ENERGY AND EXERGY ANALYSIS OF A COMBINED REFRIGERATION AND WASTE HEAT DRIVEN ORGANIC RANKINE CYCLE SYSTEM

by

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Energy and exergy analysis of a combined refrigeration and waste heat driven organic Rankine cycle system were studied theoretically in this paper. In order to complete refrigeration process, the obtained kinetic energy was supplied to the compressor of the refrigeration cycle. Turbine, in power cycle, was driven by organic working fluid that exits boiler with high temperature and pressure. Theoretical performances of proposed system were evaluated employing five different organic fluids which are R123, R600, R245fa, R141b and R600a. Moreover, the change of thermal and exergy efficiencies were examined by changing the boiling, condensing and evaporating temperatures. As a result of energy and exergy analysis of the proposed system, most appropriate organic working fluid was determined as R141b.

Key words: *energy, exergy, organic Rankine cycle, waste heat, refrigeration cycle.*

1. Introduction

The systems, which convert various heat energy into electrical energy, are called “Organic Rankine Cycle (ORC)”. In organic Rankine systems, organic compounds are used as working fluid instead of water. Organic Rankine systems have a very important place in converting low or middle grade waste heat or natural energy sources into useful electrical energy and have several advantages over the traditional Rankine cycles which use water as working fluid. First of all, in ORC systems, high molecular weighted, low corrosive, low critical temperature and low critical pressure organic compounds are used as working fluid. Fluid molecules collide turbine blades with low speed because of high molecular weight of the working fluid and this is important for the turbine life. Another advantage is low thermal and mechanical stress on the system equipments because of low temperature and pressure of the working fluid. These and similar advantages make attractive the ORC systems when generating power from low and middle grade heat sources.

Like other power generating systems, exergy and energy performance analysis has an important place in feasibility study of the ORC systems. There are several studies on the ORC systems about performance and working fluid selection. Sun *et al*. [1] analyzed performance of an ORC system by using dry, isentropic and wet fluids. Evaporation pressure, condensing pressure, outlet temperature of hot fluid, net power, thermal efficiency, total cycle irreversible loss and total heat recovery efficiency parameters of the system have been investigated with critical temperatures of the fluids and by changing evaporation temperatures. In another study, Kaşka [2], analyzed energy and exergy of an organic Rankine system which is used in waste heat recovery in steel industry. The variation of energy and exergy parameters of the system has been examined by changing outlet temperature of the heat source, evaporator pressure, superheating, dead state temperature and condenser pressure of the system. Ozdil *et al*. [3] thermodynamically analyzed ORC system that is located in southern of Turkey with actual plant data. In the analysis relationship between pinch point temperature and exergy efficiency was observed and as a result of the study, it is specified that, exergy effiency will increase if the pinch point temperature is decreased. In a different study, ORC system has been used in a 12 L diesel engine to recover waste exhaust gas and as cooler for the engine by Javan *et al.* [4]. The system analyzed for maximizing power generation from ORC system with using waste exhaust gas to evaporate organic fluid and hot water of the engine is used to actualize preheating the organic fluid in ORC system. Many studies have been done about using geothermal, solar or waste heat sourced ORC systems [5-9].

In this study, a typical refrigeration cycle is combined with a waste heat sourced ORC system. Generated power in ORC system is provided to the compressor in refrigeration system to complete cooling process. Similar system is analyzed theoretically by Cihan [10], using dry and isentropic fluids which are R600, R600a, R601 and R245fa, in the manner of coefficient of performances of power and refrigeration cycle and first law efficiency of thermodynamics. In another similar study, Lian *et al.* [11] investigated the change of overall system coefficient of performances, compression and expansion ratios in compressor and turbine due to boiling, evaporation and condensation temperatures for four different organic fluid (R290, R600, R600a and R1270). There are different studies about this subject using ejector instead of compressor in refrigeration cycle [12-14].

In organic Rankine systems fluid selection is important and many studies have been done to specify proper type of fluid and many researchers have suggested dry or isentropic fluids for this type of systems [15-16]. In expansion process at turbine, condensation may occur when temperature and pressure of working fluid decrease. This causes reduction in turbine life if liquid molecules of working fluid collides with turbine blades. Because of that reason, in this study dry, R600, R600a, R245fa, R123 and isentropic, R141b, organic fluids have been used as working fluid. Thermophysical properties of working fluids which used in the system is shown in tab. 1. As it seems from table, critical pressures and temperatures of the working fluids are less than R718 (Water). On the contrary, molecular masses of these working fluids are higher than R718.

Table.1 Thermophysical Properties of working fluids used in cycle

Working Fluid	Type of Fluid	Molecular Mass [g/mole]	Critical Temperature [K]	Critical Pressure [MPa]
R600a	Dry	58,12	407,8	3,63
R600	Dry	58,12	425,1	3,8
R245fa	Dry	134,05	427,2	3,64
R123	Dry	152,93	456,8	3,66
R141b	Isentropic	116,95	477,5	4,21
R718	Wet	18,00	647,1	22,00

2. System Description

The schematic diagram of the combined refrigeration and organic Rankine cycle is shown in fig. 1. As it shown in the figure, the system consists of evaporator, compressor, expansion valve in refrigeration cycle and pump, boiler, turbine in organic Rankine cycle. The system has a condenser for common use in refrigeration and ORC. Power is generated in turbine by high temperature and pressure working fluid and delivered to the compressor to complete refrigeration process by a mechanical shaft. Processes in the system can be listed in following:

1-2: Compression process in compressor,

3-4: Reducing pressure of working fluid at constant enthalpy in expansion valve,

4-1: Heat addition to the working fluid at constant temperature from cooled medium,

3-5: Increasing pressure of liquid fluid in pump,

5-6: Heat addition process at constant temperature by using waste heat source in boiler,

6-7: Generating power in turbine by using high temperature and pressure fluid vapor

8-3: Heat rejection to the cold source at constant temperature in condenser,

2-7-8: Adiabatic mixing of compressor and turbine leaving fluid at mixing chamber.

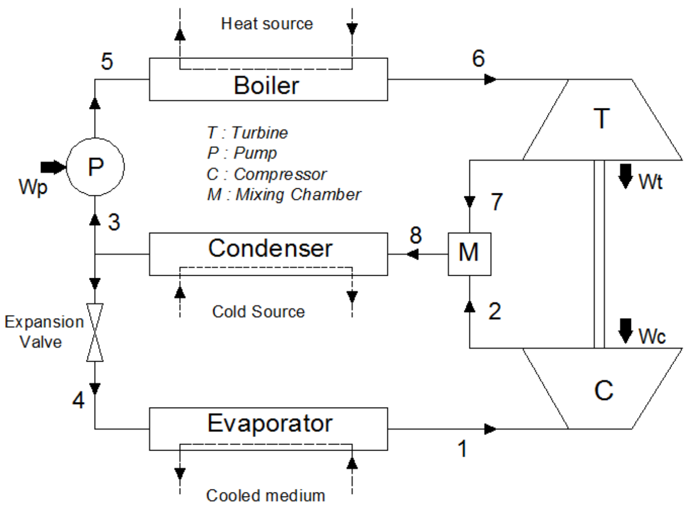


Figure 1. Schematic diagram of combined refrigeration and organic Rankine cycle

T-s diagram of the purposed system for R123 is also shown in fig. 2. Boiling temperature, evaporation temperature and condensing temperature is 100 °C, 0 °C and 45 °C respectively.

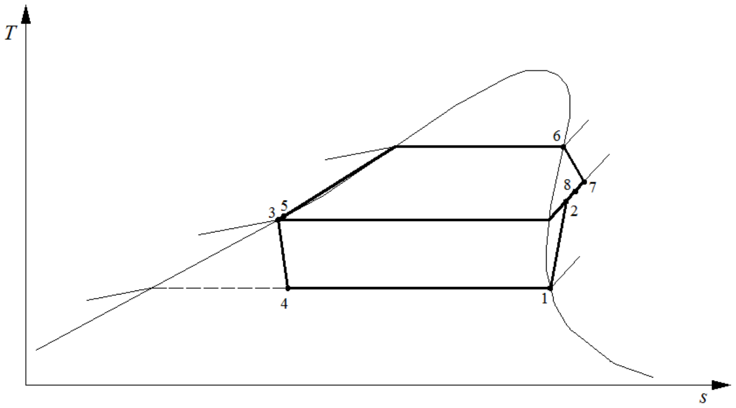


Figure 2. T-s diagram of the combined refrigeration and ORC system for R123

3. Theoretical Calculations

As it seems from fig.2, to simplify calculations, saturated state is assumed at evaporator, boiler and condenser exit and heat and friction losses in the system are neglected. Calculations are carried out with the aid of EES software [17]. For the purposed system, shown in fig. 1, mathematic model is given below.

The thermal efficiency of the ORC system η_{ORC} can be defined as:

$$\eta_{ORC} = \frac{\dot{W}_{turb}}{\dot{Q}_{boi} + \dot{W}_{pump}} \tag{1}$$

$\dot{W}_T$, $\dot{Q}_B$ and $\dot{W}_P$ will be calculated from following equations:

$$\dot{W}_{turb} = \eta_{turb,isen} * \eta_{turb,mec} * W_{turb,isen} \tag{2}$$

$$\dot{Q}_{boi} = \dot{m}_{ORC} * (h_6 - h_5) \tag{3}$$

$$\dot{W}_{pump} = \dot{m}_{ORC} * (h_5 - h_3)/\eta_{pump} \tag{4}$$

The coefficient of performance of refrigeration cycle can be defined as:

$$COP_C = \frac{\dot{Q}_{eva}}{\dot{W}_{comp}} \tag{5}$$

$\dot{Q}_{eva}$ and $\dot{W}_C$ can be found from following equations.

$$\dot{Q}_{eva} = \dot{m}_{ref} * (h_1 - h_4) \tag{6}$$

$$\dot{W}_{comp} = \frac{\dot{W}_{c,isen}}{\eta_{c,mec} * \eta_{c,isen}} \tag{7}$$

Heat rejected from condenser can be calculated by following equation:

$$\dot{Q}_{con} = \left(\dot{m}_{ref} + \dot{m}_{ORC}\right) * (h_8 - h_3) \tag{8}$$

Net power output from the turbine will be equal to net power input to the compressor. Hence, the following equation obtained:

$$\dot{W}_{c,isen} = \eta_{stc} * \dot{W}_{t,isen} \tag{9}$$

η_s can be defined with following equation:

$$\eta_{stc} = \eta_{c,mec} * \eta_{c,isen} * \eta_{t,isen} * \eta_{t,mec} \tag{10}$$

Total system thermal efficiency, η_{sys}, can be defined by equation below:

$$\eta_{sys} = \frac{\dot{Q}_{eva}}{\dot{Q}_{boi} + \dot{W}_{pump}} \tag{11}$$

Specific exergy flow of any state, ψ_i, can be defined as:

$$\psi_i = h_i - h_0 - [T_0 * (s_i - s_0)] \tag{12}$$

Thus, exergy flows can be defined as:

$$\dot{E}_i = \dot{m}_i * \psi_i \tag{13}$$

Exergy destroyed in boiler and boiler exergy efficiency :

$$I_{boi} = \dot{m}_{ORC} * T_0 * \left[s_6 - s_5 - \left(\frac{q_{boi}}{T_{boi}}\right)\right] \tag{14}$$

$$\eta_{e,boi} = 1 - \frac{I_{boi}}{\dot{E}_6 - \dot{E}_5} \tag{15}$$

Destroyed exergy in turbine and exergy efficiency of the turbine can be defined as:

$$I_{turb} = \dot{E}_6 - \dot{E}_7 - \dot{W}_{turb} \tag{16}$$

$$\eta_{e,turb} = \frac{\dot{W}_{turb}}{\dot{E}_6 - \dot{E}_7} \tag{17}$$

Exergy destroyed in pump and exergy efficiency of the pump can be calculated equations below:

$$I_{pump} = \dot{E}_5 - \dot{E}_3 + \dot{W}_{pump} \tag{18}$$

$$\eta_{e,pump} = \frac{\dot{E}_5 - \dot{E}_3}{\dot{W}_{pump}} \tag{19}$$

Exergy destroyed in compressor and exergy efficiency of the compressor:

$$I_{comp} = \dot{E}_2 - \dot{E}_1 + \dot{W}_{comp} \tag{20}$$

$$\eta_{e,comp} = \frac{\dot{E}_2 - \dot{E}_1}{\dot{W}_{comp}} \tag{21}$$

At expansion valve destroyed exergy can be defined as following:

$$I_{exp} = \dot{E}_3 - \dot{E}_4 \tag{22}$$

Destroyed exergy in the mixing chamber:

$$I_{mix} = \dot{E}_2 + \dot{E}_7 - \dot{E}_8 \tag{23}$$

Destroyed exergy at the evaporator and exergy efficiency of the evaporator can be defined as:

$$I_{eva} = \dot{m}_{ref} * T_0 * \left[s_4 - s_1 - \left(\frac{q_{eva}}{T_{eva}}\right)\right] \tag{24}$$

$$\eta_{e,eva} = 1 - \frac{I_{eva}}{\dot{E}_4 - \dot{E}_1} \tag{25}$$

Destroyed exergy at the condenser and exergy efficiency of the condenser can be calculated by following equations:

$$I_{con} = (\dot{m}_{ref} + \dot{m}_{ORC}) * T_0 * \left[s_3 - s_8 + \left(\frac{q_{con}}{T_{con}}\right)\right] \tag{26}$$

$$\eta_{e,con} = 1 - \frac{I_{con}}{\dot{E}_8 - \dot{E}_3} \tag{27}$$

Total system exergy efficiency can be defined by the equation below:

$$\eta_{e,sys} = \frac{\dot{E}_{cooling}}{\dot{E}_{in}} \tag{28}$$

In eq. 28, $\dot{E}_{cooling}$ and $\dot{E}_{in}$ will be calculated from following equations:

$$\dot{E}_{cooling} = \dot{E}_1 - \dot{E}_4 \tag{29}$$

$$\dot{E}_{in} = \dot{E}_6 - \dot{E}_5 + \dot{W}_P \tag{30}$$

Table 2. Input parameters and boundary conditions

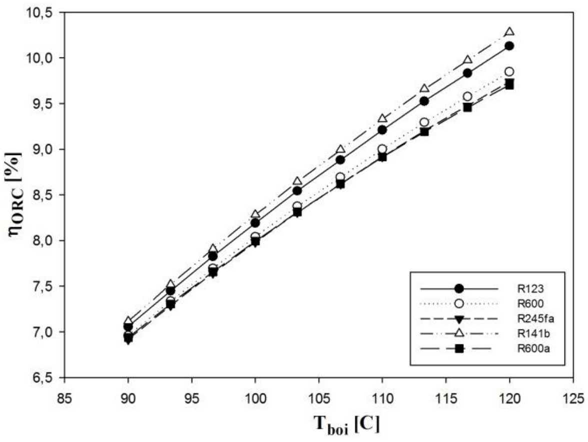
Parameter	Typical Value	Ranges
Working fluid mass flow rate in ref. cycle, $\dot{m}_{ref}$	1.0 kg/s	-
Dead state temperature, T_0	293 K	-
Dead state pressure, P_0	101,3 kPa	-
Evaporator temperature, T_{eva}	0 °C	-5 to 5 °C
Boiler temperature, T_{boi}	100 °C	90 to 120 °C
Condenser temperature, T_{con}	45 °C	40 to 50 °C
Turbine mechanical efficiency, $\eta_{t,mec}$	85%	-
Turbine isentropic efficiency, $\eta_{t,isen}$	75%	-
Compressor mechanical efficiency, $\eta_{c,mec}$	80%	-
Compressor isentropic efficiency, $\eta_{c,isen}$	70%	-
Pump isentropic efficiency, η_{pump}	80%	-

In tab. 2, input parameters, including isentropic and mechanical efficiencies of components [18-20], and boundary conditions are listed to carry out calculations. As it seems from table, system has been investigated under variation of evaporator, condenser and boiler temperatures.

4. Results and Discussion

4.1. Effect of the Boiler Temperature

a)



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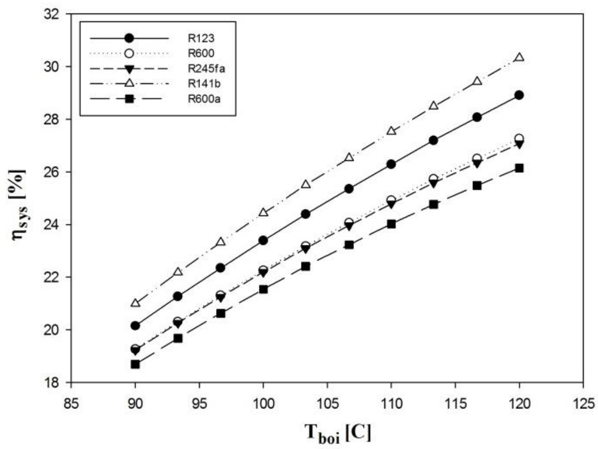

c)

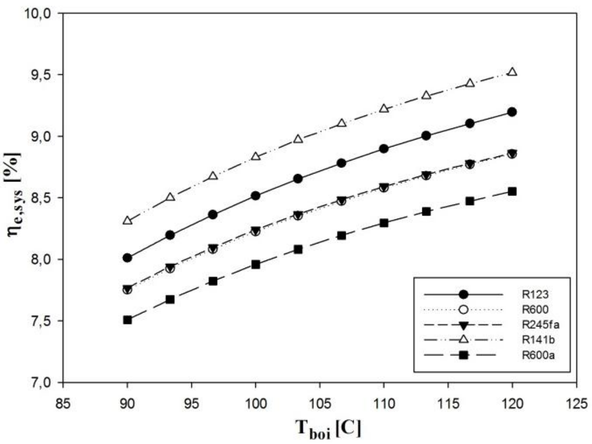

Figure 3. Performance parameters versus boiler temperature: a) Thermal efficiency of ORC; b) Total system thermal efficiency; c) Total system exergy efficiency.

To investigate effect of the boiler temperature, evaporator and condenser temperatures are fixed at 0 °C and 45 °C, respectively. In fig. 3, variation of thermal efficiency of ORC, total system thermal efficiency and exergy efficiency of the total system versus boiler temperature has been shown. Graphics indicate that, boiler temperature has a positive effect on given system performance parameters for all decided working fluids. As shown in fig. 3a, thermal efficiency of the ORC system will increase when boiler temperature increases. Since the desired input power to compressor is equal to output power of turbine, at high temperatures system can produce the same power with low mass flow rate in ORC. This causes reduction in $\dot{Q}_{boi}$ and increase in thermal efficiency. The best performance has been shown by R141b since all working fluids are very close to each other. It seems from fig. 3b that, because of previous reason, total system thermal efficiency will be greater for all working fluids if boiler temperature increases. R141b has greater values, over 30% efficiency, compare the other fluids, where the worst performance has been shown by R600a. Variation of exergy efficiency of the total system versus boiler temperature has been shown in fig. 3c. Figure indicates that for all decided working fluids exergy efficiency of the system becomes greater while boiler temperature increases. R141b, again, shows the best performance with over 9,5% at 120 °C, while R600a has the smallest value of efficiency approximately 8,6% at the same temperature.

4.2. Effect of the Evaporator Temperature

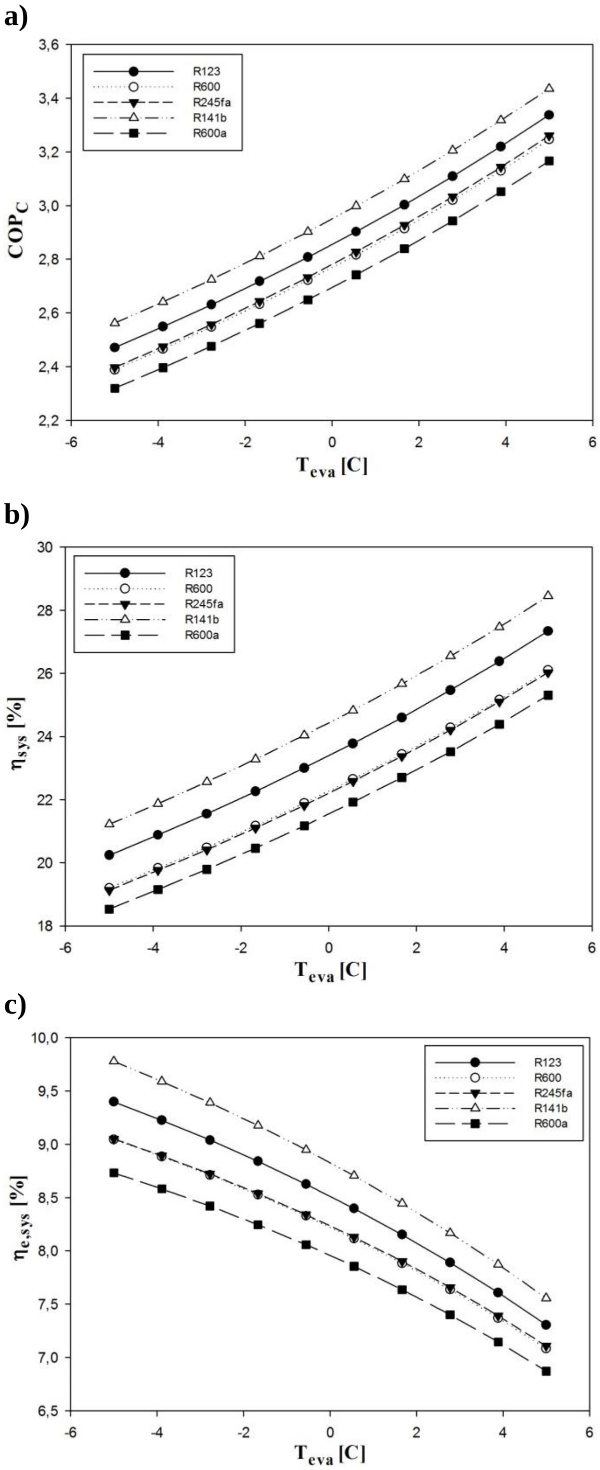

Figure 4. Performance parameters versus evaporator temperature: a) Coefficient of performance of ref. cycle; b) Total system thermal efficiency; c) Total system exergy efficiency.

The effect of the evaporator temperature on performance parameters is shown in fig. 4. At this time, boiler and condenser temperatures are fixed at 100 °C and 45 °C respectively. In fig. 4a, variation of COP values of the refrigeration cycle due to evaporator temperature has been shown. As it seems from figure, COP values become larger if the evaporator temperature increases. By using R141b, the greatest values can be obtained since the smallest values of COP will be determined if R600a is preferred as working fluid for this system. At another figure, fig. 4b, the change of total system thermal efficiency due to evaporator temperature has been shown. As it can be seen from figure that increase in evaporator temperature affects positively At given range, system reaches maximum value of efficiency up to 28% with R141b at 5 °C where the smallest values have been determined by using R600a. In fig. 4c, variation of total system exergy efficiency against evaporator temperature has been indicated. As it seems from figure exergy efficiency of the system is negatively affected as a result of closing dead state temperature by increase in evaporator temperature at given range. The greatest value of exergy efficiency, near 10 % has been obtained, again, with R141b at the smallest evaporator temperature which is -5 °C at given range. The worst performance has been shown by R600a for this parameter.

4.3. Effect of the Condenser Temperature

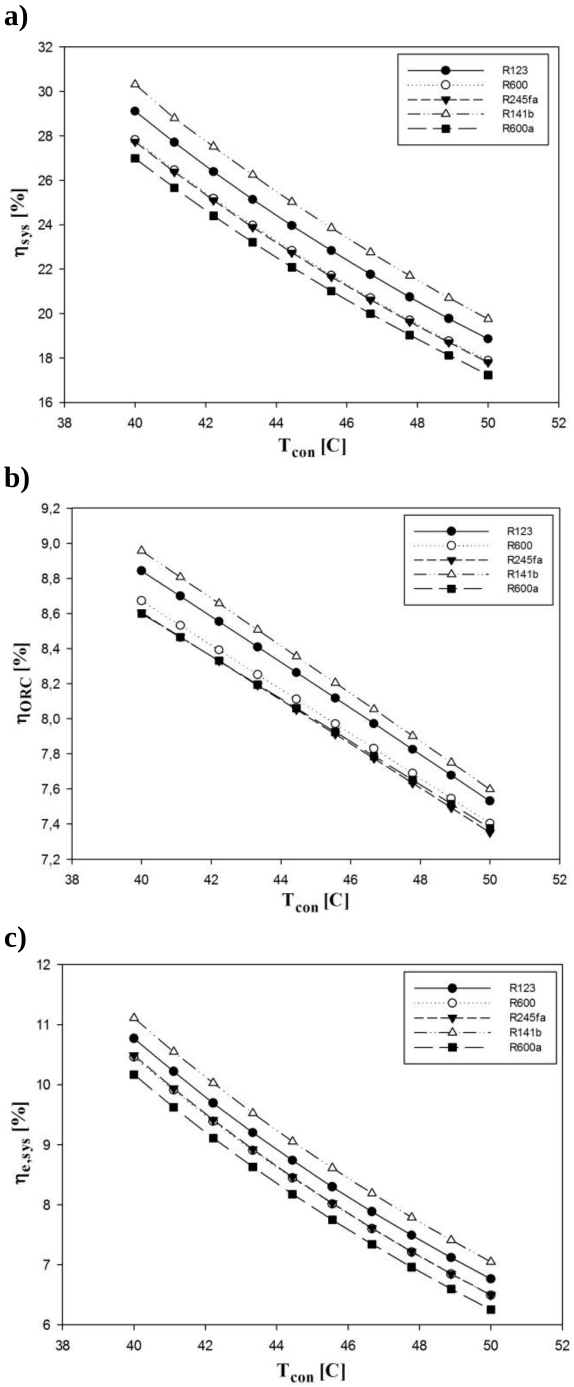

Figure 5. Performance parameters versus condenser temperature: a) Total system thermal efficiency; b) Thermal efficiency of the ORC; c) Total system exergy efficiency.

To determine effect of the condenser tempeature to the system performance parameters boiler and evaporator temperatures are fixed at 100 °C and °C respectively. As it is clearly seen from fig. 5 that, if the system is operated at high condenser temperatures performance of the system will be low. In fig. 5a and fig. 5b, variation of thermal efficiency of the total system and thermal efficiency of the ORC system due to condenser temperature has been shown. Because of high cooling load in compressor, high turbine work output is required and this causes increase in heat input at boiler as a result of high condenser temperature. It can be seen from eqs.1 and 11 that, high boiler heat input reduces ORC and total system thermal efficiency. Between five different working fluid, R141b is the best at either low and high condenser temperatures. The smallest values are obtained by using R600a and R245fa for the total system and ORC system respectively. In fig. 5c, one can see that exergy efficiency of the system will decrease as a result of high exergy input to the system. As it is well known, high condenser temperatures cause reduction in coefficient of performance (COP) at typical refrigeration cycles.

Table 3. Exergy Losses and Exergy Efficiencies of the Components

Component	Destroyed Exergy (kW)	Power I/O (kW)	Exergy Efficiency (%)
Evaporator	0,1322	135,4	98,7
Condenser	56,48	696,5	2,1
Expansion Valve	0,8515	----	0
Boiler	10,48	577,4	90,8
Compressor	50,05	47,42	58,4
Pump	2,891	1,592	81,7
Mixing Chamber	0,012	----	0
Turbine	24,69	47,42	65,8
Total System	145,6	----	8,5

For R123, destroyed exergy and exergy efficiency of components shown in tab. 3 ($T_{boi} = 100\ ℃$, $T_{eva} = 0\ ℃\ and\ T_{con} = 45\ ℃$). As a result of the calculations, the greatest exergy lost occurred in condenser because of high heat rejection resulting high heat loss. At mixing chamber because of adiabatic mixing, exergy loss nearly zero. Exergy loss value of the total system is 145,6 kW and exergy efficiency of the system 8,5% for these boiler, evaporator and condenser temperatures.

5. Conclusions

In this study, exergy and energy analysis of a combined refrigeration and waste heat sourced organic Rankine cycle has been done. Five different working fluids has been used as working fluid, which are R123, R600, R600a, R245fa and R141b, energy and exergy performance of the system analyzed due to change in boiler, evaporator and condenser temperatures at specified intervals.

It can be easily understood from the results that, R141b is the most suitable organic fluid between these five different fluids for the given parameters. Frequently used organic fluid in ORC systems, R245fa, seems to be not appropriate organic fluid for this system since its performance very low compare to R141b and R123.

Since evaporator and condenser temperatures is constant, if the system is operated at high boiler temperatures both energy and exergy performance of the system will be better. High evaporator temperatures decrease cooling load resulting greater coefficient of performance values but despite that it adversely effects the exergy efficiency of the system. Thermal efficiency and exergy efficiency of the system will dramatically decrease when the condenser temperature is increased. It causes almost 10%, 2% and 4% reduction in thermal efficiency of the system, thermal efficiency of the ORC system and exergy efficiency of the system respectively for each working fluid. According to that results, it can be understood that the best working fluid for the purposed system is R141b and the optimum working temperature conditions are high boiler temperature, medium evaporator temperature and low condenser temperature.

In this study, dry and isentropic type working fluids were chosen particularly, despite that R141b was specified as the best working fluid, life-cycle cost analysis (LCCA) is significant beside energy and exergy analysis, before making the last decision for working fluid. In life-cycle cost analysis, the most appropriate result can be attained when investment cost (cost of turbine, compressor, heat exchangers, pump, expansion valve, etc.), assembly and commissioning cost, operating cost, maintain and service cost are considered.

Nomenclature

h	– enthalpy [$kJkg^{-1}$]
s	– entropy [$kJkg^{-1}K^{-1}$]
T	– temperature [C]
P	– pressure [kPa]
$\dot{m}$	– mass flow rate [kgs^{-1}]
$\dot{W}$	– power [kW]
$\dot{Q}$	– heat transfer rate [kW]
I	– destroyed exergy [kW]
ψ	– specific exergy flow [$kJkg^{-1}$]
$\dot{E}$	– exergy flow [kW]

Greek symbols

η	– efficiency [%]

Acronyms

ORC	– organic Rankine cycle

Subscripts

ORC	– organic Rankine cycle
turb	– turbine
comp	– compressor
eva	– evaporator
con	– condenser
mix	– mixing chamber
exp	– expansion valve
boi	– boiler
C	– cooling
ref	– refrigeration cycle
t,isen	– isentropic situation of turbine
c,isen	– isentropic situation of compressor
t,mec	– mechanical efficiency of turbine
c,mec	– mechanical efficiency of compressor
stc	– combined turbine and compressor
sys	– total system
e	– exergy
1-8	– state points of the cycle
0	– dead state
in	– input

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