CTST'19 22th Congress of Thermal Sciences and Technology
11-14 September 2019, Kocaeli

ORGANIC RANKINE CYCLES FOR NATURAL GAS FIRED ENGINE WASTE HEAT RECOVERY USING BOTH EXHAUST GAS AND JACKET WATER HEAT SOURCES

Ahmet DOĞAN*, Nurullah KAYACI, Ali PINARBASI***

* Yıldız Teknik Üniversitesi, Makine Fakültesi, Makine Mühendisliği Bölümü,
34349 Beşiktaş, İstanbul, ahmdogan@yildiz.edu.tr
**Namık Kemal Üniversitesi, Çorlu Mühendislik Fakültesi, Makina Mühendisliği Bölümü,
59860 Çorlu, Tekirdağ

Abstract: The sizing and design of organic Rankine cycles (ORC) that can generate electricity using the waste heat source obtained from natural gas-fired internal combustion engine (ICE) is highly significant. In ORC power plants, the engine can recover waste heat from exhaust gas or jacket water. Also, the appropriate organic fluids for a waste heat ORC application (i.e. driven by heat sources up to 453°C) are chosen based on many criteria such as cycle efficiency, fluid densities, safety and environmental impacts, and boiling and critical temperature.In the present study, thermodynamic models of ORC system with various working fluids are simulated in EES recover waste heat energy from exhaust gas and jacket water at the same time. The effects of different parameters such as thermal and exergy efficiency, the net power output, mass flow rate of working fluids, the ratio of evaporator heat transfer rate to preheater heat transfer rate (Qevp/Qph on the system performance are examined. Results show that toluene, benzene and cyclohexane present the highest thermal efficiency and net power output values ranging from 9.35% to 21.6% , and from 293.3 kW to 604.9 kW. For the meantime, the three fluids also show the lowest working fluid mass flow rate values ranging from 3.75 kg/s to 6.75 kg/s.
Keywords: Organic Rankine Cycle, Internal Combustion Engine, Waste heat recovery, Thermodynamic analysis

INTRODUCTION

Since the beginning of the 70s, the ORC has been a well-known technology; it is used in lots of practical applications, mainly for solar power, biomass power, combined heat and power, geothermal and waste heat recovery applications. [1] The working principles and components of the ORC systems are similar to conventional steam Rankine power plant, but instead of water, an organic fluid is used as the working fluid in the cycle. The organic fluid used is generally a hydrocarbon (n-pentane, n-butane, etc.), refrigerant, siloxane, perfluorocarbons (fully fluorinated hydrocarbons), perfluorocarbon or partially flouro-substituted straight chain hydrocarbon, etc. [2].

Internal combustion engines are a well-known technology used in power generation in the range of kW to MW scales owing to their reliability, low specific cost and high thermal efficiency. Although ICE can convert around 40% of fuel energy to useful work, the rest of the energy is being wasted through exhaust waste heat, engine cooling block and heat loses[3]. Increasing fossil fuel consumption due to population increase made the use of waste heat energy very important. Organic Rankine Cycle (ORC) systems have been widely proposed as bottoming power cycles to recover waste heat from ICEs and therefore reduces emissions and improve overall system efficiency without fossil fuel addition [4].

A number of studies have been conducted on recovering heat from the jacket water and exhaust gas of ICEs owing to large amount of power and high temperature[5-7]. Yu et al. [8] analyzed an actual ORC bottoming system of a
diesel engine with a simulation model. The waste heat of engine exhaust gas and jacket water were recovered in the single loop ORC system using R24fa as a working fluid. The results showed that the thermal efficiency of diesel engine can be increased by 6.1% and this correspond to 14.5 kW additional power. Seyedkavoosi et al. have a research about performance analyses of duel loop ORC for waste heat recovery from the ICE. The ORC system is modeled with two step configuration and analyzed with different working fluids. They stated that R-123 is the most suitable working fluid under the considered condition with 21% exergy efficiency and 468 kW of net output power[9]. Le et al.[10] analyzed thermodynamic and economic optimization ORC system with comparison of different working fluids for industrial waste heat recovery. Thermodynamic analysis illustrate that n-pentane and R245fa have highest exergy efficiency with 53.2% and 53% respectively. The economic analysis of cycle with n-pentane shows the lowest investment cost and shortest payback period. Wang et al. [11]. investigated different working fluids performances such as butane, R113, R141b, R123, R11, R245fa with standard single loop ORC cycle and with additional internal heat exchanger ORC cycle, and it is found that the thermal efficiency of cycle with R141b, R11 and butane is slightly higher.

This study is focused on assessing the extra power achievable by using an exhaust gas and a jacket water as

a heat source for just one vapour power cycle, hence with a rather simplified design of the heat recovering unit, under the hypothesis of operating the engine at full load while considering different cycles based on different organic fluids and configurations.

SYSTEM DESCRIPTION

Engine Parameters

The selected ICE in this paper is an inline six-cylinder turbocharged engine manufactured by Caterpillar Energy Solutions model number TCG2032.The parameters of the engine and exhaust gas composition are shown in Table 1 and Table 2.

Table 1 Operating conditions of the ICE

Property	Unit	Value
Engine load	%	100
Exhaust temperature	°C	453
Exhaust gas mass flow rate	kg/h	22832
Electrical Power	kW	4300
Fuel consumption ($\dot{Q}_{e,con}$)	kW	9824
Thermal efficiency	%	43,8
Jacket water inlet temperature	°C	93
Jacket water mass flow rate	kg/h	22065

Table 2 Composition of the engine exhaust gas

Composition	Volumetric Ratio (%)	Molecular Weight (kg/kmol)	Spesific Heat (kj/kg°C)
Argon (Ar)	0,1	40	0,5206
Carbondioxide (CO_2)	4,9	44	1,061
Water (H_2O)	11,2	18	2,013
Oxygen (O_2)	9,9	32	0,993
Nitrogen (N_2)	73,9	28	1,07

ORC system description as the bottoming system

The schematic diagram of the bottoming ICE-ORC system shown in Figure 1 are subcritical. The system consist of an expander, a generator, a condenser, a pump, a preheater and an evaporator.For the ORC loop, benzene, R141b, R245fa, toluene and cyclohexene are chosen to be working fluid candidates. The properties of these fluids are shown in Table 3.

The system works as follows; the working fluids in liquid state at point 1 is pumped to evaporation pressure, then heated by engine jacket water to point 3 and next evaporated and superheated by exhaust gas in the evaporator. Afterwards, high pressure and high temperature superheated vapor flows into expander to generate useful work then low pressure vapor exits from expander and enters into condenser that it transform in to saturated liquids at condensers exit by cooling water. Consequently entire cycle completes.

The ORC system is recovering waste heat from jacket water and exhaust gas together.Thermal energy of the exhaust gas is completed recovered therefore the minimum allowable exhaust gas outlet temperature is set to 120 OC in. Whereas the inlet temperature of the jacket water is constant at 93 OC in system, the outlet temperature varies depending on working fluid flow rate.

Conditions and assumptions

The simulation of the ORC system in this study is done in the EES software. In order to simplify the calculation some of the parameters are chosen as constant. Assumptions and conditions are given as follows:

- The outlet of the exhaust gas is set to 120 oC to avoid acif corrosion
- The efficiencies of the pump and expander are set to be 0.8.
- The condensation temperature of working fluids at condenser outlet is fixed at 35 oC.
- The pressure loss and heat loss of the heat exchangers and pipelines are neglect.

Thermodynamic Analysis

Thermodynamic model consist of mathematical models, balance equations and efficiency terms calculations. Enthalpy and exergy can be found for all state points depending on statement points temperatures and pressures.

Waste heat of engine can be expressed as follows:

$$\dot{Q}_{exhaust} = \dot{m}_{ex} c_{p,ex} (T_{ex,in} - T_{ex,out}) \quad (1)$$

Working fluid mass flow rate can be expressed as follows:

$$\dot{m}_{wf} = \frac{\dot{Q}_{exhaust}}{q_{evaporator}} \quad (2)$$

Evaporator heat intake per unit mass can be expressed as follows:

$$q_{evaporator} = h_{oe} - h_{ie} \quad (3)$$

General exergy terms for each state point can be expressed as follows:

$$ex_i = (h_i - h_0) - T_0 (s_i - s_0) \quad (4)$$

General exergy destruction terms for each state point can be expressed as follows:

$$\dot{E}x_d = T_0 \dot{S}_{gen} \quad (5)$$

General thermal exergy terms for each state point can be expressed as follows:

$$\dot{E}x^{Q_i} = \dot{Q}_i (1 - \frac{T_0}{T_s}) \quad (6)$$

Net output power of turbine can be expressed as follows:

$$\dot{W}_{net} = \dot{W}_{tubine} - \dot{W}_{pump} \quad (7)$$

Net electricity production of ICE can be expressed as follows:

$$\dot{W}_{cogen} = \dot{Q}_{e,cogen}\eta_{e,epe} \tag{8}$$

Balance Equations

Balance equations are developed separately for system in Figure 1.System friction and heat loss inside pipes, kinetic and potential energy changes are neglect. It is assumed that the system is steady state conditions. Components are assumed adiabatic. Balance calculations are performed for each component with consideration of enthalpies, exergies, mass flow rates, pressures and temperatures. It is presented in Tablne 4.

Thermal Efficiency

Thermal efficiency and second law efficiencies are important criteria for optimization of cycle. Efficiency terms of ORC system could be calculated with the equations presented in Table 4.

On the other hand, ICE and ORC combined system overall efficiency is the most significant thermal output of the system. Feasibility report will be comprised depending on cogeneration system overall recovery terms.

Cogeneration system combined overall efficiency can be express as follows:

$$\eta_{cogen} = \frac{\dot{W}_{cogen} + \dot{W}_{net}}{\dot{Q}_{e,con}} \tag{9}$$

Reversible work is represents the maximum power output which could be provide at working conditions. Reversible work can be express as follows:

$$\dot{W}_{rv} = \dot{Q}_{loss}(1 - \frac{T_b}{T_s}) \tag{10}$$

Cogeneration system heat recovery efficiency caan be express as follows:

$$\eta_{hrv} = \frac{\dot{W}_{net} + \dot{W}_{cogen}}{\dot{W}_{rv} + \dot{W}_{cogen}} \tag{11}$$

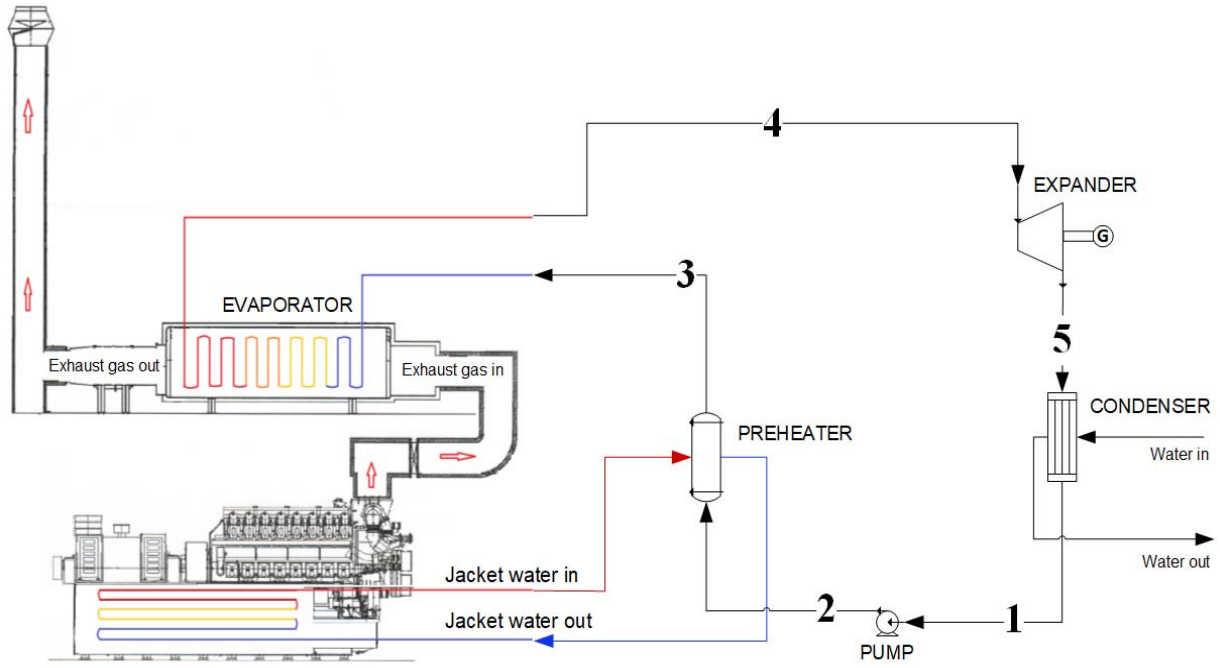


Figure 1 Schematic diagram of the bottoming ICE-ORC system

Table 1 Working fluid properties

Working Fluid	Formula	Molecule Weight (kg/mol)	T_{crit}[a] (°C)	P_{crit}[b] (kPa)	ALT[c]	GWP[d]	ODP[e]	ASHRAE 34 Classification
Cyclopentane	$(CH_2)_5$	84.16	280.6	4075	12	Low	0	A3
Benzene	C_4H_{10}	58.12	289.1	3796	12	Low	0	A3
R123	$C_2HF_3Cl_2$	152.93	183.83	3672	1.3	Low	0.012	B1
R141b	$C_2H_3FCl_2$	116.95	206.96	4212	9.3	Low	0.086	A2
R245fa	$C_3H_3F_5$	134.05	154.20	3651	7.2	Low	0	B1

ASHRAE 34 Classification: 1:non-flammable, 2:low-flammable, 3:high-flammable;A:lower toxicity, B:higher toxicity
a Tcrit: critical temperature
b Pcrit: critical pressure
c ALT: atmospheric life-time (years)
d GWP: global warming potential, relatively CO2 (100 years)
e ODP: ozone depletion potential, relatively R11

Table 4 Balance equations for ORC system with preheater

Balance Equation Type	Component	Balance Equation
Mass Balance Equations (M)	**Overall System**	$\dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = \dot{m}_5 = \dot{m}_{wf}$
Energy Balance Equation (E)	**Turbine**	$\dot{W}_{turbine} = \dot{m}_4 h_4 - \dot{m}_5 h_5$
	Condenser	$\dot{Q}_{cond} = \dot{m}_5 h_5 - \dot{m}_1 h_1$
	Pump	$\dot{m}_2 h_2 - \dot{m}_1 h_1 = \dot{W}_{pump}$
	Evaporator	$\dot{m}_3 h_3 - \dot{m}_2 h_2 = \dot{m}_{wf} q_{evaporator}$, $\dot{m}_{wf,r} q_{evaporator} = \dot{Q}_{exhaust}$, $\dot{Q}_{exhaust} = \dot{m}_{ex} c_{p,ex} (T_{ex,in} - T_{ex,out})$
	Preheater	$\dot{m}_2 h_2 + \dot{m}_{jw} h_{jw,in} = \dot{m}_{jw} h_{jw,out} + \dot{m}_3 h_3$
	Overall System	$\dot{Q}_{exhaust} + \dot{W}_{pump} + \dot{Q}_{preheater} = \dot{Q}_{cond} + \dot{W}_{turbine}$
Exergy Balance Equation (Ex)	**Turbine**	$\dot{m}_4 ex_4 = \dot{m}_5 ex_5 + \dot{W}_{turbine} + \dot{E}x_{d,turbine}$
	Condenser	$\dot{m}_5 ex_5 = \dot{m}_1 ex_1 + \dot{E}x^{Q_{loss}} + \dot{E}x_{d,condenser}$
	Pump	$\dot{m}_1 ex_1 + \dot{W}_{pump} = \dot{m}_2 ex_2 + \dot{E}x_{d,pump}$
	Evaporator	$\dot{m}_3 ex_3 + \dot{E}x^{Q_{exhaust}} = \dot{m}_4 ex_4 + \dot{E}x_{d,evaporator}$
	Preheater	$\dot{m}_2 ex_2 + \dot{m}_{jw} ex_{jw,in} = \dot{m}_3 ex_3 + \dot{m}_{jw} ex_{jw,out} + \dot{E}x_{d,preheater}$
	Overall System	$\dot{E}x^{Q_{exhaust}} + \dot{W}_{pump} + \dot{E}x^{Q_{preheater}} = \dot{E}x^{Q_{cond}} + \dot{W}_{turbine} + \dot{E}x_{d,overall}$
Entropy Balance Equation (En)	**Turbine**	$\dot{m}_4 s_3 + \dot{S}_{gen,turbine} = \dot{m}_5 s_5$
	Condenser	$\dot{m}_5 s_5 + \dot{S}_{gen,condenser} = \dot{m}_1 s_1 + \frac{\dot{Q}_{cond}}{T_b}$
	Pump	$\dot{m}_1 s_1 + \dot{S}_{gen,pump} = \dot{m}_2 s_2$
	Evaporator	$\dot{m}_3 s_3 + \frac{\dot{Q}_{exhaust}}{T_s} + \dot{S}_{gen,evaporator} = \dot{m}_4 s_4$
	Preheater	$\dot{m}_2 s_2 + \dot{m}_{jw} s_{jw,in} + \dot{S}_{gen,jw} = \dot{m}_3 s_3 + \dot{m}_{jw} s_{jw,out}$
	Overall System	$\frac{\dot{Q}_{exhaust}}{T_s} + \dot{S}_{gen,overall} = \frac{\dot{Q}_{cond}}{T_b}$

Table 5. Efficiency terms of ORC with preheater

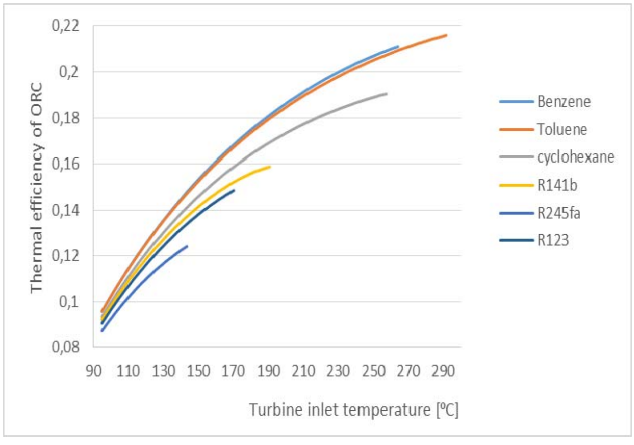
Component	First Law Efficiency	Second law efficiency
Pump	$\eta_{I,pump} = \frac{h_2 - h_1}{w_{pump}}$	$\eta_{II,pump} = \frac{ex_2 - ex_1}{w_{pump}}$
Evaporator	$\eta_{I,evaporator} = \frac{h_4 - h_3}{q_{evaporator}}$	$\eta_{II,evaporator} = \frac{ex_4 - ex_3}{q_{evaporator}(1 - \frac{T_{s,wf}}{T_{s,ex}})}$
Condenser	$\eta_{I,condenser} = \frac{q_{cond}}{h_5 - h_1}$	$\eta_{II,condenser} = \frac{q_{loss}(1 - \frac{T_b}{T_{c,wf}})}{ex_5 - ex_1}$
Preheater	$\eta_{I,Preheater} = \frac{h_3 - h_2}{q_{jw}}$	$\eta_{II,preheater} = \frac{ex_3 - ex_2}{q_{jw}(1 - \frac{T_{s,wf}}{T_{s,jw}})}$
Overall System	$\eta_{I,overall} = \frac{w_{net}}{q_{evaporator}}$	$\eta_{II,overall} = \frac{w_{net}}{w_{rv}}$

RESULTS AND DISCUSSION

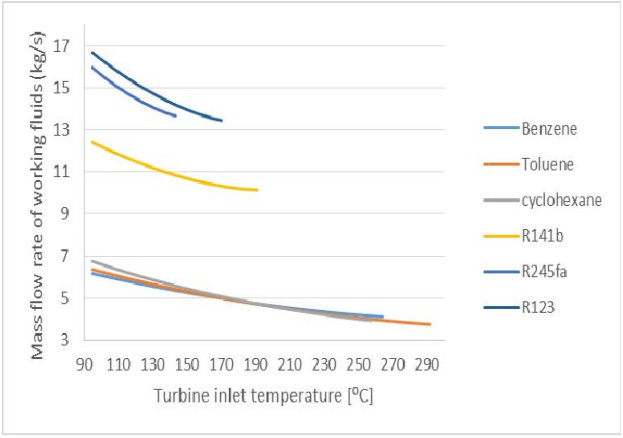
In this chapter, the thermodynamic performance of the ORC system is assessed based on the proposed model, by a script written in EES software. The thermal efficiency, exergy efficiency, net power output and so on are calculated for the system with different working fluids. The ORC system is investigated for 6 different fluids which are benzene, toluene, cyclohexane, R141b, R245fa and R123 under varies operating conditions. The results of this simulations can provide guidance in order to deciding of a suitable working fluid for the ORC system.

The first aim of the results should be to understand how thermal efficiency of the ORC system affected by turbine inlet temperature. The turbine inlet temperature of the each fluid is limited to their critical temperature.

According to Figure 2 (a) the thermal efficiency of the ORC system increases with increment of the turbine inlet temperature for all working fluids. As expected for the same turbine inlet temperature benzene shows higher achievable efficiencies with. However due to toluene has the highest critical temperature, it can achieve to maximum thermal efficiency of % 21.6 which is slightly higher than benzene with % 21.1.



(a)



(b)

Figure 3 Variation of thermal efficiency (a) and mass flow rate working fluid (b) with turbine inlet tempereture

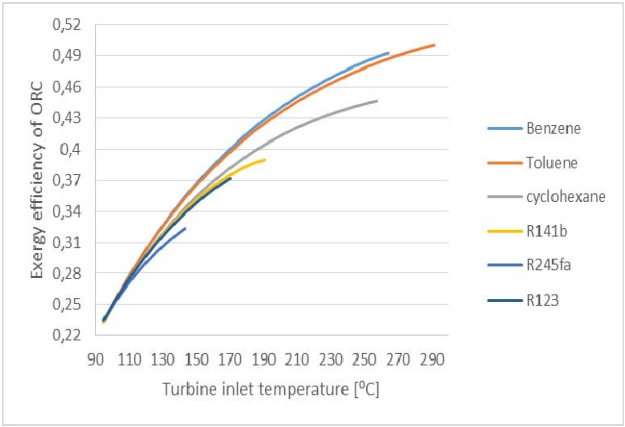
In figure 2 (b) assessed value of mass flow rate of working fluids are plotted with detail to the temperature at turbine inlet. It can be note that benzene, toluene and cyclohexane needs lower mass flow rate as a result of the highest enthalpy rise between state 3 and 4. In order to recover maximum energy of the flue gasses, it is expected that the flow rate of R123, R245fa and R141b fluids are high. Figure 4 (a) illustrate variation of turbine inlet pressure on the exergy efficiency of ORC for different fluids. The trend of exergy efficiency is same as the thermal energy.

The simulation results are presented in Figure 3 (b), which shows the ORC net power output under different turbine inlet temperature. Higher turbine inlet temperature means that more net power output can be obtained with variable jacket water temperature and same mass flow rate of exhaust gases. Among all the considered working fluids, Toluene shows the highest net power output value of about 600 kW at turbine inlet temperature 291°C. At the same turbine inlet temperature benzene presents higher net power output value of 588 kW at 288 °C. Moreover net power output of ORC for all working fluids are almost same until turbine inlet temperature 150 °C.

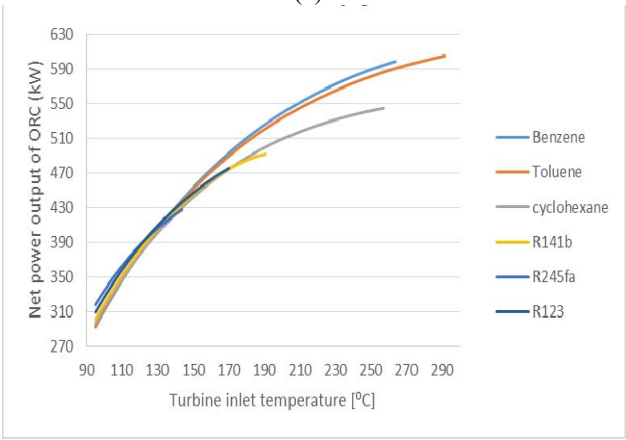
While toluene shows the highest exergy performance with 50.05 %, R245fa has the lowest exergy efficiency with 32.35% which correspond to 143.3 oC turbine inlet temperature.

The exergy efficiency of the different ORC components as a function of the temperature with benzene is presented in Figure 3 (d). It can be seen that the highest exergy loss occur in the evaporatofollowed by the preheater. Furthermore the exergy destruction of the turbine and pump are noticeably lower than the heat exchanger equipment.

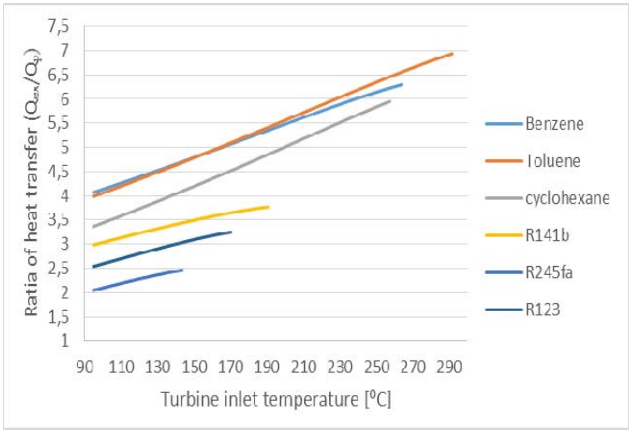
The ratio of the heat transfer of evaporator and preheater under ICE full load are indicated in Figure 3 (c). İncreasing of turbine inlet temperature result in slightly increase in the Qex/ Qp ratio for all working fluid. Since the preheater outlet temperature at working fluid side is kept constant at 88 degrees, the energy recovered from the exhaust gas is the key parameter to effect the ratio. Therefore the toluene with the highest critical temperature is the most energy recovering working fluid from the exhaust gas.



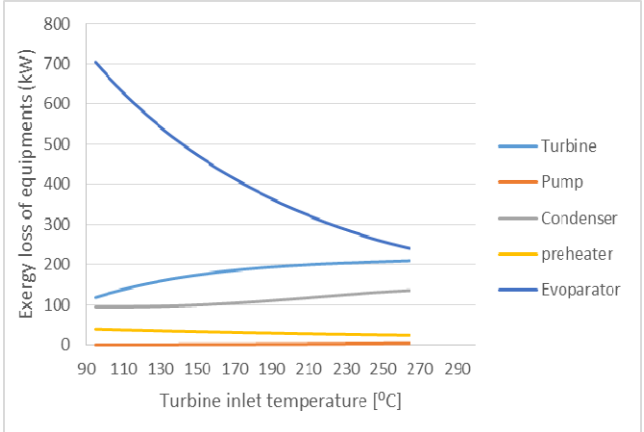
(a)



(b)



(c)



(d)

Figure 4 Variation of exergy efficiency (a), ratio of heat transfer (b), net power output of ORC (c), exergy destruction of equipments (d)

CONCLUSIONS

In this study, thermodynamic modeling of novel single loop ORC system for heat recovery from an ICEs exhaust gas and jacket water. Comprehensive thermodynamic analyses has been conducted and some of the main concluding remarks are as follows:

- The highest exergy loses rate was found to occur in the evaporator of the ORC.
- While toluene and benzene were found to have the highest performance among all working fluids, R123 and R245fa was required high pumping power and showed lower thermal efficiency.
- The recovering heat from in exhaust gas was higher than heat in jacket water for all working fluids. The rate of was Qex/Qp >1 for whole simulations.
- Toluene and benzene always presents better performance than other fluids and minor difference with each other. The ORC system gets maximum net power output of 604.9 kW and 598.5 kW and maximum thermal efficiency of 21.6%and 21.1% for toluene and benzene.

NOMENCLATURE

ORC Organic Rankine Cycle
ICE Internal Combustion Engine
GWP Global Warming Potential[100 years]
ODP Ozone Depletion Potential
$T_{ex,in}$ Engine exhaust gas inlet temperature to evaporator [°C]
$T_{ex,out}$ Engine exhaust gas outlet temperature to evaporator [°C]
$T_{c,wf}$ Condenser surface temperature for working fluid side [°C]
$T_{s,wf}$ Evaporator surface temperature for working fluid side [°C]
$T_{s,ex}$ Evaporator surface temperature for exhaust gas side [°C]
e_{xi} Exergy value of statement points [kj/kg]
Exd,i Exergy destruction rate of equipments [kj/kg]
ExQi Exergy of heat intake [kj/kg]
s_i Entropy value of statement points [kj/kg°C]
$S_{gen,i}$ Total entropy production of components
P_i Pressure value of statement points [kPa]
η_I Thermal efficiency
η_{II} Second law efficiency
η_p Isentropic efficiency of pump
η_t isentropic efficiency of türbine
Q_{con} Engine fuel energy consumption [kW]
$Q_{exhaust}$ Total heat intake from engine exhaust gas [kW]
W_{net} Net output power of türbine [kW]
$W_{turbine}$ Power outlet of türbine [kW]
W_{rv} Total reversible work [kW]
WpumpTotal work for pump [kW]
m_{ex} Mass flow rate of engine exhaust gas (kg/s)
m_{wf} Mass flow rate of organic fluid (kg/s)
m_i Mass flow rate of statement points (kg/s)
c_p Specific heat [kj/kg°C]

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