MARMARA UNIVERSITY

FACULTY OF ENGINEERING

THERMODYNAMIC INVESTIGATION OF ORGANIC RANKINE CYCLE ENERGY RECOVERY SYSTEM AND NEW GENERATION WORKING FLUIDS APPLICATIONS

CENK INAN, ECE NUR GULER

GRADUATION PROJECT REPORT

Department of Mechanical Engineering

Supervisor

Assoc. Prof. Dr. Mustafa

YILMAZ

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by

CENK INAN

ECE NUR GULER

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ABSTRACT

The Organic Rankine Cycle (ORC) has developed as a promising innovation for effective and sustainable energy conversion in different applications. This paper gives a comprehensive overview of the Organic Rankine Cycle, focusing on its standards, components, working liquids, and assorted applications. Not at all like conventional Rankine cycles utilizing water as a working liquid, ORC utilizes natural substances with lower boiling points focuses, encouraging control era from low to medium temperature heat sources. The cyclic process includes compression, heat expansion, addition, and heat rejection, making it appropriate for saddling squander heat, geothermal, and solar thermal energy. We dive into the selection criteria for natural working liquids, considering their thermophysical properties, natural affect, and security angles. Moreover, the paper investigates recent advancements in ORC innovation, including system optimization, heat exchanger design, and integration with renewable energy sources. The flexibility of ORC system is highlighted through their application in biomass, industrial waste heat recovery and solar power plants. As the worldwide interest of sustainable energy solutions escalate, the Organic Rankine Cycle stands out as an essential innovation, advertising a practical and proficient implies of converting low-grade heat into valuable power as electricity whereas contributing to the transition to a more sustainable energy landscape.

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1. Rankine Cycle

The Rankine cycle or Rankine Vapor Cycle is the process widely used by power plants such as coal-fired power plants or nuclear reactors. In this mechanism, a fuel is used to produce heat within a boiler, converting water into steam which then expands through a turbine producing useful work. The process was developed in 1859 by Scottish engineer William J.M. Rankine.[1] This is a thermodynamic cycle which converts heat into mechanical energy—which usually gets transformed into electricity by electrical generation. [1]

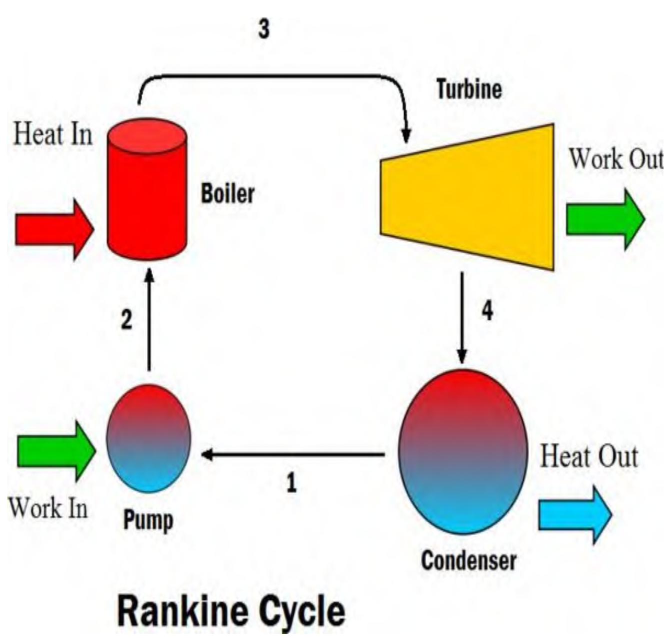


Figure 1 A simple schematic with components for the Rankine Cycle.[1]

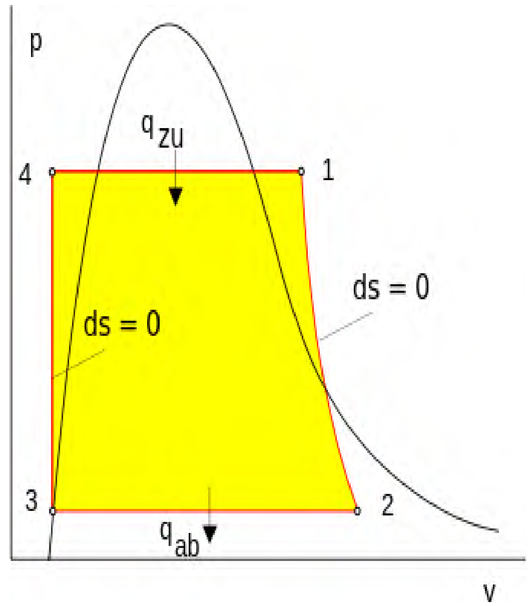


Figure 2 The pressure volume diagram of the Rankine cycle. This illustrates the changes in pressure and volume the working fluid (water) undergoes to produce work. [2]

The steps in the Rankine Cycle as shown in Figure 1 and the corresponding steps in the pressure volume diagram (figure 2) are outlined below:

Pump (Compression - Steps 3 to 4):

- The cycle begins with the compression of the working fluid (commonly water) in a pump.
- The pump requires external work input to increase the pressure of the fluid from a low-pressure state (state 3) to a higher pressure (state 4).
- This process is adiabatic, meaning it occurs without heat exchange with the surroundings.
- The result is a high-pressure liquid ready for the next stage.

Boiler (Heat Addition - Steps 4 to 1):

- The high-pressure liquid from the pump is directed to a boiler, where it undergoes isobaric (constant pressure) heating.
- Heat is added to the working fluid, causing it to undergo a phase change from liquid to vapor (saturated steam).
- This process occurs at the boiling point and results in high-temperature, high-pressure steam (state 1).

Turbine (Expansion - Steps 1 to 2):

- The high-pressure steam is expanded through a turbine, converting the thermal energy into mechanical work.
- The expansion process is adiabatic, and the steam undergoes a decrease in pressure and temperature.
- The turbine is connected to a generator, producing electrical energy as a result of the mechanical work performed by the expanding steam.

Condenser (Heat Rejection - Steps 2 to 3):

- The low-pressure steam exiting the turbine is directed to a condenser.
- In the condenser, the steam undergoes isobaric (constant pressure) heat rejection.
- Heat is transferred from the steam to a cooling medium, often water from the environment (such as a lake or river), causing the steam to condense back into a liquid.
- The condensed liquid is then pumped back to the boiler to repeat the cycle.

Every part of the Rankine Cycle is important in changing heat into power. The effectiveness of the cycle depends on the pressure, temperature, and type of fluid used in each stage. The Rankine Cycle is used to make steam power in power plants and factories.

The efficiency of the Rankine cycle is limited by the high heat of vaporization by the fluid. The fluid must be cycled through and reused constantly; therefore, water is the most practical fluid for this cycle. This is not why many power plants are located near a body of water—that's for the waste heat.

As the water condenses in the condenser, waste heat is given off in the form of water vapor—which can be seen billowing from a plant's cooling towers. This waste heat is necessary in any thermodynamic cycle. Due to this condensation step, the pressure at the turbine outlet is lowered. This means the pump requires less work to compress the water—resulting in higher overall efficiencies. [3]

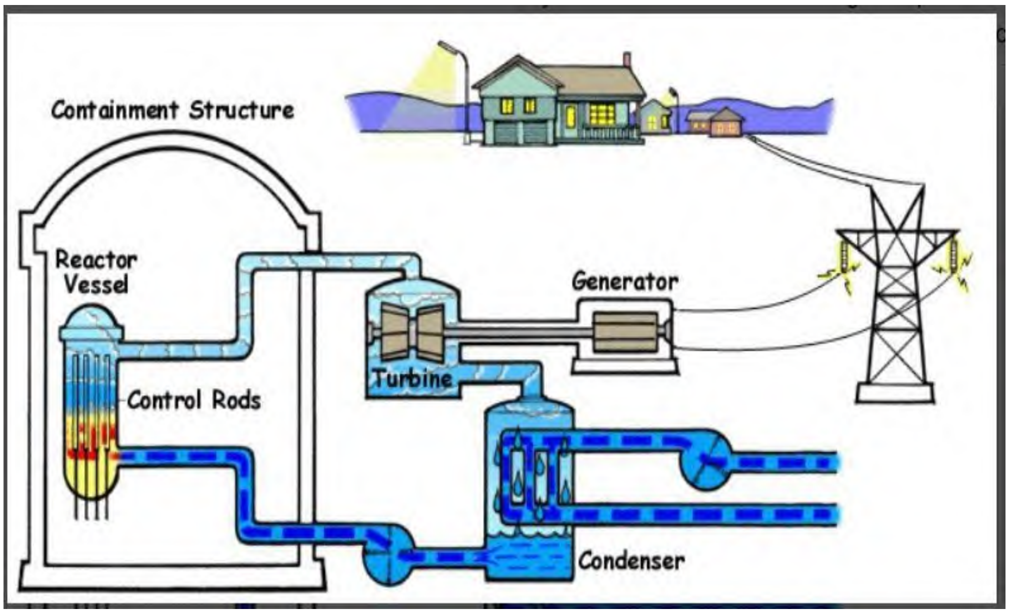


Figure 3 Rankine cycle in a nuclear power plant.[3]

2. Process of an Ideal Rankine Cycle

Whole the process consists of individual open systems (compression by pump, expansion by turbine, heat input through boiler, and heat rejection through condenser.).

Before proceeding with analyzing each step, let us take at a look the thermodynamics of open systems: we have already known the energy balance relation for a system, as follows;

$$Q_{in} + W_{in} + m_{in} e - (Q_{out} + W_{out} + m_{out} e) = \frac{dE_{sys}}{dt} = \frac{\Delta E}{\Delta t} \qquad (1)$$

Because the system is steady flow which means that there is no change of energy of the system with respect to the time. Then, we can rewrite the relation above as follow;

$$Q_{in} + W_{in} + m_{in} e - (Q_{out} + W_{out} + m_{out} e) = \cancel{\frac{dE_{sys}}{dt}} = 0$$
$$Q_{in} + W_{in} + m_{in} e = Q_{out} + W_{out} + m_{out} e \tag{2}$$

Here Q_{in} , W_{in} , m_{in} , Q_{out} , W_{out} , m_{out} are the values of heat, work, and mass input for terms with sub "in" and are the heat, work, and mass output for the terms with sub "out", respectively. Here latter e represents the flow energy of the mass that formed by enthalpy, kinetic, and potential energy. It is important to underline that, during our calculations, we ignore kinetic and potential energy changes due to too small value of changes.

It is crucial point that not only energy balance occurs, but also mass conservation takes place, as well. Therefore, it is true to write relation as;

$$m_{in} = m_{out} = m \tag{3}$$

By this perspective, we can analyze the entire cycle by observing sub-systems individually.

Isentropic Compression: 1-2

Condensed water coming from the condenser is pumped to the boiler at boiler pressure with the help of feed pump.

To do so work, W_{in} is supplied to feed pump.

$$\begin{aligned}
&\cancel{Q_{in}} + W_{in} + m_{in} e = \cancel{Q_{out}} + \cancel{W_{out}} + m_{out} e \\
&W_{in} = e(m_{out} - m_{in}) \\
&e = h + \cancel{ke} + \cancel{pe} \\
&W_{in} = (h_{out} m_{out} - h_{in} m_{in}) \\
&W_{in} = (h_{out} - h_{in}) m
\end{aligned} \tag{4}$$

This is work for compression of the water to increase its pressure to boiler pressure and in unit of kJ. To obtain specific work which is work done per mass, the relation is divided mass;

$$w_{in} = h_2 - h_1 \tag{5}$$

Because the output of the pump is numbered as 2, to clarify the calculation we represent it as h_2 and h_1 for input.

Heat Addition Process: 2-3

In this process, the steam (water for ideal Rankine cycle) is heated at constant pressure. Ideally, whole process takes place between only two pressures as high and low. Heat addition occurs in the boiler until the saturation temperature is reached. The high-pressure, high-temperature fluid from the compressor is then heated at constant pressure in the boiler, where it absorbs heat from an external source (typically combustion of fuel). This process results in the working fluid becoming a high-temperature, high-pressure vapor.Steam might exceed this value of temperature and becomes superheated. When we analyze “heat addition” we figure out the amount of Q_{in} as follows;

$$\begin{aligned} Q_{in} + \cancel{W_{in}} + mh_{in} &= \cancel{Q_{out}} + \cancel{W_{out}} + mh_{out} \\ Q_{in} &= m(h_3 - h_2) \end{aligned} \tag{6}$$

To obtain specific heat given to the steam, division by mass should be done.

$$q_{in} = (h_3 - h_2) \tag{7}$$

Isentropic Expansion Process: 3-4

Expansion process is the process delivering the mechanical energy that is converted to electricity. High pressure, high temperature steam is supplied to the turbine.

The high-temperature, high-pressure vapor expands isentropically in the turbine, producing mechanical work. This work is often used to drive a generator, producing electrical power.

This steam expands isentropically into steam turbine up to the condenser pressure. Steam turbine develops mechanical work, W_{out} due to expansion of steam. W_{out} can be found by following previous procedure. It can be obtained as:

$$\begin{aligned} \cancel{Q_{in}} + \cancel{W_{in}} + mh_{in} &= \cancel{Q_{out}} + W_{out} + mh_{out} \\ W_{out} &= m(h_3 - h_4) \end{aligned} \tag{8}$$

It is important to state that during both expansion and compression, it is accepted there is any heat loss or gain.

Heat Rejection Process: 4-1

The heat rejected in the condenser is essential for completing the thermodynamic cycle and maintaining the continuous operation of the power plant. The purpose of this heat rejection is to efficiently convert the high-temperature, high-pressure vapor leaving the turbine back into a liquid state, allowing it to be compressed again in the pump to restart the cycle.

The low-pressure, low-temperature vapor leaving the turbine enters the condenser, where it is condensed into a liquid by rejecting heat to a cooling medium (commonly water or air) at constant pressure. This process prepares the fluid for the next cycle by returning it to a liquid state.

$$\begin{aligned} \cancel{Q_{in}} + \cancel{W_{in}} + mh_{in} &= Q_{out} + \cancel{W_{out}} + mh_{out} \\ Q_{out} &= m(h_4 - h_1) \end{aligned} \tag{9}$$

Up to here, analyzing An Ideal Rankine Cycle is represented. For Rankine cycle, lots of alternative working fluids exist. Generally, water is used as the working fluid in Rankine Cycle. There are several reasons;

- Abundance and Cost: Water is abundant and relatively inexpensive compared to many other fluids. Its widespread availability makes it a practical choice for large-scale power generation.

- Thermo-physical Properties: Water exhibits excellent thermophysical properties for the Rankine cycle. It has a high specific heat capacity, high latent heat of vaporization, and a wide range of temperature and pressure conditions over which it remains in liquid form. These properties make water efficient for absorbing and transferring heat during the various stages of the cycle.

- Phase Change Characteristics: The phase change from liquid to vapor and vice versa (during condensation) is an essential aspect of the Rankine cycle. Water's high latent heat of vaporization allows for significant energy storage during the phase change, contributing to the efficiency of the cycle.

- Safety: Water is non-toxic and non-flammable, making it a safe choice for power generation. In the event of a leak or malfunction, water poses minimal environmental and safety risks compared to some other working fluids.

- Environmental Impact: The environmental impact of water as a working fluid is generally lower than that of many alternative fluids. Water vapor is a natural component of the Earth's atmosphere, and the use of water in the Rankine cycle does not introduce environmentally harmful substances.

While water is a common choice for many Rankine cycle applications, especially in steam power plants, other fluids can be used for specialized applications or when specific performance criteria need to be met. Different working fluids may be chosen based on factors such as temperature range, pressure conditions, environmental concerns, and the specific requirements of the power generation system. Our project is mainly based on this point: Organic Rankine Cycle (use of organic working fluid instead water) is analyzed and examined.[3]

3. Types of Rankine Cycle

The thermodynamic efficiency of the Rankine cycle can be expanded by increasing the heat input to the cycle. This could be done by expanding the temperature to alter the phase of steam to superheated steam. There are numerous variations like this to extend the thermodynamic productivity of the cycle.

Underneath are a few of the cycle types designed to increase the thermal efficiency of the cycle.

1. Real Rankine Cycle

The Real Rankine cycle is a thermodynamic cycle that is commonly used in power plants to generate electricity. It operates on the principle of converting heat into work. These processes are irreversible compared to the ideal cycle which causes increase in entropy as shown below:

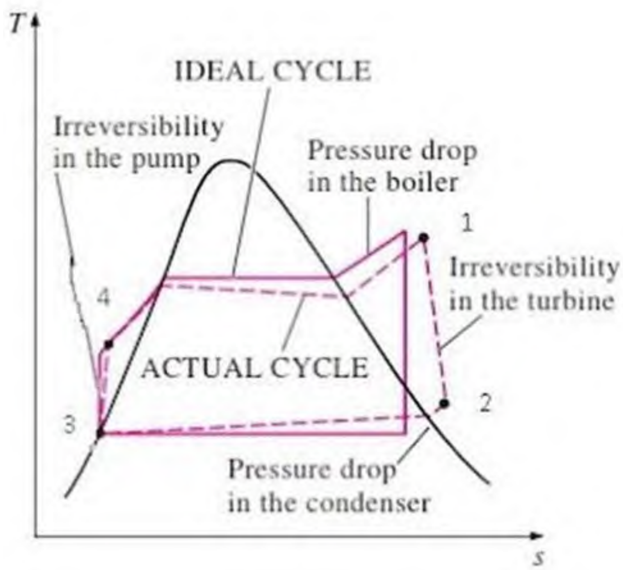


Figure 4 Ideal and Actual Cycle Comparison [4]

The real Rankine cycle includes things like processes that can't be undone, pressure losses in turbines and pumps, heat losses in the condenser, and not everything going exactly as planned with the working fluid. These things in the real world can affect how well the Rankine cycle works, making it different from the ideal version called the ideal Rankine cycle. These conditions increase the required power and decrease the generated power. Engineers consider these real-life restrictions when designing and running power plants to make them work better and produce more power.

Efforts are often made to improve the efficiency of the Rankine cycle by using new materials, improving insulation, making better parts, and finding better ways to exchange heat. Also, different types of Rankine cycles and combined cycles with gas turbines are used to make power generation more efficient.[4]

2. Regenerative Rankine Cycle:

The Regenerative Rankine Cycle aims to improve efficiency by incorporating feedwater heaters. Steam extracted from various turbine stages is utilized to preheat the feedwater before it enters the boiler. By increasing the average temperature of heat addition, the cycle reduces the amount of fuel required for steam generation, thus enhancing overall efficiency.

Benefits of Regenerative Cycle

- When the working temperature range for steam in this cycle is lowered, the stress on the equipment is also lowered.

- The turbine wears away less because of the type of fluid it uses.

Downsides of the Regenerative Cycle

- The steam goes faster because less work is done in the boiler, due to the regenerator presence.
- The big power plant costs a lot because it needs large parts to work properly.

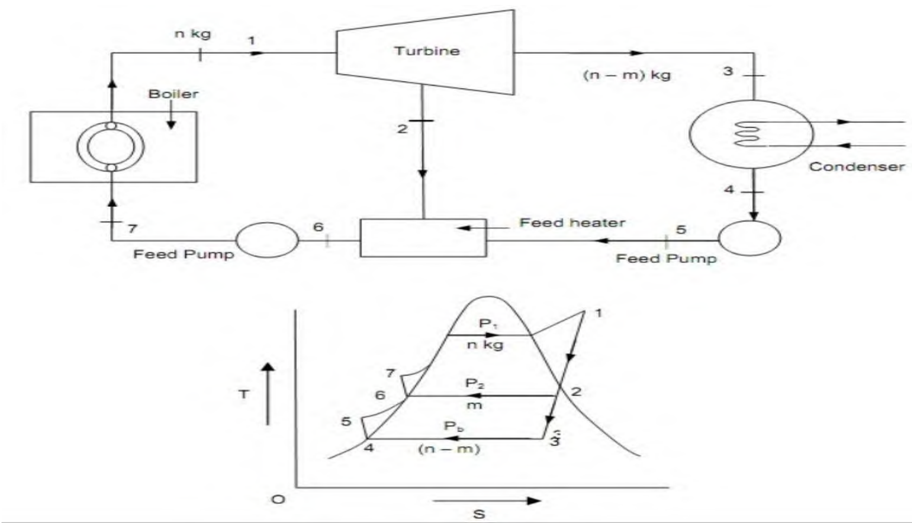


Figure 5 Regenerative Rankine Cycle [5]

3. Reheat Rankine Cycle:

In the Reheat Rankine Cycle, steam is extracted from the turbine after partial expansion and sent back to the boiler for reheating at constant pressure before re-entering the turbine for further expansion. Reheating helps to maintain high steam quality throughout the expansion process, reduce moisture content and improving overall efficiency.

Reheating removes the moisture carried by the steam during the final stages of the expansion process in the cycle. With this cycle design, the steam turbines are kept in series to perform the work. Here is a brief on how this cycle works.

1. The high-pressure steam from the boiler enters the first turbine
2. The steam is passed to the boiler and reheated again
3. The second turbine is at low pressure to which this reheated steam flows.

The purpose of the Rankine cycle with reheat is to increase the average temperature of the steam in the cycle. Reheating the steam through another stage improves the efficiency of the cycle only half as much as the preceding stage.

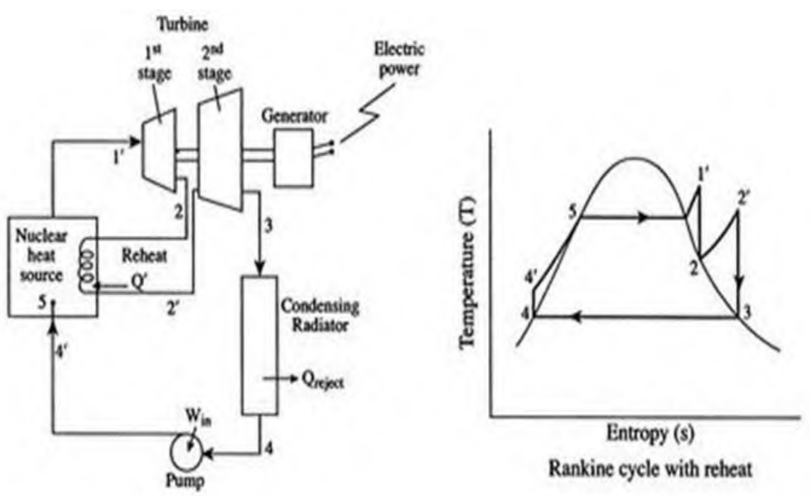


Figure 6 Reheat Rankine Cycle [6]

Benefits of using Reheat Rankine Cycle

- Increased the thermal efficiency of the cycle by making the steam turn into water as it expands, which helps to protect the turbine blades
- Increases the amount of work the turbine can do compared to the amount of work put into it

Downsides of using Reheat Rankine Cycle

- These cycles require a long set of pipes. So, first you have to pay a lot to install it, and then you have to keep paying a lot to maintain it
- When something gets heated again, the condenser may get bigger.[6]

4. Supercritical Rankine Cycle:

The Supercritical Rankine Cycle operates beyond the critical point of the working fluid (commonly water). Above this critical point, the fluid exhibits properties of both a liquid and a gas. Operating at supercritical pressures allows for higher thermal efficiency and reduced cooling requirements, contributing to improved overall cycle efficiency.

In a supercritical Rankine cycle, the heat is moved to a high-pressure supercritical fluid, which changes to a supercritical phase. Liquid in this stage goes to a turbine and expands, which helps make electricity. The steam is cooled down and turned back into water, then used again in the process.

A supercritical fluid, also known as SCF, is a kind of fluid that is at a really high temperature and pressure. This is the point where liquid and gas don't exist together. These fluids are discovered on the Earth by black smokers, which are a type of underwater hot spring.

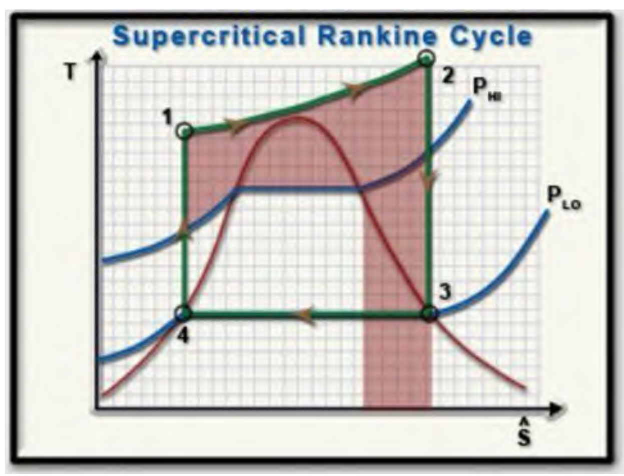


Figure 7 Supercritical Rankine Cycle [7]

Benefits of Supercritical Rankine Cycle

- This cycle works better than all the others.

The outflow of the turbine is high quality. Downsides of Supercritical Rankine Cycle

- Normal boilers may not be able to handle very high temperatures and pressure. So, a special boiler is required. Due to this, there is an overall cost bump.[7]

5. Organic Rankine Cycle

The Organic Rankine Cycle (ORC) is a useful and efficient way to make power from heat that's low to medium temperature heat sources. The ORC is different from the regular Rankine cycle because it uses organic fluids that boil at lower temperatures instead of just water. This means it can make electricity at lower temperatures. This feature makes it great for using waste heat from factories, hot water underground, solar power, and burning plants for energy.

The ORC works like the regular Rankine cycle, but it uses special liquids that boil at lower temperatures than water, such as hydrocarbons or fluorinated compounds. These fluids can make power even when it's not very hot. This means we can use heat that would usually be wasted, which we couldn't do with regular steam power.

The process starts with turning the organic liquid into vapor in the evaporator, using heat from the heat source. The hot gas moves a machine that creates energy. Later, the low-pressure gas turns back into a liquid in the condenser and gives off heat which can be used again or let go. Finally, the liquid is sent back to the evaporator to start the process over again.

The ORC has many benefits, like being able to work with different heat sources and running well at lower temperatures than steam-based cycles. Its ability to adapt to different heat sources and generate power in small, decentralized ways makes it a great technology for situations where traditional steam cycles won't work because there isn't enough heat.

The ORC can be used in many different ways, like getting energy from waste heat, extracting energy from the earth's heat, and making power from sunlight. Industries are trying to use less energy and be more environmentally friendly. The Organic Rankine Cycle is becoming more popular because it is a good way to turn low-temperature heat into electricity, which helps reduce the need for non-renewable energy sources. This is a big help in finding sustainable and renewable energy solutions. The Organic Rankine Cycle (ORC) is a good technology for using low-temperature heat to make power. It can be used in different industries and in making renewable energy.

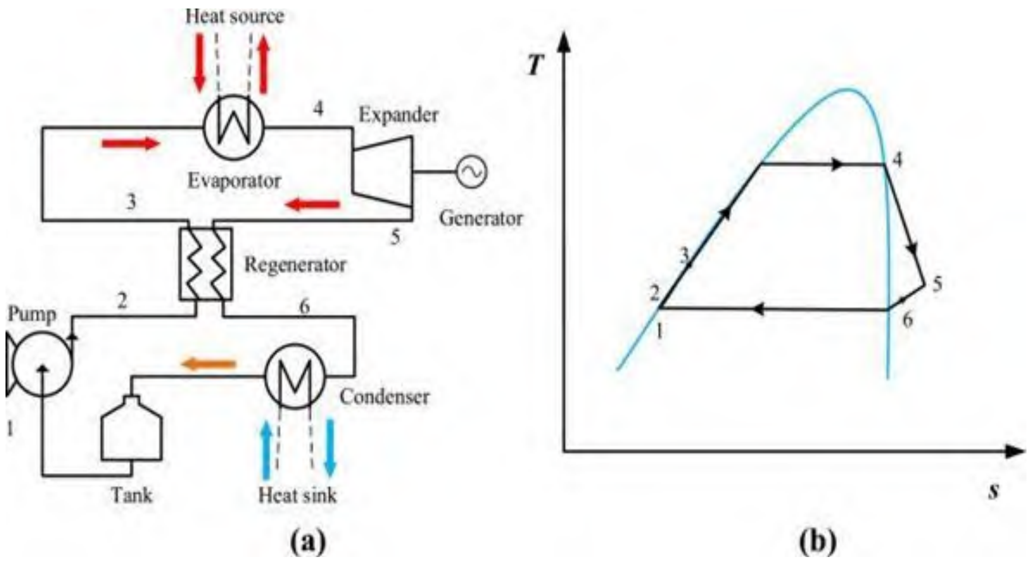


Figure 8 Organic Rankine Cycle [8]

Benefits of Organic Rankine Cycle

- We don't need to make the water too hot, so we don't need a water treatment system for the fluid.

- The turbine will have low temperature and pressure at the inlet.

Downsides of using Organic Rankine cycle

- ORCs produce less electricity power than a regular cycle when they're running in the same way.
- The organic fluids used for this process are combustible. A small leak can cause harm to the environment.

Each Rankine cycle variation offers distinct advantages and applications, enabling adaptation to specific operational requirements, increasing efficiency, and enhancing the utilization of different heat sources in various industrial applications and power generation systems.[8]

2 Literature Review of Organic Rankine Cycle

1. Introduction

Electricity demand is one of the most significant needs to be addressed in today's world. The advancement of technology and the increasing use of electronic equipment continually elevate power consumption. Various methods are applied for electricity generation to meet this demand, including hydro, thermal, nuclear, and renewable sources. For instance, in thermal power plants, electricity generation is achieved by burning fossil fuels, transferring the released energy to fluids, and then rotating turbine blades with the resulting high-pressure steam. The Organic Rankine Cycle (ORC) is a technology that converts energy from any thermal source into electrical energy. These thermal sources may include solar, geothermal, biomass, or waste heat. The primary distinction between ORC and the conventional Rankine cycle, which gives the system its "Organic" designation, lies in the preference for an organic fluid instead of steam-water in the cycle. The ideal fluid for ORC is generally "Isentropic" or "Dry Fluid" with zero or positive slope, requiring no superheating. Unlike the conventional Rankine cycle, which necessitates superheating, the ORC system has a lower turbine inlet temperature. Consequently, the use of organic fluids reduces the high thermal resistance requirements for turbine blades and lowers the overall cost. In recent years, Turkey has made significant progress in exploring ways to meet its increasing energy demand, with notable advancements in geothermal power plants. Currently, there are over 50 geothermal power plants in operation and under construction. Particularly concentrated in the Aegean region around Aydın, Denizli, and Manisa provinces, these plants position Turkey, as seen in Figure 1, with an installed power capacity of 1347 MW in the 1 GW and above capacity category in the global ranking of geothermal energy power plant capacities.[9]

In the literature, various studies on Organic Rankine Cycle (ORC) analysis and fluid selection are evident. Researchers have conducted thermodynamic analysis and parameter optimization for ORC systems designed using different organic fluids. The results indicate that R123 fluid achieved the highest efficiency with the least irreversibility for a constant 277 °C heat source temperature [10]. They conducted an optimization study using the genetic algorithm method to select the appropriate fluid, suggesting Novec649 for irreversibility performance and benzene for heat transfer area [11].

Energy and exergy analysis for waste-heat-driven ORC were performed using R245fa fluid, revealing the significant impact of evaporator pressure on energy and exergy efficiency [12]. Another study focused on ORC with mixed fluids, investigating ΔTPP,e and ΔTPP,k (evaporator-condenser pinch point temperature difference). They determined that the optimum ΔTPP,e for mixed fluids should be between 3-6 °C, considering exergo-economic performance as an evaluation criterion [13].

Research has also explored methods to determine organic fluid and operating conditions in ORC based on ΔTPP. They found that when there is an appropriate difference between the inlet temperature of the heat source and the critical temperature of the fluid, maximum power can be achieved, and the fluid evaporates near the critical region (Yu et al., 2015). A study focused on the analysis of ORC producing electricity from geothermal energy sources, using R245fa as the working fluid. A demo model was created for an analysis requiring a total turbine power of 250 kW[14].

One study conducted the thermodynamic analysis of ORC using thermal heat data obtained from biomass for a business. The system payback period was calculated as 3.24 years for a net electricity production of 891.76 kW. They identified the condenser as the equipment with the highest unit exergy cost in the ORC system [15].

Using EES software, researchers performed ORC thermodynamic analysis, focusing on the effect of water in different phases at the evaporator inlet on system efficiency. They emphasized the significant role of evaporator components in determining system performance [16]. Another study analyzed the energy and exergy of the system using HFE7100 and FC72 fluids in ORC. They aimed to maximize system efficiency with the optimal fluid selection in different temperature ranges, concluding that FC72 was the most suitable fluid [17].

Comparisons were made among organic fluids used in the system, highlighting the crucial influence of fluid type on the thermal efficiency concerning turbine inlet temperature. The results indicated that isentropic fluids were more effective than dry and wet fluids [18].

Researchers worked on ΔTPP,e optimization using Analytic Hierarchy Process (AHP) and entropy methods for ORC with different fluids. They evaluated the economic and thermal performance of the ORC. The results indicated that they achieved maximum power output with R141b and maximum thermal and exergy efficiency values with R11 [19].

Another study utilized different fluids for energy and exergy analysis in ORC, considering evaporation and heat source temperatures as performance parameters. Among the evaluated criteria, R600a fluid demonstrated the best performance in terms of thermal efficiency, net power, total irreversibility, and exergy efficiency [20].

A study focused on the irreversibility values in ORC components, indicating that an increase in evaporation pressure from 250 kPa to 400 kPa led to a 32% increase in turbine irreversibility due to the high pressure ratio. They determined that under 250 kPa and 20 °C superheating temperature, 79.6% of total irreversibility occurred in the evaporator, 10.9% in the condenser, and 9.4% in the turbine [21].

Additionally, ORC energy analysis was conducted using R134a and R152a fluids. The study revealed that the system designed with R152a required more heat input compared to R134a. The maximum thermal efficiency for R134a and R152a was found to be 8.123% and 9.351%, respectively [22].

They performed the thermodynamic analysis and design of a regenerative Organic Rankine Cycle (ORC) system operating with dry fluid using EbsilonProfessional software. The aim was to determine the ideal fluid under different operating conditions. In their study, it was observed that in a dry fluid system with a constant minimum temperature, increasing the maximum temperature resulted in higher thermal efficiency.

For the regenerative ORC system, the best and worst thermal efficiency values were found to be in R113 and R227ea fluids, respectively [23].

In the context of waste heat application, they conducted thermodynamic and economic analysis of a recuperative ORC using flue gas at 160°C as the heat source, employing MATLAB programming. They utilized a fuzzy multi-criteria evaluation method for the optimal organic fluid selection and determination of the optimum superheating temperature. Through optimization, they demonstrated that butane fluid achieved the best performance with an optimal evaporation temperature of 100°C and superheating temperature of 5°C [24].

In the study of geothermal ORC applications, they examined the impact of ΔTPP,e on thermodynamic performance. They emphasized the importance of ΔTPP,e as a significant parameter for both thermodynamic and economic performance. They found that a lower ΔTPP,e provided more turbine net power but increased the heat transfer area, leading to negative economic effects [25].

Utilizing a multi-objective approach, they determined the optimum $\Delta TPP,e$ values in ORC systems, working on two objective functions: economy and environment. They defined the economic function as the ratio of total heat transfer area to net power and the environmental function as the ratio of total irreversibility to exergy drop at the hot source. The study concluded that with R245fa as the working fluid, the optimum $\Delta TPP,e$ should be between 7-10 °C [26].

They investigated the performance of an ORC system for geothermal sources in the temperature range of 50-100 °C, using four different fluids. Performance parameters included first and second law efficiencies and net work. The study found that R141b exhibited the best performance in terms of first and second law efficiencies, while R134a performed the best in terms of net work [27].

For the 2.7 MW Afyon Geothermal Power Plant, they conducted ORC design, thermoeconomic performance evaluation, and optimization using real plant data. The study determined the exergetic cost of produced electricity as 0.0233 $/kWh

with a payback period of 3.6 years, and the optimized exergetic cost as 0.0176 $/kWh with a payback period of 2.87 years [28].

They explored ORC configurations designed with two-phase flash expansion and zeotropic fluids, aiming to maximize net power. The study emphasized that maximum net power was not always achieved at the point of highest thermal efficiency, suggesting that investigating net power maximization was more critical than thermal efficiency maximization. The results indicated that zeotropic fluid-based ORC was more advantageous at low heat source temperatures and high condenser fan power [29].

Reviewing the methods used in the literature, Engineering Equation Solver (EES) software was commonly employed for thermodynamic analysis, while genetic algorithm methods were often used for thermodynamic optimization. Some studies focused on the performance of ORC using a single organic fluid, while others compared the performance of multiple organic fluids based on different parameters. Different studies proposed different fluids as optimal, reflecting variations in identified performance parameters.

In summary, the literature reveals numerous studies on the performance of organic fluids in ORC systems. Most studies recommend fluids showing good performance under one or two parameters. However, a fluid performing well in terms of turbine work does not always exhibit good thermal efficiency performance, possibly due to excessive heat input requirements in some fluids, resulting in lower-than-expected thermal efficiency values. Therefore, the working fluid has a crucial role for the cycle. [30]

2. The Working Fluid

The selection of the organic fluid depends on the temperature profile of the heat source, safety considerations, and the overall efficiency goals of the ORC system. The main thermodynamic irreversibility in ORC is caused by the temperature difference between evaporator, waste heat stream and the boiling refrigerant.

Organic fluids that can boil at higher temperatures or boil at critical temperatures reduce that difference between waste heat stream and the cycle working fluid. Therefore as a result, there will be higher thermodynamic efficiency. Chlorine containing fluids with high critical temperatures have been proposed in the past as ORC fluids as R114, R113, R11, R141b and R123 have higher critical temperatures than R245fa and would result in substantially higher thermal efficiencies.

However, these fluids are not good for human and environmental health. Since they all either flammable or toxic, ozone layer depleting, have substantial global warming impact, they have been banned or soon will be banned.

Nowadays, the engineers and scientist are working on finding more efficient, sustainable and non-damaging fluids.

- ***R-134a (1,1,1,2-Tetrafluoroethane):***

R134a, also called tetrafluoromethane, is a type of refrigerant that is used in place of harmful R-12 CFC refrigerants in cooling systems. Ever since scientists found out that CFCs harm the ozone layer, R-134a which has no ozone depletion potential is replaced by. R-134a is a non-flammable hydrofluorocarbon with a critical temperature of 101.1°C and a critical pressure of 40.7 bar. It has good thermodynamic properties for moderate-temperature heat sources. Due to regulations and standards, it is preferred not to use this fluid and its derivatives to reduce the CO_2 footprint due to environmental limitations, even if they provide high efficiency and should be replaced by different working fluid, which will be mentioned.

- ***R-245fa (1,1,1,3,3-Pentafluoropropane):***

R-245fa is a type of chemical called hydrofluorocarbon (HFC). It is used for new air conditioning systems in buildings and industrial plants to keep things cool and save energy. It can also be used in powerful systems with centrifugal compressors and can be used instead of HCFC R-123. R-245fa is a non-flammable hydrofluorocarbon with a critical temperature of 154.0°C and a critical pressure of 5.29 bar. It has favorable thermodynamic properties for medium-temperature heat sources and is considered safe with low toxicity. Commonly utilized in ORC systems for recovering waste heat from industrial processes operating at medium temperatures. It's also employed in solar thermal applications.

- ***R-507A***

R507A is a mixture of R-125 (pentafluoroethane) and R-143a (1,1,1-trifluoroethane) in a 50/50 weight ratio. R507A as a working fluid is favorable for the ORC, is good for making energy from waste heat and works well with other things. Yet it's not good for the environment because it has a high global warming potential. This might make it not a good choice for the future. As rules change to reduce greenhouse gases, there is more focus on switching to environmentally friendly options. When choosing a refrigerant for an ORC system, it's important to think about how well it works, its impact on the environment, if it follows the rules, and how safe it is.

- ***R-1233zd(E)***

R1233zd (E) is a promising fluid to use in ORC systems. It works well, is friendly for environment, and is safe to use. However, like any type of refrigerant, whether it is a good choice depends on the specific needs of the situation. This includes how the system works, how safe it is, and if it follows the rules and regulations. Doing careful checks and tests can help figure out if R1233zd (E) is the best option for a specific ORC application.

- ***R-1234yf***

R1234yf is a refrigerant which used in air conditioning systems, lately it is common in automotive industry. As a refrigerant it is expect that to have low global warming potential and R-1234yf has smaller impact compared to R-134a. With low flammability, R-1234yf is generally more efficient than R-134a and in cooling systems it leads lower energy consumption. In ORC systems it is crucial to have balance between efficiency, environmental impact and safety. Therefore R-1234yf is a suitable choice to have it as a working fluid.

No.	Working Fluid	Triple Point Temperature	Normal Boiling Point Temperature	Critical Point Parameters			Working Fluid Class [30]	Equation of State
		t_{trp} °C	t_{nbp} °C	t_{cr} °C	p_{cr} MPa	ϱ_{cr} kg/m³		
1	R113	−36.22	47.59	214.06	3.39	560.00	ANZCM	[31]
2	R114	−92.52	3.59	145.68	3.25	579.97	AZCM	[32]
3	R123	−107.15	27.82	183.68	3.66	550.00	ACNMZ	[33]
4	R124	−199.15	−11.96	122.28	3.62	560.00	ACNZM	[34]
5	R1234ze	−104.53	−18.95	109.37	3.63	489.24	ACNZM	[35]
6	R134a	−103.30	−26.07	101.06	4.06	512.00	ACZ	[36]
7	R152a	−118.59	−24.02	113.26	4.51	368.00	ACZ	[37]
8	R227ea	−128.60	−16.34	101.75	2.93	594.25	ANCMZ	[38]
9	R236fa	−93.63	−1.44	124.92	3.20	551.30	ACNMZ	[39]
10	R365mfc	−34.15	40.15	186.85	3.22	473.84	ANZCM	[38]
11	R245ca	−81.65	25.13	174.42	3.39	523.59	ANCMZ	[40]
12	R245fa	−102.10	15.14	154.01	3.65	516.08	ACNMZ	[41]
13	R601a	−160.50	27.83	187.2	3.38	236.00	ANCMZ	[41]
14	R141b	−103.47	32.05	204.35	4.21	458.56	ACNMZ	[41]
15	R142b	−130.43	−9.12	137.11	4.05	446.00	ACNMZ	[41]
16	R236ea	−103.15	6.19	139.29	3.5	563.00	ANZCM	[42]
17	R600a	−159.42	−11.75	134.66	3.63	225.5	ACNMZ	[43]
18	RC318	−39.80	−5.97	115.23	2.78	620.00	AZCM	[32]
19	R1234yf	−53.15	−29.45	94.7	3.38	475.55	ACNZM	[44]
20	R290	−187.63	−42.11	96.7	4.25	220.48	ACZ	[45]

Figure 9 Fluid Properties [9]

3 Thermodynamic Analysis and Modeling of ORC

1. Energy Analysis

Pump Phase
In the pump phase of the ORC, the organic working fluid is in its liquid state. The pump increases the pressure of the fluid, raising its energy level in preparation for the next phases. The pump work required can be calculated using the equations:

$$W_{pump,in} = m(h_{out} - h_{in}) = v(P_{out} - P_{in})$$

Evaporator Phase
In the evaporator phase, the high-pressure liquid working fluid is heated and vaporized using an external heat source. This phase typically occurs at constant pressure and temperature. The heat input can be calculated as:

$$Q_{in} = m(h_{out} - h_{in})$$

Turbine Phase
In the turbine phase, the high-pressure vapor expands through the turbine, producing mechanical work that drives a generator. The work output of the turbine can be calculated as:

$$W_{turb,out} = m(h_{in} - h_{out})$$

Condenser Phase
In the condenser phase, the low-pressure vapor exiting the turbine is condensed back into a liquid by rejecting heat to a cooling medium (e.g., water). The heat rejected can be calculated as:

$$Q_{out} = m(h_{in} - h_{out})$$

Cycle Efficiency
The efficiency of the ORC can be defined as the ratio of the net work output to the heat input:

$$\mu_{cycle} = \frac{W_{turbine} - W_{pump}}{Q_{in}}$$

By analyzing each phase and applying appropriate thermodynamic principles, the ORC systems can be designed and optimized for various applications, such as waste heat recovery, geothermal power generation, and solar power plants.

2. Exergy Analysis

In exergy analysis, we evaluate the efficiency and performance of a system based on the destruction and utilization of exergy, which is the maximum theoretical work that can be obtained from a system as it interacts with its environment. General formula for exergy analysis:

$$\sum \dot{Ex}_{in} - \sum \dot{Ex}_{out} - \dot{Ex}_{dest} = \Delta \dot{Ex}_s$$

For steady state system, exergy difference, $\Delta \dot{Ex}_s = 0$

$$\sum \dot{Ex}_{in} - \sum \dot{Ex}_{out} = I$$

So the in and out exergy is equals to exergy destruction which is I, irreversibility.

$$\mathrm{Ex}_{dest} = T_0 \Delta \mathrm{S}_{\mathrm{gen}}$$

For the isentropic processes, there is no entropy change so, $Ex_{dest} = 0$

Pump Phase

The exergy input to the pump can be calculated as the product of the mass flow rate and the change in specific exergy between the inlet and outlet of the pump, since there is no exergy destruction because of the isentropic compression:

$$Ex_{pump} = \dot{m}\,(e_{out} - e_{in})$$

Where e are the specific exergy of the fluid entering and leaving the pump:

$$e_i = h_i - h_0 - [T_0(s_i - s_0)]$$

Evaporator Phase

The exergy input to the evaporator is the heat transfer into the system, which can be calculated similarly to the energy input:

$$Ex_{dest} = Ex_{mass} + Ex_{heat}$$

$$Ex_{mass_{eva}} = (h_{in} - h_{out}) - T_0(s_{in} - s_{out})$$

$$Ex_{heat_{eva}} = Q_{in}(1 - \frac{T_0}{T_H})$$

$$Ex_{dest_{eva}} = (h_{in} - h_{out}) - T_0(s_{in} - s_{out}) + Q_{in}(1 - \frac{T_0}{T_H})$$

$$Ex_{dest_{eva}} = T_0(s_{out} - s_{in}) - (\frac{Q_{in}}{T_H})$$

Turbine Phase

The exergy output from the turbine is the work output of the turbine since there is no exergy destruction because of the isentropic compression:

$$Ex_{turbine} = \dot{m}\,(e_{in} - e_{out})$$

Where e for each step,

$$e_i = h_i - h_0 - [T_0(s_i - s_0)]$$

Condenser Phase

The exergy output from the condenser is the heat transfer out of the system:

$$Ex_{dest} = Ex_{mass} - Ex_{heat}$$

$$Ex_{mass_{eva}} = (h_{in} - h_{out}) - T_0(s_{in} - s_{out})$$

$$Ex_{heat_{eva}} = Q_{out}(1 - \frac{T_0}{T_L})$$

$$Ex_{dest_{eva}} = (h_{in} - h_{out}) - T_0(s_{in} - s_{out}) - Q_{out}(1 - \frac{T_0}{T_L})$$

Cycle Efficiency

The second law efficiency measures how well a system uses energy compared to how much work could be done with that energy. In the Organic Rankine Cycle (ORC), the second law efficiency measures how effectively the cycle turns heat into useful work, taking into account any inefficiencies.

$$\mu_{II,cycle} = \frac{Ex_{recovered}}{Ex_{expended}} = 1 - \frac{Ex_{destroyed}}{Ex_{expended}}$$

$$Ex_{expended} = Ex_{heat_{in}} + W_{pump_{in}}$$

$$Ex_{recovered} = W_{turbine_{out}}$$

The second law efficiency provides insight into the irreversibilities and losses within the system. A higher second law efficiency means that energy is used more efficiently and there are fewer energy losses. Improving the second law efficiency means finding ways to reduce wasted energy by designing better, optimization of operating conditions, and selection of appropriate components.

4 System Description and Modeling

The ORC cycle starts with the pump inlet. The fluid in a liquid state is pumped from low pressure to high pressure side. The fluid enters directly into evaporator, where it absorbs heat from the geothermal source. In this cycle we use two different heat source to heat the fluid. After the fluid comes out from the first one, it goes into the second one which uses solar energy as a heat source. The fluid vaporizes. The high-pressure vapor expands through a turbine, where produces mechanical work by using thermal energy. Expanding the fluid which becomes low pressure vapor enters the condenser. As a last phase the condenser, gains heat from the vapor and cools it down, then condenses back into the liquid state.

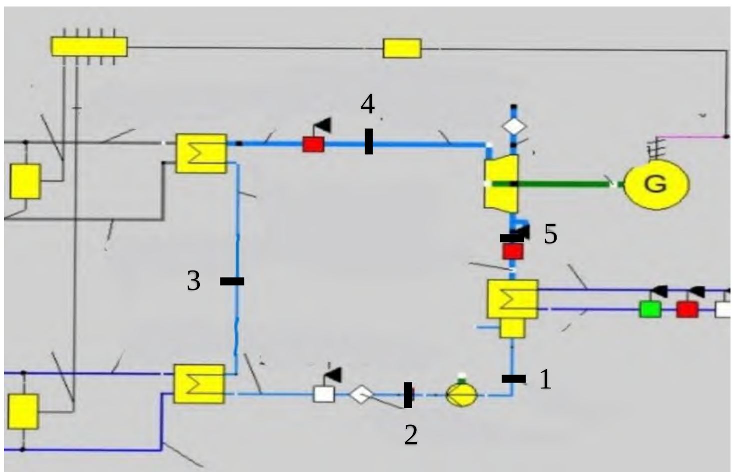


Figure 10: ORC Model

As a geothermal source, we consider Gönen Geothermal Field which located in Balıkesir, Türkiye. Heating has been provided in Balıkesir, Gönen since 1987. The Gönen geothermal source has a significant importance in presenting an opportunity for sustainable energy production, economic development in the region, as geothermal energy is renewable, and has a lower environmental impact compared to fossil fuels. So far, 17 research and production wells have been drilled at a depth varying between 133 and 180 meters in the Gönen geothermal field. Amongst of all the wells, G-9 well is chosen and used as the thermal source in the project. The well bottom temperature is taken 94 ºC and the mass flow rate of the water is taken as 8 kg/s.

The analysis of an ORC based on thermodynamic laws and the energy, exergy analyses were performed for the working fluids investigated. Here are the assumptions:

- ✓ All processes are operating at steady state.
- ✓ The thermal and friction losses are negligible.
- ✓ The kinetic and potential energy changes are negligible.
- ✓ The working fluid is in saturated liquid at condenser exit.
- ✓ Pressure drops of working fluid in the evaporator and condenser is neglected.
- ✓ The heat loss from the ORC components is negligible.
- ✓ The isentropic efficiency of expander (turbine) ηt and the pump ηp are 0.80.
- ✓ Atmospheric condition is taken as 1 atm and 293 K.
- ✓ Geothermal heat source temperature is taken as 94°C from Gönen Geothermal Source, G9 well.

Table 1 Thermo-physical properties of the selected fluids

Parameters	**R245fa**	**R507A**	**R1233zd(E)**	**R1234yf**	**R134a**
Molecular mass (g/mol)	134.05	98.9	130.5	114.0	102.03
Formula	$CF_3CH_2CHF_2$	CF_3CHF_2, CF_3CH_3	$C_3ClF_3H_2$	$C_3H_2F_4$	CF_3CH_2F
Critical Point Temperature (°C)	154.01	70.62	166.6	-29	101.1
Critical Point Pressure(MPa)	3.65	3.79	3.6237	3.3822	4.059
Boiling Temperature (˚C)	58.8	-52.1	18.1	-30	-26.3
Solar T source (˚C)	175	117	193	135	125
T sink (˚C)	20	20	20	16	18

5 Results

Although it is thermodynamically suitable to use R134a in ORC systems, it should not be used due to the damage it causes to the environment. Therefore, in this project, fluids that can replace R134a, which are R245fa, R507A, R1234yf and R1233zd (E), are being investigated and their performance and suitability is checked.

1. R245fa

We take the source temperature T_{source} sink temperature T_{sink} for the solar system as 175℃, 20℃, respectively. T_{source} for the geothermal system is taken as 94℃ and $T_{environment}$ as 20℃ for all working fluids.

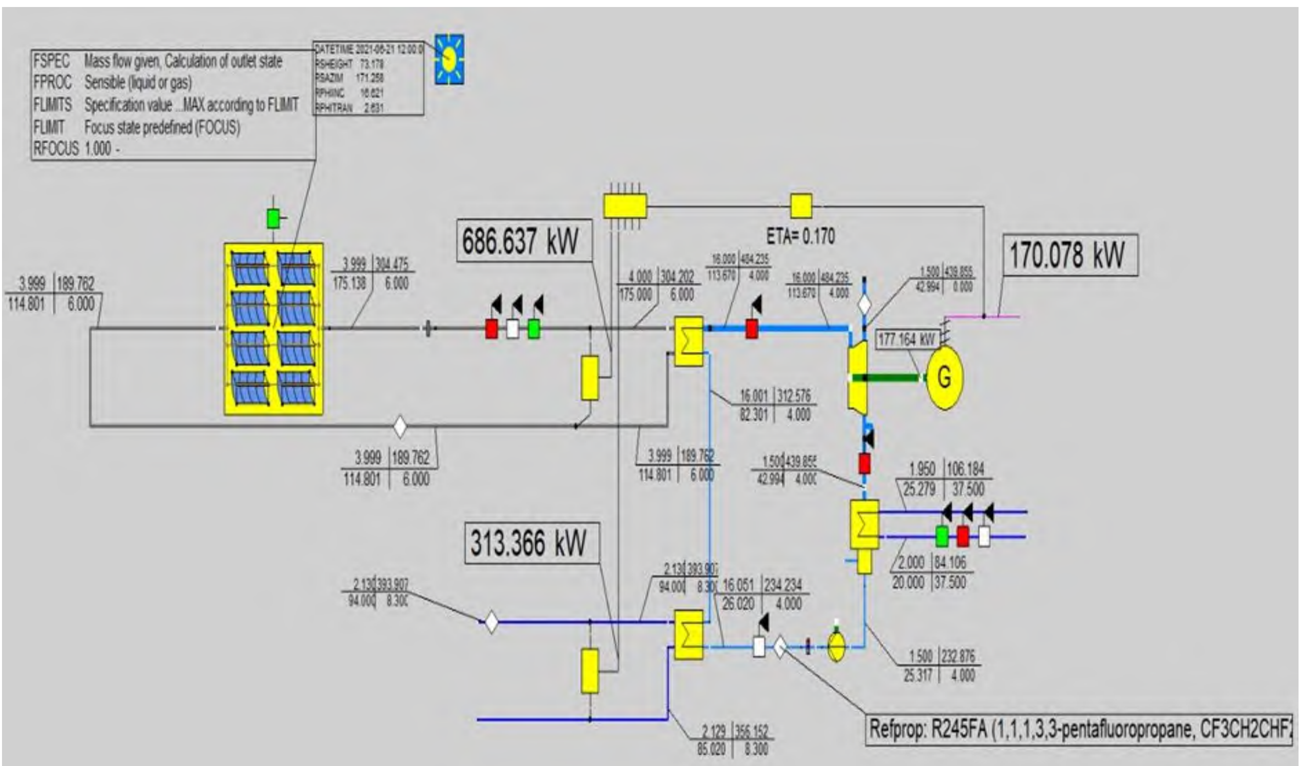


Figure 11 R245fa ORC

Table 2 Output parameters for R245fa

n_th	0,172
n_exergy	0,5793
W_out[kW]	177,4
W_in[kW]	5,412
Q_in[kW]	1000
Q_out[kW]	828
n_pump	0,8
n_turbine	1

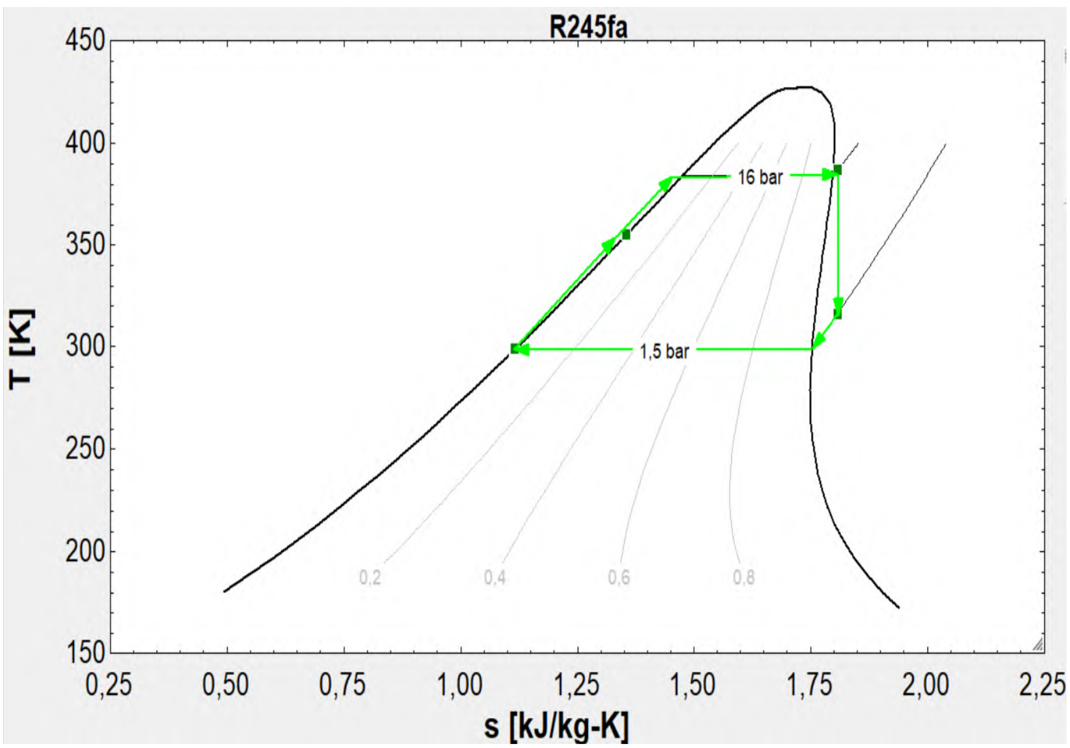


Figure 12 T-s Diagram for R245fa [11]

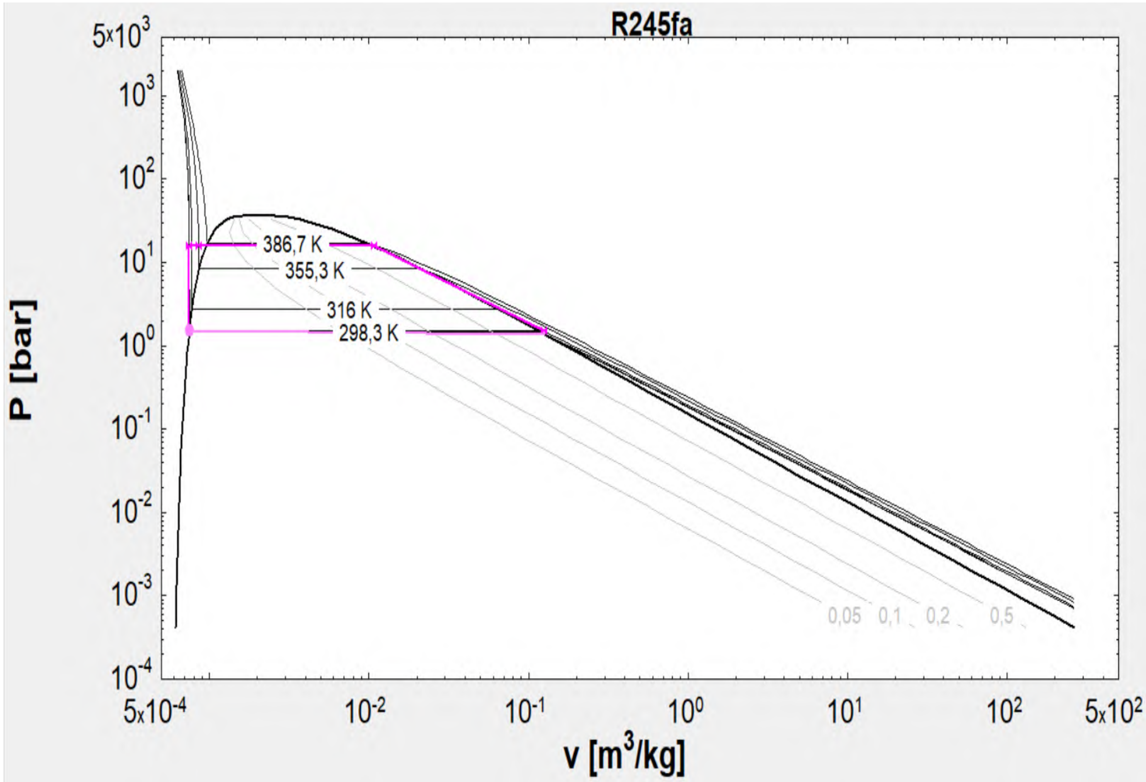


Figure 13 P-v Diagram for R245fa [12]

In general, R245fa is a commonly used and well-known fluid for ORC systems. It is good for the environment, safe to use, and works well with materials. Its good qualities make it useful for many ways to heat recovery and power generations, especially in medium-temperature conditions. However, it's important to carefully think about its limitations and whether it works well with specific operating conditions in order to make ORC systems using R245fa work better and more reliably.

2. R1234yf

We take the source temperature T_{source} sink temperature T_{sink} for the solar system as 135℃, 16℃, respectively. T_{source} for the geothermal system is taken as 94℃ and $T_{environment}$ as 20℃ for all working fluids.

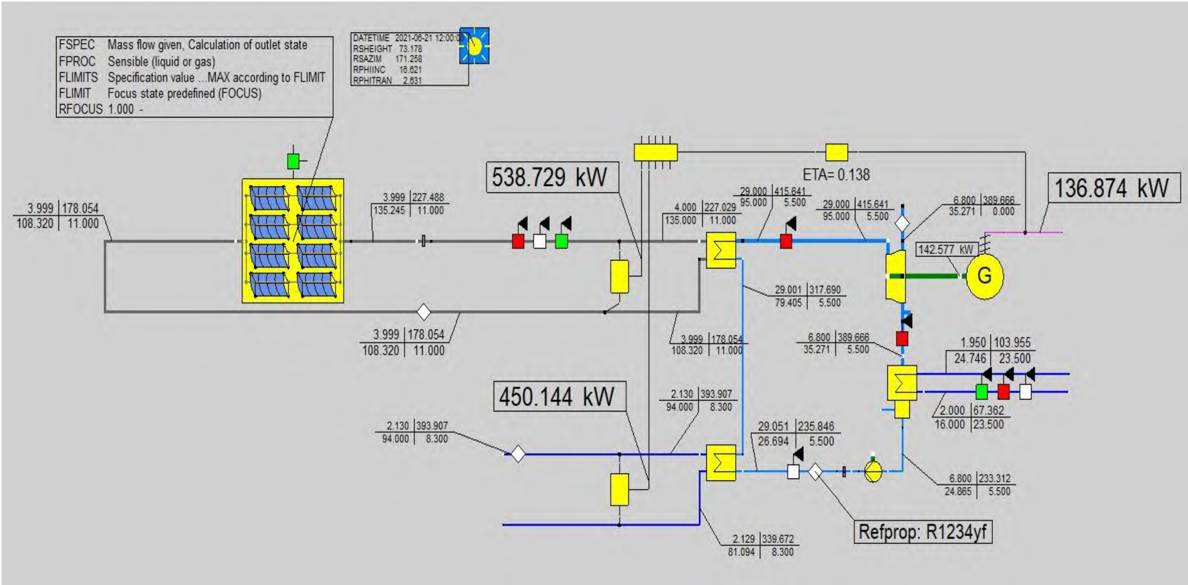


Figure 14 R1234yf Organic Rankine Cycle [13]

Table 3 Output parameters for R1234yf

n_th	0,1303
n_exergy	0,4903
W_in[kW]	13,74
W_ou[kW]t	142,6
Q_in[kW]	988,6
Q_out[kW]	859,8
n_pump	0,8
n_turbine	1

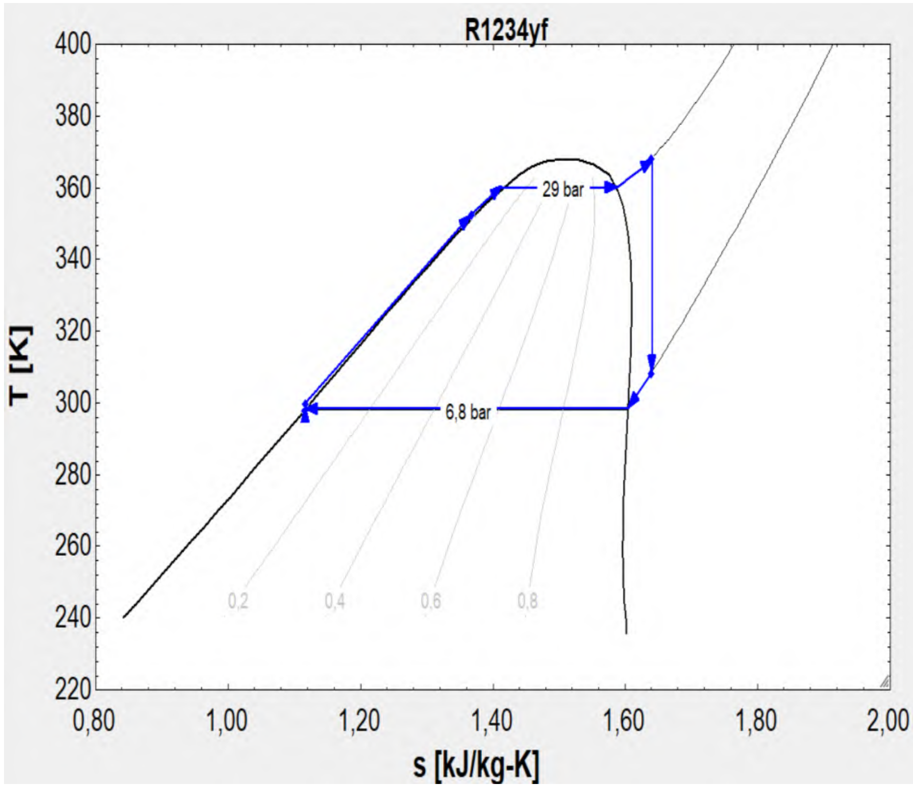


Figure 15 T-s diagram for R1234yf [14]

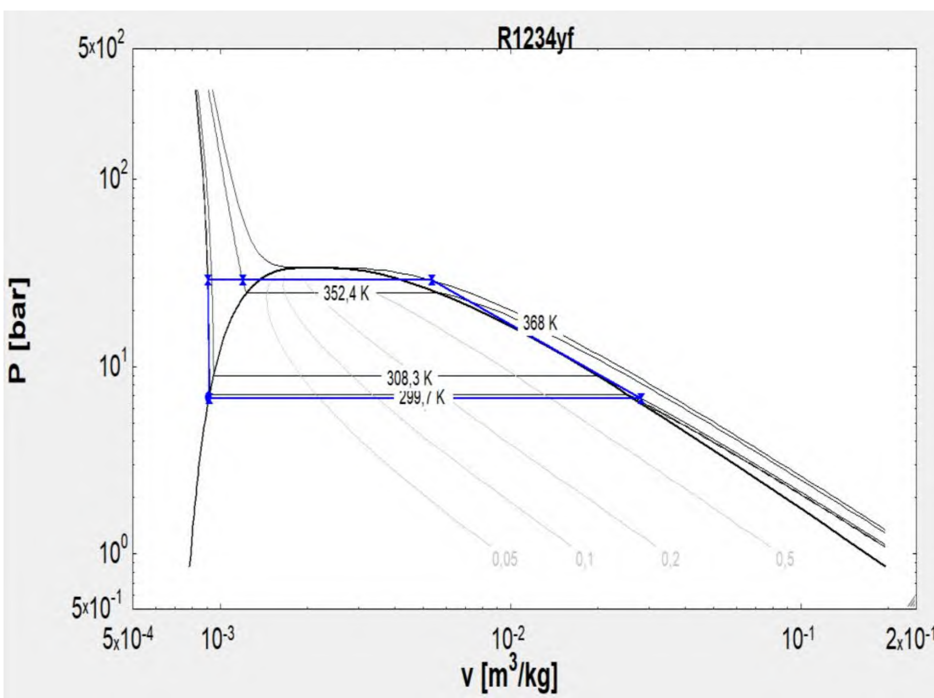


Figure 16 P-v diagram for R1234yf [15]

3. R507A

R507A as a working fluid is favorable for the ORC, is good for making energy from waste heat and works well with other things. Yet it's not good for the environment because it has a high global warming potential. This might make it not a good choice for the future. As rules change to reduce greenhouse gases, there is more focus on switching to environmentally friendly options. When choosing a refrigerant for an ORC system, it's important to think about how well it works, its impact on the environment, if it follows the rules, and how safe it is.

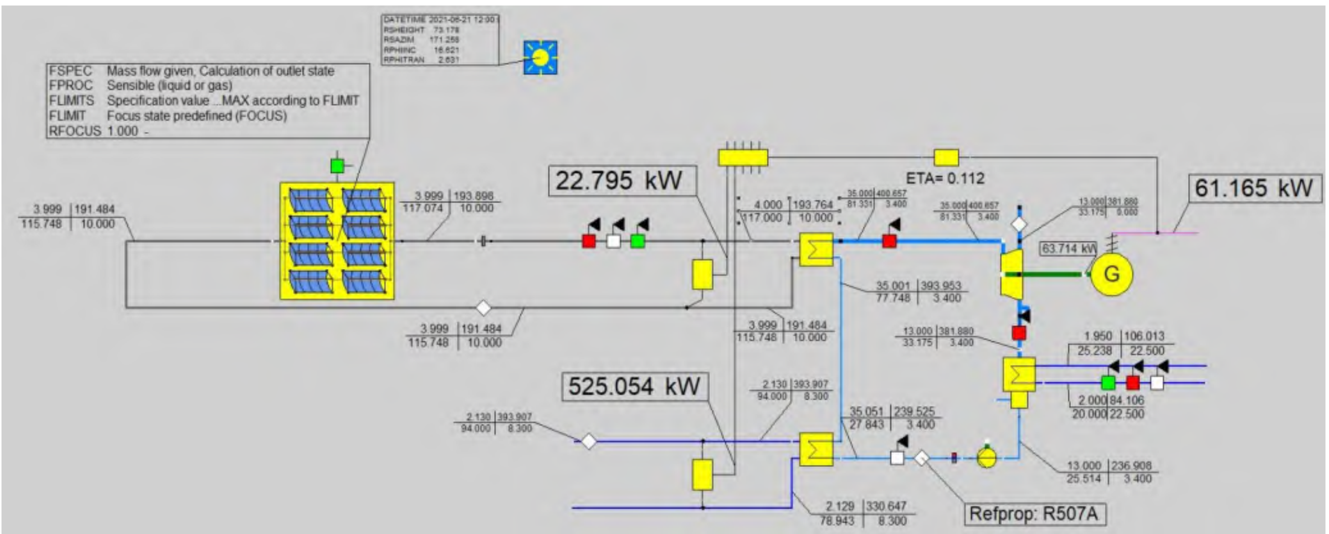


Figure 17 R507A Organic Rankine Cycle [16]

Figure 18 T-s Diagram for R507A [17]

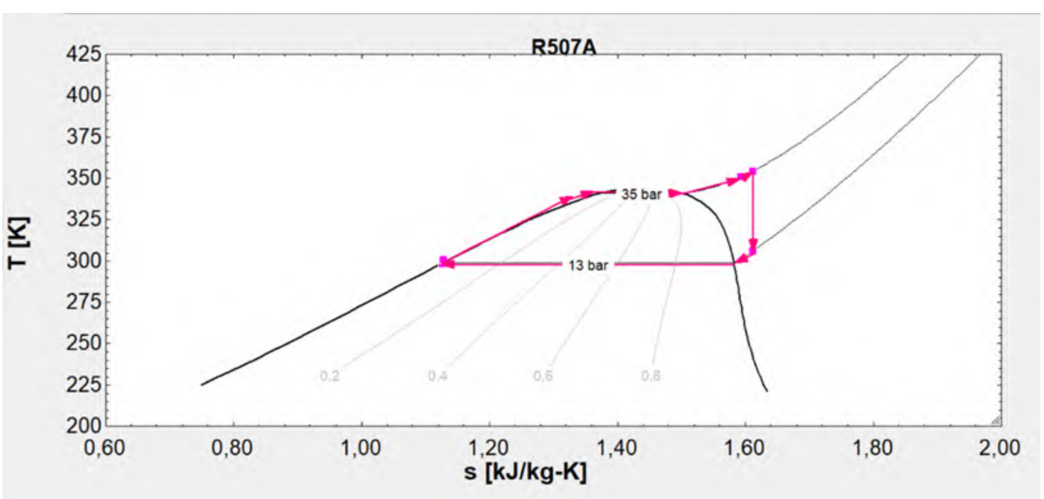


Figure 19 Pv diagram for R507A

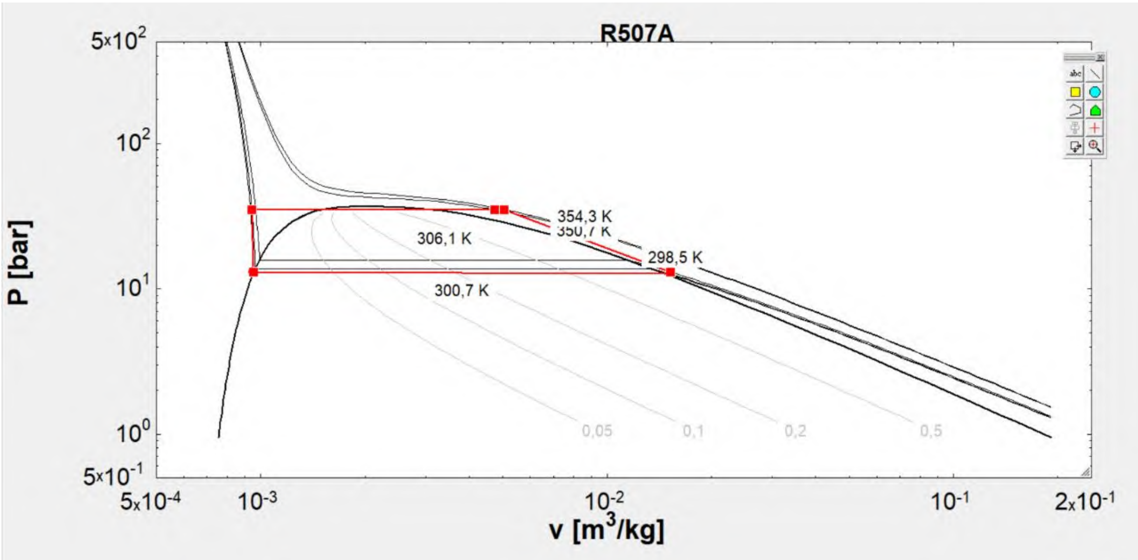


Table 4 Output parameters for R507A

n_th	0,1011
n_exergy	0,5312
W_in[kW]	8,247
W_out[kW]	63,72
Q_in[kW]	548,6
Q_out[kW]	493,2
n_pump	0,8
n_turbine	1

If we compare R507A with the other working fluids, thermal and 2nd Law efficiencies are not promising, also it is associated with environmental concerns. Therefore R507A is less appropriate as a working fluid to use in the cycle.

4. R1233zd (E)

R1233zd (E) is a promising fluid to use in ORC systems. It works well, is friendly for environment, and is safe to use. However, like any type of refrigerant, whether it is a good choice depends on the specific needs of the situation. This includes how the system works, how safe it is, and if it follows the rules and regulations. Doing careful checks and tests can help figure out if R1233zd (E) is the best option for a specific ORC application.

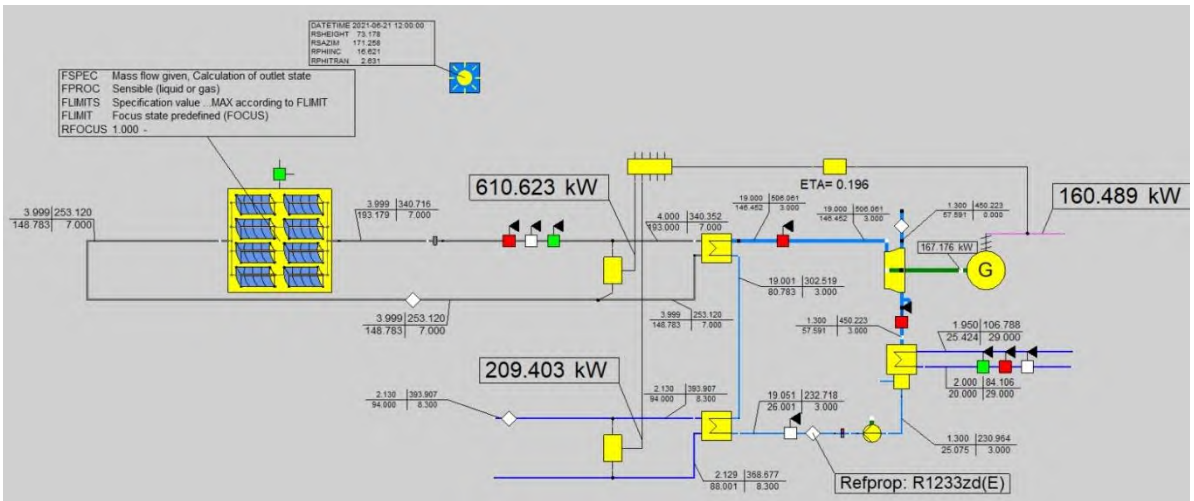


Figure 20 R1233zd(E) Organic Rankine Cycle

Table 5Output parameters for R1233zd(E)

n_th	0,1977
n_exergy	0,6105
W_in	5,248
W_out	167,4
Q_in	820,1
Q_out	657,9
n_pump	0,8
n_turbine	1

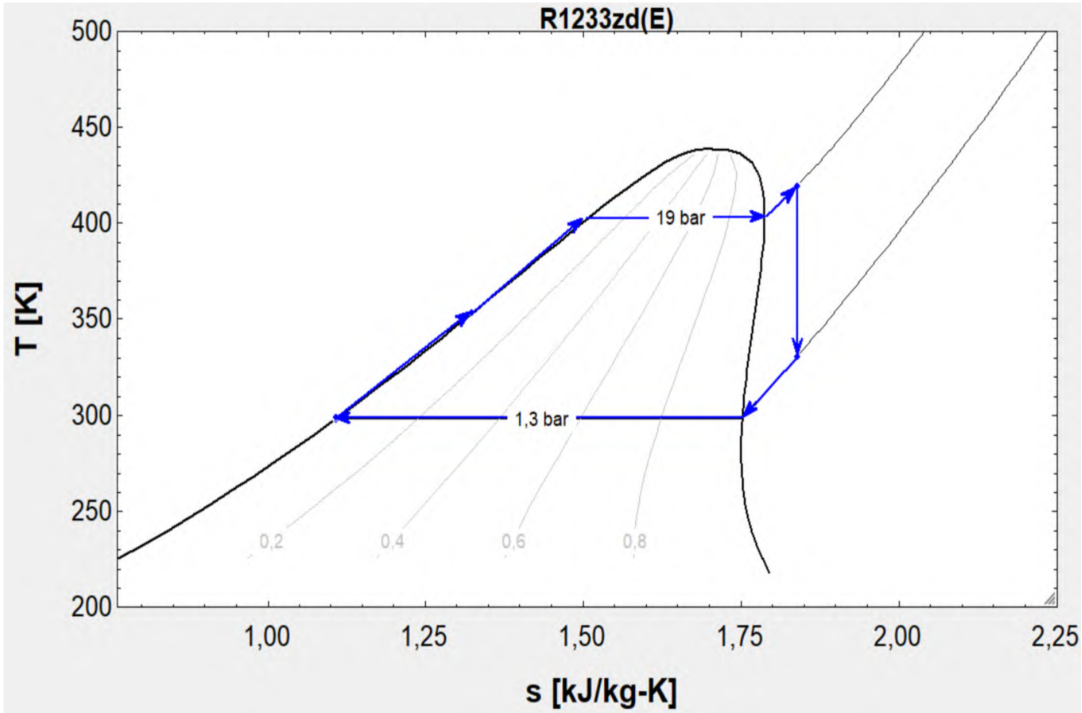


Figure 21 T-s Diagram of R1233zd(E) [19]

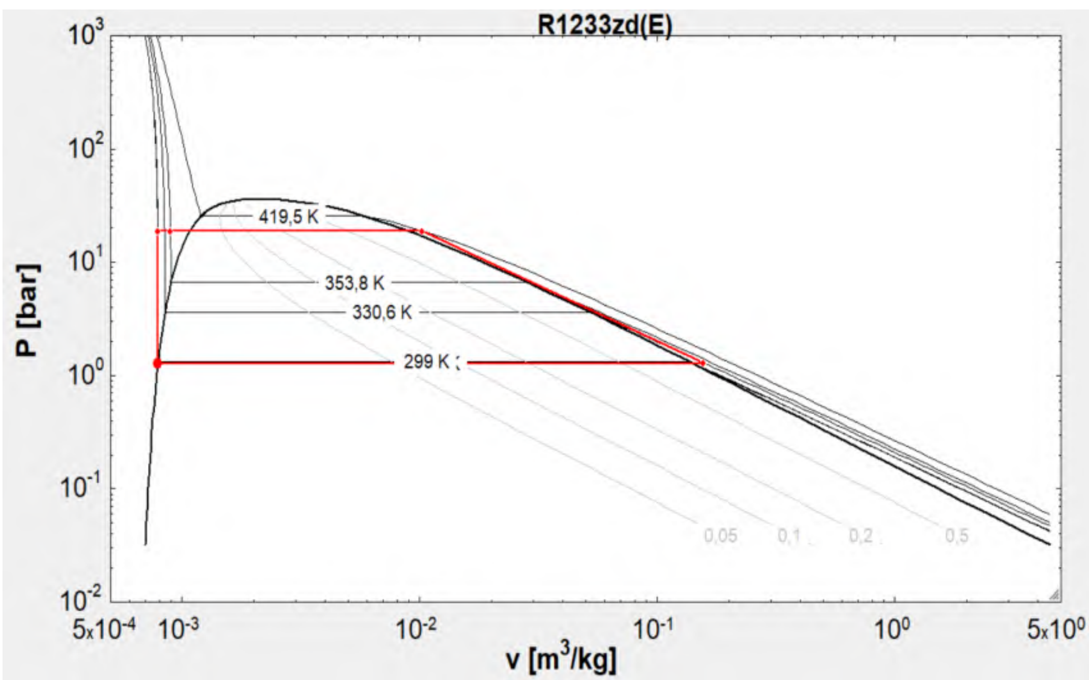
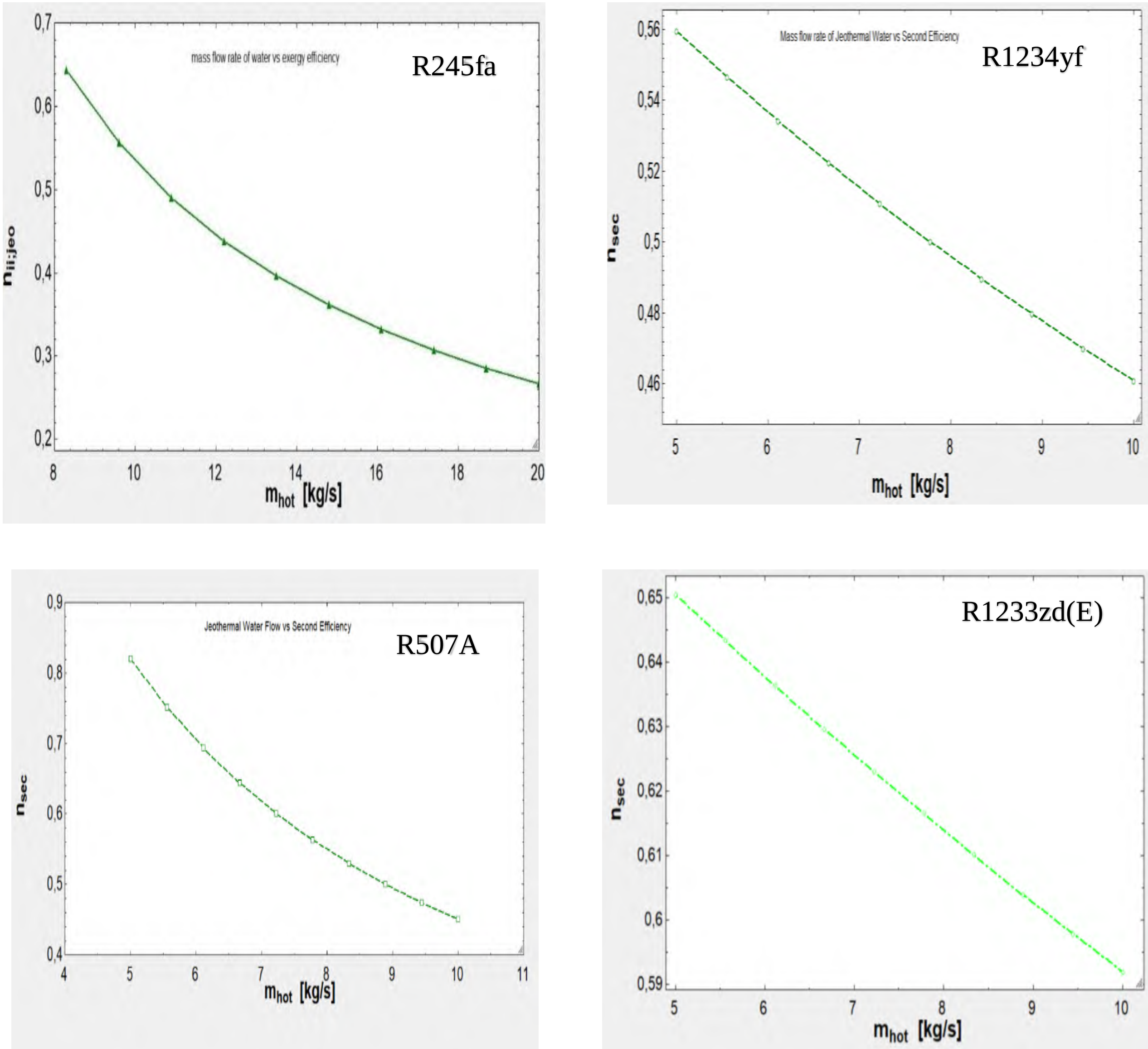


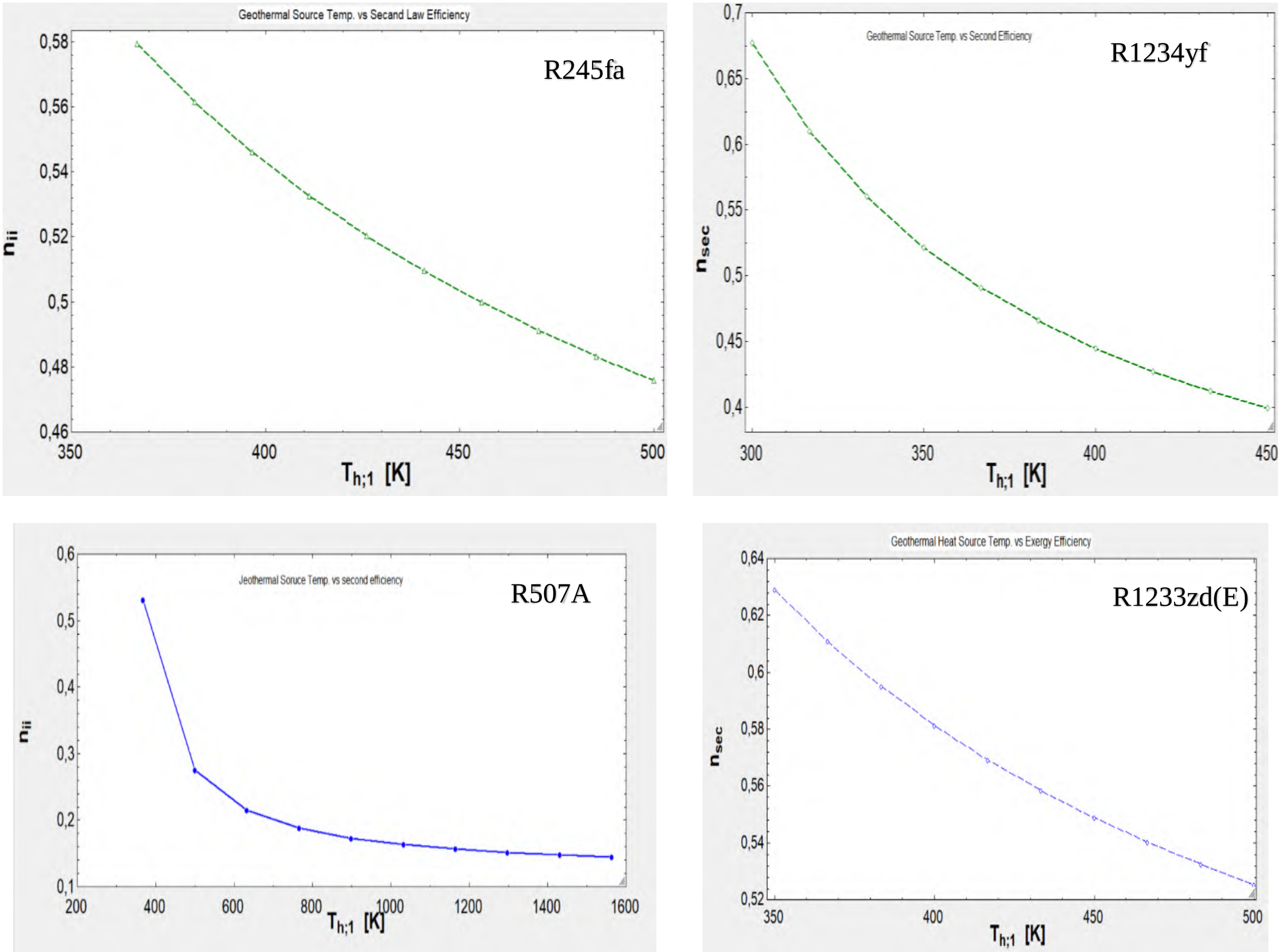
Figure 22 P-v diagram for R1233zd(E) [20]

Table 6 Second law vs geothermal source mass flow rate



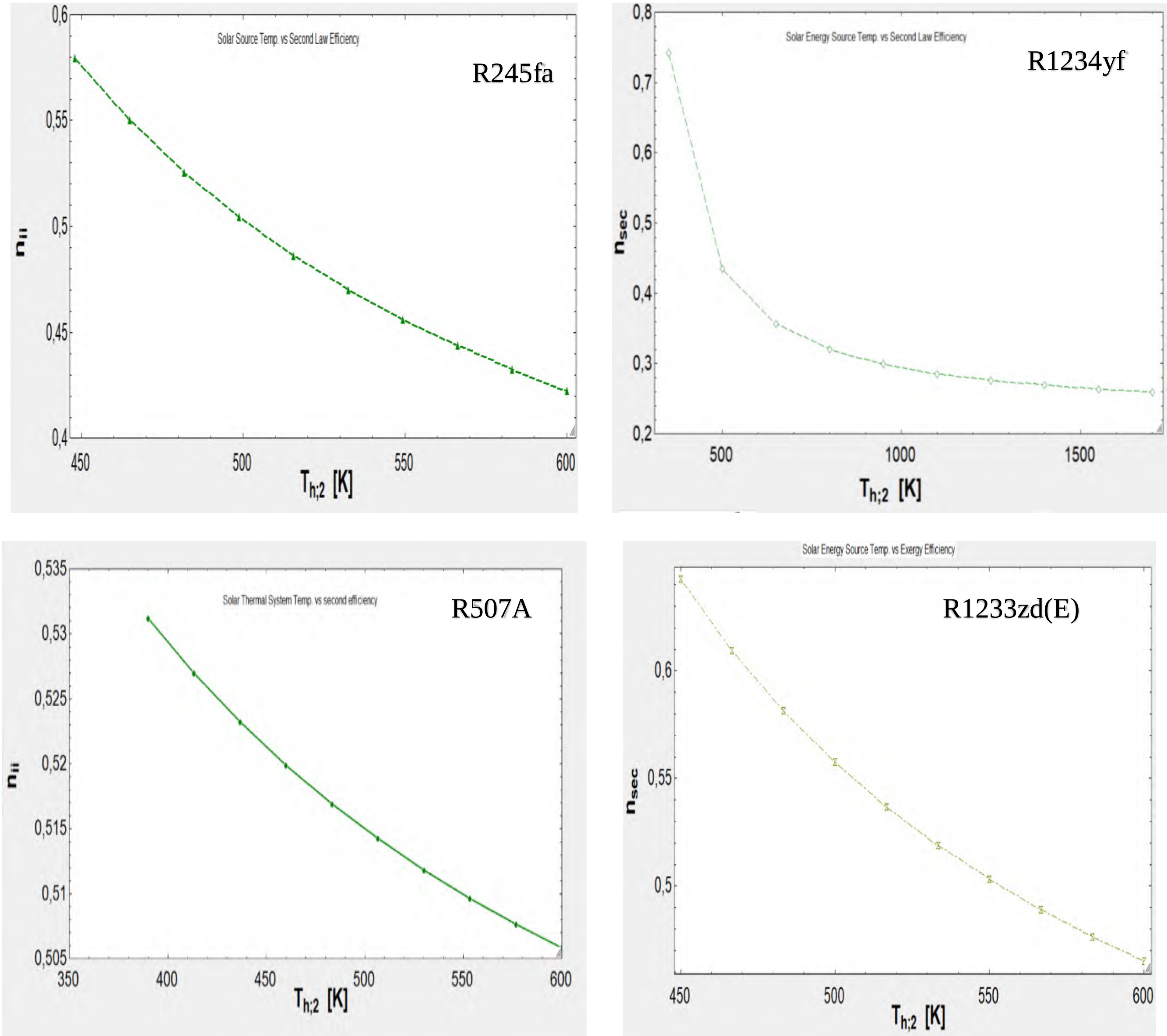
The mass flow rate of the geothermal source fluid which is water in this case also effects the exergy efficiency. The mass flow rate of the geothermal source fluid necessarily increases the heat input of the system which leads to increase of the irreversibility of the system. Therefore the irreversibility causes decrease in the exergy efficiency due to high heat input rate. But in some cases, the increase of the heat input causes high outlet temperature of the evaporator which is the inlet temperature of the turbine. High temperature input relates to high enthalpy rate, the increasing enthalpy means higher enthalpy difference for the turbine work output. In such a case, higher work output may be obtained that increases the 2nd Law efficiency.

Table 7 Second law vs geothermal source temperature



The exergy efficiency of the ORC decreased with the rise of the geothermal source temperature. The reason for the decreasing of the efficiency is related to heat transfer between the working fluid in the cycle and the geothermal water. When increasing the temperature of the source the temperature difference between the working fluid and water increases which causes increase of heat absorption by the evaporator in this case. Heat absorption also causes the increase of the irreversible loss of the system.

Table 8 Second law vs solar energy source temperature



The second law efficiency is related to Carnot Efficiency which depends on the temperature difference between the heat source and the heat sink. This leads to higher irreversibilities during the heat transfer process. High irreversibilities causes available, usable energy losses. Losing usable energy effects the exergy efficiency which in this case, it reduces the exergy efficiency.

Table 9 Total output parameters for each fluids.

Parameters / ***Fluids***	**R245fa**	**R1234yf**	**R507A**	**R1233zd(E)**
n_th	0,172	0,1303	0,1011	0,1977
n_exergy	0,5793	0,4903	0,5312	0,6105
W_in[kW]	5,412	13,74	8,247	5,248
W_out[kW]	177,4	142,6	63,72	167,4
Q_in[kW]	1000	988,6	548,6	820,1
Q_out[kW]	828	859,8	493,2	657,9
n_pump	0,8	0,8	0,8	0,8
n_turbine	0.8	0.8	0.8	0.8

6 Discussion

To choose appropriate fluid for the cycle, there are four important parameters which are effects on environment, 2nd Law efficiency, safety of the working fluid and cost.

1. Environment

- **Global Warming Potential (GWP):** The working fluid which is chosen should have low GWP to reduce its impact on climate change. High GWP fluids like hydrofluorocarbons (HFCs) and hydro chlorofluorocarbons (HCFCs) have vital impact on global warming as they penetrate the atmosphere. Using a fluid which has low GWP in the cycle helps with reducing greenhouse gas emissions.
- **Ozone Depletion Potential (ODP):** The working fluid must be selected as ozone-safe to ensure that it does not contribute to ozone depletion.
- **Biodegradability:** With time, in the components of the ORC may wear out which causes fluid leaks or spills. To consider the fluid leaks or spills, the biodegradability is also an important issue. The working fluid may contact with nature directly, in this case it should be dissolved easily to prevent harmfull impact on ecosystems.

2. Second Law Efficiency

- Main goal of the ORC system is heat recovery and power-generation by using low-to-medium grade heat sources such as solar thermal energy and geothermal energy. To reach that goal it is essential to use waste heat efficiently. 2nd Law Efficiency shows how efficiently the waste energy can be used. The 2nd Law efficiency must be optimized with the consideration of the other parameters such as thermal efficiency, environmental impacts and cost.

3. Safety

- **Flammability:** Safety is paramount when selecting a working fluid, especially in applications where ignition sources are present. Choosing non-flammable or low-flammability fluids reduces the risk of fire hazards and ensures the safety of personnel and equipment.
- **Toxicity:** Toxicity is another critical safety consideration. Working fluids should have low toxicity to minimize health risks to personnel in the event of leaks or exposure. Prioritizing fluids with low toxicity levels enhances workplace safety and compliance with health and safety regulations.

Table 10: Environmental Properties of each fluid

Fluid	Environmental Effects			Safety	Regulatory Considerations
	GWP	ODP	ALT		
R245fa	1030	0	7.6 years	Non-flammable Low toxicity	ASHRAE safety group A1
R1234yf	1	0	11 days	Mildly flammable Low toxicity	ASHRAE Safety Group A2L
R507A	3985	0	29 to 52 years	Non-flammable Low toxicity	ASHRAE Safety Group A1
R1233zd(E)	5	0	26 days	Non-flammable Low toxicity	ASHRAE Safety Group A1
R134a	1300	0	14 years	Non-flammable Low toxicity	ASHRAE Safety Group A1

4. Cost

We can classify the investment of an ORC as two categories: equipment investment and working fluid investment. During the cost analysis, we will proceed by following this classification.

In order to show the equipment's cost of the ORC system for preliminary design, the cost equations, which are widely used for analysis, synthesis and design of chemical processes, are applied [1, 2, and 3].

4.1 Equipment Investment

Equipment investment includes the investment in turbine, condenser, pump, and evaporator. The calculation of investment for turbine and pumps is based on the capacity of each component. On the other hand, the heat transfer surface area is taken into the consideration as calculation investment of the heat exchangers.

The purchased cost of heat exchanger can be estimated by the following equation [1]:

$$\log\left(C_{p,}\right) = K_1 + K_2\log(A) + K_3[\log(A)]^2$$

Where A is the total heat transfer surface area for the heat exchangers. K_1, K_2, and K_3 are the correction factors that are given in the table of "correction factors for purchased cost of heat exchanger [1]".

The purchased cost of pump and expander can be estimated by the following equation [1]:

$$\log\left(C_p\right) = K_1 + K_2\log(W) + K_3[\log(W)]^2$$

C_p represents the basic purchased cost.

Correction Factors	Value	Correction Factors	Value
K_1	3.2138	B_1	1.8
K_2	0.2688	B_2	1.5
K_3	0.0796	F_M	1.25
C_1	-0.0650	$CEPCI_{1996}$	382
C_2	0.0503	$CEPCI_{2016}$	606
C_3	0.0147		

Figure 23Correction factors for purchased cost of heat exchanger[1]

Correction Factors	Value for Pump	Value for Expander
K_1	3.4771	2.2476
K_2	0.135	1.4965
K_3	0.1438	−0.1618
C_1	−0.245832	0
C_2	0.259016	0
C_3	−0.01363	0
B_1	1.89	0
B_2	1.35	0
F_M	1.4	0
$CEPCI_{1996}$	382	382
$CEPCI_{2016}$	606	606

Figure 24Correction factors for purchased cost of pump and expander[1]

When material and pressure corrections are involved, Bare Module Cost of equipment is estimated by:

$$C_{BM} = C_P(B_1 + B_2 F_M F_P)$$

Where F_M and F_P are the material correction and pressure correction, respectively.

4.2 Working Fluid Investment

The investment cost of the working fluid is calculated simply the multiplication of total mass of the working fluid and its price per kg.

$$C_{BM} = p_w M_w$$

Where p_w is the price of the working fluid in unit of $/kg.

4.3 Total cost of ORC system:

At the beginning of the cost analysis, we have determined which type components are going to be used for our project. The type of components that we decided to use is such:

- Kaplan Turbine
- Centrifugal Pump
- Plate Heat-Exchanger
- Water Cooled Condenser

After decision of type, we have tried to evaluate the prices by getting contact with the manufacturer companies.

Total investment cost can be calculated as:

$$M_{t,c} = M_c + M_t + M_p + M_{hx}$$

The installation cost is taken as the %10 of the total cost.

$$M_{i,c} = 0.10(M_{t,c})$$

According to that, the initial investment cost can be evaluated as:

$$M_{inv} = M_{i,c} + M_{t,c}$$

The operating cost of the system consists of the cost of the refrigerant circulating in the system, the electricity cost of the pump and the cost of maintenance and repair [4]

The maintenance and repair cost of the power plant is included in the calculations as 2% of the initial investment cost [5].

$$M_{r,c} = (0.15)M_{inv}$$

4.3.1 Net Earnings (NE) of ORC:

NE is the multiplication of unit grid price and annual net power output. Net power output refers the net electricity produced by the ORC and formulized as:

$$NE = pe\left(W_t - W_p\right)t_0$$

Where pe is the unit grid price whose value is 0.185$/kWh, for Turkey. t_0 is the annual period of operation for the system which is taken as 8640h.

4.3.2 Payback Period (PP) of the ORC:

PP is the time required for accumulated benefit to become equal to the initial investment cost. PP is calculated by:

$$PP = \frac{ln\left(\frac{NE - COM}{NE - COM - iC_{tot}}\right)}{ln(1 + i)}$$

Where COM (cost of maintenance) is taken as 1.5% of the total cost, COM=1.5%C_0. Also, i represents the annual interest rate that is taken 24%, in 2024 for Turkey.

4.3.3 Return on Investment (ROI):

ROI is the ratio of NE to C_{tot}, and can be formulized as:

$$ROI = \frac{NE}{C_{tot}}$$

4.3.4 Levelized Energy Cost (LEC):

Levelized cost of energy (LCOE) is a method used to calculate the unit energy cost of electric generating plants. This method is calculated by taking the initial investment cost, operation and maintenance cost, and fuel/fluid expenses used. In this way, the minimum price at which the energy produced should be sold to avoid loss can also be calculated. LCOE can actually be evaluated as the unit energy price resulting from dividing all expenditures by the electrical energy produced [6].

$$LEC = \frac{\frac{i(1+i)^{T_S}}{i(1+i)^{T_S}-1} C_{tot} + COM}{\left(W_t - W_p\right) t_0}$$

During the calculations, the system service life T_S is taken as 20 years.

4.3.5 Present Value of Total Profit in System Service Life:

NPV is the difference between the sum of discounted expenses according to a certain discount rate and the sum of the discounted net revenues and the present value of scrap.

$$C_{pv} = \sum_{t=0}^{20} \frac{NE - COM}{(1+i)^t} - C_{tot}$$

We have made cost analysis for the R245fa and used the Bare Module ratio to other fluids (R507a, R1234yf, and R1233zd (E)) to evaluate others' cost analysis.

- We have made a cost analysis of R245fa by extracting real data from related manufacturer companies and accorded the methods in our country economical manner.
- The mass of the fluid, which is used in the cycle, is determined by the multiplication of volume of the heat exchanger and the density of the fluid.
- Other working fluid cycles' cost analyses have been evaluated by using Bare Module ratio.
- The service life is taken as 20 years and operation duration is taken as 8640 h.
- During the service life (20 yrs), the price of the electricity is taken as 0.185$/kWh.

R245fa Cost Analysis

- Kaplan Turbine ($\dot{W}_{out}$=177kW) :80,500.00$
- Centrifugal Pump ($\dot{W}_{in}$=5.412kW): 17,500.00$
- Heat Exchangers (700.00$+750.00$+3,000.00$):4,450.00$
- Solar Collector (680kW capacity): 350,000.00$
- Installation Cost (10% of Equipment Cost): (0.10x452,450.00$) = 45,245.00$
- Scrap Cost= Installation Cost = 45,245.00$
- Working Fluid Cost (For R245fa) = (25$/kg)(100kg)=2,500.00$
- Total Cost=Installation Cost+ Equipment Cost=500,195.00$
- NE (Net Earnings) = (0.185$/kWh)(Wout-Win)(24day/hours)(360days)=280,000.00$
- Present Value of Total Profit in System: (i:0,24,service life:20yrs)=630,125.62$

R1234yf Cost Analysis

- Kaplan Turbine ($\dot{W}_{out}$=142.6kW) :58,906.476$
- Centrifugal Pump ($\dot{W}_{in}$=16.74kW): 24,053.77$
- Heat Exchangers (700.00$+1,000.00$+2,350.00$):4,050.00$
- Solar Collector (538.729kW capacity): 320,000.00$
- Installation Cost (10% of Equipment Cost): (0.10x407,009.4762$) = 40,700.947$
- Scrap Cost= Installation Cost = 40,700.947$
- Working Fluid Cost (For R245fa) = (85$/kg)(100kg)=8,000.00$
- Total Cost=Installation Cost+ Equipment Cost=455,710.4232$
- NE (Net Earnings) = (0.185$/kWh)($\dot{W}_{out}$-$\dot{W}_{in}$)(24day/hours)(360days)= 205,969.824$
- Present Value of Total Profit in System: (i:0,24,service life:20yrs)=618,940.70$

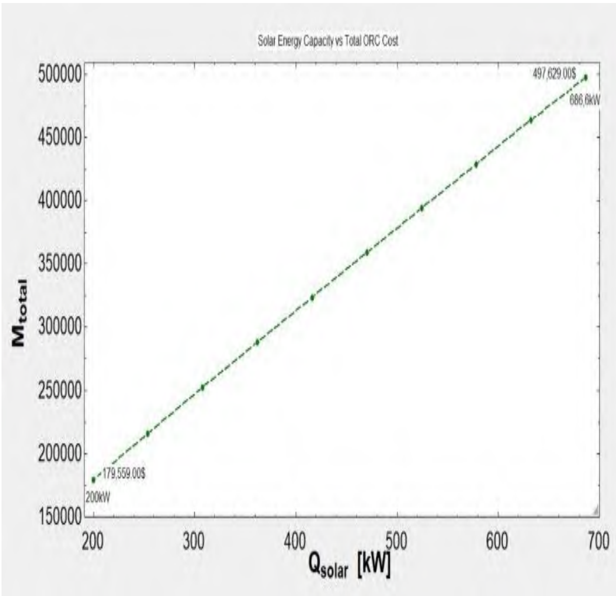
R507a Cost Analysis

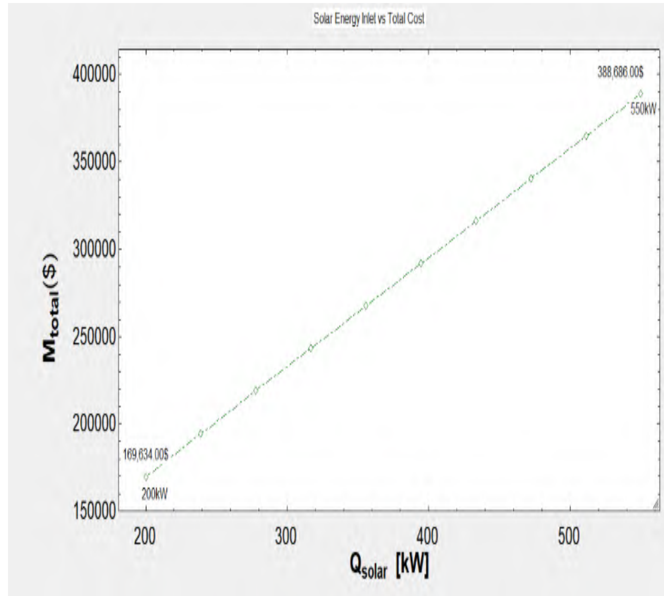
- Kaplan Turbine ($\dot{W}_{out}$=63.714kW) :26,743.206$
- Centrifugal Pump ($\dot{W}_{in}$=8.247kW): 4,845.06$
- Heat Exchangers (350.00$+1,200.00$+200.00$):1,750.00$
- Solar Collector (22.795kW capacity): 40,000.00$
- Installation Cost (10% of Equipment Cost): (0.10x73,338.266$) = 7,333.8266$
- Scrap Cost= Installation Cost = 7,333.8266$
- Working Fluid Cost (For R245fa) = (45$/kg)(100kg)=4,500.00$
- Total Cost=Installation Cost+ Equipment Cost=85,172.0926$
- NE (Net Earnings) = (0.185$/kWh)($\dot{W}_{out}$-$\dot{W}_{in}$)(24day/hours)(360days)= 88,658.4528$
- Present Value of Total Profit in System: (i:0,24,service life:20yrs)=279,236.92$

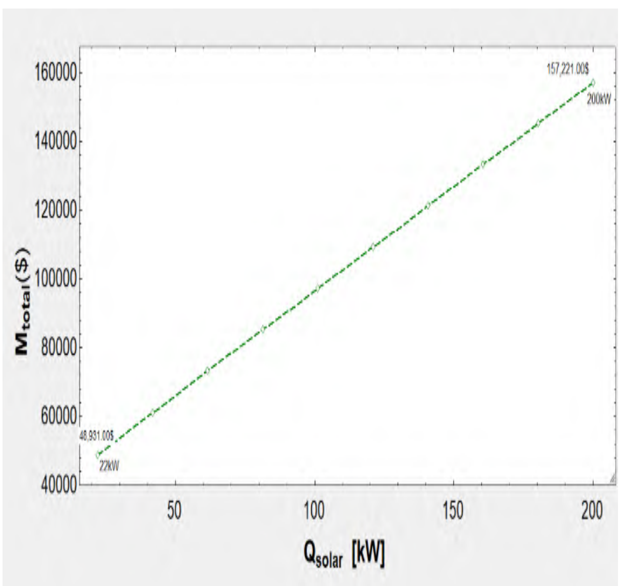
R1234yf Cost Analysis

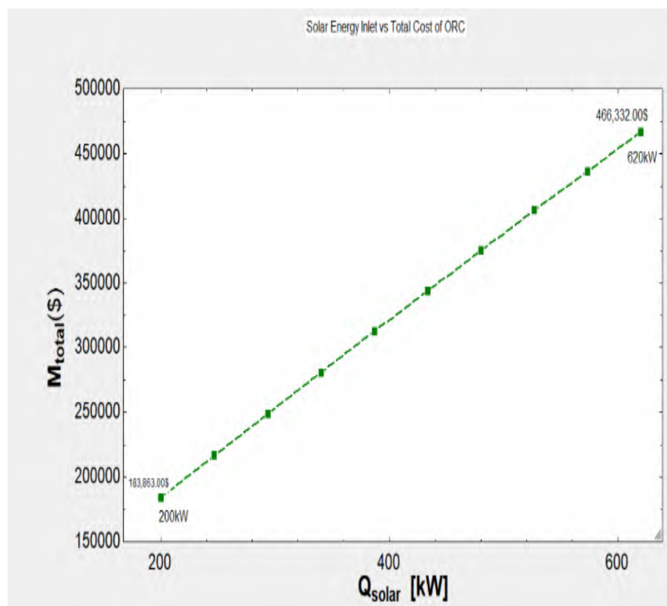
- Kaplan Turbine ($\dot{W}_{out}$=167.176kW) :79,520.00$
- Centrifugal Pump ($\dot{W}_{in}$=5.248kW): 23,850.00$
- Heat Exchangers (700.00$+1,000.00$+2,350.00$):3,750.00$
- Solar Collector (538.729kW capacity): 300,000.00$
- Installation Cost (10% of Equipment Cost): (0.10x407,120.00$) = 40,712.00$
- Scrap Cost= Installation Cost = 40,712.00$
- Working Fluid Cost (For R245fa) = (85$/kg)(100kg)=3,000.00$
- Total Cost=Installation Cost+ Equipment Cost=450,832.00$
- NE (Net Earnings) = (0.185$/kWh)($\dot{W}_{out}$-$\dot{W}_{in}$)(24day/hours)(360days)= 258,825.7152$
- Present Value of Total Profit in System: (i:0,24,service life:20yrs)=613,008.16$

Table 11: Solar Energy Input vs Total Cost

<table>
<tr><th>R245fa</th><th>R1234yf</th></tr>
<tr><td>


</td><td>


</td></tr>
<tr><th>R507A</th><th>R1233zd(E)</th></tr>
<tr><td>


</td><td>


</td></tr>
</table>

Table-11 shows how solar heat energy source that forms the majority of the total cost, affect the total cost. For ORCs, which our study is based on, there are 2 parabolic trough collectors are used for R245fa, R1234yf, and R1233zd(E). Only 1 parabolic trough collector is utilized for R507A.

When we adjust the capacity of each solar heat energy system to 200kW, the cost of each ORC is evaluated as follows;

- R507A (22kW->200kW): The total cost has increased amount of almost 220% of the original total cost for the study cycle (Figure-17).
- R1233zd(E) (620kW->200kW): The total cost has decreased amount of almost 61% of the original total cost for the study cycle (Figure-19).
- R1234yf (550kW->200kW): The total cost has decreased amount of almost 56% of the original total cost for the study cycle (Figure-13).
- R245fa (700kW->200kW): The total cost has decreased amount of almost 66% of the original total cost for the study cycle (Figure-10).

All in all, total cost ORC is mainly affected by the solar heat source energy collectors. In order to increase benefit and reduce the total cost, use of solar collectors should be minimized and geothermal energy source must be mainly utilized.

In order to make an objective comparison among the organic fluids, all parameters should be kept as same, except the type of fluid itself. Therefore, we have proceeded with analyzing the ORCs that have the same solar heat energy source capacity. Systems, which are mentioned, all have different solar energy input. In order to compare fluids with each other, the range of values where the solar energy inputs are the same was examined. By following such study, Table-13 which represents the output for each ORC fluid, is achieved.

Conclusion

The Kyoto Protocol fights against the climate change and targets to reduce greenhouse gas emission. The Kyoto protocol was adopted in Kyoto, Japan on 1997 but the protocol was entered into force on 2005. This agreement influences the choice of the working fluid of the project by limiting those with high global warming potential. In the light of the protocol, R245fa, R507A and R134a is not appropriate to use as working fluid because of their high global warming potential.

Another important criterions to choose appropriate working fluid are the ozone depletion potential and atmospheric lifetime. R1234yf and R1233zd(E) ensure the not to harm the environment by their low ODP and ALT. Therefore they can be used as alternatives instead of R245fa, R507A and R134a.

ASHRAE (the American Society of Heating, Refrigerating, and Air-Conditioning Engineers) provides classification and guidelines of refrigerant to make sure the working fluid are safe to use. According to ASHRAE Safety Classification of Refrigerants Chart, R1233zd(E) is more appropriate to use compares to R1234yf. R1234yf can still be flammable even at low levels.

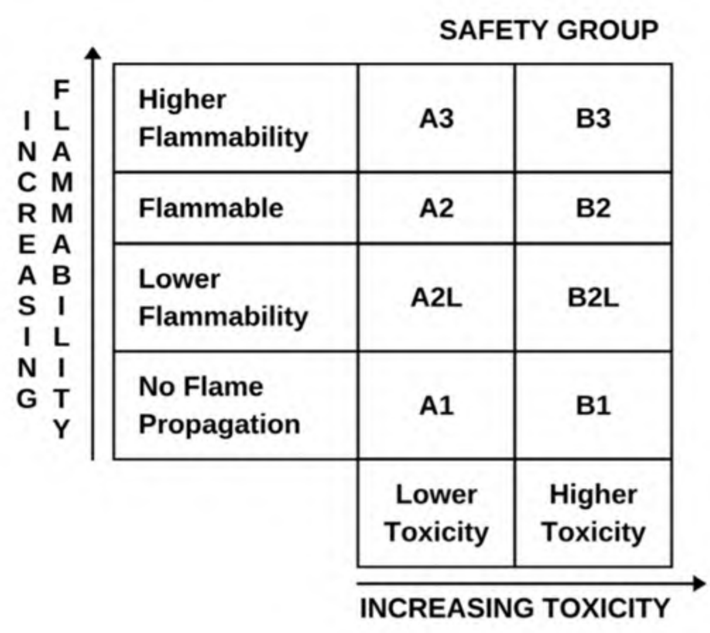


Figure 25: ASHRAE Safety Classification of Refrigerants Chart

Table 12: Outputs of each cycle

Parameters / Fluids	R245fa	R1234yf	R507A	R1233zd(E)
n_{th}	0,172	0,1303	0,1011	0,1977
n_{exergy}	0,5793	0,4903	0,5312	0,6105
W_{out}	177,4	142,6	63,72	167,4
Total Cost	500,195.00$	455,710.4232$	85,172.0926$	450,832.00$
Present Value	630,125.62$	618,940.70$	279,236.92$	613,008.16$

Up to now, we have analyzed the ORCs with respect to their organic fluids, components' capacities, and solar system energy input.

In our study, there two main important goals; the first one is that we need to select an organic working fluid for such cycles instead of R134a (Chlorofluorocarbon-CFCs) fluid, whose reactions have responsible in depletion of ozone layer. In this manner we have analyzed the ORCs and examined which organic fluids satisfy mostly and in an affordable way.

The other concern is the reduction of solar heat source energy. The point here is eliminating use of fossil fuels such as;

- Coal: Use of coal contributes to emission of high amount of CO2 which brings about the greenhouse effect, so global warming. Besides, SO2 and NOx lead to acid rains.
- Natural Gases: Although the less emission of CO2 than coal and fuel in use, natural gases also contribute to the greenhouse effect. This is also accelerated by the leakage of CH4. Another crucial point is that NOx pave the way of Ozone depletion and air pollution.
- Petroleum and its Derivatives: VOC acts a major role in Ozone depletion besides the emission of CO2 (that causes greenhouse effect) and SO2 (forms acid rains and air pollution).

In order to prevent such cases, other heat sources (solar and geothermal heat sources) are utilized. The point in our study is the reduction of solar heat energy source, which is costly to use.

As it is obviously seen in the Table-11 that the majority of the total cost is the solar heat energy source. In our study; the type of solar collectors is “parabolic trough collectors” and there are 2 solar collectors for R245fa, R1234yf, and R1233zd(E). On the other hand, there is only 1 single solar collector is used for R507A. As it is also confirmed form the Table-12, total cost is smallest for ORC whose working fluid is R507A.

In order to compare and select suitable working fluid that should be environmental-friend, eco-friend, and beneficial in use, let the solar heat energy system consist of 1 collector and energy capacity of 200kW for each ORC (R245fa, R507A, R1233zd(E), and R1234yf).

Table 13: Outputs for 200kW solar energy input for each fluid

Parameters / ***Fluids***	**R245fa**	**R1234yf**	**R507A**	**R1233zd(E)**
n_{th}	0.1273	0.1086	0.106	0.1434
n_{exergy}	0.5785	0.4798	0.4942	0.5041
W_{out}	70.75	84.32	85.15	63.97
Total Cost	179,559$	169,634$	157,221$	183,863$

We read from the Table-13 that R507 can be thought as the most affordable working fluid, again. R245fa has the highest second law of efficiency. R1234yf has the less second of efficiency.

Considering all the parameters we evaluated R1233zd(E) is the most suitable fluid for an Organic Rankine Cycle. R1233zd(E) has the highest thermal and second law efficiency besides the reasonable cost. This makes this fluid is the optimal choice for both performance and long-term financial benefits. Apart from thermodynamic and financial reasons, the environmental effects of this fluid is also appropriate. It has low global warming and ozone depletion potential, it does not last long in nature when it is compared to other fluids.

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APPENDIX

Codes Used in EES for R245fa

```
m_dot=4[kg/s]
n_pump=0,8
P1=1,5[bar]
P[1]=1,5[bar]
T1=25,317+273
T[1]=25,317+273
h1=enthalpy(R245fa;T=T1;P=P1)
s1=entropy(R245fa;T=T1;P=P1)
s[1]=entropy(R245fa;T=T1;P=P1)
v[1]=volume(R245fa;T=T1;P=P1)

P2=16[bar]
P[2]=16[bar]
h2_s=enthalpy(R245fa;s=s1;P=P2)
h2_a=((h2_s-h1)/n_pump)+h1
s2_a=entropy(R245fa;h=h2_a;P=P2)
s[2]=entropy(R245fa;h=h2_a;P=P2)
T[2]=temperature(R245fa;P=P2;h=h2_a)
T2=temperature(R245fa;P=P2;h=h2_a)
v[2]=volume(R245fa;h=h2_a;P=P2)

T3=82,301+273
T[3]=82,301+273
P[3]=P2
h3=enthalpy(R245fa;T=T3;P=P2)
s3=entropy(R245fa;T=T3;P=P2)
s[3]=entropy(R245fa;T=T3;P=P2)
v[3]=volume(R245fa;T=T3;P=P2)

{Q_solar=686,637[kW]}
h4=((Q_solar/m_dot)+h3)
P[4]=P2
T4=temperature(R245fa;P=P2;h=h4)
s4=entropy(R245fa;P=P2;h=h4)
P[5]=P1
h5=enthalpy(R245fa;s=s4;P=P1)
s[5]=s4
T[5]=temperature(R245fa;P=P1;s=s4)
T5=temperature(R245fa;P=P1;s=s4)
```

```
v[5]=volume(R245fa;s=s4;P=P1)

m_hot=8,3[kg/s]
T6=367[K]
P6=2,13[bar]
h6=enthalpy(Water;T=T6;P=P6)
s6=entropy(Water;T=T6;P=P6)

T7=358,02[K]
s7=entropy(Water;T=T7;P=P6)
h7=enthalpy(Water;T=T7;P=P6)
x_hot=m_hot*((h6-h7)-(T_0*(s6-s7)))
Q_jeo=m_hot*(h6-h7)
x_heat_67=(1-(T_0/T_h_1))*Q_jeo
x_heat34=(1-(T_0/T_h_2))*Q_solar
x_expended=x_heat_67+x_heat34
x_kazanilan=W_out-W_in
n_sec=(x_kazanilan)/(x_expended)

W_out=m_dot*(h4-h5)
wout=(h4-h5)
Q_out=m_dot*(h5-h1)
qout51=(h5-h1)
W_in=m_dot*(h2_a-h1)
win=(h2_a-h1)
Qin_23=m_dot*(h3-h2_a)
qin23=h3-h2_a
Qin_34=m_dot*(h4-h3)
qin34=h4-h3
Q_in=Qin_23+Q_solar
n_th=(W_out-W_in)/(Q_in)
T_0=293[K]
T_h_1=367[K]
x_mass_23=((h2_a-h3)-(T_0*(s2_a-s3)))
x_heat_23=(1-(T_0/T_h_1))*qin23
x_dest23=x_heat_23+x_mass_23
x_dest_23=T_0*(s3-s2_a-(qin23/T_h_1))
T_h_2=390[K]
x_mass_34=((h3-h4)-(T_0*(s3-s4)))
x_heat_34=(1-(T_0/T_h_2))*qin34
x_dest34=x_heat_34+x_mass_34
```

```
x_dest_34=T_0*(s4-s3-(qin34/T_h_2))
T_l=293[K]
x_mass_51=((h5-h1)-(T_0*(s4-s1)))
x_heat_51=(1-(T_0/T_l))*qout51
x_dest51=x_mass_51-x_heat_51
x_exp=x_heat_23+x_heat_34
x_recovered=wout-win
n_ii=(x_recovered)/(x_exp)

Cp_t=10^((2,7051)+(1,4398*(log10(W_out)))-(0,1776*((log10(W_out))^2)))
Bare_t=Cp_t*(3,4*(0,90625))
M_turbine= (Bare_t/341635,5165)*80500[$]
Cp_p=10^(3,3892+(0,0536*(log10(W_in)))+(0,1538*((log10(W_in))^2)))
Bare_p=Cp_p*(1,89+(1,35*1,6*9,66667))
M_pump=(Bare_p/73913,6084)*17500[$]
p=0,185[$/(kW*h)]
t_s=24[h]
M_electricity=(W_out-W_in)(p)(t_s)*(360)[$]
M_condenser=(700)*(Q_out/828)
M_geothermal=(700)*(Qin_23/313,366)
M_solar=(3000)*(Qin_34/686,637)
M_collector=(350000)*(Qin_34/686,637)
M_initial_investment=M_turbine+M_pump+M_condenser+M_geothermal+M_solar+M_collector
M_install=0,1*M_initial_investment
M_total=M_install+M_initial_investment
```

Codes Used in EES for R507A

```
m_dot=3,4[kg/s]
n_pump=0,8
P1=13[bar]
P[1]=13[bar]
T1=25,514+273
T[1]=25,514+273
h1=enthalpy(R507A;T=T1;P=P1)
s1=entropy(R507A;T=T1;P=P1)
s[1]=entropy(R507A;T=T1;P=P1)
v[1]=volume(R507A;T=T1;P=P1)

P2=35[bar]
P[2]=35[bar]
h2_s=enthalpy(R507A;s=s1;P=P2)
h2_a=((h2_s-h1)/n_pump)+h1
s2_a=entropy(R507A;h=h2_a;P=P2)
s[2]=entropy(R507A;h=h2_a;P=P2)
T[2]=temperature(R507A;P=P2;h=h2_a)
T2=temperature(R507A;P=P2;h=h2_a)
v[2]=volume(R507A;h=h2_a;P=P2)

T3=77,748+273
T[3]=77,748+273
P[3]=P2
h3=enthalpy(R507A;T=T3;P=P2)
s3=entropy(R507A;T=T3;P=P2)
s[3]=entropy(R507A;T=T3;P=P2)
v[3]=volume(R507A;T=T3;P=P2)

Q_solar=22,795[kW]
h4=((Q_solar/m_dot)+h3)
P[4]=P2
T4=temperature(R507A;P=P2;h=h4)
s4=entropy(R507A;P=P2;h=h4)
P[5]=P1
h5=enthalpy(R507A;s=s4;P=P1)
s[5]=s4
T[5]=temperature(R507A;P=P1;s=s4)
T5=temperature(R507A;P=P1;s=s4)
v[5]=volume(R507A;s=s4;P=P1)
```

```
m_hot=8,3[kg/s]
T6=367[K]
P6=2,13[bar]
h6=enthalpy(Water;T=T6;P=P6)
s6=entropy(Water;T=T6;P=P6)

T7=351,943[K]
s7=entropy(Water;T=T7;P=P6)
h7=enthalpy(Water;T=T7;P=P6)
x_hot=m_hot*((h6-h7)-(T_0*(s6-s7)))
Q_jeo=m_hot*(h6-h7)
x_heat_67=(1-(T_0/T_h_1))*Q_jeo
x_heat34=(1-(T_0/T_h_2))*Qin_34
x_expended=x_heat_67+x_heat34
x_kazanilan=W_out-W_in
n_sec=(x_kazanilan)/(x_expended)

W_out=m_dot*(h4-h5)
wout=(h4-h5)
Q_out=m_dot*(h5-h1)
qout51=(h5-h1)
W_in=m_dot*(h2_a-h1)
win=(h2_a-h1)
Qin_23=m_dot*(h3-h2_a)
qin23=h3-h2_a
Qin_34=m_dot*(h4-h3)
qin34=h4-h3
Q_in=Qin_23+Qin_34
n_th=(W_out-W_in)/(Q_in)
T_0=293[K]
T_h_1=367[K]
x_mass_23=((h2_a-h3)-(T_0*(s2_a-s3)))
x_heat_23=(1-(T_0/T_h_1))*qin23
x_dest23=x_heat_23+x_mass_23
x_dest_23=T_0*(s3-s2_a-(qin23/T_h_1))
T_h_2=390[K]
x_mass_34=((h3-h4)-(T_0*(s3-s4)))
x_heat_34=(1-(T_0/T_h_2))*qin34
x_dest34=x_heat_34+x_mass_34
x_dest_34=T_0*(s4-s3-(qin34/T_h_2))
```

```
T_l=293[K]
x_mass_51=((h5-h1)-(T_0*(s4-s1)))
x_heat_51=(1-(T_0/T_l))*qout51
x_dest51=x_mass_51-x_heat_51
x_exp=x_heat_23+x_heat_34
x_recovered=wout-win
n_ii=(x_recovered)/(x_exp)

Cp_t=10^((2,7051)+(1,4398*(log10(W_out)))-(0,1776*((log10(W_out))^2)))
Bare_t=Cp_t*(3,4*(0,63))
M_turbine= (Bare_t/341635,5165)*80500[$]
Cp_p=10^(3,3892+(0,0536*(log10(W_in)))+(0,1538*((log10(W_in))^2)))
Bare_p=Cp_p*(1,89+(1,35*1,6*1,69))
M_pump=(Bare_p/73913,6084)*17500[$]
p=0,185[$/(kW*h)]
t_s=24[h]
M_electricity=(W_out-W_in)(p)(t_s)*(360)[$]
M_condenser=(700)*(Q_out/828)
M_geothermal=(700)*(Qin_23/313,366)
M_solar=(3000)*(Qin_34/686,637)
M_collector=(350000)*(Qin_34/686,637)
M_initial_investment=M_turbine+M_pump+M_condenser+M_geothermal+M_solar+M_collector
M_install=0,1*M_initial_investment
M_total=M_install+M_initial_investment
```

Codes Used in EES for R1233zd(E)

```
m_dot=3[kg/s]
n_pump=0,8
P1=1,3[bar]
P[1]=1,3[bar]
T1=25,075+273
T[1]=25,075+273
h1=enthalpy(R1233zd(E);T=T1;P=P1)
s1=entropy(R1233zd(E);T=T1;P=P1)
s[1]=entropy(R1233zd(E);T=T1;P=P1)
v[1]=volume(R1233zd(E);T=T1;P=P1)

P2=19[bar]
P[2]=19[bar]
h2_s=enthalpy(R1233zd(E);s=s1;P=P2)
h2_a=((h2_s-h1)/n_pump)+h1
s2_a=entropy(R1233zd(E);h=h2_a;P=P2)
s[2]=entropy(R1233zd(E);h=h2_a;P=P2)
T[2]=temperature(R1233zd(E);P=P2;h=h2_a)
T2=temperature(R1233zd(E);P=P2;h=h2_a)
v[2]=volume(R1233zd(E);h=h2_a;P=P2)

T3=80,783+273
T[3]=80,783+273
P[3]=P2
h3=enthalpy(R1233zd(E);T=T3;P=P2)
s3=entropy(R1233zd(E);T=T3;P=P2)
s[3]=entropy(R1233zd(E);T=T3;P=P2)
v[3]=volume(R1233zd(E);T=T3;P=P2)

Q_solar=610,623[kW]
h4=((Q_solar/m_dot)+h3)
P[4]=P2
T4=temperature(R1233zd(E);P=P2;h=h4)
s4=entropy(R1233zd(E);P=P2;h=h4)
P[5]=P1
h5=enthalpy(R1233zd(E);s=s4;P=P1)
s[5]=s4
T[5]=temperature(R1233zd(E);P=P1;s=s4)
T5=temperature(R1233zd(E);P=P1;s=s4)
```

```
v[5]=volume(R1233zd(E);s=s4;P=P1)

m_hot=8,3[kg/s]
T6=367[K]
P6=2,13[bar]
h6=enthalpy(Water;T=T6;P=P6)
s6=entropy(Water;T=T6;P=P6)

T7=361,001[K]
s7=entropy(Water;T=T7;P=P6)
h7=enthalpy(Water;T=T7;P=P6)
x_hot=m_hot*((h6-h7)-(T_0*(s6-s7)))
Q_jeo=m_hot*(h6-h7)
x_heat_67=(1-(T_0/T_h_1))*Q_jeo
x_heat34=(1-(T_0/T_h_2))*Qin_34
x_expended=x_heat_67+x_heat34
x_kazanilan=W_out-W_in
n_sec=(x_kazanilan)/(x_expended)

W_out=m_dot*(h4-h5)
wout=(h4-h5)
Q_out=m_dot*(h5-h1)
qout51=(h5-h1)
W_in=m_dot*(h2_a-h1)
win=(h2_a-h1)
Qin_23=m_dot*(h3-h2_a)
qin23=h3-h2_a
Qin_34=m_dot*(h4-h3)
qin34=h4-h3
Q_in=Qin_23+Qin_34
n_th=(W_out-W_in)/(Q_in)
T_0=293[K]
T_h_1=367[K]
x_mass_23=((h2_a-h3)-(T_0*(s2_a-s3)))
x_heat_23=(1-(T_0/T_h_1))*qin23
x_dest23=x_heat_23+x_mass_23
x_dest_23=T_0*(s3-s2_a-(qin23/T_h_1))
T_h_2=466[K]
x_mass_34=((h3-h4)-(T_0*(s3-s4)))
x_heat_34=(1-(T_0/T_h_2))*qin34
x_dest34=x_heat_34+x_mass_34
```

```
x_dest_34=T_0*(s4-s3-(qin34/T_h_2))
T_l=293[K]
x_mass_51=((h5-h1)-(T_0*(s4-s1)))
x_heat_51=(1-(T_0/T_l))*qout51
x_dest51=x_mass_51-x_heat_51
x_exp=x_heat_23+x_heat_34
x_recovered=wout-win
n_ii=(x_recovered)/(x_exp)

Cp_t=10^((2,7051)+(1,4398*(log10(W_out)))-(0,1776*((log10(W_out))^2)))
Bare_t=Cp_t*(3,4*(0,9316))
M_turbine= (Bare_t/341635,5165)*80500[$]
Cp_p=10^(3,3892+(0,0536*(log10(W_in)))+(0,1538*((log10(W_in))^2)))
Bare_p=Cp_p*(1,89+(1,35*1,6*13,6154))
M_pump=(Bare_p/73913,6084)*17500[$]
p=0,185[$/(kW*h)]
t_s=24[h]
M_electricity=(W_out-W_in)(p)(t_s)*(360)[$]
M_condenser=(700)*(Q_out/828)
M_geothermal=(700)*(Qin_23/313,366)
M_solar=(3000)*(Qin_34/686,637)
M_collector=(350000)*(Qin_34/686,637)
M_initial_investment=M_turbine+M_pump+M_condenser+M_geothermal+M_solar+M_collector
M_install=0,1*M_initial_investment
M_total=M_install+M_initial_investment
```

Codes Used in EES for R1234yf

```
m_dot=5,5[kg/s]
n_pump=0,8
P1=6,8[bar]
P[1]=6,8[bar]
T1=24,685+273
T[1]=24,685+273
h1=enthalpy(R1234yf;T=T1;P=P1)
s1=entropy(R1234yf;T=T1;P=P1)
s[1]=entropy(R1234yf;T=T1;P=P1)
v[1]=volume(R1234yf;T=T1;P=P1)

P2=29[bar]
P[2]=29[bar]
h2_s=enthalpy(R1234yf;s=s1;P=P2)
h2_a=((h2_s-h1)/n_pump)+h1
s2_a=entropy(R1234yf;h=h2_a;P=P2)
s[2]=entropy(R1234yf;h=h2_a;P=P2)
T[2]=temperature(R1234yf;P=P2;h=h2_a)
T2=temperature(R1234yf;P=P2;h=h2_a)
v[2]=volume(R1234yf;h=h2_a;P=P2)

T3=79,405+273
T[3]=79,405+273
P[3]=P2
h3=enthalpy(R1234yf;T=T3;P=P2)
s3=entropy(R1234yf;T=T3;P=P2)
s[3]=entropy(R1234yf;T=T3;P=P2)
v[3]=volume(R1234yf;T=T3;P=P2)

Q_solar=538,729[kW]
h4=((Q_solar/m_dot)+h3)
P[4]=P2
T4=temperature(R1234yf;P=P2;h=h4)
s4=entropy(R1234yf;P=P2;h=h4)
P[5]=P1
h5=enthalpy(R1234yf;s=s4;P=P1)
s[5]=s4
T[5]=temperature(R1234yf;P=P1;s=s4)
T5=temperature(R1234yf;P=P1;s=s4)
v[5]=volume(R1234yf;s=s4;P=P1)
```

```
m_hot=8,3[kg/s]
T6=367[K]
P6=2,13[bar]
h6=enthalpy(Water;T=T6;P=P6)
s6=entropy(Water;T=T6;P=P6)

T7=354,094[K]
s7=entropy(Water;T=T7;P=P6)
h7=enthalpy(Water;T=T7;P=P6)
x_hot=m_hot*((h6-h7)-(T_0*(s6-s7)))
Q_jeo=m_hot*(h6-h7)
x_heat_67=(1-(T_0/T_h_1))*Q_jeo
x_heat34=(1-(T_0/T_h_2))*Qin_34
x_expended=x_heat_67+x_heat34
x_kazanilan=W_out-W_in
n_sec=(x_kazanilan)/(x_expended)

W_out=m_dot*(h4-h5)
wout=(h4-h5)
Q_out=m_dot*(h5-h1)
qout51=(h5-h1)
W_in=m_dot*(h2_a-h1)
win=(h2_a-h1)
Qin_23=m_dot*(h3-h2_a)
qin23=h3-h2_a
Qin_34=m_dot*(h4-h3)
qin34=h4-h3
Q_in=Qin_23+Qin_34
n_th=(W_out-W_in)/(Q_in)
T_0=293[K]
T_h_1=367[K]
x_mass_23=((h2_a-h3)-(T_0*(s2_a-s3)))
x_heat_23=(1-(T_0/T_h_1))*qin23
x_dest23=x_heat_23+x_mass_23
x_dest_23=T_0*(s3-s2_a-(qin23/T_h_1))
T_h_2=408[K]
x_mass_34=((h3-h4)-(T_0*(s3-s4)))
x_heat_34=(1-(T_0/T_h_2))*qin34
x_dest34=x_heat_34+x_mass_34
x_dest_34=T_0*(s4-s3-(qin34/T_h_2))
```

```
T_l=289[K]
x_mass_51=((h5-h1)-(T_0*(s4-s1)))
x_heat_51=(1-(T_0/T_l))*qout51
x_dest51=x_mass_51-x_heat_51
x_exp=x_heat_23+x_heat_34
x_recovered=wout-win
n_ii=(x_recovered)/(x_exp)

Cp_t=10^((2,7051)+(1,4398*(log10(W_out)))-(0,1776*((log10(W_out))^2)))
Bare_t=Cp_t*(3,4*(0,7655))
M_turbine= (Bare_t/341635,5165)*80500[$]
Cp_p=10^(3,3892+(0,0536*(log10(W_in)))+(0,1538*((log10(W_in))^2)))
Bare_p=Cp_p*(1,89+(1,35*1,6*3,265))
M_pump=(Bare_p/73913,6084)*17500[$]
p=0,185[$/(kW*h)]
t_s=24[h]
M_electricity=(W_out-W_in)*(p)*(t_s)*(360)[$]
M_condenser=(700)*(Q_out/828)
M_geothermal=(700)*(Qin_23/313,366)
M_solar=(3000)*(Qin_34/686,637)
M_collector=(350000)*(Qin_34/686,637)
M_initial_investment=M_turbine+M_pump+M_condenser+M_geothermal+M_solar+M_collector
M_install=0,1*M_initial_investment
M_total=M_install+M_initial_investment
```