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Thermodynamic and Environmental Analysis of a Dual-Pressure Organic Rankine Cycle for Waste Heat Recovery from a Turbocharged Diesel Engine Fueled with Ethanol and Methanol Blends

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ABSTRACT: This paper refers to retrieving the lost heat of exhaust gas and working fluid of a turbocharged Diesel engine employing a Dual-Pressure Organic Rankine Cycle. The efficiency and emission of Diesel compression ignition engines are studied by considering a one-dimensional two-zone thermodynamic model. In the proposed system, exhaust gas and intercooler waste heat have been utilized in the high and low-pressure evaporators, respectively. The used cycle has the capability of reducing the irreversibility of the heat transfer process in the evaporators and increasing the turbines' power. Furthermore, hybrid fuels are used in the turbocharged Diesel engine to decrease the level of environmental pollutants. Accordingly, an exergy-based thermodynamic approach is employed to analyze a Diesel engine's performance and its emissions. The proposed engine includes Diesel fuel mixed with methanol and ethanol with different volume fractions of 5% and 10%. The results indicate a reduction in power and maximum brake torque of the engine as well as a remarkable decrement in the emission of pollutants such as nitrogen oxides and carbon monoxide (equal to 15%) by using the alcoholic compounds with Diesel fuel. Also, R123 is an appropriate coolant utilized in the Dual-Pressure Organic Rankine Cycle for recovering the lost heat of the Diesel engine.

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1- Introduction

Looking for fossil fuels' substitutions is required to pollutant reduction. Waste heat recovery can be one of the best approaches to solve the global energy crisis and growing- greenhouse gas emissions. Various efforts have been carried out to increase energy utilization and decrease the emission of greenhouse gases [1-3]. Internal Combustion Engines (ICEs) play a pivotal role in industry, agriculture, and transportation by supplying the required power for their startup and operation [4]. To date, a typical engine converts only 20–40% of the entire fuel into power, and the rest gets out of the system into the ambient by exhaustion and coolant [5]. Climate change is one of the main issues that humanity is currently facing due to carbon dioxide emissions caused by the use of fossil fuels. Organic Rankine cycles may play an important role in reducing these emissions since they can be operated by using industrial waste heat of renewable energies [6]. Hence, recapturing the wasted heat can be taken as a useful method to raise fuel consumption. One of the best solutions in this regard is the Organic Rankine Cycle (ORC), in which various heat sources with wide temperature ranges can be utilized. Turbocharged Diesel Engines (TDEs) are widely used for the generation of power [7, 8]. Diesel fuel leads to global environmental degradation effects such as air pollution, acid precipitation, climate change, greenhouse effect, etc. [1]. To reduce these effects, some researchers proposed the combination of the alcohols with the Diesel fuel [9, 10]. The pollutants such as CO, unburnt hydrocarbons (UHC), and CO_2 are significantly decreased while ethanol and/or methanol contents rise using the fuel blend [11].

Improving thermal efficiency and reducing carbon emissions are the permanent themes for internal combustion (IC) engines. In the past decades, various advanced strategies have been proposed to achieve higher efficiency and cleaner combustion with the increasingly stringent fuel economy and emission regulations, The Energy crisis and environmental pollution have become globally increasing concerns. The world has around 1.2 billion passenger cars and 380 million commercial vehicles, and these numbers are expected to increase significantly [12]. The reduction of fossil fuels and their destructive effects on the environment has necessitated the use of clean combustion. Higher lean combustion and compression ratio cause a high efficiency of fuel conversion in CI engines. Nevertheless, the high-temperature combustion chamber includes rich and poor areas, which produce soot and NO_x in conventional diesel working conditions, respectively [13]. Using other fuels instead of Diesel fuel can be an appropriate solution to cope with pollution. A combination of

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various fuels is another remedy in this regard. Today, using renewable fuels like Hydrogen and biofuels has a promising result in the reduction of emissions and fossil fuel consumption [2]. Utilizing alcoholic fuels and their combination with different fuels is economical and environmental. In order to decrease the level of pollutants emitted from CI engines, two alcoholic fuels of methanol and ethanol are combined with common diesel fuel [14, 15]. The quality of performance of novel fuels in CI engines also needs to be evaluated by exergy analysis and recognition of the irreversibility of the system [16, 17]. For several powers, Sarjovaara et al. [18] performed a dual fuel combustion study including 15% Diesel fuel and 85% ethanol. The results showed a reduction in NO_x and combustion efficiency by injecting more ethanol. Also, due to no optimized combustion stage, a fluctuation in thermal efficiency was observed. Hua et al. [19] investigated the Challenges and opportunities of the Rankine cycle for waste heat recovery from the internal combustion engine. recent advances in Rankine cycles for ICE-WHR are summarized and discussed. To evaluate results from various existing studies, a uniform evaluation standard, thermodynamic perfection, was proposed based on the benchmark of the heat source-based ideal thermodynamic cycle, which is determined by achieving ideal thermal matching to external boundary conditions. Kose et al. [20]used ORC to recover waste heat from gas turbines, and the impact of turbine inlet pressure and temperature on the system were analyzed, energy and exergy efficiencies of the overall triple combined system (GT-SRC-ORC) was calculated by depending on pressure, temperature and working fluids. As a result, the best working fluids among the selected fluids was found as R141b. Candan et al. [21] studied the effects of methanol combination on single-cylinder diesel engine performance and pollutants emissions. They added methanol with fractions of 5, 10, and 15% to diesel and showed a rise in NO_x emission and a fall in emission of Carbon dioxide, HC, and Carbon monoxide. Also, it declined shaft torque and achieved power. Jeongwoo et al. [22] assessed the effects of ethanol substitution on the dual fuel single-cylinder diesel engine performance and pollutants emissions of under various loads. The proportion of Ethanol alters between zero and 50% of the total applied energy, and each fuel is directly injected into the cylinder. The results indicated NO_x and PM reduction by injecting more ethanol. Also, despite raising the ethanol proportion to 63% for middle loads, ethanol with high proportion was not used due to insufficient energy for slight loads and surge of pressure in big loads. Sayin [23] investigated the combination influence of ethanol and methanol with diesel fuel in CI engines. He employed a Diesel engine by single-cylinder direct injection with different rotational speeds and figured out that blending ethanol and methanol with Diesel raises the emission of BSFC and NO_x and causes a fall in brake's thermal efficiency and production of HC and CO. Fayyazbakhsh et al. [24] researched refinement of diesel fuel formulation of oxygenated additives method to increase the efficiency of the engine and diminish its emissions. The combination of oxygenated fuels with alcohol fuels such as methanol, ethanol, and n-butanol was studied by considerations of their effects. Negative influence was seen on exhausted pollutants in the obtained result by increasing the level of CO, CO_2, and NO_x as well as the engine speed in some cases. Also, adding the oxygen fuels enhanced and reduced BSFC and BTE, respectively. To improve the influence of the triple mixtures (biodiesel-diesel-alcohol), Venue et al. [25] used DEE (diethyl ether) in an experiment to amplify the ignition and raise the combustion duration, BSFC, and in-cylinder pressure. Nevertheless, NO_x and PM underwent a decrease since the ignition delay became shorter. While adding DEE to MBD raises CO, PM, and CO_2, and reduces HRR, combustion duration, BSFC, and in-cylinder pressure. Ozgur et al [26] investigated the effects of mixed diesel fuel and 20% ethanol with a speed range of 1000 to 2600 rpm. Based on the achieved results, utilizing the blend causes a fall in torque, power, and CO production and a rise in BSFC and NO_x production. Moreover, energy and exergy efficiencies saw a reduction when E2 was used.

The effect of injection of 0 to 40% methanol/ethanol on a single-cylinder diesel engine was investigated by Jamrozik [27]. The injection of up to 30% improved the thermal efficiency and had no sensible effect of IMEP and resulted in a striking impact on CO, HC, and CO_2 production. However, further injection diminished the cylinder pressure and destabilized the operation. It also enhanced the efficiency and had no effect on IMEP. Köse et al [28] studied Kalina and Organic Rankine Cycles, which are the most important low-temperature energy conversion systems in the utilization of low-temperature industrial west heat with a temperature of 250 □ and a mass flow of 10 m^3/s, were examined and compared to each other in terms of energy, exergy efficiency, economic outlook, and environmental effects. Ma et al [29] studied the second principle of thermodynamics. They found DMDF less destructive in terms of thermal exergy and saw a lower level of exhaust exergy destruction in high loads under the effect of various temperatures on the methanol's incomplete combustion. In addition, the decrement of irreversible exergy and exhaust waste enhanced the second law efficiency at higher temperatures of methanol and coolant. Exergy and energy analysis was carried out by Khoobbakht et al. [30] on a CI engine operated by biodiesel-diesel and ethanol-diesel combination. They substantiated that raising the proportion of biodiesel and methanol reduced the exergy efficiency and destroyed about 43.09% of fuel exergy. A 36.61% and 33.81% energy and exergy efficiencies were reported in their results, respectively. The effects of various operating parameters on the DMDF engine were investigated by Baodong Ma et al. [29] by an exergy analysis. According to their achievement, exhaust chemical exergy destruction decreases and the exergy efficiency improves by enhancement of intake temperature. Moreover, when the temperature of coolant and methanol is higher, the efficiency of exergy improved due to the reduction of irreversible exergy waste of the combustion and the waste of exhaust exergy. Various studies declare that more than 50% of the energy consumption that is used in the industry is wasted as heat gas or vapor [31]. Using low-cost techniques for power generation from waste heat increases the system's efficiency. Comprehensive reviews of Organic Rankine Cycles (ORCs) triggering via waste heat were conducted by Campana et al. [31], Castelli et al [32], Mahmoudi et al [33], Nami et al [34], Kolahi et al [35]. Pan et al. [36] studied the combination of ORC with a heat pump for loss heat rehabilitation. They resulted that, the proposed system reduces pollutant emissions such as CO and CO_2 by 11% and the electrical efficiency increases by 6.5%. Malouf et al. [37]reported the net power increase of ORC for low-temperature waste heat recovery by 10% compared to the conventional ORCs, when they used the waste heat of the flue gases (<120 C) exiting industrial processes for the evaporator. The combination of the supercritical CO_2 (S-CO_2) with an ORC for ICE waste-heat recapturing is proposed by Song et al [38]. They concluded that the combined system's output power by waste heat recovery is 40-70% higher than the simple case. Yu et al. [39] are Investigation and optimization of a two-stage cascade ORC system for medium and low-grade waste heat recovery using liquefied natural gas cold energy. The thermodynamic and economic models are developed to investigate the proposed system performance under steady-state conditions. The influence of the important thermodynamic parameters, e.g., the ORC turbine inlet temperature, the high pressure of first-stage ORC and the outlet temperature of condenser are investigated. Ge et al. [40] demonstrated the dual-loop ORC using the working fluid of the zeotropic mixtures for loss heat retrieval of the ICE. They found that, using the mixture of cyclopentane/cyclohexane and benzene/toluene in HP branch and R600a/R601a mixtures in the LP branch, the system net power output increased by 2.5–9.0 % and 1.4–4.3 %, respectively. Song and Gu [41] investigated the dual-loop ORC with the wet steam expansion for loss heat retrieval of the engine. They achieved that, by this method, the net power output increased by 11.6 %.

Organic Rankine Cycle has gained attraction in Diesel engines due to its outstanding capability to improve thermal efficiency and fuel economy by recovering heat from low-temperature waste heat sources, steady-state ORC models are suitable for analyzing stationary diesel engine applications, highway, and marine-based vehicular diesel engine applications due to nearly steady exhaust conditions [42]. Recently, the applications of the Dual-Pressure Organic Rankine Cycle (DPORC) at industry have increased Because of a lower total irreversibility and a higher net power [43, 44]. The application of both loss heat retrieval of turbocharged diesel engines with compositions of ethanol and methanol and the two-evaporator ORC are the options to increase fuel efficiency. A meticulous review of the literature reveals a plethora of investigations devoted to the study of the performance and products of diesel engines fueled with the combination of methanol-diesel and ethanol-diesel. These investigations substantiate the reduction of pollutants' emission and various behavior of NO_x emission, combustion trends, and energy efficiency by increment of ethanol or methanol. The impacts of alcohol-diesel on engine performance and emission of pollutants have also been assessed. However, no study has compared the effects of ethanol-diesel and methanol-diesel combined fuels on an engine's energetic and exergy performance and the level of its products.

Therefore, this paper aims to retrieve the lost heat of exhaust gas and working fluid of a turbocharged Diesel engine by employing a Dual-Pressure Organic Rankine Cycle (DPORC). Additionally, R123, R1234ze, n-pentane, n-hexane, Iso-pentane, D4, and MDM have been used as working fluids in the proposed DPORC. In section 1, an introduction of the present work and similar studies is presented. In section 2, the proposed system is introduced, the Dual-Pressure Organic Rankine Cycle (DPORC) utilizing the exhaust and coolant waste heat of the turbocharged diesel engine worked with ethanol/methanol mixture. In the following, the introduction of the modeled engine and an energy and exergy-based thermodynamic approach is employed to analyze a Diesel engine's performance and its emissions, then thermodynamic validation is studied. The used cycle has the capability of reducing the irreversibility of the heat transfer process in the evaporators and increasing the turbines' power. When the calibration process is concluded, In section 3, the study focuses on the use of methanol/ethanol blends, hybrid fuels are used in the turbocharged Diesel engine to decrease the level of environmental pollutants. The results of environmental analysis and Thermodynamics analysis of the waste heat recovery process have been investigated. Finally, in section 4, a conclusion of the results and suggested works for future studies are presented. In this regard, the aims are as follows:

To utilize the exhaust and cooling fluid loss heat of a turbocharged diesel engine by ethanol/methanol mixture employing a DPORC

To study the influences of decision variables on the net output power, exergy, and energy efficiency.

To study the key parameters and pollutants with alcoholic mixtures.

To evaluate more factors like outlet pressure and temperature of the intercooler to appraise the simulation.

To study the factors related to the exergetic performance of the developed engine like exergy efficiency and destruction.

To demonstrate the effects of revolution on the exergy destruction.

2- Modeling and validation

Figure 1 demonstrates the diagram of the proposed system. Detailed specifications of simulation parameters are brought in Table 1. To recover the waste heat, an inline 6-cylinder turbocharged diesel engine has been selected [38]. Tables 2 and 3 demonstrate the parameters of the Diesel engine and the exhaust gas composition, respectively. Additionally, R123, R1234ze, n-pentane, n-hexane, Iso-pentane, D4, and MDM have been used as working fluids in proposed DPORC [39, 40]. The seven low global warming potential (GWP) coolants are used for the thermodynamic evaluation of the DPORC. The thermodynamic and environmental features of the seven coolants are shown in Table 4. Simulation of the process is

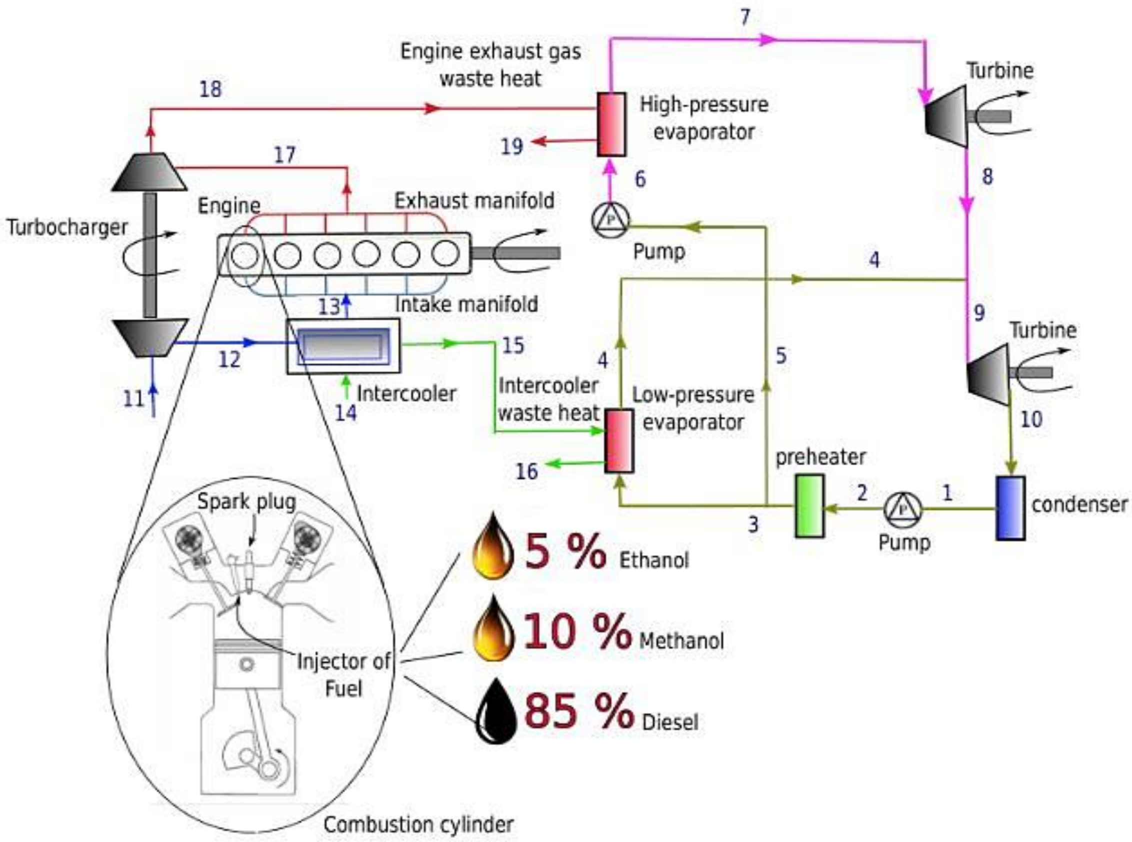


Fig. 1. The Dual-Pressure Organic Rankine Cycle (DPORC) utilizing the exhaust and coolant waste heat of the turbocharged diesel engine worked with ethanol/methanol mixtures

Table 1. Some given parameters for DPORC systems.

Parameter	Value
Inlet temperature of HP turbine	470-540 K [45]
HP turbine inlet pressure	4000-15000 kPa
Temperature of LP evaporator	320-350 K
Isentropic efficiency of turbines	0.8
Isentropic efficiency of compressor	0.75

Table 2. Engine specification [46]

Engine name	Weichai WP12.336E40 [46]
Valves timing	EVC: 21˚ ATDC EVO: 49˚ BBDC IVC: 34˚ ABDC IVO: 20˚ BTDC
Maximum lift of valves (mm)	Intake valve: 10.24 Exhaust valve: 12
Maximum torque (Nm)	1600 at 1000-1400 rpm
Maximum power (kW)	247 at 1900 rpm
Compression ratio	17:1
Displacement (L)	12
Stroke (mm)	155
Bore (mm)	126
Number of cylinders	6

Table 3. Reference environment's molar ratio

Material	Y_i^e (%)
N_2	75.67
O_2	20.35
CO_2	0.3
H_2O	3.12
other	0.83

Table 4. Elaborated features of the seven organic fluids [47, 48].

Working fluids	M (Kg/kmol)	T_{cr} (K)	P_{cr} (MPa)	ODP	GWP	Behavior	References
R123	153.93	486.54	3.668	0.02	77	isentropic	[47]
R1234ze	114.04	382.52	3.636	0	6	dry	[47, 48]
n-pentane	72.15	469.65	3.363	0	6>	dry	[47, 48]
n-hexane	86.18	508.82	3.03	0	6>	dry	[47, 48]
Isopentane	72.15	460.35	3.38	0	20	dry	[47, 48]
D4	236.43	586.5	13.22	0	small	dry	[47, 48]
MDM	236.53	564.09	14.15	0	small	dry	[47, 48]

conducted by EES software.

DPORC consists of high and low-pressure branches to retrieve the loss of heat from the exhaust gas and intercooler water, respectively. The saturated coolant is pressurized by the low-pressure pump and is then separated into two sections (line 2-3 and line 2-5). Firstly, the Organic fluid of the lower-pressure branch enters into the low-pressure evaporator afterward the heat transfer process is done (process 3–4). The other section enters into the pump and higher pressure is achieved (process 5–6), and then the working fluid of this branch enters into the high-pressure evaporator and is heated. The state of this point is supercritical. The supercritical fluid enters the high-pressure turbine and is produces the acquired mechanical power. The pressure of working fluid at the outlet of the high-pressure turbine is equal to the low-pressure vapor state (lower branch). The LP vapor from the LP evaporator outlet and high-pressure branch vapor enters into low-pressure turbine and auxiliary power is produced. Finally, after condensation at the condenser, the lower-pressure working fluid is pumped evaporators. After achieving the required power, the cycle is complete. Due to the high range temperature of the engine exhaust gas and medium range temperature of intercooler water, the DPORC consists of higher and lower evaporators. The heats of the exhaust gas and intercooler water are transferred to the HP and LP loops through the HP and LP evaporators, respectively. Fig. 2 depicts the T-s diagram (temperature-entropy) of the Dual-Pressure Organic Rankine Cycle (DPORC). Above mentioned descriptions of the proposed system are shown in Fig. 2. The supercritical and subcritical points of DPORC are demonstrated in this figure.

The specifications of waste heat parameters of the engine versus speeds are presented in Table 5. Also, the features of diesel, ethanol, and methanol fuels are demonstrated in Table 6.

The turbocharged diesel engine operated by ethanol/methanol mixture consists of various components such as compressors, turbines, valves, and manifold tubes. At line 11 air enters the compressor and after the compression process enters the intercooler (line 12) to lower its temperature, the working fluid used for recovering waste heat in the intercooler (lines 14-16) is water. Then, the intake flow goes through the inlet manifold- passes the inlet valve, and flows into the cylinder. After combustion, the hot gases exit from the outlet manifold. These gases had a good potential for heat sources to use at DPORC evaporators. Additionally, the exhaust gas temperature depends on the ethanol and methanol blending ratio. Exhaust gases after exiting the turbine are used as the heat source in the high-pressure evaporator and then released into the environment (line 19). The brake torque is calculated by the energy of combustion. For the simulation, it is essential to define the model of the combustion and heat transfer process meticulously. To fulfill this matter, the Woschny approach [52] is used. The Wiebe model is employed to calculate the rate of energy emission per crank angle for the combustion sub-model

2- 1- Engine modeling

For the studied engine, three stages of premixed combustion, mixing-controlled combustion and late combustion are taken in the Wiebe approach, whose constants

Table 5. Waste heat parameters of the engine [46].

Engine speed (rpm)	Exhaust gas		Engine coolant water	
	Temperature (°C)	Flow rate (m^3/min)	Temperature (°C)	Flow rate (lit/min)
1600	623	12.84	90	147
2200	661	18.72	90	237
2600	697	22.2	90	249

Table 6. Characteristics of the employed fuels.[49-51]

	Diesel	Methanol	ethanol
Chemical formula	C_7H_{16}	CH_3OH	C_2H_5OH
Molecular weight (kg/kmol)	100	32	46
Auto-ignition temperature (°C)	257	464	423
Cetane number	52.6	4	6
Lower heating value (Mg/kg)	42.76	20.27	28.40
Stoichiometric air/fuel ratio	14.7	6.66	8.96
Density (g/cm^3)	0.84	0.79	0.78

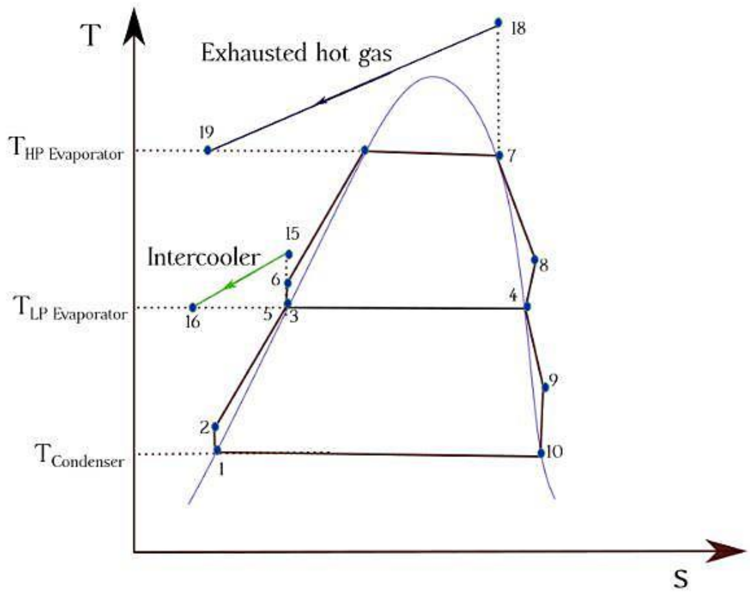


Fig. 2. T–S diagram of DPORC

are calculated as below [53]:

$$WC_p=\left[\frac{\text{Premix Duration}}{2.302^{\frac{1}{(E_p+1)}}-0.105^{\frac{1}{(E_p+1)}}}\right]^{-(E_p+1)} \tag{1}$$

$$WC_M=\left[\frac{\text{Main Duration}}{2.302^{\frac{1}{(E_M+1)}}-0.105^{\frac{1}{(E_M+1)}}}\right]^{-(E_M+1)} \tag{2}$$

$$WC_T=\left[\frac{\text{Tail Duration}}{2.302^{\frac{1}{(E_T+1)}}-0.105^{\frac{1}{(E_T+1)}}}\right]^{-(E_T+1)} \tag{3}$$

The rate of release of energy per crank angle is stated as [53, 54]:

$$\begin{aligned} &x(\theta)=(CE)(F_p) \\ &[1-expexp(-(WC_p)(\theta-SOI-ID)^{(E_p+1)})] \\ &+(CE)(F_M)[1-expexp(-(WC_M)(\theta-SOI-ID)^{(E_M+1)})] \\ &+(CE)(F_T)[1-expexp(-(WC_T)(\theta-SOI-ID)^{(E_T+1)})] \end{aligned} \tag{4}$$

Using Chen-Flynn sub-model, the friction model is expressed as[53, 54]:

$$FMEP=C_FMEP+C_1P_CP+C_2C_p+C_3C_p^2 \tag{5}$$

By having at disposal, the brake power and then BSFC are obtained by the following relations [54]:

$$P_e = M_t\left(\frac{2\pi\pi N}{60}\right) \tag{6}$$

$$BSFC=\left(\frac{\dot{m}_f\times 3600}{P_e}\right) \tag{7}$$

By recruiting experimental results through the burn rate and preparing them for the software, the input variables are assessed. By employing GT-Power the simulation is carried out in one dimension. The engine's performance and products are calculated by a thermodynamic two-zone model. Although the recruited model has a low precision compared to its 3-D counterparts, it can save calculation time. Also, for the calculation of NOx and CO production levels, the Zeldovich model and EngCylCO sub-model are employed.

2- 2- Energy analysis

The power consumed by the LP/HP pumps and the generated power by the LP/HP turbines are respectively written as follows:

$$\dot{W}_{p,Lp} = \dot{m}\left(h_2 - h_1\right) \quad \eta_{isen,p,Lp} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{8}$$

$$\dot{W}_{p,HP} = \dot{m}_5\left(h_6 - h_5\right) \quad \eta_{isen,p,HP} = \frac{h_{6s} - h_5}{h_6 - h_5} \tag{9}$$

$$\dot{W}_{T,LP} = \dot{m}_9\left(h_9 - h_{10}\right) \quad \eta_{isen,T,LP} = \frac{h_9 - h_{10}}{h_9 - h_{10s}} \tag{10}$$

$$\dot{W}_{T,HP} = \dot{m}_8\left(h_8 - h_7\right) \quad \eta_{isen,T,HP} = \frac{h_7 - h_8}{h_7 - h_{8s}} \tag{11}$$

The transferred heat in the LP/HP evaporators is calculated by:

$$\dot{m}_g\left(h_{18} - h_{19}\right) = \dot{m}_{HP}\left(h_7 - h_6\right) \tag{12}$$

$$\dot{m}_c\left(h_{15} - h_{16}\right) = \dot{m}_{LP}\left(h_4 - h_3\right) \tag{13}$$

The heat released in the condenser is calculated as:

$$Q_{cond} = \dot{m}_2\left(h_{10} - h_1\right) \tag{14}$$

2- 3- Exergy analysis

To gain precise results about the quality of work, it is crucial to calculate the irreversibility and exergy efficiency of the system. Regarding this, the stream exergy is articulated as:

$$e_x = h_x - h_0 - T_0\left(s_x - s_0\right) \tag{15}$$

Where $_0$ and $_x$ subscripts refer to the reference state (dead state) and component index. It is notable that the chemical exergy is neglected. Exergy destructions of LP/HP pumps and LP/HP turbines are respectively calculated by:

$$D_{p,LP} = \dot{m}_1\ T_0(s_2 - s_1) \tag{16}$$

$$D_{p,HP} = \dot{m}_5\ T_0(s_6 - s_5) \tag{17}$$

$$D_{T,LP} = \dot{m}_8\ T_0(s_8 - s_7) \tag{18}$$

$$D_{T,HP} = \dot{m}_{10}\ T_0(s_{10} - s_9) \tag{19}$$

LP/HP Evaporators exergy destruction rate:

$$D_{ev,LP} = T_0\left[\dot{m}_{LP}(s_4 - s_2) - \dot{m}_c(s_{c,in} - s_{c,out})\right] \tag{20}$$

$$D_{ev,HP} = T_0\left[\dot{m}_{HP}(s_7 - s_6) - \dot{m}_{h,gas}(s_{h,gas,in} - s_{h,gas,out})\right] \tag{21}$$

Condenser exergy destruction rate:

$$D_{cond} = \dot{m}_2 T_0(s_{10} - s_1) \tag{22}$$

Ignoring the exergy of waste heat of the lubricant and engine, the exergy destruction is achieved via the following equation. if the engine can be obtained as:

$$D_d = A_{in} - (A_{exh} + A_{cw} + A_{bp}) \tag{23}$$

By taking only thermochemical exergy into account of fuel exergy, we have [55]:

$$A_{in} = A_{in}^{tm} + A_{in}^{ch} \tag{24}$$

$$A_{in}^{tm} = (h - h_0) - T_0(s - s_0) \tag{25}$$

$$\begin{aligned} A_{in}^{ch} = \dot{m}_f \times [&1.0401 + 0.1728\tfrac{h}{c} + 0.0432\tfrac{o}{c} \\ &+ 0.2169\tfrac{s}{c}(1 - 2.0628\tfrac{h}{c})] \times LHV \end{aligned} \tag{26}$$

The thermomechanical and thermochemical terms of exhaust gas exergy are calculated as below[52]:

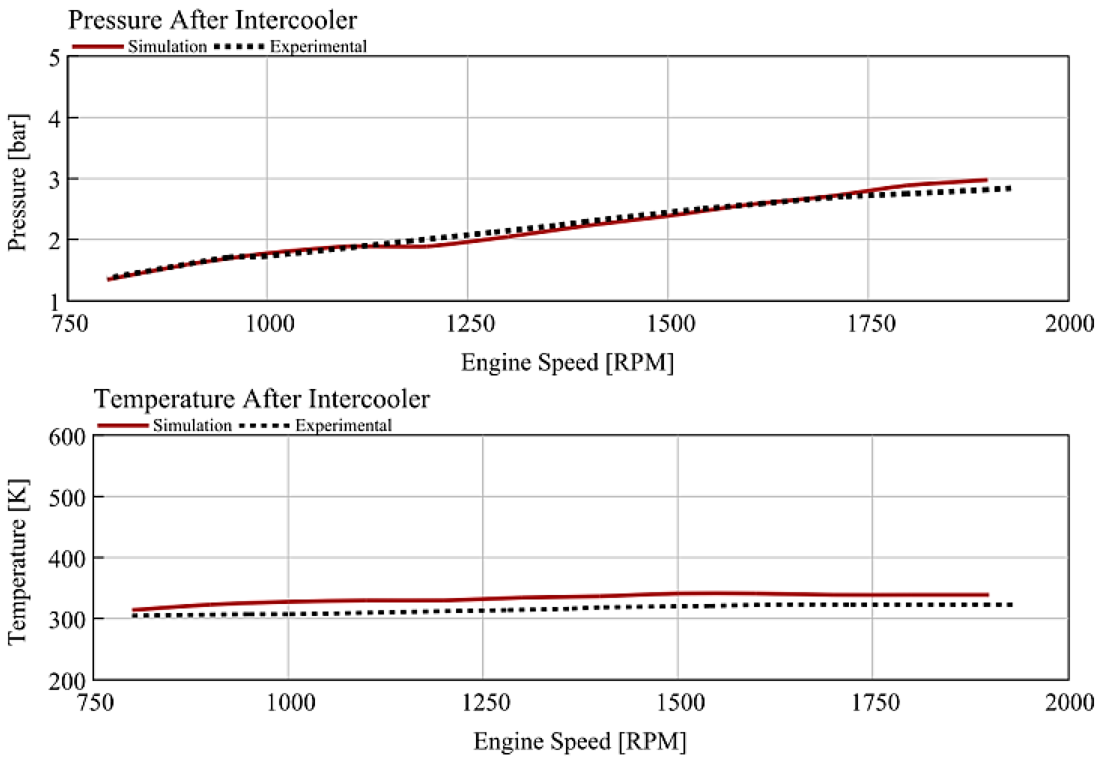


Fig. 3. Comparison of reported results with those of the simulation for pressure and temperature after intercooler.

$$A_{exh} = A_{exh}^{tm} + A_{exh}^{ch} \tag{27}$$

$$A_{exh}^{tm} = \dot{Q}_{eg} - [\dot{m}_{eg} \times T_0 \times \left\{ C_{pe} \times \ln\ln\left({T_{exh}}/{T_0} \right) - R_{eg} \times \ln\ln\left({P_{exh}}/{P_0} \right) \right\}] \tag{28}$$

$$A_{exh}^{ch} = \bar{R} T_0 \sum_{i=1}^{n} a_i \ln\ln\left(\frac{Y_i}{Y_i^e} \right) \tag{29}$$

On the other hand, the exergy of the coolant is assessed by as:

$$A_{CW} = \dot{Q}_{CW} \left(1 - \frac{T_0}{T_{CW}} \right) \tag{30}$$

$$\dot{Q}_{CW} = \dot{m}_{CW} C_{CW} \left(T_{15} - T_{16} \right) \tag{31}$$

The brake power exergy in the engine is stated as follows:

$$A_{BP} = \frac{(2\pi \times N \times T)}{60000} \tag{32}$$

Finally, the exergy efficiency is written as [52]:

$$\eta_{II} = \left(1 - \frac{A_d}{A_{in}} \right) \times 100 \tag{33}$$

2- 4- Thermodynamic validation

2- 4- 1- Model Validation of Turbocharged Diesel Engine

The revealed data of an experiment are taken into consideration to confirm the precision of the employed model. The results achieved for the compressor and turbine show the most deviation from the experimental data due to a 1-D calculation. To cover this error, the turbocharger efficiency and intercooler wall coefficients are considered. After this attempt, the results are compared and illustrated in Fig. 3.

The precision of simulation is about 6% regarding the farthest deviation, which belongs to the intercooler's output temperature at 1500 rpm. The origin of the error is because of the physical complexity of the intercooler in ICE. In other words, in the one-dimensional model, it is considered a tube

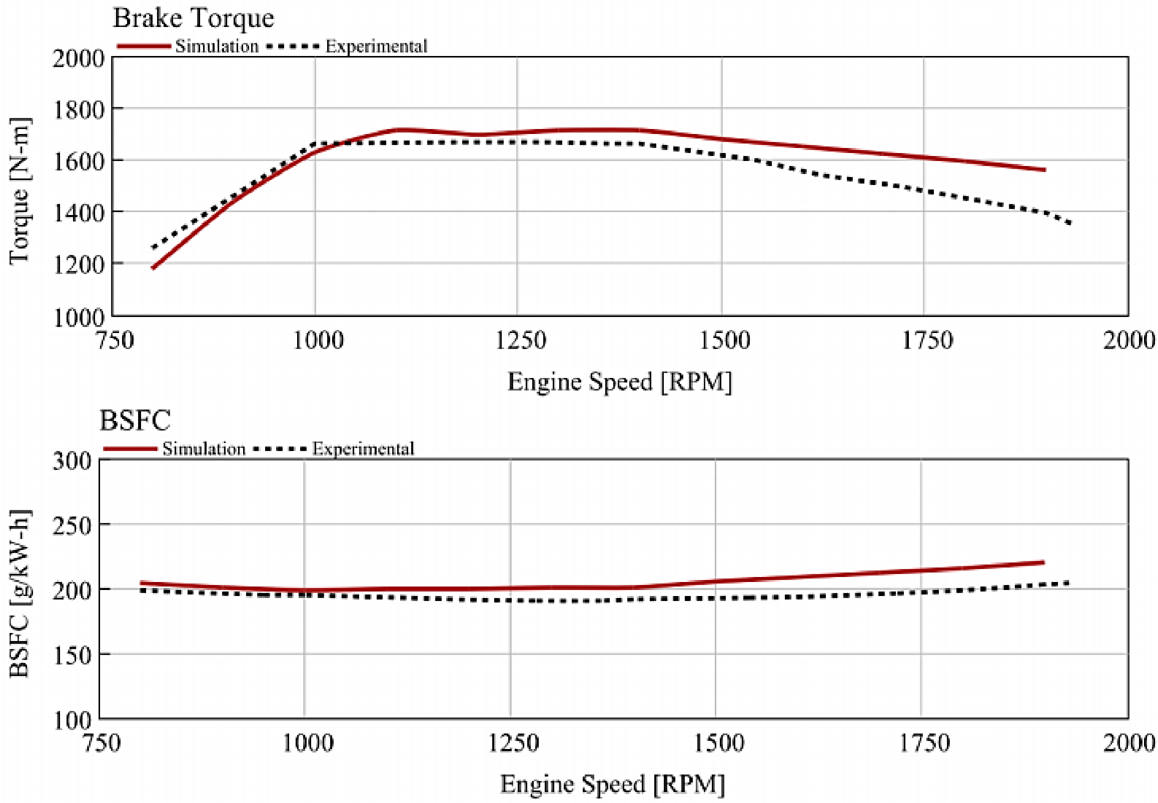


Fig. 4. BSFC and Brake torque for numerical and experimental [54] works.

containing a high rate of heat transfer instead of as a heat exchanger. The in-cylinder heat transfer is the main element of deviation in 1-D simulations because of the mentioned reason. Then, the above-mentioned calibration is carried out to cater to the compatibility of experimental and numerical results of the brake torque which are displayed in Fig. 4.

The reason for the discrepancy of compared results related to brake torque and hence in BSFC is due to a poor prediction of the sub-model to model the heat transferring between the wall and high-temperature gases. As can be seen in Fig. 3 and 4, the engine model is capable of predicting the Weichai WP12.336E40 engine's performance. Accordingly, the highest error with 12% is for the brake torque at 1900 rpm, for the pressure after the intercooler, the highest error with 11.05% at 1200 rpm, the highest error with 6.19% is for the temperature after the intercooler at 1400 rpm and the highest error with 8.85% is for the specific brake fuel consumption at 1900 rpm.

To validation of the Dual-Pressure Organic Rankine Cycle results is compared by Ref.[45]. The considered operational conditions such as pressures and temperatures are the same with Ref.[45], and Table 7 indicates the compared results of the present work and DPORC with Ref.[45]. As shown, the comparison of present results and Ref. [45]indicate a lower difference between them, which demonstrates good accuracy.

3- Results and discussion

3- 1- Environmental analysis

When the calibration process is concluded, the study focuses on the use of methanol/ethanol blends. Based on the reviewed literature, using 5 to 10% ethanol and methanol in Diesel fuel culminates in the most desirable results in ICEs [55]. Regarding this, M5, E10, E5, and M10 fuels are used in the present investigation.

Fig. 5 illustrates the results of using alcoholic compounds for brake-specific fuel consumption and brake torque. The fuel's heating value decreases by using additives in comparison with D100, which decreases the maximum torque. More decrement is shown by the injection of additives, and roughly 10% reduction occurs in torque when M10 fuel is used instead of D100. Since the spray pattern is similar for each fuel, the difference in brake torque is the only cause of nuance in the specific brake fuel consumption plot.

Due to the existence of an oxygen bond in the alcohols the quality of combustion increases and causes an enhancement in the brake-specific fuel consumption. Since the Diesel's heating value is more than that of additives, the fuel injection goes up to a certain value of output power. On the other hand, the levels of NO, CO, and CO_2 are illustrated in Fig. 6 to 8. Also, air to fuel ratio as an influential factor in the concentration of CO and CO_2 is delineated. It is inferred from

Table 7. Comparison of present results with Ref. [45]

	Ref. [45]	Present Work	Ref. [45]	Present Work	Ref. [45]	Present Work	Ref. [45]	Present Work
Fluids	$\dot{m}_{wf}$ (kg/s)	$\dot{m}_{wf}$ (kg/s)	P_{net} (kW)	P_{net} (kW)	$\eta_{ex,DPORC}$ (%)	$\eta_{ex,DPORC}$ (%)	$\eta_{en,ORC}$ (%)	$\eta_{en,ORC}$ (%)
R245fa	0.06	0.053	13.8	13.6	72.3	64.3	9.5	9.01
Pentane	0.02	0.017	14	13.87	72.3	67	9.6	9.1
Cyclohexane	0.02	0.015	14.7	14.3	73	67	10.1	9.7
Toluene	0.02	0.019	14.9	14.5	73	67.2	10.3	9.65
Average Error	+0.04		+0.2575		+6.275		+0.51	

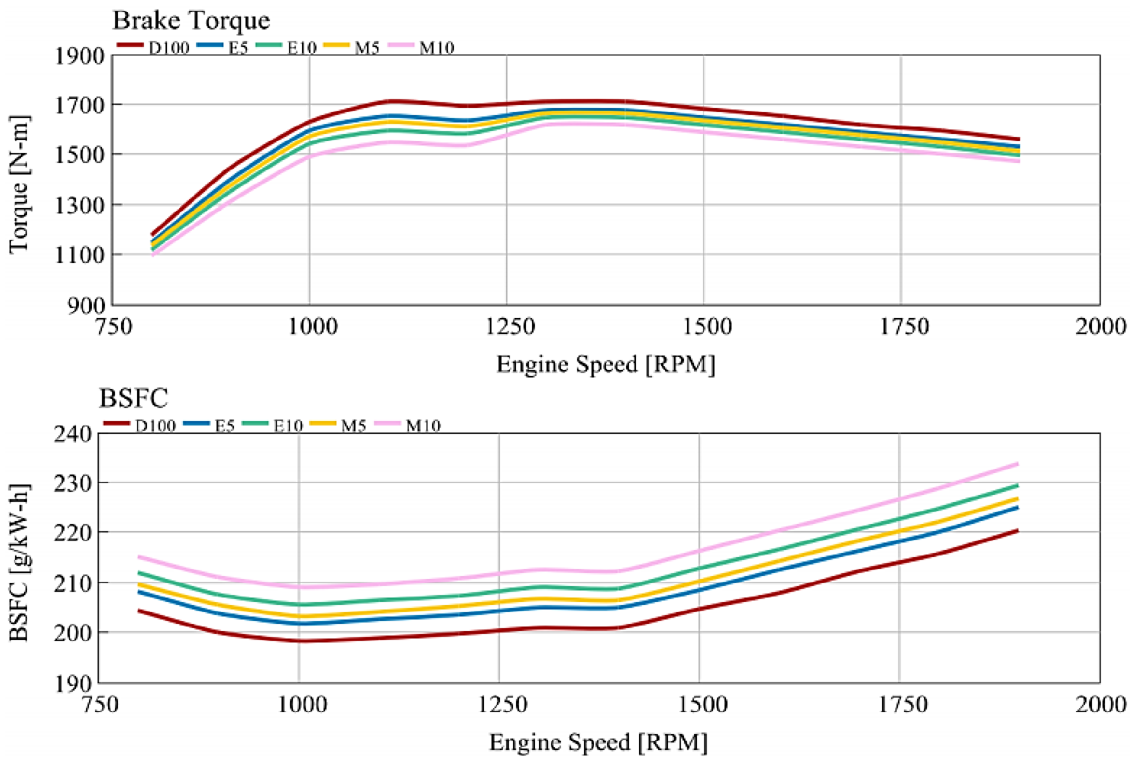


Fig. 5. Torque and BSFC values for D100, E5, E10, M5 and M10

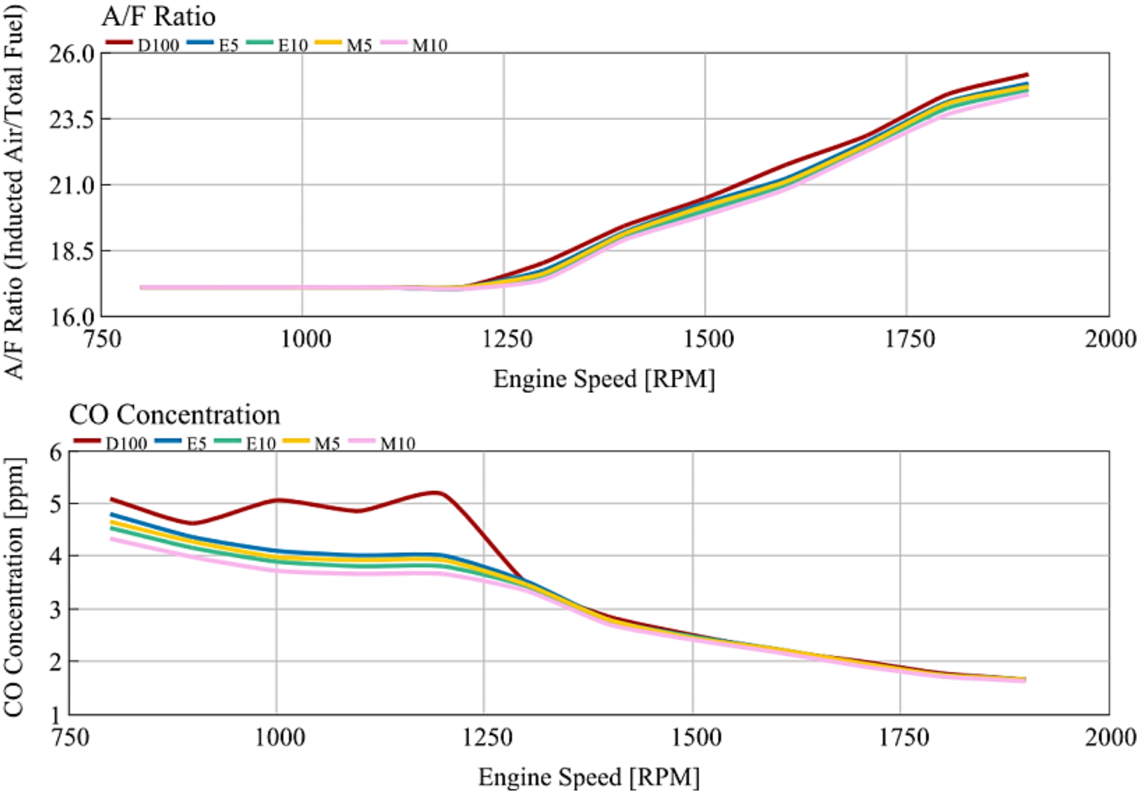


Fig. 6. Concentration of CO and Air/fuel ratio of various fuels.

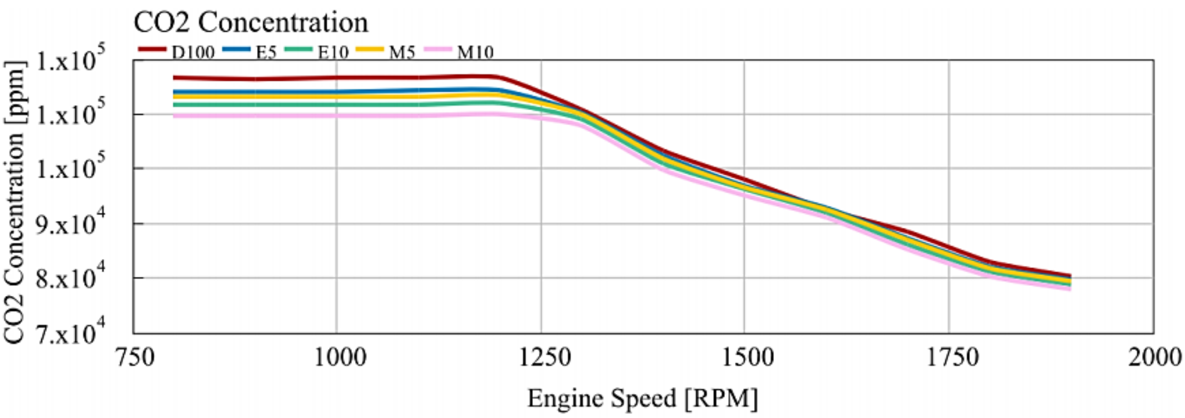


Fig. 7. CO2 emission for different fuels

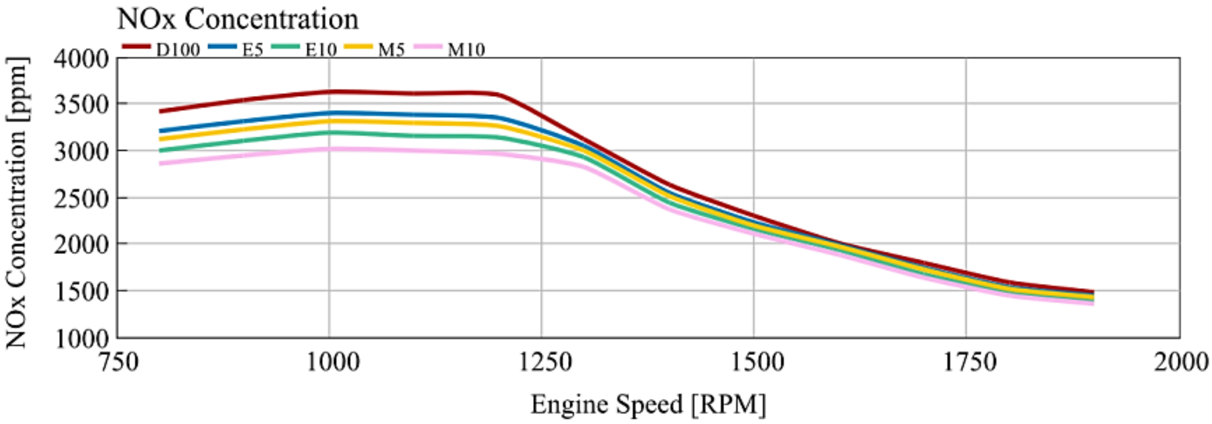


Fig. 8. NOx emission for different fuels

the results that utilizing an alcoholic mixture decreases the level of engine emissions by 15%. The fluctuations in the D100 fuel CO diagram also correspond to maintaining a minimum air-to-fuel ratio of 17.

Variation of CO emission for combined fuels is represented in Fig 6. The emission of CO falls by increasing the engine speed. Also, the injection of additives into diesel fuel causes more oxygen concentration and more complete combustion. According to Fig. 7, the concentration of CO_2 is reduced by enhancement of the engine speed. According to Fig 8, alcohols' high flammability leads to fast combustion at a lower temperature, which decreases NO_x concentration. Also, by increasing the engine speed, due to reduced production time, the NO_x decreases up to 15%.

3- 2- Thermodynamics analysis of waste heat recovery process

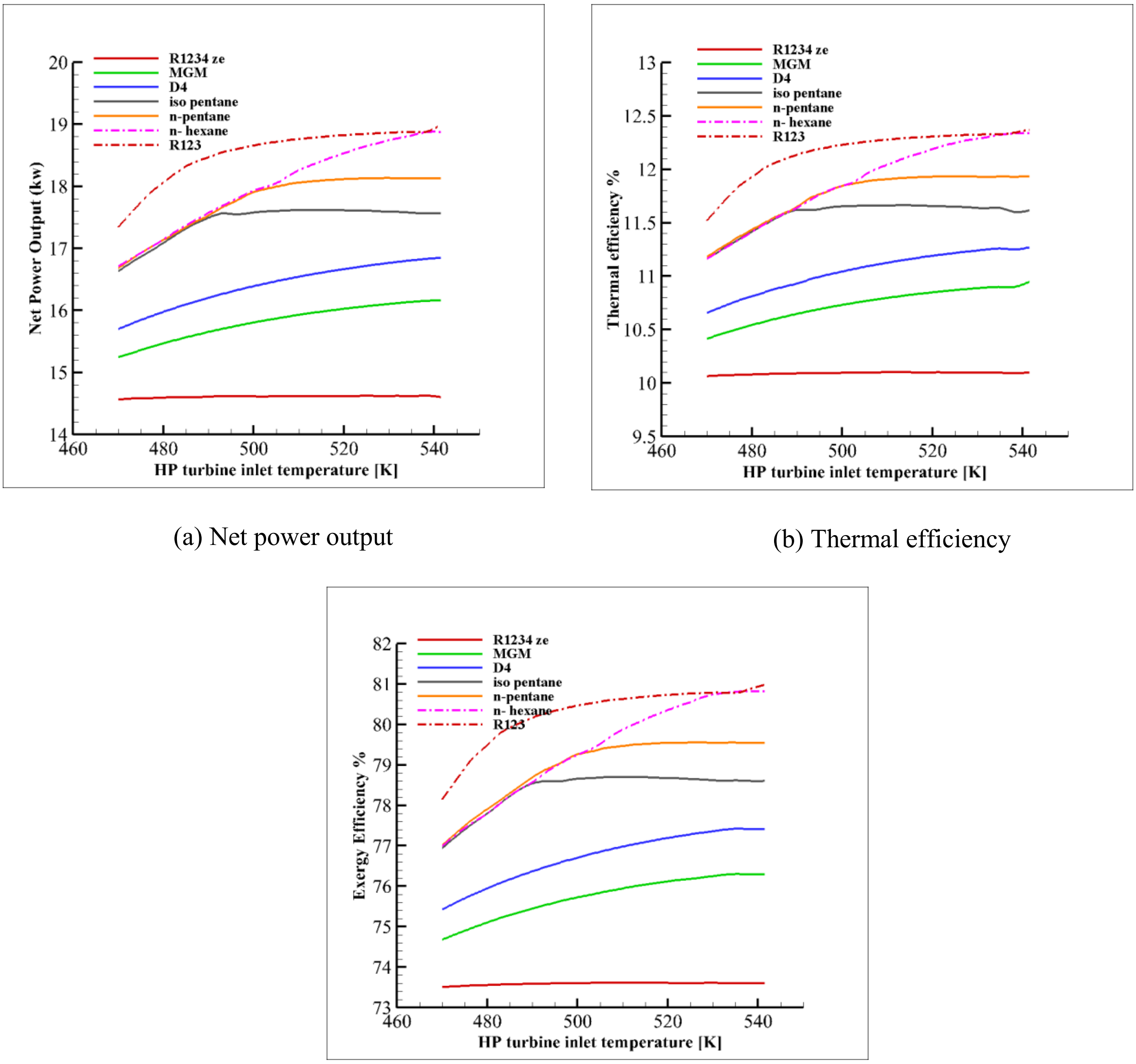
By conducting a parametric study, the effects on the thermodynamic performance of DPORC and such important parameters of the waste heat recovery cycle as low-pressure evaporation temperature, high-pressure turbine inlet pressure, and inlet high-pressure turbine temperature are demonstrated. It should be mentioned that the parametric study is carried out considering, R1234ze, MGM, D_4, iso-pentane, n-hexane, and R123.

The highest temperature of the proposed DPORC occurs at the inlet of the high-pressure turbine. At the turbine inlet, the working fluid that is heated by the combustion gases enters the HP turbine and then the mechanical power at the expander is generated. Afterwards, its temperature and pressure decrease to the lower pressure of the LP evaporator outlet. Figure 9 (a-c) shows the alteration of the net output power, energy efficiency, and exergy efficiency of the DPORC versus the HP turbine inlet temperature for the seven considered working fluids. Figure 9 (a) indicates that the net output power generally increases with the evaporator temperature. The iso-pentane and n-pentane working fluids are the same as the trend of R123. The proposed DPORC using R123 achieves the output net power of 18.84 kW which is more than the power output of 17.9 kW and 17.38 using n-pentane and iso-pentane, respectively.

It can be shown that the performance of the DPORC (output net power, energy, and exergy efficiency) for iso-pentane, n-pentane, and R123 improves. This is because when the HP turbine inlet temperature is high, the temperature difference between coolant and exhausted engine gas is low and the HP turbine ratio pressure is maximized, then causing the highest power achieved. Also, the calculated net power output of D4, MDM, and n-hexane indicates that the net turbine power increases by increasing the temperature at the turbine inlet, continuously. But the power of the cycle with R1234ze is almost constant.

Fig 9(b) and 9(c) present the effects of evaporator temperature on energy and exergy efficiencies of DPORC, respectively. As shown in Figs 9(b) and 9(c), the behavior of thermal and exergy efficiency curves is the same as the net power output curves behavior shown in Fig 9(a). Based on Eq. (11), when the HP turbine inlet temperature is very small, the flow rate of HP branch is high, and then engine exhausted outlet gas temperature $T_{g,o}$ is high, which indicates, the highest heat from engine gas is returned to working fluid causing a highest power.

The effects of HP inlet turbine pressure on the performance (output net power, energy, and exergy efficiency) of the proposed cycle are indicated in Fig 10. Fig. 10(a) demonstrates the alteration of net power output with high-pressure turbine inlet pressure at various working fluids. It is observed that net power output generally decreases by increasing of HP



(a) Net power output

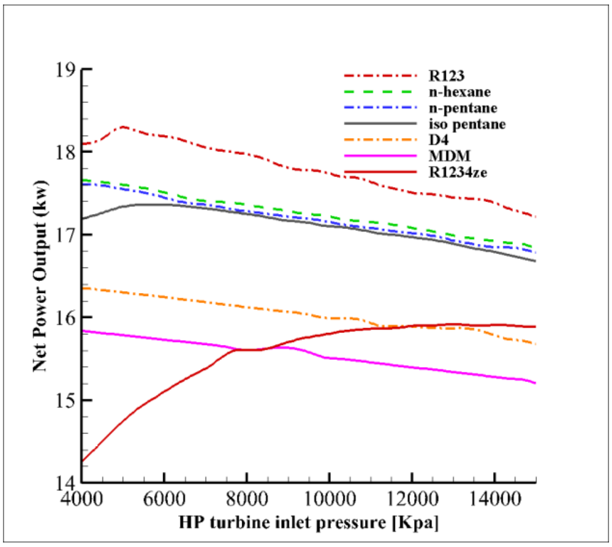
(b) Thermal efficiency

(c) Exergy efficiency

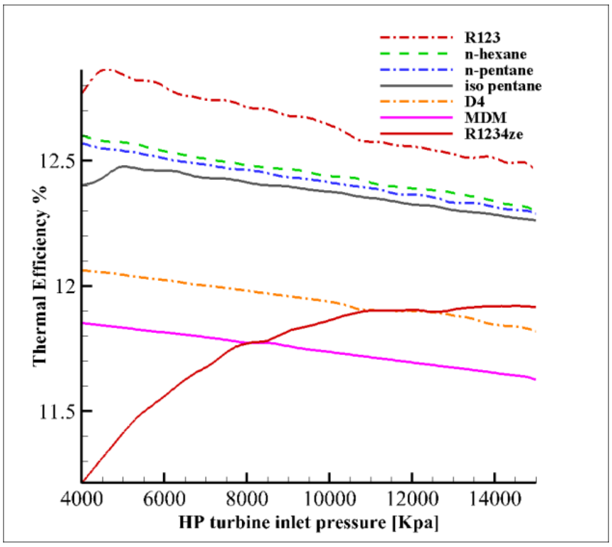
Fig. 9. Effect of HP turbine inlet temperature on performance of proposed system

turbine inlet pressure except for R1234ze. As the HP turbine inlet pressure rises, the high-pressure branch mass flow rate increases. While the LP branch mass flow rate does not change, since the enthalpy difference at pumps remains constant, the power consumption of the pump increases with the increasing mass flow rate. As mentioned in the previous section, the variation curve behavior of thermal and exergy efficiencies is as same as the net power output behavior as indicated in Fig. 10(b, c). For R1234ze working fluid, the net power output decreases with an increase in the HP turbine inlet pressure.

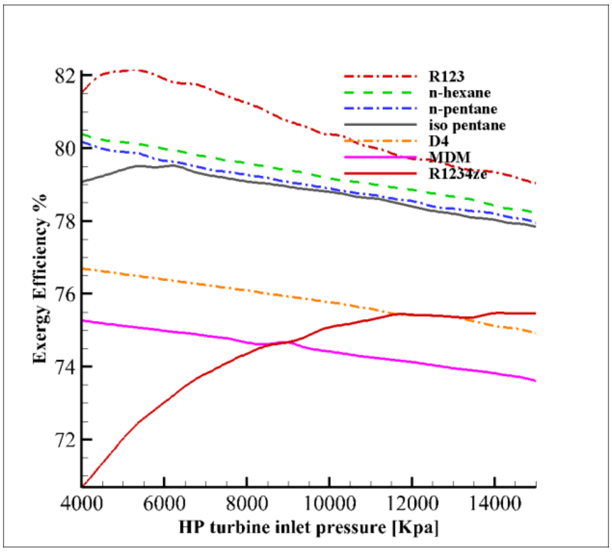
The changes in net power output and thermal and exergy efficiencies as a function of the LP evaporator temperature are given in Fig. 11(a-c). As shown, R123 has the best performance (i.e., more net power output, energy, and exergy efficiencies) than other working fluids in all cases. Based on Fig. 11(b, c), by increasing of LP evaporator temperature, the net power output, energy, and exergy efficiencies increase. By increasing the LP evaporator temperature, the LP flow rate decreases. Afterward, cause the decreases the consumed pump work and increases the net power output, energy, and exergy efficiency.



(a) Net power output



(b) Thermal efficiency



(c) Exergy efficiency

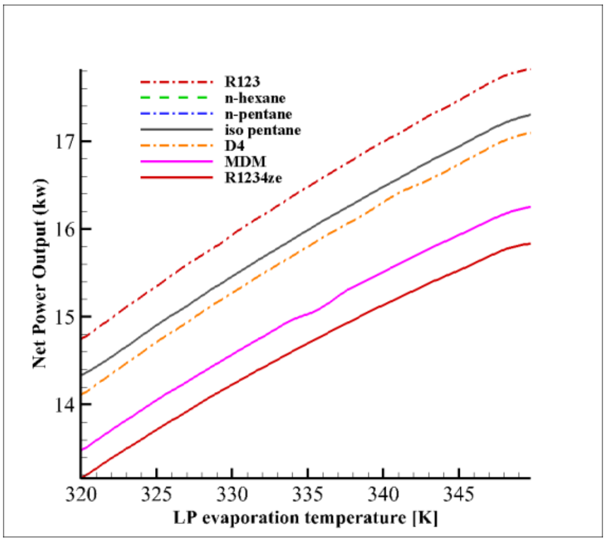
Fig. 10. Effect of HP turbine inlet pressure on the performance of a proposed system

3- 3- Exergy analysis

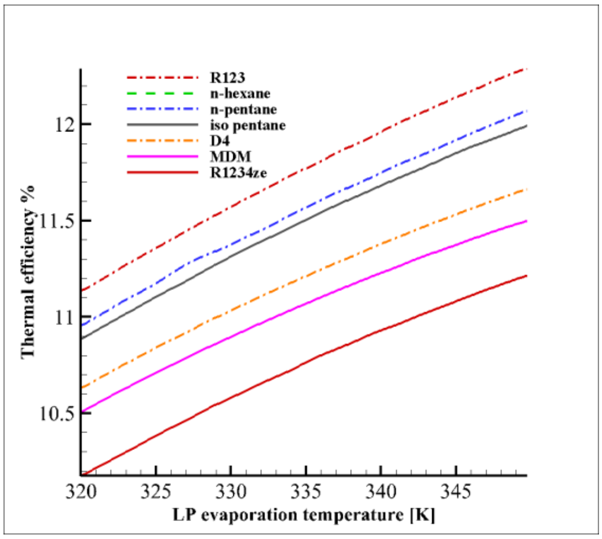
The exergy efficiency of the engine with utilized blends is shown in Fig. 13. Basically, the exergy efficiency decreases by injecting alcoholic additives into the diesel fuel. However, the change of exergy efficiency is not sensible due to the lower LHV and brake power related to additives.

On the other hand, Fig. 12 demonstrates the results of exergy analysis for D100. The exergy of destruction, brake power, exhaust gas, and cooling water are 43% and 35%, 19%, and 3%, respectively. The very irreversible combustion process leads to a high exergy destruction in the engine. The brake power, and consequently the exergy efficiency, is reduced by the addition of the compounds, while due to their lower LHV, the exergy of input fuel is decreased. Hence, the results for blends follow the same scenario of D100, as seen in Fig 13.

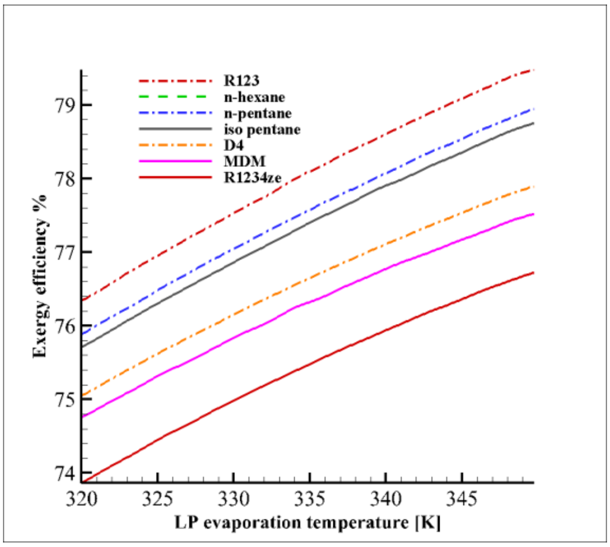
Figure 14 indicates the exergy destruction based on the engine's angular velocity variation. Accordingly, exergy destruction falls at higher speeds because of enhancing the exergy of the brake power. The aAddition of compounds to diesel fuel causes more destruction by decreasing the input exergy and braking power.



(a) Net power output



(b) Thermal efficiency



(c) Exergy efficiency

Fig. 11. Performance of the proposed system under the influence of LP evaporation temperature

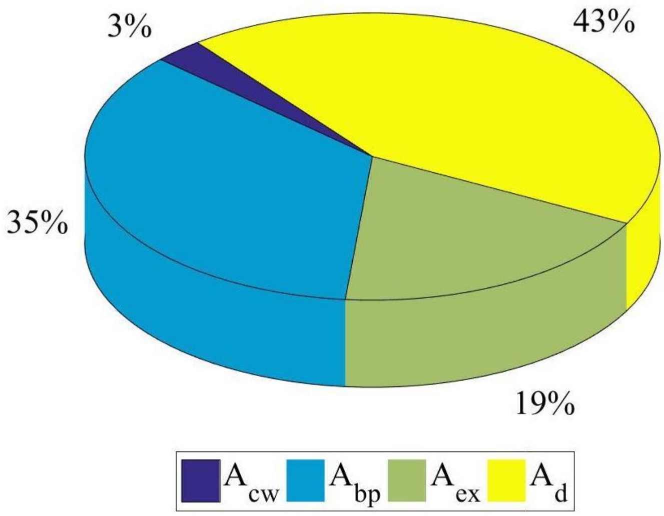


Fig. 12. Results of exergy analysis for D100.

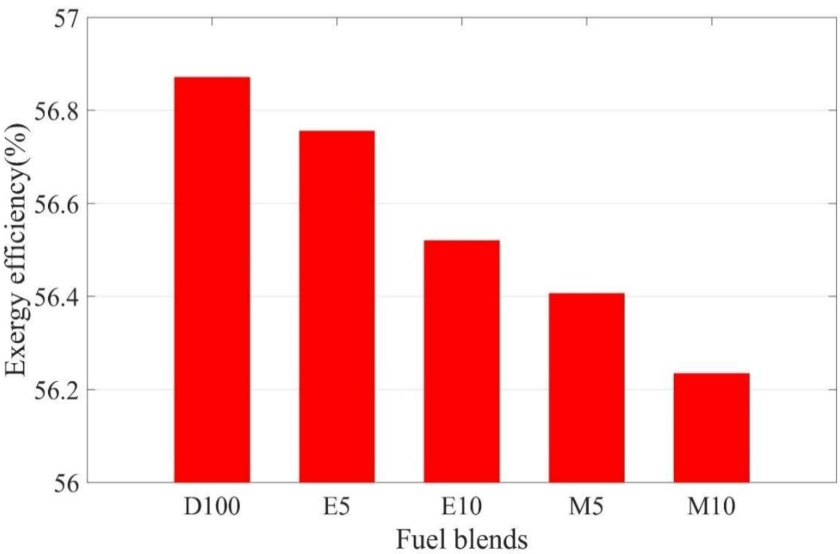


Fig. 13. Exergy efficiency of the engine with various fuel compounds.

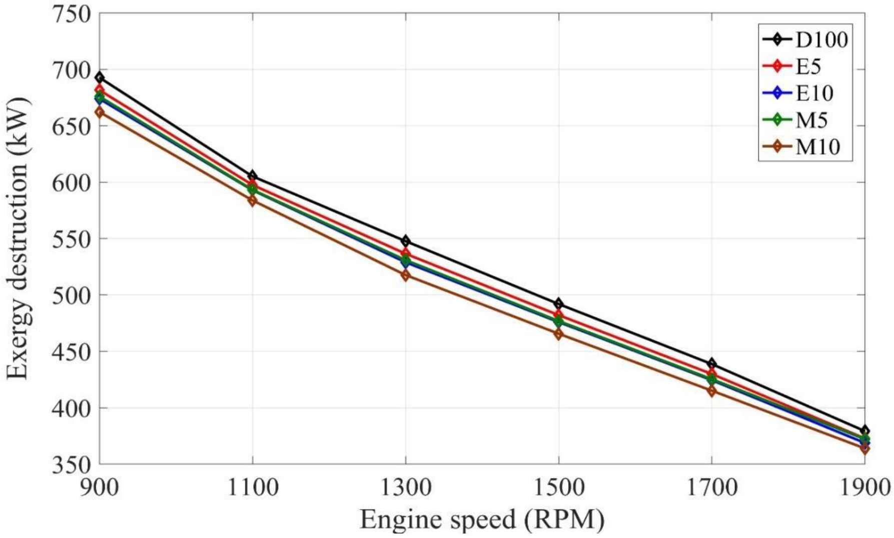


Fig. 14. Exergy destruction versus engine speed

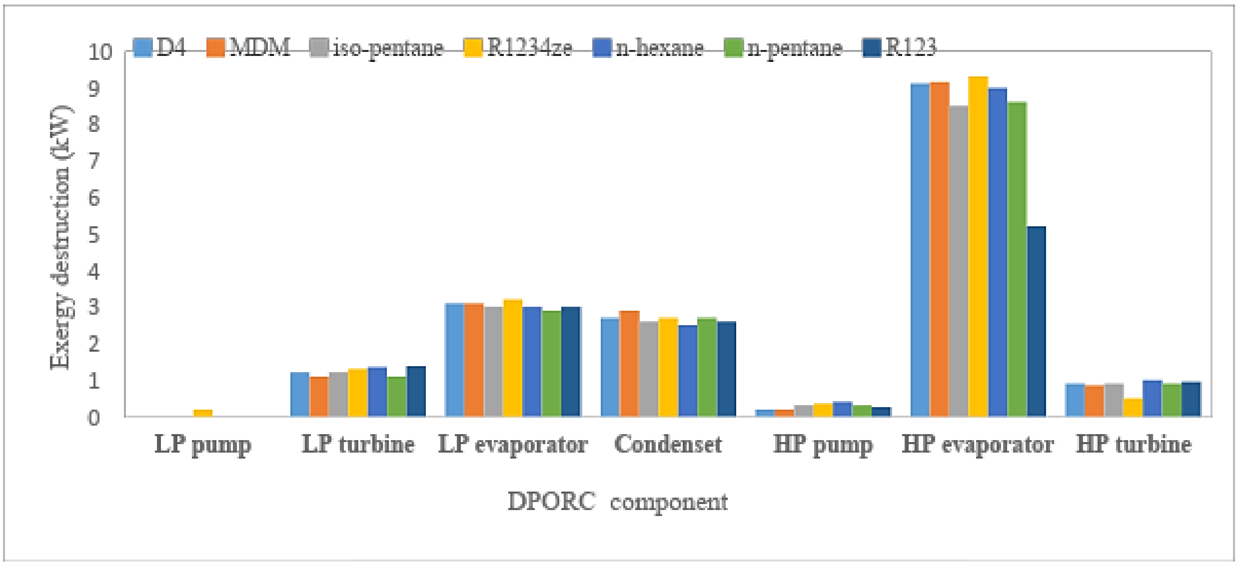


Fig. 15. Comparison of exergy destruction for seven working fluids for DPORC system by adding of D100 to Diesel.

To investigate the performance of DPORC based on under-studied working fluids, the exergy destruction values are determined for all working fluids. Fig. 15 presents the values where it can be observed that the DPORC based on R123 has lower exergy destruction. The proposed DPORC by R1234ze indicates the highest exergy destruction. The results show that the DPORC by R123 has an appropriate heat transfer value in two evaporators and condensers as well as lower exergy destruction in turbines and pumps. To summarize, the HP evaporator has the highest exergy destruction between the DPORC components.

4- Conclusions

In the present study, an innovative system of DPORC combined with a turbocharged Diesel engine fueled with ethanol and methanol blends is prepared to utilize of the engine exhaust waste heat and water cooler. For future studies, we suggest using IBC (Inverted Brayton Cycle) Instead of DPORC to recover waste heat from the engine and compare its performance with DPORC. Another suggestion we have is a change in the structure of DPORC, to clarify this, we can eliminate turbocharger and intercooler, therefore exhaust gas enters directly into the high-pressure evaporator, and engine coolant working fluid enters directly into the low-pressure evaporator.

The salient achievements of the investigation are as follows:

A 15% reduction of NO_x emission is obtained by using M10 at 1900 rpm.

A decrease in concentrations of CO occurs as ethanol and methanol are added and improve the combustion.

A reduction in the concentration of CO_2 is achieved by raising the engine speed.

The exhaust gas temperature has increased with the addition of alcoholic to diesel fuel which causes the highest potential of heat source for recovering.

Decrement of engine emissions reduces the engine torque. Thus, all aspects of employing additives need to be taken into consideration to reach a desirable result.

The net output power and exergy and energy efficiencies in the proposed system improve by approximately 5%, 7.5%, and 8.2%, respectively.

The net output power and exergy and energy efficiencies of the DPORC increase with LP evaporator temperature and HP turbine inlet.

The net output power and exergy and energy efficiencies decrease with increasing HP turbine inlet pressure.

The addition of alcoholic fuels has no sensible effect on the exergy efficiency.

The addition of alcoholic fuels diminishes the exergy destruction.

The addition of alcoholic fuels up to 10% can be suitable for emission reduction of the diesel engine.

According to the results, R123 is the appropriate working fluid used in the DPORC for recovery of the engine waste heat. HP evaporator had highest exergy destruction in the proposed system.

Nomenclature		*Subscripts*	
A	exergy rate (kW)	bp	shaft power
c	mass fraction of carbon	c	cylinder
CE	combustion efficiency	cp	maximum cylinder pressure
E	Wiebe exponent	cw	cooling water
h	mass fraction of hydrogen	eg	exhaust gas
$\dot{I}$	exergy rate	ex	exhaust
ID	ignition delay (deg)	f	fuel
LHV	lower heating value of fuel (kJ/kg)	in	input
m	mass (kg)	M	Main combustion
$\dot{m}$	mass flow rate (kg/s)	P	premixed combustion
M_t	engine brake torque (NM)	T	Tail combustion
N	engine speed (rpm)	*Greek letters*	
o	mass fraction of oxygen	η_{II}	exergy efficiency
P	pressure (bar, kPa)	θ	crank angle
P_e	engine brake power (W)	*Abbreviations*	
R	specific heat constant	DPORC	Dual-Pressure Organic Rankine Cycle
s	mass fraction of Sulphur	ORC	Organic Rankine cycle
T	temperature (℃ , k)		
T	engine torque (NM)		
WC	Wiebe constant		
Y_i	molar rate		

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