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# Performance Analysis of a Combined Dual-Stage Waste Heat Recovery System Integrated to an Internal Combustion Engine

Zunaid Ahmed
*Department of Mechanical Engineering, The Assam Royal Global University, Guwahati, Assam, India*

***Abstract: A combined dual-stage waste heat recovery system integrated to an internal combustion engine is studied. The system consists of high-temperature steam Rankine cycle (SRC) and a low-temperature organic Rankine cycle (ORC), both combined to recover the waste heat of the engine exhaust gases and engine coolant. In the ORC, organic working fluids R245fa, R600 and R601a are selected for analysis and sub-critical cycle adopted.***
***For the comparative study of the selected working fluids, energy and exergy analysis are conducted based on the engine data, pre-set parameters and mathematic model with net-output power, utilization rate, thermal efficiency and exergy efficiency as the objective functions for optimization.***
***Keywords: Heat recovery, combined system, Organic Rankine cycle, Exergy analysis***

## I. INTRODUCTION

Internal combustion engines (ICE), that converts energy from heat to work, has vast applications in road vehicles, marine transport, and power plants. In ICE, all the energy released during combustion of the fuel cannot be converted into useful work because of some thermodynamic limitations. About two-thirds of the total fuel combustion heat in automotive ICEs is wasted by the exhaust gases and engine coolant, resulting in energy waste and emission problem [1],[2],[3]. Recovering the waste energy could greatly improve the engine fuel efficiency and reduce environmental pollution.

As energy crisis and environment pollution are increasingly severe, many technologies have been proposed to save energy and reduce emission in the field of ICE.

Among these technologies, organic Rankine cycle (ORC) is an effective one because of its flexibility, economy and good thermal performance [4],[5]. Organic Rankine cycle (ORC) is seen as a high-effective way to recover the low-medium temperature heat (80$^oC$ to 300$^oC$), such as biomass, solar, geothermal, and industry reject heat [6]. ORC has been also used in heat recovery systems of ICE [7]– [11].

These systems mainly use single-stage heat recovery. In ICE the waste heat released are at different temperatures, that is, exhaust gas temperature is high (450-600$^oC$) while coolant temperature is low (80-85$^oC$). Therefore, matching of the exhaust and coolant with organic working fluid is a problem.

In single-stage systems and the engine coolant was usually used as the preheating heat source, resulting in little utilization of the engine coolant waste heat. There are also issues of decomposition of organic working fluid and unsafe direct heat exchange with high temperature waste heat.

To solve these problems an integrated dual-stage is proposed, one to recover the high-temperature exhaust waste heat and the other to recover the heat from the low-temperature coolant.

The high-temperature stage uses steam Rankine cycle (SRC) with water as the working fluid to recover the heat from exhaust gases; the low-temperature stage uses an organic Rankine cycle (ORC) with an organic working fluid to recover the heat from engine coolant; and both the stages being integrated.

In this study, ORC working fluids *R245fa*, *R600* and *R601a* are selected for the system's comparative analysis. For the selected working fluids, energy and exergy analysis are conducted based on the engine data, pre-set parameters and mathematic model with net-output power, utilization rate, thermal efficiency and exergy efficiency as the objective functions for optimization. The evaporator pressure, condenser pressure and mass flow rate are taken to be the decision variables.

## II. DESCRIPTION OF SYSTEM

For the analysis, a commercial diesel engine is selected as the topping cycle. Main engine parameters are shown in Table 1. with the assumption of full load condition.

Table 1
Main Engine Parameters

| Parameters | Unit | Value |
|---|---|---|
| Engine power output | (kW) | 1000 |
| Engine Efficiency | (%) | 41.56 |
| Exhaust gas temperature | ($^{O}$C) | 565 |
| Engine coolant temperature | ($^{O}$C) | 83.5 |
| Exhaust gases mass flow rate | (Kg/s) | 1.135 |
| Engine coolant mass flow rate | (Kg/s) | 1.350 |

The configuration diagram of combined SRC-ORC system integrated with the ICE system is shown in Fig.1. The system comprises of a high-temperature stage employing a SRC circuit to utilize a large part of the exhaust heat and a low-temperature stage employing an ORC circuit to utilize the heat of coolant and remaining part of exhaust heat. The working fluids used in the SRC and ORC are water and organic fluid respectively. Water is effective in recovering high-temperature waste heat while organic fluid is more effective in recovering low-temperature waste heat. The SRC is a high temperature circuit that comprises of feed-water pump ($P_H$), evaporator ($E_H$), steam turbine ($T_H$) and condenser ($C_H$). The ORC is a low temperature circuit that consists of organic fluid pump ($P_L$), liquid-liquid heat exchanger ($H_{L1}$), evaporator ($E_L$), gas-gas heat exchanger ($H_{L2}$); vapour turbine ($T_L$) and water-cooled condenser ($C_L$). The two circuits are combined using the heat exchanger ($E_L/C_H$) behaving as evaporator for the ORC and condenser for SRC. In the SRC, high-pressure superheated steam is generated in the $E_H$ by utilizing the heat of high-temperature exhaust gases of ICE, which is then expanded in the steam turbine $T_H$ to produce mechanical work. The lower pressure steam in then condensed in $E_L/C_H$ by giving off its latent heat to organic fluid of ORC. The water is then sent back to $E_H$ by a feed-water pump. In the ORC, the high-pressure and low-temperature liquid organic fluid is heated in the $H_{L1}$ by the hot cooling water of the ICE. The organic working fluid is then evaporated in the heat exchanger $E_L/C_H$ by absorbing the latent heat of steam. The organic vapour generated is further superheated in $H_{L2}$ by the remaining heat of exhaust gases that are at a lower temperature. The high-pressure organic fluid then expands in the vapour turbine $T_L$ to produce mechanical work. The low-pressure vapour is then condensed in a condenser $C_L$ by external cooling water. The liquid organic fluid is the sent to $H_{L1}$ by a refrigerant pump. In Fig.1 the closed circuit 1-2-3-4-1 represents the state points of SRC while the closed circuit 6-7-8-9-10-6 the state points of ORC. The path $g_1$-$g_2$-$g_3$ represents the state points of exhaust gases while passing through $E_H$ and $H_{L2}$ sequentially. The path $w_1$-$w_2$ the state points of engine coolant (water) while passing through $H_{L1}$. The path $w_3$-$w_4$ the state points of cooling water while passing through $C$. The T-s diagram of the ORC processes are shown in Fig.2 and that of SRC in Fig.3

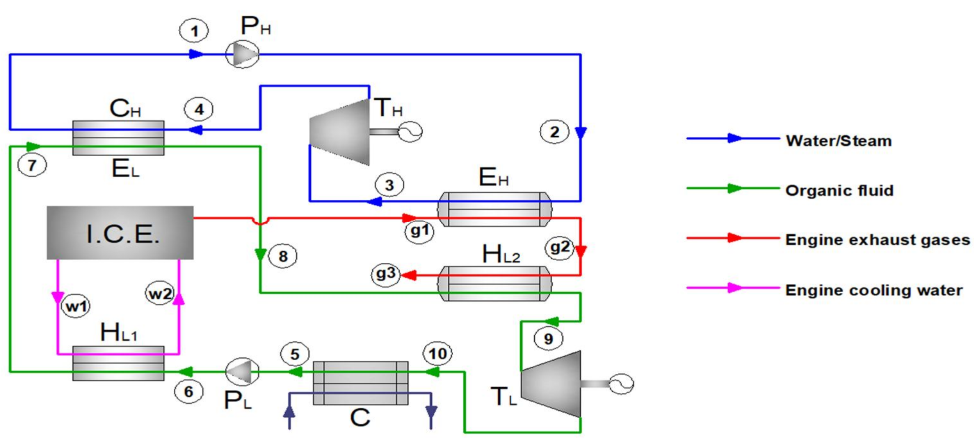

Fig.1 Configuration layout of engine waste heat recovery system

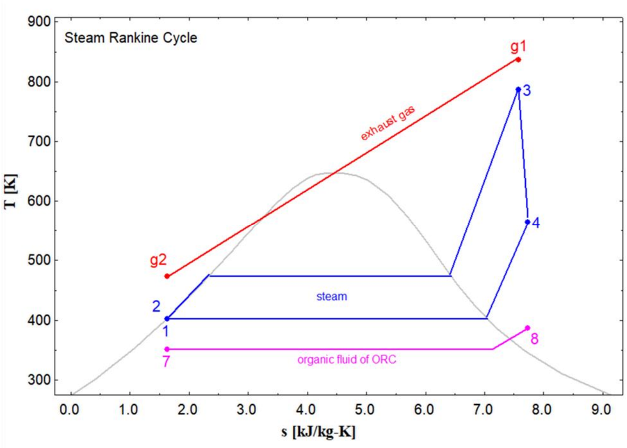


Fig.2. T-s diagram of the SRC system

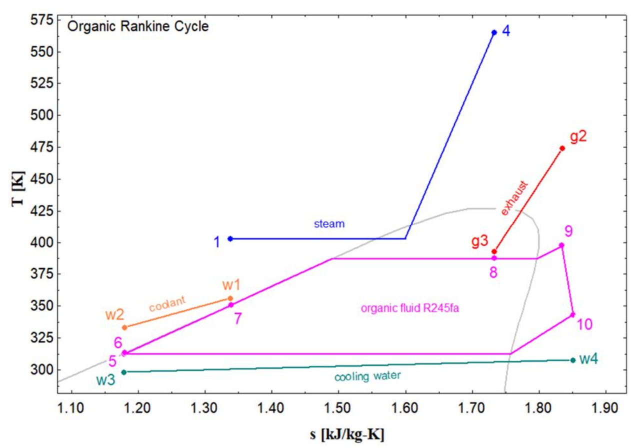


Fig.3. T-s diagram of the ORC system

System performance is significantly affected by the working fluid. The selection of criteria for sub-critical waste heat recovery having low-medium temperature, such as used in ORC, are good thermal and physical properties, high chemical stability safety and environmentally friendly [12]- [15]. Based on the selection criteria, three dry or isentropic organic fluids (refrigerants) are chosen to be used as working fluids in the ORC system. These are *R245fa*, *R600* and *R601a* (isopentane). These organic fluids have high decomposition temperature i.e., higher than the exhaust gas temperature.

## III.MATHEMATICAL MODEL

The assumption for creating a mathematical model of the system are: processes are steady-flow; pressure drop and heat losses are neglected; kinetic and potential energy of working fluid are insignificant; ambient conditions are 298 K and 100 kPa; isentropic efficiencies of turbines are 0.85 and isentropic efficiencies of pumps are 0.75; the minimum pinch point of gas-liquid heat exchanger is 15 K and that of liquid-liquid heat exchanger 5 K; superheat of organic fluid is 10 K; condensation temperature of SRC is 403 K; condensation temperature of ORC 313 K; evaporation temperature of organic fluids in heat exchanger $H_{L2}$ will depend on the temperature of exhaust gases entering heat exchanger and therefore the corresponding evaporation pressure will be different for each organic fluid; the temperature of exhaust gases leaving heat exchanger ($H_{L2}$) is constrained to 393 K in order to avoid the low-temperature corrosion [9]. The evaporation temperature of water in SRC is estimated after taking into consideration the pinch point with respect to exhaust gas temperature and the corresponding evaporation pressure is calculated.

Energy and exergy analysis are carried out for each component of the integrated system based on the first and second thermodynamic laws [16,17].

The exergy at each point $i$ is calculated by using:

$$\dot{E}_i = \dot{m}[(h_i - h_0) - T_0 \cdot (s_i - s_0)] \tag{1}$$

where 0 implies the ambient state and $\dot{m}$, $h$, $s$, $T$ implies mass flow rate, specific enthalpy, specific entropy, temperature respectively.

The mass flow rates of steam in SRC, organic fluid in ORC, exhaust gases, engine coolant and cooling water of condenser are designated by $\dot{m}_{f,src}$, $\dot{m}_{f,orc}$, $\dot{m}_g$, $\dot{m}_w$ and $\dot{m}_c$ respectively. $h_i$, $T_i$ implies specific enthalpy and temperature respectively at every state point $i$ as shown in Fig.1, Fig.2 and Fig.3.

*For pump $P_H$:*

The isentropic efficiency,

$$\eta_{P_H} = \frac{h_{2'} - h_1}{h_2 - h_1} \tag{2}$$

where $2'$ represents ideal point

Power consumed by pump

$$\dot{W}_{P_H} = \dot{m}_{f,src} \cdot (h_2 - h_1) \tag{3}$$

Exergy destruction due to irreversibility,

$$\dot{I}_{P_H} = \dot{E}_1 + \dot{W}_{P_H} - \dot{E}_2 \qquad (4)$$

*For evaporator $E_H$*

Heat recovered from exhaust gases,

$$\dot{Q}_{g_1} = \dot{m}_{f,src} \cdot (h_3 - h_2) = \dot{m}_g \cdot c_{p_g} \cdot (T_{g_1} - T_{g_2}) \qquad (5)$$

where $c_{p_g}$ is specific heat of exhaust gases; $T_{g_1}$ and $T_{g_2}$ are tinlet temperature and outlet temperature of exhaust gases respectively.

Exergy destruction due to irreversibility,

$$\dot{I}_{E_H} = (\dot{E}_2 + \dot{E}_{g_1}) - (\dot{E}_3 + \dot{E}_{g_2}) \qquad (6)$$

*For steam turbine $T_H$ :*

The isentropic efficiency,

$$\eta_{T_H} = \frac{h_3 - h_4}{h_3 - h_{4'}} \qquad (7)$$

where $4'$ represents ideal point

Power produced,

$$\dot{W}_{T_H} = \dot{m}_{f,src} \cdot (h_3 - h_4) \qquad (8)$$

Exergy destruction due to irreversibility,

$$\dot{I}_{T_H} = \dot{E}_3 - \dot{W}_{T_H} - \dot{E}_4 \qquad (9)$$

*For pump $P_L$:*

The isentropic efficiency,

$$\eta_{P_L} = \frac{h_{6'} - h_5}{h_6 - h_5} \qquad (10)$$

where $6'$ represents ideal point

Power consumed by pump,

$$\dot{W}_{P_L} = \dot{m}_{f,orc} \cdot (h_6 - h_5) \qquad (11)$$

Exergy destruction due to irreversibility,

$$\dot{I}_{P_L} = \dot{E}_5 + \dot{W}_{P_L} - \dot{E}_6 \qquad (12)$$

*For heat exchanger $H_{L1}$*

Heat recovered from engine coolant

$$\dot{Q}_w = \dot{m}_{f,orc} \cdot (h_7 - h_6) = \dot{m}_w \cdot c_{p_w} \cdot (T_{w_1} - T_{w_2}) \qquad (13)$$

where $c_{p_w}$ is specific heat of coolant (water); $T_{w_1}$ and $T_{w_2}$ are tinlet temperature and outlet temperature of coolant respectively.

Exergy destruction due to irreversibility,

$$\dot{I}_{H_{L1}} = (\dot{E}_6 + \dot{E}_{w_1}) - (\dot{E}_7 + \dot{E}_{w_2}) \qquad (14)$$

*For evaporator $E_L$ – condenser $C_L$ :*

$$\dot{Q}_f = \dot{m}_{f,src} \cdot (h_1 - h_4) = \dot{m}_{f,orc} \cdot (h_8 - h_7) \qquad (15)$$

Exergy destruction due to irreversibility,

$$\dot{I}_{C_H - E_L} = (\dot{E}_4 + \dot{E}_7) - (\dot{E}_1 + \dot{E}_8) \qquad (16)$$

*For heat exchanger $H_{L2}$*

Heat recovered from exhaust gases

$$\dot{Q}_{g_2} = \dot{m}_{f,orc} \cdot (h_9 - h_8) = \dot{m}_g \cdot c_{p_g} \cdot (T_{g_2} - T_{g_3}) \qquad (17)$$

where $T_{g_2}$ and $T_{g_3}$ are inlet temperature and outlet temperature of exhaust gases respectively.

Exergy destruction due to irreversibility,

$$\dot{I}_{H_{L2}} = (\dot{E}_8 + \dot{E}_{g_2}) - (\dot{E}_9 + \dot{E}_{g_3}) \qquad (18)$$

*For vapour turbine $T_L$ :*

The isentropic efficiency,

$$\eta_{T_L} = \frac{h_9 - h_{10}}{h_9 - h_{10'}} \tag{19}$$

where $10'$ represents ideal point

Power produced,

$$\dot{W}_{T_L} = \dot{m}_{f,orc} \cdot (h_9 - h_{10}) \tag{20}$$

Exergy destruction due to irreversibility,

$$\dot{I}_{T_L} = \dot{E}_9 - \dot{W}_{T_L} - \dot{E}_{10} \tag{21}$$

*For condenser $C_L$:*
Heat rejected

$$\dot{Q}_c = \dot{m}_{f,orc} \cdot (h_{10} - h_5) = \dot{m}_c \cdot c_{p_w} \cdot (T_{c_2} - T_{c_1}) \tag{22}$$

where $T_{c_1}$ and $T_{c_2}$ are inlet temperature and outlet temperature of cooling water respectively.
Exergy destruction due to irreversibility,

$$\dot{I}_c = (\dot{E}_{10} + \dot{E}_{c_1}) - (\dot{E}_5 + \dot{E}_{c_2}) \tag{23}$$

Parameters indicating system performance are utilization rates of engine exhaust waste heat ($\eta_{u,g}$) and coolant waste heat ($\eta_{u,w}$), net power output ($\dot{W}_{net}$), thermal efficiency ($\eta_{th}$), exergy efficiency ($\eta_e$).

$$\eta_{u,g} = \frac{\dot{Q}_{g_1} + \dot{Q}_{g_2}}{\dot{Q}_{g_{max}}} \tag{24}$$

where $\dot{Q}_{g_{max}}$ is the maximum heat available in the exhaust gases

$$\dot{Q}_{g_{max}} = \dot{m}_g \cdot c_{p_g} \cdot (T_{g_1} - T_0) \tag{25}$$

$$\eta_{u,w} = \frac{\dot{Q}_w}{\dot{Q}_{w_{max}}} \tag{26}$$

where $\dot{Q}_{w_{max}}$ is the maximum heat available in the engine coolant

$$\dot{Q}_{w_{max}} = \dot{m}_w \cdot c_{p_w} \cdot (T_{w_1} - T_0) \tag{27}$$

$$\dot{W}_{net} = (\dot{W}_{T_H} - \dot{W}_{P_H}) + (\dot{W}_{T_L} - \dot{W}_{P_L}) \tag{28}$$

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{g_1} + \dot{Q}_{g_2} + \dot{Q}_w} \tag{29}$$

$$\eta_e = 1 - \frac{\dot{I}}{\dot{E}_{in}} \tag{30}$$

where

$$\dot{E}_{in} = (\dot{E}_{g_1} - \dot{E}_{g_3}) + (\dot{E}_{w_1} - \dot{E}_{w_2}) + \dot{W}_{P_H} + \dot{W}_{P_L} \tag{31}$$

$$\dot{I} = \dot{I}_{P_H} + \dot{I}_{E_H} + \dot{I}_{T_H} + \dot{I}_{C_H - E_L} + \dot{I}_{P_L} + \dot{I}_{H_{L1}} + \dot{I}_{H_{L2}} + \dot{I}_{T_L} + \dot{I}_c \tag{32}$$

For the comparative study of the selected working fluids, energy and exergy analysis are conducted based on the engine data (Table 1), pre-set parameters (assumptions) and mathematical model equations (1) to (32), with net-output power, utilization rate, thermal efficiency and exergy efficiency as the objective functions for optimization. The evaporator pressure, condenser pressure and mass flow rate are taken to be the decision variables. A code in written in EES [18] to simulate the system. EES has an in-built library of thermophysical properties of organic fluids that can be called to obtain thermodynamic properties at each state point of the system. Analysis and optimization are carried out on the simulated system to achieve the solutions. The simulation is run for each organic fluid selected for comparative performance solutions.

## IV. PERFORMANCE ANALYSIS OF SYSTEM

Performances of R245fa, R600 and R601a are evaluated based on the assumptions and mathematical model. The evaporator pressure, condenser pressure and mass flow rate of steam in SRC system are evaluated as 1600 kPa, 269 kPa and 0.148 kg/s and is the same for analysis of each organic fluid. The optimum operating parameters i.e., evaporator pressure ($P_{b,orc}$), condenser pressure ($P_{c,orc}$) and mass flow rate ($\dot{m}_f$) of each organic working fluid in order to maximize the objective functions (performance parameters) are shown in Table 2. The evaporator pressure is the highest for fluid R600 whereas the condenser pressure is the lowest for R601a. The highest fluid mass flow rate is required for fluid R245fa.

Comparative performance of the working fluids i.e., net output power ($\dot{W}_{net}$), utilization rates of the engine exhaust waste heat ($\eta_{u,g}$) and coolant waste heat ($\eta_{u,w}$), thermal efficiency ($\eta_{th}$) and exergy efficiency ($\eta_e$) are depicted in Table 3. Comparative energy analysis of selected working fluid is shown in table 4. Comparative exergy analysis of selected working fluid i.e., irreversibility of each component is depicted in Table 5.

Table 2: Optimum operating parameters

| Organic Fluid | $P_{b,orc}$ *(kPa)* | $P_{c,orc}$ *(kPa)* | $\dot{m}_f$ *(kg/s)* |
|---|---|---|---|
| R245fa | 1740 | 248 | 2.474 |
| R600 | 2017 | 378 | 1.284 |
| R601a | 981 | 151 | 1.240 |

Table 3: Performance parameters of working fluids

| Organic Fluid | $\dot{W}_{net}$ *(kW)* | $\eta_{u,g}$ *(%)* | $\eta_{u,w}$ *(%)* | $\eta_{th}$ *(%)* | $\eta_e$ *(%)* |
|---|---|---|---|---|---|
| R245fa | 142.61 | 82.32 | 39.68 | 21.38 | 54.24 |
| R600 | 141.73 | 82.40 | 38.42 | 21.36 | 54.18 |
| R601a | 140.74 | 82.25 | 35.10 | 21.60 | 53.59 |

Table 4: Comparative energy analysis

| Energy Source | R245fa | | R600 | | R601a | |
|---|---|---|---|---|---|---|
| | (kW) | % | (kW) | % | (kW) | % |
| Exhaust gases | 652 | 66.41 | 652 | 66.41 | 652 | 66.41 |
| Coolant | 330 | 33.59 | 330 | 33.59 | 330 | 33.59 |
| Total | 981 | 100.00 | 981 | 100.00 | 981 | 100.00 |
| Energy Utilized | | | | | | |
| Exhaust gases - SRC | 439 | 44.70 | 439 | 44.70 | 439 | 44.7 |
| Exhaust gases - ORC | 98 | 9.97 | 98 | 10.03 | 97 | 9.88 |
| Coolant - ORC | 131 | 13.3 | 127 | 12.91 | 116 | 11.82 |
| Energy Loss | | | | | | |
| Exhaust gases | 115 | 11.74 | 115 | 11.69 | 116 | 11.82 |
| Coolant | 199 | 20.26 | 203 | 20.68 | 213 | 21.71 |
| Total | 981 | 100.00 | 981 | 100.00 | 981 | 100.00 |
| System Energy Transfer | | | | | | |
| Energy In (Utilized) | 667 | 100.00 | 663 | 100.00 | 652 | 100.0 |
| | | | | | | |
| Net Work | 143 | 21.38 | 142 | 21.36 | 141 | 21.60 |
| Energy Out | 525 | 78.62 | 522 | 78.64 | 511 | 78.40 |
| Total | 667 | 100.0 | 663 | 100.00 | 652 | 100.00 |

The maximum net output power of the system is around 142 kW for all the fluids. As for the utilization rate of the exhaust waste heat ($\eta_{u,g}$), it is calculated to be around 82% for all the fluids. This is because the final outlet temperature of the exhaust is 120$^{O}$C (acid due point temperature), which means complete recovery of the exhaust waste heat is avoided to keep the temperature above this. The utilization rate of coolant heat is maximum fluid R245fa. As seen in Table 4, the thermal efficiencies of all the fluids are around 21%, since the energy utilization rate is more less the same for the all the fluids. The irreversibility of each component shows its influence on the system. It is obvious from Table 5, that evaporator ($E_H$) for all fluids makes the biggest contribution to the irreversibility of the whole system, due a large temperature difference on the organic fluid stream. The smallest exergy destruction is in feed-water pump $P_h$. Therefore, the heat exchange processes between the working fluids with the exhaust and engine coolant and non-isentropic expansion processes influence the system exergy performance most. Whereas, non-isentropic compression processes influence the system exergy performance least. Therefore, optimization of evaporator ($E_H$) is significant for improving system exergy performance, such as adjusting and optimizing its design parameters (pinch point temperature difference, type, material, layout and so on).

Table 5: Comparative exergy analysis

| Exergy In | R245fa | | R600 | | R601a | |
|---|---|---|---|---|---|---|
| | (kW) | % | (kW) | % | (kW) | % |
| Exhaust gases | 264.53 | 92.38 | 264.66 | 92.13 | 264.42 | 93.41 |
| Coolant | 17.74 | 6.20 | 17.29 | 6.02 | 16.08 | 5.68 |
| Pumps | 4.07 | 1.42 | 5.33 | 1.85 | 2.57 | 0.91 |
| Total | 286.34 | 100.00 | 287.28 | 100.00 | 283.07 | 100.00 |
| Exergy Destroyed | | | | | | |
| Evaporator, $E_H$ | 58.34 | 20.37 | 58.34 | 20.31 | 58.34 | 20.61 |
| Heat exchanger, $E_L/C_H$ | 21.71 | 7.58 | 21.78 | 7.58 | 21.11 | 7.46 |
| Condenser, $C_L$ | 19.70 | 6.88 | 20.02 | 6.97 | 21.89 | 7.73 |
| Vapour turbine, $T_L$ | 12.33 | 4.31 | 12.36 | 4.30 | 11.43 | 4.04 |
| Heat exchanger, $H_{L2}$ | 7.41 | 2.59 | 7.42 | 2.58 | 7.47 | 2.64 |
| Vapour turbine, $T_H$ | 6.43 | 2.25 | 6.43 | 2.24 | 6.43 | 2.27 |
| Heat exchanger, $H_{L1}$ | 4.14 | 1.45 | 4.03 | 1.40 | 4.1 | 1.45 |
| Organic fluid pump, $P_L$ | 0.90 | 0.31 | 1.2 | 0.42 | 0.54 | 0.19 |
| Feed-water pump, $P_H$ | 0.05 | 0.02 | 0.05 | 0.02 | 0.05 | 0.02 |
| Exergy Out | | | | | | |
| Condenser | 8.65 | 3.02 | 8.563 | 2.98 | 8.383 | 2.96 |
| Turbines | 146.68 | 51.23 | 147.06 | 51.20 | 143.31 | 50.63 |
| Total | 286.34 | 100.00 | 287.25 | 100.00 | 283.05 | 100.00 |

## V. CONCLUSIONS

In this study, combined dual-stage waste heat recovery system integrated to an internal combustion engine is proposed which consists a high-temperature stage and a low-temperature stage. The high-temperature stage uses water to recover the high-temperature exhaust. The low-temperature stage uses an organic working fluid to recover the waste heat of the engine coolant, residual heat of the and low-temperature exhaust in series. Organic fluids R245fa, R600 and R601a are chosen as candidate working fluids of the low-temperature stage in this study. Based on energy and exergy analysis above, several conclusions: Maximum net output power and thermal efficiency are obtained with R245fa with corresponding values are 142.21 kW and 21.38

% at optimum evaporator pressure and condenser pressure of 1740 kPa and 248 kPa respectively. Maximum exergy efficiency of 51.23% is obtained for working fluid R245fa with maximum exergy destruction occurring in the evaporator. Based on net output power ($\dot{W}_{net}$), utilization rates of the engine exhaust waste heat ($\eta_{u,g}$) and coolant waste heat ($\eta_{u,w}$), thermal efficiency ($\eta_{th}$) and exergy efficiency ($\eta_e$) for all working fluids, R245fa is found to be a better working fluid.

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