis presents a comprehensive literature review on Rankine cycle solar power generation systems regarding cycle configurations, thermal storage and working fluids. The design, validation and parametric study of a direct steam generation solar power system is conducted for Libyan climate conditions, particular with dynamic simulation of a steam/water accumulator. Moreover, a cascade steam-organic Rankine cycle solar power system integrated a steam accumulator was studied. Finally, the cascade steam-organic Rankine cycle solar power system integrated with two water accumulators was modelled. All the designs presented in this thesis was simulated by using Simulink\Simscape software in order to see the dynamic behaviour of systems under different climate conditions.

As a case study, the electricity demand profile of a Libyan hospital was chosen as a target in the design of system. The results demonstrated that by using 1544m^2 of parabolic trough solar collectors and 160m^3 of steam storage tank with a proper control of flow rate, the power output profile of the system can meet the electricity demand of the Libyan hospital even at night. In the cascade steam-organic Rankine cycle solar power system coupled with the steam accumulator with and without using a recuperator, the findings reveal that the recuperative system outperforms the non-recuperative system in terms of power generation and thermal efficiency. In the cascade steam-organic Rankine cycle solar power system integrated with two water storage tanks, results show that the top cycle output is higher than ORC power output in the nominal operation mode. At the peak radiation time, the top cycle generates 136 kW, while the ORC generates 37 kW. In the discharging mode in the evening, a separate operation the bottom ORC system can still generate 31 kW for six hours based with using a 51 m^3 hot water tank.

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List of Publications

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Nomenclature

Abbreviation

Added	Heat flux added
CSP	Concentrating Solar Power
DSG	Direct steam generation
HTF	Heat transfer fluid
LMTD	Log-mean temperature difference
NTU	Number of Transfer Units
ORC	Organic Rankine cycle
PCM	Phase change material
PTC	Parabolic trough collector
TES	Thermal energy storage
RECUP	Recuperator
SRC	Solar Rankine cycle

Greek symbols

Δ	Difference or variation
ε	Effectiveness [%]
θ	Incidence angle, °
η	Efficiency [%]
τ	Time [hour]
ρ	Density [kg/m ³]
η_{poly}	Polytropic efficiency
$\eta_{exponent}$	The efficiency exponent
$\phi_{flow,coeff}$	The nominal mass flow coefficient
$\phi_{flow,coeff,thr}$	The mass flow coefficient threshold
ϕ_{α}	Stodola's Ellipse constant
ω	The absolute angular velocities
ϕ_{con}	The condensation energy in accumulator
δ	Delta
α_L	The heat transfer coefficient for the liquid zone

Variables

A	Area, m ²	$\dot{m}_{con}$	The mass exchange rates for condensation
A_L	Liquid inlet	$mech$	Mechanical
AU	Product of area and overall heat transfer coefficient, W K ⁻¹	$\dot{m}$	Mass flow rate, kg/s
A_V	Steam inlet	m_{Liquid}	Mass liquid in Chamber
B_L	Liquid outlet	N	Rotational speed, RPM
B_V	Steam outlet	Nu	Nusselt number
$B^{\eta_{exponent}}$	Pressure ratio threshold	P	Pressure, MPa

C_1	Heat loss term, $\text{Wm}^{-2}\text{K}^{-1}$	P_r	Prandtl number
C_2	Heat loss term, $\text{Wm}^{-2}\text{K}^{-1}$	P_{ratio}	Pressure ratio
C_p	Specific heat capacity, $\text{Jkg}^{-1}\text{K}^{-1}$	$\dot{Q}$	Heat transfer rate, W
col, in	Collector Inlet Temperature	S_w	The surface area of the accumulator
col, out	Collector outlet Temperature	T	Temperature, $^{\circ}\text{C}$
con	Condensation state	T_{am}	Ambient temperature
D	Diameter, m	T_{in}	Inlet temperature
ele	Electrical	T_{out}	Outlet temperature
f_{VL}	The volume fraction of the accumulator	v	Specific volume, m^3kg^{-1}
G	Solar irradiance, Wm^{-2}	V	Volume, m^3
h	Enthalpy, kJ/kg	vap	Vapour state
K_{θ}	Incident angle modifier	$\dot{W}$	Power, kW
k_1	Negative of the first-order coefficient of the efficiency		
k_2	Negative of the second-order coefficient of the efficiency		
L	Length of Chamber		
M	Mass, kg		

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Chapter 1 Introduction

1.1 Background

There is a significant environmental issue facing our society and the world nowadays. Most modern energy generation comes from environmentally unsustainable sources such as oil, coal and natural gas. These unsustainable energy sources directly contribute to greenhouse gas emissions, which cause environmental damage and global warming. Governments and agencies are now studying alternative strategies and the steps that must be taken to reverse this depressing picture as a result of these environmental concerns. Political action, research, and the development of sustainable energy generation are required for the solution. All of these arguments highlight the need for clean, practical renewable energy sources, such as wind energy, biomass, geothermal and solar energy. Among them, solar energy is one of the most appealing ones, given its enormous potential and steadily declining production costs [1].

There are two main solar power generation technologies. The first is photovoltaics, which uses the photovoltaic effect to produce power directly. Small-scale production, residential generation, off-grid installations in distant locations, and industrial-scale plants are highly suitable for this sort of generating. The absence of widely applicable, economically viable energy storage systems is a significant limitation of this technology. Solar thermal power plants or concentrating solar plants (CSP), which utilize the thermal energy converted from solar radiation, provide an alternative method of using solar energy. This technology appears to be the one most likely to meet a sizable portion of the future demand for renewable energy [2]. Two technology roadmaps have been created by the International Energy Agency for CSP generation. According to the roadmap from 2010, the solar thermal power plant was expected to generate around 11% of the world's electricity by 2050. This objective was retained

in the revised 2014 roadmap, which encouraged the development of solar thermal power plant systems [3]. By 2050, solar energy might supply 27% of the world's electricity needs, when the solar thermal power plants and photovoltaic systems are taken into account. This roadmap recommends the steps that might be taken to restrict long-term global warming to 2 °C [3]. The most plentiful energy source on earth is solar radiation. The amount of radiation energy that reaches the earth's surface is 3,500 times greater than the projected global energy need for 2050. Solar radiation is a resource with a relatively low density, so its efficient harvesting is challenging. In order to do this, concentrating solar plant systems often use glass mirrors or reflective film to focus radiation onto a receiver element and so utilize it more effectively. Only the direct fraction of the solar radiation is useful in these systems, since dispersed radiation cannot be concentrated. Due to the receiver's lower surface area and the concentrated radiation, it can attain higher temperatures than non-concentrating solar plants. While concentrating solar plant systems can operate at temperatures beyond 400 °C, flat plate solar collectors typically do not operate above 150 °C. The Heat Transfer Fluid (HTF), which allows the absorption of the thermal energy from the focused radiation, is circulated inside the receiver. Along with producing electric energy, this heated HTF can also be utilised for cooling, salt-water desalination, and the creation of steam for industrial processes. In a Rankine cycle, the steam is typically generated by the heated HTF to generate electricity.

Solar thermal concentrating power plant systems can be integrated to operate in combination with other renewable energy sources, such as geothermal, biomass and waste heat recovery or conventional fuel burners. This integration may reduce the investment risk and increase the dispatchability at the same time. Additionally, thermal power plants that are currently in operation could be integrated with solar

fields in order to decrease the fuel consumption in these plants. The addition of a thermal energy storage (TES) system is another option for increasing the dispatchability of solar thermal plants, enabling them to overcome solar transients like the passage of clouds and even continue to provide energy well into the night. This is a significant advantage of concentrating solar plants over other renewable energy technologies like photovoltaics and wind energy. Around 4 GW of concentrating solar plants' electric energy production capacity was deployed globally in 2014, compared to 150 GW for photovoltaics. Concentrating solar power plants using TES systems may alter this situation due to their dispatchability on demand [4].

Despite appearing competitive in the solar energy market, concentrating solar plants and photovoltaics are ultimately complementary. The energy requirement must be met during the entire day in order to sustain a larger solar matrix. While TES systems can be used by concentrating solar plants to produce electricity early in the morning, late at night, and also well into the night, the photovoltaic output is more productive in the middle of the day. The significance of TES in that concentrating solar plants increases with the proportion of others type of solar energy in the global energetic matrix.

The majority of concentrating solar power plants use thermal oil as their heat transfer fluid. In order to feed a turbine, the heated oil travels through heat exchanger systems in the power block, where it preheats, boils, and, in most cases, superheats water or steam. In these situations, the solar field operates with thermal oil, and the power block operates with water/steam. These two circuits are distinct from one another and operate using separate fluids. The Direct Steam Generation (DSG) technique, where steam is produced directly in the solar field, is a relatively new. This steam, which is sent straight to the turbine, can either be saturated or superheated.

There are many concentrating solar power plant design options. They can be utilized in both big/small installations and hybrid/solar-only configurations, employ thermal oil or water/steam as the heat transfer fluid, etc. To ensure the success of this technology, especially in relation to the direct steam generation system operation, many questions still need to be resolved, and the related processes must be optimized.

In summary, the electricity demand is growing daily and has become an essential part of human life. Existing supplies can meet this demand, but the environmental concerns about fossil fuels continue to increase. Although society and governments generally support renewable energy, far more research and development are required in this area if we are to meet the demand adequately.

As a renewable energy solution, a direct steam generation solar power system has higher efficiency and is cheaper compared to an indirect steam generation system. However, the former cannot generate power after the sun sets. An auxiliary heater system has been employed for the direct steam generation solar power system in order to overcome the fluctuation in solar radiation; nevertheless, these auxiliary heaters are very expensive and known to cause certain problems in the solar power plant, so there is a need to find solutions regarding the smooth operation of direct steam generation solar power systems. Therefore, this research aims to carry out a dynamic modelling study on shifting the power output profile of a direct steam generation solar power system with steam/water storage by using Simulink/Simscape tools.

1.2 Solar energy potential in Libya

The position of Libya and its solar radiation resources are quite favourable regarding the use of solar energy. Lying between latitude 19-34° North and longitude 9-26° East, Libya is located in the middle of North Africa. Around 90 per cent of Libya's total

area is desert, and the majority of the nation is situated in the Sunbelt. The Sahara Desert covers the majority of Libya (located at latitude $24^{\circ} 17'$ and $30^{\circ} 3'$ North and longitude $11^{\circ} 44'$ to $23^{\circ} 58'$ East) [5]. The nation's solar radiation resources are considered to be abundant. Figures 1.1 and 1.2 show a country-wide distribution map of the average yearly Global Horizontal Irradiation (GHI) and Direct Normal Irradiation (DNI), respectively.

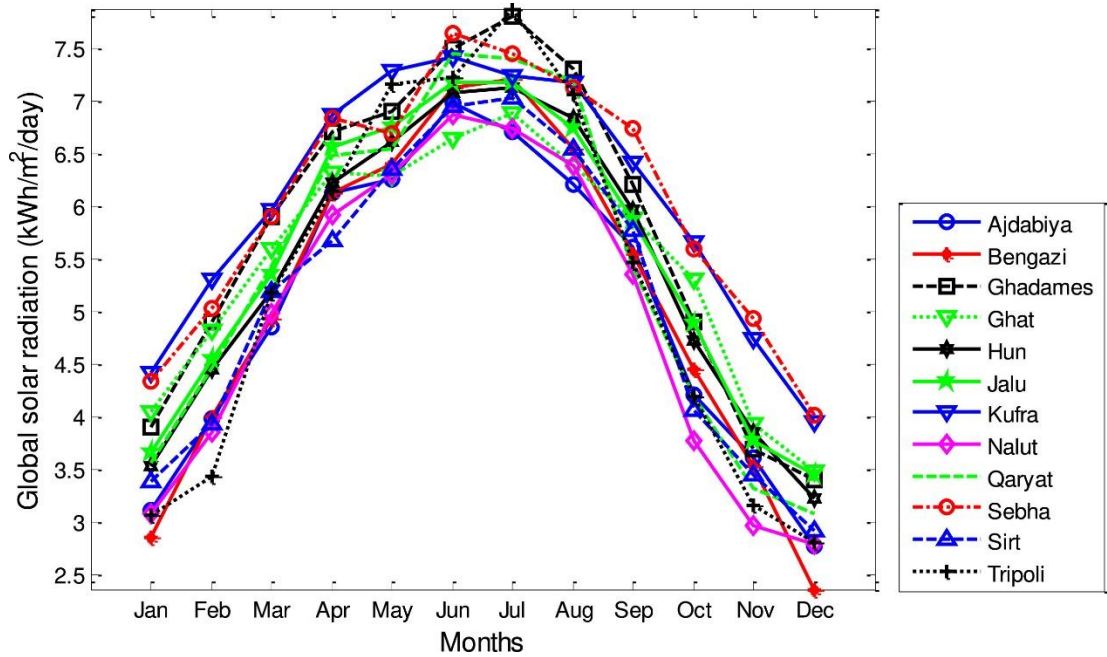


Figure 1.1: Global Horizontal Irradiation for 12 cities in Libya [5]

SolarGIS [6] collected and averaged these data for the years 1994 and 2010. It is clear that the average annual GHI in the Northern regions is around $2000 \text{ kWh/m}^2/\text{year}$, and over $2600 \text{ kWh/m}^2/\text{year}$ in the Southern region, so it appears that the majority of cities in Libya have a good average annual GHI, which will open the door to do more solar energy system investment in the future, thereby leading to decreased electricity shutdowns in some cities.

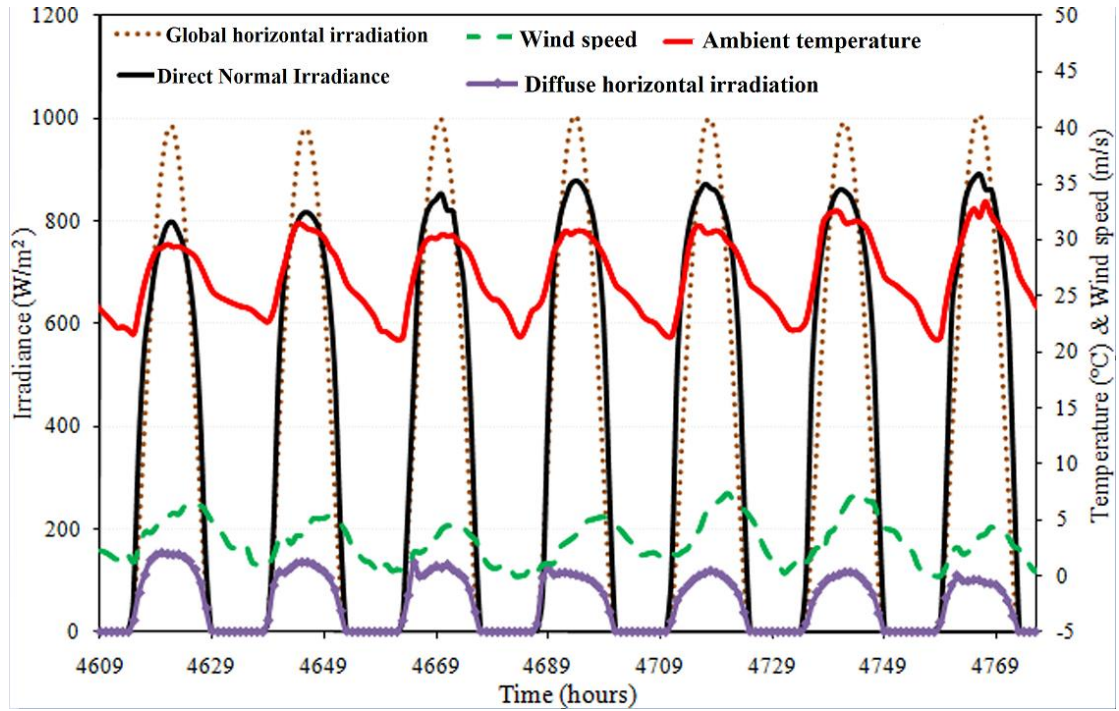


Figure 1.2: Direct Normal Irradiance in Libya [6]

On the other hand, the South has an average annual DNI of 2500 kWh/m^2 , which is higher than the coastal region's $1900 \text{ kWh/m}^2/\text{year}$ [7]. The NASA Surface Meteorology and Solar Energy database also provides data on the insolation, sky clearness index and average number of sunshine hours. Figure 1.3 displays the monthly average sky clearness index for ten different locations in Libya. According to the obtained data, the Al-Kufra location has the highest clearness value (0.68), while Tripoli has the lowest (0.57).

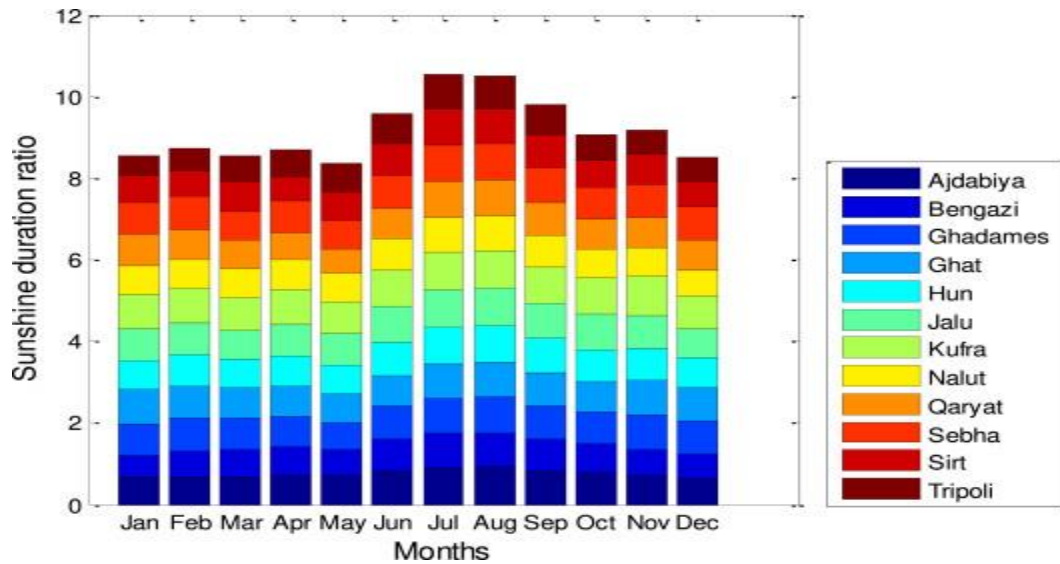


Figure 1.3: Annual sunshine duration ratio for 12 cities in Libya [5]

Additionally, there are more than 3,500 hours of sunlight annually, on average, nationwide. Only areas with $\text{DNI} > 1800 \text{ kWh/m}^2/\text{year}$ [6] can be deemed economically advantageous for concentrating solar power technology plants. Due to this aspect, depending on the quantity and quality of the solar radiation accessible in Libya, the concentrating solar power technology appears to be a very promising option for the country's energy generation.

1.3 Issues related to Libyan energy supply:

Libya's population and economic expansion have increased the strain on the country's energy supply. This increased demand for electricity necessitates further large investments in the production of electricity, including power plants and power cables. Based on information gathered from the General Electric Company of Libya (GECOL), Figure 1.4 depicts the demand for electricity in Libya. Electric consumption is expanding quickly, by roughly 8-10% annually, as shown, and is predicted to reach 9 GW by 2020 [8]. The country's entire electricity industry is run by the government-owned GECOL. Since its founding in 1984, the GECOL company has installed and managed every power plant in Libya. The installed electricity

generation capacity of Libya is 6.6 GW [8]. Most of Libya's conventional fossil fuel power facilities, such as steam turbine, gas turbine and combined cycle power plants, are used to produce electrical energy. Steam power plants make up 20% of the total installed power capacity, while gas turbine and combined cycle power plants account for 43% and 37%, respectively. Additionally, a few small diesel power plants help to supplement the energy supply, particularly in the rural areas and several villages located in the south of the country [8].

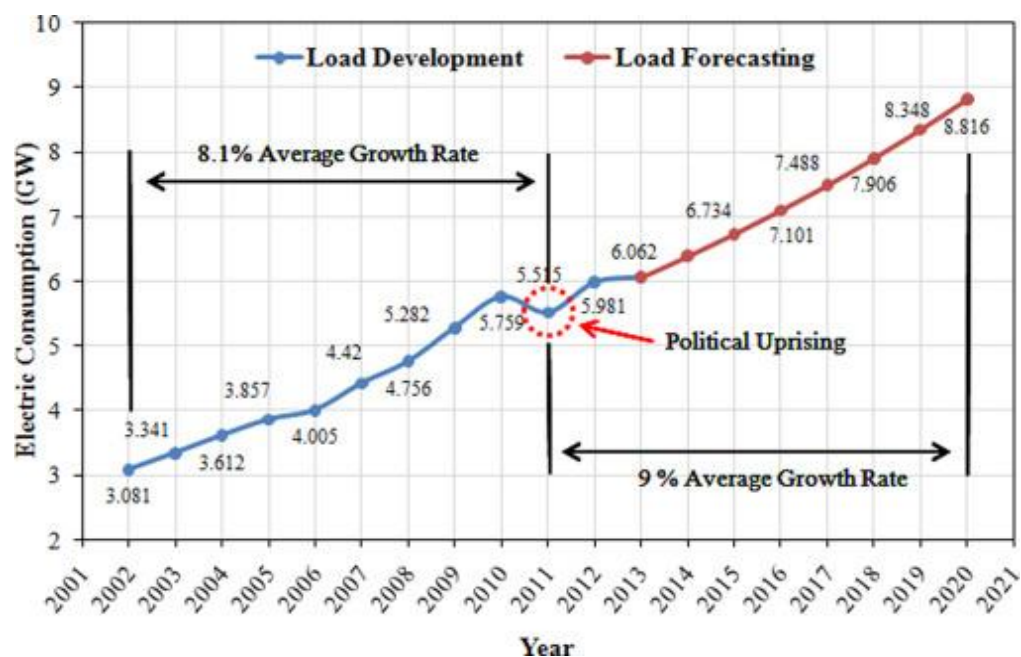


Figure 1.4: The demand for electricity in Libya [6]

Since the demand for electrical energy will soon outstrip the production capacity, GECOL plans to construct additional combined cycle and steam cycle power plants. GECOL intends to upgrade and expand the nation's electricity transmission grid in addition to expanding its generation capacity. The only solution to lessen this reliance on fossil fuels and the environmental issues that accompany their burning is to use renewable energy sources, notably solar energy, which are sufficiently abundant in Libya. The development of renewable energy for electricity production is proposed in

Libya. The company's main goal is to produce 2.219 GW, or 10%, of the total electricity needs, from renewable sources by 2025.

Forty percent of Libya's population, which was mostly supported by agriculture, lived in the countryside, according to the 2018 census. In 2018, 40% of the population of Libya lived in rural areas without access to power. It was determined that just six out of the 20 communities in Libya without electricity were located in distant, difficult-to-reach places. In addition to Libya's domestic energy issues, there is a growing international pressure to employ ever-more sustainable energy sources due to concerns about global warming as well as the rising global prices of fossil fuels. Governments worldwide, including that of Libya, are working to boost the share of renewable energy sources in the world's energy mix. As a result, businesses and researchers are constantly seeking ways to enhance and create novel approaches to delivering power and other energy services.

1.3.1 Challenge:

The main issue regarding the solar electricity generation systems is how to control the thermal plants to maintain power output to match the temporal demand for energy despite the fluctuations in solar radiation. In a solar thermal power plant, unlike a typical power plant where fuel is utilised as the easily manipulated variable, solar radiation cannot be changed and, paradoxically, it functions a racket heat resource because it varies not only daily but also seasonally. However, Ruth's accumulator storage unit has been adopted in the Puerto Errado 1 and Planta Solar 10 plants to cover some periods of limited insolation, and steam is generated at variable pressures in this storage unit to power the turbine for one or two hours [7]. This method of energy storage calls for the design of more configurations of solar thermal power plants which can overcome the fluctuations in solar radiation over a 24-hour period.

Overall, the fluctuations in the levels of solar radiation during a day can strongly affect the performance of a solar electricity generation system. The fluctuations in ambient temperature and solar radiation pose a real challenge when seeking to control a solar thermal power system.

1.3.2 Motivation:

A direct steam generation solar power system appears to be promising technology among other solar electricity generation systems. In this system, water is directly heated in the solar fields, so there is no need to use a second heat transfer fluid loop, and the use of heat exchangers is avoided, thereby reducing the system's total cost. Single-stage steam accumulators are used in the conventional Rankine cycles and in solar Rankine cycles as heat storage units to improve the efficiency of the system or, rather, counteract some of the fluctuations in solar irradiance [9]. The saturated steam generated in accumulators is directly injected into the steam turbine, as in Planta Solar 10 plant in Spain. However, technical challenges remain regarding direct steam generation solar power systems. First, during the heat discharge process, the flashing steam pressure and mass flow rate drop, leading to off-design operations and sophisticated system control techniques. Second, because moisture is present during the expansion phase, the wet steam turbine is inefficient. Typically, the moisture of the exhaust steam should not exceed 14% [10]. These problems can be alleviated or even solved by an innovative cascade steam-organic Rankine cycle based on a direct steam generation solar power system integrated with a single-steam accumulator or two-stage liquid accumulators combined with a control strategy regarding the system. Overall, the proposed system using a cascade Rankine cycle and a single-steam accumulator or two-stage liquid accumulators is promising, but the control of the

proposal system and its influence on the system dynamic behaviour needs to be investigated, which is the main target of this study.

1.4 The main aim and objectives of the thesis

Even though steam Rankine cycle solar power plants have been demonstrated for large-scale applications, there remain significant obstacles that need to be overcome in order to encourage their wider market adoption, especially for high-temperature applications and a coupling with organic Rankine cycle and thermal energy storage (TES) systems. In this context, this thesis aims to conduct a dynamic modelling study on shifting the power output profile of a direct steam generation solar power system with steam/water storage via using the Simulink/Simscape program. The following goals have been created in order to accomplish the main objective of this study:

- To undertake a comprehensive literature review on Rankine cycle solar power systems regarding cycle configurations, thermal energy storage, applications, working fluids; and a control strategy regarding the direct steam generation solar power system are presented and discussed, together with the current gap regarding cascade Rankine cycle for solar power applications.
- To conduct a thermodynamic analysis of the integrated parabolic trough collectors and steam Rankine cycle system. A thorough modelling of the design and off-design condition for a solar power system with direct steam generation will be conducted.
- To conduct a comparative study of direct steam generation and indirect steam generation solar power systems under southern Libyan climate conditions.

- To model and conduct a dynamic simulation study on the use of a steam accumulator as a heat storage unit in a single-stage steam Rankine cycle solar power system.
- To conduct a dynamic simulation study on the use of a steam accumulator as a heat storage unit in a cascade steam-organic Rankine cycle solar power system in order to use the condensation heat rejected from the steam cycle to drive the bottom ORC system, which thereby leads to the extended total power generation of the proposed system. By maintaining the same component dimensions, the proposed cascade system is modelled and contrasted with various existing systems. The advantages of the amendments to the system will be analysed and explained.
- To study the transient performance of a cascade steam-organic Rankine cycle solar power system with the use of a hot water temperature accumulator and a secondary low water temperature accumulator for a typical weather condition.

1.5 The main contribution of this research

Recently, the research on user demand-based systems has peaked. This project addresses several of the research gaps in the open literature by combining the user demand profile with the intermittence of direct steam generation (DSG) solar Rankine cycles to meet the demand energy profile for a Libyan hospital, as a case study. A good control strategy of DSG, integrated with a steam accumulator and a domestic hot water (DHW) system by using a throttle valve, has demonstrated that the system has the ability to meet the demand energy profile for a couple of chosen days under the typical solar irradiance conditions in Libya. In addition, an innovation solar-driven cascade steam-organic Rankine cycle, integrated with a steam accumulator as the heat storage unit, has been modelled and investigated in more detail, with a good control strategy. The dynamic response of each component of the proposed system has been

discussed and presented for a simulation period of a whole day. A new innovative method for the thermal storage of a solar-driven cascade steam-organic Rankine cycle, integrated with two heat storage accumulators through a two-step heat discharge, has been designed and simulated in this thesis, making it possible to extend the shift power output time of the direct steam generation solar power system.

1.6 The research methodology and software tools utilized in this study

1.6.1 Research method

The objectives of this study, which are listed as above, were achieved by employing a combination of research methods as follows; by reviewing the literature in depth to comprehend the scope of this project and find pertinent key information from earlier published studies, a thorough literature review of concentration solar collectors, direct and indirect steam generation systems, control strategy, heat storage units, steam Rankine cycle, and cascade steam-organic Rankine cycle was conducted. Modelling Simulink/Simscape software was used to model the system's transient behaviour mathematically. The components should react in accordance with the operating parameters as the operating conditions, such as the heat source temperature and energy demand from a building. This led to the modelling of an off-design operation. From the installed Cool-Prop and Ref-prop software in the Simulink/Simscape tools, the working fluid properties were taken. The EnergyPlus program was used to obtain the Libyan weather data.

1.6.2 Software tools

Simulink/Simscape is used throughout this thesis as it has a deep numerical analysis feature for each block of the built thermal network. It is a graphical programming environment built on MATLAB that is utilized to design, simulate, code and research

dynamic systems. It adopts a block diagram approach to designing, modelling and simulation, and can be used to model and control systems in the domains of thermal, electrical and mechanical engineering. Simscape, a tool for physical modelling, is part of the Simulink environment. Users can simulate the behaviour of simple to sophisticated models by utilising this programme. It is a highly sought-after software program that it is widely employed in the industry to create virtual prototypes of systems. Example of basic Simscape network with heat transfer between thermal liquid and Physical block and two-phase fluid models is shown in Figure 1.5 below. In this figure, it has been connected three different parts, two phase fluid's part, thermal part and thermal liquid part. Each block has two ports, port A means inlet flow and port B means outlet flow of the block. Each part has own sensors, blocks and fluid properties block. Two phase fluid's part uses in cases such as thermal power plant, refrigeration cycles and steam systems. Thermal liquid part uses in cases like heat systems, engines and thermal oil cycles, while thermal parts uses in heat transfer fluid between two cycle such as conductive heat transfer and convective heat transfer.

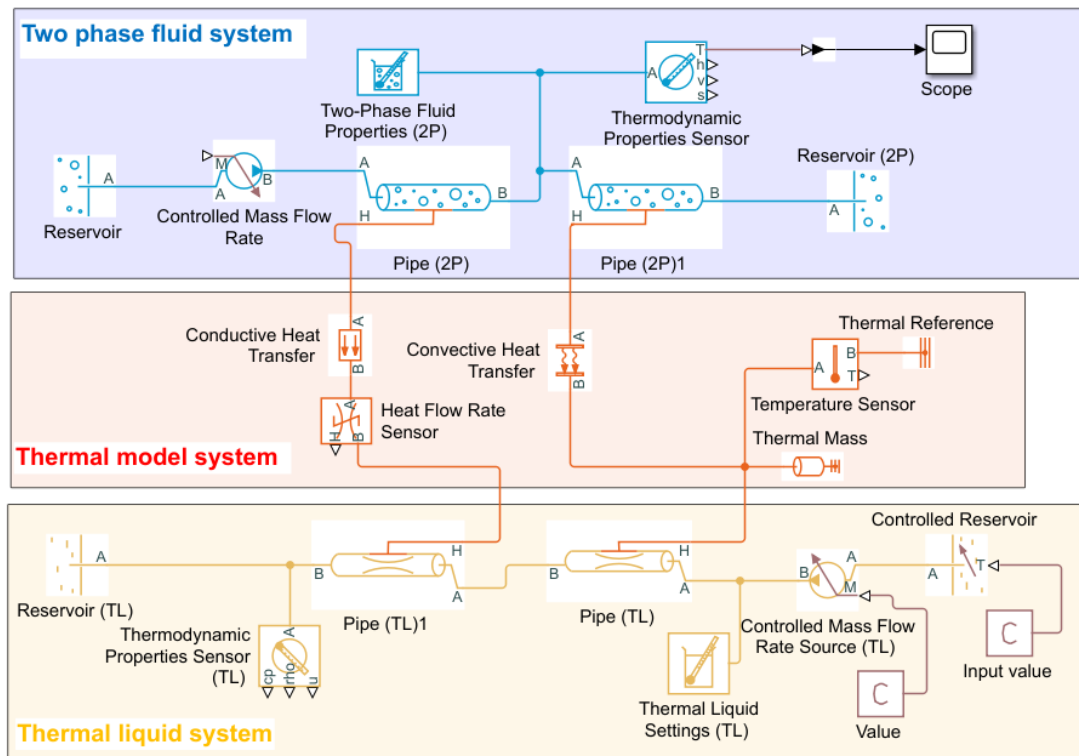


Figure 1.5: Example of basic Simscape network with heat transfer between thermal liquid and Physical block and two-phase fluid models

By arranging the basic parts into a schematic, researchers can mimic systems, such as mechanical parts, hydraulic actuators, bridge rectifiers, electric motors, and thermal refrigeration systems. Simscape add-on products offer more sophisticated parts and analysis capabilities. Simscape software aids the development of control systems and the testing of system performance and behaviour. The Simscape language, which is based on MATLAB and allows text-based writing of physical modelling components, domains, and libraries, allows users to develop bespoke component models. MATLAB variables and expressions can be used to parameterize the users' models, and Simulink can be used to develop control schemes for the users' physical system. Simscape offers C-code creation for deploying user models in different simulation environments, such as hardware-in-the-loop systems. Simulink/Simscape tools facilitate the design, simulation, prototyping, and implementation of advanced

controls for an accurate digital physical system, increase system reliability, and save weeks of customer start-up time and months of design time.

1.7 Outline of the thesis

In this thesis, a model is designed and developed for combining the user demand profile with the intermittence of direct steam generation (DSG) solar Rankine cycles to meet the demand energy profile for a Libyan hospital, as a case study. In addition, an innovation solar-driven cascade steam-organic Rankine cycle, integrated with a steam accumulator as the heat storage unit, will be modelled and investigated in more detail, with a good control strategy. The dynamic response of each component of the proposed system will be discussed and presented for a simulation period of a whole day via using Simulink/Simscape tools. A new innovative method for the thermal storage of a solar-driven cascade steam-organic Rankine cycle, integrated with two heat storage accumulators through a two-step heat discharge, will also be designed and simulated in this thesis. The thesis is structured as follows:

Chapter Two reviews the solar thermal power generation systems and components. The direct and indirect steam generation solar power systems and also a solar Rankine cycle and an organic Rankine cycle are compared, respectively. Four different type of concentration solar collectors and heat storage units are presented in this chapter. Finally, solar-driven cascade Rankine cycles are discussed and the research gaps in the literature are identified.

In Chapter Three, the main modelling chapter, the design and off-design performance of a direct steam generation solar power system are modelled. All equations and thermal components in Simscape are described and presented in this chapter. The final

section focuses on modelling a direct steam generation solar Rankine cycle and comparing the results with an indirect steam generation cycle.

Chapter Four focuses on the proposed integrated steam accumulator with a solar Rankine cycle system and how the different sizes of storage capacity affect the power output of the system. Also, an innovative direct steam generation solar power system, coupled with a steam accumulator as the heat storage unit and DHW process system, is proposed to meet the energy demand profile for a Libyan hospital, as a case study.

A novel solar-driven cascade steam-organic Rankine cycle, integrated with a steam storage unit, is modelled and simulated in Chapter Five. Four different working fluids are investigated. Also, the effect of using a recuperator in the ORC system is compared with a non-recuperative ORC system.

Chapter Six presents a novel method for a solar-driven cascade steam-organic Rankine cycle integrated with a hot temperature accumulator and low temperature accumulator as a unique heat storage unit. The system is controlled and simulated in the transient mode. Four different working fluids are investigated. Also, a comparison study is made between a solar-driven cascade steam-organic Rankine cycle integrated with a hot temperature accumulator and low temperature accumulator system and the solar-driven cascade steam-organic Rankine cycle integrated with a steam storage unit in Chapter Five.

In Chapter Seven, the thesis ends with a summary of the key conclusions and results, together with recommendations for future research.

Chapter 2 A review of the solar Rankine cycle on the aspects of configurations, working fluids and heat storage

2.1 Introduction

It is essential to increase the conversion efficiency and power outputs in order optimally to exploit the potential of solar resources. This is the landscape where the solar Rankine cycle and solar organic Rankine cycle come into play. Although almost identical to the classical water/steam Rankine cycle, the solar organic Rankine cycle can convert heat at low pressure and temperature into electricity due to the careful selection of an alternative working fluid and its low boiling point [11,12]. Other benefits include easy control strategy operation for these systems, favourable operating pressures and low maintenance operation. Over the last decade, the solar Rankine cycle has become a mature technology. Many reference installations are already available from several manufacturers. Furthermore, historical installations have demonstrated the advantages of the solar Rankine cycle. Even though numerous solar Rankine systems are currently in operation, the strategic problem of how to enhance the solar Rankine cycle thermodynamic performance and, consequently, its competitiveness remains [12]. There are five focus points when optimising a solar Rankine cycle:

- The working fluid selection;
- The configurations of the cycles;
- The control strategy;
- The component layout and sizing;
- The solar heat collector types.

Firstly, the criteria for fluid selection for the solar Rankine cycle have been recommended based on the thermodynamic performance of the cycle. Many studies

have reported that it is necessary to consider technical criteria related to safety, risk, and cost when selecting working fluids. Also, screening schemes are set up and multi-objective optimization approaches are introduced [13]. Secondly, the configuration of a solar Rankine cycle plays an important role in improving the output and efficiency of the cycle. In addition, the configuration of a solar Rankine cycle, the development of new, improved expander designs has attracted considerable attention. Thirdly, other studies have focused on the optimal control strategy for the solar Rankine cycle, such as the control pressure, temperature, fluid quality, and mass flow rate as well as the mechanical power output of the system [14]. Fourth, the type of solar collector chosen has a direct impact on the system's output and efficiency. The choice of the cycle architecture, followed by component sizing, is the final degree of flexibility. Cycle adaptations, similar to the traditional Rankine cycle, can increase the performance and cost-effectiveness by adding reheaters and recuperators or working in a supercritical mode. In the vapour compression cycles, this development route can also be observed. As a result, investigating solar Rankine cycle adaptations is a logical next step. A brief overview of some of the solar Rankine cycle architectures can already be found in the review articles [15].

The review chapter considers several titles and subtitles as follows: The introduction and background of solar thermal generation systems are introduced in Sections 2.1 and 2.2. Section 2.3 presents the control strategies for solar direct steam generation systems. Comparison between direct and indirect steam generation solar direct steam generation systems are discussed in Section 2.4, while Section 2.5 compares the theory behind the solar steam Rankine cycle and the solar organic Rankine cycle. Different types of solar collectors in the direct steam generation solar power systems are discussed in Section 2.6. The storage methods for direct steam generation solar power

systems are discussed in Section 2.7. Different solar Rankine cycle configurations are presented in detail in Section 2.8 of this chapter. Finally, dynamic modelling of solar Rankine cycle is presented in section 2.9.

2.2 Background of solar Rankine cycle

One of the most competitive and common power generation systems is a solar Rankine cycle system, which works based on the solar irradiance levels during a day [16, 17]. Thermal solar power plants are identical to conventional ones except that, instead of a steam boiler, a field of concentrating sun collectors is used. A conventional boiler is also included in hybrid plants, and it operates on conventional fuel, usually natural gas, whenever it is required. As a result, solar power plant thermal analysis is similar to that of any other plant, and the same thermodynamic relations are used. Drawing the cycle on a T-s diagram substantially simplifies the analysis is presented in the figure 1.2. The inefficiencies of the pump and steam turbine should be considered in these situations. A simple solar steam Rankine cycle consists of four main components: (1) the solar collectors' fields, (2) a steam turbine, (3) a condenser unit and (4) a pump working fluids [18]. As shown in Figure 2.1 the four main components of a Solar Rankine cycle work at high pressure and a high temperature level and a low pressure and low temperature level, respectively. The steps of the cycle are as follows:

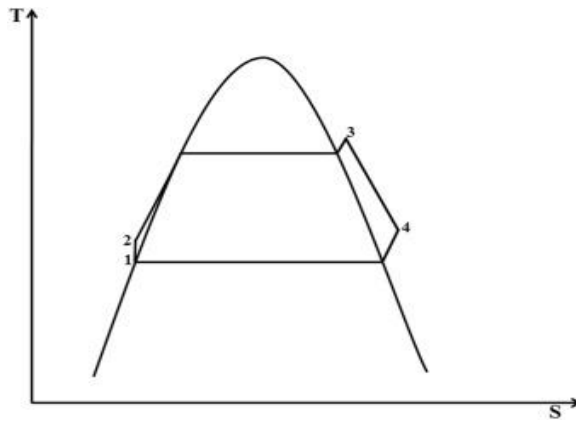


Figure 2.1: T-s diagram of the steam Rankine cycle. Point 1: pump inlet, point 2: evaporator inlet, point 3: turbine inlet, point 4: condenser inlet [18]

From point 1 to 2, the working fluids are pumped and sent to the solar collector by the pump. From point 2 to 3, the working fluids heat up and generate superheat vapor by the solar fields. From point 3 to 4, the saturated vapor enters the turbine to expands and generates electricity by a generator. From point 4 to 1, the exhausted vapor of the turbine is condensed to a low pressure and low temperature at saturated liquid and then pumped by the pump to the solar fields to be reheated. The cycle then repeats [19, 20].

The Rankine cycle is an organic Rankine cycle (ORC) or steam Rankine cycle, depending on the working fluid. The traditional Rankine cycle uses water as the working fluid and is extremely important since it contributes to the generation of 85 percent of all electric power used in the form of steam engines around the world [21, 22]. If the utilized heat sources are high temperature ($> 250\text{ }^{\circ}\text{C}$), the most suitable working fluid is water. At lower temperatures, the steam Rankine cycle is less efficient and more costly. Furthermore, solar Rankine cycles using water\steam as the working fluid, with high outputs $>1\text{ MW}$, are better than solar organic Rankine cycles [23, 24]. On the other hand, solar organic Rankine cycles utilize organic fluids, such as refrigerants and hydrocarbons, which are a better solution in the case of medium and

low heat sources due to the low boiling points of organic fluids [25, 26], but the outputs of these systems range from a few watts to several megawatts [27]. This kind of solar thermal system is being increasingly used in low outputs shafts systems [28, 29]. By the end of 2013, the installed capacity of the ORC power plants reached approximately 1,700 MW [30]. Since the 1970s, experimental and theoretical studies have been conducted regarding the solar organic cycles [31], while the history of using solar steam Rankine cycles to pump water in the 19th century has been reported in Paris [32]. However, both solar organic Rankine cycles and solar Rankine cycles have been widely applied in solar thermal energy systems, with high efficiency [33, 34].

There are several solar Rankine cycles which use steam as the working fluid around the world. Examples of this kind of solar plant that are currently in operations include solar thermal power plants with direct steam generation (DSG) in Southeast Asia Solarlite GmbH, located in Mecklenburg-West Pomerania, operated by Solarlite in Thailand, producing 5 MW electric power with a collector field area in excess of 45000 m² [35]. Another example is Planta Solar 10 - PS10 located in Sevilla Spain, operated by Abengoa, producing 11 MW of electricity, with a solar field area 75,000 m² [36]. Also, the same company has another Planta Solar 20 - PS20 in Spain, that generates 20 MW of electricity and has a solar field area 150,000m² [37]. On the other hand, solar organic Rankine cycles are currently in operation, such as a rural electrification project in Lesotho, Africa, that produces 3 kW of electric power, with a parabolic-trough collector field of area 75 m² [38]. Another example in Arizona, USA, produces 1 MW of electric power with a collector field area in excess of 10,000 m² [39, 40].

2.3 Control of Rankine cycle solar power systems

The level of energy obtained from the sun varies throughout the day. Under conditions of intense solar radiation, the received solar energy increases and decreases over time. Heat storage units with control strategies methods are fitted into the systems to prevent the interruption of the solar heat input [41]. However, because heat storage has a limited capacity for energy storage, the heat source temperature of the solar Rankine cycle system may fluctuate while in use. Since the solar Rankine cycle alone cannot stop the unstable trend, the model must include some advanced control methods to deal with the effect of heat source temperature variation, taking into account the total cost of the system. Since solar irradiation is a naturally transient condition, the design of the control system of a solar thermal power plant is vital for its proper operation. This is even more true for direct steam generation systems, where the presence of a two-phase flow inside the absorber tubes increases the magnitude of the transient phenomenon [42]. One of the condensed approaches takes the ideal solar field into account and simply adjusts the mass flow rate of the steam generation. A water level control is utilised in the power cycle system to lessen the thermal shock and, more significantly, preserve system security [43]. The basic, most crucial control method for a coal power plant's drum boiler, which is the model for the uniqueness of the power cycle system in the solar thermal power station, is the water level management. The primary control structure of the solar thermal power plant comprises the temperature control of the receiver and the water level control of the power cycle system [44, 45]. Some of control approaches for a solar steam Rankine cycle are as follows:

- Control of the outlet superheated steam pressure loop: A final separator is reached before living, superheated steam is delivered to the turbine. The valve

aperture can be changed to maintain a constant pressure. As a result, the solar field operates under a constant pressure mode;

- Control of the separator level loop: The liquid level of the medium separator is managed by the feed valve at the solar field's inlet. The steam quality is dependent on the feedwater flow rate under certain conditions, including a constant recirculation flow rate. The medium separator level is consequently controlled by changing the feedwater valve aperture. DNI (Direct Normal Irradiance) transients have a significant impact on this loop: if the irradiation declines, the steam generation declines and tank liquid level rises;
- Control of the final separator liquid level in the drum: it is utilized to overcome a high liquid level in the separator;
- Control of the outlet superheated steam temperature loop: A valve that injects feedwater into the final collector regulates the temperature. To lower the temperature, "fresh" water is fed into the highly heated steam and evaporates immediately;
- Control of the Feedwater valve pressure drop loop: The pressure drop across the feedwater valve is controlled by the feedwater pump; it is kept at a set value by regulating the rotational speed. Control is required since the feedwater tank pressure can change and affect the valve pressure drop;
- Recirculation control loop: The recirculation pump in the vaporizer is controlled to keep the flow rate constant. Irradiation transients on the receivers induce significant fluctuations in the pressure drop across the pump, necessitating this adjustment. Because a minimum flow rate is always guaranteed, this method of control makes the operation safer even though it excludes steam fraction control at the vaporizer.

The maximum electrical output of the Thai Solar One (TSE-1) solar thermal power plant is 5 MW. With 12 evaporator loops, seven superheater loops, and a central steam

separator, the solar field is run under the recirculation mode [46]. The facility is situated in an area with a difficult climate for CSP, and the DLR recently evaluated the performance of the vaporizer, providing a little information about management approach. Each loop has a valve at the input that regulates the mass flow. A feedforward control method is employed to calculate the mass flow setpoint using measurements of the sun beam irradiation. It is demonstrated that flow stability is guaranteed for days with stable solar beam irradiation and days with slightly transient solar beam irradiation, but also that the flow instability between the parallel loops and local overheating occurrences result from highly transient solar beam irradiation.

Puerto Errado 1 in Spain, a Fresnel. The SUPERNOVA collector was created with the intention of utilising steam that has been superheated to 450 °C [47] and operates under a recirculation mode. Focused area control (which is unique to Fresnel), steam mass control, and injection cooling control are utilised to regulate the outlet temperature. The latter reference demonstrates that, even on days with somewhat high irradiance transients, good temperature stability is attainable. The operation without mass flow control is also examined because the authors note that big power plants using steam turbines do not adjust to steam mass flow control through pressure variation. For a day with good irradiation conditions, the results show good temperature stability. A control approach for the cooling injection based on real-time measurements of irradiation and collector efficiency and energy balance is mentioned in the simulation study section of the work in question. Therefore, this is a feedforward control approach that also includes a feedback control element.

A recirculation architecture was used by the CNIM company to develop the Fresnel collector. The proposed system includes an auxiliary gas boiler between the superheater and the turbine to handle large solar beam irradiance drops, steam drum

control of the turbine inlet steam pressure (including flashing when the vaporizer flow declines), and injection cooling in the superheater to control the solar field outlet temperature. There are no additional specific details available regarding the control system.

2.4 Comparison between steam Rankine cycle and organic Rankine cycle

In this subsection, the solar steam Rankine cycle and solar ORC systems will be compared in terms of their overall performance, costs, complexity of the system components, size of the components, operating conditions and working medium. A number of advantages of solar steam Rankine cycles over solar ORC systems can be summarized by the following points [48]:

- In a solar steam Rankine cycle, water is used as the working fluid. Water has many features, such as being environmentally-friendly, cheap, non-toxic, non-flammable and chemically stable, as well as being abundantly available. Another feature is that water has low viscosity, thereby leading to pressure drops and lower friction losses in the system piping and heat exchangers;
- A solar steam Rankine cycle has low consumption of the power pump relative to the turbine output power;
- The thermal efficiency of a solar steam Rankine cycle is higher than that of a solar ORC system;
- A solar steam Rankine cycle is preferable to a solar ORC system when the output of power is more than 1 MW.

However, there are several unique advantages of ORC power systems, including their ability to exploit low-grade heat sources with temperatures as low as 60-170 °C [49]. According to [50] and [51], further to the solar ORC system's features mentioned

above, other advantages of using solar ORC rather than a steam Rankine cycle can be listed under the following points:

- In small-scale systems which generate up to 1 MW, solar ORC is more economic than a solar steam Rankine cycle;
- In general, a solar ORC system operates at low pressure and low temperature compared with a solar steam Rankine cycle. This decreases the equipment complexity and cost by drastically reducing the heat strain and safety measures regarding the system components;
- Unlike steam cycles, ORCs do not require any specific water treatment. It is also unnecessary to utilise a deaerator to prevent the corrosion of system metallic equipment caused by dissolved oxygen in the boiler feedwater;
- The major goal of the superheating process is to prevent the production of moist droplets at the end of the expansion process, which could cause turbine blade degradation. Due to the low heat transfer characteristics of the gaseous/vapour medium, this necessitates the use of a superheater with a relatively large heat transfer surface area, which adds to the power system's cost. After the expansion process, the majority of the organic fluids remain in the dry area; hence, superheating is unnecessary;
- With higher mass flow rates and higher isentropic efficiency, organic fluids result in lower enthalpy losses across the turbine. Furthermore, a reduced enthalpy loss results in a lower turbine turning speed, allowing a direct electric generator drive without the requirement of a gearbox;
- Solar steam Rankine cycles have a lower condensation pressure than atmospheric pressure, often less than 0.1 bar, which increases the risk of air leaking into the steam condenser. As a result, technical challenges arise, raising the pressure

and, as a result, lowering the system performance. Organic fluids with low critical temperatures, on the other hand, allow the ORC to operate at condenser pressures that are higher than ambient;

- The pressure ratio across the steam turbine is very high in solar steam Rankine cycles, so multi-stage turbines are utilized. In a solar ORC system, this pressure ratio is low and simple single-stage turbines are commonly used;
- Finally, a solar ORC system is a simple configuration, with low maintenance, high safety and very flexible operations.

The bottom line is that both types of cycles have their own features based on the proposed project. A solar ORC system is preferable compared to steam a Rankine cycle if low-temperature heat sources are available, and a small power output is required. Meanwhile, a solar steam Rankine cycle is preferable when generating above 1 MW.

2.5 Comparison between the direct and indirect steam generation solar Rankine cycles

In solar thermal plant systems, the solar thermal energy can be absorbed directly by the solar collectors' fields to heat up the working fluids in the Rankine cycle, which is usually named direct steam generation (DSG), or indirectly, using a heat transfer fluid (HTF). In this system, there are two cycles: one for a heat transfer fluid solar cycle and the second for a steam power generation cycle. Figure 2.2 shows the two technologies, HTF technology and DSG technology, respectively.

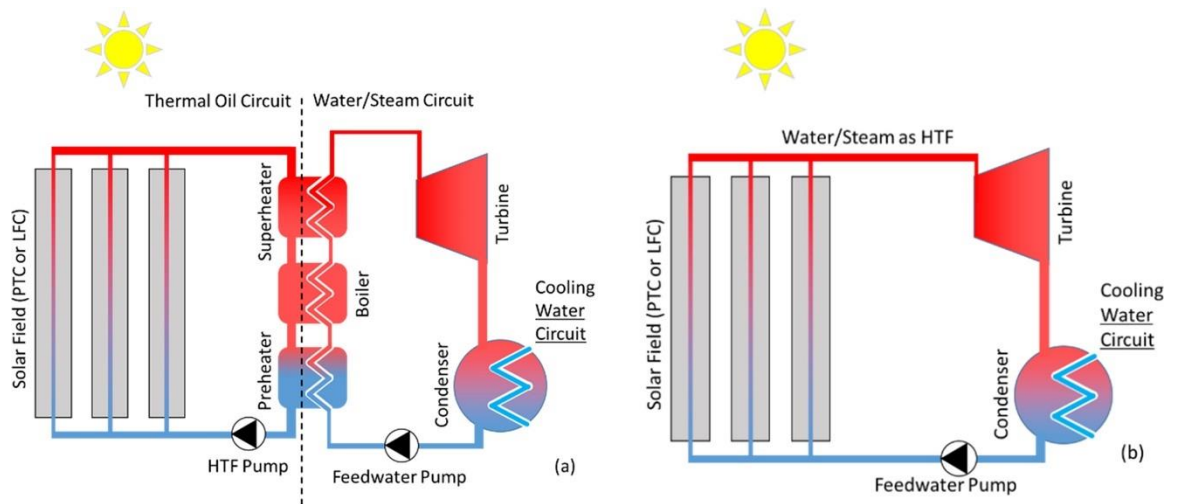


Figure 2.2: Solar Rankine cycle a) an indirect steam generation system and b) a direct steam generation system [51]

The DSG system has been used in some projects, such as the INDITEP (Integration of Direct Steam Generation technology for electricity production) project [52], Planta Solar 10 - PS10 is located in Sevilla, Spain, operated by Abengoa, and produces 11 MW of electricity, with a solar field area of 75,000 m² [53] and Puerto Errado 1 Thermo solar Power Plant 1.4 MW project. All of these projects based in direct steam generation technology. However, this type of solar Rankine cycle is still under development [53].

Most of the solar Rankine cycles under investigation use different HTFs to heat the SRC working fluids [54]. The indirect steam generation systems were commercially introduced in 1984 [55]. Different HTFs have been used in solar Rankine cycles. Water, molten salt and Therminol-VP1 are the most widely used fluids [56,57]. Several advantages have been reported related to the application of an indirect steam generation system, such as its very easy coupling with a thermal energy storage system if needed [58] and avoidance of the mechanical constraints when heating up the working fluids in the solar collectors [59]. At a given pinch-point temperature (ΔT_{PP}), the lower the mass flow rate of the heat transfer fluids, the higher their inlet

temperature in the solar fields [60]. On the other hand, the higher the mass flow rate of the HTF, the higher its outlet temperature. Thus, that it is difficult to lower the average temperature of the HTF, resulting in the disadvantage of operating the solar collector at high temperatures, which is less efficient compared to operating at lower temperatures [61]. Also, indirect steam generation systems need extra components, such as a heat transfer fluid pump, in the solar fields cycle and heat exchangers systems. All of these extra components require a high initial investment, which will reduce the thermal efficiency and net power output of the system [62,63].

In solar direct steam generation systems, there are some advantages compared with indirect steam generation systems, if they work under the same boundary conditions, such as low thermal inertia, and high efficiency with high evaporation temperatures can be obtained with high-power output from the system. On the other hand, solar direct steam generation systems have several disadvantages, including the unsteady power output when the solar irradiance faces cloudy conditions, as the fluctuating solar irradiance will affect the flow rate that enters the steam turbine, resulting in fluctuations in the turbine speed if there is no control in the system [63,64]. A comparison between the solar direct and indirect steam generation systems is summarized in Table 2.1.

Table 2.1: Comparison between solar direct and indirect steam generation systems [63]

	Direct steam generation method	Indirect steam generation method
Environmental risks	Low	High
Solar field size	Smaller	Larger
Technology stage	Demonstrative	Commercial
Control effort	Higher	Lower

Leaks	Higher	Lower
Temperature gradients	Higher	High
Configuration	Simple	Complex
Efficiency	Higher, promising	Medium, limited
Heat exchangers	No	Yes
Process Stability	Less stable	Stable

In general, more attention has been paid to solar direct steam generation systems as a promising system due to the aforementioned advantages with the aim of overcoming its disadvantages.

In the solar direct steam generation system (DSG), there are three regions in the solar field: the preheating, boiling and superheating sections. The solar field's input section, which receives subcooled feedwater, is used for preheating. It oversees the preheating of the water till it reaches the saturation temperature. The saturated water subsequently enters the boiling section which, depending on the plant and operational mode, produces saturated steam or a saturated water/steam mixture. The last element of the solar field is the superheating section, which is responsible for superheating the steam until it is safe for it to enter the turbine [65]. The once-trough, injection, and recirculation modes are the three basic operational modes for a solar direct steam generation system [66]. The nominal operational condition is frequently associated with the once-trough (see Figure 2.3a). The solar field is provided with subcooled water in this mode, and the preheating, boiling, and superheating all happen in one direct path. Although the principle is straightforward, in practice, this method has proven to be difficult and costly, and has performed poorly in tests [67]. Zarza et al. [68] tested the injection mode, which involves injecting feedwater into many spots

along the solar field in order to control the outflow steam (see Figure 2.3b). Due to challenges related to maintaining control, this complicated, expensive technique produced an unsatisfactory result and was discontinued in subsequent operations. The recirculation mode divides the solar field into two sections, one for preheating and boiling and the other for superheating the steam (see Figure 2.3c). A water-steam separator connects the two sections, with connections to the preheating and boiling section's outflow and the superheating section's input. Saturated water from the separator's bottom is pumped to the preheating and boiling section's input. In the correct circumstances, saturated steam from the upper part of the separator is channelled to the superheating section, and subsequently to the turbine. The superheating section has feedwater injections to control the superheated steam outlet state. Due to the larger recirculation of water from the separator to the solar field, this last operational mode is highly adjustable, but also has higher parasitic loads [69]. Eduardo et al. [70] reviewed the findings of numerous investigations on solar direct steam generation control systems and techniques in linear concentrating solar power plants. The results of the simulation and experimental studies, as well as operational plant control systems, were examined. The recirculation mode is the most widely researched operational mode for the solar direct steam generation system. Although it contains more components, particularly the steam separator, this mode is fairly easy to manage, so limited research to date has been conducted on the once-through mode. However, the cost savings for the solar direct steam generation system make it potentially appealing. Due to the large number of sensors, the injection mode proved to be quite an expensive way to accomplish the needed control capabilities. The Thai Solar One has been in the recirculation mode in Thailand since 2011. Flow stability is demonstrated for days with steady and mildly transient DNI, while extremely transient

DNI causes flow instability between parallel loops and local overheating occurrences. The recirculating mode is employed in the LFC plants of Puerto Errado 1 and 2, and the output temperature is controlled using focused area control, steam mass flow, and injection cooling control. Even on days with rather high irradiance transients, good temperature stability is attained.

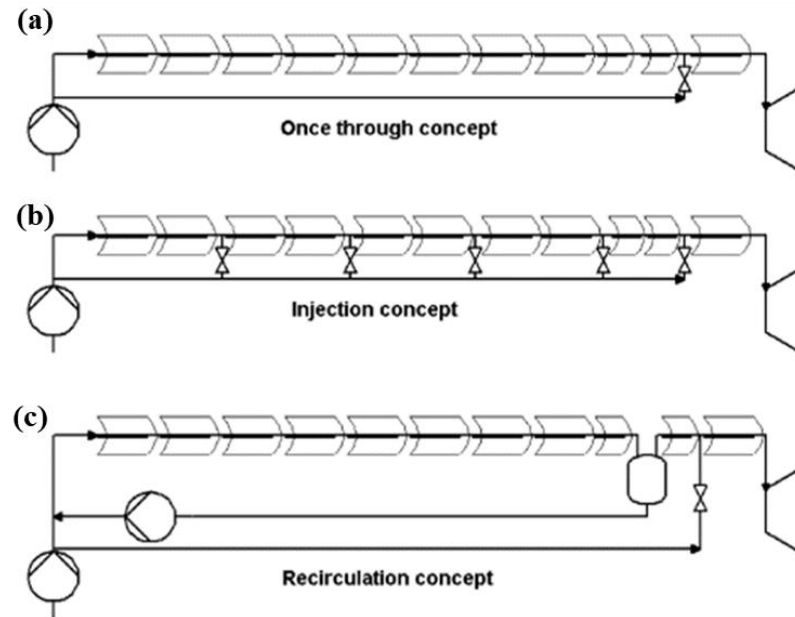


Figure 2.3: The schematic diagram of water\steam recirculation a) once-through, b) injection mode and c) recirculation mode of the DSG process [68]

2.6 Options for solar collectors that concentrate the sun's rays

In Concentrated solar power systems, there are four commercially available types of concentrating solar collector technologies: Parabolic trough collector (PTC), Linear Fresnel reflector (LFR), Parabolic dish collector (PDC), and Central receiver (CR) [71]. Fresnel lenses, composed of glass for marine purposes, were invented by the scientist Fresnel in 1822. Giovanni Francia built a Fresnel collector prototype with two-axis tracking for a steam-generating system [71]. A focal cavity receiver and an array of long parallel curved/flat mirrors/reflectors make up a solar linear Fresnel reflector (see Figure 2.4). The reflectors have a single-axis tracking mechanism (East to West) that concentrates sunlight onto the receiver plane at 30-60 times its normal

intensity [71]. To absorb the optimum amount of solar energy, the receiver is situated at the focal distance. A bank of parallel high-pressure boiler grade tubes is coated with a selective coating and contained in an insulated inverted trapezoidal chamber to make up the receiver. To allow focused sun radiation and prevent cavity heat loss, the cavity aperture is covered with a glass shield. The concentrated solar energy is transmitted to a heat transfer fluid (HTF), such as water, thermal oil, molten salt, and other thermic fluids. The HTF may be heated to temperatures ranging from 100-300 °C [71]. Compared with other concentrating technologies, such as the central receiver, parabolic dish collector and parabolic trough collector, the thermal efficiency of the Linear Fresnel reflector is relatively lower due to the higher convective heat loss in the receiver tube.

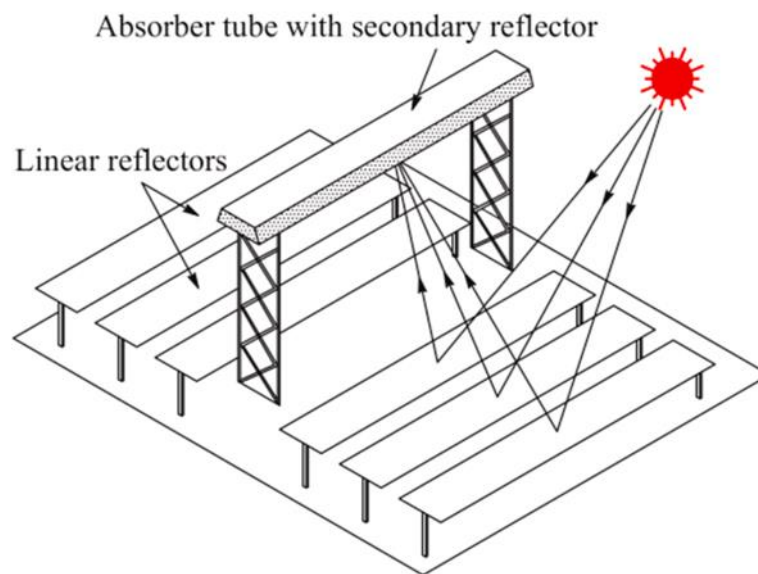


Figure 2.4: A schematic of the Linear Fresnel reflector concentrator [71]

One of the most widely tested methods for process heating and power generation systems is the parabolic trough solar collector. The solar energy is focused on a line receiver at the parabola's centre via a parabolic-shaped linear reflector in the parabolic

trough collector (see Figure 2.5). Compared to other receivers, the straight-line tube receiver offers reduced pressure drops. To focus solar energy onto the receiver, the parabolic trough collector is typically placed along a North-South axis, while the concentrator follows the sun's east-west passage. The sun's radiation can be concentrated by the parabolic trough concentrator at 30 to 100 times its usual intensity [71, 72]. To lessen convective and radiative heat loss, the receiver is composed of a high thermal conductivity metallic tubing and surrounded by an evacuated transparent glass tube. By passing HTF through the receiver, the concentrated solar energy can be captured. In order to produce high-pressure superheated steam for process heating and power generation applications, the HTF is heated as it circulates through the receivers and returns to a number of heat exchangers in a central area [72, 73]. The parabolic trough collector may produce fluid temperatures of 100-400 °C [74].

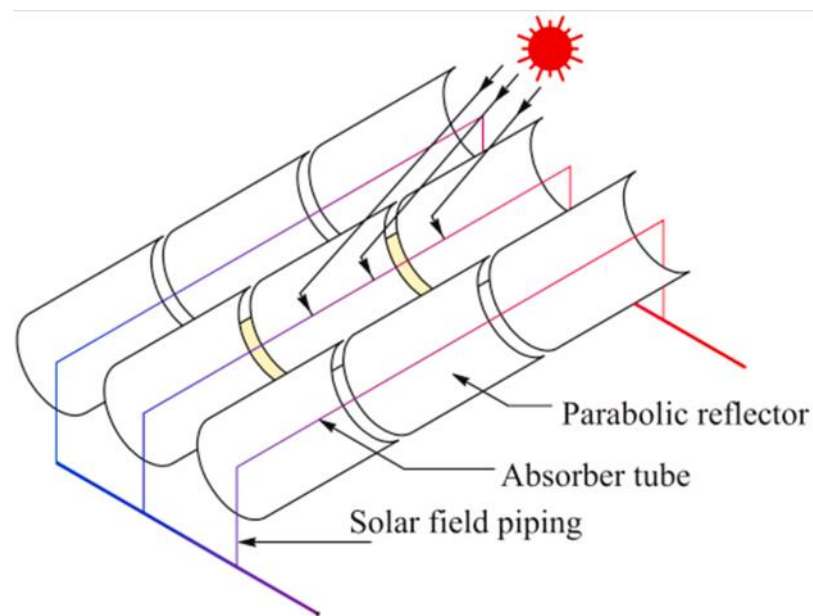


Figure 2.5: A schematic of the Parabolic trough concentrator [71]

In 1971, power generation from the central receiver was invented [71]. By concentrating solar energy onto a central receiver, such as a power tower, high-temperature heat is produced from sunlight. The technique concentrates solar radiation onto a tall tower in the centre of the heliostat field using a large number of flat, two-axis trackable mirrors known as heliostats. Figure 2.6 depicts a central receiver tower solar collector. The energy can be concentrated up to 1,500 times more than the solar radiation that was accidental [71]. With higher operating temperatures, the central receiver plant's solar to electrical conversion efficiency might increase up to 28% and the receiver transfers the concentrated heat energy that it has absorbed to heat transfer fluids such as liquid sodium, air, water, and molten salts [75]. This type of solar collector can be installed on land surfaces or even on hills or small mountains [71].

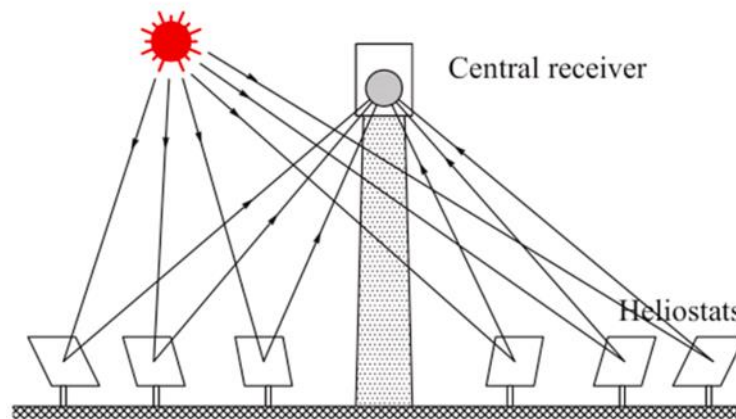


Figure 2.6: A schematic of the Central receiver tower concentrator [71]

An array of mirrors in the shape of a parabolic dish comprises a parabolic dish collector, which uses a two-axis tracking mechanism to direct solar radiation onto a receiver at the parabola's focal point. To remove heat from the receiver, the heat transfer fluid is pumped throughout the receiver. The parabolic dish collector's concentration ratio ranges from 300 to 3000 [71]. The parabolic dish collector is effective at producing temperatures between 500 and 1200 °C [71]. A regular receiver

is used in place of a modified cavity receiver with a secondary reflector to enhance the receiver's performance even further. The cavity receivers come in a variety of sizes and forms. Because the aperture area of a modified cavity receiver is less than the absorber surface area, the thermal losses in the receiver are seen to be minimised. In general, this kind of solar collector is modern and still developing, although it has demonstrated its value. A schematic of a solar parabolic dish collector concentrator is shown in Figure 2.7.

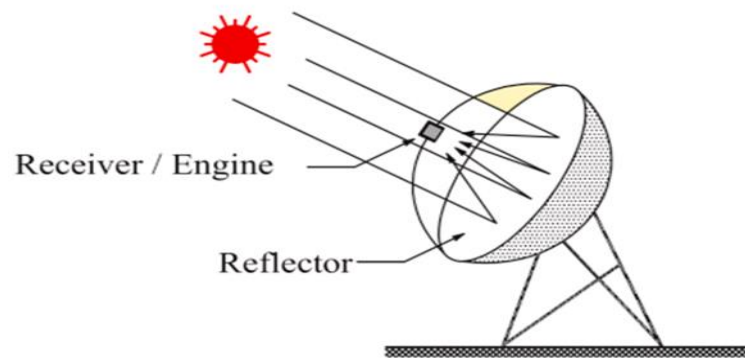


Figure 2.7: A schematic of a solar parabolic dish collector concentrator [71]

2.7 Storage methods for direct steam generation

The availability of low-cost storage devices is critical for solar thermal power plants to expand their market penetration. For parabolic trough systems, many concepts for integrated heat storage units are currently being developed. These ideas can be categorised based on their storage capacity. The goal of buffer storage systems is to compensate for the quick and unpredictable transients in solar insolation. Short reaction times and rapid discharge rates are typical requirements for these buffer storage systems, whereas their thermal capacity is typically in the 5-10-minute range. Buffer storage systems are designed to protect power plant components from the

impacts of the high thermal transients caused by abrupt changes in solar insolation. The classification of various concepts for sensible heat storage used for solar thermal power plants is shown in Figure 2.8 below.

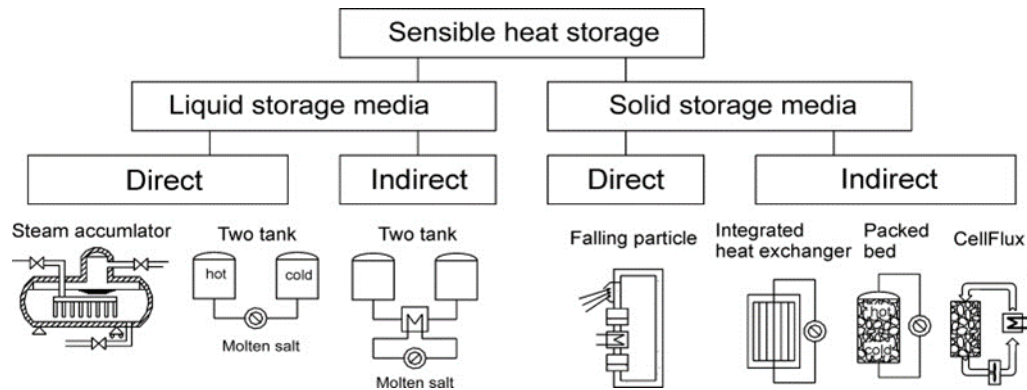


Figure 2.8: Classification of various concepts for sensible heat storage [78]

The addition of buffer storage capacity simplifies power plant operation and provides a longer response time for backup systems, which are designed to compensate for longer periods of reduced insolation. Steam accumulators are ideal for supplying saturated steam at pressures up to 100 bars, which is ideal for buffer storage in solar steam systems. The use of steam accumulators in solar thermal power plants can benefit from decades of practical experience with similar storage devices in fossil-fired power plants. In the process sector, these storage devices are utilised to balance steam demand and generation. Over the course of more than 60 years on the Berlin Island grid, a power plant with a capacity of 50 MW electric power was operated with steam accumulators with a capacity of 67 MWh storage [76, 77, 78].

2.7.1 The fundamentals of steam accumulators

Because of the poor volumetric energy density, the direct storage of saturated or superheated steam in pressure vessels is not cost-effective. Steam accumulators, on the other hand, store sensible heat in pressurised saturated liquid water [78]. They

benefit from liquid water's great volumetric storage capacity for sensible heat as well as its high specific heat capacity. Figure 2.9a below shows the layout of a steam accumulator used for a solar thermal power plant. Steam is created by reducing the saturated liquid's pressure during discharge. High discharge rates are feasible since water is employed as both a storage and a working medium, although the capacity is restricted by the pressure vessel's volume.

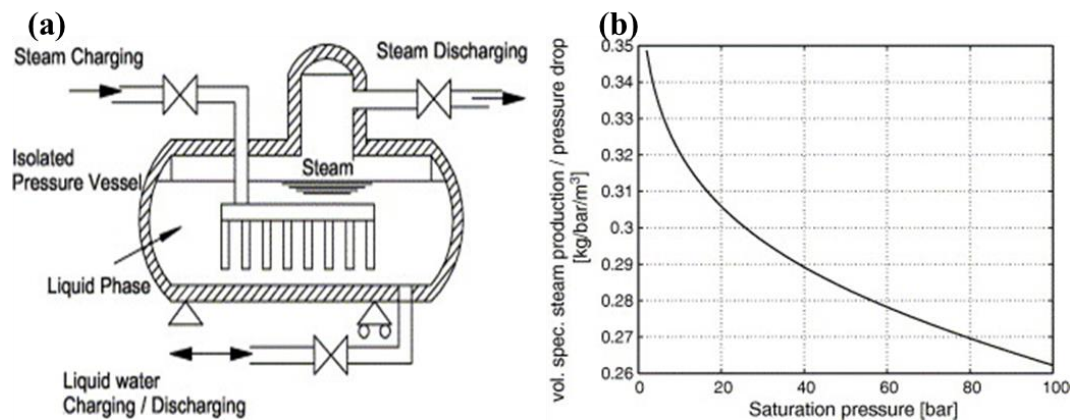


Figure 2.9: Schematic of a) a cross-sectional steam accumulator and b) the specific volume value of steam provided by a steam accumulator dependent on initial pressure [78]

The fluctuation of the saturation temperature caused by the pressure drop during discharge has a significant impact on the volume specific thermal energy density; typical values are in the range of 20-30 kWh/m³. During the charging process, the temperature of the liquid water is increased either by the condensation of superheated steam or by supplying saturated liquid water into the system. When superheated steam is employed, the pressure in the vessel rises during the charging operation, while the liquid storage mass varies relatively slightly. The pressure remains constant while saturated liquid is supplied into the steam accumulator. A heat exchanger is incorporated into the liquid volume in order to charge a steam accumulator indirectly.

The heat exchanger's medium does not have to be water, as heat from a lower-pressure source can be utilised instead. Saturated steam escapes the storage vessel during the discharge procedure in sliding pressure systems (also known as Ruths storage systems). The amount of steam created from 1 m³ of liquid water during a pressure drop of 1 bar is shown in Figure 2.9b as a function of the initial saturation pressure. In Figure 2.9b, the range of steam production corresponds to a thermal energy of 0.26 kW h/bar/m³ to 0.19 kW h/bar/m³. The saturation pressure has a variety of effects on the rate of steam production: at higher pressures, the pressure-dependent change in saturation temperature reduces, resulting in a lower change in sensible energy in the liquid water volume. With increasing pressure, the density of saturated liquid water decreases, resulting in a lower volumetric energy density. At greater pressures, however, the mass specific energy required for evaporation is lower. Figure 2.9b shows that, for a given pressure drop, steam accumulators operating at a lower pressure level provide more saturated steam; however, if the pressure at the end of the discharge process is fixed, a storage system starting at a higher initial pressure level may provide more steam due to the larger pressure variation.

2.7.2 Steam accumulators coupled with a solar concentrating power plant

Direct steam generation (DSG) in parabolic trough collectors has been demonstrated in recent years, with temperatures reaching 400°C, at pressures of up to 100 bars [79]. The use of steam accumulators in this technology is a simple method for adding buffer storage capacity to solar-thermal power plants that use this collection idea. Figure 2.10 shows the flow sheet for a solar thermal power plant with an integrated steam accumulator. If the collector field's steam production exceeds the turbine's demand, the excess steam is condensed in the steam accumulator. During periods of low insolation, steam is drawn from the steam accumulator to supplement or replace the

solar field's steam output. Operating solar steam systems in the so-called recirculation mode, in which only a portion of the water delivered into the collector field is evaporated, is advantageous. Wet steam passes through a separator after exiting the evaporation section. The liquid phase is pumped back to the entrance of the evaporator section of the collector field, while the gas phase is supplied into a saturated steam turbine or superheated in another section of the collecting field. When it comes to steam accumulators, combining storages and phase separation into a single component is a cost-effective solution. In Figure 2.10b, wet steam from the evaporation section of the collector field enters the steam accumulator/separator of a DSG-power plant operating in recirculation mode.

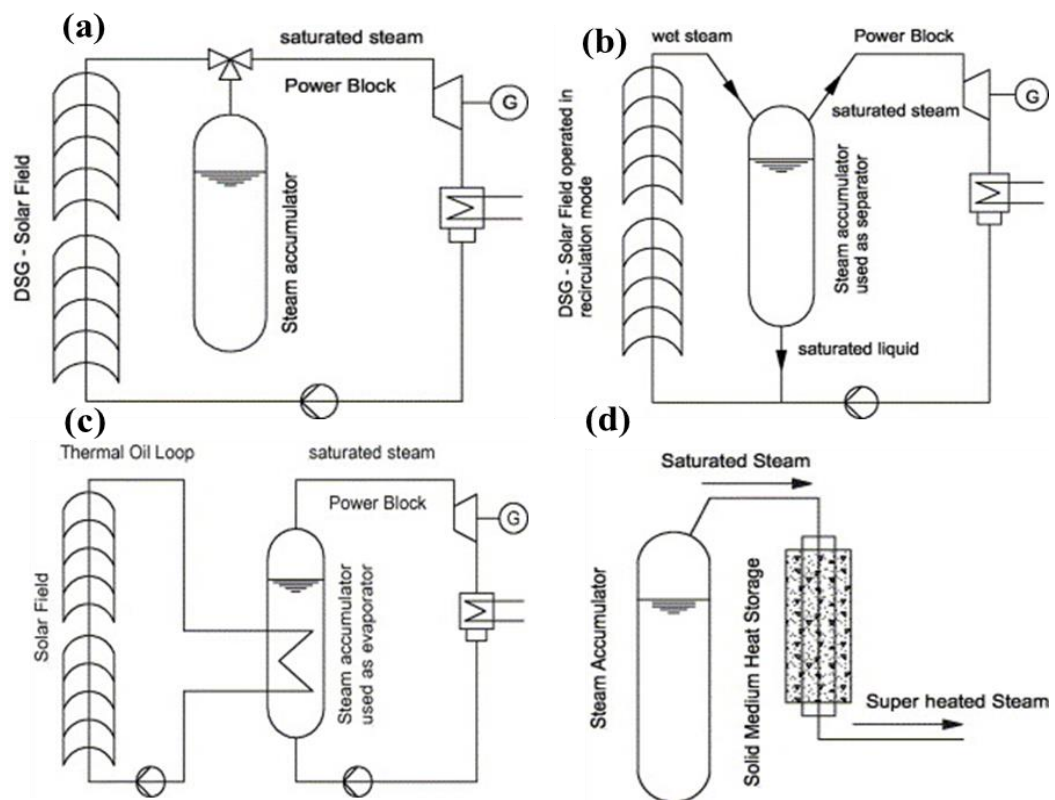


Figure 2.10: Scheme of different methods for the use of a steam vessel a) steam accumulator integrated with a solar thermal power plant, b) used as a phase separator at the exit of the collector field, c) Indirect charging method for a steam accumulator which is also used as a heat exchanger; and d) integration of a steam accumulator with another sensible storage system to supply a superheated steam to the system [78]

The phases are split here. The amount of saturated liquid leaving the vessel during the charging operation is less than the amount entering the volume through the wet steam feed line. Because the pressure in the steam accumulator/separator remains constant, no special charging equipment for steam condensation inside the volume is required. By lowering the pressure in the volume and withdrawing steam, the steam accumulator is discharged. This concept demands almost no additional hardware, the modifications mainly affect the operation strategy of the power plant. This approach requires essentially little additional hardware, and the only changes are to the power plant's operation strategy. Steam accumulators can also be used in solar collectors that employ thermal oil as the heat transfer medium, as long as steam is generated for power generation or as process steam (see Figure 2.10c). Combining the steam accumulator with an existing component, once again, is a cost-effective way to add buffer storage capacity. A heat exchanger is required to transport thermal energy from the thermal oil to the water/steam loop in this type of solar thermal system; the pressurised liquid water volume in this heat exchanger can be employed for energy storage. The thermal oil flowing in a tube registers in the liquid volume charges the steam accumulator indirectly. The charge status of the steam accumulator changes as the liquid water level changes.

The steam accumulator, which also serves as a heat exchanger, is charged indirectly. Saturated steam is provided by steam accumulators. If superheated steam is required, a second storage system can be connected to the steam accumulator's outlet. For sensible heat storage, candidate concepts for this second storage system use either solid storage media, like concrete, or non-pressurized liquids, like molten salt. In most solar steam applications, superheating requires only a tiny portion of the total thermal energy. Because superheating steam requires only 10-15 percent of the thermal energy

in a solar-thermal power plant with parabolic troughs [79], the contribution of the extra storage system is substantially lower than that of the steam accumulator (see Figure 2.10d).

2.7.3 Concepts of constant pressure

During discharge, the steam pressure in a Ruths-type steam accumulator lowers. A storage system that provides steam at a constant pressure is beneficial in some applications. A separate flash evaporator (see Figure 2.11a) can be used to avoid a pressure drop: the saturated liquid water from the steam accumulator is depressurized externally, cold water is fed into the storage vessel to keep the water level constant, the mixing of hot and cold water must be minimised, and the thermal stress caused by filling the pressure vessel with cold water must be considered.

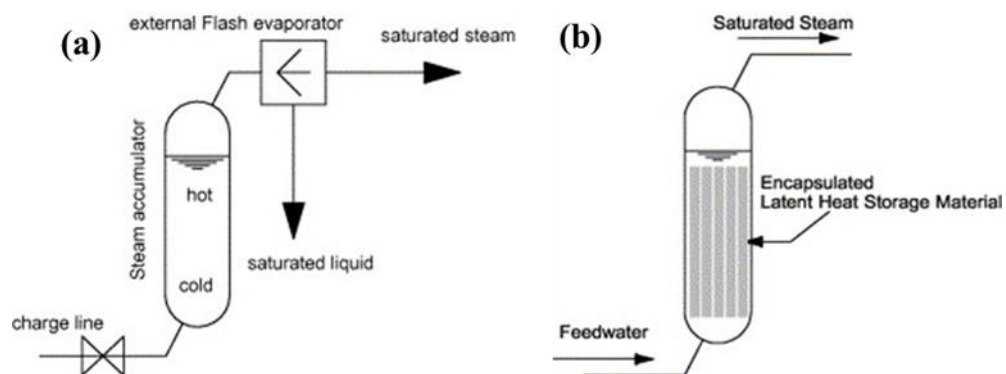


Figure 2. 11: Layout of the integrated steam accumulator; a) external depressurization in the vessel and constant pressure concept and b) integrated of pressure steam vessel and phase change material water [78]

The integration of phase change material (PCM) into the storage tank as a partial replacement for liquid water is another possibility for constant pressure storage (see Figure 2.11b). Isothermal energy storage uses the thermal energy associated with the phase change between liquid and solid states. Due to the low thermal conductivity of PCMs, thin layers of this material are required to guarantee an adequate heat transfer

rate. The encapsulation of PCM in small containers put inside the liquid volume is one option for meeting this demand. PCMs are appealing not only because they eliminate the thermo-mechanical stress caused by temperature transients, but also because their characteristic volume-specific storage capacity is in the range of roughly 100 kWh/m³. The inclusion of PCM helps to enhance the storage capacity of a particular pressure vessel as compared to the similar value for water (20–30 kWh/m³).

2.7.4 Solar direct steam generation with steam accumulators

The DISS test facility for Direct Steam Generation in Parabolic Troughs is shown in Figure 2.12. The pressure range for this test facility is 30 to 100 bars, with steam outlet temperatures up to 400 °C, and the highest thermal output is 2 MW water [78]. A separator drum separates the evaporation and superheating sections of the collector. The separator can be used as a steam accumulator by storing hot water in this pressure vessel. When the amount of liquid water exiting the drum is less than the mass flow of liquid phase delivered by the solar collector to the separator during the charging process, the water level in the drum rises. This design eliminates the need for special steam condensation equipment and allows charging at a constant pressure. In Figure 2.12, a volume specific saturated steam production of roughly 90 kg/m³ during the discharge, assuming a pressure decrease from 100 to 55 bars. Because the drum's saturated liquid water capacity is approximately 4 m³, roughly 360 kg of saturated steam can be produced. Because the collector's nominal mass flow rate is 1.2 kg/s, the steam accumulator may replace the collectors for around five minutes.

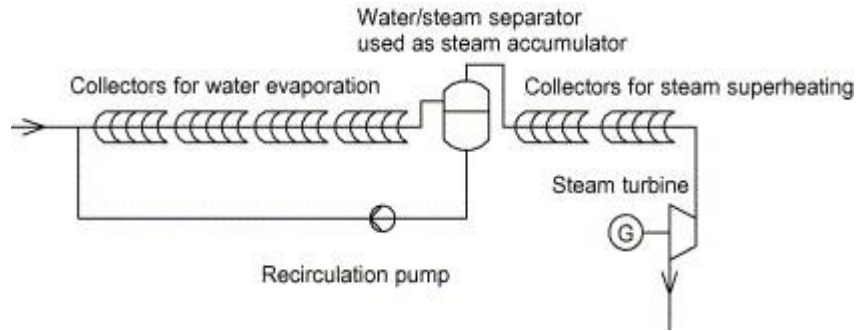


Figure 2.12: Sketch of the use of as steam accumulator as a separator in the DISS test loop project water [78]

Some of operational direct steam generation solar power plants integrated with steam accumulators around the world water are list in Table 2.2.

Table 2.2: Some of operational direct steam generation solar power plants integrated with steam accumulators around the world water [79]

Project name	Type	Location	Storage	Steam pressure (MPa)	Solar field temperature (°C)	Power capacity (MW)
Thai Solar Energy 1	PTC	Huai Kachao, Thailand	None	3.0	Inlet:201 Outlet:330	5.00
Puerto Errado 1	LFC	Calasparra, Spain	1h Steam Accumulator	5.4	Inlet:140 Outlet:270	1.4
Planta Solar 10	Power Tower	Sevilla, Spain	1h Steam Accumulator	4.0	Inlet:250 Outlet: 170	11.0
Planta Solar 20	Power Tower	Sevilla, Spain	2h Steam Accumulator	4.5	Inlet:250 Outlet:300	20.0
Puerto Errado 2	LFC	Calasparra, Spain	0.5h Steam Accumulator	5.5	Inlet:140 Outlet:270	30

Khi Solar One	Power Tower	Uppington, South Africa	2h Steam Accumulator	11	Outlet: 530	50.0
ISEGS	Power Tower	Primm, United States	fossil backup	16.0	Inlet: 248 Outlet: 565	377.0

2.8 Solar Rankine cycle configurations

Different types of solar Rankine cycle configurations and combined solar thermal power systems are discussed and presented in this section. Several configurations and amendments are made in order to increase the efficiency of the solar Rankine cycle. These amendments can be applied in the Rankine cycle itself, including internal heat exchangers that are placed between the exit line of the turbine and the outline of the pump. It is called a regenerator or recuperator. This heat exchanger is used to absorb the exhaust heat from the outlet of the turbine and preheat the high pressure cold stream entering the solar fields. Figure 2.13 shows a solar Rankine cycle with a recuperator system [80].

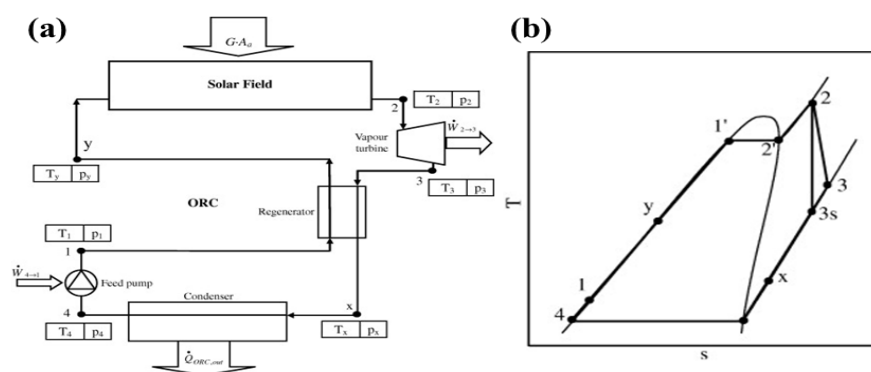


Figure 2.13: Schematics view of a) a solar Rankine cycle with recuperator system, b) a T-s diagram of the system [80]

Another method that has been used to increase the solar Rankine cycle efficiency is a reheating system, which works to reheat the exhaust stream of a high-pressure turbine

and then enters a low-pressure turbine to produce extra power and increase efficiency. The reheating system is used only for solar steam Rankine cycles. Figure 2.14 shows a solar Rankine cycle with a reheating system [81].

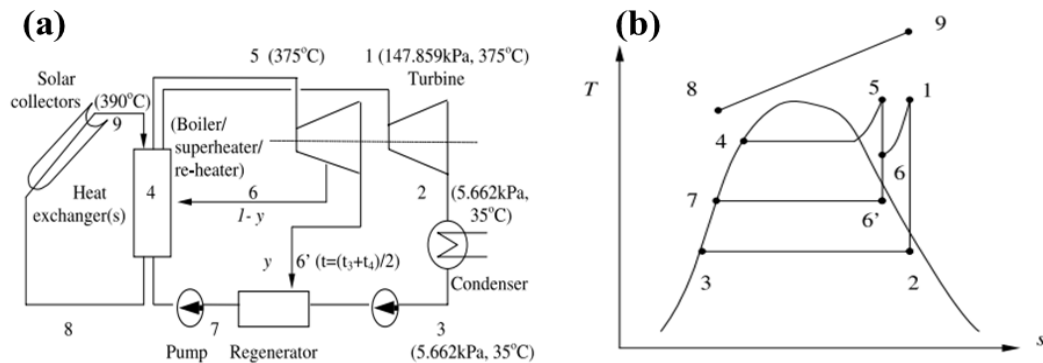


Figure 2.14: Schematic representation of a) a solar Rankine cycle with a reheating system, and b) a T-s diagram of the system [81]

Pengcheng Li et al. [82] proposed and optimised a cascade steam-organic Rankine cycle power generation system powered by a hybrid system of liquefied natural gas and solar thermal energy. In this proposed system, solar collectors are used to vaporise the working fluid in the evaporator in the top cycle (I). The heat generated by cycle I speeds up the bottom cycle's working fluid evaporation (II). Cycle II's cold source is LNG. Investigated are the impacts of working fluids on the operation of the system. Examples include evacuated tube collectors (ETC) and flat plate collectors (FPC). It has been shown that equivalent efficiency is the best indicator of how well solar collectors and LNG work together. Using isopentane/R125 and ETC, a maximum equivalent efficiency of 5.99 percent can be attained. Each expander in the cascade ORC has a volume ratio that is significantly lower than that of the single-stage ORC, thereby simplifying the design and production of the expanders. Moreover, the hybrid

system has a faster return on solar power investment than a solar ORC (see Figure 2.15).

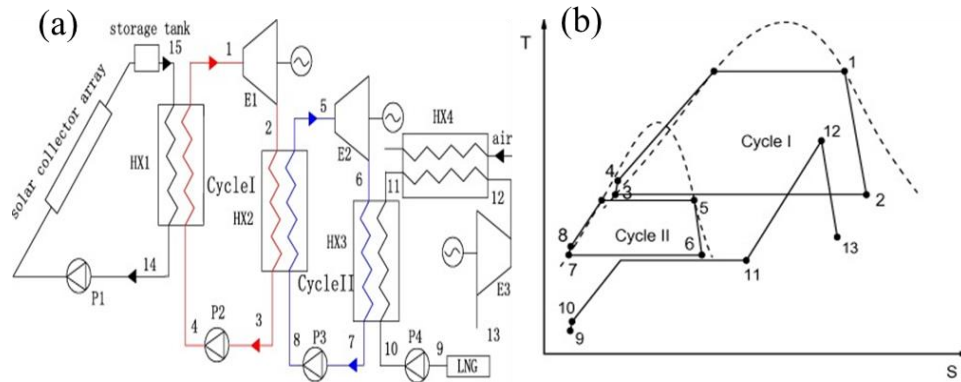


Figure 2.15: Schematic diagram of a) the cascade system utilizing solar energy and LNG, and b) a T-s diagram of the proposal system [82]

Jing Li et al. [83] explored a new direct vapour generation solar organic Rankine cycle system's efficiency and the effect of working fluids on the system. In this novel system, a PCM storage unit is used as a heat storage system to maintain the stability of the power output of the system. Also, this system avoids using a heat transfer cycle compared with conventional solar ORC systems. Furthermore, the system reacts well to fluctuations in solar radiation. A feasibility study of the system was performed. The effects of different working fluids on system performance were also studied. The findings show that, as the fluid critical temperature rises, the collector efficiency often declines, while the ORC and system efficiencies rise. The ORC, collector, and overall thermal efficiencies of R236fa are 10.59, 56.14, and 5.08 percent correspondingly, at an evaporation temperature of 120 °C and solar radiation of 800 W/m², while the same values for benzene are 12.5, 52.58, and 6.57 percent. At decreased solar radiation, the difference in collector efficiencies between R236fa and benzene increases. The latent and sensible heat of the working fluid are highly connected with the heat collection. R123 has the best overall performance of the fluids and appears to be suitable for use in the suggested system in the near future. Figure 2.16 shows the proposed system:

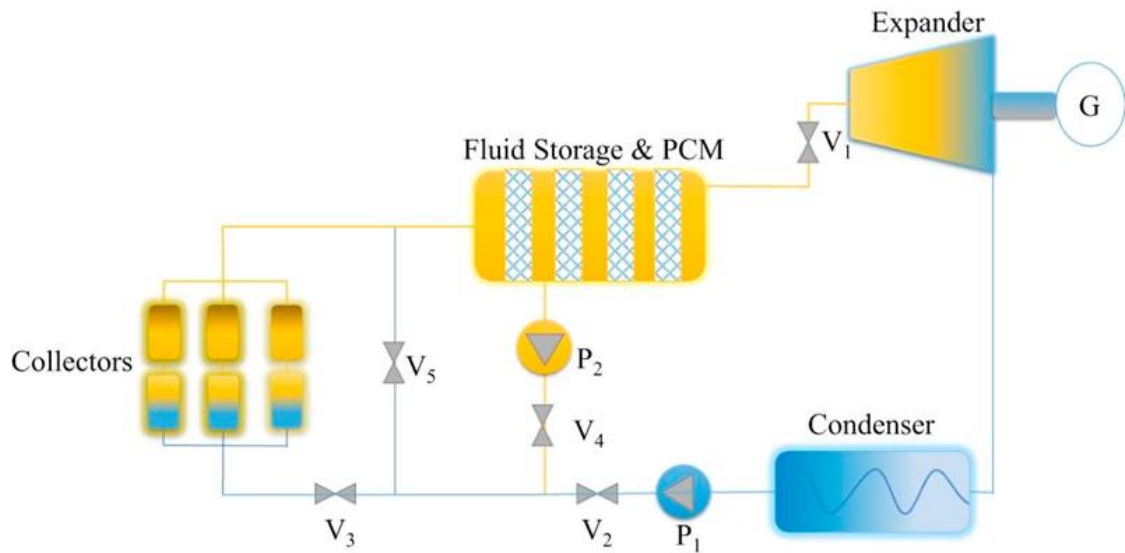


Figure 2.16: Layout of the proposed solar ORC system integrated with a PCM unit as heat storage [83]

A combined novel solar steam electricity generation system using a cascade cycle is proposed based on PTC solar collectors by Jing Li et al. [84], as shown in Figure 2.17. The top cycle is the steam Rankine cycle, and the bottom cycle is the organic Rankine cycle. A screw expander (SE) has been used in the top steam cycle. It is characterized by good applicability in terms of power conversion in steam-liquid mixture states. In this system, steam is generated directly in the solar field and expands in the screw expander. The heat released by the steam condensation section is utilized to push the bottom ORC. The system has a liquid storage tank, a PCM unit and a separator. The superheated steam section was avoided in this proposed system, and it can work under moderate operating pressure and temperatures with low technical requirements with regard to heat collection. Ten ORC fluids have been studied in this system. There are four examples of hot/cold side temperatures: 473/313 K, 473/293 K, 523/313 K, and 523/293 K. The findings show that a pressure ratio (PR) that matches the SE built-in PR cannot be obtained at the ORC evaporation temperature, which corresponds to the theoretical maximum solar power efficiency. For a more efficient system and simpler

ORC turbine, off-design operation of the SE is advised. For the proposed system, an efficiency of 13.68-16.62 percent can be attained.

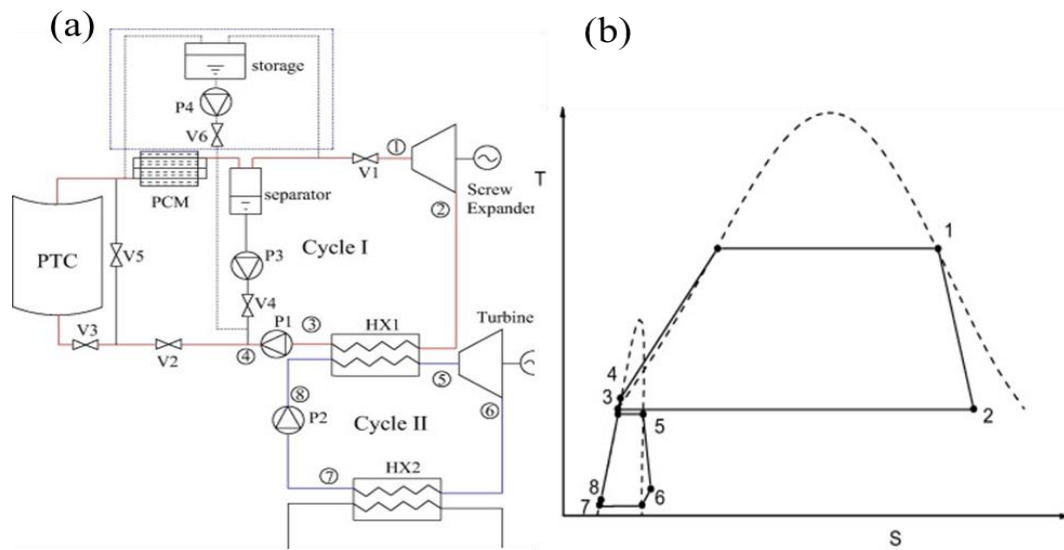


Figure 2.17: Schematic diagram of a) solar steam cascade system integrated with a PCM unit and water storage tank, and b) a T-s diagram of the proposal system [84]

A comparison study and economic investigation of direct and indirect steam generation solar power, using a cascade Rankine cycle system and integrated with water as the thermal storage fluid, was analysed by Jing Li et al. [85]. The study placed special emphasis on the thermo-economic evaluation and the heat storage unit. The optimal operating pressure and temperatures of the system were clarified based on steam screw expander characteristics. The temperature-dependent permissible stress of steam accumulators was analysed, and the total cost was investigated. Also, a comparison was made between the two systems using thermal oil at a power output of 1 MW and a storage time of 6.5 hours. The outcomes of this study show that the direct steam generation system is superior from a thermodynamic and financial standpoint. For the solar-ORC to operate at its optimum level of solar thermal power efficiency, the hot side temperature typically does not rise above 250 °C. The greatest efficiency

at 750 W/m^2 radiation is 14.3%, which corresponds to a temperature of about 240°C . Pressure vessels cost 2.55 million RMB in materials. The optimal temperature for the indirect steam generation system is around 230°C , or roughly 13.2 percent, and the anticipated oil cost is 7.92 million RMB. It is advised that screw expander-driven solar electricity generation systems should employ steam accumulators, based on this study (see Figure 2.18).

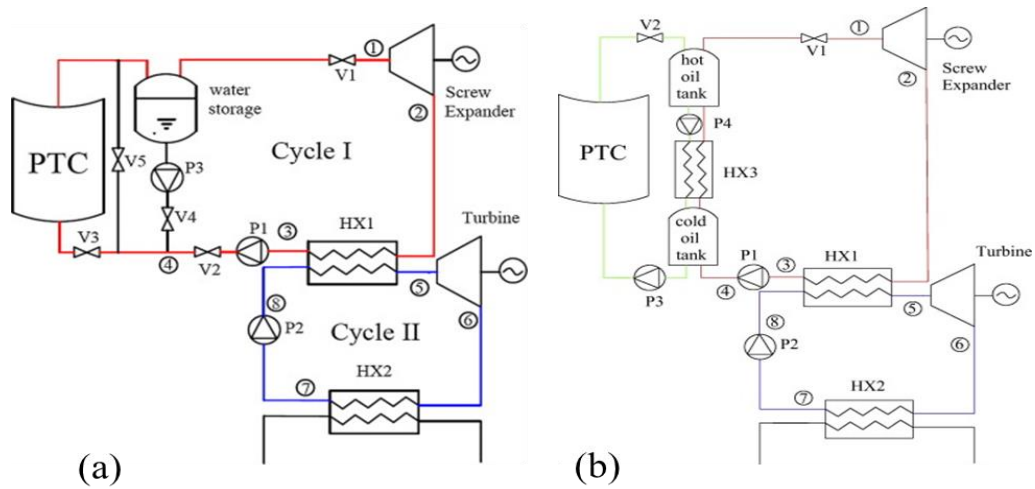


Figure 2.18: Schematic diagram of a solar-driven cascade Rankine cycle a) direct system and b) indirect system [85]

Similarly, a solar-powered cascade Rankine cycle system with respect to the characteristics of a steam screw expander has been modelled and optimized by Pengcheng Li et al. [86], in whose research an approximate steam screw expander over-expansion model was established, which reveals variations in screw expander isentropic efficiency with changing operating pressure ratio. The system's overall, off-design behaviour is then modelled. Investigated was the effectiveness of solar electricity under various circumstances. System optimization was carried out. The model system is shown in Figure 2.19.

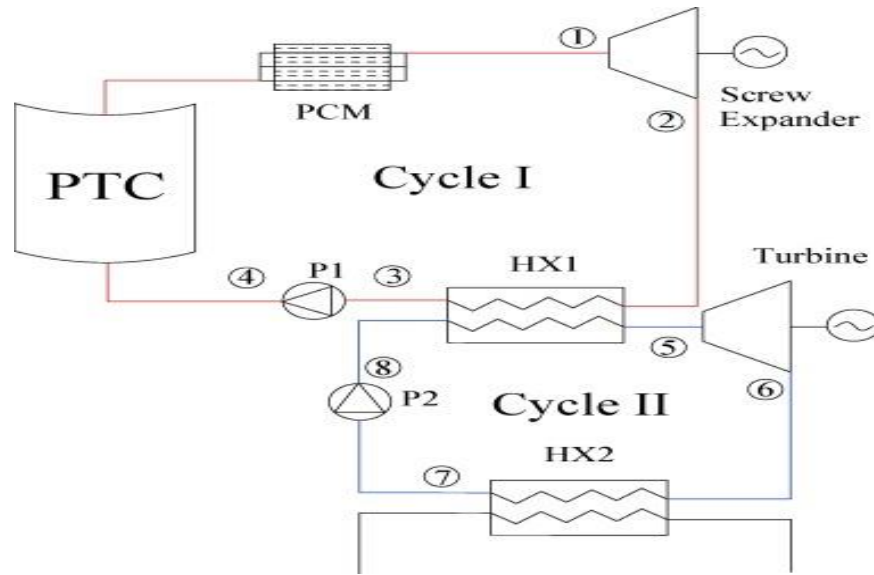


Figure 2.19: A solar-driven cascade Rankine cycle system with a PCM storage unit [86]

According to the findings of this study, the ideal hot side temperature is between 499 and 543 K when the beam solar radiation is between 600 and 800 W/m². In the case of 5.0 screw expanders, a maximum solar thermal power efficiency of 13.74–14.45% can be attained. The screw expander's efficient part-load behaviour limits the impact reduced on power conversion and, when it is 3.5, a maximum efficiency of 13.12–15.11% is attainable. Annual optimization is also put into practise in Phoenix, Sacramento, Cape Town, Canberra, Lhasa, and Barcelona. The annual variation in solar energy output ranges from 201.8 kWh/m² to 347 kWh/m².

Jing Li et al. [87] proposed and modelled a direct steam generation solar power system based on a cascade steam-organic Rankine cycle integrated with two-stage accumulators powered by parabolic trough collectors (see Figure 2.20). There are two storage tanks in this novel system, a hot water tank and a cold water tank, for the charging and discharging modes. When solar radiation is present, the high temperature tank facilitates smooth power conversion and stores thermal energy. To increase the storage capacity, a low temperature tank is used. The system's heat discharge mode is one of its most impressive aspects. To power the Solar-ORC, heat is first released

through the vaporisation of water in the high temperature tank. The high temperature tank temperature can drop by a maximum of roughly 20 °C. Heat is then solely used to power the organic Rankine cycle as water at lower temperature flows from the high temperature tank to the low temperature tank (ORC). For a global analysis of the system's performance across seven regions, nine organic fluids are used and investigated in this system. The results show that the low temperature tank can increase the storage capacity from 1.0 MWhe to 8.4 MWhe when compared to a traditional single-stage tank. In Phoenix and Lhasa, the corresponding payback period for the additional accumulator and collectors is projected to be roughly five years. The low temperature tank has the ability to significantly increase the system's overall cost-effectiveness.

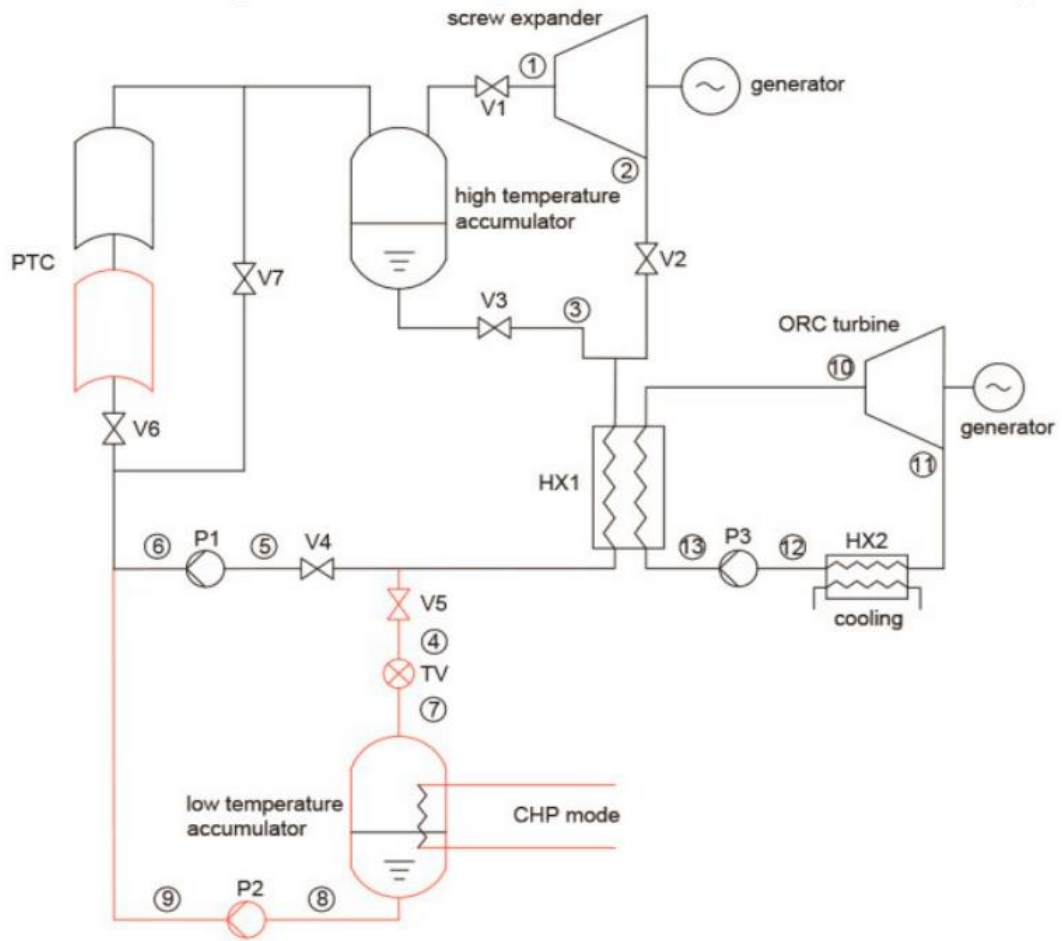


Figure 2.20: Solar-driven cascade steam-organic Rankine cycle integrated with two-stage accumulators powered by parabolic trough collectors [87]

Similar research, but with a different working mode, was carried out by Jing Li et al. [88], who explored a novel approach to the thermal storage of direct steam generation solar power systems through a two-step heat discharge process (see Figure 2.21). This research aimed to solve the problem related to direct steam generation solar power whereby the steam turbine suffers from inefficient off-design operation during heat discharge, thereby restricting the development of this technology. Two heat storage tanks were used in the system and two steam-organic Rankine cycles were integrated together. One feature of the system is the unique two-step heat discharge. In order to power the steam Rankine and ORC, heat is first released through water vaporisation

in a high temperature accumulator (HTA), which causes a drop in the hot temperature accumulator of about 30 °C. The heat is only used to power the ORC as water at a lower temperature is transferred from the high temperature accumulator to a low temperature accumulator through a heat exchanger. The temperature of the water falls by another 130-190 °C. The system's fundamentals are demonstrated. With an accumulator volume of 2,500 m³, a comparison with the traditional DSG system is made at a nominal power of 10 MW. The system's thermodynamic performance was investigated. It is expected that using the second stage heat discharge will have a similar payback period.

According to the results, the second step heat discharge can, in the majority of situations, enhance the storage capacity by 460 percent in less than five years. Overall, the suggested solution increases the DSG system's cost-effectiveness, but there are some limitations of their work that need to be evaluated in the future. One of the limitations is that the investment in the turbines, solar field, and high temperature accumulator must be taken into account when analysing the thermo-economic performance of the system. Another gap is that wet steam turbine mathematical models could be developed in order to improve the ORC evaporation temperature. Finally, there is no control strategy for the novel system, especially regarding the steam cycle and dynamic response behaviour for each component of the system for more than one day.

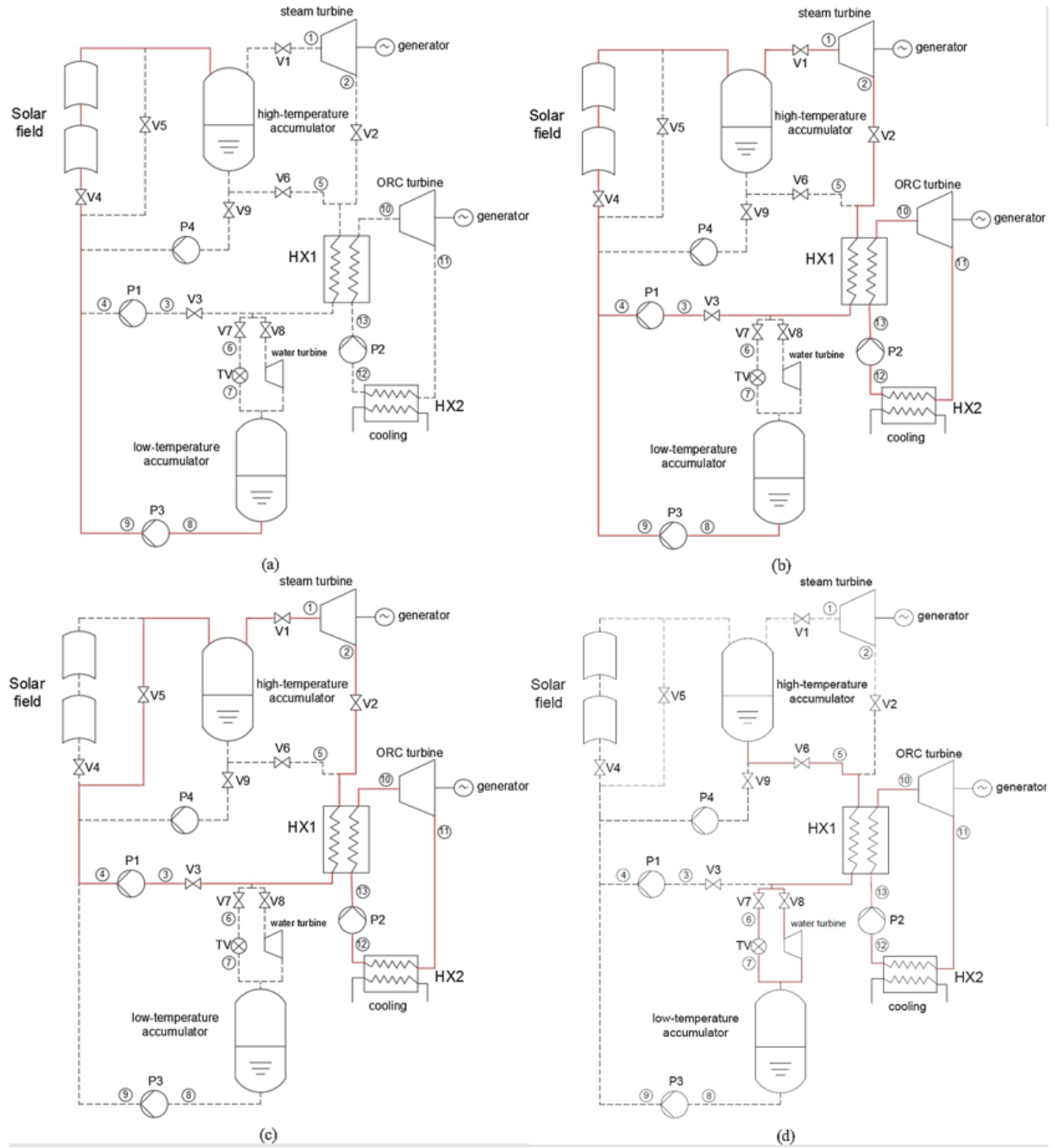


Figure 2.21: Diagrams of the different operating modes: (a) Mode 1 charging, (b) Mode 2 two-cycle work based on solar irradiation, (c) Mode 3 two-cycle work based on discharging and (d) Mode 4 only ORC work [88]

In the same approach, an innovative solar thermal power generation system using a cascade steam-organic cycle and two-stage tanks has recently been designed and proposed by Guangtao Gao et al. [89]. The author focused on the design of the steam condensation temperature in the heat exchanger placed between two cycles in this

system, and found that the new proposed system offers a higher heat storage capacity compared with direct steam generation solar power with a single heat storage tank. Because it influences the Solar-ORC efficiency during normal operations and the power conversion of the bottom organic Rankine cycle during the novel heat discharge process, the steam condensation temperature in the proposed system is an essential parameter. With respect to a novel indication, the equivalent heat-to-power efficiency, the current work develops a methodology for designing the steam condensation temperature. The efficiencies in the various operation modes are reconciled to create the thermal efficiency equivalent. Regarding the steam condensation temperature, the impacts of the main steam temperature, Baumann factor, mass of storage water, and ORC working fluid are examined. The thermal efficiency equivalent, according to the results, is a more accurate indicator than the solar-thermal ORC's efficiency. The ideal temperature for steam condensation, which corresponds to the maximum, is typically greater than the thermal efficiency equivalent, calculated using the solar-maximum ORC's thermal efficiency. As the main steam temperature (inlet-temperature turbine), Baumann factor, and mass of storage water decrease, the steam condensation temperature decreases as well. The thermal efficiency equivalent increases with an increment in the main steam temperature (inlet-temperature turbine) and a decrease in the Baumann factor and mass of storage water. In this study, Pentane working fluid achieved better results in ORC fluid than benzene and R245fa. The optimum steam condensation temperature and thermal efficiency equivalent of pentane are 139-190 °C and 20.93%-24.24%, respectively, provided that the main steam temperature (inlet-temperature turbine) ranges between 250 °C and 270 °C, the Baumann factor varies from 0.5 to 1.5, and the mass of storage water changes from 500 to 1,500 tonns.

In 2020, Pengcheng Li et al. [90] was published similar work where the system included a regenerator. In this research, a thorough performance comparison of DSG systems with and without regenerators was made. In organic Rankine cycle (ORC) systems, a Regenerator is typically worth discussing. Not only does the addition of a regenerator affect the ORC efficiency, RC-ORC efficiency, and heat exchanger area, but it also impacts on net profit, heat storage capacity, discharge length, and efficiency. The findings of the research show that, for a given power output, the regenerator reduces the aperture area, particularly for MM, R365mfc, and pentane, as a result of the increased ORC, RC-ORC, and discharge efficiencies as well as a decrease in the heat input. Depending on the ORC fluids, the discharge time is shortened by 0.01 to 1.78 hours, and the net profits 4.196.48 million USD is displayed by use R365mfc, followed by MM and pentane. Conversely, it is ineffective for benzene (5.61-4.31 million USD).

In 2021, Guangtao Gao et al. [91] suggested using a revolutionary system to produce concentrated solar power. It has three characteristics: a two-tank water storage system, a cascade organic Rankine cycle (CORC), and two-phase water/steam as the heat transfer fluid (see Figure 2.22). Water is replaced with an organic fluid for power conversion, and steam is created in the solar field and condensed in a hot tank. The system can handle the challenge posed by damp steam turbines and provides a smooth cycle operation using a resilient control method. The technical requirement is reduced since it can operate with steam or water of lower purity when those substances are simply used for heat transfer. Evaluations of the heat discharge process and thermodynamic performance under standard operating conditions are made. Examined are the effects of the storage tank size and ORC working fluid.

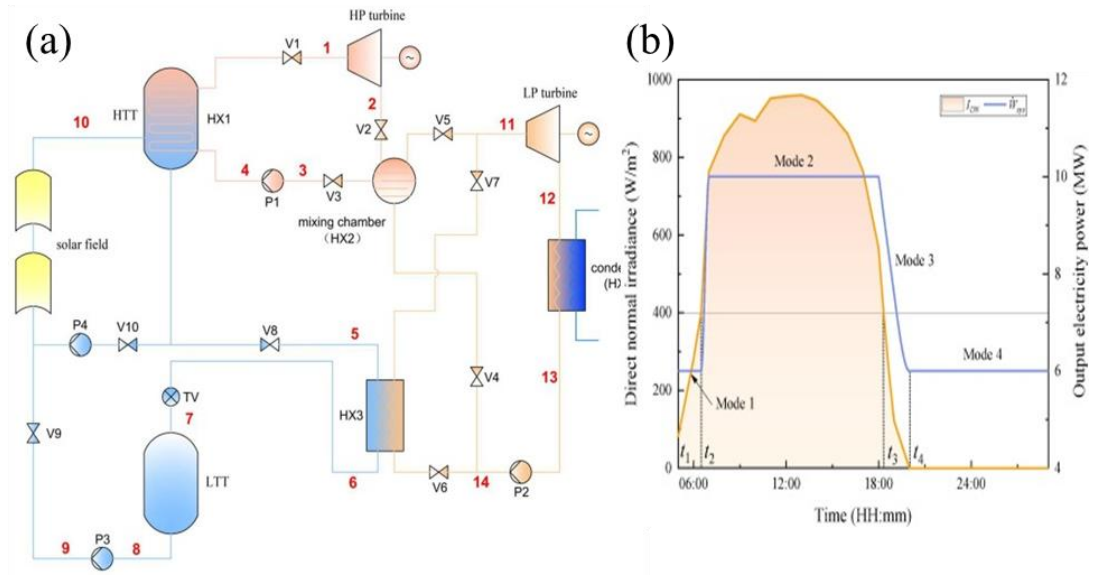


Figure 2.22: Schematic diagram of a) the proposed solar power system using a cascade organic Rankine cycle and two-tank water/steam storage, and b) the different modes' power output [91]

According to the results of the study, the thermal efficiency of the charge and discharge operations, storage capacity, and overall performance are all strongly influenced by the temperature of the mixing chamber. One can achieve a CORC efficiency of roughly 27.4%. Depending on the ORC fluid and storage tank volume, the equivalent heat-to-power efficiency can range from 13.35 to 18.81 percent. The new system requires less technical expertise to generate power and offers an efficiency level that is comparable to a traditional direct steam generation system.

From an energy and exergo-economic perspective, a cascade power generating system that consists of a partial evaporation Rankine cycle, an organic Rankine cycle, a parabolic trough solar collector, and a storage tank was researched and optimised by Hamed Habibi et al. [92]. In order to make it easier to couple the partial evaporation Rankine cycle with parabolic trough solar collectors and the storage tank while operating close to the necessary temperature of 300 °C, and prevent the production of superheated vapour in the upper Rankine cycle, a screw expander has been used in place of a normal steam turbine in the partial evaporation Rankine cycle (see Figure

2.23). The pressure limit at the expander's inlet (4 MPa), which corresponds to a steam saturation temperature of 250 °C, makes this arrangement particularly suitable. Toluene and four other organic fluids have been investigated, and parametric analysis was applied to the impact of the solar irradiance levels, total solar collection area, partial evaporation Rankine cycle and condensation temperatures and the quality of the inlet steam expander on the thermo-economic performance of the proposed system as well as multi-objective optimization by using an algorithm tool considering the net power output, efficiency and capital cost rate as objective functions.

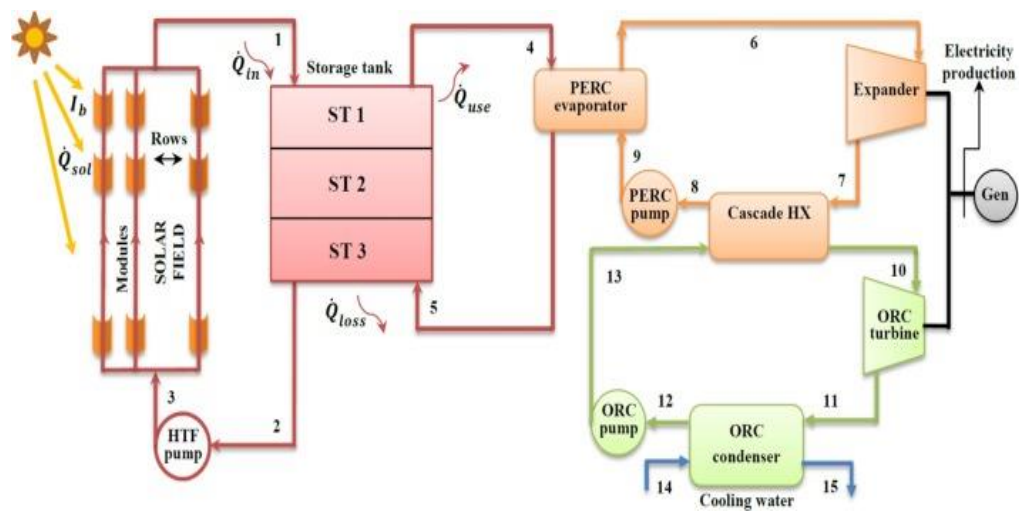


Figure 2.23: A Solar-driven cascade Rankine cycle system with thermal oil cycle integrated with a sensible heat storage tank [92]

The result of the study shows that, as the optimum design, the proposed solar-driven cascade Rankine cycles powered by 5540 m² of parabolic trough collectors' area and a storage tank volume of 184.7 m³ are able to generate 782 kW of power output. The exergy efficiency of the proposed system was 18.61%. The cost rate was 228 \$·h⁻¹. Compared to the base case cycle, the optimum design case can improve the net power output, exergy efficiency and unit cost of electricity of the system to 65%, 2.9%, and 27.26%, respectively. Moreover, ORC generated 440.6 kW, which is higher than the partial evaporation Rankine cycle's 349.2 kW.

The exergetic, energetic and economic analysis of the two-stage organic Rankine cycle, combined with the parabolic trough collectors and PCM as the storage unit and an auxiliary heater, was modelled and evaluated annually [93]. The effects of the three pairs of ORC fluids, which were R134a, R500, and R1234yf on the solar fraction and efficiency of the desired system were investigated (see Figure 2.24). The findings of this study show that the pair of ORC fluids R245fa/R500 had higher annual energy and exergy efficiencies, with 12.76% and 13.87%, respectively. According to the findings, parabolic trough collectors was the primary cause of energy destruction. Additionally, the condenser's energy destruction was significantly lower than in typical systems since the top ORC's excess heat was utilised to power the bottom ORC. The sensitivity analysis also revealed that the NPV is least sensitive to the PTC cost and most sensitive to the costs of the energy carriers. In general, the combination of R245fa/R500 and dimethylol propionic acid performed best in terms of energy, efficiency, and cost.

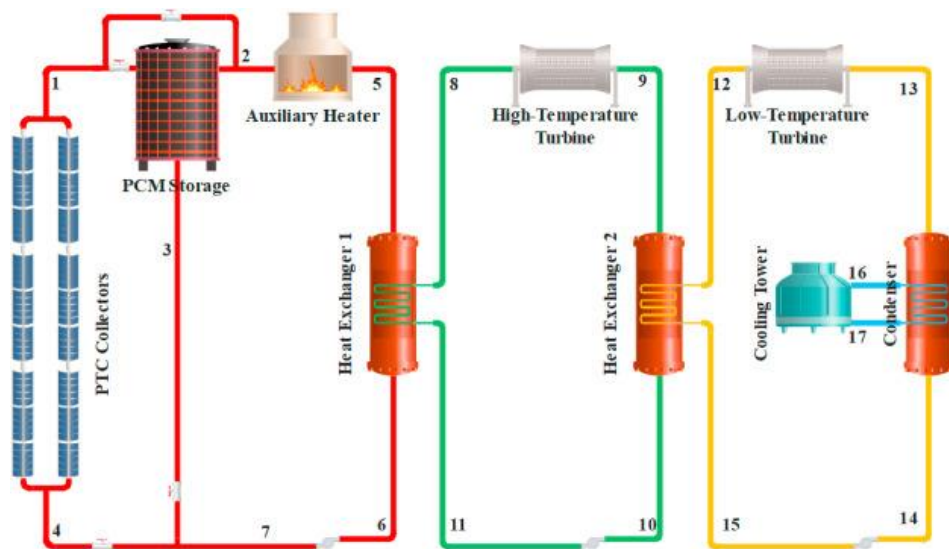


Figure 2.24: Schematic of the cascade organic Rankine system integrated with PTC, a phase change material tank and an auxiliary heater [93]

2.9 Dynamic modelling simulation research on solar Rankine cycle

Dynamic modelling simulation research on solar Rankine cycle began in the 1980s. These studies have focused on own different parts of the solar Rankine cycle. Using APROS and Simulink, Yu Q et al. [94] performed a dynamic simulation of a novel thermal oil-heated steam generation system for a solar thermal power plant developed by Balcke-Duerr. The steam generation system's rated operation and start-up phases were modelled. The simulation findings showed a boiling crisis and an economizer phenomena. However, they did not take into account how variations in solar irradiation would affect the output of the steam generation system in their paper. Soares et al. [95] studied a dynamic multi-configuration model of a solar steam Rankine cycle system by using ThermoSysPro library. However, in their paper the detailed modelling method of the solar boiler and superheater regions was not given.

In the MATLAB/Simulink environment, Hossin K et al. [50] created a dynamic simulator for an integrated solar combined-cycle plant and presented a model predictive control (MPC) approach. While developing a dynamic model of a solar steam generator, the long-term solar irradiance disturbance was not taken into account. Li X et al. [96] simulated a model of a steam generator system in a solar steam Rankine cycle plant by using Thermo-Power library software. But the detailed modelling method was not provided in this study either. APROS was used by Xu E et al. [97] to run dynamic simulations of a 1-MWe parabolic trough solar power plant on clear, partly cloudy, and heavily cloudy days. A steam generator model is included in the plant model, although the precise modelling approach has not yet been disclosed. Other studies have concentrated on the direct steam generation system [63, 69] In conclusion, the dynamic modelling of solar field regions of steam Rankine cycles has received little attention in the literature. Furthermore, to the best of our knowledge,

the literature has not yet fully addressed two features of dynamic solar Rankine cycle simulation. First, the impact of changing heat sources, notably solar irradiation, on the performance of the steam Rankine cycle has not been adequately taken into account. Second, a thorough modelling strategy has not been offered. On the basis of currently available simulation tools, previous research has created dynamic models of the solar Rankine cycle with precise architectures; however, the specifics of the modelling techniques used have not been disclosed. It is important to present a generic modelling approach in detail for a specific direct steam generation system in a solar Rankine cycle in order to simplify the dynamic performance study. Table 2.3 presents a list of some dynamic modelling studies of the solar Rankine cycle.

Table 2.3: List of some dynamic modelling studies of solar Rankine cycle

Study	configuration	Application	Software
[45]	Basic cycle	Solar Rankine cycle	MATLAB/Simulink
[47]	Basic cycle	Solar steam Rankine cycle	MATLAB/Simulink
[50]	Basic cycle	Solar Rankine cycle	MATLAB/Simulink
[53]	Basic cycle	Solar Rankine cycle	MATLAB/Simulink
[94]	Basic cycle	Solar Rankine cycle	MATLAB/Simulink
[98]	Basic cycle	Solar steam Rankine cycle	MATLAB/Simulink
[99]	Two cycles	Solar steam Rankine cycle and oil cycle	STAR-90 platform

2.10 Research gaps

Direct steam generation solar systems are considered a promising option nowadays. Researchers have paid more attention to how these systems are good used, particularly in countries with very high solar radiation. Based on previous literature review studies related to direct steam generation solar power systems, there were many lack knowledges and scientific gaps that must be covered through this thesis, as follows:

- ❖ Although there are many studies on the integration of the steam accumulator with the solar steam Rankine cycle to generate electricity, up to date there is no study regarding the integration of the steam tank with the solar steam Rankine cycle and control the steam flow rate to meet the electricity demand of users. It is considered a very interesting topic due to the large availability of water and it is cheap compared with ORC working fluids system. Furthermore, there is no research done regarding with the size effect of steam accumulator on the power output of a direct steam generation system. Also, most studies focus on short analysis time simulations. However, the transient performance of solar-DSG in comprehensive consideration of the off-design behaviour of the thermal storage unit, turbine, pump and heat exchangers has not been reported yet. It is still needed to clarify how flexible a solar-DSG system can operate and how it can fulfil the consumers' peak demand.
- ❖ There are only a few studies regarding the solar-driven cascade steam-organic Rankine cycle based on the direct steam generation method. It is a novel work in solar thermal power systems. However, these studies have been simulated using short-term analysis. Researchers used a screw expander in the top cycle in their studies, and no study has been found regarding using a steam turbine in the top cycle. In addition, their studies lack methods to control the system,

which will be covered in this research. Furthermore, there is no parametric analysis regarding the solar-driven cascade steam-organic Rankine cycle based on the direct steam generation method, and the effects of different kinds of ORC fluid on the net power output are very rare. These effects, including variations in input parameters of the system such as solar radiation intensity, pinch point in the heat exchanger, condensation temperature, and turbine inlet steam quality, will be studied on the output parameters of the system.

- ❖ Thermal storage for direct steam generation solar power systems through the use of two Rankine cycles and two storage tanks is a unique method that has been presented nowadays. However, such novel systems still need more investigations regarding the control strategy, movement of liquid and vapour fluid in the tanks, effect of the recuperator on the system performance, and working fluids for the ORC cycle.

2.11 Summary

A review of the available research on solar power generation technologies is provided in this chapter in terms of their configurations, control system, and heat storage methods. An examination of the literature reveals that several authors have studied dynamical models independently and for short time simulation, but no studies have taken into account how these solar direct steam generation systems interact, particularly with regard to the consumers' electricity demand profiles. The literature reviews on component modelling were presented in detail in order to construct a compressive, low-capacity direct steam generation solar power system design model. It is demonstrated that, in order to match the electrical demand profile from the buildings, a proper control approach must be created. However, in solar-based power plants, the control should be on the heating side, and the solar steam Rankine cycle

should synchronise itself with the controlled heat input. The studies sought to meet the energy demand by manipulating the solar Rankine cycle system's operation dynamics. Although a solar steam Rankine cycle with a heat storage unit appears promising with regard to meeting the demand and producing electricity at night, the cycle's low overall efficiency extends the payback period. In order to improve the system's power output, a novel approach that integrates a second cycle as a cascade Rankine cycle while maintaining the flexibility of the solar Rankine cycle must be devised.

The heat rejected by the steam condenser in a solar steam Rankine cycle can be used to drive another low temperature system, such as a refrigeration cycle, a DHW system or an ORC. As ORC systems show good performance and can work at low temperatures and with low pressure sources, the integration of a solar steam Rankine cycle with an ORC system can yield a better, higher power output in the system compared with a single solar steam Rankine cycle.

Finally, a steam accumulator seems to have a good effect in terms of covering periods of fluctuating solar irradiance and improving the solar Rankine cycles' performance. Integrating a steam accumulator into a solar single Rankine cycle or solar-driven cascade Rankine cycles with leading systems works by being flexible and extending the period of generation. Steam accumulators are a viable option for decreasing the impact of changing irradiance on the power generation of solar thermal systems since they have fast reaction times and high discharge rates due to the rapid evaporation and condensation of water/steam under non-equilibrium conditions. There are various ways in which steam accumulators can be integrated into power plants. A good control strategy will reduce the risk related to the wet steam turbine in a cascade cycle.

Chapter 3 Evaluation of Simulink/Simscape modelling for dynamic simulation of the Rankine cycle solar power system

3.1 Introduction

Solar thermal concentrating power systems based on parabolic trough collectors or other types of concentrating solar collectors' technology are considered a promising power system, especially in rich direct normal irradiance countries across the world. Concentrating solar power technologies can generate higher electric power outputs with additional reliability and stability compared with non-concentrating solar power systems and photovoltaic systems. Many countries around the world have installed direct steam generation solar power systems, such as the US, Thailand, Spain, Turkey, and China [99,100]. These systems were first developed in Spain and the US, and their experiences in terms of operation and construction were fully comprehensive based on local meteorological data. Meanwhile, several research projects have focused on and inspected the characteristics of parabolic trough collector technologies. Studies in solar concentrating power plants' off-design literature have considered many aspects, such as the optimization of solar steam Rankine cycle designs for given heat-source conditions; the expansion engine design for nominal conditions was subjected to off-design operation assuming that the components' performance would stay constant and equal to that at the design point; or dry condenser pressure off-design system. The literature review reveals that, in contrast to the turbines and heat exchangers, the design and off-design performance of turbines in steam Rankine cycle engines for solar stationary power generation is very limitedly documented. The solar steam Rankine cycle performance affected by various design parameters has not received enough attention. In this research, we aim to investigate the potential of a direct steam generation solar power system using Rankine cycle and parabolic trough collectors as

promising technology in Libya. This system was simulated under typical climate conditions in the southwest of Libya. The goal of the study is to analyse such a system by looking at four major elements in order to see if they can be widely deployed and adopted in the Libyan market. Firstly, operational and simulated the proposed system under Libyan meteorological data source varying conditions. Secondly, configurations, working fluids, and operating circumstances are all evaluated thermodynamically. Thirdly, selection and size of components. Finally, the better locations cross the country.

Direct steam generation system solar power (DSG) is modelled in this section. The DSG system will operate under variable boundary conditions in terms of ambient temperature, mass flow rate, heat flux, and pressure. Therefore, to facilitate the wider adoption of these systems, apart from optimizing the system's thermodynamic performance at the nominal design point, special attention should also be paid to the optimization of its performance at off-design conditions. Direct steam generation solar power efficiency depends on:

- The ability of reflection and absorption solar energy in solar fields.
- The ability of the condenser or steam generator heat exchangers to reject or extract heat transfer from the working fluid.
- The efficient operation of the expansion device, under varying pressure levels and mass flow rates.
- The ability of the cooling temperature and mass flow rate sources to condensation the working fluids.

The cooling water flow rate at the steam condenser can be readily adjusted to absorb a variable heat-rejection load at off-design operation by a variable-speed pump, which

lends greater controllability to this component and shifts the focus to solar fields. In this chapter, the methodology used throughout this thesis by using Simulink/Simscape in MATLAB program is discussed and presented. The objective of this chapter is to obtain the daily and dynamic responses of the direct steam generation solar power system under the local meteorological conditions. Furthermore, the performances of optimal design parameters in power block were explored based on the simulations. To achieve these objectives, the detailed model of DSG was built in the following;

3.2 Description of system being investigated in Simulink/Simscape

The direct steam generation solar power system is investigated in this study. Schematic of the system is shown in Figure 3.1. Whereas Figure 3.2 shows T-s diagram of the cycle when solar field outlet temperature T_3 is 270 °C and condensation pressure P_4 is 0.005MPa. This system adopts the steam turbine as a power machine. Specifically, the solar steam Rankine cycle assumed for the mathematical simulations mainly comprises solar collectors (PTC), steam turbine, a wet condenser, and a working fluid pump respectively. The steam is first produced in solar collectors and then enters to the turbine exporting power during expansion. The exhaust is condensed to saturation liquid in the wet condenser, and it is pressurized and pumped back by the pump to solar collectors' fields to be heated it again and the cycle is repeated. In the solar collector's area, there are two regions, boiler region and superheater region. Preheating and evaporation of the water occur in the boiler area, while the superheating the steam occurs in superheater region. The superheater region can guarantee only superheated steam enters to the steam turbine. The liquid level in the boiler region is controlled in this system, and the pump works based on this control strategy. More detail of control strategy for direct steam generation solar power system is presented in section 3.5 below.

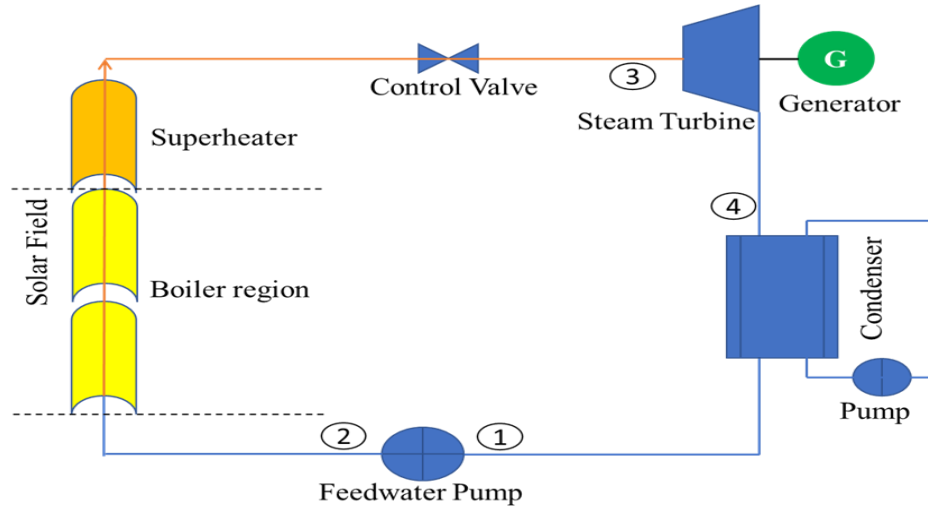


Figure 3.1: Configuration of the direct steam generation solar power system

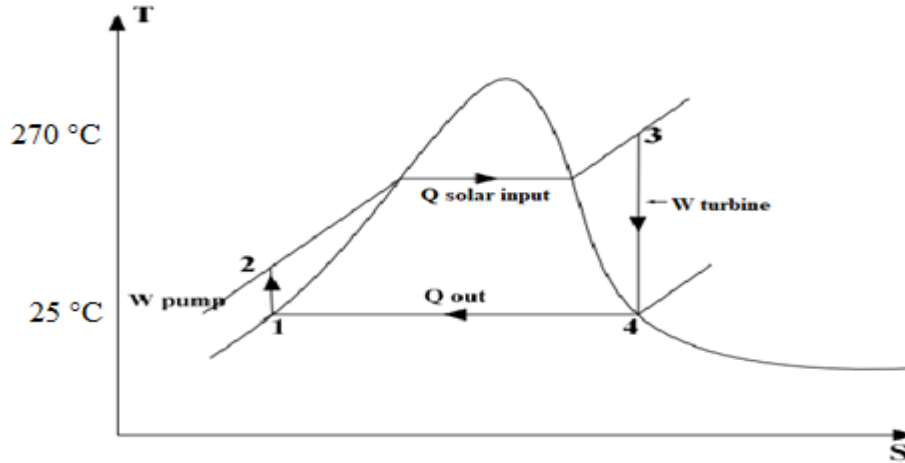


Figure 3.2: T-s diagram of direct steam generation solar power system

3.3 Thermodynamic modelling

3.3.1 Modelling of solar thermal collector

Total area of solar collectors is an important operation parameter for direct steam generation power systems. One such example is the solar collector produced by the Industrial Solar Technology (IST) Corporation. IST erected several process heat installations in the United States with up to 2700 m² of collector aperture area and it has been widely evaluated in literatures [101]. In this study, IST parabolic trough collectors have been selected to power the DSG system and a detailed thermal

modelling for the module IST-PTC has been presented. These equations can be combined together and finally, the thermal efficiency of the solar collector can be calculated in every case. IST parabolic trough collectors are imaging concentrating collectors with high concentration ratio and they exploit only the direct beam part of the incident solar irradiation [101]. Thus, the available and useful solar energy is calculated as the product of the outer aperture (A_{PTC}), the solar beam irradiation (G_b) and collector efficiency.

$$\dot{Q}_s = A_{PTC} \cdot G_b \cdot \eta_{PTC} = \dot{m}_{steam} \cdot C_{p_{steam}}(T_{col,out} - T_{col,in}) \quad (3.1)$$

The performance formula of a single Industrial parabolic trough solar collector (IST-PTC) given by the manufacturer is [101].

$$\eta_{PTC} = k_o - k_1 \times \frac{T_{in} - T_{am}}{G_b} - k_2 \times \frac{(T_{in} - T_{am})^2}{G_b} \quad (3.2)$$

The use of tracking is something vital for PTC. In this study, the collector axis was assumed to be in N-S axis and tracking is horizontal in E-W axis. Equation. (3.3) shows the solar incident angle for every moment with this tracking system [101].

$$\cos(\theta) = \sqrt{\cos^2(\theta_z) + \cos^2(\delta) \cdot \sin^2(\omega)}, \quad (3.3)$$

According to theory, the greatest advantage of this arrangement is the very small shadowing effects in the case of using many collector modules. These are presented in the first and last hours of the day. According to some studied in open literature review [101, 102, 103, 104], IST-parabolic trough collector type has some advantages such as very simple structure, it can be installed on the roof of large building, and it is cheaper compared with other solar concentrating collectors. Schematic of solar parabolic trough collector and comparison efficiency of various collectors are shown in Figure 3.3.

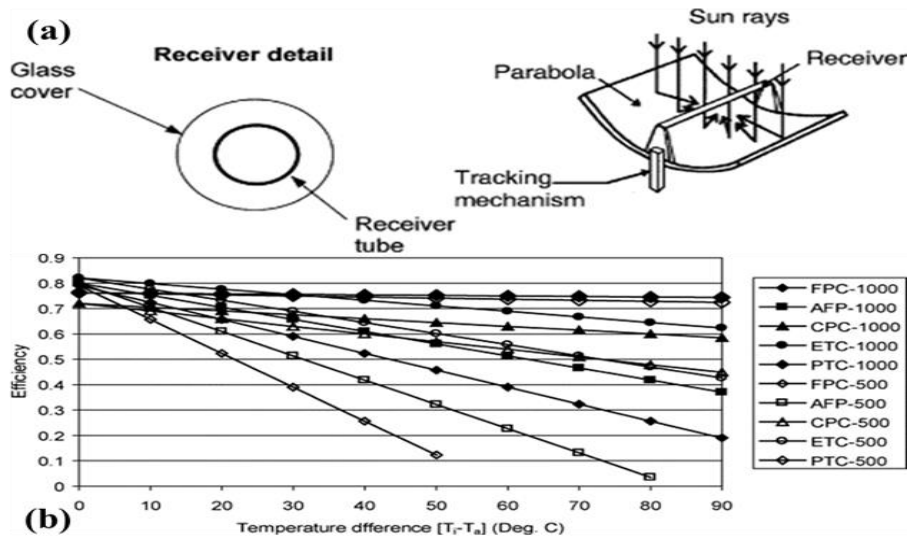


Figure 3.3: Schematic of (a) a parabolic trough collector and (b) Comparison of the efficiency of various collectors [101]

The subsystem of solar collectors' area model in Simulink\Simscape is shown in Figure 3.4 below. The subsystem consists some of thermal, two-phase fluids blocks, and Simulink blocks. There are two regions in this model, one is a boiler region, and another is superheated region. Characteristics of the IST parabolic trough collector is presented in Table 3.1.

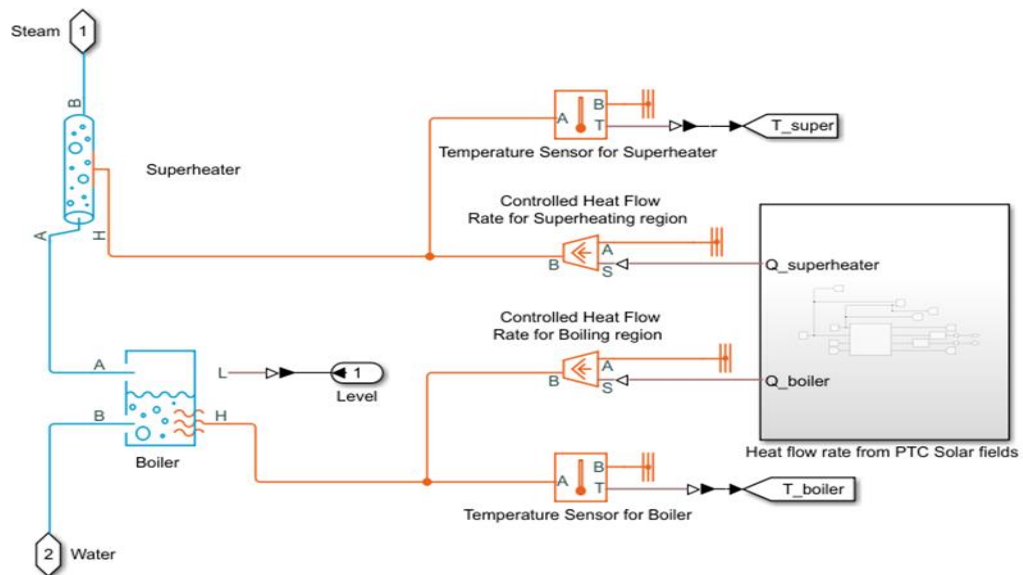


Figure 3.4: Schematic of parabolic trough collectors for boiler and superheater regions developed in Simscape

Table 3.1: Characteristics of the IST PTC system [101]

Parameter	Value & type
Gross length of every module Collector	6.1 m
Collector aperture	2.3 m
Receiver material	Steel
Collector rim angle	70°
Reflective surface	Silvered acrylic
Absorptance	0.97
Glass envelope transmittance	0.96
Absorber outside diameter	50.8 mm
k_0 : intercept efficiency	0.762
k_1 : negative of the first-order coefficient of the efficiency ($W/m^2 \text{ } ^\circ C$)	0.2125
k_2 : negative of the second-order coefficient of the efficiency ($W/m^2 \text{ } ^\circ C^2$)	0.001672
Mode of tracking	E–W horizontal
Collector orientation	Axis in N–S direction

3.3.2 Steam turbine modelling

In direct steam generation solar power systems, the turbine plays a vital role in the performance of the cycle. Volumetric-type or scroll expanders are usually preferred rather than turbines and screw expanders in small-scale solar organic Rankine cycles, that is because of their availability, lower mass flow rates, reduced number of moving parts, reliability and ability to operate at the higher pressures [105]. In medium or large-scale solar Rankine cycles, screw expanders and steam turbine are the best options. Selection of the expansion device type depends on many factors like expansion process, pressure ratio and type of working fluid. Therefore, a detailed dynamic model of the turbine is unnecessary for the simulation and a steady-state

model is sufficient for this purpose. In this study, steam turbine is selected to produce a few hundred kW. Figure 3.5 shows a two-phase fluid network turbine block in Simscape.

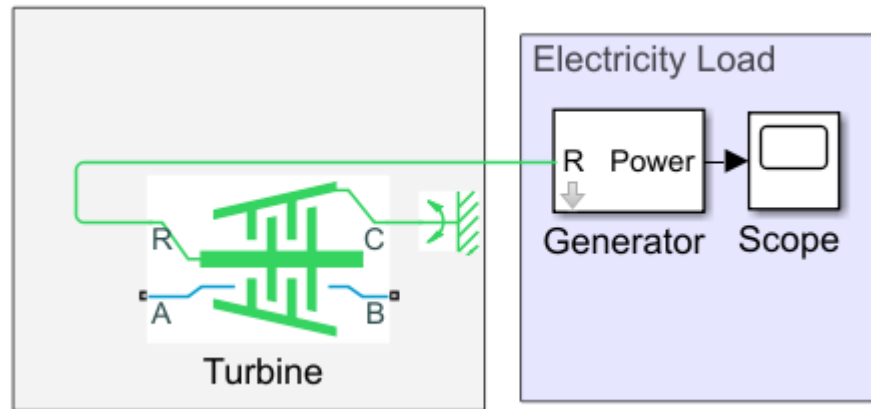


Figure 3.5: Block diagram of steam turbine connected with electricity load system in Simscape

From Figure 3.5 port A indicates inlet flow into the turbine, port B is outlet flow of the turbine. Ports R and C are the mechanical rotational conserving ports associated with the shaft and casing, respectively. Fluid flow from port A to port B drives a positive rotation of port R relative to port C. Steam enters to turbine at high temperature and pressure in superheated state. The potential energy (in the form of pressure) in the steam is converted to kinetic energy (in the form of momentum) by imparting its momentum to the turbine blades, which causes associated shaft to revolve. A generator converts the mechanical work created by the revolving shaft into electrical energy. An analytic model for a turbine is derived using conservation of mass and energy together with Stodola's law. An adiabatic expansion is assumed in which isentropic efficiency is present. The mechanical efficiency of the turbine is also considered in the model. The efficiency exponent in Stodola's ellipse equation can be computed by:

$$\eta_{exponent} = \left(\frac{\eta_{poly} + 1}{\eta_{poly}} \right) \quad (3.4)$$

Where, η_{poly} is polytropic efficiency and it is equal to 1 in this study. The nominal mass flow coefficient is calculated by:

$$\phi_{flow,coeff} = \frac{\dot{m}}{\sqrt{\frac{P_{in}}{v_{in}}}} \quad (3.5)$$

The pressure ratio in the steam turbine is calculated by:

$$P_r = \frac{P_{out}}{P_{in}} \quad (3.6)$$

Where P_{out} and P_{in} are outlet and inlet pressures of the turbine. Stodola's Ellipse constant is calculated by:

$$\phi_{\alpha} = \frac{\phi_{flow,coeff}}{\sqrt{1 - P_r^{\eta_{exponent}}}} \quad (3.7)$$

The mass flow coefficient threshold can be calculated by:

$$\phi_{flow,coeff,threshold} = \phi_{\alpha} \cdot \sqrt{1 - B^{\eta_{exponent}}} \quad (3.8)$$

$B^{\eta_{exponent}}$ is pressure ratio threshold for flow reversal and it is equal to 0.999 in this study. The mechanical power output of the turbine can be calculated by:

$$\dot{W}_{turbine} = \dot{m} \cdot (h_{in} - h_{out}) \quad (3.9)$$

Whereas here $\dot{m}$, h_{in} and h_{out} are the mass flow rate, inlet and outlet enthalpies in the turbine respectively.

The expansion in the steam turbine is modelled with the turbine isentropic efficiency and it can be expressed by:

$$\eta_{isent} = \frac{h_{in} - h_{out}}{h_{in} - h_{outs}}, \quad (3.10)$$

Where h_{in} , h_{out} and h_{outis} are indicate inlet, outlet enthalpies in the steam turbine and enthalpy of isentropic process at the exit

3.3.3 Steam Condenser modelling

A steam condenser is a closed vessel heat exchanger that is used to convert low-pressure turbine exhaust steam to water. It does this with the help of cooling water from the cooling tower that flows inside it. To improve efficiency, the pressure within a steam condenser is kept below atmospheric pressure. It's widely utilised to reduce exhaust backpressure at the turbine end. When cooling water is scarce, an air-cooled condenser is frequently employed. However, an air-cooled surface condenser is far more expensive and cannot reach the same low steam turbine exhaust pressure (and temperature) as a water-cooled surface condenser [106,107]. The steam condenser subsystem used in this model is presented in Figure 3.6. The heat reject by the steam condenser is expressed by:

$$\dot{Q}_{con} = \dot{m}_{steam} \cdot (h_{in} - h_{out}) = \dot{m}_{water} \cdot C p_{water} \cdot (T_{in} - T_{out}) \quad (3.11)$$

Where, $\dot{m}_{water}$ is the mass flow rate of water, h_{in} and h_{out} are inlet and outlet enthalpies of the steam condenser. A saturated fluid chamber block in the thermal Simscape library is selected to be as wet condenser in this study. The chamber volume is divided into the saturated vapor and liquid volume. It has three ports, port A presents inlet steam to the condenser which is always in the vapor state, port B present saturated liquid and it is always in the saturated liquid state. Port H is the thermal conserving port, and it supplies the coolant temperature into the chamber from the cooling temperature source. Mass of saturated vapor and saturated liquid can be calculated by:

$$\dot{m}_{Vapor} = (1 - L) \cdot \left(\frac{Volume}{V_{Vapor}} \right) \quad (3.12)$$

$$\dot{m}_{Liquid} = L \cdot \left(\frac{Volume}{V_{Liquid}} \right) \quad (3.13)$$

The fluid mass in the chamber is obtained from the equation:

$$\dot{m}_{water} = \dot{m}_{inlet} + \dot{m}_{outlet} \quad (3.14)$$

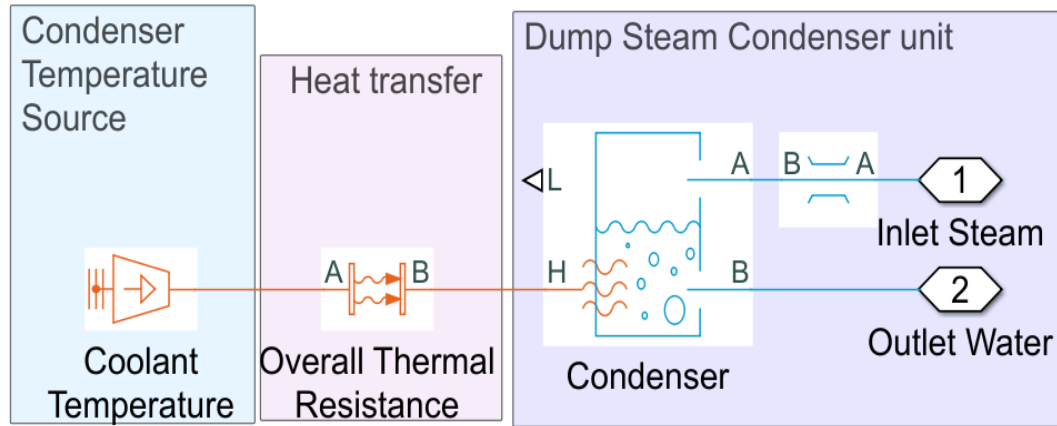


Figure 3.6: Layout of condenser model subsystem in Simscape

3.3.4 Pump modelling

In a direct steam generation solar power system, the pump is accountable for pressurizing and pumped the working fluid in the system and supplying the needed mass flow rate throughout the system. Centrifugal and volumetric-type pumps, such as diaphragm and piston pumps, can be used depending on the mass flow rate and pressure requirements. Other requirements, such as leak-proofing, the capacity to supply lower mass flow rates at higher pressures, and compatibility with the organic working fluid, should be addressed in solar Rankine cycles. Keeping a low pump electrical power consumption in solar Rankine cycles is critical because, as with other power cycles, this enhances the system's net amount of electrical power generated after consumption by the pump. In this research, the centrifugal pump is chosen between the available options for solar Rankine cycle [108]. The block named

controlled mass flow rate source in the Simscape library was chosen to be the pump and is shown in Figure 3.7 below.

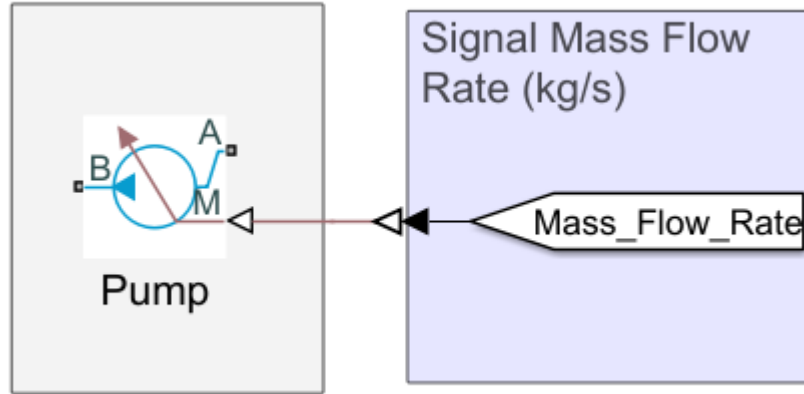


Figure 3.7: Layout of feedwater pump model subsystem in Simscape

The mechanical power required for the water pump is expressed by:

$$\dot{W}_P = \frac{\dot{m}_{water} \cdot v \cdot (P_{out} - P_{in})}{\eta_P} = \dot{m}_{water} \cdot (h_{out} - h_{in}) \quad (3.15)$$

where $\dot{m}_{water}$ is mass flow rate of water, v is water specific volume at the pump inlet, η_P is the pump efficiency and P_{in} and P_{out} are the pressures of the water at the pump inlet and outlet, respectively.

3.4 Auxiliary blocks in the Model

3.4.1 Two-phase fluid thermal properties block in Simscape

The component sets the thermophysical properties of a several fluids in a two-phase fluid network. It is generated fluid property tables from Cool-Prop or REFPROP database [109]. Properties includes viscosity, pressure up to 100MPa, temperature, specific heat and density. Each block has only one working fluid properties. Several types of the two-phase fluid are available in this block. A model cycle must be connected to this block and type of working fluid must be determined in order to

simulated very accurate results. Two-phase fluid thermal properties block in Simscape library was presented in Figure 3.8 below

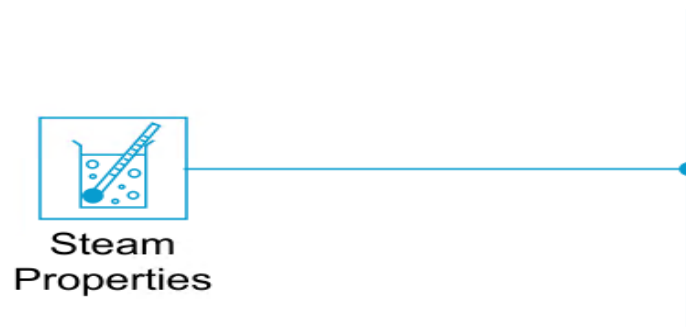


Figure 3.8: Two-phase fluid thermal properties block in Simscape [104]

3.4.2 Solver Configuration

The solver configuration component is specified solver parameters which needed for a model cycle. Each cycle or network in Simscape diagram needs a Solver configuration and own settings information for modelling simulation. Each topologically distinct Simscape block diagram requires exactly one Solver Configuration block to be connected to it [109]. Figure 3.9 shows how a solver configuration component available in the Simscape library is connected to a cycle line

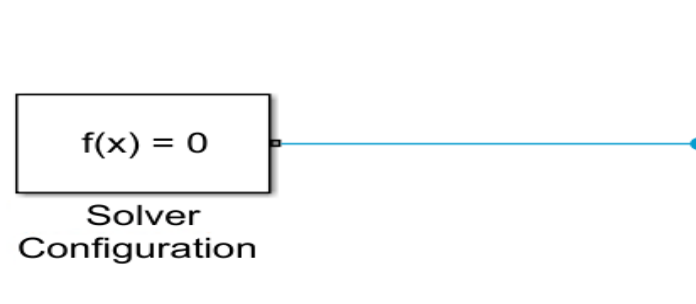


Figure 3.9: Solver Configuration block in Simscape [104]

3.4.3 Generator load part

This subsystem in Figure 3.10 below is responsible transfer mechanical power from the turbine to the generator to generate electricity in the cycle by the shaft. It consists

of Torque sensor, ideal angular velocity source, Simulink blocks like shaft speed and converter power.

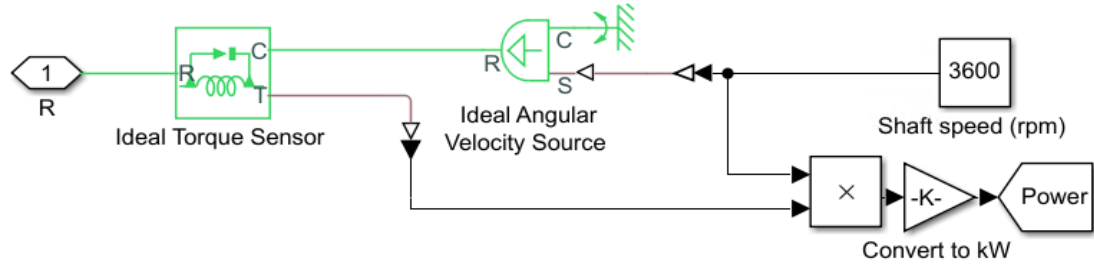


Figure 3.10: Generator load part diagram in Simscape.

A torque Sensor component is responsible to convert a variable passing through the sensor into a control signal proportional to the torque. Port T is a physical signal port that outputs the measurement result, ports R and C are mechanical rotational conserving. Ideal Angular Velocity Source is responsible ideal source of angular velocity which produces velocity differential at its terminals proportional to the input physical signal. The velocity is measured by

$$\omega = \omega_R - \omega_C \quad (3.16)$$

where ω_R, ω_C are the absolute angular velocities at ports C and R

3.5 Control strategy

Solar energy is transient and fluctuating energy during the day. Therefore, the direct steam generation solar power system requires a reliable control scheme to ensure a smooth power output. In this study, liquid level of working fluid has been controlled in the boiler region by using PI control as shown in Figure 3.11.

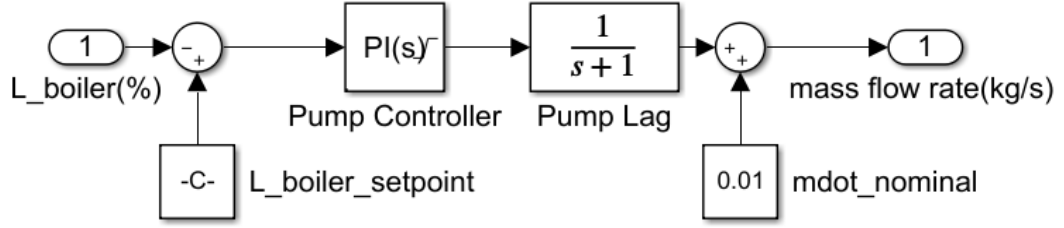


Figure 3.11: Schematic diagram of PID control strategy for liquid level in boiler region

The boiler region in the parabolic trough has a combination of two-phase fluids which means fluid consists of liquid and vapor, if this two-phase fluid goes into the steam turbine, it will damage turbine's blades and decrease efficiency of the cycle. A control system to ensure no two-phase fluid goes into the steam turbine is important. The steam flows to the superheated region which uses a simple proportional, integral, and derivative (PID) controller to keep the solar power production at a steady rate and maintain the liquid level volume at 0.47% of the boiler region with a steady mass flow rate.

The steam production flow rate in the proposed direct steam generation solar power system can be expressed by:

$$\dot{m} = \frac{\dot{Q}_s}{h_{out} - h_{in}} \quad (3.17)$$

Where h_{in} and h_{out} are the inlet and outlet enthalpies in the solar collecting area respectively.

Finally, the thermal efficiency of the proposed direct steam generation solar power system shows how effectively the solar irradiance can be transferred into electricity power and it can be calculated as:

$$\eta_{th} = \frac{\dot{W}_{turbine} - \dot{W}_p}{\dot{Q}_s} \quad (3.18)$$

3.6 Validation of the modelling

The accuracy of the model in predicting the physical behaviour of the actual system is the main issue with numerical modelling. The developed Simulink/Simscape simulation model is tested in this section using data from the open literature. The results of the current model were compared to existing theoretical and experimental data in terms of the direct steam production solar power system's various operating parameters and performance metrics. Temperatures during the operating cycle, system effectiveness, pressure, steam/water mass flow rate, heat transfer rate, net output power, and turbine power are all included. In this subsystem, the validation of solar direct steam generation power system was validated with a real solar thermal power plant.

3.6.1 Validation against experimental real data of the TSE1 solar power plant

The validation of the simulation model against real data of an existing of the TSE1 solar power steam generation plant in Kanchanaburi, in Thailand were presented and addressed in this subsection. In Kanchanaburi city of Thailand a solar concentrating thermal power plant called “KTSE-9100” powered by parabolic trough collectors based on direct steam generation has been erected to generate electricity in the city. The real power plant is 9 MW in size and is constructed in two phases, the first of which includes the 5 MWe plant described in this study and the second of which includes a second 4 MWe plant. The first plant can produce 5 MWe of nominal power, which is supplied into the public grid for a price that can reach 11.6 THB per kWh. The German company Solarlite GmbH provided the parabolic trough collector field and served as the solar field's EPC contractor. The steam turbine for the facility was shipped from MAN Turbo. The station is owned and run by the client Thai Solar

Energy (TSE), who is in charge of the power block. According to the Very Small Power Producer (VSPP) legislation, the licence for supplying power to the public grid has already been given. Construction of the power plant started in February 2010. In 2011, the mechanical completion of the plant and the solar field were both finished [98].

3.6.2 TSE1 solar power plant description

Figure 3.12 shows the aerial view of TSE1 solar thermal power plant has been installed in Kanchanaburi in Thailand which is obtained from Solarlite company.



Figure 3.12: Aerial view of TSE1, Kanchanaburi, Thailand [98]

Both solar superheater and the solar evaporator in the system use SL4600 parabolic trough collectors in this plant. A direct steam generation system in recirculation mode is being used. Within the evaporator, water is warmed and partially evaporated. The steam drum is where this two-phase flow of liquid and steam is separated [98]. The superheater field is where the superheated steam travels after becoming saturated. As a result, the superheater field influences steam temperature while the evaporator field influences live steam mass flow. Condensed superheated steam powers the turbine. The liquid phase from the steam drum and the condensate from the turbine combine as they re-enter the evaporator field. The MAN manufacturing company modified the

5 MW turbine to accept low partial load pressures during commissioning in order to maximise its flexibility. The control valves at the turbine's inlet regulate the pressure in the solar field when the turbine is typically operated in fixed pressure mode. In order for the turbine to operate at its maximum rated power during typical operation, the solar field output must match or surpass the turbine swallowing capacity. Instead, sliding pressure control is achieved by keeping the control valves at the turbine inlet always fully open, which causes the live steam pressure to be a side effect of the mass flow generated by the solar field. To keep the turbine running as long as feasible, fixed pressure control was used during normal operations and sliding pressure control during transient operations.

3.6.3 Validation of Simulink/Simscape model

The findings of the constructed Simscape/Simulink model are compared against actual data from the solar TSE1 power plant in this subsection. The developed model results were compared with the data from the actual plant at the nominal design point, as stated in Table 3.2 to perform the validation. Table 3.3 shows that there is excellent agreement between the model outputs and the data from actual plants. The highest difference between the findings of the generated model and the data from actual plants is less than 2.5%. This illustrates that the generated model accurately captures both the internal workings and the overall system performance of the actual solar TSE1 power plant. The TSE1 power plant operating conditions used in the simulation model.

Table 3.2: Fixed parameters of the TSE1 plant [98, 107]

Term	Unit	Value
Nominal electrical power	MW	5
Nominal steam pressure	MPa	3.0

Nominal thermal power	MW	19.5
Nominal steam temperature	°C	330
SL4600 collector segment length	m	12
SL4600 collector aperture width	m	4.6
PTR70 receiver diameter	mm	70
Nominal thermal efficiency of the system	%	0.26
Nominal mass flow rate of steam	kg/s	6.1

Comparison of solar direct steam generation power plant simulation model findings with the actual solar TSE1 power plant is presented in Table 3.3 below.

Table 3.3: Comparison of real data of the TSE1 plant with Simscape simulation model

Parameter	Model results	Real DSG plant data	Deviation (%)
Nominal electrical power (MW)	5	5	0.02
Nominal steam pressure (MPa)	3.1	3	3.3
Nominal steam temperature (°C)	331	330	0.30
Thermal efficiency (%)	0.2564	0.26	0.1
Nominal thermal power (MW)	19.5	19.5	0

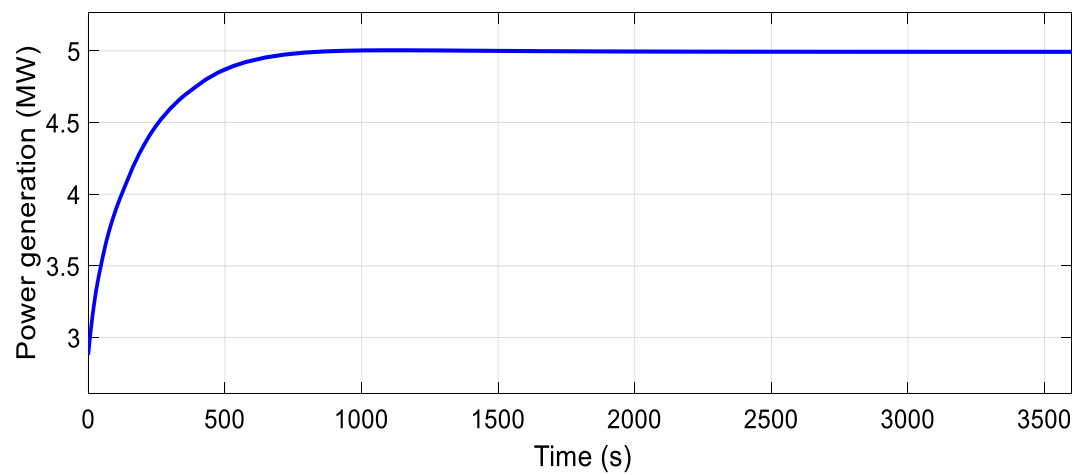


Figure 3.13: Results of the plant's power output simulation for an hour starting from an initial condition

Figure 3.13 displays the power generation output in the Simscape model for 3600 seconds of simulation. It can be observed that in the first 700 seconds of the simulation, the steam turbine reaches its design power output, which is close to 5 MW as nominal electrical power in the TSE1 power plant.

Dynamic response of the other three parameters to show the feature of dynamic modelling by this software for one hour of the simulation. These parameters are turbine inlet pressure, turbine inlet temperature, and steam quality into the steam turbine, as shown in Figure 3.14. It is clear to see that there is very good matching between the simulation data and the real power plant data.

This dynamic analysis follows a one-hour calibration of the model and is performed at every second timescale as Simscape runs and analyses the model for every second, meaning that this hour of solar radiation is factored into the model parameters with a variable value. Since the grid load may change at a quicker time step, as well as weather conditions, it would be interesting to also argue the reliability of the model at different time steps.

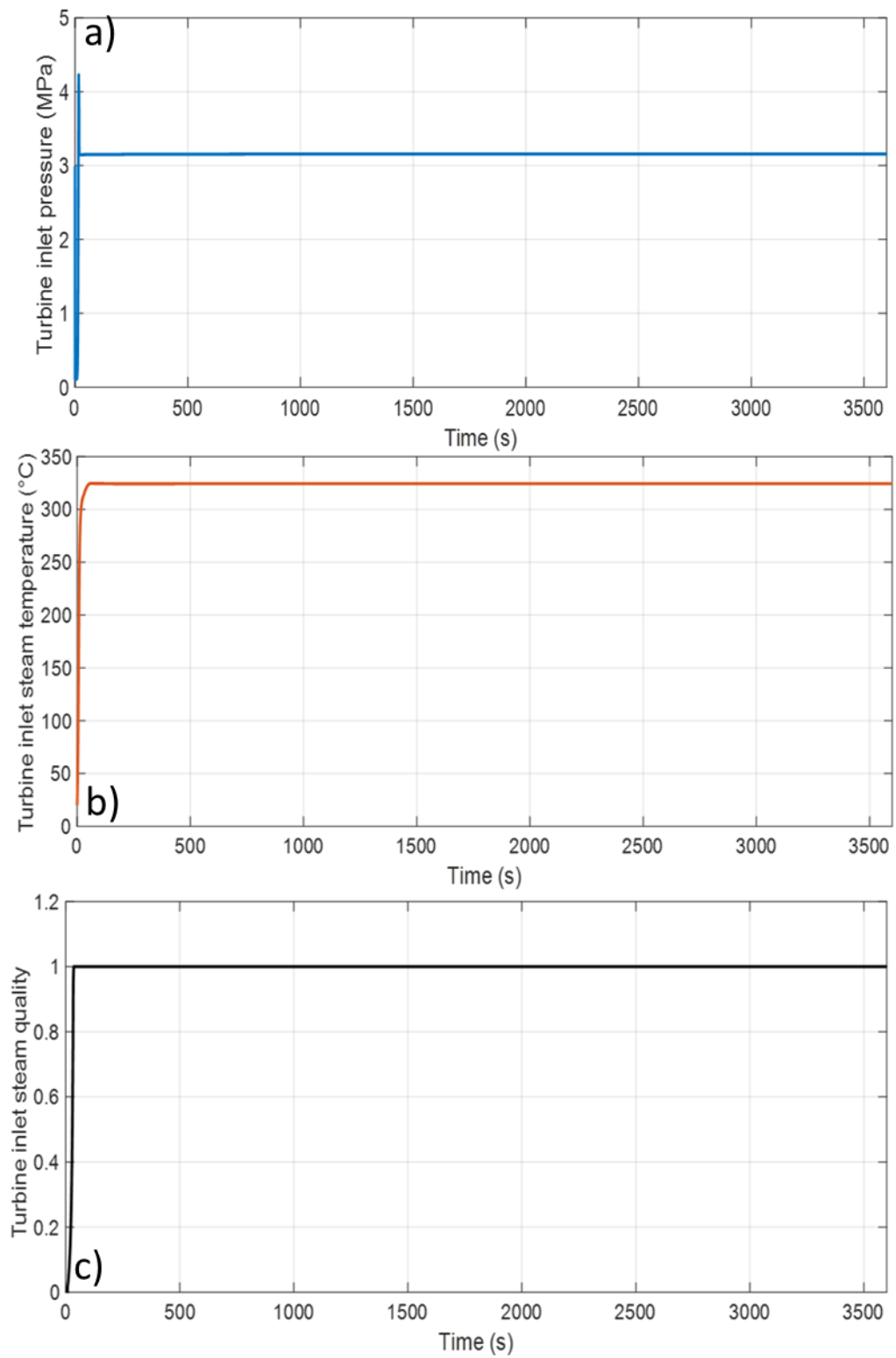


Figure 3.14: Results of the plant's power output simulation for an hour starting from an initial condition

3.7 Simulation results and discussion

Results of the direct steam generation solar power system's parametric analysis are reported in this part along with simulations of the system's design and off-design performances using Simscape and Simulink software. Additionally, the system's daily and monthly performances are examined. In this analysis, REFPROP and CoolProp software were used to establish the thermal properties of water and steam or other working fluids. All equations in Simscape are written and evaluated simultaneously. Figure 3. 15 below shows the flow chart of the calculation step in Simscape model in this chapter.

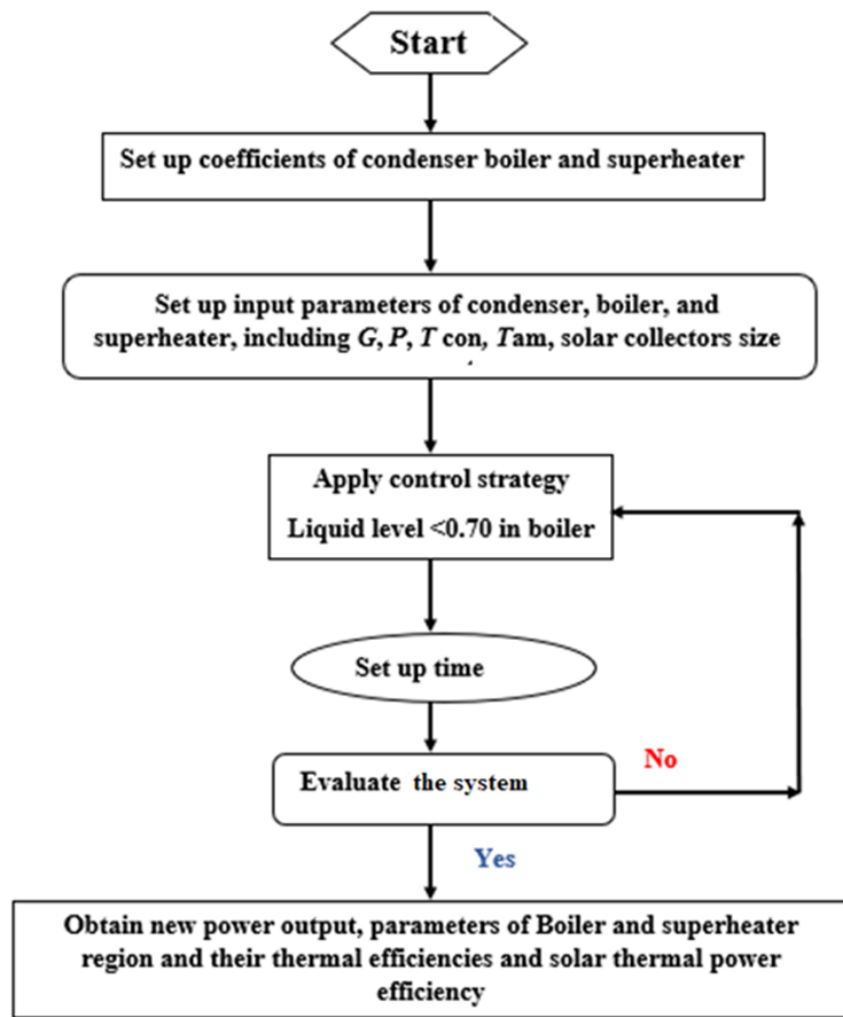


Figure 3.15: Flow chart of the calculation step in Simscape model

3.7.1 Direct steam generation solar power design conditions

First, the design condition of the direct steam generation solar power system needs to be determined to analyse the performance of the system. In this system, the solar collection area has been selected to be constant. It consists of 100 parabolic trough collectors, each with an aperture area 13.2 m^2 . Also, the condensing temperature in this system is in line with the ambient temperature in Libya. The ambient air temperature has an influence on the design conditions selection. The ambient air temperature variation during a typical day in Libya has slight variation and also changes from season to season. The mean temperature is around 20°C to 45°C and the ground water temperature is from 10°C to 25°C . Therefore, the condensing temperature is selected as 25°C . A water-cooled condenser is selected due to its better performance compared with an air-cooled condenser system. In addition, ground water is abundant in the Libyan desert according to [110, 111]. The DSG model configuration in Simulink/Simscape is shown in Figure 3.16. Each block is Simulink/Simscape component that illustrates solar collector area, valve turbine, condenser and pump which are the main components of the DSG system that reproduce the behaviour of the component simulated. All liquid qualities are kept via solid blue lines and are passed from one component to another. Solid blue lines represent fluid connections between model components. Figure 3.17 shows T-s diagram of the direct steam generation solar power system.

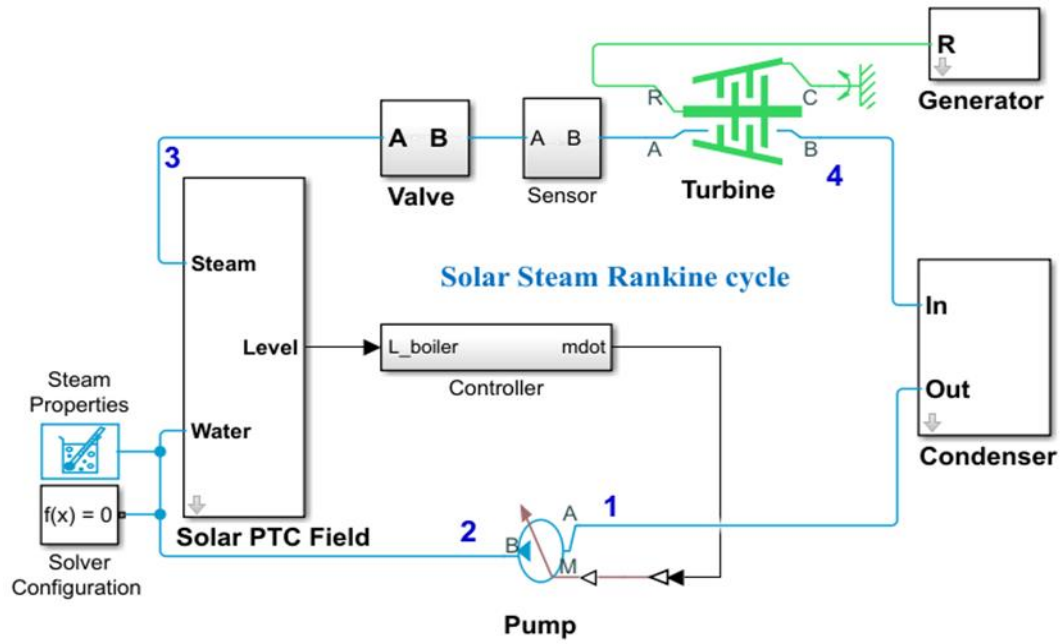


Figure 3.16: The developed Simulink/Simscape model of the direct steam generation solar power system

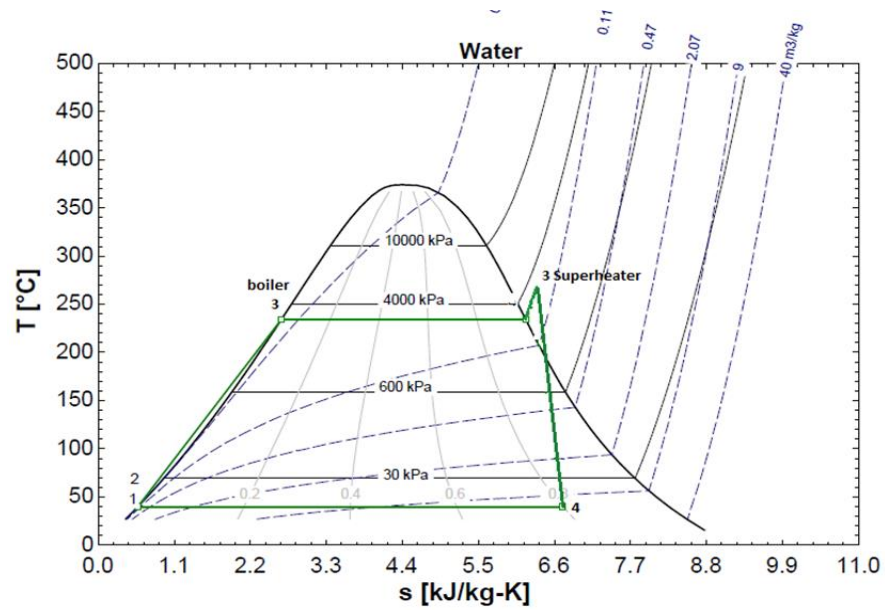


Figure 3.17: T-s diagram of the direct steam generation solar power system

Design conditions and parameters of the proposed direct steam generation solar power system is listed in Table 3.4.

Table 3.4: Design conditions of the proposed direct steam generation solar power system

Parameter	Value
No of solar PTC collector modules	100
Single solar PTC collector area	13.2 m ²
Condensing temperature	25 °C
Nominal inlet pressure turbine	3 MPa
Nominal mass flow rate	0.30 kg/s
Turbine isentropic efficiency	0.85
Electromechanical efficiency of the generator	0.95
Pump isentropic efficiency	0.85
Condenser pressure	0.005 MPa
Rotational speed	3600 rpm
Solar field outlet temperature	270 °C

3.7.2 Direct steam generation solar power off-design operation modes

As steam turbine was adopted here in this system, the investigation, and the characteristics performance prediction of the steam turbine when it works at off-design modes are evaluated. Figure 3.18 shows an off-design steam turbine created using the law of the ellipse method. This approach is frequently used to compare and portray various absolute flow rates and pressures. It is noted that the power output decreases when using a lower inlet pressure and steam mass flow rate.

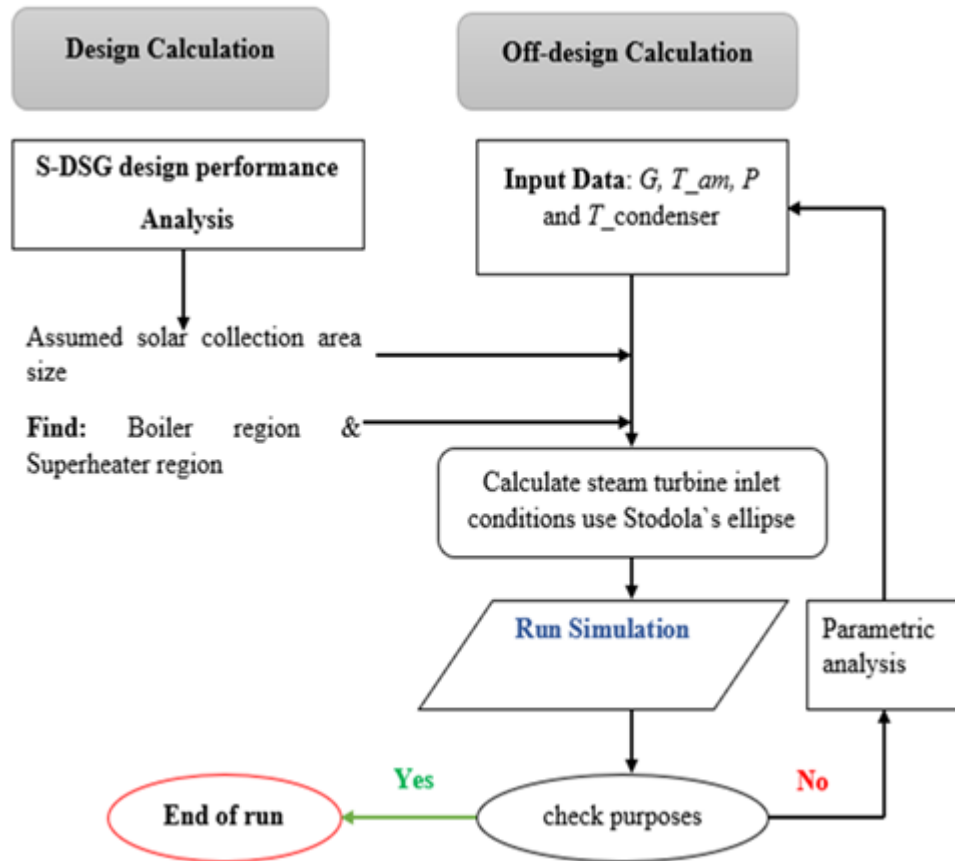


Figure 3.18: Off-design calculation steps

The turbine inlet pressure effect on the power output and torque needs to be evaluated as well as affect the mass flow rate on the turbine inlet pressure. Figure 3.19 shows effect of the turbine inlet pressure on the mechanical power output and created torque. The inlet pressure design point is selected to be 3 MPa, based on this value the system can generate more than 165 kW. It is noticed that increment of the turbine inlet pressure leads to higher torque and power output as shown in Figure 3.19.

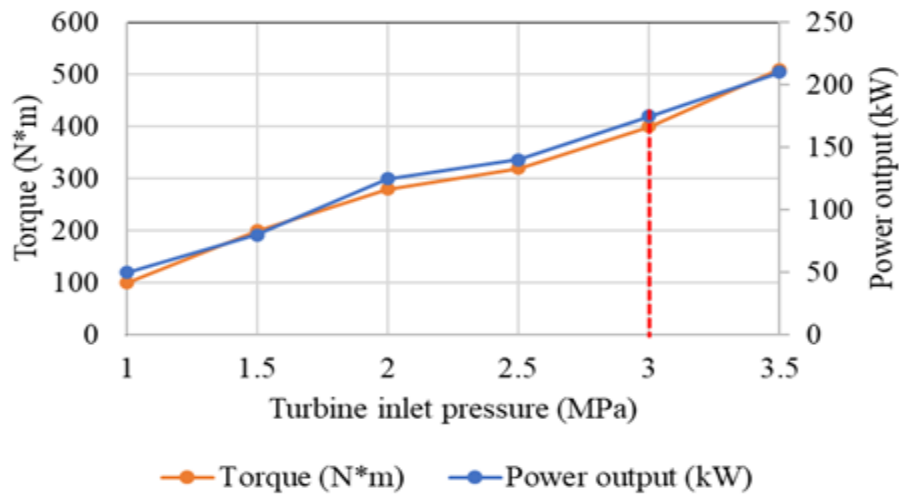


Figure 3.19: Effect of turbine inlet pressure variation on the power output and Torque

The turbine inlet pressure is affected by nominal mass flow rate as well as shown in Figure 3.20 below. To reach 3 MPa inlet pressure required 0.33 (kg/s) mass flow rate.

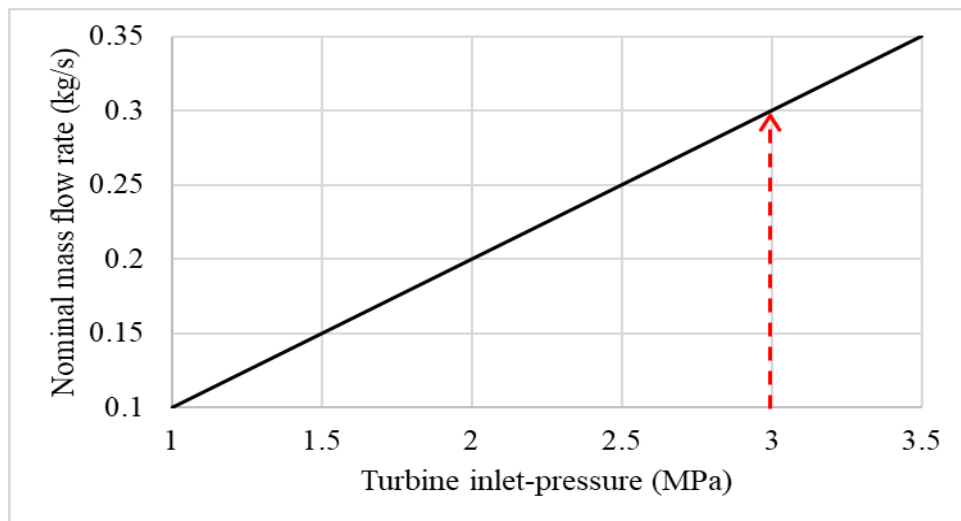


Figure 3.20: Effect of turbine inlet pressure variation on the nominal mass flow rate

Turbine isentropic efficiency has a significant effect on the system performance and as such, must be taken into account in off-design performance. From Figure 3.21, it is clearly to be seen that by increasing the turbine inlet pressure, the turbine isentropic efficiency increases. From Figure 3.21 also shows higher the turbine inlet pressure

increases turbine rotation speed as expected but it must be noted that electrical load and frictions factors can effect on the turbine rotation speed. The turbine rotation speed is in steady when the inlet pressure is around 3-3.5.

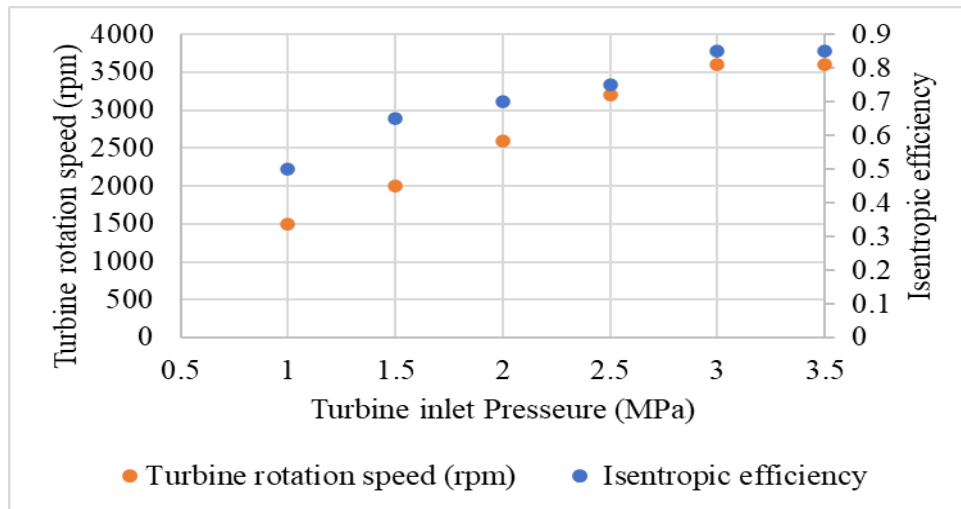


Figure 3.21: Effect of pressure ratio on the turbine isentropic efficiency and rotation speed

Figure 3. 22 display Solar irradiance and ambient temperature variation of the sunny typical day on 20th May in Libya [6].

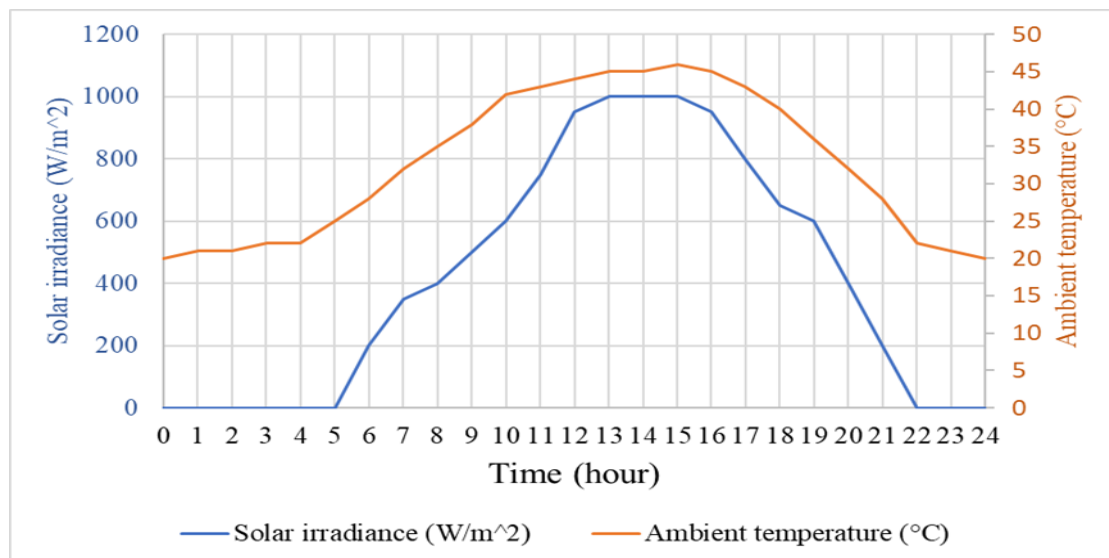


Figure 3.22: Solar irradiance and ambient temperature variation of the sunny typical day in May [6]

3.7.3 Direct steam generation solar power system parametric analysis

Several operating parameters have an impact on how well the direct steam generation solar power system performs. Therefore, to study the impact of five different operational factors on the system performance, a parametric analysis is conducted. These include the mass flow rate throughout the system, condensation temperature, steam pressure, and the steam temperature in the solar field in addition to the steam turbine isentropic efficiency. Determining which of the characteristics has a greater impact on the performance of the direct steam generation solar power system is another target of the parametric analysis. Following research using the created Simscape-Simulink simulation model, the consequences of these parameters are examined. Each time, only one parameter's value is altered within the parameters indicated in Table 3.5 leaving the other parameters unchanged and set to their default settings. In the simulations, the variations in the ambient temperature and solar irradiation are utilized. Table 3.5 shows the investigated operational parameters' fundamental values and ranges

Table 3.5: The investigated operational parameters' fundamental values and ranges

Parameter	Investigated value	Basic value
Turbine isentropic efficiency, %	70 to 85	85
Turbine inlet-pressure, MPa	3 to 4.5	3
Condensation temperature, °C	20 to 50	25
Nominal mass flow rate, kg/s	0.33 to 0.40	0.33
Solar collecting area, m ²	990 to 1544	1320

3.7.3.1 Influence of the turbine's isentropic efficiency

The steam turbine's isentropic efficiency has a direct effect on the power generation of the system, leading to impacts on the thermal efficiency and performance of the system. Figure 3.23a and b illustrates the effect of the steam turbine's isentropic efficiency on the power output and thermal power efficiency of the direct steam generation solar power system. It appears that, when the steam turbine's isentropic efficiency increases, both the power output and thermal power efficiency of the system increase. Increasing the turbine's isentropic efficiency from 70% to 85% leads to improvements in the power output of 252 to 310 kW and in the thermal power efficiency of 21% to 26%, respectively. This is mainly because of the enhancement in the direct steam generation solar power system as a result of the higher steam turbine performance.

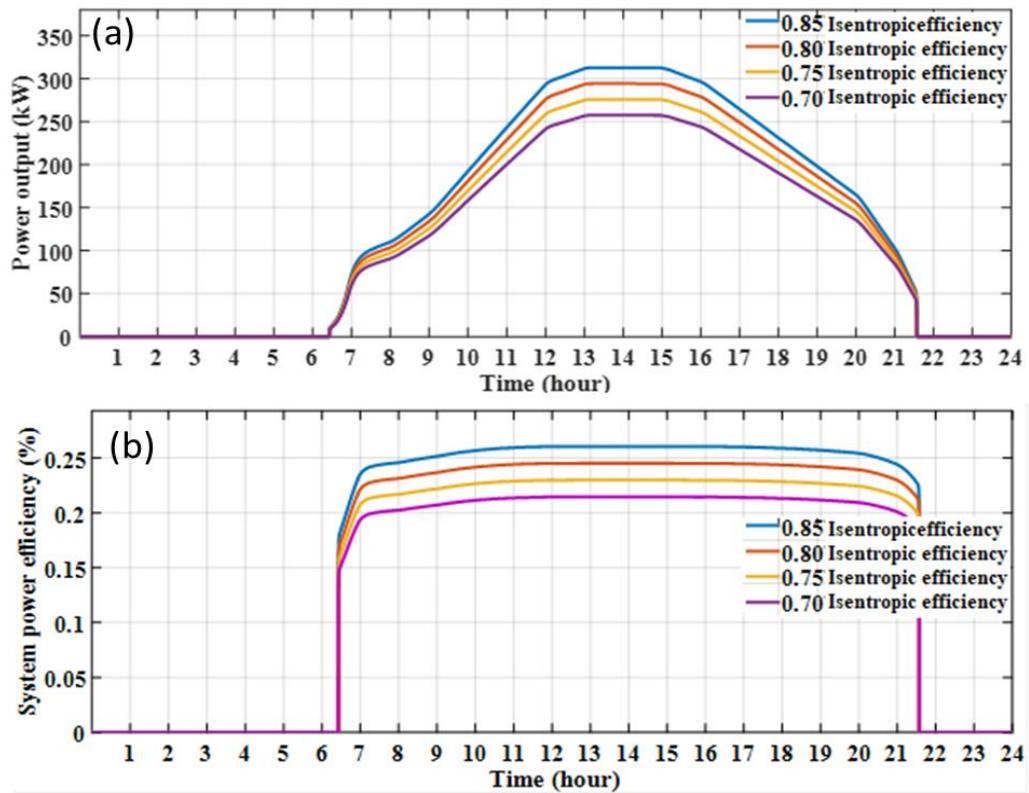


Figure 3.23: Effect of the turbine isentropic efficiency on the system, (a) power output and (b) solar thermal power efficiency

From Figure 3.23a and b, it can be seen that isentropic efficiency of the steam turbine has great effect on the proposal system performance.

3.7.3.2 Influence of the mass flow rate

A key important control parameter in the direct steam generation solar power system is the feed working pump speed, which can be controlled easily according to the work conditions or desired power output. Increasing the mass flow rate in a solar thermal power system will increase the power output. Figure 3.24a, b and c show an example of how the mass flow rate can affect the direct steam generation solar power system's performance. However, other parameters, such as the turbine isentropic efficiency, condensation temperature and liquid control level value in the boiler region are kept constant during this stage of the evaluation. The net power output is found to peak for a mass flow rate of 0.33 kg/s when the solar irradiance is 1000 W/m², while the net power output monotonically decreased when the mass flow rate was raised to 0.44 kg/s as illustrated in Figure 3.24a.

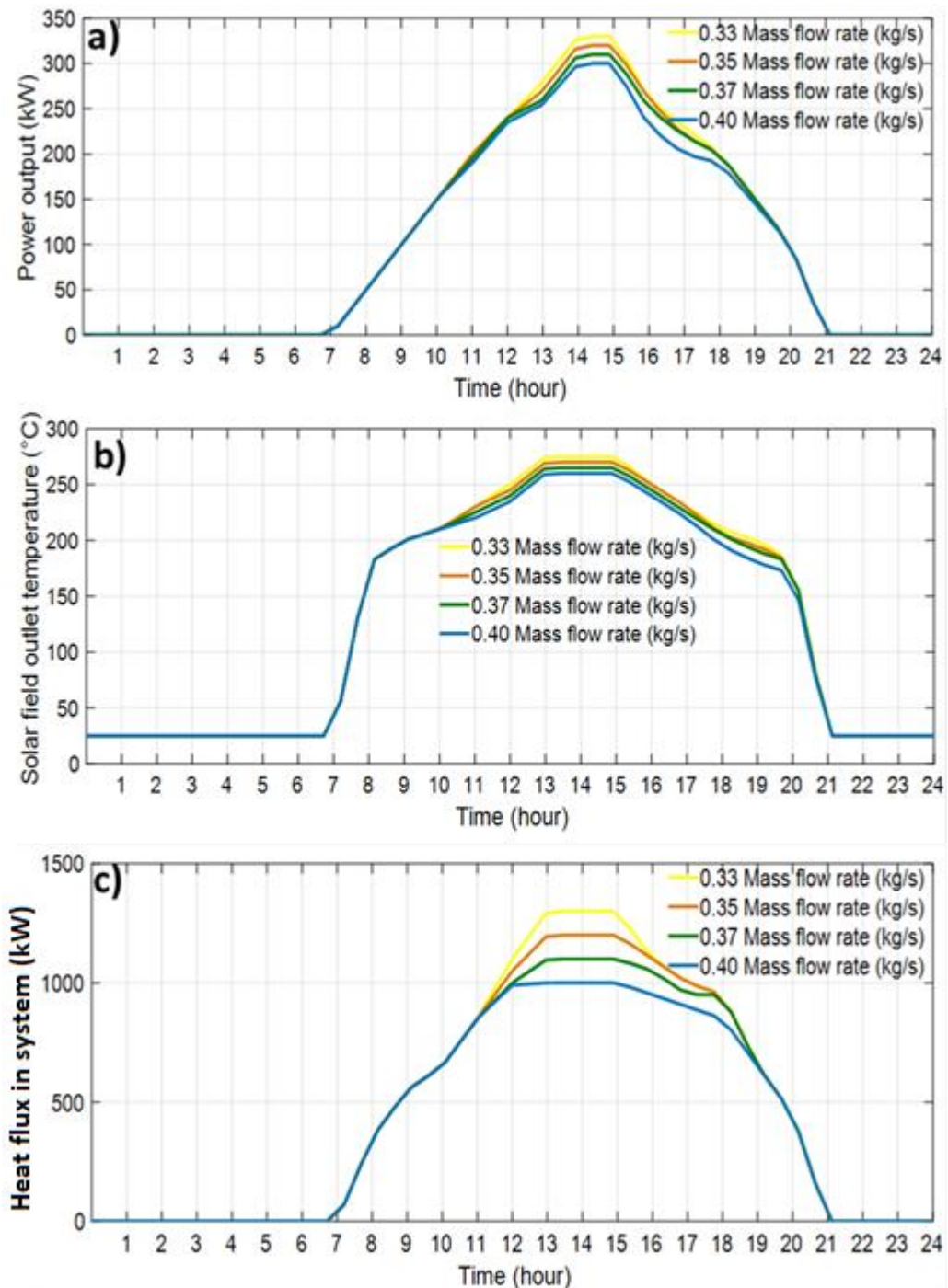


Figure 3.24: Effect of the nominal mass flow rate on the system, (a) power output, (b) solar field outlet temperature and (c) thermal energy

The mass flow rate also affects the solar field outlet temperature in the system; however, that effect is not as strong as that on the net power output. As shown in Figure 3.24b, this effect of the mass flow rate slightly changes the solar collector outlet temperature from 268 to 277°C due to the solar collection area being kept constant in

the system as well as control strategy of the value of liquid the volume in the boiler region which has been applied. From Figure 3.24c, it can be seen that the mass flow rate clearly impacts on the heat flow rate in the system. The peak heat flow rate can be achieved with a mass flow rate of 0.33 kg/s when the solar irradiance is at its peak of 1,000 W/m².

3.7.3.3 Influence of the turbine inlet-pressure

One of the key factors influencing turbine performance is the steam inlet pressure of the turbine. In this analysis, the pressure control valve was used to control inlet turbine pressure. The valve set up to be opened on different phases such as from 0.9 to maximum value 3MPa, 0.9 to 3.5MPa, 0.9 to 4MPa and 0.9 to 4.5 MPa. The steam consumption increases and turbine efficiency decreases when the steam inlet pressure falls. In a condensing steam turbine and a back pressure steam turbine, a 10% increase in steam pressure will result in a 1% and 4% reduction in steam consumption, respectively. For a condensing steam turbine and a back pressure steam turbine, a 10% rise in pressure has an impact on efficiency of roughly 1.5% and 0.45%, respectively [112]. Figure 3.25a and b show the effect of the turbine inlet-pressure on both the net power output and system thermal efficiency. As can be seen, both the system thermal power efficiency and net power generation are improved significantly as the turbine inlet-pressure increases. The maximum efficiency is obtained at the maximum turbine inlet-pressure in the system. A range of 3 MPa to 4.5 MPa of steam turbine inlet-pressure was selected to see how these values affect the system performance. The figure shows that up to 320 kW of the net power output can be achieved when the turbine inlet-pressure is 4.5 MPa, while 310 kW of the power when the turbine inlet-pressure is set at 3 MPa under a solar irradiance of 1,000 W/m². Figure 3.25b shows

the effect of the turbine inlet-pressure on system thermal efficiency. The turbine inlet-pressure has a largely similar effect on system thermal efficiency.

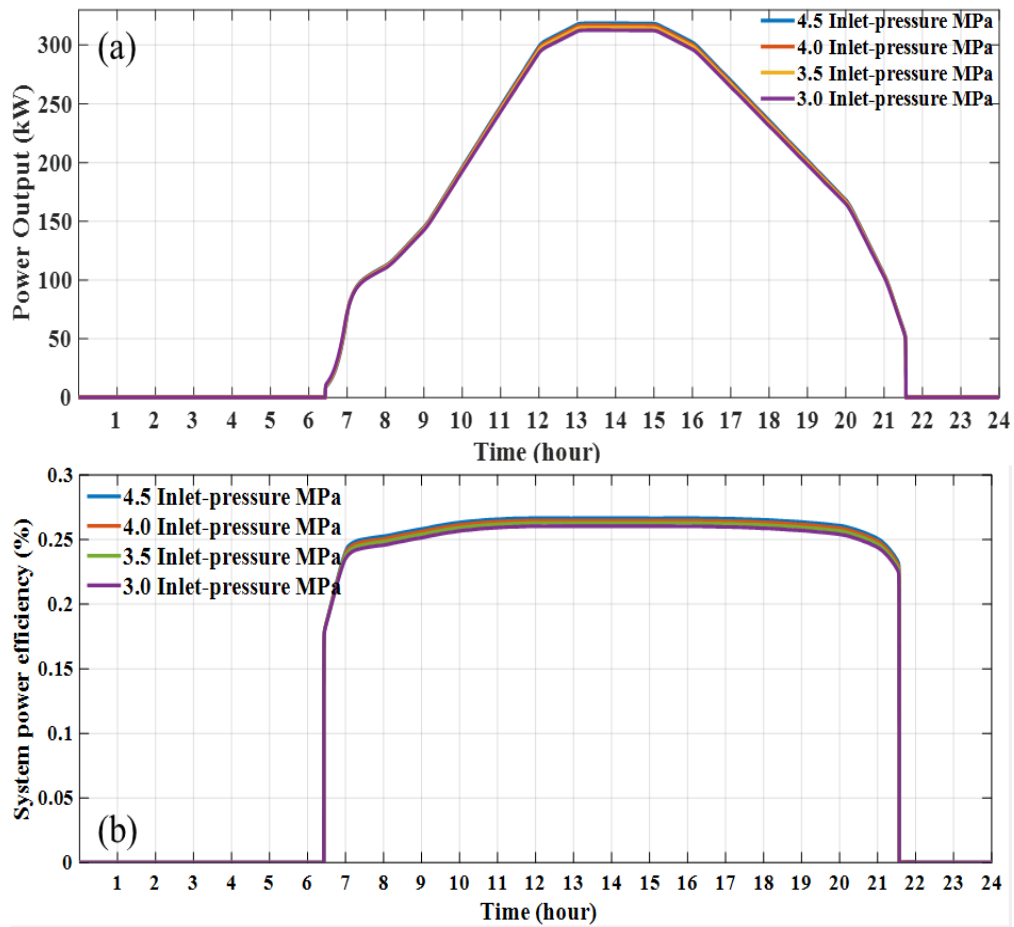


Figure 3.25: Effect of the turbine inlet pressure on the system, (a) power output and (b) solar power efficiency

3.7.3.4 Influence of the condensation temperature

Figure 3.26 shows the impact of the condensation temperature on the direct steam generation solar power system and overall system efficiency. The figure shows that the condensation temperature has a significant impact on the direct steam production solar power system and overall efficiency. According to the figure, as the condensation temperature rises, both the direct steam generation solar power system's and the overall efficiency decline. It is clear that the system can operate most efficiently at the lowest condensation temperature. The overall system's efficiency,

using water and steam, significantly increases from 23% to 26%, respectively, when the condensation temperature is reduced from 50°C to 20°C, as shown in Figure 3.26b. As a water wet condenser was selected for the system, the coolant temperature source is assumed to be come from the underground and ranges in temperature from 11°C to 40°C in the Libyan desert. The impact of the condensation temperature on the power output is depicted in Figure 3.26a. The reduction in the enthalpy difference in the steam turbine causes the net power production to fall as the condensation temperature rises. Figure 3.26a shows that 310 kW can be generated when the condensation temperature is 20°C, while 270 kW is generated if the condensation temperature is 50°C. If water is available in the installed location of a solar thermal power system, a water-cooled condenser is a significantly better choice compared with an air-cooled condenser, as it performs well.

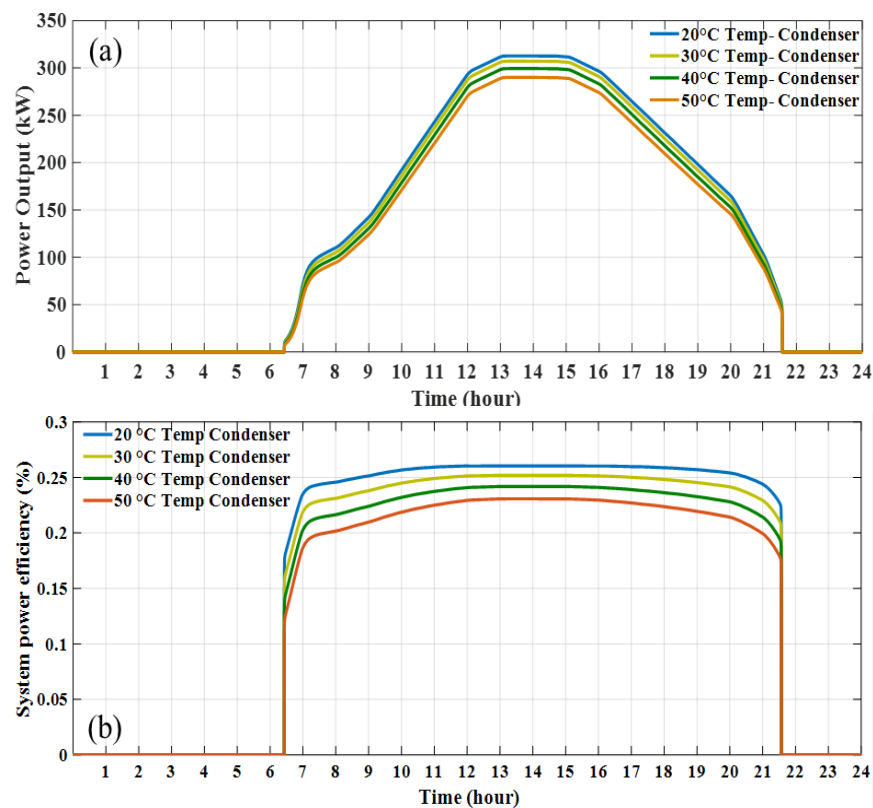


Figure 3.26: Effect of the condensation temperature on the system, (a) power output and (b) solar power efficiency

3.7.3.5 Influence of the solar collection area size

The effect of the collector types on the direct steam generation solar power system performance have been examined. However, in this analysis, only the IST-PTC collector type was considered. Four different collecting areas were tested as 990m^2 , 1056m^2 , 1320m^2 and 1544m^2 . Figure 3.27a, b and c show the effect of solar collecting area on the power generation, energy flow through the system and solar field outlet temperature.

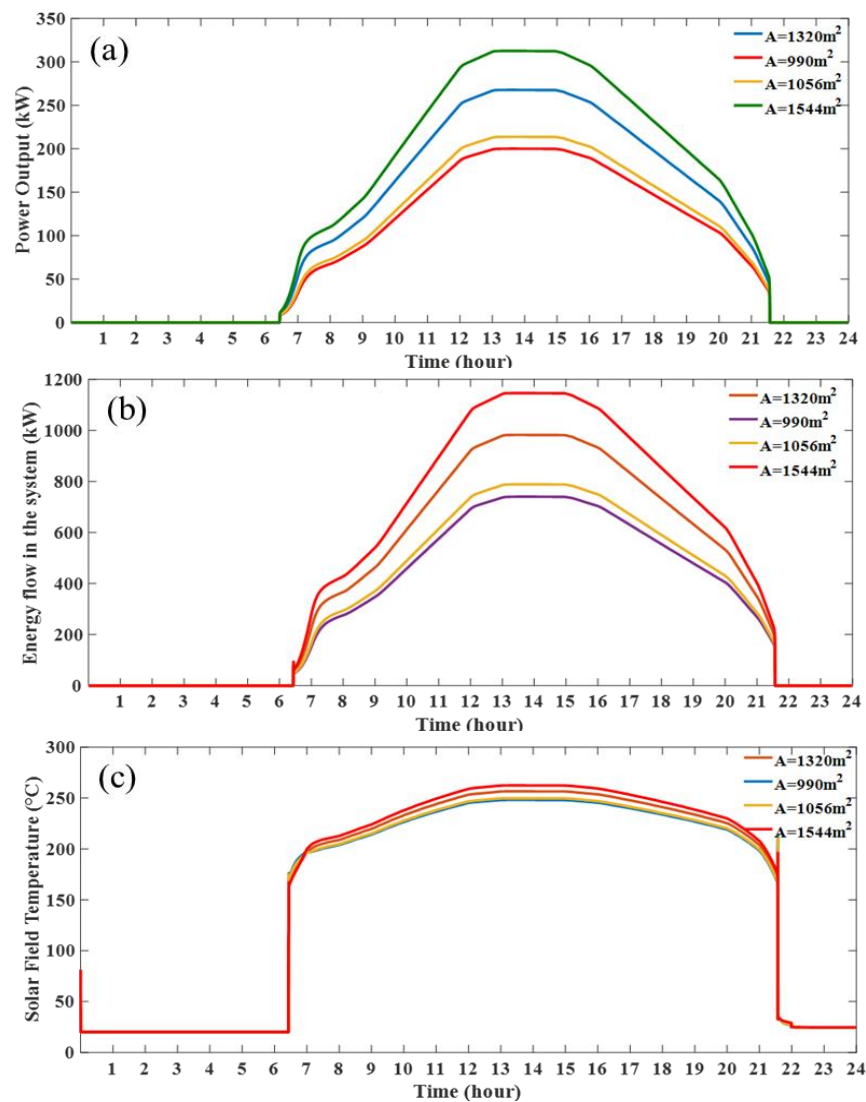


Figure 3.27: Effect of the solar collection area on the system, (a) power output, (b) thermal energy flow and (c) solar field outlet temperature

The results from Figure 3.27a reveal that the larger the area of the solar collecting, the higher the power output of the system. For example, 1544m² of solar collecting area can collect 1150 kW of the useful heat energy, which leads to generate 335 kW of the net power output in peak time of solar radiation as shown in Figure 3.28b. Also, the results indicate that 1320m² of solar collecting area generates 270 kW of electricity, followed by 1056m² of collecting area produces 210 kW, while 990m² only reached to 200 kW on the sunny typical day. From Figure 3.27c, and by keeping the parameters constant in all case of different collecting areas, the findings present that 1544m² of area, the maximum solar field outlet temperature reaches to 275 °C, while 1320m² of collection area the maximum solar field outlet temperature is 270 °C which is very close to the first case. On the contrary, the results reveal that 990m² of solar collecting area can only achieved to reach 240 °C of the maximum solar field outlet temperature under sunny typical day condition. From Figure 3.27a in this section, it can observe that 1320m² of solar collecting area is an appropriate area to meet electricity demand, and this collecting area will be constant in next chapters of the thesis.

3.8 A comparison study of direct and indirect steam generation solar power system under Libyan climate conditions

In this subsection, it was carried out comparison study between a direct and an indirect steam generation solar power systems under Libyan climate conditions in order to determine which the system is more appropriate and can be adopted in the future for Libyan government. In this analysis, it has been examined both systems based on parabolic trough collector technology for solar field, other types of solar collector were not considered. Our analysis in this section based on constant solar collecting area.

3.8.1 Cycles configuration

Figure 3.28a illustrates the schematic diagram of the proposed Direct steam generation system (DSG) driven by a parabolic trough collector. The (DSG) system consists mainly of solar collector, turbine, condenser and pump which are connected sequentially. In this DSG system, water is used as a heat transfer fluid and is heated up to the desired temperature in the heating circuit using the parabolic trough collector. The outlet temperature of the solar collector is reached to 250°C. The water (state 1) is first pumped to a high pressure (state 2) in the working fluid pump. The high-pressure fluid then passes through the evaporator where it is heated by the solar collector (PTC). During this process, the temperature of the steam is increased to its saturation temperature where it is fully converted to steam (states 3a, 3b and 3c). The superheat steam with high temperature and pressure then flows across the turbine to produce mechanical work which can be converted into electricity via a generator. The expanded steam (state 4) then passes through a condenser where heat is rejected to the cooling water. In this process, the steam is cooled down to water and then fully condensed (state 1). The water is then pumped again to repeat the cycle. In direct steam generation, the parabolic trough solar collector is divided into three parts, namely, preheating, boiling, and superheating.

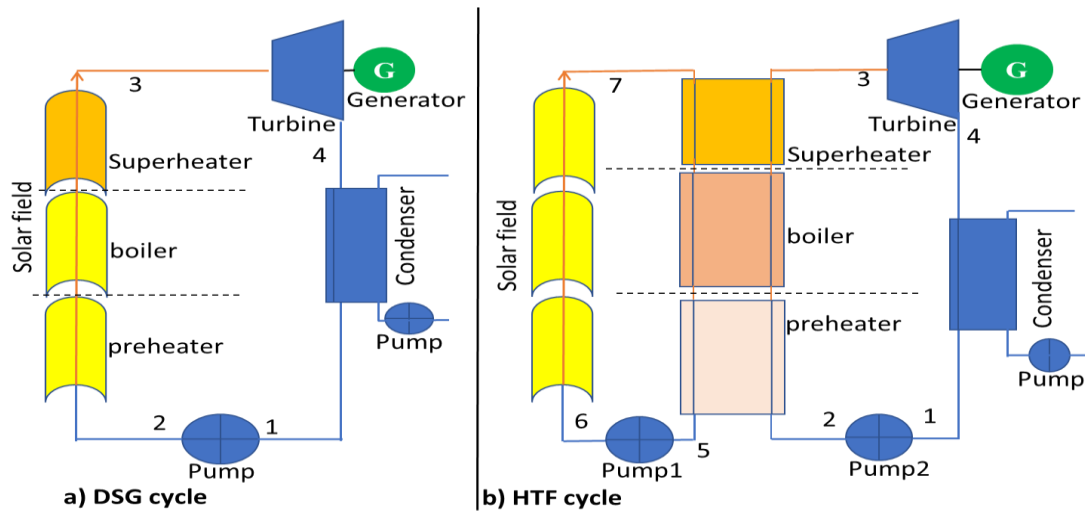


Figure 3.28: Layout of a) Direct steam generation system, and b) Indirect steam generation system

Figure 3.28b shows the diagram of the indirect steam generation system (HTF). The (HTF) system consists mainly of two circuits separate: solar field and Rankine cycle. The system consists mainly of include solar collector (PTC), (HTF) pump, heat exchanger, turbine, condenser and water pump which are connected sequentially. In this (HTF) system, water is used as working fluid in Rankine cycle and Thermo-oil is used as a heat transfer fluid (HTF) in solar collector. Thermo-oil is heated up to the desired temperature in the solar field. The outlet temperature of collectors is reached to 250 °C. The water at (state 1) is pumped to a high pressure (state 2) in the pump 2. The high-pressure fluid then passes through the evaporator (heat exchanger) where it heats from the solar collector (state 7, 5) by using Thermo-oil as heat transfer fluid in solar collector. During this process, the temperature of the working fluid is increased to its saturation temperature where it is fully converted water liquid to steam (states 3a, 3b, and 3c) in Rankine cycle. The vapour with high temperature and pressure then flows across the turbine to produce mechanical work which can be converted into electricity via a generator. The expanded exhausted steam (state 4) then passes through a condenser where heat is rejected to the cooling water. In this process, the steam is

cooled down and fully condensed (state 1). The cycle is repeat the again. The flow chart of this comparison study for both systems is shown in Figure 3.29 below.

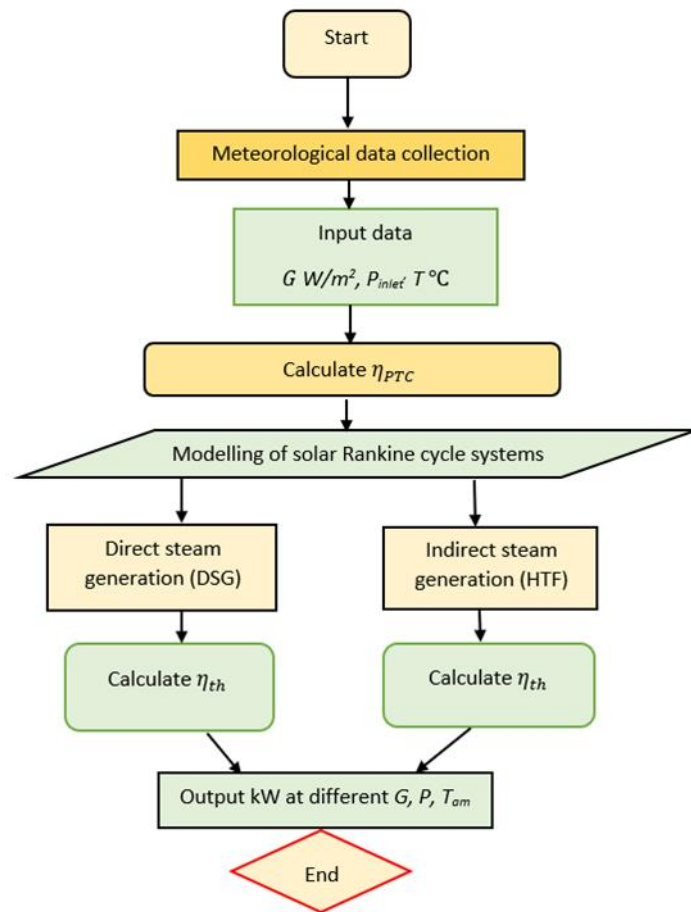


Figure 3.29: Flow chart calculation for both systems

The T-s diagram for both direct and indirect steam generation solar power system is displayed in Figure 3.30a and b below.

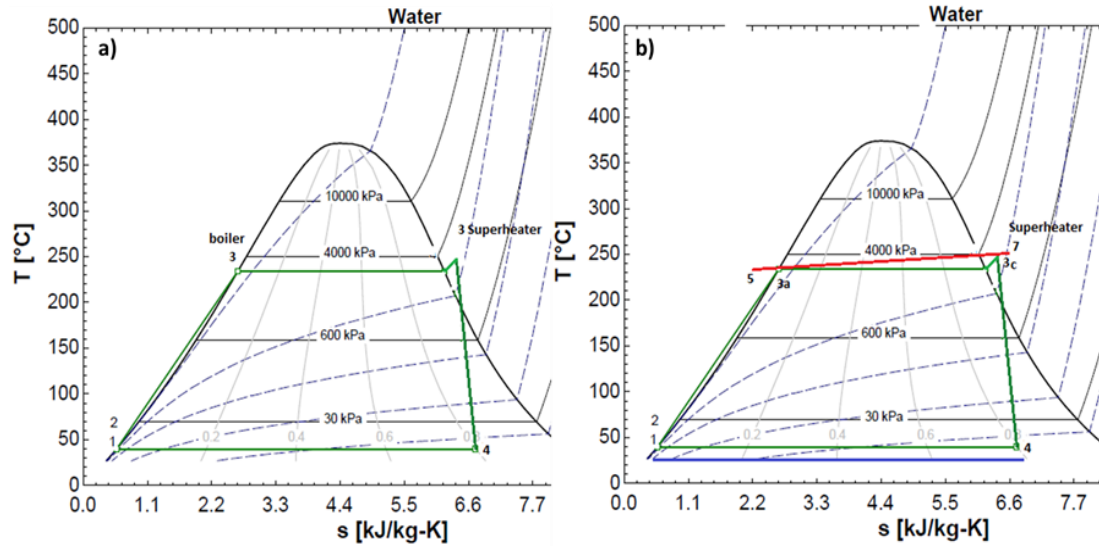


Figure 3.30: Corresponding T, s-diagram for a) Direct system, and b) HTF system

3.8.2 Hourly performance of direct and indirect steam generation solar power system

It is necessary to evaluate the 15th of July as it is considered the highest day in the month for direct and indirect steam generation in terms of outputs. The sun is available on this day for around 17 hours, which is considered a long time on a sunny day, starting from 6:00 am until 21:00 pm. according to meteorological data as shown in Figure 3.31. Both systems achieved very high results on this day, especially in the peak period, which starts from 10:00 am to 17:00 pm. The direct steam generation system provides 250 kW from 13:00 am to 14:30 pm., while indirect steam generation provides 240 kW at the same time, as shown in Figure 3.32. The solar beam irradiance reaches 1000 W/m^2 from 12:00 am to 14:30 pm. Also, the direct system has higher thermal efficiency compared with the indirect system. For example, the DSG reached 0.25 at the peak time of solar irradiance, while the maximum thermal efficiency of the indirect system reached 0.23 at the same time, as displayed in Figure 3.32 below. In the early morning, when solar irradiance is 350 W/m^2 , the DSG generates up to 50

kW, which is higher compared with the indirect system that generates 40 kW at the same time. System work when the solar irradiance is more than 300 W/m².

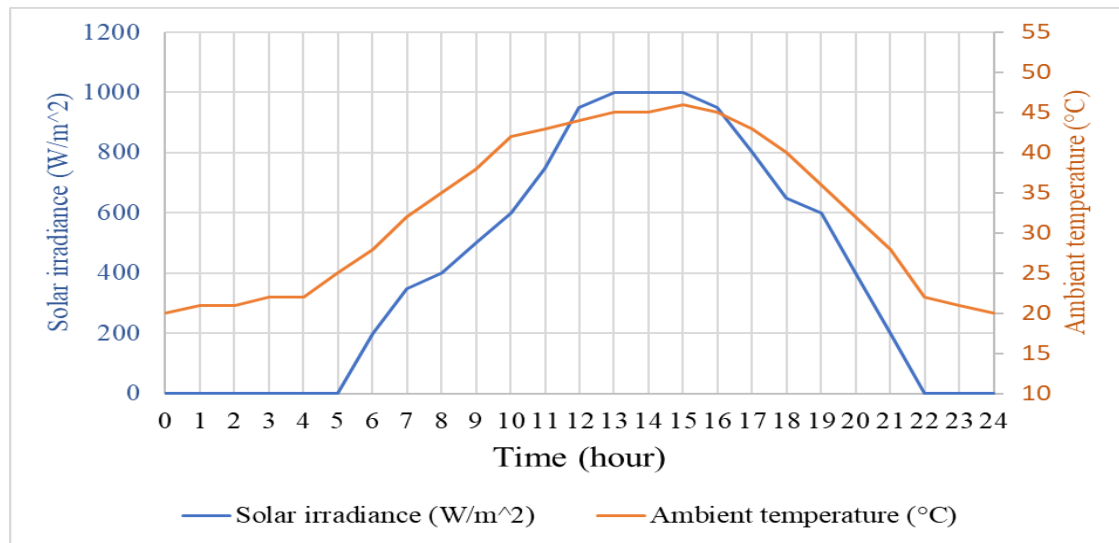


Figure 3.31: Hourly average temperature and solar irradiance on 15th July for MURZUG city [6]

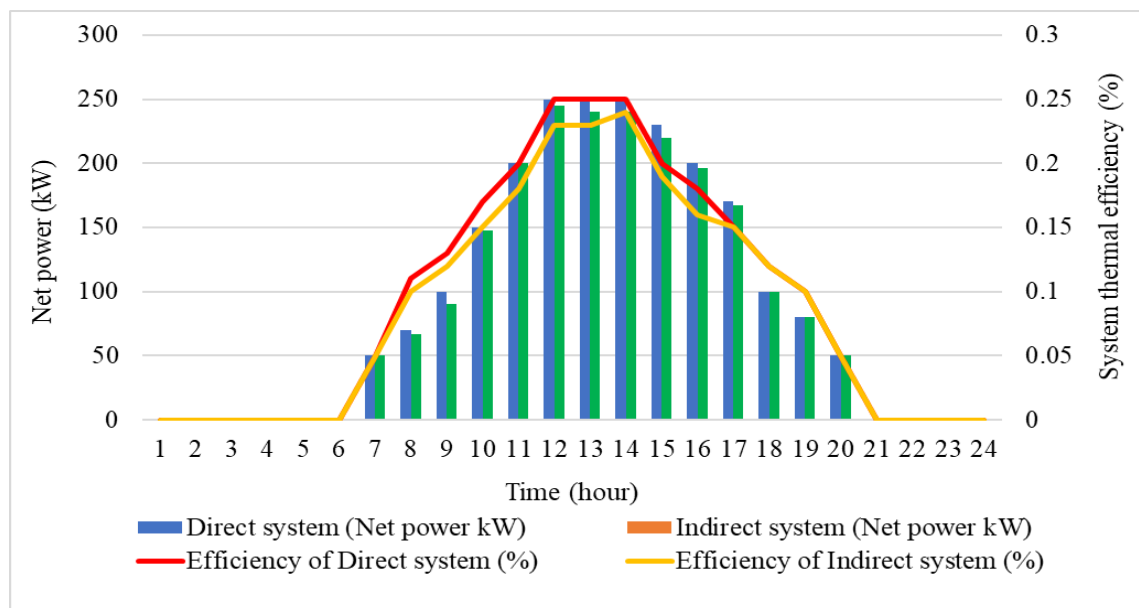


Figure 3.32: The hourly power outputs and system thermal efficiency of the direct and indirect steam generation system

During whole day of the simulation, the DSG has highest for both thermal efficiency and power generation compared with the indirect system. However, it is clearly to see

in figure 3.34, at time 7:00 AM, 18:00, 19:00 and 20:00 PM the power output for both systems are approximately same value. One reason behind that is the effect of ambient temperature on the indirect steam generation system which plays important role in both systems.

3.8.3 The monthly output of direct and indirect steam generation under Murzug city climate conditions

The warm months in South-Libya usually begins in April and ends in October. According to average of solar irradiance and ambient temperature [6], the hottest months in the year are July and then August for the selected city with average ambient temperature and solar irradiance of about 45 °C and 1000 W/m² respectively as showed in Figure 3.33.

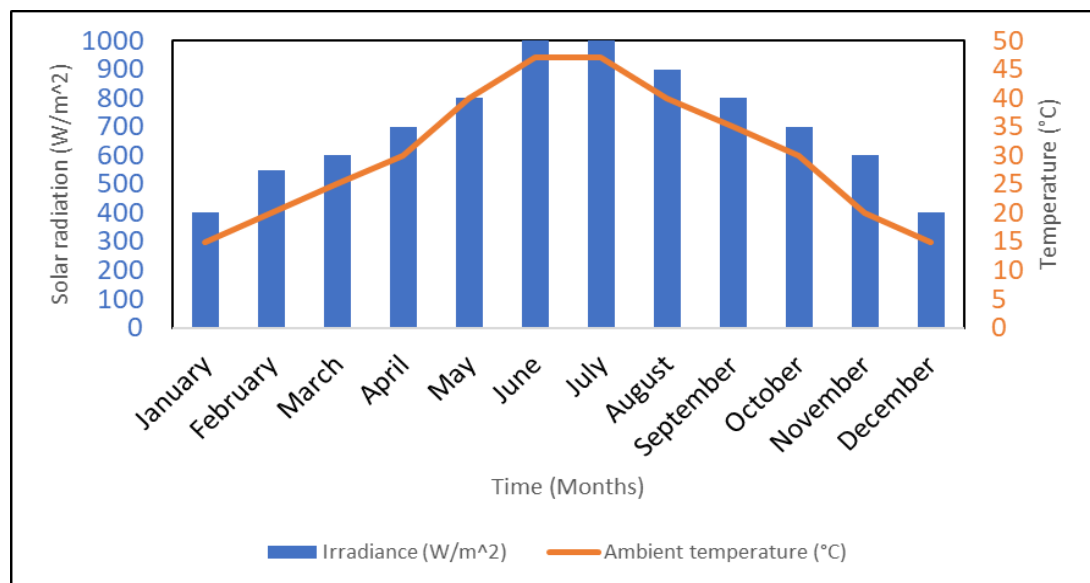


Figure 3.33: Monthly average temperature and solar irradiance in MURZG city [6]

The output of DSG and HTF systems started increasing in February and reached its highest output in July due to the high level of solar irradiance, which has an effect on the output of systems as shown in Figure 3.34. In July, output for direct steam

generation provided 255 kW and a thermal efficiency of 0.25, while indirect steam generation provided 240 kW and a thermal efficiency of 0.23 under the average solar irradiance of 1000 W/m². After that, the outputs for both systems decrease gradually from the ends of August to December due to the decrease in solar irradiance and ambient temperature during this period.

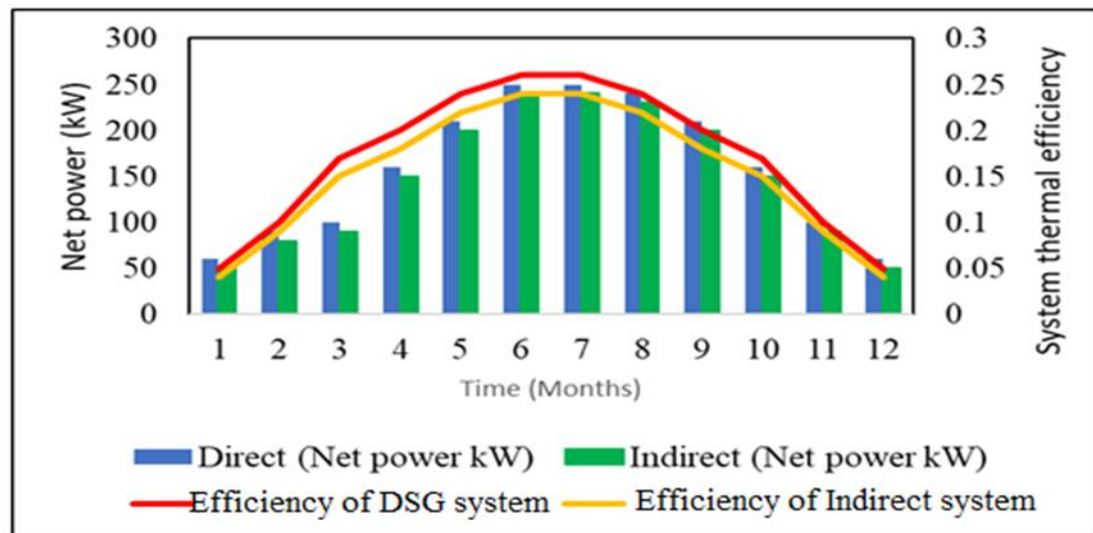


Figure 3.34: Monthly averaged outputs and thermal efficiency of direct and indirect steam generation system

This subsection provided basic criteria to assess the energy advantages of direct and indirect steam generation based solar thermal electricity systems under southwest Libyan climate conditions. It can be summarized that the direct steam solar power system has higher thermal efficiency and higher power output compared to the indirect system and it is an optimal system for Libyan climate conditions, which means that the DSG system will be viable for the Libyan government in the future.

3.9 Summary

This chapter details the research on the design and off-design performance of a direct steam generation solar power system powered by parabolic trough collectors with the aim of meeting the changing daily demand of Libyan public buildings. Without a heat storage tank, the daily power generation under this analysis provides good results. This chapter demonstrates how it is possible to control the mass flow rate of the circulation water depending on the liquid level in the boiler region to ensure that only superheated steam enters the turbine. The mass flow rate control approach has been used to model the direct steam generation solar power system under variable solar irradiation and ambient temperatures in southwest Libya. With a 1,320 m² collector area, the simulation's power output fluctuated between 250 to 270 kW during the day with 0.25 thermal efficiency. The following inferences can be made in light of the study that has been presented:

- It has been modelled and simulated the DSG system in Simulink/Simscape software in details under Libyan climate conditions.
- Validation of the proposed system with the TSE1 plant which is the first commercial power station has been made. The simulation results are very close to the power plant parameters.
- The model simulation shows that DSG is a feasible alternative to the conventional parabolic trough plants with thermal oil.
- The best performance of the system is reached when solar radiation is high and condensation temperature is low.
- The optimisations results show that 1,320m² of solar collecting area can produce 270 kW of power in a DSG system.

Therefore, in the next chapter, we will examine the proposed solar direct steam generation system integrated with a steam accumulator as a heat storage tank. The proposed system will be analysed and simulated under typical climatic conditions in Libya. The following simulation study will show how steam accumulators can affect system performance based on a new control strategy. It has been reported that steam accumulators are a promising technology for solar electricity generation systems based on direct steam generation to provide thermal energy for the Rankine cycle if there is a fluctuation in solar radiation during the day.

Chapter 4 Modelling and dynamic simulation of a direct steam generation solar power system with a steam accumulator

4.1 Introduction

To date, concentrating solar power (CSP) plants have become one of the most attractive technologies in the world. This is due to certain particular advantages, such as compatibility, high efficiency, eco-friendly and strong scaling-up potential [113]. However, due to the strong dependence on the available solar radiation resources, there appears to be a lack of continuity with electric power generation. To overcome this problem, beyond the auxiliary fuel backup system, the common practice is to incorporate a thermal energy storage (TES) system to store energy during periods of strong sunshine and release it under conditions of poor sunlight or at night. So, the TES system is a very important element of CSP plants, as it provides not only dispatchable electricity but also stability to the electric grid, and will greatly enhance the penetration of CSP technologies into the electric grid [114]. Under such conditions, the exploitation of an efficient, cost-effective TES system is especially important. An important way to study the performance of the TES system is to refer to dynamic modelling and simulation from the viewpoint of process analysis.

According to the motion state of the storage medium, the TES system can be broadly divided into two concepts: the active concept and the passive concept [115]. For the active concept, the storage medium is always moving during the operation. If subdivision is necessary, the active concept can also be divided into direct and indirect systems. For the direct system, the storage medium is also used as the heat transfer fluid (HTF). The TES system of a typical molten salt solar tower power plant belongs to this group of direct systems. For the indirect systems, the storage medium and HTF

are different, such as the popular solar parabolic trough power plants located around the world. At present, most of the installed solar parabolic trough power plants have adopted this concept [94]. For the passive concept, the storage medium is always immovable, and the concrete TES system is a typical passive TES system.

Since 2005, several small-scale experimental CSP plants have been successfully established with the financial support of the government in Yanqing CSP experiment base (40.4 N, 115.9E) in China, including: a 1 MWe Yanqing solar tower power plant with an active indirect TES system (using water/steam as the heat transfer fluid and synthetic oil as the storage medium) [116]; a 1MWe solar parabolic trough power plant with an active indirect TES system using oil as the heat transfer fluid and molten salt as the storage medium [117]; and a 1MW solar tower power plant with an active direct TES system (using molten salt as both the heat transfer fluid and storage medium) [118]. It is well known that the selection of TES system depends on the types of CSP plants, heat transfer fluid, cost of the storage medium and the local environment. Moreover, the TES system must be coupled appropriately with the power generating cycle, which provides an excellent opportunity to study the performance of the three different TES systems by conducting simulations and experiments.

The modelling of a TES system requires a profound knowledge of the thermal and fluid dynamic phenomena involved. In the last few decades, many scholars have conducted much significant work concerning the TES system of CSP plants. Generally, the previous work can be divided in two categories. The first concerns the modelling of a TES system from the of viewpoint of process analysis, including a static and dynamic analysis, operation modes under different working conditions, etc.

[119]. The other concerns the selection of the storage medium, storage concept design and optimization of TES systems [120].

Steam accumulators can store heat at either a sliding or a constant pressure. Several power plants have adopted this type of heat storage method, such as Puerto Errado power plants 1 and 2, Planta Solar 20 plant and Planta Solar 10 in Spain [121]. When heat is required to cover a limited insolation period at Planta Solar 10, steam is generated at variable pressures, from 4 MPa to the minimum pressure permitted by the system to operate the turbine at 50 percent partial load [97]. Saturated steam can be directly produced for power conversion using steam accumulators. This prevents the superheated area region in the solar collectors' fields, which is normally a difficult part of the control systems, where the steam accumulator in this kind of design also serves as the phase separator. Using this kind of accumulator storage can guarantee that only saturated steam enters the steam turbine [122,96]. One of the best commercial solutions regarding heat storage in a direct steam generation system is a steam accumulator [123], as this type of the heat storage unit has several advantages, such as the rapid evaporation and condensation of water/steam under non-equilibrium conditions, its high discharge rates, and fast reaction times, so it is considered the best solution for heat storage in a direct steam generation system when the solar irradiance fluctuates or is unavailable [124]. Using several methods, steam accumulators can be built into a solar Rankine cycle, as shown in Figure 4.1 [125].

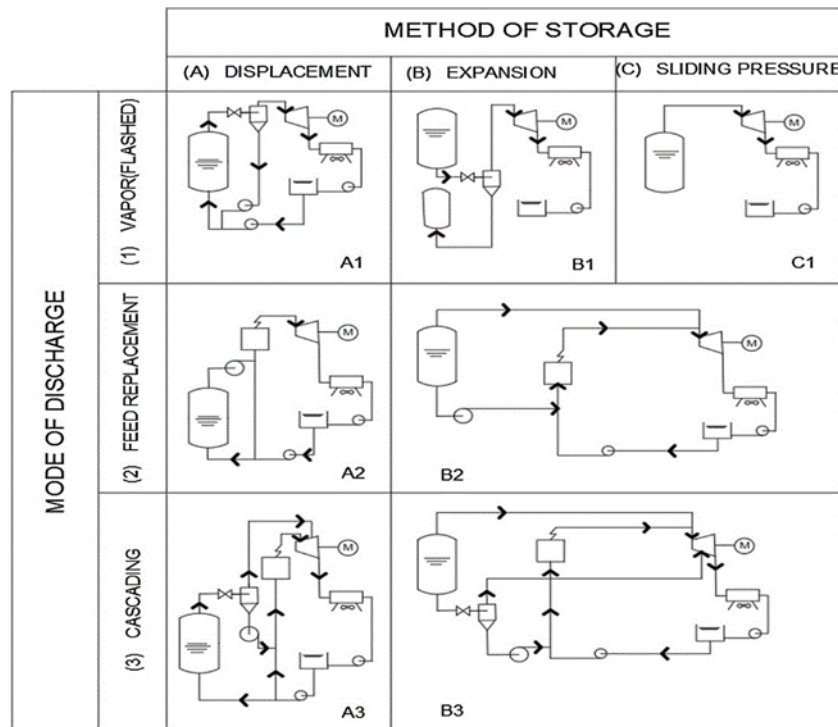


Figure 4.1: Main direct storage system variants for steam power plants, combining storage techniques and discharge modes [125]

4.2 Characteristics of steam accumulators

As mentioned in section 4.1 above, steam accumulators were used in the first commercial demonstration CSP plant PS10 in Spain, with a thermal capacity of 20 MW, which is equivalent to an effective operational capacity of 50 min at a 50% rated load [126]. However, the steam accumulators can only provide saturated steam. The warm water will remain at the bottom of the steam accumulator once the cycle finishes. This warm water can be used for many purposes, such as heating and cooling or in refrigeration systems. If superheated steam is needed, an additional system is required to heat up the saturated steam from the steam accumulator [127]. Moreover, steam accumulators have been employed to upgrade conventional power plants, and have demonstrated an ability to enhance the performance of these plants [128]. The characteristics of the steam accumulators can be listed as follows;

Pressure inside the steam vessel increases by time when the accumulator is charged. The difference between the initial and the operating pressure influences how much energy the steam accumulator releases. The steam flow and pressure affect the steam accumulator's regenerative rate, the higher the flow and pressure, the higher the regenerative rate. As a result, while the steam accumulator is in discharge mode, the pressure and temperature inside it decreases over time. A schematic diagram of a steam accumulator is shown in Figure 4.2.

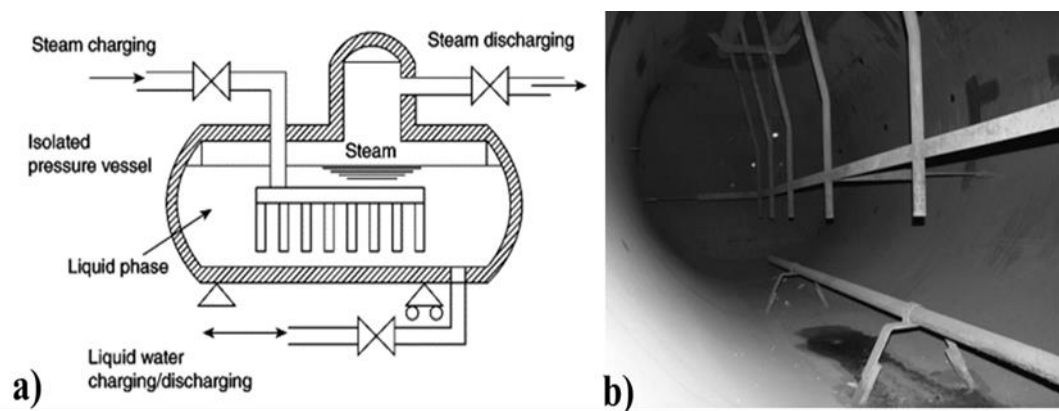


Figure 4.2: Schematic of a) steam accumulator layout and b) a view of the interior of a steam accumulator [128]

4.3 Configuration of solar power system with a steam accumulator

In this section, the proposed system will be described. Figure 4.3 shows the main configuration of the solar direct steam generation system coupled with a steam accumulator in a Simscape environment. The system consists of a solar collector field, steam accumulator, steam turbine, condenser, and pump. A steam accumulator is used as a heat storage unit to provide heat input to the Rankine cycle in a controllable manner. Due to the high thermal efficiency of parabolic trough collectors (PTC), they were chosen and used to heat the working fluid, which results in a temperature rise and steam generation, as mentioned in Chapter 3. The generated steam is to be stored in the steam accumulator and then drives a steam turbine to produce electricity. The

steam accumulator allows the solar Rankine cycle (SRC) system to be driven when the solar radiation is insufficient or unavailable.

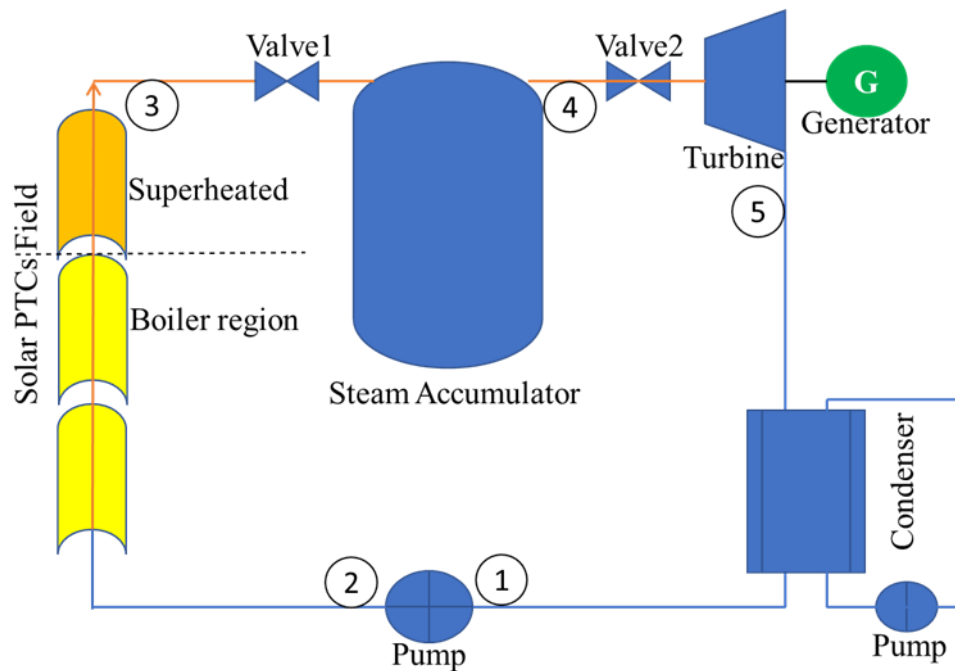


Figure 4.3: Schematic of direct steam generation solar power systems integrated with a steam accumulator

For its practical operation, there are two modes:

- The system needs to generate electricity and solar radiation is available. In this mode, valves 1 and 2 are open and the pump is run. The working fluid (water) is heated and vaporized through the collectors under high pressure and high temperature. The vapor flows into the steam accumulator to charging the accumulator and then steam into the steam turbine, exporting power in the process due to the pressure drop. The outlet vapor is cooled and condensed into a liquid state in the condenser with low pressure and low temperature, based on an ambient temperature level. Then, the liquid is pressurized using the pump. The working fluid is then sent back to the collectors.

- The system needs to generate electricity, but solar radiation is very weak or unavailable. Valve 2 is open, the pump is running, and valve 1 is closed. Heat is released from the steam accumulator and converted into power by the Rankine cycle and then the exhaust steam from the turbine will be cooled and accumulated in the liquid drum in the condenser which is 10 m^3 in this study. This mode, named the discharging steam method, may be used at night or to cover periods of fluctuating solar irradiance.

4.4 Thermodynamic modelling

For the modelling of the solar parabolic trough collectors, condenser, pump and steam turbine, the same methodology and equations are utilized as in Chapter 3. As mentioned in the previous chapter, only the IST-PTC solar collector type is used for the analysis. Here, only the equations related to the pertinent parts of the steam accumulator and the throttle valve are presented. A similar control strategy for the liquid level of the boiler region in the solar field was used in this model.

4.4.1 The steam accumulator model

A steam accumulator is a pressurized shape vessel and may be placed either horizontal or vertical. There are three ports on this steam accumulator, two for the inlet and outlet vapor phase fluids and one for the outlet liquid phase fluid, as shown in Figure 4.4. The steam accumulator is coupled with PTCs and its model is described in this section. The schematic of steam accumulator in Simscape software is shown in Figure 4.4.

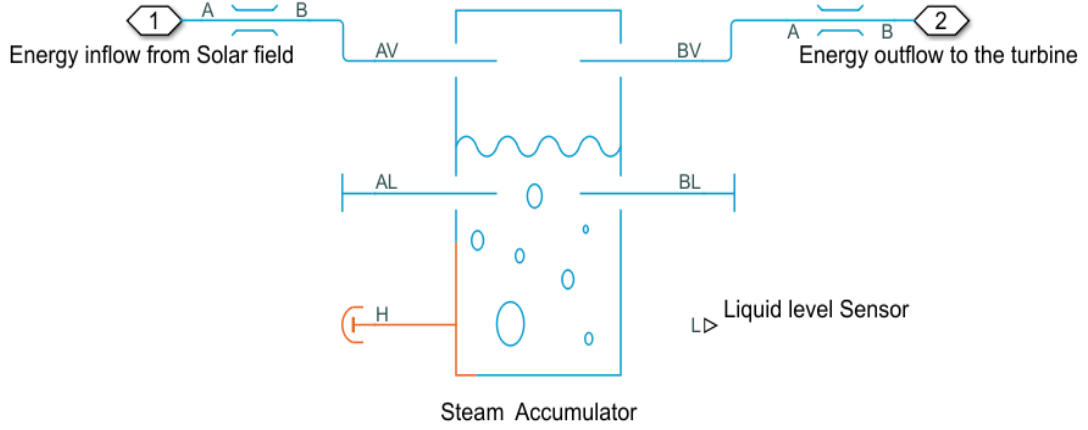


Figure 4.4: The schematic of steam accumulator in Simscape software.

The steam accumulator is considered as an important component of this system, where it is used as a heat source for the direct steam generation solar power system when solar irradiance is unavailable or weak. The energy capacity of the accumulator can be determined by its volume. The accumulator form is rigid, which means that its total volume is fixed during simulation. As the liquid and vapor inside can undergo phase changes, however, their respective volumes, and therefore their masses also, will generally vary over time. The liquid mass balance in the accumulator is given by Equation (4.1):

$$\frac{dM_L}{dt} = \dot{m}_{L_{in}} - \dot{m}_{L_{out}} + \dot{m}_{con} - \dot{m}_{vap} \quad (4.1)$$

Where, $\dot{m}_{L_{in}}$ is the liquid mass inlet to the accumulator, $\dot{m}_{L_{out}}$ is the liquid mass flow rate outlet of the accumulator, and $\dot{m}_{con}$ and $\dot{m}_{vap}$ are the mass exchange rates for condensation and vaporization at the liquid and vapor interface. Similarly, the mass balance in the vapor zone can be defined by Equation (4.2):

$$\frac{dM_v}{dt} = \dot{m}_{v_{in}} - \dot{m}_{v_{out}} + \dot{m}_{con} - \dot{m}_{vap} \quad (4.2)$$

Where, $\dot{m}_{v_{in}}$ is the vapor mass inlet to the accumulator, $\dot{m}_{v_{out}}$ is the vapor mass flow rate outlet of the accumulator, and $\dot{m}_{con}$ and $\dot{m}_{vap}$ are the mass exchange rates for condensation and vaporization at the liquid and vapor interface. The total mass flow rate for the liquid and vapor areas can be calculated by Equations (4.3) and (4.5) respectively:

$$\dot{m}_{L_{out}} = -(\dot{m}_{AL} + \dot{m}_{BL}) \quad (4.3)$$

Where, $\dot{m}_{AL}$ is inlet liquid mass flow rate to the accumulator and $\dot{m}_{BL}$ the outlet liquid mass flow rate of the accumulator in the liquid zone only. Similarly, in the total vapor mass flow rate for the vapor area:

$$\dot{m}_{V_{out}} = -(\dot{m}_{AV} + \dot{m}_{BV}) \quad (4.4)$$

Where, $\dot{m}_{Av}$ is the inlet vapor mass flow rate to the accumulator and $\dot{m}_{Bv}$ is the outlet vapor mass flow rate of the accumulator in the vapor zone. If only liquid remains in the tank, the total mass flow rate out of the liquid region can be calculated by Equation (4.5):

$$\dot{m}_{V_{out}} = -(\dot{m}_{AL} + \dot{m}_{BL} + \dot{m}_{AB} + \dot{m}_{BV}) \quad (4.5)$$

The vapor and liquid region inside can exchange heat with the wall and, through this combined with the effect of environment factors, the accumulator is not insulated. The energy flow through the ports, phase change and heat exchange allow the accumulation of specific internal energy in the accumulator. The accumulation energy balance for the liquid region can be calculated by Equation (4.6):

$$M_L \frac{du_L}{dt} + \frac{dM_L}{dt} u_L = \phi_{Lin} - \phi_{Lout} + \phi_{con} - \phi_{vap} + \dot{Q}_L \quad (4.6)$$

Where, ϕ_{Lin} and ϕ_{Lout} are the energy flow rate inlet and outlet of the liquid region.

$\dot{Q}_L$ is the heat exchange rate between the tank wall and the fluid in the liquid region.

The accumulation energy balance for the vapor region can be calculated by Equation (4.7):

$$M_V \frac{du_V}{dt} + \frac{dM_V}{dt} u_V = \phi_{vin} - \phi_{vout} + \phi_{con} - \phi_{vap} + \dot{Q}_V \quad (4.7)$$

Where, ϕ_{vin} and ϕ_{vout} are the energy flow rate inlet and outlet of the vapor region.

$\dot{Q}_v$ is the heat exchange rate between the tank wall and the fluid in the vapor region.

The total energy flow rate for the liquid and vapor region are calculated by Equations (4.8) and (4.9):

$$\phi_{Lout} = -(\phi_{AL} + \phi_{BL}) \quad (4.8)$$

$$\phi_{vap} = -(\phi_{AV} + \phi_{BV}) \quad (4.9)$$

In this accumulator, ϕ_{AL} and ϕ_{BL} are the inlet and outlet energy flow rates in the liquid zone. ϕ_{AV} and ϕ_{BV} are the inlet and outlet energy flow rates in the vapor zone. In terms of Liquid-Vapor Phase Change:

$$\phi_{vap} = \begin{cases} \frac{M_L (h_L - h_{Lsat})}{\tau}, & h_L \geq h_{Lsat} \\ 0, & h_L < h_{Lsat} \end{cases} \quad (4.10)$$

Where, M_L is the total mass in the liquid region and τ is the vaporization and condensation time constant block parameter. The subscript $Lsat$ is the saturation value for the liquid phase. The energy exchange rate due to vaporization determines the mass exchange rate between the fluid zones as liquid transforms into vapor and can be determined by Equation (4.11):

$$\dot{m}_{vap} = \frac{\phi_{vap}}{h_{v,sat}} \quad (4.11)$$

Where the subscript v,sat is the saturation value for the vapor phase. The condensation energy can be calculated by Equation (4.12):

$$\phi_{con} = \begin{cases} \frac{M_V (h_V - h_{Vsat})}{\tau}, & h_V \geq h_{Vsat} \\ 0, & h_V < h_{Vsat} \end{cases} \quad (4.12)$$

Where, M_V is the total mass in the vapor region. The energy exchange rate due to condensation determines the mass exchange rate between the fluid zones as vapor transforms into liquid and can be calculated by (4.13):

$$\dot{m}_{con} = \frac{\phi_{con}}{h_{Lsat}} \quad (4.13)$$

The heat exchange between the fluid and the wall is of a convective nature in this accumulator. The total heat rate can be calculated as the sum of the individual rates between the liquid zone and its section of the wall, and between the vapor zone and its section of the wall:

$$\dot{Q}_H = \dot{Q}_L + \dot{Q}_V \quad (4.14)$$

Where, here $\dot{Q}_H$ is the heat flow rate from environment factors. The liquid zone and its fraction of the wall surface area is defined by Equation (4.15):

$$\dot{Q}_L = f_{VL} \cdot S_w \cdot \alpha_L (T_H - T_L) \quad (4.15)$$

Where, f_{VL} is the volume fraction of the accumulator occupied by the liquid region. The product of this fraction with the surface area of the wall gives the surface area available for heat exchange between the liquid and the tank wall. S_w is the surface available for heat exchange between the wall of the accumulator and the whole of the fluid inside, α_L is the heat transfer coefficient for the liquid zone and T is the average temperature at the location denoted in subscript H for the tank wall, with L for the liquid zone.

The vapor region and its fraction of the wall surface area can be calculated by Equation (4.16):

$$\dot{Q}_L = (1 - f_{VL}) \cdot S_W \cdot \alpha_V (T_H - T_V) \quad (4.16)$$

The correspondence between the volume and mass fractions of the liquid is defined by Equation (4.17):

$$f_{VL} = \frac{f_{ML} \cdot V_L}{f_{ML} \cdot V_L + (1 - f_{ML}) v_v} \quad (4.17)$$

Where, f_{ML} is the mass fraction of the liquid zone and v is the specific volume of the phase indicated in the subscript (L for liquid, V for vapor).

The total volume of accumulator (V) is a function of the diameter (D_i), thickness (δ) and height (H) of the pressure vessel. The design of the cylinder is correlated with the design pressure [129]. The volume of the cylinder can be calculated by Equation (4.18):

$$V_{cylinder} = V_w = \frac{3600 \cdot t_H \cdot \dot{W}_{net}}{\rho_w \cdot \eta_{SRC} \cdot C p_w \cdot \Delta T} \quad (4.18)$$

V_w is water volume. t_H is storage time in hour. $C p_w$ is the heat capacity of water. ΔT is the temperature drop in the discharge process.

4.4.2 Thermal efficiency of the proposed system

It is essential to determine the thermal efficiency of the system during the day, as the input conditions are changeable, such as the solar beam irradiance, pressure, and ambient temperature. The thermal efficiency of the system can be defined by Equation (4.19) and Equation (4.20), respectively:

$$\eta_{RC} = \frac{W_t - W_p}{\dot{Q}_{solar}} \quad (4.19)$$

In discharging mode, when the system works based on a steam accumulator, the thermal efficiency of the system can be expressed by:

$$\eta_{SRC} = \frac{W_{net}}{\dot{Q}_{heat\ released}} \quad (4.20)$$

Where here $\dot{Q}_{heat\ released}$ is the rate of heat released from the accumulator to the cycle.

4.4.3 Steam accumulator validation

The simulated results should be validated using experimental data in order to confirm the dependability and accuracy of the mathematical models created from the steam accumulator system in the Simulink/Simscape programme. Table 4.1 summarizes the key design factors of the steam accumulators considered in this work and the data extracted from the open literature review [128]. The experimental data of pressure variations inside the charging steam accumulator of reference [128] are used to simulate and analyse charging and discharging transients in the horizontal cylindrical steam accumulator, which has an outside length of 11.9 m, an outer diameter of 2.9 m, and a total internal volume of 64 m³. The operating range for the accumulator is 25 to 55 bars. The steam headers at the top of the interior of the accumulator vessel are where steam is fed into the accumulator through the perforated tubes that are immersed in the water volume. In each simulation run, it is assumed that water and steam are in a state of thermal equilibrium caused by the initial pressure.

Table 4.1: Specifications of a steam accumulator [128]

Parameters	Value
Number of steam accumulators	1
Steam accumulator volume (m ³)	64
Steam accumulator diameter (m)	2.5

Initial liquid volume in the steam accumulator (%)	86
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Figure 4.5 shows model validation using data taken from [128] for the charging of the steam accumulator. It is clear to see that the validation between the experimental and simulation data were very matching in each other. The case represents the charging procedure of a steam accumulator as exhibited in Figure 4.5. The comparison between the model and the results from [128] obtains a mean absolute percentage error (MAPE) of 0.18%. The pressure growth closely aligns with the referred one, and therefore, the accuracy of the recreation is verified and meets the desired requirements.

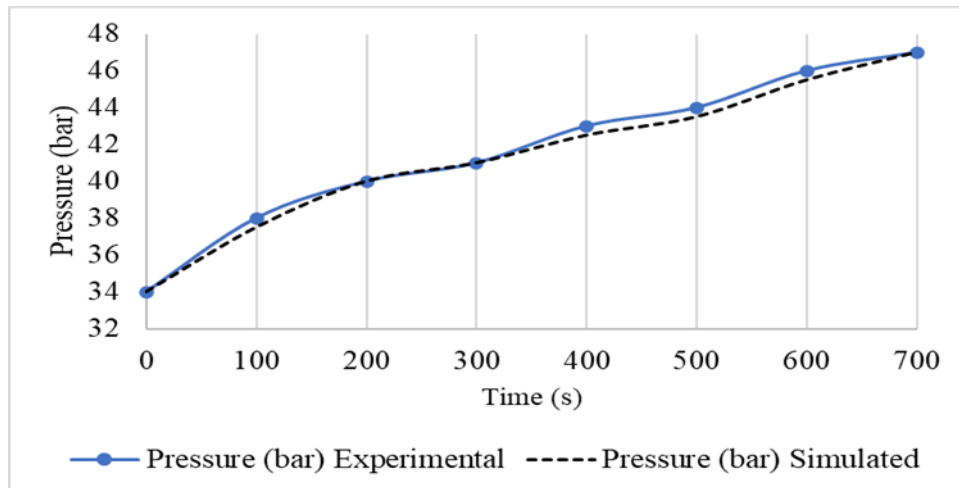


Figure 4.5: Model validation using data taken from [128] for the charging of the steam accumulator

It was found that there existed a strong agreement between the simulated and experimental outcomes. During the charging modes, the greatest deviation in the temperature and pressure appeared as extremely minor values. As a result, it can be concluded that the models provided were suitable and reliable for evaluating the behaviour of the steam accumulator in this study. The model shows good correlation with the experimental work. The small deviation in the current model as compared to [128] is due mainly to the approximation used to calculate the heat loss coefficients.

The maximum difference pressure value between the experimental work and the simulation model in Figure 4.5 for model validation is 1.2 bar at time 450 to 550s.

Figure 4.6 outlines the methodology employed in this chapter. In both systems, the design and off-design performance methodology are applied. As a first step, the steam accumulator is integrated with the steam Rankine cycle without using PTCs as the design condition, assuming inlet-pressures and temperatures. The mass flow low rate in this design was controlled based on the liquid volume fraction level in the condenser drum. Six hours of discharging method was simulated, and four different volume sizes of steam accumulators were evaluated. In the second step of the analysis, a steam accumulator integrated with a solar direct steam generation system was modelled under Libyan climate conditions. Under this design, the mass flow rate was controlled based on the liquid volume fraction of the boiler region in the solar field. The reason for this is that we only need the superheated steam to enter the steam accumulator. In this case, the steam accumulator must be filled with 90% of saturated steam, and a further 10% should be saturated liquid.

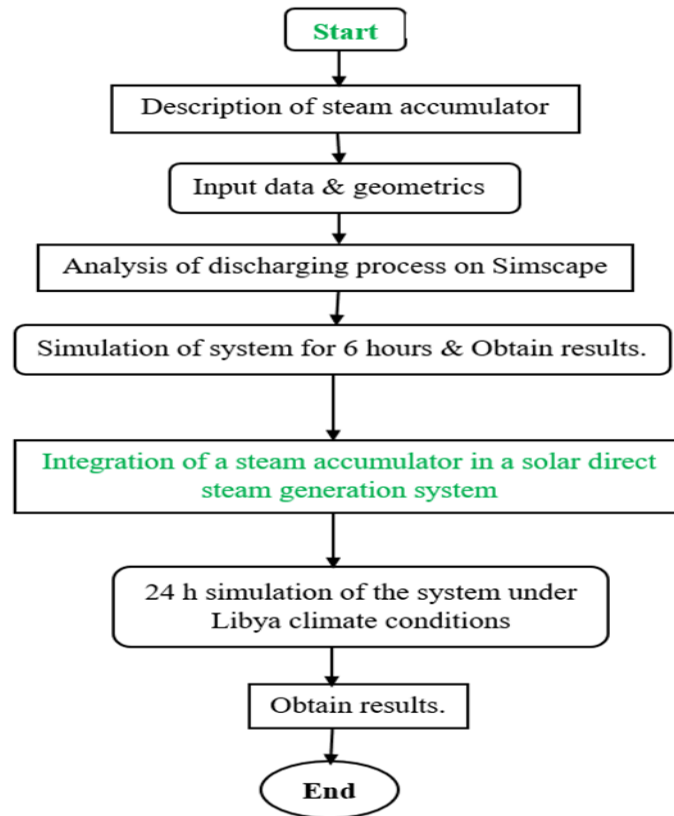


Figure 4.6: Outline methodology for the system simulation

4.5 Simulations results and discussion

In this section, the modelling results from integrating a steam accumulator into the steam Rankine cycle and the direct steam generation solar power system coupled with a steam accumulator as the heat storage unit are presented and discussed. First, we describe the integration of a steam accumulator with the Rankine cycle as the discharging model for six hours, followed by the effect of the steam tank volume on the net power output of the system. Then, the direct steam generation solar power system, coupled with a steam accumulator, are presented and analysed. A case study of the proposed system to provide energy to a Libyan hospital is modelled and presented in this section as well.

4.5.1 Discharging behaviour of steam accumulator coupled with a Rankine cycle

This section presents the physical and mathematical model of the application of a steam accumulator in a direct steam generation system, as shown in Figure 4.7. The main component of the steam accumulator is a horizontal cylindrical vessel of volume with the necessary inlet and outlet piping. The vessel is filled with steam under pressure. The phases are separated. Figure 4.7 also presents the steam inflows/outflows through the pipelines, which connect the steam accumulator with the direct steam generation system. During the charging phase, the steam with a mass flow rate enters the steam accumulator. The enthalpy of the inflow steam is equal to the enthalpy of the steam leaving the high-pressure turbine.

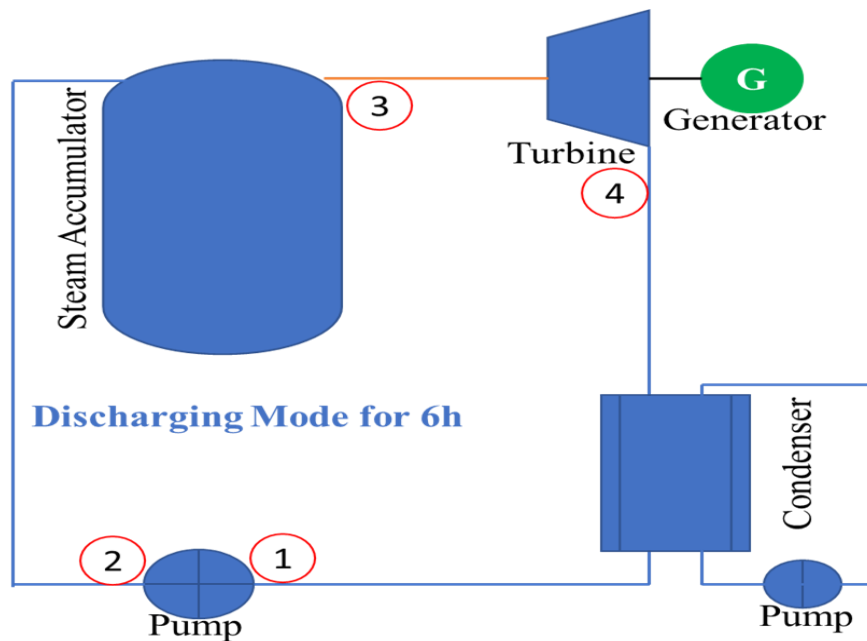


Figure 4.7: Schematic of a steam accumulator integrated with a Rankine cycle as the discharging mode

The initial parameters and geometry of the steam accumulator of the two-phase mixture in a steady state of the fully charged accumulator are presented in Table 4. 2, which provides the main data on the accumulator during the discharge method.

Table 4.2: Operating condition for a steam accumulator in this study

Internal volume	136 m ³
Pressure when fully charged	3 MPa
Internal diameter	4 m
Volume fraction of liquid water	0.05

There are four main components in this model, a steam accumulator, turbine, a condenser and a working fluid pump, as shown in Figure 4.8 below. This example was analysed for six hours (or 21,600s) and the mass flow rate was a variable value from 0.10 to 0.40 (kg/s) during the simulation. The characteristics of the discharge process for the steam accumulator integrated with the steam Rankine cycle are discussed in this section.

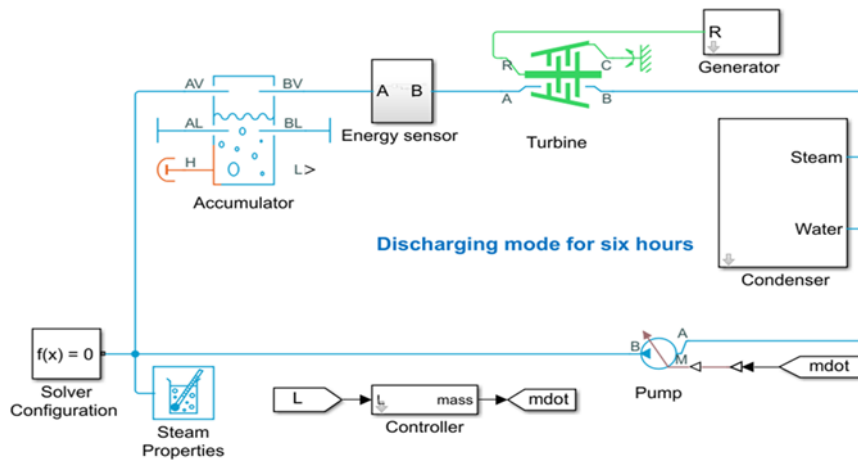


Figure 4.8: Schematic of a steam accumulator integrated with a Rankine cycle as the discharge mode in Simscape software

The results are based on the controlled mass flow rate with an initial value of 0.15 kg/s mode for the system. The total steam tank volume was set at 136 m³. For the discharge process of the steam accumulator, the steam pressure was assumed to be 3 MPa. This decreased slightly in 1 hour, and continued to decrease to 0.015 MPa during the six hours of analysis, as depicted in Figure 4.9a below.

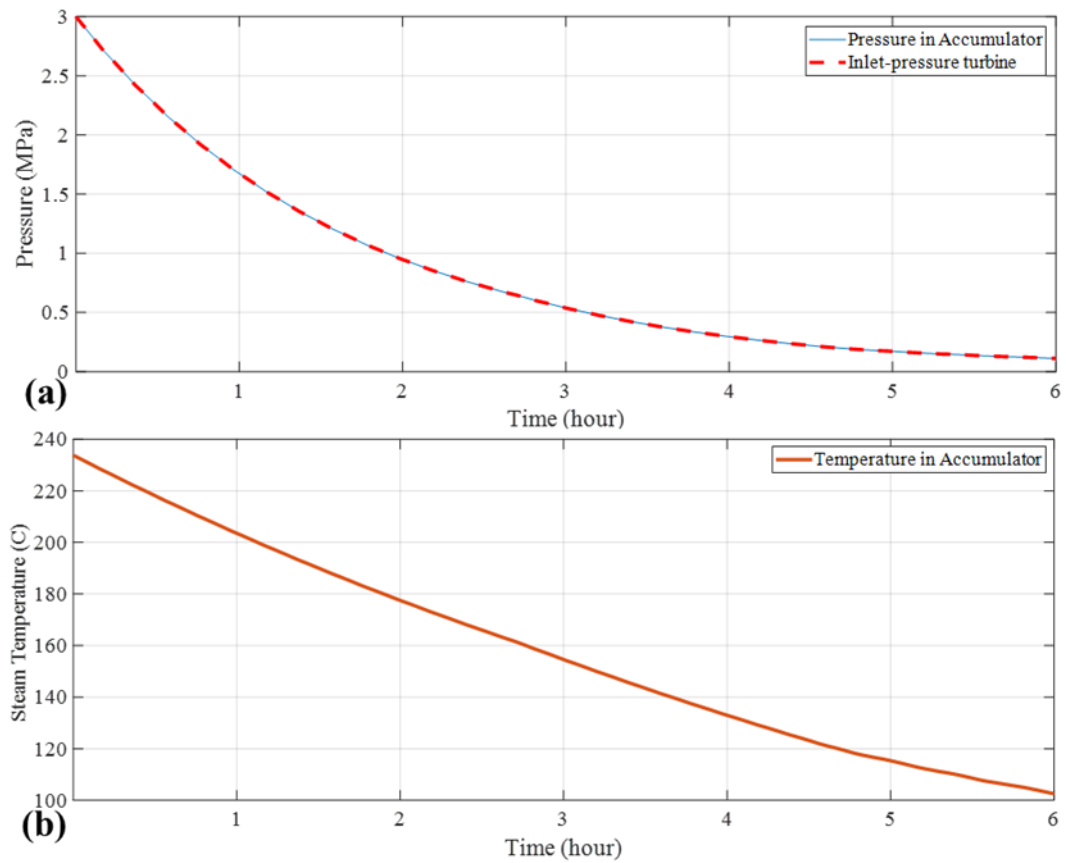


Figure 4.9: Evaluation of a) the pressure and b) the temperature during the discharge of the steam accumulator

The variations in the steam temperature during the discharge are shown in Figure 4.9b.

It is clear that the steam temperature decreases gradually, from 505 to 435 K, during the six hours of discharging, due to the decrease in the steam pressure leading to a decrease in the corresponding saturated water temperature.

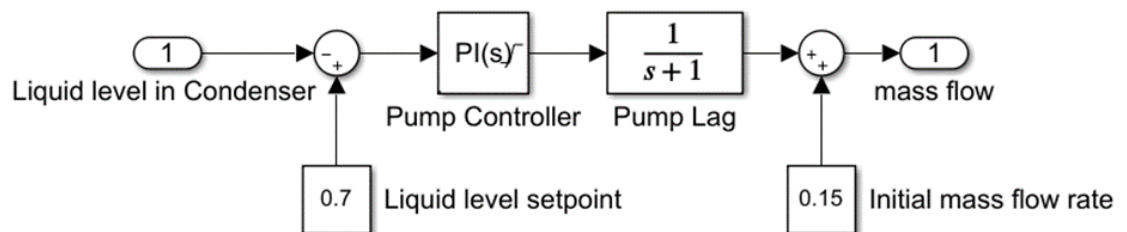


Figure 4.10: Control strategy for the liquid level in the condenser used during the discharge of the steam accumulator

Figure 4.10 shows the strategy for the mass flow control in the system during the discharge process. The mass flow rate is controlled according to the liquid level in the condenser for six hours, which is set at 0.7. The initial mass flow rate value is set at 0.15 kg/s as the design condition for this system. When the liquid level in the condenser reaches 0.7, the work fluid pump is run to pump the liquid to the steam accumulator again to keep the liquid level at the controlled level in the system, as shown in Figure 4.11a and b, respectively.

Figure 4.11a shows the mass flow rate for the turbine inlet and outlet mass flow of the working fluid pump. It is clear that the inlet-mass flow rate of the turbine decreases from 0.30 kg/s to 0.08 kg/s at the end of the discharge mode. Also, the initial starting value of the mass flow rate of 0.15 kg/s changes to the adjustable value 0.40 kg/s when the pump starts to run at the start of the test, then remains a variable value during the next five hours of operation, as shown in Figure 4.11a.

The liquid volume fraction level of the condenser was considered in this discharge process. The condenser temperature was assumed to be 25 °C, according to the average ambient temperature. Figure 4.11b indicates that the liquid volume fraction remains constant at 0.7 during the six hours of simulation. This is because we controlled the liquid level of the condenser and the working fluid pump depends entirely on the liquid level in the condenser as a control strategy of pump.

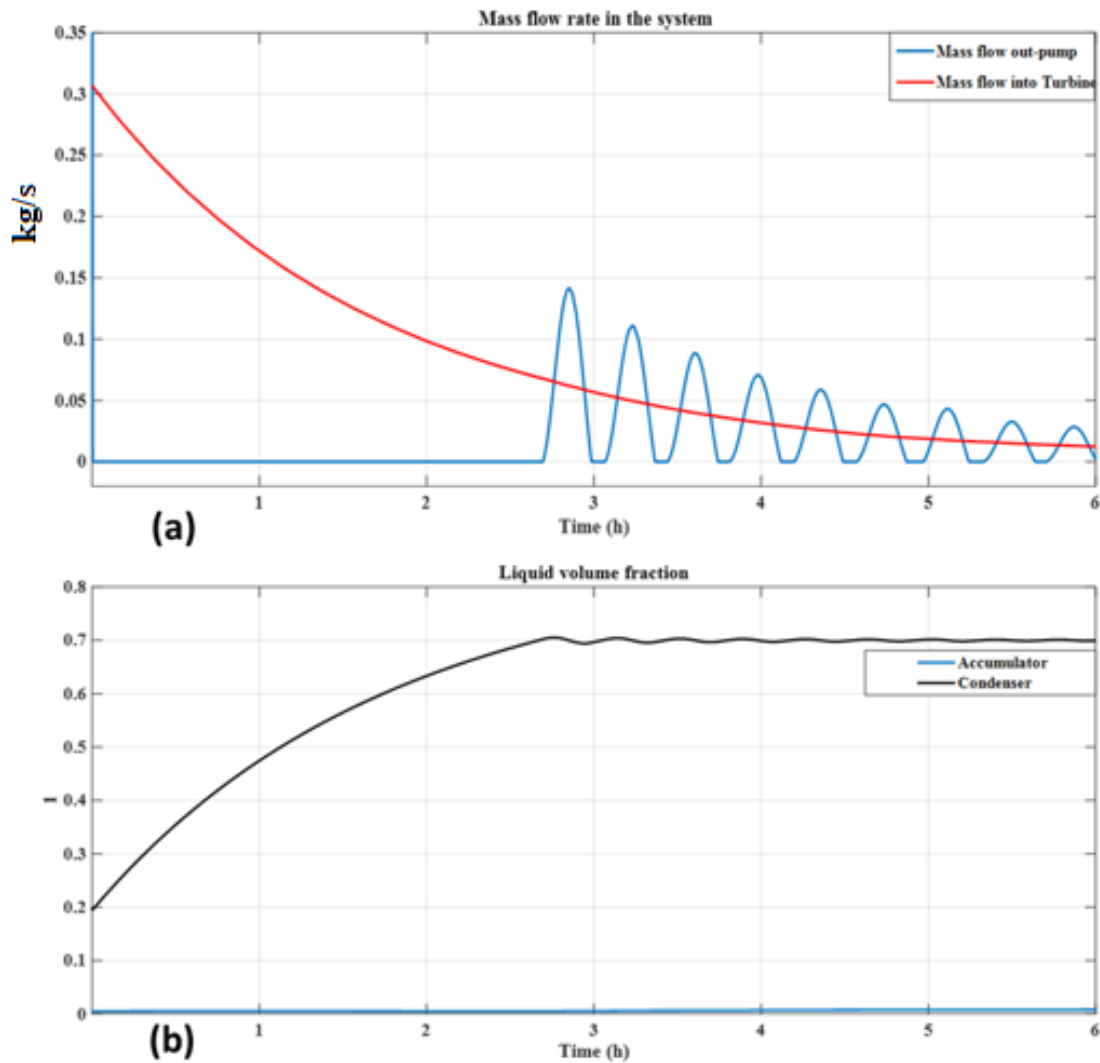


Figure 4.11: Variation of a) the mass flow rate, and b) liquid level in the steam accumulator and condenser drum

The liquid volume fraction in the steam accumulator increases during the discharge process due to the change in the mass vapor to mass liquid phases as the exhaust steam of the turbine became condensed in the condenser and is pumped back to the steam accumulator in liquid phase. Figure 4.11b shows that the liquid fraction started to increase from 0.001 to 0.05 during the six hours of analysis.

Figure. 4.12a shows that both the steam and water mass decreased and increased during the discharge process, respectively. The total charged steam mass was initially 2000 kg, which decreased to 0.5 kg during the six hours of analysis, as shown in Figure 4.12a. As the water filling coefficient increases, the heat exchange between the steam

and the water increases and the energy to be stored in the water decreases. Moreover, the liquid mass increases in the steam accumulator during the analysis, where the liquid mass level increases from 0.005 kg to approximately 1,800 kg, as shown in Figure 4.12a. Based on the variable mass flow rate of the pump during operation as shown in Figure 4.12b, the variable increase in the liquid mass in the accumulator can be observed.

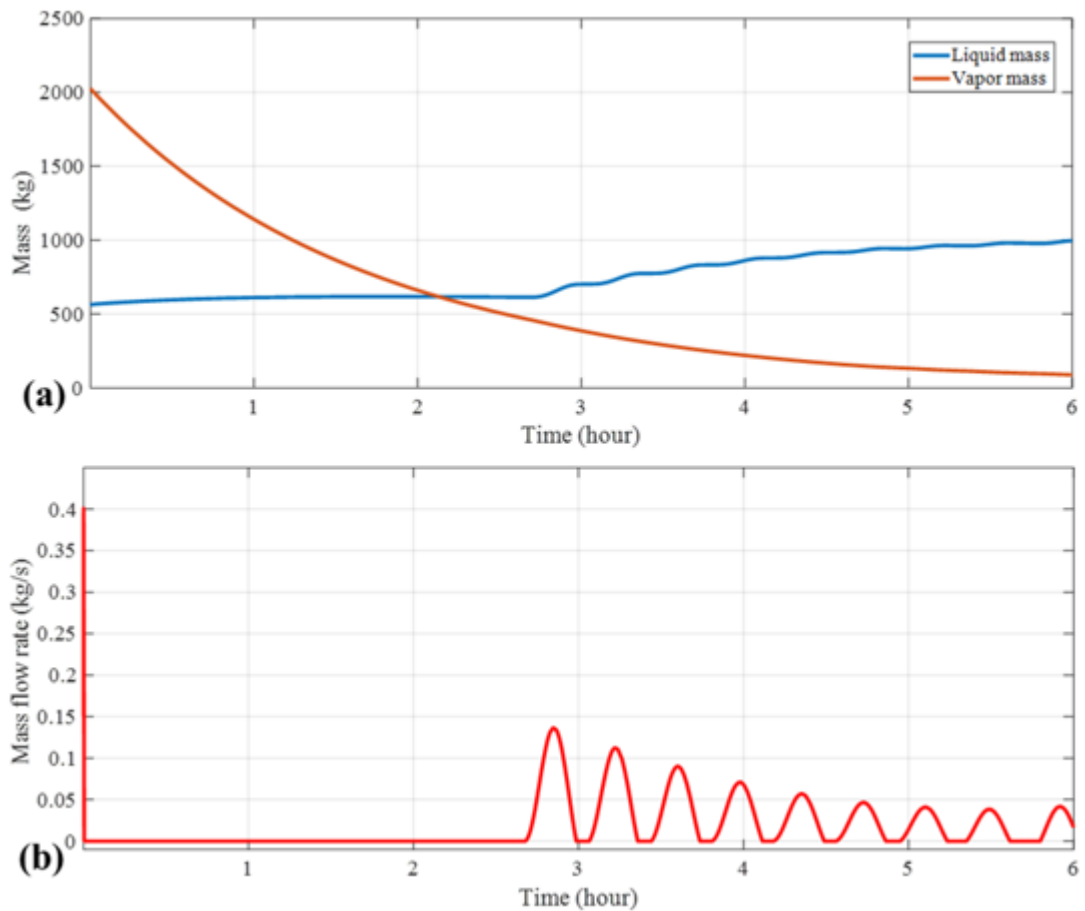


Figure 4.12: Variation of a) liquid and vapor mass in the steam accumulator, and b) mass flow rate of incoming liquid

Also, the fluid power of the turbine is 235 kW, and the mechanical power is 200 kW at the beginning. At the end of the discharge period, the mechanical power decreases to 35 kW and the fluid power drops to 45 kW due to the decreasing pressure and steam temperature in the steam accumulator as shown in Figure 4.13a.

The variations in the thermal efficiency of the system are indicated in Figure 4.13b during the six hours of the discharge method. It can be concluded that the biggest value of thermal efficiency is initially 0.24, which started to decrease during the discharge method time to reach 0.17 at the end.

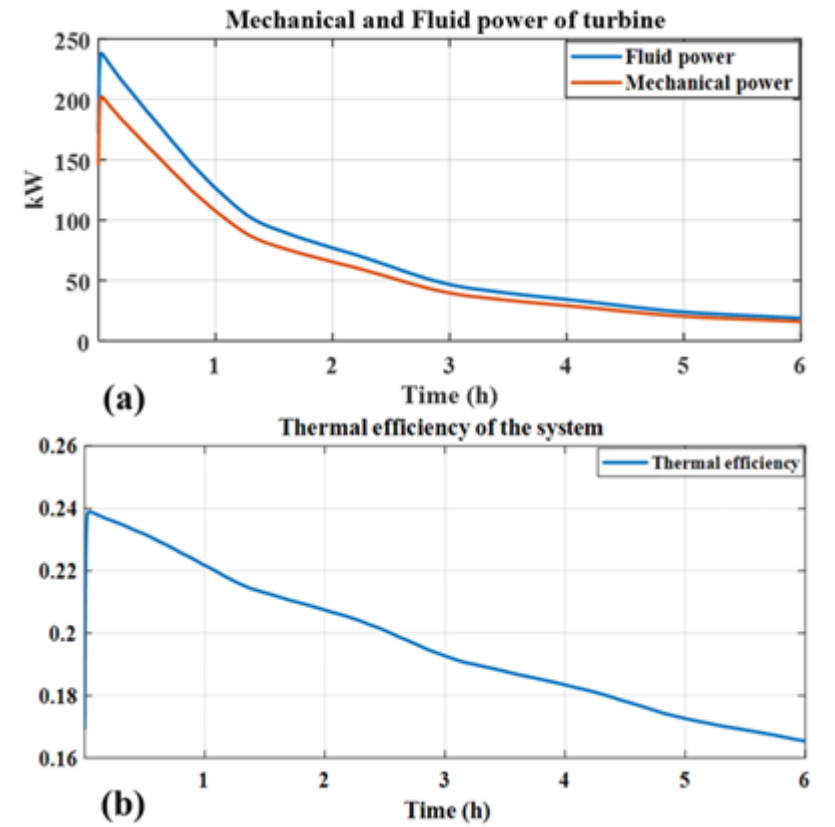


Figure 4.13: Variation of a) the fluid and mechanical power, and b) thermal efficiency of the system during the discharge mode

4.5.2 Effect of different steam accumulator storage volumes on the power output

Different sizes of steam accumulator were analysed during this discharge process, and how geometry affected the power output of the system was assessed. Four different sizes of steam accumulator with a total volume of 136m^3 , 200m^3 , 250m^3 and 300m^3 were used in this test. The mass flow rate of the system was controlled based on the liquid level in the condenser, which was 0.7 as a constant value in the three different sizes of steam accumulator. Figure 4.14 shows that the accumulator with a total

volume of 300m^3 was able to produce 203.3 kW of power output initially, which then slightly decreased to 82.86 kW during the $21,600\text{s}$ of the test time, as shown in Figure 4.14. The accumulator with a total volume of 136m^3 could generated less than the first size initially (approximately 200.9 kW), and this power output had decreased to 35.11 kW by the end, as shown in Figure 4.14. The second size of accumulator (200m^3) could generate 202.3 kW initially and 56.47 kW by the end of the simulation test, as shown in Figure 4.14. From this simulation. it can be summarized that the size of the steam accumulator can directly affect the mechanical power in the system.

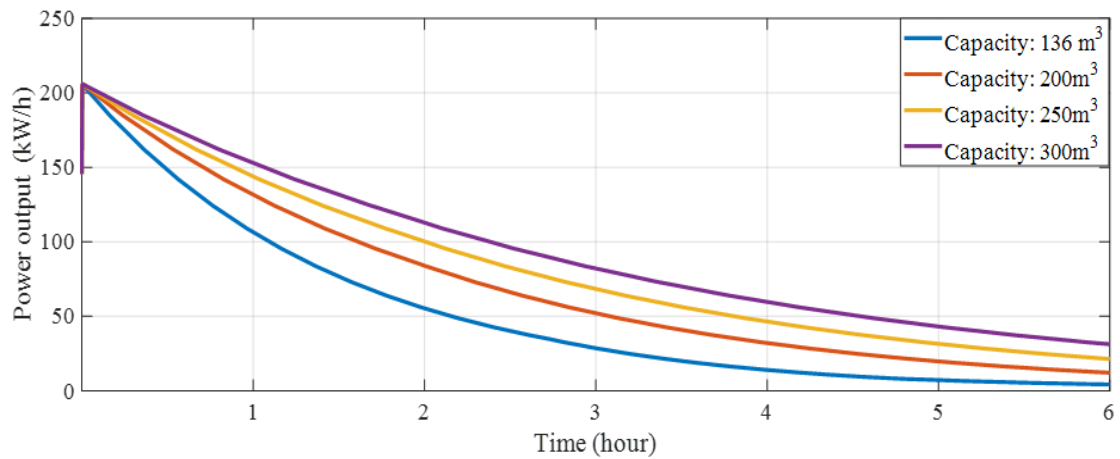


Figure 4.14: Effect of steam accumulator volume on power output during the discharge mode

4.5.3 Performance evaluation of a direct steam generation solar power system integrated with a steam accumulator

Based on the variable solar irradiance and ambient temperature in southern Libya, the system integrated with a steam accumulator was simulated throughout the day, so the changes in solar irradiance can effect directly to the system output. According to the Centre for Solar Energy Research and Studies, Libya, the boundary conditions, such as the ambient temperature and solar radiance profiles, were obtained [6]. The solar irradiance and ambient temperature were variable for one day, from 300 W/m^2 to 1000

W/m^2 , and the temperature ranged from 285 to 330 (K) in July, as shown in Figure 4.15. In this analysis, we examine the proposed system under the condition of two typical days, one when the sky is clear and the other when the sky is slightly cloudy. According to Libyan meteorological data, solar radiation is always available, but there are a few slightly cloudy days. The mathematical model developed in section 4.4 is applied to simulate the steam accumulator integrated with the direct steam generation solar power system, as shown in Figure 4.16, by using Simulink/Simscape software.

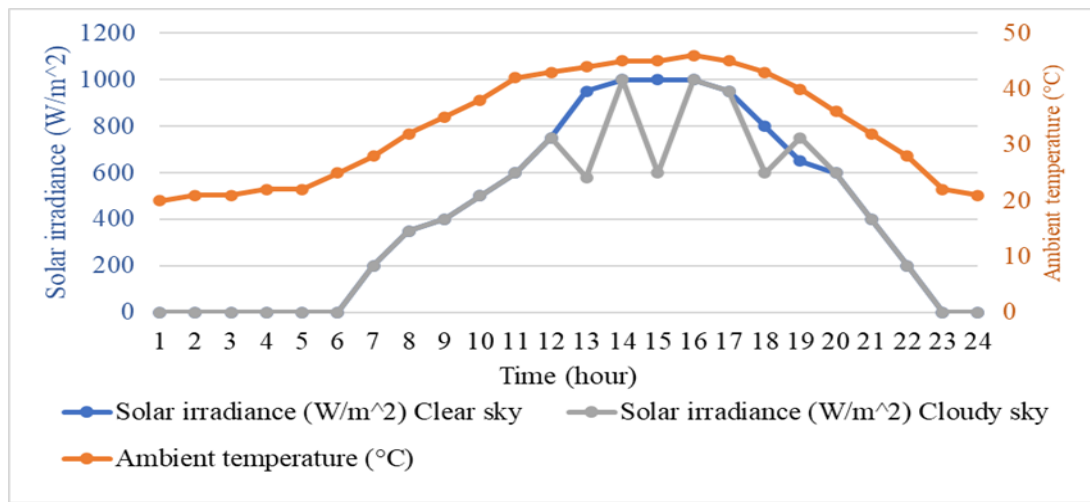


Figure 4.15: Variation in solar irradiance and ambient temperature with a cloudy and a clear sky [6]

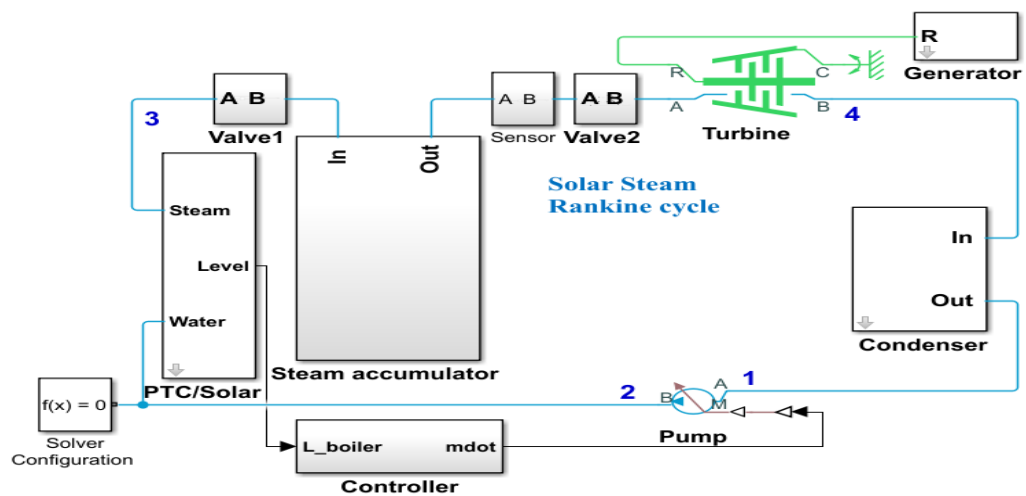


Figure 4.16: Schematic of a direct steam generation solar power system integrated with a steam accumulator using Simscape software

To clarify the influence of each variable mass flow rate, solar irradiance and the ambient temperature, the effects produced by considering a different method of operation were evaluated in turn. The first mode works when solar irradiance is available. The second mode works when solar irradiance is unavailable, which means that the system works based on the heat stored by the steam accumulator. The system is examined with a constant solar collecting area of 1320m^2 and without one, and also with four different storage tank volumes.

4.5.3.1 Dynamic simulation of the system under a clear sky

Figure 4.15 shows that the solar irradiation which was measured in July is abundant and the sky is very clear. Also, it is long, sunny day, starting from 7:00AM and ending at 09:00PM. As controlling, the system begins to operate when there is sufficient solar radiation $\geq 350\text{ W/m}^2$. If the solar radiation < 350 , the valve 1 in the system is closed and makes the system operate based on the steam storage tank alone. In the discharge mode, the operating time of the system depends on the amount of steam stored in the steam accumulator. Therefore, the different volumes of storage of the tank directly affect the power output time in different ways. To simulate the system, the initial accumulator inlet pressure and temperature selection are important factors. As a one-day simulation evaluates only one day, the temperature is not uniform in the steam accumulator due to the thermocline behaviour of the steam/water inside the accumulator. Normally, the initial temperatures of the water layers in the steam accumulator are the final temperatures of the previous day. It appears that all of the useful solar energy charged to the steam accumulator is used to drive the Rankine cycle and the rest is transferred to the ambient as heat loss from the system. The useful heat gain trend is directly affected by the solar irradiance pattern during the day. To observe the effect of the heat storage volume on system performance, the thermal

efficiency of the system was considered as well. The first examined index is electricity generation, which is the most important parameter, followed by solar thermal power efficiency. Figure 4.17 shows a comparison of the net power output of the system with and without an energy storage tank. The figure shows that the system has a higher power output without using a steam accumulator but, in this case, it has a shorter power output compared with having the steam accumulator unit. Also, Figure 4.17 shows that, the larger the capacity of the tank, the longer it takes to produce electricity from the system at the end of the day. For example, when the heat capacity storage was 160 m^3 , the system starts to generate electricity from 7:30AM to 10:15PM while, when the heat capacity storage was 136 m^3 , the system starts to generate electricity from 7:30AM to 9:30PM. The delay in the production of electricity from the system in the first hour of the day when using the steam tank is caused by the need to charge the tank with steam completely before starting the system.

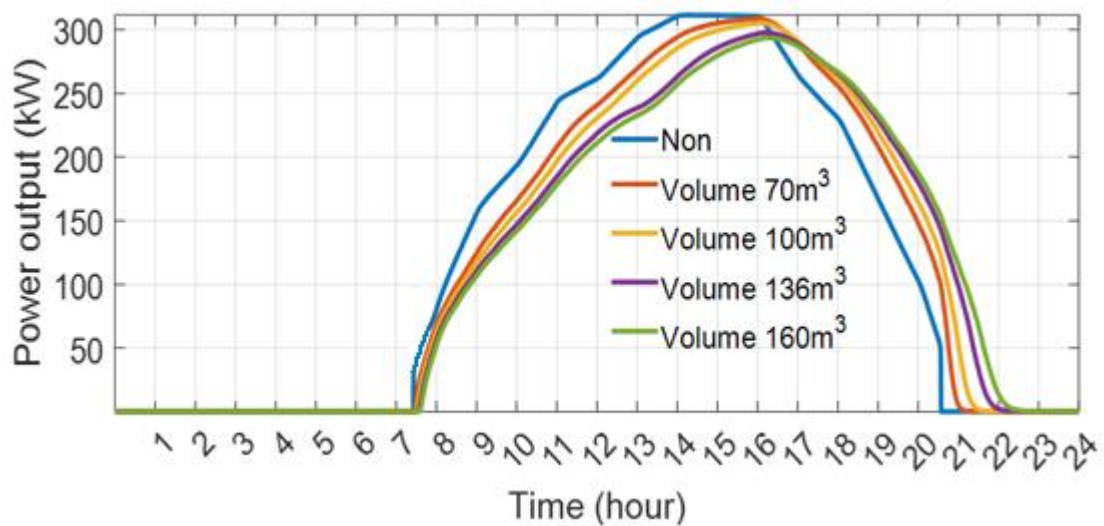


Figure 4.17: Effect of different steam accumulator storage volumes on the power output

The peak power is always the main objective emphasized in the analysis, as well as the control strategy implemented by steam tank. However, total energy production can be a different need or goal. Regarding Fig. 4.17, the integration of the power output

of the model with and without storage shows an overall power output quite similar to that with storage, this is due to the lack of steam flow rate control in the system, which means that the throttle valve plays an important role to control the power output in the solar direct steam generation system.

It can be seen that, when the system works on the charging model and solar radiation is available, all cases of differently sized storage capacities display very similar solar thermal power efficiency (0.26), as shown in Figure 4.18. In discharge mode from 8:30PM to midnight, however, the system thermal efficiency differs for each case of the volume storage tank with discharge time. The system integrated with 160m³ of steam storage capacity has the longest thermal efficiency time. The results demonstrate that the thermal efficiency of the system in discharge mode is entirely dependent on the storage capacity size of the tank.

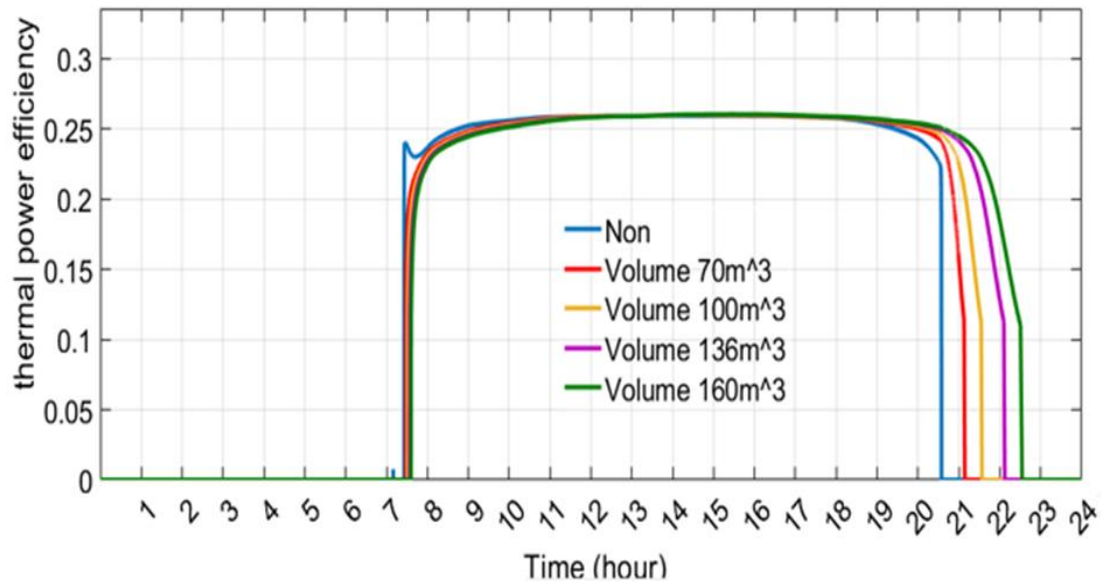


Figure 4.18: Effect of different storage tank volumes on solar thermal power efficiency

The mass flow rate also differs for each storage capacity. The findings shown in Figure 4.19 indicate that the mass flow rate during the discharge mode is completely dependent on the size of the volume of the steam tank.

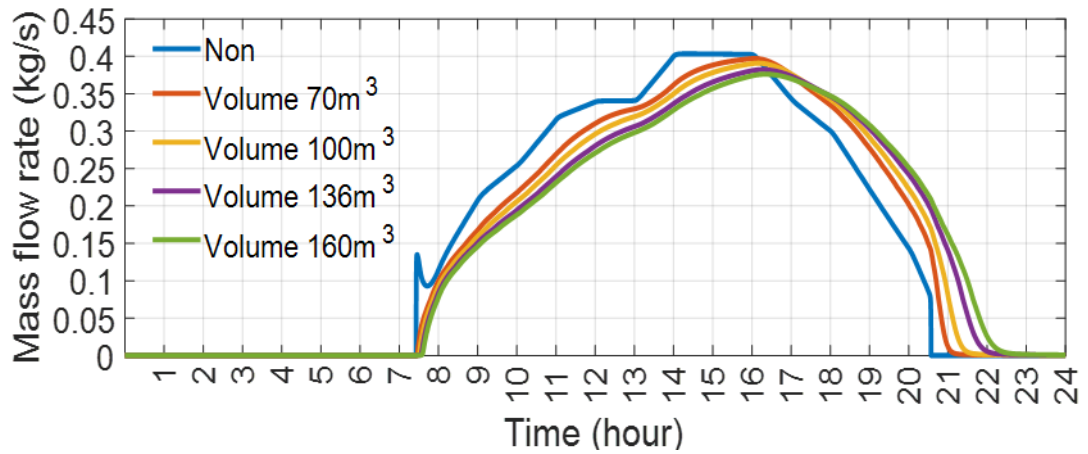


Figure 4. 19: Variations in the mass flow rate for each tank volume

Figures 4.20a and b show the charging and discharging pressure in the system and turbine inlet temperature. The pressure increases vary with increased solar radiation. The dotted lines show the design point for the turbine inlet pressure, while the solid lines indicate the response of the pressure control valve to open the design point. The designed steam turbine inlet pressure point is set at 1MPa; otherwise, the valve will close, as the steam turbine will be insufficient, according to MAN power manufacturing [130]. During the discharge mode, it appears that the steam accumulator still flashes the stored pressure, and the pressure decreases in the accumulator during its discharge time. This result is in line with data gathered from the open literature review.

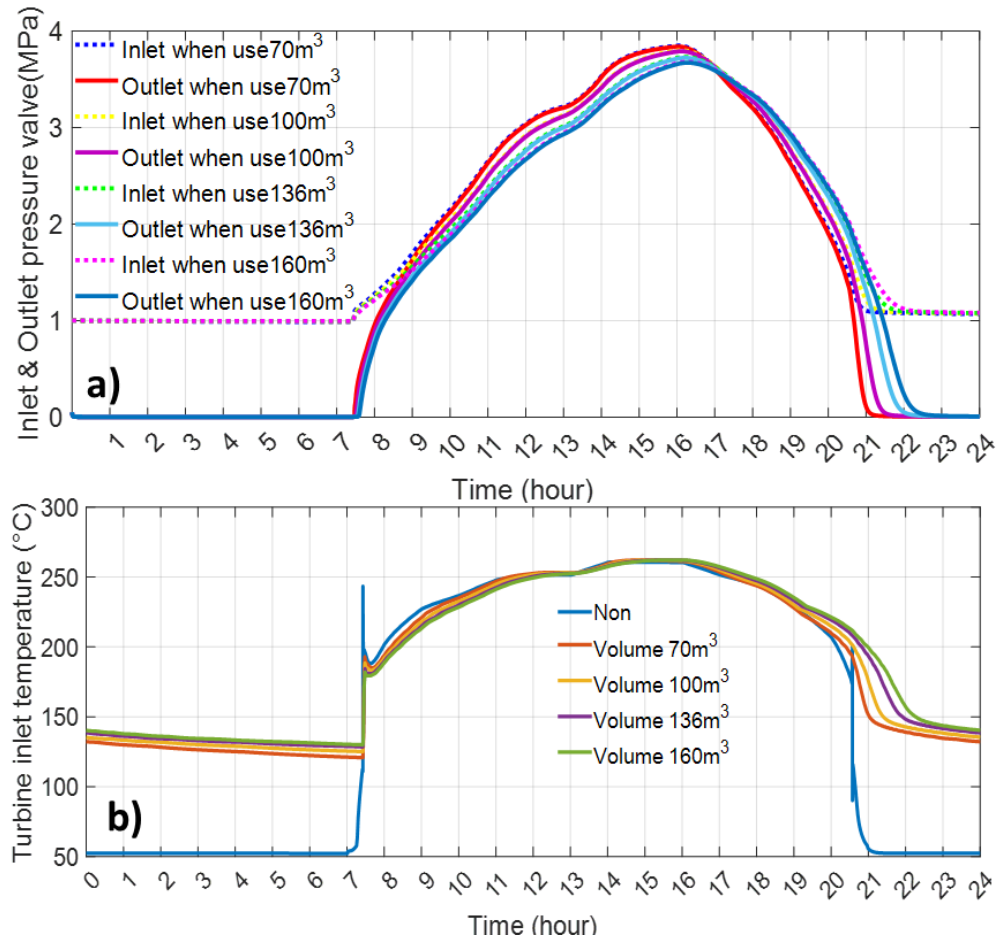


Figure 4. 20: Variations in a) the turbine inlet pressure, and b) the turbine inlet temperature for each tank volume

Figure 4.20b indicates the turbine inlet temperature. Similar to pressure, the accumulator outlet temperature is equal to the turbine inlet temperature, and decreases during the discharge period due to the steam accumulators' fast reaction times and high discharge rates.

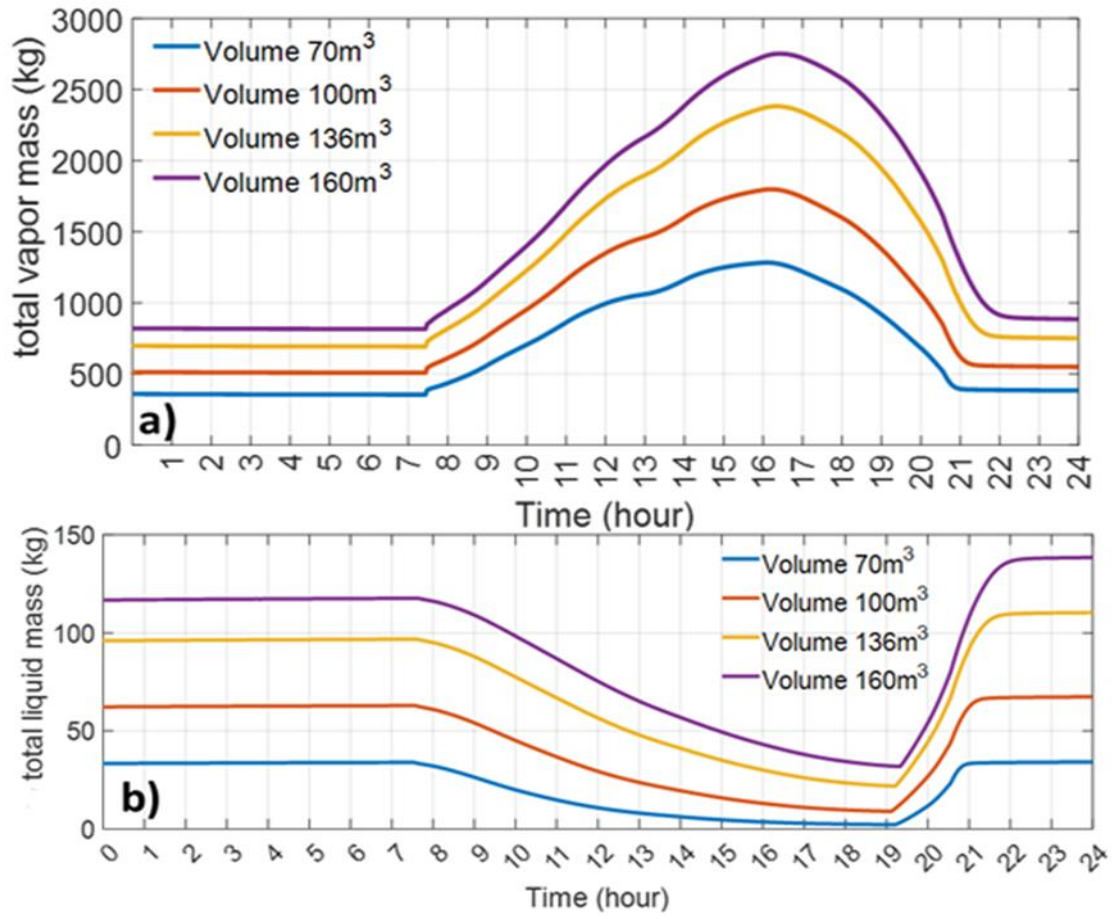


Figure 4.21: Variations of the a) vapor mass level, and b) liquid mass level inside the steam accumulator for each size

Figures 4.21a and b show the total vapor mass and the total liquid mass of the steam accumulator with four different volume capacities, respectively. Figure 4.21a shows that the vapor mass increases as the solar radiation rises and the steam accumulator starts to charge at 7:00AM, when solar radiation becomes available. The vapor mass decreases when the solar irradiance declines over time. The decline in the vapor mass in the steam accumulator caused an increase in the liquid mass, especially during the discharge mode, as shown in Figure 4.21b, because the steam stored in the tank began to condense and move to a two-phase flow stage or liquid stage. The results reveal that the difference in the volume of the tank capacity affects the amount of stored steam.

4.5.3.2 Dynamic simulation of the system under a cloudy sky

Although Libya is a country that is very rich in solar radiation, there are a few cloudy days per year. From this perspective, the proposed system must be simulated on a cloudy day. Figure 4.15 shows the cloudy solar radiation in March. The main reason for undertaking this analysis is that we need to see how the steam accumulator responds to the direct steam generation solar power system when cloud passes over the solar collectors' area. The most important parameter is electricity generation. Therefore, evaluation on a typical cloudy day is required. A comparison study with and without a built-in steam accumulator was undertaken to assess the effect of this on direct steam generation solar power output systems. The results show that the use of the steam tank could affect the power output of the proposed system on a somewhat cloudy day. The availability of a steam accumulator to compensate for the quick transitions also helps to reduce the reaction time and discharge rate requirements for the storage systems intended to supply stored energy for longer periods, as shown in Figure 4.22.

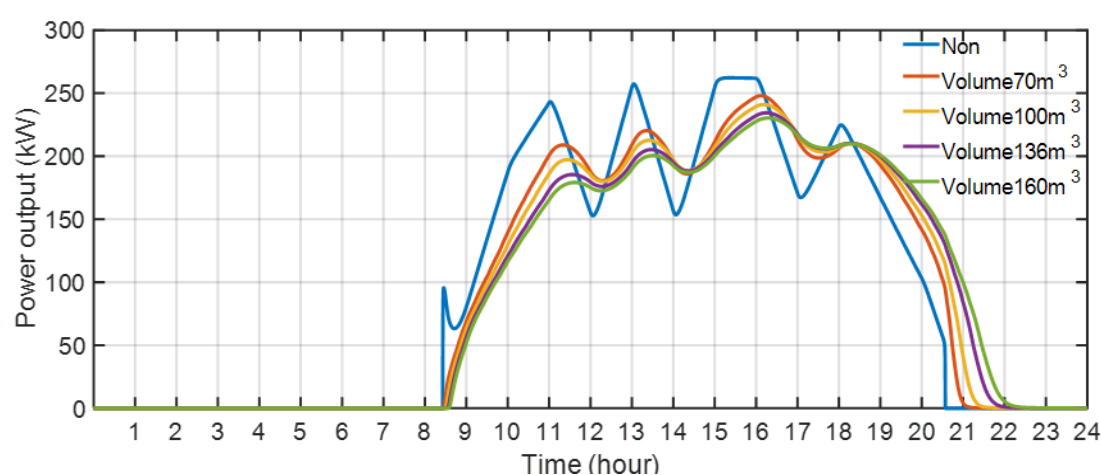


Figure 4.22: Effect of different storage tank volumes on the power output under a cloudy sky

The result of this evaluation demonstrates that, the larger the tank capacity, the lower the fluctuation in the net power output. For example, the system with a heat storage

capacity of 160m^3 had lower fluctuation in the power output compared with the other heat storage capacities.

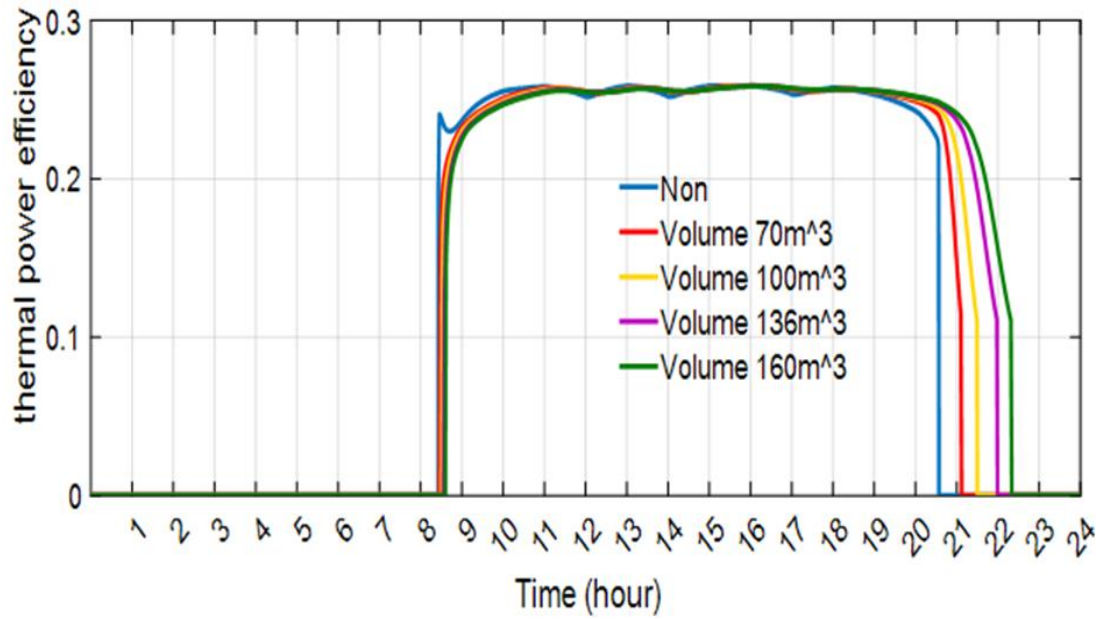


Figure 4.23: Effect of the different storage tank volumes on solar thermal power efficiency under a cloudy sky

The thermal energy efficiency of the system with different heat storage volumes is very similar even with fluctuations in solar radiation, as shown in Figure 4.23. However, there is a different extension period for the efficiency of solar thermal energy for each heat storage situation during the discharge mode. This is because of the amount of heat stored in the steam storage tank, which was obtained by the solar collectors during the charge mode. Therefore, since the power output period is affected by the heat storage volume, the thermal efficiency of the system is also affected by the heat storage volume.

As mentioned in Chapter 3 above, the control strategy for the turbine inlet pressure is applied in this chapter when the proposed system is integrated with the steam accumulator. When solar radiation fluctuation occurs, the turbine inlet pressure decreases, making the steam turbine insufficient. From Figure 4.24a, it can be noted

that the turbine inlet pressure has been controlled for all cases. The dotted lines indicate the control valve inlet pressure at the set point of 0.99 MPa., while the solid lines represent the outlet pressure of the valve, which equals the turbine inlet pressure. Conversely, when the system operates using a thermal storage tank, the system first fully charges the steam accumulator and then runs the Rankine cycle. Therefore, it appears that the valve inlet pressure point is presented under the set point, which is 0.99MPa during the charge mode, and then the valve opens when it reaches > 0.99MPa.

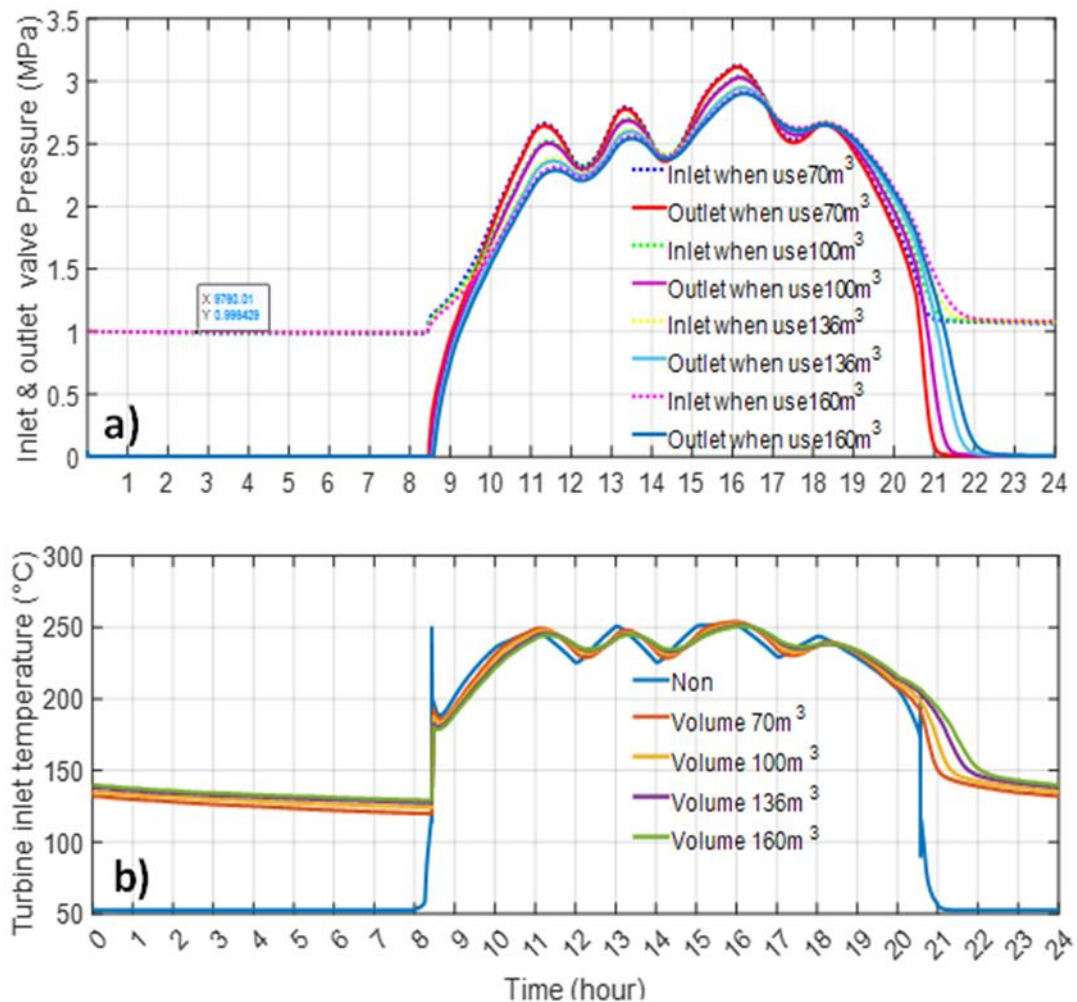


Figure 4.24: Variations in a) the turbine inlet pressure, and b) the turbine inlet temperature for each tank volume under a cloudy sky

Figure 4.24b shows the turbine inlet temperature, which is equal to the steam accumulator outlet temperature. The steam temperature inside the steam accumulator is affected by solar radiation. However, this effect is minimal, as shown in Figure 4.24b, a large steam storage volume leads to small fluctuations in the turbine inlet temperature. The reverse happens when the system does not use a steam heat storage tank. In other words, when there are some clouds in the sky and the solar collector's system is unable to operate the steam turbine, the steam tank can cover this brief fluctuation using steam stored within it until the solar collector is working again.

Figure 4.25a shows the amount of steam that was charged and stored during the day for each case. It is noted that, the greater the capacity of the thermal tank, the greater the stored steam. For instance, when the system is adopted a 160m^3 storage tank volume, the maximum total vapor mass reaches 2,100 kg. When the system is adopted a 70m^3 storage tank volume, the maximum total vapor mass inside the tank is 1,000 kg.

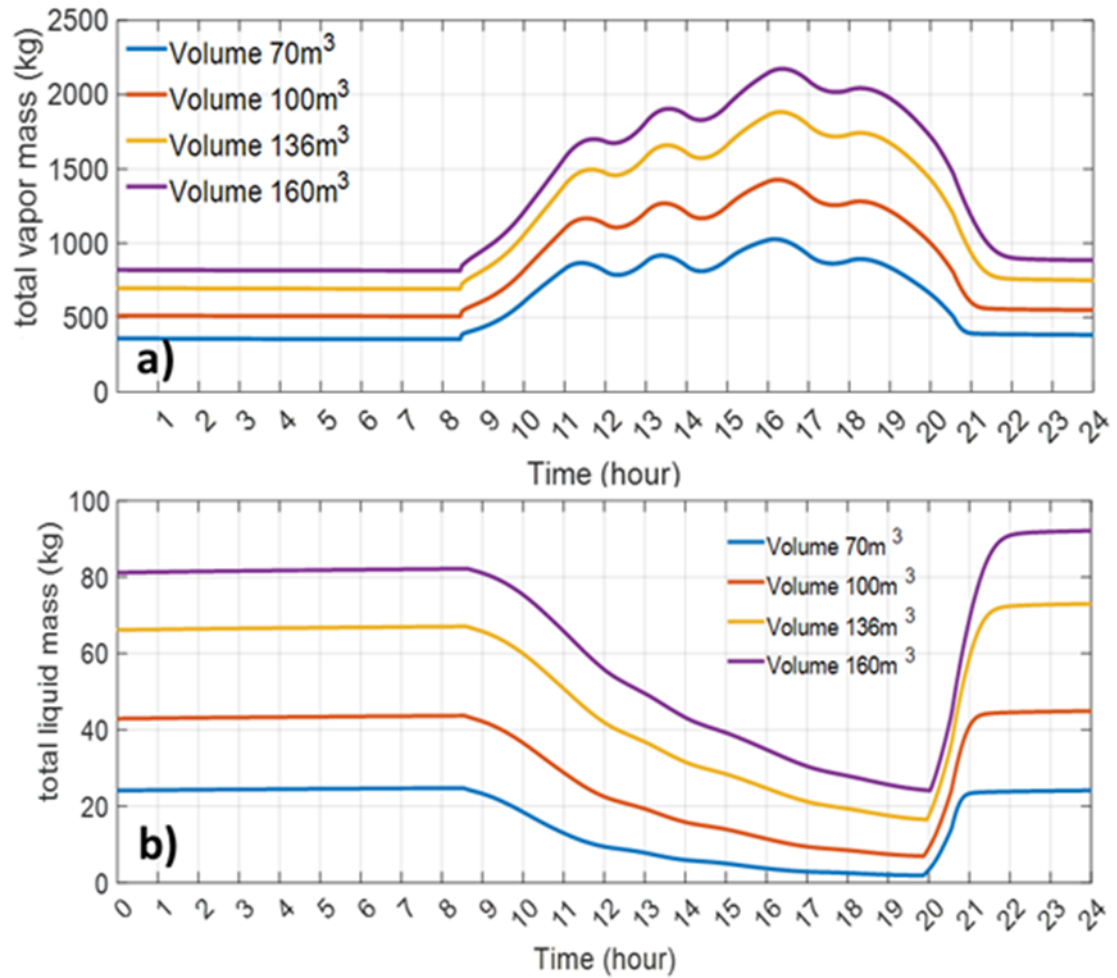


Figure 4.25: Variations in a) the vapor mass level, and b) the liquid mass level inside the steam accumulator for each size under a cloudy sky

Figure 4.25b shows the total saturated water inside the steam accumulator. This saturated water is accumulated from the saturated steam that comes from the solar collectors, which usually consists of liquid and vapor phases in the early morning, when the solar radiation is weak. Usually, the steam accumulators represent a fluid that can undergo phase changes during the operation time [128]. It can be noted that, from 8:30AM, the liquid mass level starts gradually to decrease inside the steam accumulator unit until 20:30PM. This is due to the vaporization in the accumulated saturated water when the superheated steam flows into the tank, thereby resulting in a decrease in the liquid level and a rise in the vapor level, compared with Figure 4.25a.

In the same Figure, it can be noted that, from 20:30PM, the liquid mass level starts to increase gradually inside the tank until 8:30AM, due to the condensation process that occurs in superheated steam over time.

4.5.4 Weekly evaluation of the system with and without power output control

The weekly evaluation of the system was conducted under Libyan transitory conditions. The evaluation week was chosen as the second week of March, when the average solar radiation is close to the annual mean. Figure 4.26 shows the differences in solar radiation during the assessment week. It can be seen that, between 7:00AM and 8:00PM, the amount of radiation is above 350 W/m^2 , with a maximum of 1000 W/m^2 at 1:00PM. In this chosen week, some days have fluctuating solar radiation levels. Figure 4.26 shows that the power output of the DSG integrated with the steam accumulator can solve problems regarding huge solar radiation fluctuations during the day. For example, the green line represents the net power output of the system without a control system. Nevertheless, the steam accumulator reduces the effect of solar radiation fluctuation during the day. This is due to the amount of steam heat stored inside it, and the steam accumulator has a fast reaction during the discharge mode. The orange line represents the stable net power output with a control system, as shown in Figure 4.26. The findings reveal that, by combining the steam accumulator and the throttle control valve which is placed before the steam turbine in the DSG system, the proposed system can produce a stable power output during the day, even when the solar radiation fluctuates. However, it can be noted that the net power is reduced compared with green line, because the system attempts to store a large amount of heat energy in the tank when solar radiation fluctuates and with throttle valve movement,

in consideration of a stable demand. Also, the parameters, sizes of the solar collectors' area and storage tank volumes are kept constant in this evaluation.

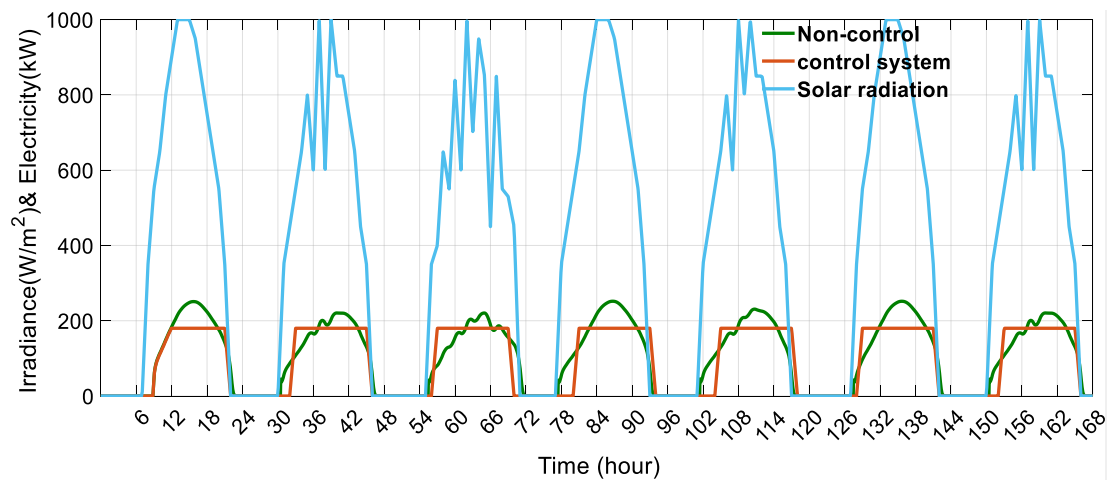


Figure 4.26: Weekly variation in solar radiation and power output

This study also shows the stability of the thermal efficiency of the DSG control system compared to the non-controlled system, as shown in Figure 4.27. There slightly reduced the controlled system's thermal efficiency, because the net power output was reduced.

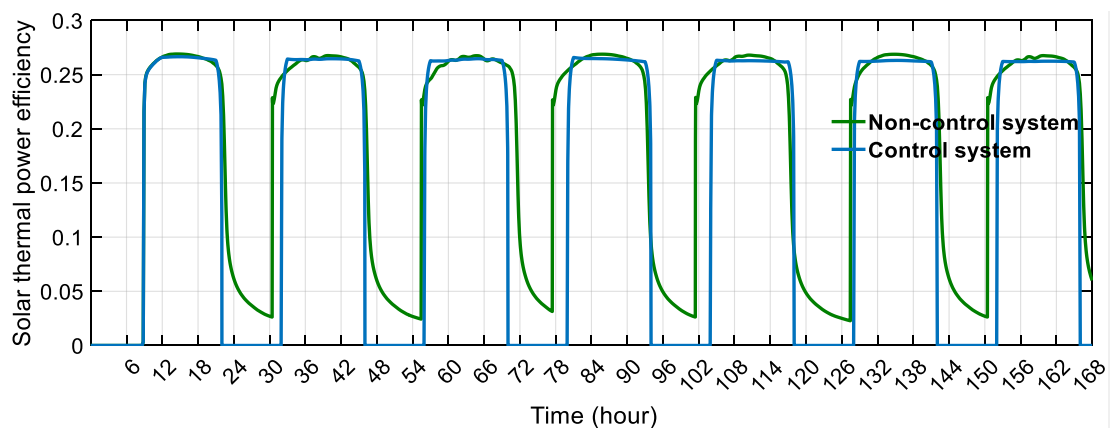


Figure 4.27: Weekly variation in solar thermal power efficiency

Figure 4.28a, b and c show the turbine inlet flow rate, energy flow to the turbine and turbine inlet temperature. It can be seen that the steam flow rate into the turbine is

stable and the heat energy flow enters the turbine, resulting in the stability of the power output in the controlled DSG system.

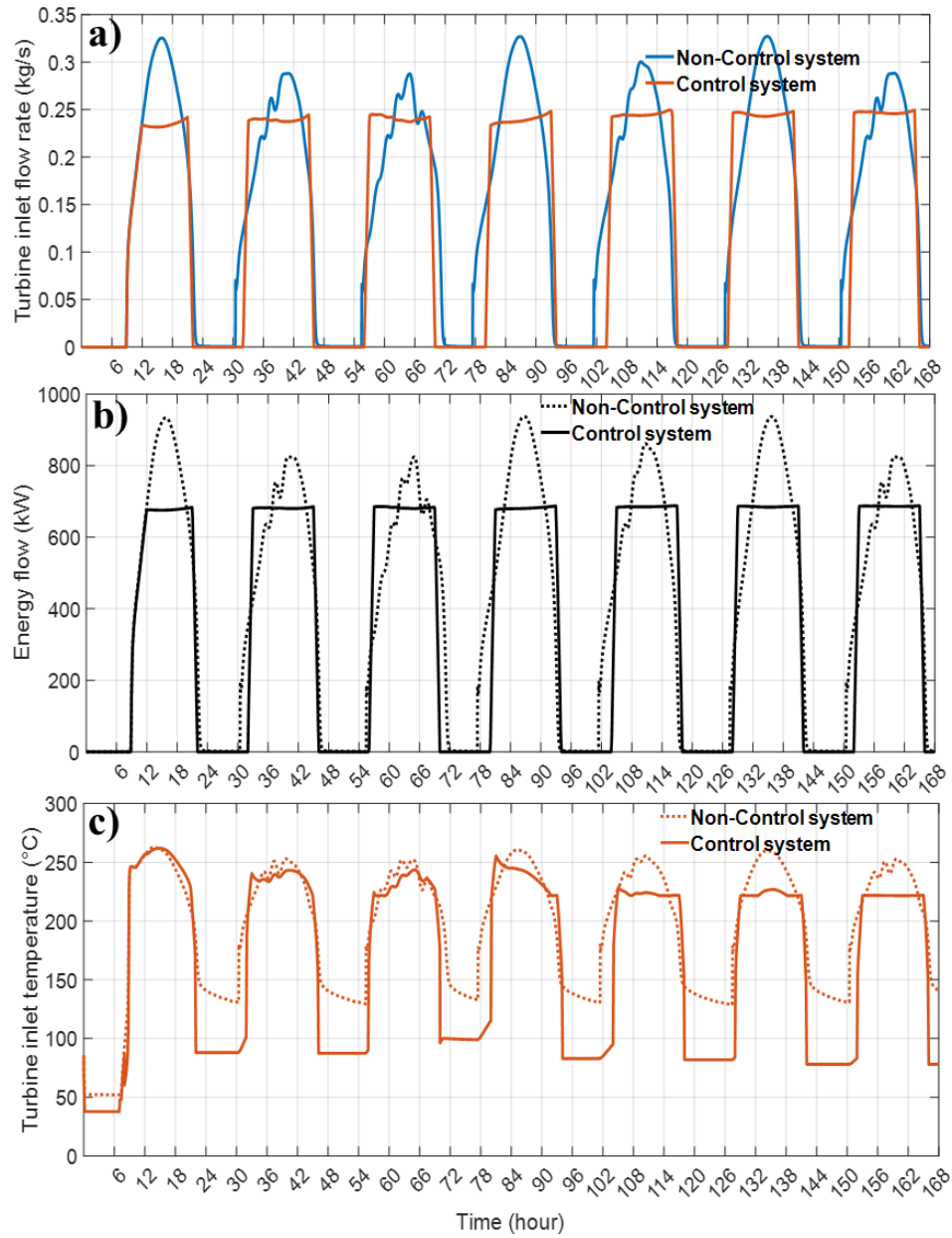


Figure 4.28: Weekly variation in a) the turbine inlet flow rate, b) the energy flow to the turbine and c) the turbine inlet temperature

Figure 4.29a and b show the total variations in the vapor and liquid mass inside the steam accumulator in the charge and discharge modes during a week. According to Figure 4.29a, the controlled DSG system can store much more steam heat energy than non-control system in order to reach the target power output. Figure 4.29b shows the

total liquid mass for the controlled and non-control systems. It can be seen that the level of the water liquid in the tank increases when solar radiation is unavailable and also in the discharge position and decreases when the tank is in the charge position and solar radiation is abundant.

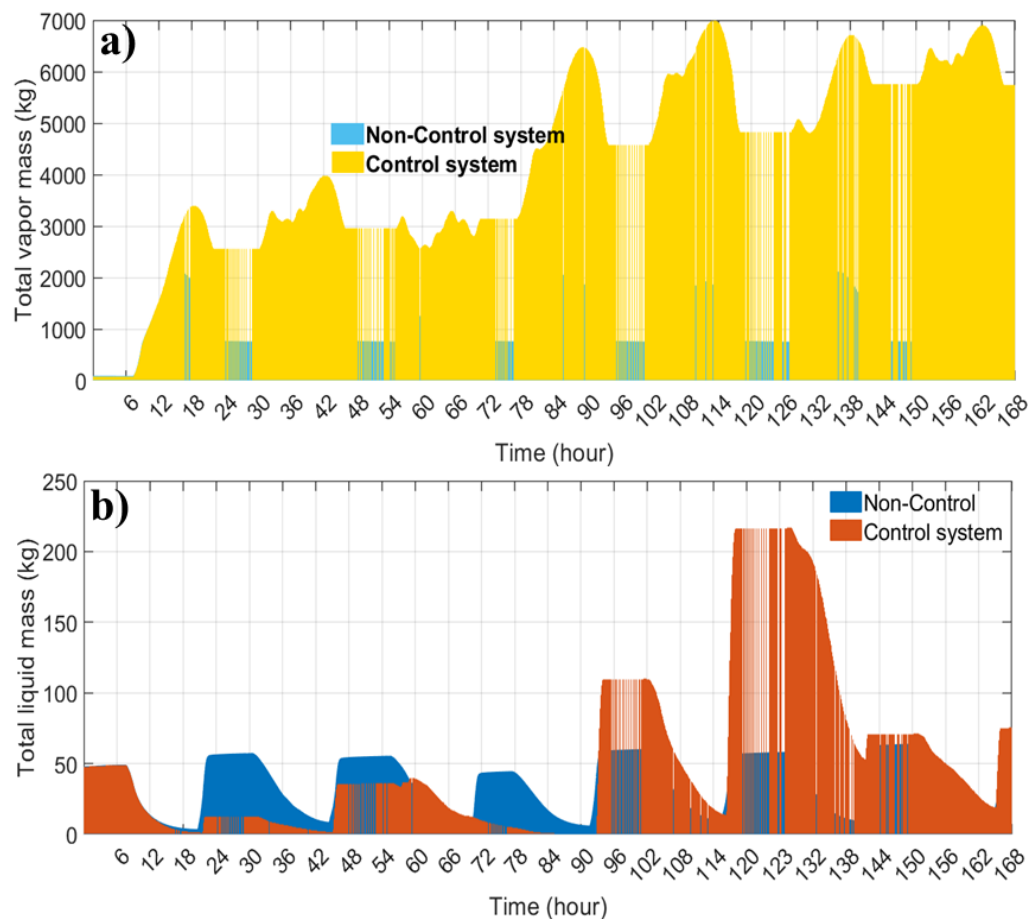


Figure 4.29: Weekly variation in a) the vapor mass level, and b) the liquid mass level inside the steam accumulator during the charge and discharge modes

Figure 4.30 shows the dynamic response of the throttle valve area in the controlled DSG system to reach stable power output during a week. It can be seen that the curve of the lock and unlock areas in the figure below leapt up at the beginning of the first day, to make the system store the required heat in the tank until the system was ready to meet the demand and achieve the required stable production.

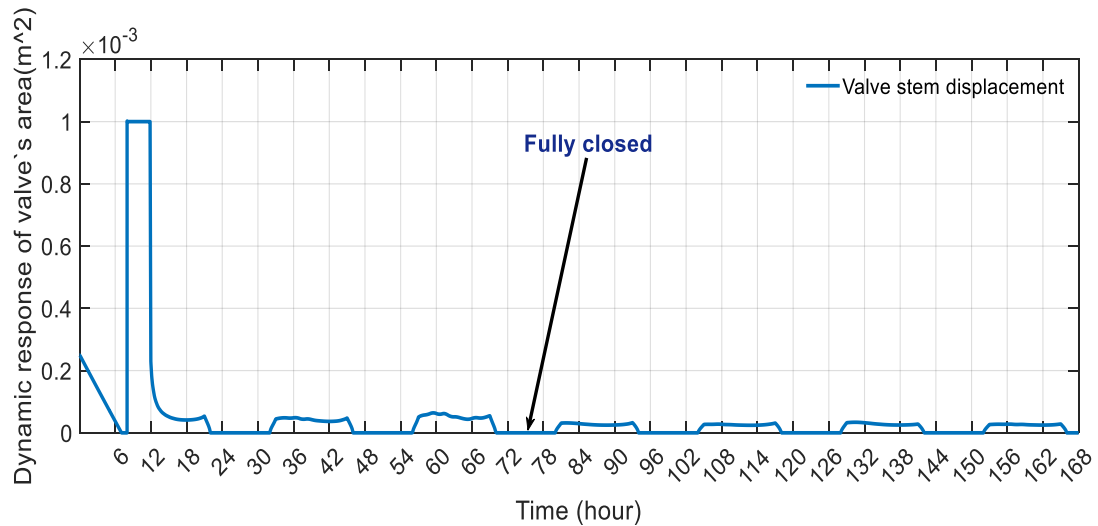


Figure 4.30: Weekly dynamic response of the throttle valve area in the controlled DSG system during the charge and discharge modes

4.6 Dynamic simulation of the system for fulfilling the energy demand of a Libyan hospital as a case study

Electricity is the most sought-after form of energy due to the simplicity of converting it into other forms and its necessity for the operation of household equipment or official devices in public government buildings. The solar system meets the electrical demand but, because the demand profile varies throughout the day, the electricity supply is not constant. To model and specify the hourly electricity demands, research has been carried out [10,131]. The demand varies from building to building but, overall, is consistent throughout all homes and public buildings in Libya, such as schools, community centres, libraries, courts, information centres, government buildings, hospitals, and public housing [131]. According to GECOL company's most recent official annual report, the nation's electricity demand climbed by 12% every year between 2003 and 2016 and the company was unable to supply this level of demand. Hospitals in Libya needs a huge investment to overhaul their electricity infrastructure and power grids, while Libya's schools and hospitals are suffering due to blackouts. These hospitals have been supplied by diesel generators, but electricity

outages still occur in them due to the difficulty of providing fuel for these generators. A PV system has been installed in some hospitals to supply electricity to the operating theatres and emergency departments only. However, some hospitals still face electricity outages, especially in the southwest of the country, so it is vital to find a solution to this problem. Figure 4.31 shows the profile of the daily energy load for some hospitals in Libya. As can be observed, the energy demand has an average peak of 200 kW and occurs between 8:00 AM and 5:00 PM. All of these data profile lines have been taken on a typical summer day for each hospital. Murzug General Hospital has the higher level of demand for energy than the other hospitals. During the night, its demand for energy is lower than 60 kW because no more patients come to the hospital. At the peak time, the demand for electricity rises because medical equipment and air conditioning are needed to serve patients in the hospital.

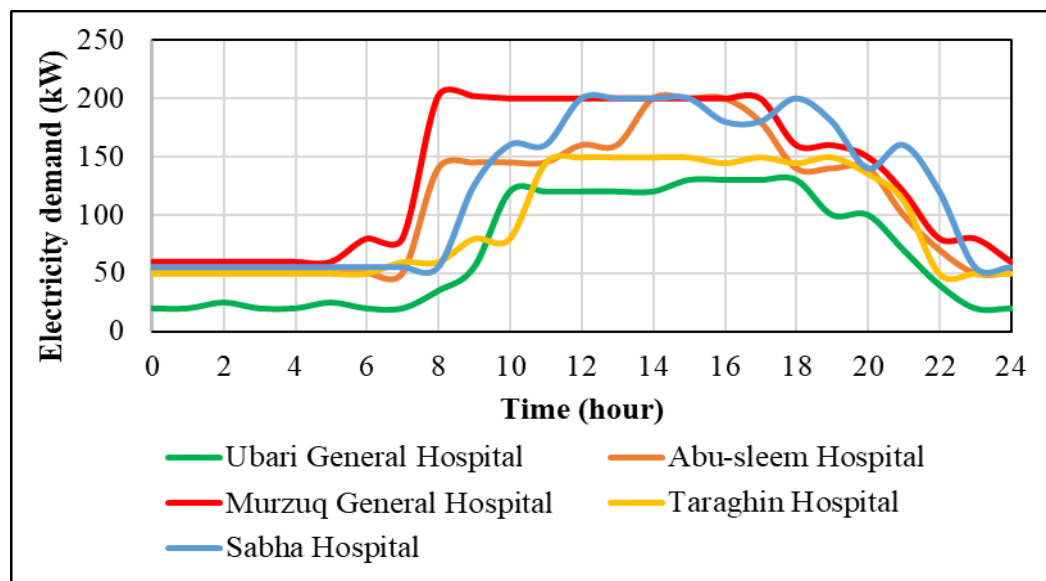


Figure 4.31: Hourly electricity demand for five hospitals in Libya [131]

4.6.1 Description of the system

Figure 4.32 illustrates the proposed DSG system to supply DHW and electricity to a hospital in Libya. This system consists of PTCs, a steam accumulator, a Rankine

There can be two operating modes for the system:

(I) The system needs to generate electricity and solar radiation is abundant. In this mode, V1 and V2 are open, and the pump runs. Both SRC and DHW operate. The water is heated and vaporized in the parabolic trough collectors.

The saturated steam flows into the steam accumulator and turbine, exporting power during the process of enthalpy drop. The outlet vapor is condensed to be saturated liquid in the DHW system and in the condenser. The working fluid is pressurized by the pump, then sent back to the PTC. The condensation heat in the DHW system is used to heat the water for the end-users.

(II) The system needs to generate electricity and DHW, but irradiation is unavailable. V2 is open and V1 is close. Both DSG and the heat exchanger for the DHW production process operate. Heat is released by the steam accumulator and converted into power.

4.6.2 Methodology

4.6.2.1 Energy demands of the hospital

The proposed system must cover the average electricity and hot water demands of Murzuq General Hospital, located in southwest Libya. The hospital covers a total area of 8,000m² and has 120 beds [132]. The daily electricity demand for air conditioning, lighting and medical equipment can reach up to 200kW and the required hot water temperature is from 40-50°C, as hot water is constantly required for the pharmacy, laboratory, etc. Five typical days in March are considered in this simulation. The demand electricity profile of the hospital is displayed in Figure 4.34 for five days. The occupancy of the hospital is 24h per day. The higher demand for electricity starts at 7:30AM and then decreases at 7:00PM. As can be seen, the day hours' demand is higher than that of the night hours, as the only department that is open at night is the

emergency department and the rest are closed. The demand for electricity is stable during the night, at 50kW, as displayed in Figure 4.34.

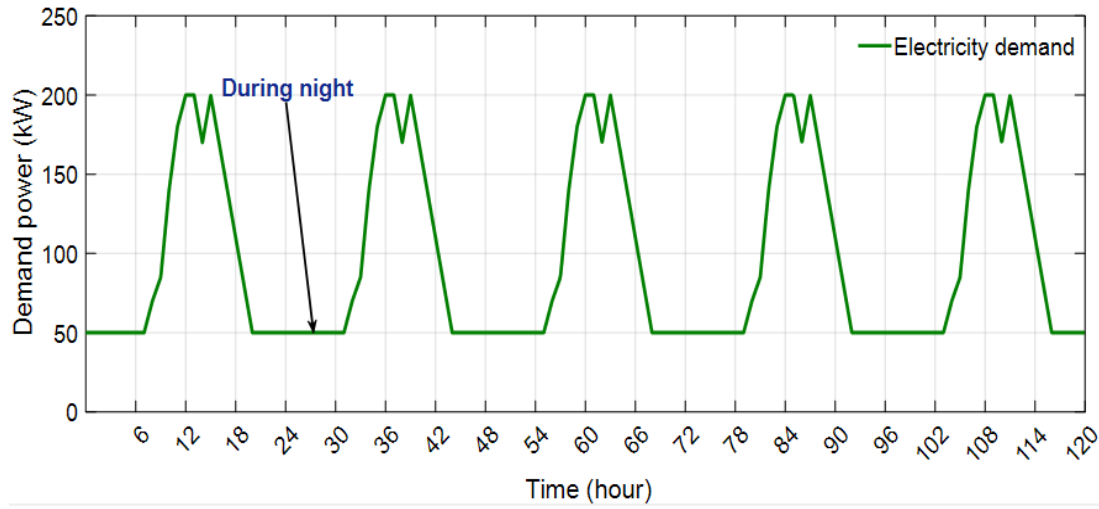


Figure 4.34: Variation in the demand energy profile for a Libyan hospital over five days [131]

4.6.2.2 Heat exchanger modelling

The design of heat exchangers should be considered because of their impact on system performance. Due to their advantages, such as affordability and high tolerance regarding operating temperature and pressure, shell and tube heat exchangers with counter flows were chosen for this investigation [133]. The DHW system is a heat exchanger unit that is placed after the turbine and before the condenser, that serves as a boiler to heat the water in the hospital and the model of the DHW in Simscape is shown in the figure 4.35. The outside and internal diameters of the tube are taken to be 0.01mm and 0.03mm, respectively, and steel is assumed to be the material of the tube. The LMTD method is used to calculate the heat exchangers area. The heat transfer rate of the heat exchanger is expressed by:

$$\dot{Q}_{rate} = U \cdot A \cdot LMTD \quad (4.21)$$

Where A is the heat transfer area and U is the overall heat transfer coefficient of the heat exchanger. LMTD is the log mean temperature difference, and it can be calculated as:

$$LMTD = \frac{\Delta T_{in} - \Delta T_{out}}{\ln\left(\frac{\Delta T_{in}}{\Delta T_{out}}\right)} \quad (4.22)$$

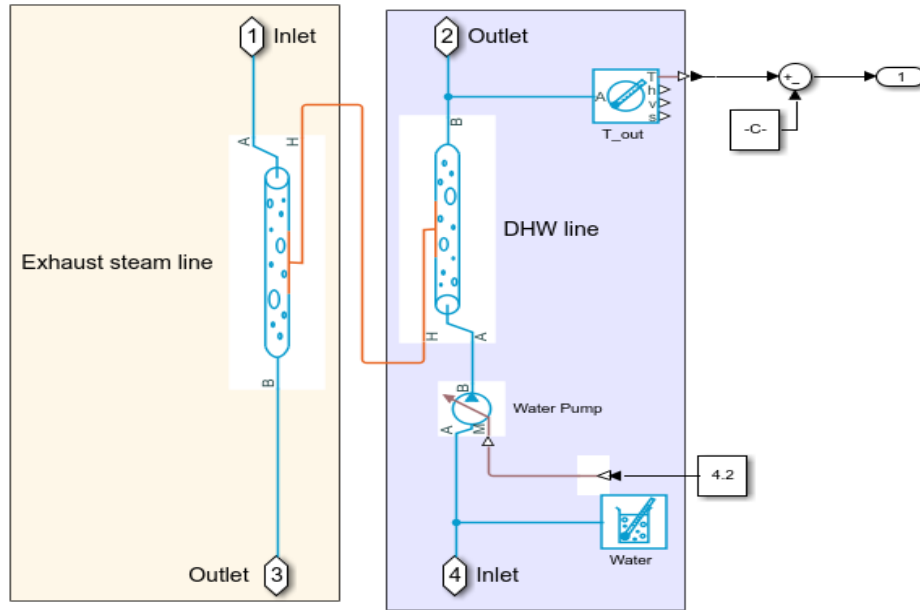


Figure 4.35: Heat exchanger model for DHW heating

4.6.2.3 Control strategy of the system

When the system runs on a cloudy day, the cloud can cause DNI reduction, which may lead to a large decline in the solar field outlet temperature so that the working fluid enters the two-phase region at the turbine inlet. This may cause severe damage to the turbine blades as a result of the liquid impact, making the system output power unstable, so it is essential to adopt a control strategy for the solar direct steam generation power system to make the system operate safely as well as to meet the electricity demand profile for three days. Figure 4.36a shows the control strategy of the system by using a throttle valve before the turbine in order to control the flow rate to meet the electricity demand profile for three different days under typical climate

conditions. Figure 4.36b shows a diagram of the control of the turbine inlet pressure. In this control, the set point of the pressure selected is 0.9MPa. It has been recommended by the MAN power manufacturing company that the lowest inlet pressure turbine should be 0.9 MPa

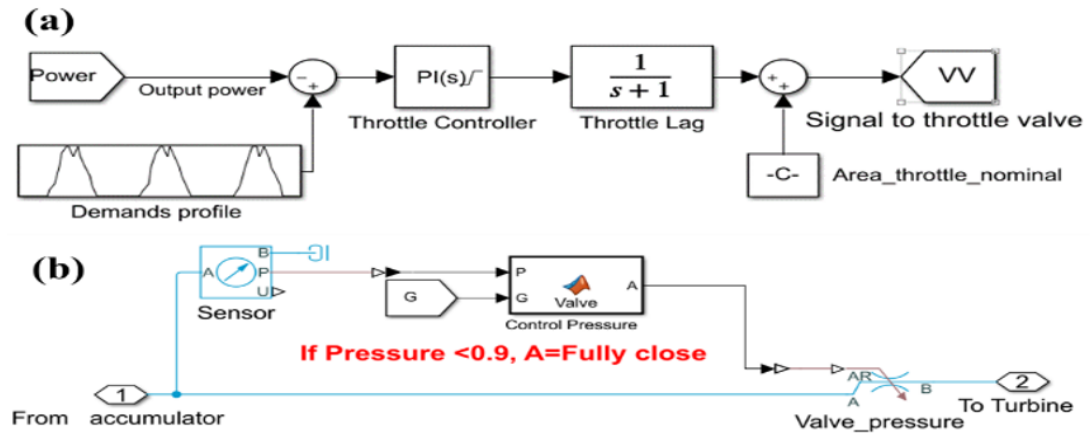


Figure 4.36: Diagram of the control strategy of the system, a) to meet the electricity profile and b) to control the turbine inlet pressure

Figure 4.37 shows the flow chart of work in this case study step by step to match the demand profile of electricity.

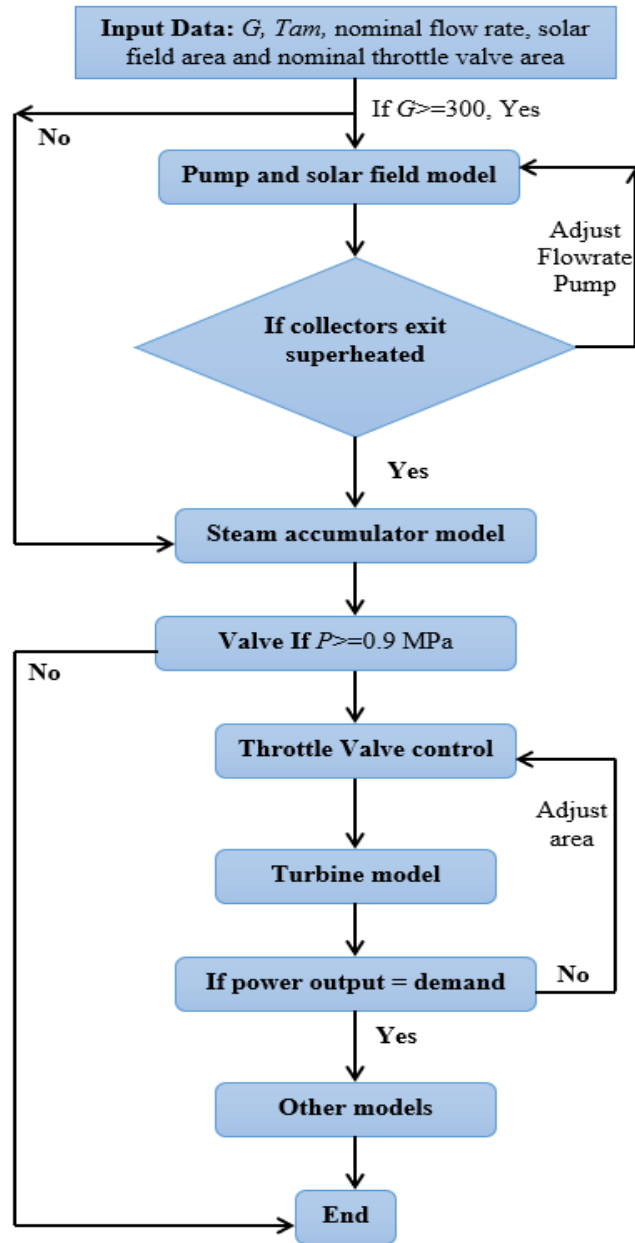


Figure 4.37: Flow chart of work in this case study

4.6.3 Simulation results and discussion of the case study

The simulation model is built using Simulink/Simscape. The influence of solar irradiance and ambient temperature under typical climate conditions is evaluated, then an hourly simulation for five days is conducted. As a design condition, the solar irradiance and ambient temperature variation profile in Libya are considered in this study. The rest of the specifications are given in Table 4.3.

Table 4.3: The rest of the proposed system specifications

Item	Value
Solar collector area m ²	1544.4
Turbine isentropic efficiency	0.85
Isentropic efficiency of the pump,	0.85
Efficiency of the Generator	0.95
Solar irradiance (W/m ²)	350-1000
DHW outlet temperature °C	40-50
Ambient temperature °C	25-40
Steam accumulator volume m ³	160

As mentioned above, five typical days were selected for this study. Figure 4.38 shows five typical days in March, and it can be noticed that there is variation in the solar irradiance and ambient temperature levels on each day.

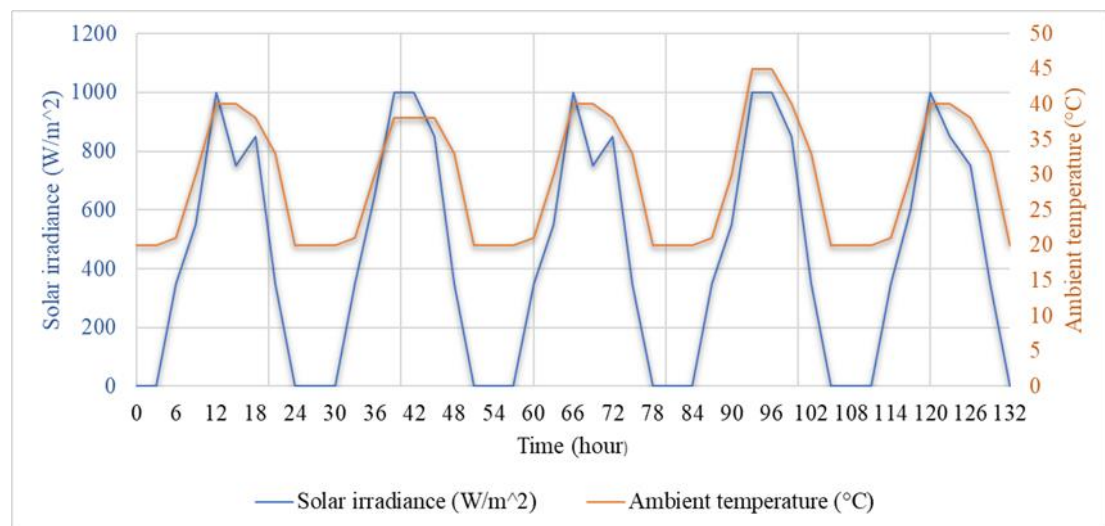


Figure 4.38: Variation in the solar radiation and ambient temperature for five days in March [6]

The solar irradiance and ambient temperature variation profile are created during five selected days (20th-24th of March) in southwest Libya.

Figure 4.39 shows the thermal efficiency of two regions of solar collectors. The boiler region is the red line, which is higher than the superheat region, that is represented by the blue line. As expected, the thermal efficiency of the collectors in the boiler region is higher than that in the superheating region because of the lower operating temperature. It can be observed that the solar field system efficiency increases as the solar irradiance increases and decreases with dropping solar irradiance during days with each typical climate condition. Figure 4.39 shows the thermal efficiency of the system. It is clear that, after the few first hours of starting up, the system has two periods of the thermal efficiency, one when solar irradiance is available and another when it is not where, during the discharge mode, in a near stable state is it 0.20 and, during the charge time, the thermal efficiency of the system is 0.23. The drop in thermal efficiency after 30 hours is caused by the valve control area when it is fully closed, which means that the pressure of the steam tank is lower than the minimum required pressure. Moreover, during the discharge mode, this value decreases and stable at 0.20. This period of stability is due to the net power output of the system being stable.

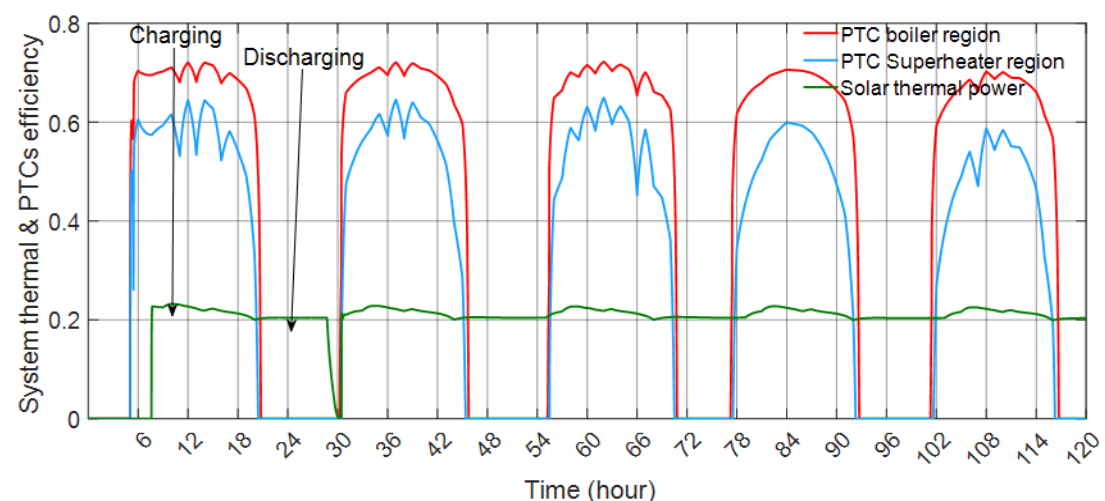


Figure 4.39: Variations in the solar collectors' efficiency and solar thermal efficiency for five days

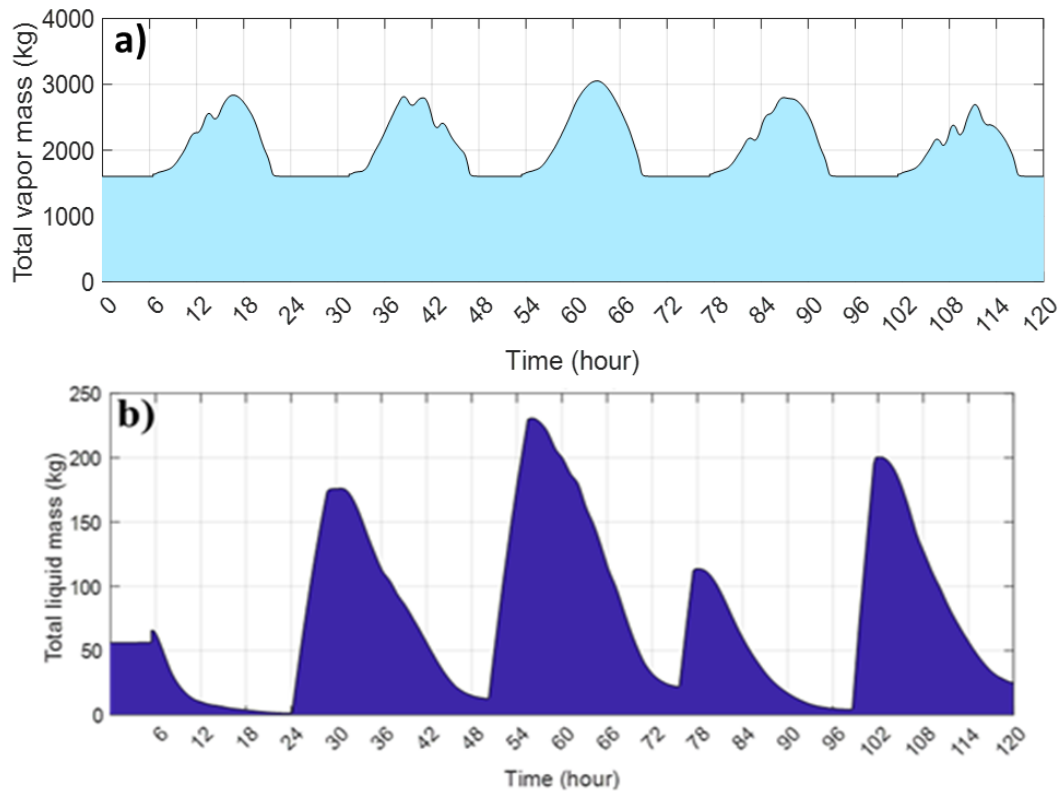


Figure 4.40: Variations in the stored volume of the steam accumulator's a) vapor mass, and b) liquid mass

Figure 4.40a shows the values of the mass of the vapor which is stored during the charge period when solar irradiance is available. The total vapor mass in the accumulator reached 5,000kg after the first day, which means that the system is ready to supply energy to the power plant. It is clear that the total vapor mass increases in the last two of the five days, due to the good solar radiation level on the day four and slight fluctuation in solar radiation on the day five where, due to the good irradiance, the system is more charged due to the amount of vapor mass stored in the tank, which thereby increases the vapor mass level. Figure 4.40b shows the value of the mass liquid in the accumulator, which reaches 200kg. This occurs when steam enters the two-phase region and consists of some liquid. It seems that the liquid level increases at night because there is no steam generation, and the condensed water returns to the steam accumulator. When the steam fluid enters the two-phase region at the steam turbine inlet, it may cause severe damage to the turbine blades as a result of the liquid's

impact, making the system output power unstable. The steam accumulator used for this was also intended to keep the system operating safely, with no damage to the turbine.

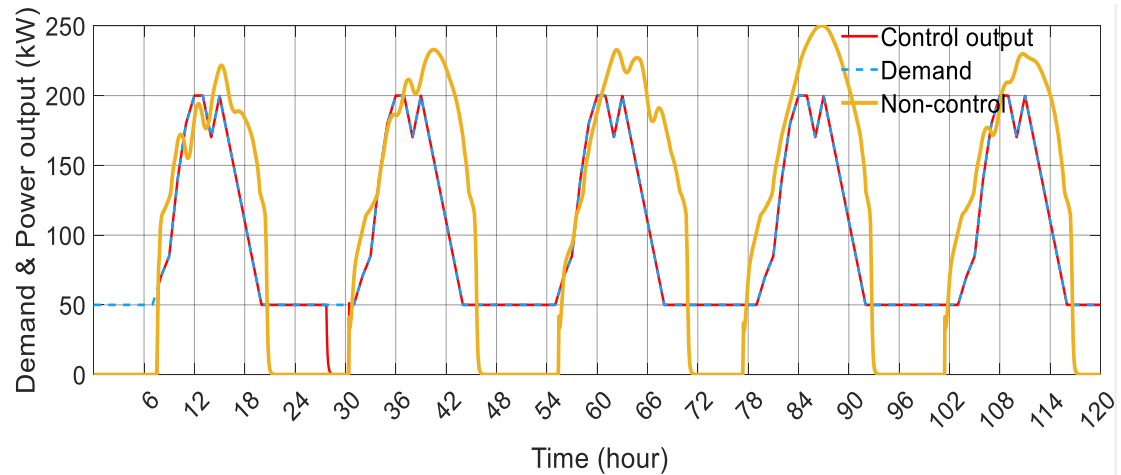


Figure 4.41: Variations in the work output for five days during the charge and discharge modes

Figure 4.41 shows the system work output over the five days. The trend is quite similar for the energy demand, as the system control is forced to produce the required electricity but, during the first six hours, the production does not match the demand. One of the reasons for this is that no heat is stored in the steam accumulator, which means that there is insufficient pressure to run the system. After fully charging the accumulator during the first day, the system is ready to supply heat to the turbine, which leads to the production of electricity during the night. Thanks to the steam accumulator providing heat during typical climate conditions and during the night, the system can match the energy demand of the hospital throughout the day. The drop in the power output at the end of the first day is caused by the valve control, when it is opened and closed upon the requested valve. Also, in the same figure it can be seen that the orange line presents non-control system which resulting much energy has been lost.

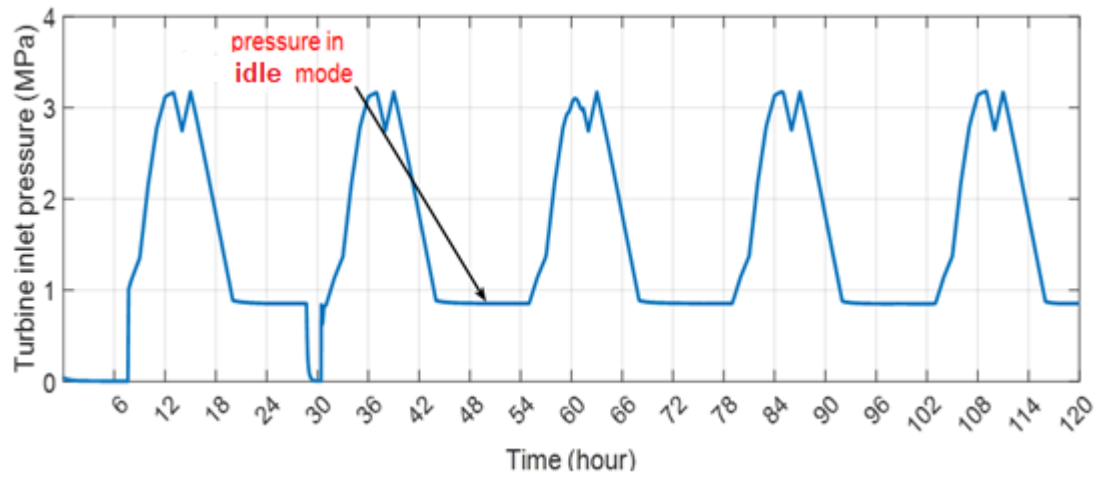


Figure 4.42: Variations in the turbine inlet pressure over five days during the charge and discharge modes

Figure 4.42 shows the variation in the steam pressure at turbine inlet. Based on the control pressure valve, which is set up before the turbine, the steam accumulator can maintain the pressure level on the operating level within a range between 0.9 to 3.2 MPa. The design point of the turbine inlet pressure level is set at 0.9MPa, as recommended for steam turbine operation. Thus, by applying this strategy, the steam turbine can work only if the inlet pressure is ≥ 0.9 MPa. As a control strategy is used by using the throttle valve to meet the electricity demand profile for each day, Figure 4.43 shows the dynamic response of the throttle valve area, and the open and closed areas to meet the electricity demand. When the pressure of the working fluid falls, the valve opens to allow a higher flow rate to meet the demand profile.

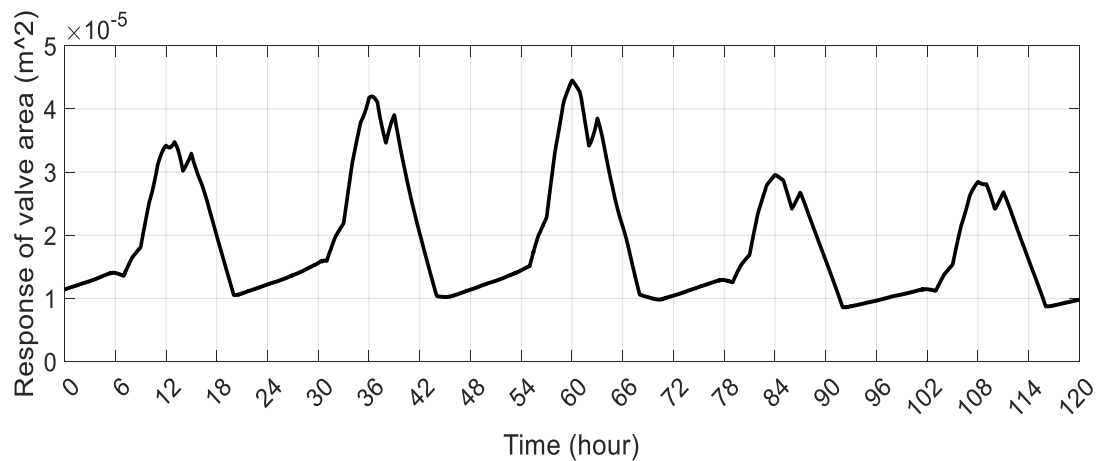


Figure 4.43: Dynamic response of the throttle valve area, and the open and closed areas to meet the electricity demand for the five days

Figure 4.44 shows generated temperature of the DHW for five working days. From 10:00AM to midnight, the outlet hot water temperature in the DHW processing reaches 50°C, with 5.5kg/s flow rate. The constant DHW flow rate is used because the hospital has a DHW storage tank and the condenser is already extracting the remaining heat from the working fluid. Thus, it can be said that the DHW is stored when the temperature exceeds 40°C, the condenser removes all of the heat in the remaining times. Also, the turbine inlet temperature is also displayed in Figure 4.44.

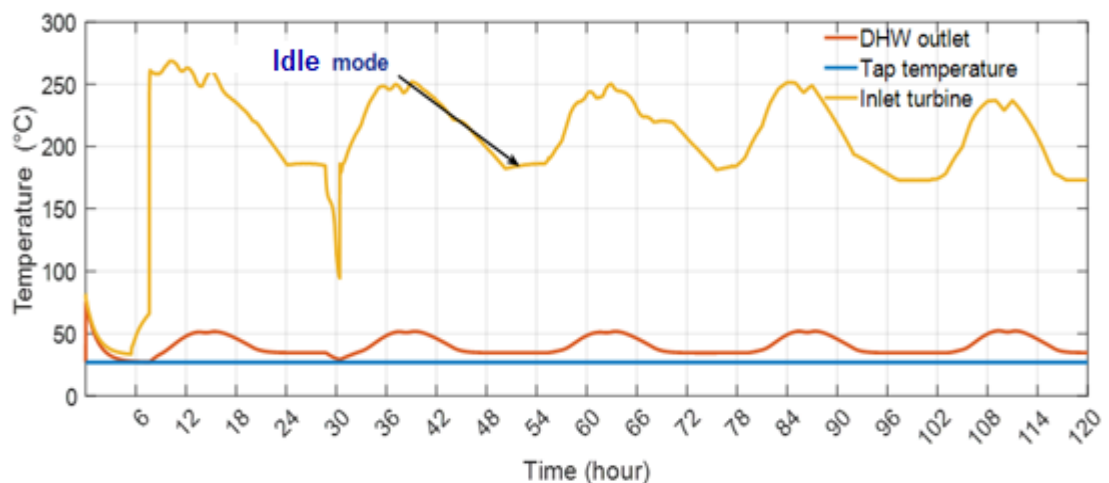


Figure 4.44: Variations in the outlet temperature in DHW for five days during the charge and discharge modes

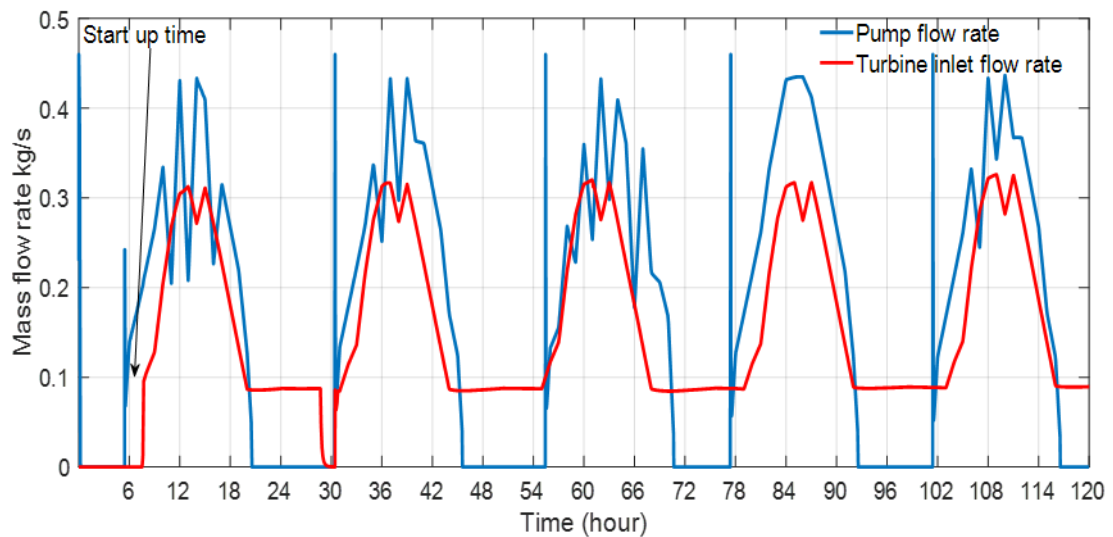


Figure 4.45: Variations in the pump flow and turbine inlet steam flow rates

The findings indicate that the steam flow rate that leaves the heat storage tank and enters the turbine has been controlled by the throttle valve to meet the demand for power. Figure 4.45 shows that the red line is the turbine inlet steam flow, which is controlled, and is stable during discharge mode, while the blue line is the mass flow rate by the pump, which attempts to keep the steam accumulator fully charged during the day, when solar radiation is abundant.

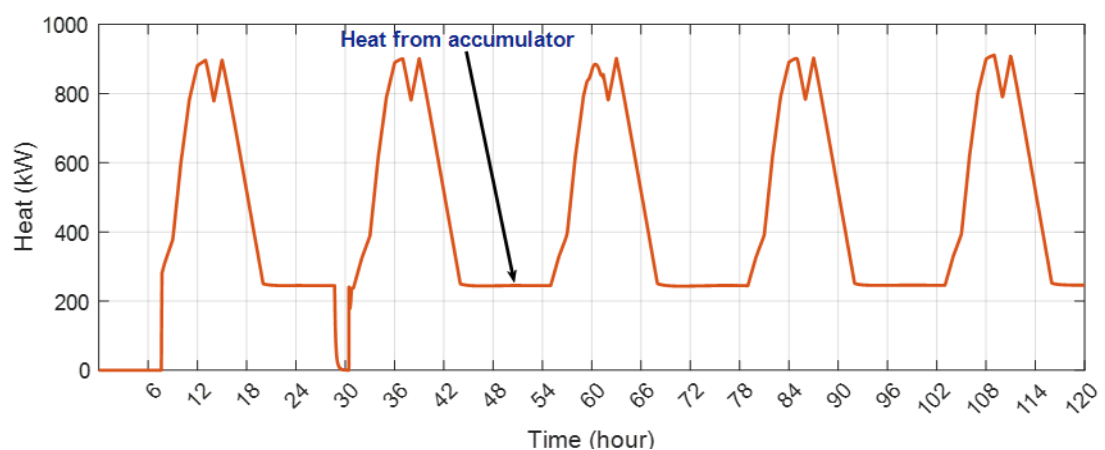


Figure 4.46: Variations in the energy flow rate in the system over the five days

Figure 4.46 shows the energy flow rate to the turbine. There are two periods, time one when solar radiation is abundant and another when the system is powered by the steam

accumulator. During the discharge mode, the steam accumulator supplies 210kW of heat energy to run the turbine, to match the demand profile.

4.6.4 Further discussion

Based on the analysis presented in this study, the performance of a solar direct steam generation system coupled with a steam accumulator and DHW system to meet the electricity and domestic hot water requirements of a Libyan hospital is numerically modelled and evaluated. The results reveal that the system has the ability to meet the maximum electricity demand, which is 200kW, even under transient climate conditions. The findings show that the proposed system is also expected to be more economical than the conventional approach and ORC solar systems, especially in countries with abundant solar radiation. Steam accumulators have proven their efficiency and ability to cover some solar clouds, and can also run a turbine for long periods after sunset, but this requires an efficient control system that depends on the throttle valve and pressure valve. The system was designed and evaluated by using a pressure control strategy ranging from 0.9 to 3.2 MPa of inlet pressure of the steam turbine in order to overcome the steam turbine's issues.

The system can produce domestic hot water with a temperature of up to 50°C to use to fulfil the hospital's needs. The thermal efficiency of the system reaches 0.235 when solar irradiance is available and 0.20 when the system worked based on the stored steam in the steam accumulator. Furthermore, it is structurally different from the traditional ORC solar system for steam turbines, and the proposed system is a simple configuration compared to other systems used to meet the required energy needs of buildings.

The additional advantages of the proposed system, such as the control strategy, are relatively simple. In the prevailing CSP systems, a superheater is necessary and

complex measures must be implemented in the process of transferring heat from oil to water under volatile solar radiation to ensure the safe operation of the turbine. In contrast to this proposed system, a vapor accumulator can separate the presence of a liquid vapor mixture or a saturated vapor state. The turbine inlet flow rate can be adjusted by controlling the throttle valve, allowing easier heat collection and storage. The operating temperature of the solar collectors of about 250°C is significantly lower than that of the prevailing CSP systems, which is 400°C or higher.

4.7 Summary

In this chapter, a comprehensive and universal steam accumulator (SA) model for different thermal storage technologies is clearly proposed based on a direct steam generation solar power system and a fixed parabolic through a solar collector's area of 1320m². In order to study the dynamic characteristics of the steam accumulator, the charge/discharge process and disturbance test were carried out for the system. An integrated steam accumulator with a Rankine cycle as the heat discharge mode for six hours has been modelled, and the effect of different steam accumulator storage volumes on the power output was evaluated. The performance of a direct steam generation solar power system, integrated with a steam accumulator under a clear sky and a cloudy sky, has been simulated. A weekly evaluation of the proposed system with and without a stable output control was performed, using the Simulink/Simscape tool. In addition, a proposed system to fulfil the energy demand of a Libyan hospital, as a case study, was designed and simulated. The key findings are as follows:

- The steam accumulator has the ability to enable the direct steam generation solar power system to overcome the fluctuation in solar radiation during the day as well as working as a separator liquid when steam consists of a two-phase regime.

- Based on the constant solar field area of 1320m^2 , the proposed system can generate up to 210kW without a control strategy and, with a controlled system, can generate a stable 180kW , even when there is fluctuation in the solar radiation.
- According to research, it was found that, to meet the electricity and hot water demand of a Libyan hospital, a $1,544\text{m}^2$ collector area and 160m^3 steam accumulator were required for the solar steam Rankine system with a throttle valve control.

Chapter 5 Dynamic simulation of a cascade steam-organic Rankine cycle solar power system with a steam accumulator

5.1 Introduction

One of the existing approaches for improving the efficiency of the current power generation cycles, such as Brayton and Rankine cycles, is to use waste heat from the top cycle as the heat source for the bottom cycle. The combined Brayton-Rankine cycle is one of the most common powers generating cycles, and has been the subject of numerous studies [134,135]. In this sort of combined cycle, the waste heat from the Brayton cycle is used to heat the bottom Rankine cycle. Compared to individual steam or gas turbine power plants, this operation results in higher entire cycle thermal efficiency and a lower environmental impact [136]. Combination SRC-ORC and high temperature ORC-low temperature ORC are two further types of power generating combination cycles that have recently been investigated, especially with regard to waste heat recovery systems. The heat source of the bottom cycle in these types is waste heat from the top Rankine cycle's condenser, a lowered source fluid temperature flowing out of the top cycle's evaporator, or a combination of the two [137,138] proposed a new dual loop ORC that combined a high temperature loop (HT) and a low temperature loop (LT) to recover waste heat from an internal combustion engine under various load conditions. The heat source for the HT loop was high-temperature exhaust gas, which was subsequently employed as the driver for the LT loop when the temperature in the HT loop was reduced. Aside from exhaust gas, the jacket cooling water and transferred heat to the LT fluid in the top cycle condenser were utilised as heat sources in the LT loop. Water was used as the working fluid in the HT loop, whereas six different organic fluids were used in the LT loop. The results indicate that the temperature of the HT evaporator and LT evaporator were respectively equal 570 and 343 K. When organic fluid R1234yf was used, 6.77 kW of output power was

achieved. [139] demonstrated a combined Rankine cycle with an air-cooled condenser that incorporated an SRC with a bottom ORC. In this work, waste heat from the SRC condenser was utilised as a low-temperature heat source for ORC. In addition, nine different working fluids were chosen for ORC, and the impacts of the ambient temperature variations and steam cycle condenser pressure changes on simulated combined cycle performance were evaluated from an energy efficiency standpoint. [140] examined the energy and exergy of the SRC and the combined steam-organic Rankine cycle (in which waste heat from the steam Rankine condenser was used as an organic Rankine cycle driver), driven by PTSC. In this study, the vacuum condensing pressure (nearly 5 kPa) and atmospheric condensing pressure (101.3 kPa) were used as the basic case for the steam cycle condenser. Due to the greater condensation temperature of steam in the second stage, it can be employed as a low temperature heat source for ORC. Finally, parametric analysis was used to study the impacts of parameter modification on the energy efficiency and exergy destruction of various components, such as direct normal irradiation and steam turbine inlet pressure. According to the review, recent studies have focused on individual SRC, individual ORC or a combined cycle of SRC-ORC, and no comprehensive study has compared the various combined SRC-ORC configurations besides a comparison with simple SRC. In addition, solar-driven cascade steam-organic Rankine cycles based on a direct steam generation system have not yet been fully investigated in detail in terms of the control and behaviour of each component in the cycle. Also, such systems are very rare, and so provide an interesting research avenue at the current time.

This unique contribution of this study is as follows:

- This chapter aims to evaluate the desired system from the aspects of energy and a dynamic behaviour response for each component.

- The dynamic simulation under different typical days of solar irradiation as well as the design of various system components, with more details and using different types of working fluid, will make it possible to obtain more accurate results and overcome any potential challenges, which cannot be achieved in the case of static or brief simulations.
- Four different working fluids (R600a, R134a, R1234yf and R407c) were investigated in this study.

5.2 Description of the system and its operating modes

A schematic diagram of the solar steam-organic Rankine system is illustrated in Figure 5.1. The system consists of the solar collectors' field, a steam Rankine cycle and a bottom organic Rankine cycle, respectively. The top cycle is the steam Rankine cycle, which consists of 100 parabolic trough collectors that absorb solar energy. It has two regions in the solar field (a boiler region and a superheated region), a steam accumulator used to store the steam heat for the system, a steam turbine to generate the electricity from the top cycle, an internal heat exchanger (IHX) that is used to transfer heat between the steam cycle and the ORC system and a water pump. Also, there are several valves in the top cycle, they are used to control the solar irradiance, pressure, and mass flow rate of the steam cycle. In the bottom cycle, which is an organic Rankine cycle, there is an ORC turbine (used to generate the electricity), an ORC condenser with a small liquid receiver and an ORC pump. In the ORC recuperation system, there is an extra internal heat exchanger named a recuperator heat system, which is used to transfer heat between the low side pressure and the high-pressure flow circuits. Its function is to improve the system performance by further sub-cooling the refrigerant being supplied to the evaporator through the refrigerant control device. In fact, it is a tube in a tube. In the solar-driven cascade Rankine cycle,

there are a high-pressure and temperature cycle and a low-pressure and temperature cycle.

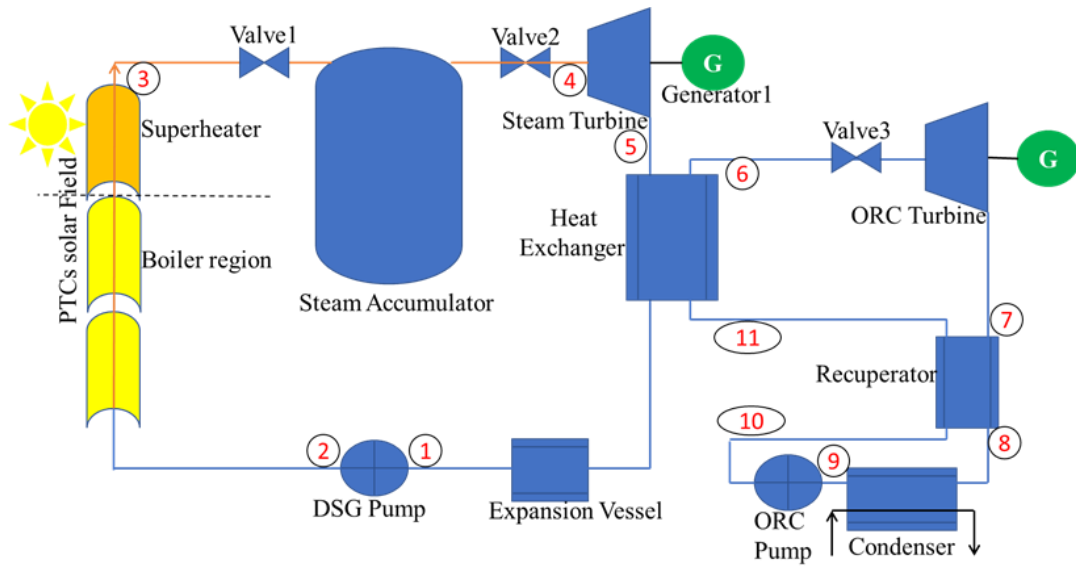


Figure 5.1: Schematic diagram of a cascade steam-organic Rankine cycle solar power system

In the proposed system, there can be two operating modes:

Mode (I): The solar irradiance is available and exceeds the 350 W/m^2 per day that the proposed system requires to produce electricity. In this mode, valve 1 is open, and DSG pump is run. The water pump in the direct steam Rankine cycle pumps the water to the solar field to be heated and vaporized. The saturated steam goes to the steam store to be fully charged. The pressure control valve is opened when the steam pressure reaches the setpoint pressure. The saturated steam flows into the steam turbine, exporting power during the process of enthalpy drop by a generator. The exhaust vapor from the turbine is condensed to saturated liquid in the internal heat exchanger, pumped by the water pump, then sent back to the solar fields. The condensation heat is utilized to vaporize the organic fluid in the bottom organic Rankine cycle. In this cycle, the ORC pump pressurizes the working fluid into the heat exchanger, which is placed between two cycles to be heated and vaporized. The

pressure valve control of the ORC turbine is open to a set point pressure, superheated stream flows into the turbine, and the exporting power during the process of enthalpy drops via the generator in the bottom cycle. The exhaust stream from the ORC turbine is condensed to saturated liquid in the ORC condenser, accumulates in the small liquid receiver and is then pumped back by the ORC pump to the heat exchanger, after which the cycle is repeated. In a recuperation ORC system, a recuperator is used to preheat the working fluid before it enters the heat exchanger, thereby increasing the efficiency of the ORC system.

Mode (II): The proposed system still requires electricity to be generated, but solar irradiation is unavailable or under 350 W/m^2 . Valve 1 is closed. The steam accumulator will be in full steam charged mode. The stored heat is released by the steam accumulator and converted into both the steam Rankine cycle and ORC system, respectively.

5.3 Thermodynamic modelling

5.3.1 Modelling of the bottom ORC system

In this chapter, the integrated bottom ORC system is modelled and simulated since it displays a perfect balance between efficiency, power output and costs, especially for medium or small power outputs, which is the main focus of this chapter. Figure 5.2 shows integrated ORC system to the bottom of the direct steam generation solar power system.

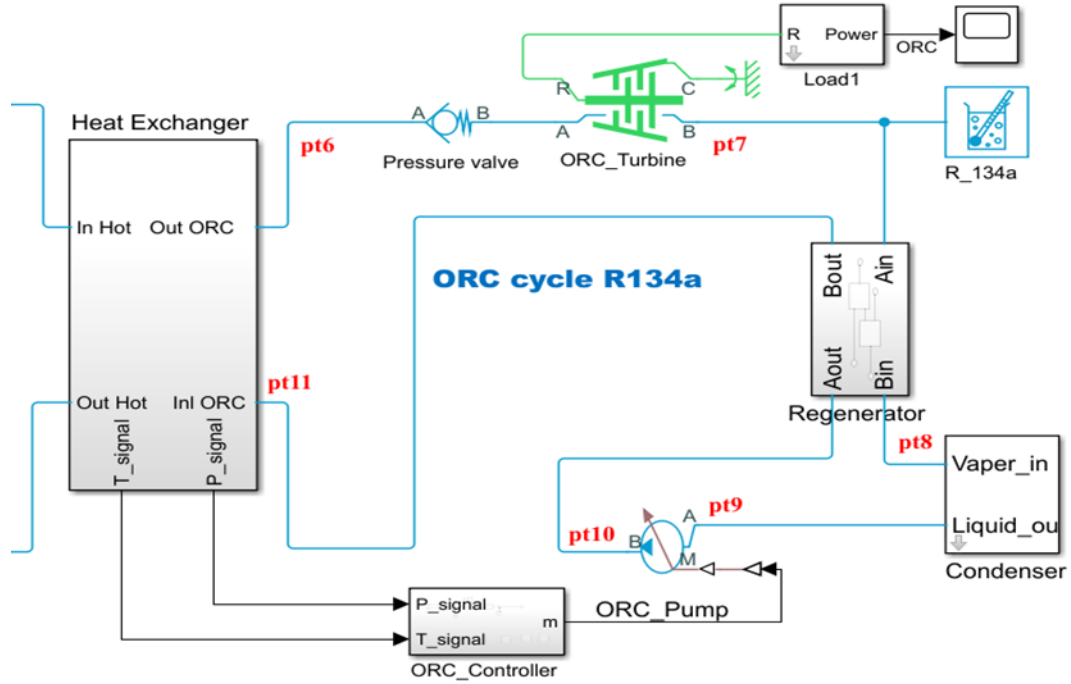


Figure 5.2: integrated ORC system to the bottom of the direct steam generation solar power system in Simscape.

A T - s diagram of a recuperative ORC system is shown in Figure 5.3. Both the steam Rankine and organic Rankine are taken into consideration during subcritical cycles in the proposed system. Figure 5.3 illustrates the thermodynamic processes represented by the T - s diagram when, for instance, R134a is used as the ORC fluid. The steam Rankine cycle is represented by the blue line and the organic Rankine cycle by the red line. The numbers represent the water and 134a thermodynamic states that correspond to the marks on Figure 5.1. Additionally, the heat exchangers and pipes' thermal and frictional losses are ignored. The simulation ignores changes in kinetic and potential energy.

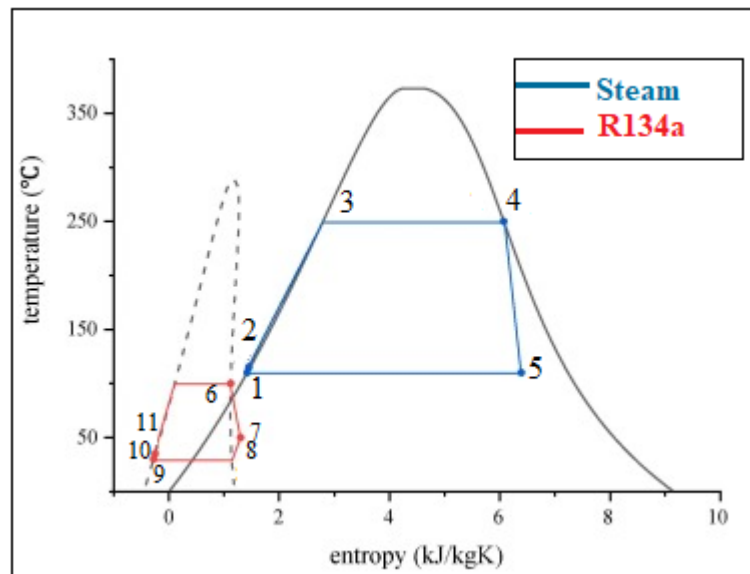


Figure 5.3: T-s diagram of the solar steam-organic Rankine system

For the top steam Rankine cycle, a wet steam turbine is used. For many years, notably in nuclear power plants, this kind of turbine was in use. Modern turbines can now handle binary-phase steam at dryness levels lower than 90% as a result of the extensive development. The enormous power capacity of steam turbines, which can be two orders of magnitude more than that of positive displacement expanders, is one of their advantages. Due to this benefit, the power block in the entire solar plant has low-cost and fast payback period.

A steady-state thermodynamic model of the ORC engines, shown in Figure 5.1, was modelled using the Simulink/Simscape program, by applying the energy and mass balance equations for each component of the proposed system. The integrated bottom ORC model is described by Equations 5.1 to 5.13. Several assumptions when modelling the integrated bottom ORC with the solar Rankine system are listed below:

- All heat exchangers were assumed to be of a counter-flow design, and adiabatic.
- The pressure drops on the working fluid side in the heat exchangers, and piping is negligible.

- The transient operation of the ORC components.
- The pinch point temperature in the heat exchanger is assumed to be 10K.

The heat flows into the bottom ORC system, in the heat exchanger (evaporator), which is process 9-6 or 11-6 in the recuperative ORC system. In the integrated solar-driven cascade steam-organic Rankine cycle, the heat rejected by the steam side in the internal heat exchanger is used to drive the ORC. The heat rejection in the steam side of the internal heat exchanger is equal to the heat input to the organic evaporator side. The heat flows rate input can be expressed by using the working enthalpy increase between states 10-6 or 11-6, as follows:

$$\dot{Q}_{evap} = \dot{m}_{ORC} (h_6 - h_{10}) = \dot{m}_{ORC} C p_{ORC} (T_6 - T_{10}) \quad (\text{No-recuperator}) \quad (5.1)$$

$$\dot{Q}_{evap} = \dot{m}_{ORC} (h_6 - h_{11}) = \dot{m}_{ORC} C p_{ORC} (T_6 - T_{11}) \quad (\text{Recuperator}) \quad (5.2)$$

The superheating degree of ORC is used to control the ORC flow rate in the system. The superheated working fluid enters the ORC turbine to enable the expansion process to generate the electricity. It is better to use the ORC turbine for bottom integrated cycles [141]. The power output of the bottom ORC turbine in this integrated proposed system can be calculated by:

$$\dot{W}_{T,ORC} = \dot{m}_{ORC} (h_6 - h_7) \eta_{mech} \eta_{ele} \quad (5.3)$$

Where, h_6 , h_7 are the inlet and outlet enthalpies of the ORC turbine, respectively, η_{mech} is the mechanical efficiency of the turbine and η_{ele} is the electricity generator efficiency.

When the expansion process has been completed, the low-pressure exhaust stream flows into an ORC condenser to be cooled down at the liquid state or into the recuperator and then the condenser. In the latter state, the recuperator is used to preheat the working fluids after the ORC pump and before it enters the heat exchanger

(evaporator) to increase the performance of the bottom ORC system. The heat gained by the recuperator is calculated by:

$$\dot{Q}_{recup} \dot{m}_{ORC} (h_{11} - h_{10}) = \dot{m}_{ORC} (h_7 - h_8) \quad (5.4)$$

Adding an extra component, such as an internal heat exchanger or recuperator, might affect the total cost of the system. However, these extra components can enhance and improve the performance of the system. The heat rejected by the ORC condenser is expressed by:

$$\dot{Q}_{cond} = \dot{m}_{ORC} (h_7 - h_8) = \dot{m}_{ORC} C_{p_{ORC}} (T_7 - T_8) \quad (\text{Non-recuperative ORC}) \quad (5.5)$$

$$\dot{Q}_{cond} = \dot{m}_{ORC} (h_8 - h_9) = \dot{m}_{ORC} C_{p_{ORC}} (T_8 - T_9) \quad (\text{Recuperative ORC}) \quad (5.6)$$

The work consumed by the working fluid pump in the ORC system is calculated as:

$$\dot{W}_{ORCpump} = \frac{\dot{m}_{ORC} (h_9 - h_{10})}{\eta_{ORCpump}} \quad (5.7)$$

The thermal efficiency of the system is calculated as follows:

$$\eta_{thermal,syst} = \frac{\dot{W}_{DSG,turbine} + \dot{W}_{ORCturbine}}{\dot{Q}_{solar}} \quad (5.8)$$

5.3.2 Modelling of heat exchanger

Two counter-flow heat exchangers were used to model the heat transfer between the solar steam Rankine cycle and ORC system (heat exchanger 1) and another heat exchanger in ORC (heat exchanger 2), named a recuperator, and are used for the proposed heat recovery system. The sketch of the heat exchanger in the Simscape model is shown in Figure 5.4.

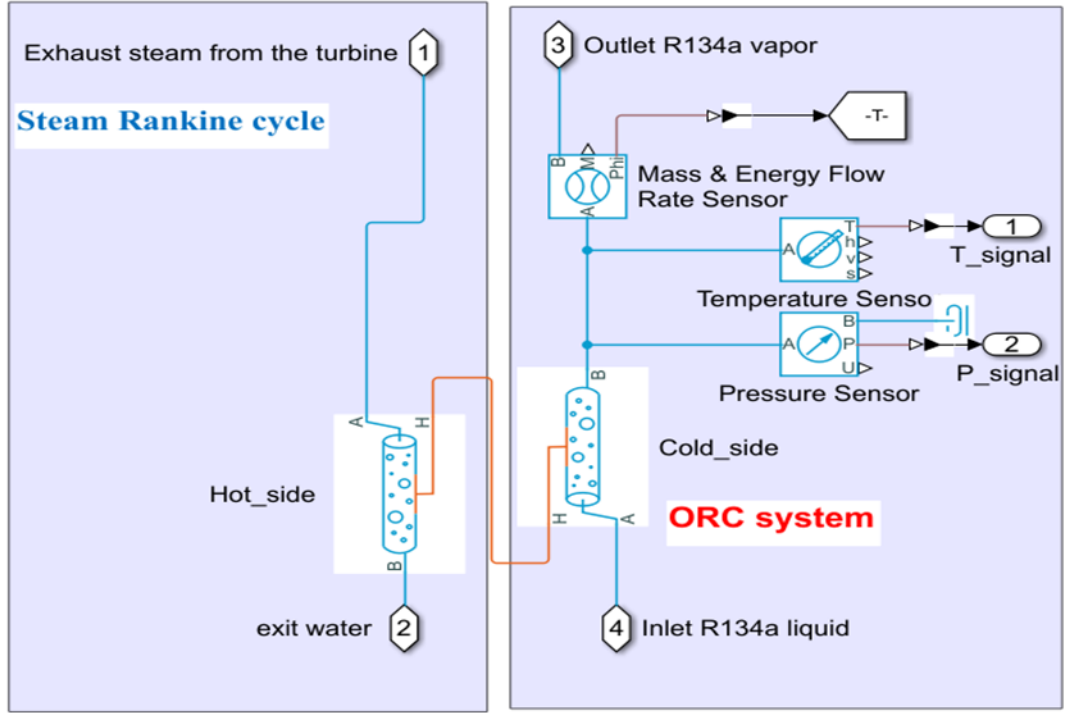


Figure 5.4: The sketch of the heat exchanger in the Simscape

The Effectiveness-NTU method was selected to model these heat exchangers [133], and their mathematical equations are as follows. The maximum possible heat transfer, which occurs in the heat exchanger, and the actual heat transfer rate can be calculated from Equation 5.9 and 5.10:

$$\dot{Q}_{Max} = C_{Min} \cdot (T_{hot,in} - T_{cold,in}) \quad (5.9)$$

$$\dot{Q}_{HX} = \varepsilon_{HX} \cdot C_{Min} \cdot (T_{hot,in} - T_{cold,in}) \quad (5.10)$$

The following defines the transfer unit number used to analyse the heat exchanger and it can be determined by:

$$NTU_{HX} = \frac{U_{HX} \cdot A_{HX}}{C_{Min}} \quad (5.11)$$

The following is the determined relationship between the counter-flow heat exchanger efficiency and NTU [133]:

$$\varepsilon_{HX} = \frac{1 - \exp[-NTU_{HX}(1-C_r)]}{1-C_r \exp[-NTU_{HX}(1-C_r)]} \quad (5.12)$$

The heat capacity ratio can be determined by:

$$C_r = \frac{C_{Min}}{C_{Max}} \quad (5.13)$$

5.4 Working fluid selection for the bottom ORC system

The choice of working fluid can have a significant impact on the performance, cost, and operating conditions of ORC systems. This chapter has examined a variety of working fluids to find the best candidate(s) for the low temperature solar-driven cascade steam-organic Rankine applications. Several working fluids were pre-selected due to the high computational cost of optimization. In addition to the working fluids' excellent thermodynamic performance features (high vapour density, low specific heat capacity, and high latent heat), their environmental impact has also attracted increasing attention. This is because rigorous restrictions were implemented, requiring the gradual phasing-out of refrigerants with high ozone depletion potential (ODP) and global warming potential (GWP). Refrigerant usage in the air conditioning and refrigeration business is subject to a number of restrictions in the EU under the so-called F-gas Regulation, which has recently been expanded to include power production units, such as ORC systems [142]. To maximise the market penetration of ORC systems, technical solutions that are not bound by such rules are needed. Due to the high operating temperatures of the working fluids, which may accelerate the fluids' chemical breakdown unless carefully studied at the design stage, there exist additional hurdles in the combined ORC systems when recovering heat from waste exhaust gases. Based on the foregoing, the refrigerants R134a, R1234yf, R407c and R600a were chosen for this investigation because they possessed favourable thermodynamic properties and low ODP and GWP values.

Table 5.1 provides a summary of the most popular fluids that have been previously taken into account in screening tests for low-temperature applications or fluids utilised in commercial ORC power plants. However, by first deleting all fluids with non-zero ODP or GWP values more than 1,500, this list can be reduced. Due to their extremely low evaporation temperature in the subcritical cycles, fluids with a critical temperature below 80 °C are then eliminated. Additionally, only dry and isentropic fluids are taken into account to reduce the risk of turbine blades damage resulting from wet fluids. Four working fluids were identified as prospective candidates after the initial screening; their thermodynamic, safety, and environmental characteristics are listed in Table 5.2. The best fluid for the current ORC system was then selected after a comparative evaluation of these operating fluids. The working fluids most frequently used in low temperature ORC systems include:

Table 5.1: Summary of the most popular fluids that have been previously in ORC systems [142]

Substance	GWP (100 yr)	ODP	T _c (°C)	Group
R134a	4470	0	101.06	HFC
R600a	~20	0	134.67	HC
R1234yf	4	0	94.70	HFO
R407c	1774	0	86.74	HCFC
R22	1810	0.05	96.15	HCFC
R245fa	1030	0	154.05	HFC
R125	3500	0	66.0	HFC
R115	7370	0.44	79.95	CFC
R236fa	9810	0	124.92	HFC
R218	8830	0	71.95	PFC

Table 5.2: Environmental and thermodynamic characteristics of the chosen fluids [144]

Working Fluids	Critical Pressure (MPa)	Critical Temperature (°C)	Boiling Point (°C)	Molecular Mass (kg/kmol)
R134a	4.60	101.06	-26.07	102.03
R1234yf	3.3	94.7	-29	114.0
R600a	3.629	134.66	-11.75	58.12
R407c	4.6	86.74	-43.5	86.2

5.5 Simulation results and discussion

A steam accumulator, used as a heat storage unit and integrated with a cascade steam-organic Rankine cycle system, is expected to be applicable to direct steam generation solar power systems powered by parabolic trough collectors based on the analysis undertaken for this study. In this chapter, the integrated proposal system has been modelled and simulated during a whole day with respect to the solar irradiance level by using Simulink/Simscape software in order to assess the dynamic response and behaviour of each component of the proposal system, which is a unique method that has not been studied in detail previously. R600a, R134a, R1234yf, and R407c are selected as the ORC fluids. These fluids have been widely investigated. Some specific parameters for the proposed system are listed in Table 5.3 below. It has been evaluated the dynamic response of the solar-driven cascade steam-organic Rankine cycle by using Simscape from initial condition to steady state with given constant 1000 W/m^2 of solar irradiance as depicted in Figure 5.5. The figure shows two power outputs of

the proposed system, one is from steam Rankine cycle and another from ORC system.

The system reaches to steady state after five hours of simulation.

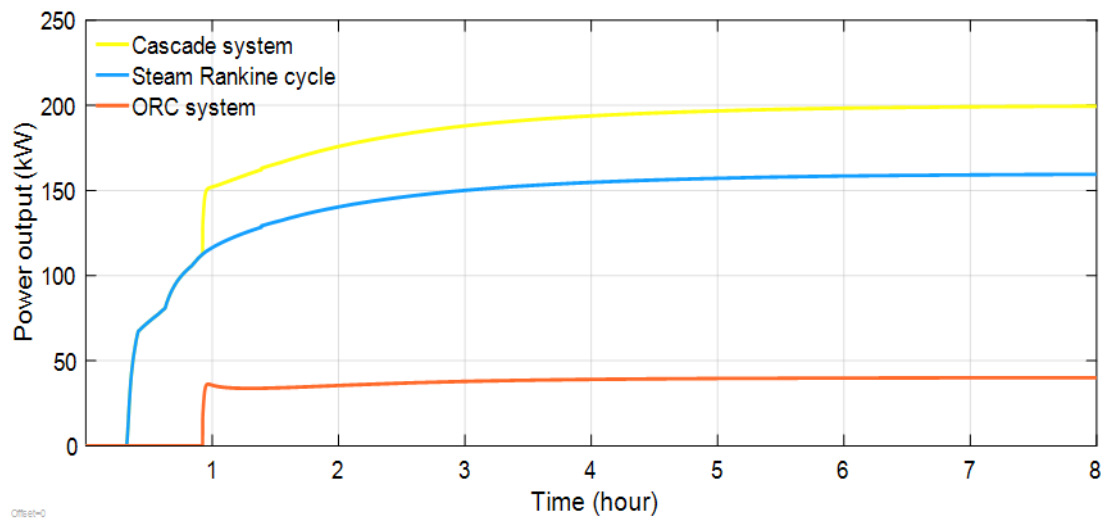


Figure 5.5: Dynamic response of power outputs of the proposed system, one is from steam Rankine cycle and another from ORC system.

The ambient temperature and direct normal irradiance on typical meteorological days in Libya, obtained from EnergyPlus software [6], are presented in Figure 5.6. The schematics diagram of the solar-driven cascade steam-organic Rankine cycle in Simscape is depicted in Figure 5.7.

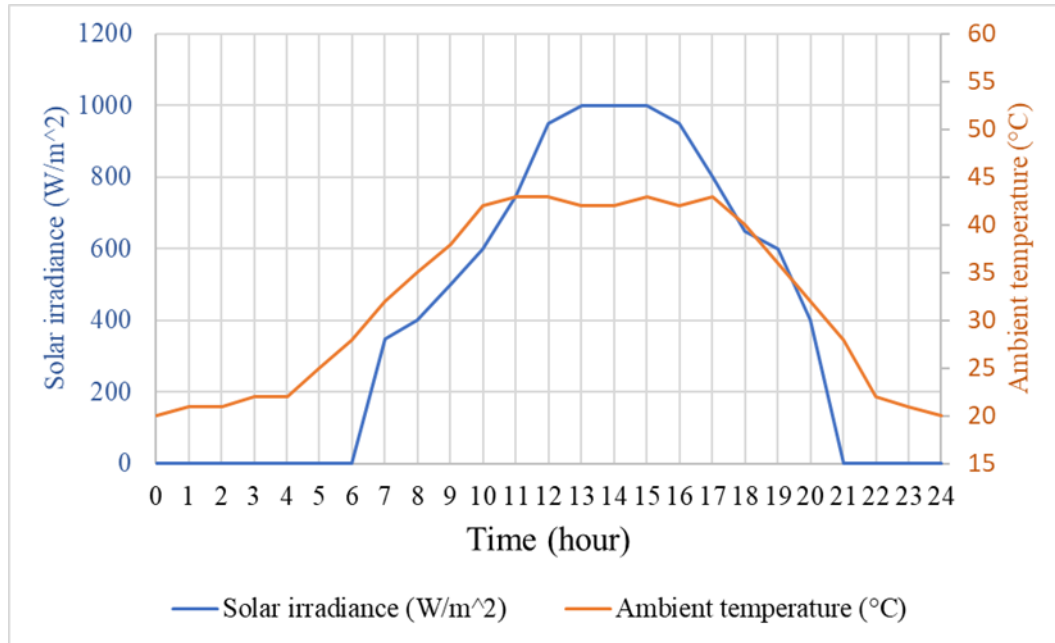


Figure 5.6: Weekly variations for solar irradiance and ambient temperature [6]

In this configuration, the first cycle model is same construct as in the chapter three, and also it was controlled as same previous control strategy. The second configuration cycle is ORC system. It was constructed to be after steam turbine in the top cycle as shown in Figure 5.7. There is intermediate heat exchanger connected between exhaust steam line and ORC evaporator side. The ORC mass flow rate was controlled based on the temperature and pressure in exit evaporator in ORC side.

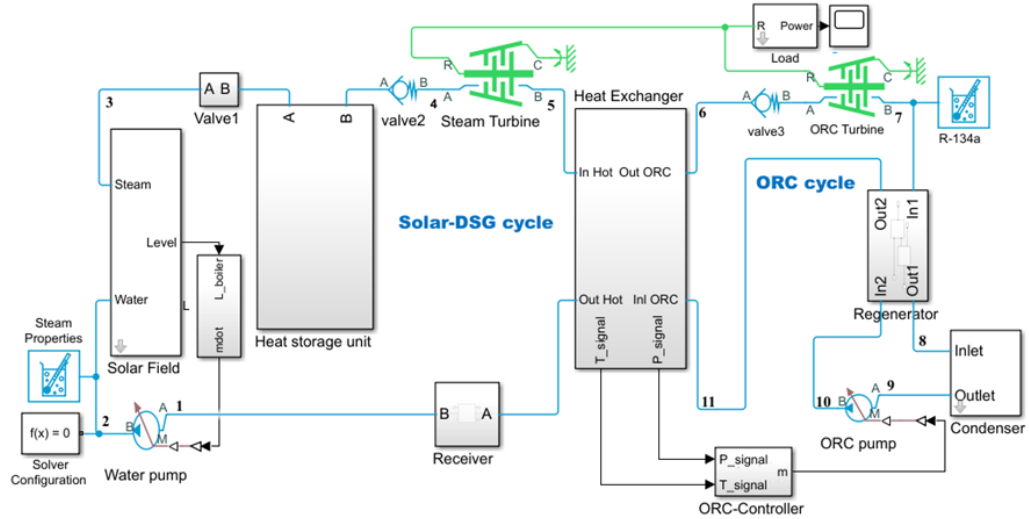


Figure 5.7: Simscape model of a cascade steam-organic Rankine cycle solar power system

Table 5.3 below summarizes the design condition of the solar-driven cascade steam-organic Rankine system.

Table 5.3: Design condition of the solar-driven cascade steam-organic Rankine system

Parameter	Value
Solar collectors' area	1320m ²
Steam flow rate	0.33 kg/s
Length of heat exchanger	350m
Inner and outer diameter of the pipe in the heat exchanger	0.01/0.03m
Volume of the steam accumulator	100m ³
Steam turbine Isentropic Efficiency	0.85
ORC turbine Isentropic Efficiency	0.85
Mechanical generator Efficiency	0.95
ORC mass flow rate	3 kg/s
Condenser temperature	25°C
Pinch point temperature difference	10°C

5.5.1 Effect of the recuperator on the performance of the solar-driven cascade steam-organic Rankine cycle

As is known, a recovery system can improve the performance of solar ORC systems. For this reason, we conducted a comparative study of a solar cascade steam-organic Rankine cycle with and without using a recuperative system in the bottom solar ORC cycle. This subsection summarizes both the recuperative and basic solar ORC ideal outcomes. It is clear that the recuperative ORC may greatly increase both the ORC thermal efficiency and the system efficiency by using R-134a working fluid. The regenerator system in ORC systems powered by waste heat recovery applications is not always beneficial, according to several studies. Nevertheless, a solar energy-driven recuperative ORC system is better than a basic solar-ORC. The results of this study show that a generator system can significantly improve the performance of the system, as has been demonstrated by many research studies. The power output of the proposed system is improved from 210 to 220 kW at the peak time of solar radiation by adopting a regenerator during a week-long simulation, where the heat emerging from the turbine at point seven is exploited before the working fluid enters the condensation process in the condenser and thus acquires a lot of heat in a process, called preheating, before entering the evaporator. Through this process, the heat wasted by the condenser can be reduced and utilized in the process of preheating the working fluid, which increases the performance of the ORC system, which in turn leads to an increase in the total power output as well as the efficiency of the proposed system. Figure 5.8a shows the variation in the power output with and without using the regeneration system. The overall thermal efficiency of the proposed system improved from 0.17 to 0.20 at the peak time of solar radiation, as shown in Figure

5.8b below. This demonstrates how significantly the recuperator may raise the system's overall efficiency and ORC thermal efficiency. Adding a regenerator to the ORC systems will naturally increase the capital cost of a system. The outcomes, however, indicate that a regenerator should be set up for the ORC system in this study, that is powered by solar energy.

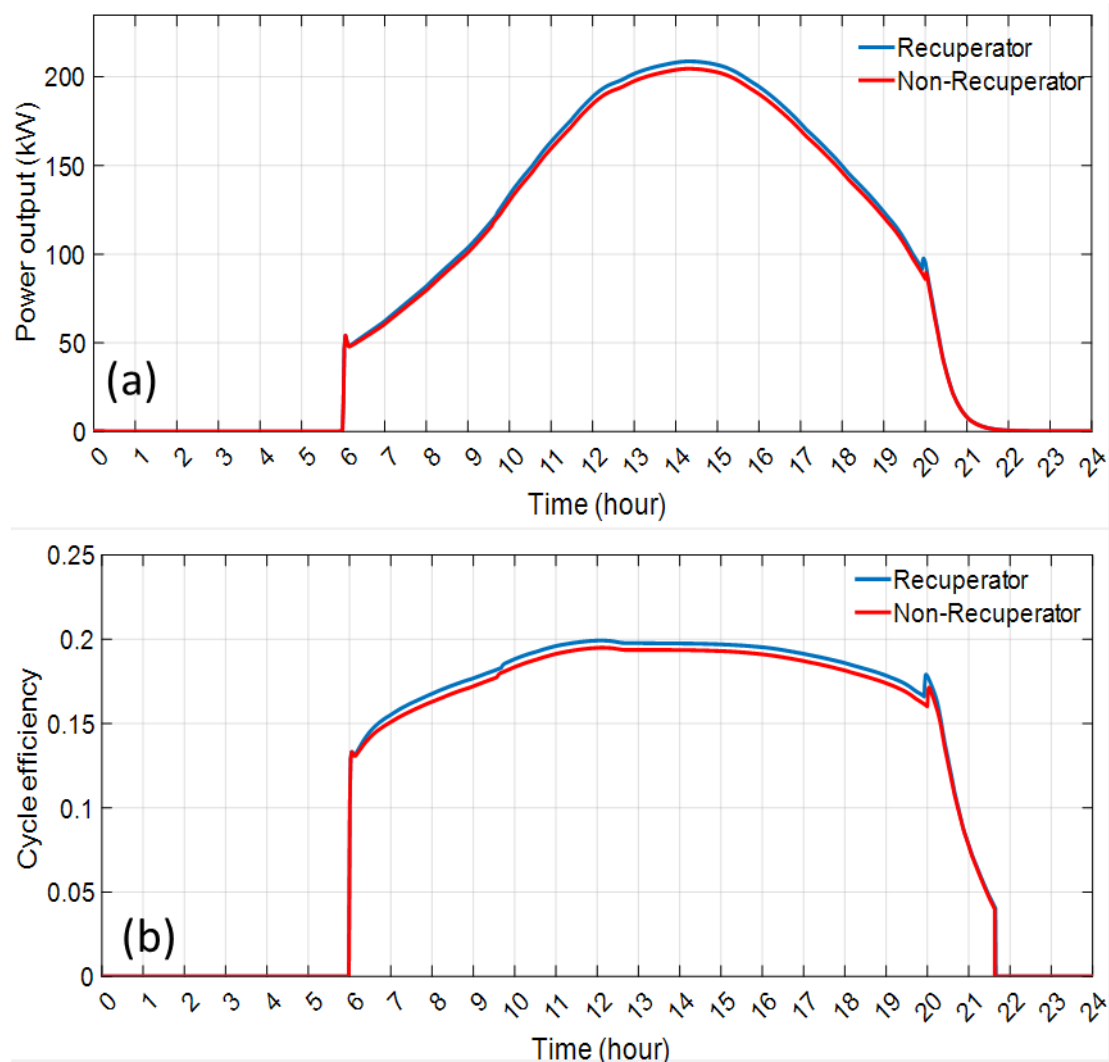


Figure 5.8: Variation a) the power output, b) thermal efficiency with and without using the regeneration system

Figure 5.9a shows variations in temperature in the heat exchanger for both sides with and without using a regenerator system. The hot side temperature represents the

exhaust steam turbine temperature in the solar DSG cycle, while the cold side temperature represents the ORC evaporation temperature in the ORC cycle. It can be seen that the hot side temperature, with and without a regenerator, is the same value and the hot side remains largely unaffected by the use of the regenerator, with only a very slight, almost negligible effect while, for the cold side temperature, there is a slightly effect caused by using the regenerator. For example, the cold side temperature without a regenerator is 76 °C at the peak time for solar radiation, as shown at 13:00 on the first day, and this temperature is improved by using a regenerator to 79 °C at the same time.

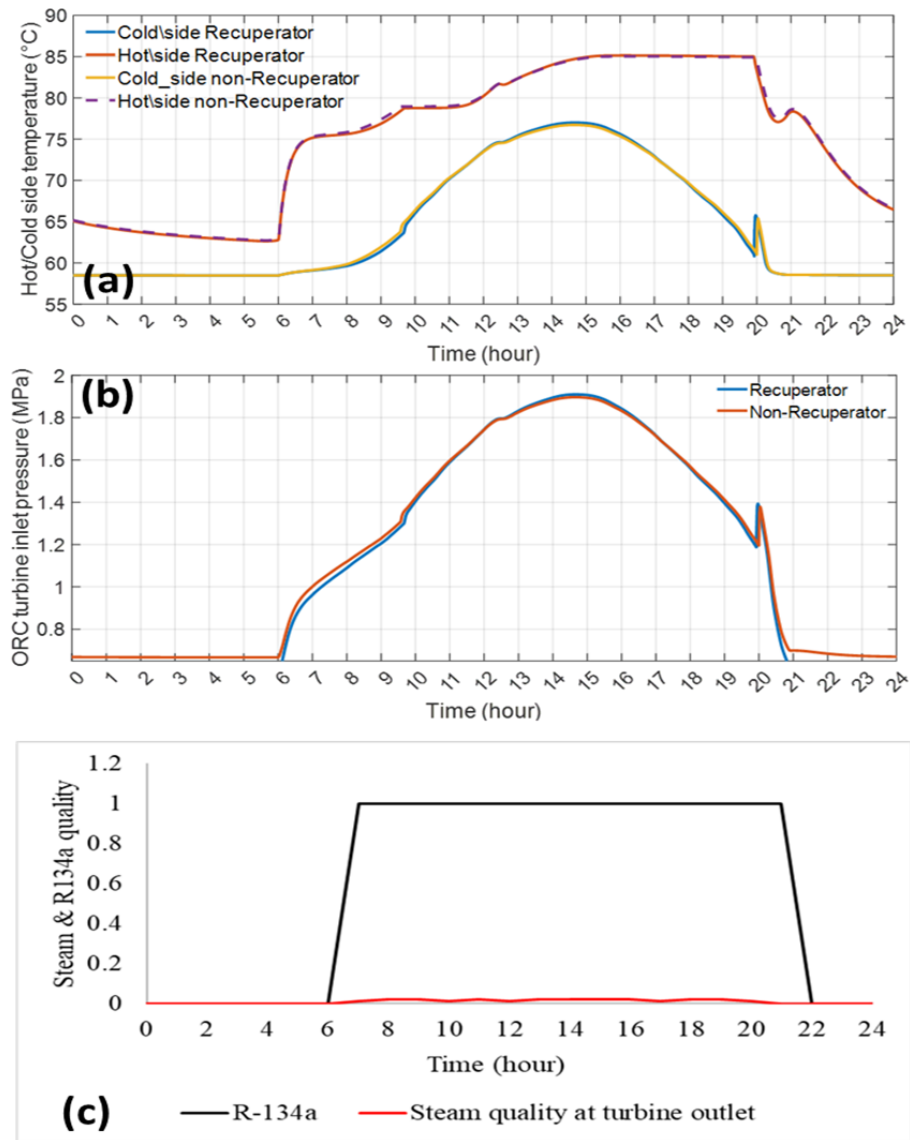


Figure 5.9: Daily variation a) Hot\Cold sides temperature in the HX, b) ORC turbine inlet-pressure and c) Quality for the exhaust steam wetness of the steam turbine and quality of the R134a working fluid in the ORC turbine with and without using regenerator

Figure 5.9b shows a comparison of the ORC turbine inlet pressure with and without using a regenerator during the day-long simulation. As expected, the ORC turbine inlet pressure with a regenerator is higher than that for the system without a regenerator, where the evaporation pressure in the recuperative ORC system reaches 1.9MPa at the peak time for solar radiation while, in the non-recuperative ORC system, the maximum evaporation pressure is 1.7 MPa for the same period. The main

reason for this is the increase in the temperature on the cold side when we use a regenerator. The evaporation pressure is equal to the ORC turbine inlet pressure for both systems in this study. Figure 5.9c presents variation quality for the exhaust steam wetness of the steam turbine and quality of the R134a working fluid in the ORC turbine. It is clear that the exhaust steam wetness of the steam turbine is approximately 0.018 and, most of the time, is 0, while the R134a working fluid turbine inlet is 1 when the system is working and zero when the system is turned off. These results for both recuperative and non-recuperative ORC systems.

Figure 5.10a shows the variation in the heat added to the ORC system by the heat exchanger for both recuperative and non-recuperative ORC systems. We can say that the heat added value is the same for both systems. The heat gained by the ORC evaporator is increased through the rise in solar radiation and decreased by the decline in solar radiation during the day. The maximum value of the heat gained by the ORC evaporator is 750 kW for both recuperative and non-recuperative ORC systems during the day. Each day has a different heat gained value, which is completely dependent on the solar radiation level. The reason for the high jump line in Figure 5.10a at 20:00 PM is due to the response of the control valve installed in the evaporator when it is opened and closed according to the superheating degree control strategy in the bottom ORC system. Figure 5.10b depicts variations in the ORC mass flow rate for both the recuperative and non-recuperative ORC systems. The ORC mass flow rate is adjusted based on the degree of superheating in the outlet temperature of the ORC evaporator in order to guarantee that only the ORC vapor stream enters the ORC turbine. It should be noted that the recuperative ORC system has a higher mass flow rate than the non-recuperative ORC system. For example, on the first day of the simulation, the maximum mass flow rate for the recuperative system was 3.8 kg/s, while that for the

non-recuperative ORC system was 3.6 kg/s. One reason for this is because the inlet evaporator temperature in the recuperative ORC system is increased, as required to increase the speed of the ORC pump. The reason for the high jump line in Figure 5.10b at 20:00 PM is due to the response of the control valve installed in the evaporator when it is opened and closed according to the superheating degree control strategy.

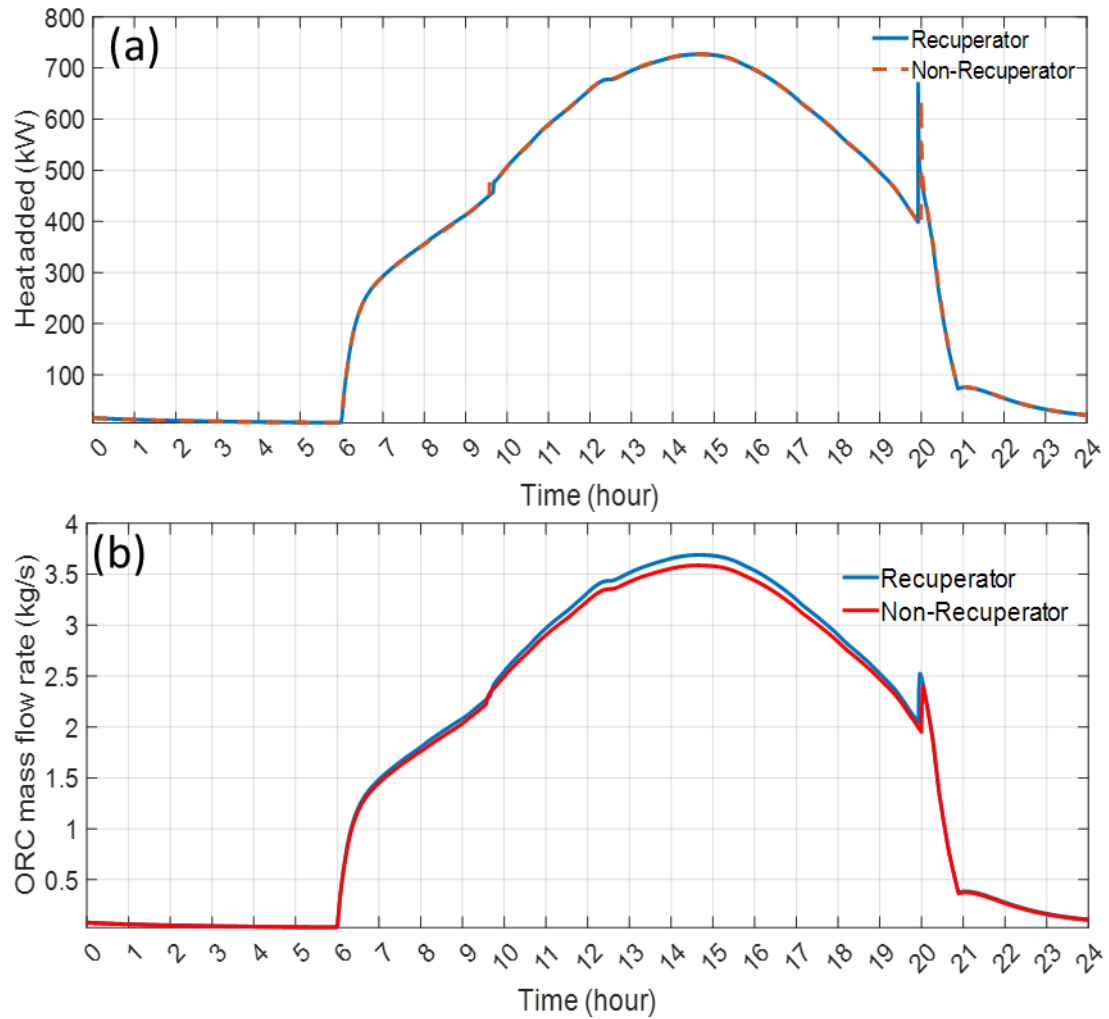


Figure 5.10: Daily variation a) Heat added to the ORC system by the heat exchanger and b) ORC mass flow rate for both the recuperative and non-recuperative ORC systems

Figure 5.11 depicts variations in the total vapor mass in the steam accumulator for both the recuperative and non-recuperative ORC systems. In this study, as mentioned above in Table 5.3, the total capacity of the steam accumulator is adopted as being 100m³. The regenerator does not affect the total steam accumulated value in the tank.

The main reason behind that because we adopted a water receiver after the heat exchanger and before the water pump in the DSG cycle, which works to accumulate the exhaust water of the hot side in the heat exchanger. The exit hot side water temperature in the heat exchangers for both the recuperative and non-recuperative ORC systems differ by only one or two degrees and become the same degree in the water receiver due to it effected by the ambient temperature.

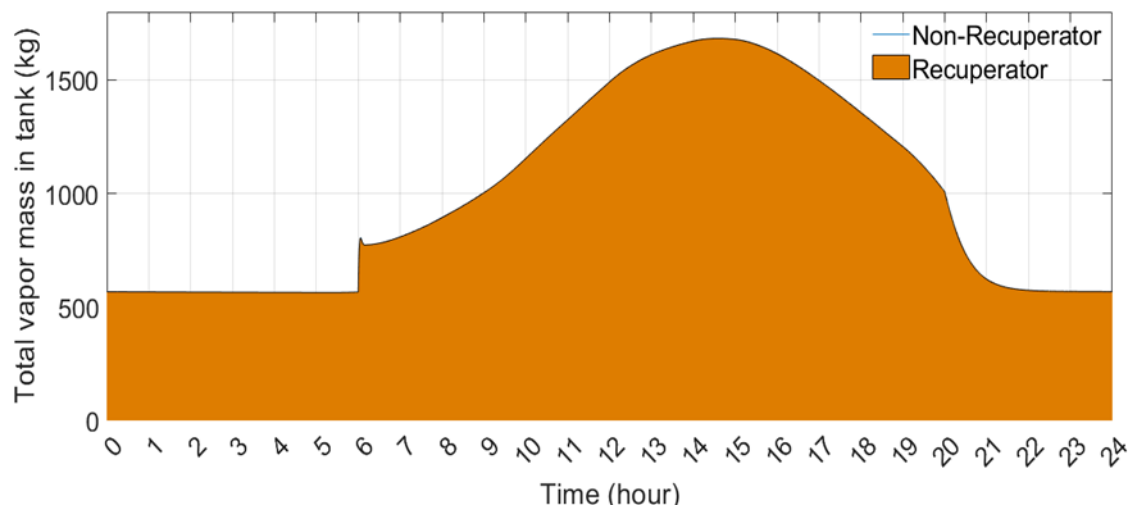


Figure 5.11: Daily variations in the total vapor mass in the steam accumulator for both the recuperative and non-recuperative ORC systems

5.5.2 Parametric study

As solar DSG is affected by several parameters, the solar-driven cascade Rankine cycle will be affected by certain parameters as well. In this subsection, we will examine the effect of two parameters on the performance of the proposed system. These parameters are the degree of the superheating temperature in the heat exchanger and condensation temperature parameters. The target of the parametric analysis is also to determine which of the parameters has the greatest effect on the performance of the proposed system. In each case, only the value of one parameter is changed within the

specified range while the remaining parameters remain constant and equal to the base values. This analysis was undertaken on a typical summer's day.

5.5.2.1 Effect of the superheating degree in the heat exchanger on the performance of the solar-driven cascade steam-organic Rankine system

The superheating degree in the heat exchanger is the key parameter that can effect on the performance of the solar-driven cascade steam-organic Rankine system. For this reason, we examined four different degrees of superheating in the heat exchanger. The findings of this study reveal that a minimum degree of superheating working fluid can achieve a higher power output. In Figure 5.12, it can see that 10°C of superheating generates up to 200kW, while 25°C degrees of superheating produces 180 kW during the peak time of solar irradiance. For the following analysis, we adopted the degree of superheating temperature in the heat exchanger to be 10°C, which is the best value that had an effect on the system performance.

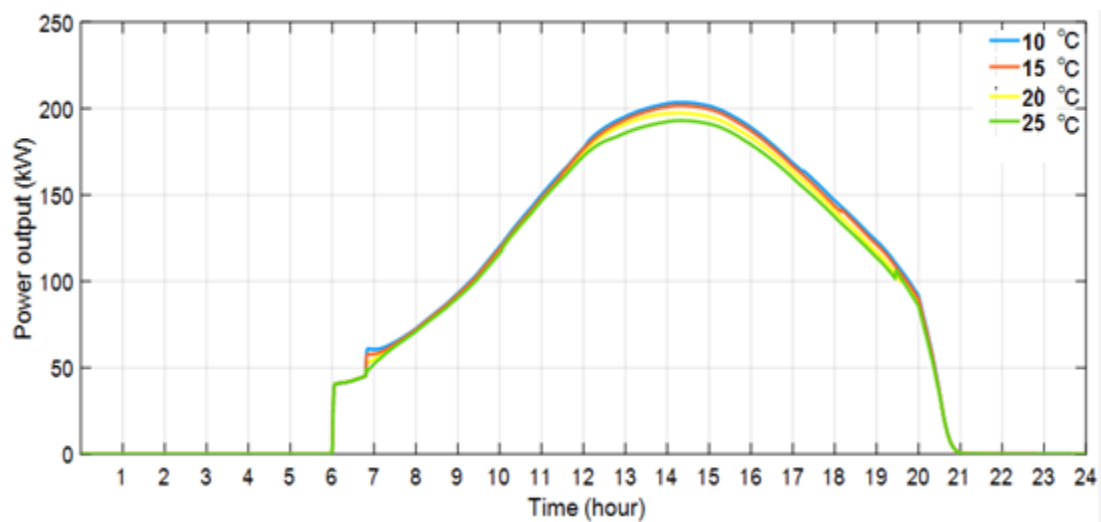


Figure 5.12: Effect of the superheating degree in the heat exchanger on the power output of the solar-driven cascade steam-organic Rankine system

Also, the thermal efficiency of the system is affected by the degree of superheating, as shown in Figure 5.13. For instance, when the degree of superheating was 10 °C, the thermal efficiency of the solar steam-organic Rankine cycle achieved the maximum

valve of 0.19 at the peak time of solar irradiance while, when the degree of superheating was 25 °C, the thermal efficiency of the solar steam-organic Rankine cycle achieved a maximum value 0.17 during the same period. The main reason for this is because the total power generation output by using 10°C was higher compared with the power output when 25°C was used.

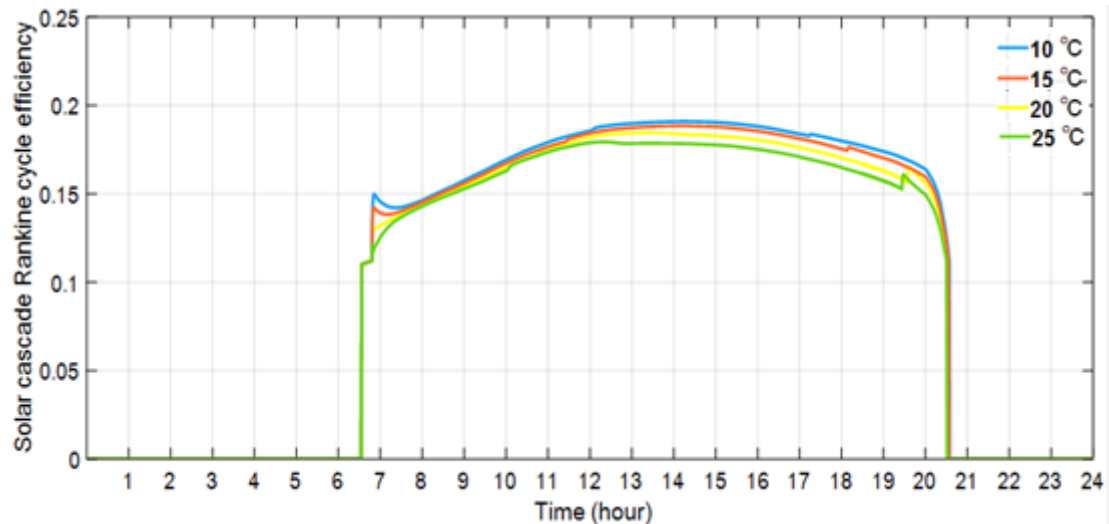


Figure 5.13: Effect of the superheating degree in the heat exchanger on the thermal efficiency of the solar-driven cascade steam-organic Rankine system

5.5.2.2 Effect of the condensation temperature on the cascade Rankine cycle

The effect of the condensation temperature on the solar steam-organic Rankine cycle is displayed in Figure 5.14. The condensation temperature has a significant effect on the proposed system. Four condensation temperature degrees were evaluated (20, 25, 30 and 35°C). The findings reveal that, when the condensation temperature decreases to 20 °C, the power generation increases to 210 kW. The solar steam-organic Rankine cycle efficiency increases when the condensation temperature decreases, as shown in Figure 5.15.

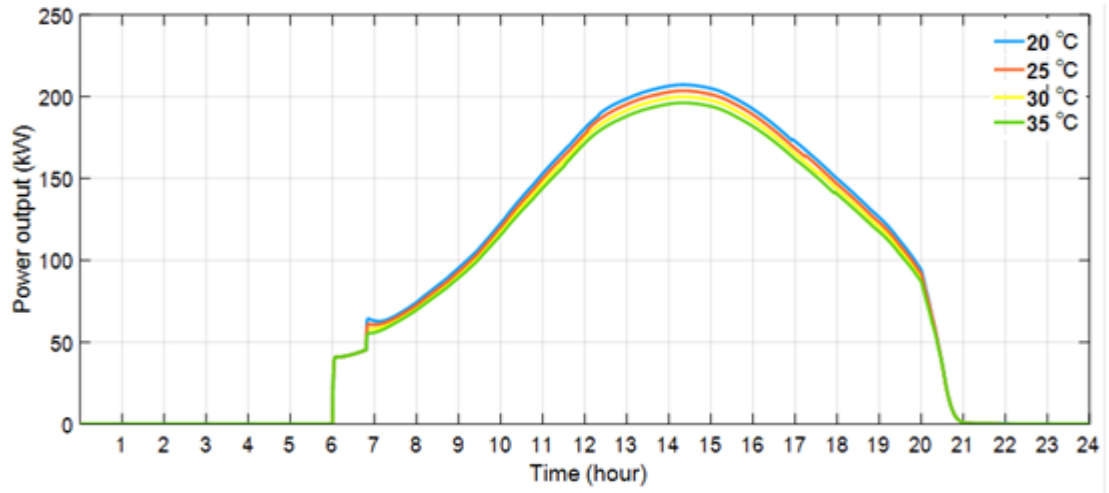


Figure 5.14: Effect of the condensation temperature on the power output of the solar-driven cascade steam-organic Rankine system

It is clear that, at the lowest condensation temperature, the highest efficiency is possible. The overall system efficiency by using R134a increases by 0.19 when the condensation temperature drops from 20°C at the peak time of solar irradiance while, when the condensation temperature is 35°C, the thermal efficiency of the system decreases to 0.17 at the same time as the peak solar irradiance occurs, as shown in Figure 5.15.

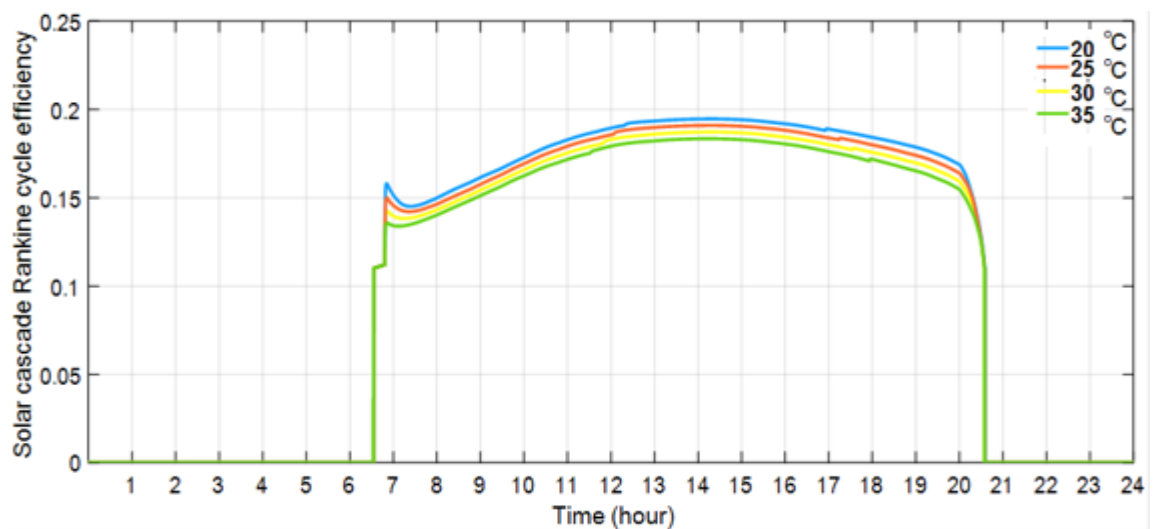


Figure 5.15: Effect of the condensation temperature on the thermal efficiency of the solar cascade steam-organic Rankine system

5.5.3 Effect of working fluids on the performance of the bottom ORC system

To investigate the effect of working fluids on system performance, several working fluids should be applied under the same boundary conditions. Four different working fluids in the bottom ORC system were investigated in this research. Table 5.2 provides a list of the most important parameters for the examined working fluids. The saturation pressure of 25 °C indicates the condensation pressure because the condensation temperature is considered to be 25 °C. The saturation temperature at 1 bar is another crucial characteristic since, if vacuum operation is not permitted in the system, 1 bar is the lowest possible condensation pressure. Table 5.2 demonstrates that, while other working fluids have significantly greater condensation temperatures, R134a has the lowest condensation temperature, at 1 bar. A high condensation temperature implies the low thermal efficiency of the ORC. Regarding the effectiveness of the system, the impact of ORC fluids is investigated. The Chena geothermal power plant, which has constructed two ORC plants with a combined capacity of up to 400 kW, uses R134a as a common working fluid [143]. It was chosen as the representative fluid as a result. Analyses of R600a, R407c, and R1234yf follow. R134a is a brand-new, promising fluid with negligible potential for ozone depletion and low potential for global warming. R134a is one of the highest rated fluids, according to research, because of its excellent efficiency and environmental friendliness [144]. Widespread research on R600a, R407c, and R1234yf confirms their great efficacy and good viability [145].

The findings reveal that the fluid with the higher critical temperature represents higher cascade Rankine cycle efficiency. For each fluid, the efficiency is closely related to the level of solar radiation. Figure 5.16 shows the variation of the total power output with various degrees of solar irradiance for the four different working fluids when the

condensing temperature was 25 °C. The highest total power output was obtained by increasing the solar irradiance. R600a has the highest effect on the proposed system with 210 kW, followed by R134a with 202 kW. Both R1234yf and R407c obtained an extremely similar level of power output. It should be noted that the fluids with a high boiling point had a greater effect on the total power output of the system.

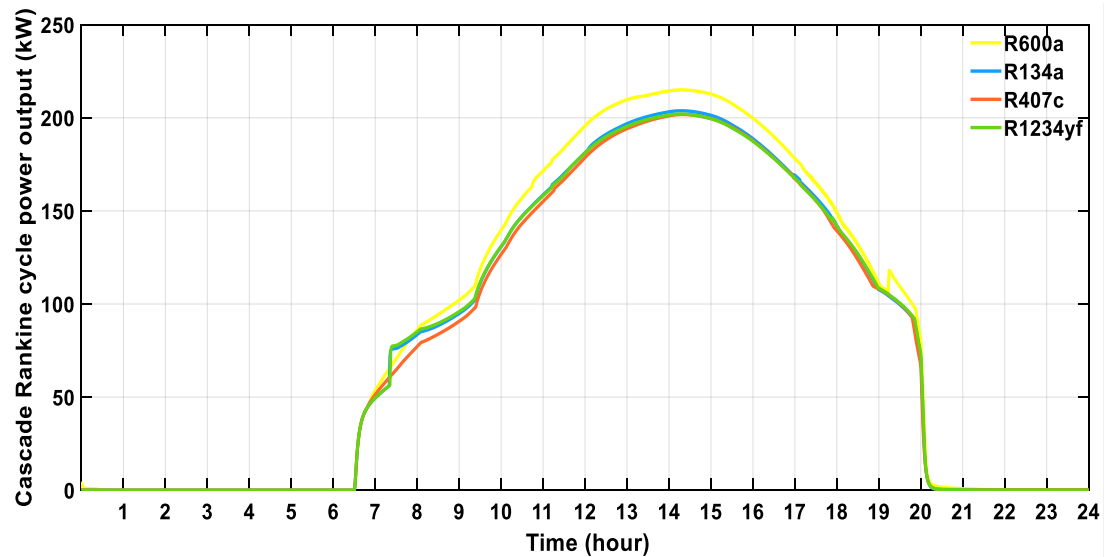


Figure 5.16: Comparison of total power outputs for the solar steam-organic Rankine system for four different working fluids

Figure 5.17 shows the variation in the solar-driven cascade Rankine cycle's efficiency under various levels of solar radiation for four different working fluids. The highest thermal efficiency of 0.20 was achieved using R600a, followed by R134a 0.18. However, it can be seen that R134a, R407c and R1234yf had extremely similar peaks of solar irradiance.

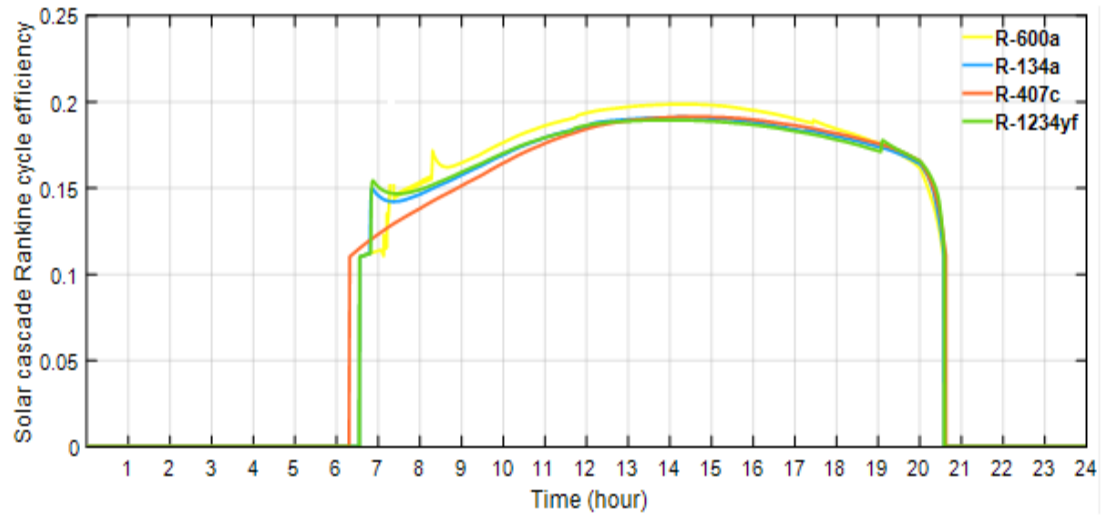


Figure 5.17: Variation in solar-driven cascade Rankine cycle efficiency under various levels of solar radiation for different working fluids

The variation in the ORC mass flow rate under various levels of solar radiation for the different working fluids are shown in Figure 5.18. The figure shows an increase in the ORC mass flow rate as the solar radiation increases. It should be noted that, for the same solar collector's area under identical boundary conditions, the ORC cycle with R1234yf and R407c requires a much higher mass flow rate compared to R134a and R600a. A jump lines occur when the control valve is opened and closed according to the set point.

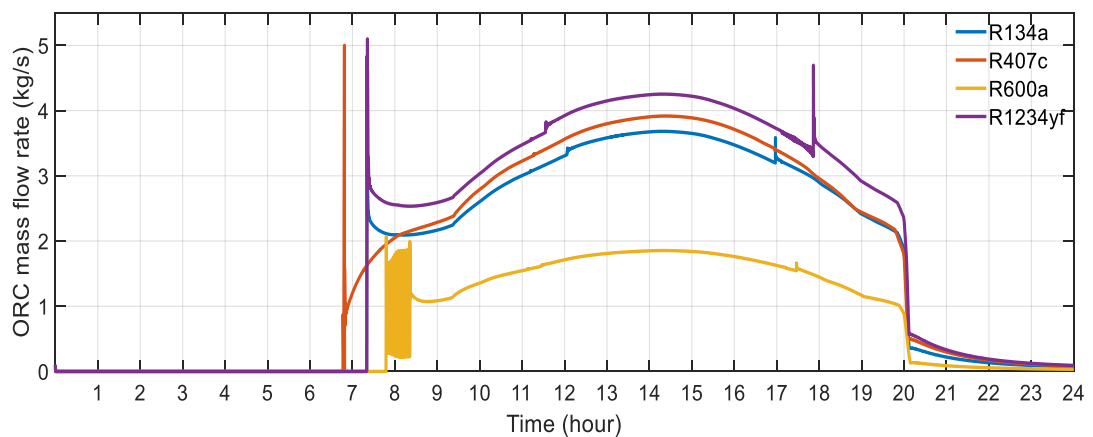


Figure 5.18: Variation in the ORC mass flow rate under various levels of solar radiation for different working fluids

Figure 5.19 shows the variation in the hot and cold sides temperatures in the heat exchanger under various solar radiation levels for different working fluids. The heat transfer fluids in the heat exchanger are related to the characteristics of the specific working fluid. The pinch point temperature difference in the heat exchanger is selected to be 10°C in this study.

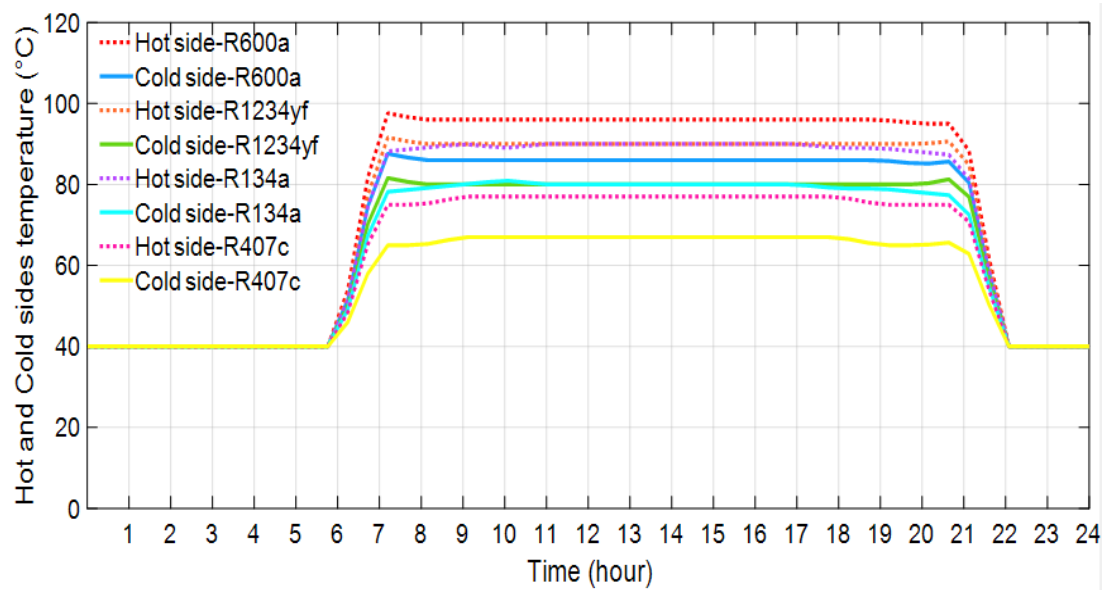


Figure 5.19: Variation between the hot and cold sides temperatures in the heat exchanger under various levels of solar radiation for different working fluids

According to the findings of this study, working fluids with higher critical temperatures have a tendency to have higher thermal efficiency, yet their condensation pressures can be lower than the ambient pressure. There is a trade-off between operational complexity and thermal efficiency, and vacuum condensation will make operation more difficult. For the ORC system, working fluids with higher critical pressure and higher saturation pressure for condensation temperatures are preferred. Computer-aided molecular design methods might identify more effective working fluids for the ORC, if this problem is taken into account.

5.5.4 Performance comparison between the steam and cascade steam-organic Rankine cycle systems

In order to compare a direct steam generation solar power system with a solar-driven cascade steam-organic Rankine cycle, some of parameters must possess the same value, such as the solar collectors' area, steam storage capacity of the accumulator, ambient temperature and solar irradiance. The solar irradiance fluctuates on some of the days, as shown in Figure 5.20.

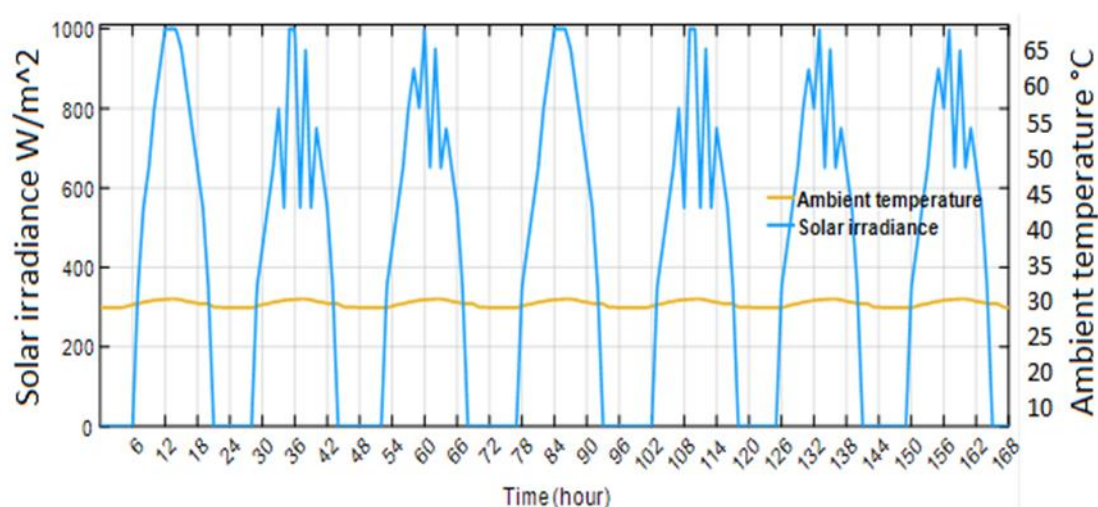


Figure 5.20: Weekly variations for solar irradiance and ambient temperature [6]

This comparison was based on a week with typical weather conditions. Figure 5.21 shows the weekly variation in the amount of solar heat collected by the solar field for the direct steam generation solar power system and cascade steam-organic Rankine cycle system. It can be seen that both systems collected the same value of heat, varying in line with the solar radiation level. The main reason for this is that the solar collector area is a constant and the boundary conditions for both systems are the same.

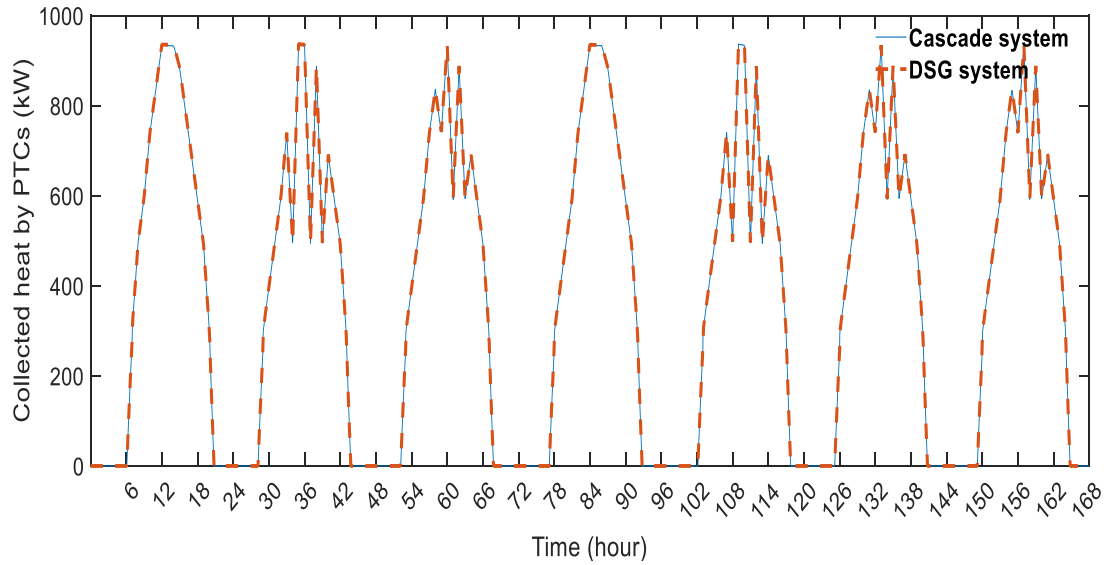


Figure 5.21: Comparison of collected heat by solar field for the solar steam-organic Rankine system and the direct steam generation solar power system during a week

Figure 5.22 shows the weekly variation in the power output for both systems. It is clear that the solar steam-organic Rankine cycle generates significantly more power compared with the direct steam generation solar power system. For example, on the first day of the simulation, the cascade Rankine cycle was able to generate up to 200 kW at the peak of radiation, while the single Rankine cycle generated only 175 kW during the same period. The extra power generated was 30 kW, and it was generated by the bottom ORC cycle. The results demonstrated that the exhausted steam of the steam turbine can generate extra power within the system. In the conventional Rankine cycle, the exhaust steam expansion process after the turbine was cooled by the condenser, which needs more power to reject a huge waste heat to the atmosphere to cooldown mixture liquid to supercooling liquid. This heat rejection is due to the conversion of the two-phase fluid into saturated liquid. However, in some systems, a recuperative system has been used to increase the power generation. This increase in the power output was very limited. However, in this study, we found that the heat rejected by the condenser can drive the ORC system smoothly, leading to the

generation of more power output compared with the conventional Rankine cycle, which uses a recuperative system.

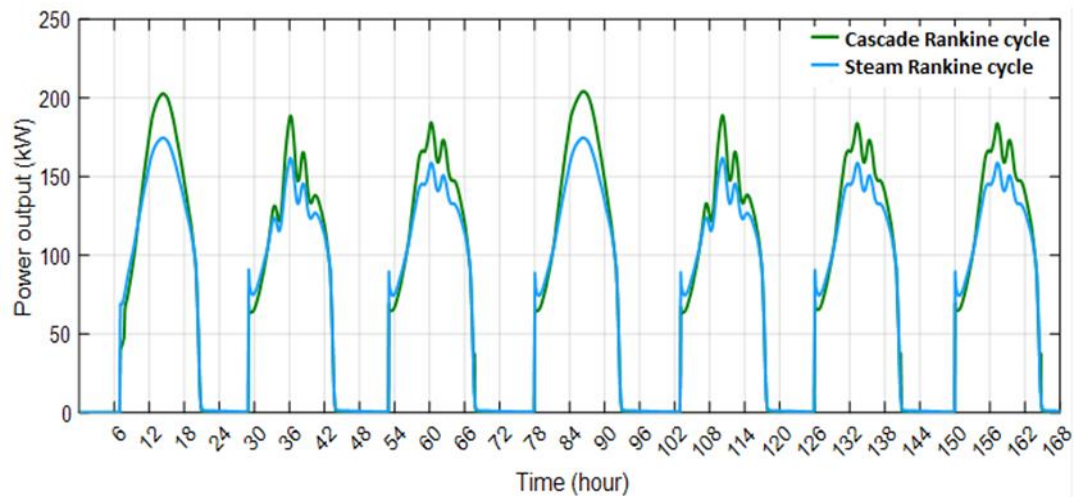


Figure 5.22: Comparison of power outputs for the solar steam-organic Rankine system and the direct steam generation solar power system during a week

The variations efficiencies of the solar steam-organic Rankine cycle and solar steam Rankine cycle are presented in Figure 5.23. Due to the higher power output in the solar steam-organic Rankine cycle, its thermal efficiency exceeds that of the solar steam Rankine cycle.

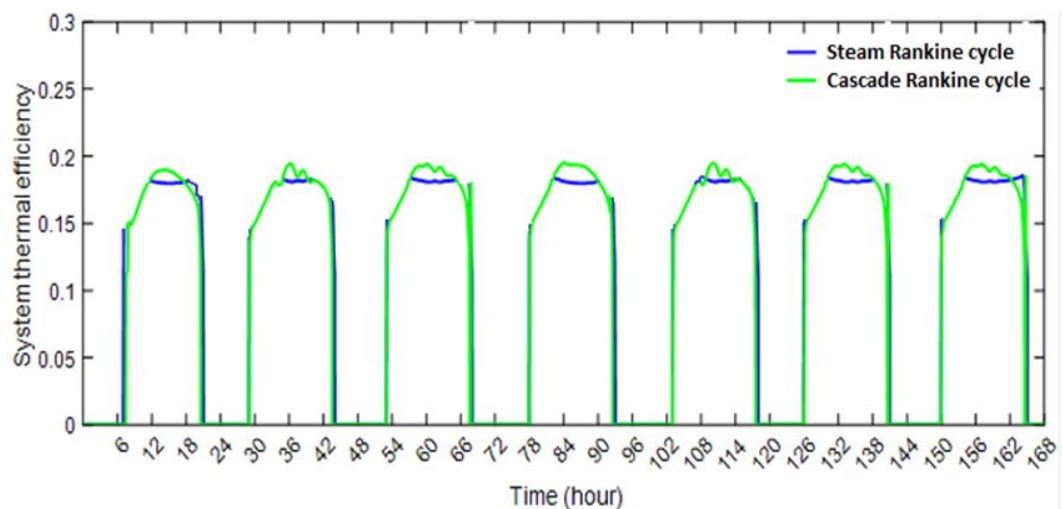


Figure 5.23: Comparison of solar thermal efficiencies for the solar steam-organic Rankine system and the direct steam generation solar power system during a week

5.6 Summary

In this chapter, an innovative solar-driven cascade steam-organic Rankine cycle power generation system, integrated with the steam storage tank, is proposed. The proposed system has been simulated and compared with and without using an internal heat exchanger (recuperator). The innovative system offers a significantly higher power output capacity than the conventional direct steam generation solar power system. The steam condensation temperature, which is the exhaust steam temperature from the steam turbine in this system, is a crucial parameter because it directly affects the performance and efficiency of the bottom ORC and during normal operation. This chapter developed a methodology for the design of the degree of superheating in the heat exchanger with respect to a new indicator, namely, the equivalent heat-to-power efficiency, which is a compromise between the efficiency of the different operation modes. The effects of different working fluids, condenser temperatures and solar irradiance levels was investigated as well. The results show the following:

- A week-long dynamic comparison of the performance of the Solar-DSG and Solar-cascade systems was conducted, and the results show that the solar-driven cascade Rankine cycle has both higher thermal efficiency and power output than the solar-DSG.
- The effect of working fluids on the performance of the bottom ORC system was investigated. The findings show that R600a has both highest thermal efficiency and power output followed by R-134a among three different working fluids.
- We examined the effect of two parameters on the performance of the proposed system. These parameters are the degree of the superheating temperature in the heat exchanger and condensation temperature

parameters. The findings demonstrated that, when the condensation temperature decreases, both the thermal efficiency and power output of the system increase. Also, the highest thermal efficiency and power output can be achieved by decreasing the degree of the superheating temperature in the heat exchanger.

- The effect of the recuperator on the performance of solar-driven cascade steam-organic Rankine cycle was analysed. The results show that the recuperator can improve both the thermal efficiency of the system and its total power output.

Chapter 6 Dynamic simulation of a cascade steam-organic Rankine cycle solar power system integrated with two water accumulators

6.1 Introduction

This chapter presents and includes modelling work of a new solar-driven cascade steam-organic Rankine cycle integrated with two water accumulators in order to extend the power generation at nights. Furthermore, the goal of this work is to provide a revolutionary method for the extended, affordable storage of DSG systems that permits a good level of thermal efficiency and power output, as well as a substantially larger water temperature decrease. Two-stage accumulators and steam-organic Rankine cycles are set up in a design that allows for two-step heat discharge to achieve this purpose. This heat release is made feasible by the steam Rankine cycle and ORC system, which are coupled with two storage tanks, and it is a unique method.

The system has certain definite benefits over currently used systems, such as the following:

- The power conversion is adaptable, and the two-step heat discharge is distinctive. Either the steam Rankine cycle or the organic Rankine cycle can generate electricity. It is important to note the benefits of resilient power generation, especially given the fluctuating nature of consumer demand for electricity which is often higher during the day and lower at night.
- In the case of two-stage tanks, the storage capacity can be significantly expanded and the water in the hot water tank can be moved to the lower water tank with a temperature differential of more than 140 °C.

- The control strategy of the proposed system is very easy, and the superheated section is avoided in the solar field as it is challenge part in direct steam generation solar power plants.

6.2 Configuration of the innovative system

A schematic diagram of the innovative system is shown in Figure 6.1. The system consists of solar parabolic trough collectors, two accumulators (high and low water temperature), three heat exchangers and two Rankine cycles (high and low temperature). Steam is directly generated in the parabolic trough collectors. The top cycle is the steam Rankine cycle, which mainly comprises a solar field, hot water temperature accumulator, steam turbine, intermediate heat exchanger, control valves, water pump and low water temperature accumulator. The bottom cycle is the organic Rankine cycle, consisting of an ORC turbine, recuperator, condenser and ORC pump. Due to the independent power output from the two Rankine cycles and two integrated accumulators, this innovative system can operate in numerous modes during a day.

Mode (I): A water pump (DSG pump) in the top steam Rankine cycle works, while the other valves and ORC pump are closed and switched off. The water normally accumulates in the low-temperature water accumulator before the system charges and generates electricity. The DSG pump draws accumulated water from the low-temperature accumulator, pressurizes it and moves it to the solar collectors to be heated. In the early morning, saturated liquid passes through the solar collectors then arrives at and fills the high-temperature water accumulator. This stream may consist of some vapor, but the majority is saturated liquid. Later, when the solar irradiance is abundant, the mass flow rate can be controlled to reach the desired temperature by adjusting the water pump. The purpose of this control is to reach the desired temperature and pressure before steam enters the turbine.

Model (II): Solar heat collection and power generation. In this mode, the system works when solar irradiance is abundant, and the desired temperature and pressure are achieved. Valve1, valve2, valve3 and the control pressure valves are open, while valve4 is closed. The water pump and ORC pump are in run mode. The solar collectors heat and partially evaporate the water. Saturated steam enters the topmost turbine and generates electricity as it expands. In a heat exchanger, the exhaust condenses into a saturated liquid and is compressed before being returned by a water pump. The working fluid in the ORC is evaporated using condensation heat. Water from the low-temperature accumulator is transferred to the high-temperature accumulator and heated using solar collectors in the meantime. The mass in the low-temperature accumulator gradually diminishes. Under ideal circumstances, the state of the water when it leaves the solar field is binary. However, as the sun's radiation changes, it can take the form of either a liquid or a vapour. To provide a constant temperature in the high-temperature accumulator and steady power conversion in the steam Rankine cycle and ORC system, the flow rate through the water pump can be adjusted depending on the radiation, so the total generation power output comes from the steam turbine and ORC turbine at the same time in this mode. Subcritical cycles for the SRC and ORC are taken into account for the proposed system.

Model (III) Heat discharging mode. There are two modes in this case. First, the solar radiation is unavailable, the heat stored in the hot temperature accumulator will release and flows into the top cycle. Valves 1, 2, 3 are open and ORC pump runs. The stored water in the hot temperature accumulator is vaporized and drive both the steam turbine and ORC system. The temperature and pressure in the hot accumulator will gradually drops. Second, if the pressure lower than 0.9MPa, the top cycle will be off work. Valve 1, 2, 3 are close, while valve 4 is open and ORC pump runs. The hot water in the hot

temperature accumulator releases into the low temperature accumulator and passing through the heat exchanger. The heat exchanger absorbs the heat energy and drive the ORC only. In the second step of the heat discharge, the total electricity production is from the ORC.

Figure 6.2 illustrates the thermodynamic processes represented by the T-s diagram when, for instance, R-134a is used as the ORC fluid. The steam Rankine and ORC are represented by the blue and red lines, respectively. The numbers represent the water and organic fluid thermodynamic states that correspond to the markings in Figure 6.1. Additionally, the heat exchangers' and pipes' thermal and frictional losses are ignored. The simulation ignores changes in kinetic and potential energy.

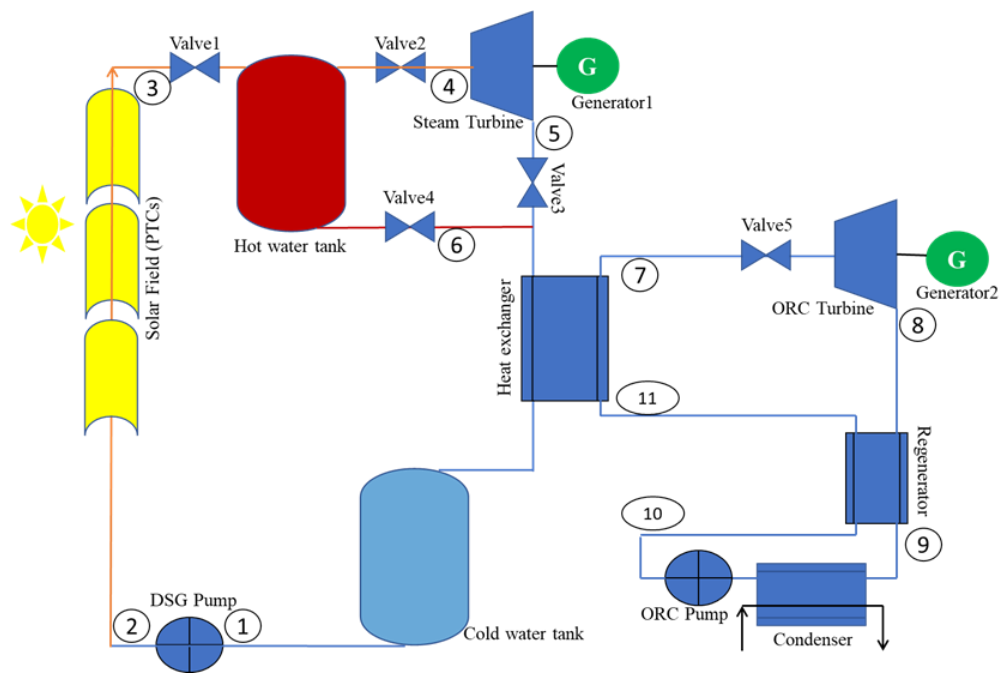


Figure 6.1: Layout configuration of the innovative system

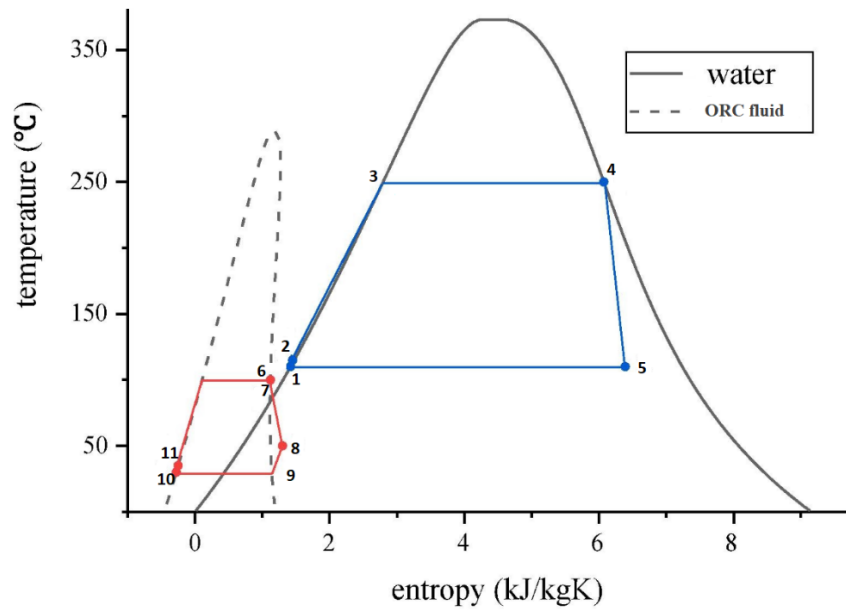


Figure 6.2: T-s diagram of the solar cascade steam-organic Rankine cycle system and two-stage tanks

6.3 Control strategy for the proposed system

A brief description of control strategy for the solar cascade steam-organic Rankine cycle combined with storage tanks in the nominal operation mode or in discharging operation mode can be summarized in this subsection. This control strategy is easier compared the other control strategies that applied in solar concentrating thermal power plant and this control strategy is based on the valves are used in the proposed system and pumps.

- ❖ Valves 1, 2 and 3 are opened in the event of abundant solar radiation and it is equal or $> 350 \text{ W/m}^2$. The water pump is working to fill up the hot water tank according to desired temperature which is 250°C .
- ❖ Valves 2 and 3 are still open if the radiation is lower than 350 W/m^2 . While valve 1 is closed to stop the hot steam energy stored in the hot water tank to goes to the solar collectors.

- ❖ If the inlet steam turbine pressure is lower than 0.9MPa in discharging mode, valves 1, 2 and 3 are closed, and valve 4 is open to let hot water energy stored in the hot water tank drains to the cold-water tank.

6.4 Modelling of the proposed system

The modelling and parameters of parabolic trough collectors, steam and ORC turbines, heat exchangers, pumps and the same method of calculation in chapters 3, 4 and 5 are used in this chapter. However, the volume of hot and cold-water accumulators and the thermal efficiency of proposed system are provided below:

6.4.1 Modelling of the water accumulator

Different from Chapter 5, in this novel system, water accumulator is to be used as heat storage unit rather than the steam accumulator. The total liquid volume for each accumulator can be calculated by Equation 6.1

$$V_{tank} = V_{water} = \frac{3600t_H \cdot \dot{W}_{net}}{\rho_{water} \cdot \eta_{SRC} \cdot Cp_{water} \cdot \Delta T} \quad (6.1)$$

Where, V_{water} is the water volume in the tank. t_H represents heat storage time in hour. ρ_{water} is the density of water, η_{SRC} indicate the solar Rankine thermal efficiency, Cp_{water} is the heat capacity of water and ΔT is the temperature drop during heat discharging mode.

The total liquid volume of the cold-water accumulator is equal to the total liquid volume of the hot-water tank in this proposed system.

6.4.2 Thermal efficiency of the proposed system

In this novel system, there are two thermal efficiencies period, one under nominal mode conditions and another one during heat discharging mode. The thermal efficiency of direct steam generation cycle is determined by Equation 6.2:

$$\eta_{DSG} = \frac{\dot{W}_{net,DSG}}{\dot{Q}_{solar}} = \frac{\dot{m}_{steam}(h_4 - h_5) - \dot{m}_{steam}(h_2 - h_1)}{\dot{m}_{steam}(h_4 - h_2)} \quad (6.2)$$

Where, $\dot{W}_{net,DSG}$ is the net power output from the direct steam generation solar power system and $\dot{Q}_{solar}$ is the heat energy collected by solar field. The thermal efficiency of ORC cycle under nominal conditions can be calculated by Equation 6.3:

$$\eta_{ORC} = \frac{\dot{W}_{net,ORC}}{\dot{Q}_{Heat,useful}} = \frac{\dot{m}_{ORC}(h_7 - h_8) - \dot{m}_{ORC}(h_{10} - h_9)}{\dot{m}_{ORC}(h_7 - h_{11})} \quad (6.3)$$

Where here, $\dot{W}_{net,ORC}$ is the net power output of ORC cycle, and $\dot{Q}_{Heat,useful}$ is the total heat added to the ORC by the heat exchanger.

The total thermal efficiency of the system in this normal operating mode can be calculated by Equation 6.4:

$$\eta_{sys} = \frac{\dot{W}_{net}}{\dot{Q}_{solar}} = \frac{\dot{W}_{net,DSG} + \dot{W}_{net,ORC}}{\dot{m}_{SDG}(h_3 - h_2)} \quad (6.4)$$

In discharging mode, only the bottom ORC system works. The ORC is powered by the energy stored in the hot water accumulator. So, the ORC power output in this discharging mode is calculated by Equation 6.5:

$$\dot{W}_{ORC,d} = \dot{m}_{ORC}(h_7 - h_{8s}) \quad (6.5)$$

The thermal efficiency of the system during discharging mode can be determined by Equation 6.6:

$$\eta_{sys,d} = \frac{W_{ORCnet,d}}{\dot{Q}_{added,d}} \quad (6.6)$$

Where here, $\dot{Q}_{added,d}$ is the total heat added to ORC system during discharging mode and it can be expressed by Equation 6.7:

$$\dot{Q}_{added,d} = M_{water} \cdot (h_6 - h_1) \quad (6.7)$$

And the ORC operating time in this discharging mode can be determined by Equation 6.8:

$$t_{ORC} = \frac{M_{water}}{\dot{W}_{ORCnet,d}} \quad (6.8)$$

where M_{water} is the hot water weight transferred from the hot-water tank to the cold-water tank in discharging mode.

Figure 6.3 shows the flow chart for the thermodynamic analysis of the solar cascade steam-organic Rankine cycle with two tanks. The parameters of two Rankine cycles are the same parameters that used in Chapter 5, the only different is parameters for two storage tanks because in this chapter the storage tank is used to accumulates the hot water not the steam.

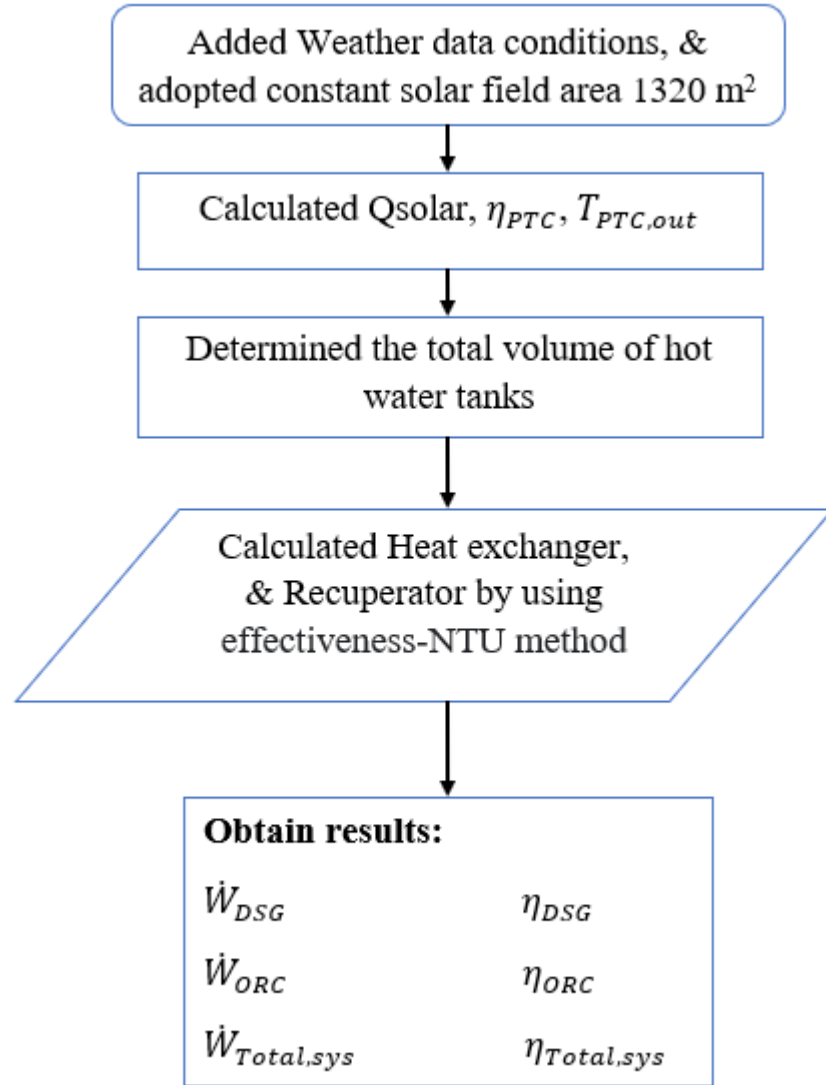


Figure 6.3: The flow chart for the thermodynamic analysis of the solar cascade steam-organic Rankine cycle with two tanks

6.5 Simulation results and discussions

The solar cascade steam-organic Rankine cycle intergraded with two-stage water tank powered by parabolic trough collectors is comprehensively built and simulated in this chapter of the thesis. It is expected to be applicable to direct steam generation solar systems in the future. However, we considered parabolic trough collector type only in this analysis. The fluid chosen for the ORC is R-134a. This fluid has undergone extensive research and as described in Chapter 5; it has been used in real ORC

systems. The critical temperatures of R-134a 101 °C. In order to provide a steady temperature and pressure during the evaporation process, subcritical and supercritical ORC are taken into consideration. The specific design parameters of the proposed system are listed in Table 6.1.

Table 6.1: The specific design parameters of the proposed system

Parameter	Value
Solar collectors' area	1320m ²
Steam flow rate	0.33 kg/s
Length of heat exchanger	350m
Inner and outer diameter of the pipe in the heat exchanger	0.01/0.03m
Hot water mass flow rate in discharging mode	1 kg/s
Steam turbine Isentropic Efficiency	0.85
ORC turbine Isentropic Efficiency	0.85
Mechanical generators Efficiency	0.95
ORC mass flow rate	3 kg/s
Condenser temperature	25°C
Pinch point temperature difference	10°C
Volume of the hot-water tank	50m ³
Volume of the cold-water tank	50m ³

Figure 6.4 below shows the proposed system model in the Simulink/Simscape software. The red block represents hot water tank while, blue block is cold water tank.

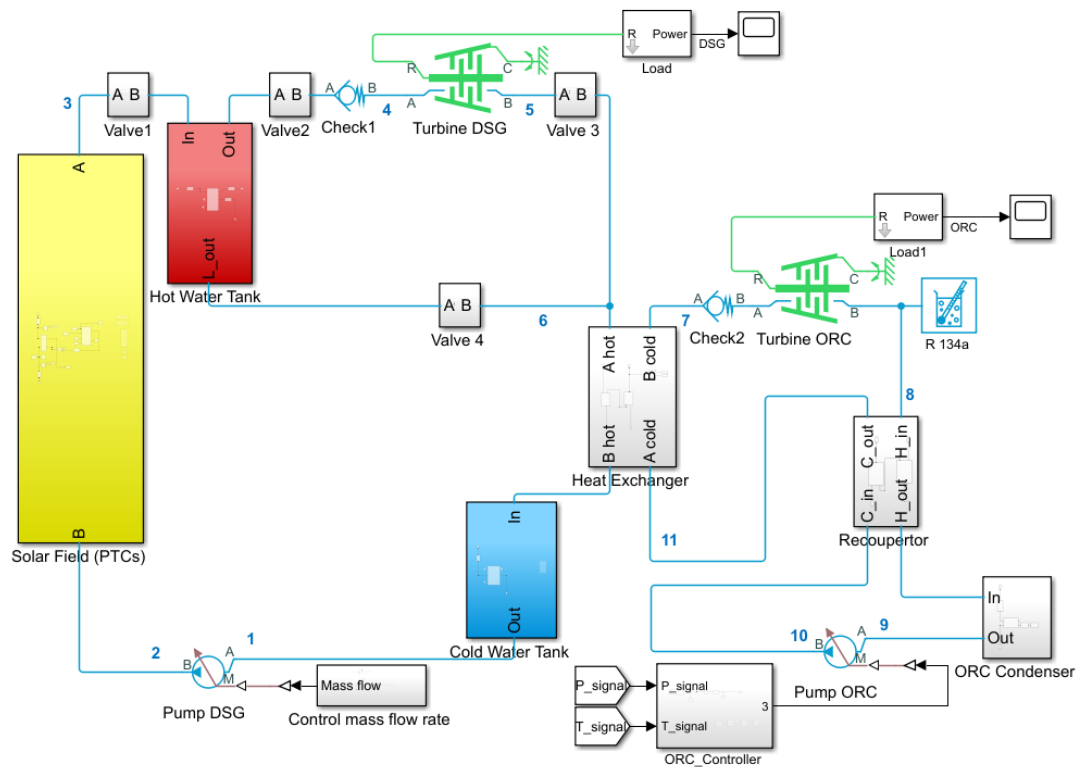


Figure 6.4: Layout of the solar cascade steam-organic Rankine cycle system and two-stage tanks in Simscape

A quasi-steady model was created using Simscape software to examine the dynamic behaviour of the solar cascade steam-organic Rankine cycle with two tanks system over the course of a day. The dynamic simulation is a helpful tool to comprehend the variations in the performance of the two-tank solar cascade steam-organic Rankine cycle system as well as the energy management procedures throughout the day's operation. The time step in this model's numerical modelling process is set to 5 seconds. To assess the system's dynamic performance, three different weather conditions from Libya that represented average summer days were chosen. The proposed system was simulated on typical summer days in Libya. Before the daily following results give, the evaluate the dynamic response of system from initial condition to steady state based on constant solar irradiance is given in order to see dynamic response of the system by using Simscape as shown in Figure 6.5.

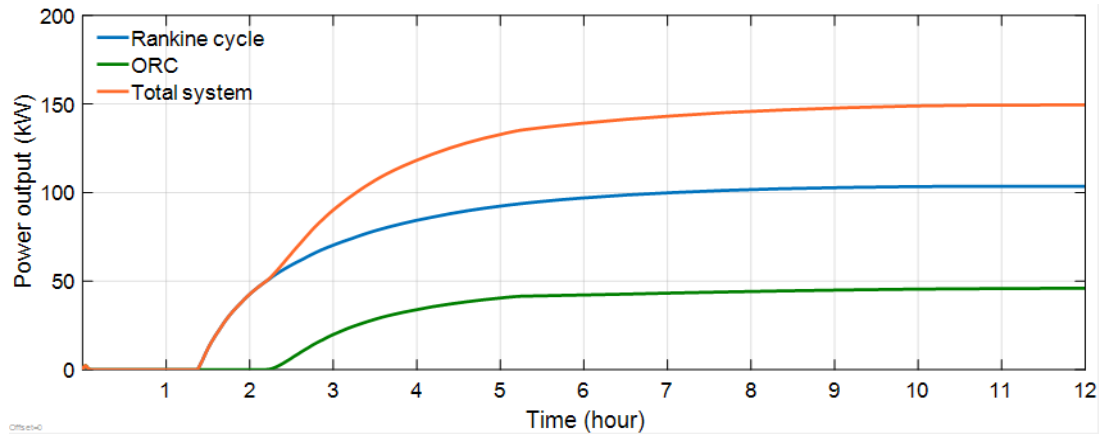


Figure 6.5: Dynamic response of the power output

Since this system is fully functional based on the level of solar radiation during a day, the performance of the system will depend on the climatic conditions. Figure 6.6 depicts weather data on three typical summer days in Libya.

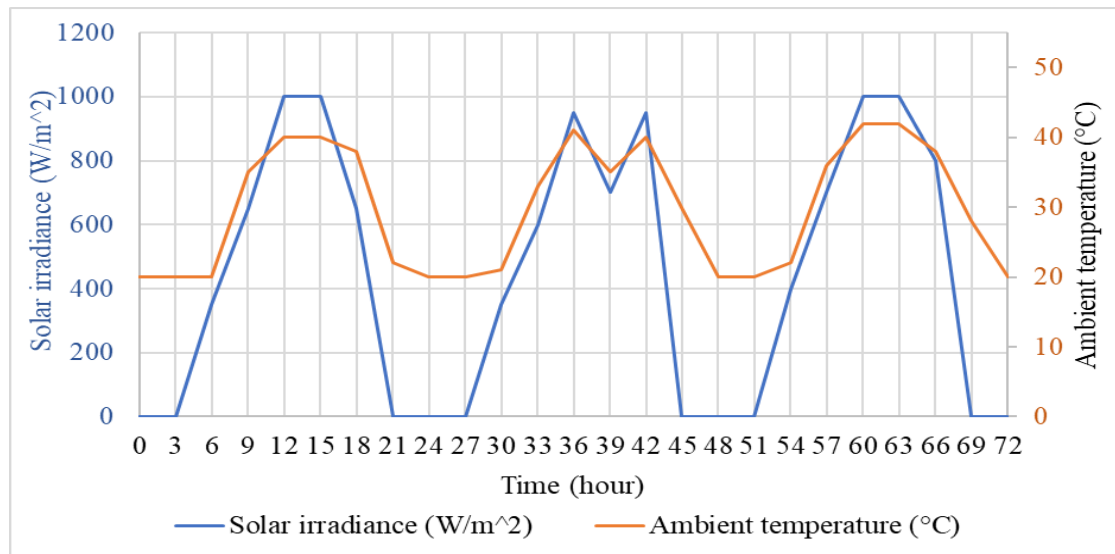


Figure 6.6: The weather data condition on three typical summer days in Libya [6]

Figure 6.7 depicts the variations of different temperatures in the proposed system over a course of three summer day in Libya. It can be seen that the outlet temperature of the solar field is equal to the temperature of the hot water tank which is represented

by the purple line. The outlet temperatures of the solar field and hot water tank increases with the increase of the radiation and its maximum value reaches to 250 °C in the peak of irradiance and it decreases when the radiation decreases. In discharge mode, the temperature of the hot water tank begins to decrease during the evening as shown in the first day of simulation from 21:00pm. The orange line represents inlet steam turbine temperature in topping cycle. When radiation is unavailable valve 2 closed, and steam turbine is off work.

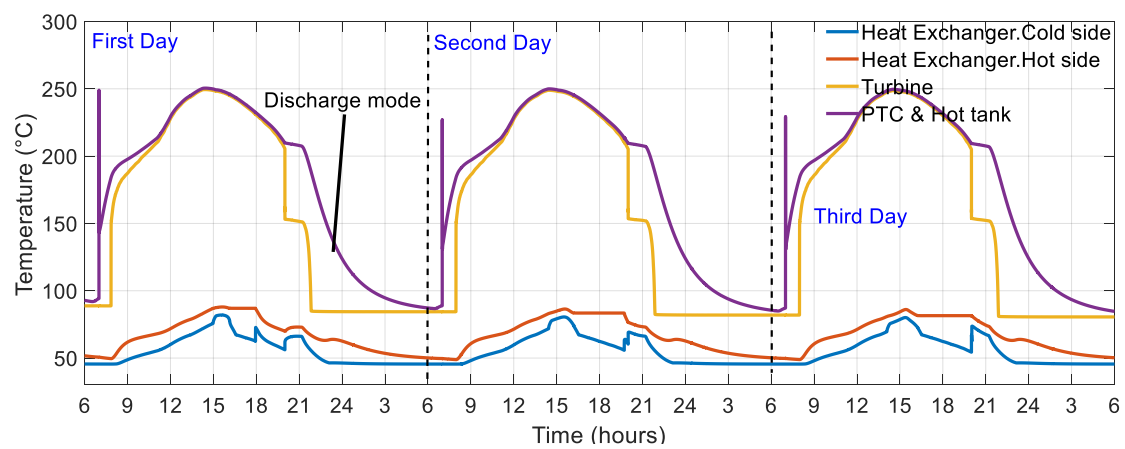


Figure 6.7: Variations of the outlet steam turbine temperature, cold and hot side temperature in the heat exchanger and the hot water tank temperature

Also, the figure shows variations of the outlet steam turbine temperature which is red line, and its maximum value is 80 °C at the peak time. The blue line indicates the cold temperature side in the heat exchanger which is equal to the inlet ORC turbine temperature. Figure 6.8 presents variations of the inlet and outlet steam quality in the DSG turbine and inlet fluid ORC turbine quality. The inlet steam turbine quality is effected directly by the radiation, and after the hot water tank fully charged, the inlet steam turbine quality is 1 during the daytime and this value decreasing by decreasing solar radiation and temperature in the hot water tank. The outlet steam turbine quality is 0.80 in the daytime. The orange line represents quality of water in the cold-water

tank which is almost 0. The inlet fluid ORC turbine quality is constant to 1 during charging and discharging mode.

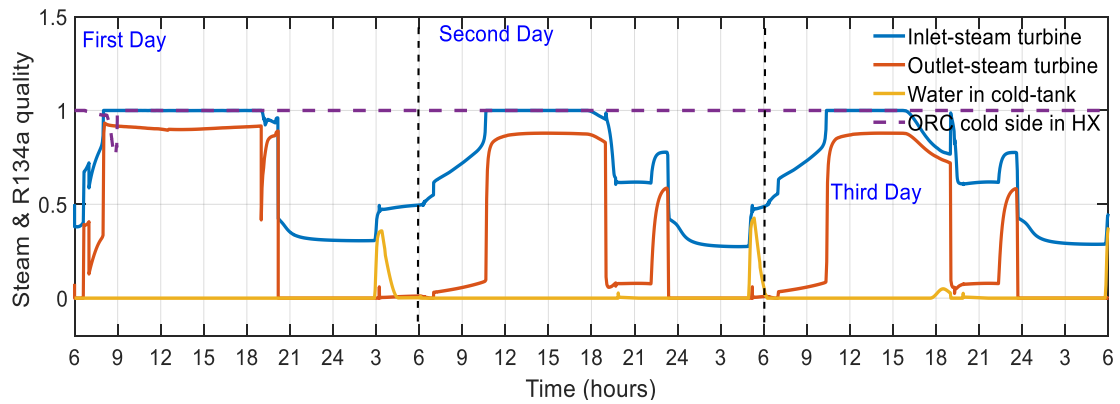


Figure 6.8: Variations of the quality for inlet and outlet steam turbine, inlet ORC turbine and the cold-water tank

Figure 6.9 depicts the variations of the pressure in the hot water tank, inlet steam turbine pressure, outlet steam turbine pressure and inlet ORC turbine pressure. As the radiation and the temperature in the hot water tank increases, the pressure in the hot water tank increases as well. The maximum pressure in the hot water tank is 3.5 MPa in the peak of the daytime. The pressure in the hot tank and inlet steam turbine pressure are extremely same from 9:00 am to 9:00 pm. The outlet steam turbine pressure is 0.6 MPa. From the same figure, it can be noted that the pressure in the hot water tank slightly decreases from 1.8 MPa in 20:30 pm to 1.7 MPa in 21:30 pm and after that it continues to decrease in the discharging mode. The inlet ORC turbine pressure which is represents by orang line is almost 1.6 in both charging and discharging modes. This is because the bottom ORC system is completely works based on the waste heat recovery in the solar steam Rankine cycle is this proposed system.

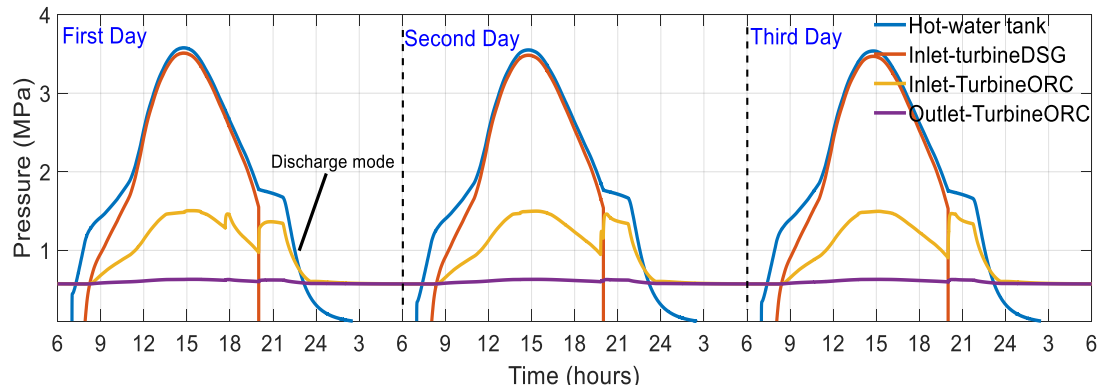


Figure 6.9: Variations of the pressure in the hot water tank, inlet steam turbine pressure, outlet steam turbine pressure and inlet ORC turbine pressure

The important thing in this system is the power generation outputs. Figure 6.10 shows variations of power outputs of the proposed system. There are two power outputs, one from steam Rankine cycle and another from the ORC system. The top steam Rankine cycle operates when the solar radiation is available, while the bottom ORC system works in both daytime and even to 03:00 am. In the normal operation when the radiation is available, both power outputs rise with increased radiation. The maximum steam Rankine cycle power is 136 kW in the peak time of the radiation, while the maximum ORC power output is 37 kW at the same time, so the total power output in this mode is 173 kW. It is clear to see that the steam Rankine cycle power output is higher than the ORC power output. In the discharging mode when the solar irradiance is not available, the power output of the proposed system is based on the steam Rankine cycle and the ORC system. The steam is still in the hot water tank and circulated in the Rankine cycle as part of the first step of heat discharge. The hot water tank experiences a similar temperature and pressure reduction as a standard one-stage tank while maintaining a nearly constant overall mass. The traditional nature of the first step's heat discharge means that it does not significantly increase the benefits of the suggested approach's power generation. Due to the following factors, it is expected

that the second step heat discharge will start at a hot water tank temperature of 230 °C. The water saturation pressure decreases from about 3.5 to 1.8 MPa as the hot water tank temperature decreases from 250 °C to 230 °C. When this happens, the mass flow rate will drop to roughly 60% of the rated value. The sum of the heat transfer in the hot water tank and the cycle efficiency, given the volume of the tanks, determines the total power that can be generated in the second step heat discharge. In the second step of heat discharging mode, the power output from ORC is 31 kW and it is seemed constant value. Moreover, the ORC cycle continues to generate electricity for up to six hours. The total power output of the ORC in discharging mode, it is completely dependent on the total volume of hot water tank and mass flow rate trough valve 4.

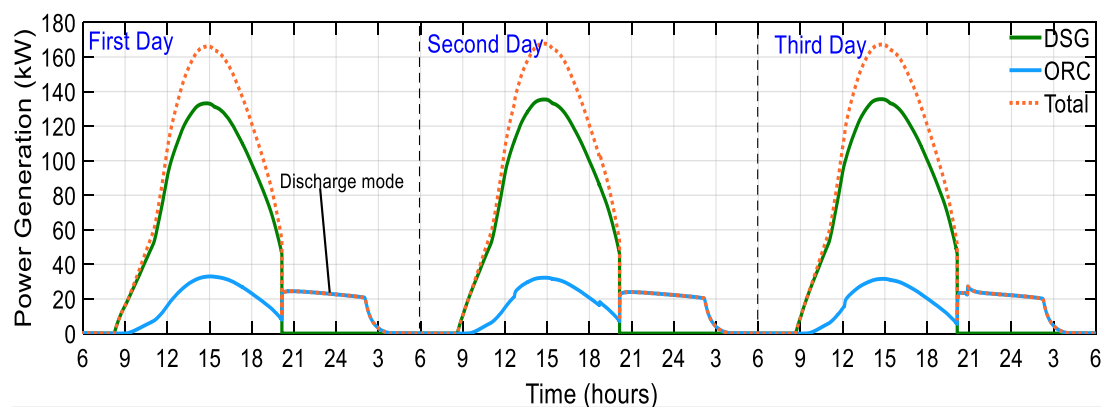


Figure 6.10: Variations of the power outputs in the system for three days

Figure 6.11 shows variations of the thermal efficiency for parabolic trough collectors, steam Rankine cycle efficiency, ORC efficiency and cascade steam-organic Rankine cycle efficiency for three days. All efficiencies of the system are increase and decline by rise and reduce the solar radiation. In the peak time of the radiation, the solar collector efficiency has achieved as 0.73. While the top steam Rankine cycle efficiency and the bottom ORC efficiency is 0.12 and 0.05 respectively. So, the thermal efficiency of cascade steam-organic Rankine cycle is 0.17 in the nominal

operation mode. The thermal efficiency of ORC in discharging mode is 0.05, and it seems to be constant value during the time of this mode.

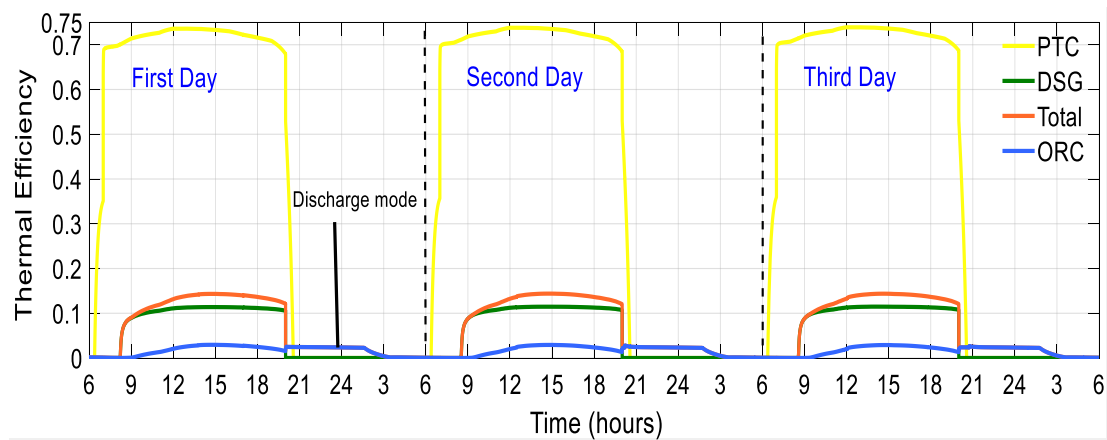


Figure 6.11: Variations of the thermal efficiency for parabolic trough collectors, steam Rankine cycle efficiency, ORC efficiency and cascade steam-organic Rankine cycle efficiency for three days

Figure 6.12 presents variations of heat absorbed by solar field and heat added to the ORC system by the heat exchanger for three days. The heat collected by solar field is based on the solar radiation level during a day. There is no auxiliary heater in this system. So, by given design conditions of the system, the total amount of the collected heat by solar field is 990 kW at the peak time of the radiation. The heat added to the ORC system varies between 650 kW to 540 kW. These values are quite low compared to amount of solar absorbed heat. This heat value is completely based on the exhausted steam outlet of the steam turbine via the internal heat exchanger in nominal operation mode. In discharging mode, the situation is different. The valve 3 is closed and the valve 4 open. The heat added by the heat exchanger comes from the hot water tank and this value varies between 550 kW to 100 kW at the end time of discharging mode as shown in Figure 6.12 below. The supplier heat useful for the ORC system can be adjusted according to the demand energy by controlled mass flow rate of the hot water tank.

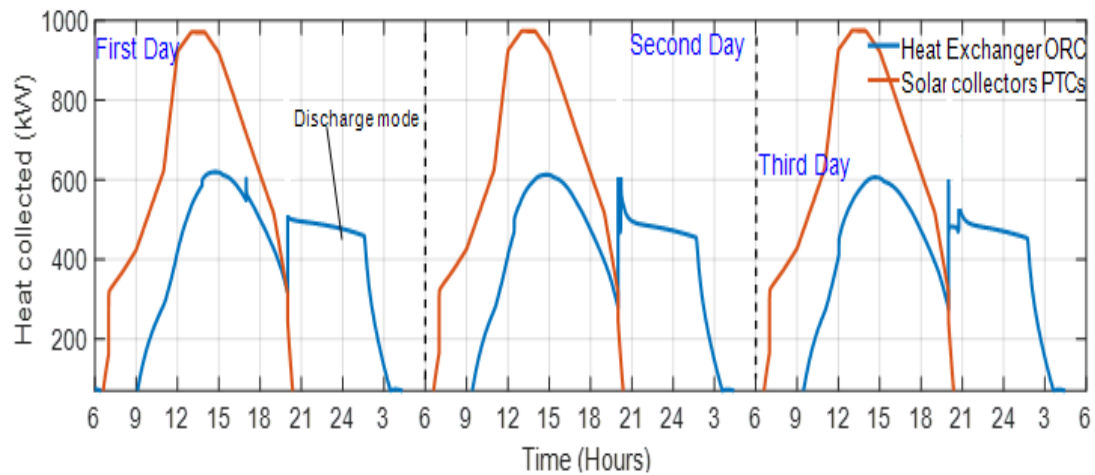


Figure 6.12: Variations of heat absorbed by solar field and heat added to the ORC system by the heat exchanger for three days

In the proposed system, mass flow rate throughout the system is playing an important role especially during discharging mode or early operation mode. There are three different mass flow rates in this system as shown in Figure 6.13. The first one for steam Rankine cycle which is controlled by the water pump. In this cycle, the maximum flow rate is 0.4 kg/s at the peak time of the radiation. the second one is ORC mass flow rate in nominal operation mode, and maximum mass flow rate is 3 kg/s which is higher than steam flow rate. The main reason behind that because the ORC system is forced to absorb the exhaust heat from the steam cycle and the ORC fluid has low latent heat capacity. The final flow rate is the hot water flow rate from the hot water tank to the cold-water tank passing through the heat exchanger during discharging mode. In this mode, hot water flow rate is 0.9 kg/s, while the ORC mass flow rate in this mode is 2.3 kg/s as shown in Figure 6.13.

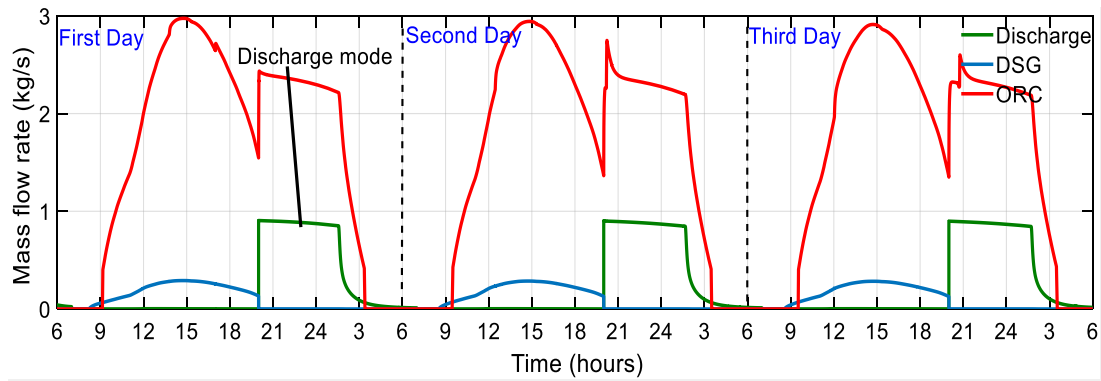


Figure 6.13: Variations of mass flow rate in the system for three days

There are four gate valves in the proposed system. These valves are used to control the operation of the system during charging and discharging modes. Figure 6.14 shows the dynamic response for each valve in the system during operation modes. It can be seen that valves 1,2, 3 are open and close at the same time during operation modes, while they are open when the solar radiation is available from 7:00 am to 20:30 pm. In the same figure we can see valve 4 is work opposite these valves, where it is open when the radiation unavailable and it is closed when the solar radiation is an abundant, leading hot water tank get fully charging.

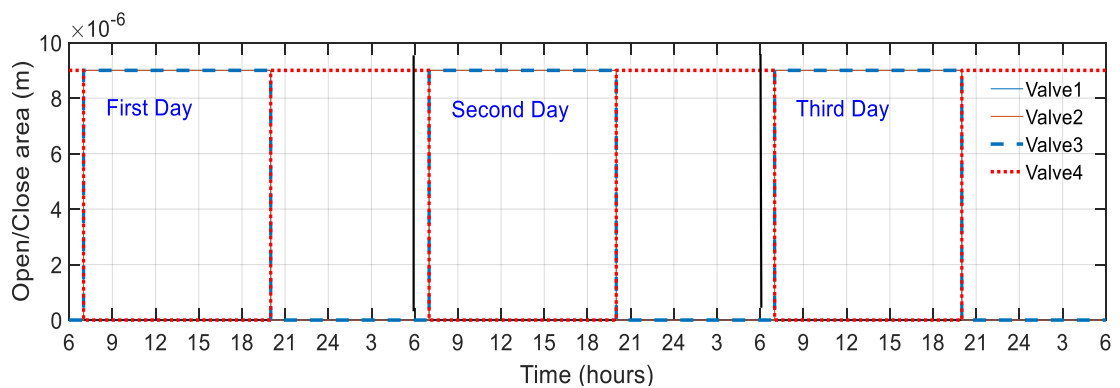


Figure 6.14: Dynamic response for each valve in the system during operation modes

The filling and emptying of the storage tanks, which results in a fluctuating state of charge throughout the day, is shown in Figure 6.15. The system is left off during the early morning hours when the sun's thermal power is rising but the intensity of its

irradiation is low, and all of the heated water is gathered inside the hot tank (while the cold tank is being emptied). When the minimum output power conditions for both systems are met, the steam Rankine and ORC cycles are activated. The hot tank is filled throughout the day's middle hours until being eventually emptied in the late afternoon and evening.

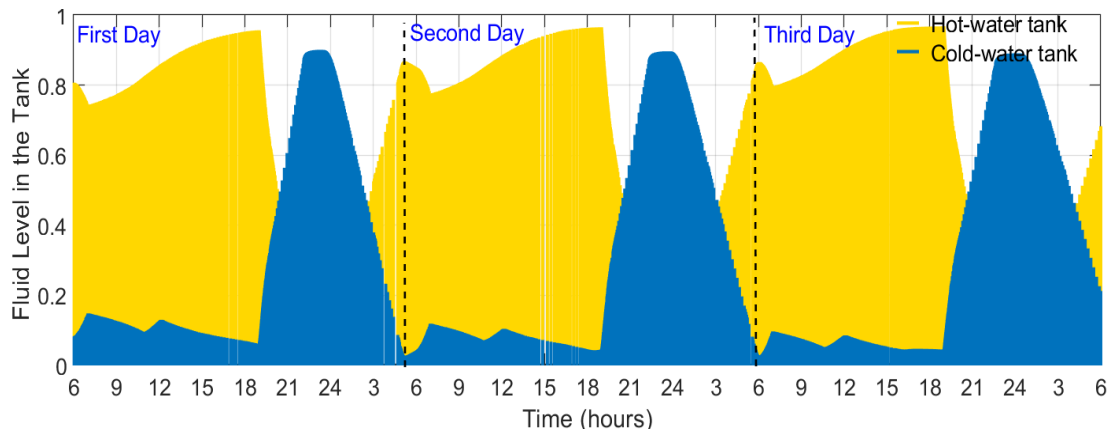


Figure 6.15: Variations of daily profile of the storage tanks state of charge

Figure 6.16 depicts variations of the total mass liquid in the hot and cold-water tank during charging and discharging mode for three days of the simulation. It is clear to see that when hot water tank starts to be charged the total mass liquid inside the tank gradually increases, while it's declined in the cold-water tank, which means all mass liquid in the cold-water tank is vacuumed by water pump to the hot water tank. On a bright day, water typically accumulates in the cold-water tank before the system starts to charge and generate power. In the morning, under low radiation, water is pressured into the solar field and subsequently delivered to the hot water tank. Stable saturated liquid at the design temperature is the control goal in this mode. The flow rate can be controlled to achieve desired temperature in the hot water tank. The total mass liquid in the hot water tank is 40 tonnes in full charging mode which is marketed by yellow line.

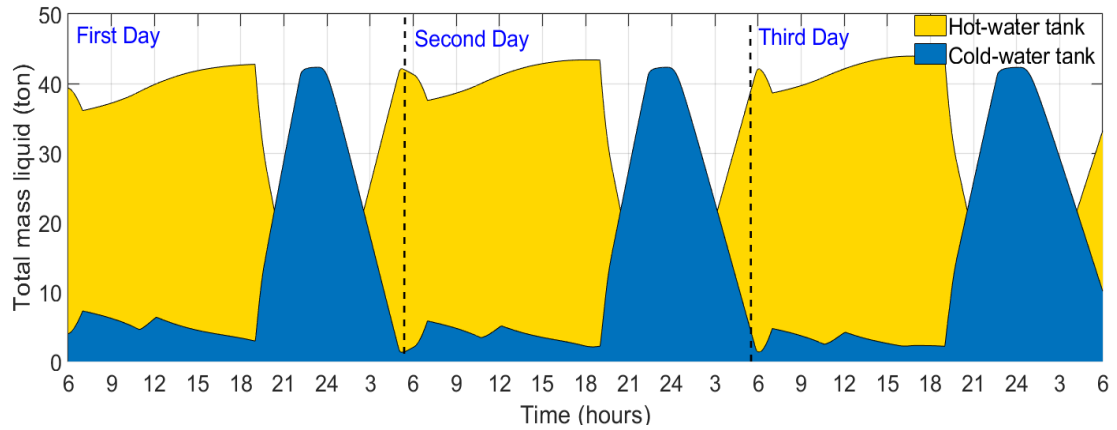


Figure 6.16: Variations of the total mass liquid in the hot and cold-water tank during charging and discharging mode for three days

In discharging mode, the total mass liquid in the hot water tank starts to decrease with the time, and the cold-water tank is gradually charged by low temperature water after powered to ORC system by the heat exchanger as depicted in Figure 6.16.

In the hot water tank, almost 10% of the total volume of the tank is vapor mass. This is because the hot water tank serves as heat storage tank and separator of two-phase fluids at the same time. Inside the hot tank, superheated steam is remaining at the top of the tank while saturated liquid remains at the bottom of the tank because of the steam has a low density compared with saturated water density. Figure 6.17 shows variations of the vapor mass in the hot water tank and cold-water tank during whole day of simulation for three different days.

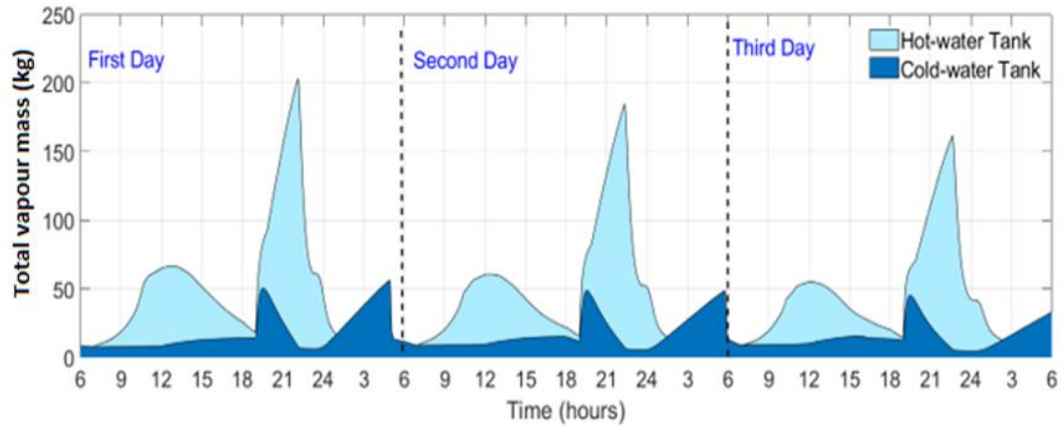


Figure 6.17: Variations of the vapor mass in the hot water tank and cold-water tank during whole day

In the same figure, it can be noticed that there is a very small amount of the vapor mass in the cold-water tank, this is because the exhaust temperature from the heat exchanger may contains some vapor, but this value is almost negligible.

6.6 Performance comparison between the single solar steam Rankine, cascade steam-organic Rankine cycle and solar cascade steam-organic Rankine cycle with two hot water tanks systems

In order to compare a direct steam generation solar power system with a solar-driven cascade steam-organic Rankine cycle, some of parameters must possess the same value, such as the solar collectors' area, steam storage capacity of the accumulator, ambient temperature and solar irradiance. Figure 6.18 shows a typical weather condition in May that has been selected for simulation analysis these three systems.

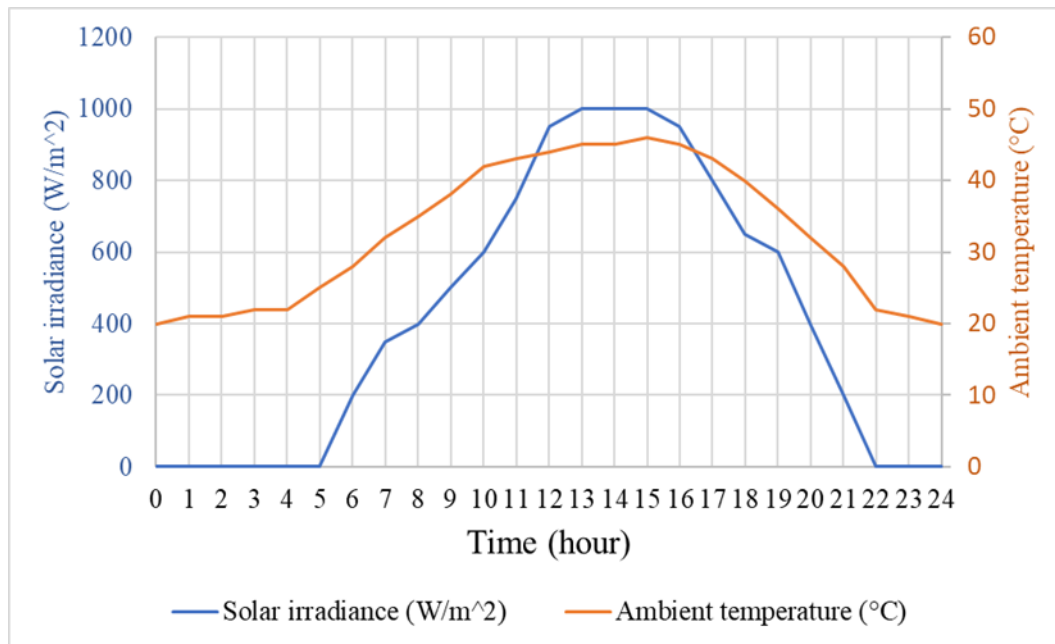


Figure 6.18: A typical weather condition in May

This comparison was based on a typical weather condition as shown in Figure 6.18 above. Figure 6.19 shows the daily variation in the power output for three different systems. It is clear that the solar steam-organic Rankine cycle generates significantly more power compared with the direct steam generation solar power system and the solar steam-organic Rankine cycle integrated with two hot water tanks when the solar irradiance is available. For example, the cascade Rankine cycle was able to generate up to 200 kW at the peak of radiation, while the single Rankine cycle generated only 175 kW during the same period and the solar steam-organic Rankine cycle integrated with two hot water tanks generates 165 kW. In the cascade Rankine cycle the extra power generated was 30 kW, and it was generated by the bottom ORC cycle. The results demonstrated that the exhausted steam of the steam turbine can generate extra power within the system.

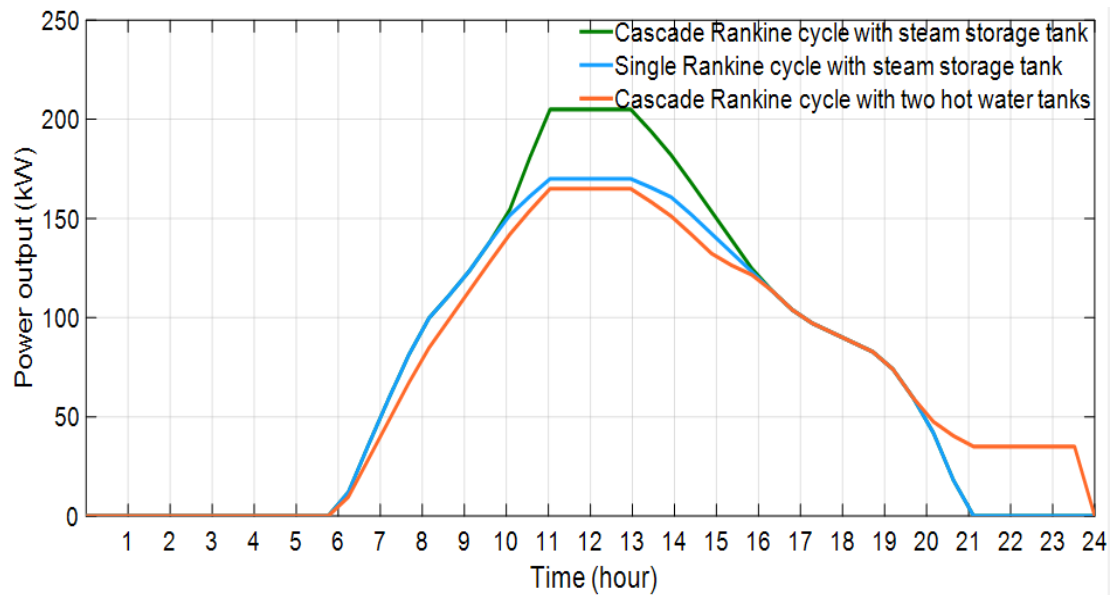


Figure 6.19: Variations of power output for three different configurations systems

During night, only the solar steam-organic Rankine cycle integrated with two hot water tanks can work and generate about 35 kW for up to 3 hours as shown in Figure 6.19. The main reason for this work during night is because that the proposed system stored hot water in the first tank during daytime and the hot water released to the cold water to drive the ORC system by heat exchanger. However, in this study, we found that the heat rejected by the condenser can drive the ORC system smoothly, leading to the generation of more power output compared with the conventional Rankine cycle, which uses a recuperative system which means it is a good idea to apply the cascade Rankine cycle in solar thermal systems.

6.7 Summary

In this chapter, the solar steam-organic Rankine cycle integrated with two storage tanks was modelled and simulated. The storage issue in the solar power plants using direct steam generation can potentially be resolved using the proposed system. The two-stage tank technology is fundamentally different from the current solar thermal

storage options. R134a has been selected as working fluid for the bottom ORC system. The regenerator is adopted in the bottom ORC system to improve the performance of the proposed system. The following is a summary of the findings from the current study:

- The steam Rankine cycle power output is higher than ORC system power output in the nominal operation mode. Where the top cycle generates 136 kW, while the ORC generates 37 kW at the peak of the radiation time. The total power output in this mode is 173 kW.
- The ORC system can generate 31 kW in the discharging mode for six hours.
- The thermal efficiency of solar collectors is reached to 0.73 at the peak time of the radiation. The thermal cascade steam-organic Rankine efficiency is 0.17. While the top steam Rankine cycle efficiency and the bottom ORC efficiency is 0.12 and 0.05 respectively.
- Dynamic comparison between the single solar steam Rankine, cascade steam-organic Rankine cycle and solar cascade steam-organic Rankine cycle with two hot water tanks systems was carried out. Results demonstrated that solar cascade steam-organic Rankine cycle with two hot water tanks system has longest period of generates electricity.

Chapter 7 Conclusion, main outcomes, and future work

7.1 Conclusion

The present civilization is aware that in the future, renewable energy will supply all of the electrical energy consumed globally as these technologies continue to advance.

In nations and places with abundant direct normal irradiation, parabolic trough concentrating solar power based on a direct steam generation system is a viable method to supply electrical power with greater stability and reliability. In solar electricity generation systems, this technology has attracted more and more attention due to its economics and is more environmentally friendly technology compared to other solar thermal energy systems. However, two technical roadblocks, including the unstable steam Rankine cycle and the constrained storage capacity, impede its development. The first is brought on by varying solar radiation. The latter is linked to the slight cooling of water in tanks to prevent inefficient power generation during the discharge process in the off-design state. This thesis has modelled and simulated a study of a novel direct steam generation solar power system integrated with a steam accumulator as a heat storage unit via Simulink/Simscape software to overcome the unstable steam Rankine cycle and the constrained storage capacity in this system. Dynamic modelling of a solar-driven cascade steam-organic Rankine cycle integrated with steam accumulators as heat storage units was simulated using Simulink/Simscape software under transient weather conditions, which is the main contribution of this work. Another issue in DSG solar power systems is that expansion in the turbine might happen in the liquid-vapour two-phase region. The use of a cascade Rankine cycle can be a solution, in which an organic Rankine cycle (ORC) is the bottom cycle, and a steam Rankine cycle is the top cycle with a higher condensation temperature to

mitigate the issue of wet expansion. The bottom cycle can also run separately using low-temperature residue steam\water in the accumulator to offer further capacity in shifting power output profile. Therefore, the design of a cascade steam-organic Rankine cycle integrated with steam\water accumulator was studied from operating and thermodynamic perspectives, aiming to fully unlock the potential of such advanced, high-efficiency cogeneration systems.

This thesis presented a comprehensive literature review on solar Rankine cycle power systems regarding cycle configurations, thermal storages, and working fluids. Also, a parametric study of direct steam generation solar power systems compared with indirect steam generation under Libyan climate conditions was designed and validated with a real solar DSG plant in Thailand. In this thesis, integrated a steam accumulator in the direct steam generation solar power system was simulated. Moreover, a solar-driven cascade steam-organic Rankine cycle integrated a steam accumulator, and the effect of the generator on the performance of the system was investigated. A solar-driven cascade steam-organic Rankine cycle integrated with two water accumulators was modelled. All dynamic models in this thesis were simulated using Simulink\Simscape software under variable solar irradiance and ambient temperature. The conclusions obtained from the present work are summarised as follows:

- Results show that the direct steam generation solar power system outperforms the indirect system in terms of power generation and thermal efficiency based on the same solar collecting area. The results indicate that 1,320m² of solar collecting area can produces 250 kW of power in the direct system, while the indirect system produce only 240 kW under 1000 W/m² of solar irradiance. The findings show that the direct steam solar power generation system will be a promising technology for Libyan applications in the future. After seasonal

analysis and a parametric study were carried out, the direct steam generation system achieved a good result in the summer, spring, autumn, and even winter in Libya. The optimisations results show that $1,320\text{m}^2$ of solar collecting area can produces up to 270 kW of power in the direct steam generation system.

- The steam accumulator has the ability to enable the direct steam generation solar power system to overcome the fluctuation in solar radiation during the day as well as working as a separator liquid when steam consists of a two-phase regime. Based on the constant solar field area of 1320m^2 , the proposed system can generate up to 210 kW without a control strategy and, with a controlled system, can generate a stable 180 kW, even when there is fluctuation in the solar radiation.
- According to research, it was found that, to meet the maximum electricity 200 kW and hot water demand of a Libyan hospital, a $1,544\text{m}^2$ collector area and 160m^3 steam accumulator were required for the solar steam Rankine system with a throttle valve control. By controlling the steam flow rate into the steam turbine and having steam accumulator as heat storage tank, the system can match the power demand even the solar irradiance is fluctuating.
- A week-long comparison of the performance of the Solar-DSG and Solar-cascade systems was conducted, and the results show that the solar-driven cascade Rankine cycle has both higher thermal efficiency and power output than the solar-DSG. The cascade Rankine cycle was able to generate up to 200 kW at the peak of solar radiation, while the single Rankine cycle generated only 175 kW during the same period. The extra power generated was 30 kW, and it was generated by the bottom ORC cycle. The results demonstrated that

the exhausted steam of the steam turbine can generate extra power within the system.

- In the solar-driven cascade steam-organic Rankine cycle coupling with the steam accumulator with and without using a recuperator, the findings reveal that the recuperative system outperforms the non-recuperative system in terms of power generation and thermal efficiency. The overall thermal efficiency of the proposed system improved from 0.17 to 0.20 at the peak time of solar radiation, and power output improved from 210 kW to 220 kW. R600a as the working fluid for the bottom ORC has achieved higher power output compared to R134a, R1234yf and R407c in the solar-driven cascade steam-organic Rankine cycle coupling with the steam accumulator. Furthermore, the findings of this study reveal that a minimum degree of superheating working fluid can achieve a higher power output. For example, 10 °C of superheating can generate up to 220 kW, while 25 °C of superheating can produce only 180 kW during the peak time of solar irradiance. The degree of superheating temperature in the heat exchanger is adopted to be 10 °C which is the best value which has an impact on the system's performance. The condensation temperature has a significant effect on the proposed system. Four condensation temperature degrees were evaluated (20, 25, 30 and 35 °C). The findings reveal that, when the condensation temperature decreases to 20 °C, the power generation increases to generate up to 220 kW and the solar steam-organic Rankine cycle efficiency increases to 0.20 %, which means that the condensation temperature has a significant effect on the proposed system.
- In solar steam-organic Rankine cycle integrated with two water storage tanks system, results show that the top DSG output is higher than ORC system power

output in the nominal operation mode. Where the top cycle generates 136 kW, while the ORC generates 37 kW at the peak of the radiation time. The total power output in this mode is 173 kW. In discharge mode, the ORC system can generate 31 kW in the discharging mode for six hours based on a 51m³ of the hot water tank.

- A performance comparison between the single solar steam Rankine, cascade steam-organic Rankine cycle, and solar cascade steam-organic Rankine cycle with two hot water tank systems was carried out in this study under typical May weather conditions. The results show that the solar steam-organic Rankine cycle generates significantly more power compared with the direct steam generation solar power system and the solar steam-organic Rankine cycle integrated with two hot water tanks when solar irradiance is available. The cascade Rankine cycle was able to generate up to 200 kW at the peak of radiation, while the single Rankine cycle generated only 175 kW during the same period, and the solar steam-organic Rankine cycle integrated with two hot water tanks generated 165 kW. In the cascade Rankine cycle, the extra power generated was 30 kW, and it was generated by the bottom ORC cycle. The results demonstrated that the exhausted steam from the steam turbine can generate extra power within the system.

The main outcomes of this thesis can be summaries:

- ❖ Integrated steam accumulator with the direct steam generation solar power system.
- ❖ A cutting-edge direct steam generation system that uses steam accumulator and a cascade steam-organic Rankine cycle powered by solar parabolic trough collectors.

- ❖ Two-step heat discharge method for thermal storage in the solar-driven cascade steam-organic Rankine cycle with two tanks.

7.2 Limitations and recommendations for future work

During the work of this research, and with the quite huge simulations that were done, there are several limitations that need to be addressed in future work to make the direct steam generation solar power system work with a stable power output and fulfil the demand for energy during the whole day. The recommendations for future work can be summarised as follows:

- ❖ The direct steam generation solar power system integrated with steam accumulator as heat storage tank has demonstrated the ability to provide electricity and domestic hot water for a commercial building in Libya. So, the feasibility study of direct steam generation solar power system needs to be investigated.
- ❖ The proposed system of DSG system integrated with steam accumulator and the throttling valves demonstrated that stability of power generation, this method can be applied for solar cascade steam-organic Rankine cycle to make the system generates stable power output.
- ❖ The techno economic evaluation of the solar cascade steam-organic Rankine cycle needs to be investigated by considering investments in different solar collector types, turbines or expanders, regenerator and storage tanks.
- ❖ The effect and assessment and working fluid comparison on the performance of solar cascade steam-organic Rankine cycle needs to be analysed. In this study, only four working fluids R134a, R1234yf, R704c and R600a were investigated in the bottom cycle, more working fluids such as dry fluids n-butene, isobutene and wet fluids cyclopropane, propyne.

❖ In solar cascade steam-organic Rankine cycle, an ORC turbine could be utilised in the top cycle as an alternative to the wet steam turbine. This use would require a secondary heat transfer between the organic fluid and the water/steam in the HTA, but a better turbine efficiency is anticipated with less technical difficulty and the total cost of the system will increasing.

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