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**A novel a-Si PV/T solar cogeneration system using the
coupled thermal storage/ORC as an alternative to battery**

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Abstract

The recent intensive focus on energy and environmental issues has resulted in an increase in studies of renewable energy applications. Solar energy has been identified as one of the most promising types, with solar energy technologies attracting a high level of interest in order to satisfy current energy demands. It is known that solar heat collection usually has a low to medium temperature in comparison with combustion heat in steam power plants. Therefore, the Organic Rankine Cycle (ORC) is appropriate for the generation of electricity from solar heat. By using the solar ORC system with a heat storage unit, electricity generation can continue during the night, however, it is known that solar to electricity conversion efficiency of these systems is low.

Photovoltaic (PV) technology has become a popular technology in recent years for electricity generation; however, electricity conversion efficiencies of this technology are still relatively low, resulting in a large proportion of collected solar radiation being dissipated as heat. To utilise this heat dissipation for other purposes, photovoltaic/thermal (PV/T) technology has become a very attractive option for solar-driven combined heat and power systems. Heat from a PV/T system can be used directly or stored for use at other times to drive a power generation cycle. The main problem with the low to medium temperature PV/T system is its lower electricity conversion efficiency at a raised temperature. However, this problem can be solved by using amorphous silicon (a-Si) PV cells instead of crystalline silicon cells. This thesis aims to investigate the potential and characteristics of a PV/T system coupled with thermal storage, using the ORC as an alternative to battery in matching the energy demand profile of buildings. In order to carry out the investigations, a thermodynamic analysis of the combined system is presented, including the transient modelling of a

hot water storage tank. Components were modelled considering off-design operation in transient state associated with hourly EnergyPlus weather data. After completion of the modelling, a detailed experimental investigation was conducted, with a focus on the performance of a scroll expander, as the most critical component of the ORC, its operation metrics and performance.

As the key factor is matching the user electricity demand profile, this thesis further uses a controlling methodology for the solar ORC system to control electricity output according to this profile. By adopting this methodology, the collected and stored solar heat can be utilized more effectively, thus allowing for the provision of high electricity output for users during peak hours.

The thesis comprises a literature review of the whole system's components and presents a detailed modelling of a solar ORC system, considering both daily and annual electricity demand profiles of residents. Additionally, it investigates the direct incorporation of a solar organic Rankine cycle with a vapour compression refrigeration unit to reduce electricity conversion losses. Finally, the modelling of a scroll expander is comprehensively conducted; expander generator coupling performance is assessed and a realistic modelling methodology is presented.

Results show that it is possible to meet the electricity demand even at night by controlling the mass flow rate of the circulation water. In order to provide 24-hour electricity to a small community with twelve dwellings, 550 m² collector area and 88 m³ heat storage unit are required for a solar ORC system. Solar energy to electricity conversion ratio was found quite low for the solar ORC systems, but for proposed innovative a-Si PV/T-ORC system, electricity generation is increased 103 % compared to solar ORC system for operations in July. a-Si PV/T-ORC system also integrated with a vapour compression system to provide both electricity and cooling.

Results show required solar collector area and heat storage unit reduce considerably by adapting PV/T-ORC-VCC unit. Finally, new scroll expander-generator coupling modelling approach allows users to predict rotational speed and electricity output.

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List of publications

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3. **C.Kutlu**, J.Li, Y.Su, Y. Wang, G.Pei, S.Riffat, Investigation of an innovative PV/T-ORC system using amorphous silicon cells and evacuated flat plate solar collectors, *Energy*, 203, (2020), 117873.
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5. **C.Kutlu**, J.Li, Y.Su, Y. Wang, G.Pei, S.Riffat, Annual performance simulation of a solar cogeneration plant with sensible heat storage to provide electricity demand for a small community: A transient model, *Hittite Journal of Science & Engineering*, 6 (2019), 75-81.

Conference papers

1. **C.Kutlu**, M.T.Erdinc, J.Li , Y.Su, G.Pei, G.Gao, S.Riffat, Scroll expander modelling by combining semi-empirical and geometric models: calculation of flow rate, *Alternative Energy Sources, Materials & Technologies (AESMT'19)*, 03 - 04 June, 2019, Sofia Bulgaria.
2. **C.Kutlu**, J.Li, Y.Su, Y. Wang, G.Pei, S.Riffat, Investigation of an innovative PV/T-ORC system using amorphous silicon solar cells and evacuated flat plate collectors: A comparison study, *17th International Conference on Sustainable Energy Technologies*, 21-23 August 2018, Wuhan China
3. **C.Kutlu**, J.Li, Y.Su, Y. Wang, G.Pei, S.Riffat, Annual performance simulation of a solar cogeneration plant with sensible heat storage to provide electricity demand for a small community: A transient model, *17th International Conference on Sustainable Energy Technologies*, 21-23 August 2018, Wuhan China.
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Nomenclatures

		Subscripts	
A	Area, m^2		
AU	Heat transfer coefficient-heat transferring area, W K^{-1}	a	Armature
BVR	Built-in volume ratio, –	ad	Adapted
c_1	Heat loss term, $\text{W m}^{-2}\text{K}^{-1}$	am	Ambient
c_2	Heat loss term, $\text{W m}^{-2}\text{K}^{-1}$	b	Base
c_p	Specific heat, $\text{J kg}^{-1}\text{K}^{-1}$	bo	boiling
d_{st}	Tank diameter, m	c	Chamber
Δk	De-stratification conductivity, $\text{W m}^{-1}\text{K}^{-1}$	crit	Critical
f_{sg}	Overall dynamic friction coefficient of scroll and generator, N m s	col	Collector
G	Solar Irradiance, W m^{-2}	con	Condenser
h_s	Scroll height, m	cw	Collector water
I_a	Current, A	DB	Dittus Boelter
J	Moment of inertia, kg m^2	dis	Discharge
K_θ	Incident angle modifier, –	e	End
K_e	Constant of back electromotive force, V s rad^{-1}	em	Electromechanical
K_t	Torque constant, N m A^{-1}	eva	Evaporator
L	Tangential distance, m	ex	Exhaust
L_a	Armature inductance, H	e ₁	Evaporating region

M	Mass, kg	e_2	Single phase region
$\dot{m}$	Mass flow rate, kg s ⁻¹	f	Flank
N	Rotation speed, RPM	fr	Friction
N_c	Number of chamber pairs, -	i	Inner
Nu	Nusselt number, -	in	Internal
P	Pressure, Pa	$i0$	Inner initial
Pit	Scroll pitch, m	(i)	Number of chambers
Pr	Prandtl number, -	h	hydraulic
R	Resistance, Ω	$leak$	Leakage
Re	Reynolds number, -	m	Mean
$\dot{Q}$	Heat transfer rate, W	mec	Mechanical
r_b	Basic circle radius, m	nom	Nominal
St	Strouhal number, -	o	Outer
T	Torque, N m	$o0$	Outer initial
$\bar{T}$	Mean temperature, °C	old	Oldham
t	thickness, m	or	Orbiting
U_T	Overall heat transfer coefficient of tank, W m ⁻² K ⁻¹	p	plate
$\bar{U}$	Mean flow velocity, m s ⁻¹	r	Radial
V	Volume, m ³	rc	Refrigerant in cooling cycle

v	Specific volume, $\text{m}^3 \text{kg}^{-1}$	ref	Refrigerant
$\dot{W}$	Power, W	s	scroll
Greek letters		sh	Shell
γ	Isentropic exponent, -	sha	Shaft
δ	leakage clearance size, m	st	Storage
θ	orbiting angle, rad	su	Supply
ρ	Density, kg m^{-3}	suc	Suction
ϕ	Involute angle, rad	sw	Swept
ω	Angular velocity, rad s^{-1}	t	Tangential
		th	Theoretical
		w	Water
		0	initial

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CHAPTER 1 Introduction

1.1 Background

Renewable energy technologies have received special worldwide attention, particularly in developed countries. Although fossil fuels will undoubtedly remain the most dominant energy source over the next decades, attention must be given to the provision of cleaner, more secure and sustainable energy sources, as strongly supported by public opinion. This trend has firmly established renewable technologies as a necessary participant in energy production reflect in an exponential growth in recent years in the sector. Solar energy has been defined as one of the most promising types of renewable energy sources. Solar energy systems can not only be used for electricity generation but are also applicable in various energy demanding systems such as refrigeration, desalination, hydrogen production and improvement of indoor environment conditions [1]. However, Figure 1.1 shows that energy generation by renewables represents only a small proportion against the total share. It is reported that global energy demand has grown by a yearly average of around 1.8% since 2011, largely in developing countries [2]. It follows therefore that carbon release will increase exponentially, if this proportional figure remains the same, given that global emissions of CO₂ have expanded from 14 to 33 Gt from 1970 to 2015.

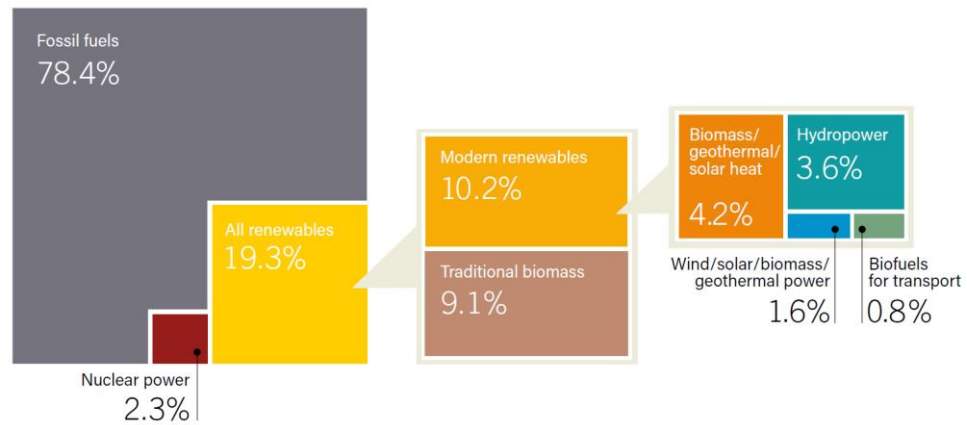


Figure 1.1 Global energy contribution by sources to overall consumptions in 2015 [2]

In most parts of the world, electricity is the most important energy source for people due to its ability to easily convert other energies. As shown in Figure 1.2, the total share of renewable electricity production is 24.5%, however, less than 2% of electricity is produced by solar PV and solar based power plants.

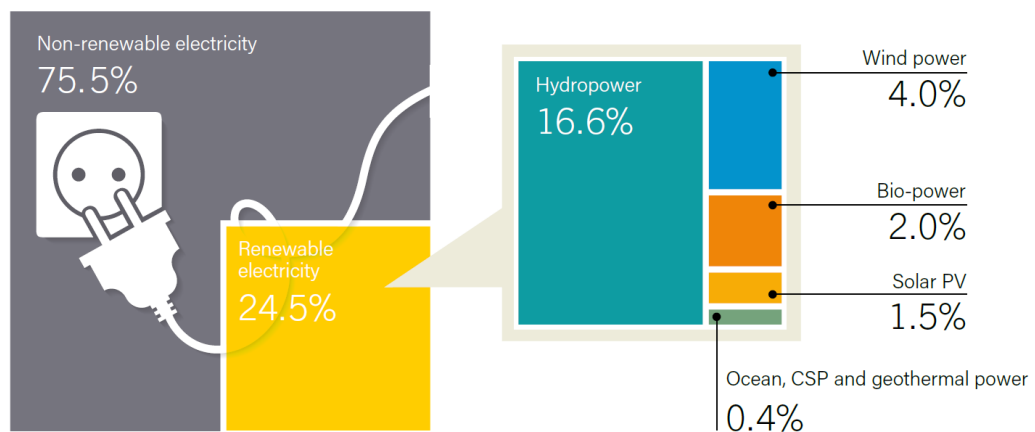


Figure 1.2 Estimation of global renewable energy share in 2016 [2]

Further, Figure 1.3. shows the prediction of global electricity demand for various energy sources in 2050 [3]. In this estimation, by the year 2050 solar energy systems will occupy the highest share. Likewise, the International Energy Agency predicts that by the year 2050, solar energy systems will be supplying about 17 % of the global electricity demand in the 2 degree rise scenario in which they defined it [4]. The PV market has seen strong growth over the last twenty years. In parallel, production costs have fallen exponentially as a result of this increased production, predominantly in China. The price of crystalline silicon (c-Si) solar cells has decreased by 10-15% per year. The biggest PV power station in the world was recently constructed in US with an installation area of over 25 km² and a 0.5 GW_p peak capacity [5].

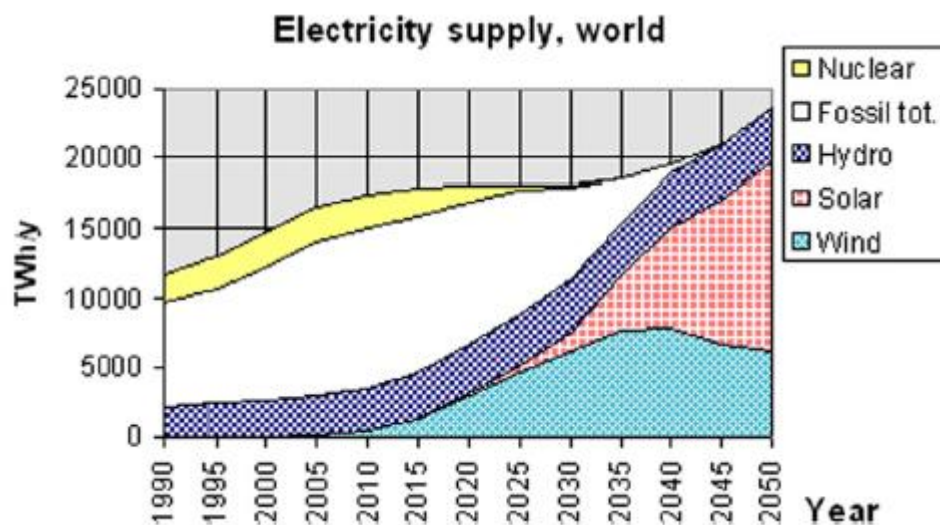


Figure 1.3 Development of the world energy supply [3]

The most popular sustainable way is electricity generation by Photovoltaics (PV) technology [1]. PV cells convert solar energy into electricity with an efficiency of about 5 to 25 %. This low electricity conversion efficiency shows that important

amounts of solar radiation are reflected or converted into thermal energy [6]. This process causes a rise in cell temperature and a decrease in electricity conversion efficiency. If thermal energy leads to an increased temperature in solar cells when it is rejected and is used properly, this normally prevents temperature increases in PV cells, and usually simultaneously improves the efficiency of the system. As a result of this phenomenon, photovoltaic/thermal (PV/T) systems are applicable. In these hybrid systems, both photovoltaic electricity and thermal energy are co-produced [7]. During the past few years PV/T technology has received considerable international attention and numerous design principles have been developed, however, the most investigated one uses water as a heat transfer medium [8].

Electricity generation from solar energy comes both from PV-based semi conductive materials and large scale solar thermal power plants with concentrated collectors, referred to as concentrated solar plants (CSP). The largest CSP plant in the world is in the US and covers 14 km² with a rating of 0.4 GW [5]. When it comes to utilization of a renewable energy source, operating with cogeneration or multigeneration can be greatly advantageous [9]. The term ‘cogeneration’ can be defined as simultaneously generated electricity and heat from one or more sources; the heat is used for another purpose such as desalination, heating and cooling, hot water and hydrogen. The term also defines systems combining heat and power (CHP). CHP systems are used for satisfying multiple user demands. The main applications of the CHP are electricity, space heating, hot water for residents, space cooling, refrigeration and potable water applications [1].

In order to produce electricity from heat, power generation cycles are used. There are numerous cycles available for different specifications and conditions. Figure 1.4

shows thermal efficiencies of different types of heat engines with various heat source temperatures.

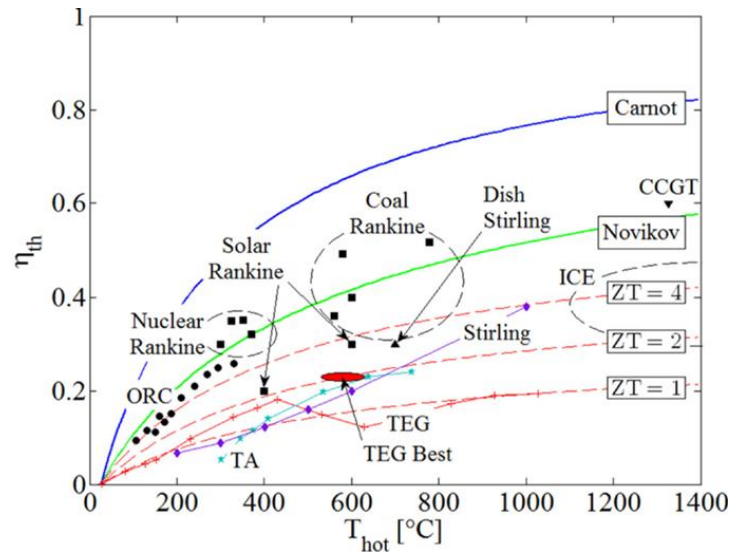


Figure 1.4 Different heat engines' thermal efficiencies [5]

This figure includes heat source temperatures from 100 °C to 1400 °C, the maximum limit of the thermodynamics is shown by the Carnot efficiency, indicated as blue line. Heat sink temperature is a fixed value of 25 °C. Figure 1.4 also presents the working ranges of power cycles, such as those up to 350 °C, where ORC and Kalina cycles can be operated using waste heat, geothermal and solar sources. , Large scale CSP Rankine cycles can operate from 400 °C to 600 °C, whereas coal-based Rankine cycles can operate up to 900 °C.

In conclusion, electricity has become a crucial need for human life and electricity demand is increasing day by day. This demand can be met by existing suppliers, however, not without further increase in environmental concerns about fossil fuels. While renewable energy is strongly encouraged by societies and governments, there

is the need for significantly more development in this field to fully satisfy demand. For residential users, PV technology offers a solution, however, PV is not able to produce power when the sun sets. Recently, lithium batteries have been employed for applications as these can be charged in the daytime and used as an electricity supply at night; these batteries, however, are very expensive and known to present some problems in applications. This study therefore aims to investigate a PV/T-ORC system in accordance with household demand using a coupled heat storage/ORC unit as an alternative to a lithium battery.

1.2 Aim and Objectives of the Study

This thesis aims to investigate the characteristics and performance of a PV/thermal cogeneration system incorporated with a coupled thermal storage/ORC unit as an alternative to battery in matching the energy demand profile of residential buildings. Consequently, there are five principal objectives to this work with the purpose of achieving this aim:

1. To review whole system components of solar ORC systems separately and to determine the research gap by reviewing recent studies in the literature.
2. To carry out thermodynamic analysis of the combined system, to include solar array, heat storage unit and organic Rankine cycle. A model is generated according to transient responses of the components and user control schemes. Then, the model is built to control electricity output to match electricity demand profile of a building. To control electricity output to better match demand profile, a strategy is developed to control evaporating temperature of the ORC.

3. To increase the power output from the solar ORC unit, a novel a-Si based PV/T-ORC system is proposed, modelled and compared with other existing systems by keeping the same component dimensions. The advantages of the modification are presented and discussed.
4. To investigate the solar ORC system with heating when required over a year considering transient state. Further, to investigate various cooling purposes of the solar ORC unit in a building, using a vapour compression cycle to provide electricity and cooling demand profile for a day and for different component sizes.
5. To investigate the scroll expander in detail via geometrical modelling and a new methodology to obtain and present more output parameters (electricity output, rotational speed) with considering generator coupling effects and make it more flexible for different capacities of the scroll expanders. This will include a validation study with experimentally obtained parameters.

1.3 Innovative aspects and contribution of this thesis

Studies on user demand-based systems have currently peaked. This project is aimed at addressing a number of research gaps in the literature by combining user demand profile and intermittence of solar energy. As it is known that solar energy is not applicable for electricity generation during the night, a coupled heat storage/ORC unit is adapted as an alternative to lithium battery. The schematic of the proposed solar ORC system is shown in Figure 1.5. Using a heat storage unit allows for night-time production of electricity, however, its finite storage energy will reduce with time. Thus, electricity production should effectively match the demand profile for heat to remain in the storage for a longer production time. Therefore, the controlling

procedure for the ORC output is important. Therefore, a circulation water flow rate is controlled to match with the user demands. A solar ORC system is modelled to provide all-day electricity considering user demands. The model includes comprehensive modelling of the components in transient state. In order to increase power output or reduce storage size, a novel a-Si based PV/T-ORC system is presented by modifying evacuated flat plate (EFP) collectors with a-Si cells. This novel system has all advantages of solar ORC and doubled the power output. Finally, a scroll expander model is improved using its scroll geometry. This model considers expander-generator interaction and gives the rotational speed of the expander, and the electricity output directly.

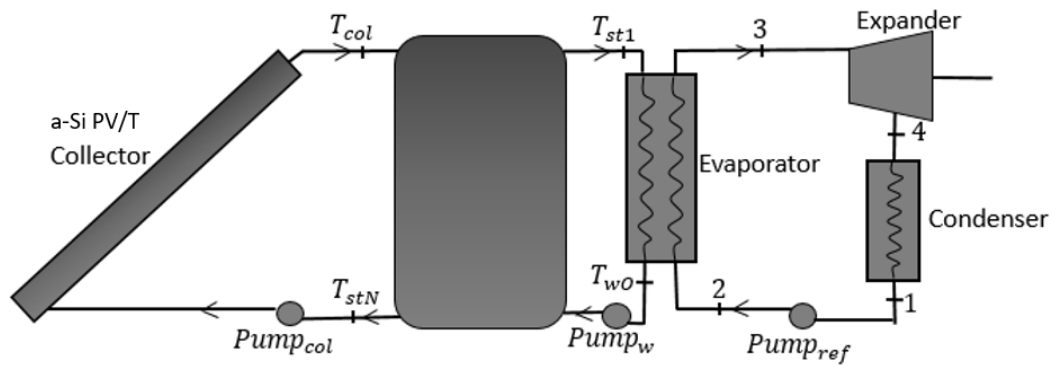


Figure 1.5 Schematic view of the solar ORC system

1.4 Methodology

This thesis mainly comprises the following stages: Literature review: A comprehensive literature review of flat plate collectors, PV modules, water storage units and ORC systems are performed in order to comprehend the scope of this project and identify relevant key data from previously published studies. Modelling: Mathematical transient modelling of the system is undertaken using MATLAB

software. As operating conditions such as heat source temperature and demand from the residents, the components should respond according to operating parameters. Thus, off-design operation is modelled. Working fluid properties and fluid property plots are taken from EES (Engineering Equation Solver) and Refprop software. In order to obtain hourly weather data, Energy Plus is used. Figure 1.6 shows step by step research work in the thesis.

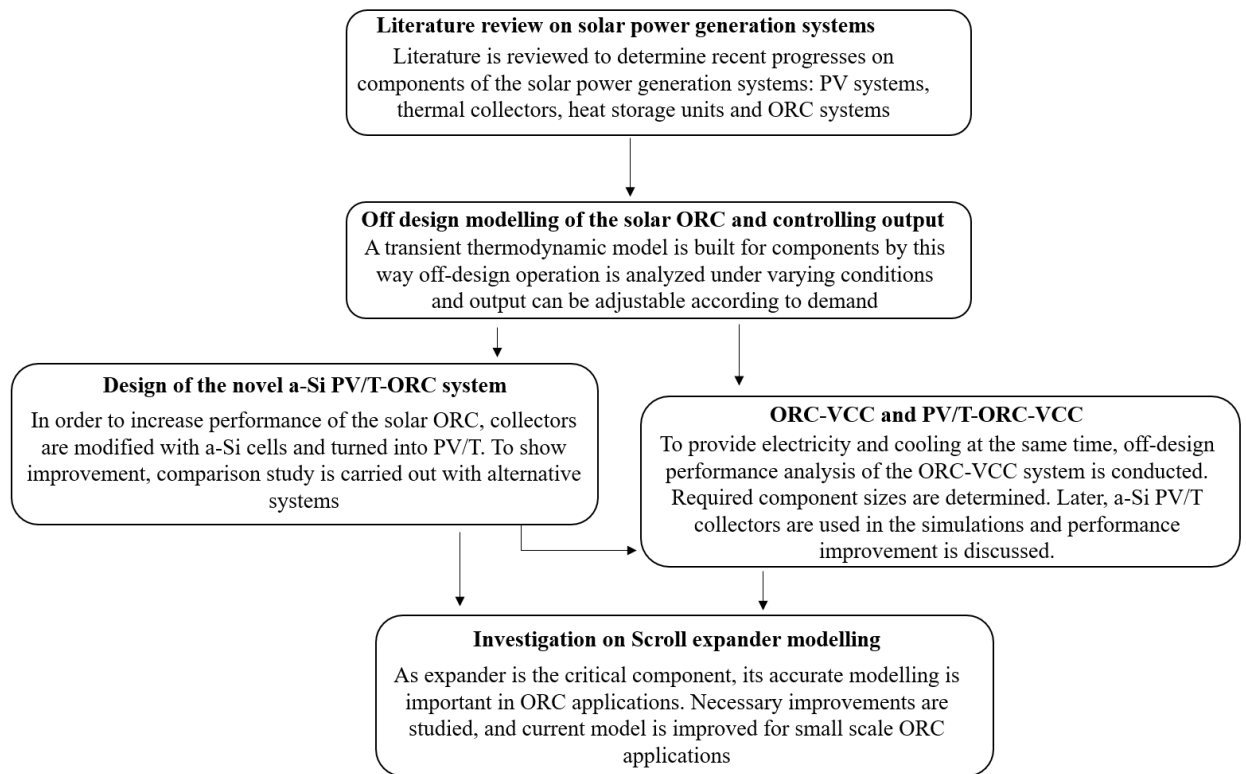


Figure 1.6 Followed methodology of the thesis

1.5 Outline of the Thesis

This thesis comprises seven chapters. Chapter 1 introduces essential research background information and identifies the research gaps, key aims and objectives, innovation and contribution of the project and a summary overview of methodology and outline of the thesis.

In Chapter 2, solar based power generation systems and components are reviewed in detail. The chapter begins with an introduction of the background of solar systems and the ORC. This focuses on currently used PV materials in the market, PV/T applications and their modelling features, solar thermal power generation systems, heat storage units and their modelling approaches and finally, published scroll expander models as discussed in the literature.

Chapter 3 is the main modelling chapter of the thesis in which the solar collector, heat storage unit considering thermocline, heat exchangers and the ORC unit are modelled considering transient state of analysis. Further, annual performance of the solar CHP system is simulated, and performance figures of the unit are given for winter and summer conditions.

In Chapter 4, an innovative a-Si PV/T system incorporated with an organic Rankine cycle is proposed. This chapter presents an innovative a-Si PV/T-ORC system design, providing better electricity output performance by combining the advantages of PV and solar ORC systems. The system performance is compared with stand-alone PV, poly-Si PV/T-ORC and solar ORC systems.

A solar ORC-VCC system is presented in Chapter 5 to simultaneously provide electricity and cooling for an office building. Using this configuration, electricity conversion losses are eliminated and effect of storage sizes and controlling methodologies are discussed.

A new modelling methodology for a scroll expander is presented in Chapter 6. An investigation into the characteristics and prediction of the expander is presented using scroll geometry and experimental and simulation results are then compared.

In Chapter 7, the conclusions derived from the modelling and simulation results are provided to show the extent to which the given objectives have been met. Further, possible future developments are outlined with recommendations.

CHAPTER 2 Review of the Solar Power Generation Systems

2.1 Introduction

Photovoltaic systems are composed of PV cells that convert solar energy into electricity. The remainder of the solar energy which cannot be converted into electricity remains as heat in the system. This heat causes an increase in temperature of the PV cells and leads to a reduce in conversion efficiency. To keep conversion efficiency at a sufficiently good level, PV cooling is necessary for lots of cases. In order to remove heat from PV cells, heat transfer fluids are employed. Using air as a heat transfer fluid is a very simple and low cost method, however, at low latitudes where the ambient temperature is higher than 20 °C, it is less effective than using a liquid [10]. Heat should be sent away in order to obtain effective cooling and research is being undertaken into the most effective way to use this heat for other purposes. Hybrid photovoltaic/thermal (PV/T) systems are applicable at this point. The heat carried by working fluid could be used for charging a heat storage unit to use later or to provide domestic hot water (DHW) in buildings [11]. Figure 2.1 shows a schematic view of a solar flat plate collector integrated with a thermal heat storage unit. The thermal heat storage unit is one of the main parts of the solar-driven power generation, heating and cooling systems. Designing an effective thermal storage unit is a priority, given that it provides continuous operation and for some applications it needs to store heat effectively for use at night [12]. It is also reported that these systems have a great influence on performance system cost and reliability [13].

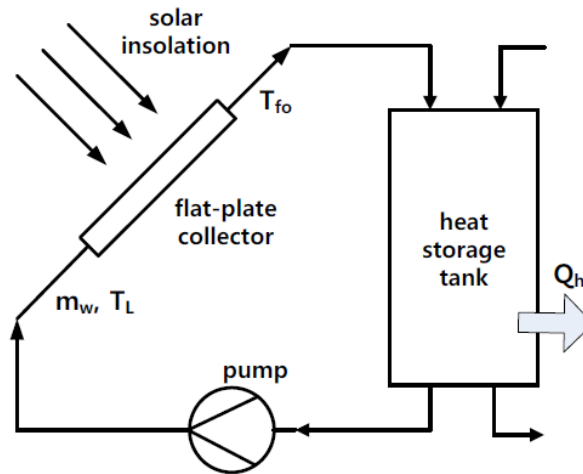


Figure 2.1 Schematic view of a collector and a heat storage unit [12]

A considerable amount of waste heat which is low temperature to use is dissipated into the environment or cannot be used. In recent years, ORC technology has become a promising technology in order to convert low grade temperature of heat into electricity. Figure 2.2 shows journal papers and patent quantity about the ORC from 1970. A small peak can be seen in the 1980s, resulting from the oil crisis which prompted researchers to search for alternative energy. In the last decade, the quantity of research papers has reached at all times high. Waste heat to the ORC can be taken from several heat sources such as solar radiation, geothermal energy, biomass, and waste heat from any other areas. Although steam power plants use water, ORC uses a refrigerant as a working fluid in order to maintain operating pressure level under low temperature conditions.

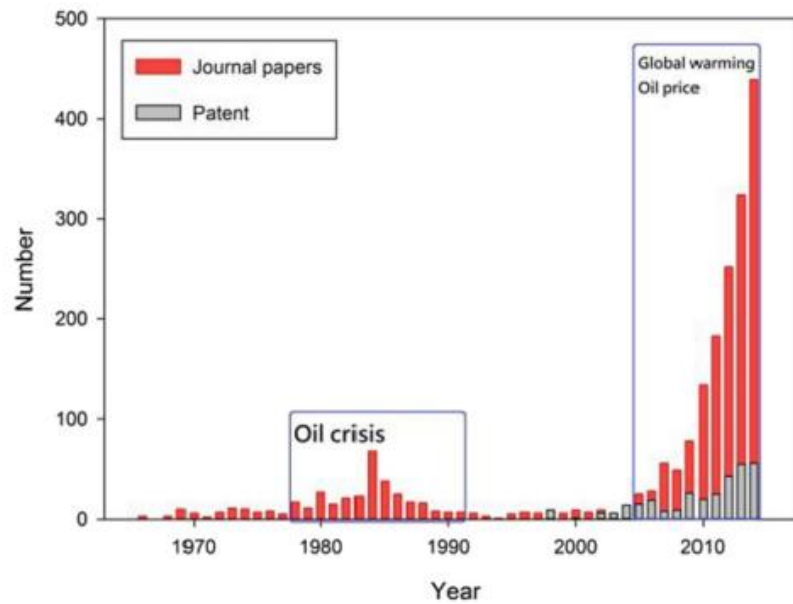


Figure 2.2 Quantity of journal papers and patents on ORC [14]

In order to design the more efficient systems required, more effective components need to be identified and researchers have invested significant time in looking at ways to improve system performance of the ORC.

This chapter consist of six main sections. Firstly, electricity demand profiles for different regions and reasons for variation are discussed. Secondly, PV technology and hybrid PV/T systems are reviewed including effects and influences of the simulation models which are steady state, lumped and discretization methods. Thirdly, the solar collectors are reviewed for thermal applications. Fourthly, heat storage units are reviewed and comprehensively discussed including sensible and latent storage units. Finally, ORC system is explained, and studies are reviewed.

2.2 Electricity demand profiles

In most parts of the world, electricity is the most important, sought after energy source for residential consumers as given, buildings have 29% of total electricity consumption through European countries [15]. Ability of the easily conversion to other energies and household appliances need it in order to work make electricity the most demanded form of energy. The demand of the electricity is supplied by grid, but the supplied electricity is not stable during the day because of demand profile changes. Some studies have been conducted to model and specify the hourly electricity demands [15],[16]. Regarding to demand, its magnitude may differ from country to country but the general trend is similar for all houses [17]. Figure 2.3 shows active use of percentages of the house appliances for five European countries. Since the figure does not consider populations, it is quite useful to understand the general profile of the house residents' behavior. Electrical loads of some appliances can be grouped under activities such as preparing food and cleaning appliances or leisure activities. Typical electricity consumption of the hob, oven, microwave, and kettle are 2.4 kW, 2.13 kW, 1.25 kW and 2 kW, respectively. These appliances are the main reason of the peak periods. Personal computer and TV's electrical loads are 0.14 kW and 0.12 kW respectively [18].

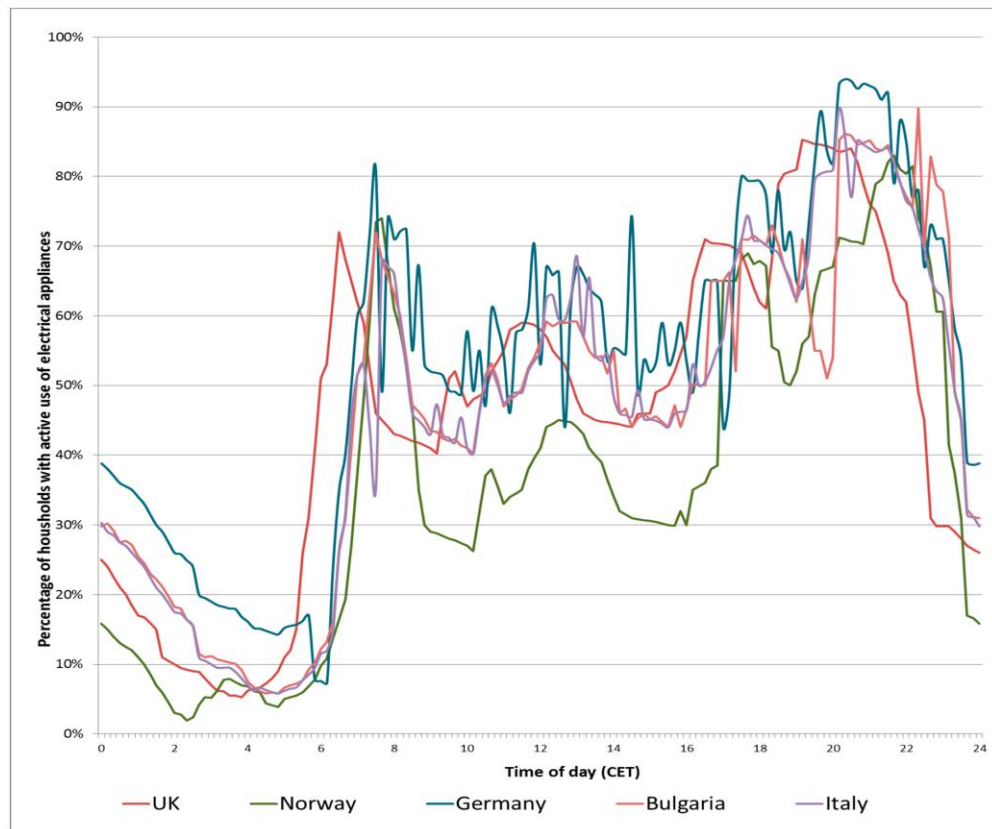
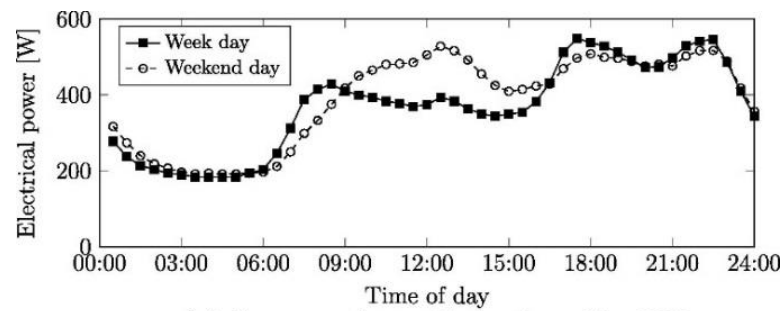
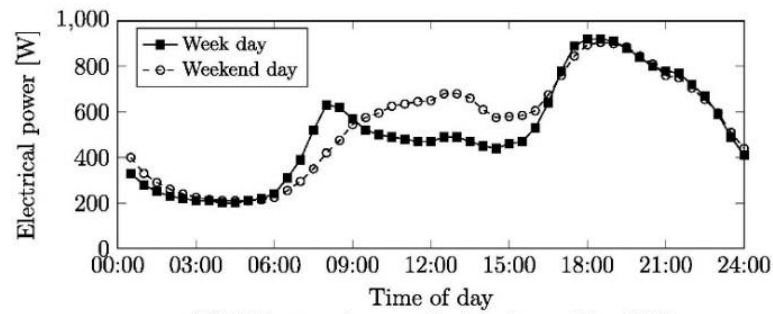


Figure 2.3 Electrical appliances usage profile of some European countries [18]

After determining of the trend of the electricity usage profile, it can be useful to compare two different seasons and climates' effect on the use of electricity. Fig. 2.4 shows average 24h domestic electricity demand for a dwelling in the UK [19]. The figures are based on measured data and taken from UKERC energy data centre [20].



(a) Summer domestic load profile, UK.



(b) Winter domestic load profile, UK.

Figure 2.4 Domestic load profile in UK [19]

As seen from the Figure 2.4, peak demand occurs in the evening for both summer and winter. Although the trend looks similar, but consumption increases in winter, the main reason of this increment may come from the boiler pump consumption. Another study's results about UK households given in Fig. 2.5, reveals that electricity demand peaks in December and the lowest one is shown in May. However, peak demand happens between 6 -7 pm for all months.

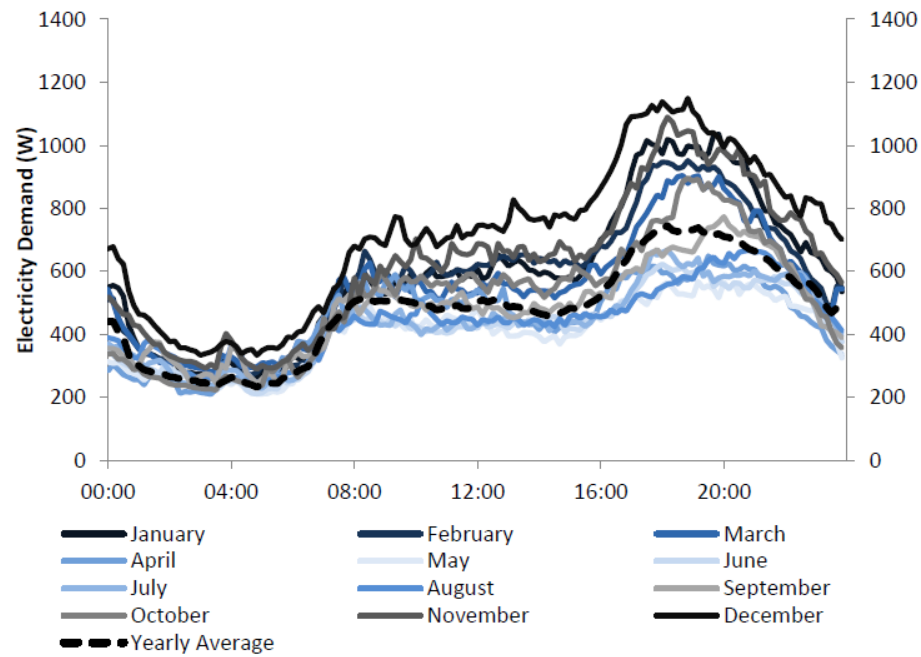
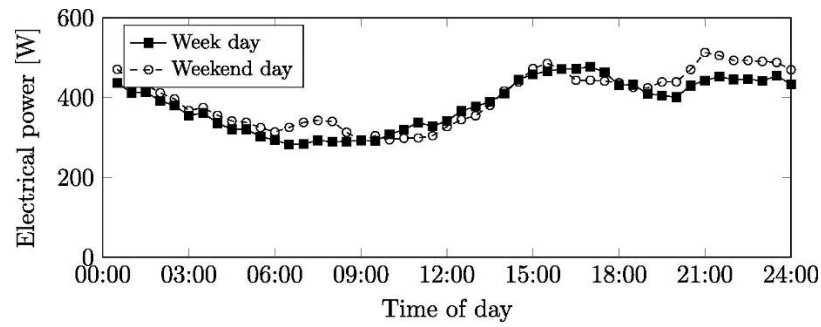
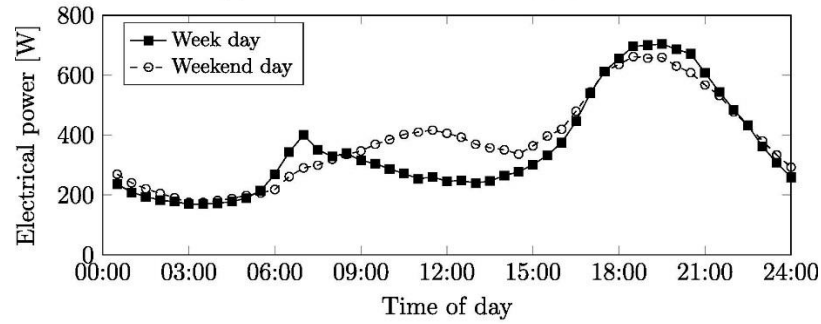


Figure 2.5 Monthly average hourly electricity demand from a household in UK [21]

It should be noted that this trend may differ according to geographical locations which receive higher solar radiation with higher ambient temperatures; the demand will be affected because of air conditioning needs. In Cyprus, for example, winter demand is similar to that of the UK but summer demand is generally flatter with high overnight and day time consumption by the air conditioners [22]. Figure 2.6 shows average electricity demand profile in Cyprus. Although summer demand profile is flatter than the UK, winter demand profile is similar with the rest of the European countries.



(a) Summer domestic load profile, Cyprus.



(b) Winter domestic load profile, Cyprus.

Figure 2.6 Domestic load profile in Cyprus [19]

2.3 PV cells and PV/T collectors

Among renewable energy technologies, PV technology requires very low maintenance and repair expenses, which make this technology operationally one of the most low-cost options, however, it also comes with the highest capital costs [23]. A PV/T system consists of a solar collector and PV module as major components. In the PV system, the PV module constitutes 70% of the capital costs. Therefore, the lifetime of the PV module, power degradation and initial power output are the main parameters for determining the payback period of the whole system [24].

The first practical solar cell was developed by Bell laboratories in 1954, in response to the need for a long-term energy supply for a space program project. In that year an efficiency of 6% for a single-crystal silicon cell was reported. Research into PV technologies has been ongoing for years, resulting in improvement of the performance

of cells and the significant reduction of capital costs [25]. Recently, the solar PV technologies as outlined in Figure 2.7, have been classified as fourth generation. Hybrid, organic and dye-sensitized materials are not included in this thesis. This study focuses on crystalline silicon type and thin film PV cells.

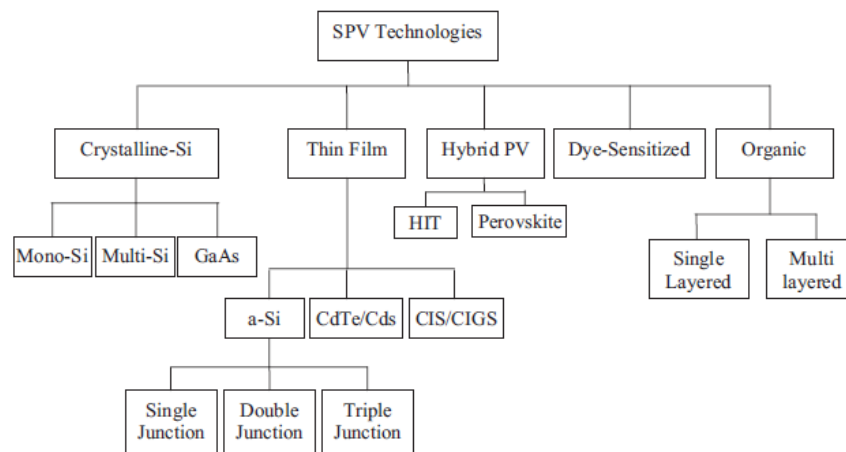


Figure 2.7 Solar power cell materials [25]

Given their higher conversion efficiency and low cost, crystalline silicon cells have become the most preferred cell material in the world market. However, the use of thin film cells has the potential to become widespread. The percentage of production of thin film cells and silicon cells in world markets is given in Figure 2.8. In addition, there is an increasing interest in hybrid technologies, which are a combination of crystalline and thin film technologies. Discrimination amongst types of silicon cells is a result of the purity of used silicon. It means that in order to obtain a more electricity conversion efficiency, silicon molecules should be more perfectly aligned. Purification of silicon is an expensive process and so, in practice, conversion efficiency must be considered alongside other important parameters like cost and

space. Two forms of crystalline silicon forms are available on the market. They are monocrystalline and polycrystalline solar cells.

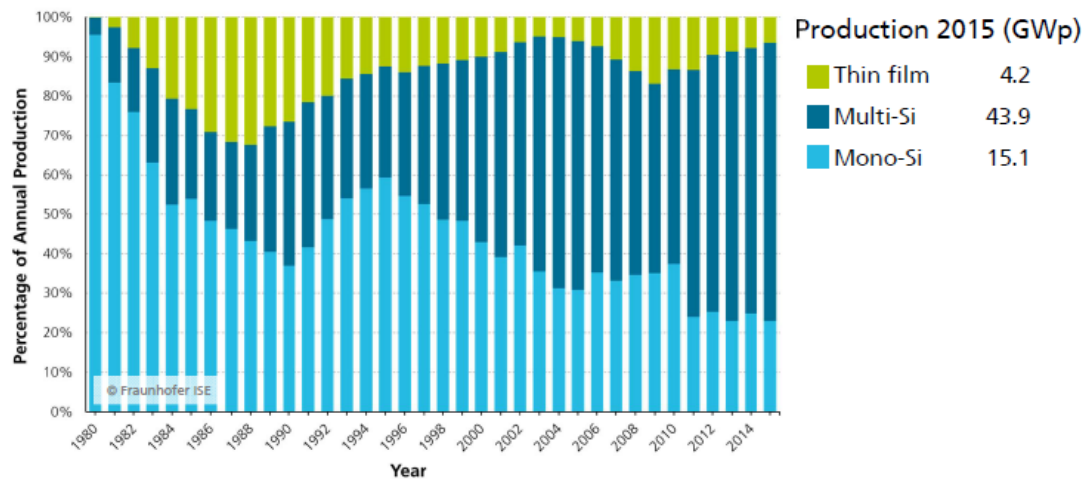


Figure 2.8 Percentage of production of thin film and crystalline silicon cells in world market [26]

2.3.1 Crystalline Cells

Crystalline type cells are classified into monocrystalline and polycrystalline cells. Monocrystalline, also known as single-crystalline silicon (single-crystal-Si), photovoltaic panels are old and relatively more efficient panels. Cells are made up of a single silicon crystal, and are more efficient but expensive as compared to the polycrystalline and thin film cells. Mono-Si cells are made out of cylindrical silicon ingots. In order to provide optimum performance and low cost, four sides of the cylindrical ingots are cut out to make silicon wafers which give mono-Si solar panels their characteristic rounded corner appearance [27]. The advantages of these solar panels are high conversion efficiency rates which vary between 15-20%, space-efficiency and a relatively longer lifespan. However, mono-Si has some disadvantages

such as high capital cost caused by the production process, as a result of which manufacturers are looking for other materials such as polycrystalline which has a lower production cost in comparison.

The first polycrystalline silicon (poly-Si) was seen in the market in 1981. In its production process, molten raw silicon is poured into a square mould, which is cooled and cut into perfectly square wafers [27]. Although these are less efficient than mono-Si cells, p-Si cells are widely used in manufactured modules due to their lower production cost, resulting in a subsequent high demand thereof amongst investors.

2.3.2 Amorphous silicon cells

Thin-film solar cells are developed by thin layers deposited on the substrate such as glass, polymer or metal through a sputtering process [25]. Thin films can easily be integrated into building components due to their flexible and lightweight structures. However, the conversion efficiency of thin film solar cells is still a cause for concern. Thin-film modules have reached efficiencies between 7–13% in prototype and production modules operate at about 9%.

Amorphous silicon (a-Si) cells, on the other hand, are ideal in the s-ORC applications with given advantages. First, they are widely used and are the most developed of thin film cells. In China more than 2.0 GW of a-Si cells can be produced annually [28], and the global market is likely to reach USD 5310.2 million by 2025 [29]. Second, a-Si cells are flexible. The flexibility can prevent fractions (breakup) of cells. For example, in a PV/T system solar cells are coupled with aluminium, copper or other metals in practice. Temperature fluctuations cause the metal to expand and shrink which results in the production of tension stress on the cells. Silicon cells are weak in

their ability to carry this thermal stress resulting in their shortened lifespan, but a-Si cells may be more reliable on this issue because of their lower thermal resistance. This flexibility facilitates its ease of integration into solar collectors. Third, a-Si cells can use metals as substrate. The thermal resistance of the PV/T module is reduced. Feasibility of PV/T with a-Si cells has been demonstrated for medium temperature applications by more than 15 month continuous operations [30]. Finally, a-Si cells profit at high operation temperature because of thermal annealing. They have a lower conversion efficiency compared to other silicon based cells [31], which is partially caused by the Staebler-Wronski (S-W) effect [32]. The photons with the photoelectron energy below the optical gap of a-Si cells directly pass through the a-Si layer and cannot be absorbed by the intrinsic layer, thus contributing essentially no photocurrent and limiting the conversion efficiency of the a-Si cells [33]. However, the S-W effect can be reversed. At a high temperature, the performance of a-Si cells is degraded fast but reaches a higher efficiency at the degraded steady state [34]. It is reported that the positive power temperature coefficient is obtained in the long term operation. For example, at the degraded steady state, the a-Si cell efficiency ratio at 90 °C, 50 °C and 25 °C is about 0.8, 0.73, 0.64 [35]. Notably, the positive power temperature coefficient is desirable in the medium temperature applications such as PV/T-ORC systems.

Figure 2.9 shows a sample of a-Si cell deposited on stainless steel. Another advantage of a-Si is that it is relatively cheap in comparison to mono-Si and poly-Si. A disadvantage, however, is its low conversion efficiency compared to other silicon based cells which also results in a lower space-efficiency [31]. Energy yields of a-Si, hybrid and c-Si PV modules, in hot months the average efficiency of thin film modules

is greater than in cold months. Table 2.1 provides a reference, showing a comparison of the cells.

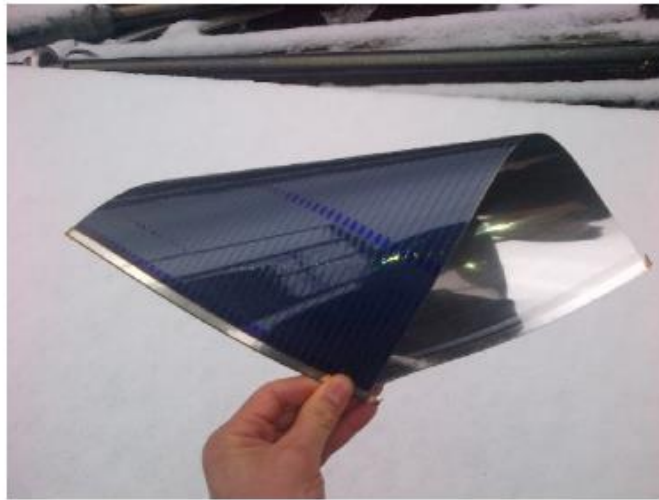


Figure 2.9 A sample of a-Si cell deposited on stainless steel [36]

Li et al. [36] studied a hybrid solar power generation system which consists of a PV/T based on a-Si cells and presented a-Si cells are a good choice for solar cogeneration systems. These cells allow for higher operating temperatures to be reached with a high resistance temperature during performance. Figure 2.10 shows variations in efficiency of c-Si and a-Si cells at various operating temperatures.

Table 2.1 Comparison of PV cells [27]

	Mono-Si	Poly-Si	A-Si	CdTe
Module efficiency	15-20%	13-16%	6-8%	9-11%
Required area for 1 kWp	6-9 m ²	8-9 m ²	13-20 m ²	11-13 m ²
Lowest price*	0.75 \$/W _p	0.62 \$/W _p	0.69 \$/W _p	

Performance drop under high temperature	Drops 10-15% at high temperatures	Less than monocrystalline	Tolerates extreme heat	Low impact on performance
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*Lowest price values are taken from reference website, it was collected in 2013.

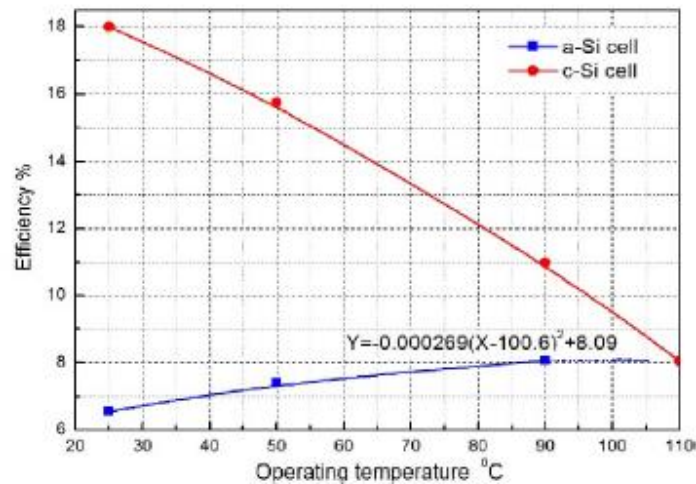


Figure 2.10 c-Si and a-Si cells efficiencies at operating temperatures [36].

2.3.3 PV/T collectors

Hybrid PV/T systems are used for sending away heat from PV cells into working fluid. Regarding to classification of the PV/T systems generally it is based on type of working fluid and configuration. In these systems, collectors which are actually a special kind of heat exchangers. According to Kalogirou [37], there are two types of collectors; stationary and concentrating collectors. Various solar collectors are available in the market, comparison of collectors' specifications is given in Table 2.2.

Table 2.2 Solar thermal collectors [38]

	Type of collector	Absorber Type	Concentration ratio	Operation temperature °C
Stationary	Flat plate	Flat	1	30-80
	Evacuated tube	Flat	1	50-200
	Compound parabolic	Tubular	1-5	60-240
Single - axis tracking	Linear Fresnel reflector	Tubular	10-40	60-250
	Parabolic trough	Tubular	15-45	60-300
	Cylindrical trough	Tubular	10-50	60-300
Two-axes tracking	Parabolic dish reflector	Point	100-1000	100-500
	Heliostat field	Point	100-1500	150-2000

In a recently published review paper by Jia et al. [39], classification of common types of PV/T collectors are given as Flat-plate PV/T system which has three branches, the first one is air type PV/T, the second one is liquid type PV/T and the third one is bifluid based type PV/T.

Next to the flat plate type PV/T is the concentrator type PV/T systems. Concentrator type PV/T collectors comprise of PV cells but, the light from reflectors are focused on it. The concentrators can be parabolic, Fresnel and dish types. For high-temperature applications, large area reflectors focus the light on a smaller absorber area to increase temperature and enhance the radiation intensity achieved by solar cells. Efficient concentrator PV/T collectors can produce high current compare to conventional PVs, and their applications are common because of this advantage despite their expensive

cost compare to flat plate collectors[40]. Additionally, it should be known that concentration PVs are expensive but required area would be small [41]. Since concentrated solar energy only partly converted to electricity, the rest of the energy increases the temperature of cells, this high temperature can damage the PVs [42]. So, cooling of the PVs is a vital process for this kind of systems. There are many details in the papers on this issue, but the project will not use concentrator PV/T type system.

Within the PV/Ts the last type can be categorised as various applications and modifications which includes phase change material integrated, nanofluid used, working fluid evaporation type, thermoelectric generator integration and solar still types.

Although as given that the PV cells can be used in many types of collectors, such as concentrator type collectors, this subsection only provides a detailed explanation of water flat plate collector (FPC) systems. Because air-cooled FPC are not used in solar cogeneration and concentrated type collectors need sun tracking system and would not be useful for building applications.

Flat plate collectors are used commonly in building applications such as solar water heating, space heating by integrated with a heat pump and because their constructions are simple, no need to sun tracking and maintenance costs are low. Numerous design principles have been developed but the most investigated one is using water as a heat transfer unit because their overall efficiencies achieve higher than air systems [8]. Thermal and electrical performance of PV/T collector is highly depending on design and operation considerations like collector designed geometry and its material [43], water circulation mode and size of the PV and absorber material areas [44], [45] , and application temperature [46].

Flat plate collectors are made of fluid tubes which is connected to a flat plate absorber that collects solar radiation and transfers heat energy to the tubes. In order to reduce both radiation and convection losses, a glass cover protects the absorber, whilst conduction losses are limited using thermal insulation. Figure 2.11 shows PV/T classifications. This represents in (1) sheet and tube, (2) channel, (3) free-flow and (4) two-absorber types [47]. The most common type is the sheet and tube type collector. For this reason, the main components of this type are analysed and modelled in this thesis.

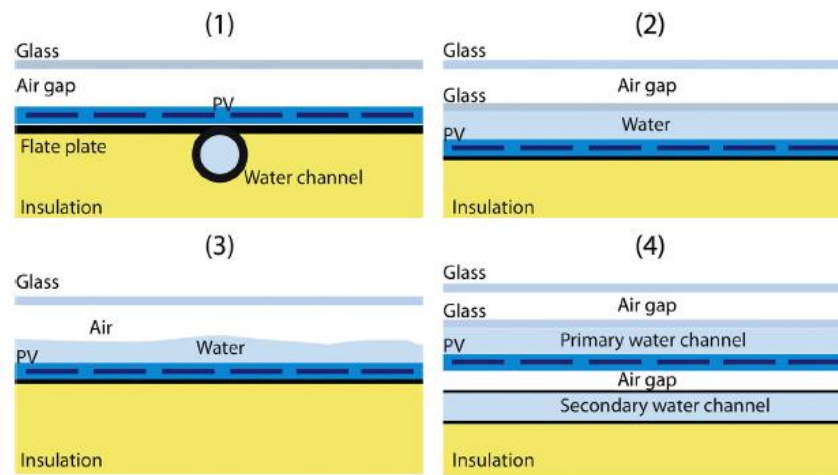


Figure 2.11 PV/T classifications [47]

The sheet and tube type collectors are easily coupled with PV modules as there are various heat and energy analyses to be found in key literature. Figure 2.12 shows the heat transfer mechanism of the PV/T system. It can be seen that part of the absorbed heat is lost through convection and radiation to the environment.

For uncovered collectors, the absorber is in direct contact with the ambient and heat losses will be significantly higher and fluid temperature cannot reach the desired high temperature. Thus, a transparent cover is used to prevent direct contact between the

absorber and the environment. The air gap should be thin in order to maintain the insulation ability of the air. Commonly, the gap is considered between 15 and 40 mm and the collector closed with glass generally has a maximum area of 3 m^2 given the structural and wind loads on the cover [48]. The glass most commonly used is a thickness of 3-4 mm [49]. Apart from glass, there are some lighter and stronger materials which can be used for providing a greater aperture area such as polyvinyl fluoride and polycarbonate [47].

As aforementioned, uncovered collectors have lower performance than covered ones because of higher conduction and radiation losses. However, this higher thermal performance causes lower electrical performance. So it is important to investigate the performance of PV/T components as a function of environmental and technological parameters [47].

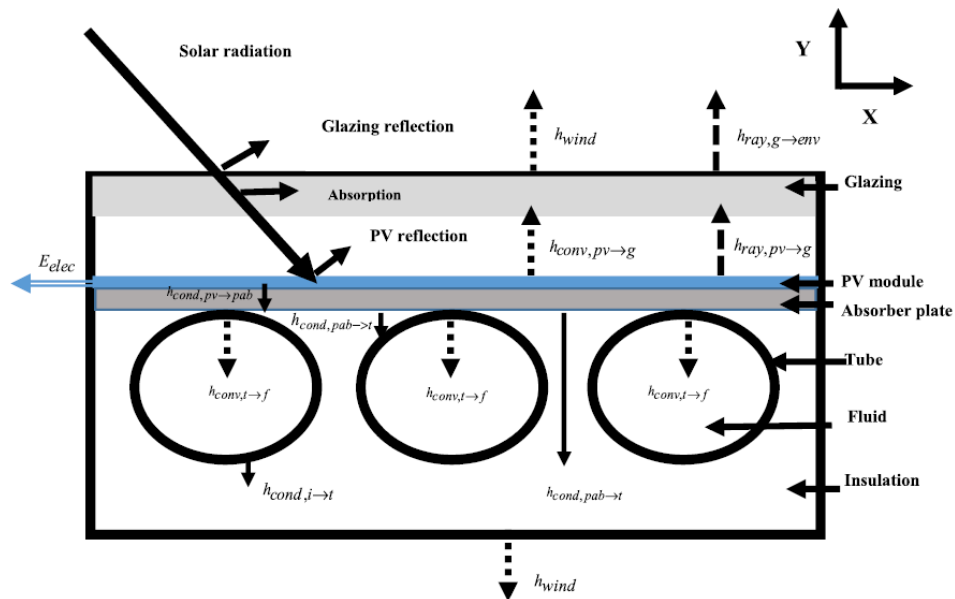


Figure 2.12 PV/T water collector coupled with PV module [50]

PV technology is implemented in the PV/T in order to convert solar power to electricity. The most common technology used in the PV market is c-Si which may be defined as high thickness cells nearly 0.2-0.5 mm which provide higher efficiency compared to thin film cells. Commercial c-Si cells have an electrical efficiency between 13-22% and a temperature coefficient of power (β) from 0.3% to 0.5%/K [51]. Another cell technology is that of a-Si cells with lower temperature coefficients of approximately 0.2%-0.3%/K and from 200 to 600 nm thickness of silicon deposit [52].

In a PV/T system, solar heat is collected by the photovoltaic elements to the fluid. As an absorber, its important property is thermal conductivity which should be as high as possible to allow heat flux from cell to fluid. Other important property of an absorber is specific heat capacity which should be low for allowing fast reaction times in regard to fluctuating weather conditions [47]. For these reasons, PV/T absorber material is usually metallic. Table 2.3 highlights the main features of absorber materials. Steel and polymers are rarely used. It is also important to mention that PV modules and absorbers have to connect and the most common method for coupling them is a gluing technique detailed in study by [47].

Table 2.3 Specifications absorber plates [47]

Material	Thickness (mm)	Density (kg/m ³)	Conductivity (W/mK)	Heat capacity (J/kgK)
Copper	~0.3	8920	380	350
Aluminium	~1	2700	160	900

2.3.4 Models for simulating the flat plate collectors

Solar collectors can be coupled with heat storage or heat pumps. It is known that the dynamical characteristics of each component in these systems have an influence on thermal performance. Proper design has to involve a control strategy for analysing energy savings during operational time. Moreover, solar collectors are heavily influenced by varying external conditions, such as solar radiation changes over time or cloud cover which may prevent solar radiation instantly. The performance of solar collectors can therefore be affected by numerous factors. Thus, given the dynamic behaviour of solar collectors, modelling with them provides a great opportunity to develop regulation strategies. In addition, a numerical model allows for the reproduction of dynamical performance which in turn offers more reliable external tests [53]. In order to present dynamical behaviour of solar collectors, several numerical models have been developed. There are some different approaches to analysing solar collector performance.

Steady state model: This model neglects the thermal capacitances so only gives accurate results with long time averaged data. It is defined as the simplest model and this approach is very useful for performing long term simulations [54].

Lumped capacitance model: This model is the simplest dynamical model which considers inertia effects but with global thermal capacitances. Experimental studies reveal that if the input variations are small this approach gives accurate results. This means that solar irradiation, water inlet and air temperature fluctuations should be small. [55] compared experimental data with simulation data and results showed in days with partial cloud cover, simulation results are not accurate as given in Figure 2.13. However, when constant solar irradiation was present, the collector outlet water temperature was accurate.

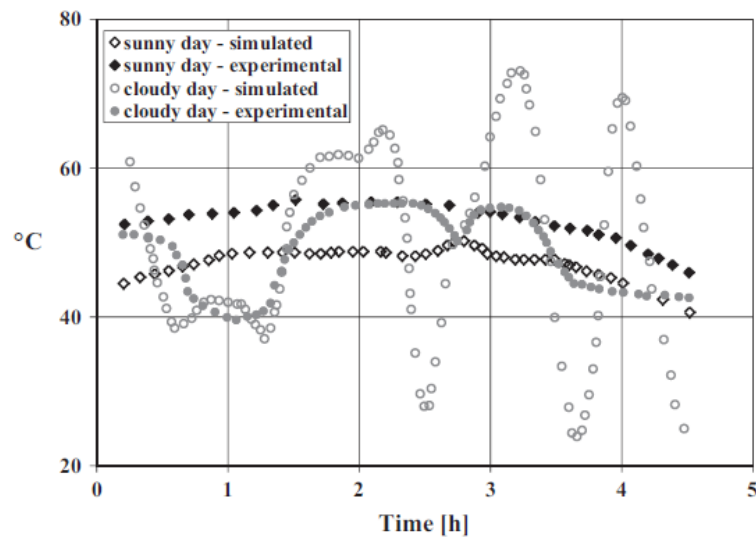


Figure 2.13 Comparison of the collector out temperatures for cloudy and sunny day [55]

Discretized model: The lumped model was modified by Klein [56] by considering the water temperature variation in the solar collector. For this model, the collector is discretized along the fluid flow direction and unsteady energy balance equations are solved for each element. Figure 2.14 shows a 2n-node model which both thermal capacitances of the elements of absorber and glass were considered. Results from some linear partial differential equations for the 2n-node were used to later develop 3n-node and 4n-node models using various numerical methods such as the Runge-Kutta method.

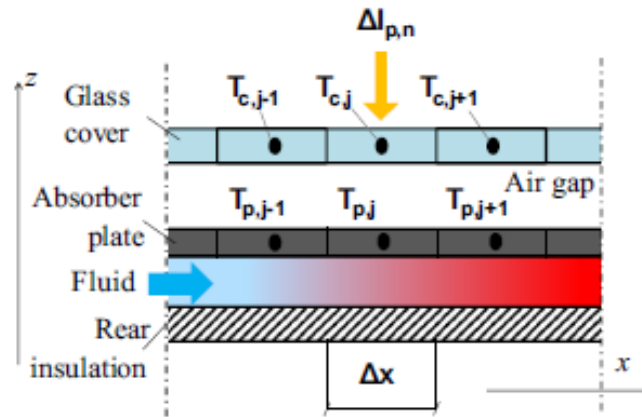


Figure 2.14 2n-node model [54]

Zima and Dziewa [57] studied a mathematical model with thermal capacitances of tube and fluid. In this model, the authors consider the capacitances of only fluid and tube and an adapted model is given in Figure 2.15. As an extension of this model, an analysis of the inertia effects to include insulation and air gap as categorized as 5n-node model was undertaken [58],[59]. Their model was discretized and equations calculated by the implicit finite difference solution method. In conclusion, the discretized model introduces water temperature variation along the solar collector.

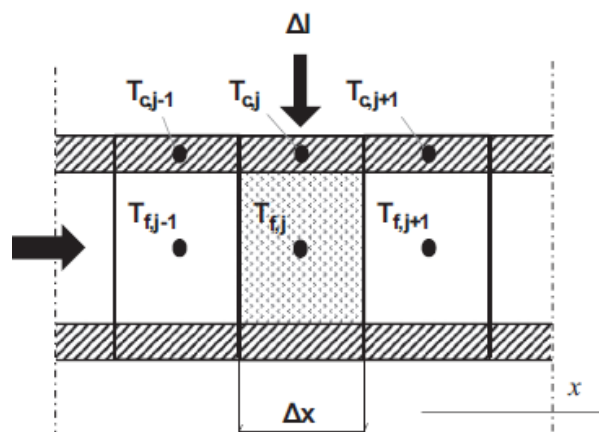


Figure 2.15 Discretized control volumes [57]

This method is very useful for predicting an accurate water outlet temperature from the collector. In order to demonstrate daily energy yield, according to [60], there is no requirement to analyse the more complex models because results of 1 node and 3 node models are very similar. However, outlet fluid temperatures can be different. So, it is important to reveal working and design conditions before analysing. Kutlu et al. [61] followed same methodology but their solution model was improved with the 6 node analysis (cover glass, PV plate, absorber plate, tube, working fluid and insulation). Fig. 2.16 shows the schematics of the study to show considered nodes.

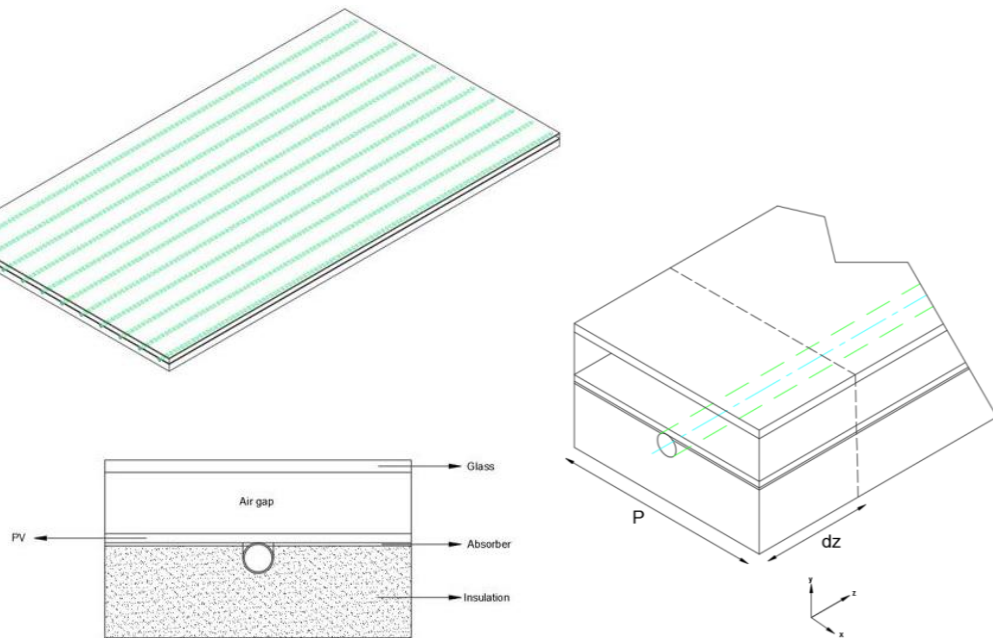


Figure 2.16 Schematic of the PV/T collector with 6 node analysis [61].

By this methodology, temperature variation of the all components in the collector can be seen. Fig. 2.17 is given as an example of the analysis when 2 m^2 collector is coupled with the 60 L water tank. The subscripts ab, f, g, ins and am indicate absorber, fluid, glass, insulation and ambient, respectively.

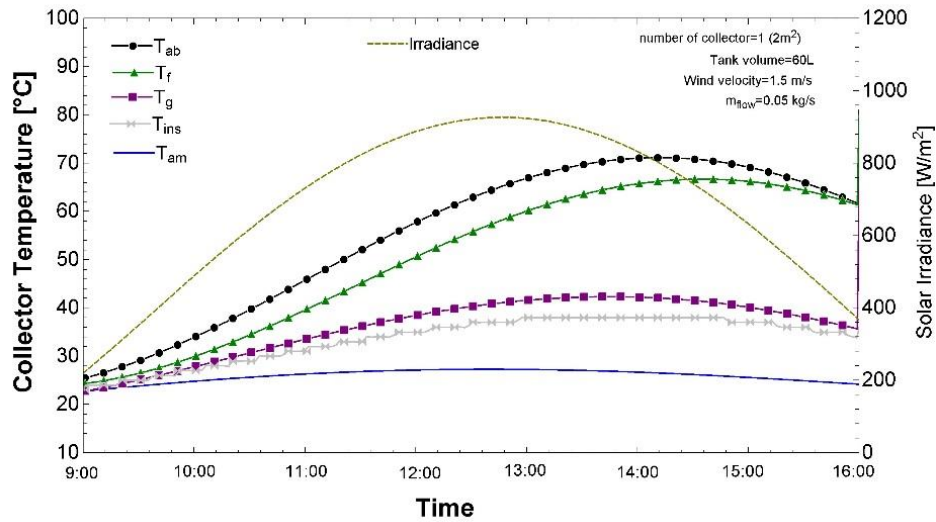


Figure 2.17 Temperature change of the nodes by time [61].

2.4 Solar collectors for thermal applications

2.4.1 Evacuated tube collectors (ETC)

Flat plate collectors are known as the most popular type of collectors due to their simple design and low maintenance costs. But they have a relatively low thermal efficiency and collector outlet temperature. It is reported, however, that evacuated tube solar collectors have considerably lower heat loss and investment cost than those of flat plates [37]. Moreover, the tubular design of ETCs eliminates the tracking of sun problems because not necessary. Classic FPCs are useful for sunny days and their performance decreases in colder days, so their performance is significantly influenced by weather conditions such as clouds, wind speed and so on. In contrast, the ETC has better thermal performance and is suitable for even cold conditions [62], [63].

Evacuated tube collectors are made of parallel glass pipes. Each pipe constitutes two concentric tubes, first an inner tube and then an outer tube, as shown given in Figure 2.18. The inner tubes are coated with a material and the outer tubes are transparent.

The transparent outer tube allows solar rays to pass through and these are subsequently absorbed by the inner tube. The reflection properties of outer tube and the inner tube have to be minimum to obtain a higher performance in order to hold the heat inside the inner tube, evacuation helps the solar energy to go inside however without allowing happens transfer so heat is not lost. [62].

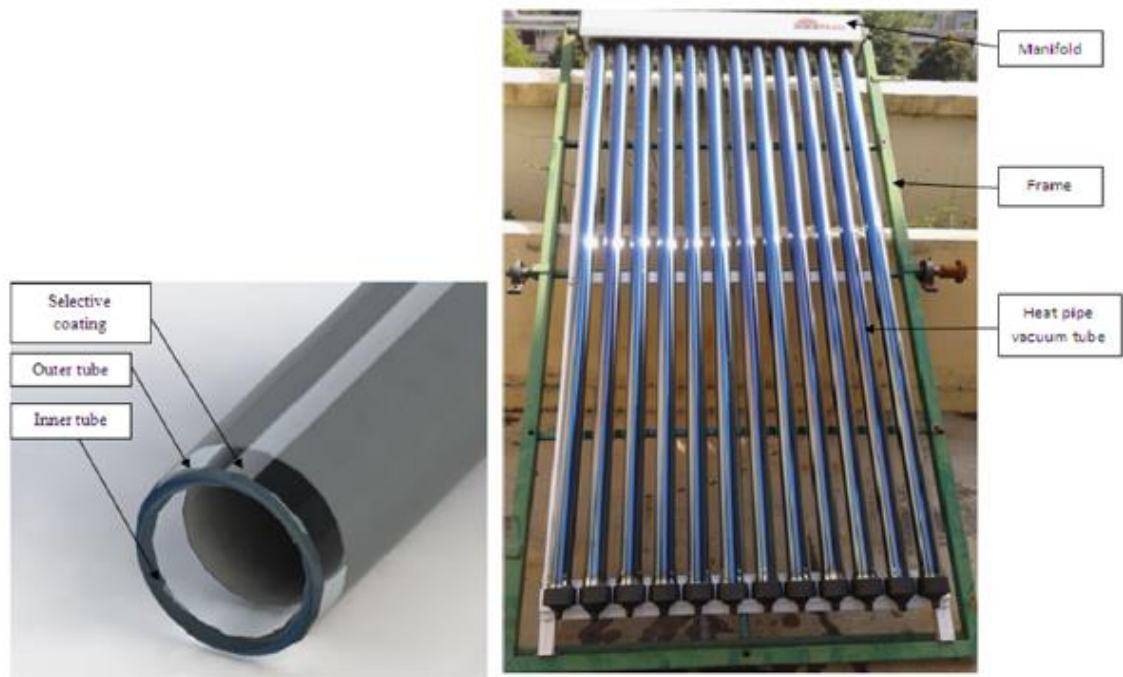


Figure 2.18 Evacuated tube collector[62],[64]

2.4.2 Parabolic trough collectors

Parabolic trough collectors (PTC) are linear solar concentration systems consist of long mirrors which are parabolic-shaped and a receiver tube to heat fluid. Solar rays are concentrated onto the receiver tube, a schematic is given in Figure 2.19, thus, the solar heat can be absorbed by the fluid. A glass envelope is often placed around the heat collection element to improve the collector's thermal efficiency by restricting the convection losses. The gap between tube and glass envelope is evacuated to improve

efficiency. PTC's concentration ratio is commonly 50, and the heat transfer fluid temperature can reach up to 400°C. [65].

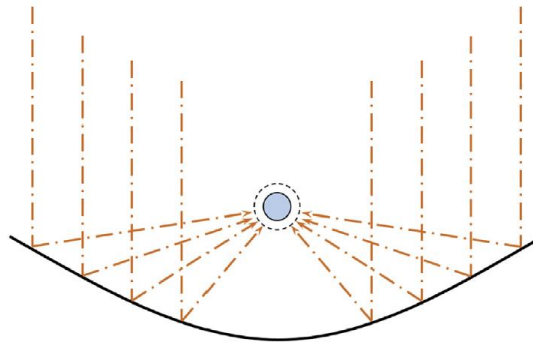


Figure 2.19 Schematic of a parabolic trough collector [65].

2.4.3 Linear Fresnel collectors

Linear Fresnel collectors (LFC) are described as line-focused concentrating systems. Differ from the PTCs, LFCs are sum of many mirrors that possible to moved them independently to focus the solar radiation onto a receiver, instead of a continuous parabola-shaped reflector like PTCs. Its schematic is given in Figure 2.20. This collector allows single-axis tracking. Because of the architecture, single optic LFCs systems' concentration ratios are slightly lower than PTCs, and LFCs are generally known as lower optical performance but higher cost savings potentials [65].

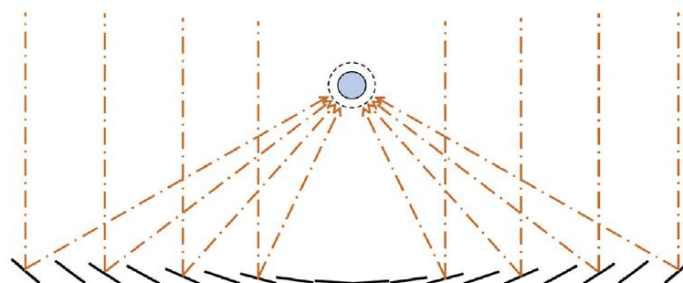


Figure 2.20 Linear Fresnel collector [65]

2.4.4 Compound parabolic collectors (CPC)

One of another trough-type technologies is compound parabolic collector that concentrates solar radiation on a receiver tube. Schematic is given in Figure 2.21. This shape's advantage is that CPCs can operate without continuous tracking. CPCs are sometimes having low concentration ratios (<5) and they can be used for medium-temperature applications (up to 200°C). CPCs also can be used with linear Fresnel collector for effective concentration ratio improvement.

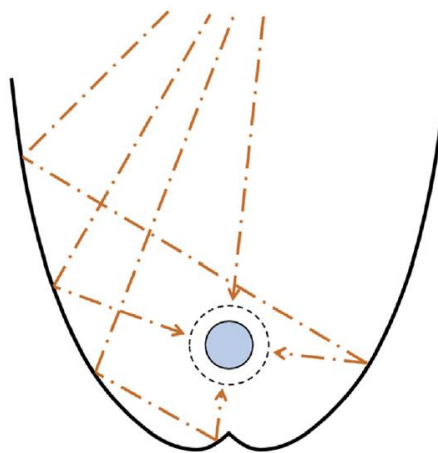


Figure 2.21 Compound parabolic collectors [65]

2.5 Heat Storage Units

The thermal heat storage units are the main parts of the solar-driven power generation, heating and cooling systems. A heat storage unit is used in order to combine the PV/T system and the ORC. Solar radiation is absorbed by a solar collector as heat and then transferred to the fluid. The heat is carried by the working fluid and is used to charge a thermal heat storage tank. This charged heat can then be used at a later time. Designing an effective thermal heat storage unit is a priority, given that it provides continuous operation and for some applications it needs to store heat effectively for

use at night [12]. It is also reported that these systems have a great influence on performance, system cost and reliability [38]. The process within a heat storage system involves three steps: charging, storing and finally discharging of heat. Schematics of these processes are presented in Figure 2.22.



Figure 2.22 Complete cycle of a thermal energy storage [66]

Using a thermal energy storage unit in an ORC system provides significant benefits. The main benefit of using solar ORC against PV is revealed when adopting cost-effective thermal energy storage in place of electrochemical batteries. It is reported that thermal energy storage unit costs are lower than electric batteries and exhibit a much longer usable lifespan. In order to design a selection of the thermal energy storage (TES) component, the consideration criteria are given [65]:

- Energy density should be high because this reduces storage size.
- The heat transfer rate should be high to facilitate rapid charging and discharging.
- There must be easy control and flexibility.
- The selection of fluid must take into account the need to avoid environmental hazards, and;

- Low system cost will provide advantage over electrical batteries.

Thermal energy may be stored in various ways, such as sensible and latent heats. A general view of the main techniques in thermal storage is given in Figure 2.23. Heat storage methods can be categorised as sensible heat and latent heat storage systems.

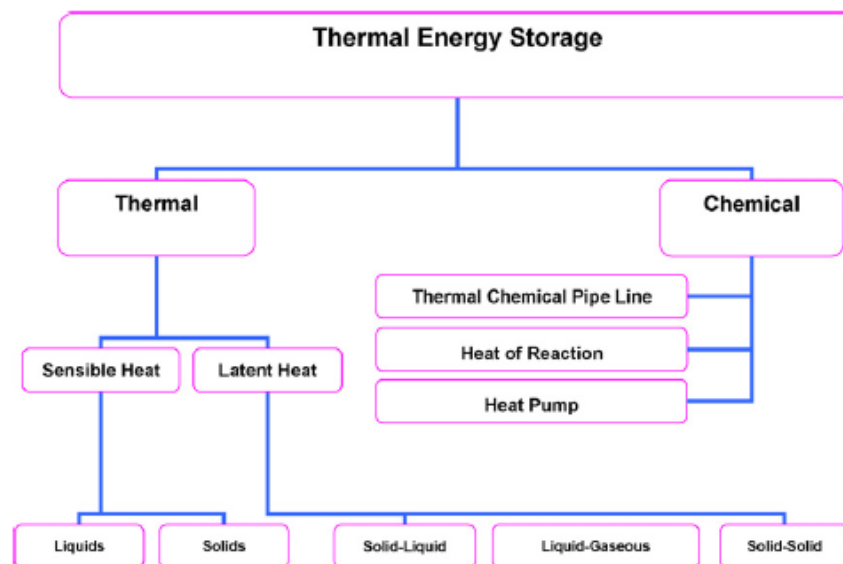


Figure 2.23 Thermal energy storage types [67]

2.5.1 Sensible heat storage (SHS)

Sensible thermal energy is stored by increasing the temperature of a liquid or solid. SHS systems utilize the heat capacity of material and changes in material temperature that occur during the processes of charging and discharging. The storage material can be air, water, oil, or brick. Each of these materials has certain advantages and disadvantages. However, when the material is chosen, the main consideration is the heat capacity and the available space for storage. Some of the used materials for SHS systems are tabulated in Table 2.4. Used materials for CSP plants are given in Table 2.5. When selecting a material, the first consideration must be high thermal capacity

and low expense. Other properties also need to be considered for selecting a suitable material, including specific heat, thermal conductivity and diffusivity, density and finally chemical stability [68]. It is possible to store sensible heat in liquid or solid materials. Water is deemed to be the best SHS liquid available because it is cheap and has a very high specific heat.

In this thesis, flat plate collectors are cooled by water and a water tank is used for heat storage. It thus takes as its focus the use of water for heat storage medium in thermal energy systems.

Table 2.4 Selected liquid and solid sensible heat storage materials [67]

Medium	Temperature	ρ (kg/m^3)	C_p ($J/kg \cdot K$)
Rock	20	2560	879
Brick	20	1600	840
Water	0-100	1000	4190
Engine oil	Up to 160	888	1880
Ethanol	Up to 78	790	2400
Isopentanol	Up to 148	831	2200

Table 2.5 Materials used in sensible heat storage in CSP [69]

	λ (W/m · K)	C_p (J/kg · K)	ρ (kg/ m^3)	Cost (€/kg)
Molten salt	0.15-2.0	1500	900-2600	0.63
Cofalite	1.4-2.1	800-1034	3120	0.01
Castable ceramics	1.35	866	3500	4.5
High temperature concrete	1	916	2750	0.08

As a sensible heat storage material, water has excellent thermodynamic properties, such as high density and very good specific heat value. Moreover, water is non-toxic, cheap and easy to store without serious precautions. The working range of 0–100°C makes it a suitable heat storage material especially in housing applications.

Containers for hot water stores are made of stainless steel, copper or plastic. It is possible to place equipment inside the container, like electric heating elements, heat exchanger coils, stratification devices and baffle plates. To reduce heat loss, the outside of the store is generally insulated with materials that have a low conduction property. Heat loss from the water storage should be sufficiently low that the system performance and stored heat level does not fall considerably. The efficiency of most systems with water storage can be increased if a high level of thermocline is built up in both the discharging and charging periods. For instance, a large stratification level in the hot water tank in the solar water heating system provides that the collector inlet temperature maintains lower and the collector performance becomes higher, and that

the required auxiliary heating is decreased to a low level as a result of high temperatures of the water rejected from the top of the water container [70].

In order to reduce the required volume for the water tank, its storable heat capacity must be as high as possible. Regarding the heat capacity, volume of the tank and the thickness and geometry of the insulation influence the space needed for it. The rate of heat loss water tank is calculated by tank's heat loss coefficient and the temperature difference between the water and the ambient temperature. The heat loss coefficient can be determined by heat transfer theory by combining convection and conduction heat transfers which are related with the conductivity and thickness of the insulation and surface area of the tank. Especially, the theoretically determined heat loss coefficient shows the rate of transferred heat through the insulation material. Furbo's [70] detailed study elaborates further on some particular cases of water tank storage. However, it is crucial to consider thermal stratification in water tanks, and the next section will discuss this in detail.

2.5.1.1 Thermocline of water tanks

The theory of the thermal stratification in tanks has been developed over many years. Researchers have conducted numerous analytical and numerical simulations of thermocline in the water tanks. These studies have shown that thermal stratification can significantly enhance energy storage functioning [71].

To explain this phenomenon, it should be assumed that the inlet fluid of the tank adjusts mixing level until the density of the inlet fluid starts to approach the density of the adjoining fluid. Because of the buoyancy effect and gravity, water of differing temperatures will settle at different heights in the tank corresponding to the difference

of density; lower-density water of high temperature rises to the upper layer, and cold water sinks to the lower layer. Thus, stratification separates the cool and warm fluids like a barrier and maintains the density and temperature gradient. In Figure 2.24, three different degrees of stratifications are seen in the tank. In (a) stratification is high, in (b) it is moderately stratified and in (c) is fully mixed (uniform).

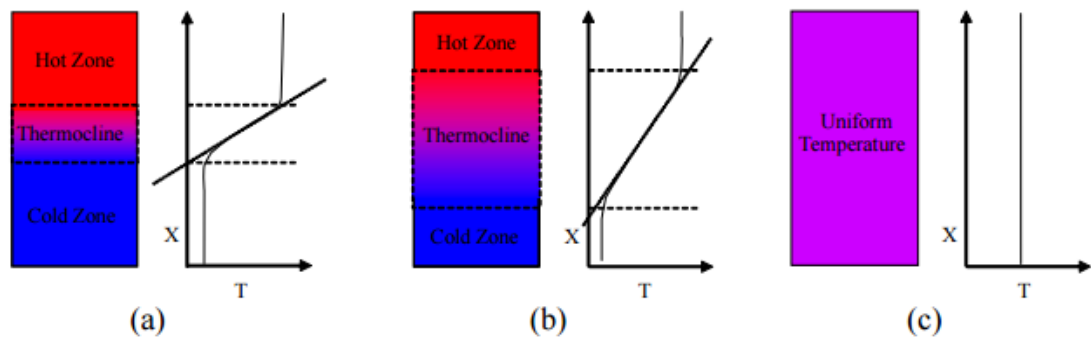


Figure 2.24 Different degrees of stratification [72]

Maintaining the stable thermal stratification level in the water tank is important. In order to achieve thermal stratification in the tank, there are several methods that can be applied [73]. The first is to heat the vertical walls, creating a thermal boundary layers that draws hot water into the upper parts. The second is to use a heat exchanger inside or outside of the tank that can facilitate the exchange heat between the water and the circulating fluid. The final approach is to allow a flow of inlet water into the tank at a suitable height.

Stratification in the water storage is affected by certain parameters, including flow rates, the thickness of tank insulation and inlet and outlet geometries of the tank. Lavan and Thompson [74] studied the effects of inlet and outlet geometries on the TS found that the inner geometry of the tank has a strong influence, while the shape of the outlet

is not important. To assess the effects of baffle plates in the tank, Shah and Furbo [75] conducted an experimental study that explored the effect of various plate designs. It illustrated that energy quality in the tank is significantly related to its geometry.

The design of the water tank should be investigated in detail to simulate stratification and geometrical effects on the individual heat storage system. However, an analysis of the whole solar cogeneration system reveals the effects are decreased but at the same time cannot be neglected for detailed study.

2.5.2 Latent Heat Storage (LHS)

LHS units use the physics of the phase alteration of materials. Solid to liquid and liquid to solid phase transition is generally used for melting or solidify the materials. The heat supplied during melting is transported to the LHS material; when the heat is released, the material solidifies at a fixed temperature and the heat can be reused. In a LHS system, the materials are specified to as phase change materials (PCM). In Table 2.6, the typical range of melting enthalpy and temperatures of common material classes used as PCM are tabulated. Although many of them have been studied as storage material only a few of them have been commercialized [76] because of some problems like corrosion, phase separation, low heat conductivity and long-term stability. These problems have not been completely solved yet. Generally the PCMs are selected as availability, suitable melting enthalpy in required temperature range and of course its cost [77].

Table 2.6 Melting temperatures of common latent heat storage materials

Material	Melting Temperature (°C)	Melting enthalpy (MJ/m³)
Water-salt solutions	-100-0	200-300
Water	0	330
Paraffins	-20-100	150-250
Salt hydrates	-20-80	200-600
Sugar alcohols	20-450	200-450
Carbonates	400-800	600-1000

The main specifications for a PCM are a suitable melting temperature and a large melting enthalpy. But, in most of the applications the requisites increase, and these are grouped as technical, physical and economic requirements. The relevant physical properties are repeatable phase change, good thermal conductivity and minimal supercooling while the technical ones are low vapour pressure, chemical stability, and low volume change. Finally, economic requirements include cost and lifespan of the raw materials to be applied as PCMs [78].

2.5.2.1 Applications of PCMs in solar energy systems

Usage of PCM units in the solar systems have attracted researchers' attention for years. Its large heat storage capacity and low heat loss potential make PCM studies popular both in power generation systems and domestic heat storage applications.

In houses, heat storage units for DHW applications are generally achieved by sensible heating of a water cylinder given that the mentioned units are popular to supply instant hot water for residents. Therefore, the heat capacity of the storage is restricted by the available space, which is related to the volume. At this point, the PCM can be a solution because PCMs can store sensible and latent heat in DHW applications which can results in a smaller volume for the same heat capacity storage. To investigate the potential of the PCM in the DHW applications, many researchers have built models and carried out testing. Researchers mainly focused on the various types of PCMs properties and configurations with the water tank. Mehling et al.[79] prepared an experiment and presented that adding PCM at the top of water tank increases storage capacity and compensate heat losses. Talmatsky and Kribus [80] conducted a modelling study of PCM with water storage and analysed the performance considering the domestic hot water demand, given in Figure 2.25. They concluded that using PCM has no significant energy efficiency advantage for end users, and they explained that reheating of the water by the PCM during night time is responsible for increased losses to the environment.

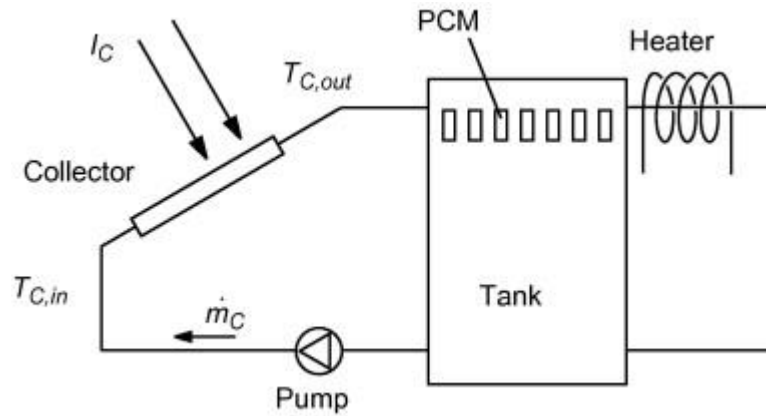


Figure 2.25 Schematic view of PCM integrated solar DHW system [80]

Padovan and Marzan [81] reported that optimization is very important for using PCM in DHW system; thus, they proposed a generic algorithm optimization to evaluate sensitivity of the parameters, such as tank geometry and PCM melting temperatures. Kumar et al. [82] prepared an experiment to test the effect of water flow rate and inlet temperature of the heat transfer fluid into the water tank with PCM. They concluded that the addition of PCM capsules can increase the stratification level of the water tank which is a useful parameter for heat storage applications. Finally, Shirinbakhsh et al. [83] comprehensively analysed the effect of DHW demand on performance of solar domestic hot water systems. They concluded that embedding PCMs in the storage tanks is not a promising solution and performance is dependent up on the operating conditions. Therefore, they recommended that the DHW demand profile should be taken into consideration [84].

For solar power generation systems, PCM heat storage is a promising solution because of the intermittent nature of solar energy. This intermittent feature causes the power supply fluctuations even in large scale solar power plants. Packed bed storage system is an option for the solar thermal systems to store the energy during its availability and supply that stored energy at the time of requirement [85]. Figure 2. 26 shows layout

of the packed bed storage. The storage medium stores the thermal energy in latent and sensible forms. The working mechanism of heat transfer is to supply or extract heat to or from the storage by using heat transfer fluid. Manfrida et al. [86] used same type of heat storage unit in their power generation system and presented generalized energy equations of PCM in detail.

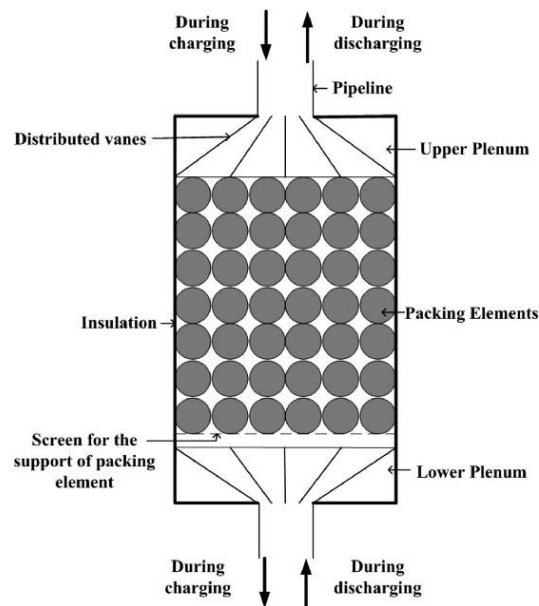


Figure 2.26 Schematic of a typical packed bed storage unit [85]

PCM modules can be placed in the tank as cylindrical shapes as well. Pirasaci et al. [87] presented a simple thermal model for the design of a latent heat storage unit for direct steam generation. They considered the effectiveness of the storage as a design criterion and investigated the influence of the other parameters. Figure 2.27 shows PCM tubes in a storage tank.

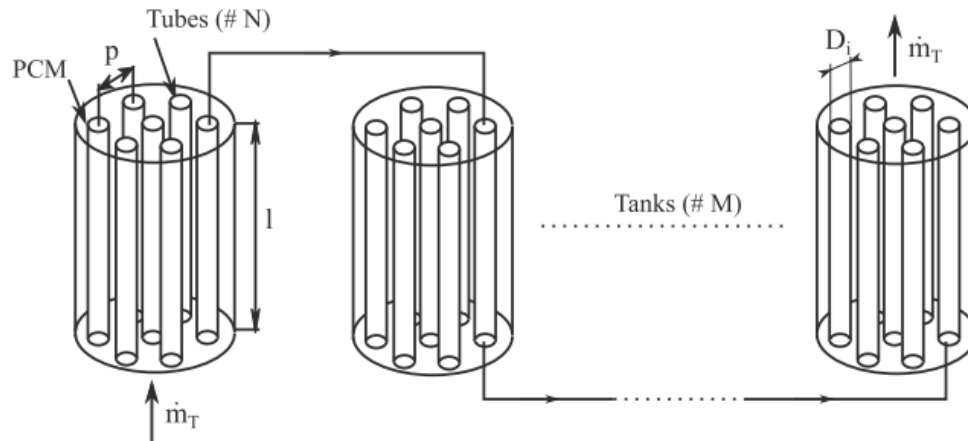


Figure 2.27 Layout of the PCM tubes in the storage tank [87]

2.6 Organic Rankine Cycle

The steam Rankine cycle is the fundamental principle for the most important large-scale heat-to-power transformation systems. An Organic Rankine Cycle (ORC) shares the same principle, but it uses an organic liquid instead of water as the working fluid. Although water has excellent thermal properties, it is not possible to use it as working fluid in renewable heat applications such as solar, geothermal, waste heat etc. because these thermal sources cannot provide high enough temperature for steam Rankine cycle. When low temperature/grade heat is available ORC system is applicable. Figure 2.28 presents a comparison of the working temperatures of water and some organic fluids. The figure shows that organic fluids have less latent heat compare to steam, a specification that makes them available for applications with low temperature and relatively limited heat gain. The advantages of the ORC over the steam cycle are that the ORC applications, the evaporations process occurs at lower temperatures and pressure, and there is no risk of blade erosion at the outlet of the expander [88].

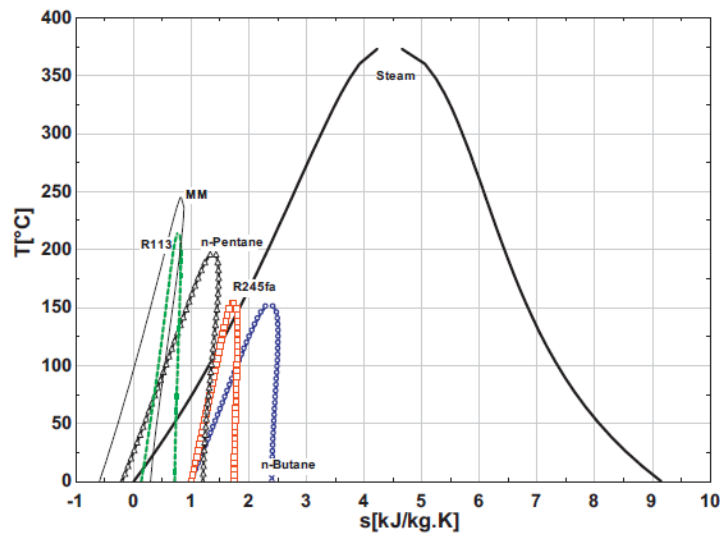


Figure 2.28 T-s diagrams of some working fluids [88]

A basic ORC system has the following components: a pump, an evaporator/boiler, an expander and a condenser. Figure 2.29 presents a schematic and T-s diagram of a basic ORC. Working fluid enters the pump as a saturated liquid at condensing pressure after which its pressure is increased by the pump to evaporation pressure level. The pump consumes power at a level based on pump efficiency. The pump efficiency, in turn, depends on the fluid inlet and outlet conditions. In the evaporator, heat transfer occurs and heat taken from the heat source and transferred to the working fluid. Depending on the type of heat source and applications, evaporator design and performance may vary. The working fluid leaves the evaporator at evaporating pressure as saturated vapour or superheated vapour and moves into the expander. The expander produces work and decreases fluid pressure to the condensing pressure. The expander is the most important component of the ORC cycle. Its isentropic efficiency and mass flow rate are the key factors in overall performance. Pitch point assumptions are applied to determine the condenser and evaporator temperatures for theoretical studies.

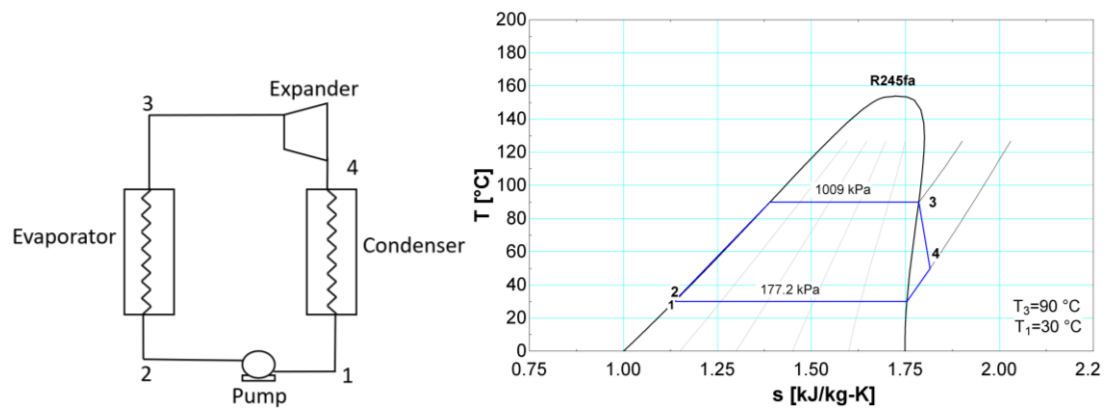


Figure 2.29 Schematic view and T-s diagram of a simple ORC unit

Used working fluid in the ORC system plays a significant role because it is related to system thermal performance and economics of the plant. Several researchers studied the effect of the working fluid and operating conditions selection on system performance.

Tchanche et al. [89] studied 20 fluids and their analysis tested the following parameters for a low temperature solar ORC system: efficiency, GWP, ODP, mass flow rate, pressure ratios, flammability and toxicity. They concluded that R134a is the most suitable refrigerant in small-scale solar ORC applications. The authors selected an evaporating temperature of 75 °C, condensing temperature of 35°C, and isentropic efficiencies of the turbine and pump of 0.7 and 0.8, respectively. These conditions and system setup are similar with the target of the current project, which has meant reviewing of this prior study particularly important. The system schematic of the study is presented in Figure 2.30. The solar collector charges the water heat storage unit and an interface water cycle carries the heat to the evaporator to release the working fluid. As mentioned, the authors found R134a to be the most suitable working fluid for a low temperature solar ORC.

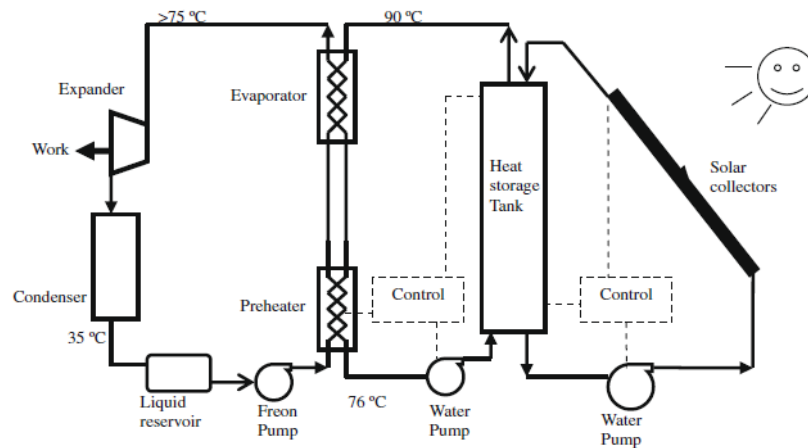


Figure 2.30 Schematic of the solar ORC system [89]

R245fa is commonly used and effective working fluid for low temperature solar systems. Wang et al. [90] designed an experimental system working with R245fa and they used both evacuated and flat plate collectors. Average expansion power output of 1.73kW and 3.2% power generation efficiency is observed with the flat plate collector conditions. Twomey et al. [91] tested scroll expander model in a small ORC and used an expander model. They found 676 W maximum instantaneous power produced by system with 3.47% ORC efficiency. Declaye et al. [92] presented experimental characterization of an open-drive scroll expander which used R245fa as working fluid. They found maximum expander efficiency of 75.7% and maximum shaft power of 2.1 kW. And the maximum cycle efficiency of 8.5% is observed when evaporating temperature was 97.5 °C. Lastly, Yang et al. [14] conducted an experimental study of a 3 kW ORC using open-drive scroll expander with R245fa fluid. System thermal efficiency and maximum expander shaft output power were found 5.92% and 2.64 kW, respectively.

The key component of the ORC is the expander. In the large-scale power generation system various turbines are used for different conditions and purposes. However, in low temperature solar ORC systems expanders are generally used.

2.6.1 Models for system control

Solar heat is a variable energy during a day. Received solar energy reaches higher levels under high solar radiation and it reduces by the time. In order to avoid interruption of the solar heat input, heat storage units are adapted in the systems. However, heat storage can store finite energy that means heat source temperature for the ORC may change during operation. The effect of variation of heat source temperature requires some control methods in the model, as the ORC alone cannot prevent the unstable trend. Hu et al. [93] explained and compared the control strategies in their paper. Adjustable inlet guide vanes of the turbines, given in Figure 2.31, and evaporation pressure are considered as control variables to adapt the system to variable heat source temperature and flow rates.

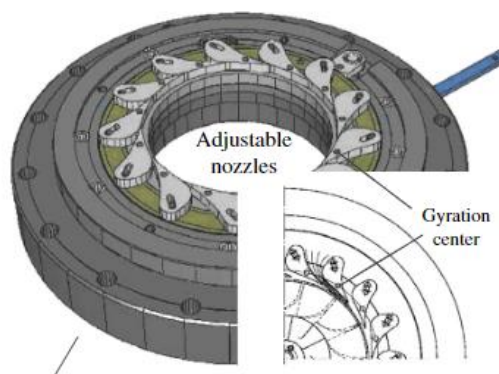


Figure 2.31 Adjustable nozzles in radial inflow turbine [93]

Regarding to conventional Rankine cycles, there are two types of operating control strategies suggested in the literature, such as constant pressure and sliding pressure operations. For sliding pressure operation, evaporating pressure is the main controlled parameter to maintain steady expander operation [94]. Gabrielli [95] built an off design model to determine optimum design parameter for varying geothermal heat

source. Fu et al. [96], [97] studied the effect of heat source temperature variation on the performance by using sliding pressure operation strategy. Their study considers the economizer performance which only includes single phase heat transfer. Figure 2.32 shows T-s diagram of their system operation. Kutlu et al. [98], [99] used same methodology in their simulations to adjust system output according to heat source (water tank) temperature.

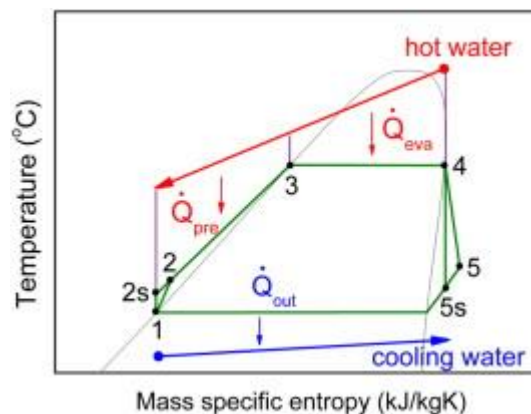


Figure 2.32 T-s diagram of sliding pressure operation [96]

Controlling expander rotation speed is also common in the literature studies. Ibarra et al. [100] adjusted both working pressure and expander rotation speed to match with load requirement. Quoilin et al. [101] defined the pump rotation speed and the expander rotation speed as control variables. Adjustment strategies were compared under variable heat source conditions and they suggested that optimisation is required.

2.6.2 Expanders for ORC applications and their models

ORC systems can be used effectively in low temperature heat sources, especially for small scale applications. They are suitable for various low temperature applications such as solar thermal, geothermal and waste heat recovery power generation systems.

Given the nature of these sources, the useful heat fluctuates during operation time and the system components must adapt to its operating regime to ensure optimal system operation [102]. Alternatively, components may be operated in off-design conditions without significant performance reduction.

The key component of the ORC unit is the expansion device. This produces work and reduces fluid pressure to condensing pressure. Thus, the expansion device is the most important component of the ORC cycle and its isentropic efficiency and mass flow rate are key factors in the overall performance. For small-scale units, an efficient and cost effective expansion device must be chosen. It is known that scroll machines produce low noise, little vibration and are high efficiency machines and therefore scroll compressors have been used for years in many applications. Thus, scroll expanders are quite popular for small scale ORC applications. Moreover, tolerance of the off-design operation makes this type of expander the best option for fluctuating heat source applications such as solar ORC applications [103].

In order to simulate scroll expanders, three types of modelling have been presented and discussed in a review paper prepared by Song et al. [104]. They state that empirical and semi-empirical models are more stable and time-saving in computations. Furthermore, a geometric model can reflect the performance of the scroll expander during the whole working process. However, the geometric model is more time-consuming in computation. Dickens et al.[102] compared three different modelling methods of ORC in their experimentally validated study, namely, constant efficiency method, polynomial-based method and semi-empirical method. They presented that semi empirical models are the most reliable for off-design operation of the ORC. Lemort et al. [105] defined three types of expander models in similar conclusions. The first is the empirical (black box) model, which is suitable for dynamic simulations but

unable to produce extrapolation beyond calibration range. The second is the semi-empirical (grey box) model, which is suitable for designing steady state models. The third is the deterministic (white box) model, which is the most powerful tool for optimizing the system and the one based on a comprehensive description of the expander with differential equations. However, this model requires that certain parameters – such as geometrical ones – be measurable with a high degree of accuracy.

Empirical model: The empirical model for a positive displacement expander includes two polynomial regressions: isentropic efficiency and filling factor. Some studies provide detailed equations and some constants for this model [92][106]. In order to build a generic non-dimensional performance curve of the scroll expander, a fitted expression of the efficiency is modelled by selected input parameters. Equations and parameter definitions were inspired by Pacejka's equation. Mathematical significance is described in Figure 2.33. This figure shows a typical performance curve of the scroll expander and the empirical correlations were built based on given parameters.

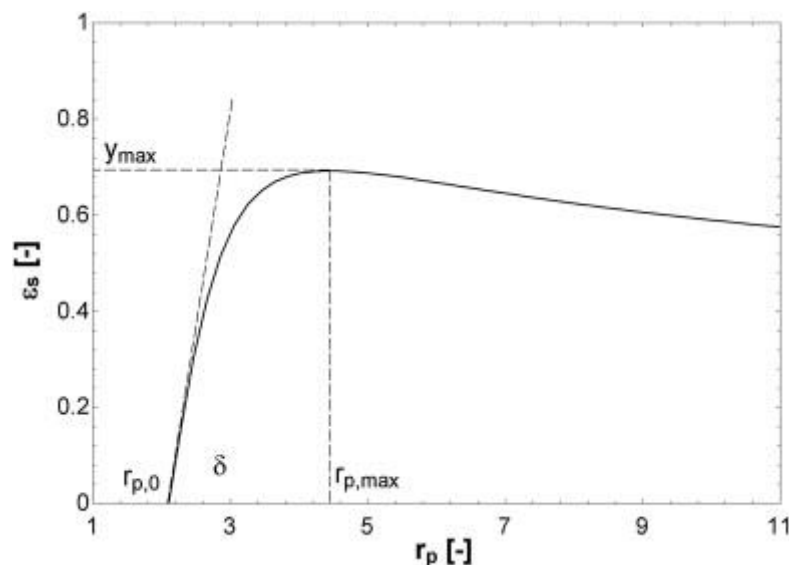


Figure 2.33 Mathematical significance of the parameters used in empirical method [92]

This model's empirical equations based on three parameters: supplied pressure of the expander, pressure ratio of the inlet and outlet conditions and rotational speed. The results of the empirical equations of isentropic efficiencies are given in Figure 2.34.

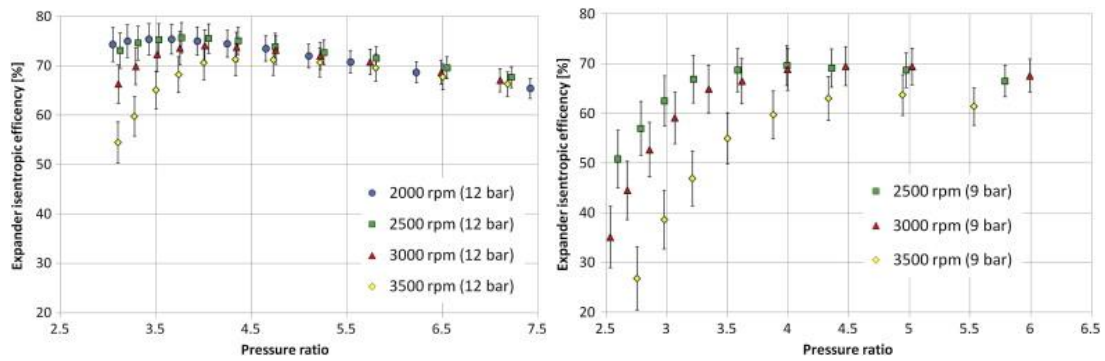


Figure 2.34 Expander isentropic efficiency variation of the empirical model [92]

Semi empirical model: Expander semi empirical model was developed by Lemort et al. [107] and the most preferable model recently. Guiffarida [108] improved the model for all refrigerant to use. This expander model is used to model the expander performance under the varying conditions. The main difference from empirical model is that this model can be applicable for ORC systems independent from working fluid or load. Thus, the model can be updated. The schematic expression of the model is given in Figure 2.35. As can be seen from the figure, all thermodynamic processes were explained in stages. In order to evaluate the mass flow rate, it is required to predict the exhaust temperature and shaft power. Model results were compared with the experimental results with an open-drive scroll expander. The maximum deviation was found 5%, which illustrated that this model has a good accuracy [104]. It offers a simulation speed, modelling accuracy and extrapolation capabilities [109]. Moreover, this semi empirical model can be adapted into screw expander models [110], [111],

reciprocating piston expanders [112] and root expander [113] by using the same formulation processes of each stages .

This model consists of several hypothetical stages.

- 1) Adiabatic pressure drop in supply side ($su \rightarrow su,1$): This process is related with pressure drop between the expander inlet and suction chamber.
- 2) Isobaric supply cooling down ($su,1 \rightarrow su,2$): There is a heat transfer occur between the fictitious envelope and refrigerant.
- 3) Internal leakage ($su,2 \rightarrow ex,2$): Entered mass is divided into two parts in the expander.
- 4) Adiabatic and reversible expansion to the adapted pressure ($su,2 \rightarrow in$):
- 5) Adiabatic expansion at constant volume ($in \rightarrow ex,2$):
- 6) Isobaric exhaust cooling-down or heating-up ($ex,1 \rightarrow ex$): Heat is exchanged between the fluid exiting the expansion chamber and the isothermal envelope.

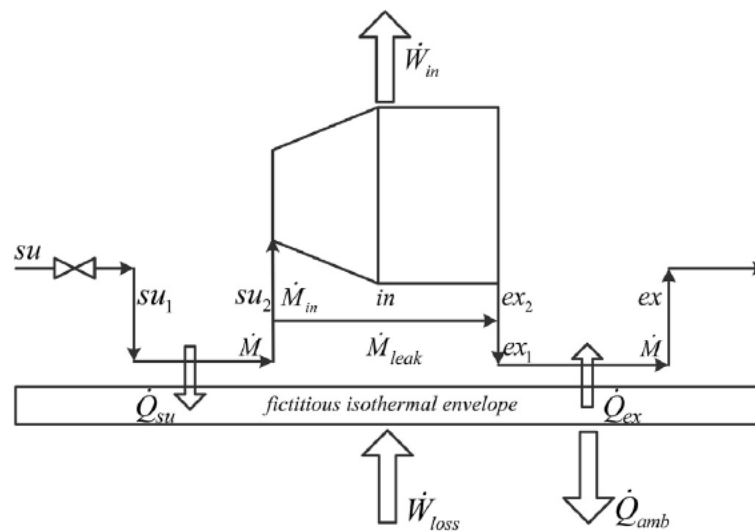


Figure 2.35 Semi-empirical scroll expander model schematic given by Lemort et al. [107]

Although this semi-empirical model is quite popular, dependence of studied expander geometry and used refrigerant make this approach is inflexible for various applications such as different capacity and operating conditions in ORC systems. Some papers have investigated how to improve the model by upgrading parameters. Giuffrida [108] looked at improving the flexibility of the model to capable to model with all working fluids . He suggested an approach to changing the working fluids by modifying the heat transfer coefficients. Since other parameters are not dependent on the working fluid, the study modified only the heat transfer coefficients, with all other parameters, which are dependent on scroll geometry, remaining the same as the reference paper. Lemort et al. [114] experimentally showed that the leakage area can be varied according to inlet pressure but presented another empirical equation for the area calculation under different conditions. Similarly, Mendoza et al. [115] presented an empirical equation for leakage area when using ammonia as working fluid. Giuffrida [116] used a load dependent leakage area equation and Ziviani et al. [117] proposed a modification of the leakage area using some constant values, pressure difference and rotation speed, however, parameters would still need to be determined via an experiment. Gao et al. [118] experimentally tested two scroll expanders with different displacement volumes. They reported that isentropic efficiencies of the expanders were different even under same operating conditions. In order to use different capacity or brand of expanders, the researchers need to determine their machine's parameters first, then can built a model for parametric studies. Twomey et al. [91] successfully adapted modelling for their scroll machine using same methodology and simulate a solar ORC system. Ayachi et al.[119] developed and characterized a semi empirical model for high pressure operations using R245fa. Miao et al. [120] enhanced its expander model by scroll geometry and defined leakage area and other parameters

from experiment. Recently, Muye et al.[121] characterized and model of the scroll expander using R134a refrigerant to operate with any working fluid.

Geometric model: Geometric model of the scroll expander consists of three main sections. Forming scroll geometry is one of the main sections to reveal geometric features. Thermodynamic model deals with the refrigerant side. Mechanical model is to analyse frictions and rotation. Sections can be summarised as: Scroll geometry, thermodynamic model and mechanical model. Scroll geometry is one of the main factors effecting the scroll compressor and expander performances. A reliable and simplified model is necessary to merge with a thermodynamic model. Compared to scroll expanders, compressor machines have been more studied and provide us significant knowledge which can be applied for designing the scroll expanders [122]. The geometric model consists of equations to identify scroll profiles and calculates volumes of the chambers and leakage areas according to orbiting angle. To build an accurate and flexible mathematical model, some of the geometric features need to be known. Scroll geometry is shown in Figure 2.36 as an example. The scroll expander geometric model is investigated in Chapter 6 in detail.

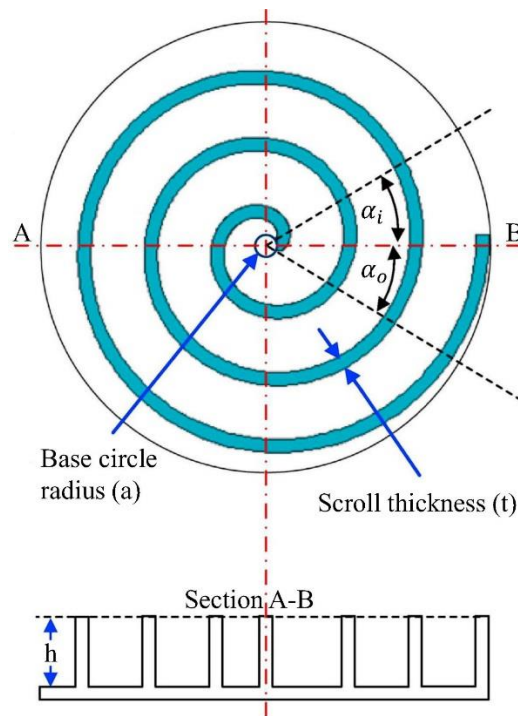


Figure 2.36 Geometry of a constant wall thickness scroll [123]

Scroll geometry and its dynamic mathematical model was presented by Morishita et al. [124]. They defined the chamber volume as a function of one rotation of orbiting scroll. Their model inspired to next models. Yanagisawa et al. [125] established the first full scroll compressor model in 1990. This model allows to analyse each chamber separately thus, their model was convenient to establish a thermodynamic model of a scroll machine. Later, Chen et al. [125] improved the model considering different involute angles. There are many papers have been published for scroll expander geometry but papers from Bell must be studied in detail from his thesis [126], model development [127], optimisation [128] and analytic solutions of scroll machines [129].

2.7 Solar ORC system applications and configurations

Utilization of solar energy for power generation systems has been mostly conducted by concentration collectors. As concentration collectors use mirrors and lenses to focus solar heat, high temperature of operating conditions are achieved in power generation plants. Concentration solar collector integrated systems can be studied from the excellent review papers [130], [131]. A simple indirect solar power plant layout is given in Figure 2.37.

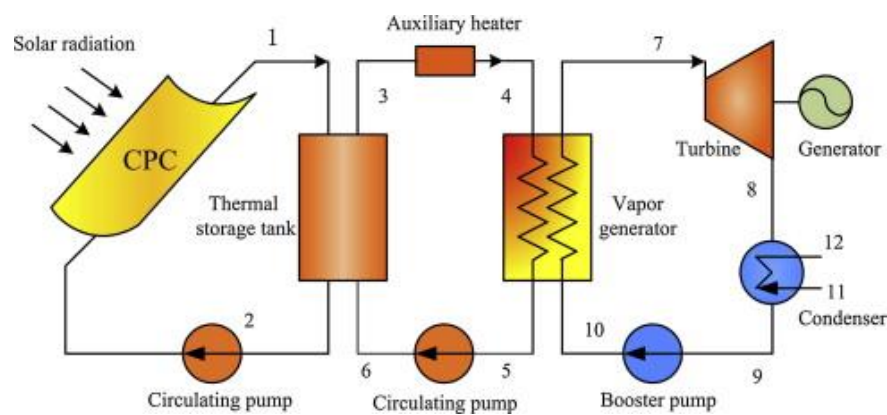


Figure 2.37 Schematic diagram of solar ORC [94]

Solar energy is transferred to working fluid, probably a thermal oil. This heat is stored in a heat storage unit but at the same time or later, this heat is used to evaporate the working fluid in ORC block. There are many configurations found in literature such as combined with geothermal source, waste heat source and biomass heat sources.

For the usage of solar fields in the power plants, designers used many configurations according to their needs. One of the common solar power plant configurations is a two-tank molten salt storage plant. Its schematic is shown in Figure 2.38a. This configuration also used in operational solar power plants. Figure 2.38b shows an example of subdivided solar power plant.

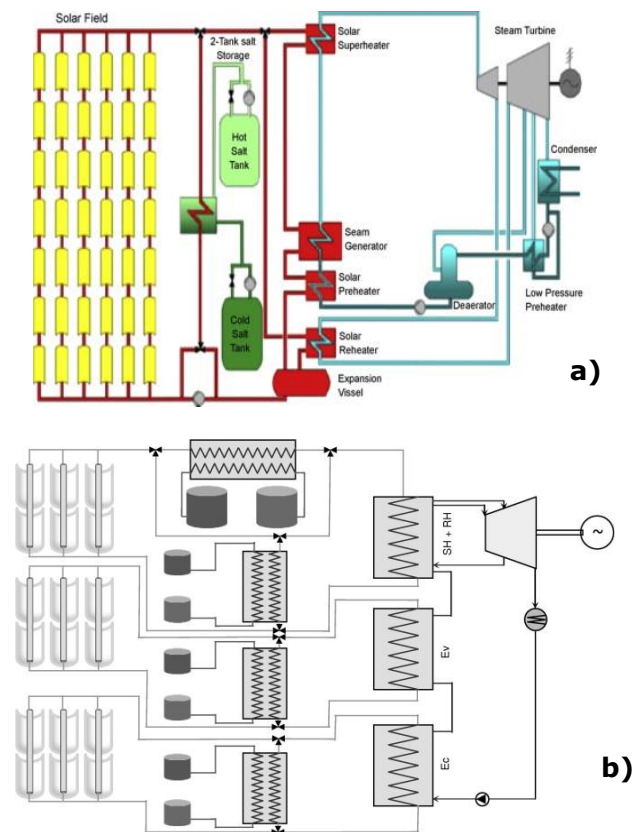


Figure 2.38 Schematics of **a)** two-tank molten tank plant [132] and **b)** power plant with subdivided solar field [133].

The development of solar ORC (s-ORC) systems are restricted by its efficiency, which is significantly lower than main stream PV efficiencies. Studies show that concentrating collector used systems can reach higher operating temperatures and high operating temperatures result with better overall efficiency. A summary of some solar ORC studies and their overall efficiencies are given in Table 2.7.

Table 2.7 Summary of some solar ORC studies

Study	Type of collector	Operating temperature	Overall efficiency	Notes
[134]	Parabolic trough collectors	80-200 °C	3.5-8 %	
[135]	Parabolic trough collectors	135-147 °C	6-12 %	combined system (geothermal-solar)
[136]	Parabolic trough collector	80-300 °C	3-11 %	
[137]	Concentrating photovoltaic	90- 150 °C	6.35-10 %	CPV-ORC, 10.52 %
[138]	Concentrating PV (no collector calculations)	80-130 °C	10-15 %	
[139]	Evacuated compound parabolic concentrator	-	12-13.5 %	
[140]	Evacuated flat-plate	Oil temp. 180 °C	9-10 %	
[141]	Evacuated flat-plate collector	95-145 °C	4.4-7.4 %	
[141]	Evacuated-tube heat-pipe collector	95-145 °C	2.5-4 %	
[91]	Evacuated tube type	85-120 °C	3.47 %	

[142]	Evacuated tube collectors	180-200	5.5 %	
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Plenty of studies have been carried on solar ORC systems, however, usually parabolic through collectors have been preferred: Wang et al. [94] studied the off-design performance of the solar ORC system under variation of the ambient temperature and heat transfer fluid flow rates of vapour generator. They concluded lower environment temperature could improve the system performance. Chacartegui et al. [143] examined a 5MW parabolic trough plant with ORC power block with a thermal storage unit. They conducted off-design and cost analysis and findings show that the investment cost for direct thermal energy storage systems is a 17% lower than the investment cost for indirect storage system. Tzivanidis et al. [144] studied a parametric analysis of a solar ORC plant by using parabolic trough collectors in order to optimize the system energy and financial outcomes. Their results showed that increasing the total collecting area decreases the solar thermal efficiency.

However, for small scale power generation applications in buildings, the concentrators such as compound parabolic concentrators and parabolic trough collectors requires direct beam radiation and may have a high cost for the tracking system. To overcome the challenges, evacuated flat plate collector (EFPCs) have been recently used for s-ORC applications. It is a relatively new technology and promising with high thermal efficiency even under low ambient temperature and solar radiation levels. Studies on the EFPCs in district heating and s-ORC applications have been reported. Calise et al. [140] designed and simulated a CHP system by using EFPCs. They reported that EFPCs can operate higher than 50% thermal efficiency in summer and around 20% in winter. These efficiencies are promising values considering the high operating

temperatures. Freeman et al. [141],[145] simulated this collector for s-ORC applications and conducted that their CHP unit can considerably provide electricity demand of a domestic UK home. Kutlu et al. [99] built a yearly simulation model and presented that the EFPCs were successfully implemented in ORC system even for low irradiance periods for CHP operation. Finally, Gao et al. [146] reported that the average EFPC efficiency is higher than 50.8% when operation temperature is above 100 °C. Also flat plate collectors have been used in solar ORC systems. Wang et al. [90] prepared an experimental rig to compare two collector types and they found overall power generation efficiency was 4.2% for evacuated solar collectors and about 3.2% for flat plate solar collectors. Wang et al. [147] studied a solar-driven regenerative solar ORC with flat plate collector to compare working fluids. Their results show that R245fa and R123 are the most suitable working fluids due to higher system performance.

2.8 Summary

In this chapter, a literature review on solar power generation systems is presented. The review of the literature shows that dynamical models have been investigated separately by several authors, however, no studies have considered these systems working together, especially considering electricity demand profile from the users. To build a compressive small-scale solar ORC system model, literature studies for component modelling were presented in detail. It is shown that a suitable controlling methodology is needed to be developed to match electricity demand profile from the buildings. The studies aimed to provide electricity demand with manipulating the ORC system's operation dynamics, but in solar based power plants, controlling should be on the heating side and ORC should synchronise itself with controlled heat input.

Although, s-ORC with heat storage unit looks promising to provide demand profile, low overall efficiency causes longer payback period. To increase the power output of the system, a novel system needs to be developed by keeping the advantage of the solar ORC's flexible operation ability. Low thermal efficiency of the conventional flat plate collectors at elevated temperatures can be overcome by using evacuated flat plate collectors. As a-Si cells show good performance under high temperature operation, combination of PV/T-ORC system can yield better overall efficiency compared to single s-ORC system. Finally, it seems scroll expander model has potential for improvements. The model is accurate to find shaft power at given rotation speed. These parameters would be enough for new system development and steady-state or quasi-steady-state operations, but in more detailed dynamic simulations, the model can be improved. The dependency of the empirical equations should be reduced and generator effects on electricity output need to be included in the analysis to find electricity output for small-scale ORC applications (around 1-2 kW)

CHAPTER 3 Off-design performance modelling of a solar organic Rankine cycle with hot water storage

3.1 Introduction

The main objective of this chapter is to build a comprehensive thermodynamic model of the off-design analysis based on fulfilment of user demand profile by controlling the operation parameters.

- The ORC system is modelled by considering the ORC pump and expander performances according to variations of the operating conditions.
- Sliding pressure operation strategy is implemented for allowing and controlling the electricity generation under varying heat source temperatures.
- A transient heat storage unit is modelled with considering the thermal stratification of the water tanks. It is analysed using a one-dimensional temperature distribution model.
- In order to provide the electricity demand and keep the useful heat in the tank, mass flow rate of the circulation water is controlled during different time periods. Thus, the system is analysed under off-design conditions.

3.2 System description

Proposed solar ORC system is shown in Figure 3.1. The system consists of three sub-systems, namely, solar collectors, water tank and ORC unit. The solar collectors, storage material and type of expander were carefully selected on the following basis:

Evacuated flat plate (EFP) collectors are chosen for heat collection. Using EFP collectors has various advantages over other types of collectors, such as, parabolic trough collectors in power generation plants. They do not need a sun tracking system and evacuated types can be used not only in countries where direct beam is available, but also on a grand scale. Their thermal performance is good even under low radiation and low ambient temperature conditions compared to standard flat plate collectors thus, there is a potential for use in winter. Therefore, evacuated type collectors are good candidate for the power generation plants with a heat storage unit.

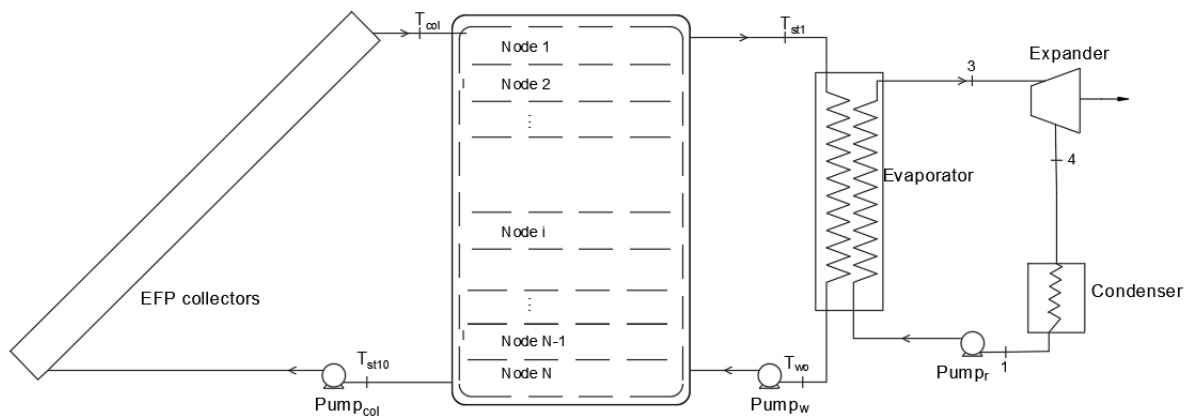


Figure 3.1 Schematic view of the solar ORC system

In large scale solar thermal electricity generation systems, there are many alternative materials for thermal storage, namely, molten salts, thermal oils and water. It is suggested that molten salt is the best choice for thermal storage in high temperature operations ($>400^{\circ}\text{C}$) [148]. Thermal oil is also promising in the temperature range between 300°C and 400°C , for lower operating temperature, water can be properly used because water has good thermal properties and has a much lower cost compared to other fluids [149]. In the present modelling, the temperature range of the operation

which is below 150 °C makes water a proper storage media. The working fluid which is pressurized water remains in liquid phase in all cases while operating with 5 bar pressure [150].

Working fluid in the ORC plays an important role because it is related to thermal performance and economics of the power plant. A number of researchers studied the effect of the working fluid selection on system performance [151],[145]. R245fa is a very common and effective working fluid for low temperature solar systems according to some theoretical analyses. Its performance has been investigated especially in small scale systems with commonly using a scroll expander [14],[92], [152].

The scroll type expander was selected as an expansion device in the present analysis because it is particularly well adapted to small-scale Rankine cycle applications that are lower than 25 kWe power output. Also, it offers major advantages over other types such as low rotational speeds, reliability and robustness (less number of moving parts), and the ability to handle high pressure ratio [114].

In Figure 3.1, high performance evacuated flat-plate collectors are used for heating the water which comes from the bottom of the tank (T_{st10}) by converting solar radiation to heat and filling the tank to the topside (T_{col}). Water is used as the heat transfer fluid instead of thermal oil in this model because of its more favourable thermal properties and its ability to be directly discharged into the tank without heat exchangers make more effective. The water storage tank has two inlet and two outlet; usage of these ports depends on the working periods.

The examined system should provide the required average electricity needed for an average house in a small community. According to the reference [19], 24h of a day can be divided into three different time periods, as shown in Figure 3.2. By including

the electricity generator efficiency, the approximate required work outputs per house should be minimum 0.5 kW for the day time, 0.75 kW for the early night and 0.3 kW for the late night period. The peak energy demand is observed in the early night period so the design conditions of the system are selected by considering the highest electricity demand.

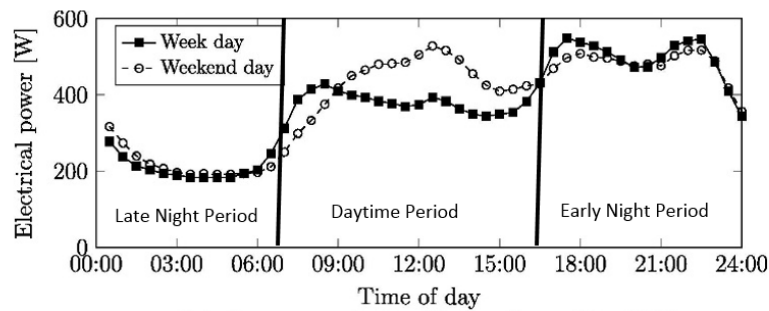


Figure 3.2 Proposed three periods [19]

In this chapter, firstly the ORC working conditions are to be determined using design parameters subjected to performance of the expander [153]. Condenser operating temperature is found by environment temperature, but evaporator temperature totally depends on heat source temperature and flow rate, thus, the analysis should be conducted with heat source temperature which cannot be same during the day because of charging and discharging. The effect of variation of heat source temperature requires some control methods in the model, as the ORC alone cannot prevent the unstable trend. Thus, sliding operation is adapted.

Figure 3.3 shows the outline of the methodology in this chapter. A general off-design performance methodology is used [154]. As a first step, the ORC is designed for on-design conditions. As the most critical and higher electricity requiring period is at

evening, which is called early night, design of the heat exchangers should be considered according to this period. Later, off-design performance is investigated for other periods by using previously dimensioned heat exchangers. Lastly, a parametric study is conducted to determine suitable number of collectors and water tank volume.

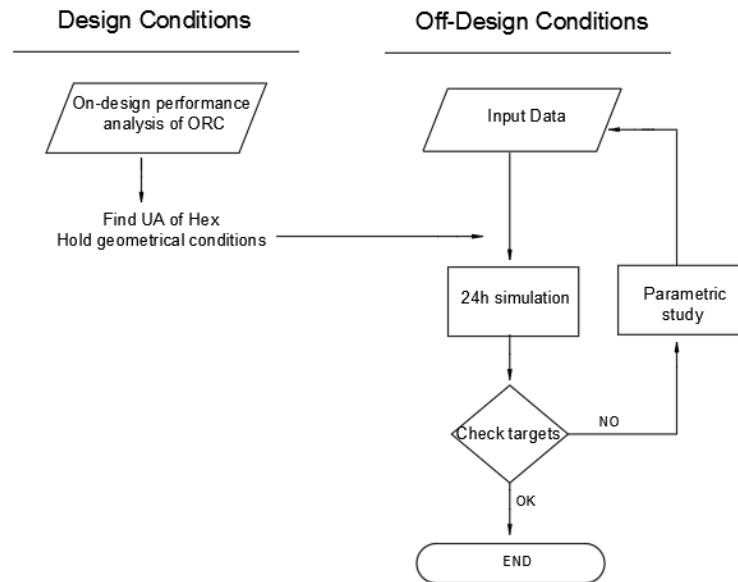


Figure 3.3 Followed flowchart of this chapter

3.3 Mathematical modelling of components

3.3.1 Modelling of solar collector

The chosen solar collector for this system is TVP SOLAR HT-Power [155], high efficiency evacuated flat plate collector. This collector has been already evaluated for its potential in ORC systems by Freeman et.al [19], [145] and Calise et.al [140]. In the previous papers diathermic oil was used as a heat transfer fluid, however, in this study pressurized water is used as a heat transfer fluid for reasons explained in previous sections. For modelling of the EFP collector, same assumptions as in the

reference [145] is followed. Thus, the thermal efficiency of the solar collector can be calculated by Eq. (3.1):

$$\eta_{col} = \eta_0 \cdot K_\theta - c_1 \frac{\bar{T} - T_{am}}{G} - c_2 \frac{(\bar{T} - T_{am})^2}{G} \quad (3.1)$$

The collector thermal efficiency parameters are taken from [140] which are $\eta_0=0.82$, $K_\theta=0.91$, $c_1=0.399$, $c_2=0.0067$. η_0 is solar collector zero-loss optical efficiency, K_θ indicates incident angle modifier, c_1 and c_2 are collector heat loss coefficients. Efficiency curve for the collector is shown in Figure 3.4.

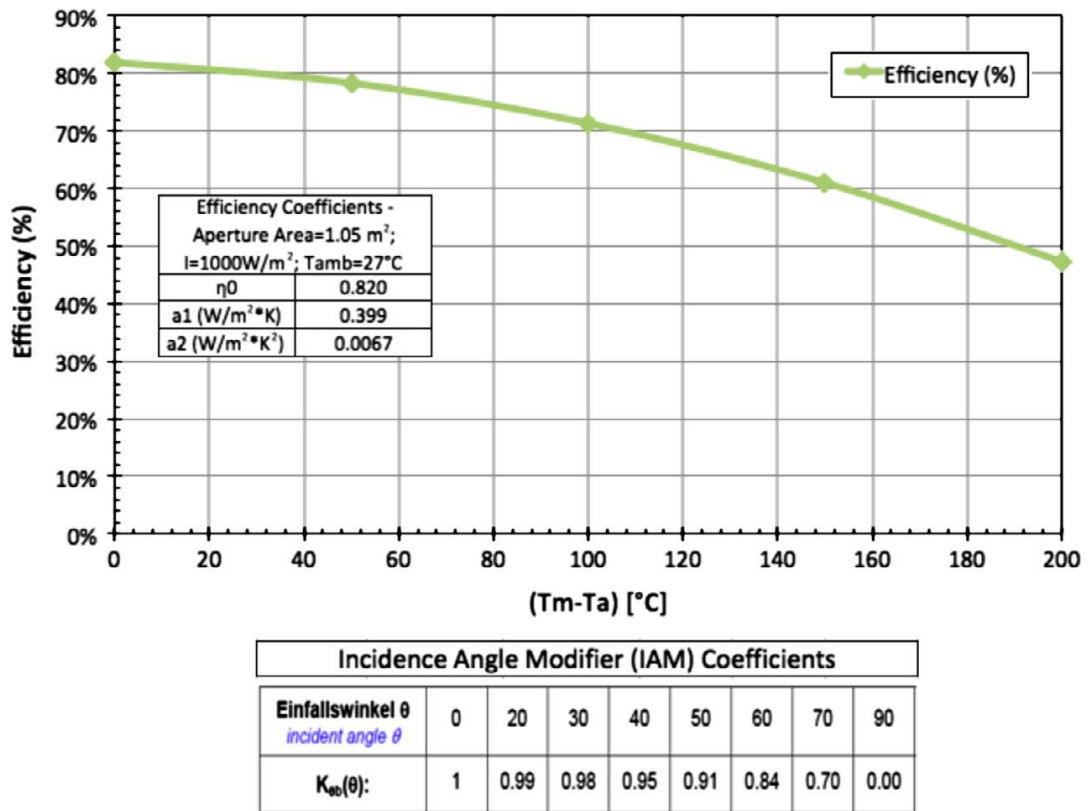


Figure 3.4 Efficiency curve for TVP SOLAR HT-Power panels [140]

The amount of solar radiation absorbed by the collector is equal to the enthalpy increase of the heat transfer fluid, the equation is given in Eq. (3.2):

$$\dot{Q}_{col} = \eta_{col} \cdot A_{col} \cdot G = \dot{m}_{cw} \cdot c_{pcw} \cdot (T_{col} - T_{stN}) \quad (3.2)$$

Where t_{col} and t_{stN} are collector water outlet temperature and water return temperature from the tank, respectively.

3.3.2 Thermocline water tank modelling

Water storage tank and its thermodynamic modelling is described in this section. Heat storage unit is one of the important components in the solar ORC system because this unit is used as the heat source for the ORC. Its energy capacity, which is related to its volume, determines the energy storage level in the system and influences ORC performance and electricity output. There are many studies have investigated the thermocline behaviour in water storage tanks and have analysed from 1D to 3D models [156]. Usually, 1D models use CFD based or experiment-based correction factors. However, in this modelling, the most acceptable approach, the isothermal mixing zone methodology is used for the simulations. The water tank volume is divided into a number of equal elements to obtain temperature distribution in the storage tank [157], and the node of each element can be seen in Figure 3.1. For each control volume, an energy balance equation is written by considering the heat losses to the ambient. By solving all the energy balance equations simultaneously, temperature distribution inside the tank can be determined. The following equations give the energy balances. These equations have already been used in previous studies [158],[159],[160]. Moreover, in this study ten node mixing zones are used. Eq. (3.3) is the energy balance for the first node, Eq. (3.4) is the energy balance for the internal node “i” and Eq. (3.5) for the last node.

$$\begin{aligned}
M_{st1} \cdot c_{pw} \cdot \frac{\partial T_{st1}}{\partial t} & \quad (3.3) \\
& = \dot{m}_{cw} \cdot c_{pcw} \cdot (T_{col} - T_{st1}) + \dot{m}_w \cdot c_{pw} \cdot (T_{st2} - T_{st1}) - U_t \cdot A_{st1} \\
& \quad \cdot (T_{st1} - T_{am})
\end{aligned}$$

$$\begin{aligned}
M_{st(i)} \cdot c_{pw} \cdot \frac{\partial T_{st(i)}}{\partial t} & \quad (3.4) \\
& = \dot{m}_{cw} \cdot c_{pcw} \cdot (T_{st(i-1)} - T_{st(i)}) + \dot{m}_w \cdot c_{pw} \cdot (T_{st(i+1)} - T_{st(i)}) \\
& \quad - U_t \cdot A_{st(i)} \cdot (T_{st(i)} - T_{am})
\end{aligned}$$

$$\begin{aligned}
M_{stN} \cdot c_{pw} \cdot \frac{\partial T_{stN}}{\partial t} & \quad (3.5) \\
& = \dot{m}_{cw} \cdot c_{pcw} \cdot (T_{st(N-1)} - T_{stN}) + \dot{m}_w \cdot c_{pw} \cdot (T_{wo} - T_{stN}) - U_t \\
& \quad \cdot A_{stN} \cdot (T_{stN} - T_{am})
\end{aligned}$$

T_{wo} is the water temperature coming from the evaporator to the tank's bottom node.

U_t is the overall heat loss coefficient of the well-insulated tank which was taken as 0.8 W m⁻²K⁻¹ [160]. The tank has a cylindrical shape with diameter d_{st} and height L , and the outer areas of nodes are given in equations as below:

$$A_{st1} = \frac{\pi d_{st}^2}{4} + \frac{\pi d_{st} L}{N} \quad (3.6)$$

$$A_{st(i)} = \frac{\pi d_{st} L}{N} \quad (3.7)$$

and the last node:

$$A_{stN} = \frac{\pi d_{st}^2}{4} + \frac{\pi d_{st} L}{N} \quad (3.8)$$

The static mode of the storage tank means there are no external forced flows entering or leaving the tank. Therefore, conduction heat transfer between the nodes should be considered. Heat loss to the environment also creates thermal stratification in the tank, as fluids near the wall are cooled due to heat loss and these lower temperature fluids, which have lower density, go through the bottom of the tank. This phenomenon has been previously studied by other researchers [161], [162]. Armstrong et.al [163] investigated the influence of the wall material specification on de-stratification and showed that thermal conduction of the wall material has a strong influence on this. Cruickshank et.al [162] formulated the energy balance equation when there are no flows entering or exiting the tank:

$$\begin{aligned} M_{st(i)} \cdot c_{pw} \cdot \frac{\partial T_{st(i)}}{\partial t} & \quad (3.9) \\ &= \frac{(k + \Delta k) A_{c(i)}}{\Delta x_{i+1 \rightarrow i}} \cdot (T_{st(i+1)} - T_{st(i)}) + \frac{(k + \Delta k) A_{c(i)}}{\Delta x_{i-1 \rightarrow i}} \\ & \cdot T_{st(i-1)} - T_{st(i)} - U_t \cdot A_{st(i)} \cdot (T_{st(i)} - T_{am}) \end{aligned}$$

Where $\Delta x_{i+1 \rightarrow i}$ and $\Delta x_{i-1 \rightarrow i}$ are a center-to-center distance between nodes, k and Δk are the thermal conductivity of water and the de-stratification conductivity. Newton [164] derived empirically of this conduction term Δk using tank wall lateral area A_{lwall} :

$$\Delta k = k_{wall} + \frac{A_{lwall}}{A_{c(i)}} \quad (3.10)$$

3.3.3 Organic Rankine cycle

An ORC system mainly consists of working fluid pump, evaporator, expander and condenser. The system schematic is given in Fig. 3.1. The refrigerant enters the pump as a saturated liquid '1' at condensing pressure, then its pressure is increased by refrigerant pump to the evaporator pressure level '2'. Evaporating pressure depends on the heat source temperature and the ORC working strategy which will be explained later. The next component of the ORC is the evaporator where the heat is supplied from the water tank, at the outlet of the evaporator, the fluid phase is changed to saturated vapour '3'. After that, the fluid goes into the expander. The expander generates shaft work and reduces the fluid pressure to the condensing pressure and finally, the refrigerant enters the condenser at point '4'. To indicate the state points, a T-s diagram of the ORC cycle is given in Figure 3.5. In following subsections, each component of the ORC is modelled to simulate system with varying conditions.

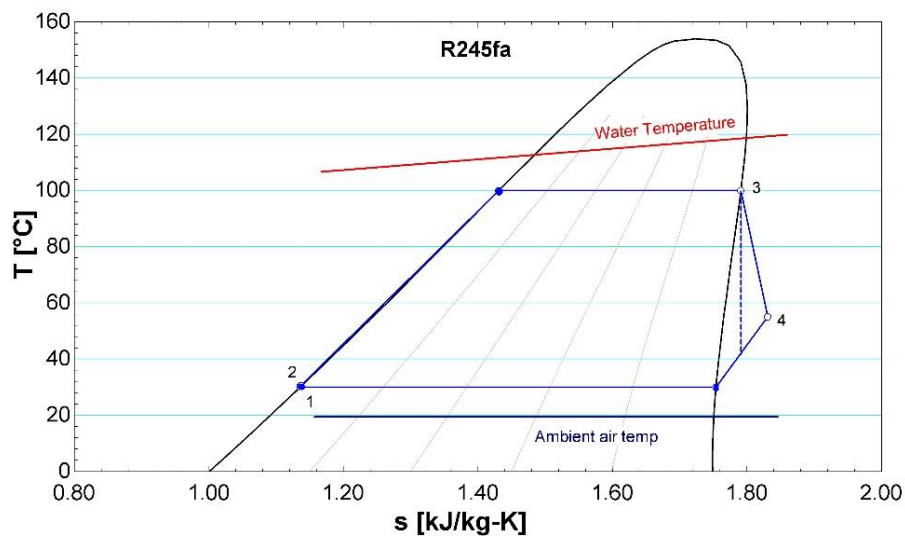


Figure 3.5 T-s diagram of ORC

3.3.3.1 Pump modelling

The pump is used to increase pressure of the working fluid. Its key parameter isentropic efficiency is not constant as the discharge pressure and mass flow rate vary with evaporation temperature which is related to heat source temperature (water tank). Quoilin et al [165] have used some empirical equations for modelling their dynamic ORC system. In this analysis same equations are followed; Therefore, the isentropic pump efficiency is defined in the Eq. (3.11) and the pump empirical equation is given in Eq. (3.12)

$$\eta_{pump} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (3.11)$$

$$\eta_{pump} = a_0 + a_1 \cdot \log(X_p) + a_2 \log(X_p)^2 + a_3 \log(X_p)^3 \quad (3.12)$$

X_p indicates the pump capacity fraction, its equation is given by:

$$X_p = \frac{\dot{v}_{su} \cdot \dot{m}_{ref}}{\dot{V}_{su,p,max}} \quad (3.13)$$

The overall pump efficiency is therefore:

$$\eta_{overall} = \eta_{me,p} \cdot \eta_{pump} \quad (3.14)$$

The relevant parameters in Eqs. (3.12), (3.13) and (3.14) are listed in Table 3.1.

Table 3.1 Pump model parameters

$\dot{V}_{su,p,max}$	0.25 l/s	$\eta_{me,p}$	0.9
a_0	0.93	a_1	-0.11
a_2	-0.2	a_3	-0.06

3.3.3.2 Expander modelling

Since desired power is generated in the expander, the expander is the most critical component in ORC systems. According to the literature review, there are some models are available from several applications. As a scroll expander is chosen before, a study is selected and its empirical equations are taken from ref. [92]. In selected study, the empirical equations and expander performance depend on three independent parameters, namely, inlet pressure of the expander, pressure ratio and rotational speed of the expander. In order to obtain a generic non-dimensional performance curve of the expander, input variables were carefully selected, and only heat losses were disregarded in this model. According to the expander model, isentropic efficiency and filling factor are defined in Eq. (3.15) and Eq. (3.16).

$$\eta_{exp} = \frac{\dot{W}_{shaft}}{\dot{m}_{ref}(h_{su} - h_{ex,s})} \quad (3.15)$$

$$\phi = \frac{\dot{m}_{ref} v_{su}}{\dot{V}_s} \quad (3.16)$$

With three parameters, isentropic efficiency and filling factor can be found from empirical correlations [92]:

$$\eta_{exp} = y_{max} \cdot \sin \left(\xi \cdot \arctan \left(B \cdot (r_p - r_{p,0}) - E \cdot \left(B \cdot (r_p - r_{p,0}) - \arctan \left(B \cdot (r_p - r_{p,0}) \right) \right) \right) \right) \quad (3.17)$$

$$\phi = \phi_n - b_0 \cdot \ln \left(\frac{N_{rot}}{3000} \right) + b_1 \cdot r_p^* + b_2 \cdot p^* \quad (3.18)$$

$$B = \frac{\delta}{\xi y_{max}} \quad (3.19)$$

$$E = \frac{B \cdot (r_p - r_{p,0}) - \tan\left(\frac{\pi}{2\xi}\right)}{B \cdot (r_p - r_{p,0}) - \arctan\left(B \cdot (r_p - r_{p,0})\right)} \quad (3.20)$$

$$r_p^* = \frac{r_p - 4}{4}, \quad N_{rot}^* = \frac{N_{rot} - 3000}{3000}, \quad p^* = \frac{p - 10}{10} \quad (3.21)$$

$$r_{p,0} = r_{p,0,n} + a_0 \cdot N_{rot}^* \quad (3.22)$$

$$\delta = \delta_n + a_0 \cdot p^* + a_2 \cdot N_{rot}^* \quad (3.23)$$

$$r_{p,max} = r_{p,max,n} + a_3 \cdot p^* + a_4 \cdot N_{rot}^* \quad (3.24)$$

$$y_{max} = y_{max,n} + a_5 \cdot p^* + a_6 \cdot (N_{rot}^* - N_{rot,n}^*)^2 \quad (3.25)$$

$$N_{rot,n}^* = \frac{N_{rot,n} - 3000}{3000} \quad (3.26)$$

Each parameters can be expressed as a polynomial function of the non-dimensional pressure and rotation speed [106]. Explanations of some of the parameters and constants can be taken from the given ref. [92]. Figure 3.6 gives the expander isentropic efficiency variation with pressure ratio for the given operating conditions. It is seen that pressure ratio of the scroll expander has an influence on isentropic efficiency. As the condenser temperature or pressure is related with the environment temperature, environmental changes also affect the system performance. However, constant condensing temperature can be assumed for this analysis. Thus, expander performance depends on the evaporating pressure which is related with the

temperature of heat source. In order to obtain higher performance from the expander, the evaporating temperature is controlled between 80°C and 100°C. The expected working range of the expander under given conditions is also shown in Figure 3.6.

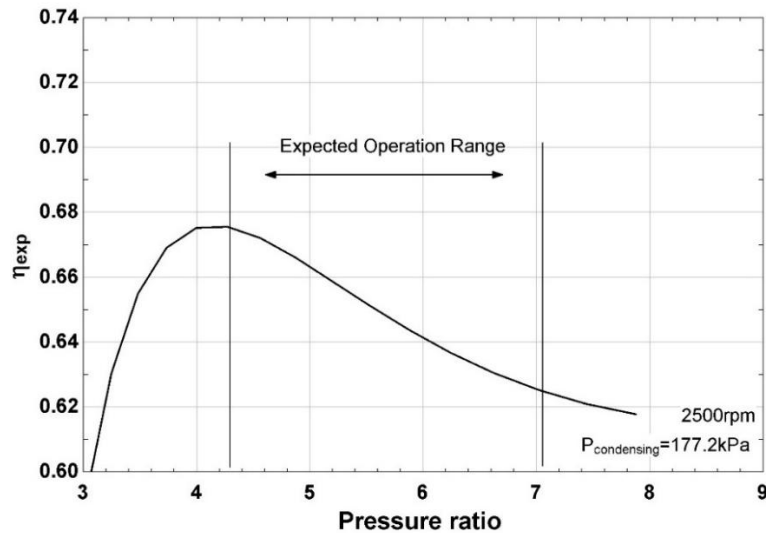


Figure 3.6 Expander isentropic efficiency variation

3.3.4 Modelling of the heat exchangers and control strategy

In the ORC unit, there are two heat exchangers are used for different purposes. By neglecting pressure losses in the heat exchanger and to make the study more practical, concentric tube type heat exchangers are chosen for this model, as various authors chose because of the same reasons [165], [166], [167]. To find the effectiveness of the heat exchangers, the effectiveness-NTU method is used in the analysis. Some equations are used [168],[169],[170] to find heat transfer coefficients for single and two phase states in the literature. In this modelling, Gnielinski equation where the fluids exist in a single state, as given in Eqs. (3.27) and (3.28), which are used and given in Ref.[171] for turbulent flows.

$$h = \frac{\left(\frac{f}{8}\right) (Re - 1000) Pr}{1 + 12.7 \left(\frac{f}{8}\right)^{0.2} (Pr^{2/3} - 1)} \left(\frac{k}{d}\right) \quad (3.27)$$

$$f = (0.79 \ln Re - 1.64)^{-2} \quad (3.28)$$

In order to determine heat transfer coefficient when boiling of the refrigerant R245fa takes place, Kenning-Cooper correlation in Eq. (3.29) is used as given by Sun and Mishima [172] based on their empirical findings.

$$h_{bo} = [1 + 1.8X^{-0.87}] 0.023 Re_l^{0.8} Pr_l^{0.48} (k/d) \quad (3.29)$$

Where X is the Martinelli factor which is given from vapour quality x:

$$X = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.1} \quad (3.30)$$

As it is known, the most important heat exchanger unit in the ORC system is the evaporator. In this modelling it is more important because pressure control strategy is closely related to adjusting the evaporation temperature. The proposed double pipe evaporator includes two regions in one exchanger such as single phase and evaporating regions. The working fluid temperature is increased to the desired level in the single-phase region and then the phase is changed into saturated vapour in the evaporating region. A schematic view of the evaporator is given in Figure 3.7. In off-design operation, total length of the evaporator must be constant, but length of the regions may differ according to heat source conditions. Since the evaporator exposed to hot water flow from the heat storage tank the source temperature cannot stand constant because the storage tank temperature falls during the ORC operation period. To

calculate the evaporating temperature when heat source temperature varies, the sliding pressure control method is used in the analyses.

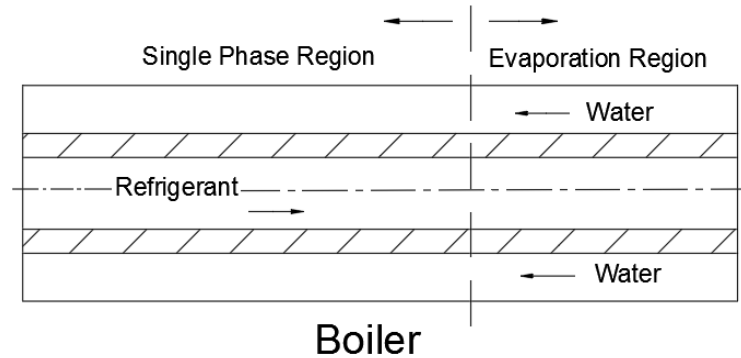


Figure 3.7 Schematic view of evaporator (boiler) single phase and evaporating regions boundary changes

To analyse the off-design operation of the evaporator by effectiveness-NTU method, Eq. (3.31) is used.

$$NTU = \frac{U \cdot A}{C_{min}} \quad (3.31)$$

Where U is overall heat transfer coefficient and NTU is number of transfer units.

The T-s diagram related with variable evaporation temperature is given in Figure 3.8. Variation of evaporation temperature results with variable work output of the expander. It also effects the rejected heat from the heat source so it will be used in analysis to get balanced energy conversion with providing energy demands. Once the dimensions of the evaporator are determined according to design conditions, sliding pressure control procedure is applied to find the evaporating temperature in off-design

conditions. This control strategy follows; area of the evaporating region and evaporating temperature are assumed by the user. The heat transfer coefficients are found according to given conditions then the effectiveness-NTU method is applied into the evaporating region until a proper evaporating temperature is found. Proper temperature is found by comparing the assumed parameters' effectiveness and new effectiveness. Difference between assumed and calculated values is continues to iteration until difference would be smaller than a certain value. After satisfying the evaporating side, the area of the single phase region is found. The same procedure is followed and if the state is not convincing the loop returns to beginning and the area of the evaporation region is altered. The related flow chart showing the procedure is given in Figure 3.9.

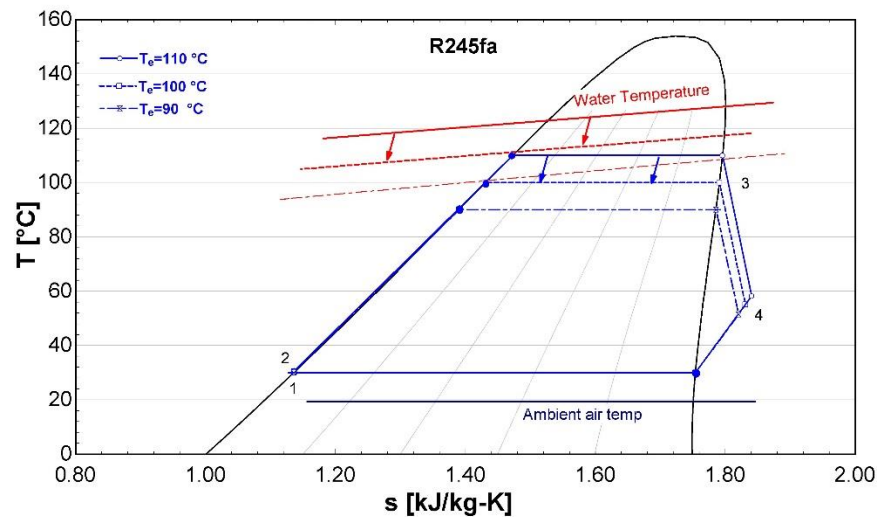


Figure 3.8 T-s diagram for sliding pressure operation

The other heat exchanger device in the ORC block is condenser. It is used for rejecting heat from refrigerant to the environment; water or air cooled condensers are available

in the literature but in this study the air cooled condenser was used because small scale solar cogeneration system condensing loads are not at high levels, and also using the air cooled condenser is more practical. The heat load of the condenser depends on the inlet condition of the refrigerant it also depends on evaporating temperature, expander efficiency. However, performance investigation of the condenser is not within the scope of this study because heat load can be easily adjustable by fan speeds.

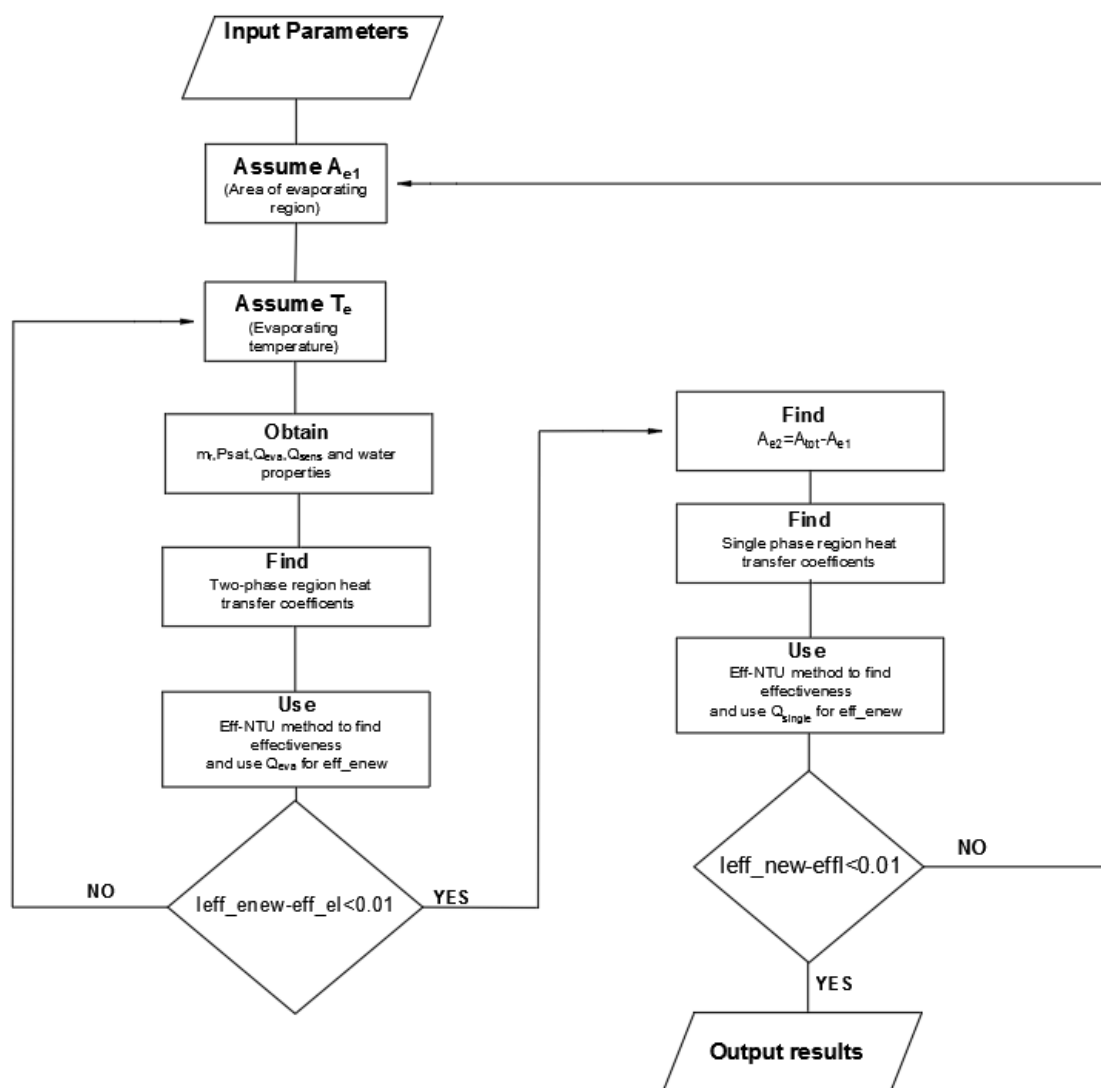


Figure 3.9 Off-design operation flow chart to operate sliding pressure

3.4 Results and discussions on the performance characteristics for the selected days

In the analysis, Engineering Equation Solver (EES) software was used to obtain the thermal properties of the refrigerants. Regarding the following methodology of the transient states, the initial temperatures in all subsystems apart from node temperatures in the tank, have been set as equal to the ambient temperature. The equations were solved by the developed mathematical program which was written in the software MATLAB. The differential equations in the water tank modelling were discretized by using Eq. (3.32). This method solves the quasi-steady problems in every time step and time interval which was selected as 1 minute. In every time step, the produced work output, mass flow rates, fluid temperatures and available solar energy are calculated to assess the performance.

$$\frac{\partial T_{st}}{\partial t} = \frac{T_{st}^{t+\Delta t} - T_{st}^t}{\Delta t} \quad (3.32)$$

3.4.1 Design conditions

In order to evaluate the system performance, firstly, design conditions need to be determined. Since condensing temperature depends on the ambient temperature in air cooled condenser, the ambient air temperature has an influence on the design conditions selection. Figure 3.10 shows hourly ambient air temperature variation during a typical day in June in Istanbul. The ambient temperature has a slight variation during the day and mean temperature is around 20°C. Therefore, the condensing temperature is selected as 30°C. According to the specifications of the selected expander model with constant condensing temperature, the ORC behaviour, by

varying evaporating temperature is presented in Figure 3.11. Since the heat storage unit is a finite source, it is important to select the matched requirements according to Figure 3.2 as a design point for avoiding excessive consumption of this finite source.

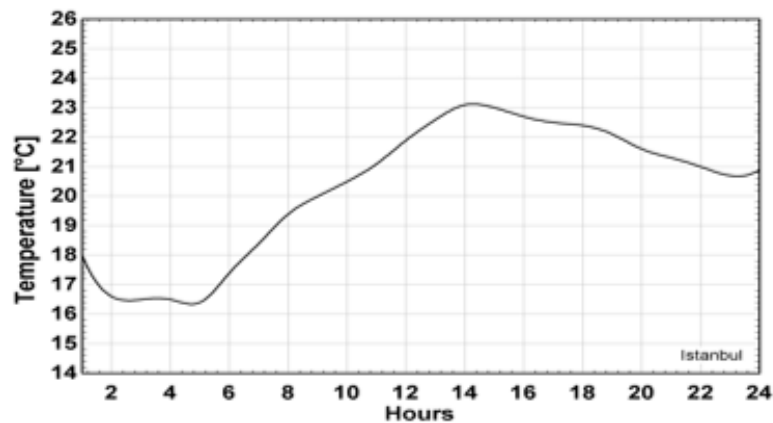


Figure 3.10 Ambient temperature profile for a selected day in Istanbul

After selecting the design condensation temperature of 30°C, the design evaporation temperature should be corresponding to the built-in pressure ratio. However, the electricity demand needs to be considered for the peak period, so the expander needs to operate at higher pressure ratio. Due to characteristic of the scroll expander, operation at higher pressure ratio has a slightly lower performance than operation at built-in ratio. It can be acceptable because peak period covers only 30% of the day, at the rest of the day, the expander operates with high performance.

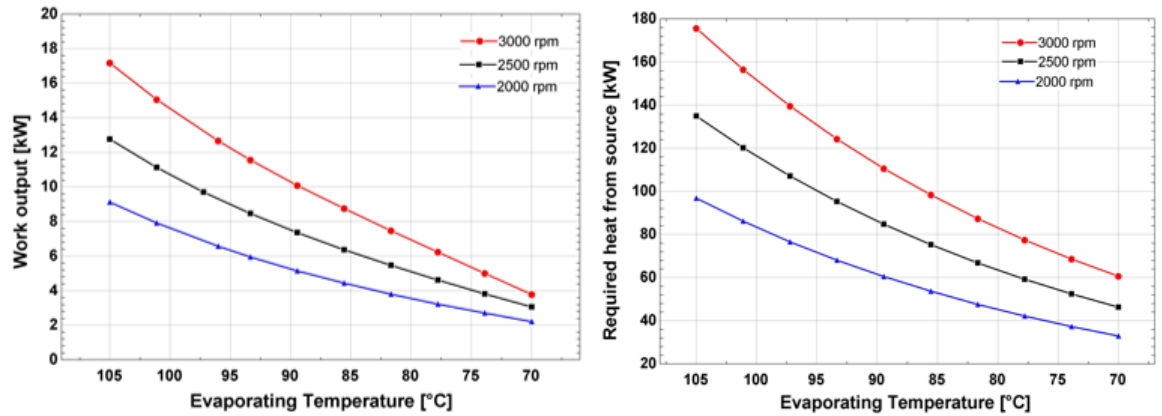


Figure 3.11 Effect of the evaporator temperature on the shaft power and required heat input

To meet the electricity demand for twelve dwellings at early night period, the expander speed is selected as 2500 rpm and evaporating temperature as 96 °C. As a result of these selections, work output and extracted heat from the water tank are expected to be 9.3 kW and 103.2 kW, respectively. Given these conditions the evaporator needs to be dimensioned to predict the performance in all day simulation which refers to off-design conditions. The temperature of the water tank will go down by time, especially at night, and as a result the heat source temperature will not be constant. The heat exchanger has been designed using Eqs. (3.27)-(3.31) on which Figure 3.12 has been based. It shows total length of the evaporator is dependent upon design inlet temperature and this length increases with lowering of the temperature. It should, however, be noted that these plots are drawn for 96 °C of evaporating temperature and when water inlet and evaporating temperatures approach, the required length of the heat exchanger will normally increase. It is expected that the heat source temperature which is the water tank temperature is around 130 – 100 °C during operating times, thus, water inlet temperature is selected as 110 °C for design conditions.

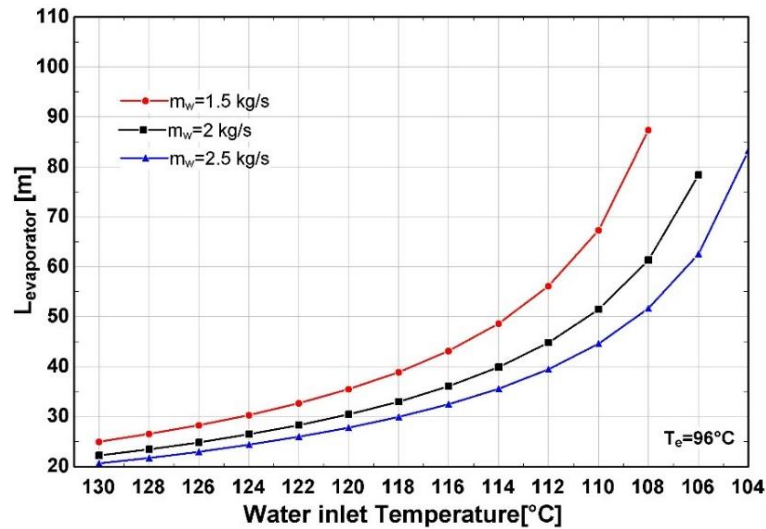


Figure 3.12 Designed coil length variation of the evaporator with water inlet temperature

The other parameter affecting the evaporator length is the mass flow rate of the water. A higher mass flow rate has the positive effect of shrinking the dimensions. However, it should be considered that higher mass flow rates can destroy the thermocline in the water storage tank. Therefore, water mass flow rate is selected as 2 kg/s as a design parameter which leads to an evaporator of 51 m. The effect of water mass flow rate on the system performance will be discussed in detail in a later section. Selected design parameters are summarised in Table 3.2.

Table 3.2 Design conditions of the solar ORC

Work output: 9.5 kW	Heat from source: 103.2 kW
Evaporating temperature: 96 °C	Expander speed: 2500 RPM
Condensing Temperature: 30 °C	m_w : 2 kg/s
Water inlet Temperature: 110°C	Evaporator length: 51 m
Evaporator water side, d_o : 0.3 m	Evaporator refrigerant side, d_i : 0.012 m

3.4.2 Operation at off-design conditions

Before simulating the whole system, the reaction of the heat exchangers when the system operates at off-design conditions is investigated. Firstly, the effect of water inlet temperature originating from top of the tank needs to be analysed. Furthermore, its effect also depends on mass flow rate. Figure 3.13 shows the effect of the water inlet on evaporating temperature with various mass flow rates. And this analysis originates the controlling the power output methodology. Since the design conditions are 96°C and 110 °C of evaporating and water inlet temperatures, respectively, the heat exchanger has been dimensioned to satisfy these conditions. Sliding pressure operation control strategy is applied according to the flow chart in Figure 3.9. This method is also applied in order to compare different water mass flow rates. It is observed that the evaporating temperature decreases when using lower water mass flow rate. The lower evaporating temperature yields both lower work output and lower extracted heat from the finite source.

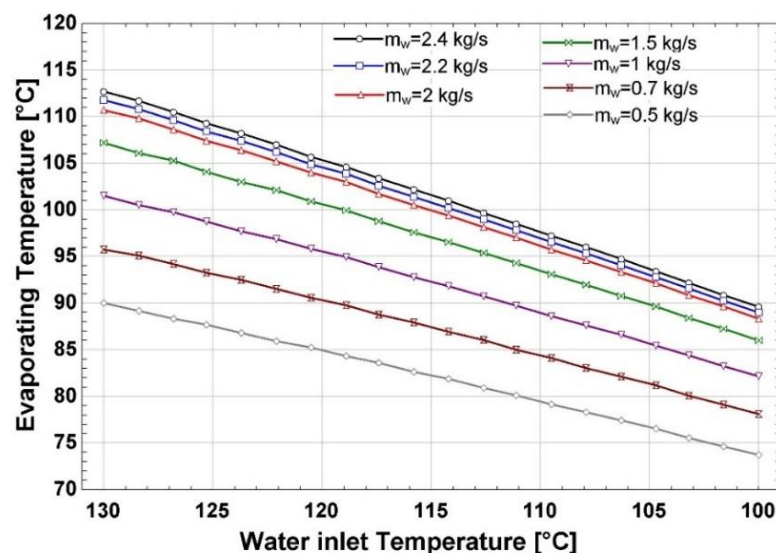


Figure 3.13 Evaporator temperature change by various circulation water mass flow rates and temperatures

Figure 3.14 shows the effect of the water inlet temperature on work output and ORC thermal efficiency. The ORC thermal efficiency has an important influence on the system metrics and as such, should be considered in the off-design performance. However, in this case, conservation of stored heat is important for early night period operation. A mass flow rate of 2 kg/s has a higher work output and efficiency but using this flow results in more extracted heat from the source. Therefore, a mass flow rate of 0.5 kg/s can be selected for day time and late night periods. It is seen from Fig. 3.14 that 0.5 kg/s mass flow rate is proper to fulfil the demand when the inlet temperature is between 120°C and 105°C.

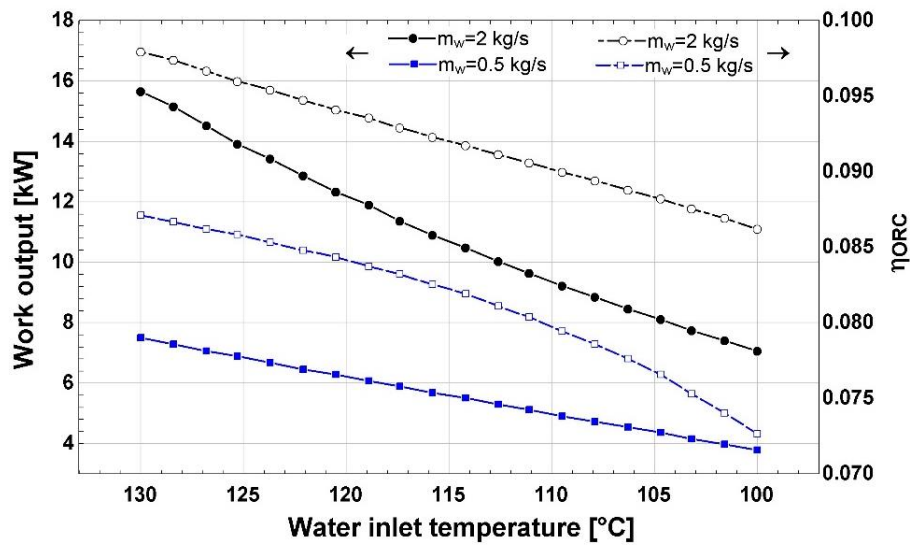


Figure 3.14 Work output and ORC thermal efficiency change by circulation water mass flow rate $m_w = 2$ kg/s and $m_w = 0.5$ kg/s and temperature

3.4.3 Hourly performance simulations during a selected day

In order to provide a performance assessment of the system, solar collector array and heat storage dimensions needs to be determined. The system is simulated for a clear

day, relatively good solar irradiance but shorter day time which is presented in Figure 3.15. The present system is analysed for a small community level application; it is chosen for twelve dwellings, so the area of solar collectors can be selected between 400 m^2 and 600 m^2 . To observe good results 550 m^2 is chosen. Electricity demand reaches peak level in early night period and this peak demand claims approximately 10 kW output for 7 hours. Therefore, the system requires quite a large heat storage unit. According to a preliminary assessment of the system, pressurized water tank volume should be higher than 70 m^3 . Since thermocline phenomena is considered in the present model, dimensions of the pressurized tank have an influence on the performance. Whilst thermocline is affected by many factors, this study only considers the one-dimensional temperature distribution model. The storage tank is selected as a cylinder with a diameter of 4 meter and height of 7 meter.

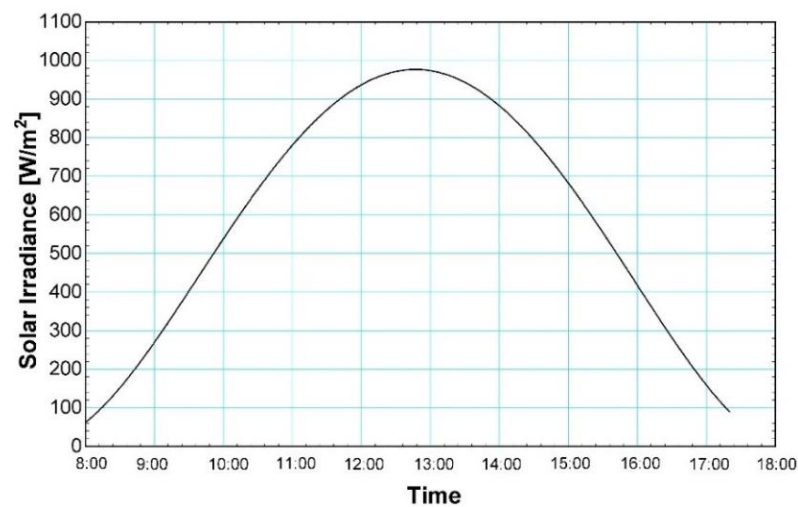


Figure 3.15 Solar radiation variation of the chosen day

The system operation is based on the following strategy: day time period starts at 08:00, the collector pump runs and solar heat is stored in the tank, meanwhile the ORC produces electricity. Collector water mass flow rate is selected as 0.02 kg/s per

collector and is taken from the data sheet and the total mass flow rate poured into the tank is determined by the number of collectors. The ORC side water mass flow rate is chosen as 0.5 kg/s to match electricity demand in the designed heat exchanger. Moreover, this prevents excessive use of the heat source. Day time period ends at 17:15 when solar irradiance is not sufficient and peak demand period starts. This period covers the main target of the study and ends at 24:00. Only ORC works and water mass flow rate are set at 2.4 kg/s to satisfy the excessive demand by reaching higher evaporation temperature. The last period is late night period from 24:00 to 08:00. During this period, the water mass flow rate is switched to 0.5 kg/s again as production of a high amount of electricity is not required.

According to Figure 3.13 and Figure 3.14, it is expected that the tank temperature, especially the first node temperature, should be higher than 100°C both to provide the required production and to avoid low expander performance. Otherwise, the performance of the expander will be degraded significantly, as shown by the characteristic curve in Figure 3.6. Therefore, initial tank temperature is selected as 100°C for simulations. One of the important aspects is selection of on-off criterion. To provide operation at the same conditions for other days the stop criterion has to be defined. The late night period production can be dispensable to conserve the stored heat in the tank for next day. It is found that when stop criterion is assigned as a condition in simulation it produces good results. Middle node of the tank, fifth node, is selected as stop consideration. When the temperature of the middle node reaches the initial condition, the working fluid pump is shut off and the tank is subjected to static mode only cooling until 08:00.

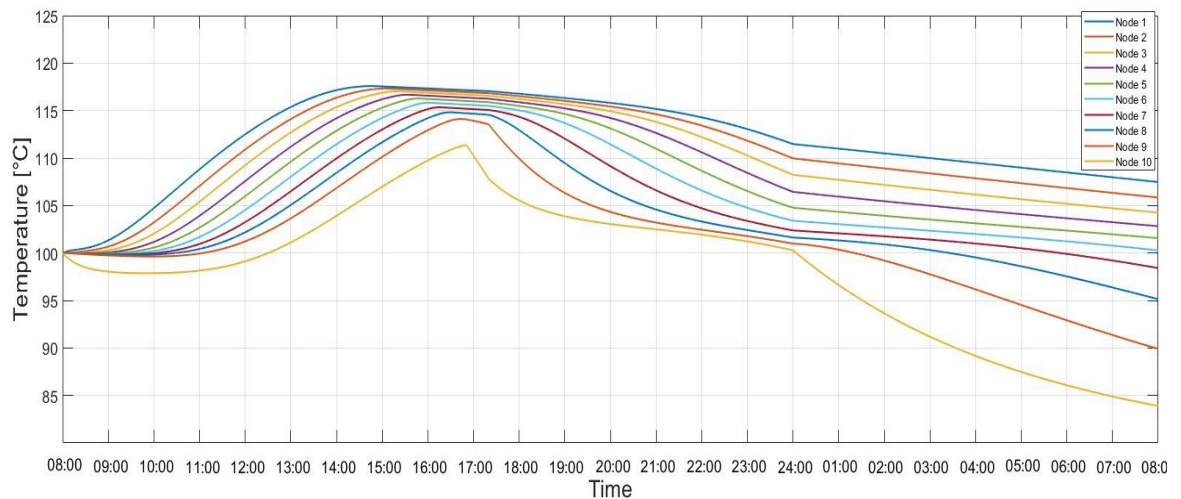


Figure 3.16 Water elements temperature changes during first simulation

One of the most important issues for the daily simulation is the selection of the initial temperature in the tank. According to previous sections, it can be concluded that temperature levels have an influence on the work output. Since selecting a proper initial tank temperature is significant for the results, it is required to eliminate this uncertain situation. Otherwise, it results in over-or underestimation of the work output.

In order to determine the reasonable initial condition, a number of simulations need to be conducted until initial and final temperatures reaching a stable level in the simulation. After finishing the first simulation, the second simulation's initial conditions are selected as the previous one's final temperatures. This iteration continues until the initial temperatures are matched with the final temperatures. Normally, the temperature gradient in the water tank is not the same at all levels; however, as a starting point, it is assumed that the initial temperature is 100°C for all nodes. After applying the control strategy described in the previous sections, Figure 3.16 is plotted and it shows temperature distribution in the tank during the first 24

hours, and Figure 3.17 shows work output results for the first 24 hours. Although the first node temperature is higher than in the early night period between 11:00 and 17:00, produced work is quite lower because of controlling of the evaporation temperature by mass flow rate. 0.5 kg/s mass flow rate is used in day time and late night periods, whereas 2.4 kg/s mass flow rate is used in early night period in all simulations. It is also shown that work generation is not ended for this day because the fifth node temperature does not reach the initial temperature and stop criterion can not be activated.

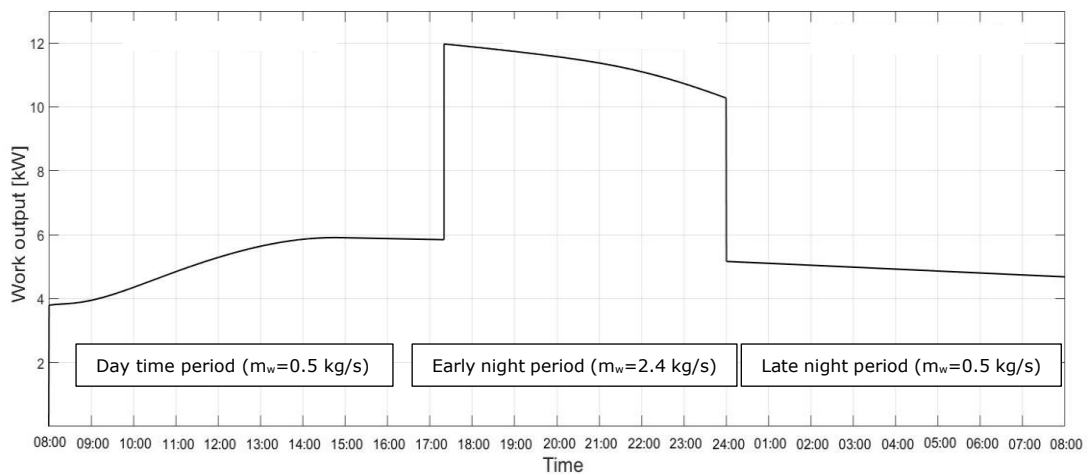


Figure 3.17 Shaft output during the first simulation

According to temperature distribution, it is observed that the last node temperature has a different trend compared to other nodes. The reason can be explained with the temperature of the water outlet from the evaporator which is discharged into the last node of the tank. This colder fluid decreases the last node's temperature. However, its influence is quite insufficient to the other nodes because it has a very low mass flow rate compared to the tank volume. In the early night period, water mass flow rate is

increased, which leads to an increase in the temperature of the water outlet from the evaporator. As a result of these, the degree of thermocline in the tank decreases. However, it is increased again by the lower flow rate in the late night period.

Figure 3.16 and Figure 3.17 show the first simulation results which are based on the assumption of the same initial temperatures for all nodes in the tank. Using final temperatures as the next simulation's initials, eight simulations have been conducted and temperature variations of the initial temperatures are given in Figure 3.18. By the 8th simulation, temperatures become a stable level, which means inlet and final temperatures are same. It can be said that all of the useful solar heat charged to the tank are used for driving the ORC and the rest are transferred to the ambient as heat losses. To explain in more detailed, Figure 3.19 is plotted. It shows power outputs in certain simulations. It can be seen that the cumulative work outputs are stabilized by the 8th simulation. It is likely because the 8th simulation is more realistic for the selected typical day so it is chosen as a reference day of the present study. In the third simulation, work output falls dramatically, which can be explained by the assigned stop criterion. In that simulation, the stop criterion is activated because temperature of the middle node falls to 100 °C at and the work generation is interrupted to conserve the stored heat in the tank. It is seen that stored heat from the third simulation is consumed in the fourth simulation and meets the demanded electricity. It can be predicted that using a stop criterion, the system can balance itself for the following simulations with fluctuated during late night period production.

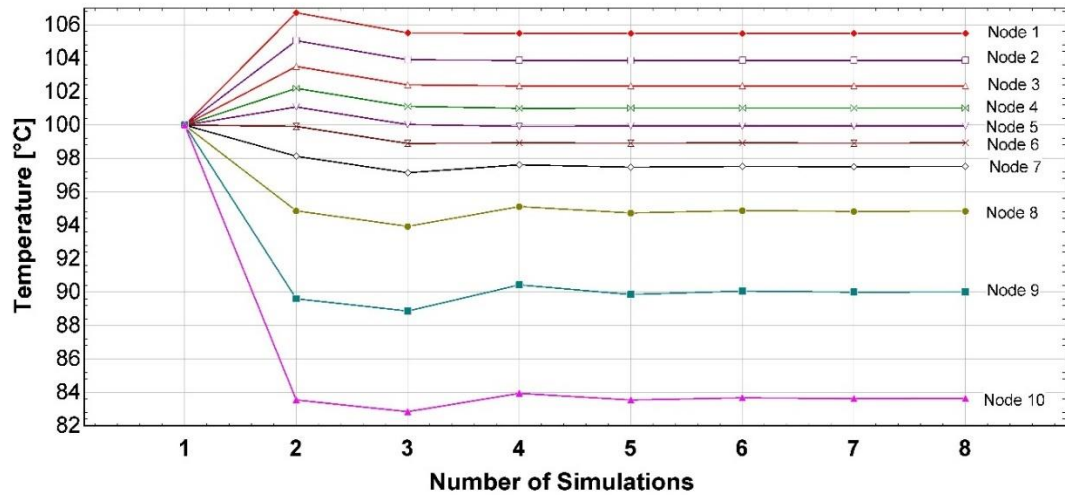


Figure 3.18 Initial temperature variations by the number of repetitive simulations

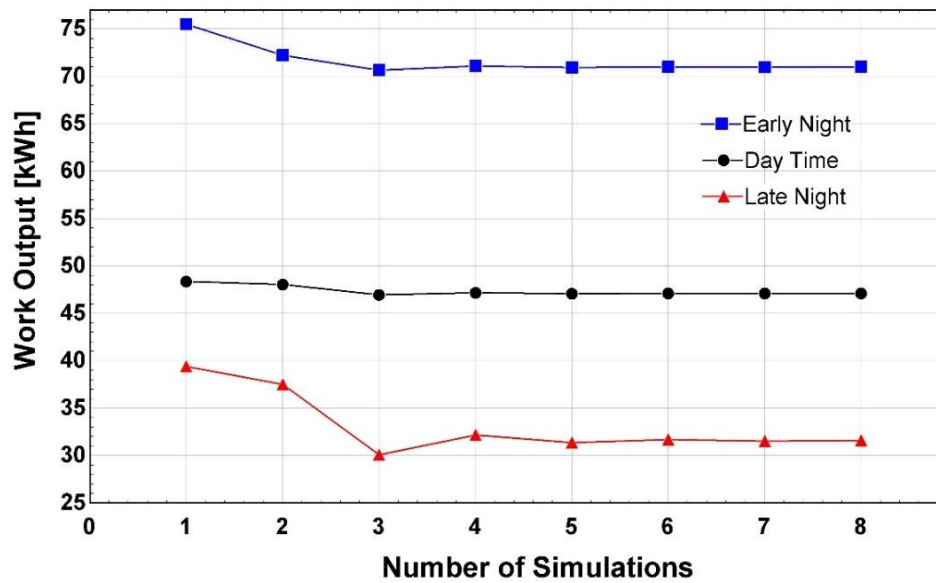


Figure 3.19 Variation of shaft output with number of repetitive simulations

After determination of the initial temperatures, the system is ready for the investigation. Figure 3.20 shows the temperature distribution in the tank in hours. An interesting trend is observed between 08:00 and 10:45. Although collector output is

discharged into the first node, during the first half hour this only affects the last node. Later, other nodes are affected and finally, it gets mixed with the first node at 10:45. The reason for this trend is density difference. At the beginning, collector outlet temperature is only matched with the last node, however, later its temperature increases and systems operate as usual. The same phenomenon can be seen between 15:00 and 17:00 for all simulations.

The rest of the day has a similar trend with the Figure 3.16. The only difference is the period between 07:00 am and 08:00 am. During the last one hour, the system is switched to the static mode. It means the tank is only subjected to heat loss to the ambient.

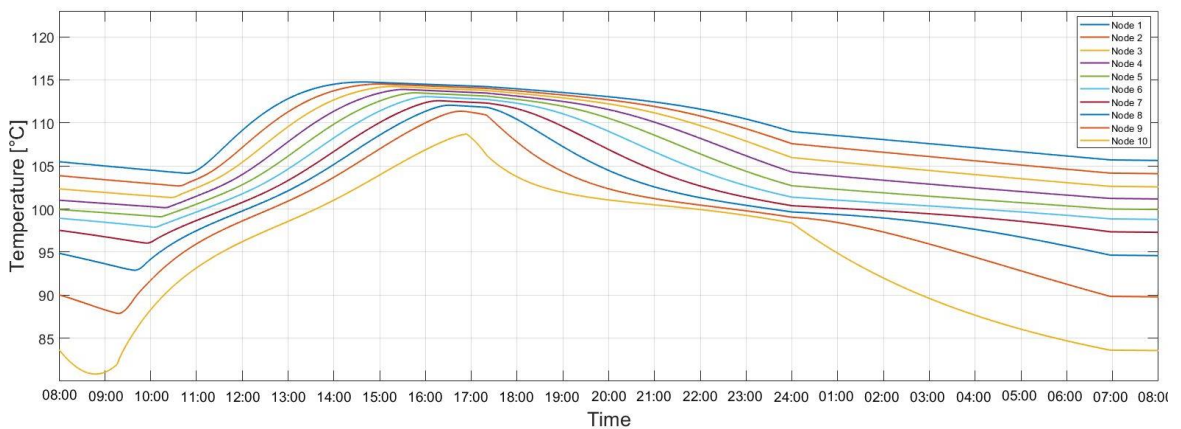


Figure 3.20 Temperature variations of the nodes for 24 hours operation

Figure 3.21 shows the work output of the system during 24 hours. The trend is quite similar with the Figure 3.17 but during the first two hours, the production is higher and more stable compared to it. One of the reasons is the temperature difference. Previously, all temperatures were assumed as 100 °C. However, the first node

temperature is determined as nearly 105°C, which results in a higher work output. Also, stable generation comes from the steady first node temperature which is already explained in the Figure 3.20. Moreover, it can be seen that work production is interrupted at 07:00 am because temperature of the middle node falls to 100 °C. The stop criterion is activated at that time, the work generation is interrupted to conserve the stored heat in the tank.

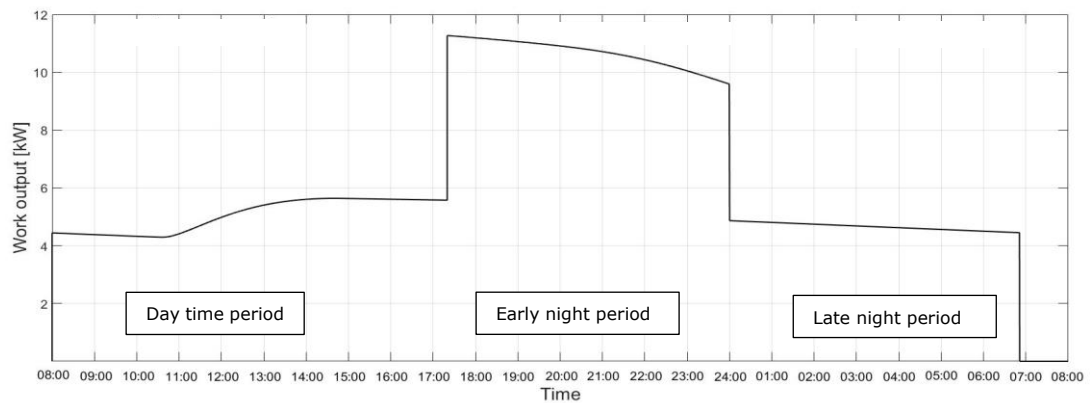


Figure 3.21 Shaft power output of the system for 24 hours

Figure 3.22 shows variation of the collected useful heat from the solar collectors, rejected heat for the ORC and heat loss to the ambient by time. The heat loss varies between 7.15 kW and 8.22 kW. These values are quite low compared to amount of collected heat. Evacuated flat plate collector's efficiency reaches maximum value of 0.68 during the operation. The amount of collected useful heat peaks at 12:40 and about 367 kW. The consumed heat for driving the ORC varies in different periods. During the day time period, it increases because water temperature of the first node is getting higher with higher solar irradiance. Then, it falls slightly as first node temperature is decreasing. During the early night period, evaporating temperature is increased. As a result of this increment, the consumed heat increases. In the late night

period, the evaporating temperature is controlled for the purpose of decreasing it again, and it yields to lower heat ejection from the water tank. Figure 3.22 also shows that all the useful collected energy is discharged during the simulation. This result makes the study more accurate because it eliminates the stored or excessive use of the energy in the tank. The initial temperatures has been chosen properly to avoid over-or underestimation of the work output for given typical conditions.

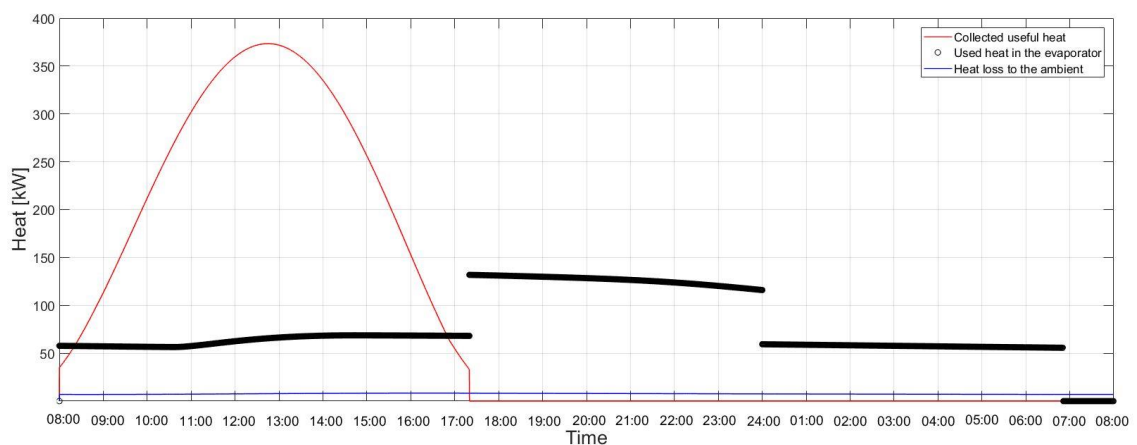


Figure 3.22 Collected heat, used heat for the ORC and heat loss from the tank

To evaluate the off-design performance of the system, performance of the expander during the second day has been analysed and it is shown in Figure 3.23a. During the daytime and late night periods, the isentropic efficiency of the expander varies slightly. Referring to the Figure 3.6, since expander operation pressure difference range at these periods are close to expander design pressure ratio (low evaporating temperature despite higher water temperature during these periods), its performance is higher. However, during the peak period, it falls below 0.63 because evaporating temperature is forced to increase by the present model for controlling the expander output.

According to off-design performance of the expander, this control strategy looks proper because peak period takes only 7 of 24 hours, remaining hours system operates at the very close range of the expander's maximum performance.

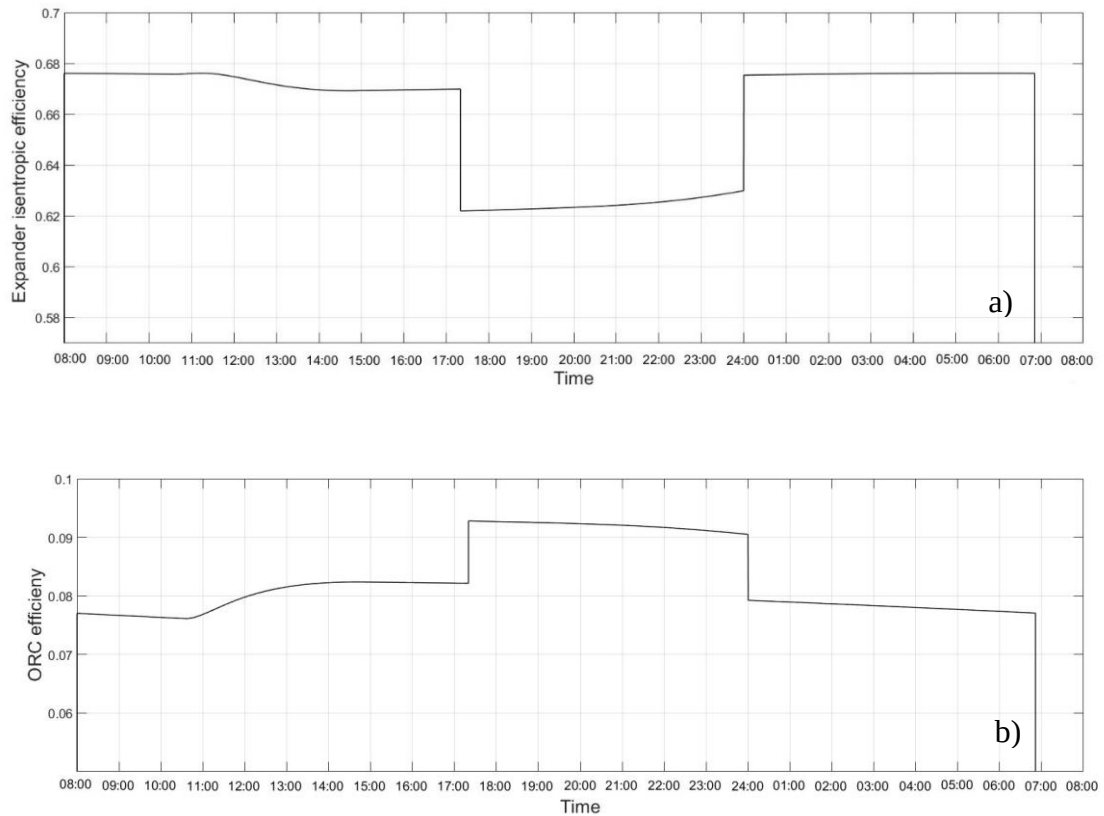


Figure 3.23 a) Expander isentropic efficiency variation. b) ORC thermal efficiency variation for 24 hours

ORC efficiency is also useful metric for evaluation of the system performance. It is related with some parameters but in the present study, main factor is evaporating temperature which is higher during the peak time period. Figure 3.23b shows the ORC efficiency for second day. In the other periods the evaporating temperature is forced to decrease by the present model. The main purpose is to avoid using the heat source

excessively and of course to meet the demand. The efficiency variation is observed between 0.076 and 0.092.

3.5 Annual performance of the solar ORC system

The solar ORC system using EFP collectors is a promising technology but dynamic simulation on its annual performance has not been conducted yet. According to the literature review, previous studies have used average monthly weather data to analyze the annual performance of systems in general. However, in this section, real daily weather data have been used and simulations have been conducted according to a transient model and considering hourly electricity demand. This analysis also focuses on controlling the ORC operations related to weather conditions and output requirements such as supporting heat during the winter months.

In order to conduct annual simulation, followed methodology is given in Figure 3.24. First, EnergyPlus weather data is collected for one year. As it is known that ambient temperature has an influence on condensing temperature, monthly control strategies are developed according to weather conditions. Moreover, daily control strategies are developed considering the demands.

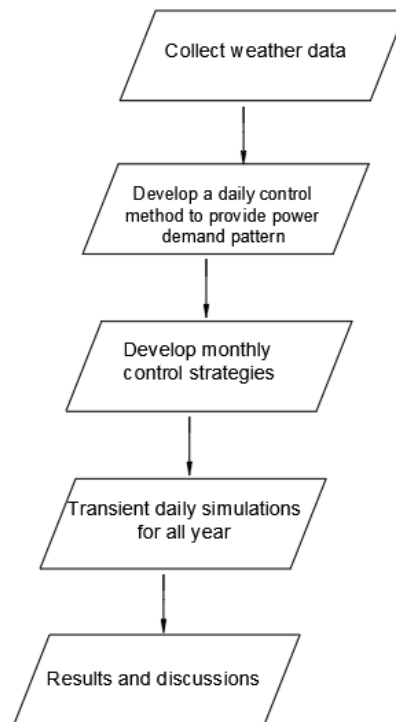


Figure 3.24 Procedure of the annual simulation

3.5.1 Weather data

The yearly weather data in this chapter relates to the climate conditions of Istanbul, Turkey. Relevant parameters such as solar irradiance and ambient temperature profiles are obtained by using the software EnergyPlus [173]. Figure 3.25 and Figure 3.26 show solar irradiance and environment temperature variations during the year, respectively. Istanbul is chosen for the analysis given that its irradiance profile is similar to some Mediterranean cities in summer but in winter, it is colder than in most other cities around the Mediterranean Basin. As seen in Figure 3.25, solar irradiance profile is intermitted during the summer period, which contributes to testing the performance of the system under variable operating conditions. Moreover, average solar irradiance is quite low in the winter period, being around 200 W/m^2 . The system is not able to produce satisfactory electricity under this irradiance values. Therefore,

winter mode (CHP mode) can be activated to produce useful condensing heat for space heating purposes.

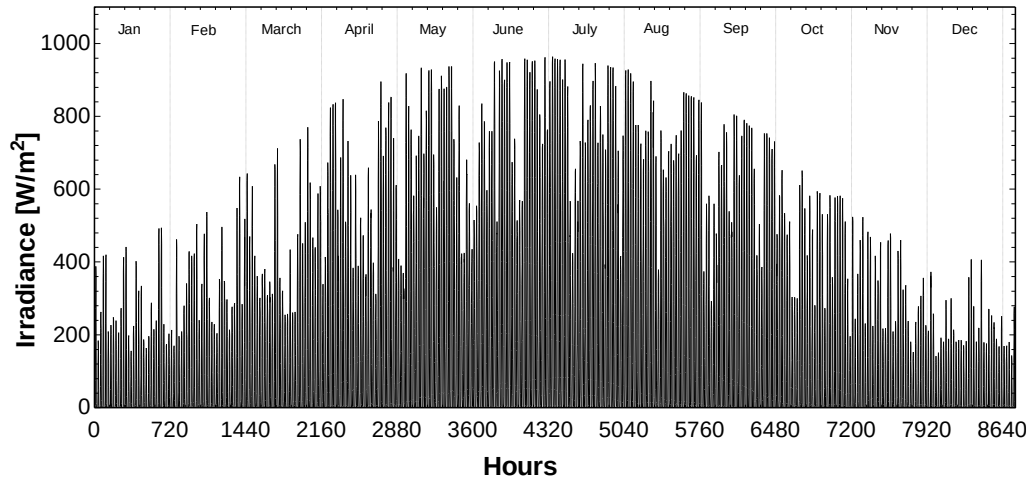


Figure 3.25 Solar irradiance variation during the year

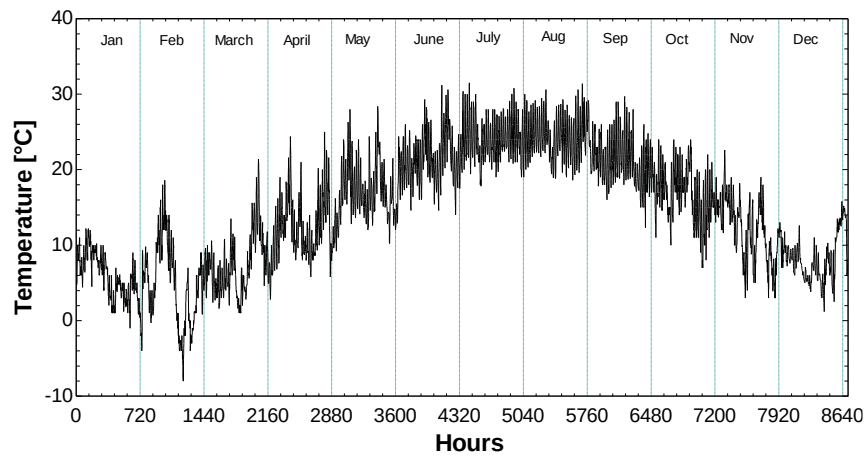


Figure 3.26 Ambient temperature variation during the year

3.5.2 Control strategies

The main aim of this analysis focuses on application of the control strategies. To maintain the demand profile which is given in previous section, 24hrs of a day is

analyzed in three different periods then heat source water circulation mass flow rate is adjusted for every period. Adjusted mass flow rates contribute to controlling the evaporating temperature, that is to say, controlling the electricity output [98]. Therefore, regarding the expander model, mass flowrate of the refrigerant R245fa is determined by the evaporating temperature. Collector mass flow rate is also adjusted according to weather conditions. In order to reach a higher temperature in the top part of the water tank, collector mass flow rates are reduced in winter months. To maintain similar operation for next days, a stop criterion must be defined. The middle water element in the tank is chosen as stop consideration; this activates to refrigerant pump is shut off and the tank is switched to static mode until 08:00 the next day. Condensing temperature is dependent upon the ambient temperature except combined heat and power (CHP) mode.

In CHP mode, the target is to produce electricity only for collector pumps due to weak solar radiation. The condensing temperature is increased, and condensing heat can be used for space heating purposes. A summary of the monthly control strategies is given in Table 3.3.

Table 3.3 Summary of the control strategies in monthly basis

Months	$\dot{m}_{col}$ (kg/s)	T_{cond}	Stop Temp.	Water flowrate (kg/s) 08:00-18:00	Water flowrate (kg/s) 18:00-24:00	Water flowrate (kg/s) 24:00-08:00
1	0.01	45	80	2	2	-
2	0.01	45	80	2	2	-
3	0.01	20	80	1	2	0.5
4	0.015	25	85	1	2	0.5
5	0.02	30	85	1	2	0.5

6	0.02	35	95	1	2	0.5
7	0.02	35	95	1	2	0.5
8	0.02	35	95	1	2	0.5
9	0.015	35	85	1	2	0.5
10	0.01	30	85	1	2	0.5
11	0.01	45	80	2	2	-
12	0.01	45	80	2	2	-

3.5.3 Annual simulations

For annual simulations, same collector area and design conditions are used. Although the main aim is conducting the yearly simulation considering the transient behavior, application of the control strategies are necessary for meeting the user electricity demand. The operation follows the given strategy; the collector pump runs, and solar energy is stored in the tank, meanwhile the ORC produces electricity between 08:00 and 18:00. When sun sets, peak demand period starts, water flow rate is adjusted to provide higher output until 24:00. The last period is from 24:00 to 08:00. During the last period, the circulation water mass flow rate is switched to 0.5 kg/s, as generation of a high amount of electricity is not required. Simulations are conducted considering both aforementioned strategies and also every per day's irradiance profile.

Figure 3.27 shows daily produced electricity as kWh. Four months (namely November, December, January and February) are adjusted for CHP mode so output is quite low. Additionally, large fluctuations are observed throughout the year regarding electricity outputs. These fluctuations occur as the solar irradiance does not follow a stable trend naturally. The control strategy allows these sharp decrements for

conserving the thermal energy in the tank for following day. Stop criterion is activated when condition is provided thus, the ORC pump is turned off and electricity production ends for this certain day.

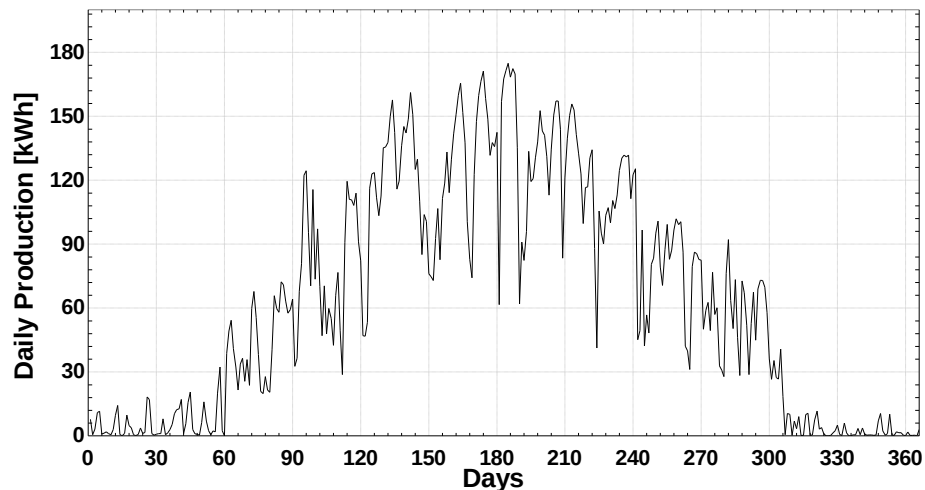


Figure 3.27 Daily produced electricity

To understand the specifications of CHP operation, Figure 3.28a is given to compare CHP and sole power generation modes. It is clearly seen from the figure that electricity output is lower in CHP mode. The main reason is a higher condensing temperature which causes a lower expansion ratio in the expander. However, high condensing temperature allows the use of condensing heat for other purposes for example space heating. Figure 3.28b shows produced condensing heat on a daily basis. The trend is quite similar with electricity generation as heat can only be obtained while the ORC is working. CHP mode is active for four months and received useful heat is recorded as 19344.7kWh.

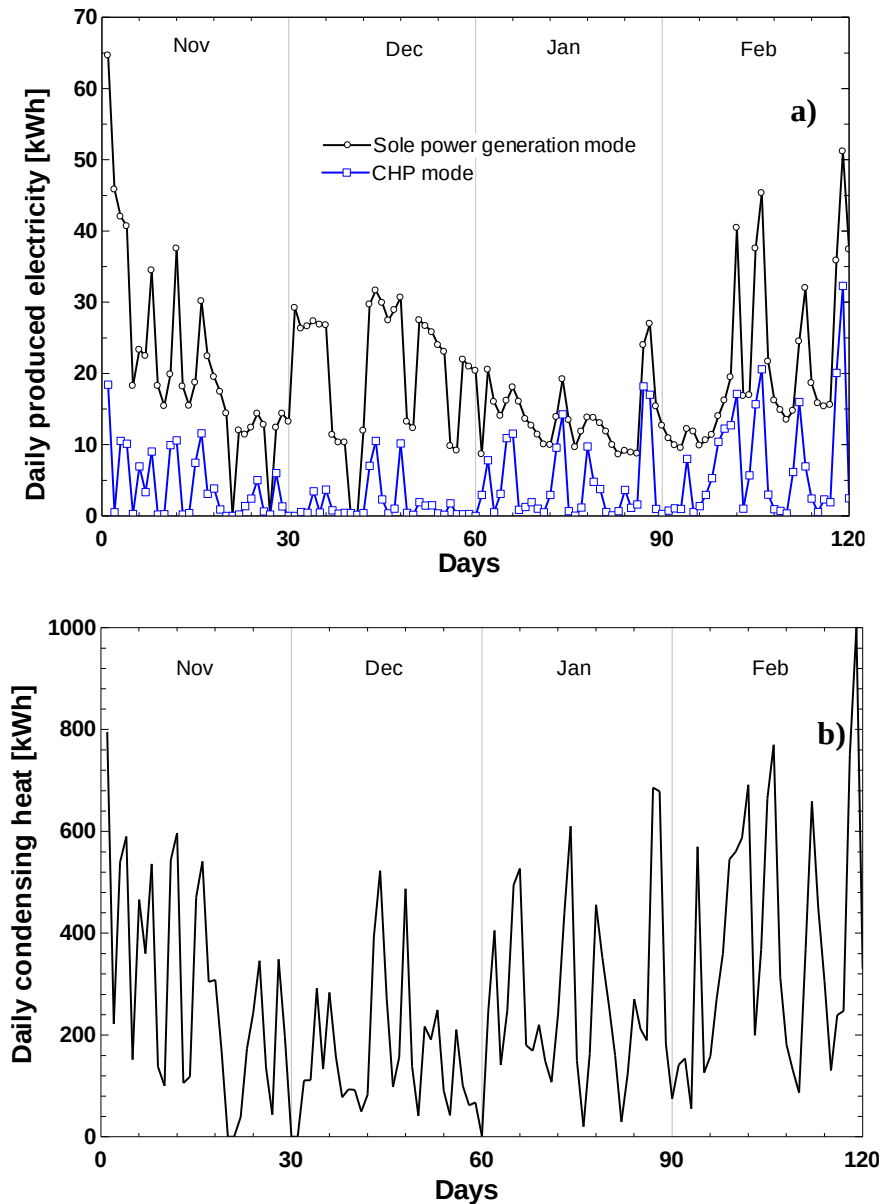


Figure 3.28 Comparison of outputs of the CHP and sole power generation modes (a) and condensing heat for space heating (b)

One of the aims of this study is matching the electricity output with demand during the operation. Therefore, electricity yield is given in Figure 3.29. The figure shows the effect of the CHP mode on production in the winter period and as expected, higher output in summer period. In order to clearly observe the output trend and continuous production profile, there should be zoomed on a specified month. Figure 3.30a

presents the electricity production profile during April. Since April is a transition month, the output profile is also of a similar nature. From April, continuous (24h) production begins to form. Figure 3.30b shows that the 24hr production is observed for 11 days. When focusing on the last 8 days, continuous production and controlling hours can be seen clearly.

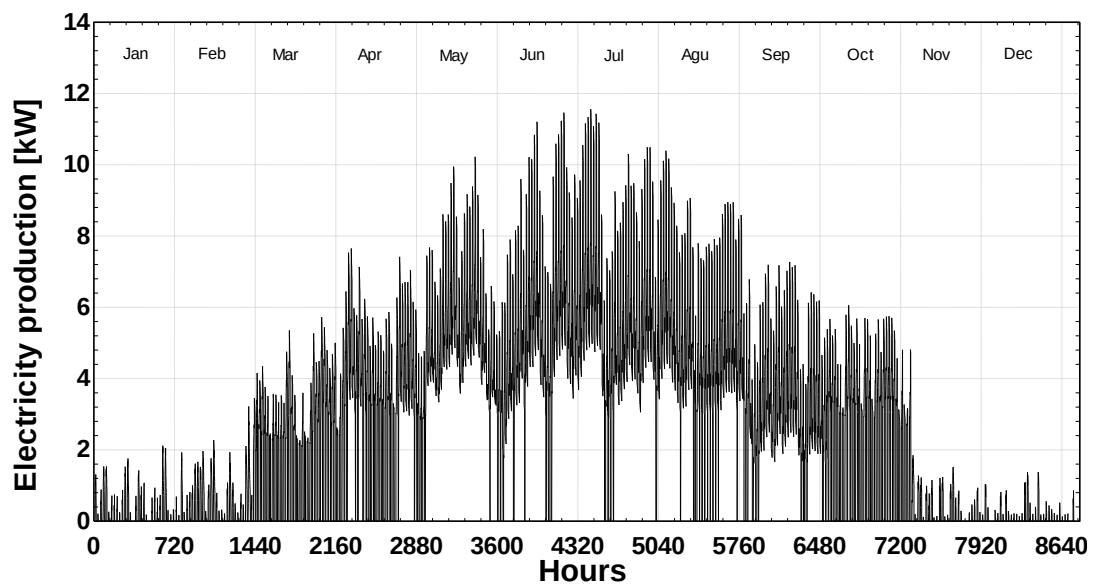


Figure 3.29 Hourly produced electricity

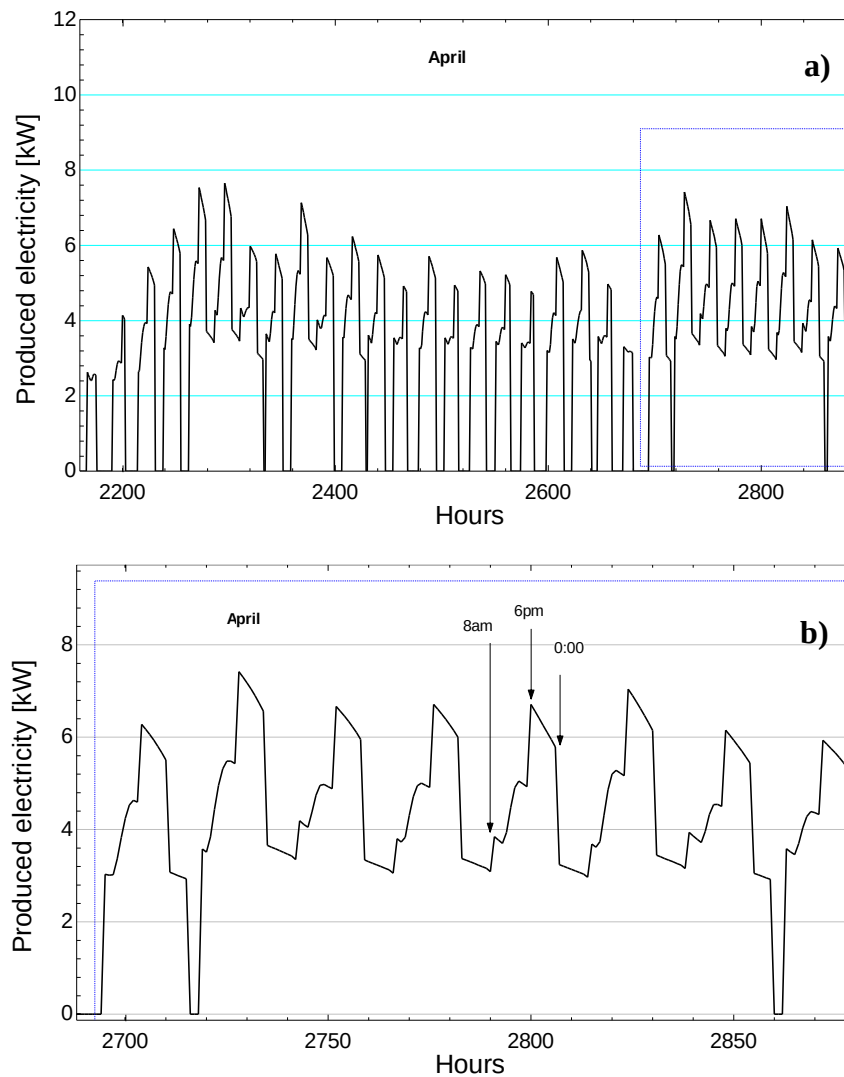


Figure 3.30 April hourly electricity production

The last figures are plotted to explain different outputs. June and September electricity generations and water tank top temperatures are compared in Figure 3.31a. Since these two months have the same condensing temperature, the only effecting parameter on output is the evaporating temperature which is directly related with top temperature. Thus, it is obvious that three days have almost the same output and top temperature in Figure 3.31b.

Heat losses to the environment from the tank are also considered, compared and calculated as 5251 kWh during June and 4451 kWh during September. The difference is approximately 15% but produced electricity difference falls by almost 40%. This figure shows wasted heat from heat loss is a serious problem which cannot be neglected as even the ORC stops, heat loss from tank to environment never ends until to be same temperature with ambient.

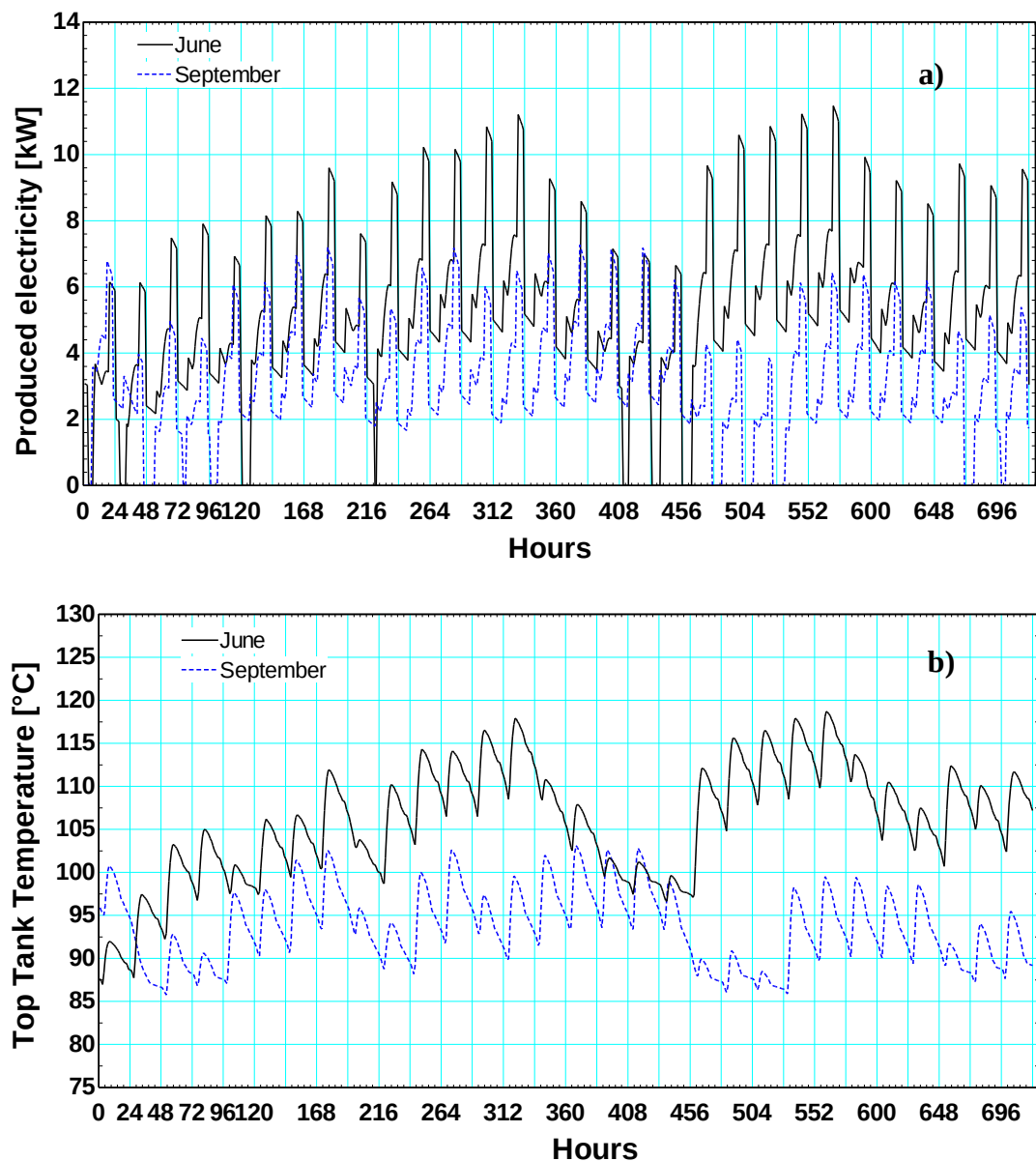


Figure 3.31 Comparison between June and September electricity outputs (a) and top tank temperatures (b)

3.6 Summary

In this chapter, a research into off-design performance of a solar ORC system integrated with a compressed water heat storage unit has been conducted based on fulfilment of the end user variable demand during a day from the point of view of control strategies. The analysed system combining the evacuated flat plate collector and the heat storage unit to provide all day power generation offers good results. The heat storage unit has been analysed using a one-dimensional temperature distribution model to represent the thermocline phenomena. However, it is known that lots of parameters affect the thermocline, so a more complex model may result in more accurate findings. Nonetheless, there is no doubt that this simplified stratification model gives more realistic results than the fully mixed uniform model. Moreover, a proper initial tank temperature distribution has been determined by repeating simulation several times in order to conduct a proper daily simulation analysis under given conditions.

This chapter shows that power output can be adjusted by controlling the mass flow rate of the circulation water and it is possible to meet electricity demand at night. The ORC has been successfully simulated at variable heat source temperature by use of sliding pressure control strategy. Throughout the simulation, by using 550 m² collector area, the power output was ranged from 4.3 to 5.7 kW in the daytime, 9-11.2 kW at early night and 4.7-4.3 kW at late night via adjustment of water mass flow rate in the evaporator of ORC.

Moreover, in this chapter, a solar-driven Organic Rankine Cycle was investigated for a year with consideration given to meeting electricity demand. Components were modelled and analyzed in a transient state for the whole year. When weak solar irradiance was available in the winter period, operation was switched into CHP mode

and the system produced heat and power simultaneously. Weather conditions were used as real data and performance was predicted under real conditions. The system also focused on control strategies. On the basis of the presented study, the following conclusions can be written:

- EFP collectors have been successfully implemented in the ORC system even for low irradiance periods for CHP operation.
- Although solar ORC performance totally depends on an irradiance profile, by controlling the system operation, it is possible to conserve the stored heat for next day with dispensable late-night production. However, this intermitted generation causes sharp falls in daily electricity production.
- 24hrs production needs sufficient solar irradiance. According to simulation 24hrs production occurs by April, but it depends on weather conditions.
- Electricity output depends on control strategies. Stop criteria assignment is one of the main parameters to obtain magnitude of output and production time in daily basis.
- Using 550 m² collector area, 23296.8 kWh electricity has been produced considering the demand pattern for twelve dwellings in year. Adjustments of the circulation flow rate covers all year to match or to maintain a similar trend of demand profile. If we consider the annual required electricity and annual produced electricity, the fraction is found around 37%.

CHAPTER 4 Incorporation of an innovative a-Si PV/T system with an organic Rankine cycle

4.1 Introduction

The solar organic Rankine cycle (s-ORC) has a great potential for meeting the residential electricity demand. Its flexible operation capacity with easier collection and storage ability make it promising technology in sustainable energy studies. By combining with a heat storage unit, power generation is not intermitted from solar radiation alterations and it is possible to continue power generation during night. Moreover, the s-ORC allows to adjust produced electricity according to resident's electricity demand [98].

The advantages of s-ORC systems are restricted by its efficiency, which is lower than the main stream PV efficiencies. Studies show that concentrating collector systems can reach higher operating temperatures and high operating temperatures result with better thermal solar to electricity efficiency. Commonly used evacuated tube collectors (ETCs) and flat plate collectors (FPCs) are suitable and preferred for solar heat collection at temperature below 100 °C. Concentrators such as compound parabolic concentrators (CPCs) and parabolic trough collectors (PTCs) can hardly use diffuse radiation and may have a high cost for the tracking system. To overcome the given challenges, evacuated type flat plate collector (EFPCs) have been recently used for s-ORC applications [140], [141], [98], [174], [99], [175].

In order to use advantages of the s-ORC and PVs in a system, an innovative PV/T-ORC system is proposed in this chapter. The advantages of the PV/T-ORC system using a-Si cells and EFPCs can be summarized below:

- Encapsulation for the PV cells can be avoided. Operating at medium-high temperature is challenging to solar cells. However, it is the encapsulation rather than solar cell suffers from medium-high operation. In the vacuum chamber, there is no need for encapsulation and a long lifetime of a-Si cells is expected.
- a-Si cells will benefit from thermal annealing at medium-high temperature. In fact, on heating at 150 °C, a-Si cells could reverse the Staebler-Wronski effect [32].
- Thermal storage using water is adopted for flexible power generation. The storage cost is relatively cheaper as compared with battery.
- Combining the EFPC with a-Si cells can result in an increment of annual electricity yield and significant decrement in payback period. So the integration is also beneficial for the s-ORC.

To prove these statements, a performance comparison is made between four different systems in this chapter. The models are built and investigation of the systems has been conducted from the point of view of electricity output and its profile. The first one is the proposed a-Si PV/T-ORC system, in which the EFPCs are covered with a-Si cells with a 0.85 cover ratio. A heat storage unit is also included. The second one is the s-ORC by using EFPCs for thermal power conversion only. The third is the PV/T-ORC system using poly-Si cells. The last one is stand-alone PV systems. These systems will be compared according to their thermal and electrical efficiencies and power outputs considering thermocline behaviour of the water tank in transient state for moderate and high irradiance weather data.

4.2 System description

In this section, the proposed systems' descriptions will be given. The schematic of s-ORC is shown in Figure 4.1 which is composed of solar field, heat storage unit and ORC block. EFP collectors are used for heating water which comes from the bottom of the tank, then is discharged into the top of the water tank. A water tank is used as a heat storage unit to provide heat for the ORC. Moreover, it allows ORC to be driven when solar radiation is insufficient or unavailable. In the ORC block, there are four main components namely pump, evaporator, expander and condenser.

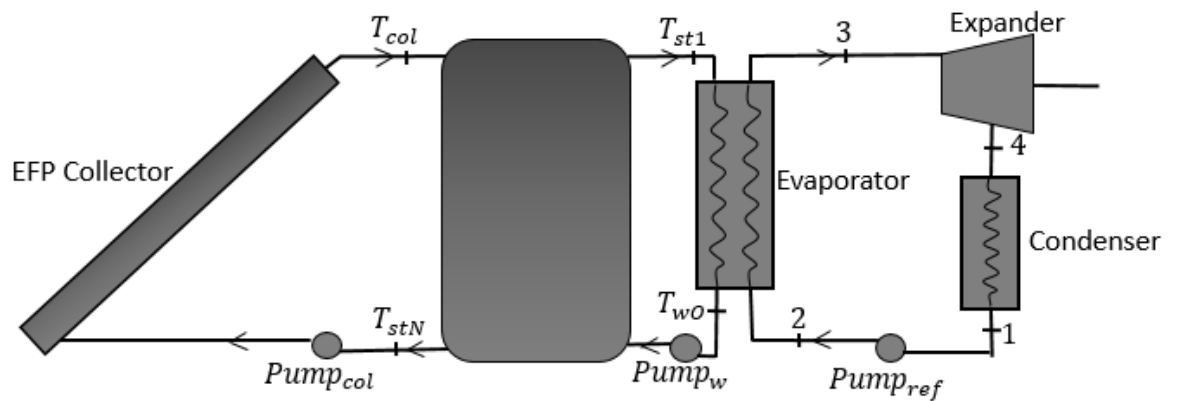


Figure 4.1 System schematic of s-ORC system

For stand-alone PV system, same collector area is taken, and poly-Si modules are placed on the solar array. For PV/T-ORC systems, heat storage and the ORC block are kept the same but only EFP collectors are replaced with PV/T collectors.

4.3 Characteristics of a-Si PV cell

In the stand-alone PV system commonly preferred poly-Si type modules have been chosen. Electrical characteristics of the modules are taken from the manufacturer's

data and presented in Table 4.1. The most common equation to determine the cell temperature is using Normal Operating Cell Temperature (NOCT). This value is given by the manufacturer and Eq. (4.1) is used for calculating the cell temperature [176].

$$T_{cell} = T_{am} + (NOCT - 20) \cdot \frac{G}{800} \quad (4.1)$$

Photovoltaic module efficiency is calculated by most known model:

$$\eta_{PV} = \eta_r [1 - \beta_r \cdot (T_{cell} - T_r)] \quad (4.2)$$

The parameters are given by producer and η_r indicates efficiency at reference temperature. Global electrical efficiency of the PV system can be calculated as Eq. (4.3):

$$\eta_{gPV} = \eta_{PV} \cdot \eta_{DC-AC} \quad (4.3)$$

Where η_{DC-AC} is the conversion efficiency of DC to AC and is reported as between 80-95% depending on solar irradiance [177], [178]. For $< 300 \text{ W/m}^2$, η_{DC-AC} is 80%. For $300 \text{ W/m}^2 < G < 500 \text{ W/m}^2$, η_{DC-AC} is 85%. For $500 \text{ W/m}^2 < G < 800 \text{ W/m}^2$, η_{DC-AC} is 90%. For $> 800 \text{ W/m}^2$, η_{DC-AC} is 95%.

Table 4.1 Some of the PV module technical specifications [179]

PV		Specifications
Model		Atersa A-214P
Type		Polycrystalline Silicon
Module efficiency	η_r	12.64 %

Power temperature coeff. β_r	-(0.46)%K
NOCT	47 °C

Regarding the a-Si cell performance, as power temperature coefficient is not constant [30], experimentally obtained results have been used [36]. Figure 4.2 gives electrical efficiencies of a-Si and poly-Si cells, to calculate the poly-Si efficiency Eqs. (4.1-4.2) and Table 4.1 have been used. Figure 4.2 presents that electrical efficiency lines intercept at 98 °C; this means that using the a-Si cells in PV/T at an operation temperature of over 98 °C has better electrical performance than using poly-Si cells in PV/T. This figure supports the aim of the recent study.

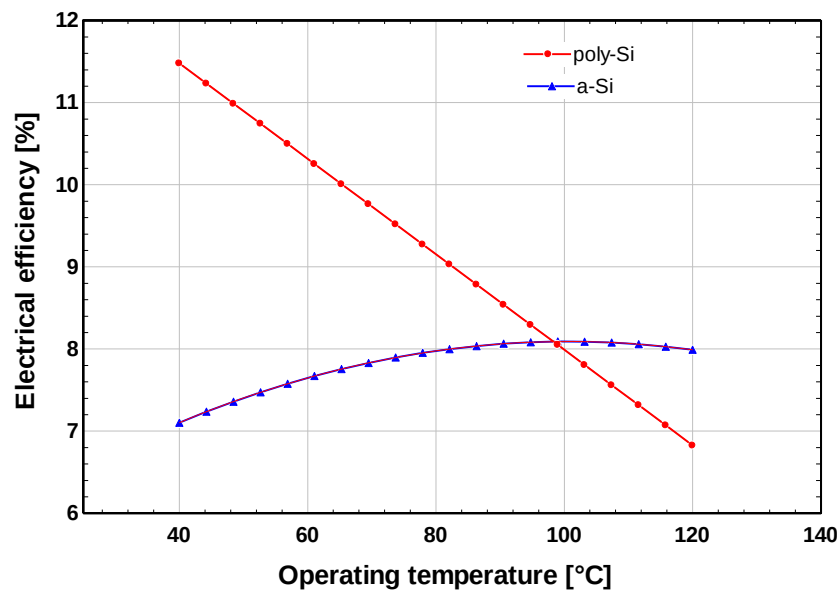


Figure 4.2 Electrical efficiency variations of a-Si and poly-Si cells [36]

4.4 Thermodynamic modelling

4.4.1 PV and PV/T system models

To obtain PV/T collectors from EFP collectors, PV cells are added. TVP SOLAR HT-Power EFP collectors [155] have been used in s-ORC system and also modified PV/T collectors in PV/T-ORC systems. EFP collector structure is shown in Figure 4.3.

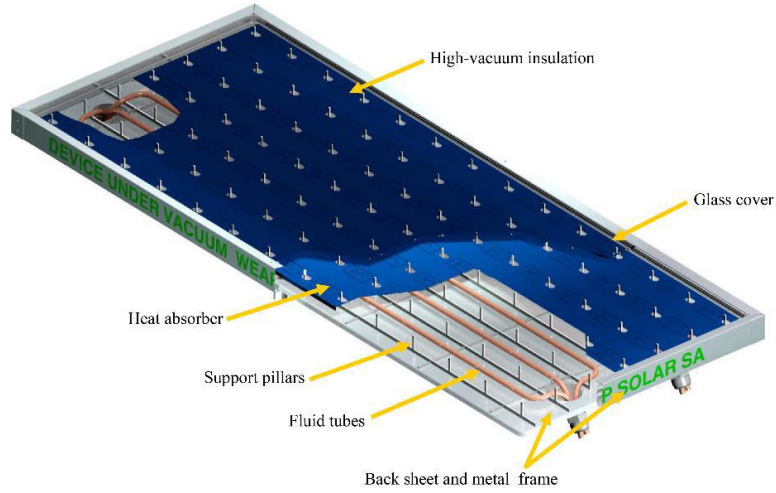


Figure 4.3 Structure of EFP collector [155]

A general equation for thermal efficiency of a solar collector is given in Eq. (4.4). T indicates mean temperature of water in the collector, G indicates solar irradiance, A and B are heat loss coefficients. For proposed EFP collectors η_0 is 74.62, A and B are 39.9 and 0.67, respectively [145].

$$\eta_{col} = \eta_0 - A \cdot \frac{T - T_{am}}{G} - B \cdot \frac{(T - T_{am})^2}{G} \quad (4.4)$$

The power output of PV cell is calculated by Eq. (4.5):

$$W_{PV} = \eta_{PV} \cdot G \cdot A_{col} \cdot x \quad (4.5)$$

x is cover area ratio of PV and collector area and it is taken as 0.85. The heat collection efficiency of a PV/T collector can be deduced [36] as :

$$\eta_{PVT} = \eta_{col} - \eta_{PV} \cdot x \quad (4.6)$$

$\eta_{PV}(T)$ is assumed as second order equation and can be expressed by

$$\eta_{PV} \cdot x = \eta_{PV,0} + a \cdot T + b \cdot T^2 \quad (4.7)$$

For PV/T collector thermal efficiency the equation can be written as

$$\eta_{PV/T} = \eta'_{c,0} - A' \cdot \frac{T - T_a}{G} - B' \cdot \frac{(T - T_a)^2}{G} \quad (4.8)$$

$\eta'_{c,0}$, A' and B' are the modified coefficients of the collector.

$$\eta'_{c,0} = \eta_{c,0} - \eta_{PV,0} - a \cdot T_a - b \cdot T_a^2 \quad (4.9)$$

$$A' = A + (a + 2 \cdot b \cdot T_a) \cdot G \quad (4.10)$$

$$B' = B + b \cdot G \quad (4.11)$$

By applying given deductions, the thermal efficiency equations for poly-Si and a-Si PV/T collectors are obtained.

4.4.2 Modelling of the expander by using semi-empirical model

In this chapter, semi-empirical model of the scroll expander is chosen. Semi empirical model is more flexible compare to empirical model which was used in the previous chapter. By this model, working fluid can be changed and analysis will be more detailed by including heat transfer from expander shell to ambient. The semi-empirical model for scroll expanders was developed by Lemort et al. [107] it is the most preferable model recently. Schematic of the model is given in Figure 4.4. As can be

seen from figure, great deal of stages are included in the whole thermodynamic process.

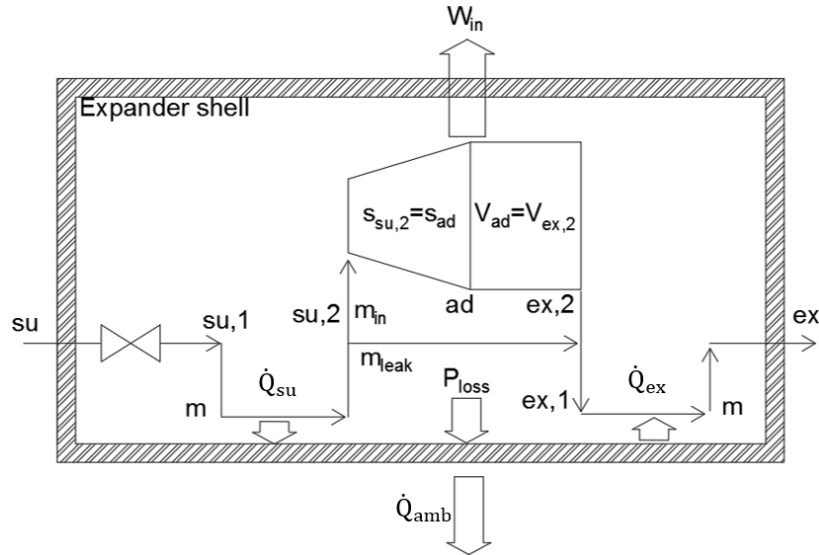


Figure 4.4 Semi empirical expander model adapted from the reference [107]

Lemort's model consist of several hypothetical stages.

- 1) Adiabatic pressure drops in supply side (su → su,1):

This process is related with pressure drop between the expander inlet and suction chamber. Subscript su indicates supply conditions. Cross sectional area A_{su} is used for an isentropic flow through nozzle.

$$\dot{m} = \rho_{su,1} A_{su} \sqrt{2(h_{su} - h_{su,1})} \quad (4.12)$$

- 2) Isobaric supply cooling down (su,1 → su,2):

There is a heat transfer occur between the fictitious envelop and refrigerant.

$$\dot{Q}_{su} = \dot{m}(h_{su,1} - h_{su,2}) = \left[1 - e^{\frac{AU_{su}}{\dot{m}c_p}}\right] \dot{m}c_p(T_{su} - T_w) \quad (4.13)$$

$$AU_{su} = AU_{su,nom} \left(\frac{\dot{m}}{\dot{m}_{nom}}\right)^{0.8} \quad (4.14)$$

$U_{su,nom}$ is nominal supply heat transfer coefficient and T_w is wall temperature which is referred to expander shell temperature.

3) Internal leakage (su,2→ ex,2):

Entered mass flow is divided into two parts in the expander. Total mass flow rate is given in Eq. (4.15). Total mass flow is described as sum of the theoretical mass flow which depends on rotation speed, swept volume and inlet conditions, and leakage flow.

$$\dot{m} = \dot{m}_{th} + \dot{m}_{leak} \quad (4.15)$$

$$\dot{m}_{th} = \rho_{su,2} V_{sw} \frac{N}{60} \quad (4.16)$$

Leakage flow paths inside the expander are assumed as a lumped hypothetical area of A_{leak} . This area is used in Eq. (4.17) to calculate leakage flow which is assumed as an isentropic nozzle flow.

$$\dot{m}_{leak} = \rho_{leak} A_{leak} \sqrt{2(h_{su,2} - h_{leak})} \quad (4.17)$$

In order to calculate throttling conditions in isentropic nozzle flow, these conditions are shown as subscript leak in the Eq. (4.17), critical pressure $P_{crit,leak}$ is used. When expander outlet pressure is higher than critical pressure, this time expander outlet pressure becomes critical pressure to calculate leak properties. To calculate the critical pressure, Eq. (4.18) is used by assuming working fluid as a perfect gas. γ indicates the ratio of heat capacity at constant pressure and heat capacity at constant volume.

$$P_{crit,leak} = P_{su,2} \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma-1}} \quad (4.18)$$

4) Adiabatic and reversible expansion to the adapted pressure (su,2 → ad):

Expansion of the volume depends on characteristics of expander. Built in volume ratio defines the adapted volume of the expander.

$$v_{ad} = BVR v_{su,2} \quad (4.19)$$

5) Adiabatic expansion process at constant machine volume (ad → ex,2):

At this stage, under and over expansion losses occur depending on the adapted pressure. In order to equalise these pressures of adapted and expander outlet, the model assumes that some fluid flows out of or into the discharge chamber instantaneously when the expansion chamber opens onto the discharge line. As it is known expander performance depends on operating conditions and most times expander exposes to off-design operation. When expander exhaust pressure is higher than adapted pressure it is called over expansion. Pressure–Volume diagrams for under and over expansion are given in Figure 4.5.

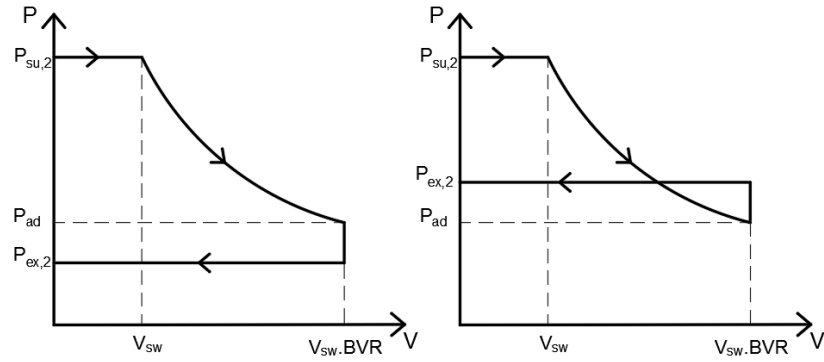


Figure 4.5 Schematic of under and over expansion adapted from [108]

6) Adiabatic fluid mixing stage (ex,2 $\rightarrow$ ex,1):

Theoretical flow rate and leakage flow are adiabatically mixed and mixture enthalpy is slightly increased. Eq. (4.20) can be used:

$$\dot{m} \cdot h_{ex,1} = \dot{m}_{th} \cdot h_{ex,2} + \dot{m}_{leak} \cdot h_{leak} \quad (4.20)$$

7) Isobaric exhaust cooling-down or heating-up (ex,1 $\rightarrow$ ex):

In this stage, heat transfer occurs between the fluid exiting the expansion chamber and the isothermal envelope. Similar with isobaric supply cooling down process.

After finding all states, generated power of the expander can be found.

The produced power is given in Eq. (4.21). This power is theoretical internal power of the expander. In order to calculate shaft output power, mechanical losses are considered.

$$\dot{W}_{in} = \dot{m}_{th}[(h_{su,2} - h_{ad}) + v_{ad}(P_{ad} - P_{ex,2})] \quad (4.21)$$

Mechanical losses are lumped to mechanical loss torque; the equation is given;

$$\dot{W}_{sh} = \dot{W}_{in} - \frac{2\pi N \tau_{loss}}{60} \quad (4.22)$$

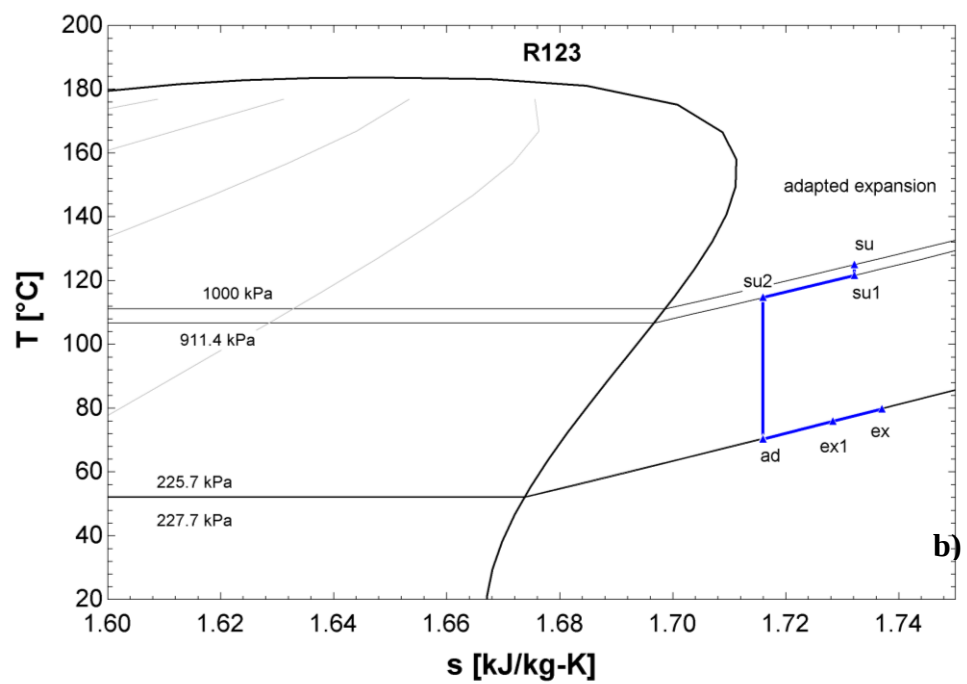
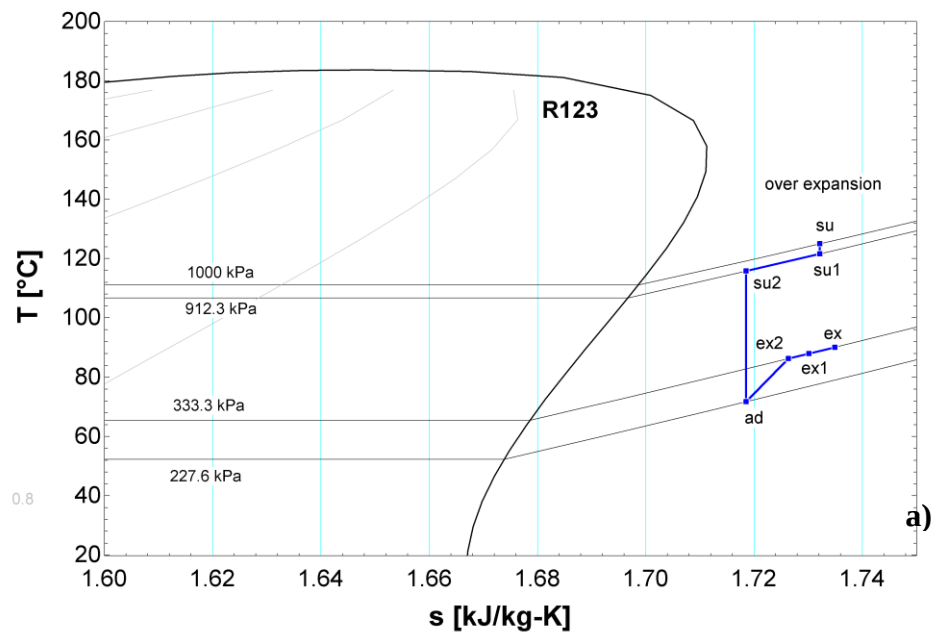
Another heat transfer is happened between the expander walls and ambient air.

$$\dot{Q}_{am} = AU_{am}(T_w - T_{amb}) \quad (4.23)$$

Overall heat balance equation is given in Eq. (4.24). By solving overall heat balance equations, all variables can be found.

$$\dot{W}_{loss} \pm \dot{Q}_{ex} + \dot{Q}_{su} - \dot{Q}_{am} = 0 \quad (4.24)$$

By using given equations and empirical parameters, expander performance can be obtained. Figure 4.6 is given to show stages during the expansion process of different cases such as, over expansion, adapted expansion and under expansion operations.



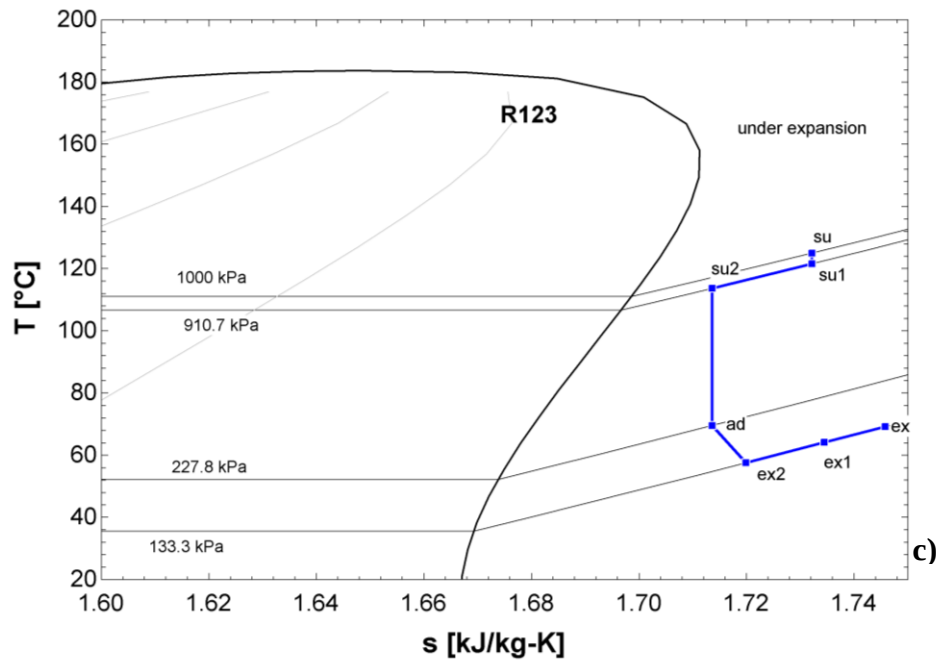


Figure 4.6 T-s diagrams to show theoretical stages of modelling for over-expansion
(a), adapted expansion (b) and under expansion (c)

Table 4.2 Empirical parameters for Lemort's semi empirical model [107]

1	Swept volume, ($V_{s,exp}$)	36.54 cm ³
2	Built in volume ratio, (BVR)	4.05
3	Leakage area, (A_{leak})	4.6 mm ²
4	Supply port cross sectional area, (A_{su})	27.43 mm ²
5	Nominal mass flow rate, ($\dot{m}_{nom}$)	0.12 kg/s
6	Heat transfer coefficient with ambient, (AU_{am})	6.4 W/K
7	Supply heat transfer coefficient, ($AU_{su,nom}$)	21.2 W/K

8	Exhaust heat transfer coefficient, ($AU_{ex,nom}$)	34.2 W/K
9	Mechanical loss torque, (T_{loss})	0.47 Nm

Required parameters were empirically identified by the authors which are given in Table 4.2. By using given equations and empirical parameters, scroll expander analysis is carried out. Figure 4.7 shows effect of pressure ratio on shaft power output for different rotation speeds. For given inlet conditions, pressure ratio can be only increased by reducing the condensing temperature. It is also seen that, rotation speed has an influence on the shaft power. Higher pressure ratio means higher expansion and yields higher output. Higher rotation speed also results in higher output. Although rotation speed is not a determined parameter in real systems, this approach requires to use a rotation speed as a known parameter. Normally, the expander shaft rotates according to inlet pressure as a driving force and frictions as counter forces. In order to make a reasonable power output prediction, rotation speed analysis should be investigated. But in this stage, rotation speed is used as a known and controllable parameter as modelling approach allows.

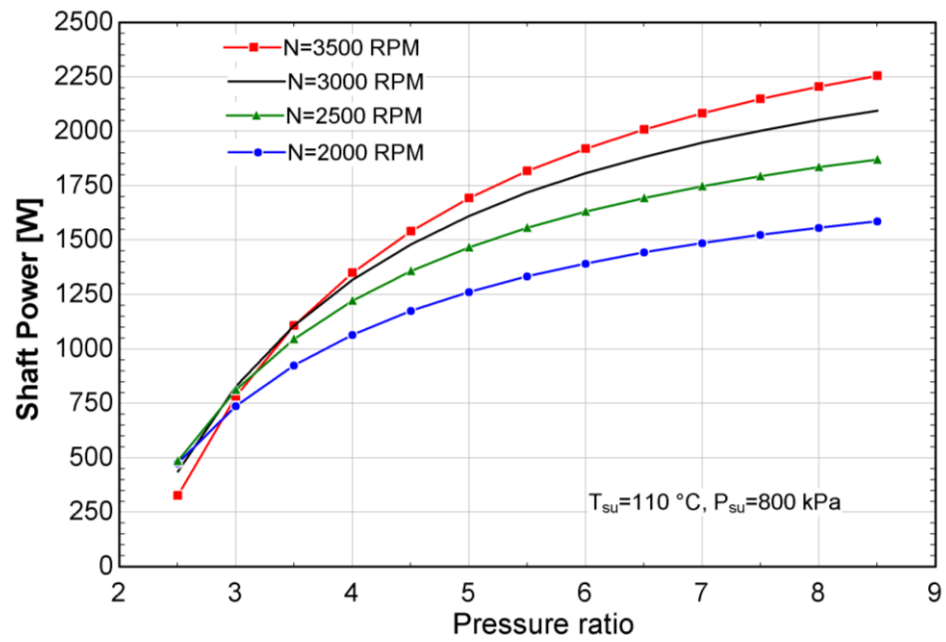


Figure 4.7 Effect of pressure ratio and rotation speed on shaft power

Isentropic efficiency variation can be found by using same input and output pressures and temperatures. Figure 4.8 shows effect of pressure ratio on isentropic efficiency. As expected, typical scroll expander performance curve is seen. Performance is good when operation close to adapted pressure range.

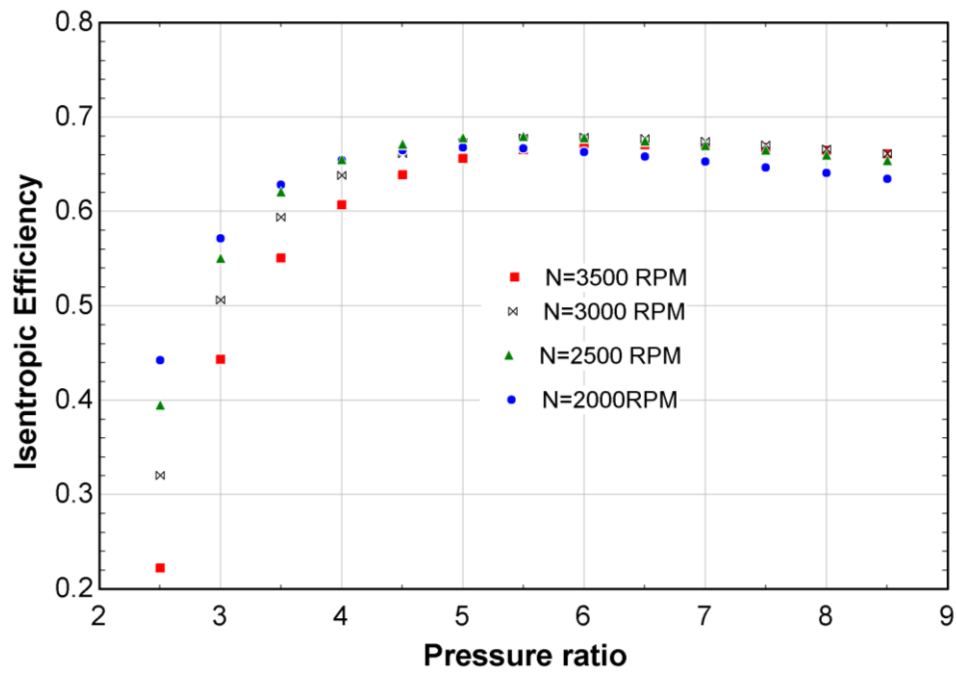


Figure 4.8 Effect of pressure ratio and rotation speed on isentropic efficiency

In order to investigate this modelling in detail, effect of rotation speed on other parameters are investigated. Figure 4.9 shows effect of rotation speed on expander shell and exhaust temperature. It also shows effect of inlet conditions to make a good understanding. By using the same pressure ratio between expander inlet and outlet, higher inlet temperature yields higher exhaust and wall temperature as expected. When rotation speed increases wall temperature increases slightly around 1°C and exhaust temperature almost remains same. This temperature rise is negligible for applications.

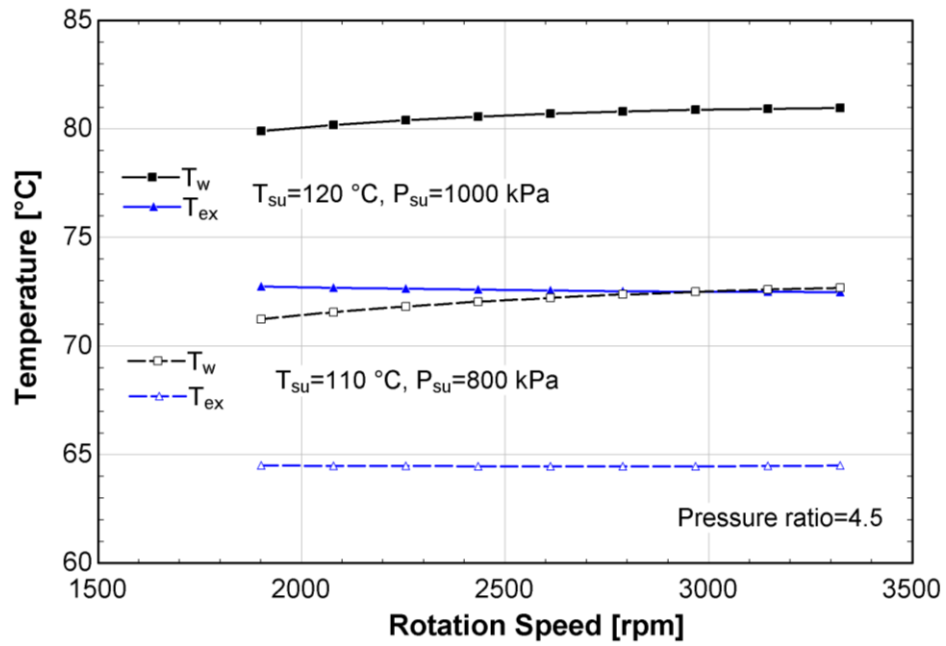


Figure 4.9 Effect of rotation speed on expander wall and exhaust temperatures

In order to explain temperature variation of the exhaust and shell temperatures, heat transfer rates are investigated. Figure 4.10 shows supply, exhaust and ambient heat transfer rates variation. Firstly, we should know that $\dot{Q}_{su}$ and $\dot{Q}_{ex}$ have different directions. When considering control volume of the system, $\dot{Q}_{ex}$ and $\dot{Q}_{amb}$ should be negative but they are given as magnitude to easily compare with supply heat. $\dot{Q}_{su}$ increases wall temperature by using refrigerant's high supply temperature but $\dot{Q}_{ex}$ decreases because it uses wall temperature to increase refrigerant exhaust temperature. High rotation speed yields higher flow rate of working fluid, this results in better heat transfer. Figure 4.10 shows $\dot{Q}_{su}$ and $\dot{Q}_{ex}$ increase with rotation speed. However, increment of $\dot{Q}_{ex}$ is faster than $\dot{Q}_{su}$ and also $\dot{Q}_{amb}$ increases slightly. As can be seen from the figure, supply heating rate increases around 0.05 kW on the other hand exhaust heating increases as double. Energy balance is maintained by the losses

because it is known that higher rotation speeds cause more friction and internal heating.

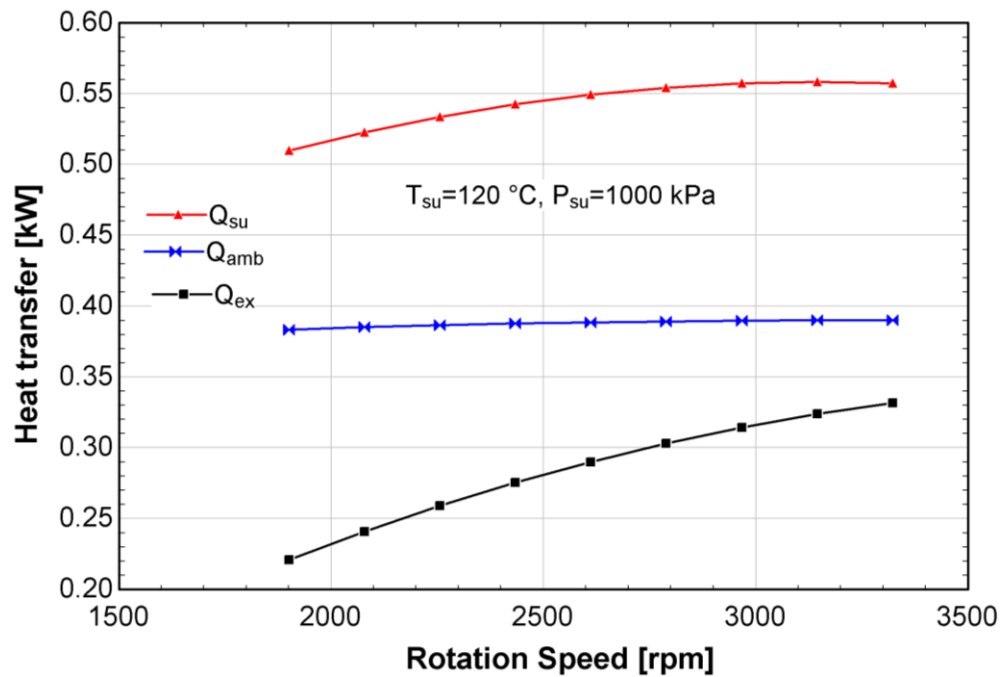


Figure 4.10 Effect of rotational speed on heat transfer rates

The parameters given in Table 4.2 are only valid for tested expander machine and can be used for working fluid R123. It is known that working fluid has an important effect on the system performance and different operating conditions may require different refrigerants to use. Therefore, Guiffarida [108] suggested an approach to model scroll expander for changing the working fluids by modifying the heat transfer coefficients which are given in numbers 7 and 8 in Table 4.2. Since other parameters are not dependent on the working fluid, the study only modifies parameters 7 and 8.

As known, thermal resistance can be calculated as Eq. (4.25):

$$U = \frac{Nu\lambda}{L} \quad (4.25)$$

Conductivity and Nusselt number depend on refrigerant type. Eq. (4.26) gives Nusselt number correlation:

$$Nu = 0.023Re^{0.8}Pr^m \quad (4.26)$$

$$Re = \frac{\rho w D}{\mu} \quad (4.27)$$

$$Pr = \frac{c_p \mu}{\lambda} \quad (4.28)$$

Hydraulic diameter and characteristic length are geometric parameters of scroll, additionally, velocity is depending on geometry. When heat transfer areas taken into accounts:

$$\frac{AU_{fluid}}{AU_{HCFC-123}} = \frac{Nu_{fluid} \lambda_{fluid}}{Nu_{HCFC-123} \lambda_{HCFC-123}} \quad (4.29)$$

And finally:

$$\begin{aligned} AU_{fluid} \\ = AU_{HCFC-123} \left(\frac{\rho_{fluid}}{\rho_{HCFC-123}} \right)^{0.8} \left(\frac{\mu_{HCFC-123}}{\mu_{fluid}} \right)^{0.8-m} \left(\frac{c_{p_{fluid}}}{c_{p_{HCFC-123}}} \right)^m \left(\frac{\lambda_{fluid}}{\lambda_{HCFC-123}} \right)^{1-m} \end{aligned} \quad (4.30)$$

AU_{fluid} can be implemented for both exhaust and suction heat transfer coefficient. Thus, given empirical heat transfer parameters can be adopted for different refrigerants. Table 4.3 shows $AU_{su,n}$ and $AU_{ex,n}$ for various refrigerants.

Table 4.3 Modified heat transfer parameters for various refrigerants from [108]

Fluid	$AU_{su,n}$ (W K ⁻¹)	$AU_{ex,n}$ (W K ⁻¹)
HCFC-123	21.2	34.2
HCFC-141b	21.57	33.43
HCFC-245ca	28.42	45.37
HCFC-365mfc	32.57	50.77
HFE-245fa2	25.33	43.82

Although this update is useful for using different refrigerants in the system, it still depends on geometry of tested expander. In order to model for different capacity and brand expanders, geometric modelling method needs to be followed.

4.5 Modelling of different system configurations

This study also includes a steady-state analysis and comparison of the performances of systems under design points. To conduct this, the effect of the heat storage needs to be eliminated. Thus, an indirect solar ORC unit has been considered in the calculations. Pinch temperature difference of 10 K has been taken between the ORC working fluid and collector water flow (heat transfer fluid). After obtaining the evaporating temperature according to pinch difference, ORC calculations can be done.

Overall thermal efficiency equation is defined for s-ORC system including ORC efficiency and collector thermal efficiency:

$$\eta_{overall,s-ORC} = \eta_{ORC} \cdot \eta_{col} \cdot \eta_{me} \quad (4.30)$$

$$\eta_{ORC} = \frac{W_{net,ORC}}{\dot{m}_{ref} \cdot (h_3 - h_2)} \quad (4.31)$$

η_{me} indicates conversion efficiency of mechanical shaft power to electrical output.

$W_{net,ORC}$ is the net power output considering pump power consumption. For PV/T-ORC systems, overall efficiency also includes PV efficiency:

$$\eta_{overall,PV/T-ORC} = \eta_{PV} + \eta_{ORC} \cdot \eta_{PVT} \cdot \eta_{me} \quad (4.32)$$

Figure 4.11 shows the flow chart of the analysis; the followed procedure is shown step by step. s-ORC is based on EFP thermal collectors. The same collectors are converted into PV/T collectors with a cover ratio; thus, PV/T a-Si and PV/T poly-Si collectors are obtained. Firstly, three systems are compared according to steady state operation assumption. Later, with the stand-alone PV system, four systems are compared by considering transient operation for two different weather conditions.

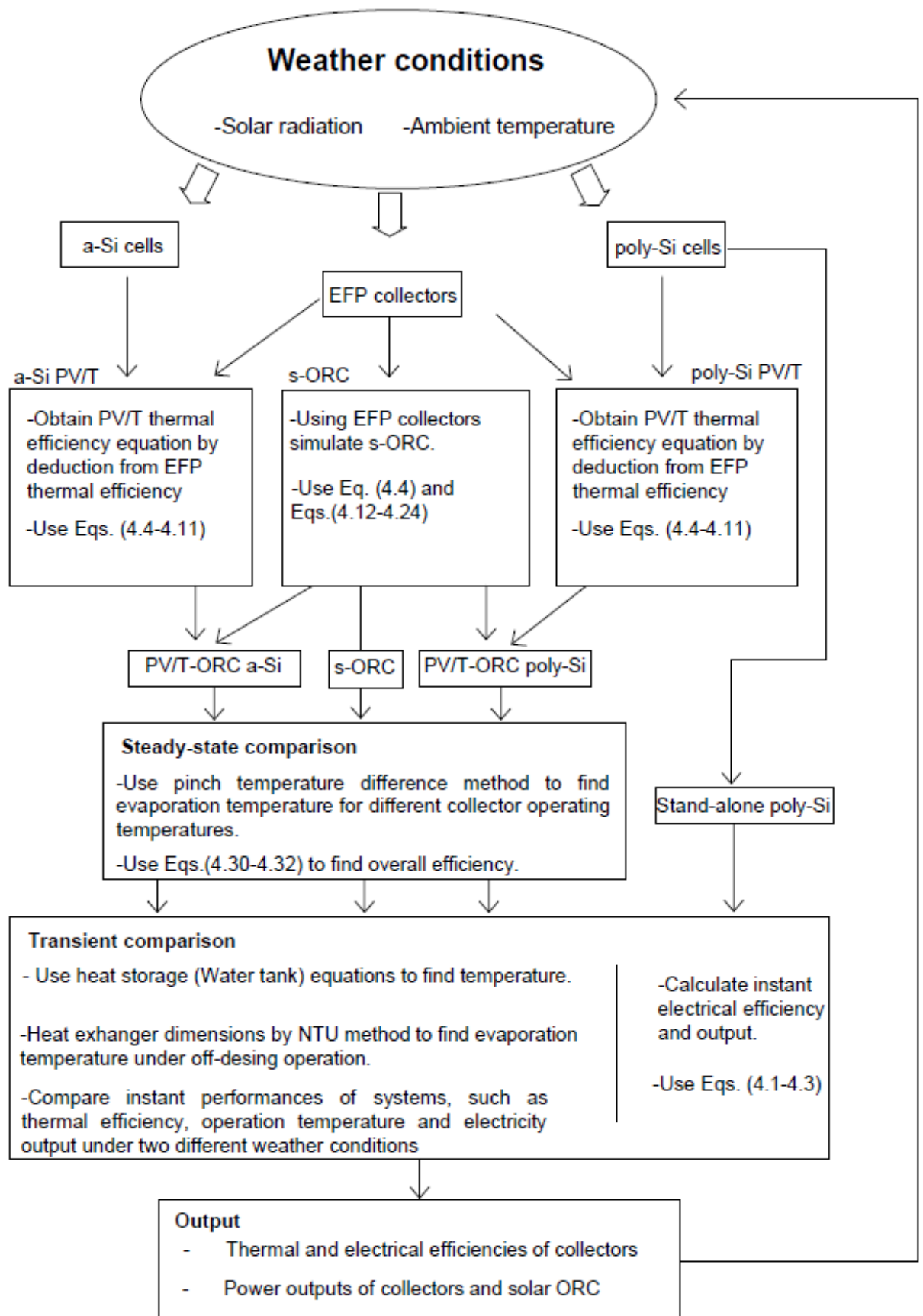
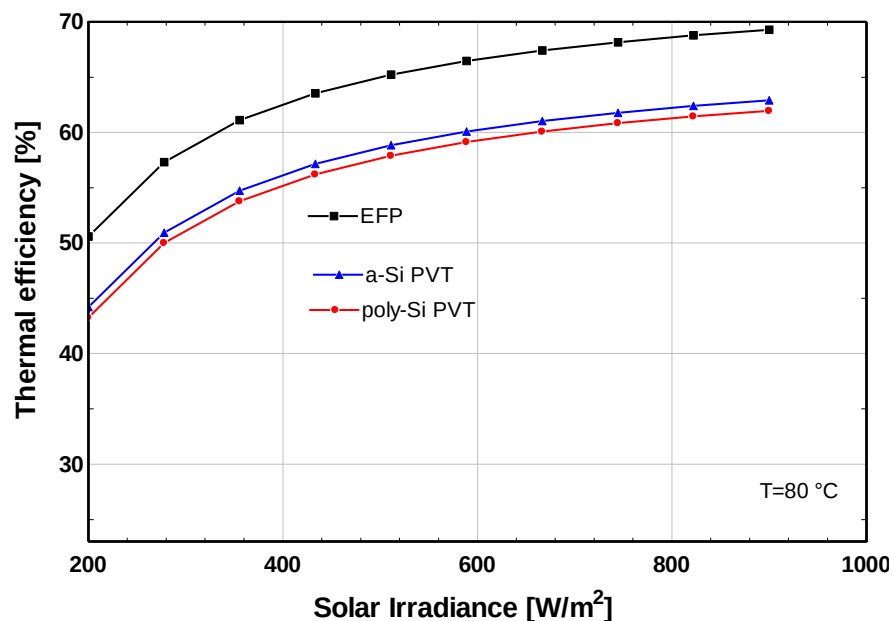


Figure 4.11 Followed flow chart of the study

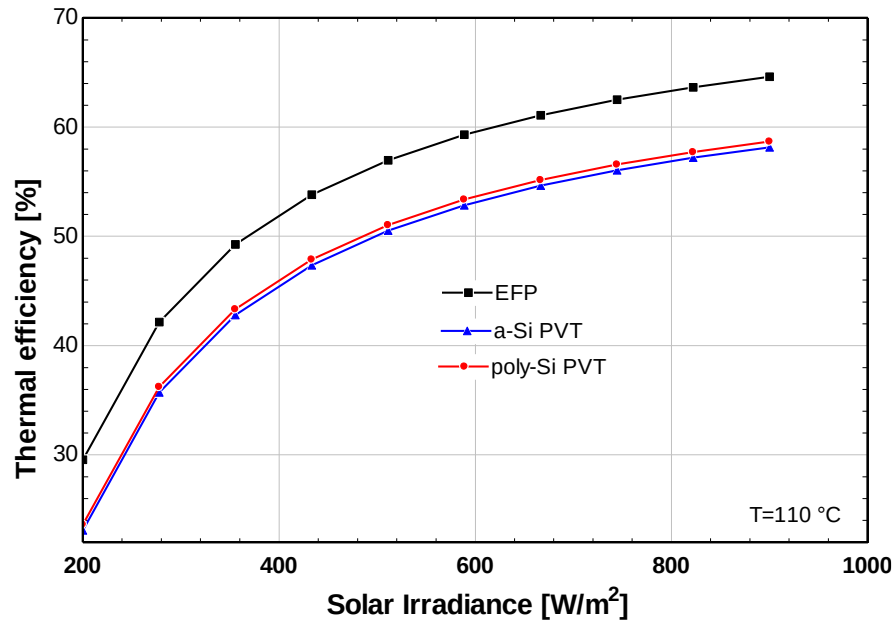
4.6 Results and discussions

4.6.1 Comparison of steady-state performances

Before conducting the daily simulations, it is necessary to analyse the system according to operating parameters. As a first step, conventional EFP collectors are covered with a-Si and poly-Si cells (a-Si PV/T, poly-Si PV/T) and their thermal efficiencies are determined by using Eqs. (4.4-4.11). Figure 4.12 shows the result of the analysis and compares the thermal efficiencies according to various solar irradiances and operating temperatures. Figure 4.2 presents electrical efficiency lines and they intercept at 98 °C; this means that using the a-Si PV/T at an operation temperature of over 98 °C has better electrical performance than poly-Si PV/T. Moreover, Figure 4.12 indicates poly-Si PV/T has a higher thermal efficiency at higher operation temperatures since the rest of the electrical conversion energy is converted into thermal energy. Thus, a-Si PV/T has less thermal performance but better electrical performance at high temperature operations.



a) For the operating temperature of 80°C



b) For the operating temperature of 110°C

Figure 4.12 Comparisons of thermal efficiency for different collector operating temperatures: 80°C (a), 110 °C (b)

As seen in Figure 4.2 and Figure 4.12, PV electrical performance and collector thermal performance depend on the operating conditions. In this sub-section, s-ORC and PV/T-ORC systems are compared according to their operating temperatures. To conduct this, some assumptions have been made for this sub-section only. Normally s-ORC and PV/T-ORC systems have a water tank as a heat storage to produce electricity during night, however, in this steady analysis, it is assumed that there is no heat storage unit. Circulated water from the collector evaporates ORC working fluid via a heat exchanger. This indirect system is analysed by the pinch temperature difference method [180]. The working fluid is in the liquid phase and from subcooled liquid becomes a saturated liquid then the saturated working fluid converted to saturated steam. This process is achieved with an optimum way by using a counter flow heat exchanger. Since the collector operating temperature varies during the day, this analysis considers that ORC evaporation temperature can be found by keeping

minimum temperature difference constant. Figure 4.13 shows T-s diagram of refrigerant and pinch difference in relation to heat transfer fluid. By doing this analysis, it can be obtained collector efficiency and ORC efficiency according to operating temperatures to calculate overall efficiency of the systems.

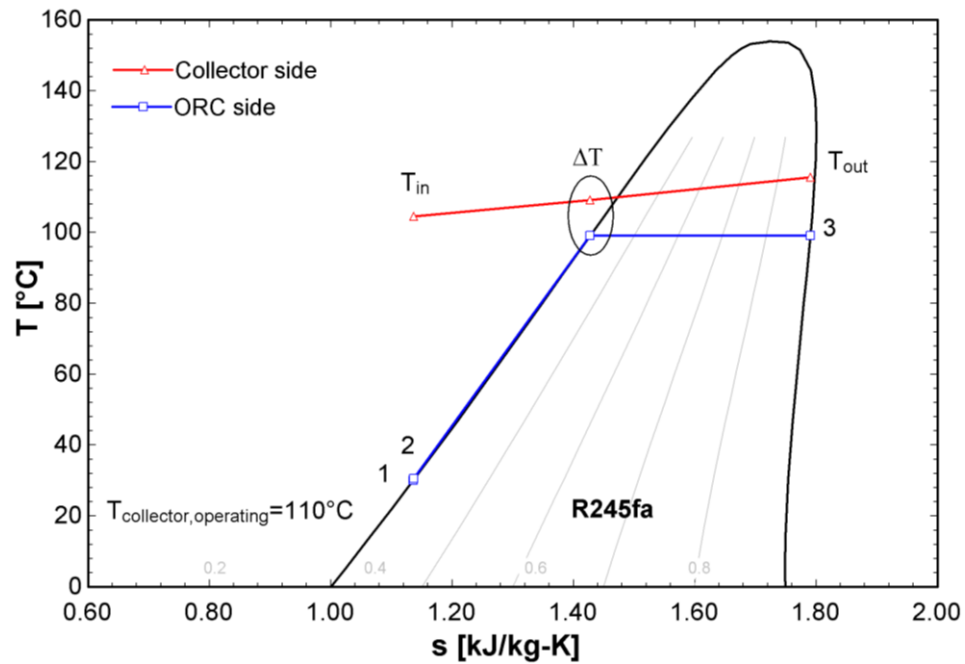
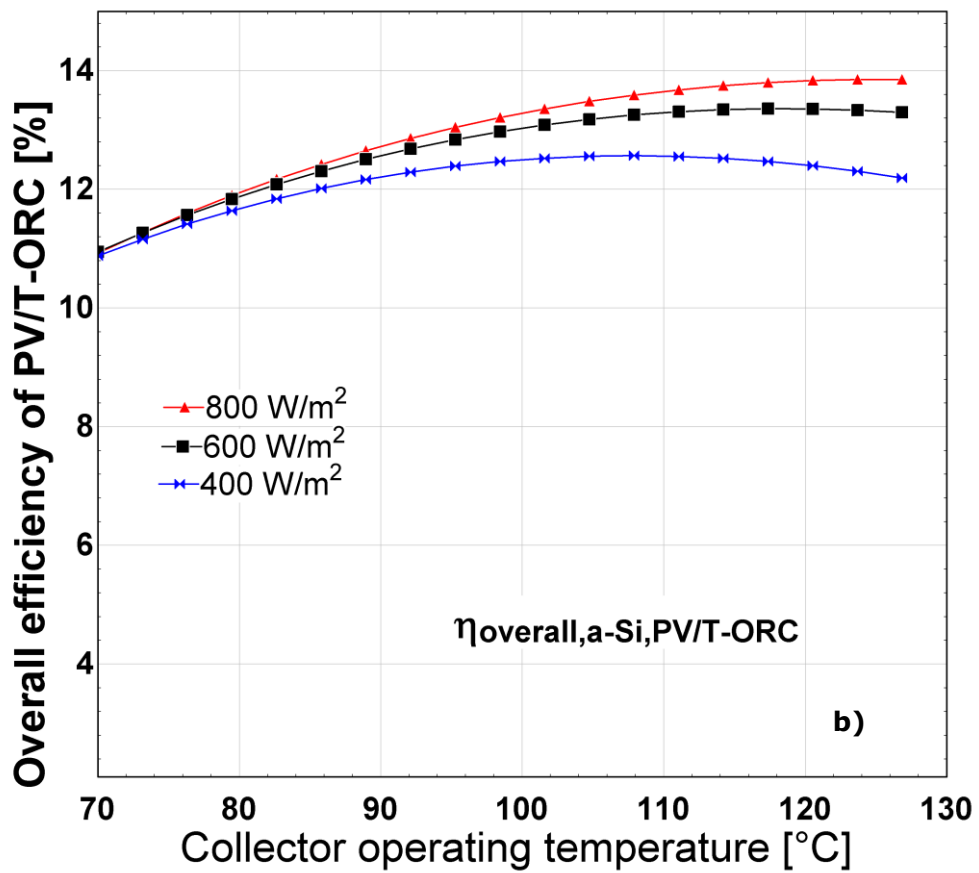
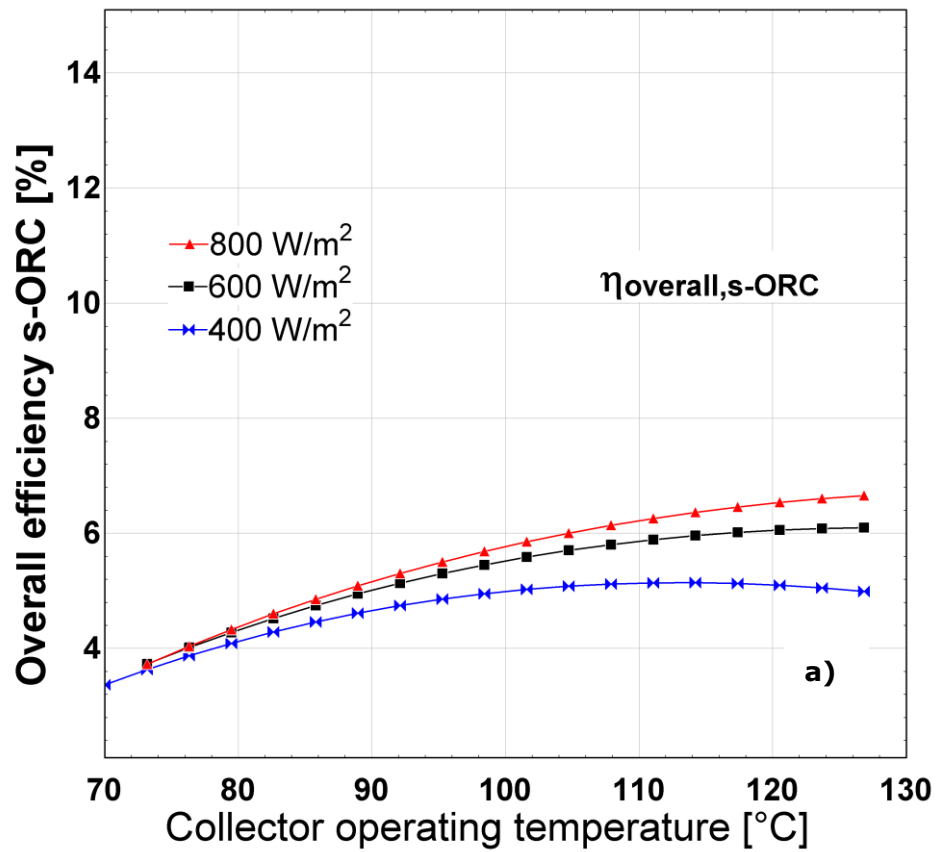


Figure 4.13 Pinch temperature difference for steady state analysis

Figure 4.14 shows variation of overall efficiency by collector operating temperature. In order to show the effect of solar radiation on the collector efficiency; system performance under low, medium and high solar radiation levels is investigated. Figure 4.14a shows results for s-ORC. Overall efficiency for s-ORC depends on collector and ORC efficiency value as calculated from Eq. (4.32). We can easily identify that the operating temperature has the same trend as the evaporation temperature, thus, higher operating temperature yields higher evaporation temperature and higher ORC efficiency. However, efficiency does not increase continuously because the collector thermal efficiency reduces with high temperature operations. Its effect is obvious

under low solar radiations, that efficiency starts to decrease after 115 °C for under 400 W/m² solar radiation applications. The PV/T-ORC system efficiency is additionally affected by PV efficiency, as with s-ORC systems, where the higher operating temperature reduces collector efficiency and increases ORC efficiency. However, another parameter has the main influence on performance. Figure 4.14b. shows PV/T-ORC a-Si system performance. Although collector thermal efficiency decreases with operating temperature, ORC efficiency and PV efficiency increase until a certain temperature for different solar radiations. Peak performance is obtained around 125°C under 800 W/m² solar radiation. Compared to the a-Si based PV/T-ORC system, poly-Si PV/T-ORC reaches peak performance in lower temperatures. As seen in Figure 4.14c, overall efficiency starts to reduce by 80 °C for 400 W/m² and by 100°C for 800W/m² solar radiation. This trend comes from poly-Si PV efficiency as its reduction is steeper in high operating temperatures. We can conclude that a-Si PV/T-ORC system is more applicable for high temperature applications. To clearly present this, Figure 4.15 is given. The figure shows comparison of overall efficiency variation for the three systems under 800 W/m² solar radiation. It can be seen that, an operating temperature of 98 °C is critical to determine proper cell material selection for PV/T-ORC.



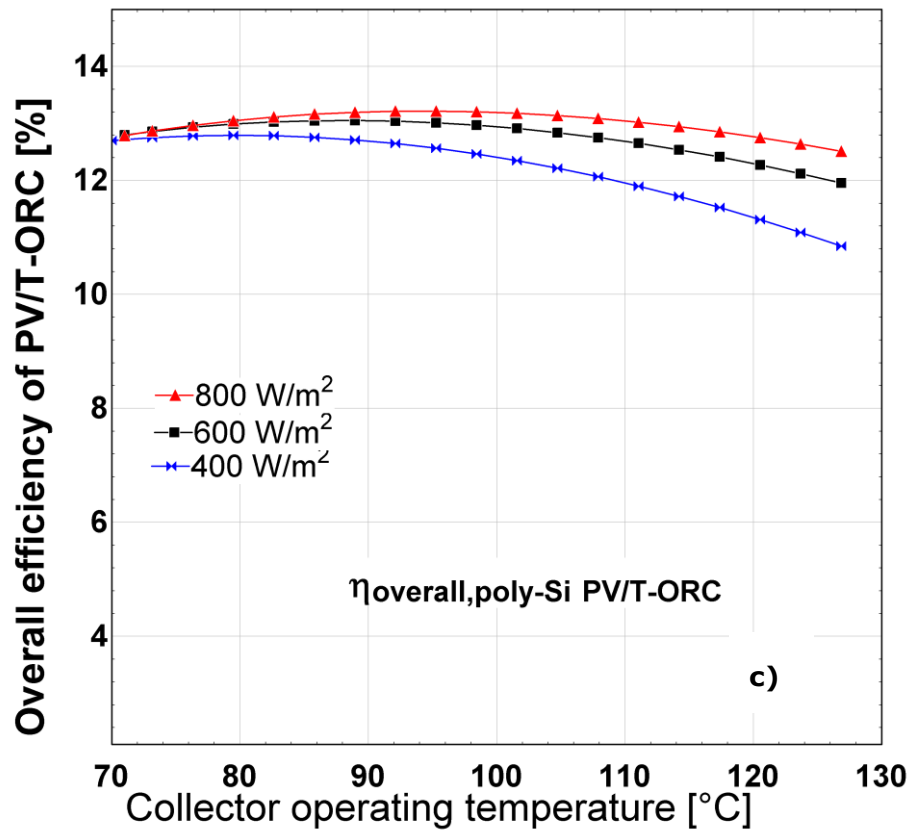


Figure 4.14 Overall efficiency variation by the collector operating temperature under different solar radiations. a) s-ORC, b) a-Si PV/T-ORC, c) poly-Si PV/T-ORC

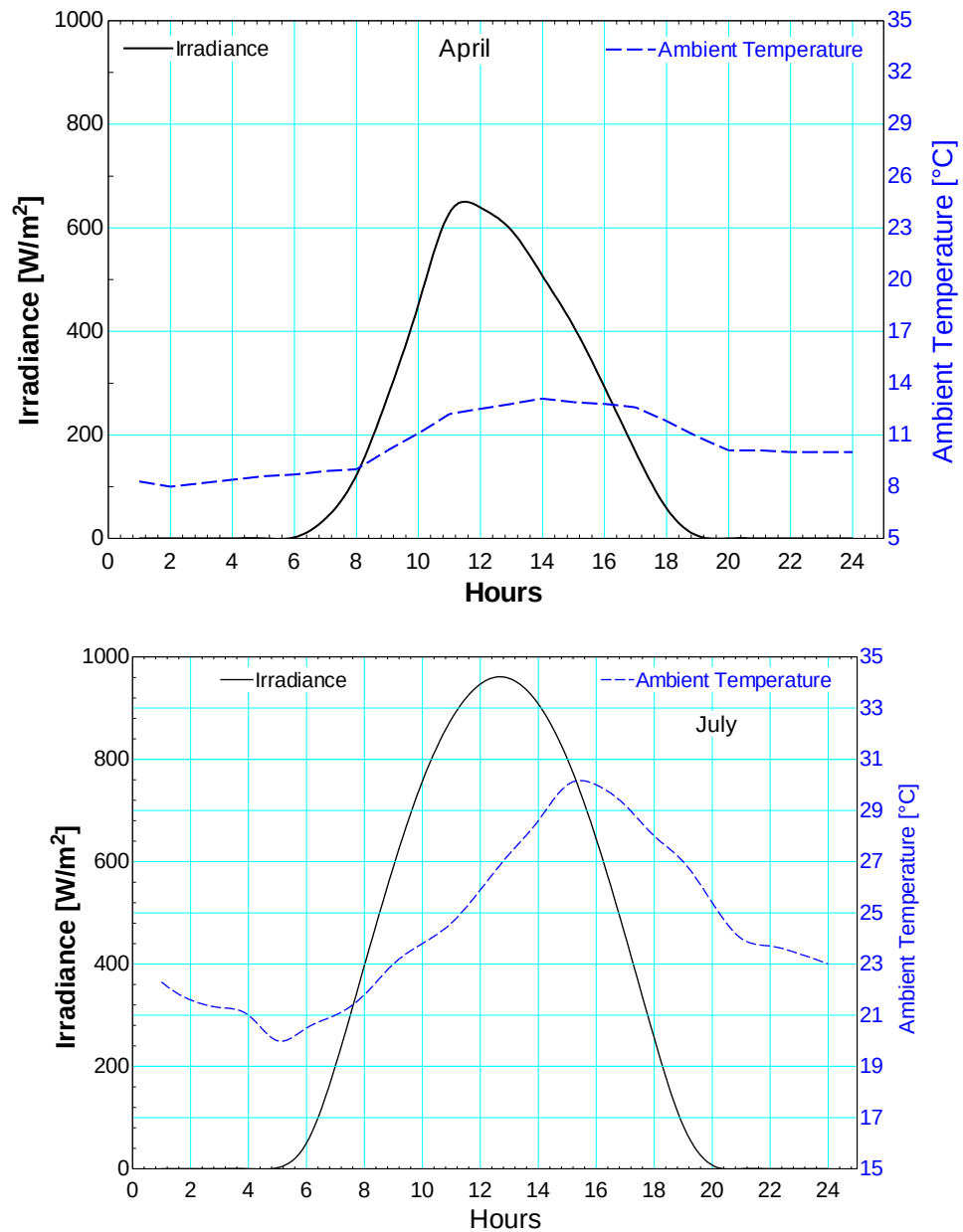
4.6.2 Comparison of transient performances

After analysing the steady-state operation and noting the key observations, this subsection presents transient simulations of the four systems. To carry out a transient operation, dimensions of the components must be determined first because the performance of system depends on them such as evaporator length, water tank capacity and other specifications. The s-ORC design parameters are given in Table 4.4.

Table 4.4 Design parameters of s-ORC

Collector area	550 m ²	Tank volume	87 m ³
Collector mass flowrate	0.02 kg/s	Water mass rate	2 kg/s
Evaporator length	51 m	Condensing temp.	T _a +10
Evaporator water side d_o	0.3 m	Evaporating Temp.	96 °C
Evaporator ref. side d_i	0.012 m	Power output	9.5 kW

To make this comparison more realistic, statistical weather data has been used. This weather data relates to the climate conditions of Istanbul, Turkey. In the study, two different weather data are used to test the proposed system's performance. One of the weather data is chosen as very good solar irradiance, the other is chosen as moderate solar irradiance with relatively low ambient temperature. Figure 4.15 shows the used weather data which are statistical days of Istanbul in April and July. Chosen days are 14th April and 26th July.

Figure 4.15 Weather data for 14th April and 26th July

As simulations are conducted for selected weather data, the initial tank temperature needs to be determined by using the method of repeating the simulations. To provide operation at close value of the design parameters, the stop criterion is also defined. The fifth element of the tank is selected as stop consideration; this is activated, the working fluid pump is shut off and the tank is switched to static mode for the particular

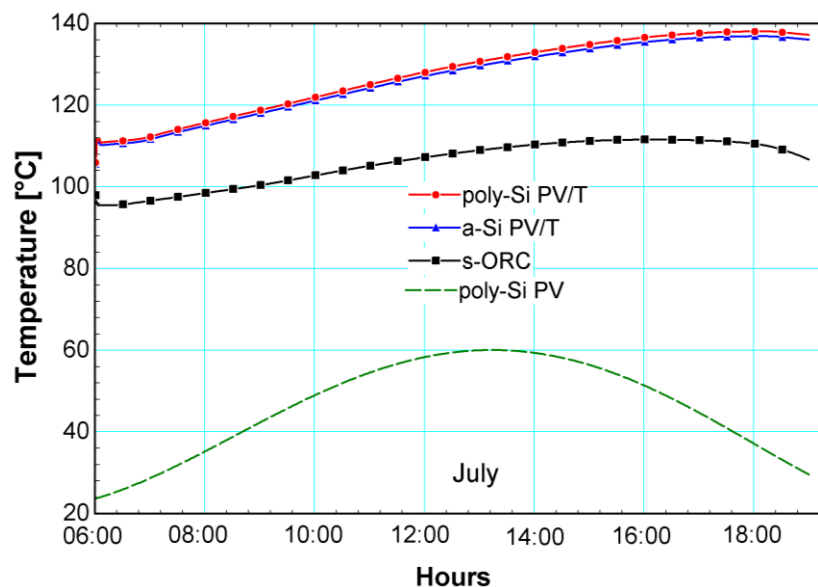
simulation. Stop temperature is chosen as 80 °C and 100°C for April and July, respectively. Simulations have been conducted for different systems such as s-ORC (EFP collectors are used), a-Si PV/T-ORC, poly-Si PV/T-ORC and stand-alone PV (poly-Si collectors are used). For s-ORC system, electricity is only produced by the expander thus system operation is continued throughout the day. However, for a-Si PV/T-ORC and poly-Si PV/T-ORC systems, electricity is produced by the cells during day time. After the sun sets, stored heat energy in the tank is converted into electricity by the expander. To reveal the given systems' performances, investigated parameters are compared in relevant figures.

Figure 4.16 shows the comparison of the proposed systems on 26th July according to collector operating temperatures, collector thermal efficiencies and PV electrical efficiencies. Collector operation temperature which is the mean temperature of the collector inlet and outlet has an influence on collector performances such as thermal and electrical efficiencies. Therefore, Figure 4.16a is plotted to compare operating temperatures of the s-ORC, a-Si PV/T-ORC and poly-Si PV/T-ORC. Since only the s-ORC consumes stored heat during the daytime, the tank temperature of this system stands lower than hybrid systems. Thus, its operating temperature is lower despite it having better thermal efficiency. It increases from 95°C to 105°C. The poly-Si PV/T collector operating temperature is slightly higher than a-Si PV/T. This means poly-Si has better thermal efficiency in higher operating temperatures but lower electric efficiency.

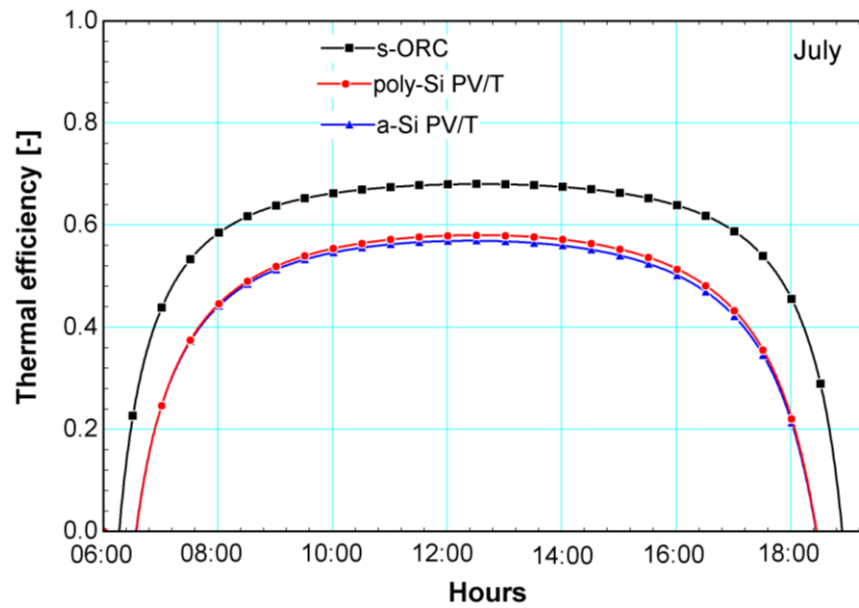
Thermal efficiencies of the collectors are given in Figure 4.16b. It is known that thermal efficiency of this type of collector depends on operating temperature, solar irradiance and ambient temperature. However, in the analysis, weather conditions are identical for specific months and only operating temperatures are different. Electricity

efficiencies make the difference for both PV/T systems, as PV conversion reduces thermal efficiency. Therefore, thermal efficiency of the collectors of the s-ORC system is higher than that of PV/T collectors.

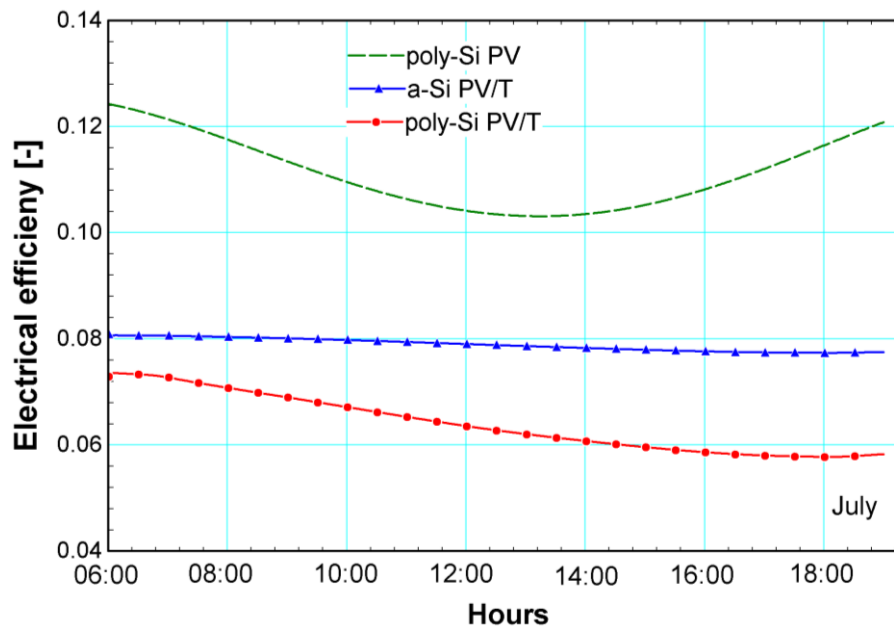
Figure 4.16c shows electrical efficiencies of cells for a-Si PV/T, poly-Si PV/T and stand-alone PV during the day. It is evident that the electrical performance of stand-alone PV is better than PV/T-ORC systems. The electrical efficiency depends on cell temperature; the PV cell temperature is lower in stand-alone PV as the heat generated in the PV cells is transported via convection through the ambient by external surfaces. However, in the hybrid systems, cell temperature is totally dependent on collector operating temperature. Thus, stand-alone PV performance is only affected by ambient conditions, such as better performance, as observed for the stand-alone PV. According to Figure 4.2, poly-Si cells have lower performance when operating higher than 98 °C. Since collector operating temperature increases from 110 °C to 140 °C during the examined day, the poly-si cell efficiency decreases to under 6%. In contrast, a-Si cell performance is quite stable and higher around 8% even in high operating temperatures.



a) Collector operating temperatures



b) Collector thermal efficiency variations



c) PV electrical efficiency variations

Figure 4.16 Comparison of systems on 26th July a) Collector mean temperatures, b)Collector thermal efficiency variations, c)PV electrical efficiency variations

Figure 4.17 shows the comparison of the instant electricity outputs for the four examined systems. As solar radiation is high in July, s-ORC and hybrid systems can produce electricity for 24 hours. Stand-alone PV is the highest and s-ORC is the lowest

instant electricity producing system. Although the stand-alone system and poly-Si PV/T have the same collector area and same PV cell material, the higher operating temperature of PV/T-ORC causes lower electricity output as collector working fluid is circulated from the water tank. a-Si PV/T-ORC produces higher output than poly-Si PV/T-ORC because a-Si cells have better efficiency in high operating temperatures. Since s-ORC only produces electricity through the expander, its performance is lower than others in daytime. After sunset, hybrid systems generate higher electricity than s-ORC because their water tank temperatures can reach a higher level. The reason is that the s-ORC consumes collected solar heat throughout day, however, hybrid systems charge the water tank during the daytime and start to consume this stored energy after sunset.

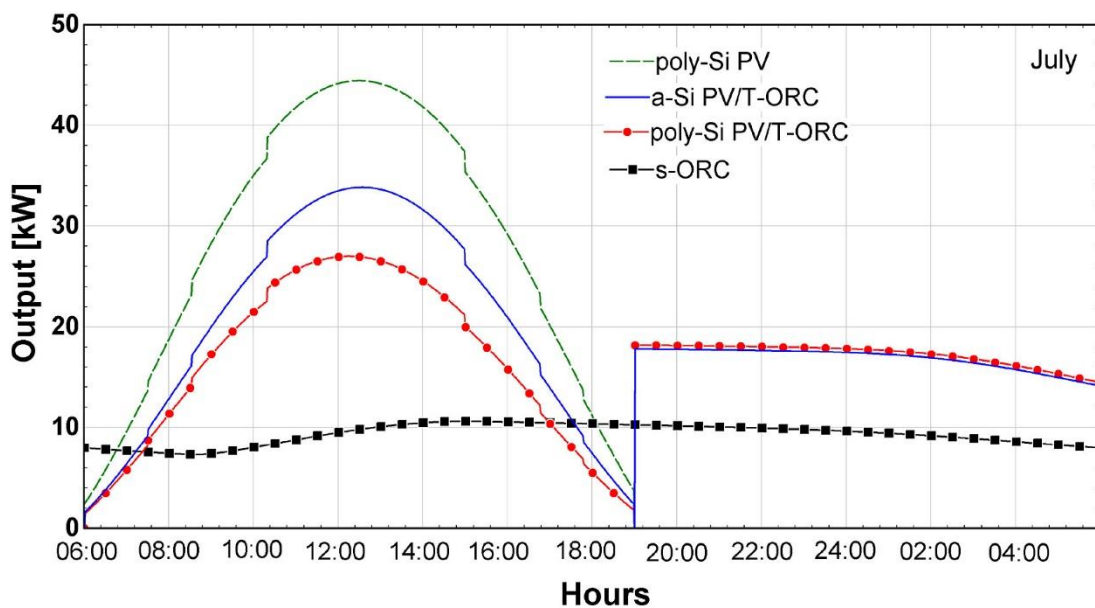
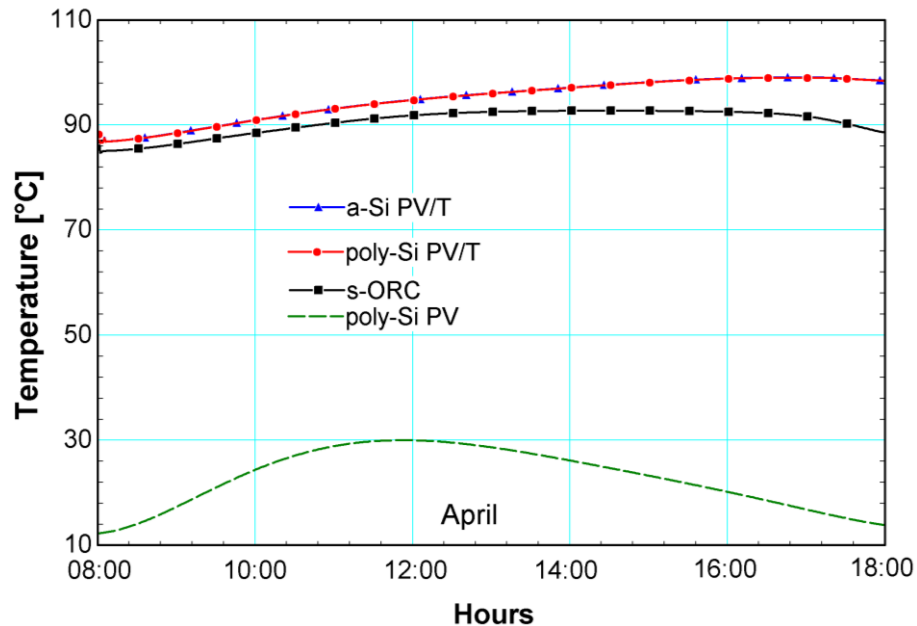


Figure 4.17 Comparison of systems on 26th July: Instant electricity outputs

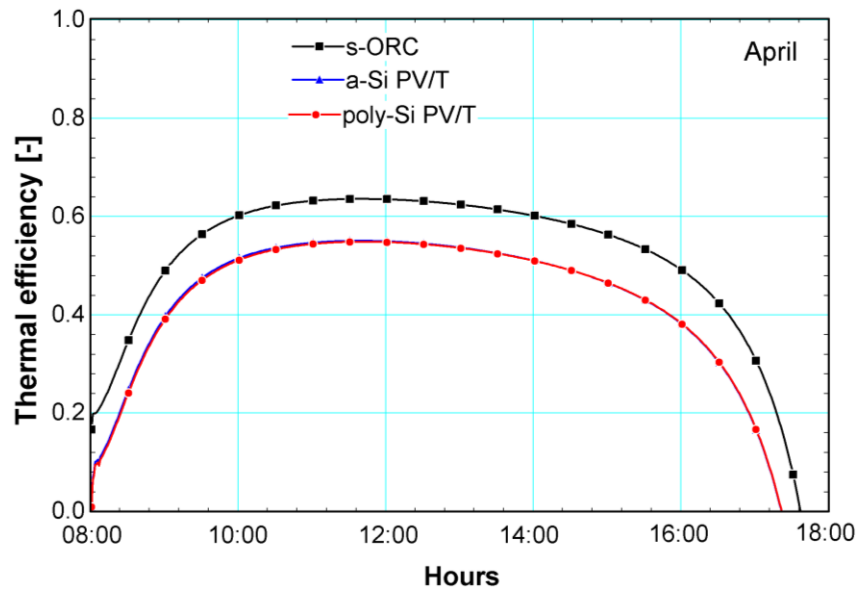
To show a detailed comparison, Figure 4.18 is given of the proposed systems when they are operating on 14th April according to collector mean temperatures, collector thermal efficiency variations and PV electrical efficiency variations. Figure 4.18a

shows collector mean temperatures. As expected, all systems' operating temperatures are lower than in July as the lower solar irradiance does not allow an increase in tank temperature as high as that in July. It is also seen from the figure that, s-ORC operation temperature is still lower compared to the hybrids. Temperature of the hybrids varies from 85°C to 100°C.

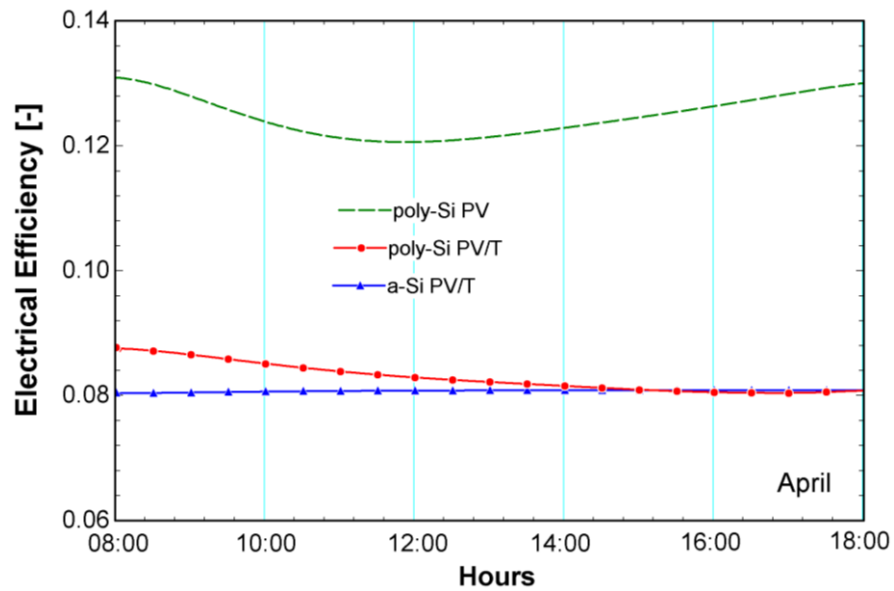
Figure 4.18b shows collectors' thermal efficiency variations. Although predicting collector operation temperatures are lower in April, all systems' performances are better in July because of higher solar radiation and ambient temperature. Figure 4.18c presents PVs' electrical efficiency variation profiles. The stand-alone PV performance is only affected by ambient conditions because better performance is observed in April as a result of lower ambient temperature. The poly-Si PV/T have higher electrical efficiency until noon as the operating temperature is increasing from 87°C to 100°C.



a) Collector operating temperatures



b) Collector thermal efficiency variations



c) PV electrical efficiency variations

Figure 4.18 Comparison of systems on 14th April a) Collector operating temperatures, b) Collector thermal efficiency variations , c) PV electrical efficiency variations

Figure 4.19 shows the comparison of the instant electricity outputs for the four examined systems on 14th April. Since the solar irradiance profile is weaker and solar time is shorter in April, electricity production is lower for all systems compared to

July. Moreover, none of the systems can produce electricity for 24 hours. For both months, the stand-alone PV is the highest and s-ORC the lowest producing system. In Figure 4.19, poly-Si PV/T-ORC system generates slightly higher electricity against a-Si PV/T-ORC in daytime due to better electrical efficiency, however, in July, the a-Si PV/T-ORC system produces considerably more. As both collectors have very close thermal efficiencies, their ORC performances are quite close. PV outputs of hybrid systems are close because operating temperatures are very close and electrical efficiencies are also very close in that temperature range. Since the s-ORC consumes heat during day, its electricity production ends around 23:00 but hybrid systems continue to produce until 5:00.

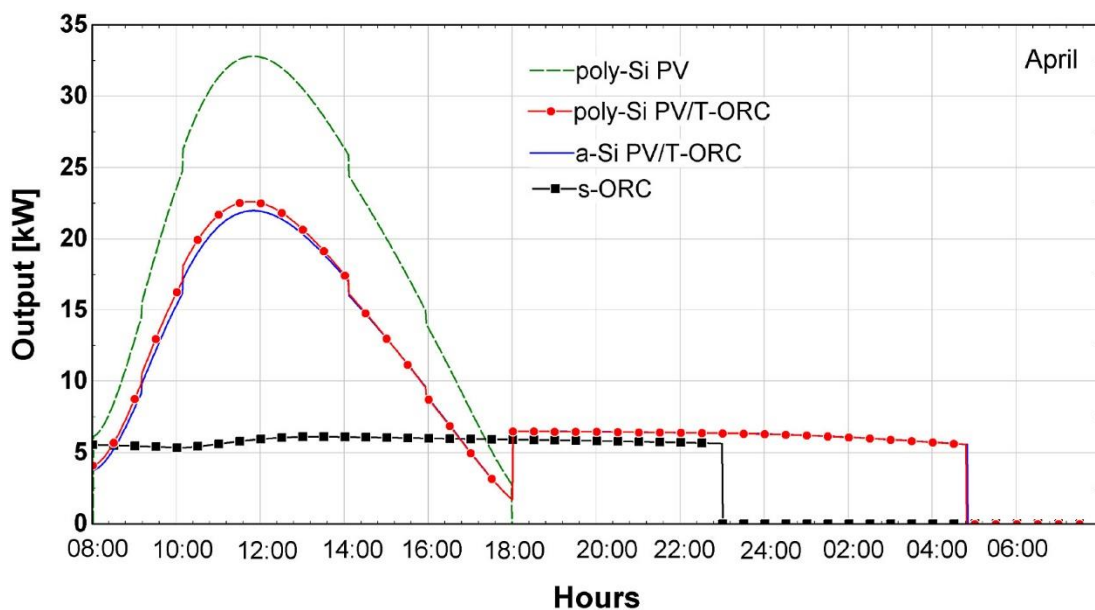


Figure 4.19 Comparison of systems on 14th April: Instant electricity outputs

Cumulative daily performances of the systems are given in Table 4.5. Compared to April, the daily produced electricity values increase in July. The s-ORC has the worst performance for both months. a-Si PV/T-ORC, poly-Si PV/T-ORC and stand-alone PV systems produce almost same electricity in April. However, the a-Si PV/T-ORC

system produces more power in July. Thus, it can be concluded that hybrid systems have an advantage over conventional s-ORC and PV systems. Moreover, the a-Si PV/T-ORC is promising when combining PV technology and ORC.

Table 4.5 Daily Produced electricity

	s-ORC	a-Si PV/T- ORC	poly-Si PV/T- ORC	poly-Si PV
Daily Output	87.27	201.74	205	204.62
[kWh]- April				
Daily Output	223.6	452.35	403.9	365.15
[kWh]- July				

4.7 A basic thermo-economic performance comparison of the examined systems

Apart from performances of power generation, a comparison on the thermo-economic performance between the a-Si PV/T-ORC and the other three systems can be conducted by using the data given in tables.

- Compared with the poly-Si PV/T-ORC, the a-Si PV/T-ORC has a lower capital cost. The costs of a-Si and poly-Si PV panels are about 0.45-0.53 USD/Watt and 0.8-0.9 USD/Watt, respectively [181]. Given an aperture area of 550 m², a-Si module efficiency of 6.5% @25 °C, poly-Si module efficiency of 13% @25 °C and solar irradiance of 1000W/m², the costs of a-Si and poly-Si panels are about 18,000 and

55,000 USD. Owing to its higher yearly output and lower capital cost, the a-Si PV/T-ORC would have a shorter payback time.

(2) Compared with s-ORC, the a-Si PV/T-ORC has a higher capital cost due to the employment of a-Si cells. However, an additional cost of about 18,000 USD for the cells is relatively low in consideration of the costs of EFP collectors ($>137,500$ USD for 550 m^2 , i.e., $>250 \text{ USD/m}^2$) and ORC module ($>1500 \text{ USD/kW}$). Because its power output is 2-3 times higher, the a-Si PV/T-ORC will be more cost-effective.

(3) Compared with the stand-alone PV system, the a-Si PV/T-ORC uses water storage for flexible power generation. If lithium-ion battery of a 6-hour storage capacity is included, the cost of the PV system will be increased by around 45,000 USD according to the market average battery price of 156 USD/kWh in 2019 [182]. Moreover, the lifetime of battery is about 3-5 years (or 1000 - 1500 cycles). Due to its higher power output and ability to meet the domestic heat demand, the a-Si PV/T-ORC can be more economic.

4.8 Summary

In this chapter, the mathematical models for the s-ORC, a-Si PV/T-ORC, poly-Si PV/T-ORC and stand-alone PV systems have been built and their performance parameters have been compared in steady and transient daily state analysis. Although the s-ORC has advantages such as flexible operation and continuous production even at night, its instant electricity output has been found to be much lower than stand-alone PVs. Combination of these two technologies has given promising results as all their advantages are combined in hybrid systems. The following conclusions were drawn:

- Hybrid systems' daily outputs were almost same as the stand-alone PV in April simulations, however, their night long production made them a better option over stand-alone PVs.
- In April, poly-Si s-ORC system has power output of 205 kWh, followed by that stand-alone PV, a-Si s-ORC and s-ORC have outputs of 204.62 kWh, 201.74 kWh and 87.27 kWh, respectively
- Using a-Si cells in the hybrid system showed great results in summer applications, since it can tolerate excessive temperatures with a stable electrical efficiency. Moreover, a-Si PV/T-ORC has not only the advantage of power outputs, but it is also expected that a-Si PV/T-ORC systems will reduce its payback time against s-ORC systems. a-Si cells' resistance to thermal annealing also makes their usage promising over poly-Si cells in PV/T-ORC applications in the point of their life time.
- In June, a-Si s-ORC system has power output of 452.35 kWh, followed by that poly-Si s-ORC, stand-alone PV and s-ORC have outputs of 403.9 kWh, 365.15 kWh and 223.6 kWh, respectively

CHAPTER 5 Direct incorporation of a solar organic Rankine cycle with a vapour compression refrigeration cycle

5.1 Introduction

It is reported that in some cities, up to 50% of the supplied electricity during summer months is consumed by air conditioning applications [183]. As solar units have a great potential to produce combined electricity, heating and cooling the solar energy-related systems have been developed in an exponentially rate over the last decades. To match both electricity and cooling demand at the same time, a combination of a solar ORC with a vapour compression cycle (VCC) system is applicable. In this kind of systems, the expander shaft of the ORC and the compressor shaft of the VCC are coupled directly to reduce two-way energy conversion losses. Different from the absorption and desiccant refrigeration systems, the ORC-VCC system is flexible by the way of outputting power when there is no need to cooling. In a hot summer, most of solar heat can be used for to refrigeration, while less heat is used for cooling in spring and fall. When there is no need for cooling, the all available thermal energy can be converted to electricity and sent to the grid [184].

Therefore, this chapter includes an analysis based on using solar ORC with VCC for an office room. In this kind of systems, the expander shaft of the ORC and the compressor shaft of the VCC are coupled directly to reduce two-way energy conversion losses. Operational flow characteristics are examined to provide required cooling load and electricity demand.

5.2 System description and operating modes

The examined system consists of four sub-systems, namely, an array of solar field, a heat storage unit, an ORC block and the VCC. The ORC unit has four main components: pump, boiler, expander and condenser. The refrigeration system also has four main components, namely, compressor, condenser, expansion valve and evaporator. Although the present analysis focuses on summer application to produce cooling and electricity at the same time, the components are selected to operate well in winter period as well for production of electricity and heating. In order to operate the system in all seasons, some operation modes need to be determined and these are given below:

Mode 1- ORC-Cooling mode: this operation mode includes the key part of the study. Original ORC-VCC operation is required during the summer period to produce power and cooling at the same time. Figure 5.1a shows the schematic view of the unit. Blue dashed lines show inactivity.

Mode 2- ORC-Heating mode: this mode can be used when heating is required by residents. If enough heat is stored in the tank to run the ORC system which is directly related with the solar radiation profile in that day, the system can operate the ORC cycle and condensation heat from the ORC can be used for space heating for the property. As shown in Figure 5.1b, the evaporator of the refrigeration system turns into an ORC condenser and blue dashed lines are inactivated.

Mode 3- Heating mode: It is obvious that solar energy may be unavailable or weak on some days in winter. The water temperature in the tank cannot be increased to a high enough level to produce satisfactory electricity for the people. In these days, the system is unable to produce electricity from the ORC. However, heating is still demanded by the residents. Lower temperature water in the tank can therefore be used

for heating, a process which is commonly studied before; thus, it will not be simulated in this thesis.

Since it is assumed that the system operates smoothly in the three modes, solar collectors, heat storage medium and expander were carefully selected on the following basis:

EFP collectors are used for heating water which comes from the bottom of the tank (T_{st10}) and is then discharged into the top of the water tank at a higher temperature (T_{col}). High temperature water in the tank is used as a heat source for the ORC at (T_{st1}). A concentric tube heat exchanger is adopted as a boiler for the ORC block and high-pressure working fluid is evaporated in this boiler. After evaporating (3), high pressure and high temperature working fluid enters the expander. As an expander device, a scroll type expander was selected. Regarding working fluid selection, Saleh [185] reported that R600 and R245yf achieved the best performance among other candidates in his parametric analysis. Thus, in this simulation, commonly preferred working fluid R245fa was chosen to use directly with the validated expander model in the simulation.

After the expander, the operating mode determines the path of the working fluid. For ORC-cooling mode, the condenser is in common use by both the ORC and VCC systems, which lead the same refrigerant circulating in both cycles, one comes from the expander (4), the other from the compressor (r4), thereafter, the two are mixed in the auxiliary component mixer before entering the condenser. Later, the refrigerant is separated through the two cycles after condensing. One path goes into the pump (1) and leaves it with higher pressure (2). The other path is for the refrigeration cycle (r1). Working fluid pressure and temperature is decreased by the expansion valve (r2) then via the evaporator, the refrigerant is evaporated and provides cooling. After the

evaporator (r3), working fluid pressure and temperature is increased by the compressor.

For the ORC-heating mode, the compressor and expansion valve are not in use. The working fluid directly enters the new condenser which was the evaporator in ORC-cooling mode after the expander. While the fluid is condensing, taken heat is used for heating the room.

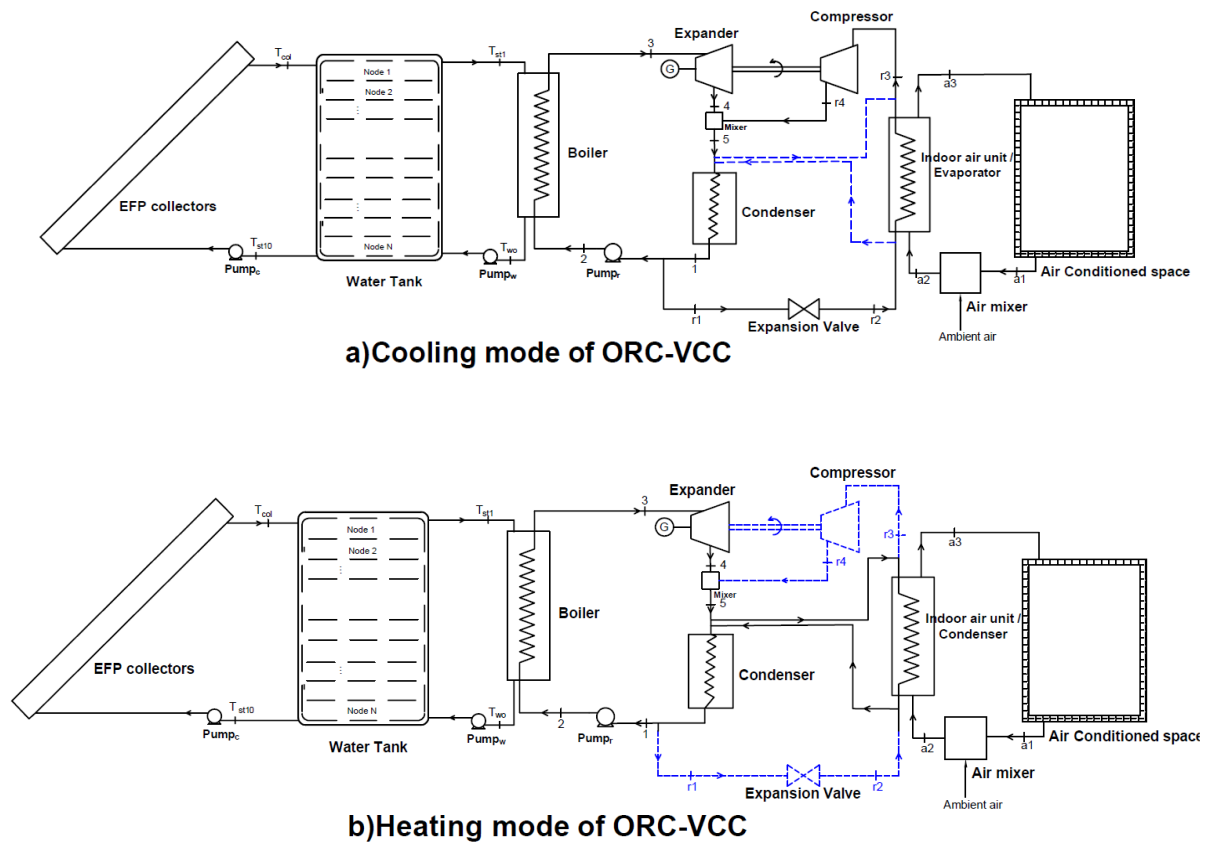


Figure 5.1 Schematic views of cooling (a) and heating (b) modes of ORC-VCC

5.3 Electricity and cooling demand for a virtual office building

For the ORC-VCC mode air cooling section, exhaust air from the conditioned space is mixed with ambient fresh air. This mixture is cooled in the evaporator and discharged into the conditioned space. The cooling load which impacts directly on the

operation of the compressor, is calculated by using the software EnergyPlus [173] for the given data of Istanbul. A 60 m² office room was assumed for the case study and Figure 5.2 gives the cooling and electricity demands for the virtual room during a typical day in July. To conduct the analysis, some assumptions were made:

- For ORC-VCC cooling mode, condenser and ambient air temperature difference is considered to be 10°C [186],[187]. This practical approach is used for obtaining appropriate cost of heat exchanger [188].
- For heating mode, condenser is indoor unit for heating. The condensing temperature is adjusted at 38°C to heat up the indoor unit blowing air [188] .
- Mechanical efficiency of the compressor η_{mec} changes between 0.95-1 and it can be assumed as 0.95 [189].
- For VCC cooling mode the evaporating temperature is fixed at 5°C with assumption of no superheating. The conditioned space has a temperature of 24°C and 50% relative humidity [190].

According to the given assumptions, the corresponding P-h diagram of the ORC-VCC system for boiling temperature of 90 °C is given in Figure 5.3.

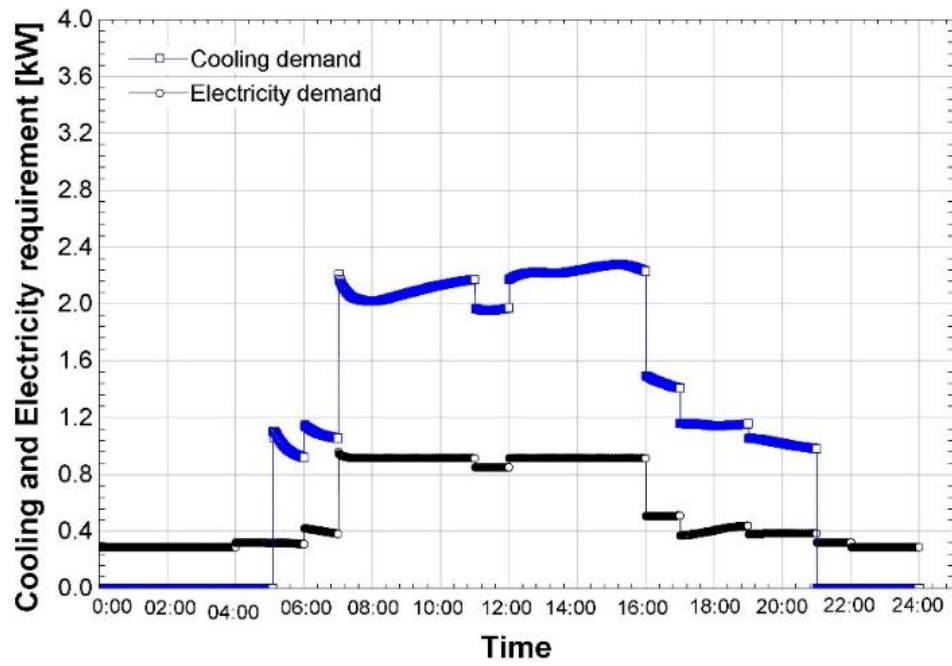


Figure 5.2 Cooling and electricity demand profile in an examined day for the assumed office room

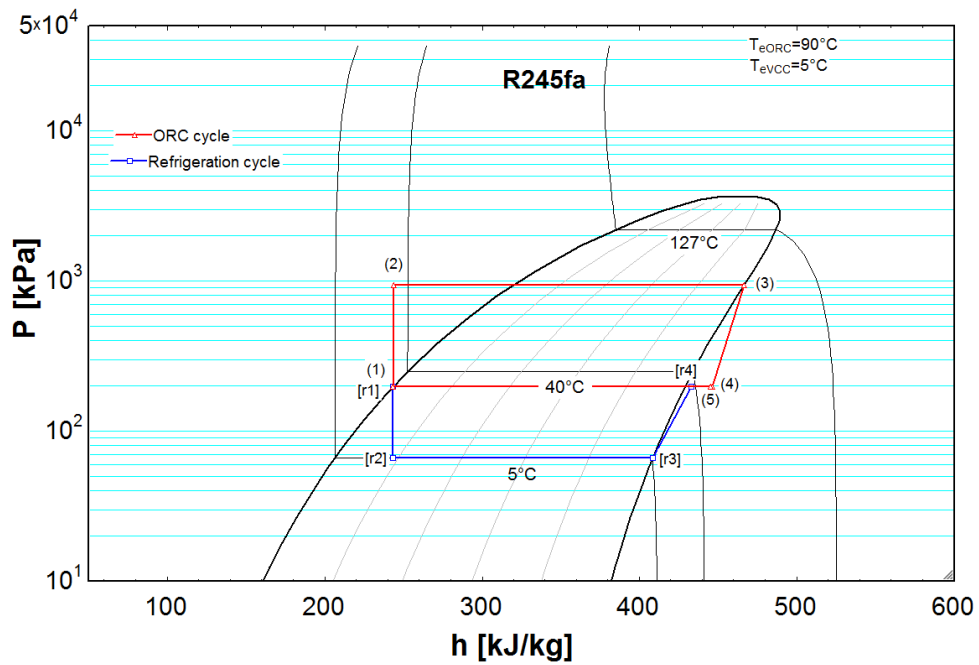


Figure 5.3 Pressure-enthalpy diagram of the proposed unit

5.4 Modelling of ORC-VCC

For modelling of the solar collector, water tank and expander, the same methodology and equations in Chapter 3 are used. Only the equations for relevant components of VCC are given below:

The most important parameter of the VCC unit is compressor electricity consumption. In this system the compressor is directly coupled with the expander shaft. Thus, the compressor work equation can be included with mechanical transmission efficiency. The equation is given as:

$$\dot{W}_{comp} = \frac{\dot{m}_{rc} \cdot (h_{r4} - h_{r3})}{\eta_{mec} \cdot \eta_i} \quad (5.1)$$

Here, η_{mec} and η_i are mechanical and isentropic efficiency of the compressor, respectively. η_{mec} changes between 0.95-1 and it can be assumed as 0.95 [189]. However, η_i changes with condensation and evaporation ratio. So, the following equation is used [191].

$$\eta_i = 0.874 - 0.0135 \cdot \frac{P_{con}}{P_{eva}} \quad (5.2)$$

where P_{con} and P_{eva} are condenser and evaporator pressures.

As previously mentioned, in cooling mode, evaporating temperature is kept constant, the refrigerant mass flow rate varies according to cooling requirements and this affects compressor work under different ambient conditions. In the calculations, cooling load is known parameter and taken from EnergyPlus simulation results. Since evaporator

inlet and outlet conditions are determined primarily, the mass flow rate of the refrigerant in the cooling cycle can be found from:

$$\dot{Q}_{cooling} = \dot{m}_{rc} \cdot (h_{r3} - h_{r2}) \quad (5.3)$$

5.5 Results and discussions

In order to evaluate the system performance, design conditions need to be determined first. These are determination of the design evaporating and condensing temperatures and later, the boiler is dimensioned as operation of the ORC depends on boiler performance. Transient mathematical model has been built in Matlab environment and results have been presented by using EES (Engineering Equation Solver) software. When design conditions are determined, the simulation is run, and collector and storage dimensions are changed until the system output matches with the desired output. The calculation procedure is briefly summarized in Figure 5.4. Useful solar heat is collected by the EFP collectors. This useful heat and collector water inlet temperature effects the output temperature which is mixed with the first element of the tank. Tank model gives temperature distribution of the water in tank. It is affected by two circulation waters one to the collector side and the other to the boiler side. Boiler model determines the evaporating temperature of the working fluid and effects on tank temperature gradient as well. Expander model calculates the mass flow rate and shaft power output. The solar ORC and VCC systems are evaluated as separate systems except their condensing parts because VCC performance mainly depends on compressor electricity consumption and it can be calculated from the cooling load which is variable. Calculated mass flow rate from the evaporator is used to obtain compressor work. The compressor work is taken from the expander shaft and the rest of the energy is transmitted to the generator to generate electricity.

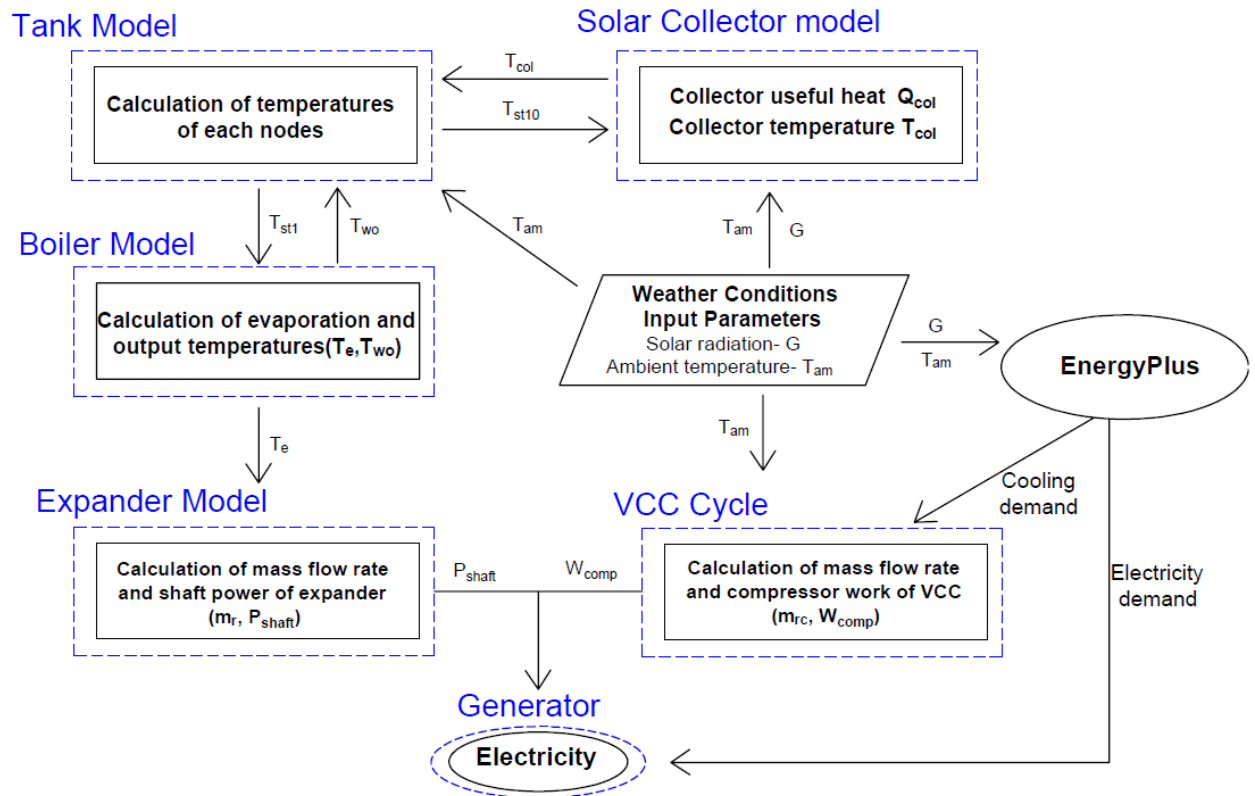


Figure 5.4 Calculation procedure of the ORC-VCC system

5.5.1 Design of the ORC-VCC system

Condensing temperature of the ORC can be determined first, as it depends on the ambient temperature for air cooled condensers. Figure 5.5 shows ambient air temperature and solar irradiance variation during a warm day in July in Istanbul. Since the water tank is a finite heat storage, it is necessary to choose matched requirements to avoid over usage of the contained heat. Thus, before selecting the design conditions and parameters, the total electricity demand of the room is required. Variation of compressor work and total demands including electrical appliances as given in Figure 5.6. The compressor work was calculated according to the cooling load given in Figure

5.2. It is found that the maximum compressor work and total demand are 0.4kW and 1.35 kW, respectively.

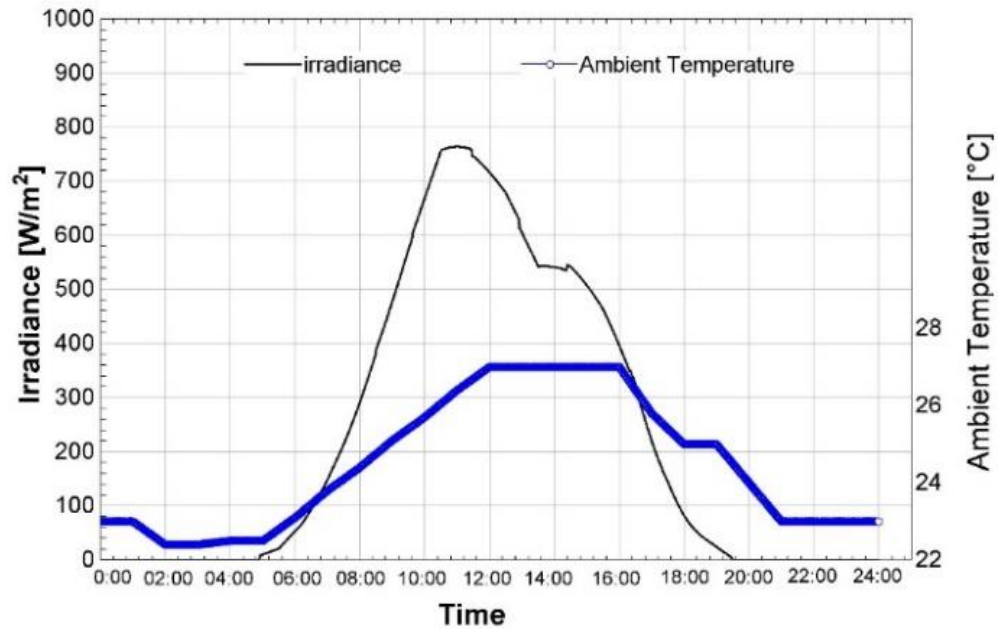


Figure 5.5 Weather data for Istanbul on a typical warm day in July

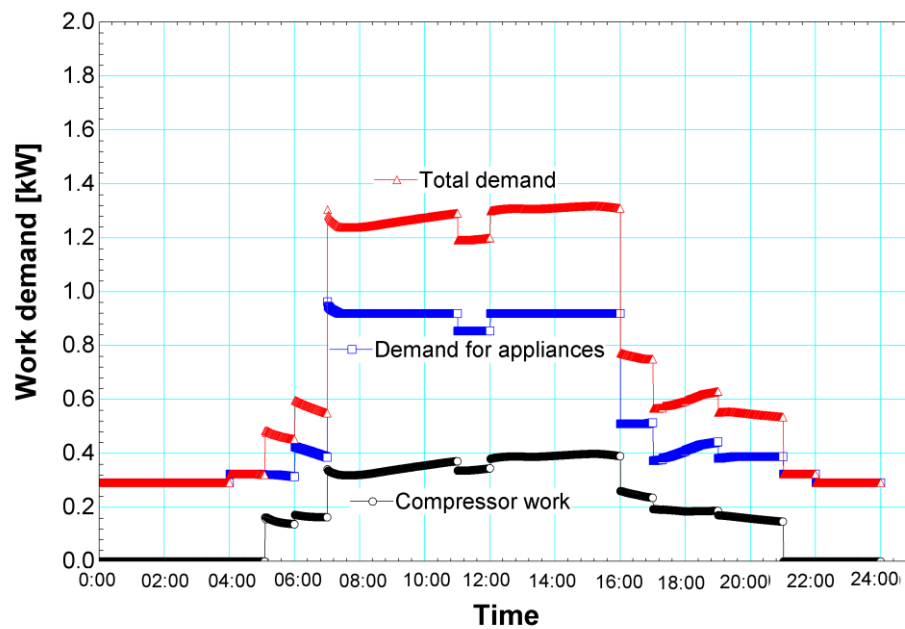


Figure 5.6 Work requirements of the system over time on a typical warm day in July

Figure 5.7a shows determination of the design parameters by considering 36 °C condensing temperature and 2500 RPM expander rotation. To provide 1.35 kW electricity, the designed output work from the expander shaft is chosen 1.437 kW when considering the generator efficiency. Thus, the design evaporating temperature is selected as 95 °C. In order to conduct boiler length same approach followed as given in chapter 3. Figure 5.7b shows required boiler lengths according to hot water temperature and water circulation flowrates to give design evaporation temperature. Water circulation flow rate and boiler length are chosen 0.3 kg/s and 21 m, respectively.

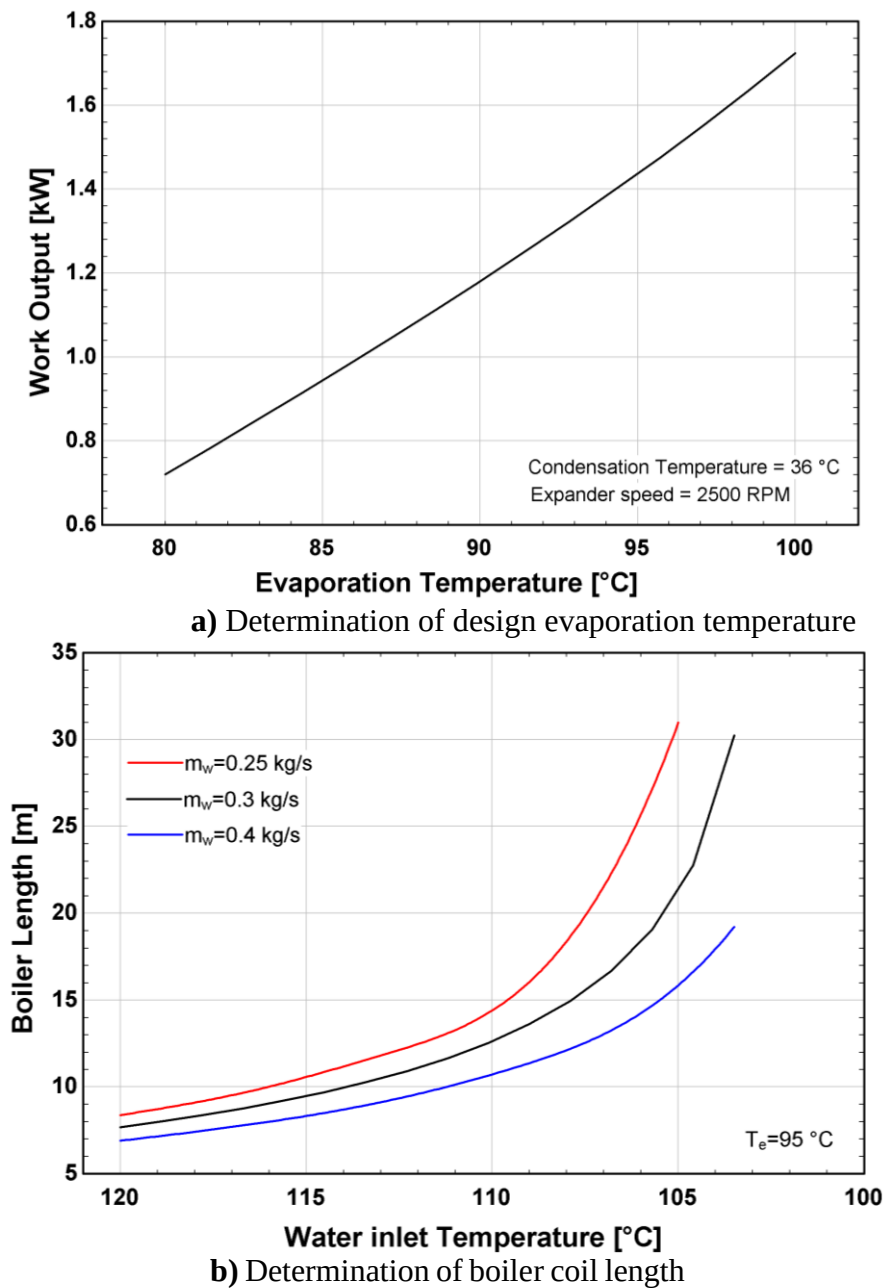


Figure 5.7 Design consideration of ORC-VCC a) evaporation temperature, b) boiler coil length

Normally, a 150 L of hot water tank is the standard household size in practice for DHW applications. However, in this work, charging and discharging heat levels are much higher compared to the normal domestic water tanks. To supply the required amount of heat to the ORC unit, the size of the heat storage tank can be selected from 1.5 m³ to 9 m³. To provide useful solar heat to run the system at given design

conditions, 60 m² of solar collector area is chosen preliminarily, to be the same as the floor area of the simulated office room. In Table 5.1, design conditions for the ORC unit are summarised.

Table 5.1 Design conditions and system specifications of the ORC-VCC unit

Collector area A_{col} : 60 m ²	Evaporating temperature in the boiler: 95 °C	Condensing Temperature: 36 °C	Work output: 1.437 kW
Collector mass flow rate $\dot{m}_{cw}$: 0.02 kg/s	Water inlet Temperature: 105°C	L_{boiler} : 21 m	Heat from source: 15.6 kW
Tank volume: 3.5 m ³	Boiler water side, d_{wi} : 0.016 m	Boiler refrigerant side, d_{ri} : 0.007 m	$\dot{m}_w$: 0.3 kg/s

Off-design performance of the system is based on the performance of the boiler given that work output is related to the evaporating temperature and this temperature is directly related to the source conditions which come from the water tank. Two parameters determine the evaporating temperature in the boiler. The first is source temperature; as solar energy is not constant during the day, the water tank is exposed to charge and discharge operations. In the day time, the tank is charged by the collected useful heat and its temperature is increased in spite of ORC runs at the same time. As a result of this increment, evaporating temperature and work output increase. The other parameter is mass flow rate of the water. The mass flow rate also has an influence on the evaporating temperature. The effects of the parameters are given in the Figure 5.8. This trend reveals the off-design operation of the system.

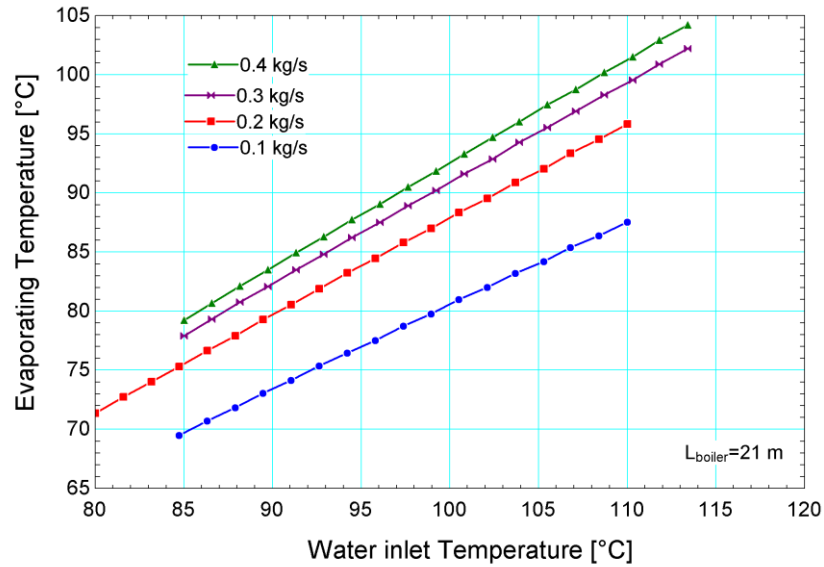


Figure 5.8 Evaporating temperature changes with water mass flow rates and inlet temperature.

Operation of the system is given as the basis: the ORC-VCC system operation starts at 06:30 to provide shaft work to the compressor; for cooling purposes, the collector pump operates and collected solar energy is accumulated in the storage, at the same time the ORC produces electricity for supplying to the electrical appliances. The water mass flow rate from tank to boiler is selected as 0.3 kg/s to match with the demand via previously designed heat exchanger. The peak period ends at 16:00 and the other period which requires less work begins. In this period, the mass flow rate of water is reduced to 0.1 kg/s to prevent excessive use of the heat source. This period is completed by 21:00 and energy in the tank is stored for the next day. There still exists an electricity requirement in the building, but this requirement can be met from the grid or a storage battery.

5.5.2 Cooling mode of the solar ORC-VCC cycle

To simulate the system, initial tank temperature selection is an important issue. As one-day simulation is carried out for a chosen day, the temperature is not uniform in the tank due to the thermocline behaviour of the water. Normally, initial temperatures of the water layers in the tank are the final temperatures of the previous day. However, this study considers only one day, thus, initial temperatures need to be determined. Firstly, same temperature is given to all water layers and simulation is executed. Final temperatures of the first simulation are used as initial temperatures of the next simulation. These simulations had to be repeated by using final temperatures as the next simulation's initials, until the end temperature distribution matches the initial temperature distribution [98]. It can be said that all of the useful solar energy charged to the tank is used for driving the ORC and the rest is transferred to the ambient as heat losses. Off 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 the system performance, Figure 5.9 is plotted by using constant water mass flow rate during all periods.

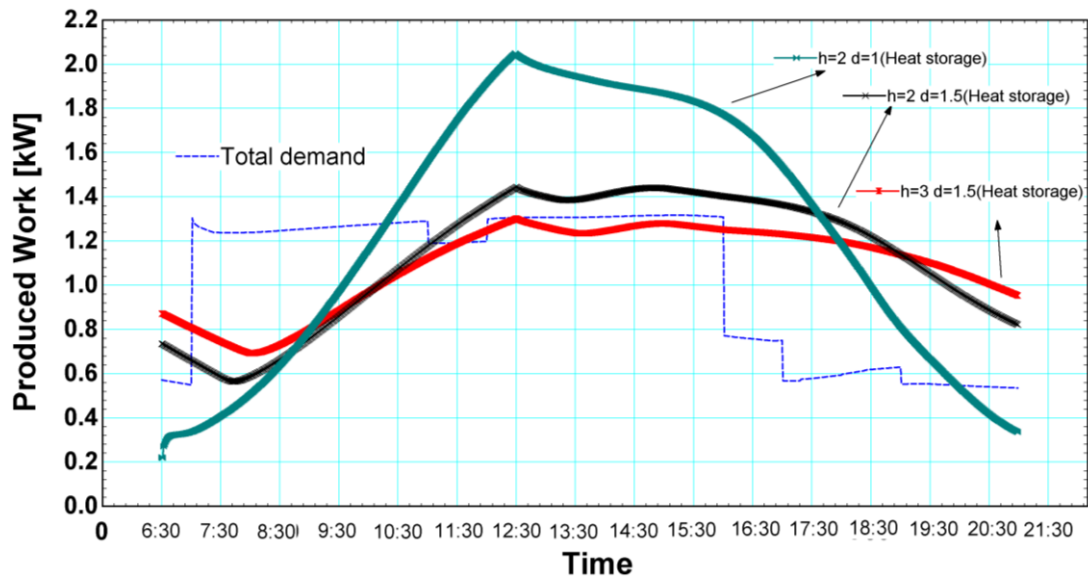


Figure 5.9 Produced shaft output with time for different water tank dimensions ($m_w=0.3 \text{ kg/s}$, $A_{\text{collector}}=60\text{m}^2$)

Since temperature increment is higher when using the constant solar collector area in smaller heat storages, peak output yields higher than the demand. A problem occurs, however, in the early morning period as output in this period cannot fulfil the demand. One solution is an increment to the storage capacity to avoid excessive temperature fall at night-time, however, using higher capacity storage decelerates the temperature increase at midday. As seen from Figure 5.9, when 3 m high and 1.5 m diameter heat storage units area used, produced work is getting closer to the demand but does not fulfil the midday demand as the temperature of the storage is not high enough.

Figure 5.10 shows expander efficiencies for using various storages. Since temperature alteration is high in small capacity storages, expander performance variation is considerable. For the bigger storages, expander performance can be further stabilised by avoiding excessive changes to the operating temperature.

In order to meet the electricity demand, the collector area is increased to 80 m^2 and simulations have been conducted for two different storage sizes. Figure 5.11 shows that increments in collector and heat storage size are still not enough to meet the demand. As a solution, the mass flow rate of the water needs to be controlled for different periods, so power output and heat rejection rates can be controlled.

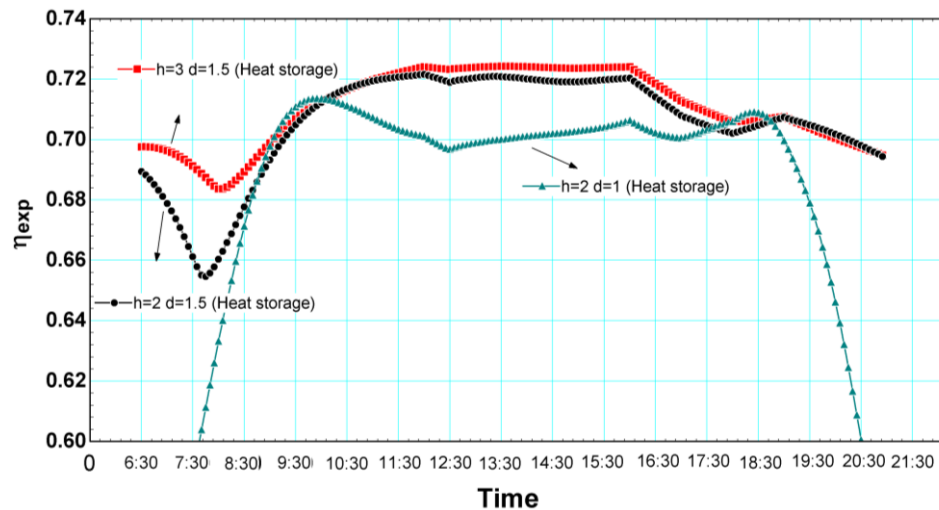


Figure 5.10 Expander efficiency by time for different water tank dimensions ($m_w=0.3 \text{ kg/s}$, $A_{\text{collector}}=60\text{m}^2$)

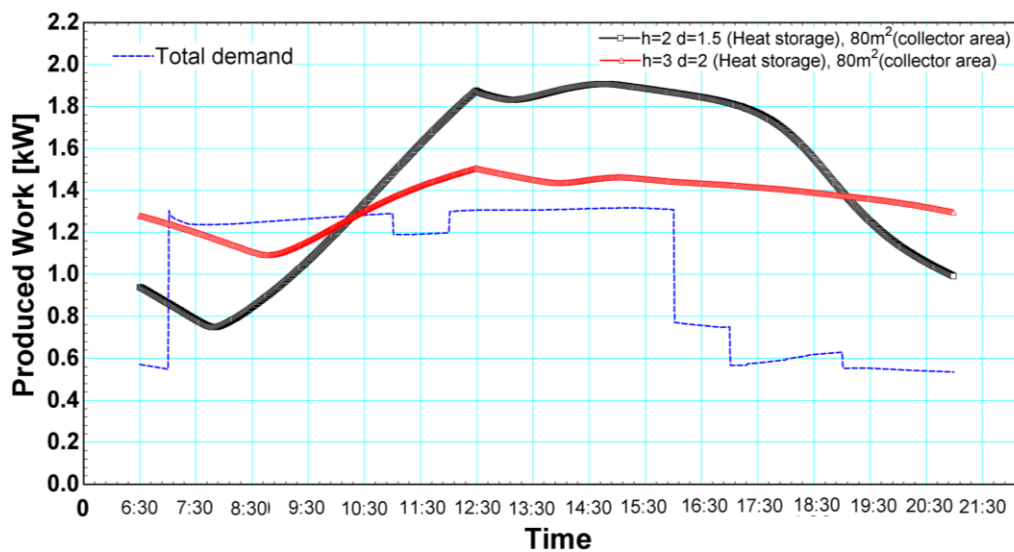


Figure 5.11 Produced work by time for different water tank dimensions ($m_w=0.3 \text{ kg/s}$, $A_{\text{collector}}=80\text{m}^2$)

In Figure 5.12, the mass flow rate is controlled for three different cases. Firstly, a case using a 60 m^2 collector area and 2 m height with 1.5 m diameter storage tank has been analysed. Although performance is improved by controlling the mass flow rate, there is still a problem in the morning period, as the temperature of the tank at the end of the day does not satisfy the requirement in early morning, until the sufficient solar radiation is received. An increment to the storage capacity to avoid excessive temperature fall in the night period can be applied, however, it may require a larger collector area. To observe the phenomenon, a second case has been analysed. When the heat storage size is held constant and the collector area is increased to 80 m^2 , over production occurred at midday, however, the problem still exists in the early morning. As a final case, the storage size is also increased with the collector area. With these final changes, produced work fulfils the demand.

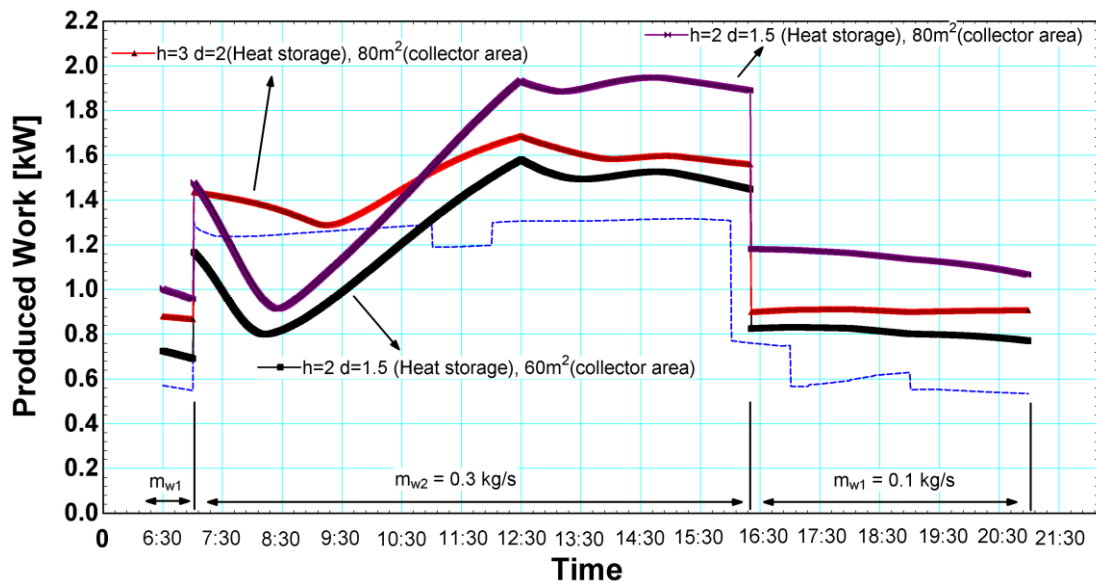


Figure 5.12 Produced shaft output by time with control strategy ($m_{w1}=0.1 \text{ kg/s}$, $m_{w2}=0.3 \text{ kg/s}$)

5.5.3 ORC-heating mode

One of the advantages of the studied system is that it can operate and show a good performance in winter days without any modification just by switching to CHP mode. In this mode, the compressor does not run and the VCC evaporator operates as the ORC condenser, which is given in Figure 5.1b. Since this heat exchanger located in the air conditioning cycle, what would normally be wasted. Thus, condensing heat can be used for heating purpose. The condensing temperature is adjusted at 38°C to heat up the indoor unit blowing air. This temperature is identical to the condensing temperature in ORC-cooling mode. More specifically, the ORC operation would not be affected by the mode switch as a result of the same condensing temperature and pressure. However, electricity output is expected to decrease, as solar heat would be lower than in the summer. Figure 5.13 shows weather condition for a sunny day in February. Solar radiation is quite good but ambient weather temperature is around 12 °C which shows heating is needed. Solar collectors and ORC work same as with the ORC-cooling mode but expander outlet is directed to the indoor heat exchanger. Figure 5.14 shows instant electricity generation by the ORC, the magnitude of the production looks not satisfying but electricity production in daily basis is 5.5 kWh. The reason is lower tank temperature. Since solar irradiance is not good in previous days, water tank temperature drops and causes lower evaporating temperature with lower electricity output. It should be noted that while electricity is produced its useful condensing heat, almost 7 kW, can be used for space heating. Required heat is transferred to the indoor air and the rest is ejected by the outdoor condenser as seen in Figure 5.1b.

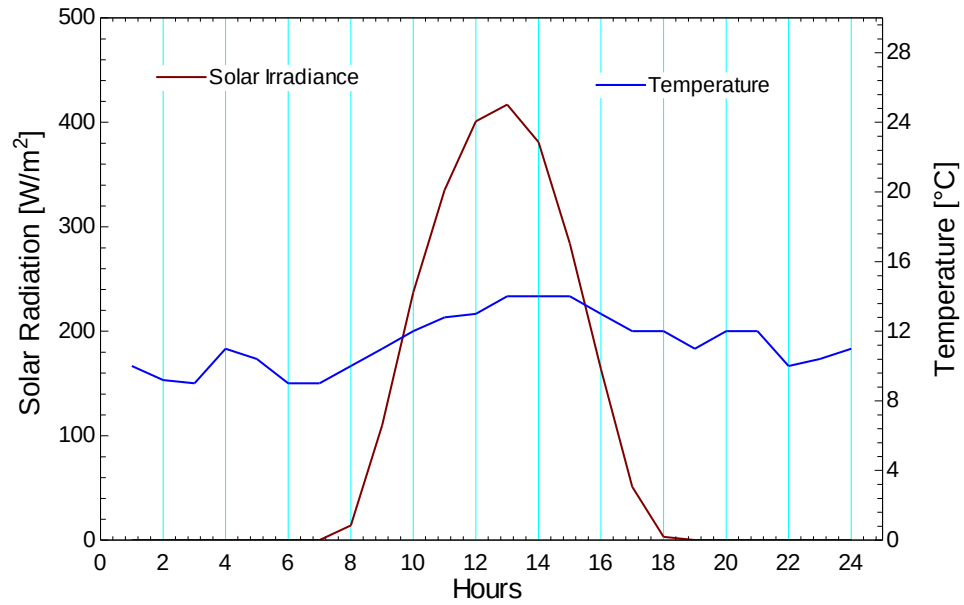


Figure 5.13 Solar irradiation and environment temperature variation on a sunny winter day

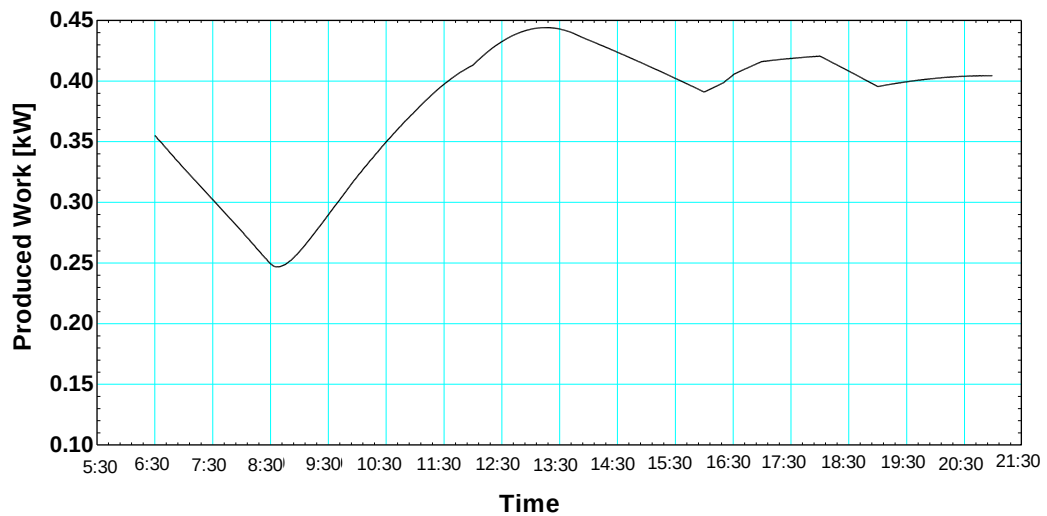


Figure 5.14 Produced work by time in ORC-heating mode on a sunny winter day ($m_w=0.3$ kg/s, $A_{\text{collector}}=60\text{m}^2$)

5.6 PV/T-ORC-VCC configuration

Modification of the solar ORC units with PV/T collectors improves the performance of the system considerably as shown in Chapter 4. Apart from achieved advantages of

ORC-VCC combination, required heat storage unit and collector area are quite big for applications. The main reason for large storage unit is that the electricity conversion efficiency of ORC is low and solar to electricity efficiency need to be increased. Moreover, large heat storage requires larger collector areas. In this case, adapting PV/T collectors look like a good solution for reducing the size of system components. In first part of this chapter, electricity output controlling methodologies and different storage sizes were investigated. In this sub section, using same methodology and same design parameters, required heat storage volume is determined.

The system configuration is given in Figure 5.15 and operation of the unit is explained as follows: in the morning, PV cells feed the VCC compressor and other electrical equipment. At the same time, collector pump operates and water tank temperature is increased. When PV electrical power reduces, ORC pump starts to operate and expander generates electricity until 21:30. Thus, stored heat is converted to electricity for a shorter time compared to solar ORC system.

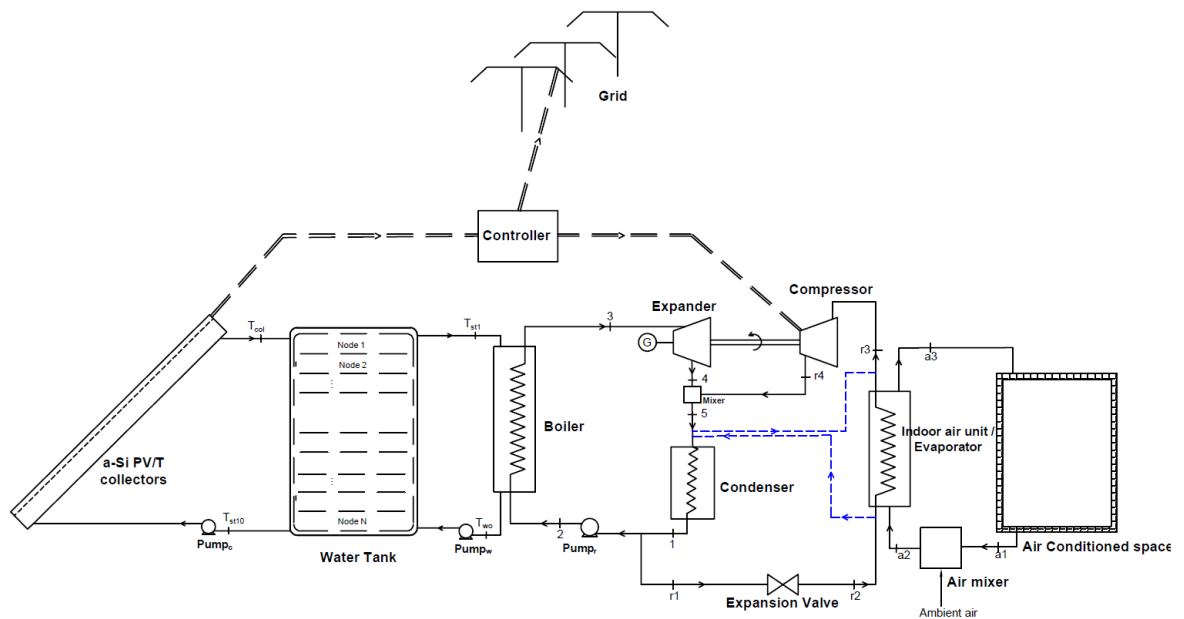
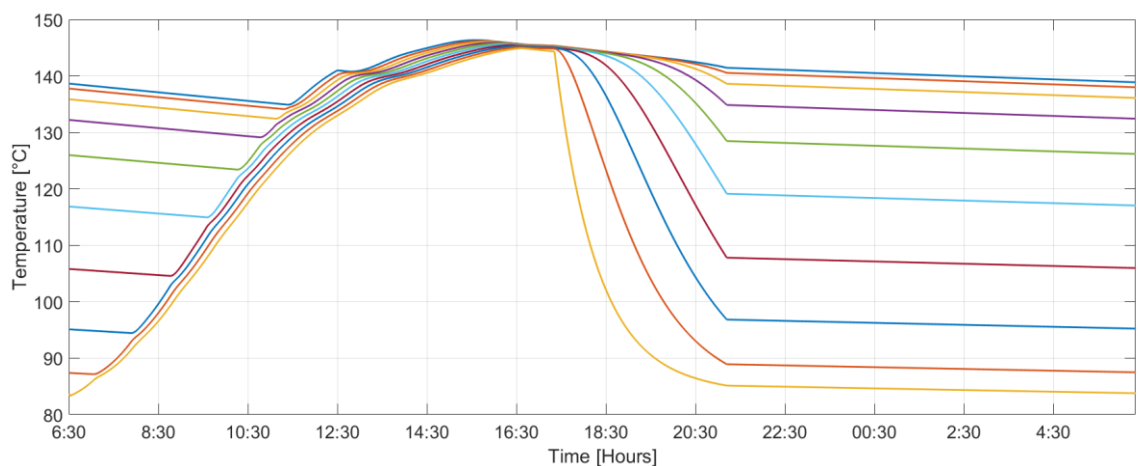


Figure 5.15 PV/T-ORC-VCC unit

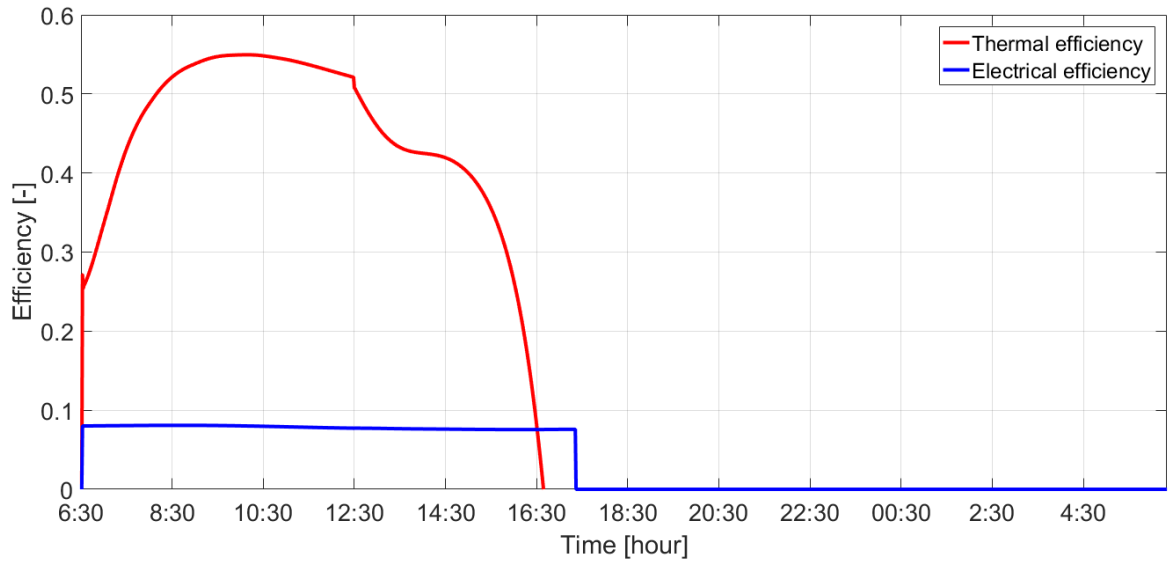
5.6.1 Simulations of PV/T-ORC-VCC unit for different size of components

In order to make a good comparison with solar ORC-VCC unit, same collector area, boiler length and heat storage size are used in the first simulation. The only difference

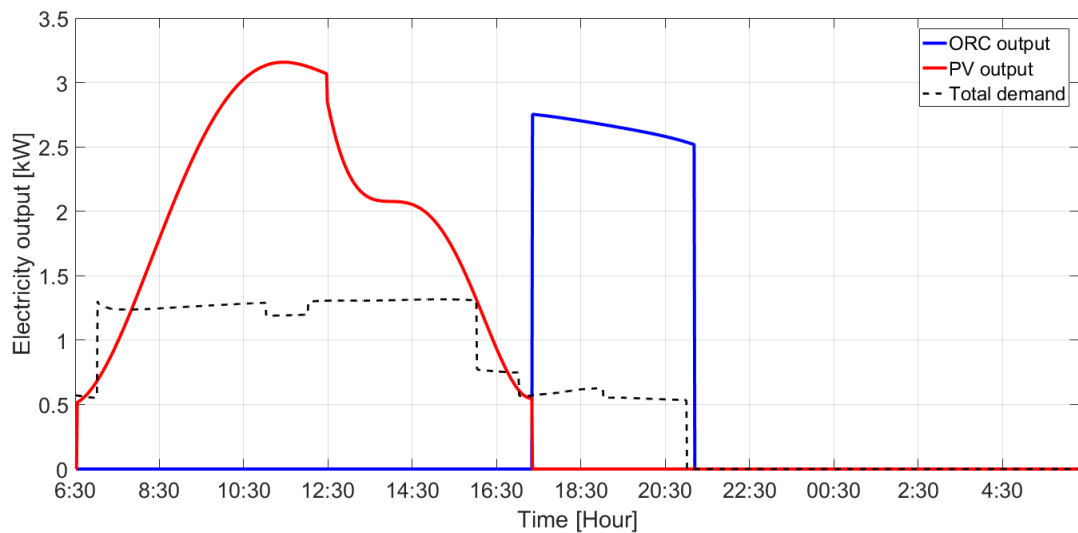
is that the water mass flowrate is reduced to 0.1 kg/s to generate less electricity because tank temperature reaches high level compared to design conditions. In the first case, 60 m² collector area and 3.534 m³ heat storage unit is simulated which are same with design parameters of the ORC-VCC unit. In order to convert the evacuated flat plate collectors into PV/T collectors, same methodology is followed with Chapter 4. Cover ratio of PV cells are selected 0.85 that means 60 m² collector area has 51 m² of a-Si cell area. In order to simulate one day application, initial temperatures are determined by applying the same approach given in Chapter 3. Figure 5.16a. shows temperature variation inside the water tank with time. As ORC circulation water flow rate is reduced to 0.1 kg/s, degree of thermocline in the water tank increases. Solar array increases the temperature with high solar collector flow rate during daytime. The main reason of this increment is ORC doesn't consume any stored heat from the water tank until 5 pm because PV cells produce required electricity and all collected heat stored in the water tank. However, when sun sets ORC starts to operate and receives heat from the tank.



a) Temperature distribution inside the water tank



b) Collector thermal efficiency and electrical efficiency variation



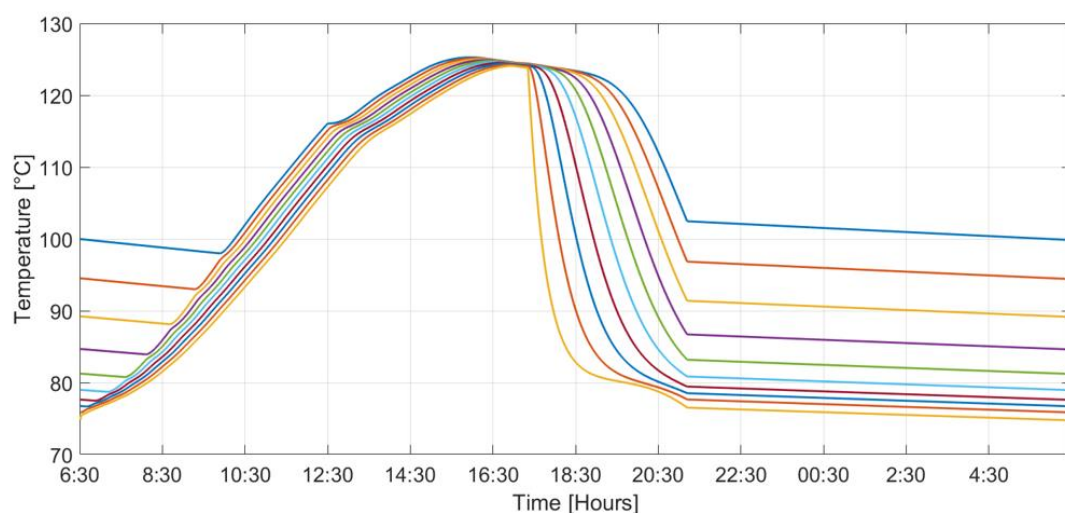
c) Produced electricity from PV and ORC

Figure 5.16 PV/T-ORC-VCC collector efficiencies and electricity output profile ($d=1.5\text{m}$, $h=2\text{ m}$ (3.534 m^3), $m_w=0.1\text{ kg/s}$, $A_{\text{collector}}=60\text{m}^2$)

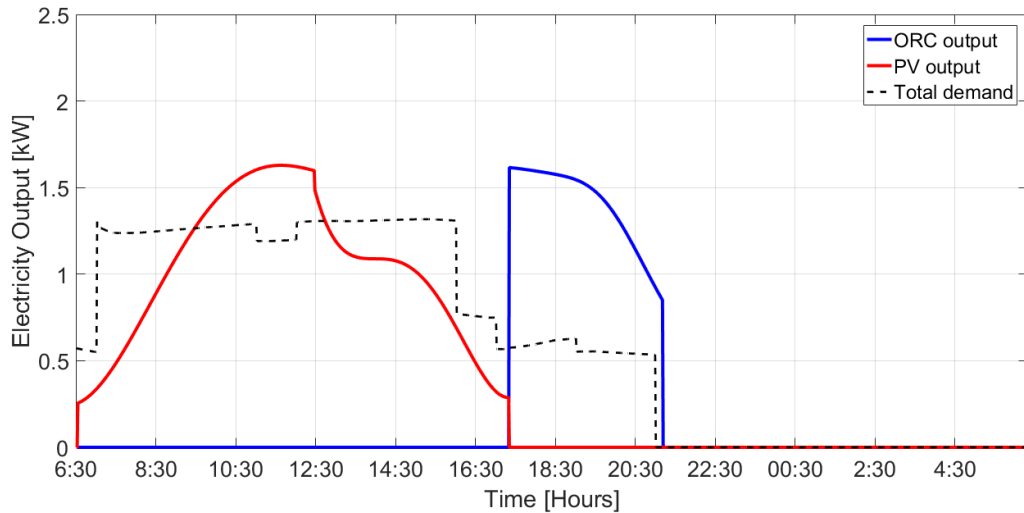
Figure 5.16b shows collector thermal efficiency and electrical efficiency of the PV cells on the day. Thermal efficiency pattern seems quite similar with the solar radiation profile as normal. However, PV electrical efficiency remains almost constant at 8%. This is the result of a-Si cell characteristics. Figure 5.16c shows electricity output and demand profile. PV output and ORC output generally higher than demand except early

morning. Since solar radiation is not strong enough at that period PV output can not match with the demand. For given conditions in a day, 10.21 kWh electricity is produced by ORC and 21.51 kWh electricity is produced by PV, electricity demand is 14.80 kWh.

In order to show effect of the sizes, the collector area is reduced to 30 m² and water tank volume is decreased to 1.57 m³. Although the storage size is reduced, maximum temperature in the tank decreases, as seen in Figure 5.17a. The main reason is using less collector area in the system, it reduces collected heat which means stored heat in the water tank. Figure 5.17b shows generated electricity profiles. In this case, PV output reduced considerably because of less area but ORC output still high enough to provide required electricity to the building. Daily outputs are 5.4 kWh and 11.04 kWh for ORC and PV, respectively. Although produced daily electricity looks enough for demand it does not match with it. In the following subsection, PV-ORC operation mode at the same time is investigated.



a) Temperature distribution inside the water tank



b) Produced electricity from PV and ORC

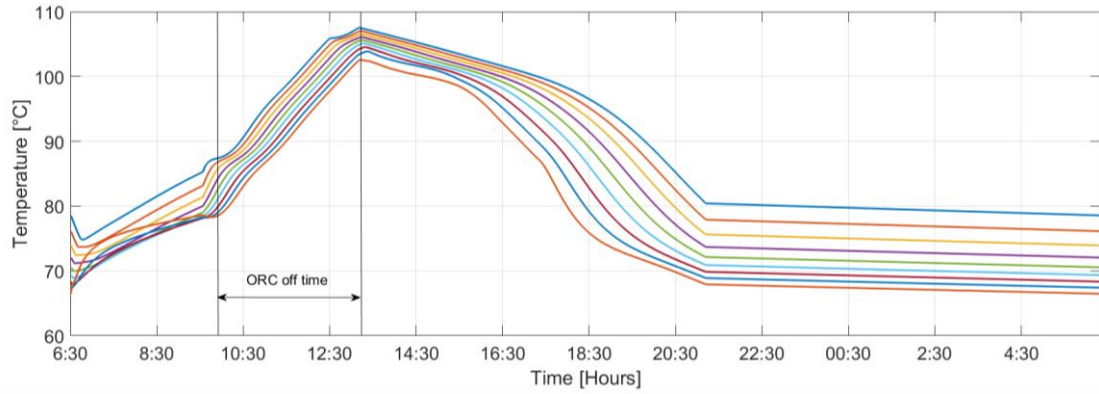
Figure 5.17 PV/T-ORC-VCC temperature gradient and electricity output profile ($d=1$ m, $h=2$ m (1.57 m³), $m_w=0.1$ kg/s, $A_{\text{collector}}=30$ m²)

5.6.2 Simulations of PV/T-ORC-VCC unit for different control methods

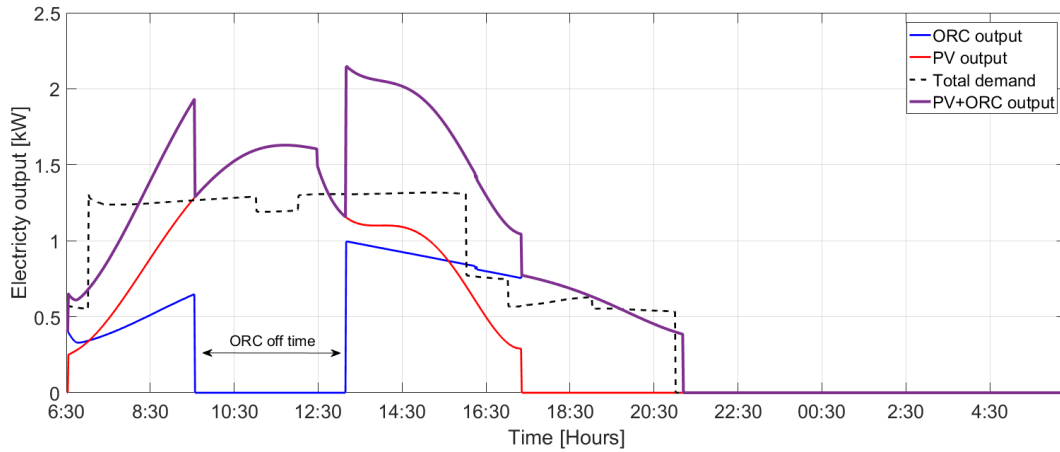
As seen from the previous subsection, modification of ORC-VCC unit with PV/T collectors reduces required collector area and heat storage volume by producing more electricity than the total demand. However, early morning demand which is around 7 am to 7.30 am is still higher than PV output even for 60 m² collector area. Different operation methodologies can be applied to solve this issue such as PV-ORC operation at the same time, the ORC operates during day and produces electricity with the PVs or it can be operated two hours in the morning then it stops until evening. By referring to Figure 5.17, PV-ORC is adapted, and Figure 5.18 is given.

In order to reduce heat consumption from the storage by ORC, for some periods ORC mode is turned to off mode when PV output is enough for the demand. Figure 5.17a shows temperature variation in the water tank. Compared to Figure 16a, lower temperature is observed. The reason of the low temperature is ORC heat consumption. By reducing water temperature, ORC output also reduces as seen from the Figure 5.17b. Since PV output is not enough for the demand, ORC operates in the early

morning with higher water circulation flow rate (0.3 kg/s) to increase power output. However, ORC boosting cannot provide required power.



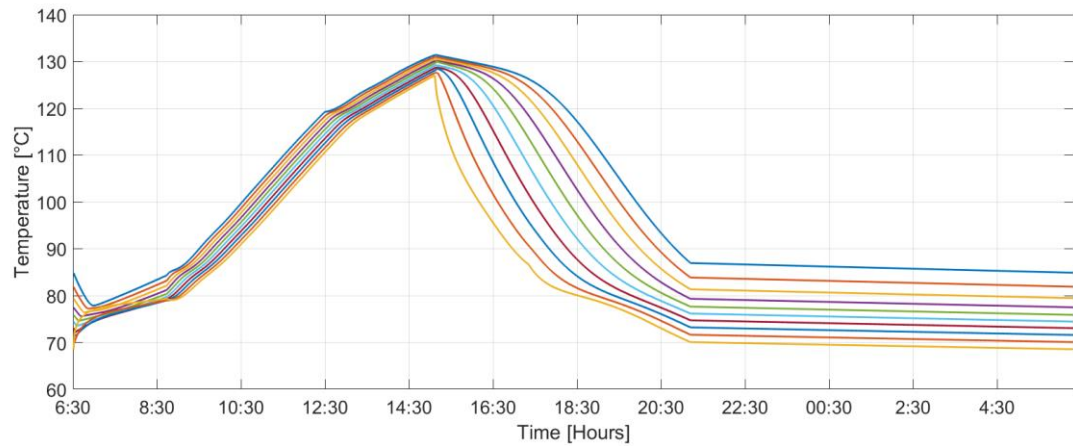
a) Temperature distribution inside the water tank



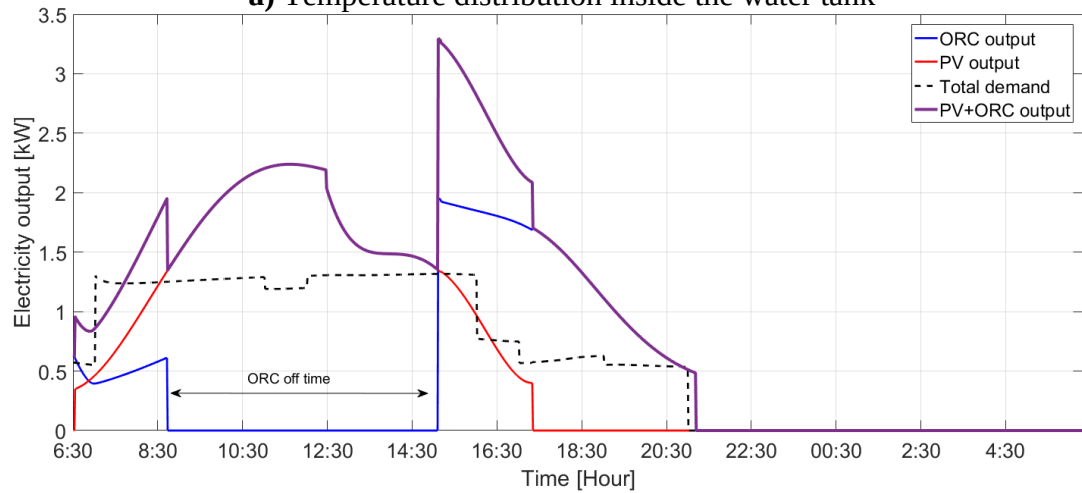
b) Produced electricity from PV and ORC

Figure 5.18 PV/T-ORC-VCC temperature gradient and electricity output profile ($d=1$ m, $h=2$ m (1.57 m³), $m_w=0.3$ kg/s and 0.1 kg/s, $A_{\text{collector}}=30$ m²)

For the final case, solar collector area is increased to 40 m² to increase heat input and PV output. Same methodology and controlling procedure are followed. Figure 5.19a shows water tank temperature variation. Compared to previous case maximum tank temperature is obtained 131 °C in new case, it was 108 °C. There are two reasons for this temperature increment. First one is heat input increases with the solar collector area increment and other one is ORC working time reduces because PV output increases.



a) Temperature distribution inside the water tank



b) Produced electricity from PV and ORC

Figure 5.19 PV/T-ORC-VCC temperature gradient and electricity output profile ($d=1$ m, $h=2$ m (1.57 m³), $m_w=0.3$ kg/s and 0.1 kg/s, $A_{\text{collector}}=40$ m²)

As it is seen from the analysis, it is not easy to exactly match with the demand profile for given components because heat exchangers have been designed for higher ORC outputs. However, it is easy to indicate that PV/T modification has promising results. By this way collector area and heat storage volume can be reduced considerably. For future studies, an optimisation will be conducted, and a logic will be generated to match with the demand profile by using PV/T-ORC-VCC operation for all summer period.

5.7 Summary

In this chapter, the modelling of the solar ORC-VCC system integrated with a sensible heat storage unit operating at off-design conditions was presented. Electricity and cooling demand from the end user was also considered from the point of view of control strategies. In the solar part, evacuated flat plate collectors have been used and collectors linked with a sensible heat storage unit. As working conditions have an influence on expander performance, electricity output is also effected. It is shown that water flow rate control strategies are promising and can improve the precision between the produced work and the demand. Sizes of the components have been determined according to summer conditions for Istanbul for ORC-cooling mode. In the second part of the chapter, evacuated collectors are replaced with a-Si PV/T collectors to benefit from the PV output by using same components and conditions. Concluded remarks are given:

- It was found that to meet electricity and cooling demand for studied office, 80 m² collector area and 9.4 m³ heat storage are required for ORC-VCC system.
- This system with selected components have been analysed for a winter day as ORC-heating mode and 5.5 kWh electricity and 104 kWh heating load potential has been obtained.
- By modifying evacuated flat plate collectors with a-Si cells, electricity output increased considerably that results to size reduction of the collector area and water tank.
- For PV/T-ORC-VCC system, to meet electricity and cooling demand of the same office, 40 m² collector area and 1.57 m³ almost satisfied the demand.

CHAPTER 6 An investigation into the characteristics and performance prediction of scroll expander

6.1 Introduction

This chapter presents the investigation of the characteristics of the scroll expander. As expander is the most critical component of the solar ORC systems, its performance characteristics need to be investigated in detail. Current popular expander model, semi-empirical model, is very useful and accurate to calculate shaft power output by considering heat losses, leakage flows and over or under expansions. However, there are different capacity and brands of expanders available in the market, as their scroll design parameters can be different from the validated expander's model, used empirical equations would be ineligible for different capacities and new method needs to be developed to replace empirically obtained parameters. Moreover, it is seen from the literature that, the effect of the generator on system performance cannot be ignored for small-scale power generation systems.

For these reasons, the expander geometric model is proposed and improved by empirical equations such as clearance and friction correlations. Two objectives have been clarified into scroll expander modelling. One of them is clearance between the scrolls. The clearances are responsible of the leakage flows inside the scroll expander because they form leakage area and decrease volumetric efficiency and isentropic efficiency of the scroll expander. As mentioned in previous chapters, mostly accepted semi-empirical model assumes leakage area as a constant parameter but it varies by

operating conditions. Thus, an investigation on leakage flow is conducted in this chapter. Other one is investigation of performance of expander-generator coupling. The recent expander model is updated to calculate rotation speed and electricity output. By this way, more realistic calculations can be performed, and the electricity output can be predicted.

The chapter starts with the experiment setup presentation. In Section 6.3, equations for geometric modelling of the scroll expander are given in detail. Geometric modelling of the expander consists of three subsections. Firstly, scroll geometry which is related with features of the scroll shape is given in Section 6.3.1, later, volume calculations and leakage area variations are presented in detail. In Section 6.4, thermodynamic equations are given such as energy balance and heat transfer equations. In Section 6.5, motion equations are given and finally, simulation results and measured data are compared in Section 6.6.

6.2 Experimental rig and expander model implementations

In order to obtain empirical correlations and validate the proposed model with experimental results, a test rig was set up at the project collaborator institution, University of Science and Technology of China. The same experiment setup has been used and presented in another study by Li et al.[192]. In the test rig, an ORC unit was tested using an oil free semi hermetic scroll expander produced by Air Squared, Inc. [193]. Figure 6.1 shows diagram of the ORC system. Originally system has two expanders but, in this case, only scroll expander line was operated. The system has three loops. First one is oil cycle; the oil is heated to a desired temperature by the thermal bath then heat is transferred to working fluid by evaporator. Second cycle is

ORC cycle. R123 is used as a working fluid and it evaporates via evaporator by transferred heat from the oil. Working fluid enters into scroll expander and drives it to rotate. Generator is magnetically connected to expander's rotating shaft and generates electricity. After expander, condenser ejects heats from the R123 refrigerant and changes its state to liquid. The final cycle is cooling water cycle, cooling water absorbs heats from the working fluid by the condenser. This heat is carried away by the cooling tower.

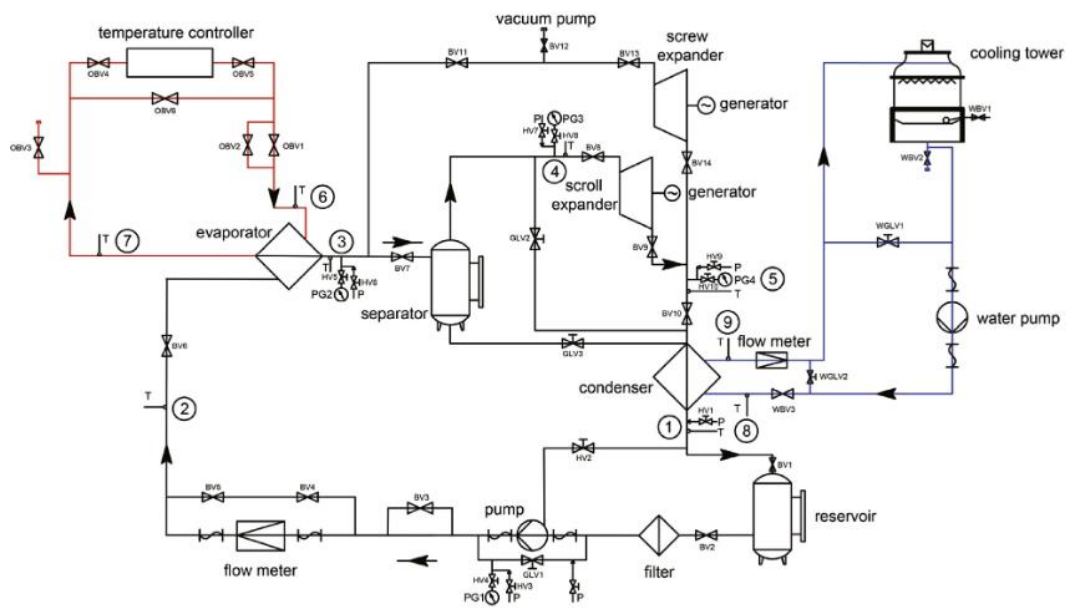


Figure 6.1 Schematic diagram of the ORC system

Figure 6.2 shows experimental layout of the system and expander generator coupling. The oil-free scroll expander E15H022A-A03 has been used. Expander's maximum output power, speed, inlet temperature and pressure are 1 kW, 3600 rpm, 175 °C and 13.8 bar, respectively. The coupled generator (AB30L) is a single-phase generator and was produced by Wanco, Inc. It is connected to the expander via a magnetic coupler in the housing. The rated volts, amps, hertz and speed are 240 V, 10 A, 50 and 3000 rpm.



Figure 6.2 Experimental layout and expander-generator coupling

To make consistent with the given nomenclature of the equations and to simplify the diagram Figure 6.3 is given. The measured parameters including temperature and pressure, are denoted by T and P next to the lines. Mass flow rate of the working fluid R123 and electricity output were measured by the flow meter and digital power meter, respectively.

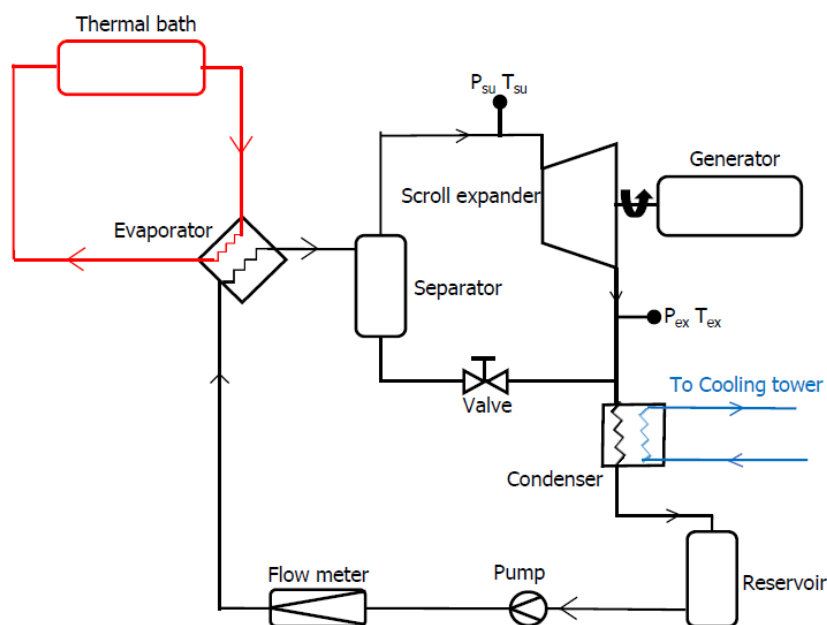


Figure 6.3 Simple diagram of the tested ORC system

In order to give accurate and reliable results, evaluation of the uncertainties of the experimental setup is very important. Copper-constantan thermocouples are used for measuring the temperatures with an accuracy of ± 0.5 °C. To measure pressure, three types of ceramic pressure transmitters produced by Huba Control Co. were utilized. The pressure ranges were -1 to 9 bar (scroll expander outlet, condenser outlet and organic fluid pump outlet), 0 – 25 bar (scroll expander inlet) and 0 – 30 bar (organic fluid pump inlet and evaporator outlet). And the accuracies were $\pm 1.0\%$. The R123 mass flow rate was measured by the flow meter (MFM2081K-60P/DN25) fabricated by KROHNE Group. The zero-point stability was ± 0.012 kg/min, and accuracy was $\pm 0.15\%MV + Cz$. The output voltage, current, electricity and frequency of the generator were measured by the digital power meter (8716C1T-RS) provided by Qingdao Qingzhi Company. And the accuracies were $\pm 0.5\%$, $\pm 0.5\%$, $\pm 0.5\%$, and

$\pm 0.1\%$, respectively. An uncertainty analysis as recommended by Holman [194] has been conducted. It's found that maximum uncertainty of the obtained power is $\pm 0.7\%$. According to datasheet of the expander given by the company, the built-in volume ratio is 3.5 and displacement volume is 14.5 cm^3 . Thankfully, Ma et al. [195] used the same expander in their work, measuring and presenting the geometric parameters of the scrolls. Thus, by using the given scroll geometry parameters, the scroll profile is obtained. The geometric parameters of the scroll expander are given in Table 6.1. by using the geometric parameters, scroll is formed, and full model can be built. Details are given in following subsections.

Table 6.1 Geometric parameters of the scroll expander of Air Squared

Parameter	Symbol	Value
Height of scroll	h_s	22.4 mm
Radius of the base circle	r_b	2.37 mm
Thickness of the scroll	t_s	4.08 mm
Number of chambers	Nc	4
Pitch of the scroll blade	Pit	14.91 mm
Start angle of the involute	ϕ_0	49.25°
Swept volume	V_{sw}	14.5 cm^3

Built-in volume ratio	V_{ratio}	3.5
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6.3 Geometric model of a scroll expander

Geometric model of the scroll expander consists of three main sections. Forming scroll geometry is one of the main sections to explain geometric features. Thermodynamic model deals with the refrigerant side and energy equations. Finally, mechanical model is to analyse frictions and rotation. Sections are summarized below:

- Scroll geometry model: By modelling the scroll geometry, volumes of the chambers can be defined which are related to built-in volume ratio, swept volume, leakage areas etc.
- Thermodynamic model: In thermodynamic model, conversation of mass and energy equations are applied on the system.
- Mechanical model: Motion equations, torques and frictions are investigated with the mechanical model.

6.3.1 Scroll geometry

Scroll geometry is one of the main factors effecting the scroll compressor and expander performances. A reliable and simplified model is necessary to merge with a thermodynamic model. Compared to scroll expanders, compressor machines have been more studied and provide us significant knowledge which can be applied for designing the scroll expanders [122]. The geometric model consists of equations to identify scroll profiles and calculates volumes of the chambers and leakage areas

according to orbiting angle of the scroll. To build an accurate and flexible mathematical model, some of the geometric features need to be known.

The shape of the scroll is an involute of circle and a scroll is defined by two involutes namely inner and outer involute. These profiles are established according to a mathematical differential equation given in Eq. (6.1). Most of the equations are taken from Bell [126].

$$\frac{\delta L(\phi)}{\delta \phi} = r_b \quad (6.1)$$

Where L indicates tangential distance, ϕ represents involute angle, subscript “0” indicates initial and r_b is base circle radius. For inner and outer involutes equations are given as:

$$L_i = r_b(\phi - \phi_{i0}) \quad (6.2)$$

$$L_o = r_b(\phi - \phi_{o0}) \quad (6.3)$$

The thickness of the scroll is determined by:

$$t = L_o - L_i = r_b(\phi_{i0} - \phi_{o0}) \quad (6.4)$$

By fixing the origin of the coordinate system at the centre of the base circle, the individual coordinate points x and y on the involute can be determined by Eq. (6.5) and Eq. (6.6). Details of initial angles and involute definition is shown in Figure 6.4.

$$x = r_b(\cos\phi + (\phi - \phi_0)\sin\phi) \quad (6.5)$$

$$y = r_b(\sin\phi - (\phi - \phi_0)\cos\phi) \quad (6.6)$$

Other two important geometric features which are shown in Figure 6.4, starting angle ϕ_s and ending angle ϕ_e . Starting angle can be different for inner and outer involutes but ending angle can be generally same for both. When we know base circle radius, initial and starting angles and ending angle, it is possible to form a scroll by using given equations.

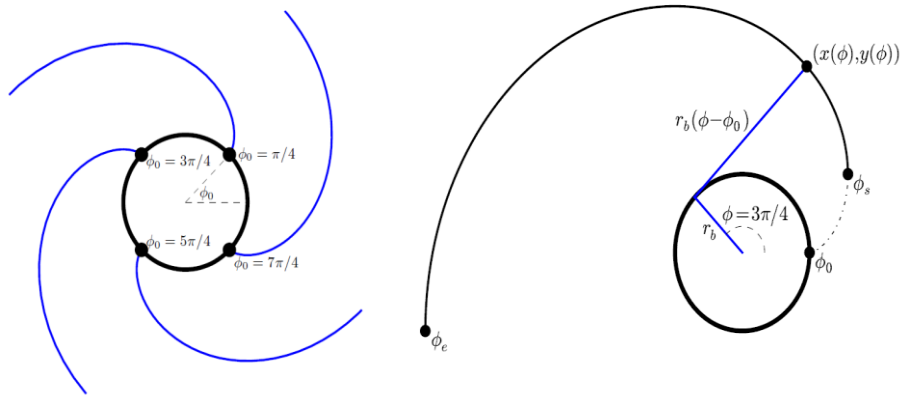


Figure 6.4 Details of initial angles and general involute definition [126].

Detailed coordinate point equations for inner and outer involutes for a fixed scroll are given Eqs. (6.7-6.10) and schematic for fixed scroll geometry is given in Figure 6.5.

$$x_{fi} = r_b(\cos\phi + (\phi - \phi_{i0})\sin\phi) \quad (6.7)$$

$$y_{fi} = r_b(\sin\phi - (\phi - \phi_{i0})\cos\phi) \quad (6.8)$$

$$x_{fo} = r_b(\cos\phi + (\phi - \phi_{o0})\sin\phi) \quad (6.9)$$

$$y_{fo} = r_b(\sin\phi - (\phi - \phi_{o0})\cos\phi) \quad (6.10)$$

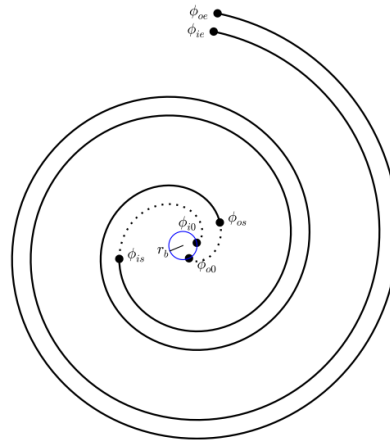


Figure 6.5 Involute forming the fixed scroll geometry [126]

Fixed scroll works with another scroll called moving or orbiting scroll which is assembled together in the scroll machines. These two scrolls are in conjugacy and orbiting scroll is then offset by a fixed amount away from base circle centre of the fixed one. Orbiting scroll orbits (not rotates) around the centre point of the base circle of the fixed scroll. There is a conjugacy between fixed and orbiting scroll. That means two scrolls contact each other perfectly at some conjugate angles. Orbiting radius can be calculated by Eq. (6.11):

$$r_o = r_b\pi - t_s = r_b(\pi - \phi_{i0} + \phi_{o0}) \quad (6.11)$$

Other relevant parameters also given in equations such as pitch and ending angle given by Ma et al.[195] and outer starting angle, displacement for compressor and built-in volume ratio are given by Bell et al. [128], [196].

$$Pit = 2\pi r_b \quad (6.12)$$

$$\phi_e = (2N_c + 1/2)\pi \quad (6.13)$$

$$\phi_{os} = \phi_e - \theta - 2 \cdot \pi \cdot N_c - \pi \quad (6.14)$$

$$V_{disp} = -2\pi h_s r_b r_o (3\pi - 2\phi_e + \phi_{i0} + \phi_{o0}) \quad (6.15)$$

$$V_{ratio} = \frac{3\pi - 2\phi_e + \phi_{i0} + \phi_{o0}}{-2\phi_{os} - 3\pi + \phi_{i0} + \phi_{o0}} \quad (6.16)$$

N_c is number of expansion pairs, θ is crank angle or orbiting angle and h_s indicates scroll height. By using given equations and specific required parameters, scroll geometry can be formed.

By using given equations and geometric parameters, scroll positions for different orbiting angles can be formed [197]. Figure 6.6 shows changing of the scroll positions for one rotation.

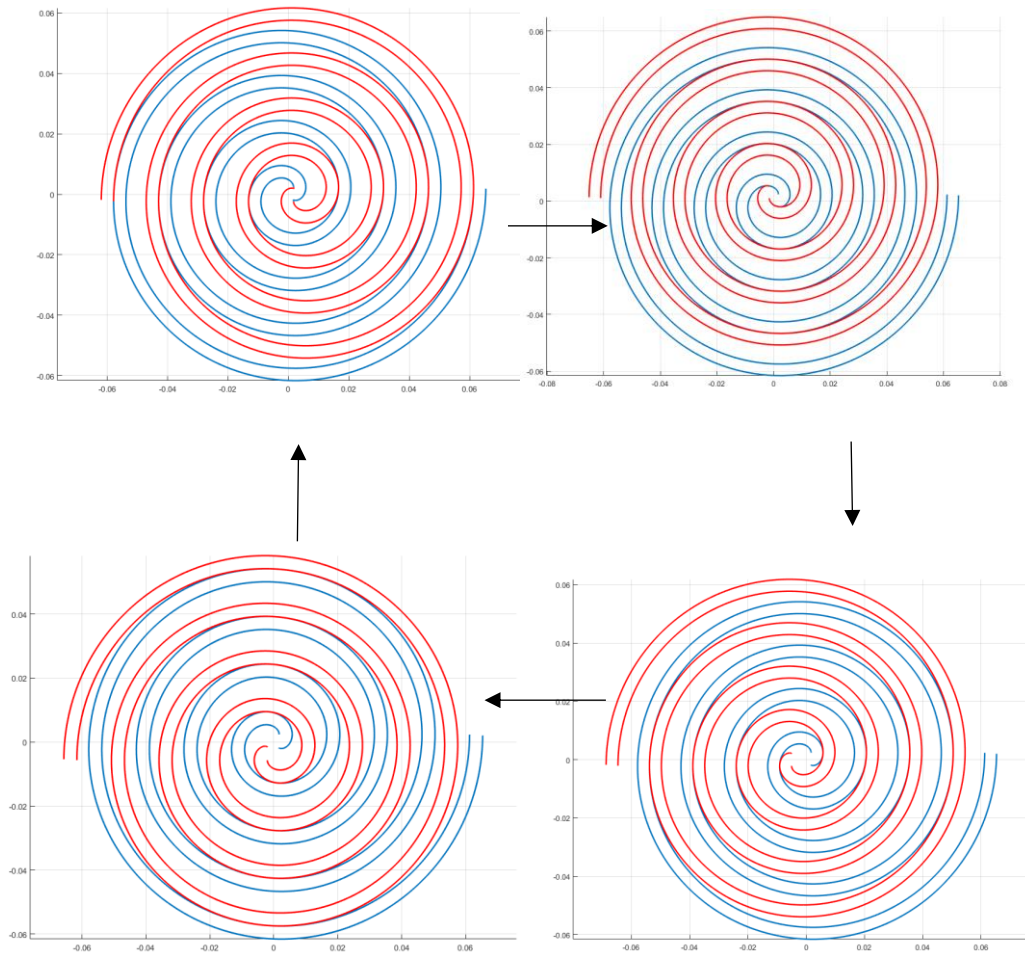


Figure 6.6 Scroll positions for orbiting angles: $\theta = 0$, $\theta = \pi/2$, $\theta = \pi$, $\theta = 3\pi/2$

6.3.2 Volumes of chambers

Volume of intake chamber of the expander is actually the discharge chamber of the scroll compressor. Additionally, expansion and discharge chambers are compression and suction chambers of the scroll compressor. In this subsection, the original scroll

compressor equations are given and results are converted into expander operation. The equations are taken from Chen et al. [198], the equation can be written as:

$$V_{intake} = h_s r_b r_o \left\{ [\phi_k^2 - (\phi_{os} + \pi)^2] - \frac{3\pi}{2} [\phi_k - (\phi_{os} + \pi)] \right\} \quad (6.17)$$

$$\phi_k = \phi_e - 2\pi - \theta \quad (6.18)$$

The expansion is the compression chamber of the compressor. The relevant equation is given in Eq. (6.19).

$$V_{expansion} = 2\pi h_s r_b r_o \left(2\phi_e - 2\theta - \frac{7}{2}\pi \right) \quad (6.19)$$

For the discharge chamber:

$$V_{discharge} = h_s r_b r_o \left(2\theta\phi_e - \theta^2 - \theta\frac{3}{2}\pi \right) \quad (6.20)$$

Since used expander for testing has two expansion chambers, the volume of the expansion chamber equation is implemented from [195] can be used to find volume variations of the expansion chambers:

$$V_{ci} = 4\pi r_b^2 h_s (\pi - 2\phi_0)(\theta - 3\pi + 2i\pi) \quad (6.21)$$

i represents number of chambers, for expansion process, the requirement for i is ($2 \leq i \leq N_c$).

By using chamber volume equations, pressure and temperature variation of the working fluids through the expansion process can be obtained. In order to be sure about the scroll geometry parameter and equations' validity a calculation procedure can be done. Regarding to used geometric parameters for calculation of scroll geometry, the reference paper didn't give any information about accuracies and resolutions about measurements. To validate measured geometric parameters, scroll company's data were used. In the website, volume ratio and displacement are given as 3.5 and 14.5 mm³/rev, respectively [193]. Eq. (6.15) can be used to calculate displacement for scroll compressor. By using given geometrical parameters and equations, the expander's displacement is found 49.46 mm³ when under compressor operation. In order to be sure about the scroll geometry parameter and equations' validity, it is known from the data sheet that volume ratio is 3.5 for this expander. $49.46/3.5=14.1314$ mm³ is found for expander displacement and company value is 14.5. Thus, results are in good agreement with 2.5 % deviation. The key point here is 5 measured parameters ($h, r_o, r_b, \phi_{i0}, thickness$) have been used to calculate this value thus we can conclude that validity of the measured geometrical parameters is in good range. To prove 2.5 % deviation, an uncertainty analysis has been done. The uncertainty of displacement volume of the scroll compressor was calculated 2.82% when all parameters ($r_b, r_o, h, \phi_{i0}, thickness$) have 1% accuracy. If accuracies were 0.5%, this time total uncertainty would be 1.41%.

6.3.3 Leakages

Normally, orbiting scroll and fixed scroll should mesh perfectly during the operation to obtain higher volumetric efficiency and performance; however, internal leakage is happened inside the expander. There are two kind of leakages have been reported in

scroll machines, flank leakage and radial leakage. Flank type leakage appears over the flank clearance between the side surfaces of fixed and orbiting scrolls. Radial leakage occurs over the radial clearance between the tip wall and base plate of the scrolls. Figure 6.7 shows the diagram of the two kind of leakages. Different clearance sizes have been used by several studies. Some studies used variable clearance size according to operation pressure, some of them used constant value for flank and another constant for radial clearance, but some authors preferred to use fixed values for both clearances.

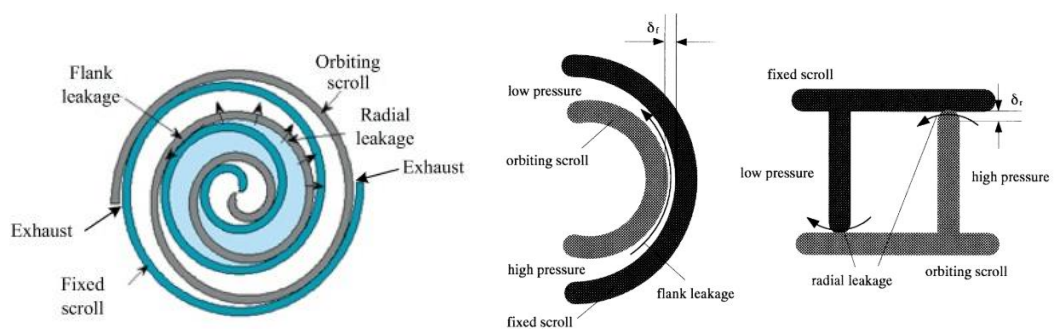


Figure 6.7 Scroll diagram and two kind of leakages [198]

Summary for clearance values has been given in Table 6.2. δ_r and δ_f indicate radial and flank clearances, respectively. Variation of the clearances according to pressure regarding to Reference [199] is given in Figure 6.8. Given equations state that gap between the scrolls depend on supply pressure but other references suggested to use constant clearance values.

Table 6.2 Summary of literature clearance values

Reference	Information	δ_r	δ_f
[198]	Scroll compressor modelling	$1.1 \cdot 10^{-6} \left(\frac{P_{dis}}{P_{suc}} - 1.67 \right) + 10^{-6}$	$-9.615 \cdot 10^{-5} \left(\frac{P_{dis}}{P_{suc}} - 1.67 \right) + 20 \cdot 10^{-6}$
[200]	Scroll compressor modelling	$\left[0.85 \cdot \left(\frac{P_{dis} - P_{back}}{P_{suc}} \right) + 1.65 \right] \cdot 10^{-6}$	$\left[-6.11 \cdot \left(\frac{P_{dis} - P_{back}}{P_{suc}} \right) + 32 \right] \cdot 10^{-6}$
[199]	Vapor injection scroll compressor	$1.1 \cdot 10^{-6} \left(\frac{P_{dis}}{P_{suc}} - 1.67 \right) + 10^{-6}$	$2.11 \cdot 10^{-6} \left(\frac{P_{dis}}{P_{suc}} - 1.67 \right) + 20 \cdot 10^{-6}$
[123]	Air scroll expander	$60 \mu m$ (5-100 μm)	$60 \mu m$ (5-100 μm)
[201]	ORC scroll expander	$40 \mu m$	$40 \mu m$
[202]	Air scroll expander	$40 \mu m$	$40 \mu m$

[203]	ORC scroll expander	$40\ \mu m$	$40\ \mu m$
[204]	Air scroll expander	$15\ \mu m$	$10\ \mu m$
[205]	Air scroll expander	$15\ \mu m$	$10\ \mu m$
[195]	Air scroll expander	$10\ \mu m$	$10\ \mu m$

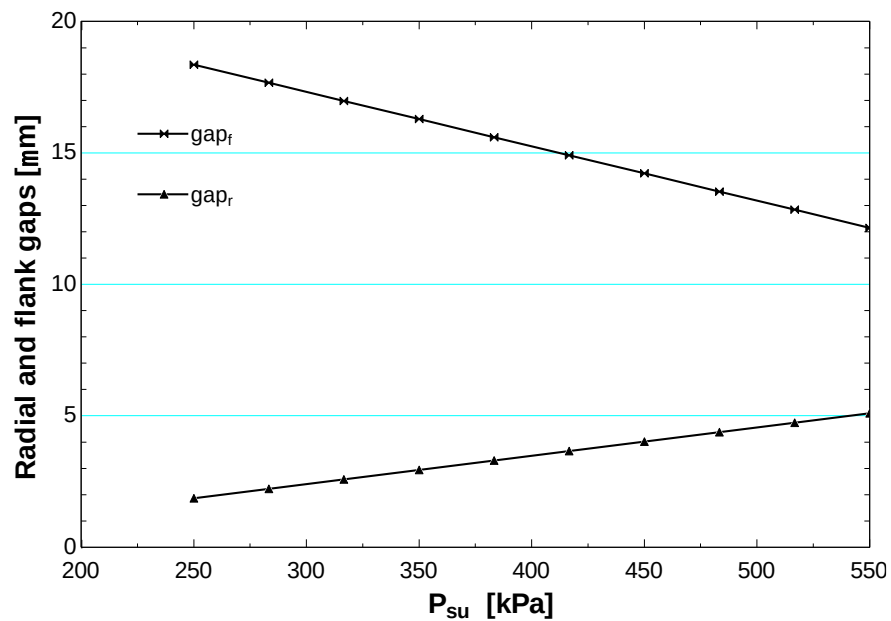


Figure 6.8 Clearance values by pressure according to Reference [199]

6.3.4 Leakage area calculations

As two kind of leakage flows are existing, radial and flank leakage area calculations are held separately. Chen et al. [198] and Wang et al.[206] presented some equations

for in-flowing refrigerant and out-flowing refrigerants. Mendoza et al. [207] experimentally characterized the scroll expander leakage area. According to tests with ammonia, the equation is presented in Eq. (6.22) and empirical constants are given in Table 6.3.

$$A_{leak} = C_1 + C_2 \cdot P_{su} + C_3 \cdot T_{su} + C_4 \cdot P_r + C_5 \cdot N [m^2] \quad (6.22)$$

Table 6.3 Empirically obtained constants for Mendoza's study [207]

C_1	C_2	C_3	C_4	C_5
-3.41E-06	7.86E-10	1.39E-08	3.48E-08	-2.04E-08

However, Ma et al. [195] and Wu et al. [203] used more convenient equations for calculation of the leakage areas. As flank leakage happens on both sizes of the scroll, the length of the gap is the scroll height, the total flank leakage area is:

$$A_{flank}(\theta) = 2h_s \delta_f \quad (6.23)$$

For the radial leakage, the area varies according to orbiting angle because chambers are enlarging and shrinking during operation. Eq. (6.24) gives relation of the radial leakage area with the orbiting angle θ and chamber number i .

$$A_{radial}(\theta) = \delta_r 2\pi r_b (2(i-1)\pi + \theta) \quad (6.24)$$

For given tested scroll expander geometry by using $30\ \mu\text{m}$ clearance for both leakages, total flank area is found $1.34\ \text{mm}^2$ and maximum radial area is found $2.8\ \text{mm}^2$. The total leakage area is $4.14\ \text{mm}^2$. Figure 6.9 is given to show leakage area variation by orbiting angle. As it is mentioned above, flank leakage area remains constant but radial leakage area increases by orbiting angle during rotation.

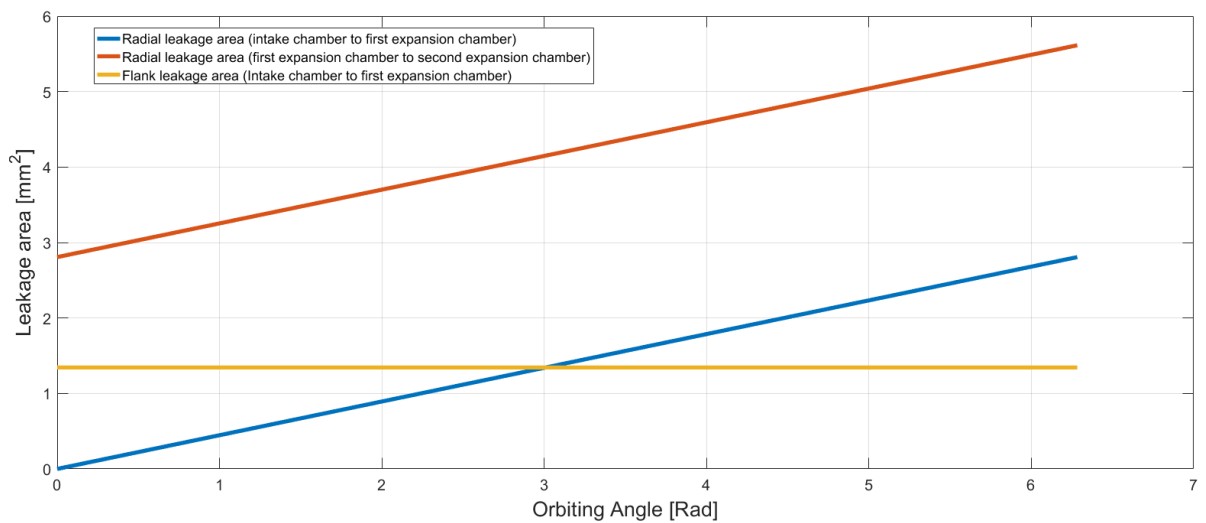


Figure 6.9 Leakage areas variation with orbiting angle

6.4 Thermodynamic Model

Thermodynamic model consists of mass flow rate calculation and energy equations of the working fluid. Heat transfer between chambers in scroll expander and heat transfer to the ambient via expander shell need to be calculated.

6.4.1 Mass flowrate

There are some models available for calculating the mass flow rate of the working fluid. For air used expanders, some authors preferred to use valve models to control

the flow rate. The semi empirical model uses a constant cross-sectional area for calculating the mass flow rate using the adiabatic pressure drop approach. This model offers that the expander intake chamber volume is filled with supplied working fluid. Since volume changes with the orbiting angle, the cumulated mass in the chamber reaches maximum at the end of one full rotation and all collected working fluid is then transferred into the next chamber. During that operation, the pressure in the intake chamber remains same as the supply pressure. Figure 6.10 shows an example for the theoretical mass in the intake chamber for another expander. As it seen from the figure, inlet pressure has an influence on the collected mass in the intake chamber because refrigerant density increases if gas has higher pressure. Higher supply pressure also increases rotation speed, but it will be investigated later.

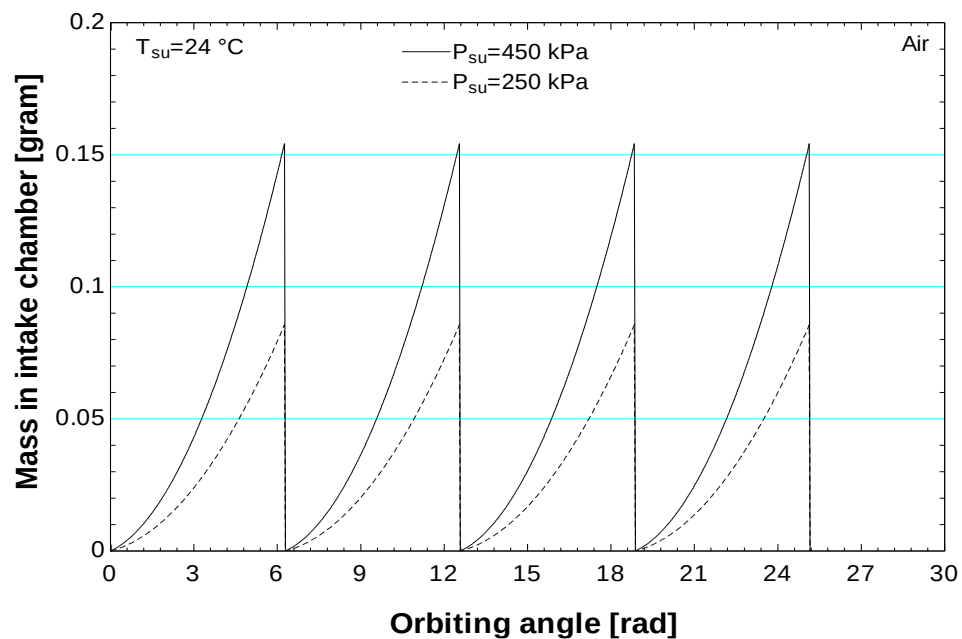


Figure 6.10 Working fluid mass taken by the scroll expander by orbiting angle

Theoretical mass flow rate can be calculated but we need to include leakage flow rates to find total mass flow rate coming into the expander. The general mass flow rate equation is given in Eq. (6.25):

$$\dot{m}_{total} = \rho_{su} V_{sw} \frac{N}{60} + \dot{m}_{leak} \quad (6.25)$$

As it is seen from the equation, mass flow rate depends on rotation speed (N), swept volume or intake volume of the expander (V_{sw}), density of the gas (ρ_{su}) and leakage flows. For the leakage mass flow rate, it can be referred to the orifice theory. Commonly used equations are given for one chamber to next one. For flank leakage Eq. (6.26) and for radial leakage Eq. (6.27) are given:

$$\frac{dm_{flank}}{d\theta} = f_f \frac{A_{flank}(\theta)}{\omega} \sqrt{\frac{2k}{k-1} P_i(\theta) \rho_i(\theta) \left(\varepsilon^{\frac{2}{k}} - \varepsilon^{\frac{k+1}{k}} \right)} \quad (6.26)$$

$$\frac{dm_{radial}}{d\theta} = f_r \frac{A_{radial}(\theta)}{\omega} \sqrt{\frac{2k}{k-1} P_i(\theta) \rho_i(\theta) \left(\varepsilon^{\frac{2}{k}} - \varepsilon^{\frac{k+1}{k}} \right)} \quad (6.27)$$

f_r and f_f are flow coefficients of the working fluid, recommended at 0.87-0.95 for air.

ε is expansion ratio which is given in Eq. (6.28):

$$\varepsilon = \max \left(\frac{P_{i+1}}{P_i}, \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}} \right) \quad (6.28)$$

During the expansion process, pressure in the expansion chamber decreases but intake chamber pressure stands as constant. Figure 6.11 is obtained for tested expander.

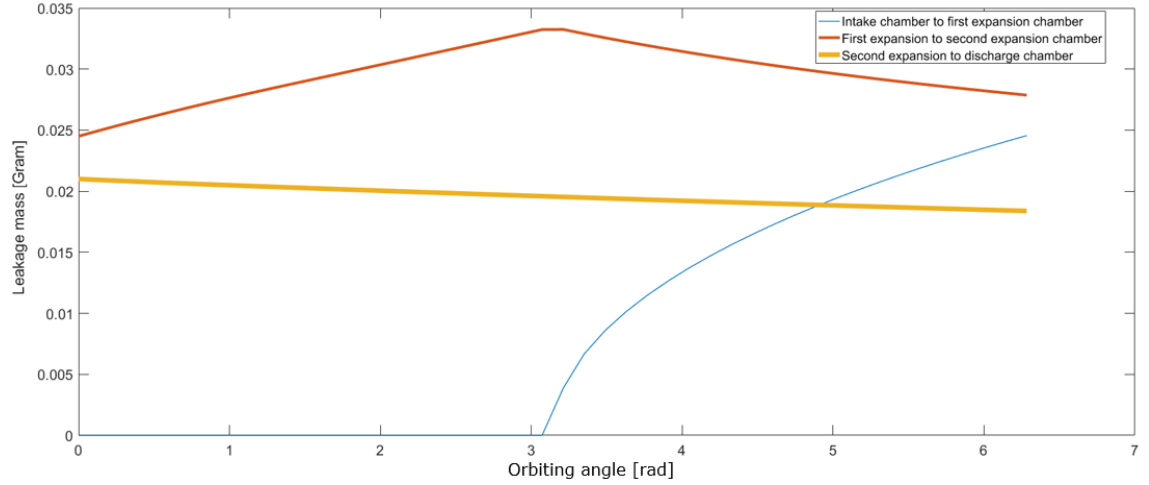


Figure 6.11 Total leakage mass variation during one rotation

6.4.2 Mass and energy balance of the working fluid

A chamber is taken as a control volume there are working fluid inlet and outlet by the flank and radial clearances. The mass conversation of the control volume given in Eq. (6.29).

$$\frac{dm}{d\theta} = \frac{dm_{ini}}{d\theta} - \frac{dm_{outi}}{d\theta} \quad (6.29)$$

m indicates mass in the chamber, m_{ini} and m_{outi} are leakage inlet to the chamber from upper chamber and leakage out to the next chamber, respectively.

Based on the energy conversion, Eq. (6.30) can be written for the working fluid:

$$\frac{d(mu)}{d\theta} = \frac{dQ}{d\theta} + \frac{dm_{in}}{d\theta} h_{in} - \frac{dm_{out}}{d\theta} h_{out} + \frac{dW}{d\theta} \quad (6.30)$$

6.4.3 Heat transfer equations

During operation of the expander, heat is transferred from working fluid to the base plate and to other chambers by scrolls. To determine the heat transfer rate, firstly heat transfer coefficient of the working fluid needs to be found. Jang and Jeong [208] presented an empirical heat transfer equation for heat transfer of a oscillating wall in a fixed heating wall for a rectangular duct. The equation is given in Eq. (6.31):

$$Nu = \left(1 + 3.5 \frac{D_h}{D_c}\right) (1 + 8.8(1 - e^{-5.35St})) Nu_{DB} \quad (6.31)$$

D_h is hydraulic diameter of the flow duct, D_c represents mean diameter of the scroll curvature, related equation is given in Eq. (6.32):

$$D_c = 2r_b(\theta + 2(i - 1)\pi) \quad (6.32)$$

St is Strouhal number which is used for correcting the oscillating movement. f_o is oscillating frequency, A_{max} is oscillating amplitude, $\bar{U}$ is the mean velocity of the fluid.

$$St = \frac{f_o A_{max}}{\bar{U}} \quad (6.33)$$

$$f_o = \frac{\omega}{2\pi} \quad (6.34)$$

$$A_{max} = r_o \quad (6.35)$$

$$\bar{U} = \frac{\dot{m}_{tot}}{r_o h_s \rho} \quad (6.36)$$

The mean hydraulic diameter is calculated from Eq. (6.37):

$$D_h = \frac{2r_o h_s}{r_o + h_s} \quad (6.37)$$

Dittus-Boeter heat convection correlation can be used for Nu_{DB} .

$$Nu_{DB} = 0.025 Re^{0.8} Pr^{1/3} \quad (6.38)$$

When heat capacities of base plate and scroll wraps are neglected, heat transfer rate of working fluid in chamber i is given in Eq. (6.39):

$$\begin{aligned}
\frac{dQ_i}{dt} = & \frac{T_{ci+1} - T_{ci}}{\frac{t_s}{\lambda_{scroll}} + \frac{D_h}{Nu_{c+1}\lambda_{wf}} + \frac{D_h}{Nu_{ci}\lambda_{wf}}} A_i \\
& + \frac{T_{ci-1} - T_{ci}}{\frac{t_s}{\lambda_{scroll}} + \frac{D_h}{Nu_{c-1}\lambda_{wf}} + \frac{D_h}{Nu_{ci}\lambda_{wf}}} A_{i-1} \\
& + \frac{T_{amb} - T_{ci}}{\frac{t_{plate}}{\lambda_{plate}} + \frac{1}{\alpha_{amb}} + \frac{D_h}{Nu_{ci}\lambda_{wf}}} A_{pi}
\end{aligned} \tag{6.39}$$

λ indicates thermal conductivities, t_{plate} is thickness of the base plate, α_{amb} is natural convection heat transfer coefficient of the ambient air. For the intake chamber, second term of the equation disappears because it is the first chamber. For discharge chamber, T_{ci+1} in the equation is changed into T_{amb} . The heat transfer area between the base plate and the chamber is:

$$A_{pi} = \frac{2V_{ci}}{h_s} \tag{6.40}$$

The heat transfer area between the chambers except discharge chamber is:

$$A_i = 2\pi r_b h_s (2(i-1)\pi + \theta) \tag{6.41}$$

For the last chamber:

$$A_i = 2\pi R_{sh} h_s \tag{6.42}$$

R_{sh} is shell radius and its equation is given in Eq. (6.43):

$$R_{sh} = \sqrt{(r_b(\phi_e + \phi_o) + r_o)^2 + r_b^2} \quad (6.43)$$

Figure 6.12 shows a map diagram of the heat transfers and leakage flows through the expander. Since leakage flow occurs by the pressure difference between chambers, it has only one direction.

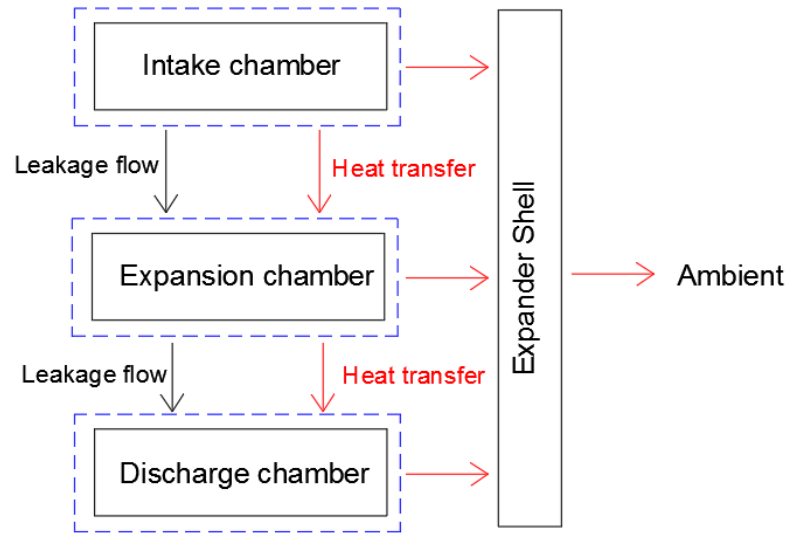


Figure 6.12 Heat transfers and leakage flows inside the expander

6.5 Mechanical model

6.5.1 Power output

To calculate the shaft output power of the expander, the equation is given by Zhang et al. [123] and Guangbin et al. [202]:

$$PW = \eta_m \cdot \frac{N}{120} \sum (V_{i+1} + V_i) (P_{i+1} - P_i) \quad (6.44)$$

Additionally, Wu et al. [203] presented an equation which includes tangential gas forces inside the expander and considers orbiting angle variations:

$$PW(\theta) = F_t(\theta) \cdot \omega \cdot r \cdot \eta_m \quad (6.45)$$

$F_t(\theta)$ is tangential force in scroll, given in Eq. (6.46) and r indicates crank radius.

$$F_t(\theta) = 2 \cdot r_b \cdot \theta \cdot h_s \cdot (P_1 - P_N) + 2 \cdot r_b \cdot \pi \cdot h_s \cdot \sum_{i=2}^{N-1} (2P_i - P_N) \quad (6.46)$$

6.5.2 Motion equations

Rotation speed is an independent parameter in the semi-empirical model but its effect on flow rate, leakages and friction losses cannot be ignored. Pressure difference between chambers in the expander generates gas forces such as, tangential, axial and radial forces. Through these forces, tangential force drives the rotation of the orbiting scroll. Ma et al. [195] presented detailed equations for analysing the forces acting on the scrolls. The tangential force is the one that drives the rotation of the moving scroll. Moreover, radial force and axial force also created by the moving of the orbiting scroll which contribute the generation of the friction force. Figure 6.13 shows all forces acting on the scroll.

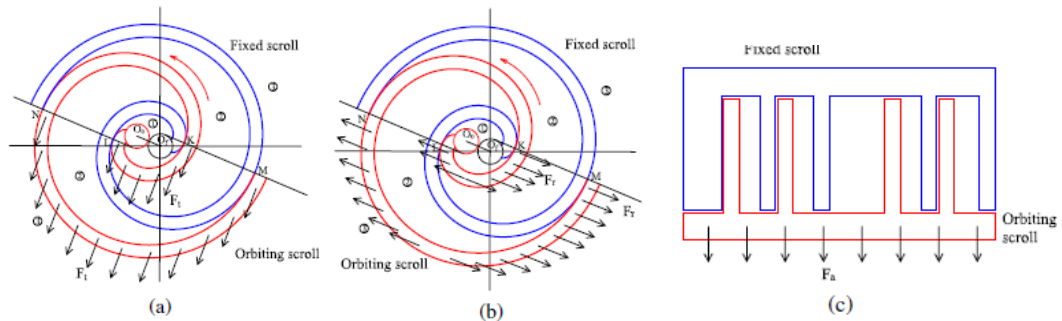


Figure 6.13 Generated forces on orbiting scroll, (a) tangential force, (b) radial force axial force, (c) axial force, adapted from [195]

Tangential force equation can be applied for each chamber:

$$F_t(i) = 2 \cdot h_s \cdot r_b \cdot [\theta + 2(i - 1) \cdot \pi] \cdot (P_i - P_{i+1}) \quad (6.47)$$

For the radial and axial forces Eq. (6.48) and Eq. (6.49) are given:

$$F_r(i) = 2 \cdot h_s \cdot r_b \cdot (P_i - P_{i+1}) \quad (6.48)$$

$$F_a = \sum (P_i S_{ci}) - P_b S_{orb-plate} \quad (6.49)$$

Each point on the orbiting scroll rotates along a circle at the same diameter but with different circular centre. The torque generated by tangential forces is given in Eq. (6.50):

$$T_t = r_o \sum_{i=1}^{N_c} F_t(i) \quad (6.50)$$

Given torque is driving moment of the rotation, in order to rotate the scroll, this torque needs to overcome friction torque T_{fr} and electromechanical torque T_{em} if generator connected. For the expander-generator rotation Eq. (6.51) is given:

$$\begin{aligned} (J_{orb} + J_{sha} + J_{arm} + J_{old} \sin^2 \theta) \frac{d\omega}{dt} - J_{orb} \sin \theta \cos \theta \omega^2 \\ = T_t - T_{fr} - T_{em} \end{aligned} \quad (6.51)$$

J represents the inertia moment of the rotating parts, subscripts orb, arm, sha, old are orbiting scroll, armature, shaft and Oldham ring, respectively.

Friction torque is not easy to define because expander constitutes from many parts. Ma et al. [195] recommended Eq. (6.52) for summing of friction in the expander and generator.

$$T_{fr} = f_{sg}\omega \quad (6.52)$$

f_{sg} is overall dynamic friction coefficient and need to be determined by fitting the experimental data. Regarding the electromechanical torque, Eq. (6.53) can be used [209]:

$$T_{em} = K_t I_a \quad (6.53)$$

I_a and K_t are current and torque constant, respectively. Generated armature current and rotation speed is correlated in Eq. (6.54), [209]:

$$K_e \omega = (R_{arm} + R_{load})I_a + L_a \frac{dI_a}{dt} \quad (6.54)$$

K_e is back electromotive force constant and its value is taken as 0.12 NmA^{-1} [195].

R_{arm} and R_{load} are resistances and L_a is armature inductance.

When inlet pressure and temperature are given as 450 kPa and 80 °C, the pressure and temperature gradient during one rotation is shown in Figure 6.14. These values are obtained by considering the processes as isentropic. These results will be used as initial parameters for full model. As this expander has four chambers, the figure shows them clearly. Intake chamber pressure remains same as mentioned but expansion chamber

reduces working fluid's pressure during expansion. Intake chamber and first expansion chambers are merged due to perfect mesh profile (PMP) modification of the scrolls. When scroll rotates, the volume of merged chamber increases. However, disconnection occurs at a disconnection angle as the merged chamber splits into a small chamber as intake chamber and larger chamber first expansion. Two chambers expand until orbiting angle reached to $\theta = 2\pi$. At the same time, discharge chamber volume reduces.

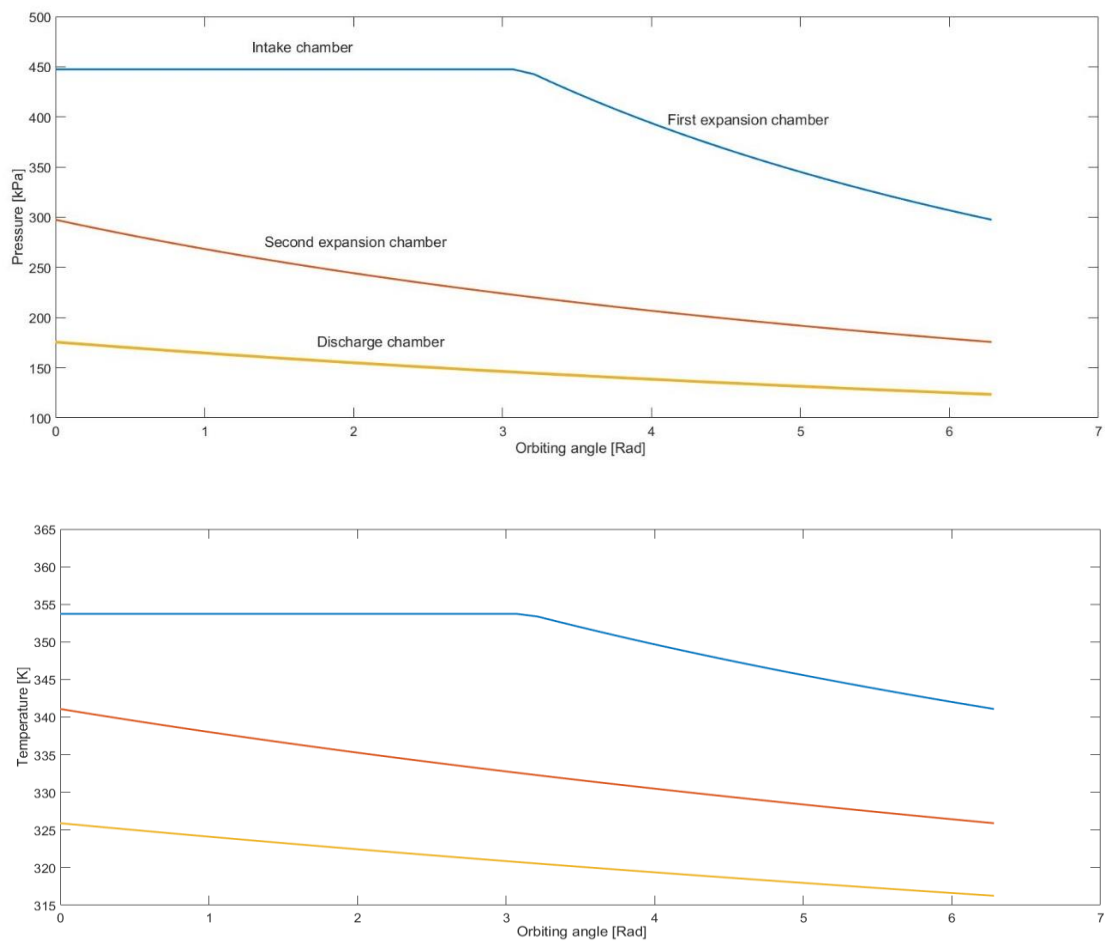


Figure 6.14 Pressure and temperature variation during one rotation (isentropic)

By using Eq. (6.47), tangential gas forces acting to each chamber can be calculated, Figure 6.15a shows total tangential forces for one rotation. Total tangential torque can be calculated by Eq. (6.50) and given in Figure 6.15b. This tangential torque value is used in torque balance equation to determine rotation speed of the expander. Once simulation features of the used expander have been conducted, experimental data analysis and validation process can be taken.

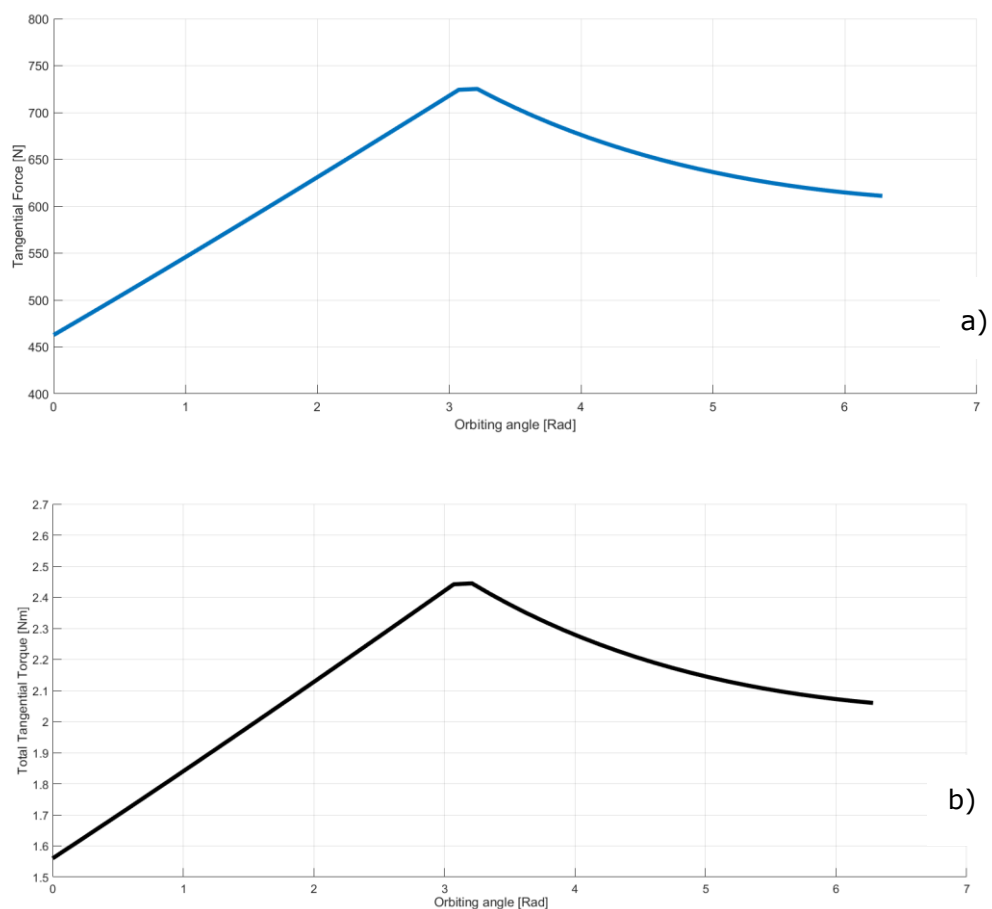


Figure 6.15 Total tangential force (a) and tangential torque (b) variation for one revolution

6.6 Experimental results and comparison with model

Experimental validation is based on leakage area calculations, rotation speed and power output validations. We have some number of data for tested ORC system. The recorded data are inlet pressure and temperature, outlet pressure and temperature, rotation speed and generator output voltage and current.

Figure 6.16 shows first set of experimental data in transient state. In this experiment, pump frequency has been increased gradually so inlet pressure is increased but temperatures and pressure outlet change slightly. By using experimental data clearance of the scroll gaps are determined first. Lately, the effect of pressure ratio on rotation speed and efficiencies will be discussed.

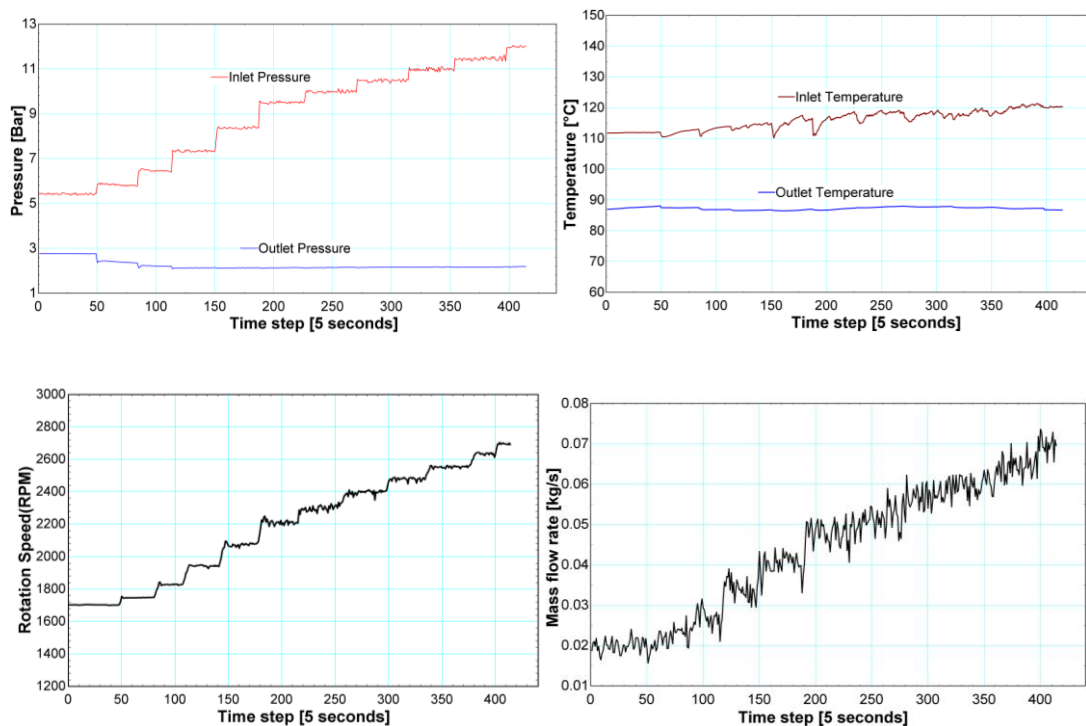


Figure 6.16 First set of experimental data

6.6.1 Scroll clearance

To eliminate empirically obtained leakage area, leakage area equations can be used but these equations require clearance values between the scrolls. As seen from Table 6.2, there is no agreement about the clearances among the studies. For the scroll compressors, the manufacturer gave some equations to calculate them, but we know that it is not possible to use same equations for expander applications because reverse operation makes possible to decrease sealing effectiveness that's why authors used higher clearance values for expanders. In order to decide real clearance value, experimental data have been used for solving the equations, but leakage area must be determined first. By using the geometrical parameters to be given in Table 6.1 the leakage areas have been calculated using gap as a variable.

Therefore, the clearance value needs to be determined experimentally to determine A_{leak} value for each operating condition. Thus, the flow chart of the method is given in Figure 6.17. As seen from the figure, experimentally measured values are needed for the calculation of $\dot{m}_{th}$ and $\dot{m}_{leak}$. Then, clearance value can be determined by taking A_{leak} from the scroll geometry calculations. The procedure is started with calculation of the theoretical mass flow rate ($\dot{m}_{th}$) by using obtained rotation speed, inlet pressure and temperature. Leakage flow rate ($\dot{m}_{leak}$) is found by subtracting the theoretical flow rate from the experimentally recorded total flow rate ($\dot{m}_{total}$). Once $\dot{m}_{leak}$ is known, using leakage area (A_{leak}) which can be found from scroll geometry as a function of the clearance, the empirical clearance equation is obtained.

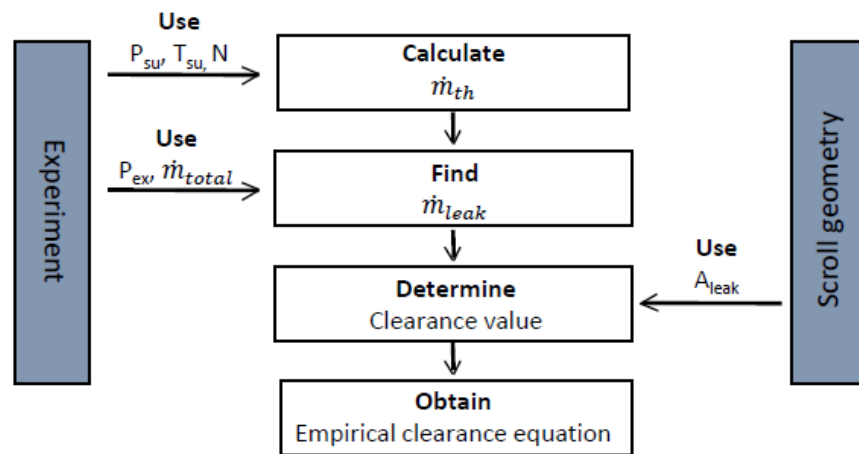


Figure 6.17 Method to obtain empirical clearance equation

It is obvious that larger clearances result in larger leakage areas. By using area equation with experimental mass flow rates, real clearance can be determined. Figure 6.18 shows clearance variation based on experimental mass flowrate. Three set of experiments' data have been used and effect of pressure ratio in clearance value is obtained. Clearance starts around $20\ \mu\text{m}$ and increases by the pressure ratio. Increment of the clearance by pressure ratio looks reasonable because scroll pairs consist of one fixed and one orbiting scroll, thus, it is possible to enlargement of the gaps by higher pressure ratios. The fitted curve is logarithmic function given in Eq. (6.55) which reaches $34\ \mu\text{m}$ for infinite larger pressure ratios. It is also reasonable to have a maximum value because scrolls are rigid materials, so gap increment is limited.

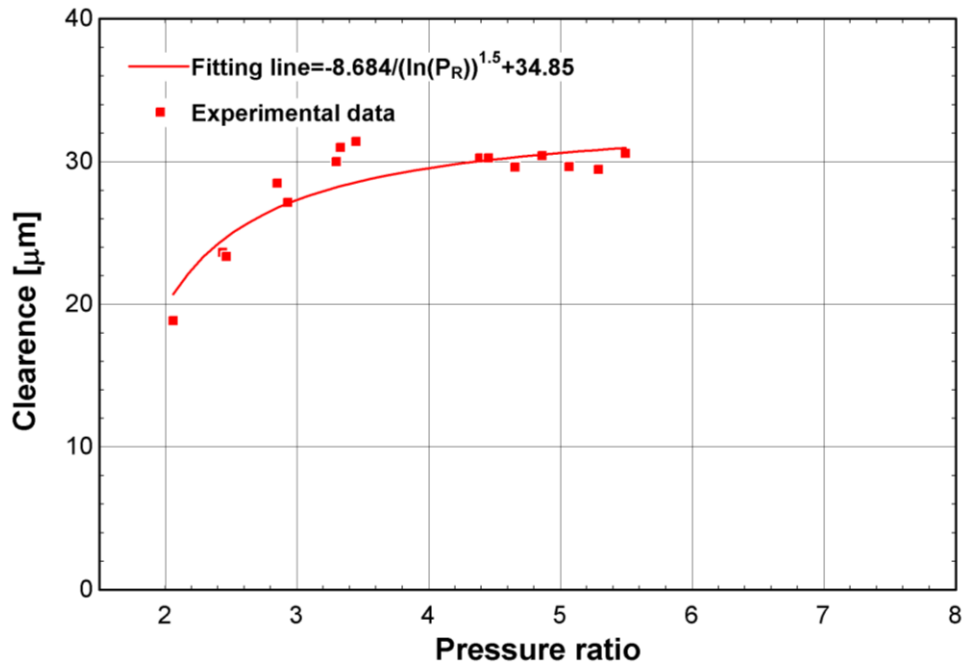


Figure 6.18 Scroll clearance variation by pressure ratio

$$Clearance = -8.684 / (\ln(P_r))^{1.5} + 34.85 \quad (6.55)$$

The figure shows that taking 30 μm of clearance can be useful for operations between 4 and 5.5 pressure ratios. However, many applications such as solar ORC and waste heat applications, the system operates at off design conditions. Therefore, taking a constant clearance value causes some deviations for predicting leakage flow calculations.

By using experiment data in Figure 6.16 and computing calculation procedure given above, Figure 6.19 has been plotted to compare measured and calculated mass flow rates. Since pressure is increased during the experiment, the system cannot reach steady operation but relation between calculated and measured value is still have a compatibility. The obtained deviation is less than 10%.

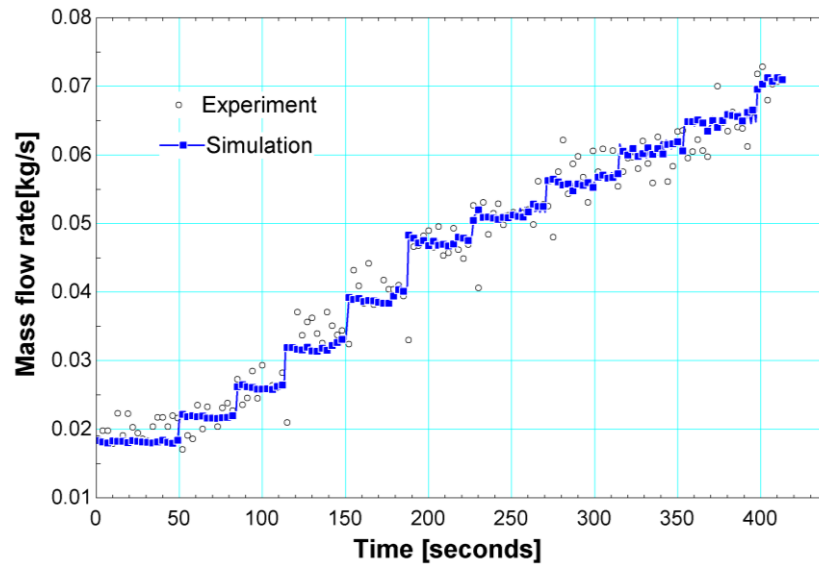


Figure 6.19 Measured and calculated flow rates

6.6.2 Volumetric efficiency

Volumetric efficiency is calculated by ratio of the theoretical flow rate and total flow rate which includes internal leakage, inside the expander. It is an important parameter because volumetric efficiency is one of the indicators of the leakages and it is known that leakage flow is also responsible of the low performance of the expander. Since leakage flow rate effected by rotation speed, Figure 6.20 is given. Rotation speed increases by the pressure ratio as expected but it must be noted that rotation speed also depends on other factors like frictions and electrical load, yet it gives information that rotation speed increases by the pressure ratio.

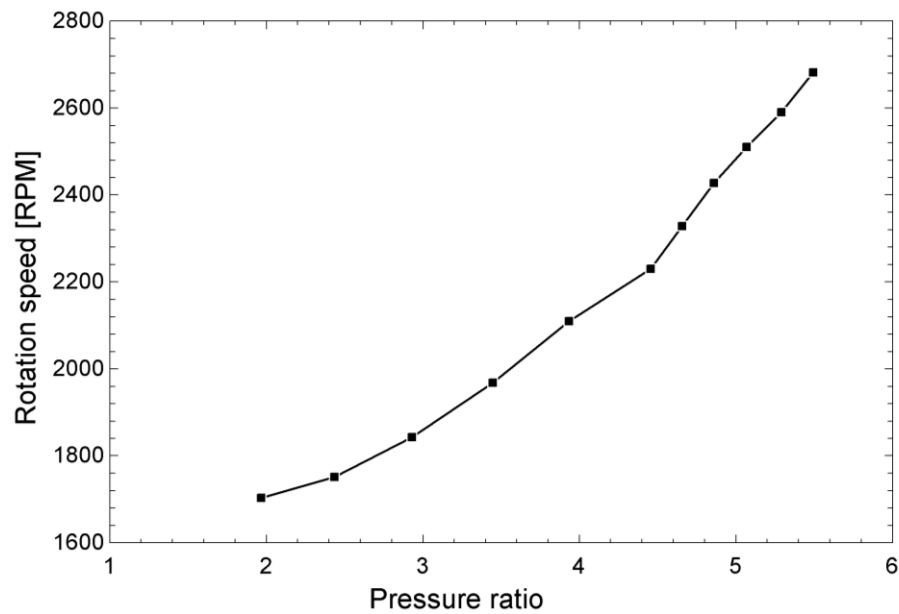


Figure 6.20 Experimentally obtained effect of pressure ratio on rotation speed

Figure 6.21 shows volumetric efficiency variation by pressure ratio. In the figure, both experimentally obtained and calculated curves are given to eliminate measured mass flow rate discrepancies. Normally volumetric efficiency is expected to be obtained an increasing trend with pressure ratio because higher rotation speed generally results better volumetric efficiency. However, efficiency decreases until pressure ratio of 3.5 then starts to increase.

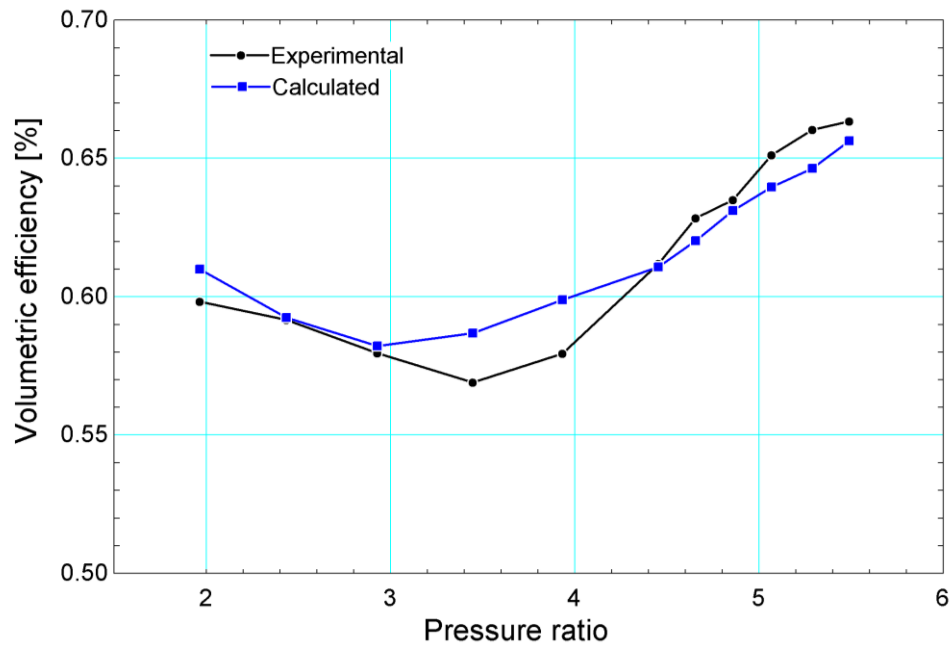


Figure 6.21 Volumetric efficiency variation with pressure ratio

In order to explain this phenomena, Figure 6.22 is given below. It shows variation of the flow rates inside the expander such as theoretical and leakage flow rates by pressure ratio. Theoretical flow rate is increased by inlet pressure because it depends on inlet density, rotation speed and swept volume of the expander. However, leakage flow rate is depended in inlet condition and also leakage area. Leakage area is not constant during operation because clearance between scrolls changes by the pressure ratio as explained with Figure 6.18. Since expander output pressure almost constant during this experiment, pressure ratio increases with inlet pressure. Thus, theoretical mass flow rate increases by pressure ratio in Figure 6.22. Leakage flow also increases but not same trend with it. Leakage flow rate increases faster than theoretical flow rate until pressure ratio of 3.5 because leakage area is increased rapidly with the pressure ratio so volumetric efficiency reduces. However, leakage area increment slows after

pressure ratio of 3.5 but theoretical flow rate continues to increment while inlet pressure increases.

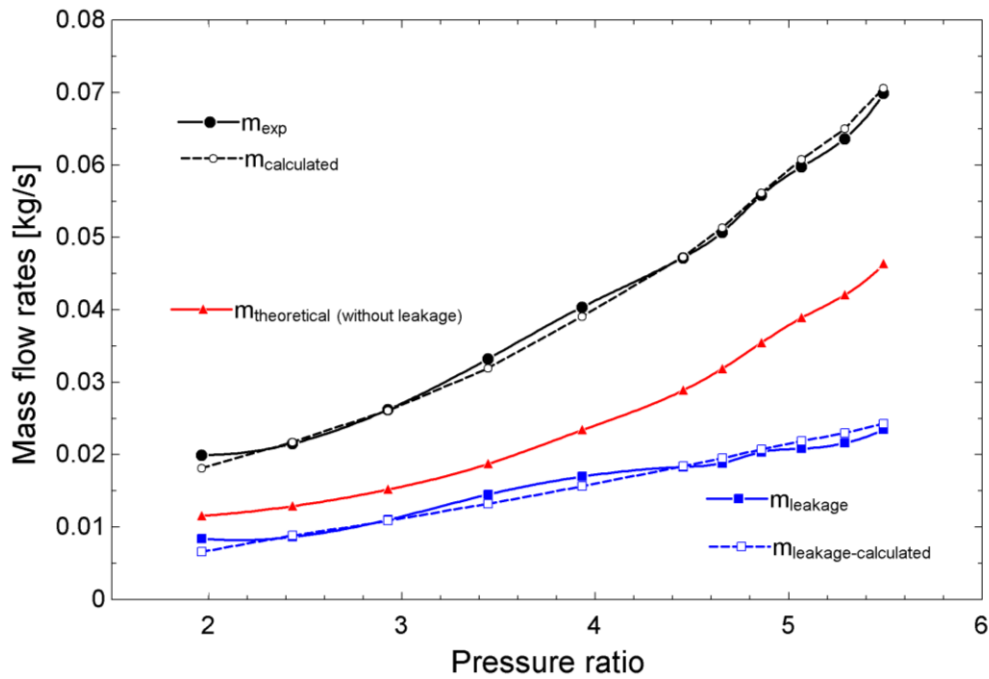


Figure 6.22 Theoretical, leakage and measured total flow rates and calculated leakage and total flow rates

6.6.3 Rotation speed and power output

Rotation speed is one of the important parameters as it affects mass flow rate and obtained currency in the generator and it has also indirect effects on system performance. To determine rotation speed of the expander, driving gas torque, electromechanical torque and friction torques should be considered because torque balance determines the rotation speed. Torque balance equation is given in Eq. (6.51) for dynamic systems. However, we can analyse the system in steady state operation. Thus, Eq. (6.51) is deduced as Eq. (6.56) by eliminating the inertia and acceleration effects of the components.

$$T_t - T_{fr} - T_{em} = 0 \quad (6.56)$$

To calculate driving torque T_t , Eq. (6.47) and Eq. (6.50) can be used. Regarding electromechanical torque, Eq. (6.53-6.54) are given but friction torque is quite complicated to determine because expander has components and frictions happens between them but not only expander components also generator has inertial friction between its components. Thus, an overall dynamic friction coefficient (f_{sg}) is used to find friction torque which is given in Eq. (6.52). f_{sg} is determined by experiments and obtained as it depends on rotation speed of the expander so its accurate value will be determined by fitting the experimental data.

Before analysing the data, we must be sure about factors affecting on the rotation speed. Given equation presents electromechanical torque depends on torque constant and current, however, to make the study more reliable, other literatures have been reviewed and Eq. (6.57) is decided to use. Because electromechanical torque is also affected by the air gap flux per pole. The equation is taken from [210]:

$$T_{mech} = K_t \Phi_a I_a \quad (6.57)$$

As torque constant and back electromotive force constant is same K_e can be used as K_t . The flux value depends on some factors, but we used experimental analysis to decide its trend with armature current. As current and voltage is recorded for all experiments, only resistant is controlled according to produced voltage and current automatically. Thus, current and voltage trends are similar for all tests as seen in Figure 6.23.

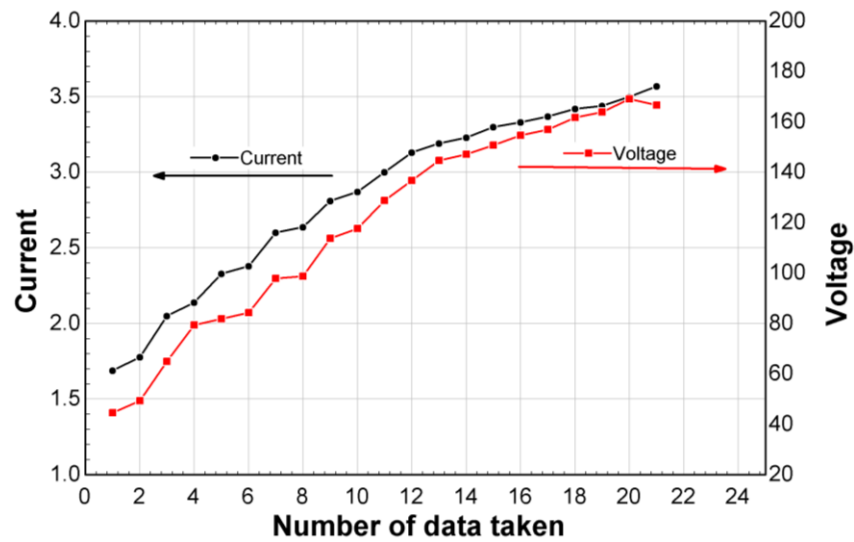


Figure 6.23 Voltage and current variation for combination of different tests

Based on 8 sets of experiments, Figure 6.24 is obtained. It shows that $K_e \Phi_d$ increases with current. Same trend has been observed from all experimental data. Legends on the figure indicates different condensing and evaporation temperatures. All experiments yield with same $K_e \Phi_d$ value variation. Thus, a fitted curve is used for $K_e \Phi_d$ in calculations.

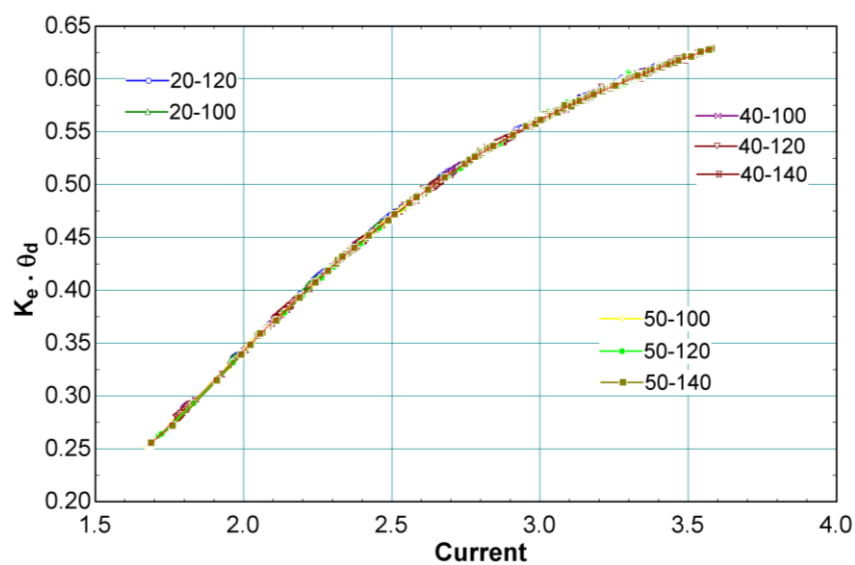


Figure 6.24 $K_e \Phi_d$ variation by current based on experiment data

After finding the generator specifications, rotation speed and power output can be found by using given equations. To calculate rotation speed and power output, steady state conditions are determined from various tests. Three sets of data are considered for comparison of simulation and measured data. Figure 6.25a shows used inlet pressures and Figure 6.25b shows measured rotation speeds.

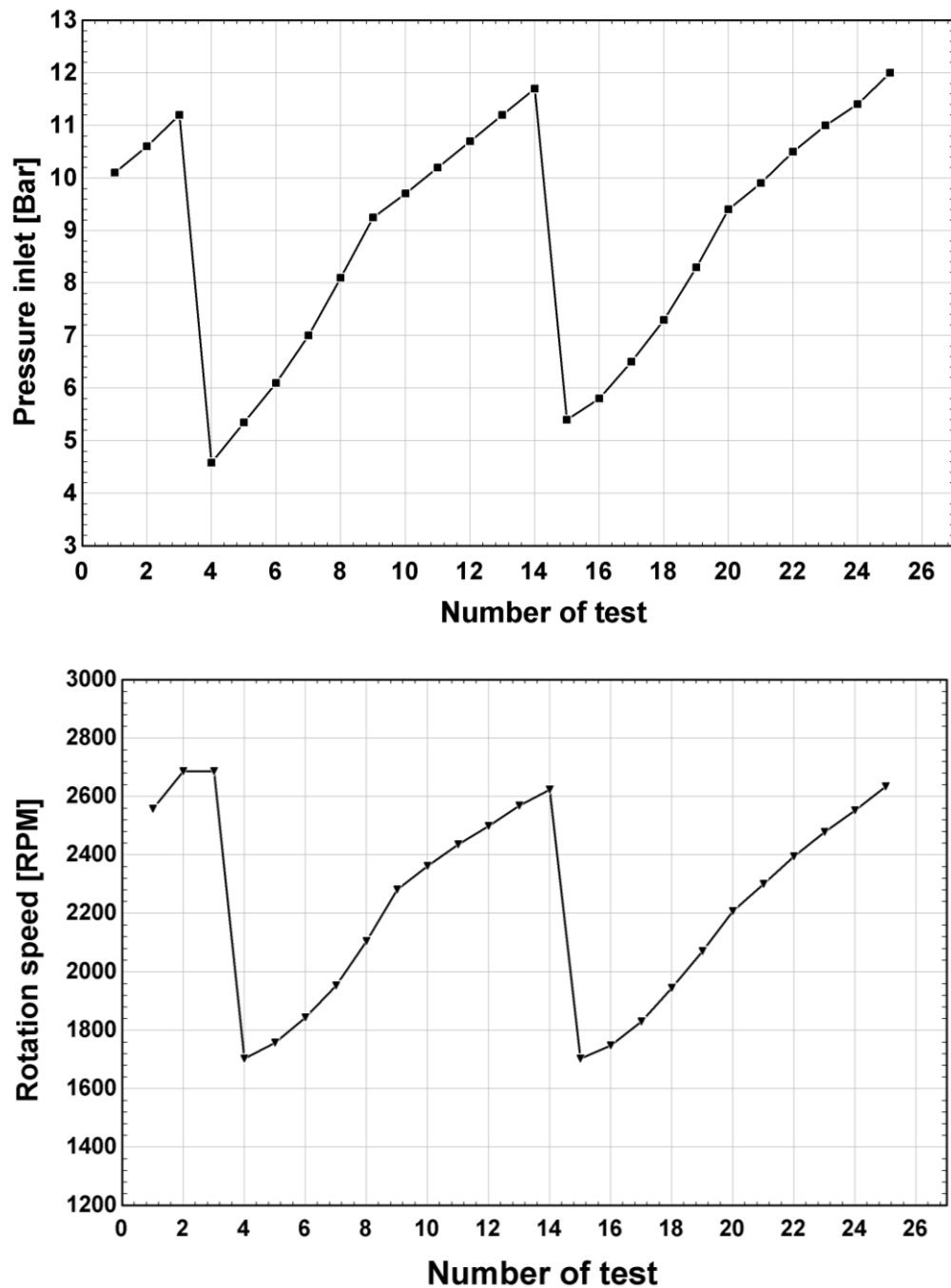


Figure 6.25 Steady state pressure inlets (a) and rotation speed (b) for three sets of data

The first step is determination of overall friction coefficient (f_{sg}) variation by rotation speed. f_{sg} is determined by experiments and obtained as it depends on rotation speed of the expander. Thus, its accurate value will be determined by fitting with the experimental data, as shown in Figure 6.26. According to fitted data, friction coefficient increases with the rotation speed. This trend has agreement with the literature that many authors observed similar trends.

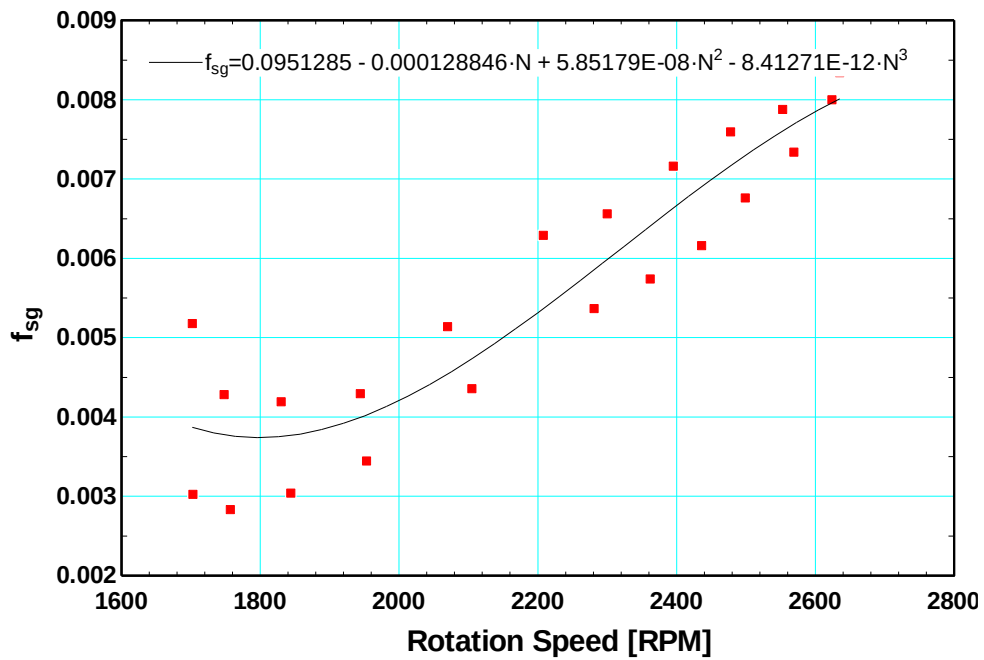


Figure 6.26 Overall dynamic friction coefficient variation by rotation speed

6.6.4 Prediction of the rotation speed and electricity output

After obtaining the empirical equations, simulation for predicting the rotation speed and electricity output can be conducted. In order to compare simulation results with the experimental data, the same operating parameters are selected. Figure 6.27 shows the followed procedure's flow chart. Procedure is based on assumption of current and rotation speed. The geometric model uses scroll geometry to indicate driving torque. Overall friction coefficient and generator data is calculated by using assumed current and rotation speed. As next step, the voltages are compared, one from the torque balance and the other from the assumed current. Then, the rotation speed is updated,

and voltage is recalculated using a new rotation speed. This procedure continues until the difference satisfies the condition for both rotation speed and current.

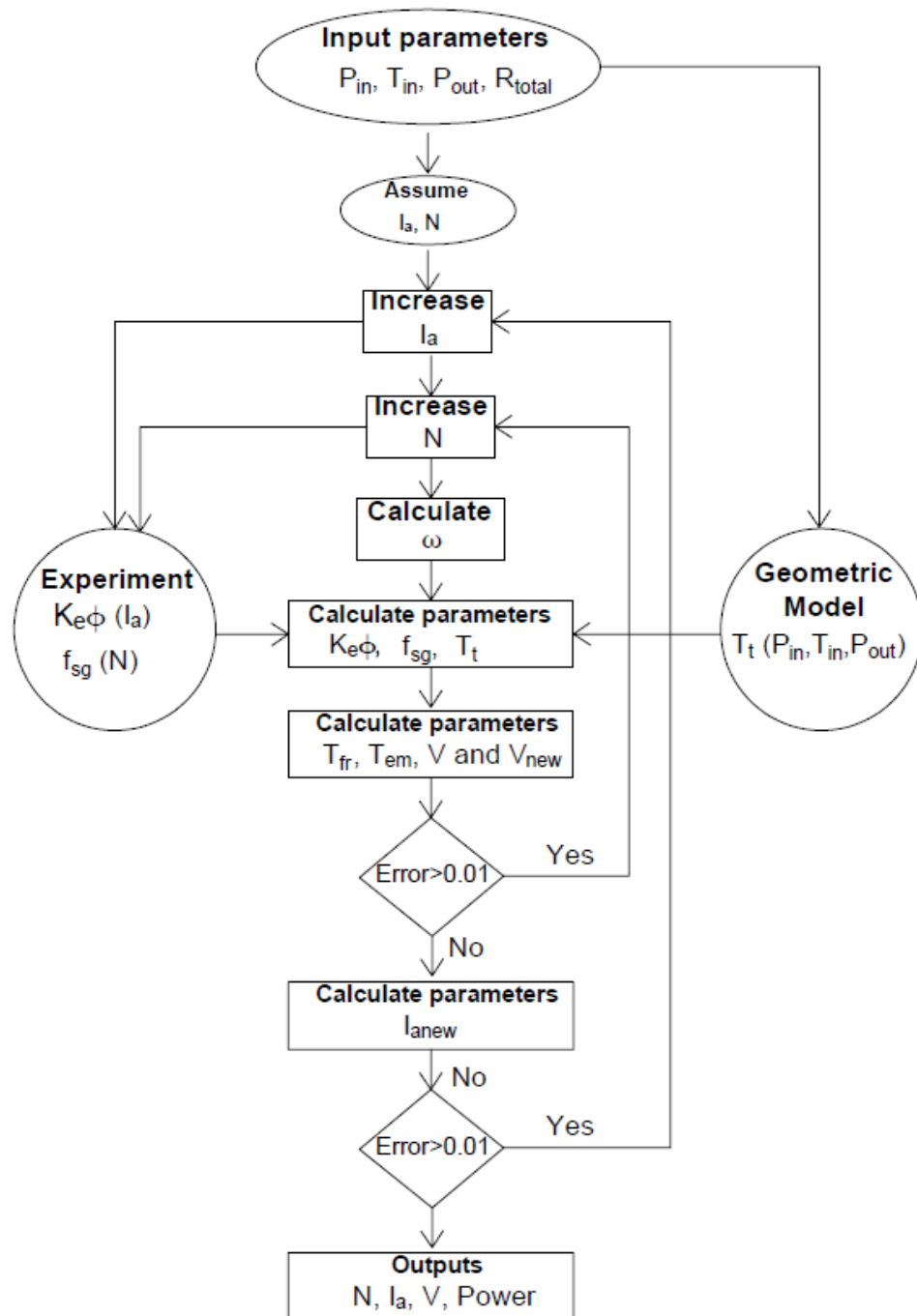


Figure 6.27 Calculation procedure of the rotation speed and power output.

Figure 6.28 shows the comparison of the simulation and experimental results of the rotation speed. Deviation is small for low inlet pressure operations but increases with higher pressure inlets. However, rotation speed also increases with pressure, thus, maximum deviation reaches 10% but this maximum is obtained only for one data. The difference can be reduced by decreasing the convergence criteria of conditions, though computational time increases in this case.

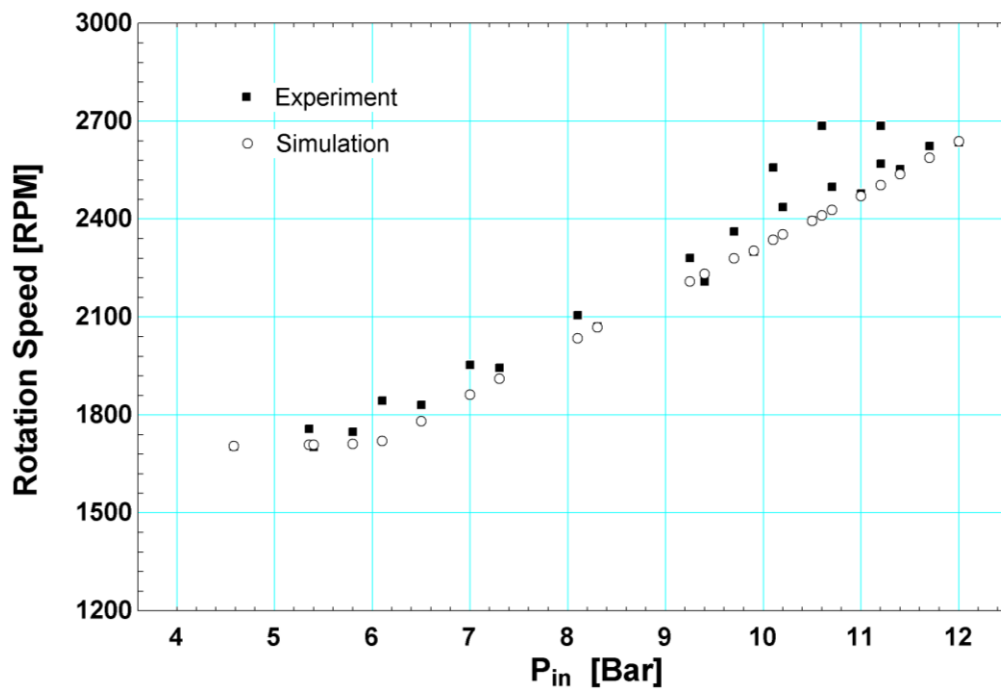


Figure 6.28 Comparison between experimental and simulated results of rotation speed

Figure 6.29 shows a comparison of simulation and experiment results of electricity output. Prediction of electricity output values are in good agreement with the test results, especially under low pressure operations. Accuracy of the model can be improved when high computational execution time is considered, however, the results prove that the given methodology can predict rotation speed of the expander and electricity output in an acceptable deviation. The proposed approach can predict rotational speed and generator output. These parameters make this analysis more applicable for system design and performance evaluation for various applications.

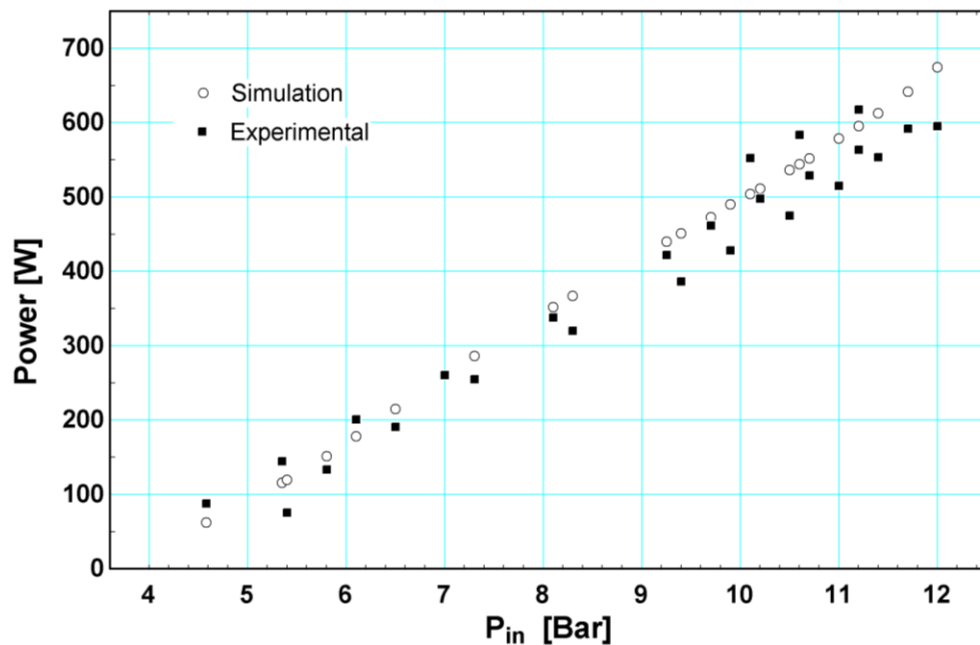


Figure 6.29 Comparison of the simulation and experimentally obtained electricity output

6.7 Summary

In this chapter, a methodology for modelling of the scroll expander equations has been given. The objective was building the detailed scroll geometry model. In order to carry out geometric model, scroll geometry features have been given in detail.

A simulation model was built, and simulation results were validated by experimental data. Moreover, the study was presented a leakage area calculation methodology. The calculation of the leakage area was examined properly by given equations, and clearance between scrolls was decided and experimentally validated. Some important conclusions are drawn as follows:

- Lumped leakage area in the semi empirical modelling can be found using scroll geometry equations. Mass flow rate was predicted with maximum 10% deviation under transient conditions.
- Clearance values between the scrolls are varied according to pressure ratios. These start around $20\ \mu\text{m}$ and increase with higher pressures. Experimentally obtained logarithmic function shows that $34\ \mu\text{m}$ is the maximum clearance when larger pressure ratios are applied.
- The overall dynamic friction coefficient of the expander generator coupling was studied, and its variation was determined experimentally using a semi empirical approach. It varied from $0.004\ \text{N m s}$ to $0.009\ \text{N m s}$.
- The model predicted rotation speed and electricity output of the expander generator coupling with good accuracy.

CHAPTER 7 Conclusions and future works

7.1 Conclusions

The work of this thesis was conducted by following a step by step investigative approach to fulfil its research aim and objectives by building a solar ORC system considering a user electricity demand profile. A methodology was given for matching the electricity output with the electricity demand profile using a heat storage unit. In the modelling stages, the components of the solar ORC unit were built to operate under off-design conditions. According to electricity demand profiles, the ORC electricity output was adapted by adjusting the circulation flow rate. A novel a-Si PV/T ORC system design was proposed, its operation metrics were presented, and electricity output was increased and compared to conventional small-scale power generation systems. This novel unit was also adapted to ORC-VCC units to improve performance. Finally, a comprehensive modelling of a scroll expander was carried out. Scroll geometry was formed using geometric parameters and important processes effecting performance such as leakage flow, heat transfer, rotation speed and electricity output trends were presented and discussed. The conclusions obtained from the current work are summarised as follow:

- A small community with twelve dwellings was considered in the case study. By using a 550 m² collector area and adapting an 88 m³ heat storage unit, it was possible to meet the electricity demand even at night by controlling the mass flow rate of the circulation water. Since the electricity requirement is lower in the daytime, the ORC evaporation temperature was forced to be lower by adjusting the water flow rate. Thus, consumed heat was reduced to maintain high temperature stored heat in the tank for evening peak demand operations.

The solar ORC system was investigated for a year's operation by considering meeting the electricity demand in transient state simulations. As the performance of the solar ORC depends on the availability of solar radiation, it is impossible to achieve 100% match with the demand profile. 24 hour electricity production was started in April because of solar radiation level reached good level from April and production reaching a maximum level in July.

- To overcome the low electricity conversion efficiency of solar ORC systems, a novel a-Si PV/T ORC system was proposed by using the same collector area and heat storage volume. This proposed system combines the advantages of the night time electricity production ability of a solar ORC with the higher electricity conversion efficiency of PV systems. To show the improvement, power outputs of a solar ORC, a-Si PV/T-ORC, poly-Si PV/T-ORC and a stand alone PV array were compared for typical summer and spring days. Using a-Si cells in the hybrid system achieved excellent results in summer applications, since it can tolerate excessive temperatures with a stable electrical efficiency. In April, poly-Si PV/T-ORC, a-Si PV/T-ORC and stand-alone PV systems' daily total output values were close, however, the solar ORC output was found to be lower. Nevertheless, the solar ORC system can produce electricity during the night-time, and this is the advantage of this system. As a PV/T-ORC can produce more electricity and have ability to produce day long electricity this makes this system promising alongside others. For summer applications, a-Si PV/T-ORC system has the highest electricity output.

- A solar ORC system was coupled with a VCC system with a common shaft to produce both electricity and cooling for hot summer days. It was found that to meet both electricity and cooling demand at the same time for a 60 m² office room, an 80 m² collector area and 9.4 m³ heat storage volume were required. When collectors were modified with a-Si PV cells, required collector area and heat storage volume can be reduced to 40 m² and 1.57 m³.
- Scroll expander modelling was examined in detail. The advantages of combining scroll geometry modelling and semi empirical modelling were presented. Further, expander leakage flows and electricity output calculations methodologies were presented using some empirical equations and rotation speed and electricity output were predicted.
- Leakage area variation was analysed under different pressure ratios using experimental data. Clearance starts around 20 μm and can increase with higher pressures up to 34 μm . Additionally, the coefficient of overall dynamic friction for expander generator coupling was studied, and its variation determined experimentally using a semi empirical approach. This varied from 0.004 N m s to 0.009 N m s. By the proposed methodology, flow rate, rotation speed and electricity output could be predicted accurately.

7.2 Recommendations for future research

As the PV/T-ORC system shows promising results over PVs and solar ORC systems, a feasibility study should be conducted. While a vacuum environment will increase the a-Si cell's lifetime, longer-term tests should be conducted to prove this.

Furthermore, the ORC-VCC system is a promising configuration as it can provide cooling and electricity to users at the same time. It therefore follows that the operation of the common shaft between the expander and compressor should be dimensioned and examined. A mechanic model should be developed to share shaft power between the compressor and generator according to the demand profile. Modification with a-Si cells has a great potential to reduce sizes, an economic analysis should be conducted to prove advantages. Finally, the scroll expander operation was presented and modelled using scroll geometry, however, to predict electricity output value there are still some parameters such as the friction coefficient that should be determined experimentally. To make this model more flexible to adapt to different operating conditions, different capacities and different brand of expanders, the friction coefficient should therefore be calculated analytically or numerically.

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