Analysis of Operating Parameters of Internal Combustion Engines with Waste Heat Recovery using Two-Stage Organic Rakine Cycle

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Abstract: In this paper, the amount of heat wasted from different parts of a 12-liter compression ignition engine and the recoverable heat from these parts were investigated. Then, two-stage configuration of organic Rankine cycle was introduced for simultaneous heat recovery from exhaust gases and coolant. Finally, parameters such as hybrid generated power, engine thermal efficiency, and brake specific fuel consumption were studied at different engine speeds under full engine load. By adopting this method, 35 kW hybrid power can be generated consequently, causing 9.5% reduction in brake specific fuel consumption.

Keywords: Cogeneration, Internal Combustion Engine, Two-Stage Organic Rankine Cycle, Waste Heat Recovery

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1 INTRODUCTION

The interest for low-grade waste heat recovery has been growing during the last decade, due to the increasing concern over energy shortage and global warming. Because of engine heat limitations, a significant amount of fuel energy is wasted as heat, just one third of fuel energy is converted to mechanical power and the rest is released in the environment as heat. In real process, the engine does not work with its maximum performance as well. Therefore, amount of wasted heat increases [1].

For instance, in a spark ignition ICE with 15 to 32% thermal efficiency, 1.7 to 45 kW is released through the radiator and 4.6 to 120 kW is released by exhaust gas [2]. Different methods have been presented for waste heat recovery of ICE including absorption cooling system, thermoelectric system, and organic Rankine cycle (ORC).

Waste heat recovery using ORC is proposed as a high efficient method in comparison with the other methods such as thermoelectricity and absorption refrigeration cycle used for air conditioning. To increase automobile efficiency, the mentioned method is used recently by some of automobile manufactures [3-5]. In the recent years, several researches have been conducted about using different working fluids and configurations and modifying the system components [6-12].

In 2010, Tahir, et al. [13] investigated the influence of using vane expander on efficiency of a compact ORC for low temperature waste heat recovery. They reached the output power of about 30 W with 4% thermal efficiency. The hot source temperature range was 60 to 100°C and the cold source temperature range was 10 to 30°C.

Vaja, et al. [14] studied different working fluids and their influence on ORC efficiency. They found that using ORC as a second producer in ICEs increases the overall heat efficiency up to 12%. They concluded that simultaneous and complete waste heat recovery from cooling system and exhaust gas is impossible, and just a part of heat wasted in engine cooling system can be recovered. The maximum amount of recoverable heat from hot source was 57.9% of heat released from cooling system. It was achieved by selecting R134a as working fluid. Using other fluids such as R11 and Benzene caused lower heat recovery.

A variety of configurations including simple cycling, pre-heating and recovery were considered to recover the wasted heat. Pre-heating and recovery cycles were intended to improve the efficiency of simple cycling. However, previous studies suggested that increasing production capacity is more useful than enhancing efficiency of waste heat recovery [15]. The simultaneous heat recovery from engine cooling system and exhaust gases are investigated in all previous works as well. Some of these works studied also heat recovery only from exhaust gases or engine cooling system. Tahani et al. (2010) showed that there are two configurations and two storeys for simultaneous heat recovery from engine cooling system and exhaust gases. Enjoying higher production capacity than the corresponding value of pre-heating cycling, the two-storey configuration was considered optimal. In addition, heat recovery from engine cooling system was impossible [16].

After having computed the amount of waste heat, the values of recovered heat, generated power, thermal efficiency and Brake Specific Fuel Consumption (BSFC) were investigated. Calculations were performed for engine speed range of 800 to 1900 RPM. The power generated by these systems was converted to electricity and can be used to remove parasitic loads from the engine. Thus, an electric generator with efficiency of 95% was coupled with the expander. CATT 2® and SCOPE® were used to obtain the thermodynamic properties of working fluid (R11) and engine characteristics curve, respectively.

2 ICE STUDIED IN THIS WORK

In this study, a commercial engine applied in vehicles was used for heat recovery. It was a 11.7-liter compression ignition six-cylinder engine with linear arrangement. The tests were performed under engine load of 100% at engine speeds ranging from 800 to 1900 rpm. The engine fuel was gas oil. The related technical characteristics were presented in table 1.

Table 1 Main engine characteristics

Parameter	Value
Engine Type (model)	6-cylinder, Linear, (DC12 06)
Max. output power (kW)	310
Exhaust gas temp. (°C)	388-565
Exhaust gas flow (kg/s)	0.13-0.49
Engine jacket temp. (°C)	80
Engine jacket flow (kg/s)	1.13-4.16

3 COMPUTATION OF WASTE HEAT IN DIFFERENT PARTS OF ICE

In order to recover waste heat from an ICE, it is necessary first to get information about the temperature and thermal energy levels of its various sources of wasting thermal energy. In an ICE, a great portion of fuel energy is released to the ambient through the exhaust gases and coolant. Remaining fuel energy is wasted by the lubrication system and charged air. In the following steps, the thermal energy wasted by coolant and exhaust gases were computed and compared with the other wasting sources.

Cooling system

To compute the amount of waste heat in engine cooling system, mass flow rate and required temperature change of cooling fluid should be known. The amount of waste power in cooling system was:

$$\dot{Q}_{\mathrm{clnt}} = \dot{m}_{\mathrm{cf}} \bar{C}_{\mathrm{p\ cf}}\, \Delta T \qquad (1)$$

The engine cooling fluid was ethylene glycol 50% with specific heat capacity of 3.3 kJ/(kg K) [15]. The required temperature change of cooling fluid was considered as 5°C of course according to the engine manufacturer recommendation for optimum operation.

Exhaust Gas

To compute the heat wasted through exhaust gases, mass flow rate and temperature of exhaust gas after turbo charger should be known versus engine speed. The value of specific heat capacity was determined based on the exhaust gas composition. In this study, T_{amb} the ambient temperature was 25°C. Released energy by exhaust gas was obtained from:

$$\dot{Q}_{\mathrm{exh}} = \dot{m}_{\mathrm{exh}} \bar{C}_{\mathrm{p\ exh}}\, (T_{\mathrm{exh}} - T_{\mathrm{amb}}) \qquad (2)$$

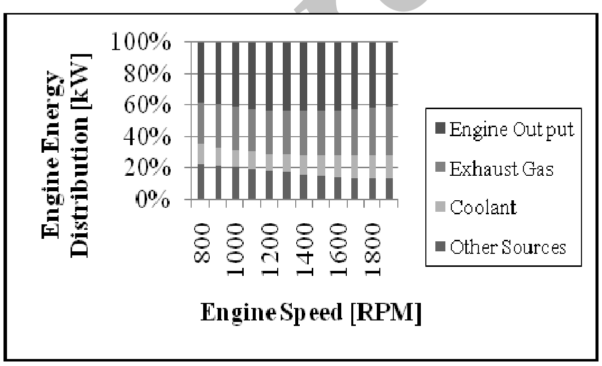

Fig. 1 Released energy by coolant, exhaust gas, engine output power and the other sources vs. engine speed

The fuel energy distribution in various sections of the studied engine versus engine speed was shown in Fig. 1. As observed, a noticeable part of fuel energy was released to the environment through exhaust gases. This part was approximately constant at various engine speeds. Considering its high temperature and energy levels, waste heat recovery was recommended from this resource. Also, the engine speed rise increases the part of wasted power by coolant and decreases the part of wasted power by the other waste resources (lubricant, charged air). Thus, heat recovery from coolant was recommended in comparison with the other waste resources. Although coolant bears high energy level at engine speeds close to the speed with maximum nominal power, its low temperature results in a few limitations. Its energy accessibility is particularly very low compared with the exhaust gases.

4 TWO-STAGE ORGANIC RANKINE CYCLE DESCRIPTION

Using the appropriate configuration according to hot source is one of the solutions to increase the Rankine cycle output. Various configurations have been presented to recover the heat wasted from an internal combustion engine using Rankine cycle. As a result, recovered heat and generated power get changed based on their properties. Two-stage configuration, shown in figure 2, has been used rarely to recover waste heat from an ICE.

In this system, two pumps and one turbo expander were used. The applied turbo expander is called dual expander containing two inlets and one outlet. As shown in Fig. 2, in this configuration working fluid flows through two different stages with different mass flow rates after leaving condenser: first to recover heat from engine cooling system and then to recover waste heat from exhaust gas.

In thermodynamic analysis of waste heat recovery by Rankine cycle, because of a cost-free energy, the configurations with higher power generation is required. The most important feature of the present configuration is its capability to recover the whole wasted heat in cooling system, compared with the pre-heater configuration which recovers only a small part of the wasted heat. The comparison between two-stage and pre-heating configurations was carried out by Tahani et al. [16].

5 SELECTION OF THE WORKING FLUID

The choice of working fluid for a given application is of high significance and has been treated in numerous studies. The following are some general relevant characteristics extracted from the mentioned studies [2]:

1. Thermodynamic performance: The efficiency and/or output power should be as high as possible for the given heat source and heat sink temperatures. This generally involves low pump consumption and high critical point.

2. Positive or isentropic saturation vapor curve: A negative saturation vapor curve ("Wet" fluid) results in forming droplets at the end of expansion process. The vapor must therefore be superheated at the turbine inlet in order to avoid turbine damages which decrease cycle performance.

In the case of positive saturation vapor curve (dry fluid), a recuperator can be used to increase cycle efficiency.

3. High vapor density: This parameter enjoys high importance especially for fluids with a very low condensing pressure. A low density leads to very large equipment at the expander and condenser level.

4. Acceptable pressures: As already stated for water, high pressure values usually increase investment costs and complexity.

5. High stability temperature: Unlike water, organic fluids usually suffer from chemical deteriorations and decomposition at high temperatures. The maximum heat source temperature was therefore limited by chemical stability of the working fluid.

6. Low environmental impact and high safety level: The main parameters to take into account were the Ozone Depleting Potential (ODP), the Greenhouse Warming Potential (GWP), the toxicity and the flammability.

7. Good availability and low cost: According to the mentioned criteria, R-11 was selected as the working fluid with characteristics summarized in table 2.

Table 2 Working fluid (R-11) main characteristics

Parameter	Value
Chemical formula	Trichlorofluoro-methane
Critical temp. (°C)	197.96
Critical pre. (MPa)	4.4
Ozone depleting potential	1
Global warming potential	4600

The relevant thermodynamic analysis is presented in the following sections. Some required assumptions were stated also in table 3.

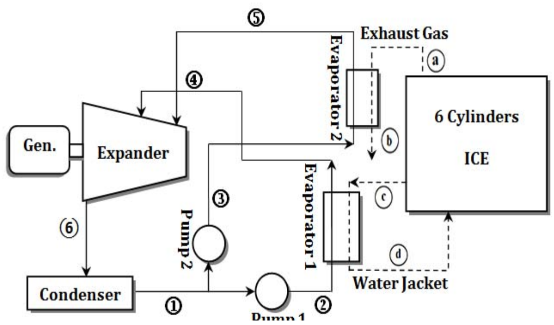

Fig. 2 Schematic of an ICE, bottoming with Two-Stage Rankine cycle to recover engine waste heat

Table 3 Assumptions made in Two-Stage cycle

Parameter	value
Condenser temp. (°C)	35
Evaporator #1 pre. (kPa)	311
Evaporator #2 pre. (kPa)	3835
Expander efficiency (%)	80
Evaporator #2 effectiveness (%)	50-80

6 THERMODYNAMIC ANALYSIS OF AN ORC COMBINED WITH ICE

In this system, there are two evaporators with two different pressures, where the first law of thermodynamics was written for them. The required mass flow rate was computed for the evaporators and the generated power and cycle efficiency were studied. Evaporator No. 1 can recover cooling system waste heat. The maximum energy exchanged in these heat exchangers by passing engine coolant and cycle working fluid through them was a key parameter considering inlet temperature values. For this purpose, the following equation was considered [17]:

$$\dot{Q}_{clnt\ max} = \dot{m}_{\mathrm{cf}}\bar{C}_{\mathrm{p\,cf}}(T_{\mathrm{c}} - T_2) \qquad (3)$$

On the other hand, the required effectiveness for this heat exchanger was computed from [17]:

$$\varepsilon_1 = \frac{q}{q_{\max}} = \frac{T_{\mathrm{c}} - T_{\mathrm{d}}}{T_{\mathrm{c}} - T_2} \qquad (4)$$

Considering Eq. (4), required temperature reduction for engine cooling fluid was computed and the total heat wasted from cooling system can be recovered by a heat exchanger with effectiveness of 40%. To determine the required mass flow rate for cooling system heat recovery, the first law of thermodynamics was written for evaporator No. 1:

$$\dot{m}_{\mathrm{f1}} = \frac{\dot{m}_{\mathrm{cf}}\bar{C}_{\mathrm{p\,cf}}\,(T_{\mathrm{c}} - T_{\mathrm{d}})}{(h_4 - h_2)} \tag{5}$$

Where h was specific enthalpy of working fluid. In addition, maximum exchangeable energy from exhaust gas was estimated as follows [14]:

$$\dot{Q}_{\mathrm{exh\,max}} = \dot{m}_{\mathrm{exh}}\bar{C}_{\mathrm{p\,exh}}(T_{\mathrm{a}} - T_3) \tag{6}$$

Effectiveness of evaporator No. 2 was assumed to vary between 50 to 80%. So actual recovered energy in this heat exchanger was estimated from [14]:

$$\dot{Q}_{\mathrm{exh\,ac}} = \varepsilon_2 \dot{Q}_{\mathrm{max\,exh}} = \dot{m}_{\mathrm{exh}}\bar{C}_{\mathrm{p\,exh}}(T_{\mathrm{a}} - T_{\mathrm{b}}) \tag{7}$$

By applying the first law of thermodynamics for evaporator No. 2, the required mass flow rate during heat recovery from exhaust gases was determined as:

$$\dot{m}_{f2} = \frac{\dot{m}_{\mathrm{exh}}\bar{C}_{\mathrm{p\,exh}}(T_{\mathrm{a}} - T_{\mathrm{b}})}{(h_5 - h_3)} \tag{8}$$

Mechanical output power was obtained using the first law of thermodynamics for dual expander as:

$$\dot{W}_{\mathrm{exp\,ac}} = \eta_{\mathrm{exp}}[\dot{m}_{\mathrm{f1}}(h_4 - h_6) + \dot{m}_{\mathrm{f2}}(h_5 - h_6)] \tag{9}$$

Total efficiency of Rankine cycle was equal to cycle output power dividing by maximum energy extracted from exhaust gases and cooling system, as computed in equation [18]:

$$\eta_{total} = \frac{\dot{W}_{\mathrm{exp\,ac}}}{\dot{Q}_{\mathrm{exh\,max}} + \dot{Q}_{\mathrm{clnt\,max}}} \tag{10}$$

BSFC known as power-specific fuel consumption or simply specific fuel consumption is a comparative ratio presenting the engine fuel efficiency. It was defined as the ratio between engine fuel consumption and its power generation. BSFC for various engine operation states and the engine combined with ORC was computed from [17]:

$$BSFC_{\mathrm{eng}} = \frac{\dot{m}_{\mathrm{fuel}}}{\dot{W}_{\mathrm{eng}}} \tag{11}$$

$$BSFC_{\mathrm{eng-orc}} = \frac{\dot{m}_{\mathrm{fuel}}}{\dot{W}_{\mathrm{eng-orc}}} \tag{12}$$

In the above equations, $\dot{m}_{fuel}$ denotes fuel mass flow rate, estimated as:

$$\dot{m}_{\mathrm{fuel}} = \frac{\dot{m}_{\mathrm{exh}}}{AF + 1} \tag{13}$$

AF denotes Air Fuel ratio. As a result obtained by cogeneration, the percentage of BSFC reduction was expressed as:

$$\%\,BSFC = \frac{bsfc_{\mathrm{eng}} - bsfc_{\mathrm{eng-orc}}}{bsfc_{\mathrm{eng}}} \times 100 \tag{14}$$

7 RESULTS AND DISCUSSION

Fig. 3 shows the output power of engine, Rankine cycle and their accessories. Rankine cycle output power increases with augmentation of engine speed up to 1900 RPM.

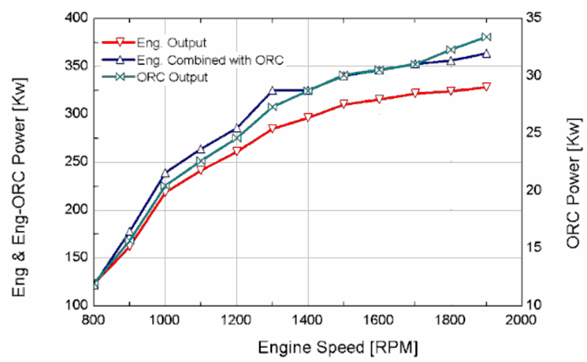

Fig. 3 Engine, ORC and total output power vs. engine speed (ε=0.8)

Fig. 4 shows ORC generated power versus engine speed with evaperator effevtiveness ranging from 0.55 to 0.80. As shown in this figure, output power was enhanced by increasing evaperator effectiveness. The size and weight of evaporator No. 2 also mounted up. Although generated power increases by enhancing the effectiveness of evaporator No. 2, the cost of components and produced overload also increases. The evaporator optimum effectiveness should be obtained through thermo-economic optimization.

Fig. 5 shows that total efficiency of Rankine cycle decreases slightly by increasing the engine speed; while the expander generated power increases by mounting the engine speed. The cause was sudden increment in exhaust gas temperature. As exhaust gas temperature rises, maximum transferable heat in Operator 2 increases, while the work generated by cycling increases with lower rate; thus, it was observed that the trend of cycling efficiency, computed from equation 10, was descending at 1000-1700 rpm. In cycles used for waste heat recovery, as mentioned earlier, enhancing the production capacity is more important than increasing the efficiency value, of course.

Fig. 6 presents a comparison between BSFC for engine operating state and engine combined with ORC. This quantity was lower for cogeneration because of more power generation at the same fuel consumption. Fig. 6 demonstrated also that the BSFC has the minimum value at 1400 RPM. Therefore, it was the optimum engine speed to obtain the best values of the ratio of fuel consumption and generated power.

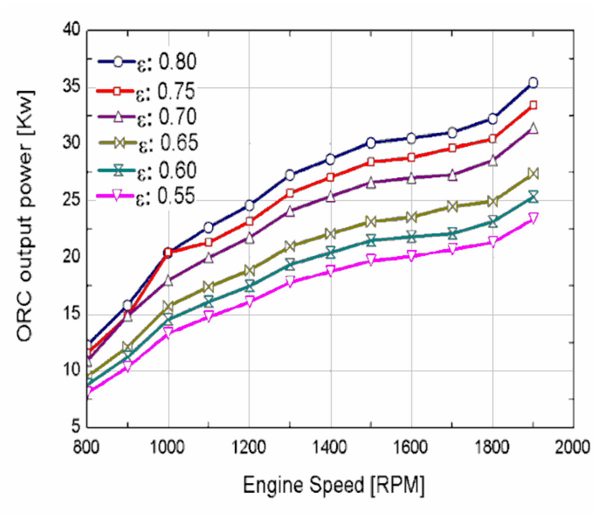

Fig. 4 Expander output power vs. engine speed with evaporator effectiveness ranging from 0.55 to 0.80

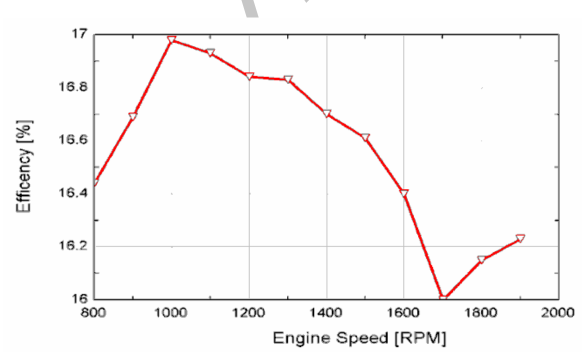

Fig. 5 Total efficiency of Two-stage organic Rankine cycle vs. engine speed (ε=0.8)

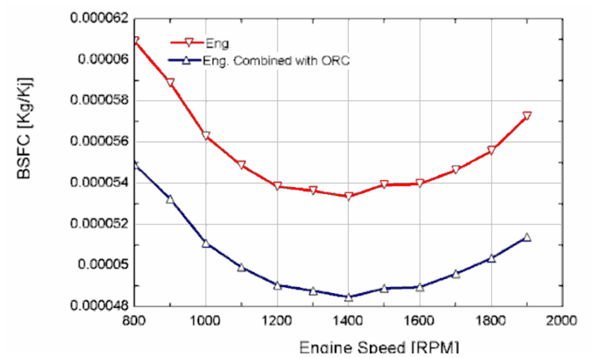

Fig. 6 Brake Specific Fuel Consumption for both Engine and Engine combined with ORC (ε=0.8)

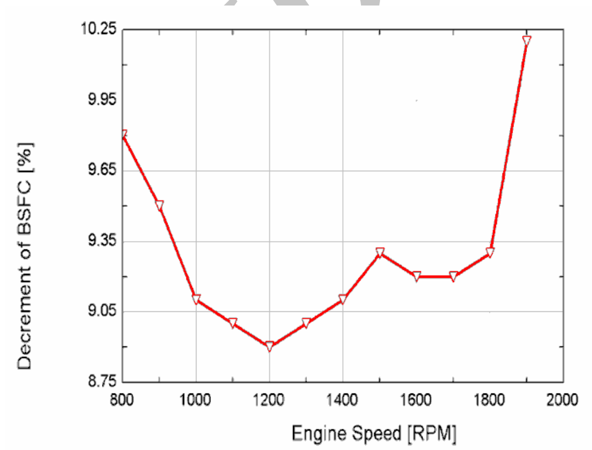

Fig. 7 Decrement of BSFC because of cogeneration (ε=0.8)

As shown in Fig. 7, the reduction precentage of BSFC obviuosly varies with engine speed increment. Thus, by engine speed increment up to 1200 RPM, the quantity was reduced; then it started to rise up and at 1900 RPM, assigned as nominal maximum engine power, it reached its maximum value. Thus, the greatest impact of cogeneration on power specific fuel consumption occurred at the mentioned engine speed.

8 CONCLUSION

In present study the amount of an ICE wasted heat through exhaust gas and cooling system was computed. It was found that at all engine speeds, almost 60% of the fuel energy gets wasted. A new configuration for Rankine cycle was introduced to recover engine wasted heat. In this configuration, the maximum amount of generated power was 35 kW which entails 10% enhancement in engine output power. It should be considered that waste heat recovery using ORC causes about 9.5% reduction in BSFC. Heat recovering systems also decrease environmental pollutants by reducing fuel consumption. Another notable point is the use of heat recovering systems at various engine speeds. These cycles result in greatest reduction in fuel consumption rate at high and low speeds, while the reduction value is smaller at lower speeds.

9 NOMENCLATURE AND SUBSCRIPTS

AF	Air-Fuel ratio
BSFC	Brake specific fuel consumption
C_p	Specific heat at constant pressure
h	Specific enthalpy [kJ/kg]
ICE	Internal Combustion Engine
$\dot{m}$	Mass flow rate [kg/s]
ORC	Organic Rankine Cycle
$\dot{Q}$	Heat flow [kW]
RPM	Engine Speed
T	Temperature [C]
$\dot{w}$	Power [kW]
Δ	Difference
Ac	Actual
Amb	Ambient
cf	Cooling fluid
clnt	Coolant system
eng	Engine
eng-orc	Engine bottoming with Organic Rankine cycle
exh	Exhaust Gas
exp	Expander
f	Working fluid
max	Maximum

REFERENCES

[1] Johnson, V., "Heat-generated cooling opportunities in vehicles", SAE Technical Papers, No.2002, 2002-01-1969.

[2] Quoilin, S., and Lemort, V., "Technological and economical survey of organic rankine cycle systems". 5th European Conference Economics and Management of Energy in Industry, April 2009.

[3] Endo, T., Kawajiri, S., Kojima, Y., Takahashi, K., Baba, T., Ibaraki, S., Takahashi, T., and Shinohara, M., "Study on maximizing exergy in Automotivee engines", SAE Paper No. 2007-01-0257, Presented at SAE World Congress &Exhibition, April 2007, Detroit, MI, USA, Session: New SI Engine and Component Design(Part 2 of 2).

[4] Kadota, M, and Yamamoto, K., "Advanced transient simulation on hybrid vehicle using Rankine cycle system", SAE Paper No. 2008-01-0310, Presented at SAE World Congress & Exhibition, April 2008, Detroit, MI, USA, Session: Advanced Hybrid Power trains (Part 2 of 3) Hydraulic Hybrids, Hybrid Vehicle Designs, and Thermal Energy Recovery and Storage.

[5] Ringler, J., Seifert, M., Guyotot, V., and Huebner, W., "Rankine cycle for waste heat recovery of IC engines", SAE Paper No. 2009-01-0174, presented at SAE World Congress & Exhibition, April 2009, Detroit, MI, USA, Session: Thermal Management Systems (Part 2 of 2).

[6] San, J. Y., "Second-law performance of heat exchangers for waste heat recovery", Energy 35, 2010, pp. 1936-1945.

[7] Kalyan K., Srinivasan, P. J., Mago, S., and Krishnan, R.,. "Analysis of exhaust waste heat recovery from a dual fuel low temperature combustion engine using an Organic Rankine Cycle", Energy 35, 2010, pp. 2387-2399.

[8] Donghong, W., Xuesheng, L., Zhen, L., and Jianming, G. "Dynamic modeling and simulation of an Organic Rankine Cycle system for waste heat recovery (ORC)", Applied Thermal Engineering, Vol. 28, 2008, pp. 1216-1224.

[9] Wipplinger, K. P. M., Harms, T. M., and Taylor, A., "Stainless steel finned tube heat exchanger design for waste heat recovery", Journal of Energy in Southern Africa, Vol. 17, No. 2, May 2006.

[10] Drescher, U., and Bruggermann, D., "Fluid selection for the Organic Rankine cycle (ORC) in biomass power and heat plants", Appl. Therm. Eng., Vol. 27, 2007, pp. 223-8.

[11] Miazza, V., and Miazza, A., "Unconventional working fluids in organic Rankine cycles for waste heat recovery systems", Appl. Therm. Eng., Vol. 21, 2001, pp. 381-90.

[12] Mago, P. J., Charma, L. M., Srinivasan, K., and Somayaji, C., "An examination of regenerative organic Rankine cycles using dry fluids", Appl. Therm. Eng., Vol. 28, 2008, pp. 998-1007.

[13] Mustafah, B., Mohd, T., Noboru, Y., and Tetsuya, H., "Efficiency of Organic Rankine Cycle System with Rotary-Vane- Type Expander for Low-Temperature Waste Heat Recovery", World Academy of Science, Engineering and technology C: Civil and Environmental Engineering, Vol. 2, No. 1, 2010.

[14] Iacopa, V., and Agostino, G., "Internal combustion engine (ICE) bottoming with Organic Rankine Cycles", Energy, Vol. 35, 2010, pp. 1084-1093.

[15] Robert, H., Perry, D., and Green, W., "Perry's Chemical Engineers' HANDBOOK", Seventh Edition, MC Graw Hill, 1999.

[16] Tahani, M., Javan, S., and Biglari, M., "A comprehensive study on waste heat recovery from internal combustion engines using organic Rankine

cycle", doi: 10.2298TSCI111219051T. Thermal Science, ISSN 0354-9836.

[17] Kalyan, K., Srinivasan, P., Mago Sundar, J., and Krishnan, R., "Analysis of exhaust waste heat recovery from a dual fuel low temperature combustion engine using an Organic Rankine Cycle", Energy 35, 2010, pp. 2387-2399.

[18] Schuster, A., Karellas, S., and Aumann, R., "Efficiency optimization potential in supercritical Organic Rankine Cycles", Energy 35, 2010, pp. 1033-1041.