Determination of Optimum Pinch Point Temperature Difference Depending on Heat Source Temperature and Organic Fluid with Genetic Algorithm

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Abstract

In this study, the effect of evaporator pinch point temperature difference ($\Delta T_{PP,e}$) value in Organic Rankine Cycle (ORC) on system performance was determined. Under different applications of ORC, optimum $\Delta T_{PP,e}$ value has been determined in ORC systems designed with different heat source temperatures. By changing the $\Delta T_{PP,e}$ value, the heat input provided to the system, the mass flow of organic fluid, the evaporation pressure and the enthalpy drop in the turbine are affected. In thermodynamic optimization, the objective function is determined as turbine power maximization. Genetic algorithm optimization technique is used. Within the scope of low and high temperature ORC applications, the optimum $\Delta T_{PP,e}$ value of different organic fluids under 10 different heat source temperatures (Low, 90-130 °C; High, 250-290 °C) has been determined. Low temperature organic fluids have been selected from dry, isentropic, wet and new-generation categories. High temperature organic fluids have been selected from the alkane, aromatic hydrocarbon, and siloxane categories. The effect of $\Delta T_{PP,e}$ on fluids of different categories was determined for low and high temperature ORCs. It has been determined that taking the $\Delta T_{PP,e}$ value constant regardless of the heat source temperature and organic fluid causes performance loss in ORC.

Keywords: Genetic Algorithm, Low-High Organic Fluids Optimum, Pinch Point, Organic Rankine Cycle, Thermodynamic Optimization

1. INTRODUCTION

The Organic Rankine Cycle (ORC) works like the Rankine cycle as its working principle, the difference is that an organic fluid other than water is used. The fluid used in ORC has a lower boiling point and a higher vapor pressure than water and can therefore be used in low temperature heat sources to generate electricity. The organic fluid is selected to best match the heat source according to its different thermodynamic properties, resulting in higher efficiency of both the process and the expander.

In this study, the performance of organic fluids was determined depending on the heat source temperature under low and high temperature applications of ORC. The optimum evaporator pinch point temperature difference ($\Delta T_{PP,e}$) was determined for each heat source temperature. $\Delta T_{PP,e}$; It is defined as the difference between the evaporator pinch point temperature ($T_{P,e}$) and the evaporation temperature of the organic fluid. It has been observed that this value ($\Delta T_{PP,e}$), which was taken as a constant in most of

the previous studies, seriously affects the ORC performance. Important studies on this subject are summarized.

Wu et al. [1] conducted a study on the determination of $\Delta T_{PP,e}$ and $\Delta T_{PP,c}$ in ORC designed using mixing fluids. They considered exergo - economic performance, which is the ratio of annual total cost to net power, as an evaluation criterion. They stated that the increase of $\Delta T_{PP,e}$ rapidly increased exergo economic performance, but reached the best performance at optimum $\Delta T_{PP,e}$ value. They concluded that the optimum $\Delta T_{PP,e}$ for mixing fluids should be between 3-6 °C.

Yu et al. [2] developed a method that can instantly determine the organic fluid and working conditions in ORC depending on the $\Delta T_{PP,e}$. They defined the $\Delta T_{PP,e}$ formed in the preheater and the $\Delta T_{PP,e}$ formed in the evaporator for this aim. They determined that the maximum power is reached when there is a suitable difference between the heat source inlet temperature and the critical temperature of the fluid, and the fluid evaporates near the critical region.

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Liu et al. [3] performed a performance analysis for geothermal different heat source temperatures in the ORC system they designed using R245fa. The effect of $\Delta T_{PP,e}$ on system performance has been determined. Net power, turbine size parameter, volume flow rate and total thermal conductivity were calculated. It has been determined that $\Delta T_{PP,e}$ is inversely proportional to total thermal conductivity and net power. It has been stated that the optimum $\Delta T_{PP,e}$ is associated with the heat source inlet temperature, and low $\Delta T_{PP,e}$ provides high net power. As a result of the change of heat source inlet temperature between 80-180 °C, it has been determined that $\Delta T_{PP,e}$ increased from 2 °C to 21 °C.

Kaşka et al. [4] conducted a study on the energy and exergy analysis of the Organic Rankine-Brayton combined cycle. They found that it is important to determine the optimum $\Delta T_{PP,e}$ temperature in heat exchangers where heat source and work fluid heat transfer occurs in ORC design. They stated that while the heat transfer to the evaporator increases linearly with the increase of the $\Delta T_{PP,e}$ value, the thermal efficiency of the ORC decreases, but depending on the $\Delta T_{PP,e}$ value, the net power produced by the ORC is the optimum point.

Sun et al. [5] examined the effect of $\Delta T_{PP,e}$ on thermodynamic performance within the scope of geothermal ORC applications. They stated that $\Delta T_{PP,e}$ is an important parameter for thermodynamic and economic performance. They have determined that low $\Delta T_{PP,e}$ will provide more turbine net power but have a negative effect on the economy as it will increase the heat transfer area. For heat source applications higher than 130 °C, it has been determined that ORC produces 1.7-2.6% more power with every 1 °C decrease in $\Delta T_{PP,e}$.

Bademlioğlu et al. [6] studied the effect of $\Delta T_{PP,e}$ on exergy performance in ORC. The effect of changing $\Delta T_{PP,e}$ between 5-20 °C on systems prepared using different organic fluids has been determined. They stated that depending on the $\Delta T_{PP,e}$ and the organic fluid, the irreversibility in the evaporator can be reduced by 62.32%.

Wang et al. [7] have worked on $\Delta T_{PP,e}$ optimization using the Analytical Hierarchy Process (AHP) - Entropy method in ORC systems. As a result of the study, they stated that they reached the maximum power output with R141b and the maximum thermal efficiency and exergy efficiency values with R11.

Sarkar [8] worked on $\Delta T_{PP,e}$ design and optimization for maximum heat recovery in ORC. He developed a method that can determine $\Delta T_{PP,e}$ and $\Delta T_{PP,c}$ instantaneously. Best results have been achieved in ammonia fluid in terms of low mass flow requirement, high exergy efficiency and low turbine size at optimum points. In terms of high-power output and heat recovery efficiency, it performed better in isopentane fluid.

Jankowski et al. [9] determined the optimum $\Delta T_{PP,e}$ value in ORC systems using the multi-objective approach technique. They worked on two objective functions: economy and environment. At the end of their studies, they reached the optimum $\Delta T_{PP,e}$ between 7-10 °C by using R245fa fluid.

Imran et al. [10] 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 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; Bian et al. [11] determined the heat transfer area as the ratio of the total net power output, and Long et al. [12] decided the total exergy efficiency. Gutierrez et al. [13] accepted gross annual profit as an objective function, Han et al. [14] as a total irreversibility loss, Pierobon et al. [15] as a thermal efficiency, Agromayor et al. [16] as a second law efficiency. Finally, Andreasen et al. [17], Fiaschi et al. [18] and Kai et al. [19] used the net power as the objective function and studied both the optimum fluid selection and the thermodynamic optimization of the system with GA.

In this study, thermodynamic optimization has been made in order to find the optimum $\Delta T_{PP,e}$ point for ORC designed using different fluids. As can be seen from the literature studies, it is stated that the maximum turbine power is not obtained due to the absorption of heat in the evaporator at the point where the thermal efficiency reaches its maximum. It has been determined that while the thermal efficiency decreases with the increase of $\Delta T_{PP,e}$ value, the turbine power is not in the same trend. It was observed that the turbine power of the system started to decrease after a certain $\Delta T_{PP,e}$ value. With the change of $\Delta T_{PP,e}$, the heat input required to be provided to the system increased, however, the mass flow rate of the organic fluid increased. But at the same time, with the change of $\Delta T_{PP,e}$, the evaporation pressure decreased and the enthalpy difference in the turbine decreased. It has been determined that the turbine power of the system starts to decrease at the point where the decrease in the enthalpy difference is more than the increase in ORC mass flow rate.

Therefore, it was observed that the optimum $\Delta T_{PP,e}$ point depends on the organic fluid and the heat source temperature. In the studies, it was determined that taking a constant $\Delta T_{PP,e}$ value caused a certain amount of error in the analysis results. In this study, the optimum $\Delta T_{PP,e}$ point of organic fluids in different categories at different heat source temperatures under various ORC applications was determined. These applications; geothermal, low temperature solar, waste heat and biomass-high temperature solar. Organic fluids have been selected for low temperature ORC from dry, isentropic, wet and new-generation organic fluids. In high temperature ORC, fluids have been chosen from among alkanes, aromatic hydrocarbons and siloxanes.

For low temperature ORC;
- Geothermal Energy Applications ($T_{h,i}$ = 90, 100, 110 °C)
- Low Temperature Solar Energy Applications ($T_{h,i}$ = 120, 130 °C)

For high temperature ORC;
- Waste Heat Applications ($T_{h,i}$ = 250, 260, 270 °C)

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- Biomass and High Temperature Solar Energy Applications ($T_{h,i}$ = 280, 290 °C)

Li [20], in his review study, examined the organic fluid performance under different application areas (geothermal, low temperature solar, waste heat and biomass-high temperature solar) of ORC according to the heat source temperatures.

By using the temperature values determined for these applications, the effect of optimum pinch point temperature on turbine power maximization on different fluids has been determined.

In previous studies, it was observed that the $\Delta T_{PP,e}$ value was taken as constant. However, the optimum $\Delta T_{PP,e}$ value changes depending on the heat source temperature and the organic fluid. Based on these two factors, it is aimed to make an optimization study by determining the turbine power maximization purpose under the optimum $\Delta T_{PP,e}$. By determining the optimum $\Delta T_{PP,e}$ points for different applications of ORC, it is aimed to reach higher system performances in thermodynamic analysis, modeling and optimization studies conducted by the researchers.

2. MATERIALS AND METHODS

2.1. Thermodynamic Analysis

Engineering Equation Solver (EES) was used for thermodynamic analysis and optimization of ORC. Energy and mass equations for $\Delta T_{PP,e}$ is introduced to EES, boundary conditions are entered for optimization using EES and genetic algorithm interface.

Table 1 and Table 2 summarizes the thermophysical and safety-environmental properties of fluids for low and high temperature ORC fluids respectively. The thermophysical properties of the fluid are taken from the "ASHRAE Standard 34" table. [21].

General definitions and equations (1-4) for the system are given below.

Mass balance (Total Mass Input = Total Mass Output);

$$\sum \dot{m}_{input} = \sum \dot{m}_{out} \tag{1}$$

Energy balance (Total Energy Input = Total Energy Output);

$$\sum E_{input} = \sum E_{out} \tag{2}$$

$$\dot{Q} - W = \dot{m} * (h_{in} - h_{out}) \tag{3}$$

Exergy balance (Total Exergy input = Final Exergy + Exergy Consumption + Exergy Destruction);

$$\dot{E}x_{in} = \dot{E}x_f + \dot{E}x_c + \dot{E}x_d \tag{4}$$

Table 1. Thermophysical and safety-environmental properties of fluids for low-temperature ORC.

Fluids	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 (ºC)	36.1	27,8	32	27,8	-24	-26.1	-29.3	-18.8
Critical Temperature (ºC)	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

Table 2. Thermophysical and safety-environmental properties of fluids for high-temperature ORC.

Fluids	n-octane	cyclohexane	benzene	toluene	MM	D4
Type	Alkanes		Aromatic Hydrocarbons		Siloxanes	
Molecular mass (g/mol)	114.23	84.161	78.108	92.138	162.4	296.6
Normal Boiling Points (ºC)	125	80	80	110	100.4	175
Critical Temperature (ºC)	296	280	289	319	245	312
Critical Pressure (MPa)	2.49	4.075	4.89	4.12	1.91	1.33
ASHRAE 34 safety group	n.a	A3	B2	A3	n.a	n.a
ODP	n.a	0	0	0	n.a	n.a
GWP	n.a	low	low	2.7	n.a	n.a

*A2L; low toxicity and mildly flammable

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In the energy analysis of the components in the system, the equations used for pump work (5), evaporator heat input (6), turbine work (7), the amount of heat discharged from the condenser (8) are given below (Isentropic efficiencies of turbine and pump, η_t and η_p, respectively).

$$W_p = (h_2 - h_1) = (h_{2s} - h_1)/\eta_p \tag{5}$$

$$Q_e = (h_3 - h_2) \tag{6}$$

$$W_t = (h_3 - h_4) = (h_3 - h_{4s})\eta_t \tag{7}$$

$$Q_c = (h_4 - h_1) \tag{8}$$

The equations used for net work (9) and thermal efficiency (10) in the system are given below.

$$W_{net} = W_t - W_p \tag{9}$$

$$\eta_{th} = W_{net}/Q_e \tag{10}$$

The irreversibility equations used for the pump (11), evaporator (12), turbine (13) and condenser (14) in the exergy analysis of the components in the system are given below. The average temperatures of the heat source and cooling water are given in Equation 15-16.

$$i_p = T_0(s_2 - s_1) \tag{11}$$

$$i_e = T_0[(s_3 - s_2) - (h_3 - h_2)/T_h] \tag{12}$$

$$i_t = T_0(s_4 - s_3) \tag{13}$$

$$i_c = T_0[(s_1 - s_4) + (h_4 - h_1)/T_c] \tag{14}$$

$$T_h = (T_{h,i} - T_{h,o})/Ln\ (T_{h,i} - T_{h,o}) \tag{15}$$

$$T_c = (T_{c,i} - T_{c,o})/Ln\ (T_{c,i} - T_{c,o}) \tag{16}$$

The equations used for total irreversibility (17), consumed exergy (18) and exergy efficiency (19) in the system are given below.

$$i_{Total} = i_p + i_e + i_t + i_c \tag{17}$$

$$e_{consumed} = [1 - T_0/T_h]Q_e + W_p \tag{18}$$

$$\eta_{II} = 1 - i_{Total}/e_{consumed} \tag{19}$$

The working principle of ORC and the demonstration of $\Delta T_{PP,e}$ is given in Figure 1. The evaporator and condenser energy balance relations (Eq.20-26) are given below. The explanations of the symbols in these equations are given below.

- $T_{p,e}$: Evaporator pinch point temperature
- $T_{3,f}$: Evaporation temperature
- $\Delta T_{PP,e}$: Evaporator pinch point temperature difference
- $T_{p,c}$: Condenser pinch point temperature;
- $T_{1,g}$: Condensation temperature
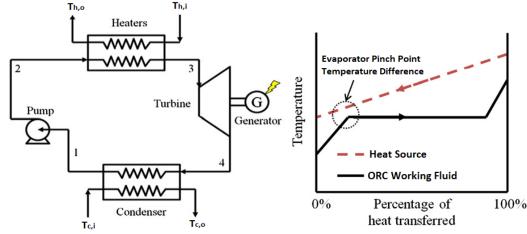- $\Delta T_{PP,c}$: Condenser pinch point temperature difference

Figure 1. ORC Working Principle and Demonstration of evaporator pinch point temperature difference ($\Delta T_{PP,e}$) [22]

Evaporator energy balance

$$\dot{m}_{ORC} * (h_3 - h_2) = \dot{m}_h * Cp * (T_{h,i} - T_{h,o}) \tag{20}$$

$$\dot{m}_{ORC} * (h_3 - h_{3,f}) = \dot{m}_h * Cp * (T_{h,i} - T_{p,e}) \tag{21}$$

$$\Delta Tpp_{,e} = (T_{p,e} - T_{3,f}) \tag{22}$$

Evaporator effectiveness (ε)

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{\dot{m}_h * Cp * (T_{h,i} - T_{h,o})}{\dot{m}_h * Cp * (T_{h,i} - T_2)} = \frac{(T_{h,i} - T_{h,o})}{(T_{h,i} - T_2)} \tag{23}$$

Condenser energy balance

$$\dot{m}_{ORC} * (h_{4a} - h_1) = \dot{m}_c * Cp * (T_{c,o} - T_{c,i}) \tag{24}$$

$$\dot{m}_{ORC} * (h_{1,g} - h_1) = \dot{m}_c * Cp * (T_{p,c} - T_{c,i}) \tag{25}$$

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$$\Delta Tpp,_c = (T_{1,g} - T_{p,c}) \qquad (26)$$

For the thermodynamic analysis of ORC, the following assumptions are employed.

- All processes are under steady state.
- Pressure losses in the evaporator and condenser are neglected. Losses in pipelines are neglected.
- In the analysis, all equipment is considered adiabatic and it is assumed that there is no heat transfer between its surfaces and the environment.
- Potential and kinetic energy changes have been neglected.
- Low-temperature ORC heat source temperatures: 90, 100, 110, 120 and 130 °C
- High-temperature ORC heat source temperatures: 250, 260, 270, 280 and 290 °C
- Heat source mass flow rate is 0.27 kg/s.
- Isentropic efficiency of the turbine and the pump are 75%.
- Evaporator effectiveness is 75%
- Cooling water inlet temperature ($T_{c,i}$) 27 °C.
- Dead point pressure and temperature, respectively, P0: 100 kPa and T0:25 °C

2.2. Thermodynamic Optimization with GA

In this study, the effect of $\Delta T_{PP,e}$ on ORC was determined by Genetic Algorithm (GA). Tournament selection method was used for the optimization of the simple ORC with the genetic algorithm. Control parameters for optimization are shown in below. Flow diagram of GA's working principle is shown in Figure 2.

Control parameters of GA for the optimization:
- Population size is 65.
- Maximum generations are 256.
- Crossover probability is 0.7.
- Mutation probability is 0.175.
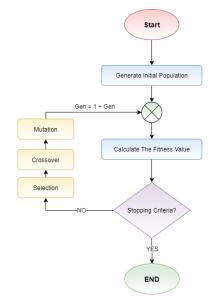- Selection process is "Tournament".

Figure 2. Flow chart of the genetic algorithms.

Thermodynamic optimization is performed using genetic algorithm. The lowest turbine power in the system is 1 kW; the highest turbine power has been set as 10 kW and 50 kW for low and high temperature ORC respectively. 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 3 summarizes the logical bounds for four decision variables for low-high temperature ORC respectively.

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 (for low-temperature ORC)
- 1 kW < W_T < 50 kW (for high-temperature ORC)
- $T_{eva} + \Delta T_{PP,e} < T_{h,i}$
- $T_{eva} + \Delta T_{PP,e} < T_{critical}$
- $T_{eva} + T_{sup} < T_{h,i}$
- $T_{c,i} + \Delta T_{PP,c} < T_{con}$
- $T_{eva,min}$: 70 °C

By changing the $\Delta T_{PP,e}$ value, the heat input provided to the system, the mass flow of organic fluid, the evaporation pressure and the enthalpy drop in the turbine are affected.

Four important parameters are affected by the change of $\Delta T_{PP,e}$ value in ORC system. These are; the heat input provided to the system, the mass flow of organic fluid, the evaporation pressure and the enthalpy drop in the turbine. It has been determined that the turbine power of the system starts to decrease at the point where the decrease in the enthalpy difference is more than the increase in ORC mass flow rate. Therefore, the objective function in this study was determined as turbine power maximization.

Objective Function;

- f(x): max (W_T); Turbine power maximization

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

3. MODEL VALIDATION

In order to determine the accuracy of the data obtained using GA, two studies investigated within the scope of literature research were used. The net power values determined by using three different organic fluids under the same design parameters were compared for two different studies in Table 4. When Table 4 is examined, it is seen that the thermodynamic model prepared can be used successfully

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Table 3. Logical bounds for four decision variables for low and high temperature ORC.

Low-temperature ORC				
Organic Fluids	Evaporating Pressure (P_{eva}) (kPa)	$\Delta T_{PP,e}$ (ºC)	$\Delta T_{PP,c}$ (ºC)	Tsup (ºC)
R601	260< P_{eva} <410	$1<\Delta T_{PP,e}<15$	$1<\Delta T_{PP,c}<10$	0<Tsup<20
R601a	330< P_{eva} <510			
R141b	300< P_{eva} <470			
R123	350< P_{eva} <550			
R152a	1840< P_{eva} <4250			
R134a	2100< P_{eva} <3900			
R1234yf	2000< P_{eva} <3300			
R1234ze	1600< P_{eva} <3410			
High-temperature ORC				
Organic Fluids	Evaporating Pressure (P_{eva}) (kPa)	$\Delta T_{PP,e}$ (ºC)	$\Delta T_{PP,c}$ (ºC)	Tsup (ºC)
n-octane	200< P_{eva} <400	$1<\Delta T_{PP,e}<40$	$1<\Delta T_{PP,c}<10$	0<Tsup<20
cyclohexane	590< P_{eva} <1150			
benzene	550< P_{eva} <1100			
toluene	270< P_{eva} <480			
D4	50< P_{eva} <130			
MM	460< P_{eva} <1270			

Table 4. Comparison of important optimization results with literature under same design parameters (GA).

Design Parameters	Heat Source Temperature: 150 ºC; Heat Sink Temperature: 20 ºC; $\Delta T_{PP,e}+\Delta T_{PP,c} = 20$ ºC Turbine and pump isentropic efficiency: 85% and 80%				Evaporation Temperature: 80 ºC $\Delta T_{PP,e} = 8$ ºC Turbine and pump isentropic efficiency: 80% and 70%	
Organic Fluids	R113		R11		R245fa	
Performance Parameters	Present Study	Literature [9]	Present Study	Literature [9]	Present Study	Literature [11]
Net Power (kW)	73.12	73.91	70.24	70.93	50.2	51.0

4. RESULT AND DISCUSSION

Figures 3 and 4 show the effect of $\Delta T_{PP,e}$ change on turbine power in geothermal and low temperature solar energy applications of ORC, respectively. When GA optimization results are evaluated for low-temperature ORC;

- It has been determined that the turbine power decreases at the point where the enthalpy difference decrease is more than the mass flow increase at the other heat source temperatures except 90 ºC. Net power increased as $\Delta T_{PP,e}$ increased, since mass flow rate increase was greater than enthalpy difference decreases at 90 ºC.
- It was observed that the allowable $\Delta T_{PP,e}$ value according to the minimum evaporator temperature under 90 ºC heat source temperature is 5 ºC maximum.
- In low-temperature applications of ORC, the highest turbine power has been reached in the system with R1234yf. Also, ORC systems with R1234ze at 90 °C and R134a at 100 and 110 °C performed better.

- While $\Delta T_{PP,e}$'s effect on turbine power tends to be similar in dry and isentropic fluids, it is very different in wet and new-generation fluids.
- It is seen that the turbine power starts to decrease after a certain $\Delta T_{PP,e}$ value at all heat source temperatures except 90 °C heat source temperature.
- Especially in low temperature solar energy applications, for wet fluid and new-generation organic fluids, the effect of $\Delta T_{PP,e}$ on turbine power is different than other fluids.
- In dry and isentropic fluids, low turbine power was obtained at low $\Delta T_{PP,e}$ values. As $\Delta T_{PP,e}$ increased, the turbine power value increased and decreased after a certain value due to the ORC mass flow rate and enthalpy drop in turbine.
- However, in wet fluid and new-generation organic fluids, a high turbine power value was achieved at the minimum $\Delta T_{PP,e}$ value allowed by the optimization limit values and it was observed that the turbine power remained at the same rate or started to decrease directly as $\Delta T_{PP,e}$ increased.
- Due to the low critical temperature of wet and new-generation fluids, the minimum $\Delta T_{PP,e}$ point increased as the heat source temperature increased.

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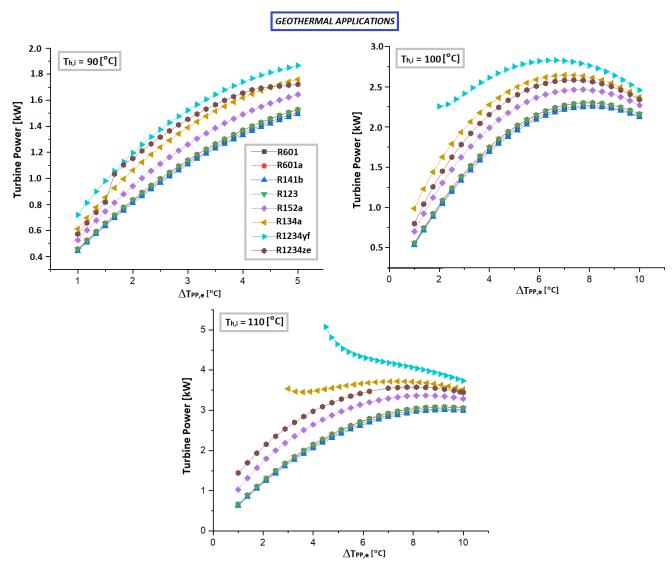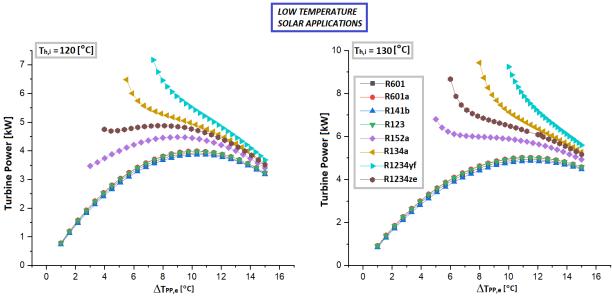
Figure 3. Effect of $\Delta T_{PP,e}$ change on turbine power for 90, 100 and 110 °C heat source temperatures in ORC's geothermal applications.

Figure 4. Effect of $\Delta T_{PP,e}$ change on turbine power for 120 and 130 °C heat source temperatures in ORC's low temperature solar applications.

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The optimum $\Delta T_{PP,e}$ points where the maximum turbine power is obtained under 5 different heat source temperatures of 8 different fluids are summarized in Table 5. It is noteworthy that the $\Delta T_{PP,e}$ value is the same in all fluids at 90 °C heat source temperature. In addition, it has been determined that dry and isentropic fluids have the same $\Delta T_{PP,e}$ value at other temperatures.

Table 5. Determination of optimum $\Delta T_{PP,e}$ value for different fluids under different heat source temperatures for low temperature ORC applications.

$T_{h,i}$	Optimum $\Delta T_{PP,e}$ (ºC)							
	R601a	**R601**	**R141b**	**R123**	**R152a**	**R134a**	**R1234yf**	**R1234ze**
90 ºC	5							
100 ºC	8.125				7.75	7	6.67	7.38
110 ºC	9.25				8.5	7.375	4.5	8.125
120 ºC	10.33				9	5.5	7.33	3.98
130 ºC	11.5				5	8	10	6

Figures 5 and 6 show the effect of $\Delta T_{PP,e}$ change on turbine power in waste heat and biomass-high temperature solar energy applications of ORC, respectively. When GA optimization results are evaluated for high-temperature ORC;

- In high temperature applications of ORC, the highest turbine power was achieved in the siloxanes group.
- The highest turbine power has been reached in the system with MM. It has been observed that benzene and toluene, which are aromatic hydrocarbons, perform worse than other fluids.
- It is seen that the turbine power starts to decrease after a certain $\Delta T_{PP,e}$ value at all heat source temperatures except MM fluid.
- Since MM has a lower critical temperature compared to other fluids, as the heat source temperature increased, the minimum $\Delta T_{PP,e}$ point increased.

The optimum $\Delta T_{PP,e}$ points where the maximum turbine power is obtained under 5 different heat source temperatures of 6 different fluids are summarized in Table 6. It was stated that very close $\Delta T_{PP,e}$ values were obtained in fluids in the same fluid group. It is seen that MM, which has a lower critical temperature compared to other fluids, has a lower optimum $\Delta T_{PP,e}$ value from 280 °C.

In the last part of the study, the loss of performance due to constant $\Delta T_{PP,e}$ values were investigated. It was seen from the literature research that the constant $\Delta T_{PP,e}$ value in low and high temperature ORC's was taken as 5 and 20 °C, respectively. In systems where the heat source temperature is higher than 90 °C, it is seen that taking $\Delta T_{PP,e}$ as constant 5 °C causes performance loss. In low temperature ORC systems, it is seen that the performance loss increases as the heat source temperature increases. There was less performance change in high temperature ORC systems compared to low temperature systems. Performance comparison of all fluids used in thermodynamic design was made under constant and optimum $\Delta T_{PP,e}$. On average, 38.7% and 5.9% higher turbine power was achieved for low and high temperature applications, respectively, in the optimum $\Delta T_{PP,e}$ condition. An example of performance comparison from low and high temperature applications is given below.

- At 120 °C, the turbine power under constant $\Delta T_{PP,e}$ (5 °C) in ORC system with R141b is 2.863 kW, while it is 3.871 kW under optimum $\Delta T_{PP,e}$ (10.33 °C). Under optimum $\Delta T_{PP,e}$, 35% performance increase was determined.
- At 270 °C, the turbine power under constant $\Delta T_{PP,e}$ (20 °C) in ORC system with benzene is 21.47 kW, while it is 22.42 kW under optimum $\Delta T_{PP,e}$ (27 °C). Under optimum $\Delta T_{PP,e}$, 4.42% performance increase was determined.

5. CONCLUSIONS

In this study, the effect of optimum $\Delta T_{PP,e}$ value on ORC performance was determined. Optimum $\Delta T_{PP,e}$ values were determined under different applications by thermodynamic optimization with turbine power maximization.

In low temperature ORC applications,

- The performance of dry, isentropic, wet and new-generation fluid groups was compared.
- The highest turbine power has been reached in the ORC system with R1234yf.
- While the effect of $\Delta T_{PP,e}$ on turbine power has a similar tendency in dry and isentropic fluids, it has been different due to the low critical temperature of wet and new-generation fluids.
- It was stated that the optimum $\Delta T_{PP,e}$ value for dry and isentropic fluids depends on the heat source temperature. In wet and new-generation fluids, it was determined that the optimum $\Delta T_{PP,e}$ value depends on both the heat source temperature and the organic fluid.
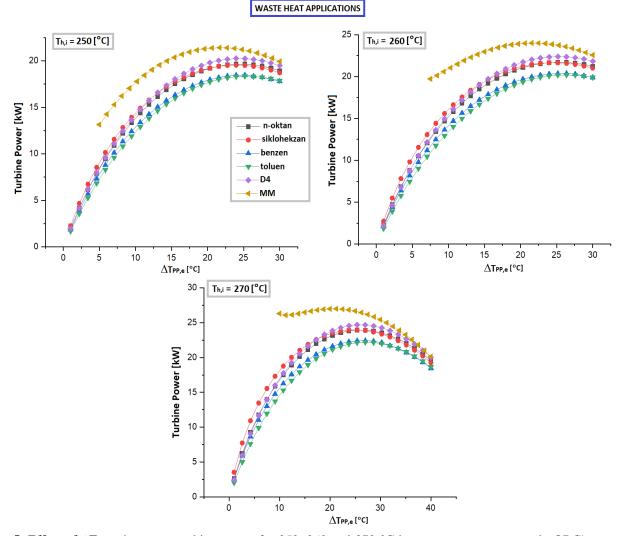
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- For 90 °C, it is seen that the optimum $\Delta T_{PP,e}$ value is the same for all fluids. In ORC systems designed at heat source temperatures above 90 °C, firstly, optimum $\Delta T_{PP,e}$ values should be determined.

In high temperature ORC applications,

- The performance of alkanes, aromatic hydrocarbon and siloxane fluid groups were compared.
- The highest turbine power has been reached in the ORC system with MM.
- Since MM has a lower critical temperature than others, as the heat source temperature increased, the minimum $\Delta T_{PP,e}$ value increased.
- Cyclohexane appears to be the fluid most affected by $\Delta T_{PP,e}$ exchange. It is stated that it has higher turbine power than others at low $\Delta T_{PP,e}$ values.
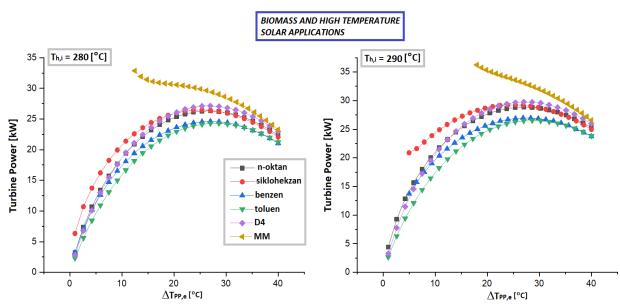
It has been determined that the effect of $\Delta T_{PP,e}$ on turbine power is greater in low temperature ORC systems. Studies in which the $\Delta T_{PP,e}$ value was taken as constant regardless of the heat source temperature and organic fluid were examined. It has been determined that 38.7% and 5.9% more turbine power will be achieved, respectively, for low and high temperature applications under optimum $\Delta T_{PP,e}$.

Taking constant $\Delta T_{PP,e}$, which has a very important place in ORC performance, causes seriously erroneous results in studies. Using optimum $\Delta T_{PP,e}$ values determined depending on the heat source temperature and organic fluid will help achieve higher ORC performances. In thermodynamic analysis, modeling and simulation studies, it is recommended to determine the optimum $\Delta T_{PP,e}$ value first.

Figure 5. Effect of $\Delta T_{PP,e}$ change on turbine power for 250, 260 and 270 °C heat source temperatures in ORC's waste heat applications.

Figure 6. Effect of $\Delta T_{PP,e}$ change on turbine power for 280 and 290 °C heat source temperatures in ORC's biomass and high temperature solar applications.

Table 6. Determination of optimum $\Delta T_{PP,e}$ value for different fluids under different heat source temperatures for high temperature ORC applications.

$T_{h,i}$	Optimum $\Delta T_{PP,e}$ (ºC)					
	n-octane	**siklohekzan**	**benzen**	**toluen**	**MM**	**D4**
250 ºC	23,96	23,96	25,17	25,17	21,67	23,96
260 ºC	25,17	23,96	26,38	26,38	21,56	25,17
270 ºC	25,38	25,38	27	27	21,25	25,38
280 ºC	27	25,38	27	28,63	12,5	27
290 ºC	27	25,42	26,88	28,63	18	27

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NOMENCLATURE

$\Delta T_{PP,e}$: Evaporator pinch point temperature difference

$\Delta T_{PP,c}$: Condenser pinch point temperature difference

Wp: Pump Work

Wt: Turbine Work

Wnet: Net Work

Qe : Evaporator heat load

Qc : Condenser heat load

i_p: Pump irreversibility

i_e: Evaporator irreversibility

i_t: Turbine irreversibility

i_c: Condenser irreversibility

i_{total}: Total irreversibility

Th: Average heat source temperature

Tc: Average coling water temperature

Th,i: Heat source inlet temperature

Th,o: Heat source output temperature

Tc,i: Cooling water inlet temperature

Tc,o: Cooling water output temperature

η_{th} : Thermal efficiency

η_{II} : Exergy efficiency

η_p : Pump isentropic efficiency

η_t : Turbine isentropic efficiency