Wasit Journal of Engineering Sciences

Journal homepage: https://ejuow.uowasit.edu.iq

The Influence of Condenser Temperature on the Energy and Exergy Efficiencies of the ORC

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Received
12-April-2022
Revised
20-June-2022
Accepted
06-Agust-2022
Doi
10.31185/ejuow.Vol10.Iss3.313

Abstract

From low-grade heat sources, the organic Rankine cycle may be exploited to create power. The thermal efficiency of the organic Rankine cycle is affected by the value of the lowest cycle temperature, which is the condensation temperature. This study looks at the effect of condensation temperature on the efficiency of energy systems that use organic Rankine cycles. At a condensing temperature of 10–20 °C, the ORC thermal efficiency is calculated. R134a working fluid was used in the study. The expander's power output was boosted to 0.09765 kW by decreasing the condensing temperature. Additionally, the thermal efficiency has been enhanced by 3.826 %. At a minimum temperature of 10 °C, the expander speed at 595 rpm. Exergy efficiency has an 18.26 %. is shown that lowering the condensing temperature increased the ORC system's thermal efficiency and energy output.

Keywords: Organic Rankine cycle, R134a, Condenser temperature, thermal efficiency; exergy efficiency.

الخلاصة: من مصادر الحرارة منخفضة الدرجة ، يمكن استغلال دورة رانكين العضوية لتوليد الطاقة. تتأثر الكفاءة الحرارية لدورة رانكين العضوية بقيمة أدنى درجة حرارة للدورة ، وهي درجة حرارة التكثيف. تبحث هذه الدراسة في تأثير درجة حرارة التكثيف على كفاءة أنظمة الطاقة التي تستخدم دورات رانكين العضوية. عند درجة حرارة التكثيف من 10-20 درجة مئوية ، يتم حساب الكفاءة الحرارية ORC. تم استخدام سائل العمل R134a في الدراسة. تم تعزيز خرج طاقة الموسع إلى 0.09765 كيلو وات عن طريق خفض درجة حرارة التكثيف. بالإضافة إلى ذلك ، تم تحسين الكفاءة الحرارية بنسبة 3.826٪. عند درجة حرارة لا تقل عن 10 درجات مئوية ، يدور الموسع عند 595 دورة في الدقيقة. تبلغ كفاءة Exergy 18.26٪. تبين أن خفض درجة حرارة التكثيف زاد من الكفاءة الحرارية لنظام ORC وإنتاج الطاقة.

1. INTRODUCTION

Because of the expanding global population and rising living standards, demand for various forms of high-quality energy (e.g., electricity, heat, and cold) has increased considerably in recent decades. Three of the most popular uses of energy are lighting, heating, and air conditioning. However, the necessity to power an ever-increasing number of domestic appliances, electric autos, and other equipment has resulted in increased energy consumption in recent decades [1]. ORC technology significantly reduces emissions and greenhouse gas emissions by vaporizing the active agent using solar, geothermal, and waste heat as heat sources [2] [3]. One easy way to enhance the Rankine cycle's efficiency is to lower the condensing temperature, which is the cycle's lowest temperature. The performance of a solar organic Rankine cycle system has been studied in relation to condensing temperature. Water is used to cool the condenser. When the condensing temperature is dropped from 25 °C to 10 °C, R134a's thermal efficiency rises by 29.78 % [4]. Exergy efficiency is suggested for analyzing low-temperature power systems and comparing performance for different heat source temperatures. Energy and exergy efficiency decrease as condenser pressure rises. Energy and exergy efficiency were respectively 7.5 % and 43.8 %. In the condenser, the rate of exergy degradation is rising [5]. Using R134a, Water, Ammonia, and R113 working fluids, a study of the change in thermal efficiency and total irreversibility rate of ORC was done at various temperatures. When the turbine pressure was raised, the total irreversibility of all fluids rose. The highest irreversibility values are found in water, while the lowest are found in R134a. According to the experts, the working fluid should operate under saturated conditions to reduce the irreversibility of the entire process [6]. R134a has demonstrated promising results as a working fluid in organic Rankine cycles [7], allowing for high power outputs. R134a-based systems have 21.3% exergy efficiencies [8]. The ORC system was designed and constructed as an evaporator for a liquid helical heat exchanger and a helical heat exchanger as a condenser for R-134a. The system efficiency in the experiment was 3.33 %, and the turbine power was 614W [9]. The conclusion was that ORC with R134a can function at lower temperatures; it is an excellent candidate for creating usable energy as a working fluid using low enthalpy heat [10]. The effect of condensation temperature on the performance of the organic Rankine cycle system was investigated in this study. The condenser is cooled by water. R134a is a particular working fluid. All compounds were subjected to energy and exergy tests in order to evaluate ORC performance. It also looked at how it affected rotational speed.

2. DESCRIPTION OF THE ORC SYSTEM

The organic Rankine cycle system under study consists of an evaporator, an expander, a condenser, and a pump. We used a counter-flow shell and tube for the condenser and evaporator. A schematic of the ORC system is shown in Figure 1. The fluid is pumped to high pressure, and after passing via a heat exchanger, it is heated and evaporated in the evaporator. High-temperature, high-pressure vapour from the evaporator enters the expander, where the enthalpy produced by the rotation of the shaft transforms it into energy. when low-pressure liquid is pumped into to the condenser and is condensed by the water, the cycle is repeated. As shown in Figure 2, temperature-sensitive and pressure measurements were put at each component's entrance. A P-h diagram is used to show the preceding steps. Table 1 lists the simulation parameter conditions for the ORC model. This research does not account for pressure losses in the evaporator and condenser. Temperatures in the condensing chamber range from 20 to 30 °C.

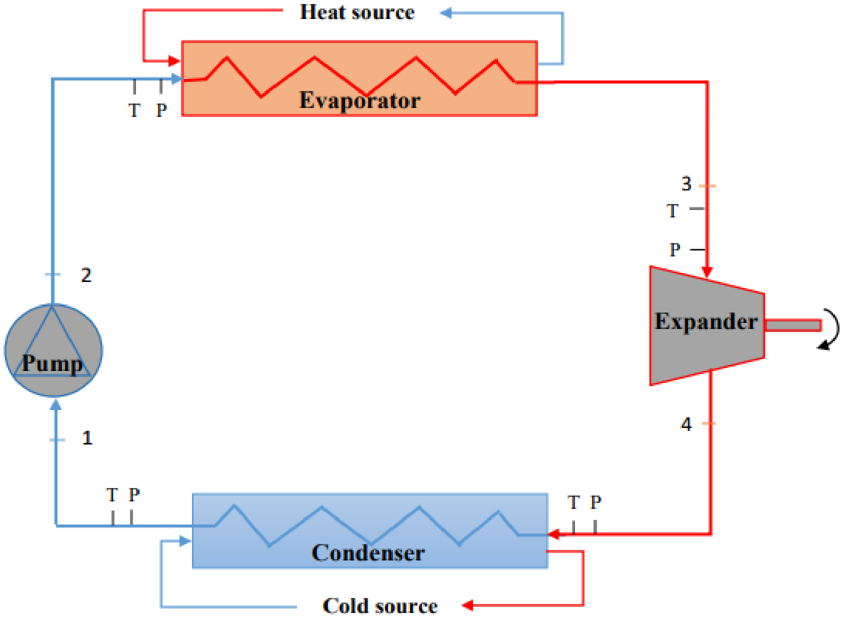

Figure 1 Schematic diagram of the ORC system.

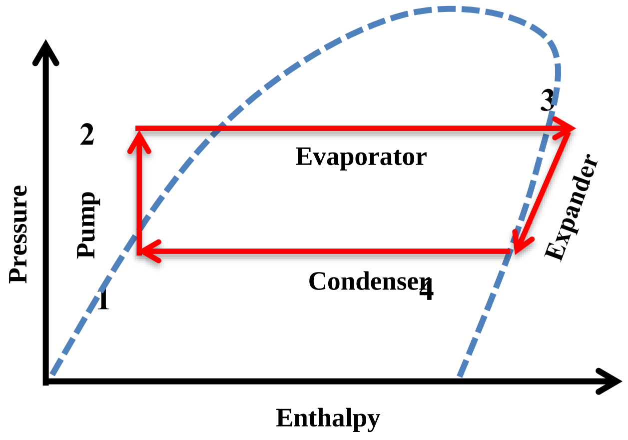

Figure 2 P-h diagram of the ORC system

3. EES SOFTWARE SIMULATION

The main purpose of this research is to undertake a thermodynamic analysis of variables such as condensation temperature. The effect of the parameters on expander power output, thermal and exergy efficiency, and rotation speed is explored in this study, as illustrated in Figure 3. The simulation was performed using EES. The energy and exergy equations were utilized to create the ORC simulation programs in EES. The input parameters are shown in Table 1.

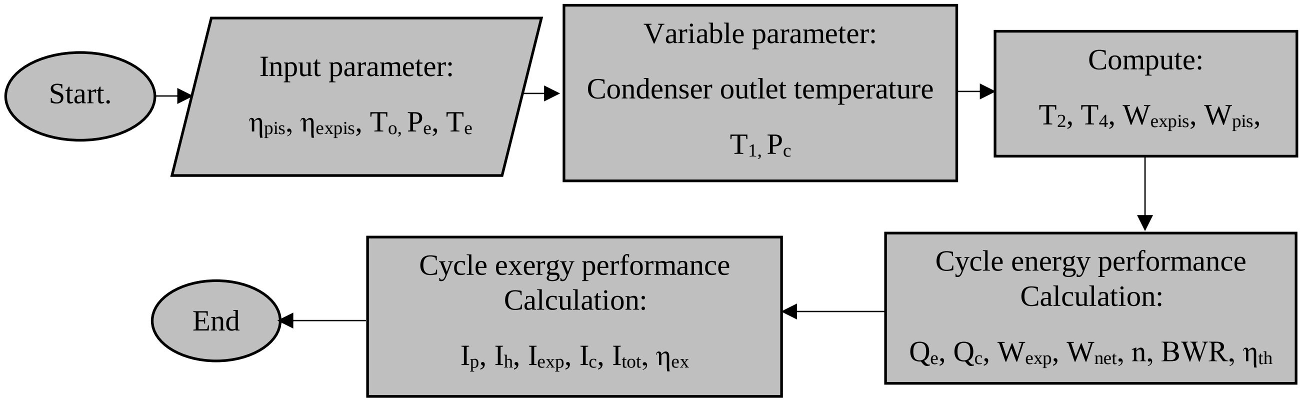

Figure 3 Flow chart of the simulation procedure.

Table 1 Specifications of the ORC conditions.

Parameter	Value	Unit
Evaporator temperature	85	°C
Evaporator pressure	0.65	MPa
Condensing temperature range	10-20	°C
Ambient temperature	15	°C
Ambient pressure	0.1015	MPa
the expander's isentropic efficiency	85%	-
Isentropic efficiency of the pump	80%	-

4. WORKING FLUID

The performance and thermodynamic design of the ORC system may be significantly impacted by the selection of an appropriate working fluid. A few of the qualities that make it ideal include the working fluid's latent heat, Ozone Depletion Potential (ODP), and Global Warming Potential (GWP) [11]. Based on the slope of the saturation vapour line, the best working fluid is selected. The operating fluid was decided upon as R134a. The standard boiling point, critical pressure, and fluid molecular weight are all factors that affect an ORC's efficacy and efficiency. Table 2 lists the physical parameters of the organic fluid.

Table 2 Thermal and physical properties of R134a [12]

Working	Type	Molecular mass (kg / kmol)	Critical Temperature (°C)	Critical Pressure (MPa)	Flammability/ Toxicity
R134a	Wet	102.03	101.1	4.06	No

5. MATHEMATICAL MODEL

In this part, we conducted two different sorts of analysis. The first kind employs the first law of thermodynamics to study energy. The thermal efficiency and power of the ORC are calculated using energy analysis. Exergy analysis, which makes use of the second law of thermodynamics, is the second kind. Exergy analysis' main objective is to assess each component's effectiveness and rate of exergy degradation (ORC). The program Engineering Equation Solver (EES) is used to perform these evaluations.

In a steady-state with all components, the first and second thermodynamic analysis rules are applied. These calculations employ data from the EES software program that measures temperature, pressure, and flow rate. On all ORC components, governing equations, energy, and exergy studies were carried out. A statistic for assessing reversibility and the ability to convert low or medium temperatures into work is the exergy destruction rate (I). All ORC compounds' energy and exergy analyses are shown in Equations (1) through (17).

The exergy balance is the difference between the net exergy transfer across a system boundary and the exergy destroyed within a system boundary because of changes in exergy that can't be changed back.

$$\sum \dot{m}_i = \sum \dot{m}_o \qquad (1)$$

$$Q + W = \sum \dot{m}_i h_i - \sum \dot{m}_o h_o \quad \text{for energy analysis} \qquad (2)$$

$$E_{heat} + W = \sum \dot{m}_i e_i - \sum \dot{m}_o e_o + I \quad \text{for exergy analysis} \qquad (3)$$

Where: $e = (h - h_o) - T_o(s - s_o)$

$$I = \sum \dot{m}_i e_i - \sum \dot{m}_o e_o + \left(\sum \left(Q_i\left(1 - \frac{T_0}{T}\right)\right) - \left(\sum Q_O\left(1 - \frac{T_0}{T}\right)\right) + W \qquad (4)$$

The equations for doing a comparative thermodynamic analysis are presented below. Energy analyses equation [13] [14]:

Although the pressure in the evaporator pipes lowers somewhat, the process is referred to be isobaric since heat is transmitted to the working fluid at a constant pressure. The condition of the working fluid as it exits the evaporator is shown in point 3, and the heat produced by the working fluid may be estimated using Equation (5).

$$Q_e = \dot{m}_r (h_3 - h_2) \qquad (5)$$

Condenser: Because the heat is rejected in the condenser, the working fluid can condense and be re-circulated in the cycle. Despite pressure dips in the condenser owing to friction losses in condenser pipes, the heat rejection process is called isobaric. Use the Equation to figure out how much heat is being rejected (6).

$$Q_c = m_r(h_3 - h_2) \qquad (6)$$

An expander, also known as a turbine or expander, converts the absorbed energy at the evaporator into usable mechanical work. Because the expander efficiency can never reach 100%, the process is called isentropic. Point 4 depicts the condition of the working fluid entering the expander, and the usable work out may be calculated using equation (7).

$$W_{exp.} = m_r(h_3 - h_4) * \eta_{exp} = m_r (h_{3s} - h_4) \qquad (7)$$

Pump: The working fluid is delivered to the evaporator at constant entropy after passing through the condenser as a saturated liquid. The process is ideal. Energy conversion efficiency, on the other hand, is never 100%. Equation (8) in point 1 is used to compute the amount of power absorbed by the pump.

$$W_p = m_r(h_2 - h_1) * \eta_p = m_r(h_{2s} - h_1) \tag{8}$$

Energy efficiency is critical for reducing both fuel use and greenhouse gas emissions. The ratio of net work out to heat absorbed in the evaporator is known as thermal efficiency.

$$\eta_{th} = \frac{W_{net}}{Q_e} \tag{9}$$

Where : $W_{net} = W_{exp.} - W_p$

Exergy destruction or irreversibility rate analyses equation [15]:

A thermodynamic system's performance is strongly impacted by the pace at which its action is irreversible. Entropy production in a genuine process is primarily caused by internal or external forces. Internal entropy is caused by frictional pressure drops in pipes that are part of the system, unchecked turbine expansions, and the transfer of energy across a small difference in temperature between the components. External entropy may be produced through both the transmission of mechanical work and heat via restricted temperature changes. This is a reference to the second law of thermodynamics' basic concept [18].

The evaporator irreversibility rate may be calculated using Equation (4)

$$I_h = T_0\, m_r\, ((s_3 - s_2) - \frac{(h_3 - h_2)}{T_H}) \tag{10}$$

The irreversibility rate of an expander may be calculated as follows:

$$I_{exp.} = T_0\, m_r\, (s_4 - s_3) \tag{11}$$

Equation (4) may be used to calculate the condenser irreversibility rate.

$$I_c = T_0\, m_r\, ((s_1 - s_4) - \frac{(h_1 - h_4)}{T_L}) \tag{12}$$

The irreversibility rate for the pump is:

$$I_p = T_0\, m_r\, (s_2 - s_1) \tag{13}$$

Equations (10), (11), (12), and (13) may be used to calculate total irreversibility as follows:

$$I_{tot} = I_h + I_{exp.} + I_c + I_p \tag{14}$$

Second law efficiency or exergy efficiency: The exergy efficiency of the ORC system is defined as the ratio of net power production to waste heat exergy before inflowing evaporation, and it's calculated using equation (15).

$$\eta_{ex} = \frac{\eta_{th}}{(1 - \frac{T_0}{T_{hw}})} \tag{15}$$

Where : $T_{hw} = \frac{T_{hwo} + T_{hwi}}{2}$

In theory, it's easier to figure out how much energy the pump used than how much energy it made (the energy produced by the expander). The back work ratio (BWR) compares how much energy the pump needs to how much energy it makes [16]. This helps figure out how much energy the pump uses in the system.

Back work ratio $$BWR = \frac{W_p}{W_{exp.}} \tag{16}$$

For figuring out the speed of the expander's rotation [17], use the equation: $$n = \frac{2 * W_{exp} * 60}{V * (\rho_{in} - \rho_{out}) * (h_3 - h_4)} \tag{17}$$

6. RESULTS AND DISCUSSION

ORC performance is determined by the condensing pressure and temperature. This is accomplished by examining how they influence the system. For the working fluid R134a, the condensing temperature was raised from 10 to 20 °C in this study. The constant evaporator pressure of R134a used in the evaporator is 0.65 MPa. All other factors remain unchanged. The important parts of the thermodynamic cycle were computed using the EES software.

The effect of increasing the condenser temperature on system performance is being investigated. it can be seen when the temperature of the condenser rises from 10 to 20 degrees Celsius, as seen in Figure 4, if R134a is in a saturated or superheated vapor state at the expander's input, as in equation (5). The net power lowers between 0.09474 and 0.02689 kW, resulting in a beneficial duty cycle due to the reduction in total work. As a result, the extender's output is lowered to between 0.09765 and 0.02789 kW.

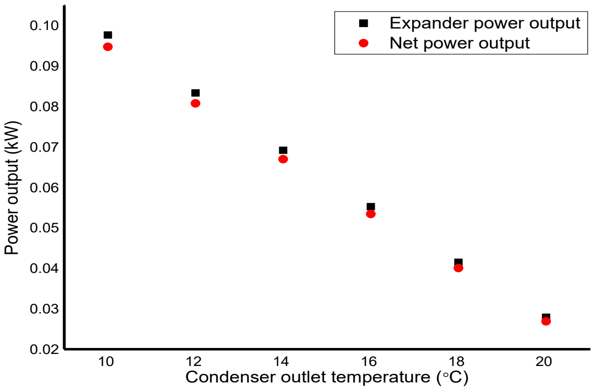

Figure 4 Variation of net power output and expander power output with condensation temperature.

While, as shown in Figures 5 and 6 and 7, reducing the pump's work consumption from the power of the expander reduces net power, resulting in a linear loss in thermal efficiency from 3.826 -1.167 % as given in Eq. (9). While the exergy's effectiveness varies between 18.26 and 5.556 %, as seen in Eq.(15). As a result, as the BWR rises, the system's efficiency falls. According to Eq. (16), the BWR is between 0.02983 and 0.03563. The reason for this is that when the condensing pressure rises, the pressure back or exit pressure of the expander rises as well, lowering the enthalpy at the expander's outlet.

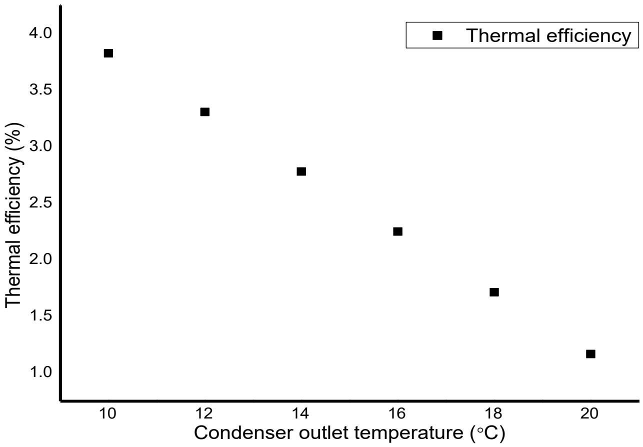

Figure 5 Variation of thermal efficiency of system with condensation temperature.

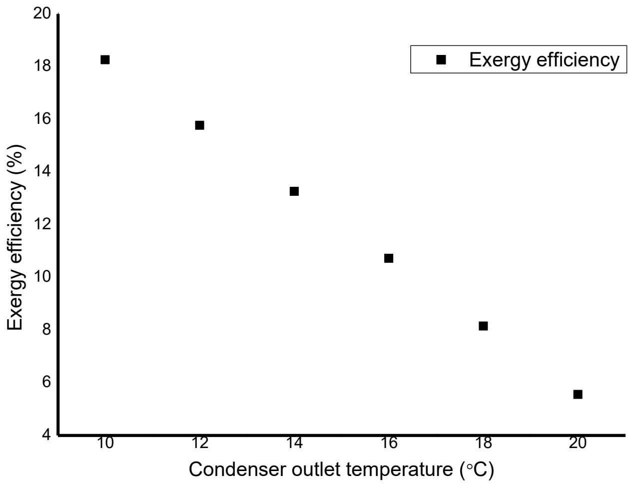

Figure 6 Variations of the exergy efficiency with condenser temperature.

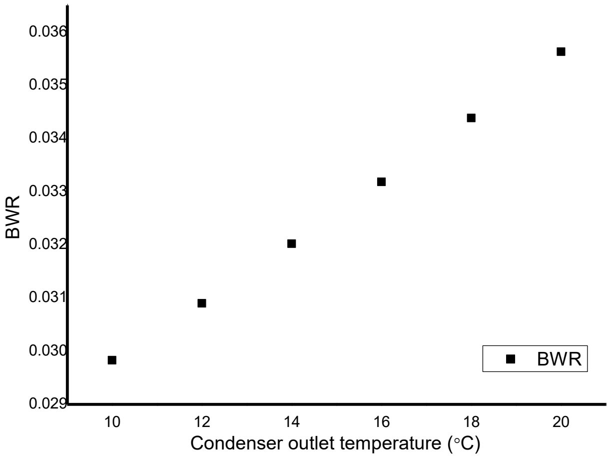

Figure 7 Influence BWR on the outlet condenser temperature.

The quantity of heat taken from the condenser and received in the evaporator is impacted by the condensing temperature, as shown in Figure 8, which is dependent on the temperature decrease of the liquid at the expander's output, as shown in equations (5) and (6). As a result, heat extracted from the condenser varies from 2.382 to 2.278 kW, whereas heat entering the evaporator ranges from 2.476 to 2.305 kW. Reducing the amount of heat entering the evaporator and extracting it into the condenser results in lower thermal efficiency. This is due to a rise in both the condenser's outflow temperature and pressure. As a result, the chilled used water enters the condenser and emerges hotter. The heat exchange between R134a and cold water causes this.

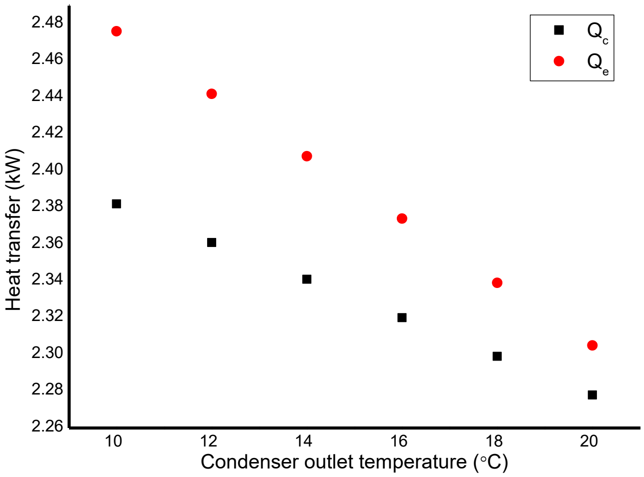

Figure 8 Variation of the amount of heat extracted from the condenser and received in the evaporator with condensation temperature.

It is simple to estimate where the highest energy will be lost using the external energy destruction ratio given in Figure 9. This helps to focus changes in system components and increase the ORC system's energy efficiency. The evaporator is the largest source of external energy destruction, accounting for the majority of the total external energy destroyed in the system, although it is declining since the evaporator temperature has not been increased. As shown in figure 9, increasing the condensation temperature has an influence on the exergy destruction or rate of irreversibility for all components in the ORC system as in eqs. (10) to (14). This is owing to the enthalpy difference decreasing via the evaporator, as well as the temperature difference between the hot water entering and leaving the evaporator. The energy efficiency improves as the temperature disparity decreases. The irreversibility rate of the evaporator is 0.4265-0.3946kW. Then, the destruction in the condenser increases as the temperature of condensation rises and therefore the heat content rises. The irreversibility rate of the condenser ranged from 0.04505 to 0.1221kW. The heat exchange between the liquid utilized and the water happens even if the expander and pump's exergy destruction, at 0.01753 and 0.0005922 kW, respectively, are nearly constant due to the little temperature change. As shown in Figure 10, the impact of the evaporator and condenser on the system causes the total rate of irreversibility to increase and range between 0.4896 and 0.5218 kW.

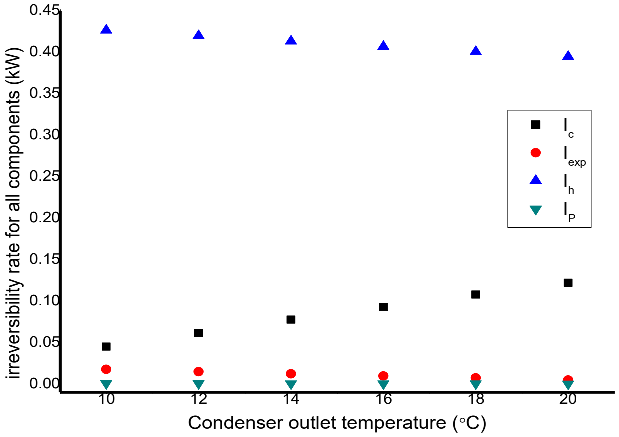

Figure 9 Variations of the irreversibility rate with condenser temperature.

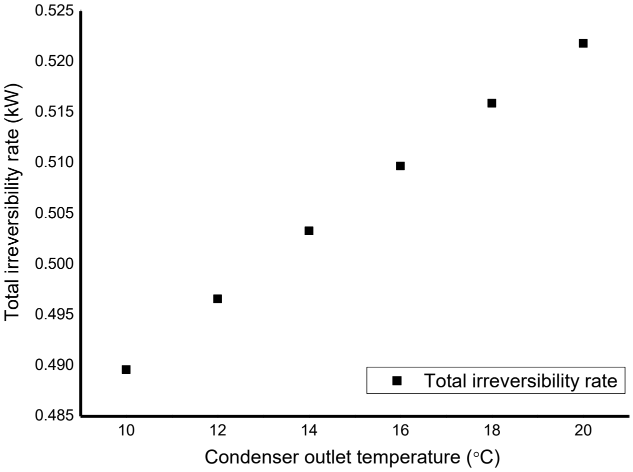

Figure 10 Variations of the total irreversibility rate with condenser temperature.

Figure 11 depicts the effect of the expander's spinning speed, as described by eq. (17), on the temperature of the condenser. Temperature and rotational speed have an inverse relationship. As the temperature increases, the condenser's spinning speed decreases because of an increase in enthalpy at the expander's departure point and an increase in coolant density. The rotational rates were 595 and 521 rpm when the condenser temperature was 10 °C and 20 °C, respectively. As can be seen, the lower the condenser exit temperature is the faster the spinning speed. The findings demonstrate that ORC is more useful in areas with lower ambient temperatures.

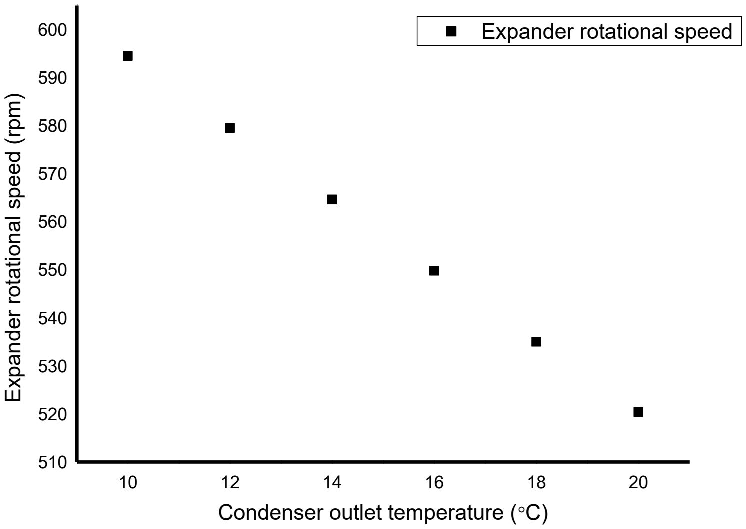

Figure 11 Variations of Rotational speed with condenser temperature.

7. VALIDATION WITH OTHER STUDY

When A study like [9] was contrasted with the present research. Although there is a little variation between the two studies, the thermal efficiency of the present research is higher than that of the prior study for the identical design of shell and tube heat exchangers. The efficiency in the present investigation was 3.826 % with a condenser temperature of 10 °C and an expander pressure of 0.65 MPa. It moves at a mass flow rate of 0.0125 kg/s. While the efficiency of the prior research was 3.8 % with a condenser temperature of 10 °C and an evaporator pressure of 0.79 MPa. the mass flow rate of 0.123 kg/s Despite the fact that the mass flow rate is lower than in the earlier research, it is observed that the present efficiency is higher. This is due to the fact that rising temperatures ensure the entry of superheated steam free of moisture for R134a when they enter the expander. And the efficiency improves as the condenser's temperature decreases.

8. CONCLUSION

Low condensation temperatures boost the performance of the organic Rankine cycle system. The output of the extender for R134a is increased by 0.09765 kW by lowering the condensing temperature from 20 to 10 °C. Furthermore, thermal efficiency has increased by 3.826%. The expander rotational speed was 595 rpm at the lowest temperature of 10°C. The efficiency of exergy is 35.12 %. At the lowest temperature of 10°C, the minimum total irreversibility or exergy destruction rate of 0.3946 kW is also achieved. To reach a condensing temperature of 10°C, the condensing cooling water temperature should be at 5–6°C. Except in the winter, getting chilled water at this temperature is difficult most of the year. As a result, absorption coolers with an ORC system can be utilized to generate cold water to solve this issue.

NOMENCLATURES

Q	Heat transfer rate (kW)	η	Energy efficiency (%)
W	Power (kW)	η_{is}	Isentropic efficiency (%)
h	Specific enthalpy (kJ/kg)	η_{ex}	Exergy efficiency (%)
I	Irreversibility or Exergy rate (kW)	EES	Engineering Equation Solver
$\dot{m}$	Mass flow rate (kg/s)		
V	Volume cell chamber in the expander (m^3)		

REFERENCE

1. Kolasiński P, (2019). "Application of the Multi-Vane Expanders in ORC Systems—A Review on the Experimental and Modeling Research Activities". *Energies,* 12 (15), 2975.

2. Hung, T. C., Shai, T. Y., and Wang, S. K. (1997). "A review of organic Rankine cycles (ORCs) for the recovery of low-grade waste heat". *Energy,* 22(7), 661-667.

3. Aghahosseini, S., and Dincer, I. (2013). Comparative performance analysis of low-temperature Organic Rankine Cycle (ORC) using pure and zeotropic working fluids. *Applied Thermal Engineering,* 54(1), 35-42..

4. Dragomir-Stanciu, D., Saghebian, S. M., and Kurchania, A. (2020). "The influence of condensing temperature on the efficiency of solar power systems with ORC". *Procedia Manufacturing, 46,* 359-363.

5. Baral, S., Kim, D., Yun, E., and Kim, K. C. (2015). "Energy, exergy and performance analysis of small-scale organic Rankine cycle systems for electrical power generation applicable in rural areas of developing countries". Energies, 8(2), 684-713.

6. Somayaji, C., P. J. Mago, and L. M. Chamra. (2006). "Second law analysis and optimization of organic Rankine cycle." *ASME power conference*. Vol. 42053.

7. Tchanche, B. F., Papadakis, G., Lambrinos, G., and Frangoudakis, A. (2009). "Fluid selection for a low-temperature solar organic Rankine cycle". *Applied Thermal Engineering*, *29*(11-12), 2468-2476.

8. Darvish, K., Ehyaei, M. A., Atabi, F., and Rosen, M. A. (2015). "Selection of optimum working fluid for organic Rankine cycles by exergy and exergy-economic analyses." *Sustainability*, *7*(11), 15362-15383.

9. Martin, A., Agustina, D., and Ibra, A. M. (2019). “Design and manufacturing of organic rankine cycle (orc) system using working fluid r-134a with helical evaporator and condenser”. In *IOP Conference Series: Materials Science and Engineering* (Vol. 539, No. 1, p. 012027). IOP Publishing.

10. Vélez, F., Chejne, F., and Quijano, A. (2014). “Thermodynamic analysis of R134a in an Organic Rankine Cycle for power generation from low temperature sources”. *Dyna, 81*(185), 153-159.

11. Kumar, A., and Shukla S. K. (2016). “Analysis and performance of ORC based solar thermal power plant using benzene as a working fluid,” *Procedia Technol*., vol. 23, pp. 454–463, 2016, doi: 10.1016/j.protcy.2016.03.050.

12. Calm, J. M., and Hourahan, G. C. (2007). “Refrigerant data update,” HPAC Heating, Piping, Air Conditioning Eng., vol. 79, no. 1, pp. 50–64.

13. Nouman, J., (2012). “Comparative studies and analyses of working fluids for Organic Rankine Cycles - ORC,”.

14. Koç, Y., Yaglı, H., and Koç, A. (2019). “Exergy Analysis and Performance Improvement of a Subcritical / Supercritical Organic Rankine Cycle (ORC) for Exhaust Gas Waste Heat Recovery in a Biogas Fuelled Combined Heat and Power (CHP) Engine Through the Use of Regeneration,” *Energies*, vol. 12, no. 575, pp. 1–22, doi: 10.3390/en12040575.

15. Shengjun, Z., Huaixin, W., and Tao, G. (2011). “Performance comparison and parametric optimization of subcritical Organic Rankine Cycle (ORC) and transcritical power cycle system for low-temperature geothermal power generation,” *Appl. Energy*, vol. 88, no. 8, pp. 2740–2754, doi: 10.1016/j.apenergy.2011.02.034.

16. Hartulistiyoso, E., Sucahyo, L., Yulianto, M., and Sipahutar, M. (2020). “Thermal efficiency analysis of Organic Rankine Cycle (ORC) System from low-grade heat resources using various working fluids based on simulation,” *IOP Conf. Ser. Earth Environ*. Sci., vol. 542, no. 012047, pp. 1–9, doi: 10.1088/1755-1315/542/1/012047.

17. Fatigati, F., Di Bartolomeo, M., Di Battista, D., & Cipollone, R. (2021). Model based control of the inlet pressure of a sliding vane rotary expander operating in an ORC-based power unit. *Applied Thermal Engineering*, *193*, 117032, https://doi.org/10.1016/j.applthermaleng.2021.117032.

18. Roy, J. P., and Ashok Misra. "Comparative performance study of different configurations of organic Rankine cycle using low-grade waste heat for power generation." *International Journal of Green Energy* 14.2 (2017): 212-228, doi:10.1080/15435075.2016.1253570.