Modelling and Optimization of small scale solar assisted Organic Rankine Cycle

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Declaration

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Copyright Statement

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Abstract

In recent decades, energy demand has grown exponentially. Fossil fuels, which today meet the bulk of global energy demand, can be gradually replaced by renewable energies, thus limiting its dangerous consequences such as climate change, environmental pollution, depletion of natural resources, etc. Solar energy can play an important role in the satisfaction of energy demand, especially in a country heavily sun-ridden.

The current study focuses on the modelling and optimization of Organic Rankine Cycle (ORC) based on different organic fluids operating in a temperature range below 50-100 °C. The ORCs are a good choice to produce small-scale energy due to the lower temperature range of 50 to 99 ° C. They are therefore simple and inexpensive. Flat plate (FPC) or evacuated tube solar collectors (ETC) can also provide the desired energy.

In this work, two system configurations are analyzed. In configuration-I (C-I), the water from the collector outlet moves to the hot water storage tank (HWST) connected in series, while in configuration-II (C-II), HWST is not used. Therefore, the hot water from the solar collector outlet enters directly into the auxiliary heater (which will be lit if necessary, otherwise) and the water return from the heat exchanger will become the input of the solar collector.

Working fluids suitable for a solar-powered ORC at a temperature of 100 ° C or lower are selected using predefined criteria such as higher fluid densities, maximum cycle efficiency, safety and environmental data, a moderate temperature and inexpensive and uncomplicated equipment.

The R125 and R245ca were found to be good fluids due to the minimum collector area for the desired yield and maximum efficiency, respectively. For the R125, the minimum required collector area is estimated to be 50 m^2 for the ETC and 68.14 m^2 for the FPC. For these areas, the optimized size of the HWST is estimated at 1350 L.

System configurations are modelled and simulated in TRNSYS for the entire year, from January 1^{st} to December 31^{ST} , to investigate optimal collector tilt, the smallest collector area for maximum solar fraction, and solar collector thermal efficiency. monthly solar collector efficiency for both configurations. The results of the simulation showed that C-II gives a comparatively higher solar collector thermal efficiency and solar fraction. For both collectors, the maximum seasonal solar fraction is obtained at an inclination of approximately 14°.Papoutsis et al. presented similar trend for a solar adsorption cooler. Their results showed that the maximum of SF is obtained at a tilt of 14° for Athens, Greece.

C-II for ETC gives a thermal efficiency of solar collector comparatively higher than that of C-I and one observes the same trend for FPC. In addition, the thermal efficiency of the ETC at 50 m^2 is higher than that of the FPC at an area of 68.14 m^2.

Key Words: *Flat plate collector; Evacuated tube collector, Solar Fraction; Tilt Angle; Collector*

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Nomenclature

Abbreviations

C: Configuration

FPC: Flat Plate Collector

PTC: Parabolic trough collector

ETC: Evacuated Tube Collector

STPP: Solar Thermal Power Plant

SF: Solar Fraction

ORC: Organic Rankine Cycle

SORC: Solar Organic Rankine Cycle

HWST: Hot water storage tank

GWP: Global Warming Potential

ODP: Ozone Depletion Potential

PV: Photovoltaic

ASHRAE: American Society of Heating, Refrigerating, and Air-Conditioning Engineers

ALT: Atmospheric Life Time

BWR: Back-Work-Ratio

CFCs: Chlorofluorocarbons

DORC: Dual-loop Organic Rankine Cycle

GHG: Green House Gases

HTA: Heat Transfer Area

HTF: Heat Transfer Fluid

ORC: Organic Rankine Cycle

WF: Working Fluid

kW: Kilowatt

Subscripts

in	inlet
out	outlet
con	condenser
p	pump
r	ratio
t	turbine

Symbols

A_i	m^2	Surface area of the ith tank
A_c		Area of Evacuated or Flat plate type solar collector
a_1		First order coefficient
a_2	$kJ/hr.m^2.K$	Second order coefficient
a_o	$kJ/hr^2.m^2.K^2$	Optical efficiency
ε_{heat}		Efficiency of boiler
I_t	$kJ/hr.m^2$	Total radiation on tilted surface
G_t	$kJ/hr.m^2$	Incident global solar radiation
T_{amb}	K	Ambient Temperature
T_{gen}	K	Regeneration temperature
T_{env}	K	External environment temperature
T_i	K	ith node temperature
T_{in}	K	Collector inlet water temperature
$\eta \emptyset$		Latitude
$\eta \emptyset$		Efficiency of solar collector
δ		Declination
ε		Effectiveness
η	%	Rankine efficiency
C	J	Heat Capacity of Fluids
h	kJ/kg	Enthalpy
$\dot{m}$	kg/s	Mass flow rate
P	Bar	Pressure
$\dot{Q}$	J	Heat Energy
s	kJ/kg.K	Specific entropy
T	K	Temperature
U	W/m^2K	Heat Transfer Coefficient
$\dot{W}$	Watts	Work

CHAPTER 1: INTRODUCTION

1.1 Global energy trend

Fossil fuels fulfilled the energy demand throughout the entire modern era. Coal, oil and natural gas remained in use for ages. However, accelerated human development threatened the sustainability of these energy sources. The persistent and widespread reliance on fossil fuel resources so far led to their depletion. It also challenged environment by an increase in air pollution and global warming. This caused a search for renewable sources of energy that ensure a sustainable future for the humanity.

For production and industrial processes global economy revolves around energy. The efficient energy production and its use has a significant consequence on our society and environment as well. As per international energy agency (IEA) the current trend for energy consumption and efficiency increases energy requirement by 70% and 60% and emissions respectively by 2050 in comparison to year 2011. The global average temperature by associated emissions will increase by $6^{\circ}C$ by 2050, which might result in massive adverse impact i.e. climate change, energy security and unendurable future. [1]

The fossil fuels burning generates nearly 21300 million tons of toxic carbon dioxide (CO_2) every year. As per reports it is claimed that natural processes absorb only about half of this amount thus there is net increase of 1065 million tons of atmospheric CO_2 every year. [2]

Renewable energy sources, an alternative to conventional fossil fuels, available in various forms and harnessed through various means. There is no region in the world that lacks one form of renewable energy source or another, however small or limited these might be. Getting energy from renewable sources like solar, wind, hydro, biomass, geothermal and industrial waste heat is becoming attractive to conventional sources. Research into efficient, effective and energy harnessing techniques from renewable sources is trending worldwide.

In 2016, clean energy installations of 160 GW introduced globally. This shows an increase of 10% compared to year 2015, but their cost is reduced almost a quarter. Furthermore, solar power provided the biggest boost and consisted of half of all new capacity. The year of 2016 was the first year in history that introduced solar capacity that surpassed all other electricity-producing technologies. [3]

Many techniques generate electricity from low temperature thermal energy for the reduction of environmental pollutants and Organic Rankine Cycle (ORC) is amongst them.

The Figure 1 shows the yearly mean of solar radiation at earth's surface in different locations over the day. The solid grey line indicates the value of 5.75 x 10^3 Wh/day and the dashed grey line indicates 2.88 x 10^3 Wh/day [4].

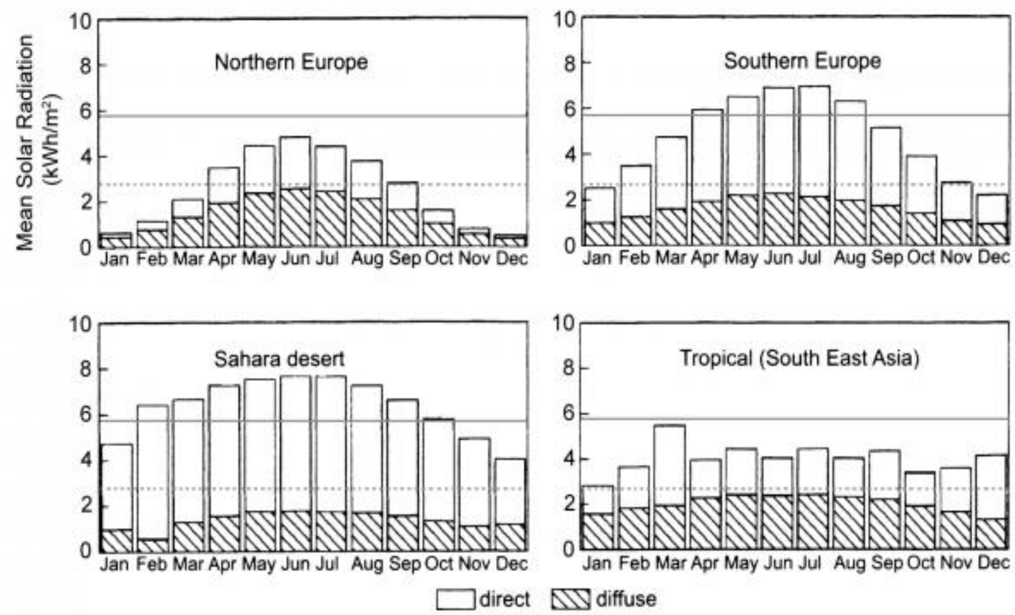


Figure 1 Global Solar Radiation at Earth's Surface

1.2 Organic Rankine Cycle Technology

The Rankine cycle technology, one of the incipient energy conversion techniques, harness energy from renewable sources efficiently and effectively. Compression of wet vapors in Carnot cycle became a problem so ORCs stepped in as an advancement. Rankine cycle comprises four main thermodynamic processes namely compression, evaporation, expansion and condensation. Four main components of the cycle namely the pump, boiler/evaporator, turbine/expander, and condenser perform these processes. The technology of ORCs is applicable in various applications such as geothermal, hydropower, wind, solar, biomass and industrial waste heat recovery etc.

High, medium and low-grade heat sources defined by their temperature facilitate such applications. Medium to high thermal sources provide high efficiency of an ideal RC system with temperatures above 370^{o}C. The ORC operates on the same principle as that of an ideal RC with the exception of using organic fluid as working fluid.

The effective utilization of low-grade source of energy, wasted otherwise, gives ORC a great advantage. The additional features of the organic Rankine technology such as dry expansion process, low adverse environmental impact, high thermal efficiency, simple expander design makes it a valuable technology. Figure 2 shows the technologies employed for electricity generation including ORC systems [5].

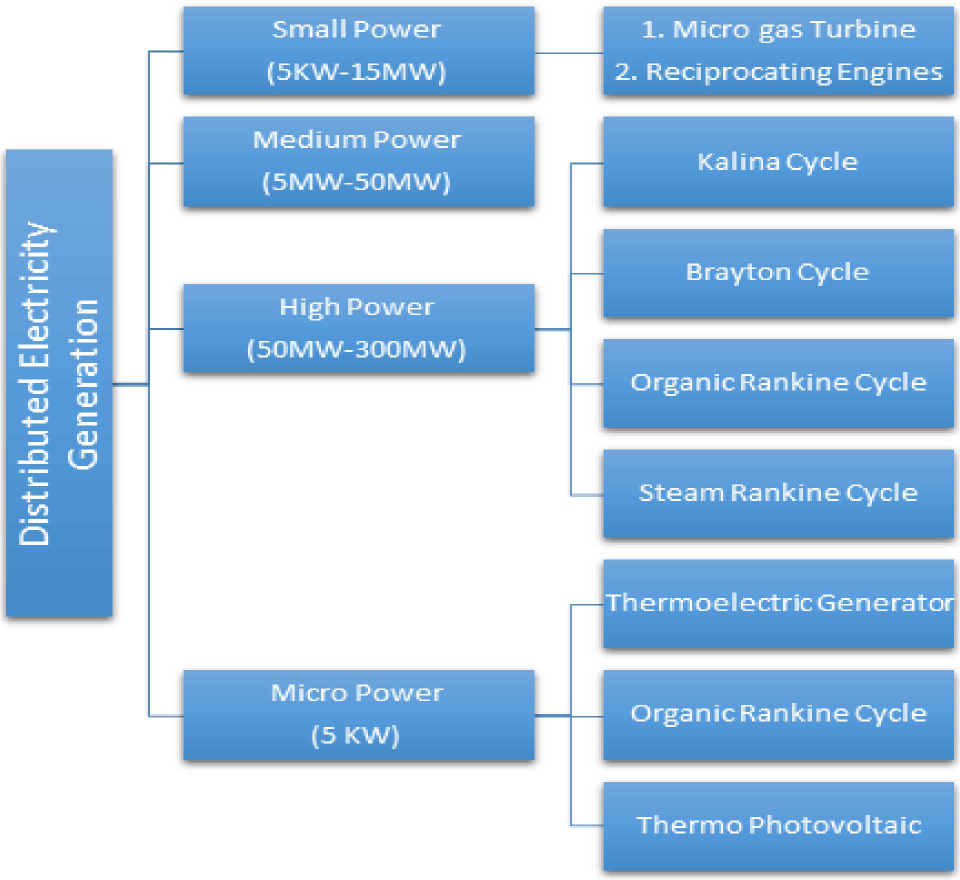


Figure 2 Electricity Generation Technologies

1.3 Motivation

For many years, the research trend shifted towards power generation from sustainable and clean resources. Industrialization, socio-economic and environmental factors mainly contribute to this trend.

Small-scale SORC are best suitable for off-grid remote areas. In comparison with the competing technology, the PV collectors, solar ORCs offer local manufacturing. They also show much more flexibility that allows the hot water production as a by-product for domestic or industrial use with a low appropriate investment cost.

For medium and large scale, the technology is much developed but only a few solutions are available for small size systems in the low kW power range. The evolution of small organic Rankine power systems is due to lack of efficient components for decades.

Pakistan faced severe energy crisis since the last two decades. The region shows sufficient solar radiation but lacks solar assisted organic Rankine cycle on small scale. This opens a doorway to research that determines the potential of solar energy to fulfil energy demand in remote areas on domestic level in Pakistan.

Results from this research work address some of the fundamental issues linked with the conventional way of energy generation from fossil fuel, moreover, present the benefits of renewable energy production.

1.4 Objectives

This research aims to:

i. Conduct a comprehensive literature review of the renewable energy resources and the technology used to harness.

ii. Develop and optimize a suitable model for ORC that works in low temperature range.

iii. Present the output of ORC system in terms of power generated and efficiency.

iv. Show the effects of variation in input parameters i.e. pressure, heat source temperature and refrigerant flow rates, for the model developed and on the solar ORC performance thus determining optimum performance conditions.

1.5 Scope

The current research presents the power production potential available in renewable energy sources with low to medium temperature range.

Factors which influence the selection of working fluid for the ORC technology are discussed in detail and selection of appropriate refrigerants is made.

In the standard formulation of this research work, the law of conservation of energy has been carefully examined.

The cost analysis and evaluation of the materials used for the current study were not taken into account.

CHAPTER 2: LITERATURE REVIEW

2.1 History

Organic Rankine cycle (ORC) works almost same as steam Rankine cycle (SRC) because all the important components it involves are of the SRC like heat exchanger, condenser, expander and pump. The main difference lies with the working fluid as an organic compound (i.e. siloxanes, hydrocarbons or refrigerants) replaces water. The ORC technology and SRC has approximately same evolution history. The first evidence of utilizing ether just as the working fluid in engine dates back in 1826. To modify ORC technology for power production the first half of the 20th century made great efforts by consuming the solar and geothermal energies. The oil crisis and environmental problems were faced in era of 70s and 80s due to which ORC systems developed at a faster rate in several aspects i.e. substituting organic fluids, innovative applications and zeotropic mixtures i.e. the use of ORC in vehicles etc. Until today such trend continued rapidly.

Israel based solar engineers, Harry Zvi Tabor and Lucien Bronicki (1961) presented one of the earliest use of ORC technology successfully for power generating system from low temperature heat source in [6].

D. Manolakos (2005) presented the design framework for low temperature solar ORC system for the sake of reverse osmosis (RO) desalination. This research discussed the low temperature thermal sources for fresh water production as by product [7].

Joan Carles Bruno (2008) optimized and modelled solar SORC engines for reverse osmosis (RO) desalination by utilizing solar thermal collectors available. For rural areas without or limited access to electricity grid, the proposed systems are potentially useful. [8].

Nishith B. Desai (2009) modified basic ORC to improve its thermal efficiency by using both regeneration and turbine bleeding. This paper accounts for 16 distinct organic fluids as working medium for the simple and modified ORCs. Appropriate methodology for the optimization and integration of an ORC as a cogeneration process is proposed. The selection of cycle layout for suitable integration with the background process relies upon heat rejection contour of the background process [9].

Agustín M. Delgado (2010) extended the theoretical study made by the authors of previous works to the case where stationary solar collectors provide the thermal energy needed by a solar ORC.

As working fluids of the ORC, twelve candidates are considered. It takes into account four distinct models of solar collectors which are stationary [10].

Sylvain Quoilin (2011) presented a thermodynamic model of an ORC with lost heat recovery. He compared not only thermodynamics, but also the thermo-economic efficiency of many working fluids for medium to low temperature range [11]. Ya-Ling He (2012) presented an integrated model for the solar thermal energy system typical of the parabolic trough with ORC. The transient energy simulation software TRNSYS has built the simulation model. The model examined the influence of several design and operating parameters on sensor field performance as well as the entire system [12].

Enrico Bocci (2012) conducted a research on the simulation of energy production of a solar thermal energy treatment circuit utilizing R245fa, a refrigerant, as working fluid. The dynamic model calculated an efficiency of almost 5%, even with leftover heat at a temperature of approximately 100 ° C. Other studies included the study of different sizes of solar collector installations and storage tanks. storage in a one-year simulation and a contrast of the results of the simulation with the experimental data [13].

B. Twomey (2013) calibrated a spiral-extension model and integrated it into a larger dynamic model of a solar thermal cogeneration system using vacuum tube sensors suitable for a larger housing or a small one commercial establishment [14].

Sylvain Quoilin (2013) clarified the latest state of ORC technology, focusing on the particularities and temperature levels of each application. The document focused on the major manufacturers, their field of activity, the important technological features of their solutions for ORC and their power production range. Analogy with the conventional steam cycle has identified the more appropriate ORC cycles for low to medium temperature applications i.e. low temperature heat recovery [15].

Suresh Baral and Kyung Chun Kim (2014) thermodynamically modelled a 1 kW solar ORC system with 15 distinct organic working fluids selected for medium to low temperature heat resources, depending on its critical temperature of 150 ° C and 240 ° C. The research consisted of comparing various parameters such as energy, heat, the solar energy cycle efficiency as well as the pressure ratio, the heat input, the mass flow, the flow rate volume at the inlet of the turbine and the area required for collecting hot water and collecting hot water [16].

Francesco Calise (2015) has designed a new solar power plant based on both ORC and CPVT technologies. The main idea was to use a high temperature CPVT system, which simultaneously produces electricity and heat to operate a deep ORC system, thus generating more electrical energy. The TRNSYS environment accurately modelled and simulated the new system, which has never been analysed from a numerical and experimental point of view [17].

Monica Borunda (2015) presented a research on a small CSP plant integrated with an ORC with a new layout in which a useful energy directly feeds the power supply and charges the thermal storage. The analysis of this new configuration used a case study with cogeneration applied to a medium temperature textile industrial process. The results revealed that this layout reduced the size of the heat storage dump. TRNSYS simulated the performance of the solar power plant to emulate the actual operating conditions [18].

James Freeman (2015) presented a techno-economic model to study the cost and potential performance of a nationally-based CSHP-ORC system with a positive displacement expander to be used in the United Kingdom. The results of the initial simulations, based on simple component efficiency data, operational control regimes and load profiles, show the sensitivity of the system's electrical output to the temperatures, flow rates and working pressures of the sub. - ORC system, as well as design and maintenance and working of the solar collector network [19].

Kyoung Hoon Kim and Chul Ho Han (2015) reviewed the most recent developments in solar collector systems and solar control energy recovery systems. The two central subsystems of a solar energy application include the solar collector and the thermal energy storage subsystem. They discussed both types of solar concentrators and concentrating types. The thermal storage tank of a solar system improves the use of trapped solar energy by giving thermal capacity to reduce load imbalance and solar availability. Researchers have worked hard to develop a better solar collector system. ORC is a reliable technique to exploit low-grade heat into useful work or into electricity. Theoretical and experimental studies show the potential for the production of mechanical energy or electricity by a system combining a solar thermal collector and an ORC [20].

R. Loni (2016) presented the heat loss of a proposed open-tube rectangular closed-tube solar receiver in copper and a numerical model. It used thermal oil as a solar working fluid and an 84% reflector collector and a 1.8 m diameter for numerical modelling. This cavity receiver associated with an ORC uses the fluid R141b as working fluid, which heats under the condition of saturated vapor [21].

Kiyarash Rahbar (2016) presented an extensive review of the CRO, accounting for cycle layouts, choice of working fluids and relaxation machines. Most ORC studies focus on the selection of an appropriate working fluid for a number of applications or the thermodynamic modelling and optimization of ORC performance metrics. Very few studies have focused on expander design, modelling, and experimental testing. Experimental research available on ORC expanders uses volumetric expanders that adapt and modify machines from available compressors and not precisely designed based on the thermo-physical properties of the selected working fluid [1].

Emily Spayde and Pedro J. Mago (2017) presented a model to study the performance of a solar regenerative organic Rankine R-ORC using five dry organic fluids: R227ea, RC318, R236ea, R218 and R236fa. Two different locations in the United States evaluated the system. The meteorological data for each location determined the available solar collector heat that R-ORC uses to generate energy. The first and second law criteria along with carbon dioxide emission (CDE) and the primary energy consumption (PEC) savings for both sites match the R-ORC performance results with a key ORC. Parametric analysis investigated the impact of turbine efficiency along with the intermediate pressure of the open feed organic fluid heater on system efficiency [22].

Maciej Z. Lukawski (2017) studied four different configurations of ORC plants utilizing heat sources ranged 100 - 230 ° C, 13 distinct working fluids and 2 different types of heat rejection systems. The optimization of the models made it possible to reach a maximum efficiency of use. Performance data and previous numerical studies from basic ORC facilities validated the results [23].

Saurabh Pathak and S.K. Shukla (2018) examined ORC performance for several heat sources, such as solar energy, wind and biomass etc. ORC is best suited for lost heat recovery applications. Previous literature has focused more on ORC performance for different operating parameters and has paid some attention to the performance of the condenser at several ambient temperatures. As the condensing temperature changes with the time of day and seasons, it is necessary to review the performance of the condenser for listed conditions [24].

Shuai Lia (2018) focused on dynamic performance of a small-scale solar ORC with thermal energy storage (TES) taking into account solar disturbances and presented a dynamic model of SORC. Fluctuation Suppression Ratio factor reflects the impact of the TES on the suppression of dynamic impact. The SORC dynamics contains resonance properties. Interaction between solar

disturbances and the thermal inertia of the system, energy superposition can cause dynamic resonance. The SORC dynamic performance analyses the crucial factor, including solar fluctuation (mean solar, amplitude, period), TES capacity, and evaporation temperature, while the RSF and efficiency of whole system serve as the signs to show stability and system performance [25].
Briefly, many researchers made tireless efforts to develop a better ORC system. Nowadays a technology used to convert low-grade heat into electricity employs ORC.

2.2 Power Cycle

A thermodynamic cycle includes an order of thermodynamic processes connected together so that system produces heat and work by varying state parameters such as temperature and pressure. Work done obeys the principle of conservation of energy in accordance with the first law of thermodynamics. The thermodynamic cycle comprises power cycles and heat/refrigeration cycles. This research is about power generation, so, this research deals with only power cycles.
The power cycles convert heat, an input to a system, into mechanical work or electrical power as output. Keeping in view the power generation, the cycles considered herein are the Carnot cycle, Brayton cycle, ideal (steam) Rankine cycle and the organic Rankine cycle.

2.2.1 Carnot Cycle

Sadi Carnot in 1824 devised the Carnot Cycle. It is an adequate power cycle which sets the caps for the thermal efficiency of any heat engine working between a heat source at temperature T_h and a sink at temperature T_l. Assuming that the reservoirs of heat are huge enough to both accept or deliver heat without changing their temperatures. It comprises the four processes given below [26]:

I. Reversible Isothermal Gas Expansion

The ideal gas in the system absorbs a quantity of heat q_{in} from a high temperature heat source T_h, expands and operates efficiently.

II. A reversible adiabatic gas expansion

This process includes an insulated system. The gas expands and work on the environment, causing the system to cool down to a lower temperature, T_l.

III. Reversible isothermal gas compression

In this process, the work is done by surrounding on gas at T_l. and cause heat loss q_{out}.

IV Reversible adiabatic gas compression

This process involves an insulated system. The surroundings keep on working on gas, which raises the temperature to T_h..

Figure 3 illustrates the graphical representation of these processes [27].

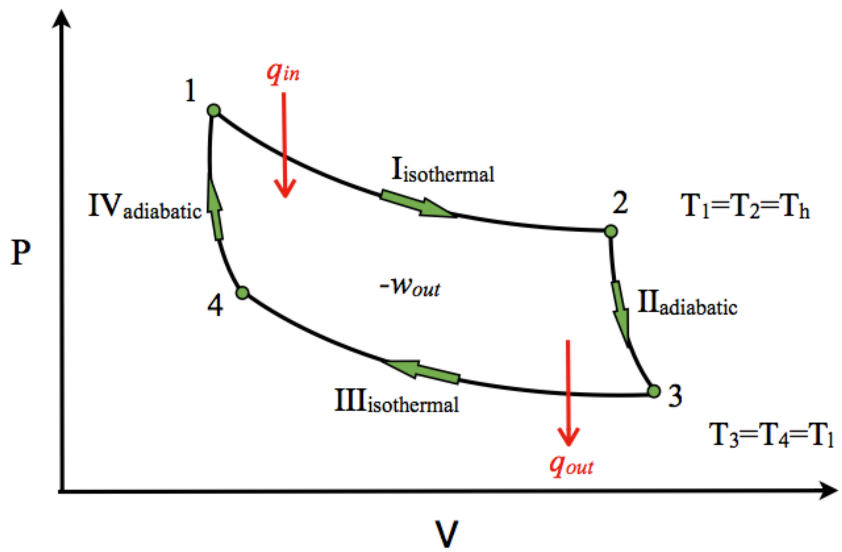


Figure 3 P-V Diagram of the Carnot Cycle

The Carnot cycle is impossible to achieve. Reasons may include:

I. The finite temperature difference causes the addition and rejection of heat between systems. In order to maintain isothermal heat transfer, cycle's maximum temperature remains much lower than the working fluid's critical temperature, which decreases the cycle's efficiency.

ii. During work generation, processes become

iii. Two-phase processes become difficult to manage during expansion and compression. Thus, actual cycles will always show lower efficiencies as compared to the ideal Carnot cycle [28], [29].

2.2.2 Brayton Cycle

The Brayton cycle has a similar draft to Rankine cycle but differs on various things like all through the process its working fluid remains in vapor state without phase change [30]. The thermal efficiency obtained from the Steam Rankine cycle compared to the Brayton cycle is higher at a temperature range of about 600^0C but it on depends upon the components and material used. Above 600^0C, the Brayton cycle offers a higher thermal efficiency with CO_2 recompression. In a dual cycle arrangement, the Brayton cycle may operate as a topping cycle serving as a source of heat for an ORC thus improving the overall thermal efficiency of the system. Using superheated carbon dioxide as its working fluid the Brayton cycle has the advantage of least toxicity and non-flammability [31].

In an ideal and closed Brayton cycle, the system consists of four processes: two isentropic processes (reversible adiabatic) alternating with two isobaric processes, illustrated in Figure 4. [32]:

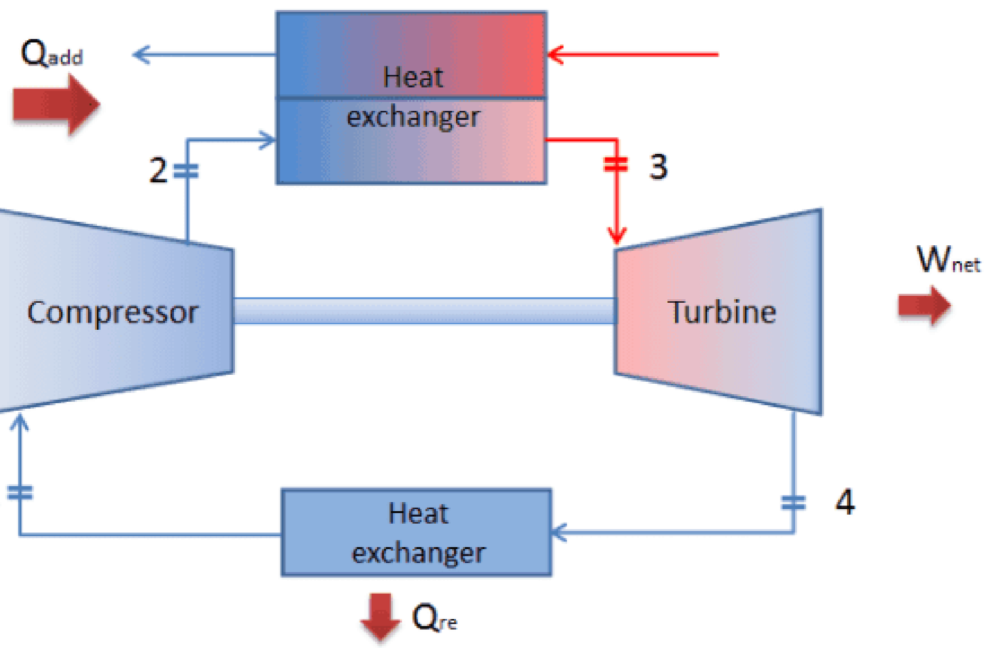


Figure 4 Brayton Cycle

1. Isentropic compression (compression in a compressor)

The working gas is adiabatically compressed from states 1 to 2 by the compressor. The surroundings work on working gas which causes an increase in its internal energy and pressure. whereas the entropy remains constant. The work needed for the compressor is given by

$$W_c = H_2 - H_1 \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

2. Addition of isobaric heat (in a heat exchanger)

In this process, heat transfer to the gas from an external source is done at constant pressure because incoming and outgoing flows are open in chamber. In an ideal open Brayton cycle, combustion chamber passes compressed air, where the fuel is burned and air or other means are heated ($2 \rightarrow 3$). The net added heat is given by

$$Q_{add} = H_3 - H_2 \ldots\ldots\ldots\ldots\ldots (2)$$

3. Isentropic expansion (expansion in a turbine)

In a turbine the compressed and heated gas expands adiabatically from state 3 to 4. The gas runs on the blades of the turbine and loses an amount of internal energy equal to the work that leaves the system. The entropy remains unchanged. The turbine's work done is given by:

$$W_T = H_4 - H_3 \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

4. Isobaric heat rejection

In this process, the heat of the gas in rejected at constant pressure. The temperature of working gas falls from point 4 to 1. The net heat rejected is given by:

$$Q_{re} = H_4 - H_1 \ldots\ldots\ldots\ldots\ldots (4)$$

2.2.3 Reverse Brayton Cycle

A Brayton cycle when driven in reverse direction is called the reverse Brayton cycle. It moves the heat of a colder body to a warmer body, rather than producing work. According to the second law of thermodynamics, heat cannot flow spontaneously from a cold system to a warm system without external intervention of the system. Flow of heat can be from a colder body to a warmer body, when external work is done. That is how refrigerators and heat pumps work. Electric motors drive them by requiring jobs from their environment to work. One of the feasible cycles is a reverse Brayton cycle like the typical Brayton cycle, but driven in reverse, thru a clean input. This cycle is also known as gas refrigeration cycle or a Bell Coleman cycle. This type of cycle is greatly used in jet airplanes for air conditioning systems using the engine's air compressors. Figure 5 shows the diagrams of the reverse Brayton cycle [33].

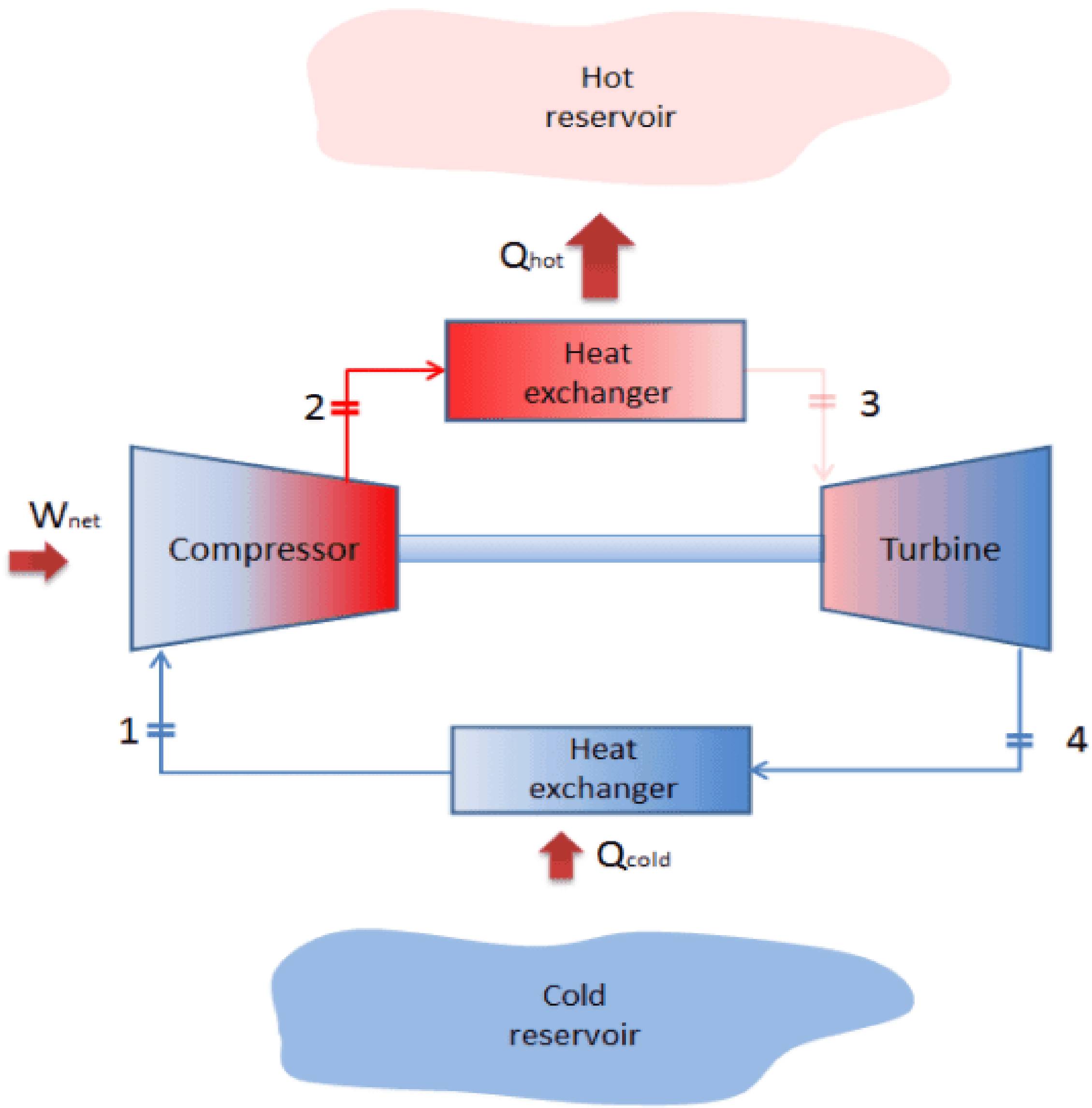


Figure 5 Reverse Brayton Cycle

2.2.4 Rankine Cycle

The power plants achieve production and generation of electrical power using steam as the working fluid. A study compared the Carnot cycle with Rankine cycle. In Rankine cycle heat addition and rejection occurs over a range of temperature. The efficiency of Carnot cycle remains higher than the Rankine cycle. The irreversibility and mechanical friction in the Rankine cycle makes it less efficient in comparison with the Carnot cycle. Highly efficient Carnot cycle is practically impossible. Hence, improvement made in the Rankine cycle for efficient performance [34].

The Rankine cycle is the thermodynamic cycle which is closed and that utilizes a heat source to obtain work as an output. Water serves as the working fluid and its efficiency is high when operating at a temperature greater than 600℃. Working fluid changes its phase between liquid and vapor phase for the Ideal Rankine cycle. The Rankine cycle works in the steps given below, as shown in Figure 6 [35]:

1. 1-2-3 isobaric heat transfer

Via the feed pump (1) high-pressure liquid enters the boiler which is heated to the saturation temperature (2). The additional addition of energy makes the evaporation of the liquid until it is completely transformed into saturated vapor (3).

2. 3-4 Isentropic Expansion

The vapors expand in the turbine, generating work, which is then converted into electricity. In real, the expansion is limited by the coolant's temperature and the erosion of turbine's blades by driving liquid into the vapor stream as the process advances into the two-phase region. The output steam qualities must be greater than 90%.

3. 4-5 Isobaric heat rejection

At the turbine (4) outlet vapor liquid mixture exists and is condensed at low pressure, generally in a surface condenser where cooling water is used. In properly designed and maintained condensers, the vapor pressure is quite below atmospheric pressure, reaching the saturation pressure of the working fluid at the temperature of the cooling water.

4. 5-1 Isentropic Compression

The condensate pressure is high in the feed pump. As the liquids have the low specific volume, the work of the pump is comparatively small and usually neglected in thermodynamic calculations.

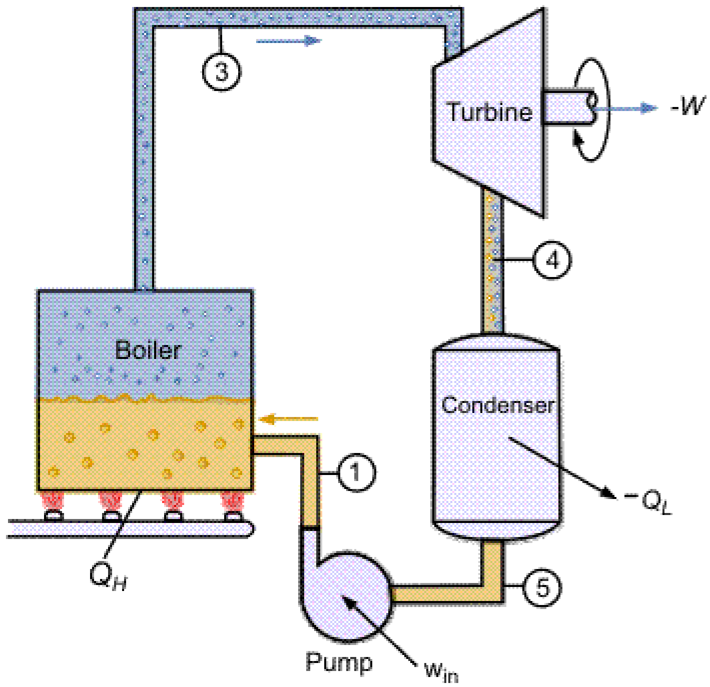


Figure 6 Simple Rankine cycle

In a steady state condition, the efficiency of the Rankine cycle depends upon:

a) The average rate of the heat addition and heat rejection from the source to the evaporator and from the condenser to the sink respectively.

b) The turbine and pump serve as critical components in improving the efficiency of the Rankine cycle.

c) Other factors such as feed water, heater, and controlled pressure in the steam condenser further enhance the thermal efficiency of the plant.

d) The introduction of a second stage steam turbine improved the efficiency of the cycle. However, it depends upon the exit temperature of steam from the low-pressure turbine.

The Rankine cycle with steam has limitations in its usage. The evaporating temperature increases to a certain limit beyond which the turbine blade characteristics are adversely affected and the fluid quality at the turbine exit decreases below the accepted 0.9 limit. A two-stage expansion with a heating stage between them eliminates the challenges of acceptable fluid quality. The two stages each consist of turbines (a high-pressure turbine with fluid from the boiler, and a low-pressure turbine with inlet fluid from the high-pressure turbine). With this modification, the overall efficiency of the Rankine cycle is improved. For a low temperature heat source, the steam Rankine cycle aborts compared to the ORC in terms of overall efficiency of the system [9].

2.3 **ORC the Power cycle**

The ORC technology has proven to be a reliable mean by which high thermal energy converts into power in an efficient manner. In the evaluation of two separate working fluids, the low-temperature exhaust from other Rankine cycle processes makes the use of ORCs as a bottoming cycle in a combined configuration highly desirable. The ORC is ideal for temperature range of 80^0C to 350^0C and small-scale power generation below 50 kW [36].

The ORC works on the same principle as that of Steam Rankine cycle, the only difference being the use of organic compounds as working fluid rather than steam. The organic working fluid has a low boiling point and high molecular mass. A major advantage of ORC over the steam or Brayton cycle is that it generates power from low thermal heat sources in a temperature even below 100^0C [37].

2.3.1 Reason

Organic Rankine Cycle uses an organic fluid instead of water. This organic fluid is typically a refrigerant, a hydrocarbon (hexane, butane, pentane etc.), a silicone oil, a perfluorocarbon, etc. It has a boiling point lower as compared to water, which makes it possible to recover heat at a temperature lower than in water. Rankine cycle with traditional steam. The thermo-physical properties differ from those of water in a number of ways, which has practical implications for designing Organic Rankine Cycle.

2.3.2 Mechanism

The configuration of the Organic Rankine cycle is less complicated to that of Rankine steam cycle: there is no water-steam tank connected to the boiler rather a single heat exchanger may be used to perform the three phases of evaporation: preheating, vaporization and lastly overheating. The variations of the layout of the cycle are also confined: the reheating and purging of the turbine are usually not adapted to the ORC cycle, rather a recuperator is installed which serves as liquid preheater between the output of the pump and the outlet the regulator, as shown in Figure 7 [37].

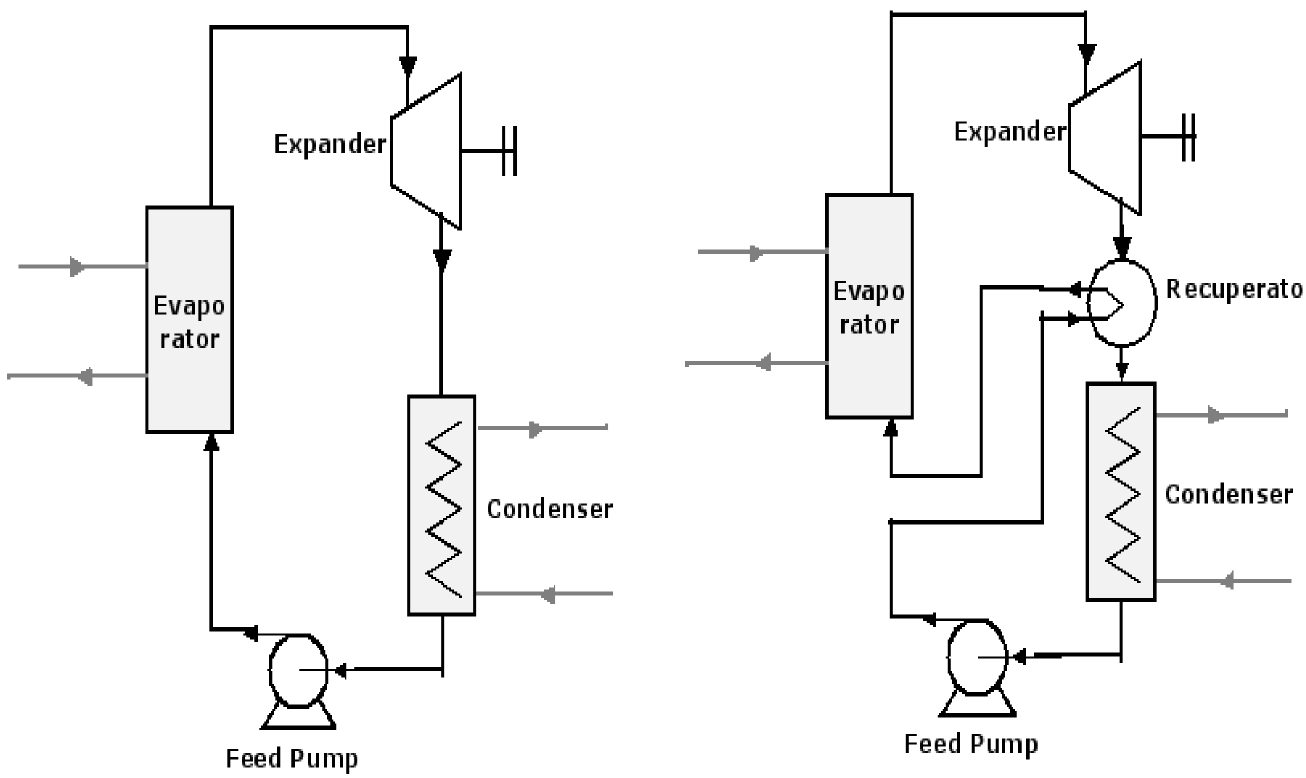


Figure 7 ORC cycle with (right) and without (left) recuperator

The cycle is quite similar to the conventional steam cycle: the organic working fluid undergoes pumping, vaporization, expansion and then condensation. The recuperator cycle uses leftover heat after the expansion takes place to preheat the liquid after pumping. This operation permits the reduction of the amount of heat required to vaporize the fluid for evaporation.

2.3.3 Comparison with Steam Rankine Cycle

The Rankine cycle, with steam as its working fluid, produces 65% of the world's electricity production [36]. Organic Rankine Cycle (ORC) is a special category in the Rankine cycle which utilizes a fluid which is not vapor. Several working fluids mostly utilized in ORC systems offer advantages as well as disadvantages with respect to the process of the steam Rankine cycle. A comparison between ORC and Steam RC has the following advantages and disadvantages [37], [9].

Table 1 Advantages and disadvantages of ORC cycles compared to SRCs

Advantages of ORCs	**Disadvantages of ORCs**
A low-boiling working fluid enables the economical conversion of thermal resources at low temperatures.	Working fluids increase the expense and mostly are less chemically stable as compared to steam.
A dry, high molecular weight working fluid eliminates costly overheating or reduces turbine resistance requirements.	At high temperatures, steam cycles provide higher efficiency than current ORC designs.
Highly dense steam fluids minimize the need of turbines and pipes.	ORC fluids show poor heat transfer properties, thus require a large heat exchanger.
Eliminates the need for de-aeration and demineralization processes.	Viscous ORC fluids result in higher pumping work and pipe dimensions.
The condensing pressure, above atmospheric pressure, reduces the need for seals to lessen air infiltration.	ORC fluids have specific and latent heats lower than water, which requires higher mass flow rates, which leads to greater needs for pipes, heat exchangers and pumps.
ORC fluids with a minute difference in density between vapor and liquid eliminate the steam drum or recirculation parts.	The experience and design expertise of ORC turbines are inferior to those of steam turbines.
Lower pressure ratios help to keep the turbine designs less complex	
The evaporation pressure, less than the steam cycle, eliminates the need for an on-site operator.	

2.3.4 Applications

ORCs are recently much uncommon than conventional steam-Rankine cycles in practical applications. Over the last decade, the decentralized generation has generated a growing interest in solving the energy problems faced by many modern power grids [38]. The number of commercial suppliers of ORC systems have increased which led to the improvement in technology and increased competition in the market. Now, solar energy has been captured and utilized in ORC commercial facilities.

Applications of electricity produced by ORC systems change considerably, with several systems being used to generate electricity. However, it is interesting to use other applications of ORC systems, such as direct working of trees in industrial plants [13], which slightly improves the overall efficiency and eliminates the need to use a generator. which can represent a significant cost on a small scale.

2.3.4.1 Waste Heat Recovery Application

Lost heat out is the unused heat produced during a combustion process which is then directly discharged into the environment. Energy concentrated industrial besides heat engines and mechanical equipment produce a large amount of waste heat. The exhaust fumes not only contain a high energy value, but also large amounts of pollutants. A study conducted in Canada's eight largest manufacturing sectors revealed a loss in energy input of up to 70% [38].

ORCs for waste heat recovery has also been used at low scales ranging 8 MW to just a hundred watts. It generally extracts low-heat heat from commercial and industrial processes that otherwise would have been wasted. They have an advantage of providing electricity directly to the industrial site, thus avoiding transportation and connectivity costs issues.

The most commonly used heat sources to which ORC WHRs are applied are [39].

- Biogas gasifier exhaust.
- Exhaust gas from waste incinerator
- Exhaust gas from internal combustion engine (ICE)
- Exhaust gas from gas turbine
- Exhaust from cement kiln.
- Steel thermal treatment systems
- Exhaust gas from glass oven

Research is under progress to design heat exchangers well adapted for recovering waste heat from tough heat sources like steel and electric arc furnaces. A pilot project was finished with the Life EU heat recovery program in energy intensive industries back in 2013. [40], [41].

Figure 9 and 10 show the heat transfer from the hot site to the power cycle in two possible ways [38]:

(1) Exhaust heat and working fluid streams are exchanged in the very same heat exchanger

(2) A thermal oil loop is added to avoid any threatening contact between the exhaust flow and working fluid which is flammable.

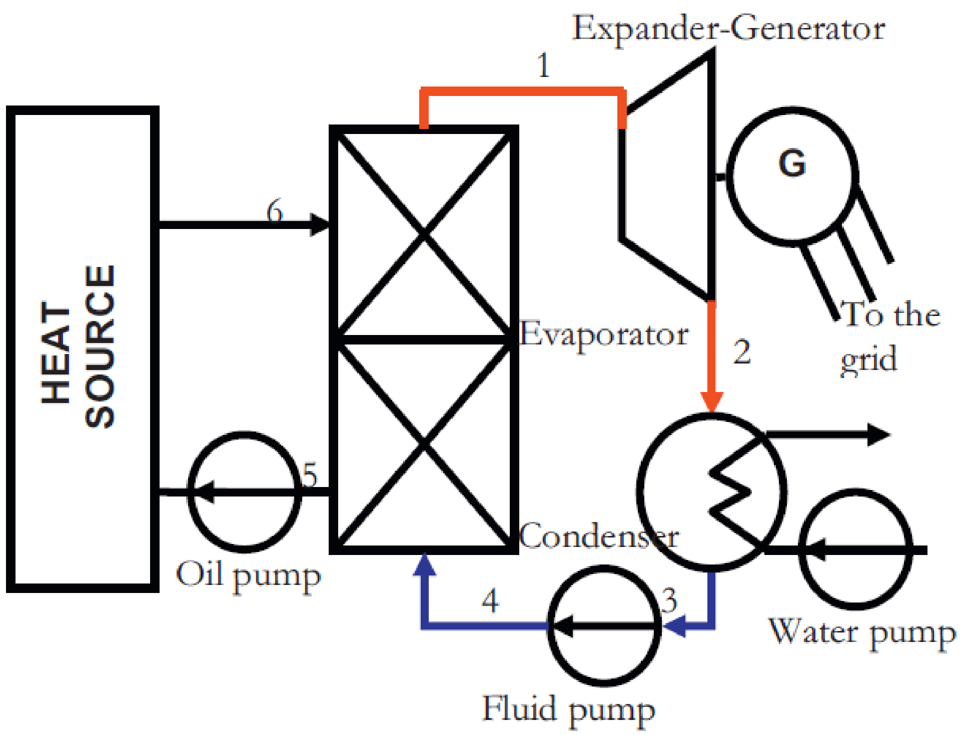


Figure 8 RC Configuration (1) in WHR process

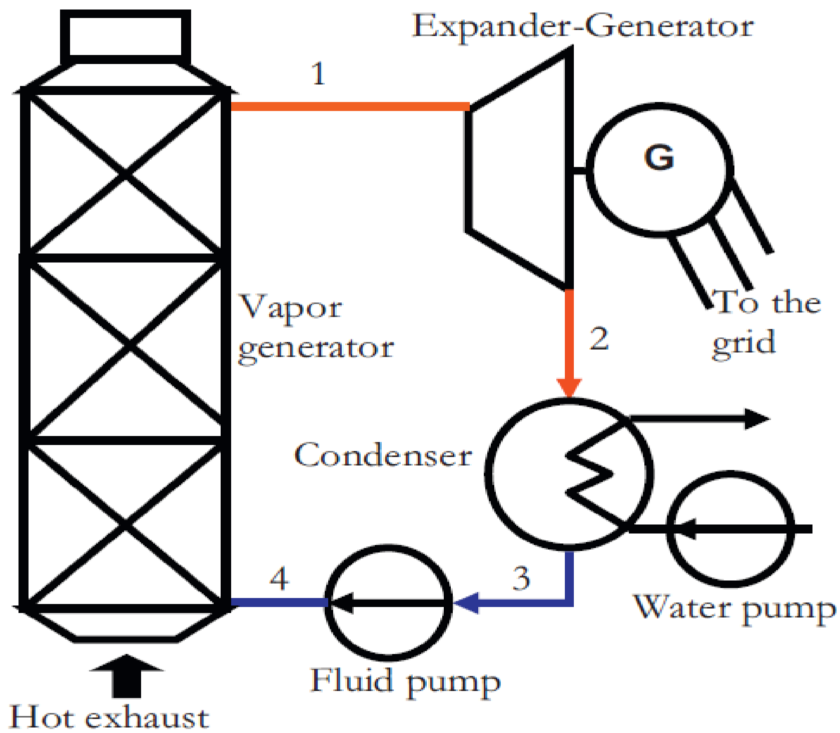


Figure 9 RC Configuration (2) in WHR Process

2.3.4.2 Geothermal Applications

Heat sources of geothermal are present in a range of temperatures ranging 300̊C to a few tens of degrees. The real lower technological limit for electricity production is almost 80°C: below this temperature the conversion efficiency becomes insignificant and geothermal power plants are less economical. To recover heat energy at a temperature which is acceptable, wells are usually drilled in the soil. Figure 10 illustrates this configuration [42]. The hot brine mixture is pumped at a lower temperature from the first and injected back to the second. Geological configuration helps in deciding whether the drilling may be several thousand meters deep, which requires working for many months continuously [43].

Low-temperature geothermal ORC plants have comparatively high accessory power consumption: pumps use 30 to 50% of total output power produced [44]. The important consumer is the brine pump which must circulate the brine with a high flow rate over long distances. The consumption of the pump pumping working fluid is also greater than that of the cycles with higher temperature because the ratio between the pump power utilization and turbine output power grows with a dropping evaporation temperature.

Geothermal heat sources at higher temperatures (>150°C) allow the combined production of heat and electricity: the condensing temperature is set at a temperature higher (i.e. 60 ° C), which allows the use of cooling water for district heating applications. In the case given below, the overall energy recovery efficiency is increased but lowering electrical efficiency. [45], [46].

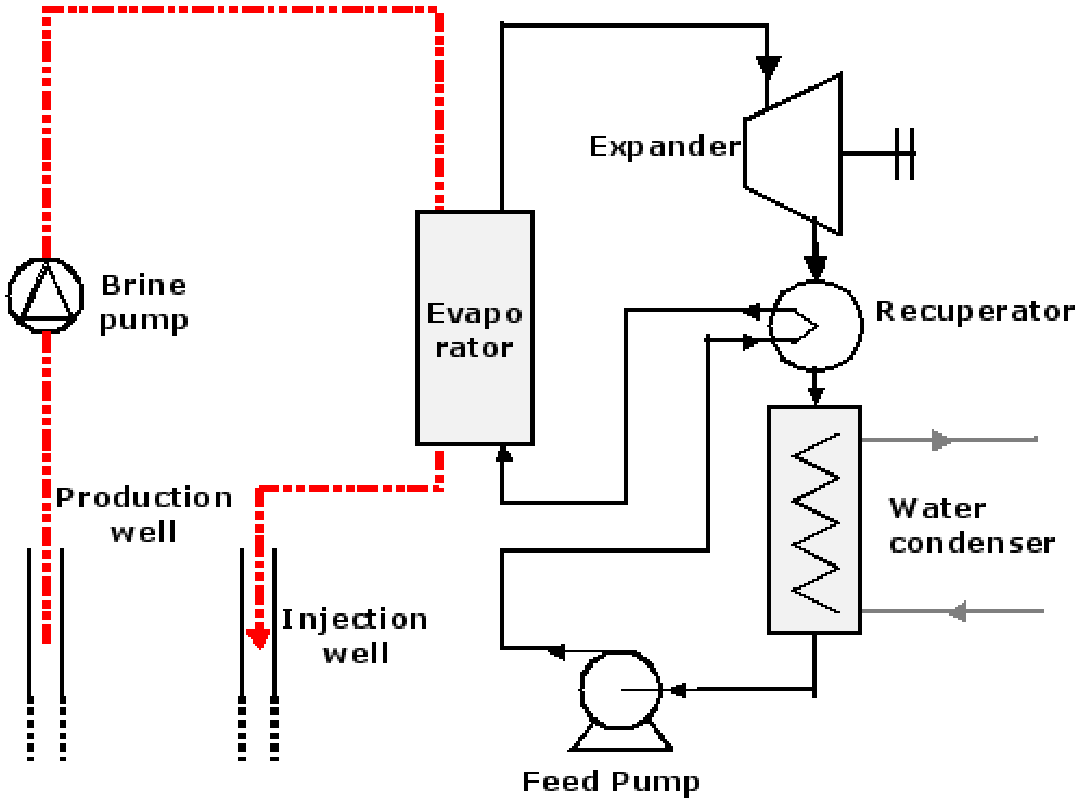


Figure 10 Working Principle of Geothermal ORC System

2.3.4.3 Biomass Applications

The fourth biggest source of energy in the world is biomass and accounts for nearly 10% of global demand for primary energy. In under developed countries, the contribution of biomass to the demand for primary energy at national level is higher, reaching 70 to 90% in some countries and is generally used in an unsustainable manner. This surplus source could be converted into electricity and heat if needed in cogeneration plants. ORC binary biomass technology is gaining much attention for its use in small-scale distributed power generation [38], [47].

Figure 11 shows a typical system consisting of a biomass boiler and an ORC module integrated by a thermal oil loop [38]. Biomass fuel burns in a process similar to basic steam boilers. As heat transfer fluid the thermal oil is used which has many advantages, including low boiler pressure, high inertia and least sensitivity to load fluctuations, safe and simple control and operation. In addition, the temperature of the hot side adopted (lower than 350 C) which ensures long life of the oil. The heat transported by the thermal oil is transferred to the ORC which then is converted into electricity.

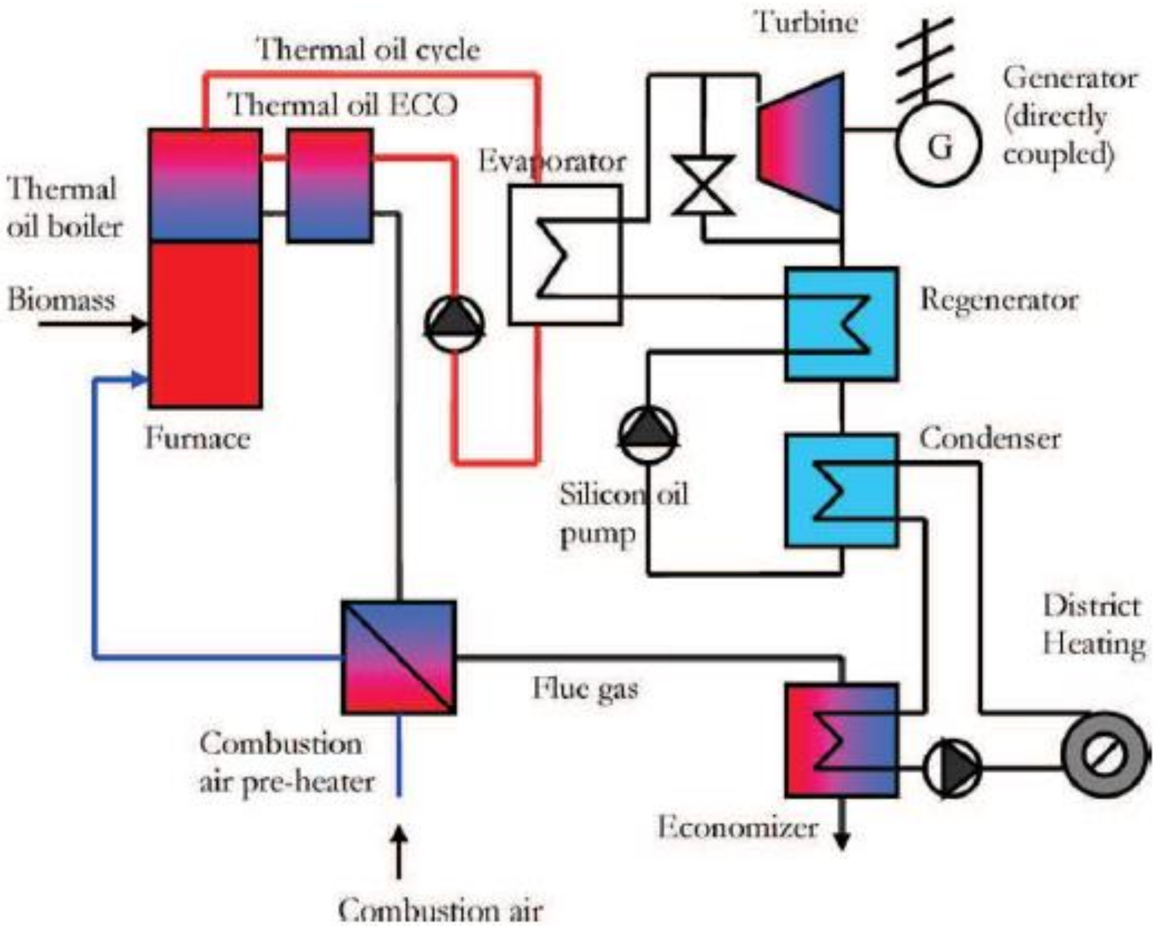


Figure 11 Working Principle of Biomass Power Plant

2.3.4.4 Solar Applications

The concentration of solar energy is a sustainable technology in which the sun is tracked, and its radiations are reflected to a linear or point collector, which transfers the heat into a fluid at quite high temperature. The heat is then sent into a power generation cycle. The three main concentration technologies are the solar tower and the parabolic trough. Satellite dishes and solar towers technologies lead to a higher concentration factor and produce higher temperatures. The steam cycle is well suited to the technologies listed above. The parabolic troughs operate at temperature lower (300 ° C - 400 ° C). So far, they were integrated with conventional steam Rankine cycles for power production [48]. This limitation remains in geothermal or biomass plants: steam cycles need high temperatures, high pressures for their operation therefore a high capacity installed to be profitable. Rankine organic cycles are reassuring technology for reducing small-scale investment costs as they can operate in a low temperature range and the total installed power is reduced to the kW scale. Figure 12 shows the operating principle of such a system [49].

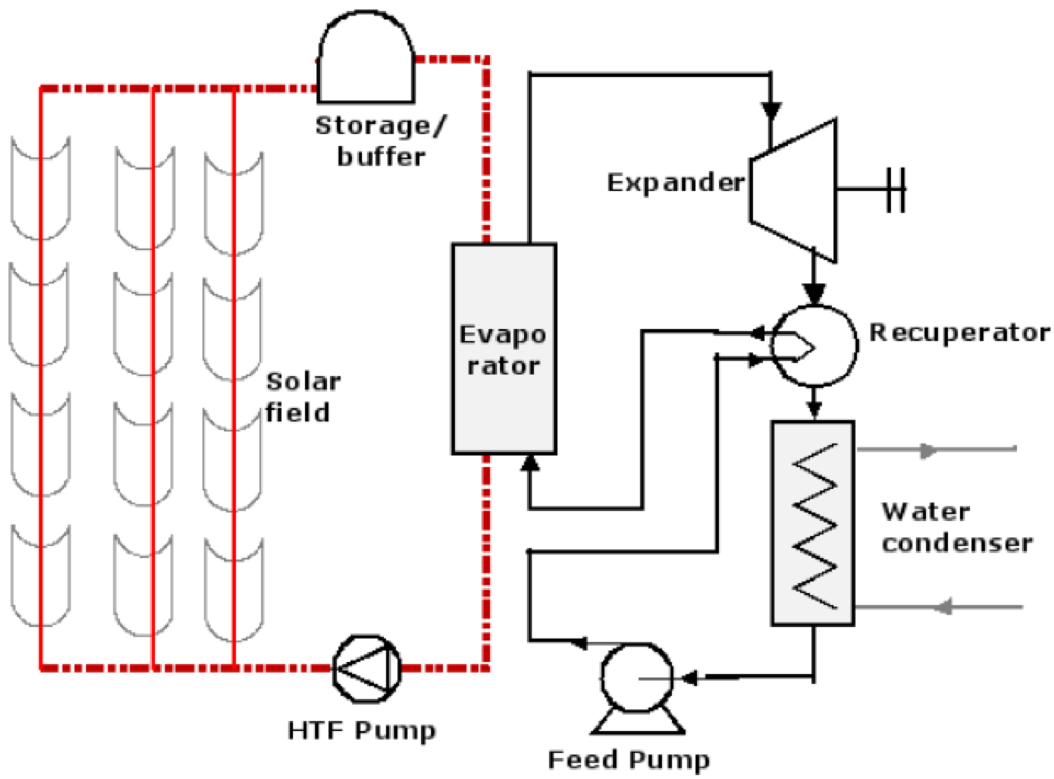


Figure 12 Working Principle of solar ORC

Utilizing low to medium temperature solar collectors are considered in ORC technology for distributed power generation on smaller scale. The research on these systems is in progress and two major possibilities are explored; ORC units / reversible heat pumps running on solar thermal roofs and distributed low cost distributed solar systems [43]. Of these, several demonstration plants have been built. In conjunction with WHR the solar CSP has been used in a 2 MW ORC plant built in Morocco [49].

2.3.4.5 Ocean thermal

Ocean thermal energy conversion systems have opted for ORCs, which generate energy from thermal gradients present in tropical regions of the ocean [50]. Although Japan has built a 100kW demonstration plant, [51] this resource still needs commercial development.

D'Arsonval first suggested the OTEC closed-cycle plant idea in 1881. It uses a low boiling point working fluid i.e. ammonia, R134a or propane and works on a single cycle. of Rankine. Hot surface seawater heats and vaporizes the working fluid in the evaporator. The steam generated runs a turbine integrated to a generator. After the expansion process completes, the low-pressure working fluid vapor is condensed in a heat exchanger with the seawater of the deep ocean which is cold. The working fluid is then pumped back to the evaporator and the cycle runs like this continuously. Figure 13 shows a flow diagram of a closed OTEC cycle[38].

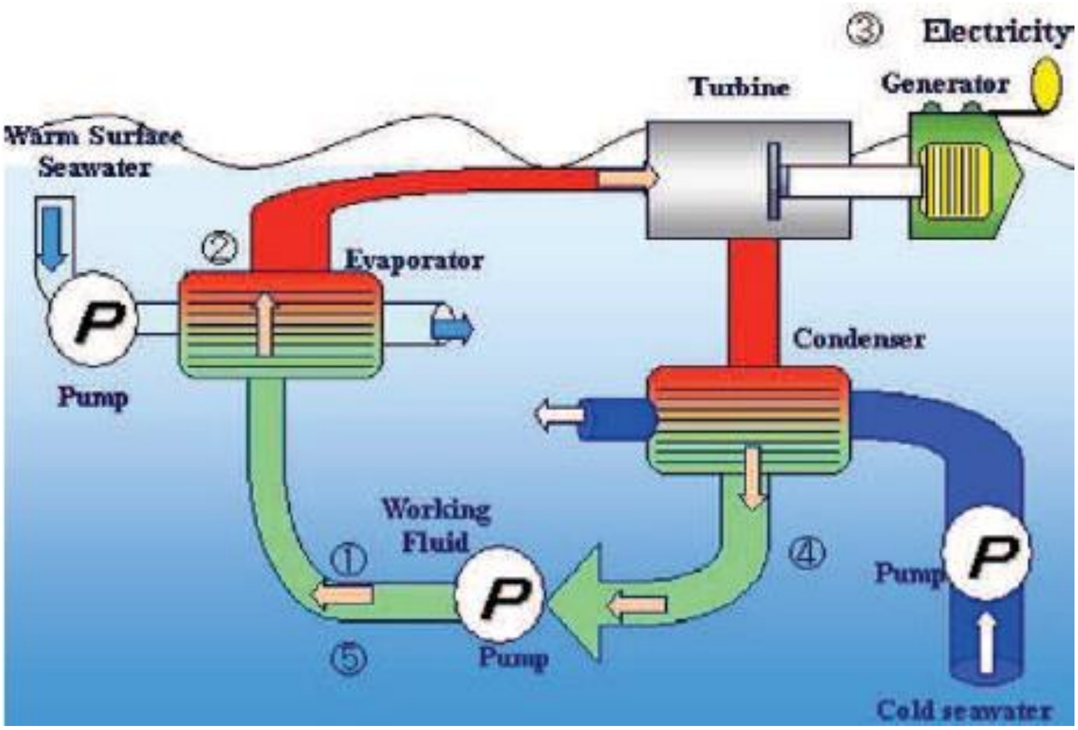


Figure 13 **Schematic of Closed OTEC**

2.4 Solar Heat Engines

Solar energy is the energy packets in the form of radiant light and the heat from the sun reaching the earth’s surface which can be captured with a wide range of technologies i.e. solar heating, photovoltaics, solar architecture and artificial photosynthesis [52].

Solar heat engines technologies are usually divided as a passive solar heating system or an active solar heating system. It all depends upon how solar energy is captured or converted into useful energy. Active solar systems use photovoltaic systems, concentrated solar power and solar water heating to trap energy. In Passive solar techniques building is oriented to the sun and a material is chosen with favorable thermal mass or favorable light scattering properties, and designing spaces in which air circulates naturally [53].

Photovoltaics (PV) also known as photovoltaic solar for the past two decades. It has been developed from a niche market where it was implemented on small-scale applications. Solar cells convert light coming from sun directly into electricity by using the photoelectric effect. Charles Fritts built the very first solar cell back in the 1880s. In 1931, Dr. Bruno Lange, a Germany based engineer, made a solar cell using silver selenide instead of using copper oxide [54].

In the 19th century, Augustin Mouchot connected a Rankine cycle engine to a steam boiler where metal reflectors concentrated sunlight to heat its external surface. At the turn of the century, Willsie and Boyle perceived a solar engine design based on a low temperature Rankine cycle using ammonia, and later Sulphur dioxide, as the working fluid. In the latter system, water used as the heat carrier fluid that was circulated through a non-concentrating collector to reach temperatures of 82 ^{O}C before evaporating the working fluid in a separate vessel via a heat exchanger. The engine claimed to produce 11 kW from a collector area of 93 m^2 [55], [56].

2.5 Working Fluid

An electricity producing ORC needs a fluid, also known as working fluid, which drives the turbine for power generation. Factors like the type of turbine, heat source, heat sink and expected output of the system determine well suited working fluid for a particular ORC system. Selection of the proper working fluid affects the system efficiency, the impact on the environment and condition of operation.

2.5.1 Selection Criteria

A major element in the operation of the ORC is the kind of refrigerant employed as its working fluid. The selection of this refrigerant determines the overall efficiency of the cycle in terms of either cost or generated power or both. Below is the criteria that needs consideration prior to selection of an appropriate working fluid for ORC purposes.

2.5.1.1 Thermal Stability

Thermal stability is the potential of a substance to remain unchanged chemically during a temperature change. The working fluid must possess a good thermal stability. It must be able to maintain its chemical integrity in cases of changes in temperature. Thermally unstable working fluids tend to decompose easily and require frequent replacement. It decreases the efficiency of the ORC system. Chemical decomposition produces gasses that do not condense and can be of adverse effect on components of the ORC system [57]. The chemical stability determines maximum working temperature of the system that the working fluid possesses. Water does not suffer chemical decomposition when used as the working fluid [58].

2.5.1.2 Critical Points

Critical point, an important parameter in fluid selection, referred as the point on the Temperature-Entropy (T-S) curve whereby there is no distinguishable difference between the properties of a working fluid either in the liquid or gaseous state. The fluid composition, density and temperature at either phase, liquid or gas remain unchanged [57]. The peak point of the T-S diagram indicates this critical point. This point indicates the appropriate region of temperature at which the working fluid is optimally operating [58].

2.5.1.3 Fluid Molecular Weight

The intended output for the ORC in terms of power produced and temperature worked at informs the selection of the fluid used for the system. From literature, for a high working temperature or high-power output requirement using a multi-stage turbine, a working fluid with low molecular weight is appropriate. Conversely, a high molecular weight working fluid is appropriate when low power output is desired [57]. During the expansion of fluid in the turbine, the working fluid with a heavy molecular mass would produce a slight decrease in enthalpy which is advantageous to the system operation but will result in a poor coefficient of heat transfer which is undesirable [59].

2.5.1.4 Fluid-Material Compatibility

The working fluid must be compatible with the material of the ORC system and the lubricating oil.

2.5.1.5 System Performance

The system performance is of prime importance while selecting a good working fluid. The chosen fluid is evaluated by simulating the system with the aid of a computer program and supplying the parameters of the fluid properties required [57].

2.5.1.6 Environmental Consideration and Ethics

The working fluid should be environment friendly in ORC applications. Factors i.e Atmospheric Life Time (ALT), Ozone Depletion Potential (ODP) and Global Warming Potential (GWP) of the chosen refrigerants determine the environment safety. In order to preserve earth and keep low rate of ozone depletion, the selected working fluid for the ORC must keep ODP, ALT, and GWP to the barest minimum. An ideal organic working fluid meets this criterion [57], [58]. In the past, various operations employed working fluids with desirable qualities neglecting their negative effect on the environment until Roland and Molina postulated that the chlorofluorocarbons (CFCs) contents of some refrigerants remain unchanged, and on reaching the stratosphere, release chlorine atoms because of the sun's intense radiation. These chlorine atoms, in turn, destroy the ozone layer thereby endangering living things through direct exposure to the sun's ultra-violet rays. Consequently, these organic compounds were prohibited [59].

2.5.1.7 Accessibility and Cost Factors

The overall cost of an ORC system matters a lot. Queries like easy production or availability of fluid and the cost of one-unit mass of the fluid taken into consideration before the eventual selection of the fluid [57]. It is near impossible to have a working fluid that adequately satisfy all the above-stated factors. Hence, choice of the working fluid comes down to the most desired property and which fluid best meets such property [58].

2.5.1.8 Safety

The ORC working fluid must be safe to handle and should possess little or no hazard to the operator and the system. Due to the high level of toxicity, flammability and high corrosive tendency, some fluids deemed inappropriate for the use of ORC operation. In as much as these above listed criteria of non-toxicity, non-flammability and minimal corrosive tendency is desired, they are not always of critical importance as their effect or prevention, as the case may be, can always be safely handled

[42]. The American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) safety classification for refrigerant used in the classification of ORC working fluid. Two key factors, namely the flammability level and the toxicity level of the fluid, govern this classification [57].

2.5.2 A Thermodynamic Comparison of Water and Organic Fluids

Water as the working fluid of is the choice for a number of fossil- fueled Rankine cycle power plants. Water is well suited for high temperature applications, but low temperature operation limits its use. The unique property of organic fluids allows them to supersede where water is lacking. The main difference between organic fluids and water is their behavior when they move from a saturated or overheated state to a turbine at a low to moderate temperature (200 to 400 ° C). The expansion of the turbine in this temperature regime reveals this behavior.

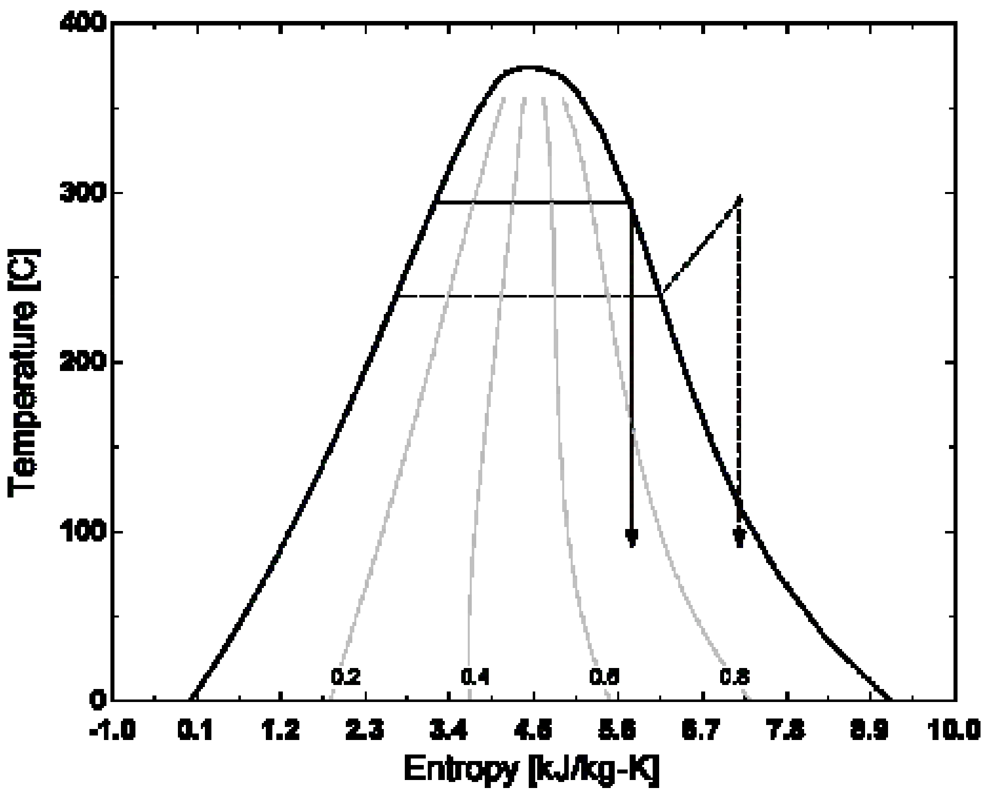


Figure 14 T-S Diagram for water undergoing an isentropic expansion at a single temperature but variable pressures

Figure 14 indicates a T-S diagram for water where two isobaric heat additions are done up to 300 ° C and after that isentropic expansions take place. The isentropic expansion of the water results in a two-phase mixture in the absence of superheat which is relatively low in quality (<80%). Low quality flow through a turbine is not acceptable as it causes remarkable material damage.

The design method used for Steam Rankine cycles shows reheated and superheated cycles to lessen poor flow in the turbine. There are several other ways of mitigating low quality flows, i.e. the mechanical separation of liquid and vapor from the intermediate expansion stages, but they have a disadvantage of being complex and expensive. As per the figure, the desired cycle would be the dotted line, which boils at a lower pressure and overheats to the desired temperature, resulting in a better turbine exhaust. This cycle makes sure that a high-quality flow is flowing through the turbine.

2.5.3 Working fluids

This study analyzed the following fluids as working fluids for the ORC system.

2.5.3.1 R-125

Pentafluoroethane has a chemical formula of CF3CHF2. It is basically a refrigerant. Although its potential for ozone depletion is zero, it has a high global warming potential, which has been explained by the United States Environmental Protection Agency (EPA) to 3 450 times to that of carbon dioxide.

At room temperature and atmospheric pressure R125 is flammable. However, this material will become more combustible when mixed with air under pressure and exposed to strong sources of ignition.

Table 2 enlists the thermodynamic properties of the R-125 [60]

Property (Units)	**R-125**
Molecular mass (kg/kmol)	120
Critical temperature (^{0}C)	66.02
Critical Pressure (MPa)	36.18
Latent heat of vaporization at 1 atm (kJ/kg)	164.1
Boiling temperature at 1 atm (^{0}C)	48.08

Safety	Non-flammable
Atmospheric Lifetime (years)	14
ASHRAE level of safety	A1
Ozone Depletion Potential	≈0
GWP	3500

2.5.3.2 R-218

It is synonym to Octafluoropropane, Perfluoropropane ,Freon 218, genetron 218, Octafluoropropane. Its chemical formula is CF3CF2CF3.

Fluorocarbon known as Octafluoropropane (C3F8) is a non-flammable greenhouse gas which can be produced either by electrochemical fluorination or by the Fowler process where Cobalt fluoride is utilized [75].

Table 3 Thermodynamic properties R-218 [75]

T_c (℃)	71.87
Critical density (kg/m³)	627.9845622
P_c (MPa)	2.64
BP (℃)	-36.79
M (g/mol)	188.01933
Triple point (℃)	-147.7

2.5.3.3 R-236ea

Its chemical formula is CF3CHFCHF2.

Table 4 : Thermodynamic properties of R-236ea[75]

T_c (℃)	139.29
Critical density (kg/m³)	563.004494625
P_c (MPa)	3.50198
BP (C)	6.19
M (g/mol)	152.03928
Triple point (C)	-273.15
GWP	1200

2.5.3.4 R-245ca

Its chemical formula is CHF2CF2CH2F.

Table 5: Thermodynamic properties of R-245ca[60]

T_c (℃)	174.42
Critical density (kg/m³)	523.599999253272
P_c (kPa)	3925
BP (℃)	25.15
M (g/mol)	134.050
GWP	693
Triple point (℃)	-273.15

2.6 Thermal energy storage for solar systems

For the proper functioning of solar-thermal energy systems an important design challenge is the ability to cope with an incident radiation intensity varying with time. A buffer can be provided by Thermal energy storage (TES) for stable operation over a time scale of minutes to hours (using large molten salt reservoirs or solid material filled beds).

Traditional TES solutions used in large-scale CSP systems equipped with Rankine steam plants generally utilize indirect two-tank systems which provides storage for longer periods of 6 to 12 hours of operation at its full capacity. In this configuration, during the charging cycle a molten salt mixture is pumped from a cold storage tank into a hot storage tank thru a heat exchanger where it is given heat from the solar field by a heat transfer oil. In order to supply superheated steam at a pressure of 100 bar, the salts temperatures in the hot and cold storage tanks must be close to 390℃ and 300℃, respectively [61].

Thermocline systems with single tank are often regarded as a lower-cost substitute to two-tank systems and low-cost filler materials i.e. quartzite and granite particles which offer higher specific heat capacity and cut down the required storage volume and cost of extra heat transfer fluid [62], [63].

More energy storage solutions considered suitable for utilization in solar thermal systems include thermochemical storage techniques that either utilize absorption processes or reversible chemical reactions i.e. hydration or carbonation of metal oxides. The storage of the chemical reaction energy demands the careful choice of a reaction adapted to the temperature of the specific application and taken into account in high temperature applications [64].

2.7 Expander

Expanders can be divided into two categories: volumetric and dynamic expander. The choice of the expander relies on the thermodynamic properties of the working fluid, the required mechanical power, the volume and mass flow rate and the volumetric expansion rate. The required output power influences the choice of the expander.

1. The dynamic turbine is better if the power exceeds 1 MW. Changes in pressure occur in the turbine because of the dynamic effects linked with the change in the amount of movement of the working fluid. The high velocity of the fluid flowing, the higher mass flow rate and the higher rotational speed characterize the turbines. A problem can arise at high speeds in the form of a complex connection to a generator producing electricity at frequency of 50-60 Hz. However, the turbines cannot withstand a multiphase flow because the speed high droplet levels will result in significant erosion of the blades. [65]
2. The volumetric expansion valve provides a considerable advantage over the turbine if the power output is less than 1 MW. It dilates a determined quantity of active substance in a closed working chamber.

When compared to a turbine, the volumetric expansion valve has lower flowing fluid velocities, lower mass flow rates and lower revolutions.

2.7.1 Scroll expander

Volumetric expanders classified as rotary - screw, spiral, finned; or alternatively - piston and diaphragm. Figure 15 illustrates the basic structure of the spiral expander [66]. The roller in orbit makes contact with the fixed roller of the expansion chambers. As working fluid enters at the high-pressure and high-temperature at the entry of the expander located at the centre of the stationary scroll, it begins to expand into the sealed pockets, thus causing the disc to orbit in orbit. To the eccentric crankshaft pin the orbital spiral is coupled at the orbiting spiral hub so that the orbiting scroll transmits its orbiting motion to the crankshaft rotation. For the sake of lubrication, the trochoid oil pump is implemented to the rear of the expander.

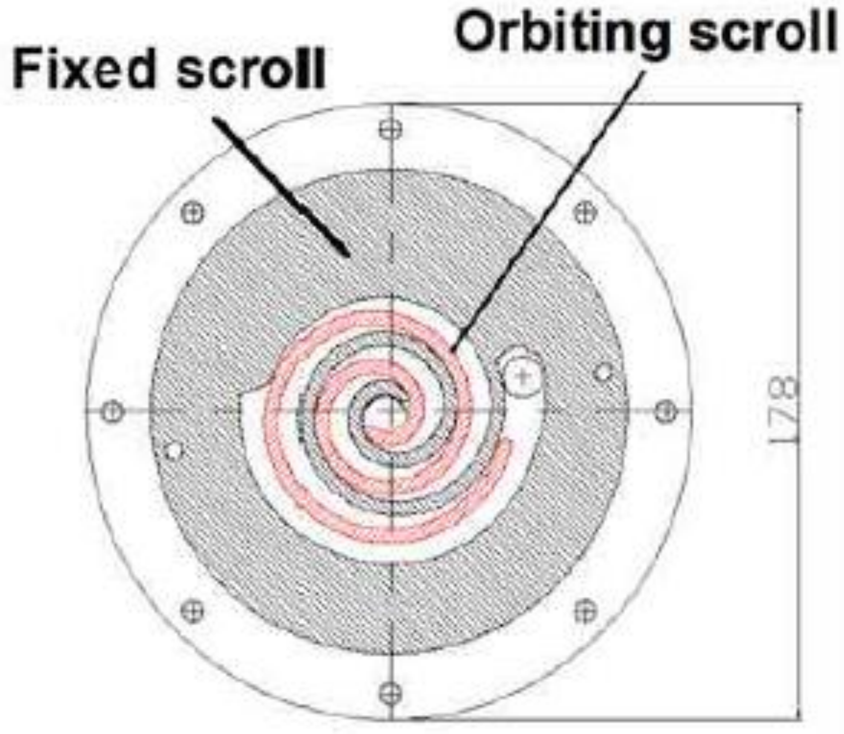


Figure 15 Scroll Expander

2.8 Heat Exchangers

Since heat exchangers absorb a large part of the investment cost of an ORC installation, it is advisable to optimize them first. Heat exchangers are an important part of the total cost. The main features of heat exchanger design are [67]

i) Efficiency (or pinch point) ii) Pressure drop.

While selecting heat exchangers to be used in ORC systems, it is possible to choose either between the tubular heat exchanger or the plate heat exchanger. For both of these types, the efficiency of the entire cycle goes up by increasing the heat transfer surface, as more heat is transferred. Higher heat transfer coefficients can be obtained with plate heat exchanger hence a lower pinch point temperature difference (PPTD) can be achieved which means that the costs are higher, but heat losses are lower due to the process's irreversibility. For small heat transfer surfaces Plate heat exchangers have better properties, although for large heat transfer surfaces (> 2000 m^2) the tubular heat exchangers when used with same working fluids can provide similar results. By increasing heat exchange surface, the PPTD drops abruptly in the shell heat exchanger as it has a different heat exchange surface on the primary and secondary sides of the heat exchangers, unlike the plate type .By increasing the inlet temperature on the heat source side a higher heat transfer value and an increase in the PPTD value is achieved which increases abruptly in the shell heat exchanger at a higher temperature and constant heat exchange surface. An appropriate heat exchanger for the ORC system are designed to use low potential residual heat i.e. a brazed plate heat exchanger [68].

Figures 16 and 17 provide the basic configurations of commonly used heat exchangers [67].

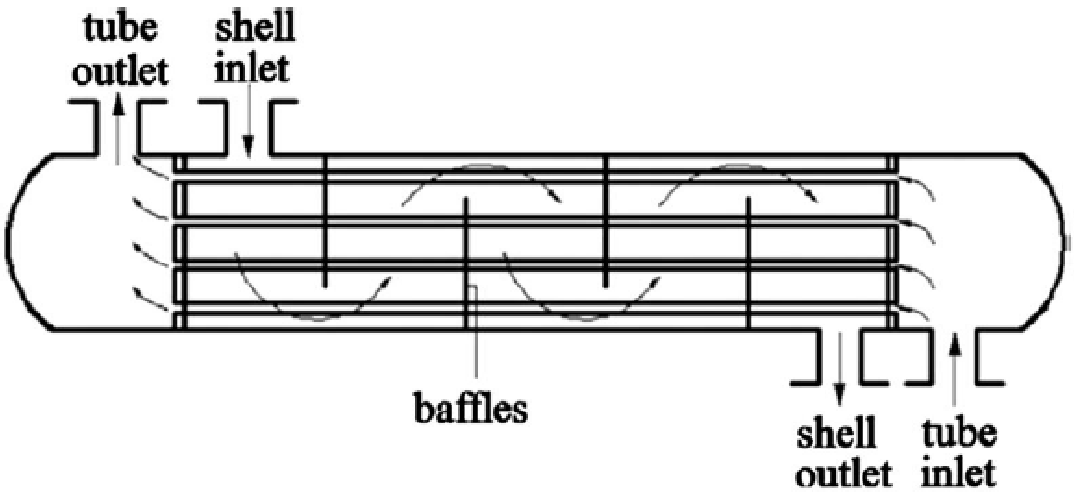


Figure 16 Shell and Tube Heat Exchanger

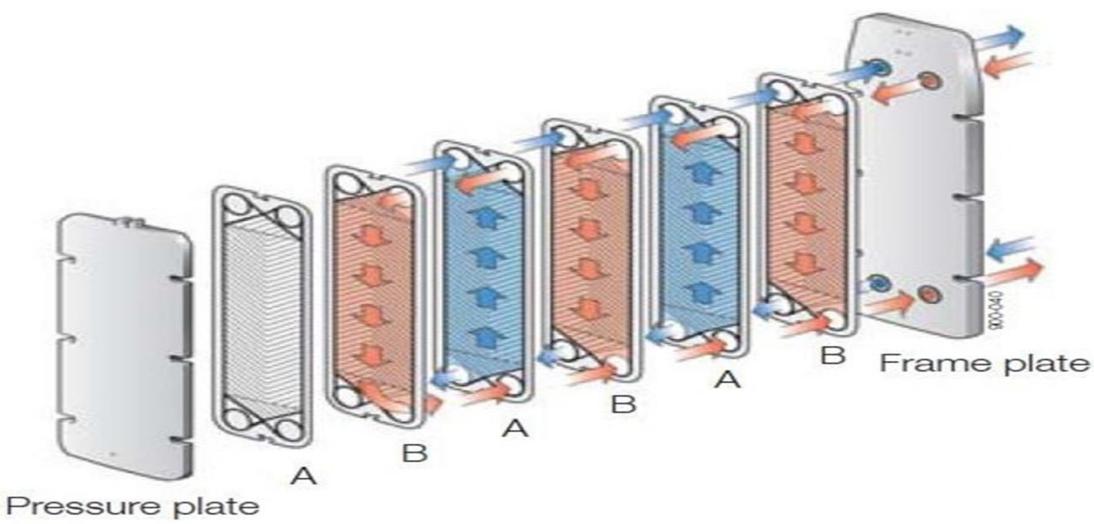


Figure 17 Plate Heat Exchanger

2.9 Recovery Heat Exchanger (evaporator)

The heat transfer rate of the evaporator is crucial. An evaporator too small will not cause complete evaporation of the refrigerant at the outlet of the evaporator, which could damage the turbine. On the other hand, a too large evaporator results in a marginal improvement in performance compared to the added cost. Similar problems arise on the condenser side. Appropriate selection and sizing of the heat exchanger is therefore beneficial for both investment and operational cost [69]

2.10 ORC condenser design

The cooling process of condenser with water or air. Figure 18 shows three various condenser diagrams [70]. The condenser calculations are almost same in both of the cases, because the used cooling fluid (air or water) is placed very near to the thermally linear behaviour. In usual ORC installation, the condenser known to be the heat exchanger amongst the blazing steam of the regenerator / turbine and the cooling means. In condenser temperature of hot fluid is way to high as compare to the cold fluid, but it will not be considered as a significant difference. In addition, this should be remembered that the relationship between enthalpy and temperature is non-linear, which requires that the vaporizer be divided into sections suitable for calculation. This is particularly valid for Kalina cycles. In the ORC cycle, there is only a change of property at the dew point, where the de-overheat ends and the condensation begins.

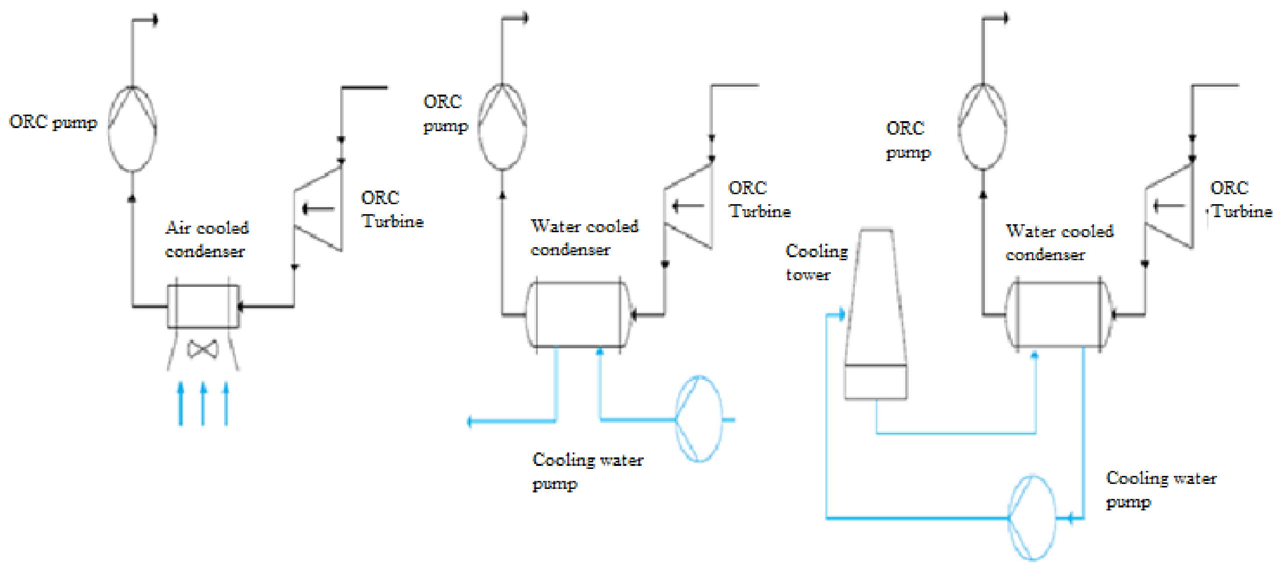


Figure 18 Configurations for Condenser

2.11 Scaling and Material selection

To scale up the system, its technical applications and research consider is the most challenging issues in binary cycle plants design. Tartar (most of the time silica or another kind of deposit) is precisely related to the fluid's chemical composition. Its distinctiveness is that the analysis can only be done on the basis of the experience of the manufacturer and the geothermal exploration, which are lacking in the face of a new reservoir in development, without historical data on drilling. To measure its real impact, it needs a devoted analysis and drilling of wells. The problem of graduation and deposits is currently facing the experience and track record of operators and is one of the sectors least involved in technology transfer [69].

Chapter 3: System Configurations and Modelling in TRNSYS

3.1 Introduction

This study investigated two system configurations of solar-based organic Rankine cycle. First configuration revolves around flow path from collector to hot water storage tank. Whereas the second configuration is modelled and analysed without using hot water storage tank. These two configurations are as followed:

3.1.1 Configuration-I

In this configuration, organic fluid recurring from an ORC system moves towards hot water storage tank connected to solar thermal collector. The auxiliary boiler connects in series with the storage tank to fulfil the remaining heat demand before an ORC system receives the working fluid. Figure 19 shows this configuration [71].

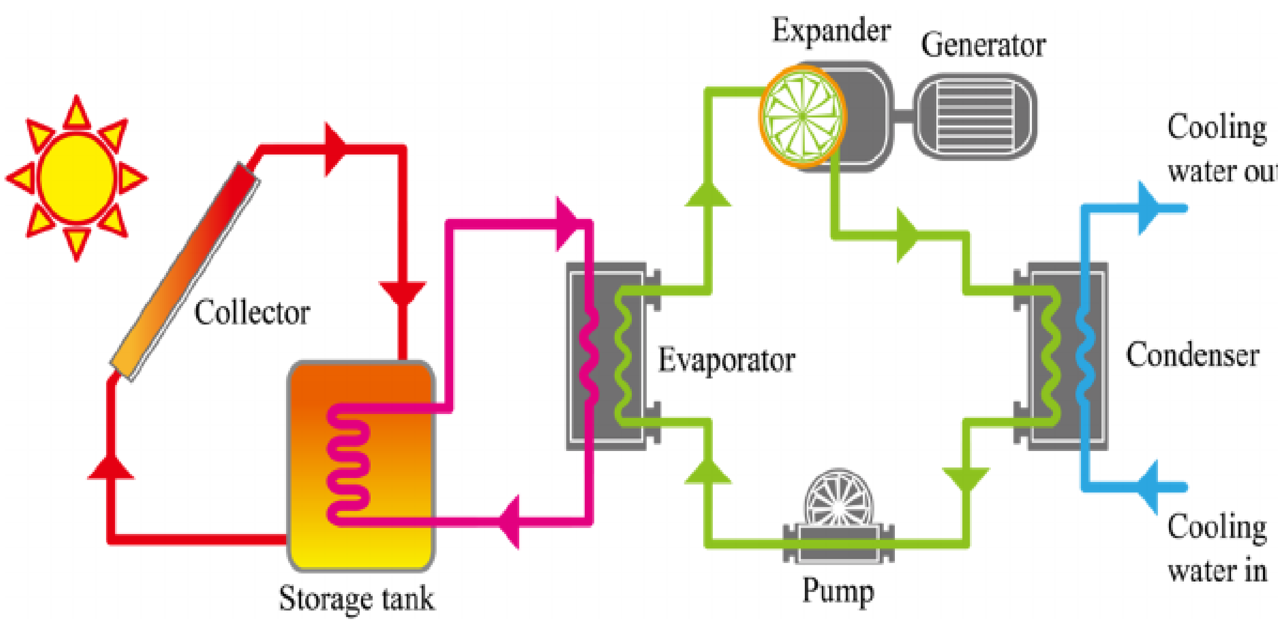


Figure 19 **Schematic for Configuration I**

3.1.2 Configuration-II

This configuration does not employ storage tank for hot water. Therefore, water from the outlet of solar collector directly enters in the heat exchanger and return to the inlet of collector as presented in Figure 20 [71].

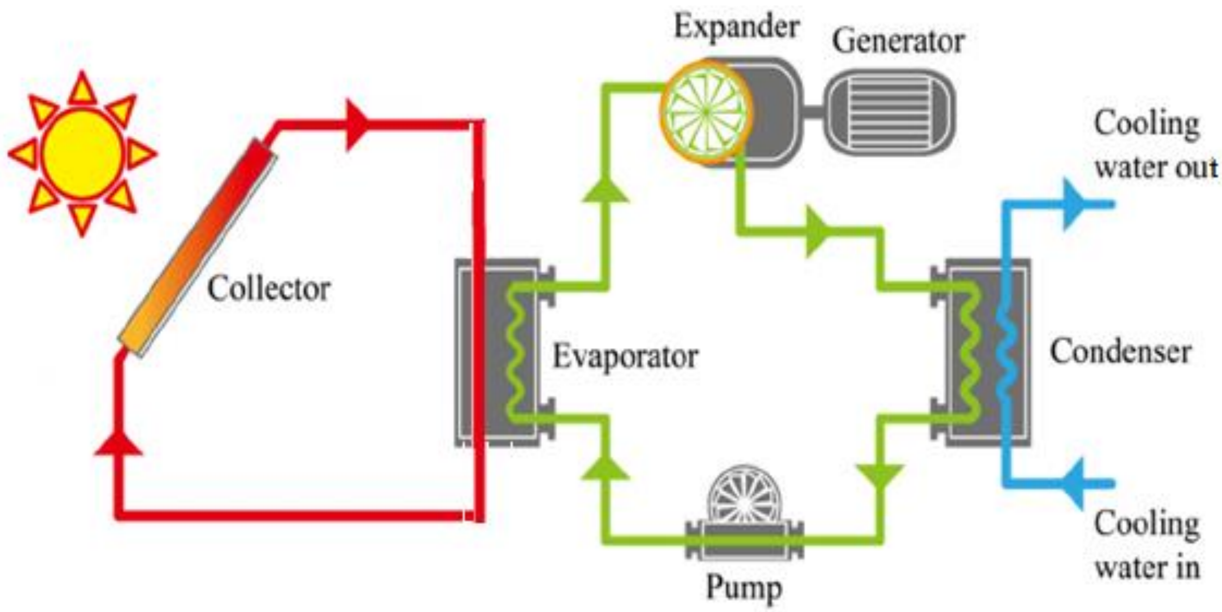


Figure 20 **Schematic for Configuration II**

3.2 Modelling in TRNSYS

TRNSYS 17, a dynamic simulation tool, models and simulates the two system configurations. It relies on predefined characteristic equations of individual system components and shows capability of simulating the complete system performance over long period. Figures 21 and 22 show the pictorial view of TRNSYS model for C-I and C-II respectively.

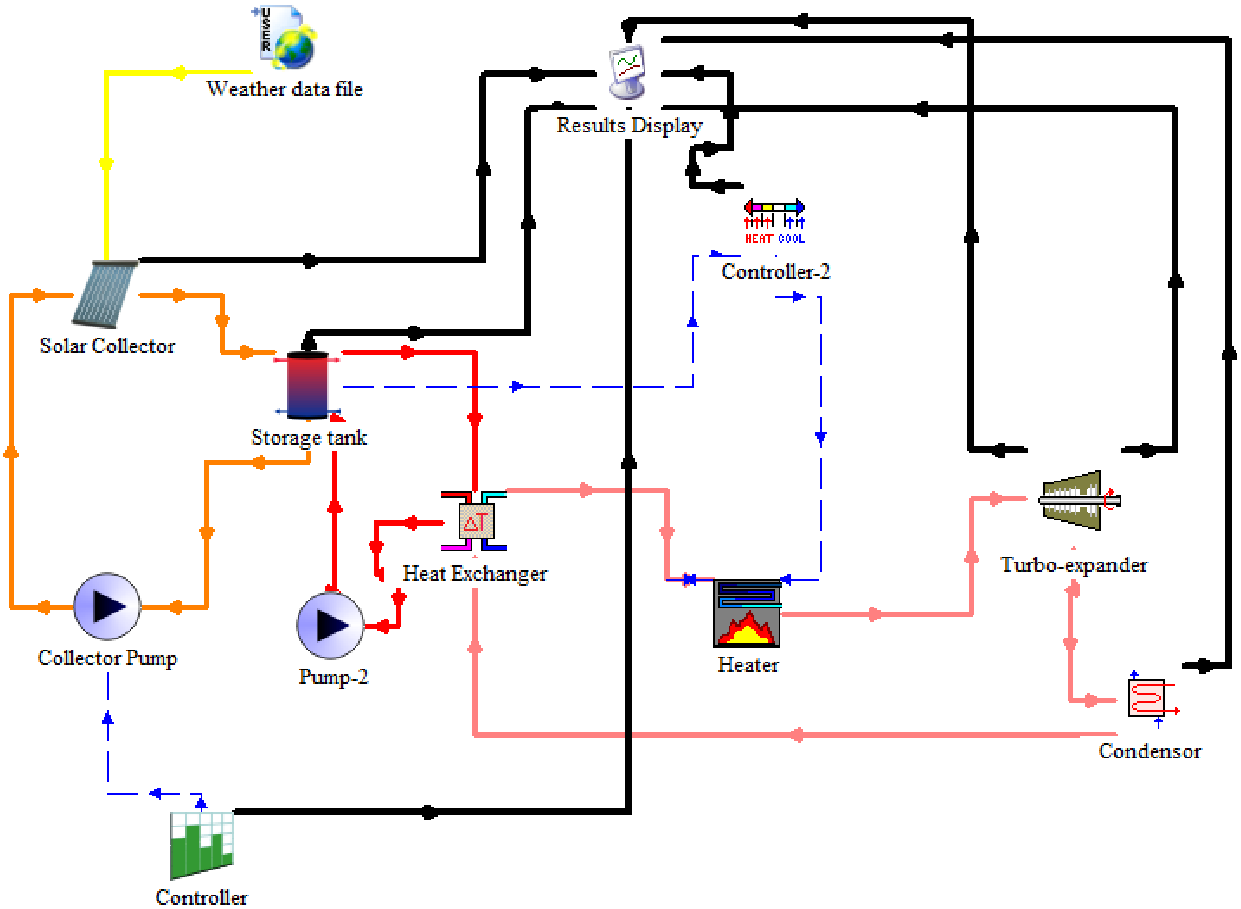


Figure 21 **TRNSYS Model for Configuration I**

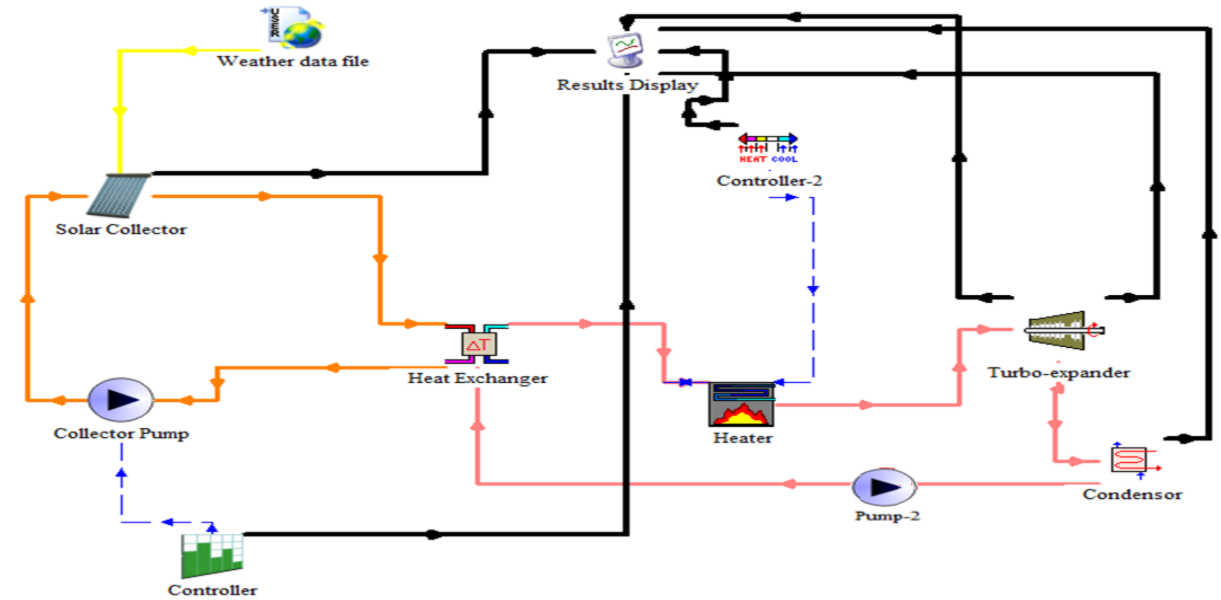


Figure 22 **TRNSYS Model for Configuration II**

3.2.1 System Components

In order to develop a complete solar assisted ORC model in TRNSYS several components play their role. Key components are discussed below:

3.2.1.1 Solar Thermal Collectors (Type 71 and Type 1b)

Flat plate collector thermal performance is analyzed by using TRNSYS component Type 1b. Similarly, evacuated tube collector thermal performance is analyzed by using TRNSYS component Type 71. In this regard data provided by manufacturer Apricus is used. Manufacturer also provide experimental based curves which have ΔT (difference between collector entering and ambient temperature on x-axis and collector efficiency on y-axis).

Solar collector thermal efficiency is written as:

$$\eta = a_0 - a_1 \frac{\Delta T}{I_T} - a_2 \frac{\Delta T^2}{I_T} \qquad (5)$$

Where:

And $\Delta = (T_{in} - T_{amb})$

I_T: Global radiations that are incident on the solar collector (Tilted surface)

a_o = Optical efficiency

a_2 and a_1 are the negative of second and first order energy loss coefficients

T_{in} is water's temperature at inlet of solar collector

T_{amb} is the ambient air temperature.

In this study values of a_o, a_1 (W/m².K) and a_2 (W/m².K²) are taken as 0.845, 1.47 and 0.01 for ETC and 0.749, 2.770 and 0.023 for FPC, respectively. These values are taken from a manufacturer "Apricus" collector's catalogue.

a_0, a_1 and a_2 define the thermal efficiency.[72]

3.2.1.2 Hot water storage tank (Type 4a)

A fluid storage tank is modelled by Type 4a (multi-node) with two internal auxiliary heaters that are optional. This robust model gives precise accuracy while keeping the parametric complexity and computational effort at a very reasonable level. In the standard library it is the most frequently used tank model.

The fluid-filled sensible energy storage tank's thermal performance may be modelled by making an assumption that the tank has N (N ≤ 100) fully mixed equal volume segments, as shown in Figure 23 [73]. N determines the degree of stratification value. When N =1, the storage tank is modelled as a fully-mixed tank with no stratification effects. Several options are available of variable or fixed inlets, equal or unequal size nodes, temperature dead band on heater's thermostats and losses that occur to gas flue of auxiliary heater.

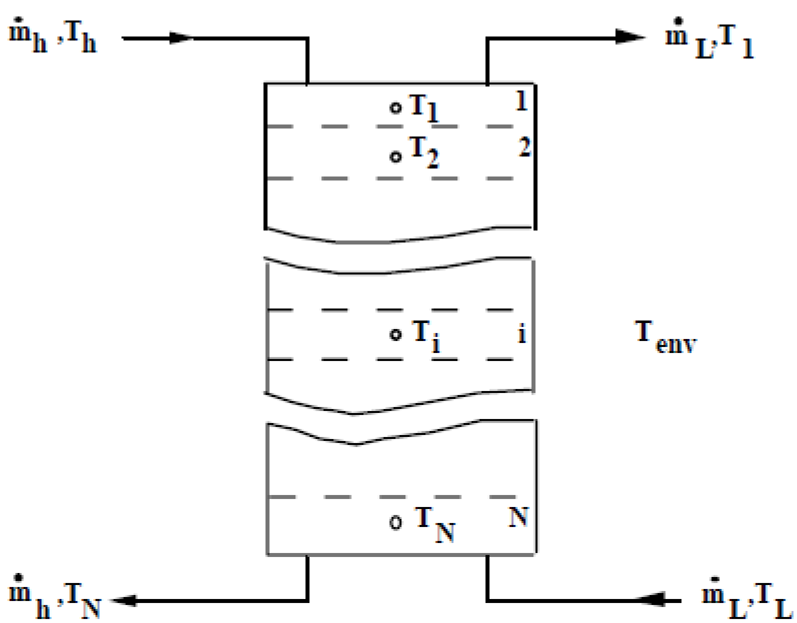


Figure 23 Hot Water Storage Tank

Where

m_h: From the heat source fluid mass flow rate to tank

m_L: Fluid's mass flow rate to the load and/or of the makeup fluid

T_h: Fluid's temperature entering the storage tank from the heat source

T_L: Fluid's temperature replacing that trapped to supply the load

N: Number of fully mixed tank segments (uniform temperature) ($N \leq 15$)

3.2.1.3 Heat Exchanger (Type 5g)

TRNSYS models a zero-capacity heat exchanger in various modes, namely parallel, meter, several cross-flow layouts and shell and tube modes. For all such modes, the input temperatures and flow rates given on the source and load sides help in the calculation of the efficiency for a fixed value of overall heat transfer coefficient. The crossflow modes take one of the given below.

1. The source fluid remains unmixed while the charge side mixes completely.
2. The charge side's fluid remains unmixed whereas the source side mixes completely.
3. None of the fluids mix.
4. The two fluids mix.

Type 5 is based on a minimum efficiency approach in modelling a heat exchanger. In this case, the user supplies the AU and the input conditions of the heat exchanger. The model determines if the load or source side shows the minimum capacity and hence calculates efficiency based on the specified flow configuration and the AU. The following equations calculate the output conditions for a heat exchanger. These equations calculate the capacity of each side of the heat exchanger [73].

$\mathrm{C_c = m_c C_{pc}}$ ……………………………………………... (6)

$\mathrm{C_h = m_h C_{ph}}$ ……………………………………………... (7)

$\mathrm{C_{max}} = \text{maximum value of } \mathrm{C_h} \text{ and } \mathrm{C_c}$ ………………..... (8)

$C_{min} = \ minimum\ value\ of\ C_h\ and\ C_C$ ……………… (9)

Where

$\mathrm{C_c}$: Fluid's capacity rate on cold side

$\mathrm{C_h}$: Fluid's capacity rate on hot side

$\mathrm{m_c}$: Fluid mass flow rate on cold side

$\mathrm{m_h}$: Fluid mass flow rate on hot side

C_{pc}: Specific heat of fluid on cold side

C_{ph}: Specific heat of fluid on hot side

Figure 24 shows the schematic of the heat exchanger.

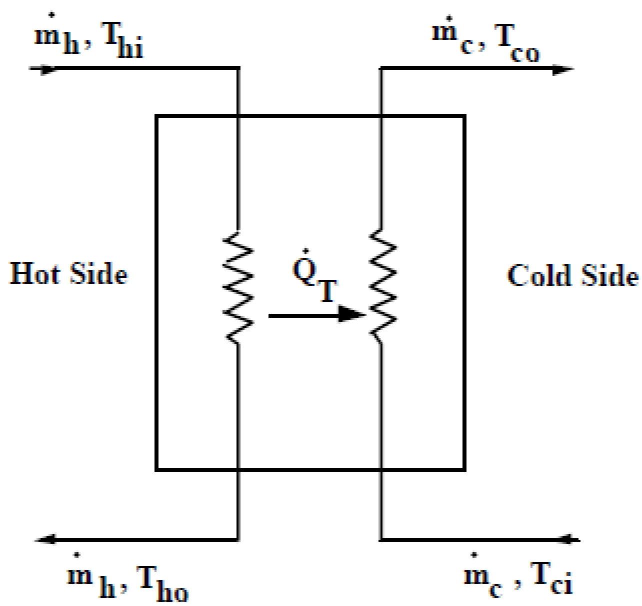


Figure 24 Heat Exchanger Schematic

In shell and tube model, both fluids remain unmixed. The following show the expression which is used to calculate the heat exchanger effectiveness at each time step which depends upon heat exchanger's configuration.

$$\varepsilon = \left[\left(\frac{1-\varepsilon_1\frac{C_{min}}{C_{max}}}{1-\varepsilon_1}\right)^N - 1\right]\left[\left(\frac{1-\varepsilon_1\frac{C_{min}}{C_{max}}}{1-\varepsilon_1}\right)^N - \frac{C_{min}}{C_{max}}\right]^{-1} \qquad (10)$$

Where

ε: Heat exchanger effectiveness

C_{min}: Minimum capacity rate

C_{max}: Maximum capacity rate

3.2.1.4 Steam Turbine (Type 591)

A steam turbine is simulated by this model that responds to pre-defined electrical load by changing steam's mass flow while entering the turbine. When the model determines the steam flow that is needed at turbine's inlet, the inlet and the steam flow is used only as mass balance control at the point where convergence takes place at each time step. Based on an isentropic efficiency approach this model is used to calculate the performance of the steam turbine as a function of turbine back pressure and steam inlet conditions. Users specify up to five injection ports and five extraction ports along the turbine's length. On these ports, the steam conditions can be provided in any order of the model - the type will dispose them properly depending on the flow pressure. The turbine will place the injection and extraction streams at the suitable locations automatically along the turbine's length in order to match the flow pressures to the local turbine pressure. The output power provided by the turbine is limited by the turbine's capacity. If you specify a higher power requirement than the machine's capacity, the machine will operate at its rated capacity and report the unfilled load [74].

3.2.1.5 Auxiliary heater (Type 751)

In TRNSYS, Type 751 models an auxiliary heater (gas boiler). The amount of energy that is needed to raise the inlet water temperature to the required temperature (provided that control signal to the boiler is ON and there is a flow through the boiler) is calculated by using the following equation:

$\dot{Q}_{need} = \dot{m}_{fluid} Cp_{fluid} (T_{st} - T_{in})$ ------------------ (13)

If control signal is ON and set temperature is less than inlet temperature, heater will not find negative value of $\dot{Q}_{need}$. The energy required is limited by capacity of the boiler.

Equation used to determine the amount of fuel utilized by the boiler after calculating the required energy is given as: [74]

$$\dot{Q}_{fuel} = Q_{need}/\eta_{boiler} \quad \text{..............................} \quad (14)$$

3.3 Methodology

The system consists of two circuits: the primary and the secondary circuit.

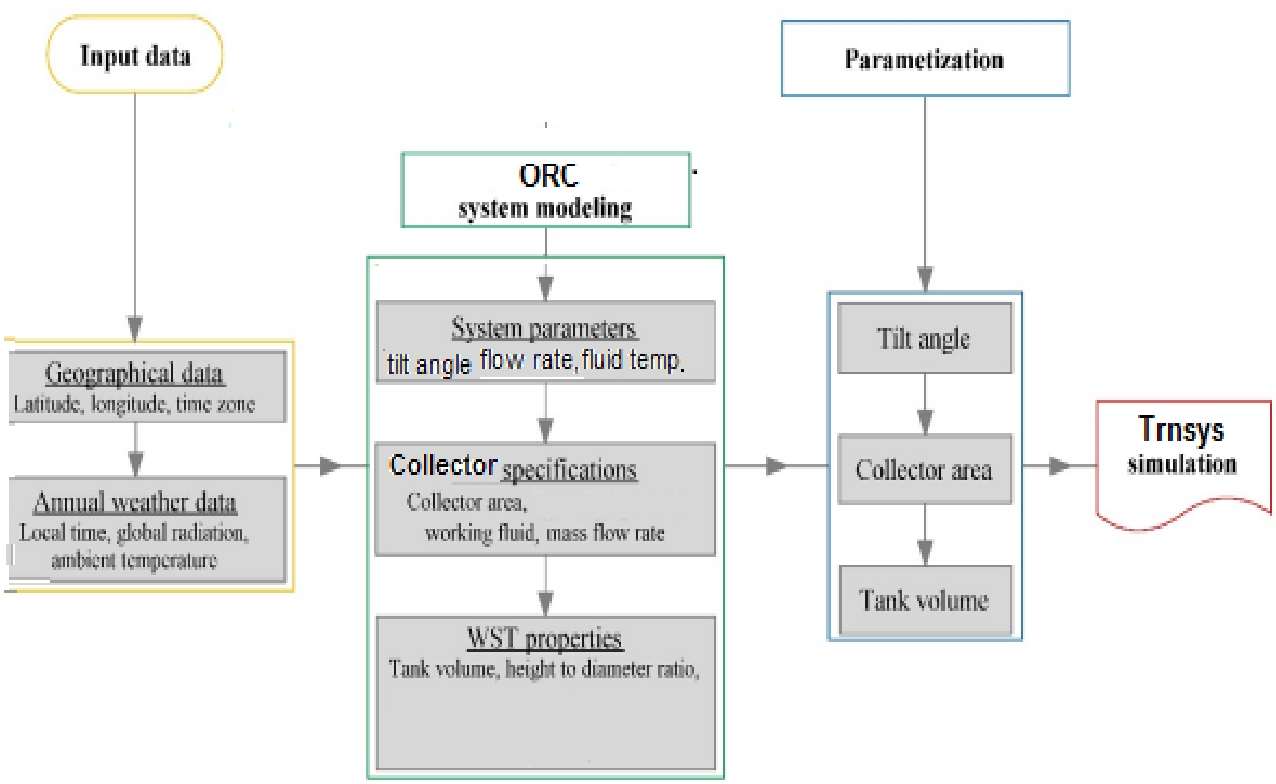
The working fluid expands in the turbine in the primary circuit from where it goes into a condenser where it is condensed.

In the secondary circuit, the pump circulates the hot water of the solar collector (Type 1b or 71) in the hot water tank (Type 4a). and two possibilities are discussed: solar collectors without solar thermal energy storage tank (CI) and solar collectors connected to a solar thermal energy storage tank (C-II).

In both cases, a type 751 auxiliary heater (a gas boiler is considered as an auxiliary device (backup gas) when the pump receives an activation signal from the controller.) The set temperature of the auxiliary heating was 100 ° C. Type 108 controller on the boiler when the outlet temperature is below 100 ° C.

A heat exchanger transfers heat between the two circuits explained above.

3.3.1 Algorithm



3.4 System Performance Indicators

Following performance indicators, based on integrated values of various energies over the whole, optimize the system:

3.4.1 Solar fraction

Solar Fraction is the quantity of energy contributed by solar thermal collector divided by total energy needed to meet the required heating and/or cooling demand.

$SF = \frac{\int Q_u}{\int Q_u + \int Q_{aux}}$ …………………………………….. (15)

Where

Q_{aux}: heat energy from auxiliary boiler

Q_u: the useful energy gain by the solar collector

Its value is zero when system is entirely running on auxiliary heater and 1 when all the required energy is supplied by the solar system. High quantity of solar fraction is always desired to impose less load on auxiliary device.

3.4.2 Collector efficiency

Solar collector efficiency estimates the long-term performance of a solar collector. The following equation calculates the efficiency:

$\boldsymbol{\eta = \frac{\int Q_u dt}{A \int G\, dt}}$**……………………………………… (16)**

Where

G: The solar radiation (global) on a horizontal plane

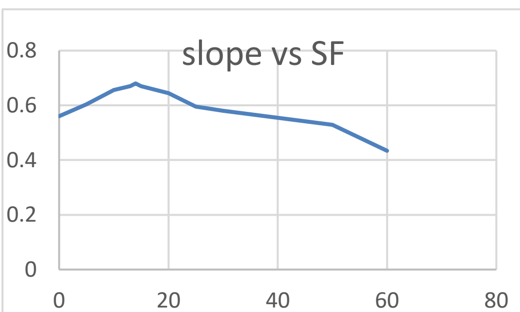
A: The collector’s area.

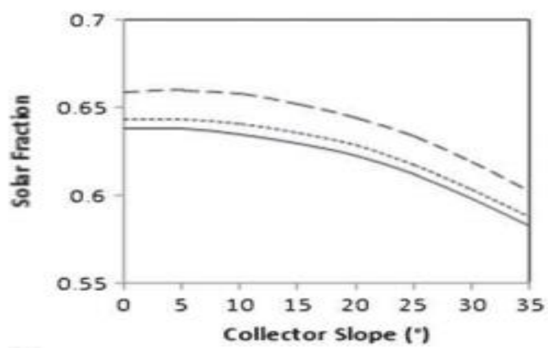
Chapter 4: Model Validation, Results and Discussion

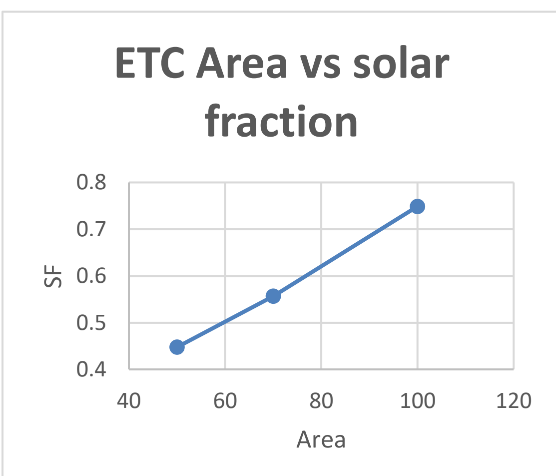
4.1 Variation of solar fraction with collector area and collector tilt

Figure 4.1 exhibits the effect of varying ETC area and tilt angle on SF.

Maximum value of SF is achieved at 14° tilt angle for each value of solar collector area. Increasing tilt angle further causes SF to decrease. Similar trend is also obtained in case of FPC. Trend shown below is supported by Wajahat khan [74] for a solar based adsorption chiller. Papoutsis et al. presented similar trend for a solar based adsorption chiller. Their results showed that maximum SF is obtained at a collector tilt of 14° for Athens, Greece [76]. Similarly simulation results of Roonak Daghigh [77] with several types of PVT parametric studies of an active solar water heating system collectors also showed that optimum angle is 12° for maximum solar heat gain for the location under consideration and simulation results of Ding Ling Zhao [78] for optimal study of a solar air heating system with pebble bed energy storage verify the trend for solar collector area vs solar fraction as shown in figure 4.1







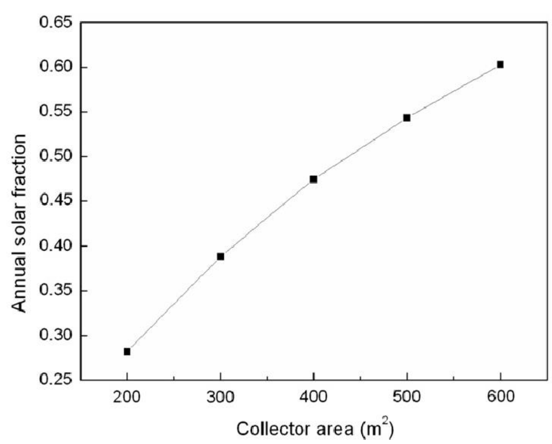
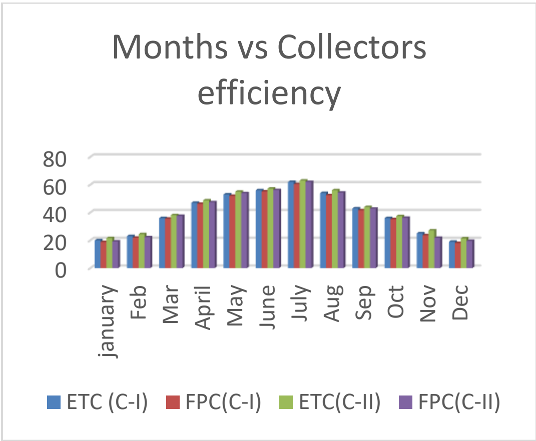


Figure 4.1 Comparison of effect of variation of collector area and collector tilt on SF with literature.

4.2 Collector Efficiency

Figure 4.2 shows comparison of collector efficiency for both configurations for ETC and FPC with literature. Trend shown below is supported by Wajahat Khan. [74]. Their analysis shows that entering water in a system without storage tank the solar collector hot water has greater values of collector thermal efficiency. It is because of temperature of storage tank is comparatively lower as fluid is returning directly from condenser.



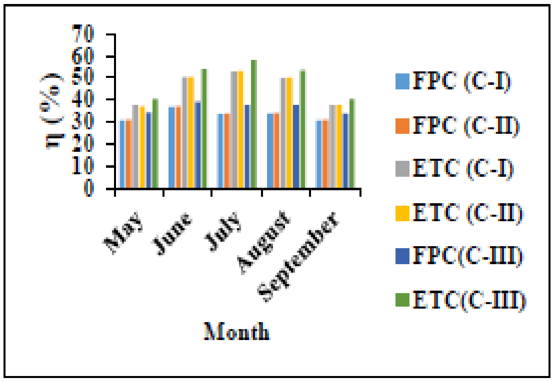


Figure 4.2 Comparison of collector efficiency with literature.

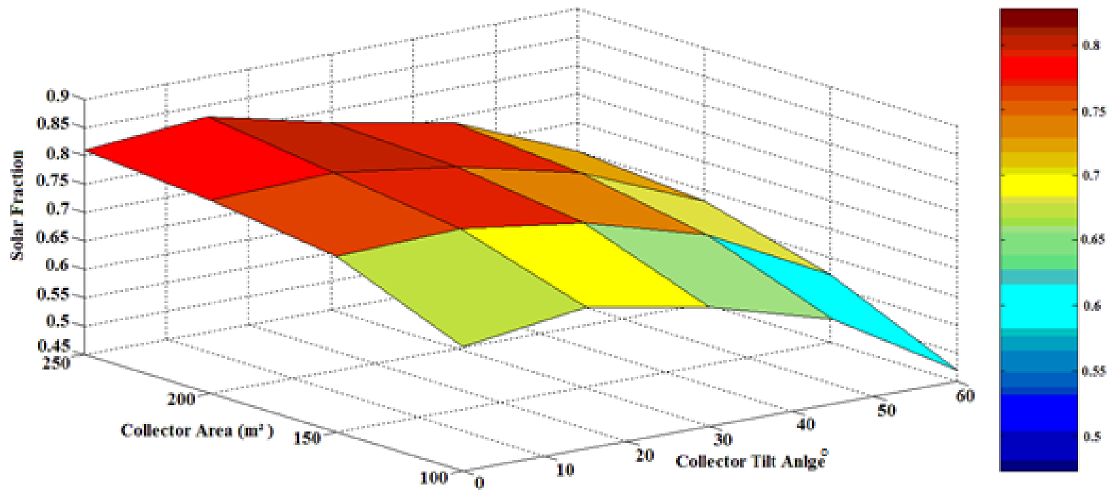
4.3 Results and Discussion

Simulations were performed in TRNSYS from 0 hour i.e. 1st hour of 1st January till 8760 hr i.e. last hour of 31st December.

Simulation time step is 0.125 hr.

4.3.1 Variation of solar fraction with collector area and slope

Figure 4.3 exhibits the effect of varying ETC area and tilt angle on SF. Maximum value of SF is achieved at 14° tilt angle for each value of solar collector area. Increasing tilt angle further causes SF to decrease. Similar trend is also obtained in case of FPC.



➢ As collector area increases SF increases. Maximum value of SF is achieved at 14° tilt angle for each value of solar collector area. Increasing tilt angle further causes SF to decrease.

4.3.2 Seasonal variation of collector efficiency

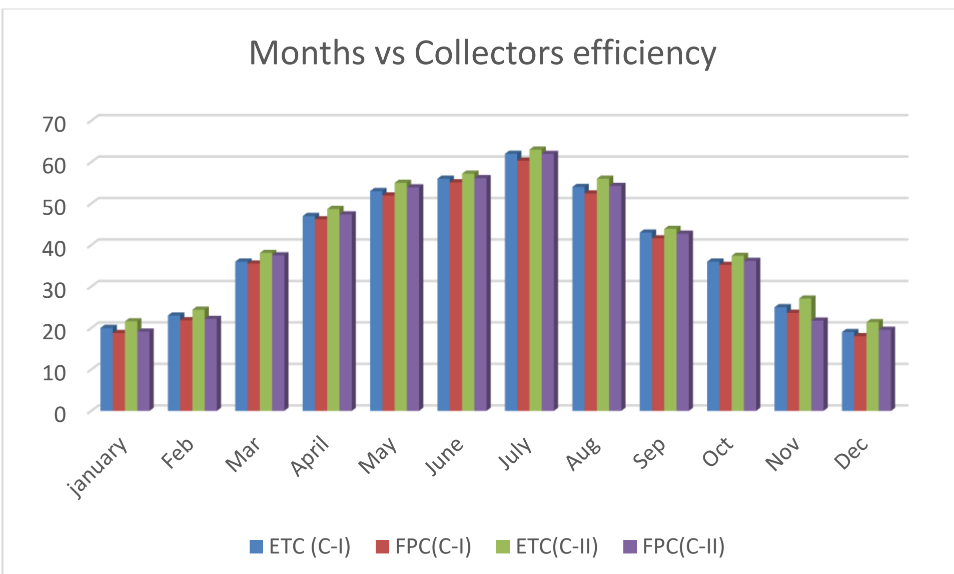
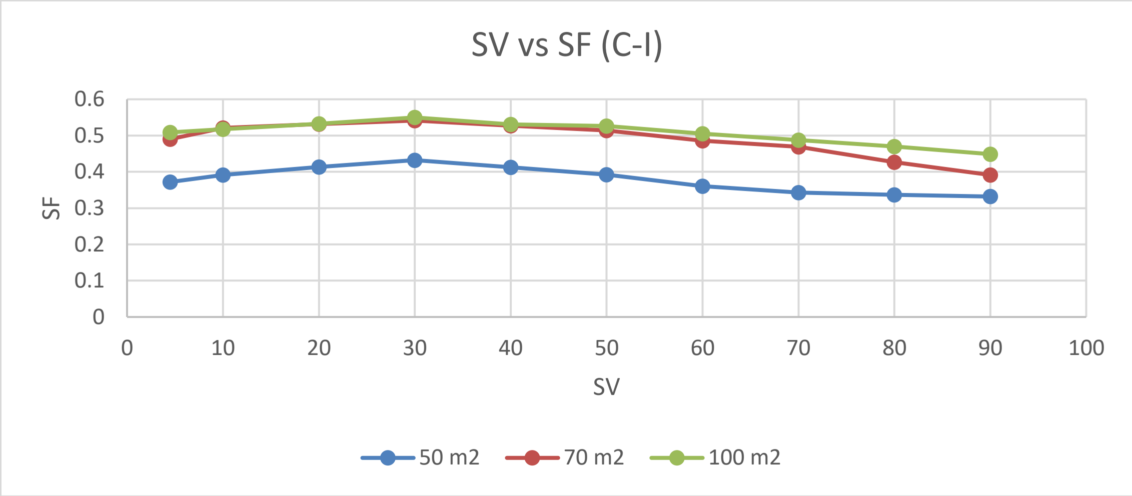


Figure 4.2 presents monthly thermal efficiency of FPC and ETC for both configurations corresponding to FPC and ETC areas of 68.14 m² and 50 m² respectively. It is evident that ETC yields higher monthly efficiency than FPC, however the effect of system configuration (C-I or C-II) on collector efficiency is marginal. It is also observed that for fixed collector areas of 68.14 m² and 50 m², system without hot water storage tank has higher values of collector thermal efficiency.

It is because that temperature of the water entering the array of solar collector in a system without hot water storage tank is comparatively lower as water entering in the solar collector from hot water storage tank.

4.3.3 Variation of solar Fraction with respect to Storage Tank size



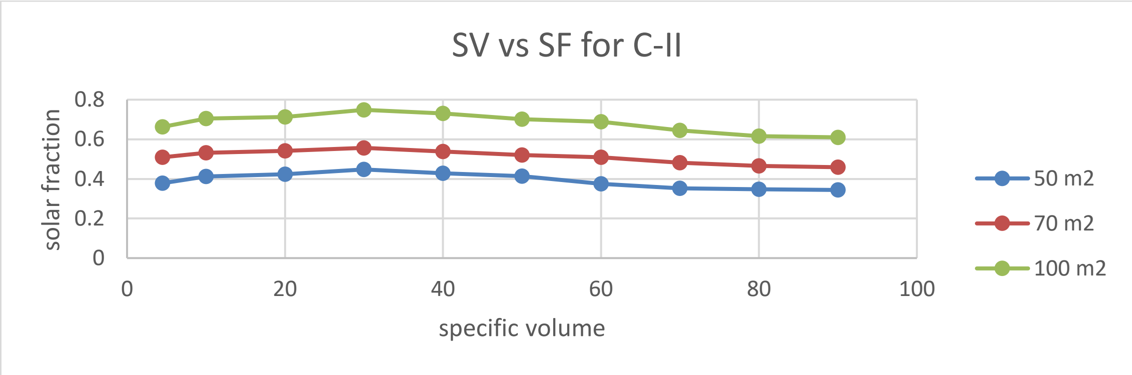


Figure 4.3 shows variation of with respect to specific storage volume for both configurations corresponding to ETC areas of 50 m^2,70 m^2 and 100 m^2.

C-II has higher SF than C-I. Moreover, for any solar collector Area, solar fraction increases as storage tank size increases up to 30 L/m^2. By further increasing size of storage tank, solar fraction decreases by increasing solar collector area.

This is also valid for FPC.

4.3.4 Most suitable Organic fluid for targeted output:

Substance (Fluid)	$T_{ou}t$ (°C)	η_{th} (%)	A_c (m^2) for ETC	A_c (m^2) for FPC	Storage Tank size (liters)
R125	89.45	3.36	50	68.14	1350
R218	81.78	7.5	58.24	83.17	1570
R236ea	55	12.83	63	87.4	1700
R245ca	51.54	12.83	68.12	92.3	1832

The table above shows that the organic fluid R125 is the best option amongst all as it requires least collector area i.e. 50 m^2 to obtain the targeted output i.e. 1KW.It is also suitable option if we consider the quality of steam at turbine outlet as T_{out} for R125 which is higher. This shows that it is comparatively drier and is least damaging to turbine blades. Moreover, it requires minimum optimum storage. Lastly, R245ca has maximum efficiency amongst all.

Conclusions

This thesis presents the modelling and optimization of two system configurations of solar assisted Organic Rankine Cycle (ORC) system for Islamabad (33.71° N, 73.06° E). Simulations of the system are done in TRNSYS for the whole year to investigate the possibilities of driving ORC with low grade heat, i.e. from an evacuated tube or flat plate solar collector. Results of the analysis showed that for the given collector area, highest solar fraction is achieved at collector tilt of 14°.
Simulation results showed that C-II gives comparatively higher collector thermal efficiency and same trend was observed for FPC. Moreover, efficiency of ETC is higher than FPC. R125 and R245ca emerged as good fluids owing to the least collector area (50 m^2 for ETC and 68.14 m^2 for FPC) for desired output i.e. 1KW and maximum fluid efficiency i.e. 12.8 3% respectively.
Corresponding to these areas optimized size of HWST is estimated to be 1350 litters. The thermal efficiency for ETC at 50m^2 is higher as compared to FPC at an area of 68.14 m^2. Increasing the collector area improves the solar fraction remarkably but its rise decreases gradually after 70 m^2.R125 requires least storage tank size i.e. 1350 L for desired output. Increasing the specific storage volume above 30 L/m^2 has no significant effect on solar fraction rather the solar fraction is adversely affected, particularly after 90 L/m^2 due to heat dissipation of the storage tank. It is also concluded that If system has to be installed without hot water storage tank then solar collector area must be increased for the purpose of achieving SF similar to that of system with hot water storage tank.
The optimum ratio between storage volume and collector area increases as the collector area increases. Therefore, a trade-off between heat collection and heat loss has to be made while attempting to increase solar fraction by improving collector area.

Future Work

Simulation results presented in this study have shown that there is good potential for solar based Organic Rankine Cycle in Pakistan. Till date there is not even a single solar based Organic Rankine Cycle system installed in Pakistan. Therefore, model presented in this study can be taken as a reference for design and installation of solar assisted Organic Rankine Cycle.

However, the following ideas should be addressed in future work:

1. Performance analysis of system by addition of PV panels in the proposed solar based Organic Rankine Cycle system so that electric load of different devices e.g. pumps can be shifted on PV panels. In this way the system will be less dependent on electricity from grid hence results in more energy savings.
2. Due to absence of any solar based Organic Rankine Cycle system in Pakistan, results presented in this study for proposed configurations have been validated with literature and need experimental validation.
3. By addition of some components, the proposed model in this study can be used to carry out year around simulations for space heating/cooling.

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