Energy and Exergy Analyses of Automotive Organic Rankine Cycles

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ABSTRACT :

Wasted heat in internal combustion engines affects both environmental and economic aspects all over the world. Reusing the wasted heat can potentially improve engine performance, reduce exhaust harmful emissions as well as reduce fuel prices. Organic Rankine cycle (ORC) system is considered a promising technology to convert low to medium heat into mechanical power and electricity. In this study, energy and exergy theoretical analysis of an organic Rankine cycle system (ORC) as a waste heat recovery (WHR) technology in internal combustion engines is carried out. ORC is simply a Rankine cycle that operates with organic fluids instead of steam in low to medium temperature applications. The ORC performance parameters are evaluated by varying the operating conditions as turbine inlet temperature and pressure ratio. The wasted heat in the exhaust gas of a 77.25ℓ heavy-duty diesel engine is used as the heat source. However, thermal oil loop is integrated in the system to assure steady state operation. The encompassed energy-exergy analyses for evaluating the potential of ORC as a WHR in truck engines are the core objective of the current study which eventually lead to further advanced analyses in the future. The results of the study indicate that ORC can used as a potential WHR system. The maximum recovery, thermal and exergy efficiencies are 8.59%, 7.8% and 23.4%. The turbine presents a maximum mechanical power of 19.42 kW.

KEYWORDS :

energy analysis; exergy analysis; organic Rankine cycle; waste heat recovery; internal combustion engine.

1. INTRODUCTION :

Carbon dioxide (CO2) expulsions from internal combustion engines account for a substantial proportion of the atmosphere's global carbon dioxide level. These emissions introduce ecological problems in the form of greenhouse gases and potential danger to lives. Because of these adverse effects, the International Community has progressively benchmarked automotive emissions from automobile engines. ICCT, as an international body, placed a target of a 1% reduction of automotive exhaust emissions in 2007 and then increased the target to 18% and 28% for 2012 and 2020, respectively, (Milkov, 2018). Published studies have shown that the transport sector relatively burns much of global fuel, accounting for over 67% of total fuel consumption in the US, (Alshammari, Elashmawy and Bechir Ben Hamida, 2021), and over 73% in the UK in 2013, Emissions of NOx and PM from the transport sector affect air quality, whereas CO2 contributes to global warming. The internal combustion engine expels 60 to 65% of its fuel energy content to the environment in the form of heat due to system inefficiencies (Karvountzis-Kontakiotis et al., n.d.). Therefore, recovering the wasted fuel energy increases engine power as well as reduces fuel consumption and harmful emissions.

The main heat sources where fuel energy is wasted are the exhaust gases, cooling systems, and relatively smaller amounts available from the exhaust gas recirculation (EGR) system (Ezoji and Ajarostaghi, 2020). Fuel energy is mostly wasted in the exhaust gas followed by the engine coolant and EGR system (Shi, Tian and Shu, 2020). Only one third of the exhaust gas heat can be recovered and converted to useful work while the recovery potential of coolant energy is much lower due to the lower coolant temperature (Alshammari, Alshammari and Pesyridis, 2019). However, the efficiency of the heat source is a trade-off between its quantity (energy contained in the heat source) and quality (temperature range of the heat source) (Alshammari, Pesyridis and Elashmawy, 2021). Recently, organic Rankine cycle (ORC) has received great attention as a WHR technology in internal combustion engines (ICEs) (Mat Nawi et al., 2019). Compared to other WHR technologies, ORC systems present equivalent thermal efficiencies under a wide range of operating conditions, relatively low cost and minor increase of pumping losses (Santos et al., 2020). Energy and Exergy analyses are crucial tools to identify key design aspects that enhance overall conversion efficiency and maximize resource utilization. Energy analysis can be utilized to globally evaluate the feasibility of ORC systems as WHR technologies in automobile applications. Exergy analysis, on the other hand, can be utilized to provide guidelines in system design at various operating conditions.

Recently, researchers have paid more attention in investigating ORC systems as bottoming cycles in internal combustion engines. Koppauer et al. (Koppauer, Kemmetmüller and Kugi, 2017) investigated the feasibility of ORC systems in reducing the fuel consumption of a six cylinder diesel engine. However, the study focused mainly on designing optimal control strategies for WHR systems. Two evaporators were set in parallel with ethanol delivered to the two evaporators by means of the fluid supply unit which consists of a pump and two proportional valves. The turbine was coupled with an electrical brake on the test bench. The results indicated the good accuracy of the developed model and a maximum turbine power of 0.6 kW was obtained. Song et al.(Song, Sun and Huang, 2017) presented a mean-line model for designing an ORC radial outflow turbine. The heat source and the working fluid were exhaust gas of a gas turbine and R123, respectively. The mean-line model was validated with a 3D CFD analyses. The obtained turbine efficiency by the 1D model was 85.9% compared to 87.1% via CFD. Alshammari et al. (Alshammari et al., 2017) presented a mean-line model for designing a backswept radial inflow turbine for automotive ORC systems. The exhaust gas of 7.2 L heavy duty diesel engine was considered. The study focused mainly on the optimization of the turbine by improving the rotor inlet blade and exit radius. The rotor geometric optimization showed that by increasing the rotor exit tip radius by 10% and adopting 54 degree backswept blades, the maximum achieved isentropic efficiency exceeded 83%. The following year, the authors (Alshammari et al., 2020) presented a novel mean-line model turbine design model equipped with an optimization algorithm. The authors considered the operation of a realistic engine-ORC system by studing the turbine design at different engine operating points. In the study, the authors focused on the losses model for micro-scale radial turbine. Based on the steady-state cycle simulation, a radial turbo-expander with a pressure ratio of seven (7) was designed for an automotive application and demonstrated a total-to-static efficiency and power output of 74.4% and 13.6 kW, respectively. Compared to the base diesel powertrain systems, the waste heat recovery system improved the brake specific fuel consumption, power and NOx emissions of the engine by 3.79%, 3.95% and 3.7%, respectively. In a separate study, Alshammari et al., (2018) developed a novel off-design performance prediction model for automotive ORC radial turbines. The model was validated with experimental results from the literature and presented less than 10% errors. Recently, Song et al. (Song et al., 2020) coupled a preliminary turbine design method with a Trans-critical CO2 system. The study focused on the thermodynamic and economical perspectives of such systems as WHR technologies in ICEs. Maximum obtained newt power and specific investment cost were 175 kW and 5180 $/kW, respectively.

Energy and exergy analyses are important tools to identify key design aspects that may improve overall conversion efficiency and maximize resource utilization. The brief literature study indicates that ORC system is a promising technology. However, it can be drawn from the literature study that exergy analysis is still needs to be carefully covered. In addition, exergy analysis should be integrated with energy analysis for more feasible results.

Therefore, the present study focuses on evaluating the performances of the ORC system by the method of energy and exergy analyses from a theoretical basis. The feasibility of ORC systems as WHR technologies in ICEs is explored at different cycle parameters i.e. turbine inlet temperatures and pressure ratios. The exergy efficiency, thermal efficiency, heat recovery efficiency and the irreversibility rate are calculated with the various turbine temperatures and the inlet temperatures and pressure ratios.

1. METHODOLOGY :

The system consists of a heavy-duty diesel engine to provide via the exhaust gas, and ORC system.

2.1 DESCRIPTION OF THE HEAVY-DUTY DIESEL ENGINE (HEAT SOURCE) :

The engine utilized in the current study is a 7.25ℓ Yuchai engine. It is a turbocharged, direct injection engine and fulfils the EURO III regulatory requirements. Table 3 shows the characteristics of the utilized diesel engine. To maintain steady state operation, a thermal oil loop is integrated in the system as an intermediate circuit between the exhaust gas and organic Rankine cycle loop. The developed model of the gas-oil heat exchanger can be found in the previous study of the authors (Karvountzis-Kontakiotis et al., 2017).

2.2 DESCRIPTION OF THE THERMAL OIL LOOP :

The intermediate thermal oil loop is placed between the exhaust gas of the engine and the ORC system via the main heat exchanger as shown in Fig. 1. The thermal oil loop requires two more components i.e. heat exchanger and pump which results in heavier system. However, the

thermal oil loop assures steady-state conditions for the ORC operation and is beneficial in order to avoid any potential decomposition of the working fluid at high exhaust enthalpy operations. In addition, the combination of ORC-thermal oil assures stabilizing the thermal oil temperature in the evaporator. The thermal oil is a synthetic organic heat transfer fluid contains a mixture of diphenylethane and alkylated aromatics. It exhibits better thermal stability, particularly at the upper end of hot oil's use range, and significantly better low-temperature pumpability. It's critical temperature and pressure are 489 °C and 24 bar, respectively.

Displaced volume	7255 cc
Stroke	132 mm
Bore	108 mm
Compression ratio	17.5:1
Number of Cylinders	6
Number of Valves	4
Maximum Torque	1600rpm–1400 @ 1100Nm
Maximum Power	2300rpm @ 206kW
Optimum bsfc point	g/kWh 205≥

Table 3: Characteristics of the heavy-duty diesel engine

2.2 DESCRIPTION OF THE ORGANIC RANKINE CYCLE :

The organic Rankine cycle system consists of six main components: three heat exchangers (evaporator, recuperator, and condenser), turbine and two pumps as shown in Fig. 1. The thermal oil first extracts heat from the exhaust gas in the evaporator and passes through the recuperator to exchange heat with the working fluid. As a result, the working fluid is converted from liquid to gaseous state. The vapor then enters the radial turbine to rotate the blade generating power by the process of enthalpy drop. After that, the vapor enters the condenser to be condensed back to the liquid state and then pumped at high pressure to the evaporator, where it is heated again, starting a new cycle. It is worth mentioning that pressure and heat losses within the connecting pipes of the evaporators and condenser are neglected and only steady state operation is considered. To avoid extreme pressures and alleviate the safety concerns, only subcritical ORC is considered in this study. The working fluid considered in the current study is NOVEC649, $CF_3CF_2C(O)CF(CF_3)_2$, (which proved effective as shown in the previous study of the author (Alshammari, Karvountzis-Kontakiotis and Pesyridis, 2018). The fluid has critical pressure and temperature of 18.69 bar and 441.81 K, respectively. The boiling point is 322.2 K. The global warming potential of the fluid is 1 while it has zero ozone depletion potential.

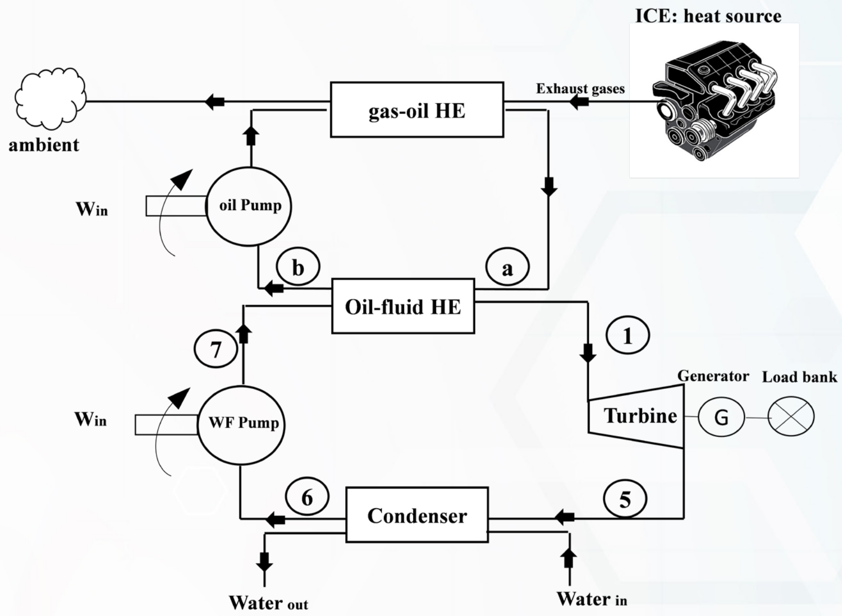

Fig. 1. Schematic of the powertrain system

2.2.1 ENERGY ANALYSIS :

The energy balance in the oil-working fluid heat exchanger is shown in Eq. (1). The subscripts are shown in Fig. 1.

$$Q_{in} = \dot{m}_{oil}\left(h_{oil,in} - h_{oil,out}\right) = \dot{m}_{wf}(h_1 - h_7) \quad \text{.......... 1}$$

The power output produced by the turbine can be obtained using Eq. (2). U4 and Cθ4 are the tip speed and tangential velocity at the rotor inlet. The isentropic total to static efficiency of the turbine is determined by Eq. (3).

$$W_{tur} = \dot{m}_{wf}(h_1 - h_5) = \dot{m}_{wf}(U_4 C_{\theta 4} - U_5 C_{\theta 5}) = \dot{m}_{wf}\Delta h_{act} \quad \text{.......... 2}$$

$$\eta_{ts} = \frac{(h_1 - h_5)}{(h_1 - h_{5,is})} \quad \text{.......... 3}$$

The energy balance in the condenser is shown in Eq (4). The condenser inlet pressure is considered as a constraint in the cycle modeling. To avoid water contamination in the condenser, the condenser inlet pressure is set to 160 kPa.

$$\dot{m}_{wf}\,(h_5 - h_6) = \dot{m}_{water} c_{p,water}\,(T_{\mathrm{out}} - T_{\mathrm{in}}) \quad \text{.......... 4}$$

The consumed power by the working fluid pump is determined by Eq. (5).

$$\dot{W}_{pump} = \dot{m}_{wf}(h_7 - h_6) \quad \text{5}$$

The cycle net power and thermal efficiency are, respectively, obtained using Eq. (6) and Eq. (7):

$$\dot{W}_{net} = \dot{W}_{tur} - \dot{W}_{pump} \quad \text{6}$$

$$\eta_{thermal} = \frac{\dot{W}_{net}}{Q_{in}} \quad \text{7}$$

2.2.2 EXERGY ANALYSIS :

Exergy analysis in the current work are available in (Yunus A. Cengel, 2006). For steady state conditions, the exergy balance can be obtained using Eq. (8). Q^. and W^. are, respectively, the heat and work of the system. $\dot{ex}_{out}$, $\dot{ex}_{in}$. and I are the exit exergy rate, inlet exergy rate and total irreversibility, respectively. The total irreversibility is the sum of components irreversibility.

$$\dot{Q} + \dot{W} = \dot{ex}_{out} - \dot{ex}_{in} + I \quad \text{8}$$

The exergy efficiency of the cycle can be calculated using Eq. (9). T_{water} is cooling medium temperature and set to 293 K.

$$\eta_{ex} = \frac{\dot{W}_{net}}{Q_{in}\left(1 - \frac{T_{water}}{\left(T_{oil,in} + T_{oil,out}\right)}\right)} \quad \text{9}$$

The heat recovery efficiency can be obtained using Eq. (10).

$$\eta_{rec} = \eta_{thermal}\frac{\left(T_{oil,out} - T_{oil,in}\right)}{\left(T_{oil,out} - T_o\right)} \quad \text{10}$$

2.2.3 RADIAL INFLOW TURBINE :

Radial-inflow turbine is one of most common types of rotor-dynamic machines. It is a work-producing device that has two essential parts (Fig. 2). The first part is the fixed part that consists of volute and stator (or nozzle) vanes. The volute introduces some swirl to the fluid, and the nozzle vanes guide the flow, accelerate it, and remove any circumferential non-uniformity. The second part is the rotating part, which is called rotor. In the rotor, the flow is further expanded to rotate the blades and produce mechanical work. Fig. 3 shows a meridional view of the turbine stage. The detailed modelling of the turbine can be found in the previous study of the author (Alshammari et al., 2020).

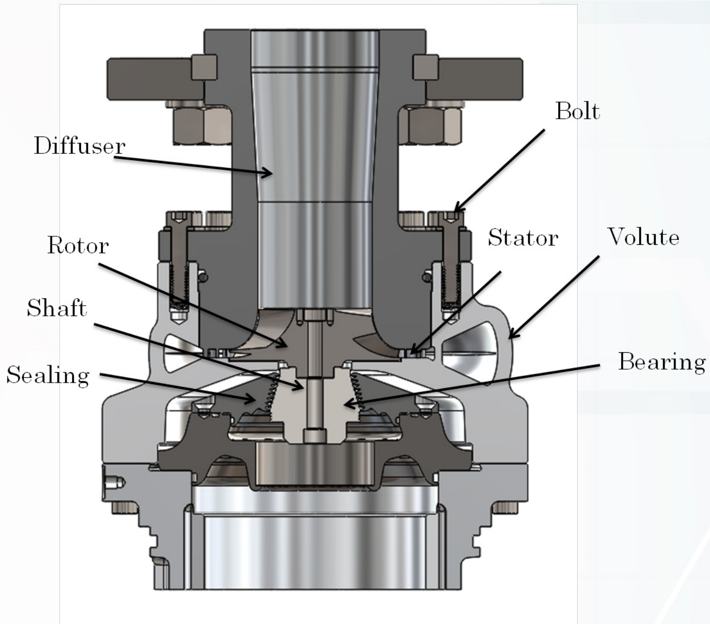

Fig. 2. Schematic of the turbine assembly

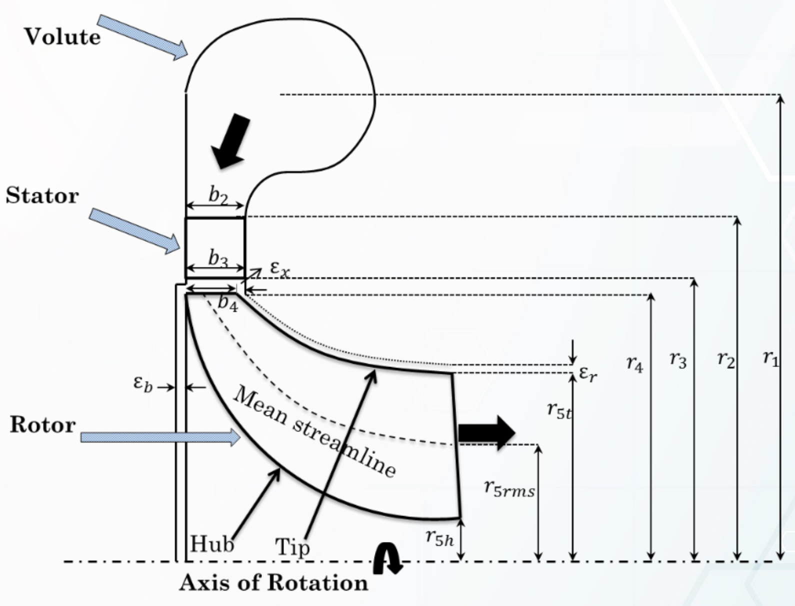

Fig. 3. Meridional view of the turbine stage.

3. RESULTS AND DISCUSSION :

3.1.1 GENERAL ANALYSIS :

The potential of recovery of wasted heat is firstly analysed. Fig. 4 represents the effects of turbine pressure ratio and inlet temperature on the working fluid mass flow rate. It is obvious from Fig. 4a that the mass flow rate increases linearly with increasing the pressure ratio. This trend is expected since more flow capacity is required in order to obtain the requested enthalpy drop. Fig. 4b, on other hand, indicates that the working fluid the mass flow rate decreases as the inlet temperature increases. Higher heat source temperature represents more available heat energy, which demands more working fluid. The decreased mass flow rate of the organic fluid with increasing the turbine inlet temperature is due to the large enthalpy at higher temperatures. Thus, a smaller mass flow rate of the organic fluid is required to receive a given heat at high temperatures.

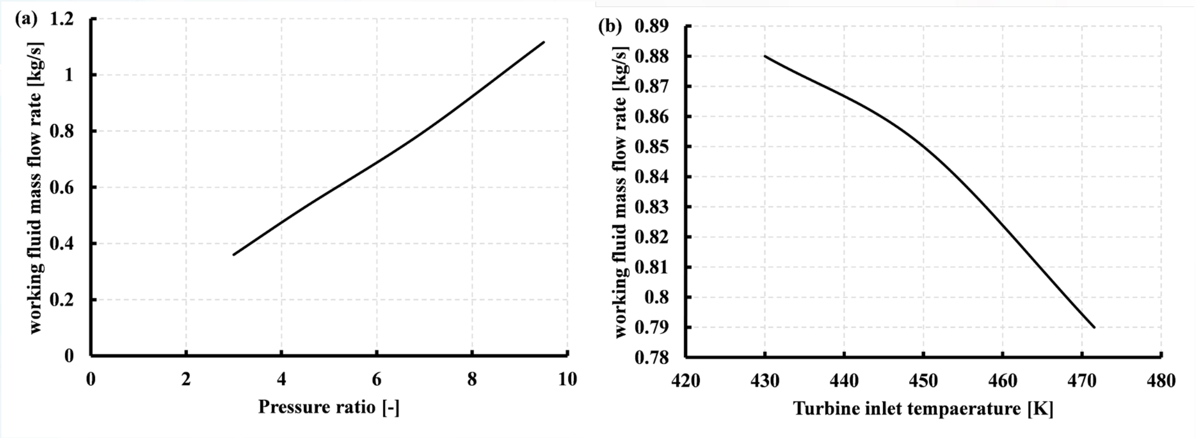

Fig. 4. Effects operating conditions on working fluid mass flow rate

The total heat recovery represents the potential of the ORC system as a WHR technology in internal combustion engines. It is evident from Fig. 5 that the total heat recovery increases with increasing the turbine pressure ratio and inlet temperature. As the pressure ratios increases, the enthalpy drop increases resulting in higher cycle net power, and consequently, higher recovery efficiency. From the definition of the recovery efficiency, Eq. (10), the turbine inlet temperature plays a vital role in improving the recovery potential. A maximum recovery efficiency of 8.59% is obtained at the design point.

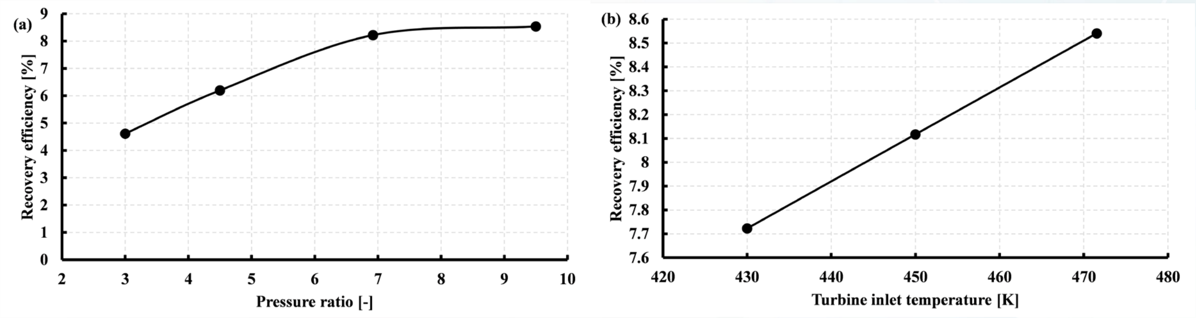

Fig. 5. Effects of operating conditions on recovery efficiency

3.2 EXERGY AND ENERGY ANALYSIS AT VARIOUS TURBINE PRESSURE RATIOS :

The expansion machine is the most critical component in the ORC system since it is responsible for power production. In automotive applications, exhaust gas conditions vary considerably especially that vehicles run mostly at partial loads. Therefore, it is of great importance to evaluate the turbine performance at off-design conditions. Fig. 6 shows that turbine efficiency increases with increasing the pressure ratio, reaching a maximum value and then decreases. This is common trend in high pressure ratio radial turbines due to the increased flow losses at high pressure ratios in which the flow reaching supersonic conditions. The turbine power, on the other hand, is a function of the enthalpy drop and turbine speed, equation (4). As shown in Fig. 6, turbine power increases significantly with increasing the pressure ratio due to the increased enthalpy drop.

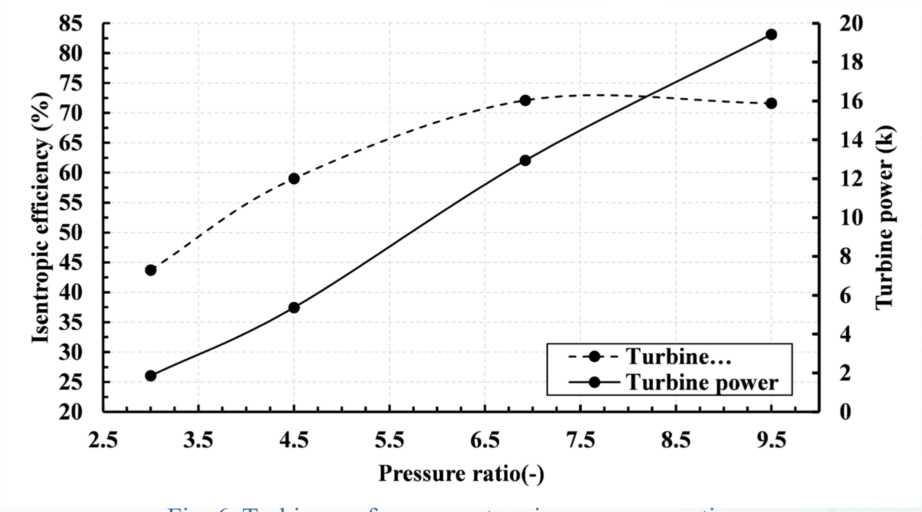

Fig. 6. Turbine performance at various pressure ratios

Fig. 7 depicts the exergetic and energetic efficiencies of the ORC systems at various turbine pressure ratios. It is evident that both exergy and thermal efficiencies increase with increasing the turbine pressure ratio presenting the same trend. Both efficiencies are functions of cycle Wnet which is a function of turbine power output. As shown in Fig. 6, turbine power increases substantially with increasing the pressure ratio. As a result, system efficiencies increase significantly. As the pressure ratio increases from 3 to 9.5, the exergy efficiency and thermal efficiency increase by 6.13% to 21.1% and from 2.25% to 7.76%, respectively.

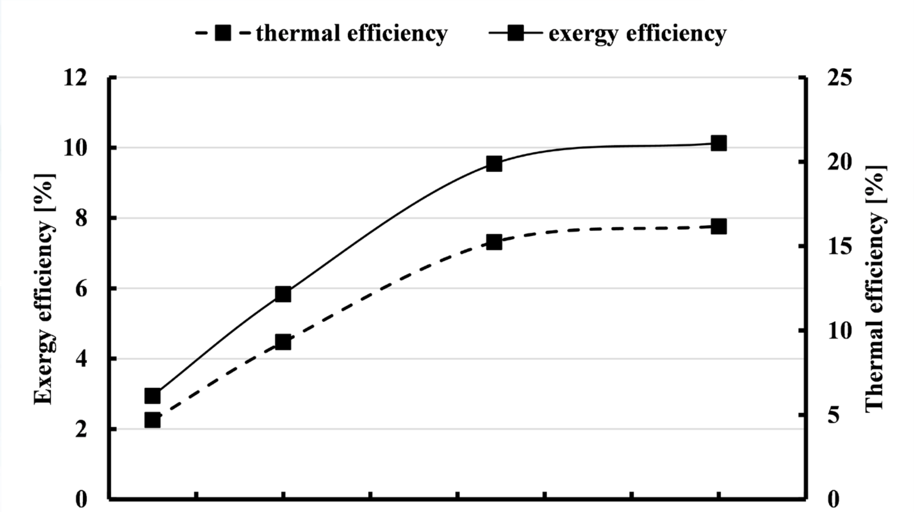

Fig. 7. Exergy and thermal efficiencies at various pressure ratios

Figure 8 depicts the exergy destruction for each component, and Fig. 9 shows the total exergy destruction. The highest exergy destruction occurs in the evaporator presenting 64% of exergy destruction rate. The reason for high exergy loss in the evaporator is the high outlet temperature of the working fluid which also clarifies the low recovery efficiency as shown in Figure.

Fig. 9 clearly indicates that increasing the turbine pressure ratio results in lower system irreversibility. This is expected due to the increased system exergy as well as energy efficiencies as shown in Fig. 7. The irreversibility rate decreases by 63% as the pressure ratio increases from 3 to 9.5.

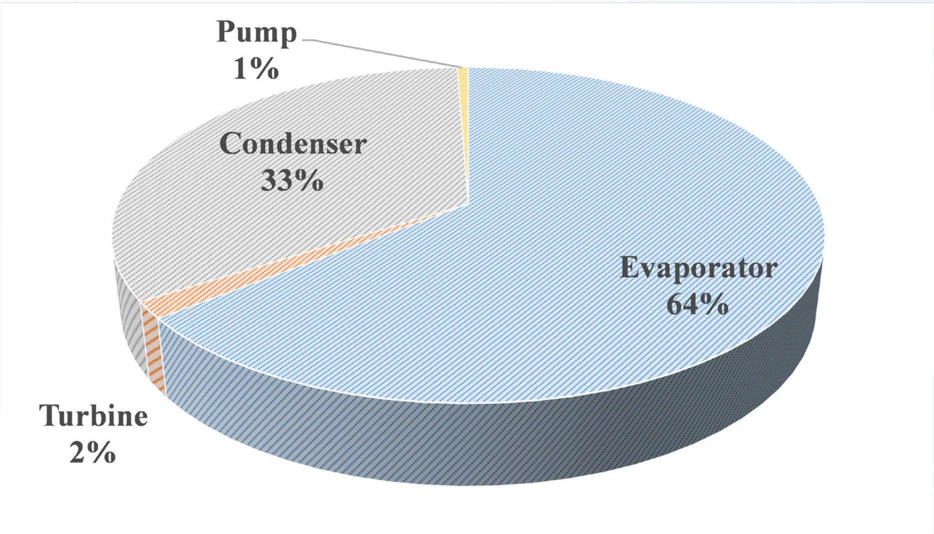

Figure 8. Exergy destruction for the ORC components

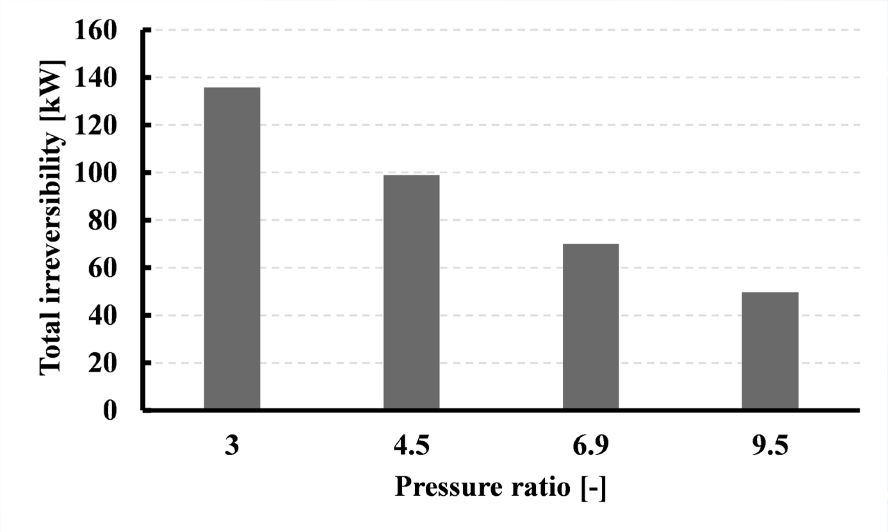

Fig. 9. System total irreversibility at various pressure ratios

3.3 EXERGY AND ENERGY ANALYSIS AT VARIOUS TURBINE INLET TEMPERATURES :

Turbine performance is explored at different turbine inlet temperatures as shown in Fig. 10. Turbine efficiency increases with increasing inlet temperature, reaching maximum value at the nominal inlet temperature (471.55 K). As temperature increases, the enthalpy drop becomes higher which results in higher turbine efficiency and power output. However, further increase in the temperature may result in efficiency deterioration due to the increased turbine losses at extreme temperatures.

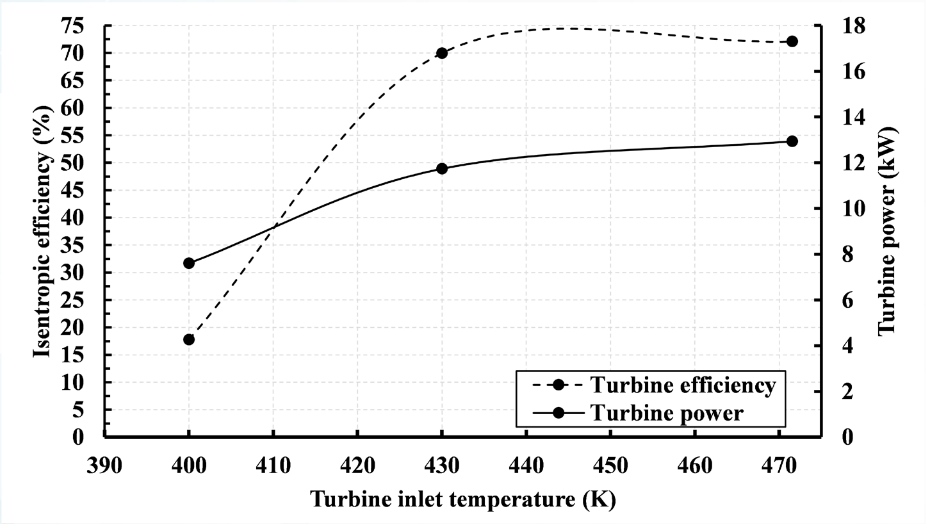

Fig. 10. Turbine performance at different turbine inlet temperatures

The exergy and energy efficiencies are depicted in Fig. 11. It is evident from the figure that increasing the turbine inlet temperature has considerable impact on both cycle exergy and thermal efficiencies. In fact, this is expected due to the higher efficiency recovery as shown in Fig. 5. Moreover, as the inlet temperature increases, the total enthalpy also increases resulting in higher turbine power, hence higher cycle net power. Similarly, system irreversibility decreases at higher temperatures as shown in Fig. 12.

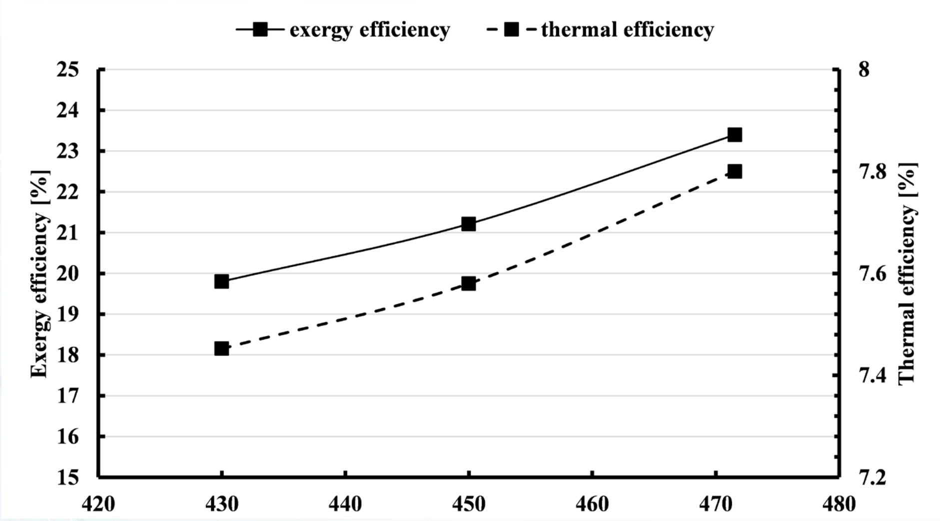

Fig. 11. Exergy and thermal efficiencies at different turbine inlet temperatures

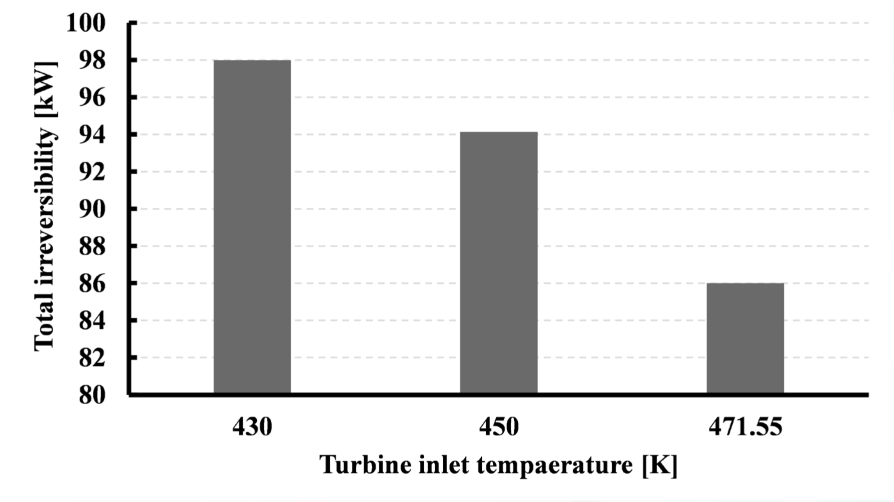

Fig. 12. System total irreversibility at different turbine inlet temperatures

4. CONCLUSION :

In the current study, energy and exergy analyses of automotive ORC systems were performed. The analyses were performed at various turbine pressure ratios and inlet temperatures. The results indicated the effectiveness of the ORC system in capturing the wasted heat in ICEs. The outcomes of the current study can be summarized as following:

- Turbine efficiency increased with increasing the pressure ratio, reaching the peak then decreased. The mechanical work, on the other hand, increased significantly due to the increased enthalpy drop. The maximum obtained efficiency and power were 72.1% and 19.42 kW, respectively.
- By capturing only 8.59% of the wasted heat, the system was able to produce mechanical power of 19.42 kW.
- Increasing turbine pressure ratio and inlet temperature had positive impact on both thermal and exergy efficiency. Maximum obtained exergy and energy efficiencies were 23.4% and 7.8%, respectively.
- The evaporator presented the highest exergy destruction of the value 64% followed by the condenser with 33%.

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	Nomenclature		
	Variables	Subscript	
7-1	Stations through ORC, Fig. 1	act	actual
act	actual	h	hub
C	Absolute velocity [m/s]	in	inlet
Cθ	Tangential velocity [m/s]	is	isentropic
ex	exergy	tur	turbine
h	Enthalpy [kJ/kg]	wf	Working fluid
I	Irreversibility	Greek Symbols	
m'	Mass flow rate [kg/s]	η	[-] Efficiency
N	Rotational speed [RPM]	Abbreviations	
P	Pressure [kPa]	ICE	Internal combustion engine
Q	Heat transfer [kJ/kg]	ORC	Organic Rankine cycle
s	Entropy [kJ/kg.k]	WHR	Waste heat recovery
T	Temperature [K]	WHR	Waste heat recovery
U	Tip speed [m/s]		
W	Work [kW]		

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