Isı Bilimi ve Tekniği Dergisi, 39, 2, 121-135, 2019
J. of Thermal Science and Technology

ISSN 1300-3615

A PARAMETRIC ANALYSIS OF THE PERFORMANCE OF ORGANIC RANKINE CYCLE WITH HEAT RECOVERY EXCHANGER AND ITS STATISTICAL EVALUATION

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(Geliş Tarihi: 07.09.2018, Kabul Tarihi: 16.04.2019)

Abstract: In this study, the performance of a case study of Organic Rankine Cycle with heat recovery exchanger using different fluids is analyzed. As the fluids worked in the cycle, the commonly used R134a, R236fa, R245fa, R600a, R717 and R718 are preferred. Cycle performances of the selected fluids are compared based on both the heat source's temperature that changes between 80°C and 109°C and the effectiveness of the heat exchanger. Furthermore, the contribution ratios and the order of importance of the parameters affecting the performance of the cycle are evaluated using the Taguchi statistical method. As a result, the effect of the waste-heat source temperature on the performance of the system is greater than the other parameters examined, and the contribution ratio of this parameter is determined as 59.80%. However, effectiveness of heat exchanger is found to be the least effective parameter and the effect ratio is calculated as 2.18%. In addition, the best and worst operating conditions are determined from the statistical analysis, and in these conditions, the thermal efficiencies of the Organic Rankine Cycle are obtained as 15.26% and 8.61%, respectively.
Keywords: Organic Rankine cycle, Working fluids, Performance analysis, Heat exchanger, Thermal efficiency, Taguchi method.

ISI GERİ KAZANIM EŞANJÖRLÜ ORGANİK RANKİNE ÇEVRİMİNİN PARAMETRİK ANALİZİ VE İSTATİKSEL DEĞERLENDİRMESİ

Özet: Bu çalışmada farklı çalışma akışkanları için ısı geri kazanımlı eşanjör kullanan örnek bir organik Rankine çevriminin performans analizi gerçekleştirilmiştir. Akışkan olarak yaygın kullanılan R134a, R236fa, R245fa, R600a, R717 ve R718 akışkanları tercih edilmiştir. Seçilen bu akışkanlar için, 80°C ile 109°C sıcaklıkları arasında değişen atık ısı kaynak sıcaklığına ve eşanjör etkenliğine bağlı olarak sistem performansları karşılaştırılmıştır. Ayrıca sistemin performansını etkileyen parametrelerin etki oranları ve önem sırası Taguchi istatiksel metodu kullanılarak değerlendirilmiştir. Sonuç olarak, atık ısı kaynak sıcaklığının sistem performansı üzerindeki etkisinin incelenen diğer parametrelere göre daha fazla olduğu ve bu parametrenin etki oranının %59,80 olduğu belirlenmiştir. Bununla birlikte eşanjör etkenliğinin sistem performansı üzerindeki etkisinin diğer parametrelere göre çok daha az olduğu bulunmuş ve etki oranı %2.8 olarak hesaplanmıştır. Ayrıca sistemin en iyi ve en kötü çalışma şartları istatiksel analiz yapılarak belirlenmiş ve bu çalışma şartlarında organik Rankine çevriminin ısıl verimi sırasıyla %15,26 ve %8,61 olarak elde edilmiştir.
Anahtar Kelimeler: Organik Rankine çevrimi, Çalışma akışkanları, Performans analiz, Isı eşanjörü, ısıl verim, Taguchi metodu.

NOMENCLATURE

c_p	specific heat [kJ/kg K]
DOF	degree of freedom
$\dot{Q}$	amount of heat transfer
ε	effectiveness of the heat exchanger [-]

h	enthalpy [kJ/kg]
MS	mean of squares
P	pressure [kPa]
SS	sum of squares
S/N	signal to noise
T	temperature [°C]
$\dot{m}$	refrigerant flow rate [kg/s]
$\dot{W}$	work [kW]
v	specific volume [m^3/kg]
η	efficiency [%]

Subscripts

s	source
con	condenser
cw	cooling water
evap	evaporator
exc	heat exchanger
p	pump
T	turbine

INTRODUCTION

Rankine cycle is an ideal cycle for power plants that use steam. Water is used in the cycle in medium and big power plants to produce energy. However, high temperature and pressure should be maintained for proper operation. As the required temperature cannot be maintained from low temperature heat sources, the cycle can be used in a limited range. In recent years, hydrocarbon compounds; which have lower critical temperature, lower pressure, higher molecular mass and less risk to cause corrosion are preferred in Rankine cycle instead of water. With the use of these working fluids, these systems are named as Organic Rankine Cycle (ORC) and became one of the common power generation processes mainly used in factories for recovered waste-heat, solar energy, geothermal energy (Ferrara et al., 2014; Hung et al., 2010; Roy et al., 2010; Zare, 2016; Yamamoto et al., 2001; Ergun et al., 2016). Ferrara et al. (2014) performed a thermodynamic analysis of a concentrated solar power plant that works with organic Rankine Cycle. In this study, thermodynamic optimizations were made for various working fluids (R134a, R245fa, Acetone) and the results in terms of system efficiency and absorbing mirrors surface were compared. Hung et al. (2010) investigated the thermodynamic performance of the ORC systems used to convert energy from renewable energy sources such as solar energy and ocean thermal energy. In this study, it was aimed that the suitable working fluids determined which may yield high system efficiencies. Roy et al. (2010) performed the parametric optimization and performance analysis of a waste heat recovery system based on ORC using R12, R123, R134a as working fluids for power generation. It was determined that R123 as a working fluid, under the considered utilization of waste heat gives the best result among the selected fluids. Zare (2016) performed a thermodynamic analysis based on the second law of two different trigeneration systems for the ORC and Kalina cycle, which utilize geothermal energy as a heat source. In that study, the exergy yields of the Kalina cycle based system was calculated as 50%, while the efficiency of the ORC-based system was determined as 46%. Yamamoto et al. (2001) investigated theoretically and experimentally the performance and characteristics of the closed type Organic Rankine Cycle (ORC) using working fluids such as HCFC-123 and water. Ergun et al. (2016) made an extensive research on various application areas of ORC systems and made suggestions about the ORC systems that can be used in Turkey.

In many studies, thermodynamic analyses of working fluids used in organic Rankine cycle have been made and fluid selection has been made for maximum ORC performance in determined operating conditions. Saleh et al. (2007) used alkanes, fluorinated alkanes, ethers and fluorinated ethers as working fluids in ORC of geothermal power plants. They found the highest thermal efficiency was 0.13 with n-butane. Quoilin et al. (2013) examined different ORC processes and made analysis for various types of working fluids. Drescher and Bruggemann (2007) developed a new software to find thermodynamic suitable working fluid for ORC in biomass power and heat plants. In this study, it was found that the fluids in the family of alkyl benzenes maximized the efficiency of the system. Tchanche et al. (2009) analyzed the thermodynamic performance of different working fluids that used in a low-temperature solar organic Rankine cycle. In this study where the performances of 20 working fluids were evaluated, R134a and then R152a, R600, R600a and R290 were determined to be the most suitable fluids for low temperature applications operating at temperatures below 90°C. He at al. (2012) developed a theoretical formula to calculate the optimal evaporation temperature of subcritical organic Rankine cycle based on thermodynamic theory. They investigated the thermodynamic performance of ORC for 22 working fluids including wet, isentropic and dry fluids. In this study, it was found that the maximum net power output of ORC changed from 9.43 kW to 9.61 kW by using R600, R245fa, R600a, R142b and R114.

R245fa, which is an isentropic refrigerant, is known for its common use recently, and there are so many theoretical and experimental studies on this refrigerant. Wang et al. (2010) designed an experimental prototype of a low temperature solar Rankine system utilizing R245fa. In this experimental study, it was determined that expander worked with an average isentropic efficiency of 45.2%. Kaynakli et al. (2017) studied on the thermodynamic analysis of a basic and simple ORC for some determined operation conditions in which auxiliary heat exchanger does not exist. In this study, it was found that the increase of the cycle thermal efficiency becomes maximum of 47.6% with the use of R245fa depending on the increase of the geothermal source temperature.

Many researchers have been done a lot of work to optimize the Organic Rankine cycle parameters. Hettiarachchi et al. (2007) presented an optimum design of an ORC driven by low-temperature geothermal water, with the screening criterion of total heat transfer area to the net power out. In addition, it was found that the choice of the working fluid can greatly affect the power plant cost, in some instances the difference could be more than twice. Cayer et al. (2010) and Zhang et al. (2011) conducted a parametric investigation for transcritical and subcritical ORC systems. Quoilin et al. (2011) focused both on the thermodynamic and economic optimization of a small-scale ORC in waste-heat recovery application. Cihan (2014) modelled a system that combines organic Rankine cycle, which uses low temperature waste heat sources, and traditional vapor-compression refrigeration cycle. R600, R600a and R601 refrigerants were used for the system. Pulyaev et al. (2013) 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.

In order to improve ORC thermal performance, an internal heat exchanger could be conducted to exchange heat between the fluid leaving the turbine and that before entering the evaporator to reduce heat rate input of the cycle. Yamankaradeniz et al. (2018) examined the energy and exergy analysis for the unit flow rate of refrigerant (R600a) of a sample organic Rankine cycle with a heat exchanger that produces energy via a geothermal source with a temperature of 140ºC. It was found that the exergy efficiency of cycle showed a 6.21% improvement when the effectiveness of heat exchanger and evaporator temperature are taken into consideration. Deethayat et al. (2015) investigated performance of a 50kW ORC with internal heat exchanger, using mixture of R245fa/R152a as refrigerant.

In some studies in the literature, statistical methods are usually used to optimize the working conditions of the thermal systems and to determine the impact rates of the parameters affecting the system. Taguchi method, which is one of the statistical methods used for this purpose, is very useful for determining the best combination among different levels of different parameters. Turgut et al. (2012) determined the optimum design parameters of the concentric heat exchanger with the injector turbulators by using Taguchi experimental design method. They investigated the effects on the heat transfer and pressure loss of the injector-shaped turbulators having different diameter, angle and number. Verma and Murugesan (2014) performed the performance analyzes of a solar assisted ground source heat pump using Taguchi technique and utility concept. Zeng et al. (2010) analyzed the influence of various design parameters on the heat transfer and flow friction characteristics of a heat exchanger with vortex-generator fins with numerical method. In this study the parameters of vortex-generator fin and tube heat exchangers were optimized by the Taguchi method. Yakut et al. (2006) analyzed the influences of the different kinds of design parameters on thermal resistance and dimensionless pressure drop for heat sinks having hexagonal fins by using the Taguchi method. Arslanoglu and Yigit (2017) used the Taguchi method to investigate the efficient parameters on optimum insulation thickness in terms of the order of importance. They found that heating degree-day value is the most effective parameter on the optimum insulation thickness with a contribution ratio of 34.53% percent of the total effect. Li et al. (2019) performed a comparative analysis on the system thermal efficiency and the multi-objective optimization results to investigate the influence of the turbine efficiency model selection on the ORC system.

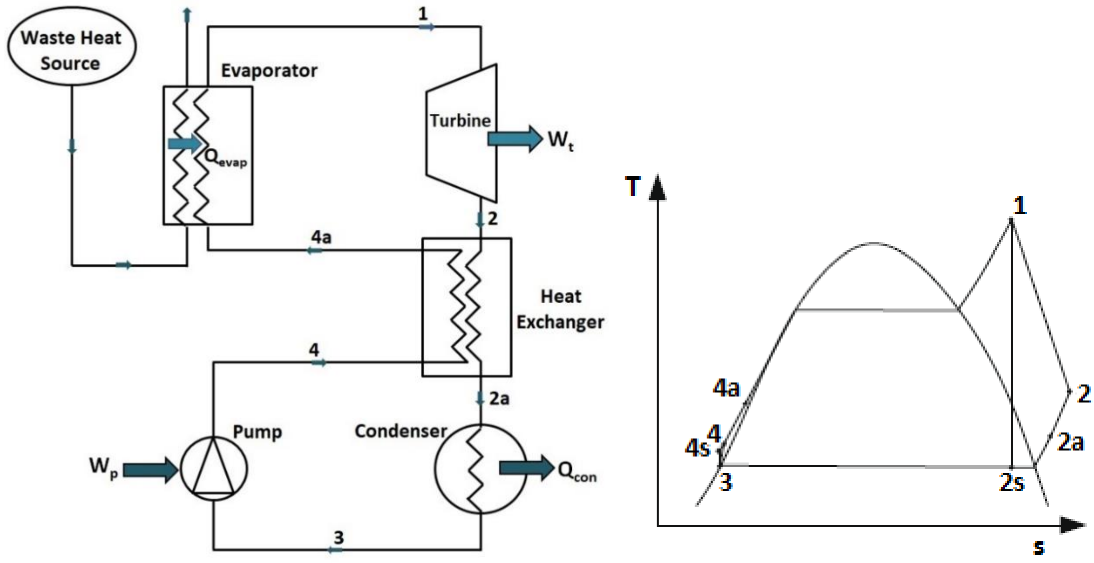
Generally, studies in the literature focus on mathematical modeling and thermodynamic analysis of ORC. The main difference of the present study from other studies in the literature is to determine the effective parameters of the system, optimum working conditions and the thermal efficiencies in these conditions by carrying out a statistical analysis, Taguchi and ANOVA methods. In this study, the performance of a case study of Organic Rankine Cycle that produces electricity from any waste-heat source is analyzed. As the working fluid to be used in the cycle, the commonly used R134a, R236fa, R245fa, R600a, R717 and R718 are preferred**.** For the selected fluids, the necessary cycle performance in order to provide 1 MW of work in turbine is examined; at the first stage according to heat source temperature (80°C-109°C), at the second stage according to the effectiveness of the heat exchanger; and the capacity changes in the system components are calculated. The parameters affecting the performance of the cycle, their contribution ratios and the order importance of these parameters are evaluated. Thus, the best and worst operating conditions of the cycle are determined within the parameters examined.

MATERIALS AND METHODS

Organic Rankine Cycle

Thermodynamically, Organic Rankine Cycles work with the same principle with ordinary Rankine cycles. 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 is composed of a pump, evaporator, turbine and condenser. The schematic diagram of the ORC with heat exchanger, the thermodynamic analysis of which is done in this study, is shown in Figure 1(a) and the T-s diagram is given in Figure 1(b).

The heat exchanger in the system is generally used for reducing the evaporator capacity (the heat energy provided from the source) and the improvement of system efficiency.



(a) (b)

Figure 1. The schematic (a) and T-s (b) diagrams of ORC with heat exchanger

The Performance Analysis of Organic Rankine Cycle

In the performance analysis of the organic Rankine cycle, the first law of thermodynamics is used in order to find the performance of the individual system components and thermal efficiency of the cycle.

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

$$\dot{Q}_{12s} - \dot{W}_{T12s} = H_{2s} - H_1 = \dot{m}\ (h_{2s} h_1) \quad (1)$$

Necessary arrangements are made to calculate the isentropic turbine work through the following equation:

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

The actual turbine work between 1 and 2, and the actual enthalpy in the turbine outlet are calculated with the help of the following equation:

$$\eta_{T,isen} = \dot{W}_{T12}/\dot{W}_{T12s} = (h_1 - h_2)/(h_1 - h_{2s}) \quad (3)$$

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.

The heat exchange in the condenser between 2a and 3 is calculated using the equation:

$$\dot{Q}_{con} = \dot{Q}_{23} = \dot{m}\ (h_{2a} - h_3) \quad (4)$$

The heat exhausted from condenser can be calculated using the values of the cooling water with the help of the following equation:

$$\dot{Q}_{con} = \dot{m}_{cw} c_{p_{cw}}\ (T_{in} - T_{out}) \quad (5)$$

In this equation, $\dot{m}_{cw}$ means the amount of the cooling water, $c_{p_{cw}}$ means the specific heat of the cooling water, T_{in} and T_{out} mean the inlet and outlet temperatures of the cooling water.

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:

$$-w_{P34s} = \int v\, dP = v_3\ (P_4 - P_3) = h_{4s} - h_3 \quad (6)$$

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

$$\eta_{P,isen} = \dot{W}_{P34s}/\dot{W}_{P34} = (h_{4s} - h_3)/(h_4 - h_3) \quad (7)$$

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.

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

$$\dot{Q}_{evap} = \dot{Q}_{41} = \dot{m}\ (h_1 - h_{4a}) \quad (8)$$

Between 2 and 4, heat transfer occurs in the heat exchanger. Thus, as the temperature in point 2a decreases, the temperature in point 4a increases. The heat transfer in the heat exchanger is calculated by one of the following equations:

$$\dot{Q}_{exc} = \dot{m} c_{p2} (T_2 - T_{2a}) \quad (9)$$

$$\dot{Q}_{exc} = \dot{m}c_{p4}(T_{4a} - T_4) \quad (10)$$

In these equations, c_{p2} is the specific heat of the working fluid at point 2 and c_{p4} is the specific heat of the working fluid at point 4. The following equation is maintained by using Eq. (9) and (10).

$$\dot{Q}_{exc} = \varepsilon\left(\dot{m}c_p\right)_{min}(T_2 - T_4) \quad (11)$$

In this equation, ε means the effectiveness of the heat exchanger.

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:

$$\eta_{thermal} = \dot{W}_{net}/\dot{Q}_{evap} = \frac{(\dot{W}_{turb} - |\dot{W}_{pump}|)}{\dot{Q}_{evap}} \quad (12)$$

Assumptions

In order to simplify the analysis, following assumptions are made:

- The system runs under steady-state conditions.
- Pressure losses in the heat exchanger and in all pipelines are negligible.
- The refrigerant at the outlet of condenser is saturated liquid.
- The efficiencies of the turbine and pump are assumed to be constant for all working fluids.
- The specific volume of the working fluid remains constant during pumping.

Taguchi Method

Taguchi method is one of most useful and reliable design and optimization technique to determine the optimal combination of different parameters for the target function. The Taguchi method contributes an effective and systematic way to obtain the result with far less experiments. In Taguchi methodology, it is important to select the suitable orthogonal array. It should be selected depending on the total degree of freedom (DOF), which can be calculated by summing the individual DOF of each process parameter. The DOF for each factor is the number of factor levels minus 1 (Yuce et al., 2016).

The results of the objective functions are converted into the S/N ratio for the statistical analysis. There are three different S/N ratio equations depending on the objective function type, i.e., the lower is the better, the higher is the better and nominal is the best. Due to the fact that, maximum thermal efficiency is the objective function, the higher is the better situation has been selected in this study. The S/N ratio equation for the higher is the better situation is given as follows:

$$S/_N = -10log\left(\frac{1}{n}\sum_{i=1}^{n} 1/{y_i}^2\right) \quad (13)$$

In this equation n represents the number of tests (i.e., number of case) and y_i defines the resulting value for the ith performance characteristics. In this study, y_i values are the thermal efficiencies of ORC for each case.

Analysis of Variance (ANOVA)

ANOVA is another statistical approach, which is used for determining the importance of each parameter on the performance characteristic. Due to the fact that ANOVA demonstrates the importance level of influencing factors on response, it also enables to check the statistical reliability of the results obtained from Taguchi method. In this study, the effect of each parameter on the thermal efficiency of ORC are determined using this method and calculation results are analyzed. In the ANOVA analysis, the significance level of the statistical analyses is 0.01 which corresponds to 99% confidence level. For the statistical reliability of the results, F-test has been carried out. The F-test (F_{factor}) values, which are the ratio between the regression mean square and the mean square error, have been determined by using ANOVA method. If the calculated F-test value is greater than F value from the appropriate standard confidence table, the relevant parameter is considered statistically significant. Following equations can be used for calculating F values, sum of squares (SS), mean of squares (MS) and DOF of each factor (Ross, 1996):

$$F_{factor} = \frac{V_{factor}}{V_{error}} \quad (14)$$

$$V_{factor} = \frac{SS_{factor}}{DOF_{factor}} \quad (15)$$

$$DOF_{factor} = k - 1 \quad (16)$$

$$SS_{factor} = \frac{\sum {\beta_{factor,i}}^2}{N} - \frac{(\sum \beta_i)^2}{n} \quad (17)$$

where F_{factor} indicates whether the factor is associated with the response. The larger F_{factor} value states that the parameter has a greater effect on the response. V_{factor} and V_{error} values are the variance of the factor and error, respectively. DOF_{factor} is number of factor's degree of freedom, SS_{factor} is the sums of squares due to factor, $\beta_{factor,i}$ is the sum of the S/N ratio at the ith level of the factor, β_i is the S/N ratio at the ith level of the factor, N is repeating number of each level's factor, n is the number of tests. In these equations, "factor" represents the name of the individual factors (Celik and Turgut, 2012). MS is equal to the ratio of the SS values of each parameter to the DOF of each parameter.

Validation

In order to ensure the reliability of the statistical analysis to be carried out within the scope of this study, the developed thermodynamic model should be verified. Therefore, using the thermodynamic model, system efficiency was calculated for the different working

Table 1. Validation parameters and thermal efficiencies

Study	Fluids	T_{evap}	ε	η
Dai et al. (2009)	R718	83.44	1	0.1254
Present Study	R718	83.44	1	0.1226
Shengjun et al. (2011)	R134a	72	-	0.0853
Present Study	R134a	72	-	0.0861
Facao et al. (2008)	R718	120	0.8	0.1259
Present Study	R718	120	0.8	0.1298
Khennich and Galanis (2012)	R134a	100	-	0.1126
Present Study	R134a	100	-	0.1086

Table 2. Thermodynamic properties of working fluids. (He et al., 2012; Cihan, 2014)

Working Fluids	Molecular Mass (g/mol)	Critical Temperature (K)	Critical Pressure (MPa)
R134a (CF_2CH_2F)	102.03	374.2	4.06
R236fa ($CF_3CH_2CF_3$)	152.04	398.1	3.20
R245fa ($C_3H_3F_5$)	134.05	427.2	3.64
R600a (C_4H_{10})	58.10	408.1	3.65
R717 (NH_3)	17.03	405.4	11.3
R718 (H_2O)	18.02	647.1	22.1

parameters discussed in this study and the results were compared with the studies in the literature (Dai et al., 2009; Shengjun et al., 2011; Facao et al., 2008; Khennich and Galanis, 2012). In general, when Table 1 is examined, it is seen that the developed thermodynamic model is very compatible with the studies in the literature and the obtained results are reliable.

RESULTS AND DISCUSSION

The thermodynamic properties of the working fluids that are used in this study are presented in Table 2.

The process parameters and conditions to be used for the thermodynamic analysis of the system components are presented in Table 3.

As the working fluid in the system receives energy from waste-heat source, it evaporates in the evaporator. In order to provide heat transfer, the evaporation temperature in the evaporator is lower than that of the waste-heat source. In the performance analysis based on the parameters in Table 3, in order to have superheated vapor in the inlet of the turbine (at the point 1), the temperature of this point can be assumed as 5°C higher than the evaporation temperature in the evaporator (Yamankaradeniz et al., 2018; Li et al., 2014). The condenser temperature of the system is taken as 30°C (Saleh et al., 2007; Kaynakli et al., 2017).

Table 3. Process parameters and conditions

Parameters	Values
Waste-Heat Source Temperature	80-109°C
Evaporation Temperature	72-101°C
Turbine Power	1 MW
Condensing Temperature	30°C
Effectiveness of the Heat Exchanger	0.4-0.9
Isentropic Efficiency of Turbine	85%
Isentropic Efficiency of Pump	80%

According to the results of the thermodynamic analysis based on the defined process conditions and given equations, the thermodynamic properties of the dry type refrigerant R245fa is given for every point of the cycle in Table 4. (T_s=109°C, ε = 0.65)

The Performance Analysis Based on Waste-Heat Source Temperature

Figure 2 shows the changes in the flow rates of the working fluids, necessary to produce 1MW turbine work, depending on the waste-heat source temperature. All the flow rates of the examined working fluids decrease as the temperature of the waste-heat source increases. R236fa is the fluid with the highest flow rate required for 1 MW turbine work whereas R718 is the fluid with the lowest flow rate (Figure 2).

Table 4. The thermodynamic properties of R245fa at each point of the cycle

R245fa	*State*	*T(°C)*	*P(kPa)*	*h(kJ/kg)*
1	Superheated Vapor	106.00	1297.32	481.43
2s	Superheated Vapor	47.83	177.18	443.72
2	Superheated Vapor	53.67	177.18	449.38
2a	Superheated Vapor	38.65	177.18	434.84
3	Saturated Liquid	30.00	177.18	239.10
4s	Compressed Liquid	30.40	1297.32	239.95
4	Compressed Liquid	30.56	1297.32	240.16
4a	Compressed Liquid	41.52	1297.32	254.87

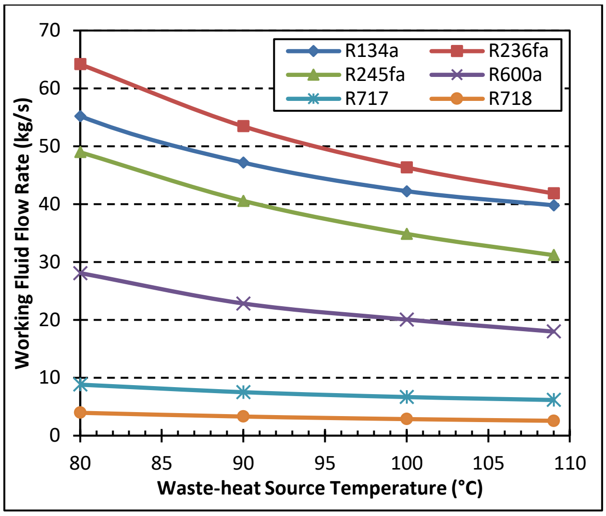
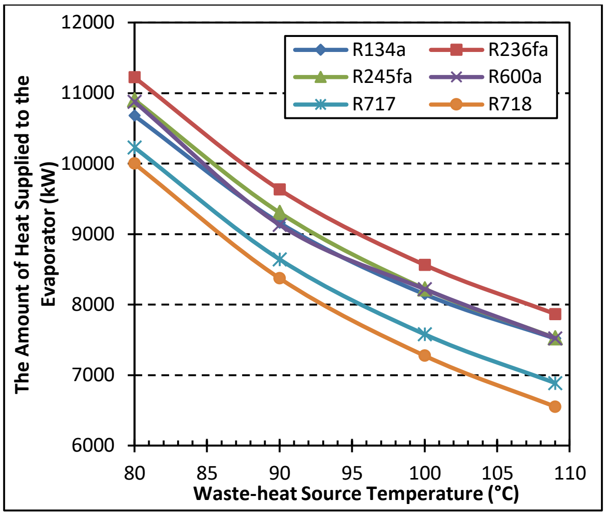


Figure 2. The change of the working fluid flow rates, necessary to produce 1MW turbine work, depending on the waste-heat source temperature

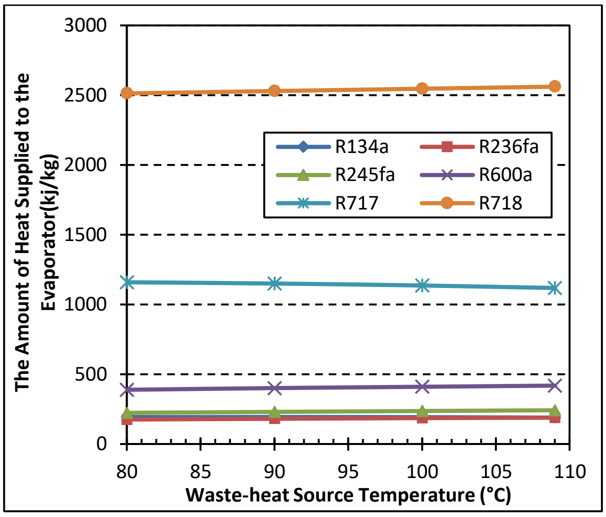
The change in the amount of heat transferred from waste-heat source to the evaporator, depending on the waste-heat source temperature is shown in Figure 3(a). In Figure 3(b) the amounts of heat transferred to the evaporator for the unit flow rate of the working fluids is given. As it can be seen in Figure 3(a), for all examined working fluids, as the temperature of the waste-heat source increases, the heat transferred from waste-heat source to evaporator decreases. However, when the amount of heat transferred to the evaporator for a unit flow rate is examined, it is seen that there is no significant change due to the waste-heat source temperature (Figure 3(b)). The reason for this is the decrease in both the flow rate of working fluid (Figure 2) and the amounts of heat transferred to the evaporator, as the waste-heat source temperature increases.

It can be seen in Figure 3(a) that, the heat that is transferred to the evaporator in order to obtain 1MW turbine work has a higher value if R236fa is used, has a lower value if R718 is used. In addition to this, related to the changes in the flow rates of working fluid (Figure 2), the amount of heat transferred to the evaporator for a unit flow rate has a higher value when R718 is used, has a lower value when R236fa is used (Figure 3(b)).

In Figure 4(a), the change of pump work required to obtain 1MW of turbine work, depending on the waste-heat source is given. In Figure 4(b), the change of pump work required for unit working fluid flow rate, depending on the waste-heat source temperature is given. For all working fluids that are examined, as waste-heat source temperature increases the necessary pump work also increases. Furthermore, the decrease in the flow rates of working fluid due to the increasing waste-heat source temperature causes the pump work for unit working fluid flow rate to increase.



(a)

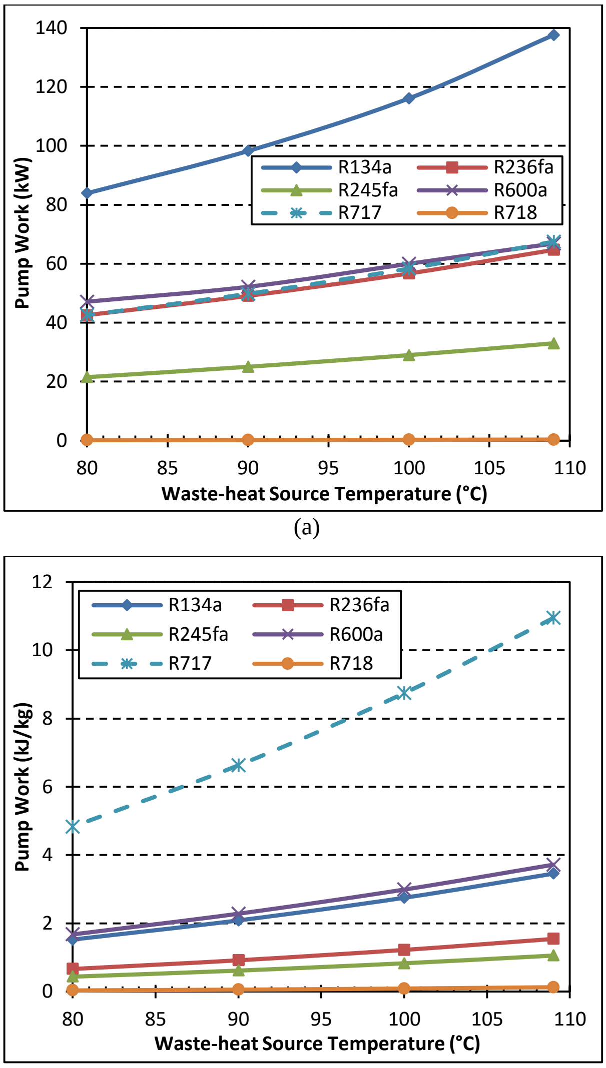


(b)

Figure 3. The change in the amount of heat that is transferred to evaporator in order to obtain 1MW turbine power, depending on the waste-heat source temperature

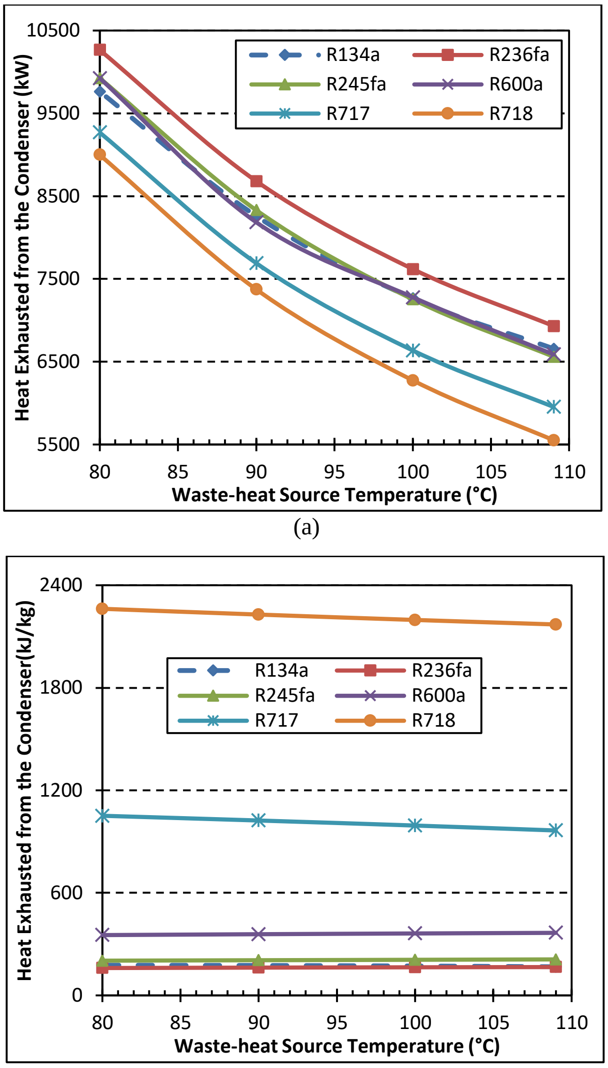
It can be seen in Figure 4(a) that, the necessary pump work required to obtain 1MW turbine work is higher if R134a is used, is lower when R718 is used. In addition to this related to the change in the flow rates of working fluid (Figure 2), the necessary pump work for unit working fluid flow rate is higher when R717 is used, is lower when R718 is used (Figure 4(b)).

In Figure 5(a) the change in the amount of heat exhausted from condenser in order to obtain 1MW turbine work, depending on the waste-heat source temperature is given. In Figure 5(b) the change in the amount of heat exhausted from condenser, depending on the unit working fluid flow rate is shown. As it can be seen in Figure 5(a) for all working fluids that are examined the heat that is transferred from the condenser decreases as the waste-heat source temperature increases. But for a unit working fluid flow rate, the heat transferred from condenser according to the source temperature increases in dry type working fluids (R236fa, R245fa, R600a), decreases in wet type working fluids (R134a, R717, R718) (Figure 5(b)).



(b)

Figure 4. The change of necessary pump work required to obtain 1MW turbine work, depending on the waste-heat source temperature



(b)

Figure 5. The change in the amount of heat exhausted from condenser in order to obtain 1MW turbine work, depending on the waste-heat source temperature

The amount of heat exhausted from condenser in order to obtain 1MW turbine work is higher if R236fa is used, is lower if R718 is used (Figure 5(a)). In addition to this, with the effect of the working fluid flow rates that change depending on the waste-heat source temperature, the amount of heat exhausted from condenser for unit flow rate is higher for R718 and lower for R236fa (Figure 5(b)).

In Figure 6, the change in the thermal efficiency of organic Rankine cycle in order to obtain 1MW turbine work, depending on waste-heat source temperature is shown. For all working fluids that are examined, the thermal efficiency of the cycle increases depending on the waste-heat source temperature as expected. As it can be seen it Figure 6, although R718 and R717 are the working fluids that provide the highest thermal efficiencies respectively, it is seen that the vapor quality in the turbine outlet decreases to 0.89 under these process conditions. These conditions can be disadvantageous for the turbine (Cengel and Boles, 2011). Because of this, it should be better to use R245fa instead of R717 and R718.

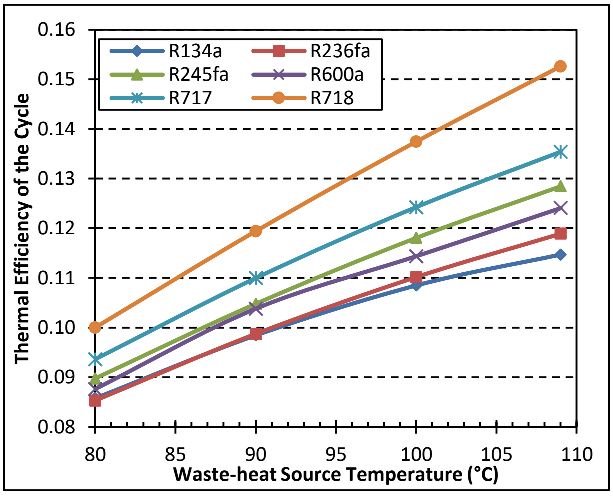


Figure 6. The change of the thermal efficiency of the cycle depending on waste-heat source temperature

Performance Analysis Based on Effectiveness of the Heat Exchanger

If wet type working fluids (R134a, R717 and R718) are used in the cycle, as the working fluids leaves the turbine in condensing temperature there is no need for a heat exchanger. However, if a dry type fluid is selected as the working fluid, the steam at the turbine outlet is in the superheated steam region, so using a heat exchanger is beneficial to the system. For this reason, in this study, the examination of the heat exchanger effect was carried out on dry type fluids (R236fa, R245fa and R600a).

In case where R245fa is selected as the working fluid, the capacities of the cycle components according to the effectiveness of the heat exchanger for different waste-heat source temperatures (80°C and 109°C) are listed in Table 5.

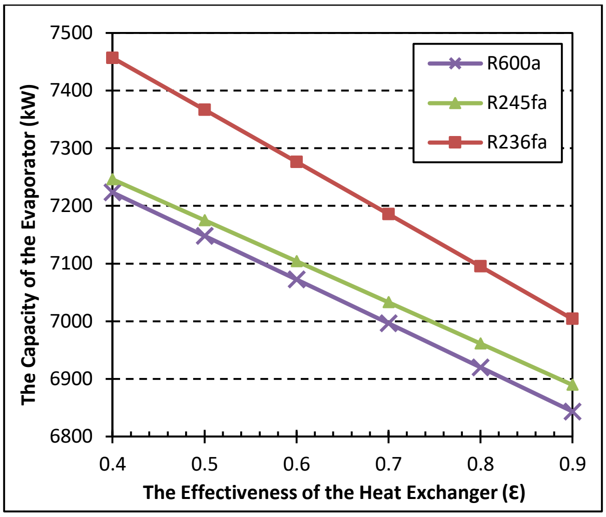
As it can be seen in Table 5, as the effectiveness of the heat exchanger increases, the capacities of evaporator and condenser decrease, and depending on the decrease in evaporator capacity, the thermal efficiency of the system increases. Also, it can be seen that the effectiveness of the heat exchanger has no direct effect on the work done by turbine and pump.

In Figure 7, for 109*°C* of waste-heat source temperature, the changes in the capacities of evaporator (Figure 7(a)) and condenser (Figure 7(b)), depending on the effectiveness of the heat exchanger are given. As expected the evaporator and condenser capacities decrease as the effectiveness of the heat exchanger increases. In this study, when working fluid selection and effectiveness of the heat exchanger are considered, a maximum of 6.07% decrease is maintained in evaporator capacity and a maximum of 6.59% decrease is maintained in condenser capacity.

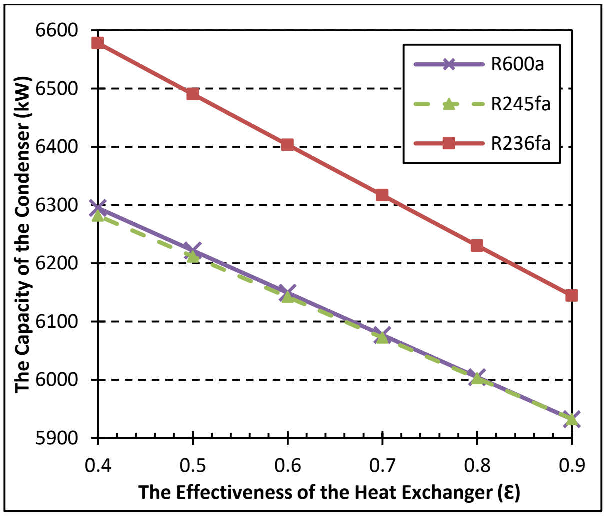
In Figure 8, the change in heat exchanger capacity depending on the effectiveness of the heat exchanger is given for different waste-heat source temperatures. In Figure 8(a) the waste-heat source temperature is taken as 109°C, and in Figure 8(b) the waste-heat source temperature is taken as 80°C. As the heat-source temperature decreases, the heat exchanger capacity increases in order to obtain 1 MW turbine work.

When Figure 8(a) and 8(b) is examined, it can be seen that the heat exchangers capacity increases as the effectiveness of the heat exchanger increases. In both waste-heat source temperatures, when R236fa is used as the working fluid the needed heat exchanger capacity is at maximum, and when R245fa is used the heat exchanger capacity is at minimum. When the waste-heat source temperature drops from 109°C to 80°C, the exchanger capacity increases by 15.67% at maximum, depending on the effectiveness of the heat exchanger and working fluid.

In order to inspect the temperature changes in heat exchanger, the change in turbine and pump outlet temperatures depending on the effectiveness of the heat exchanger is given for the examined refrigerant R245fa in Figure 9. In Figure 9(a) the waste-heat source temperature is taken as 109°C, and in Figure 9(b) the waste-heat source temperature is taken as 80°C. In both cases as the effectiveness of the heat exchanger increase, the condenser inlet temperature of the working fluid decreases, and the inlet evaporator temperature of the working fluid increases. As the temperature of the waste-heat drops from 109°C to 80°C, the temperature change of the working fluids becomes less.



(a)



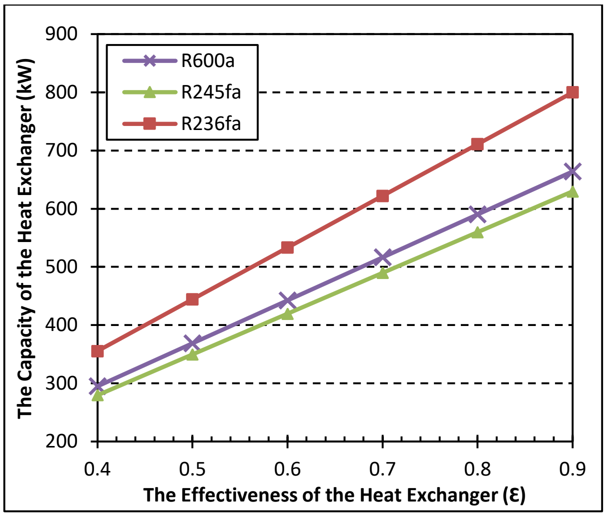
(b)

Figure 7. For 100°C of waste-heat source temperature, the change in evaporator and condenser capacities depending on the effectiveness of the heat exchanger (a) the capacity of the evaporator (b) the capacity of the condenser.

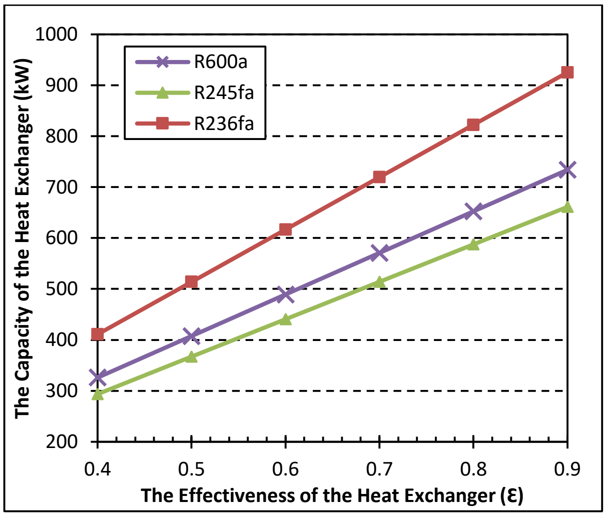
Table 5. The capacities of the cycle components according to the effectiveness of the heat exchanger for 80°C and 109°C waste-heat source temperatures.

R245fa	***Effectiveness of the heat exchanger ε****					
	0.4	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**
Q_{evap} (kW)	10609.4/7246.3	10535.3/7175.4	10461.1/7104.3	10386.7/7033.0	10312.1/6961.5	10237.4/6889.7
Q_{con} (kW)	9632.2/6281.4	9558.7/6211.6	9485.2/6141.9	9411.7/6072.1	9338.1/6002.3	9264.4/5932.4
W_T (kW)	1000.0/1000.0	1000.0/1000.0	1000.0/1000.0	1000.0/1000.0	1000.0/1000.0	1000.0/1000.0
W_p (kW)	21.5/33.0	21.5/33.0	21.5/33.0	21.5/33.0	21.5/33.0	21.5/33.0
Q_{exc} (kW)	293.8/279.8	367.3/349.8	440.7/419.7	514.2/489.7	587.6/559.6	661.1/629.6
η (%)	9.22/13.35	9.29/13.48	9.35/13.61	9.42/13.75	9.49/13.89	9.56/14.04

** The capacities that are calculated depending on the effectiveness of the heat exchanger are shown for 80°C and 109°C waste-heat source temperatures, respectively.*

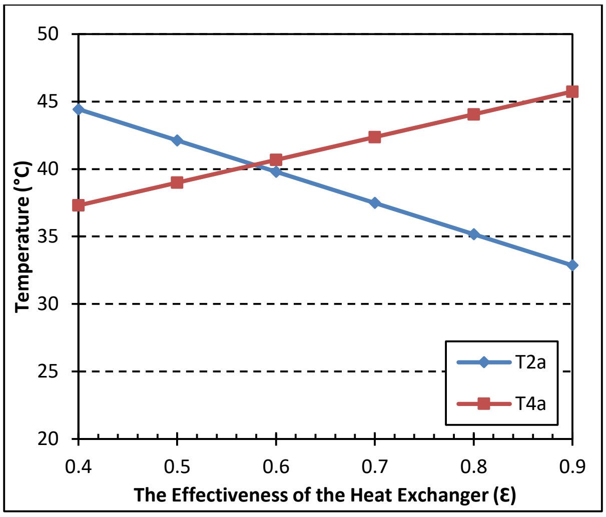


(a)

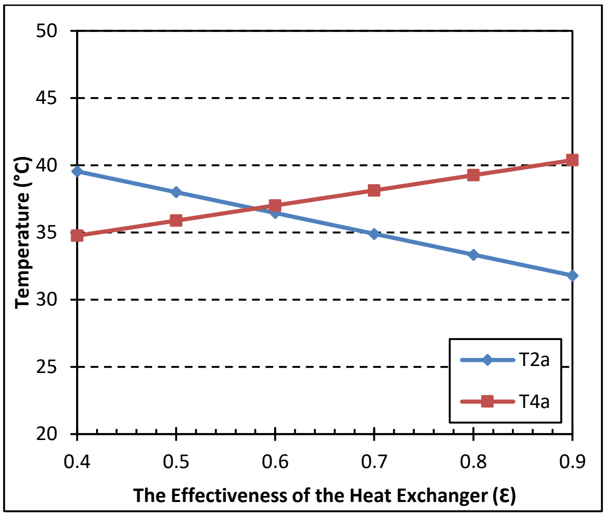


(b)

Figure 8. The change of heat exchanger capacity according to the effectiveness of the heat exchanger for different waste-heat source temperatures (a) T_s=100°C (b) T_s=80°C



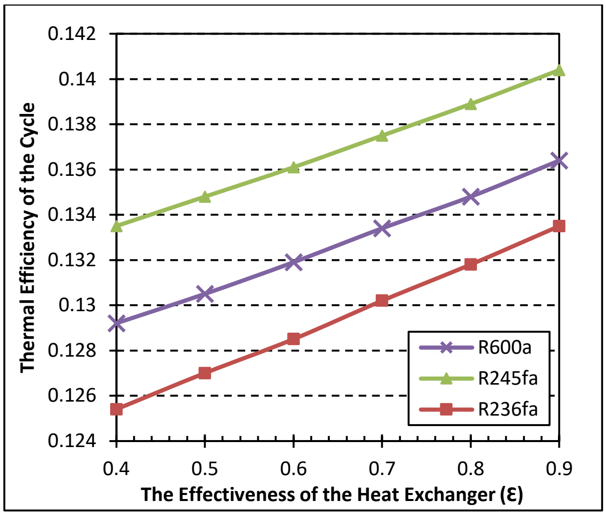
(a)



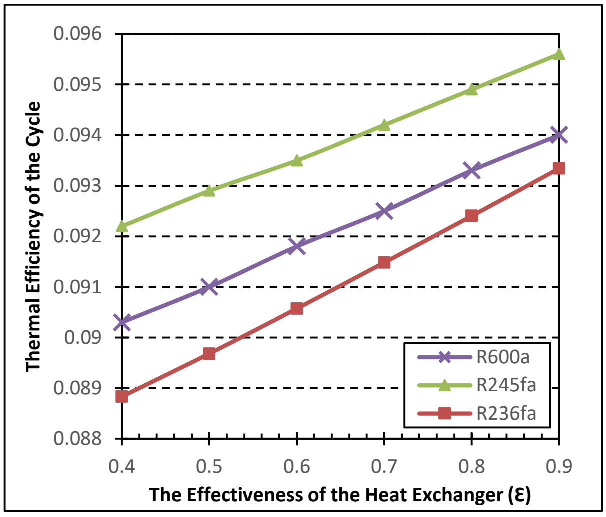
(b)

Figure 9. The changes in evaporator and condenser inlet temperatures when R245fa is used as the working fluid, depending on heat exchanger capacity (a) T_s=100°C (b) T_s=80°C

In Figure 10, the change in cycle thermal efficiency depending on the effectiveness of the heat exchanger is given for different waste-heat source temperatures. In Figure 10(a) waste-heat source temperature is taken as 109°C and in Figure 10(b) it is taken as 80°C. For all three working fluids, the thermal efficiency of the cycle increases depending on the increase of the effectiveness of the heat exchanger. It is found that at 109°C of waste-heat source temperature, heat exchanger can increase the thermal efficiency 6.46% at maximum. But when the waste-heat source temperature drops from 109°C to 80°C, the heat exchanger can increase the thermal efficiency of the cycle just 5.08% at maximum.



(a)



(b)

Figure 10. The variation of cycle thermal efficiency according to the effectiveness of the heat exchanger at different waste-heat source temperatures. (a) T_s=100°C (b) T_s=80°C

Statistical Analysis

A statistical analysis has been used to find out the effect ratios and the order of importance of the parameters (i.e. working fluid selection, waste-heat source temperature and effectiveness of heat exchanger) which are presented in the figures above. In this study, Taguchi and ANOVA method have been selected for the statistical analysis to determine the contribution ratios of each factor and how they affect the ORC thermal efficiency.
Traditionally, Taguchi method which has been widely used as a statistical analysis technique, is applied to experimental studies. Nevertheless, the number of numerical and theoretical studies using the Taguchi method has been increasing over the last decades. In this study, in addition to parametric analysis, Taguchi method was used to find out the contribution ratio of each parameter to thermal efficiencies of the ORC. In the analysis, the ranges of the parameters (levels) has been selected based on previous works in the literature (Saleh et al., 2007; Quoilin et al., 2013; Kaynakli et al. 2017; Deethayat et al., 2015; Yamankaradeniz et al., 2018) considering the critical temperatures of the fluid types and they are presented in Table 6.

It was mentioned that the importance of selecting suitable orthogonal array. Due to the fact that there are two factors with three levels and one factor with six levels, the total DOF number is obtained to be 17. According to the Taguchi design methodology, the total DOF should be lower than the DOF of the selected orthogonal array. Thus, Taguchi orthogonal array of L18 (6^1 x 3^2) is established in this study as shown in Table 7. For each case, thermal efficiencies are calculated and converted into S/N ratio that can show us which levels of control factors are more efficient. Calculated S/N ratios obtained from the thermal efficiencies are shown for each case in Table 7.

Table 7. Thermal efficiencies and S/N ratios for the L18 orthogonal array

	A	*B*	*C*	*Thermal Efficiency*	*S/N Ratio*
Case	*Levels*				
1	1	1	1	0.08618	-21.292
2	1	2	2	0.10340	-19.710
3	1	3	3	0.11330	-18.915
4	2	1	1	0.09352	-20.582
5	2	2	2	0.11710	-18.629
6	2	3	3	0.13470	-17.413
7	3	1	2	0.09999	-20.001
8	3	2	3	0.12860	-17.815
9	3	3	1	0.15260	-16.329
10	4	1	3	0.09310	-20.621
11	4	2	1	0.10970	-19.196
12	4	3	2	0.12890	-17.795
13	5	1	2	0.09387	-20.549
14	5	2	3	0.12050	-18.380
15	5	3	1	0.13350	-17.490
16	6	1	3	0.09404	-20.534
17	6	2	1	0.11240	-18.985
18	6	3	2	0.13270	-17.543

Table 6. Process parameters and their levels used in this study

	Parameters	*1*	*2*	*3*	*4*	*5*	*6*
A	Working Fluid	R134a	R717	R718	R236fa	R245fa	R600a
B	Waste-Heat Source Temperature	80	95	109			
C	Effectiveness of Heat Exchanger	0.4	0.65	0.9			

Table 8. Response table for the S/N ratios for the thermal efficiencies.

		Parameters		
	Level	***A***	***B***	***C***
S/N	1	-19.97	-20.6	-19.06
	2	-18.87	-18.79	-19.04
	3	-18.05	-17.58	-18.95
	4	-19.2		
	5	-18.81		
	6	-19.02		
Delta (max-min)		1.92	3.02	0.11
Rank		2	1	3
Contribution Ratio (%)		38.02	59.80	2.18

Table 9. ANOVA table

Parameters	***DOF***	**SS**	**MS**	**F**	**Rank**
Working Fluid	5	5.8358	1.1672	15.35*	2
Waste-Heat Source Temperature	2	27.6498	13.8249	181.83*	1
Effectiveness of Heat Exchanger	2	0.0257	0.0129	0.17	3
Error	8	0.6083	0.076		
Total	17	34.1195			

**Significant at 99% confidence level*

Average S/N ratios and ranking of parameters are presented in response table for the S/N ratios (Table 8). In this table, Delta states the difference between maximum and minimum of the S/N ratio on each parameter's. Rank is the order of parameter according to cycle's thermal efficiency. In addition; contribution ratios which represent the degree of impact is obtained to analyze the results in detail, seen in Table 8 and Figure 11. The contribution ratio is equal to the ratio of the Delta values of each parameter to the total Delta value of all parameters (Gunes et al., 2011). As it can be seen in Figure 11, waste-heat source temperature and working fluid selection play a crucial role in terms of thermal performance. As seen in Table 8, the parameter B (waste-heat source temperature) is the most effective parameter on thermal efficiency with a contribution ratio of 59.80% and parameter A (working fluid) follows this parameter with a contribution ratio of 38.02% of the total effect. Compared with these two parameters, effectiveness of heat exchanger has a slight effect on ORC thermal performance with a contribution ratio of 2.18%. This parameter is considered statistically insignificant as a result of F-test and 99% reliable test comparisons (Table 9). Therefore, waste-heat source temperature and working fluid can be considered as the main parameters for optimum design and working condition of the ORC. As mentioned before, the larger F value states that the parameter has a greater effect on the response. According to F values obtained in ANOVA analysis, Rank (order of importance) is determined in Table 9. As a result of both analysis (Taguchi and ANOVA), the order of importance of the parameters is found to be same.

S/N ratio variation of each parameter which can be used to determine the optimum parameter combination are shown in Figure 12. The small difference between the largest and smallest S/N ratios indicates that the parameter slightly affects the objective function. On the contrary, if the difference is large, it means that the parameter highly affects the objective function. Moreover, the level with the largest S/N ratio gives the optimum level of design parameters. Therefore, in this study, the optimum parameter combination is determined to be R718 (A_3) for working fluid, 109°C (B_3) for waste-heat source temperature and 90% (C_3) for effectiveness of heat exchanger. When the optimum condition ($A_3B_3C_3$) is selected for the ORC, thermal efficiency is calculated as 15.26%, which is the maximum value that can be obtained under these operating conditions. In addition to the best condition, the worst condition has also been determined as $A_1B_1C_1$. Minimum ORC thermal efficiency has been calculated as 8.61% under these working conditions (working fluid = R134a (A_1), waste-heat source temperature = 80°C (B_1) and effectiveness of heat exchanger = 0.4 (C_1)).

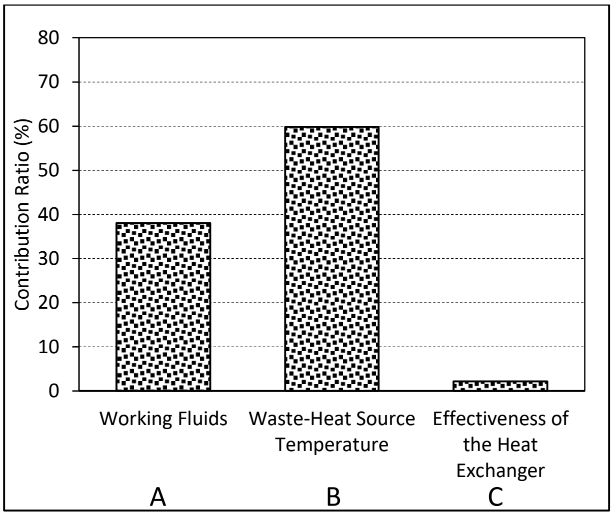


Figure 11. Contribution ratio of each parameter to the thermal efficiency

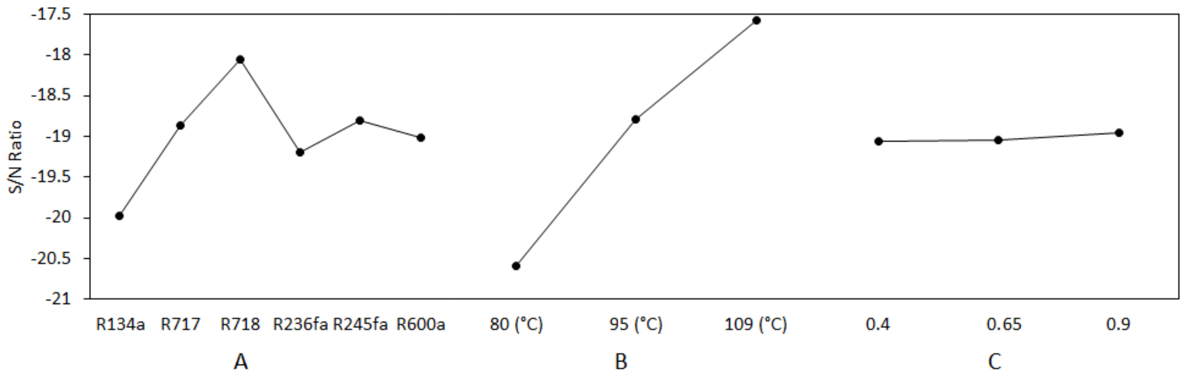


Figure 12. The effects of each design parameter on thermal efficiency

CONCLUSION

In this study, in addition to parametric analysis, the effective parameters on the Organic Rankine Cycle with heat recovery exchanger, optimum working conditions and the thermal efficiencies in these conditions are determined by carrying out a statistical analysis, Taguchi and ANOVA methods. The main results of the study are given as follows:

- Generally, the increase of waste-heat source temperature causes the cycle thermal efficiency to increase. For the conditions examined in this study, in the analysis based on waste-heat source temperature, it is seen that R245fa should be selected because it provides a higher thermal efficiency and it is at super-heated steam phase in the turbine outlet.
- In the calculations, the vapor quality at the turbine outlet (at the point 2) is obtained in the range of 0.84-0.89 for R717 and R718. Because of low vapor quality, the use of these working fluids is not recommended under these working conditions.
- When the effect of heat exchanger on the cycle efficiency is inspected, the cycle thermal efficiency can be increased 6.46% at maximum according to the working fluid selection and the effectiveness of the heat exchanger.
- When the waste-heat source temperature drops from 100°C to 80°C, the exchanger capacity increases by 15.67% at maximum, depending on the operation conditions.
- According to the effect of the effectiveness of the heat exchanger, a 6.07% of decrease at maximum is provided in evaporator capacity, and 6.59% of decrease at maximum is provided in condenser capacity.
- The flow rates of the working fluid to be used in the cycle show that; the working fluid with the lowest flow rate that is required to obtain 1 MW turbine work is R718 whereas R236fa is the fluid with the highest flow rate.
- The two most significant parameters on ORC thermal efficiency are found to be the waste-heat source temperature (59.80%) and the working fluid selection (38.02%), while the least effective parameter is found to be the effectiveness of heat exchanger (2.18%).
- The sum of the two parameters' contribution ratio (A and B) mentioned above is over 97% of the total effect, which means that they should be taken into consideration primarily when designing the cycle.
- According to Taguchi analysis, the maximum and minimum thermal efficiencies of the ORC are obtained as 15.26% and 8.61%, under these conditions $A_3B_3C_3$ and $A_1B_1C_1$, respectively.
- In the parametric analysis, waste-heat source temperature is changed between 80-109°C. When other parameters are kept constant, thermal efficiency of the cycle is increased from 9.99% to 15.25% for the R718. According to the Taguchi analysis, contribution ratio of the waste-heat source temperature on the thermal efficiency of ORC is found to be 81.04% which is the most effective parameter. On the contrary, when effectiveness of the heat exchanger is changed between 0.4-0.9, the thermal efficiency of the cycle is increased from 9.22% to 9.55% for the R245fa which is very low compared with other parameters. According to the statistical analysis result, effectiveness of heat exchanger is found to be the least effective parameter and the effect ratio is calculated as 2.18%. It is observed that the results of the parametric analysis and the statistical analysis are in a good agreement with each other.
- Consequently, instead of full factorial parametric analysis, the Taguchi method can be used in the thermodynamic analysis of ORC systems in order to find out the impact ratio of each parameters, best and worst working conditions and the thermal efficiencies under these conditions with less effort.

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