Comparison of Thermodynamic Performance of Geothermal Organic Rankine Cycle with Two Different Working Fluids

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Session 2019-21

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THESIS ACCEPTANCE CERTIFICATE

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Dedication

The thesis is wholeheartedly dedicated to my beloved parents and my supervisor Dr. Majid Ali. A special thanks to Dr. Majid Ali, Dr. Adeel Waqas, Dr. Mariam Mahmood, and Dr. Sehar Shakir for pushing me forward in times when I struggled. I am thankful for their love and measureless support. All of you have been a driving force throughout this process.

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Abstract

The organic Rankine Cycle is a potential power-generating technology that can be used to transform low and medium heat sources into electricity. As compared to conventional Rankine Cycles, ORC operate at low operating pressures and temperatures. Organic fluids or refrigerants have low critical temperatures as compared to water which operates at higher temperatures as compared to organic fluids. Organic fluids can be utilized as working fluids in ORC applications as a substitute for water in the conventional Rankine Cycle. This research provides a thorough simulation of Organic Rankine Cycle processes for the conversion of heat energy from geothermal resources. The selection of the working fluids is vital since the performance of ORC systems is dependent on the working fluids. Power generated and thermal efficiency by a baseline ORC, recuperative ORC, and ORC with pre-heater are calculated with refrigerants R113 and R245fa at different evaporation temperatures and condenser temperatures. The comparison of baseline ORC, recuperative ORC, and ORC with pre-heater has been conducted. Evaporation temperatures have been optimized to acquire the highest power output at evaporator PPTDs of 5-15 °C and geothermal source temperatures of 150 °C, 168 °C, and 180 °C. Work output the efficiency of the cycle has been enhanced by the deployment of an internal heat exchanger. Cycle-Tempo is a modern graphical tool, which is used for optimization and thermodynamic analysis of the systems to create electricity, heat, and refrigeration. The temperature of the condenser has varied from 30 °C to 50 °C. The maximum power obtained is 2942.38 kW by working fluid R113 with recuperative ORC at the source temperature of 180 °C. According to the results, the recuperative ORC is the best performing configuration among the three configurations. The working fluid R113 shows maximum thermal efficiency of 14.84% with recuperative ORC.

Key Words: Organic Rankine Cycle, Geothermal ORC, Thermodynamic analysis, Performance analysis, Internal heat exchanger

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Abbreviations

ORC	Organic Rankine Cycle
PPTD	Pinch Point Temperature Difference
h	Specific enthalpy
W	Specific work of turbine/pump
Q	Heat absorbed/rejected
A	Heat transfer area
$\dot{m}$	Working fluid mass flow rate
U	Overall heat transfer coefficient
η	Efficiency
$\dot{m}_{cf}$	Mass flow rate of cooling fluid
$c_{p,cf}$	Specific heat capacity of cooling fluid
$T_{out,cd,cf}$	Outlet temperature of condenser cooling fluid
$T_{in,cd,cf}$	Inlet temperature of condenser cooling fluid
$\dot{m}_r$	Mass flow rate of refrigerant
$h_{in,cd,r}$	Enthalpy of the refrigerant at the inlet of the condenser
$h_{in,cd,r}$	Enthalpy of the refrigerant at the outlet of the condenser
$p_{cd,r}$	Pressure of refrigerant in the condenser
in	Inlet
out	Outlet
cri	Critical
th	Thermal
T	Turbine
P	Pump
ODP	Ozone Depletion Potential
GWP	Global Warming Potential

Chapter 1: Introduction

Geothermal energy is a viable alternative to conventional fossil fuel-based power plants. Power plants utilizing the Organic Rankine Cycle are appropriate for using low-grade energy resources (below 150 °C) like geothermal energy [1].

1.1 Background

The last decade has seen a rise in energy demand globally to support economic and social growth which has had an impact on the environment and reserves of fossil fuels. The demand for cleaner alternative fuel sources was driven by these elements. Access to renewable energy is vital to meet energy requirements, and support economic growth, social advancement, and a higher standard of living. By 2030, the Sustainable Energy for All initiative of The United Nations (UN) seeks to ensure worldwide access to cheap, dependable, and Sustainable Energy, with 60% of that growth expected to take place in developing countries that possess renewable energy resources abundantly [2].

These days, it is crucial to replace fossil fuels with renewable energy which is clean energy, and the sustainability of energy systems is dependent on additional effective energy-based and efficient utilization of renewable energy resources. Geothermal resources are abundantly available and regarded as energy resources having high potential, wherein the urban regions near those resources may be geared up for uncommon conditions. Geothermal resources are considered high-capacity energy providers, wherein the urbanized zones near those resources may be geared up for uncommon conditions. The current pattern of energy usage is unsustainable because it is broadly dependent on finite, depleting fossil fuels and results in unprecedented greenhouse gas emissions (GHGs) even while greater global energy consumption is a sign of an increase in economic growth in developed countries (GHGs) [3].

During the past few decades, geothermal energy has fulfilled energy demands on domestic as well as industrial levels. It is an abundant source of energy having high potential which can be utilized to meet energy demands in the future. Geothermal energy uses the internal heat of the earth which is produced and stored in the earth [4].

1.2 Geothermal Energy

Geothermal, which may be translated as geothermal heat or energy generated by the earth, is derived from Greek terms, geo which means earth and thermal which means heat. Energy in the form of water, which can be in the form of vapor, liquid, or a combination of both is present in the form of geothermal fluid. Typically, the geothermal fluid containing geothermal energy is found more than one kilometer below the surface of the earth [5]. The temperatures of the core of the earth can exceed 6650 °C, which is due to radioactive decay, the geothermal fluid gains energy from the core [6], and the geothermal energy progresses to the surface through conduction and convection processes [7]. Up to 42 million MW of energy is expected to arise from these operations. A heat source, water, and permeable layer are three crucial components involved to produce geothermal energy in a specific region of the earth [8]. Figure 1.1 illustrates how a geothermal source is formed when water seeps into the soil through rain or melting snow and is retained in an impermeable layer [9]. For 100,000 years, geothermal energy can meet the world's energy consumption needs [10].

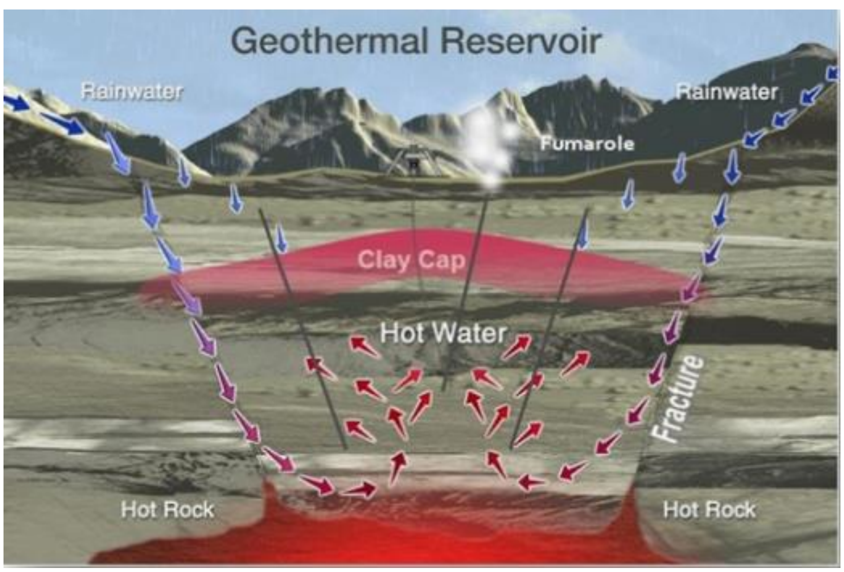

Figure 1.1 The geothermal system diagram [11]

Geothermal energy is the thermal energy that is enclosed in the interior body of the Earth and is made possible by the stratified structure of the earth. The lithosphere supports the subsequent motion of tectonic plates because of forces executed by the convecting mantle, which promotes several thermal processes at the surface of the earth. Examples of such reactions relate to hydrothermal activity all over the earth and include mid-oceanic ridges, mountain belts, and volcanic activity sites. Magma serves as a heat source for fluids that are trapped under the earth's surface and is particularly significant in the creation of volcanoes. Hot springs and geysers bring warm water to the surface in some cases. This heat originates from the magma heating the geofluid water contained under the surface [2]. It has long been a widespread alternative energy source because it is a relatively clean and renewable resource. It is generated when the transportation of heat energy available underground is carried out by heated water as it travels through heated rocks or magma bodies at distances ranging from some hundred meters to some kilometers. Borewells are drilled to bring the water to the earth's surface as steam or heated waters. The water is natural groundwater that leaks down via cracks and fractures. Borewells are used to inject the water down from the surface to introduce it artificially, in some cases. Geothermal energy exploitation on a commercial level has only recently become economically feasible because of the fluctuating crude oil prices, even though geothermal energy extraction and transmission are capital intensive [4].
Depending on its characteristics, geothermal energy may be used to produce clean power or be used for heating and cooling if the temperature is suitable [12].

1.2.1 Geothermal powerplants

The operation of geothermal power plants is analogous to that of fossil fuel and nuclear power plants with different heat sources i.e., geothermal energy in this case. Through a series of wells, hot water or steam is drawn from the earth and the geothermal fluid exchanges heat with the working fluid in the evaporator which is pressurized and allowed to expand in the turbine and provides rotational energy to the turbine to move the turbine blades on the shaft. To generate the electric current, the rotational energy generated by the shaft's blades by the expansion of working fluid in the turbine is utilized to spin magnets inside a huge coil. The expander and generator, which convert geothermal energy to electric energy, are critical components of the geothermal system. A reinjection well is used to reintroduce water into the system [13].

1.3 Working fluids

The working of the Organic Rankine Cycle (ORC) is identical to that of the Steam Rankine Cycle except that the organic fluids are utilized as working fluids in ORC systems instead of water which is used in Steam Rankine Cycles. As a result, picking the right operating fluid is crucial. The working fluid in the ORC process should accomplish higher thermal efficiency and maximize the use of the heat source(s) that are already available. Furthermore, the working fluid must be ecologically friendly, low-cost, and meet all safety requirements. For the selection of a suitable working fluid for ORC, the temperature of the heat source must be considered first.

The critical points of organic fluids determine their chemical stability. In particular, the working fluid may start to degrade when heated past its critical temperature. Particularly when the fluids are heated above their critical temperatures, they may begin to degrade. The selected working fluids have high critical temperatures and high boiling points. Due to the high critical and boiling temperature of the working fluid, it enters the turbine at a relatively high temperature and expands to lower condenser pressure. As a consequence, the turbine may obtain a greater enthalpy difference and more power output.

1.4 Problem Identification

Present fossil fuel-based technologies are harmful to the environment emitting harmful gases to the environment. Also, the prices of fossil fuels are increasing in the international market making fossil-based power production expensive. Conventional energy sources require higher energy input to operate. A geothermal source is a natural source of energy that is a low-temperature energy resource and can be operated using ORCs to fulfill the electricity demands of the world. ORCs are a viable choice for low-temperature geothermal heat sources as these cycles require low temperatures to operate. Earth can act as a heat sink so there is no need to deploy a cooling tower. These systems do not require fossil fuels, eliminating the dependency on expensive fuels. In comparison to other renewable energy sources like wind and solar, geothermal is a reliable source of energy since it is constantly accessible. Geothermal resources are being explored around the world and technologies are being created to improve these systems.

1.5 Justification of research

Geothermal energy has a lot of potential having an installed capacity of 15.6 GW worldwide.

The selection of topic for research has the following reasons:

- Utilizes low-temperature heat source
- Growth in the international market
- No GHG emissions
- Green energy resource
- Fossil fuels are not required
- Green energy
- No complex parts needed
- Cost-effective

1.6 Objectives of research

The primary objective of this research is the conversion of available geothermal energy into electrical energy.

1. To calculate the potential of low-temperature geothermal energy resources and utilize them to obtain electrical power.
2. To design and compare different configurations and find out the best possible configuration at which the maximum power output can be obtained.
3. To observe the effects of changing the condenser temperature on the power and thermal efficiency of the system.
4. To optimize the configurations of ORCs according to geothermal source temperature inlet and reinjection temperature.

1.7 Thesis outline

Chapter 1 discusses the increase in global energy demand. The chapter also emphasizes replacing existing fossil fuel-based energy production technologies with cleaner renewable energy power plants. Furthermore, the scope and objective of this work are defined.

Chapter 2 summarizes the literature review of ORC. The ORC has been compared with conventional Rankine Cycle power plants. Different topics covered in this chapter include a comparison of ORC with conventional power plants, working fluid selection, and methods to improve the performance of ORC.

Chapter 3 deals with thermodynamic modeling of the ORC in which mass and energy balance is applied to obtain the results. The chapter also includes the equations to calculate the power output and thermal efficiency of the system.

Chapter 4 includes the modeling of the system. System configurations and T-s diagrams are given in the chapter. It also explains the significance of the pinch point temperature difference on the performance of the cycle.

The results obtained by the methodology discussed in chapter 4 are represented and discussed in chapter 5.

Chapter 6 includes the conclusions of this study. Future work has also been discussed.

Summary

Geothermal energy resources and their potential have been discussed in detail in this section. Types of geothermal energy resources and comparisons with other renewable energy resources are discussed. Also, the working of geothermal power plants was discussed in detail. The working fluids for ORC were discussed in detail. The advantages, scope, and objectives of the research are also discussed at the end.

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Chapter 2: Literature Review

2.1 Comparison of ORC with Conventional Rankine Cycle

The structure of ORC systems is mostly similar to that of the conventional Rankine cycle, with a few modifications that can be both a benefit and a drawback. According to Angelino et al. [1], fluids with differing critical characteristics can be used to create configurations that are not possible with water. One of its potential applications, according to the authors, is low-temperature supercritical cycles. Quoilin et al. [2], Vankeirsbilck et al. [3], and Chen et al. [4] have published an in-depth analysis of the variations between ORC and conventional Rankine cycle systems.

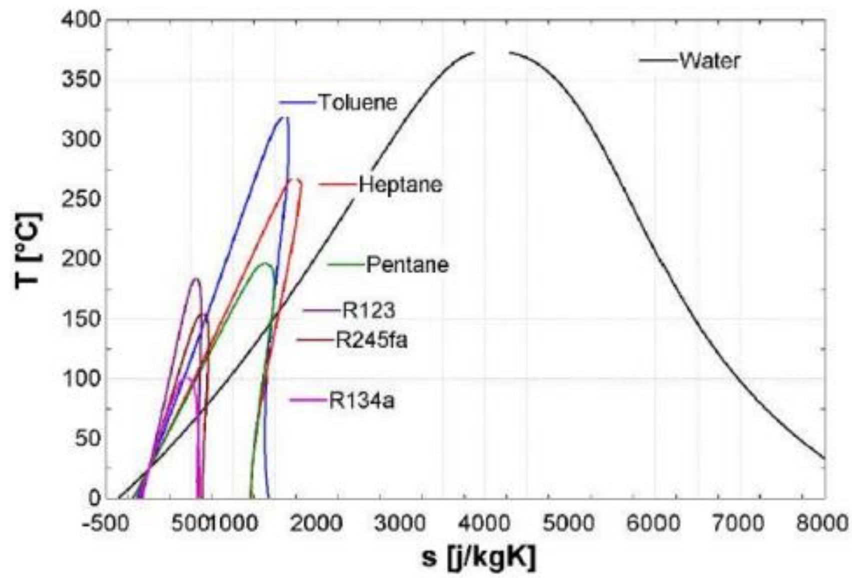

Figure 2.1 Water and various typical ORC fluids T-s diagram [2]

The entropy difference along the evaporation line is readily obvious for organic fluids, which require a substantially increased mass flow rate for a similar thermal power to water, resulting in higher pump usage. Unlike steam cycles, where the condenser pressure is usually near to vacuum, higher than atmospheric pressure is suggested in ORCs to eliminate air infiltrations [5].

Table 2.1 Comparison of ORC-Steam cycle [5]

ORC	Steam cycle
Simple architecture	High efficiency
Low evaporating pressure and temperature	Less parasitic load
No superheating	Favorable fluid characteristics
Low overall cost	
Low thermal stresses	
No makeup water treatment system	
Long lifetime of the components	

ORCs have lower thermal efficiencies than traditional steam cycles, owing to the high-temperature input limit of roughly 350 °C due to chemical instability [6] and the Carnot efficiency. While steam cycles are typically greater than 30%, ORCs are typically less than 20%. Because the ORC cycle has a lower input temperature and pressure, the enthalpy and pressure drop on the turbine are likewise lower, allowing single-stage turbines with lower rotational and tip speeds to be used. Decreased speeds provide favorable effects such as direct generator driving and lower tip speed. Table 2.1 lists the advantages of each cycle [5].

2.1.1 Characteristics of the working fluids

Water has a zero Ozone Depletion Potential (ODP), it is cheaper, non-flammable, non-toxic with a low global warming potential, and has a low viscosity. Thus, friction losses are lower, and the heat exchange coefficient of water is higher. Water treatment and a deaerator must be connected with the power plant to provide the cycle with high-purity deionized, oxygen-free water. Organic fluids are potentially hazardous and expensive as compared to water (pre-treatment costs excluded). According to Chen et al. [4], cost savings can be achieved by mass production or the utilization of low-priced hydrocarbons.

Most significantly, due to their thermal properties, R245fa and R113 were chosen for this investigation. R245fa has been demonstrated to have excellent thermal efficiency in low-temperature ORC systems in several studies (Aljundi et. al [7]). (Arribas et. al [8]).

2.2 Improving the efficiency and performance of the organic Rankine cycle

There are several ways to improve the efficiency of ORC. Altering the design of the cycle, for example, by introducing internal regeneration, operating at supercritical pressure, adding alternative evaporation pressure levels, and so on, is one option. Internal regeneration is the process by which the working fluid is heated before entering the evaporator after exiting the expander [9]. Lecompte et al. [9] offer several ORC configurations for waste heat recovery along with a summary of the current experimental findings.

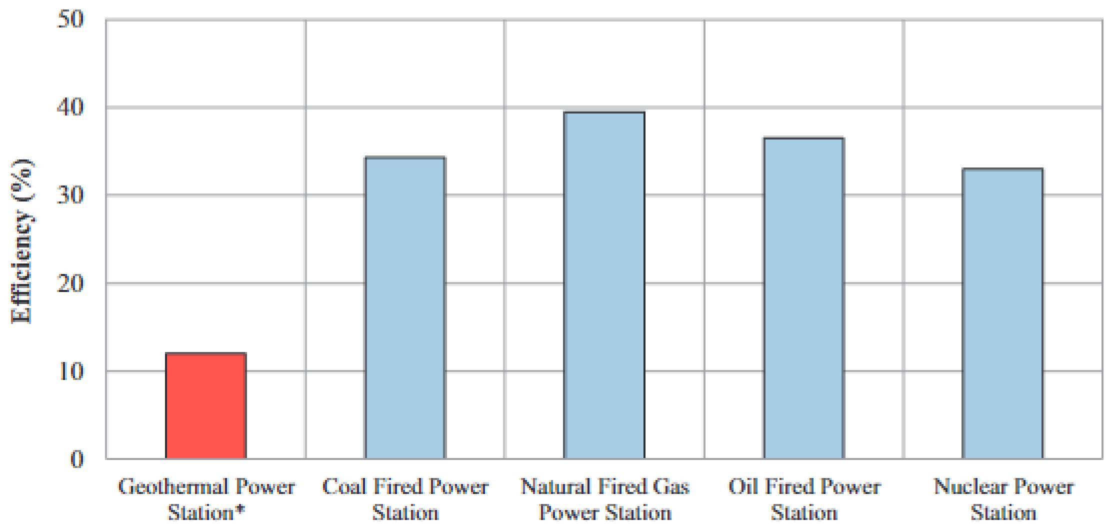

Figure 2.2 Thermal power plant efficiency [10]

Changing the working fluid can also improve efficiency, as different working fluids have distinct thermodynamic and physical properties. Working fluids can be categorized according to the shapes of T-s diagrams, and then further classified based on their thermodynamic and physical parameters, which affect cycle efficiency and performance [11]. According to Karellas et al. [12], the fluid is selected based on the cycle's process characteristics. The ideal working fluid is the one that provides the highest levels of thermal and cycle efficiency, as determined by the critical temperature and pressure as well as the boiling point at various pressures. As desirable thermodynamic and physical properties, Maizza and Maizza [13], mentioned high thermal conductivity, adequate thermal stability, high latent heat, low specific heat, low viscosity, low surface tension, low critical temperature, and moderate vapor pressure. Several properties of the working fluid such as boiling point, molecular weight, and critical pressure affect how efficiently the ORC cycle operates, according to Lee et al. [14]. However, as previously said, while choosing a working fluid, not just thermodynamic and physical parameters should be considered. Other considerations, like safety and cost, are also critical. No fluid will meet all of the criteria perfectly, thus compromises must be made [4].

The working fluid chosen may have an impact on the ORC system architectures required, such as the need for superheating. Classification of fluids can be constructed based on the vapor saturation line slope in a T-s diagram, as shown in Figure 2.1. The figure shows that for water the slope is negative, which is referred to as a wet fluid, and a certain amount of superheat is required after evaporation. Liquid droplets would form during expansion in the turbine if no superheating was used, which might damage the turbine blades and reduce the turbine's isentropic efficiency. In the case of isentropic (having slope equal to infinity) and dry (having positive slope) working fluids, no liquid phase is formed after the expansion process in the turbine, so these working fluids do not require superheating. Additionally, when utilizing a dry fluid, some superheating occurs during expansion, which can be employed for regeneration [15].

Furthermore, the temperature and pressure range in which these fluids function are critical in the development of efficient ORCs. Figure 2.1 shows that n-pentane and R245fa evaporate at substantially lower temperatures than water (at the same pressure), making them more suited for ORC applications that use low-grade heat sources.

The efficiency of the cycle can also be raised by optimizing its several parts, including the pump, expander, and heat exchangers.

2.3 Selection of Working Fluid for ORC

Since the working fluid utilized in ORC directly affects the efficiency of the system, so the selection of a working fluid is crucial. The ORC working fluid must have some properties such as it should be non-toxic, non-flammable, economical, and environment friendly, the fluid must possess the best thermodynamic characteristics at the lowest temperatures and pressures feasible, depending on the usage, the heat source, and the amount of heat to be applied [16].

There are numerous characteristics to consider while working with organic working fluids. These are the following:

- Toxicity
- Chemical stability
- Flammability
- Global warming potential
- Viscosity
- Ozone-depleting potential

Summary

A comprehensive comparison of ORC with the conventional Rankine Cycle has been made in this chapter. Methods to improve thermal efficiency have been discussed. Method to choose appropriate organic working fluid for ORC application and properties such as vaporization latent heat, specific heat, density, conductivity, and viscosity have been discussed. Desirable working fluid properties for ORC applications have been discussed.

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Chapter 3: Thermodynamic Modeling

3.1 Thermodynamic analysis

3.1.1 First Law of Thermodynamics

In the steady-state conditions, a hypothetical control volume is subjected to the following generic relations, based on the mass and energy conservation laws of thermodynamics [1].

$$\Sigma \dot{m}_{in} = \Sigma \dot{m}_{out} \tag{3.1}$$

$$\dot{Q} - \dot{W} = \Sigma \dot{m}_{out} h_{out} - \Sigma \dot{m}_{in} h_{in} \tag{3.2}$$

For each thermodynamic process, the thermodynamic modeling is based on energy and mass conservation equations [2]. Where $\dot{Q}$ is net heat input, $\dot{m}$ is mass flow rate, $\dot{W}$ represents the power generated and h represents the specific enthalpy and the subscripts *in* and *out* stand for inlet and outlet.

Table 3.1 Geothermal source and cycle properties for ORC

Factors	Value	Unit
Geothermal source inlet temperature	150, 168, 180	°C
Source pressure	101.25	kPa
Pump isentropic efficiency	0.80	%
Turbine isentropic efficiency	0.85	%

3.1.2 Turbine

In the expansion process, the turbine of the ORC transforms the energy of the working fluid into work output. Using the equation given below, the corresponding work output of the turbine can be calculated.

$$W_T = \dot{m}(h_1 - h_2) = \dot{m}(h_1 - h_{2s})\eta_T \tag{3.3}$$

where the particular enthalpies of the turbine at its intake and outflow are represented by h_1 and h_2, respectively. The specific enthalpy following an isentropic expansion at the turbine output is h_{2s}. η_T is the turbine's isentropic efficiency, which signifies the irreversibility of expansion. The isentropic efficiency of the turbine was set to 85%.

3.1.3 Pump

The working fluid exits the condenser in liquid form and enters the pump after the condenser, in which the pressure of the working fluid rises. The isentropic efficiency of the pump is taken as 80%, which is added to make the model more accurate. The relation below may be used to determine how much power the pump uses.

$$W_P = \dot{m}(h_3 - h_4) = \dot{m}(h_{3s} - h_4)/\eta_p \tag{3.4}$$

Where the specific enthalpies of the pump at the intake and outflow are represented by h_3 and h_4, respectively. η_P is the pump's isentropic efficiency.

3.1.4 Evaporator

The temperature at the outlet of the evaporator is the turbine inlet temperature. By applying the energy conservation principle on the evaporator, the following relation, can be used to calculate the mass flow rate of the working fluid.

$$\dot{m} = \frac{c_{ph}\dot{m}_h(T_{hi} - T_{e-p} - \Delta T_e)}{h_4 - h_{e-p}} \tag{3.5}$$

The specific heat capacity of the geothermal heat source is denoted by the symbol c_{ph}, $\dot{m}_h$ denotes the mass flow rate of the geothermal heat source, while T_{hi} denotes the entrance temperature of the geothermal heat source. The pinch point temperature difference is denoted by ΔT_e. The temperature and enthalpy of the working fluid at the pinch point are indicated, respectively, by the symbols T_{e-p} and h_{e-p}.

3.1.5 Condenser

Working fluid at the exit of the turbine is in the vapor phase after which it enters the condenser. It is condensed in the condenser by cooling water. The following equation represents the condensation heat of a basic ORC

$$Q_{cond} = \dot{m}(h_2 - h_3) \tag{3.6}$$

3.1.6 Recuperator

The heat at the exit of the turbine is being rejected into the environment in baseline ORC. A recuperator is introduced in the system to recover the heat of the vapor at the exit of the turbine. The pressure of working fluid rises as it leaves the pump, it enters the recuperator, and is preheated in the recuperator before it enters the evaporator. The involved heat transfer at the recuperator is calculated by the equation.

$$\mathrm{Q_{rec}} = \dot{\mathrm{m}}(h_8 - \mathrm{h}_2) \tag{3.7}$$

Where Q_{rec} represents the quantity of recuperative heat.

3.1.7 Pre-heater

Heat is added to the working fluid in the pre-heater before it enters the evaporator, which is q_{pre} during the process 4-7 to pre-heat the working fluid to saturated liquid.

$$\mathrm{q_{pre}} = \dot{\mathrm{m}}(\mathrm{h}_7 - \mathrm{h}_4) \tag{3.8}$$

3.1.8 Cycle performance

The net power output of the system is determined by the following equation, which is based on energy analysis of the system's essential components.

$$\mathrm{W_{net}} = \mathrm{W_T} - \mathrm{W_P} \tag{3.9}$$

The thermodynamic cycle efficiency of the system is obtained by the equation given below.

$$\eta_{\mathrm{th}} = \frac{\mathrm{W_{net}}}{\mathrm{Q_{in}}} \tag{3.10}$$

Summary

This chapter summarizes the equations involved in the modeling of three configurations of ORC. The mass and energy balance equations were explained and applied to the equipment for calculations involved in the system. The mathematical model to calculate power output, thermal efficiency, mass flow rate, and pump work is also given in this chapter.

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Chapter 4: System Description

Cycle-Tempo is used to develop and simulate the proposed system, considering the following assumptions and boundary conditions:

4.1 Assumptions

- Pressure drop is neglected in the heat exchangers and network of pipes.
- Steady-state operating conditions were assumed.
- Neglecting the heat losses in the equipment.
- Isentropic efficiencies of turbine and pump are given.
- The working fluid at the outlet of the condenser is in the saturated liquid state.

4.1.1 Efficiencies

- Isentropic efficiency of turbine η_T: 0.85
- Isentropic efficiency of the pump η_p: 0.80
- Mechanical efficiency of electric generator η_{elect}: 0.99

4.1.2 Design temperatures

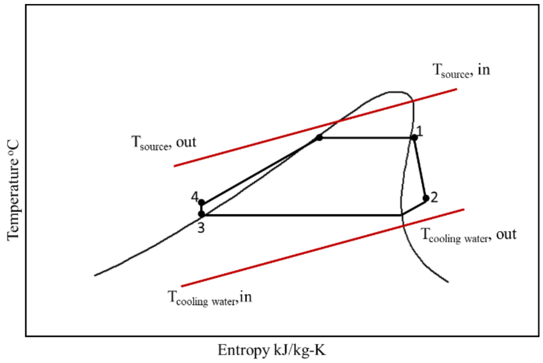
- Geothermal water inlet temperature T_{source}
- The minimum temperature difference in the evaporator between source and working fluid PPTD≥5 °C

4.2 System configuration

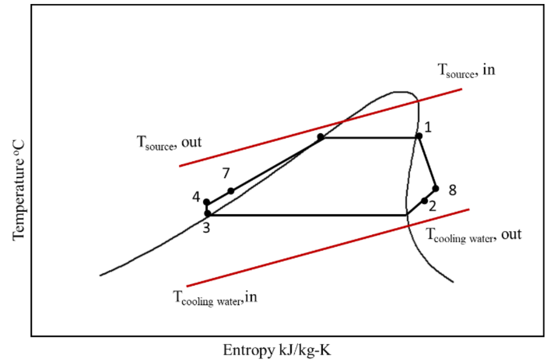
An evaporator, expander, condenser, and pump make up a baseline ORC. A recuperator is introduced in the second configuration to pre-heat the working fluid after it exits the pump and enters the evaporator.

Three configurations of ORC are considered in this study baseline ORC, recuperative ORC, and ORC with pre-heater. The corresponding configurations are given in Figure

4.2 (a), (b), and (c). Thermodynamic property changes for baseline ORC, recuperative ORC, and ORC with pre-heater are given in Figure 4.1 (a), (b) and (c) below.



(a)



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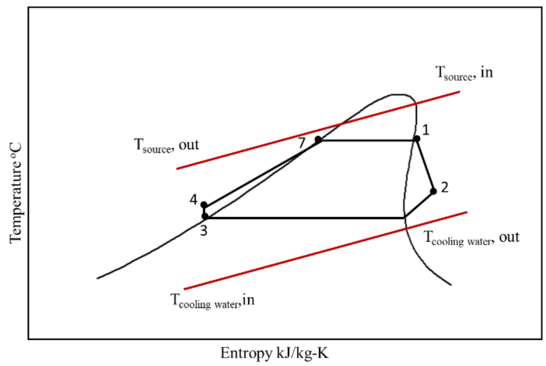


(c)

Figure 4.1 T-s Diagram of (a) baseline ORC (b) recuperative ORC (c) ORC with pre-heater

4.2.1 Considerations of expander

A vital part of the ORC is its expander, the efficiency of a volumetric expander or a turbine can be employed in the thermodynamic model. The ORC systems with an output capacity of more than 250,000 W cycles are analyzed using the turbine, and the isentropic efficiency is taken as 85% [1]. A volumetric expander will be used in a system with a capacity of less than 250kW; isentropic efficiencies of both low 35% and high 70% are considered for volumetric expanders [2]. There is a zone of overlap between 250kW and 1MW where either expander can be utilized [3].

4.3 Cycle design

Thermodynamic software to determine the thermal characteristics of the fluid at all of the states of ORC is needed for the parts that follow.

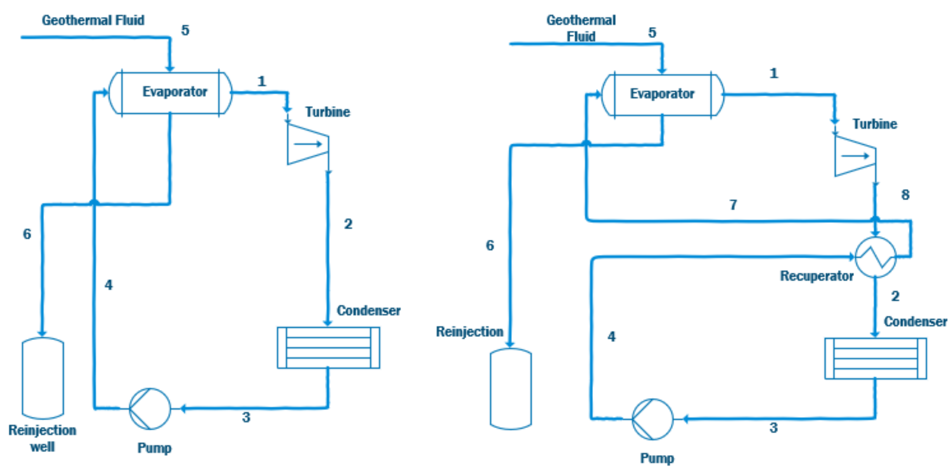
The list of software utilized is as follows:

- Cycle-Tempo
- RefProp
- (EES) Engineering Equation Solver

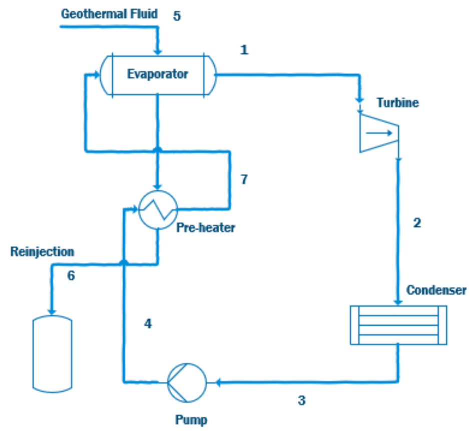
Alternatively, to complete the cycle analysis, thermal property software such as RefProp may be used in combination with Excel.

4.3.1 The modeling of ORC

The thermodynamic model of an ORC depicts the energy balance among a geothermal resource fluid, a working fluid, and a cooling fluid. The energy exchange equations for the ORC configurations are listed in Table 4.1 and the ORC's critical state positions are represented in T-s diagrams in Figure 4.1. To define the energy balances at specific state points, characteristics of the working fluid like temperature, pressure, and quality must be known at each state point.



(a) (b)



(c)

Figure 4.2 Process flow diagram of (a) Baseline ORC (b) Recuperative ORC (c) ORC with pre-heater

Table 4.1 Energy Balance Equations at Different State Points of ORC

Equipment	State at the input	State at the output	Transfer of Energy	The energy balance	Isentropic efficiency
Turbine	1 Saturated vapor	2 Saturated vapor	W_{12}	$W_{12} = \dot{m}(h_1 - h_2)$	$\eta_e = \frac{h_1 - h_2}{h_1 - h_{2s}}$
Condenser	2 Saturated vapor	3 Liquid	Q_{23}	$Q_{23} = \dot{m}(h_2 - h_3)$	$Q_{61} = U_c A_c \Delta T_{lm23}$
Pump	3 Liquid	4 Subcooled liquid	W_{34}	$W_{34} = \dot{m}(h_3 - h_4)$	$\eta_p = \frac{h_{4s} - h_3}{h_4 - h_3}$
Evaporator	4 Saturated liquid	1 Saturated vapor	Q_{14}	$Q_{14} = \dot{m}(h_1 - h_4)$	$Q_{14} = U_v A_v \Delta T_{lm14}$
Recuperator (Turbine outlet)	8 Saturated vapor	2 Saturated vapor	Q_{82}	$Q_{82} = \dot{m}(h_8 - h_2)$	$Q_{82} = U_r A_r \Delta T_{lm82}$
Recuperator (Pump outlet)	4 Subcooled liquid	7 Subcooled liquid	Q_{47}	$Q_{47} = \dot{m}(h_4 - h_7)$ $Q_{47} = Q_{82}$	$Q_{47} = U_r A_r \Delta T_{lm47}$ $Q_{47} = Q_{82}$
Net Power			$W_{12} = W_{34} + Q_{47} + Q_{14} - Q_{82} - Q_{23}$		

Notes:

* If the ORC does not need a recuperator, the recuperator portion is omitted, and the expander and pump exit to the condenser and evaporator, respectively.

4.3.2 Geothermal parameters

Firstly, geothermal resource conditions are defined. The conditions of the geothermal resource that are known, are the temperature at the inlet, pressure, and the mass flow rate for geothermal fluid. The output temperature also known as the re-injection temperature of the geothermal fluid is 80 °C. This will be adjusted to find the working fluid's ideal output temperature. It will also be necessary to determine the allowed geothermal flow.

T_{ins} Temperature of the geothermal fluid ORC (°C)

T_{outs} Temperature of the geothermal fluid at the outlet of the ORC (°C)

$\dot{m}_s$ Mass flow rate of geothermal fluid (kg/s)

4.3.3 Condenser condition

The condenser may be operated using water or air, the condensing conditions change depending on these two aspects. The inlet temperature of the condenser determines the output state of the expander, which is set by ambient circumstances for a condenser operated by air. For ORC design, the design condenser conditions are crucial [4]. A floating condenser condition is explored [5].

4.3.4 Approach Temperature and Temperature Difference at Pinch Point

The temperature difference at the exit of the working fluid and the temperature at the inlet of the cooling fluid in the condenser is termed approach temperature. Furthermore, the approach temperature can be taken from 8 to 14 °C [6].

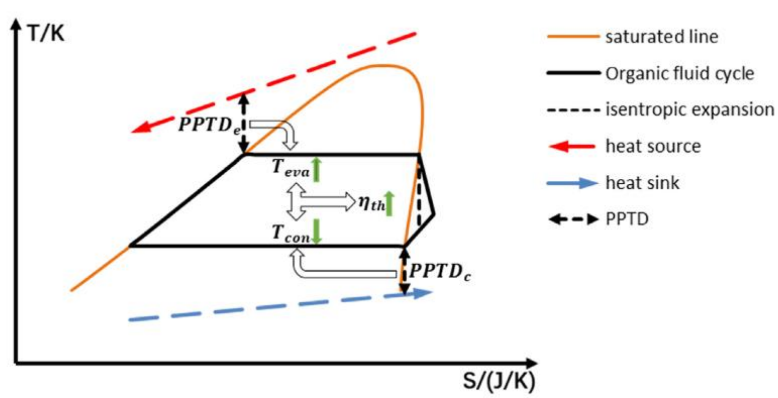


Figure 4.3 Efficiency change with moving pinch point [7]

Pinch points of the heat exchangers of organic Rankine cycles affect the heat absorption by the working fluid as well as the thermal efficiency of the cycle [7]. The condenser outlet condition is defined by the state of the cooling fluid approach inlet and the approach temperature. A minimum pinch point of 4 °C is recommended with a typical pinch point of 15 °C for the phase change heat exchangers [8]. The pinch point of the evaporator varies from 5 °C to 15 °C, whereas the condenser pinch point varies with the change in the condenser temperature. There might be a lesser pinch in the condenser since the condensing state has a greater influence on the scheme.

4.3.5 Effect of evaporator pinch

One of the key factors that affect the evaporation pressure is the evaporator pinch point. Evaporator pressure and temperature are defined by pinch point. As heat transfer in the heat exchanger increases with decreasing pinch point temperature, the temperature of the geothermal fluid may further reduce as a result. The size of the heat exchanger is significantly influenced by the pinch point.

The work performed by the ORC is reduced as the pinch point temperature raises, which also decreases the area and expense of the heat exchanger needed [3].

4.3.6 Isentropic efficiency

To determine the system's net power, turbine and pump efficiencies are required. A turbine's efficiency is set to 85 percent.

4.3.7 Pump inlet state

The thermodynamic properties can be determined using information from the pump inlet. In this instance, the working fluid is in a subcooled liquid state. The inlet temperature of the cooling fluid and the approach temperature are used to calculate the fluid temperature. To eliminate cavitation in the feed pump, a 5 °C subcooling is required. Therefore, the temperature of the input of the pump is 5 °C higher than the saturation conditions; this determines the condenser's pressure and expander's outlet temperature.

4.3.8 Outlet conditions of the evaporator

The evaporation temperature can be calculated by an iterative process that starts with an estimated temperature of the evaporator and the presumption that there is no variation in the feed pump temperature. After the identification of all the initial state points, this will be reassessed. To determine the ideal evaporation temperature, a simple iteration is employed.

4.3.9 Iterative evaporator calculation

To estimate how much heat will be required for the working fluid in the evaporation process, the evaporator temperature is used. For the estimation of the temperature of the geothermal pinch point, a temperature heat analysis is used.

The evaporation temperature T_{evap} is used to set the evaporator pressure p_{evap}. The amount of heat needed to get the working fluid to evaporate is calculated. Heat is added to the system in the evaporator and this heating process that occurs in the evaporator is the heat input to the system h_{in}. The pinch point temperature in the evaporator is estimated using the relation given below [9].

$$T_{pinch} = \frac{T_3 - T_4}{\Delta H_{25}}(\Delta H_{22'} - \Delta H_{24}) + T_3$$

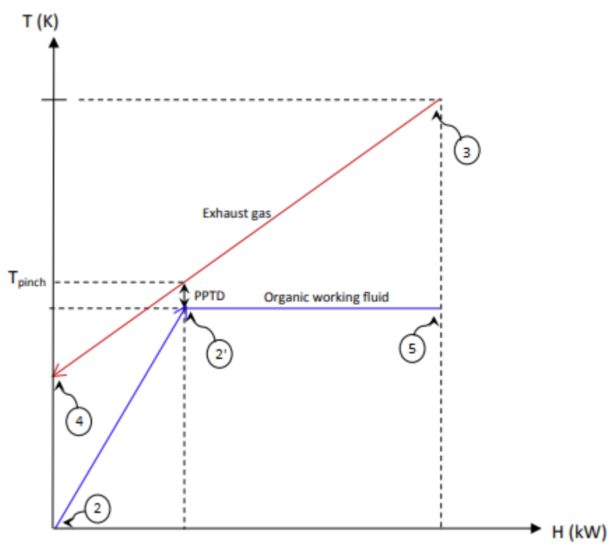


Figure 4.4 T-Q diagram used for the analysis of pinch-point in the evaporator [9]

After the determination of the evaporation output conditions, the equation for the isentropic efficiency of the expander is utilized to get the outlet enthalpy of the expander. At this stage, every state of the system is recognized.

The pressure at the exit of the expander is the same as the pump intake pressure. The specific work output of the ORC and the enthalpy at the expander outlet are calculated using the turbine efficiency equation.

$$\eta_e = \frac{h_5 - h_6}{h_5 - h_{6s}}$$

4.3.10 Pump outlet

After the determination of the high-pressure side of the ORC, the isentropic efficiency of the pump will define the output state of the pump. This will slightly alter the evaporator conditions.

$$\eta_p = \frac{h_{2s} - h_1}{h_2 - h_1}$$

4.3.11 Recuperator approximation

Heat is introduced to the system by the recuperator if a recuperator is utilized. The calculations of the evaporator are repeated with the additional heat by the recuperator. The hot fluid enters the recuperator at the outlet of the turbine and the cold fluid enters the recuperator from the outlet of the pump. The working fluid in the vapor form after the turbine losses the enthalpy to the fluid after the pump which should be equal if there is no heat wasted and the effectiveness of the recuperator is 100 percent. However, the recuperators are considered less than 100 percent effective, and the literature suggests that when investigating recuperator advantages, 80 percent effectiveness should be used[10].

$$T_{out,R,Gas} - PPTD_{recuperator} = T_{in,R,Fluid}$$

The pinch point difference of the recuperator is the difference in temperature b/w the recuperator gases exit and the inlet of cooling fluid.

$$T_{out,R,Fluid} - PPTD_{recuperator} = T_{in,R,Gas}$$

The temperature difference between the inlet of the gas and the exit of the cooling fluid is the same.

$$h_{out,R,Gas} = h_{in,R,Fluid}$$

The hot working fluid transfers the heat to the cold working fluid [3]. As a result, it lowers the amount of heat needed in the evaporator and improves the thermal efficiency of ORC. On the other hand, a recuperator is only worthwhile if the reinjection temperature is strictly limited [11]. As a result, a recuperator is unlikely to be cost-effective for a low-temperature ORC.

4.4 Selection of Working Fluid

The three kinds of working fluids taken into consideration for ORC are wet, isentropic, and dry. Dry working fluids are those working fluids having a positive slope and exiting in a superheated state after expansion in the turbine. Isentropic fluids have an infinite slope and the slope of the wet fluids is negative [12]. In this study, the working fluids R113 and R245fa have been considered. The thermodynamic properties of these working fluids are given below.

Table 4.2 Thermodynamic properties of selected working fluids

Working fluid	T_{cri} (°C)	P_{cri} (MPa)	Molar Mass (kg/kmol)	ODP	GWP	Fluid type
R113	214.06	3.392	187.38	0.9	6130	Isentropic
R245fa	154.05	3.640	134.05	0	950	Dry

4.5 Altering set-points

The different settings such as $T_{in,exp,}$ and $p_{in,exp}$, except for expander and refrigerant pump rotational speed, cannot be altered independently and must be obtained by altering other parameters.

4.5.1 Pressure ratio

The pressure ratio across the expander is proportional to the pressures before and after it. Changing the expander's inlet pressure or outlet pressure can so change the pressure ratio. Condensing pressure is what determines the latter. As a result, changing the condenser parameters changes the PR. The mass flow rate of the cooling liquid is the only changeable parameter in the condenser. Condensing pressure will drop as the cooling mass flow rate increases, and vice versa, if the heat balance between the refrigerant and cooling liquid is maintained.

$$\dot{m}_{cf}.c_{p,cf}\left(T_{out,cd,cf} - T_{in,cd,cf}\right) = \dot{m}_{r}(h_{in,cd,r}\left(p_{cd,r}\right) - h_{out,cd,r}\left(p_{cd,r}\right))$$

4.5.2 Expander inlet temperature

The expander inlet temperature can be adjusted according to the geothermal heat source. A specific expander input temperature can be established by altering the mass flow rate or inlet temperature of the heating liquid. This is due to a heat balance in the evaporator.

4.5.3 Expander inlet pressure

The heat source conditions are modified to adjust the inlet pressure and temperature of the expander. At the set values of the inlet temperature of the expander, the pressure values at the saturation conditions are obtained from RefProp.

Summary

The methodology of the research has been discussed in this section in which different assumptions are made. The configurations of ORCs have been drawn and explained accordingly. Different settings such as evaporator pressure and temperature were altered to obtain the results using Cycle-Tempo and RefProp. The performance of the evaporator is greatly influenced by the pinch point temperature difference (PPTD), a significant parameter.

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Chapter 5: Results and Discussion

Three ORC configurations are optimized and compared namely baseline ORC, recuperative ORC, and ORC with pre-heater for optimal thermodynamic parameters. Each configuration is optimized and developed in Cycle-Tempo software. The thermodynamic performance of each ORC configuration is analyzed for the working fluids R113 and R245fa.

5.1 Baseline ORC configuration

The thermal efficiencies, power output, and mass flow rates to produce a unit power output of different configurations of ORCs for two working fluids (R113 & R245fa) have been assessed at various condenser and evaporator temperatures and pressures.

5.1.1 Baseline ORC at varying condenser temperatures

Results of baseline ORC configuration indicate that the working fluid R245fa outperforms the working fluid R113 in terms of thermal efficiency and power output. The baseline ORC produces 2191.81 kW of power output with working fluid R113 and 2443.03 kW with working fluid R245fa.

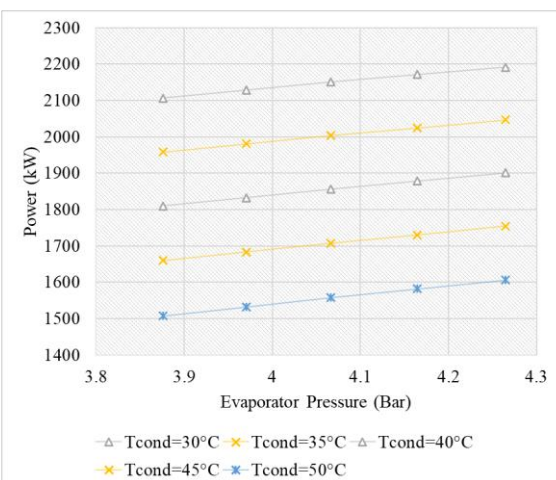

Figure 5.1 Effect of evaporator pressure on power output of baseline ORC for R113

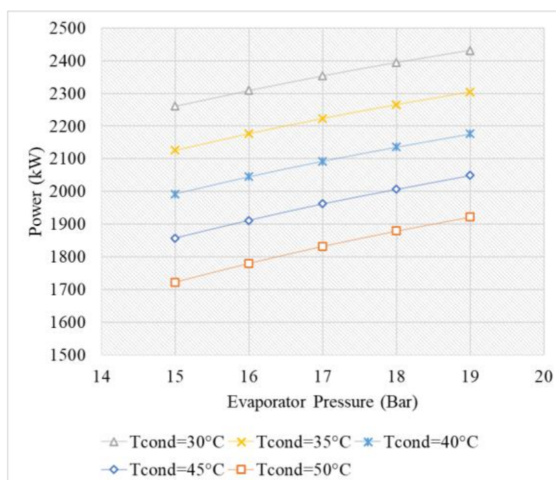

Figure 5.2 Effect of evaporator pressure on power output of baseline ORC for R245fa

Figure 5.1 presents the variation in power output in kW at different evaporator pressures and condenser temperatures obtained for the working fluid R113. Figure 5.2 presents the results obtained for variation of power at different evaporator pressures for the working fluid R245fa. It is seen that the power output increases with an increase in evaporator pressure. At the lowest condenser temperature, the power output is the highest and it decreases as the condenser temperature increases. So the maximum power outputs for both of the working fluids are obtained at the condenser temperature of 30 °C which is the lowest condenser temperature in this case.

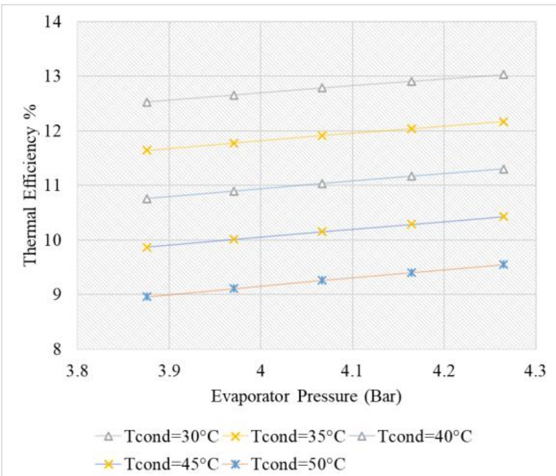

Figure 5.3 Effect of evaporator pressure on thermal efficiency of baseline ORC for R113

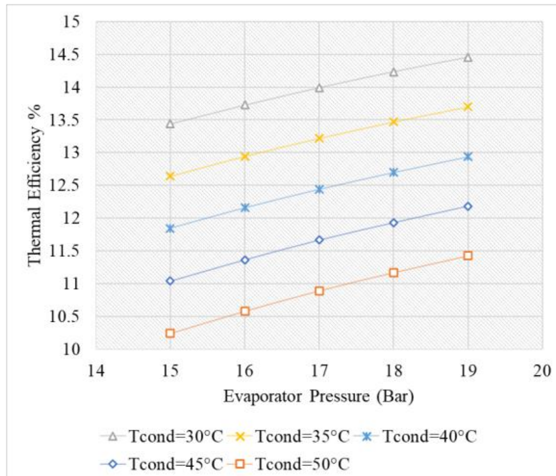

Figure 5.4 Effect of evaporator pressure on thermal efficiency of baseline ORC for R245fa

The evaporator pressure for the working fluid R113 varies from 3.87 bar to 4.26 bar and for the working fluid R245fa, it varies from 15 bar to 19 bar. At a specific evaporator temperature, the value of evaporator pressure is obtained using the software RefProp at the state points. The working fluid R245fa operates at a higher evaporator temperature as compared to the working fluid R113.

Baseline ORC achieves maximum efficiency of 13.02% with the working fluid R113 and 14.45% with R245fa. The mass flow rate to produce unit power output is higher in the case of the working fluid R113 which is 0.0648 kg/kW.s. It is 0.0451 kg/kW.s in the case of the working fluid R245fa.

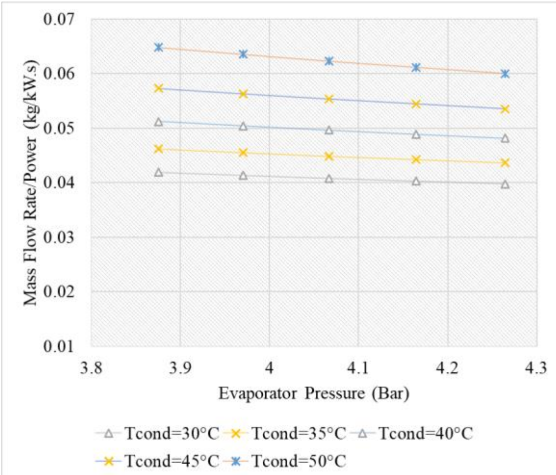

Figure 5.5 Effect of evaporator pressure on mass flow rate to produce unit power of baseline ORC for R113

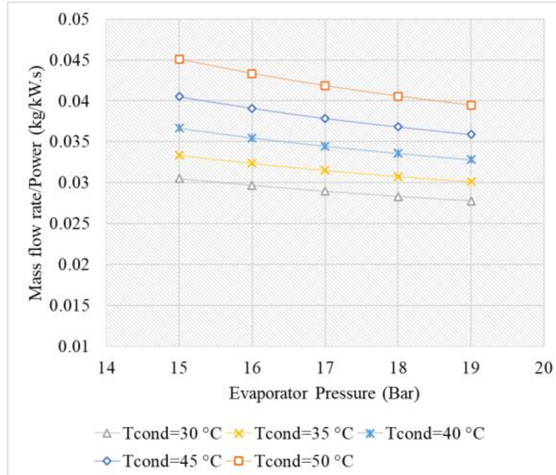

Figure 5.6 Effect of evaporator pressure on mass flow rate to produce unit power of baseline ORC for R245fa

5.1.2 Baseline ORC at source temperature of 150 ºC

Decrease in source temperature results in a decrease in power output and thermal efficiency of the system for both working fluids. Power output has decreased to 1740.58 kW for working fluid R113 and 1673.06 kW for R245fa. The cycle works at lower evaporation temperatures when the source temperature decreases as shown in the figures below.

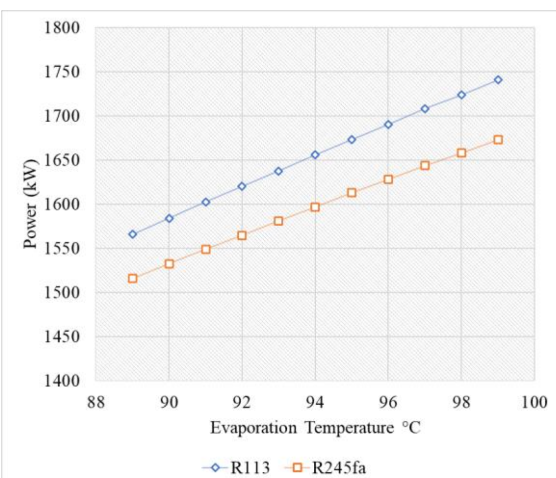

Figure 5.7 Effect of evaporation temperature on power output of baseline ORC

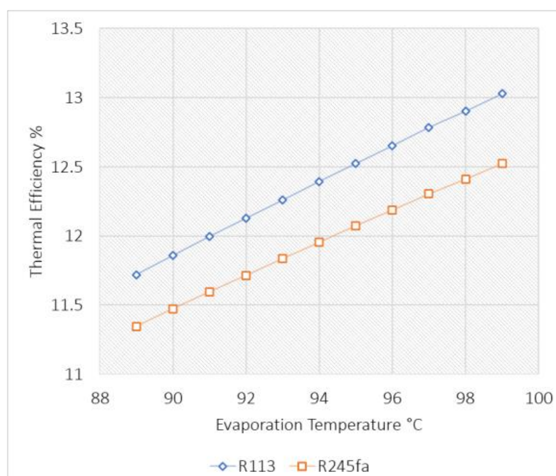

Figure 5.8 Effect of evaporation temperature on thermal efficiency of baseline ORC

Thermal efficiency decreased to 13.03 % for R113 and 12.52 % for R245fa. Mass flow rate to produce unit power output has also decreased. The highest mass flow rate for the working fluid R113 is 0.064 kg/kW.s at the highest condenser temperature and lowest evaporator temperature. As the evaporator temperature increases the mass flow rate decreases. The mass flow rate to produce unit power output also decreases with the increase in evaporator temperature. For the working fluid R245fa, the mass flow rate to produce unit power is 0.045 kg/kW.s at the evaporator temperature and pressure of 107.85 °C and 15 bar. at Figure 5.10 shows that the working fluid R245fa needs higher specific pump work as compared to the working fluid R113. R245fa operates at higher pressure as compared to R113.

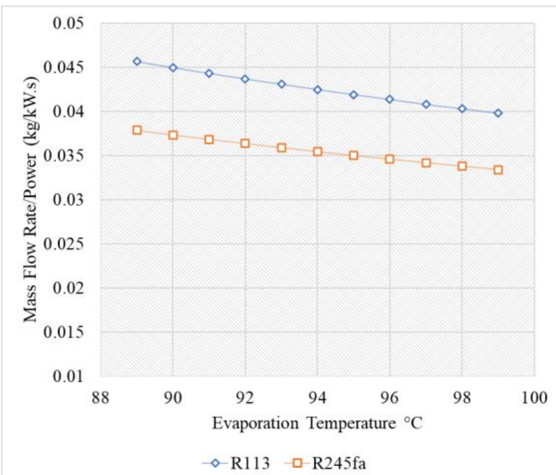

Figure 5.9 Effect of evaporation temperature on mass flow rate to produce unit power output of baseline ORC

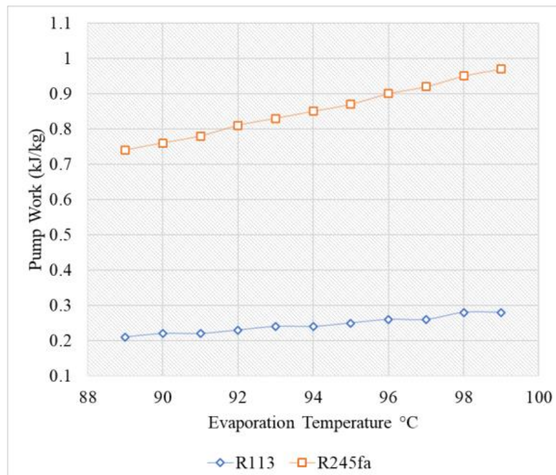

Figure 5.10 Effect of evaporation temperature on specific pump work of baseline ORC

5.1.3 Baseline ORC at source temperature of 180 °C

Power output as well as the thermal efficiency of the baseline ORC improves as the geothermal source temperature increases. The cycle operates at a higher evaporation temperature at a higher source temperature. The maximum power output increased to 2840.85 kW for working fluid R113 and 2696.7 kW for R245fa. Thermal efficiency increased to 14.84% for R113 and 14.09% for R245fa.

The mass flow rate to produce unit power output decreases as the source temperature increases. With the increase in source temperature, the evaporation temperature and pressure increase which causes an increase in specific pump work and pressure ratio for both working fluids for the same condenser pressure. The working fluid R113 operates at an evaporator pressure of 6.12 bar R245fa at 17.41 bar for this case.

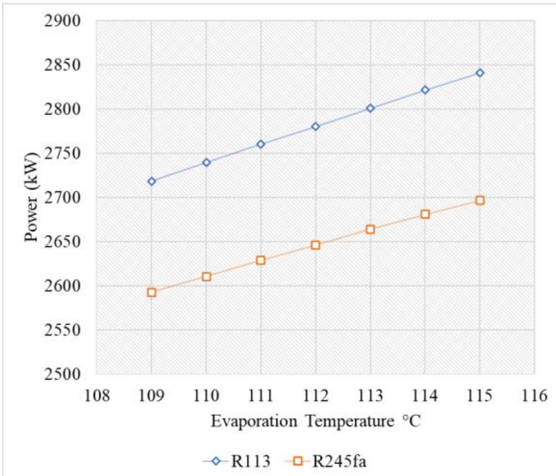

Figure 5.11 Effect of evaporation temperature on power output of baseline ORC

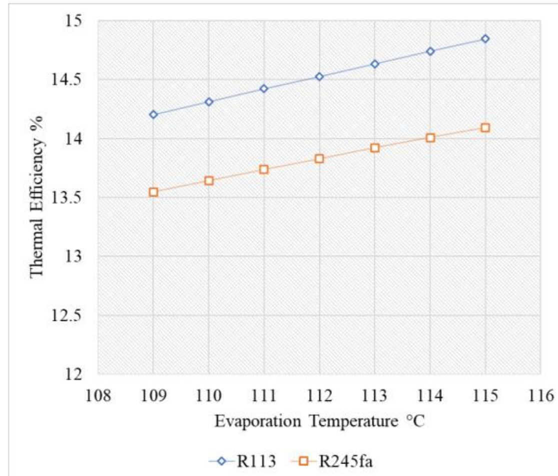

Figure 5.12 Effect of evaporation temperature on thermal efficiency of baseline ORC

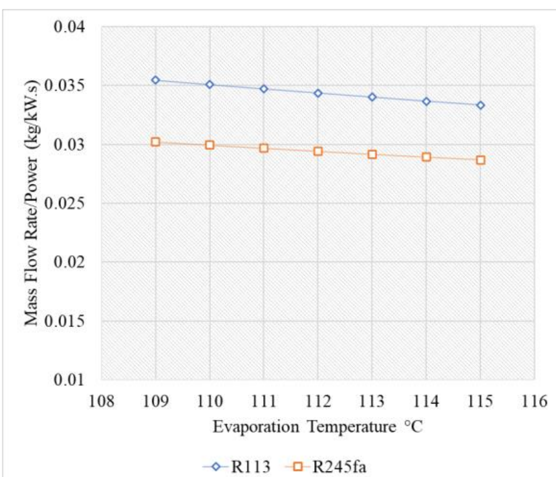

Figure 5.13 Effect of evaporation temperature on mass flow rate to produce unit power output of baseline ORC

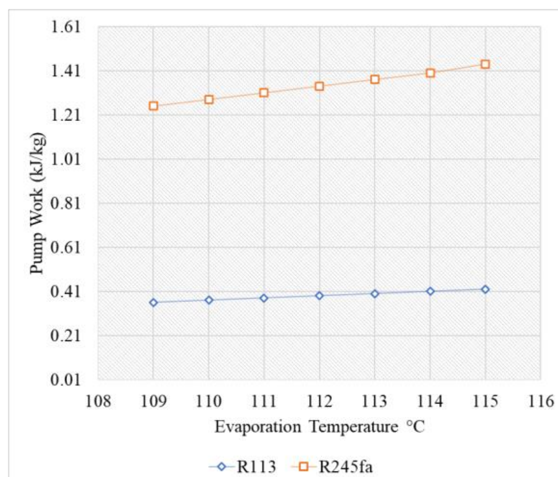

Figure 5.14 Effect of evaporation temperature on specific pump work of baseline ORC

5.2 Recuperative ORC

Recuperative ORC is a modified form of baseline ORC in which a recuperator is introduced between the turbine outlet and condenser inlet. The recuperator recovers heat from the working fluid at the turbine outlet and pre-heats the working fluid at the outlet of the pump before it enters the evaporator. The recuperator recovers the heat being rejected to the environment in the baseline ORC and adds it to the system.

5.2.1 Recuperative ORC at varying condenser temperatures

Recuperative ORC operates at lower evaporator pressure as compared to baseline ORC and produces higher power output. The results show that higher power output and thermal efficiency are obtained by recuperative ORC at lower evaporator pressure as compared to baseline ORC. A maximum power output of 2212.92 kW has been obtained with the working fluid R113 and 2579.86 kW with R245fa. R113 produced 2191.81 kW of power and R245fa produced 2443.03 kW of power output with the baseline configuration. There is a significant increase in power output for both of the working fluids in this case.

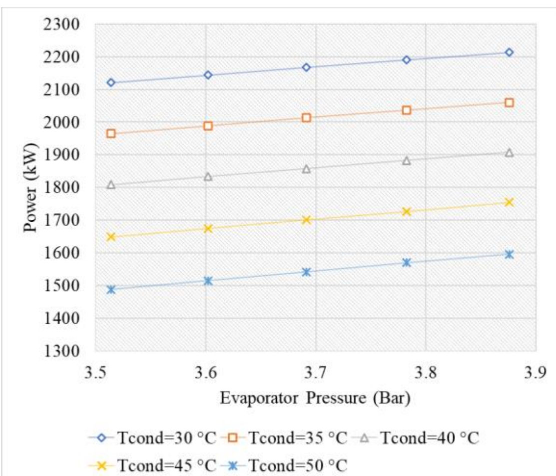

Figure 5.15 Effect of evaporator pressure on power output of recuperative ORC for the working fluid R113

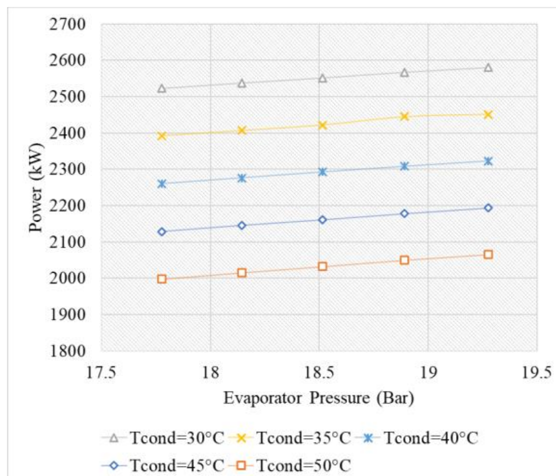

Figure 5.16 Effect of evaporator pressure on power output of recuperative ORC for the working fluid R245fa

Figure 5.15 and Figure 5.16 represent the variation in power output in kW at different evaporator pressures and condenser temperatures obtained for the working fluid R113 and R245fa respectively.

Recuperative ORC achieves maximum efficiency of 13.16% with the working fluid R113 and 14.50% with R245fa. Power output and thermal efficiency are maximum at the lowest condenser temperature which is 30 °C. The thermal efficiency of the cycle can be improved by reducing the condenser pressure and temperature. Mass flow rate to produce unit power output decreases with an increase in evaporator pressure. The results show that the mass flow rate to produce unit power increases when decreasing the efficiency of ORC. The maximum mass flow rate to produce unit power output for the working fluid R113 is 0.07 kg/kW.s and for R245fa it is 0.04 kg/kW.s.

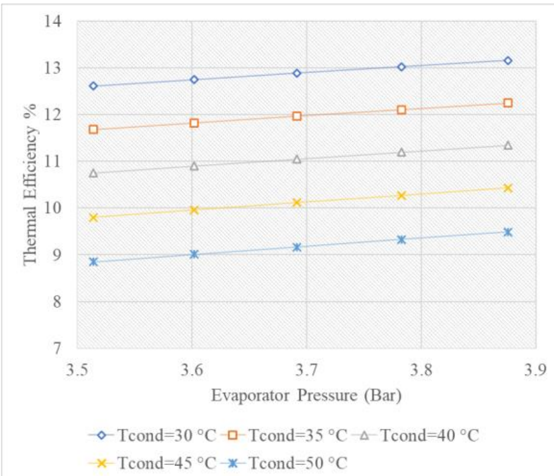

Figure 5.17 Effect of evaporator pressure on thermal efficiency of recuperative ORC for the working fluid R113

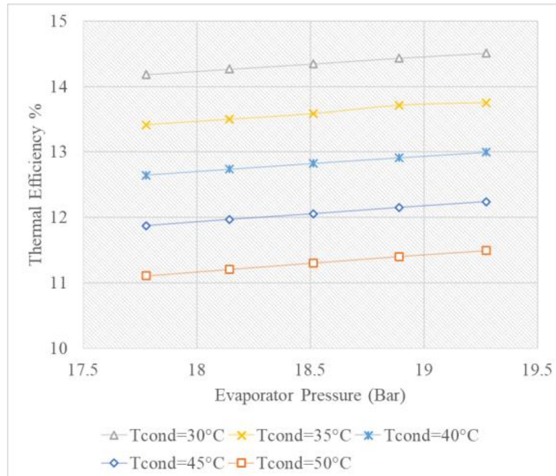

Figure 5.18 Effect of evaporator pressure on thermal efficiency of recuperative ORC for the working fluid R245fa

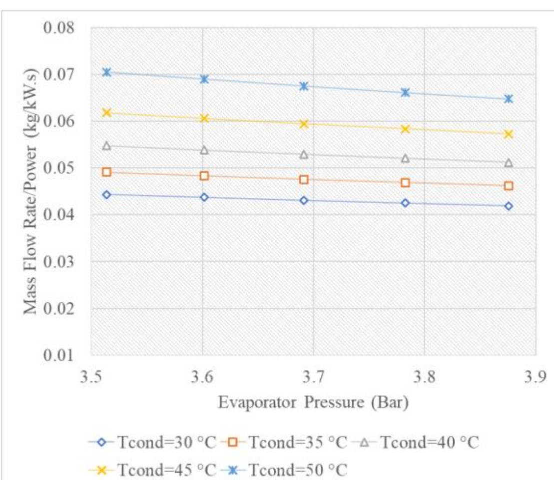

Figure 5.19 Effect of evaporator pressure on mass flow rate to produce unit power of recuperative ORC for the working fluid R113

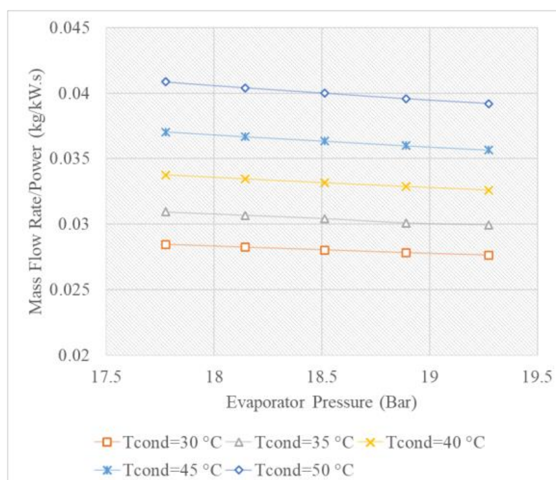

Figure 5.20 Effect of evaporator pressure on mass flow rate to produce unit power of recuperative ORC for the working fluid R245fa

5.2.2 Recuperative ORC at source temperature of 150 °C

Power output and thermal efficiency decrease because of a decrease in source temperature. Power output has decreased to 1857.25 kW for working fluid R113 and 1757.91 kW for R245fa. The cycle works at lower evaporation temperatures when the source temperature decreases.

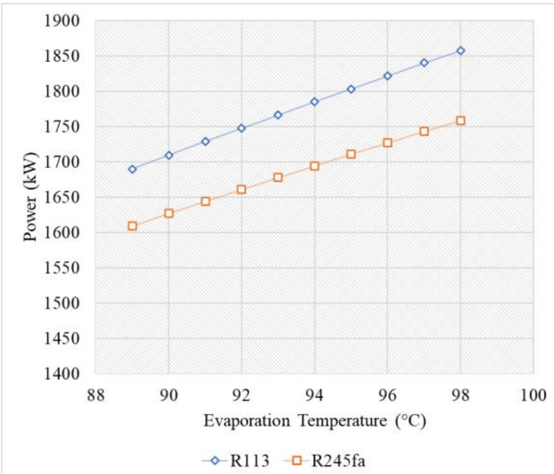

Figure 5.21 Effect of evaporation temperature on power output of recuperative ORC

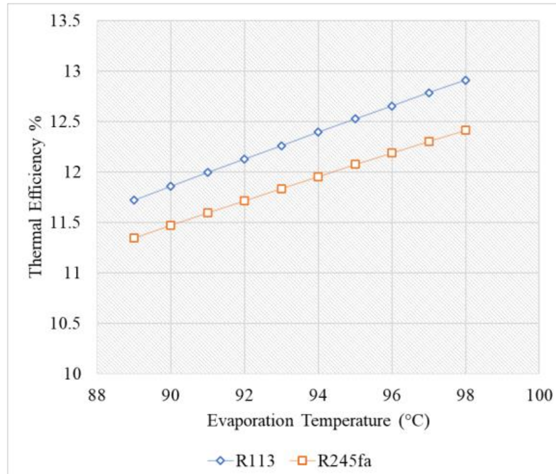

Figure 5.22 Effect of evaporation temperature on thermal efficiency of recuperative ORC

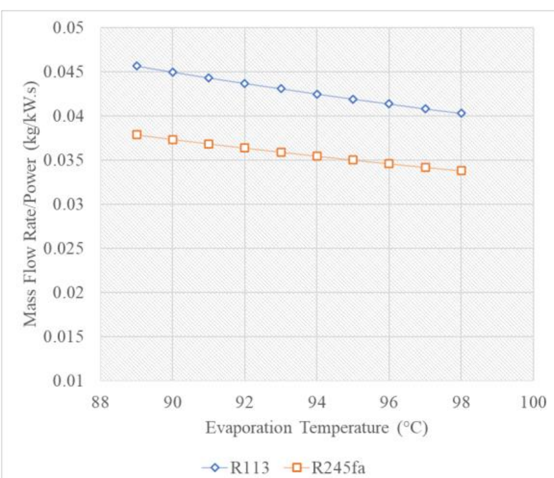

Figure 5.23 Effect of evaporation temperature on mass flow rate to produce unit power output of recuperative ORC

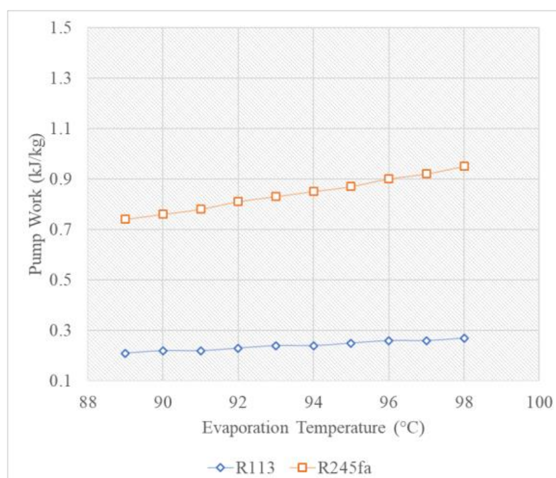

Figure 5.24 Effect of evaporation temperature on specific pump work of recuperative ORC

Thermal efficiency and mass flow rate to produce unit power decrease as the source temperature decreases. The mass flow rate to produce unit power output for the working fluid R113 is 0.045 kg/kW.s at an evaporator temperature of 89 °C and 0.038 kg/kW.s for R245fa at the same evaporator temperature. Thermal efficiency in this case dropped to 13.15 % for R113 and 14.5 % for R245fa. The cycle operates at lower evaporator temperature and pressure at lower source temperature, so the pump work decreases.

5.2.3 Recuperative ORC at source temperature of 180 °C

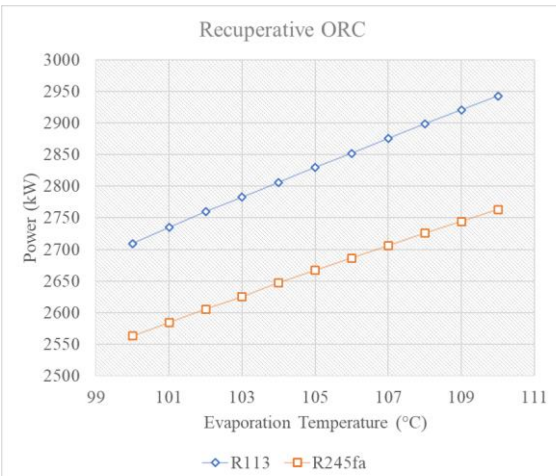

Figure 5.25 Effect of evaporation temperature on power output of recuperative ORC

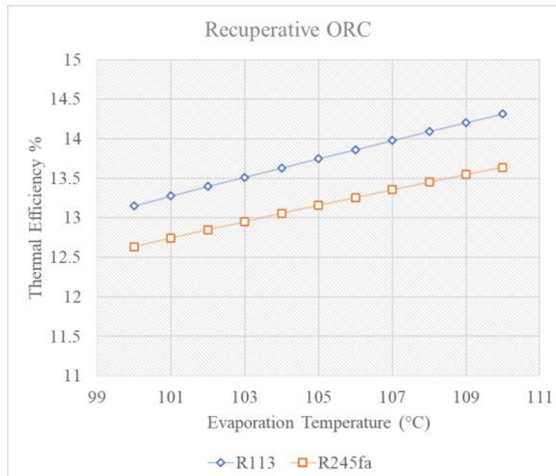

Figure 5.26 Effect of evaporation temperature on thermal efficiency of recuperative ORC

Power output and the thermal efficiency of the recuperative ORC increase as the geothermal source temperature increases. The cycle operates at a higher evaporation temperature at a higher source temperature. The power output increased to 2948.38 kW for working fluid R113 and 2763.06 kW for R245fa.

The mass flow rate to produce unit power for R113 is 0.0393 kg/kW.s and 0.033 kg/kW.s for R245fa. With the increase in evaporation temperature the mass flow rate decreases. The cycle operates at a higher evaporation temperature as compared to the previous case.

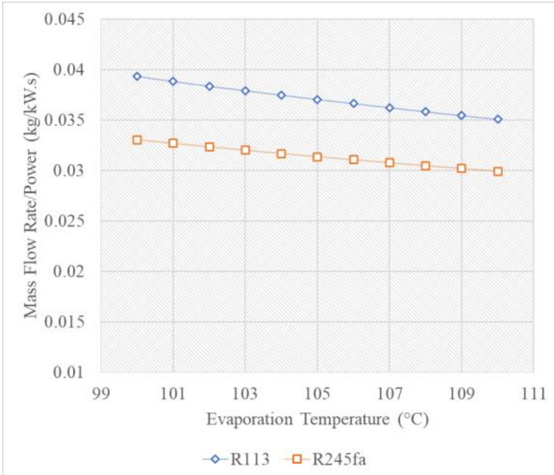

Figure 5.27 Effect of evaporation temperature on mass flow rate to produce unit power output of recuperative ORC

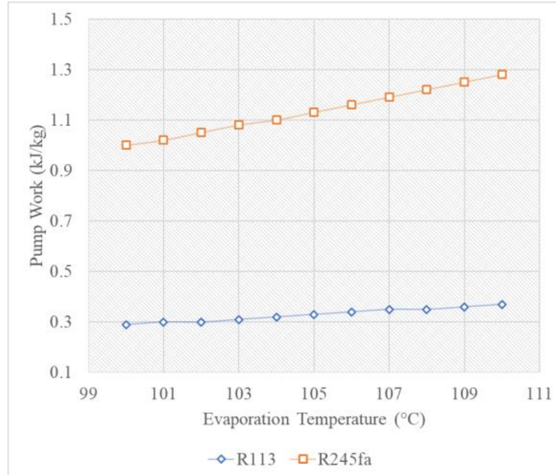

Figure 5.28 Effect of evaporation temperature on specific pump work of recuperative ORC

5.3 ORC with pre-heater

In this configuration, the geothermal source exchanges heat in the evaporator and enter the pre-heater in which the working fluid is pre-heated before entering the evaporator. This configuration produces higher power output as compared to baseline ORC but is lower than that of recuperative ORC.

5.3.1 ORC with pre-heater at varying condenser temperatures

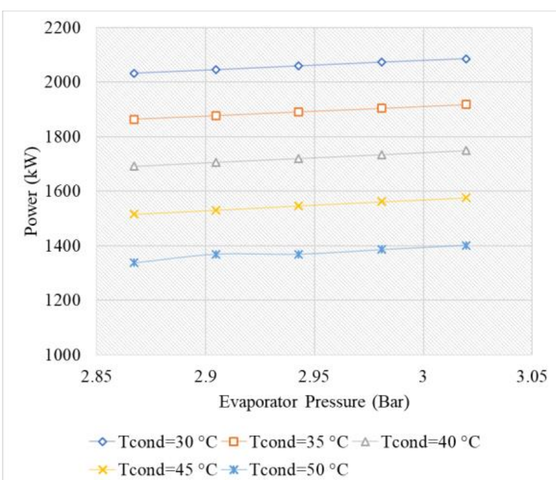

Figure 5.29 Effect of evaporator pressure on power output of ORC with pre-heater for the working fluid R113

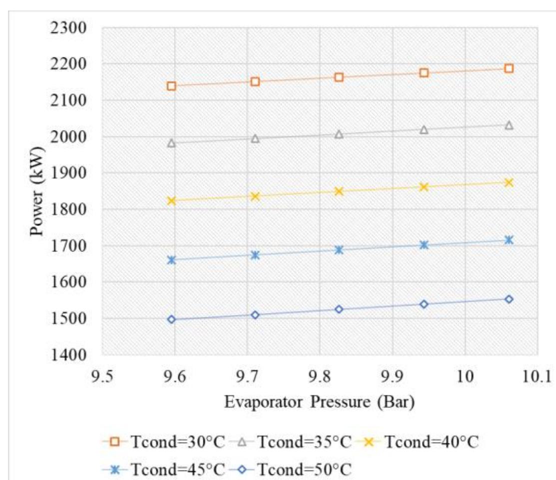

Figure 5.30 Effect of evaporator pressure on power output of ORC with pre-heater for the working fluid R245fa

It can be observed from Figure 5.29 and Figure 5.30 that the working fluid R245fa performs better than R113. Power output for R113 is 2084.92 kW and 2187.05 kW for R245fa.

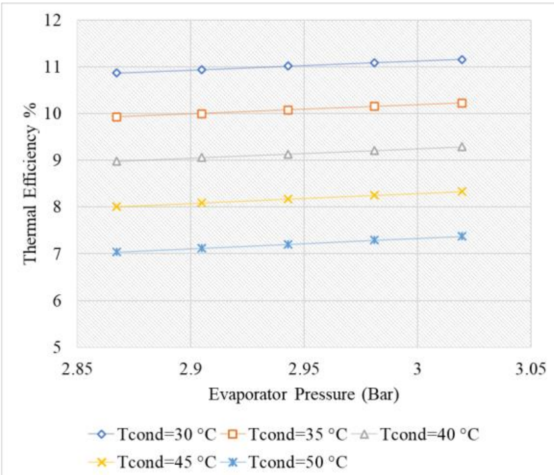

Figure 5.31 Effect of evaporator pressure on thermal efficiency of ORC with pre-heater for the working fluid R113

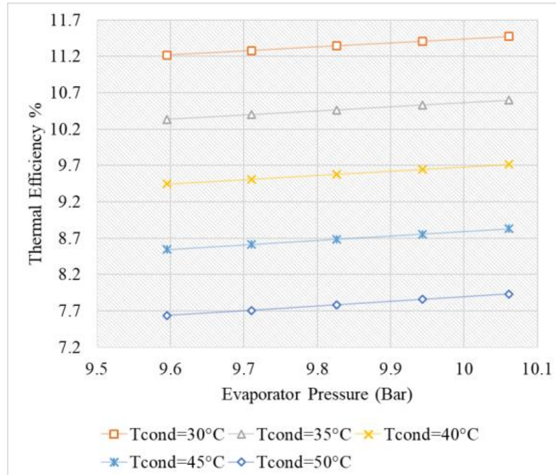

Figure 5.32 Effect of evaporator pressure on thermal efficiency of ORC with pre-heater for the working fluid R245fa

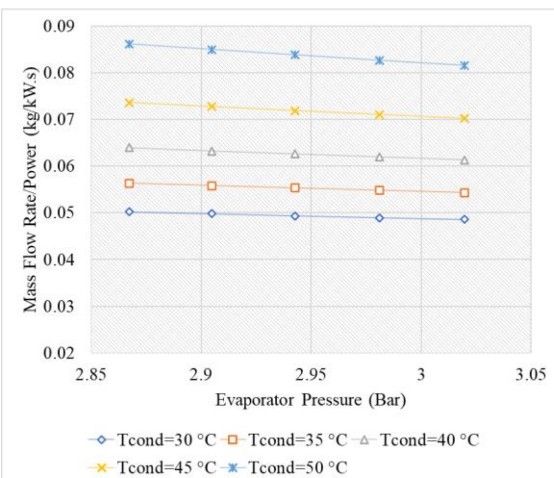

Figure 5.33 Effect of evaporator pressure on mass flow rate to produce unit power of ORC with pre-heater for the working fluid R113

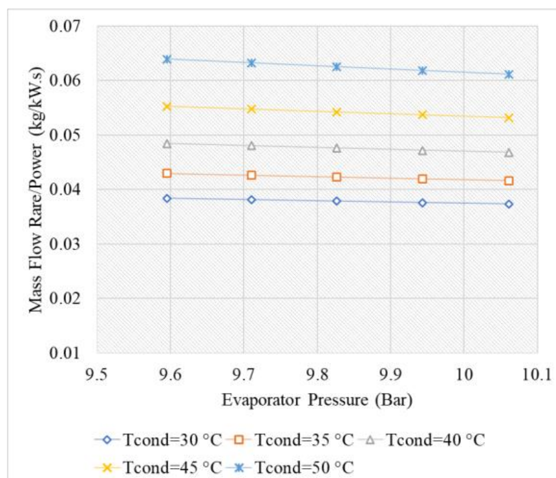

Figure 5.34 Effect of evaporator pressure on mass flow rate to produce unit power of ORC with pre-heater for the working fluid R245fa

It can be observed from the results that this configuration operates at lower evaporator pressure and temperature as compared to baseline ORC and recuperative ORC but achieves higher power output as compared to baseline ORC. A lower amount of energy is required to be added to the evaporator as the working fluid is already pre-heated. The thermal efficiency has not improved in this configuration as more energy is being added to the system in the pre-heater.

5.3.2 ORC with pre-heater at source temperature of 150 ^{o}C

The power output has decreased to 1852.13 kW for R113 and 1824.59 kW for R245fa. The power and thermal efficiency obtained from the working fluid R113 is higher than that of R245fa.

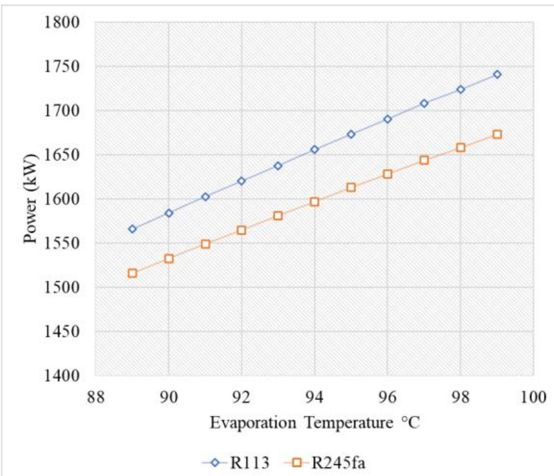

Figure 5.35 Effect of evaporation temperature on power output of ORC with pre-heater

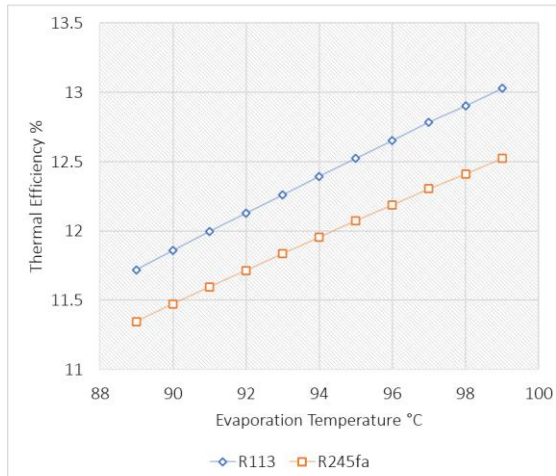

Figure 5.36 Effect of evaporation temperature on thermal efficiency of ORC with pre-heater

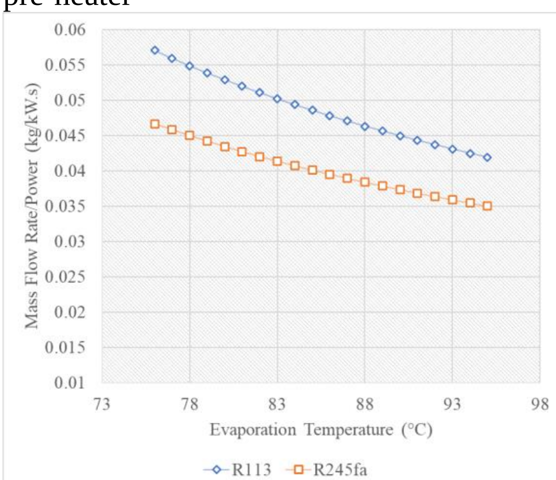

Figure 5.37 Effect of evaporation temperature on mass flow rate to produce unit power output of ORC with pre-heater

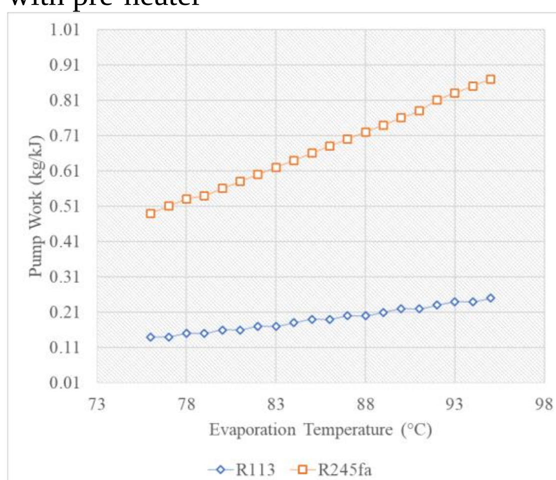

Figure 5.38 Effect of evaporation temperature on specific pump work of ORC with pre-heater

The mass flow rate to produce unit power output and specific pump work has decreased in this case as the cycle operates at lower evaporator pressure when the source temperature decreases.

5.3.3 ORC with pre-heater at source temperature of 180 °C

The ORC with the pre-heater shows an increase in power output and thermal efficiency with an increase in source temperature. For the same conditions, the power output is higher than that of baseline ORC and lower than recuperative ORC.

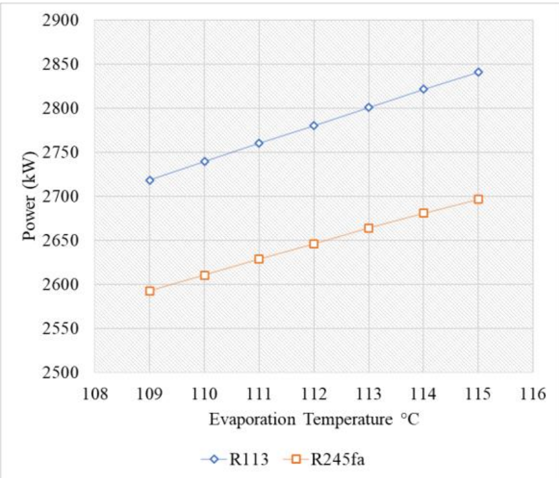

Figure 5.39 Effect of evaporation temperature on power output of ORC with pre-heater

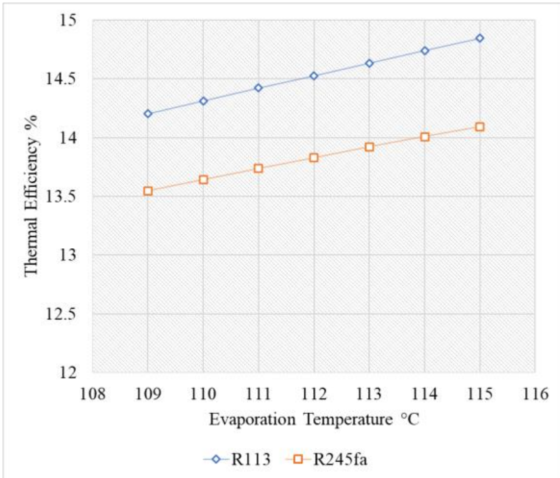

Figure 5.40 Effect of evaporation temperature on thermal efficiency of ORC pre-heater

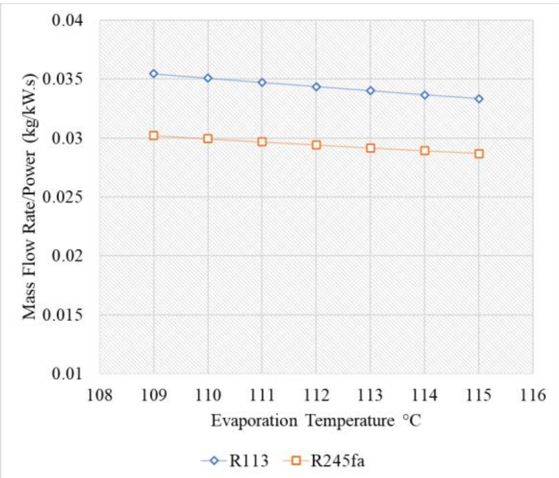

Figure 5.41 Effect of evaporation temperature on mass flow rate to produce unit power output of ORC with pre-heater

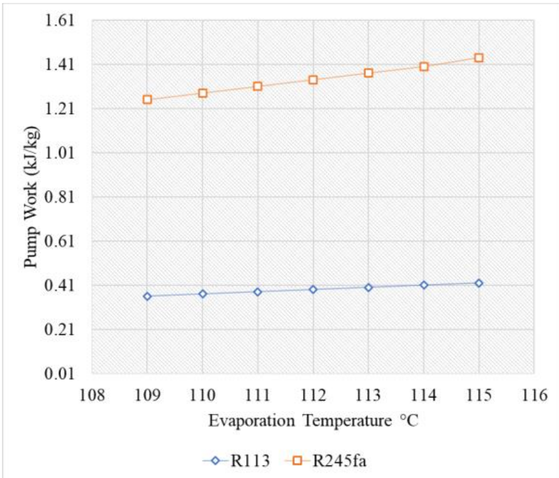

Figure 5.42 Effect of evaporation temperature on specific pump work of ORC with pre-heater

This configuration achieves higher power output at lower evaporator temperature than baseline ORC and recuperative ORC. ORC with pre-heater produces 71.89 kW more power than baseline ORC with working fluid R113 and 103.14 kW with R245fa.

The maximum power obtained was 2912.74 kW for R113 and 2799.84 kW for the working fluid R245fa at this source temperature. This power output is higher than that of baseline ORC but it is not higher than the recuperative ORC which is the best performing configuration among these three configurations.

Also, at the same evaporator temperatures, the pinch point temperature differences in the evaporator for these two working fluids are different. The working fluid R245fa shows lower power output and thermal efficiency for each case. However, at different evaporator temperatures and the same pinch point temperature difference values, the working fluid R245fa outperforms R113. The working fluid R113 operates at condenser pressure lower than the atmospheric pressure so there is a chance of leakage into the system while using this working fluid. This problem can be avoided by operating this working fluid at higher condenser pressure than that of atmospheric pressure, which increases the condenser temperature. The drawback of operating at higher condenser temperatures is the lower power output and thermal efficiency of the system.

The condenser temperature for the recuperative configuration is limited by the temperature profile of the recuperator. The temperature of the working fluid exiting the turbine should be higher than that of the working fluid exiting the pump so that the heat can be added to the working fluid at the output of the pump. Otherwise, the heat might start flowing in the other way and the temperature profiles cross.

Summary

Three configurations namely baseline ORC, recuperative ORC, and ORC with pre-heater have been considered in which the values of evaporator pressure and temperature are altered to obtain the results. The source temperature of geothermal energy was considered 150 °C, 168 °C, and 180 °C, the mass flow rate at 169 kg/s, and the geothermal fluid reinjection temperature were kept at 80 °C. The results are obtained keeping the PPTD ranging from 5 °C to 15 °C. The results for two working fluids have been plotted.

Chapter 6: Conclusions and Future Recommendations

6.1 Conclusions

This study conducts a comprehensive comparison of thermodynamic performance of two refrigerants for three configurations of ORC namely, baseline ORC, recuperative ORC, and ORC with pre-heater. Geothermal water is considered at 150 °C 168 °C and 180 °C as the open heat source and the reinjection temperature is kept at 80 °C. Cycle parameters such as evaporator pressure and temperature are optimized to get the maximal output power of a geothermal ORC.

- The results show that the recuperative ORC outperforms the other configurations in terms of power output.
- Working fluid R245fa requires higher specific pump work as compared to R113.
- The working fluid R113 achieves the highest power output of 2942.38 kW for recuperative ORC at the source temperature of 180 ^{o}C.
- The evaporator PPTD varied from 5 °C to 15 °C. By increasing the evaporator PPTD, the evaporation temperature decreases. The net power output can be increased by reducing the evaporator PPTD for ORC using lower-temperature geothermal sources.
- The primary variables determining the performance of the geothermal power plant are the cycle type, the ORC operating conditions, and the thermophysical characteristics of the working fluids.

6.2 Future recommendations

- The working fluid in the cycle is crucial, and while some alternatives have a better thermodynamic efficiency under the same operating conditions, they might not be suitable due to practical considerations like cost, toxicity, and component corrosion.

- Zeotropic fluids can be introduced to find new working fluids having higher thermodynamic efficiencies for low-temperature heat source applications.
- Together with technological improvements in the current systems, new applications in the field of renewable energy can be introduced.
- These systems have high potential in waste heat recovery systems.

Appendix

MUBEEN AHMED

From:	eesd2022@easychair.org on behalf of EESD2022 <eesd2022@easychair.org>
Sent:	Friday, January 14, 2022 11:51 AM
To:	MUBEEN AHMED
Subject:	Acceptance of full paper for 6th International Conference on "Energy, Environment & Sustainable Development (EESD2022)"

Dear Mubeen Ahmed,
Ref: EESD_2022_paper_159

Dear Sir/Madam,

I am pleased to inform you that your submitted full paper titled, "Comparison of Thermodynamic performance of geothermal Organic Rankine Cycle with two different working fluids" has been accepted for oral presentation at the 6th International Conference on "Energy, Environment & Sustainable Development (EESD2022)" to be held from January 17th – 19th, 2022 at Mehran University of Engineering and Technology, Jamshoro, Pakistan.

The program, containing the date, time, and location of your presentation, will be distributed soon. For further information on this event, please visit the conference homepage on the EESD 2022 website at: http://eesd.muet.edu.pk/

Note: Abstract proceedings of your submitted paper will be published upon successful Registration (https://rb.gy/l9hivm), and Oral Presentation of at least one author. However, accompanying co-author/participants have to register separately. Today is the last day for registration, please register as soon as possible.

In case you have any queries, please do not hesitate to contact us.

Sincerely,

Prof. Dr. Suhail Ahmed Soomro
Convenor, EESD 2022