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DETERMINATION OF OPTIMUM FLUID FOR DIFFERENT HEAT SOURCE TEMPERATURES BASED ON MULTI-OBJECTIVE FUNCTIONS IN THE ORGANIC RANKINE CYCLE

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Keywords
Genetic Algorithm, Multi-Objective, Organic Rankine Cycle, Optimum Fluid, Thermodynamic Optimization.

Abstract
In this study, the optimum fluid was determined by using Non-dominated Sorting Genetic Algorithm-II (NSGA-II) within the scope of Organic Rankine Cycles (ORC) low temperature applications. Heat source temperatures are taken as 90, 100 and 110 °C. Fluid optimization was performed by comparing the performance of 8 fluids from 4 different categories under different criteria (dry-R601 and R601a, isentropic-R141b and R123, wet-R152a and R134a, new generations-R1234yf and R1234ze). Objective functions have been established under the parameters of Energy, Exergy, Economy and Environment (4E). In ORC systems, every organic fluid has certain advantages and disadvantages. It is seen that the studies on organic fluid selection meet a single goal from the system performance parameters. However, it has been observed that the turbine power performance is not at the desired level due to the required evaporator capacity of the fluid, which performs well in terms of thermal efficiency in ORC systems. Therefore, it is necessary to determine the percentage of organic fluid that can be used by optimizing it under different objective functions. In this study, the optimum fluid was determined for ORCs operating under 90, 100 and 110 °C heat source temperatures by evaluating different objective functions together.

ORGANİK RANKİNE ÇEVRİMİNDE ÇOKLU AMAÇ FONKSİYONLARINA BAĞLI OLARAK OPTİMUM AKIŞKANIN FARKLI ISI KAYNAĞI SICAKLIKLARI İÇİN BELİRLENMESİ

Anahtar Kelimeler
Genetik Algoritma, Çok Amaçlı, Organik Rankine Çevrimi, Optimum Akışkan, Termodinamik Optimizasyon.

Öz
Bu çalışmada, Organik Rankine Çevrimleri (ORÇ) düşük sıcaklık uygulamaları kapsamında baskılanamayan sıralamalı genetik algoritma-II (NSGA-II) kullanılarak optimum akışkan belirlenmiştir. Isı kaynağı sıcaklıkları 90, 100 ve 110 °C olarak alınmıştır. Akışkan optimizasyonu, 4 farklı kategoriden 8 akışkanın farklı kriterler altında performansları karşılaştırılarak yapılmıştır (kuru-R601 ve R601a, izentropik-R141b ve R123, ıslak-R152a ve R134a, yeni nesil-R1234yf ve R1234ze). Enerji, Ekserji, Ekonomi ve Çevre (4E) parametreleri altında amaç fonksiyonları oluşturulmuştur. ORÇ sistemlerinde her organik akışkanın belirli avantajları ve dezavantajları vardır. Organik akışkan seçimi ile ilgili çalışmaların sistem performans parametrelerinden tek bir amacını karşıladığı görülmektedir. Ancak ORÇ sistemlerinde ısıl verim açısından iyi performans gösteren akışkanın gerekli evaporatör kapasitesinden dolayı türbin güç performansının istenilen seviyede olmadığı gözlemlenmiştir. Bu nedenle farklı amaç fonksiyonları altında optimize edilerek kullanılabilecek organik akışkan yüzdesinin belirlenmesi gerekmektedir. Bu çalışmada farklı amaç fonksiyonlarının birlikte değerlendirilmesiyle 90, 100 ve 110 °C ısı kaynağı sıcaklıkları altında çalışan ORÇ'ler için optimum akışkan tespit edilmiştir.

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Highlights

- The economic performance of the turbine (VFR and SP) was investigated in organic fluid selection.
- Environmental sustainability indices (EES, ESI and WER) of fluids were examined.
- The optimum utilization rate of the fluid was determined with NSGA-II.

Purpose and Scope

The most important factor determining the system performance in the Organic Rankine Cycle (ORC) is the organic fluid. But there is no fluid that is ideal in all aspects. Therefore, the selection of organic fluids should not be made on a single criterion, but by considering multiple criteria. In this study, it is aimed to determine the percentage of use of fluids by using multi-objective optimization technique for different heat source temperatures.

Design/methodology/approach

In this study, the performance of 8 fluids (dry-R601 and R601a, isentropic-R141b and R123, wet-R152a and R134a, new generations-low GWP (R1234yf and R1234ze) were optimized for ORC designed under 90, 100 and 110 ^{o}C heat source temperatures. Six different objective functions are defined with thermodynamics (thermal Efficiency, turbine power, exergy destruction and exergy efficiency), turbine economy performance (Volume flow ratio-VFR, size parameter-SP, pressure ratio-PR) and environmental sustainability indices (Environmental effect factor-EEF, Waste Exergy Ratio-WER and Exergy Sustainability Index-ESI). The performance of the fluids was determined by defining the weight function, G(x), which these objective functions affect equally.

Findings

According to NSGA-II results, the best performing fluids in different criteria according to heat source temperatures are as follows.
For 90 ^{o}C heat source temperature; R141b for maximum thermal efficiency and exergy efficiency, minimum exergy destruction and EEF; R1234yf for maximum turbine power; R152a for minimum VFR.
For 100 ^{o}C heat source temperature; R141b for maximum thermal efficiency, minimum exergy destruction; R1234yf for maximum turbine power and exergy efficiency, minimum EEF; R152a for minimum VFR.
For 110 ^{o}C heat source temperature; R1234yf for maximum thermal efficiency, exergy efficiency, turbine power, minimum EEF; R141b for minimum exergy destruction; R152a for minimum VFR.
According to the weight function result, it was determined that 51% of R141b for 90 ^{o}C; 65% of R1234yf for 100 ^{o}C; 83% of R1234yf for 110 ^{o}C could be used as the optimum fluid.

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Originality

Although there are many studies on organic fluid selection, studies that examine performance under different criteria (thermodynamic, environmental sustainability and turbine economy performance), compare different fluid categories, including new-generation, and do this for different heat source temperatures are limited. It has contributed to the literature by examining these points.

1. Introduction

Organic Rankine Cycle (ORC) is a technology that enables the conversion of energy from any thermal source into electrical energy. These thermal resources are; It can consist of solar, geothermal, biomass or waste heat. The biggest difference in naming the system as "Organic" and separating it from the classical Rankine cycle is that an organic fluid is preferred instead of steam-water use in the cycle. The ideal fluid of ORC is generally zero or positive slope "Isentropic" or "Dry Fluid" and they don't require overheating. The classical Rankine cycle requires overheating, so the turbine inlet temperature is higher than the ORC system. For these reasons, the use of organic fluids not only reduces the high heat resistance requirements for the manufacture of turbine blades, but also lowers the cost.

2. Literature Survey

Studies on thermodynamic optimization of ORC are reviewed below. Behzadi et al. (2018) conducted multi-objective optimization and exergo-economic analysis on the ORC integrated power plant in Tehran. Using MATLAB, they used Genetic Algorithm (GA)-based multi-objective optimization technique.

Woodland et al. (2020) have worked on alternative ORC configurations. These are two-phase flash-expanded ORC and zeotropic fluid ORC. Net power maximization has been taken as a function of objective. They stated that the maximum net power was not reached at the point where the highest thermal efficiency was observed, therefore, net power maximization should be examined more important than thermal efficiency maximization. Xi et al. (2015) proposed graphical criterion method for simple and recuperative ORC comparison and appropriate fluid selection. They determined their objective functions as annual cost flow and exergy efficiency with GA.

Andreasen et al. (2015) studied the selection of suitable fluid with the GA method for simple and recuperative ORC designed using binary mixtures fluids. They determined the net power output as the objective function. Yang et al. (2015) conducted a thermodynamic optimization study with GA for ORC designed using R245fa fluid. Evaporation pressure, superheating temperature and condensation temperature were chosen as design parameters.

Xi et al. (2014) used the GA method to select the appropriate fluid in ORC which was designed using zeotropic fluids. They determined the annual cash flow as the objective function. Larsen, Sigthorsson, and Haglind (2014) conducted studies on system optimization with the GA method for ORC designed using R245ca fluid. They determined the net power output as the objective function. Imran et al. (2014) conducted an optimization study by aiming thermal efficiency maximization and unit investment cost minimization with NSGA-II method. Evaporation pressure, superheating temperature and $\Delta T_{PP,e}$ - $\Delta T_{PP,c}$ values were chosen as design parameters.

In this section, the applications of ORC (geothermal, solar, waste heat) found in the literature under different heat sources are evaluated. Coskun et al. (2012) conducted an energy and exergy analysis study for geothermal heat source multigeneration systems. Performance parameters were determined as energy and exergy regeneration rate and re-injection rate. They achieved the highest energy efficiency in the combined system of electricity generation + greenhouse heating. Coskun and Al-Talabani (2017) conducted a thermodynamic analysis study on Aliağa Gas Turbines and Combined Cycle Power Plant with EES software. They determined that the most exergy destruction was in the combustion chamber, heat boiler and condenser units, respectively. They determined that the energy and exergy efficiency values of the power plant with an installed capacity of 180 MW are 32.8% and 43.4%, respectively.

Baral (2019) conducted an ORC hybrid solar-geothermal study for electricity generation in Nepal. They found that there is 17.5 kW production when using R134a, but 22.5 kW production in ORC with R245fa. Altınkaynak and Çelik (2021) conducted exergy analysis studies within the scope of ORC's geothermal applications. Analyzes were made in EES software with N-pentane fluid. They reached 34% exergy efficiency at 80 °C low well temperature.

Hu et al. (2022) studied organic fluid selection within the scope of low temperature geothermal applications of ORC. Considering the net power produced per flow rate of geothermal water, they determined that the most suitable fluid is R245fa. Wang et al. (2013) performed a thermodynamic analysis of regenerative ORC within the scope of solar applications. They used flat plate solar collectors to collect solar radiation because of their low cost.

When other organic fluids were compared, they found that R245fa and R123 were the most suitable fluids for the system due to their high system performance and low operating pressure.

Boyaghchi and Chavoshi (2018) analyzed the ORC-based solar micro-coupled power generation system in terms of exergy, economy and environmental criteria. For the R1234yf and R245fa fluid groups, a significant positive effect was found between 16.71% and 24.34%, respectively, on the thermal and exergy efficiencies in November. Al Jubori et al. (2017) conducted an optimization study for a small-scale axial turbine for a low-temperature heat source such as solar applications of ORC. They achieved the best performance values (10.5% thermal efficiency, 73.3% exergy efficiency, 6.3 kW maximum power) with R123 fluid.

Shu et al. (2014) investigated the performance of alkanes within the scope of waste heat applications of ORC. They obtained the highest power output with cyclohexane fluid. Considering the criteria such as low irreversibility and high power, they determined that the most suitable fluids are cyclohexane and cyclopentane. Kölsch and Radulovic (2015) studied the use of methanol, toluene and solkatherm SES 36 organic fluids in the ORC of diesel engine waste heat. The best thermal performance was achieved with methanol and the highest power output with toluene. Considering the heat transfer area, they suggested the use of methanol fluid. On the other hand, Khatita et al. (2015) studied power generation using waste heat recovery with ORC in the oil and gas sector in Egypt. They used the Aspen HYSYS v7.1 simulation model. Considering the thermodynamic and economic criteria together, they determined that the most suitable fluid was benzene.

In the experimental studies on the Organic Rankine Cycle, R134a or R245fa was generally used as the organic fluid. However, in recent studies, it has been seen that it is used in experimental studies in different organic fluids. Eyerer et al. (2019) compared the experimental performance of R1224yd(Z) and R1233zd(E) with R245fa. They found that a higher thermal performance was achieved with R1233zd(E). They stated that both fluids can replace R245fa in terms of low GWP value. Blondel et al. (2019) determined the performance of pure and zeotropic fluids in ORC by conducting an experimental study. Pure NovecTM649 and 80% NovecTM649-20% HFE7000 zeotropic fluids were used. 10% higher thermal efficiency value was determined in the zeotropic mixture. They emphasized that these fluids are potential candidates to replace traditionally used fluids such as R134a and R245fa due to their zero ODP and low GWP values.

Within the scope of the literature research, air-cooled and water-cooled condensers are examined in this section. Walraven et al. (2015) compared the performance of air- or water-cooled condensers in geothermal heat source applications of ORC. They found that it is economically better to use mechanical draft wet cooling towers instead of air-cooled condensers. They stated that the difference in performance was seen especially for brine inlet temperature. They found that the investment cost of water-cooled condensers is also lower. Zhao et al. (2017) compared cooling methods for heat recovery in ORC. The simulation results showed that the water-cooling method is more suitable for the vehicle ORC system than the air-cooling method. They found that water-cooled ORC has greater power output and higher thermal efficiency.

In the section below, the differences of the number of objective functions in optimization with GA are examined. In some studies, the objective function was determined through a single parameter in GA optimization. The objective functions; by determining the ratio of heat transfer area to the total net power output (Bian, Wu, and Yang 2014), total exergy efficiency (Long et al. 2014), gross annual profit (Gutiérrez-Arriaga et al. 2015), total irreversibility loss (Han, Yu, and Ye 2013), thermal efficiency (Pierobon et al. 2013), second law efficiency (Agromayor and Nord 2017), net power worked with GA on both optimum fluid selection and thermodynamic optimization of the system (Andreasen et al. 2014)(Fiaschi et al. 2014)(Kai et al. 2015). The studies in which multiple parameters are determined as the objective function are also summarized below. The multiple objective functions; by determining total exergy efficiency and product cost rate (Nazari, Heidarnejad, and Porkhial 2016), net power, volumetric flow rate and turbine efficiency (Donateo and Fazio 2014), thermal efficiency, exergy efficiency, payback period and annual emission reduction (Wang et al. 2016), exergy efficiency and total cost rate of the system (Khaljani, Khoshbakhti Saray, and Bahlouli 2015), thermal efficiency, exergy efficiency, total heat transfer area and total product cost (Boyaghchi, Chavoshi, and Sabeti 2015), exergy efficiency and total cost rate of the system (Javan et al. 2016), net power and investment cost worked with GA on both optimum fluid selection and multi-objective optimization of the system (Huster, Schweidtmann, and Mitsos 2020).

In this study,

• ORC optimization and optimum fluid selection were made using NSGA-II. Objective functions are specified by evaluating four different factors: Energy (thermal efficiency, turbine power), Exergy (exergy efficiency, total irreversibility), Economic (turbine performance-Volume Flow Ratio) and Environment (Thermodynamic Sustainability Indices).

• In Orc systems, the thermal efficiency and turbine power of fluids may be differences in maximization points. Therefore, these two parameters are discussed separately.
• No study has been found in which thermodynamic sustainability indices (WER, ESI and EEF) are considered as an objective function in NSGA-II. In this study, by determining EEF minimization as the objective function, environmental performance also played a role in determining the optimum fluid.
• The performance of fluids in different categories is compared. In the design, dry (R601 and R601a), isentropic (R141b and R123), wet (R152a and R134a) and new-generation (R1234ze and R1234yf) are used as organic fluids.
• Fluids performed differently under each objective function. Therefore, the weight function was created by evaluating the performance increase of the fluids under the objective functions and then optimum fluids are determined for different heat source temperatures.

3. Material and Method

3.1. Thermodynamic Analysis

Table 1 summarizes the thermophysical and safety-environmental properties of fluids (Calm and Hourahan 2007). Engineering Equation Solver (EES) software has been used for thermodynamic analysis and modeling of ORC. The equations used are given in Table 2

Table 1. Thermophysical and safety-environmental properties of fluids

Fluids/ Properties	R601	R601a	R141b	R123	R152a	R134a	R1234yf	R1234ze
Type	Dry		Isentropic		Wet		New-Generations	
Molecular mass (g/mol)	72.15	72.15	116.95	152.93	66.05	102	114.04	114.04
Normal Boiling Points (°C)	36.1	27,8	32	27,8	-24	-26.1	-29.3	-18.8
Critical Temperature (°C)	196.6	187.2	204.4	183.7	113.3	101.1	94.85	109.52
Critical Pressure (MPa)	3.37	3.38	4.21	3.66	4.52	4.06	3.38	3.63
ASHRAE 34 safety group	A3	A3	n.a	B1	A2	A1	*A2L	*A2L
ODP	0	0	0.12	0	0	0	0	0
GWP	20	20	725	77	124	1430	4	6

*A2L; low toxicity and mildly flammable

Table 2. ORC Thermodynamic Analysis Equations

Components	Energy Analysis	Exergy Analysis
Pump	Pump Work (kJ/kg) $w_p = (h_2 - h_1) = (h_{2s} - h_1)/\eta_p$	Pump Irreversibility (kJ/kg) $i_p = T_0(s_2 - s_1)$
Evaporator	Evaporator Duty (kJ/kg) $q_e = (h_3 - h_2)$	Evaporator Irreversibility (kJ/kg) $i_e = T_0[(s_3 - s_2) - (h_3 - h_2)/T_h]$
Turbine	Turbine Work (kJ/kg) $w_t = (h_3 - h_4) = (h_3 - h_{4s})\eta_t$	Turbine Irreversibility (kJ/kg) $i_t = T_0(s_4 - s_3)$
Condenser	Condenser Duty (kJ/kg) $q_c = (h_4 - h_1)$	Condenser Irreversibility (kJ/kg) $i_c = T_0[(s_1 - s_4) + (h_4 - h_1)/T_c]$
System	Net Work (kJ/kg) $w_{net} = w_t - w_p$	Total Irreversibility (kJ/kg) $i_{total} = i_p + i_e + i_t + i_c$
	Thermal Efficiency $\eta_{th} = w_{net}/q_e$	Exergy Expended (kJ/kg) $e_{expended} = [1 - T_0/T_h]q_e + w_p$
		Exergy Efficiency $\eta_{II} = 1 - i_{total}/e_{expended}$

η_t and η_p; isentropic efficiencies of turbine and pump
$T_{h,i}$ and $T_{h,o}$ heat source input-output; $T_{c,i}$ and $T_{c,o}$ are the cooling water inlet-outlet temperatures (Eq.1-2).

$$T_h = (T_{h,i} - T_{h,o})/Ln\,(T_{h,i} - T_{h,o}) \tag{1}$$

$$T_c = (T_{c,i} - T_{c,o})/Ln\,(T_{c,i} - T_{c,o}) \tag{2}$$

Evaporator and condenser pinch point temperature difference ($\Delta T_{PP,e}$ and $\Delta T_{PP,c}$) can be seen from the operating principle and T-s diagram of ORC given in Figure 1.

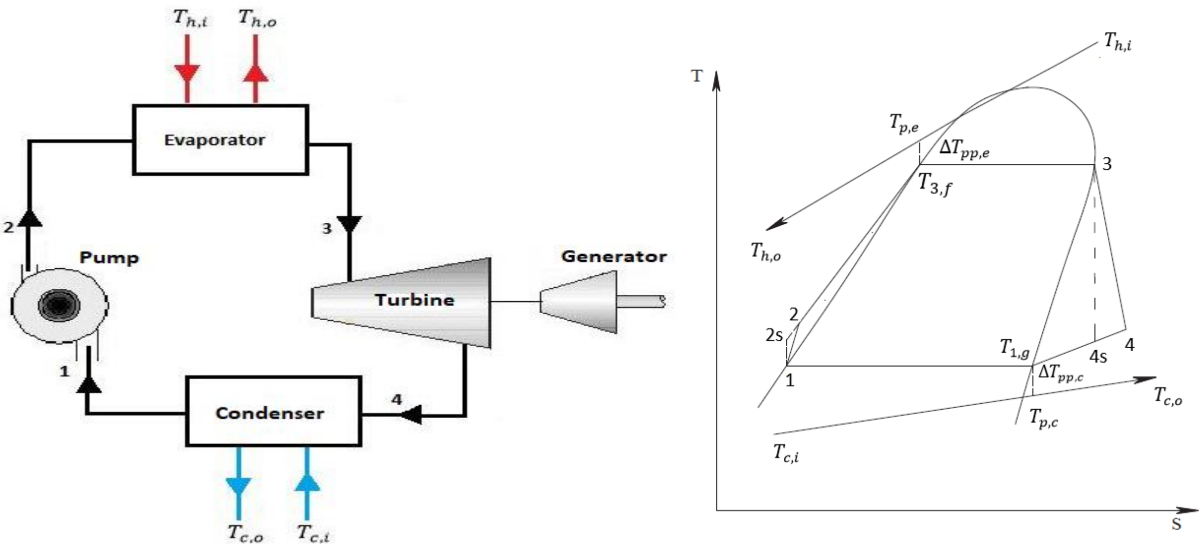

Figure 1. ORC Working Principle and Demonstration of ΔTPP,e and ΔTPP,c in T-s diagram

The evaporator and condenser energy balance relations (Eq.3-8) are given below.

- Evaporator energy balance

$$\dot{m}_{ORC} * (h_3 - h_2) = \dot{m}_h * Cp * (T_{h,i} - T_{h,o}) \tag{3}$$
$$\dot{m}_{ORC} * \left(h_3 - h_{3,f}\right) = \dot{m}_h * Cp * (T_{h,i} - T_{p,e}) \tag{4}$$
$$\Delta Tpp_{,e} = (T_{p,e} - T_{3,f}) \tag{5}$$

- Condenser energy balance

$$\dot{m}_{ORC} * (h_{4a} - h_1) = \dot{m}_c * Cp * (T_{c,o} - T_{c,i}) \tag{6}$$
$$\dot{m}_{ORC} * \left(h_{1,g} - h_1\right) = \dot{m}_c * Cp * (T_{p,c} - T_{c,i}) \tag{7}$$

$$\Delta Tpp_{,c} = (T_{1,g} - T_{p,c}) \tag{8}$$

($T_{p,e}$: evaporator pinch point temperature; $T_{p,c}$: condenser pinch point temperature)

For the thermodynamic analysis of ORC; all processes are under steady state. Pressure losses in the evaporator and condenser are neglected, all equipment is considered adiabatic. Isentropic efficiency of the turbine and the pump are 75%.

3.2. Thermodynamic Optimization with NSGA-II

NSGA-II optimization technique is used for optimum fluid selection under different heat source temperatures. Tournament selection method was used for the optimization of the simple ORC with the genetic algorithm. The NSGA-II parameter for optimization are shown in Table 3. Flow diagram of GA's working principle is shown in Figure 2.

Table 3. The NSGA-II parameter for optimization

NSGA-II Parameters	Values
Population size	64
Maximum generations	256
Crossover probability	0,7
Mutation probability	0,175
Selection process	Tournament

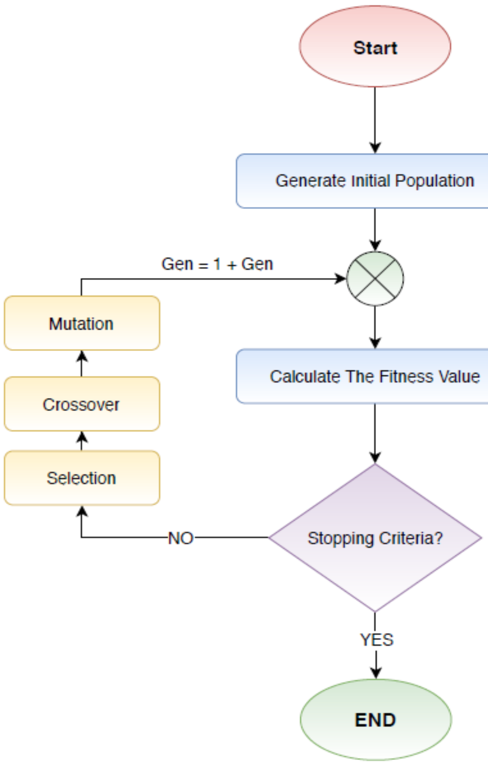

Figure 2. Flow chart of the genetic algorithms

Thermodynamic optimization is performed using NSGA-II. The lowest turbine power in the system is 1 kW; the highest turbine power has been set as 10 kW. Heat source temperatures: 90, 100 and 110 °C. Heat source mass flow rate is 0.27 kg/s. The primary working conditions are selected as decision variables which include evaporating pressure (P_{eva}), $\Delta T_{PP,e}$, $\Delta T_{PP,c}$ and superheating temperature (T_{sup}). Since organic fluids in different fluid categories are used in the design, the limit values for evaporation pressure have been determined at different ranges. In this way, better results were obtained in optimization. Table 4 summarizes the logical bounds for four decision variables.

Table 4. Logical bounds for four decision variables

Organic Fluids	Evaporating Pressure (P_{eva}) (kPa)	$\Delta T_{PP,e}$ (°C)	$\Delta T_{PP,c}$ (°C)	T_{sup} (°C)
R601	260< P_{eva} <310	1<$\Delta T_{PP,e}$<10	1<$\Delta T_{PP,c}$<10	0<T_{sup}<20
R601a	330< P_{eva} <385			
R141b	300< P_{eva} <355			
R123	350< P_{eva} <410			
R152a	1840< P_{eva} <2160			
R134a	2100< P_{eva} <2670			
R1234yf	2000< P_{eva} <3200			
R1234ze	1600< P_{eva} <1900			

Based on the energy balance and the definition of evaporator and condenser pinch point temperature difference, other following constraints are considered in the optimization. Thermodynamic optimization was applied separately for 3 different heat source temperatures. Therefore, the limitations that should be related to the heat source temperature are also specified.

- $1\ kW < W_T < 10\ kW$
- $T_{eva} + \Delta T_{PP,e} < T_{h,i}$
- $T_{eva} + \Delta T_{PP,e} < T_{critical}$
- $T_{eva} + T_{sup} < T_{h,i}$
- $T_{c,i} + \Delta T_{PP,c} < T_{con}$

With the NSGA-II, ORC system has been evaluated by considering 4 different factors as energy, exergy, economic (turbine performance) and environment. 6 different objective functions have been determined. The G(x) weight function has been determined by evaluating the percentage increase in performance achieved by the fluid reaching the desired objective function.

Energy:
- f1(x): max ($\eta_{ısıl}$); Thermal efficiency maximization
- f2(x): max (W_T); Turbine power maximization

Exergy:
- f3(x): max (η_{II}); Exergy efficiency maximization
- f4(x): min (I_T); Total irreversibility minimization

Economic (Turbine performance):
- f5(x): min (VFR); Volume Flow Ratio minimization

Environmental:
- f6(x): min (EEF); Environmental Effect Factor minimization

where x = {P_{eva}, $\Delta T_{PP,e}$, $\Delta T_{PP,c}$, T_{sup}} subjected to lower bound < x < upper bound.

In addition to the VFR, turbine Size Parameter (SP) and turbine Pressure Ratio (PR) were also examined within the scope of turbine performance. The correlations related to these values are given in equation 9-13. ORC systems with low VFR can reach high turbine efficiency values. In addition, high SP values require a high turbine size.

$$\dot{m}_{ORC} = \rho_3 \dot{V}_3 \tag{9}$$
$$\dot{m}_{ORC} = \rho_4 \dot{V}_4 \tag{10}$$
$$VFR = \dot{V}_4 / \dot{V}_3 \tag{11}$$
$$SP = \frac{\sqrt{\dot{V}_4}}{[(h_3 - h_{4S})\eta_{+}]^{\frac{1}{4}}} \tag{12}$$
$$PR = P_3 / P_4 \tag{13}$$

In addition to EEF minimization, Waste Exergy Ratio (WER) and Exergy Sustainability Index (ESI) were also examined within the scope of thermodynamic sustainability indices. The relations related to these values are given in equation 14-16.

$$WER = I_{total} / E_{Expended} \tag{14}$$
$$EEF = WER / \eta_{II} \tag{15}$$
$$ESI = 1 / EEF \tag{16}$$

4. Model Validation

The prepared model was compared with the data of two different studies using different pinch point temperatures. When the net power values obtained are examined, it is seen that the model is usable (Table 5).

Table 5. Model Validation (NSGA-II)

Design Parameters	Heat Source Temperature: 150 °C; Heat Sink Temperature: 20 °C; $\Delta T_{PP,e}+\Delta T_{PP,c}$ = 20 °C Turbine and pump isentropic efficiency: 85% and 80%				Evaporation Temperature: 80 °C $\Delta T_{PP,e}$ = 8 °C Turbine and pump isentropic efficiency: 80% and 70%	
Organic Fluids	R113		R11		R245fa	
Performance Parameters	Present Study	Literature (Jiansheng et al. 2017)	Present Study	Literature (Jiansheng et al. 2017)	Present Study	Literature (Jankowski et al. 2019)
Net Power (kW)	73.12	73.91	70.24	70.93	50.2	51.0

5. Result and Discussion

In this study, optimum fluid was determined by using NSGA-II for 90, 100 and 110 °C heat source temperature. ORC performance is determined under 6 different f(x), objective functions. Weight function G (x); It is organized under the objective functions by taking into consideration the increase in the performance of the fluids.
Figure 3 shows the thermal efficiency, turbine power, exergy efficiency and total irreversibility values of 8 different fluids at the optimum design point of 90 °C heat source temperature under different objective functions.

- It is seen that the best fluid in terms of thermal efficiency maximization is R141b. However, it is seen that R1234yf fluid is better as turbine power.
- In the system with R1234yf, the thermal efficiency was not found high due to the need for heat input. In the system with R141b, 16.9% more thermal efficiency was obtained than R1234yf. However, in the system with R1234yf, 25.1% more turbine power was obtained than R141b.
- It is seen that R141b is better in exergy efficiency maximization and total irreversibility minimization values.

Figure 4 shows the VFR, SP and PR values of 8 different fluids at the optimum design point of 90 °C heat source temperature under different objective functions. It is seen that R152a has very low VFR and SP value compared to other fluids. It was stated that the lowest value in the turbine pressure ratio values is in R1234yf.
Figure 5 shows the EEF, ESI and WER values of 8 different fluids at the optimum design point of 90 °C heat source temperature under different objective functions. It is seen that the lowest environmental impact factor is obtained in fluid R141b. It was determined that the EEF value of the system with R1234yf is 9.1% higher than the system with R141b.

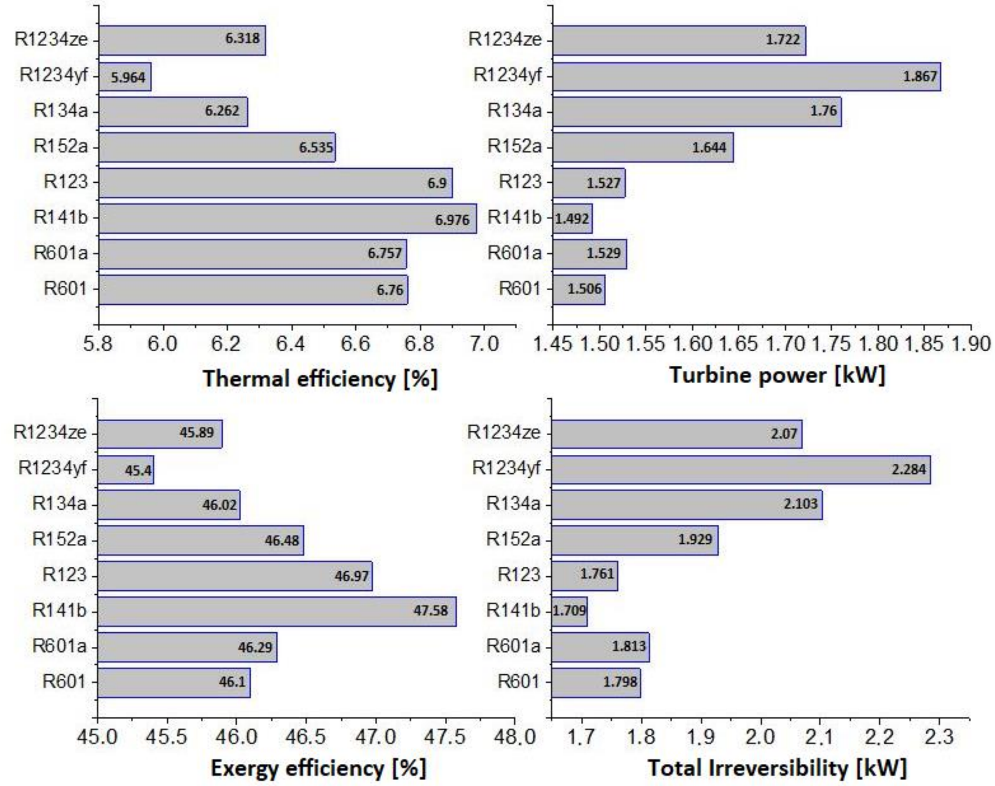

Figure 3. Determination of thermodynamic performance parameters of different fluids at 90 °C heat source temperature

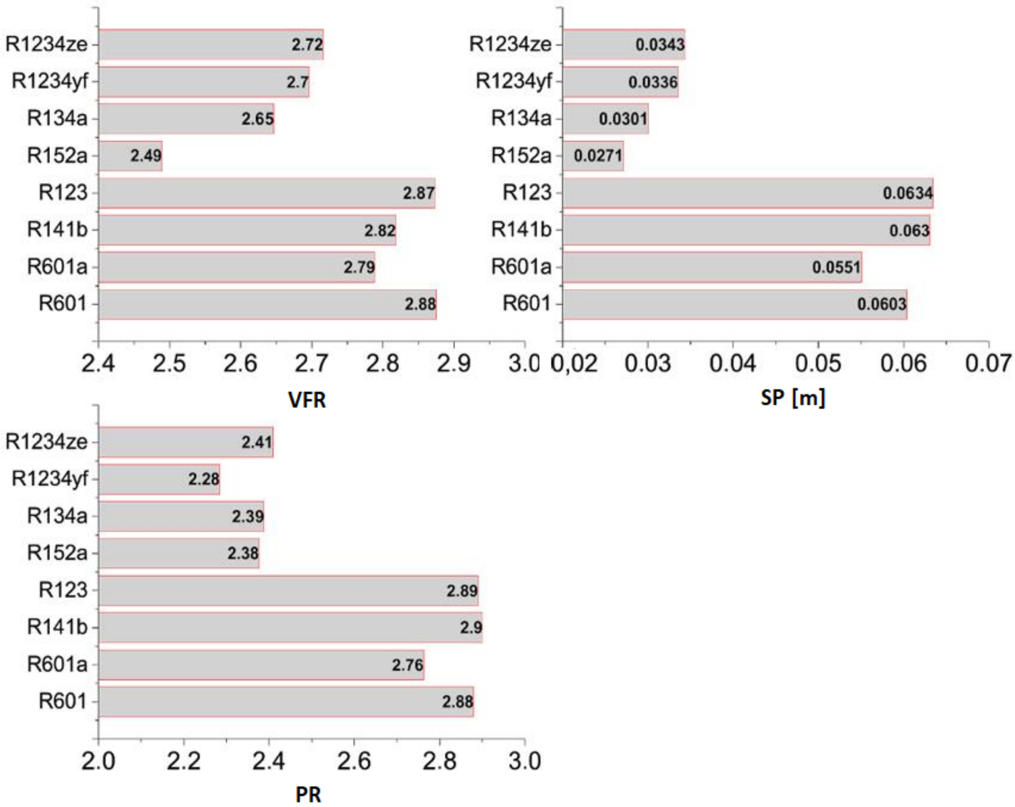

Figure 4. Determination of economic (turbine performance) parameters of different fluids at 90 °C heat source temperature

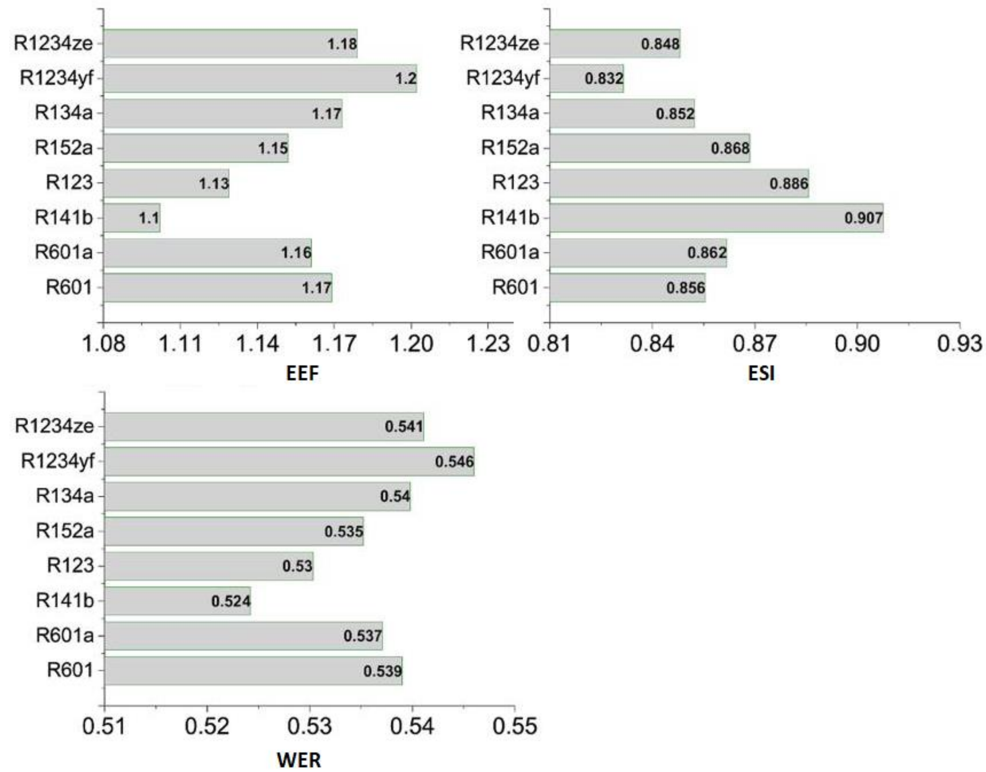

Figure 5. Determination of thermodynamic sustainability parameters of different fluids at 90 °C heat source temperature

The best performing fluids for the objective functions determined under 90 °C heat source temperature is summarized in Table 6. The average performance increase percentage of the fluid that is optimally determined under the objective functions in comparison to other fluids is specified in the table. The R141b fluid performed 7.58% better than other fluids under the thermal efficiency maximization objective function. Other objective functions were also evaluated in the same way. As a result of the weight function determined according to the performance increase percentages, R141b can be used in proportion as 51% optimally for 90° C heat source temperature (Decision points for R141b; P_{eva}: 325.4 kPa, $\Delta T_{PP,e}$: 4.94 °C, $\Delta T_{PP,c}$: 5.07 °C, T_{sup}: 0.8968 °C).

Table 6. Determination of optimum fluid under different objective function at 90 °C heat source temperature

$T_{h,i}$ 90 °C	$f_1(x)$ max (η_{th})	$f_2(x)$ max (W_T)	$f_3(x)$ max (η_{II})	$f_4(x)$ min (I_T)	$f_5(x)$ min (VFR)	$f_6(x)$ min (EEF)	**G(x) with Availability rate**
Optimum fluid Performance increase	R141b (7.58%)	R1234yf (17.36%)	R141b (3.07%)	R141b (12.34%)	R152a (10.17%)	R141b (5.49%)	**51%, R141b 31%, R1234yf 18%, R152a**

Thermodynamic analysis, turbine performance and sustainability indexes were determined for 100 and 110 °C as well as at 90 °C. Due to the limited number of pages, only optimization results are given for 100 and 110 °C. The best performing fluids for the objective functions determined under 100 °C heat source temperature is summarized in Table 7. The average performance increase percentage of the fluid that is optimally determined under the objective functions in comparison to other fluids is specified in the table. As a result of the weight function determined according to the performance increase percentages, 65% R1234yf was determined as the optimum fluid for 100 °C heat source temperature (Decision points for R1234yf; P_{eva}: 1795 kPa, $\Delta T_{PP,e}$: 6.67 $_o$C, $\Delta T_{PP,c}$: 5.21 $_o$C, T_{sup}: 9.33 $_o$C).

Table 7. Determination of optimum fluid under different objective function at 100 °C heat source temperature

$T_{h,i}$ 100 °C	$f_1(x)$ max (η_{isII})	$f_2(x)$ max (W_T)	$f_3(x)$ max (η_{II})	$f_4(x)$ min (I_T)	$f_5(x)$ min (VFR)	$f_6(x)$ min (EEF)	**G(x) with Availability rate**
Optimum fluid Performance increase	R141b (2.45%)	R1234yf (18.23%)	R1234yf (6.05%)	R141b (5.12%)	R152a (11.04%)	R1234yf (9.68%)	**65%, R1234yf 21%, R152a 14%, R141b**

The best performing fluids for the objective functions determined under 110 °C heat source temperature is summarized in Table 8. The average performance increase percentage of the fluid that is optimally determined under the objective functions in comparison to other fluids is specified in the table. As a result of the weight function determined according to the performance increase percentages, 83% R1234yf was determined as the optimum fluid for 110 °C heat source temperature (Decision points for R1234yf; P_{eva}: 2477 kPa, $\Delta T_{PP,e}$: 4.5 °C, $\Delta T_{PP,e}$: 5.29 °C, T_{sup}: 12.83 °C).

Table 8. Determination of optimum fluid under different objective function at 110 °C heat source temperature

$T_{h,i}$ 110 °C	$f_1(x)$ max (η_{th})	$f_2(x)$ max (W_T)	$f_3(x)$ max (η_{II})	$f_4(x)$ min (I_T)	$f_5(x)$ min (VFR)	$f_6(x)$ min (EEF)	**G(x) with Availability rate**
Optimum fluid Performance increase	R1234yf (2.97%)	R1234yf (56.36%)	R1234yf (19.23%)	R141b (6.23%)	R152a (14.93%)	R1234yf (28.33%)	**83%, R1234yf 12%, R152a 5%, R141b**

6. Conclusions

In this study thermodynamic optimization study was carried out using NSGA-II for optimum fluid selection. Fluids in different categories (dry, isentropic, wet, new generations) were used in the design. The optimum fluid was determined by evaluating 6 different objective functions at 90, 100 and 110 °C heat source temperatures. These are, f1(x): max ($\eta_{ısıl}$); f2(x): max (W_T); f3(x): max (η_{II}); f4(x): min (I_T); f5(x): min (VFR); f6(x): min (EEF).

It has been found that determining the optimum fluid with a single objective function may be erroneous. It has been determined that turbine power and thermal efficiency performance are not the same in some fluids due to evaporator load, condenser heat load and mass flow rate requirement. Therefore, the weight function was created by considering the performance increase it showed under 6 different objective functions. Accordingly, the percentage of using the optimum fluid has been determined.

At a heat source temperature of 90 °C, R141b performed better than the others under three different objective functions. Better results were obtained by 7.58% in thermal efficiency, 12.34% in total irreversibility, and 5.49% in EEF. As a result of the weight function, it has been determined that R141b can be used as an optimum fluid at the rate of 51%.

At a heat source temperature of 100 °C, R1234yf performed better than the others under three different objective functions. Better results were obtained by 18.23% in turbine power, 6.05% in exergy efficiency, and 9.68% in EEF. As a result of the weight function, it has been determined that R1234yf can be used as an optimum fluid at the rate of 65%. With the increase in heat source temperature from 90 °C to 100 °C, the percentage of R141b's optimum fluid availability decreased from 51% to 14%.

At a heat source temperature of 110 °C, R1234yf performed better than the others under four different objective functions. Better results were obtained by 2.97% in thermal efficiency, 56.36% in turbine power, 19.23% in exergy efficiency, and 28.33% in EEF. As a result of the weight function, it has been determined that R1234yf can be used as an optimum fluid at the rate of 83%. With the increase in heat source temperature from 100 °C to 110 °C, the percentage of R1234yf's optimum fluid availability increased from 65% to 83%. With the increase in heat source temperature from 90 °C to 110 °C, the percentage of R141b's optimum fluid availability decreased from 51% to 5%.

This study shows that in ORC systems, the optimum fluid heat source varies depending on the temperature and your purpose function. It is not possible to define a 100% ideal fluid for any heat source temperature. However, its optimized use depending on your purpose function will play an important role in increasing system performance.

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Conflict of Interest

No conflict of interest was declared by the authors.

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