Thermodynamic Analyses and Optimization Study Of Organic Rankine Cycle Usage with Internal Combustion Engine Waste Heat

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Abstract: *This paper presents a organic Rankine Cycle (ORC) usage analyze as a heat recovery system for exhaust gas heat loss of Internal Combustion Engine (ICE) which is TCG2032 model manufactured by MWM brand of Caterpillar Energy Solutions GmbH. Mathematical model was developed for basic ORC and depending on seven different working fluids which are R113, R141b, R134a, R123, R245fa, n-pentane and n-butane to represent different fundamental classes. The maximum net power output is the foremost evaluation criterion to analyze heat recovery system effectiveness. The thermal analysis results reveal that the R141b is the most efficient working fluid. R134a is one of the wet type fluid and it has the lowest net output power as comparatively from the other selected fluids. Also analysis shows that the 6-14% increase of the overall efficiency appear possible with ICE-ORC combined cycle in according to single engine, and as a result of that 400-700 kW reduce can be achieved for the fuel energy consumption..*

Index Terms: *Organic Rankine Cycle, Internal Combustion Engine Heat Recovery, Energy and Exergy Analysis of Organic Rankine Cycle, Optimization Study of Organic Rankine Cycle.*

Introduction

One of the most important reasons of pollution is fossil fuel usage increment depending on population growth. Fossil fuels have widely utilization area such as application of industrial production, climatization, electrical energy production, agricultural machinery, transportation. Internal combustion engines (ICEs) are most known mechanical energy generator and because of that, it is major consumer of fossil fuels [1].

ICEs are being used for electrical energy production from mechanical energy with using integrated generator and a typical ICE can produce average 25% useful work, and approximately 40% energy loses with exhaust gas waste heat, friction and engine block cooling [2]. Waste heat recovery systems are more important with increasing pollution depending on increment of fossil fuel usage as a response for growth population needs. [3]. Using a secondary cycle such as Organic Rankine Cycle (ORC) which is use waste heat of ICE for heat source is acceptable as heat recovery system. Integration of ORC to ICE improves overall system efficiency and reduces emissions by generating additional power without using fossil fuel resources [4]. Organic rankine cycle can be preferred against rankine cycle which is use water as working fluid for small scale power plants by reason of reduces working pressure. ORC working fluids have lower boiling temperature and high vapor pressures against water at same temperature and as a reason of that it cause to increase temperature difference between heat source and evaporator [5]. Also increasing temperature difference between heat source and evaporator has positive effects to heat exchanger

size by higher heat flux. R141b and R11 was found slightly more efficient than R113, R123, R245fa with single stage cycle by Wang et al. [6]. All different working fluids must have different working state points based on its chemical and physical parameters [7]. ORC system exergy efficiencies range was found 32% to 38% and the R123 and R141b have highest exergy efficiencies dependşng on results of Asim et al. [7] research. Xu and Yu[8] presented working fluid selection for subcritical pressure ORC depending on critical temperature criterion and proposed R245fa and R141b because of its can be used wide heat source temperature range. Imran et al.[9] examined thermo-economic analysis of basic ORC, single stage ORC and double stage ORC for five different working fluids such as R245fa, R11, R123, R134a, R141b and shows that the R245fa has lowest specific investment cost and using single stage ORC against basic ORC improve system efficiency 1.01% with an additional 187 $/kW cost, same situation happens between basic and double stage cycle improve efficiency 1.45% with 297 $/kW additional cost. He et al.[10] examined comparatively 21 different working fluid for determine optimum evaporation temperature according to maximum net output power of single stage ORC and the result of this research is R114, R245fa, R123, R601a,n-pentane, R141b and R113 are most suitable working fluids at 423 °K waste heat resource temperature. Wang et al.[11] generate a selection chart for the most efficient working fluids depending on heat source temperature as a comparison 25 different working fluids with using single stage ORC. And also they found the evaporator has the higher exergy destruction rate.

In this study, basic Organic Rankine Cycle with different type working fluids was examined comparatively for thermal efficiency, second law efficiency and fuel energy save rates. Seven different working fluids were used in this paper as represent different classes of fluids, for determine the optimal statement points for maximize efficiency.

System Description

Internal combustion engine which is heat recovery system will be applied, has three different working levels such as 50%, 75% and 100%. Exhaust gas conditions and engine parametric terms which are shared by the manufacturer were presented as Table.1 for each working level. Exhaust gas composition and specific heat for average temperature were presented at Table.2.

Table 1 *ICE Working Parameters for Different Engine Loads*

Property	Unit	Value	Value	Value
Engine Load	%	50	75	100
Exhaust Temperature	°C	510	480	453
Exhaust Mass Flow	kg/h	12268	17397	22832
Fuel Energy Consumption ($Q_{e,con}$)	kW	5486	7630	9824
Thermal Efficiency	%	46,9	44,6	43,8
Electricity Production Efficiency($\eta_{e,epe}$)	%	39,2	42,3	43,3

Table 2 *Composition of Exhaust Gas*

Composition	Volumetric Ratio (%)	Molecular Weight (kg/kmol)	Spesific Heat (kj/kg°C)
Argon (Ar)	0,1	40	0,5206
Carbondioxide (CO2)	4,9	44	1,061
Water (H2O)	11,2	18	2,013
Oxygen (O2)	9,9	32	0,993
Nitrogen (N2)	73,9	28	1,07

Basic Organic Rankine Cycle (BORC) consists of four components which are pump, evaporator, condenser and türbine as shown at Figure.1. Cycle loop begin with pump which is the only moving part of cycle. Pump pressurized the liquid and provides liquid supply to evaporator. Evaporator vaporized the liquid until working fluid reaches saturated vapor or super-heated vapor depending on design. After that, turbine produces power by expanding the vapor to condenser pressure. Expanded vapor goes into condenser and phase change occurs at constant pressure with heat rejection. Liquid outlet of condenser goes into pump and by this way; closed loop cycle was obtained for BORC.

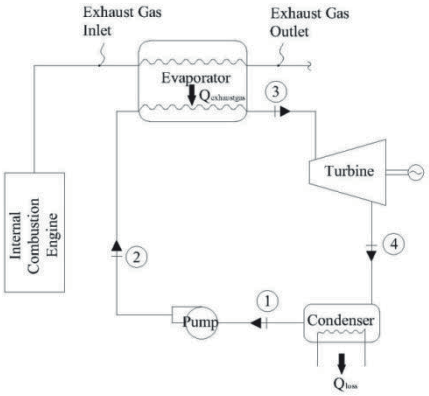


Figure 1 *Basic Organic Rankine Cycle*

The most significant design parameter for a power plant is heat source capacity. In this study, exhaust gas of an IHE was used as heat source and exhaust gas contents water vapor and CO2. Funnel wall or evaporator outside surface can be damaged with acidic water generation in case of exhaust gas condensation. Evaporator was designed based on 120 °C exhaust gas outlet temperature and by this way, it is assumed that there is no acidic condensation at funnel or evaporator. Exhaust gas capacity was found depending on temperature difference between inlet and outlet with using specific heat and mass flow rate. Specific heat of exhaust gas was calculated with the contents mass ratio which is presented at Table.2 for average temperature. Average temperature of exhaust gas assumed 300 °C and specific heat was calculated 1.116 kj/kg°C. Pump has to be pressurized liquid to under subcritical pressure of working fluid. Risks of reaching critical pressure were expressed in detail at next steps of study. Heat sink of ORC is coolant water which is produce by cooling tower. Coolant water temperature regime accepted as 20-30 °C by considering summer conditions. Because of that, condenser outlet temperature assumed 35 °C. Also it is assumed that the condenser outlet is saturated liquid which constitutes another limitation parameter such as turbine expansion pressure. There are various isentropic efficiencies which are in range between 65%-75% for turbine and in range between 75%-85% for pumps depending on

literature [2][5]. Isentropic efficiency was assumed 65% at turbine and 75% at pump for more realistic approach. And also all assumptions are express as follows:

- Exhaust gas evaporator outlet temperature: 120 °C
- Specific heat of the exhaust gas (cp): 1,116 kj/kg°C
- Condenser outlet temperature: 35 °C
- Condenser outlet is saturated liquid.
- Isentropic efficiency of pump: 0,75
- Combined Isentropic and generator efficiency of turbine: 0,65
- Pressure drop in pipe, evaporator, condenser and IHE was neglected.
- Components were accepted adiabatic.
- The system operates steady state.

Working fluid selection is directly effective to ORC system design and feasibility researches. Every working fluid has different physical and chemical characteristics which are highly important for environmental based terms and beside that, also thermodynamic characteristics are highly important for cycle type, components design, system maximum efficiency and optimization. First of all, there are basically three different organic fluids types which are dry, isentropic and wet type depending on saturation curve slope. Working fluids were observed for chemically conditions (stability under operating conditions, non-flammable, non-explosive, non-toxic), environmentally conditions (low GWP, low ODP) and thermodynamically conditions (efficient, suitable to conditions) as presented at Table 3.

Table 3 *Working Fluid Properties*

Working Fluid	Formula	Molecule Weight (kg/kmol)	Tcrit[a] (°C)	Pcrit[b] (kPa)	ALT[c]	GWP[d]	ODP[e]	ASHRAE 34 Classification
n-pentane	C5H12	72,15	196,56	3370	12	4	0	A3
n-butane	C4H10	58,12	152,01	3796	12	3	0	A3
R113	C2F3Cl3	187,38	214,06	3392	85	6130	0,9	A1
R123	C2HF3Cl2	152,93	183,83	3672	1,3	120	0,012	B1
R141b	C2H3FCl2	116,95	206,96	4212	9,3	700	0,086	A2
R245fa	C3H3F5	134,05	154,20	3651	7,2	950	0	B1
R134a	C2H2F4	102,03	101,06	4059	14	1300	0	A1

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

Listed working fluids at Table 3 were chosen for next steps of study as the most commercial and most popular fluids depending on literature research. Combined T-S diagram of selected working fluid shown at Figure 2.

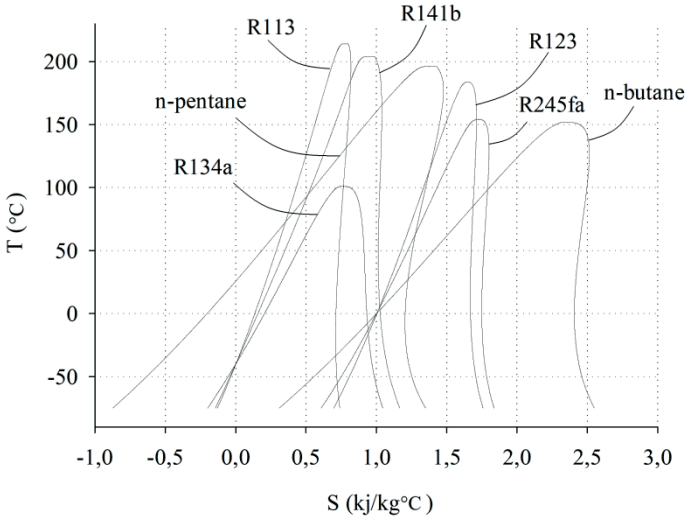


Figure 2 *T-S diagrams of selected working fluids*

Mathematical Model

Thermodynamic equations constitute the base of all mathematical models, balance equations and efficiency terms calculations. Enthalpy and exergy values are also base for efficiency terms. Enthalpy and exergy values were found for all state points with using temperature and pressure. Temperatures and pressures of all statement points are known depending on assumptions which are mentioned at design parameters and independent variables which will be specified at optimization step.

Waste heat of engine can be expressed as follows:

$$\dot{Q}_{exhaust} = \dot{m}_{exhaust} c_p (T_{ei} - T_{eo}) \tag{1}$$

Working fluid mass flow rate can be expressed as follows:

$$\dot{m}_{wf} = \frac{\dot{Q}_{exhaust}}{q_{evaporator}} \tag{2}$$

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

$$\dot{Q}_{evaporator} = \dot{m}_{exhaust} \left(h_{oe} - h_{ie} \right) \tag{3}$$

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

$$ex_i = (h_i - h_0) - T_0 (s_i - s_0) \tag{4}$$

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

$$\dot{E}x_d = T_0 \dot{S}_{gen} \tag{5}$$

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

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

Net output power of turbine can be expressed as follows:

$$\dot{W}_{net} = \dot{W}_{tubine} - \dot{W}_{pump} \tag{7}$$

Net electricity production of ICE can be expressed as follows:

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

Thermal efficiency and second law efficiencies are important deterministic criteria for optimization of cycle. On the other hand, ICE and ORC combined system overall efficiency terms is most significant output parameter of the system. Feasibility research was comprised depending on cogeneration system overall recovery terms.

Overall efficiency of cogeneration system can be expressed as follows:

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

Reduction of ICE fuel energy consumption can be expressed as follows:

$$\dot{Q}_{e,red} = \dot{Q}_{e,con} - \frac{\dot{W}_{need}}{\eta_{cogen}} \tag{10}$$

Reversible work is represents the maximum power output which could be provided at working conditions. It can be expressed as follows:

$$\dot{W}_{rv} = \dot{Q}_{loss}(1 - \frac{T_b}{T_{s,ex}}) \tag{11}$$

Cogeneration heat recovery efficiency is an indication that shows system heat recovery amount how close to the maximum possible heat recovery amount. Cogeneration system heat recovery efficiency can be expressed as follows:

$$\eta_{hrv} = \frac{\dot{W}_{net} + \dot{W}_{need}}{\dot{W}_{rv} + \dot{W}_{need}} \tag{12}$$

Optimization

In this section, thermal output parameters were comparatively examined to identify optimal independent state point conditions such as evaporating pressure and superheat temperature. Mathematical model was developed with all assumptions and limitations by using Energy Equation Solver (EES) and also it is used for all calculations.

As seen at Figure.4 (a) and (b) increasing of evaporating pressure has positive effect to cycle's thermal output parameters. But there is a critical point limitation for all working fluids as mentioned before. It is hazardous that the pass over critical pressure for delicate equipment such as turbine because of there is no vapor-liquid equilibrium above this point. It will be logical to choose different optimum evaporator pressures for all working fluids depending on its thermophysical properties. If error margin of real systems are taking consideration, it is dangerous to approach critical pressure. The other fact remains that the increase rates of thermal output parameters are decreasing with approaching critical pressure. The result of that, it is proper to choose 90 percent of critical pressures' as evaporating pressures for all working fluids.

The other independent parameter is superheat temperature has to be specified for optimal working condition. As shown at Figure.5 (a) and (b), Increasing of super heat temperature has no appreciable positive effect for the Basic ORC after the 10 °C point. Decision of optimal superheat temperature has decomposition risk separately from thermal outputs. Fluids become useless by time because of decomposition rate which is the change rate of fluid chemical structure.

Decomposition rate increases with the increment of temperature [12]. The result of that, it is proper to choose 10 °C as superheat temperature.

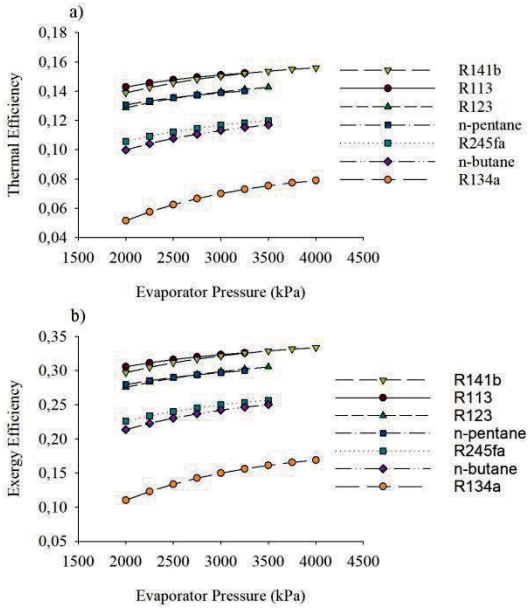


Figure 4 *Pressure based thermal parameters*

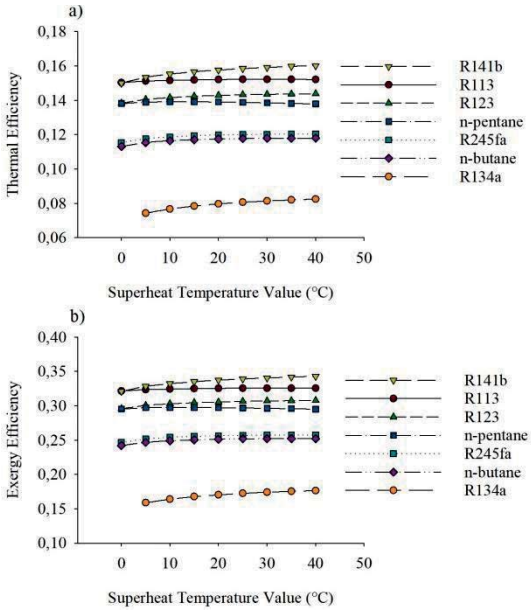


Figure 5 *Temperature based thermal parameters*

Figure 6 generated for n-pentane based cycle. Figure 6(a) represents that the second law efficiency changes of components depending on different evaporator pressure. Figure 6(b) shows that the second law efficiency changes of components depending on different superheat temperatures. Condenser second law efficiency is reducing with increment of superheat temperature. The reason of that, condenser heat loss is increasing with increment of super heat temperature.

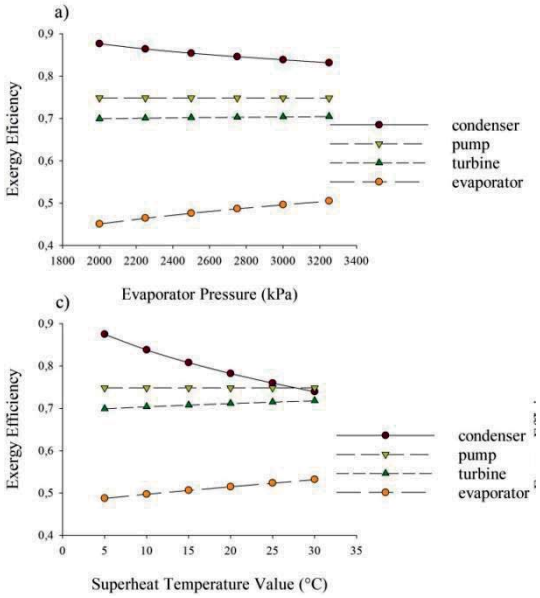


Figure 6 *Second law efficiencies of components*

Results and Discussion

In this section, working fluids were comparatively examined for thermodynamic results and overall system results. Also different engine loads examined for most efficient working conditions of fluids which are specified at optimization phase. Figure 7(a) represents net turbine output power of all selected working fluids for Basic ORC with different engine loads. The cogeneration system efficiency change of all working fluids with basic ORC presented as Figure 7(b) for different engine loads. As seen at Figure 7(b) cogeneration efficiency is increasing averagely 8 percent with increment of engine load from 50% to 75% besides that, it is increasing averagely 4 percent with increment of engine load from 75% to 100%. The reason of that, engine is more efficient at higher load levels depending on data which are published by manufacturer. Cogeneration system overall efficiency is 6-11% higher than the single engine at 100% engine load depending on working fluid type. In addition to this, the overall efficiency of cogeneration system is 4-10% higher at 75% engine load and 6-10% higher at 50% engine load.

Figure 8(a) shows the fuel usage reduction of engine for different working fluid usage depending on different engine loads. Also, results which are shown at Figure 8(b) will be basis of economical analyses of cogeneration system. Reduction of fuel consumption is increasing with different increment rates between different working fluids depending on different engine loads. For example, while R113 reduction of fuel consumption is increasing with rate of 30%, R134a was increased with rate of 25% between 50% and 100% engine load. Simultaneously, cogeneration efficiencies increment has same increment slope for different working fluids. The reason of that, reduction of fuel usage is a directly dependent result to ICE total electricity production capacity and net output power of cycle apart from cycle efficiency. Cycle thermal efficiency is indirectly effective to reducing fuel energy consumption.

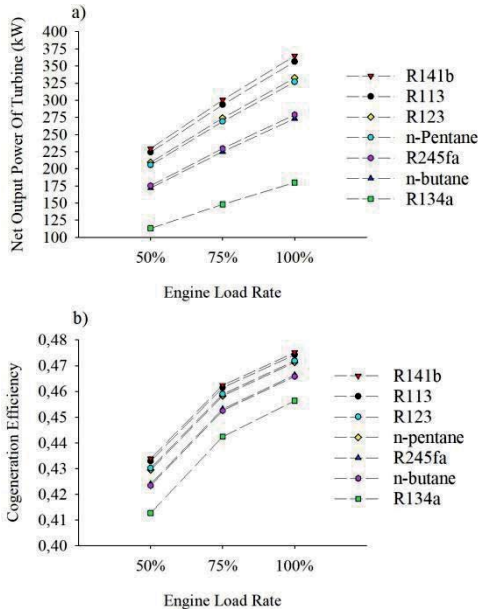


Figure 7 *Maximum outputs for different engine loads*

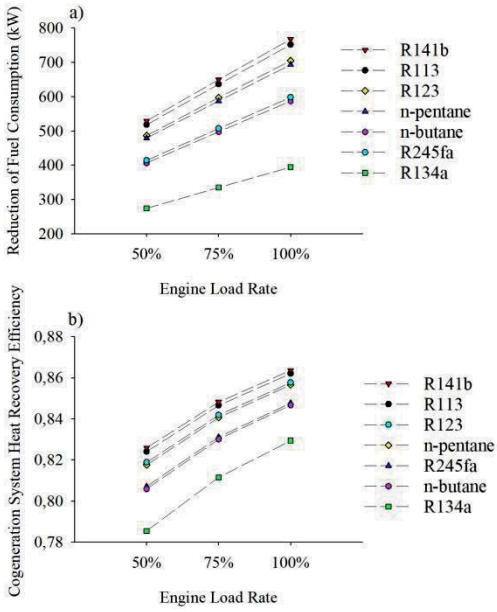


Figure 8 *Cogeneration outputs for different engine loads*

Conclusion

Purpose of this study is research basic ORC with different working fluids as a heat recovery system for exhaust gas of natural gas powered internal combustion engine which is TCG2032 model manufactured by MWM brand of Caterpillar Energy Solutions GmbH. The determinative criterion for heat recovery system effectiveness is reduction of fuel energy usage of engine which is relative to maximum net output power of turbine. Optimization study was made for each working fluid to maximize net output power of turbine with using thermal parameters. R113, R134a, R141b, R123, R245fa, n-pentane and n-butane were investigated as working fluids. The major conclusions of research are express as follows:

- R141b is the most efficient working fluid its net turbine output power higher than other selected fluids. R141b could provide 767 kW fuel energy save with cogeneration efficiency that 48% at 100% engine load. Beside that R141b has the highest evaporator outlet temperature.
- R134a is the worst working fluid for heat recovery system depending on thermodynamic results. Fuel energy usage could be reduced 394kW with using R134a based basic ORC.
- Environmental parameters of working fluids at least as important as thermal parameters. R123 has the lowest ALT value and it could save 705 kW fuel energy.
- The lowest GWP value belongs to n-butane which is save 587 kW fuel energy with basic ORC.
- Different optimal thermal working condition must be defined for each working fluid depending on its thermal and chemical properties to maximize turbine net output work.
- ORC exergy efficiencies are increasing with engine load level increment. The reason of that, exhaust gas temperature is reducing with exhaust mass flow rate increment at higher load level.

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