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THERMODYNAMIC ANALYSIS OF THE ORGANIC RANKINE CYCLE AND THE EFFECT OF REFRIGERANT SELECTION ON CYCLE PERFORMANCE

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Abstract

The Organic Rankine cycle is a power-generation system for lower temperature ranges in which organic fluids with hydrocarbon components are used instead of water. Organic Rankine Cycles, which are suitable for heat recovery applications at low temperatures, can be used for generating electric energy from various waste heat sources. In this study, a thermodynamic analysis is conducted on an example Organic Rankine Cycle that is used to generate electric energy from a geothermal source. The working fluid to be used in the cycle was selected as R134a, R236fa, R245fa and R600a, which are commonly used. For these selected organic fluids, the required cycle performance to generate 1 MW of energy from the turbine was analyzed according to the geothermal source temperature (90-140°C), and the thermal efficiency of the cycle was calculated. The obtained results are presented comparatively with the help of the graphs. R245fa was defined to be more appropriate for the cycle as a refrigerant at constant work conditions.

Key words: *Organic Rankine Cycle, Refrigerant, Thermodynamic analysis, Cycle performance, Thermal efficiency*

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1. Introduction

Steam power plants are cycles that use water as refrigerant. Thermodynamic model of these power plants is the Rankine cycle. Rankine cycle transforms thermal energy into work. Water is commonly used in the cycle as a working fluid and it is preferred in medium and big-size power plants for the generation of electric power.

Water has significant properties such as being safe, environment-friendly and it can enable high heat transfer, it also has certain disadvantages. Some of these disadvantages are that; it is highly corrosive and it has a rather high freezing temperature. Tchanche et al. [1] investigated the physical and chemical properties, the advantages and disadvantages of water and organic fluids.

In recent years, higher-molecular mass hydrocarbon compound fluids have been used in Rankine cycle instead of water. With the use of these fluids, the systems have been referred to as the Organic Rankine cycle and they have been included in the power generation processes that are most common in thermal, solar and geo-thermal power etc. applications in factories [2-4].

Thermodynamic properties of certain fluids used in the Rankine cycle are presented in Table 1. Many studies have shown that; among the hydrocarbon compound organic fluids used in the Organic Rankine cycle systems, the ones that have higher molecular mass, low critical temperature and pressure and that are dry and isentropic in the meantime are more appropriate [5-8]. The first theoretical studies with different fluids (R11, R12, R113 and R134a) in Organic Rankine cycle system were conducted by Hung [9].

Table 1. Thermodynamic properties of certain organic fluids.

Organic Fluid	Type of Fluid	Molecular Mass (g/mol)	Critical Temperature (K)	Critical Pressure (Mpa)	ODV	GWP
R123 ($CHCl_2CF_3$)	isentropic	152,93	456,8	3,66	0.060	77 (low)
R134a (CF_2CH_2F)	wet	102,03	374,2	4,06	0	1430 (medium)
R152a ($C_2H_4F_2$)	wet	66,05	386,2	4,52	0	124 (low)
R236fa ($CF_3CH_2CF_3$)	isentropic	152,04	398,1	3,2	0	9810 (high)
R245fa ($C_3H_3F_5$)	isentropic	134,05	427,2	3,64	0	1030 (medium)
R600a (C_4H_{10})	dry	58,1	408,1	3,65	0	3 (low)
R717 (NH_3)	wet	17,03	405,4	11,3	0	0 (zero)

Yamamoto et al. [10] showed that; in Rankine cycle, the utilization of the fluid R123 instead of water could provide a higher performance value. Lui et al. [11] investigated the impact of the critical temperature of organic fluids on performance. Teng et al. [12] classified the organic dry type fluids.

R245fa, which is one of the isentropic fluids, have become popular recently and there are numerous theoretical and experimental studies on it [13-14]. Cihan [15] modeled a system in which the organic Rankine cycle that operates with low-temperature waste heat is combined with the traditional vapor-compression refrigeration cycle. He used R600, R600a and R601 as fluid.

Pulyaev et al. [16] conducted the thermodynamic analysis of the generation of electricity through the organic Rankine cycle by making use of the waste heat that comes out during the transfer of the natural gas to the turbine once it has been pressurized in the combined cycle plant. Ozden and Paul [17] investigated the Saraykoy Geothermal Power Plant which generates electricity through the organic Rankine cycle.

In recent years, with the use of renewable energy resources, the important of ORC systems has become popular. For this reason, the number of studies about these systems has increased in both academic and industrial fields. This study presents a performance evaluation of the most commonly used refrigerant in the industrial ORC systems. R134a, R236fa, R245fa and R600a were determined as refrigerants to

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be used in the Organic Rankine cycle that generates electric power through a geothermal resource. For these refrigerants, the cycle performance required to obtain 1 MW work from the turbine was analyzed based on the temperature of the geothermal resource (90-140°C) and the thermal efficiency was measured. R245fa was defined to be more appropriate for the cycle as a refrigerant at constant work conditions.

2. Materials and Methods

2.1. Organic Rankine Cycle

Organic Rankine cycles function with the same principle as Rankine cycles in terms of thermodynamics. The only difference between Organic Rankine cycles and Rankine Cycles is the fact that; the fluid is an organic fluid that evaporates in lower temperatures compared to water. The cycle consists of a pump, boiler, turbine and condenser. The turbines used in Organic Rankine cycles only need a single step expansion and thus, they are more practical and economical compared to the conventional steam turbines.

The schematic diagram of the organic Rankine cycle used to generate electric power from various thermal sources (such as bio-mass, geo-thermal, exhaust gases etc.) is given in Fig. 1. The Fig. 2 presents the T-s diagram of the cycle.

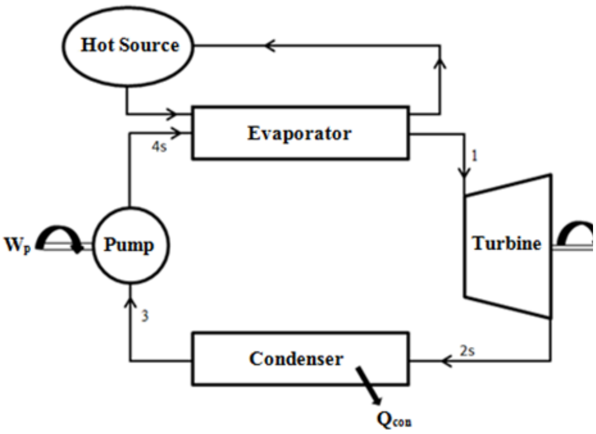

Fig. 1. Schematic diagram of the Organic Rankine cycle.

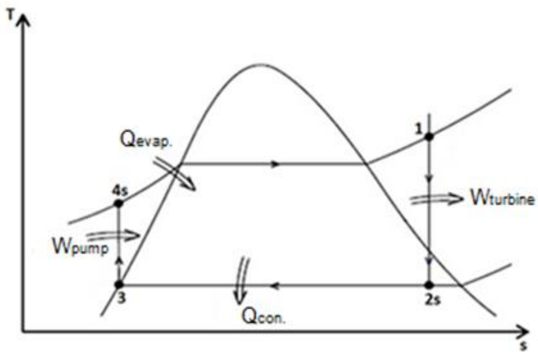

Fig. 2. T-s diagram of the Organic Rankine cycle.

An Organic Rankine cycle technology that is based on the system that generates electricity from heat uses the heat of the hot source in order to evaporate the organic fluid inside the boiler. The pressurized vapor is then delivered to the turbines. The fluid that is in vapor state at high temperature and pressure expands isentropically in the turbine, it rotates the turbine shaft attached to the electric generator and produces work. The pressure and temperature of the vapor are reduced during this process and they take their last states to enter the condenser. The vapor that comes out of the turbine is mostly a saturated liquid-vapor combination that is highly dry. Then the vapor rejects heat to environment in a condenser and condensed at constant pressure.

The vapor that comes out of the condenser as a saturated liquid enters into the pump and here, it is isentropically compressed up to the boiler operating pressure. Due to the reduction in the specific volume of the fluid, the temperature of the fluid increases slightly. The organic fluid that comes out of the pump enters into the boiler as a compressed liquid and it completes the cycle.

2.2. Thermodynamic Analysis of the Organic Rankine Cycle

For the thermodynamic analysis of the Organic Rankine cycle, the first law equations were used to estimate the performance of each element in the cycle and to detect the thermal efficiency of the cycle [18-20].

2.2.1. Turbine

As isentropic expansion occurs in the turbine between 1 and 2s,

$$Q_{12s} = 0 \quad , \quad s_1 = s_2 \tag{1}$$

If the first law of thermodynamics is applied on the turbine,

$$Q_{12s} - \dot{W}_{T12s} = H_{2s} - H_1 = \dot{m}_{ref}\ (h_{2s} - h_1) \tag{2}$$

If Eq. 2 is simplified, the isentropic turbine work can be calculated by,

$$\dot{W}_{T12s} = \dot{m}_{ref}\ (h_1 - h_{2s}) \tag{3}$$

In this equation; $\dot{m}_{ref}$ means the amount of the refrigerant and is calculated with below equation:

$$\dot{m}_{ref} = W_{T12s}/(h_1 - h_{2s}) \tag{4}$$

The actual turbine work and the actual enthalpy value at the turbine exhaust from 1 to 2 is calculated through the following equation:

$$\eta_{T,isen} = \dot{W}_{T12}/\dot{W}_{T12s} = (h_1 - h_2)/(h_1 - h_{2s}) \tag{5}$$

In this equation; $\dot{W}_{T12s}$ means the isentropic turbine work, $\dot{W}_{T12}$ means the actual turbine work and $\eta_{T,isen}$ means the isentropic efficiency of the turbine.

2.2.2. Condenser

Between 2 and 3, the heat is rejected from the condenser at constant pressure and there is no work exchange during this time.

$$W_{23} = 0 \tag{6}$$

$$\dot{Q}_{con} = \dot{Q}_{23} = \dot{m}_{ref}\ (h_2 - h_3) \tag{7}$$

The heat rejected at the condenser is delivered to the cooling water at thermal power plants. Thus; the amount of cooling water required for the condensing process is calculated through the following equation:

$$\dot{Q}_{con} = \dot{m}_c C_p\ (T_{in} - T_{out}) \tag{8}$$

In this equation; m_c means the amount of the cooling water, T_{in} and T_{out} mean the inlet and outlet temperatures of the cooling water. C_p means the specific heat of the cooling water and the C_p values of the fluids were calculated using the Engineering Equation Solver (EES) program.

2.2.3. Pump

Between 3 and 4s, isentropic compression occurs in the pump. The fluid used in the pump is deemed as an incompressible fluid ($v_3 = v_{4s}$ = constant) and the specific work (the work required to compress unit mass of working fluid) of the pump is calculated through the following equation:

$$w_{P34s} = \int v\, dP = v_3\ (P_4 - P_3) = h_{4s} - h_3 \tag{9}$$

As in this expression the specific volume of the fluid and the inlet-outlet pressures of the pump are known, the specific work of the pump is easily found. Also by making use of the equation (9), the specific pump work is found and the enthalpy of the fluid at the outlet of the pump is calculated through the following equation:

$$h_{4s} = v_3\ (P_4 - P_3) + h_3 \tag{10}$$

The actual pump work between 3 and 4 and the enthalpy value at the outlet of the pump is calculated through the following equation:

$$\eta_{P,isen} = W_{P34s}/W_{P34} = (h_{4s} - h_3)/(h_4 - h_3) \tag{11}$$

In this equation; $\dot{W}_{P34s}$ is the isentropic pump work, $\dot{W}_{P34}$ is the actual pump work and $\eta_{P,isen}$ is the isentropic efficiency of the pump.

2.2.4. Boiler

Between 4 and 1, the hot source is used to deliver heat to the organic fluid at constant pressure. The amount of heat delivered to the boiler is calculated through the following equation:

$$\dot{Q}_{boiler} = \dot{Q}_{41} = \dot{m}_{ref}\ (h_1 - h_4) \quad (12)$$

The amount of heat delivered to the boiler from the hot source is calculated through the following equation:

$$\dot{Q}_{boiler} = \dot{m}_{geo} C_{P_{geo}} \Delta T \quad (13)$$

Here; $\dot{m}_{geo}$ is the geothermal resource flow rate, ΔT is the temperature difference between the hot source and its surrounding.

2.2.5. Thermal efficiency of the cycle

When performing the thermal calculations of the elements that make up the cycle, the thermal efficiency of the cycle is calculated through the following equation:

$$\eta_{thermal} = \dot{W}_{net}/\dot{Q}_{boiler} = (\dot{W}_{T12} - |\dot{W}_{P34}|)/\dot{Q}_{boiler} \quad (14)$$

3. Results and Discussion

The design parameters and operating conditions to be used for the thermodynamic analysis of the system elements are presented in Table 2.

Table 2. Design parameters and operating conditions

Parameters	**Values**
Geothermal resource temperature	90-140°C
Turbine inlet temperature Boiling temperature	85-135°C 80-130°C
Turbine power	1 MW
Condensing temperature	30°C
Turbine isentropic efficiency	75%
Pump isentropic efficiency	80%
Ambient temperature	20°C

In the thermodynamic analysis that was made using the parameters given in Table II, the turbine inlet temperature was defined based on the temperature of the geothermal resource and was taken as 5°C lower than the temperature of the geothermal resource. As for the turbine inlet pressure, saturation pressure was taken as 10°C lower than the temperature of the geothermal resource.

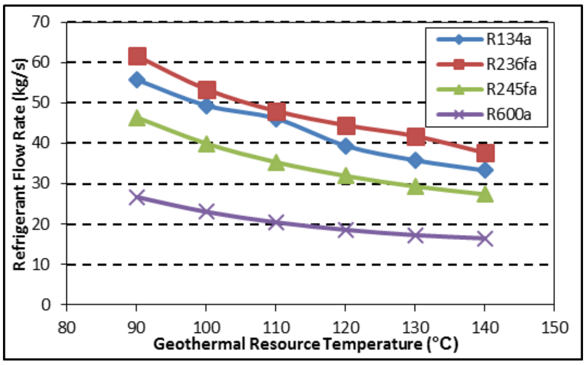

Fig. 3. Variation of flow rate of refrigerant that is required to obtain 1 MW turbine work for different refrigerants.

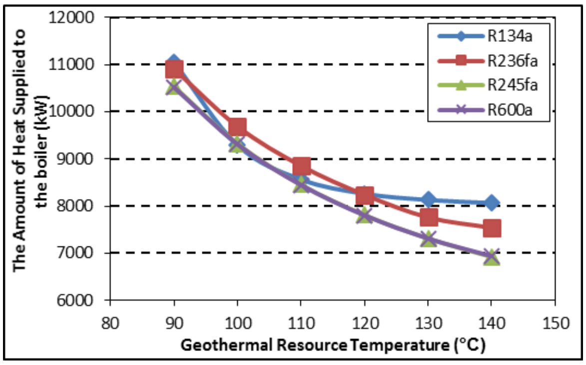

Fig. 4. Variation of amount of heat supplied to the boiler that is required to obtain 1 MW turbine work for different refrigerants.

Based on the temperature of the geothermal resource, variation of flow rate of refrigerant that is required to obtain 1 MW turbine work is given in Fig. 3 for constant operating conditions. As seen in Fig. 3, the flow rates of each four refrigerants decrease based on the increasing temperature of the geothermal resource. R236fa is the fluid with the highest flow rate required for 1 MW turbine work whereas R600a is the fluid with the lowest flow rate.

Based on the temperature of geothermal resource, variation of amount of heat supplied to the boiler that is required to obtain 1 MW turbine work is given in Fig. 4 for constant operating conditions. As seen in Fig. 4, for each of the four refrigerants, the amount of heat supplied to the boiler from the geothermal resource decreases depending on the increase observed in the temperature of the geothermal resource.

When R245fa and R600a are used in the cycle as refrigerants, the amounts of heat delivered to boiler depending on the temperature of the geothermal resource are very close to each other and the amount of heat delivered to the boiler is lower compared to the other two refrigerants. The amount of heat delivered to the boiler from the geothermal resource is higher when R236fa is used within an approximate range of 95-120°C geothermal resource temperature and when R134a is used outside this range (Fig. 4).

Based on the temperature of the geothermal resource, variation of flow rate of geothermal resource that is required to obtain 1 MW turbine work is given in Fig. 5 for constant operating conditions. For each of the four refrigerants, the geothermal flow rate decreases depending on the geothermal resource temperature. When R245fa and R600a are used as fluids, the geothermal resource flow rates required to obtain 1 MW turbine work are very close to each other and the geothermal resource flow rate required is lower compared to that of the other fluids.

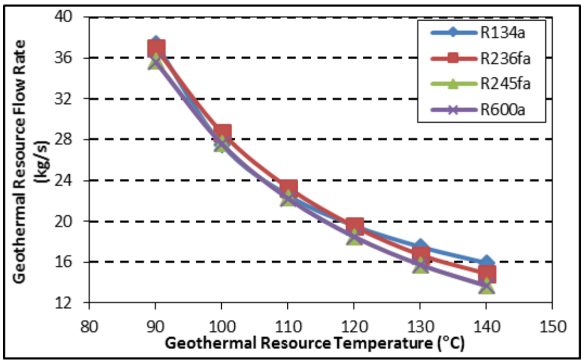

Fig. 5. Variation of flow rate of geothermal resource that is required to obtain 1 MW turbine work.

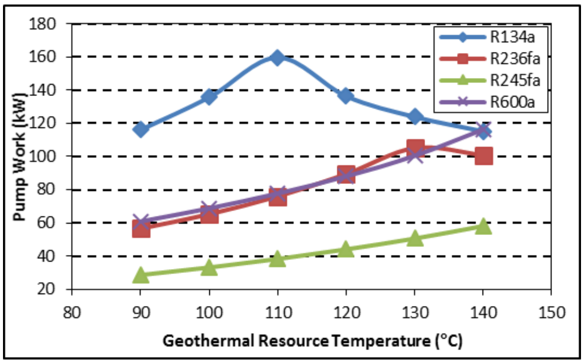

Fig. 6. Variation of the pump work that is required to obtain 1 MW turbine work for different refrigerants.

The geothermal flow rate required is higher when R236fa is used within an approximate range of 95-120°C geothermal resource temperature and when R134a is used outside this range (Fig. 5).

Based on the temperature of geothermal resource, variation of the pump work that is required to obtain 1 MW turbine work is given in Fig. 6 for constant operating conditions. When R245fa and R600a are used as refrigerants in the cycle the required pump work increases in parallel with the geothermal resource temperature. On the other hand; when R134a and R236fa are preferred as fluids, the pump work required until the critical pressure is reached at (the temperature that gives the critical pressure) tends to increase just as the other two refrigerants. As the pump’s specific heats remains constant at critical pressure, the required pump work decreases depending on the increasing geothermal resource temperature.

Based on the temperature of geothermal resource, variation of amount of heat required to be rejected from the condenser to obtain 1 MW turbine work is given in Fig. 7 for constant operating conditions. For each of the four refrigerants, the amount of heat required to be rejected from the condenser decreases depending on the geothermal resource temperature.

When R245fa and R600a are used as fluids, the amount of heat required to be rejected from the condenser are very close to each other and they are lower compared to other fluids. On the other hand; the amount of heat rejected from the condenser is higher when R236fa is used within an approximate range of 95-120°C geothermal resource temperature and when R134a is used outside this range (Fig. 7).

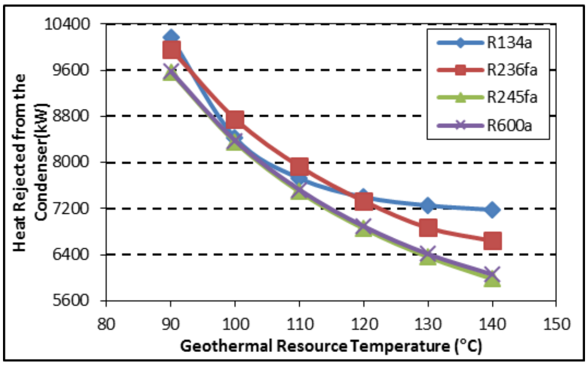

Fig. 7. Variation of amount of heat required to be rejected from the condenser to obtain 1 MW turbine work for different refrigerants.

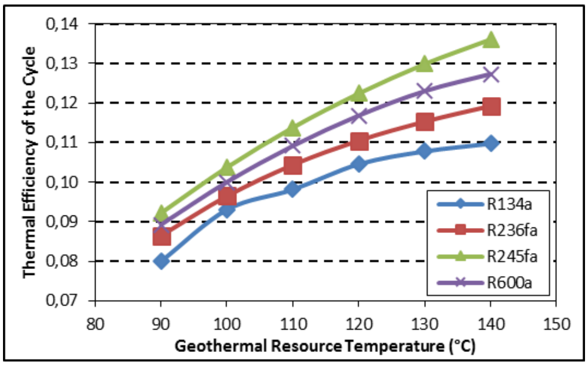

Fig. 8. Variation of thermal efficiency of the organic Rankine Cycle that provides the performance required to obtain 1 MW turbine work.

For different refrigerants, variation of thermal efficiency of the organic Rankine Cycle that provides the performance required to obtain 1 MW turbine work is given Fig. 8 based on the temperature of geothermal resource. For each of the four refrigerants, the thermal efficiency of the cycle increases depending on the geothermal resource temperature. Whereas the highest thermal efficiency is achieved when R245fa is used in the cycle as the refrigerant, the lowest thermal efficiency is obtained when R134a is used.

4. Conclusion

There are several factors that influence the performance of Organic Rankine cycles. Selection of the refrigerant to be used in the cycle is one of the most significant of these factors. In this analysis performed for the specific conditions, R245fa is recommended as the refrigerant as the thermal efficiency of the cycle would be higher.

Depending on the increase of the geothermal source temperature, the increase of the cycle's thermal efficiency becomes maximal (approx. 47.6%) with the use of R245fa and the gain in the thermal efficiency is minimal (approx. 37.2%) if R134a is preferred.

The cooling fluid flow required for 1 MW turbine work becomes maximal (approx. 41%) depending on geothermal resource temperature. Similarly, the amount of heat rejected from the condenser becomes maximal (approx. 37.5%) depending on the refrigerant used.

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