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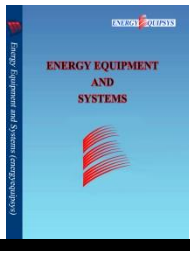

Combustion and exergy investigation of a compression ignition engine fueled with ethanol and methanol blends

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ABSTRACT

Today, the use of hybrid fuels in internal combustion engines has been considered to decrease the level of environmental pollutants. Among the various additives used to increase the efficiency and decrease pollution of Compression Ignition (CI) engines, alcohols especially methanol and ethanol have a special place. In this work, a one-dimensional, thermodynamic and two-zone model was used to evaluate the performance and emission of a compression ignition engine using ethanol and methanol in two volumes of 5% and 10% with diesel fuel by using engineering software such as GT-Power. Also, an exergy assessment is performed to study the exergy efficiency and exergy destruction of the engine for desired cases. The results show that the use of diesel fuel with alcoholic compounds reduces the heating value of the fuel compared to the D100 as well as the maximum brake torque and power and significantly reduces the engine exhaust emissions. This reduction is at most equal to 15%, which is of particular importance with respect to pollutants such as NO_X and CO. The exergy analysis shows that addition of alcoholic compounds such as ethanol and methanol by 5% and 10% to diesel fuel reduces braking power, it also reduces input exergy, which decreases exergy destruction and decreases the exergy efficiency by a maximum of 1.1%.

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1. Introduction

Due to the limitations of fossil fuels and environmental pollution, achieving high efficiency and clean combustion is inevitable. In this regard, CI engines have more fuel conversion efficiency because of their high compression ratio and lean combustion. However, under conventional diesel operating conditions, there are rich and poor areas in the combustion chamber with high temperatures that form soot and NOx emissions, respectively [1].

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Substituting diesel fuel with other fuels is a proper way to solve pollution problems. Combined fuel combustion using two different fuels illustrates this approach. Renewable fuels such as biofuels and Hydrogen are used today to reduce emissions and also fossil fuel consumption.

In recent years, use of alcoholic fuels or combine them with other fuels for internal combustion engines, due to the long-term environmental and economic benefits of fossil fuels has received more attention. Ethanol and methanol are two types of alcoholic fuels used in combination with common diesel fuel in CI engines to decrease emissions [2, 3]. Methanol and ethanol are both simple molecules. Methanol has similar combustion features with ethanol such as close octane numbers, low stoichiometric air-fuel ratios and very high vaporization heats [4]. Exergy analysis of CI engines with these alternative fuels is crucial [5]. This is because of allowing us to identify the irreversible sources and to obtain more accurate information on engine efficiency [6]. Sarjovaara et al. [7] performed experiments in the field of dual-fuel combustion, with 85% ethanol and 15% diesel fuel under different load conditions. They found that NO_X emission decreases with increasing ethanol percentage, but combustion efficiency is also reduced, and because the combustion stage was not optimized, the thermal efficiency trend was not constant. Candan et al. [8] examined the influences of methanol mixture on emissions and performance of a single-cylinder diesel engine. In their study, methanol was mixed in different ratios of 5, 10 and 15% with diesel. They found that using methanol increased NO_X emission but reduced HC, CO and CO_2 emissions. Moreover, power and torque prohibited employing methanol. Jeongwoo et al. [9] considered the ethanol substitution effect on emissions and performance of a dual fuel single-cylinder diesel engine under different load conditions. Ethanol percentage is varied from zero to 50% of the total energy applied to the cylinder, and both fuels are driven to the cylinder by direct injection injectors. They found that with increasing ethanol substitution, NO_X and PM emissions decline. Also, they could increase ethanol percentage up to 63% for mid-load, but because of the lack of sufficient energy in the lower loads and severe increase of cylinder pressure in the higher loads, a high percentage of ethanol cannot be used. The effects of combining diesel fuel with ethanol and methanol on emissions and combustion of CI engine was investigated by Sayin [10]. A direct injection single-cylinder diesel engine at 1000-1800 rpm with a torque of 30 N-m was used. He realized that using methanol and ethanol fuels blended with diesel increases BSFC and NO_X emission while reduces HC and CO emissions and brake thermal efficiency. Fayyazbakhsh et al. [11] studied the amendment of formulation of diesel fuel to improve the performance of the engine and reduce engine emissions. The effects of combination oxygenated fuels ethanol, methanol and n-butanol were studied. The results showed that in some condition, increase the content of CO, CO_2 and NO_X and the engine speed had a negative effect on exhaust emissions. Appending the oxygen fuels decreases brake thermal efficiency (BTE) and rises brake-specific fuel consumption (BSFC). Also, Venue et al. [12] tried to improve the performance of the triple mixtures (diesel-biodiesel-alcohol) employing DEE (diethyl ether) as an ignition amplifier, experimentally. Results show that adding DEE to ethanol-biodiesel-diesel (EBD) will enhance the in-cylinder pressure, BSFC and combustion duration. However, because of the reduction of ignition delay and evaporation of latent heat, the emission rate of NO_X and PM decrease. While adding DEE to methanol-biodiesel-diesel (MBD) will increment the CO, PM and CO_2, and decline in-cylinder pressure, BSFC, combustion duration and HRR. Ozgur et al [13] studied emissions and performance of diesel fuel and diesel fuel combined with ethanol at 20% by volume (E20) in a CI engine between 1000 - 2600 rpm engine speed. Results indicated that using diesel-ethanol blends decrease power, torque and CO emission while BSFC and NOx emission decreased. But, exergy and energy efficiency values were decreased by appending alcohol fuels to diesel fuel. Jamrozik [14] examined the effects of combining ethanol and methanol with diesel fuel on a diesel engine, experimentally. The percentage of alcoholic fuels in his study varies from 0 to 40%. He found that by increasing the methanol in the methanol-diesel mixture up to 30%, the thermal efficiency improves, while IMEP does not change significantly. In addition, this increment caused a significant decrease in CO emissions. More increment in methanol percentage by more than 30% disarranges the combustion process, significantly reducing cylinder pressure and destabilize the engine operation. However, increasing the ethanol in ethanol-diesel blend up to 40% improved the efficiency of the engine while IMEP remains constant. Overall combining both alcoholic fuels with diesel fuel increases the NO_2 emission of the engine.

Taghavifar et al. [15] has numerically investigated the HCCI engine fueled with a combination of methanol and diesel fuel. The main aim of their work is to investigate engine probability and exergy analysis. Their result shows that replacing diesel with 30% methanol and 20% DME at 1400 rpm causes more in-cylinder pressure and AHR. While 80% diesel with 20% methanol at 2000 rpm with 20% EGR, offers lower combustion performance with lower engine efficiency. However, because of the high workability and the least irreversibility, the proposed D50 blend with 35% mechanical efficiency can achieve the highest exergy efficiency. Ma et al [16] investigated exergy aspects of a dual fuel engine fueled with methanol and diesel to reveal the mechanism of Diesel Methanol Dual Fuel (DMDF) engine from the second law viewpoint. The results demonstrate that due to the effect of various temperatures on the imperfect combustion of methanol, heat transfer exergy loss of DMDF was lower and exhaust exergy loss in high loads was low. Additionally, it was found that by reducing combustion irreversible exergy loss and exhaust loss, exergy efficiency improved in the higher temperature of methanol and cooling water. Rufino et al. [17] used exergy analysis to study a comparison between hydrous ethanol and gasohol. Energy and exergy efficiencies were assessed in terms of engine load and speed, and air-fuel ratio. Exergy distributions of hydrous ethanol and the exergy losses were also estimated at different engine operating conditions. It has been observed that hydrous ethanol has higher values of energy and exergy efficiencies as well as lower in-cylinder exergy losses. Higher air-fuel ratios caused higher exhaust exergy losses because of the late combustion. Moreover, the exergy of cooling water decreases with an increment in the air-fuel ratio. The effect of biodiesel fuels on the CI engine has been studied by Khoobbakht et al. [18] using exergy analysis. Moreover, they investigated the effects of engine speed and load, and also, the mixture of biodiesel-diesel and ethanol-diesel in diesel fuel on the exergy efficiency. According to the results, they found that with increasing volume percent of biodiesel and methanol, the exergy efficiency declines. They also realize that 43.09% of the fuel exergy is annihilated and the average thermal and exergy efficiencies were 36.61% and 33.81%, respectively. Baodong Ma et al. [19] used exergy analysis to study the dual-fuel Diesel engine in different operating conditions, fueled with Diesel and methanol. They concluded that by increasing intake temperature the exhaust chemical exergy loss declines and the exergy efficiency improves. They also found that the higher cooling water and methanol fuel temperature increase the exergy efficiency due to the reduction of the combustion exergy destruction as well as the exhaust exergy loss.

In the performed literature review, it is revealed that both methanol-diesel and ethanol-diesel blends influences on the diesel engine performance and emission are examined. The relevant studies confirm the decrease of emissions with increasing ethanol or methanol substitution, but different trends in NO_X emission, thermal efficiency and combustion behavior can be observed. The effects of alcohol-diesel on engine performance and emissions are investigated, but the comparison of the engine performance, exergy efficiency, exergy destruction and emissions of the engine with ethanol-diesel and methanol-diesel blended fuels are not studied. Moreover, works on the exergy analysis of methanol-diesel and ethanol-diesel fuels are still rare to see in the literature. Accordingly, in the present work, a one-dimensional thermodynamic two-zone model was developed to evaluate the performance and emission of a compression ignition engine using ethanol and methanol in two volume ratios of 5% and 10% with diesel fuel. The main objectives of the present work are as follows:

- To investigate the engine performance parameters and pollutants with diesel fuel and alcoholic compounds.
- To examine more parameters such as pressure and temperature after intercooler to validate the model.
- To examine the exergy performance parameters of the proposed engine. The exergy efficiency and destruction are considered in this regard.
- To investigate the influences of the engine speed on the exergy destruction for diesel fuel and alcoholic compounds.

Nomenclature

P	pressure (bar, kPa)

T	temperature (℃, K)
N	engine speed (rpm)
M_t	engine brake torque (NM)
P_e	engine brake power (kW)
WC	Wiebe constant
E	Wiebe exponent
ID	ignition delay (deg)
CE	combustion efficiency (%)
A	exergy rate (kW)
LHV	lower heating value of fuel ($^{kJ}/_{kg}$)
h	mass fraction of hydrogen
c	mass fraction of carbon
o	mass fraction of oxygen
s	mass fraction of Sulphur
m	mass (kg)
$\dot{m}$	mass flow rate ($^{kg}/_{s}$)
R	specific heat constant
T	engine torque (NM)
Y_i	molar rate (%)
A_{cw}	exergy rate of cooling water (kW)
A_{bp}	exergy rate of brake power (kW)
A_{ex}	exergy rate of exhaust gases (kW)
A_d	destroyed exergy rate (kW)

Subscripts

f	fuel
M	main combustion
T	tail combustion
P	premixed combustion
c	cylinder
cp	maximum cylinder pressure
in	input
ex	exhaust
cw	cooling water
bp	shaft power
eg	exhaust gas

Greek letters

η_{II}	exergy efficiency (%)
θ	crank angle (deg)

2. Modeling and validation

2.1. Engine modeling

In this study, GT-Power software is used to perform a one-dimensional simulation of engine performance. This application presents a one-dimensional thermodynamic two-zone model to solve the performance and emission of the engine. One-dimensional simulations are less accurate than three-dimensional types, but due to solving equations in one dimension, they have less calculation time than three-dimensional simulations. This led to the use of such simulations to identify and study the general behavior, the feasibility of proposed designs and analysis of the sensitivity of various parameters to each other. A turbocharged 6-cylinder diesel engine owned by Weichai Co. has been used. The specification of the engine is presented in Table 1.

Table 1. Engine specification

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

The specifications of diesel, methanol and ethanol are presented in Table 2.

The following assumptions are made during the simulation:

- The exhaust gases and combustion air are assumed to be ideal gas mixtures.
- Kinetic and potential energy effects are ignored.

Moreover, the following assumptions are made in the used heat release method:

- Inside the cylinder, the air-fuel mixture charge is an ideal gas.
- The charge in the cylinder is assumed a uniform single zone with constant composition from the intake valve closing to the exhaust valve opening.
- The released energy because of fuel combustion is assumed as heat flow to the cylinder.

The engine model shown in Figure 1 consists of various components for modeling the flow of inlet and outlet manifold pipes, valves, turbines and compressors. In fact, the intake flow enters the inlet manifold and after passing through the inlet valve, enters the cylinder. The equations of combustion and heat transfer are defined in the cylinder part. From the released energy by combustion and the available energy after heat transfer to the cylinder walls, the brake torque of the cylinder is calculated. To model the cylinder, in this work combustion model and the heat transfer specifications must be accurately defined. To specify the model of heat transfer in the cylinder, the Woschny model was used. The Wiebe model was used for the combustion sub-model, to calculate the energy release rate per crank angle. This is the appropriate model for characterizing the combustion of a compression ignition engine.

Wiebe function for compression ignition engine includes three-step of combustion, premixed combustion, mixing-controlled combustion and late combustion. The Wiebe constants per step of combustion to calculate the energy release rate are defined as

$$WC_P = \left[\frac{Premix\ Duration}{2.302^{1/(E_P+1)} - 0.105^{1/(E_P+1)}}\right]^{-(E_P+1)}, \quad (1)$$

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

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

and the energy release rate per crank angle Eq. (4) is defined as

$$\begin{aligned} x(\theta) = {} & (CE)(F_p)\left[1 - \exp\left(-(WC_p)(\theta - SOI - ID)^{(E_P+1)}\right)\right] \\ & +(CE)(F_M)\left[1 - \exp\left(-(WC_M)(\theta - SOI - ID)^{(E_M+1)}\right)\right] \\ & +(CE)(F_T)\left[1 - \exp\left(-(WC_T)(\theta - SOI - ID)^{(E_T+1)}\right)\right] \end{aligned} \quad (4)$$

Table 2. Fuel properties of diesel, methanol and ethanol. [20],[21],[22]

	Diesel	Methanol	ethanol
Chemical formula	C_7H_{16}	CH_3OH	C_2H_5OH
Molecular weight ($kg/kmol$)	100	32	46
Autoigintion temperature (℃)	257	464	423
Cetane number	52.6	4	6
Lower heating value (MJ/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

The engine friction is the next crucial parameter in the simulation, for the friction sub-model Chen-Flynn model was used. The friction model for this purpose is given as

$$FMEP = C_{FMEP} + C_1 P_{CP} + C_2 C_P + C_3 C_P^2 \ , \qquad (5)$$

where FMEP is friction mean effective pressure in bar, C_{FMEP} a constant value of FMEP, C_1 maximum cylinder pressure coefficient, P_{CP} maximum cylinder pressure in bar, C_P is the mean piston speed in $m/_S$ and C_2 is the mean square speed of the piston.
By knowing the engine brake torque M_t, the brake power can be calculated as

$$P_e = M_t \left({}^{2\pi N}/_{60} \right) \quad . \qquad (6)$$

Therefore, by calculating the brake power, brake specific fuel consumption (BSFC) can be calculated as

$$B.S.F.C = \left({}^{\dot{m}_f . 3600}/_{P_e} \right) \quad . \qquad (7)$$

The input parameters of the Wiebe model are calculated automatically by using experimental data from the burn rate. These data are provided for the software. Also, the zeldovich model is used to calculate NO_x emissions, and for CO emission calculation the EngCylCO sub-model is used in GT-POWER software.

2.2. Exergy analysis

Exergy is defined as the maximum work that can be extracted from a system as it reaches equilibrium with the reference environment [23]. In the current study, exergy analysis is accomplished to get more accurate information about irreversibilities and engine efficiency.
The net (mechanical) work output of a control volume is determined using

$$\dot{W} = \omega T \ , \qquad (8)$$

where ω is the angular velocity, and T is the torque. The destroyed exergy of the engine can be obtained as (the waste heat of lubricating oil exergy and the heat loss of engine are negligible)

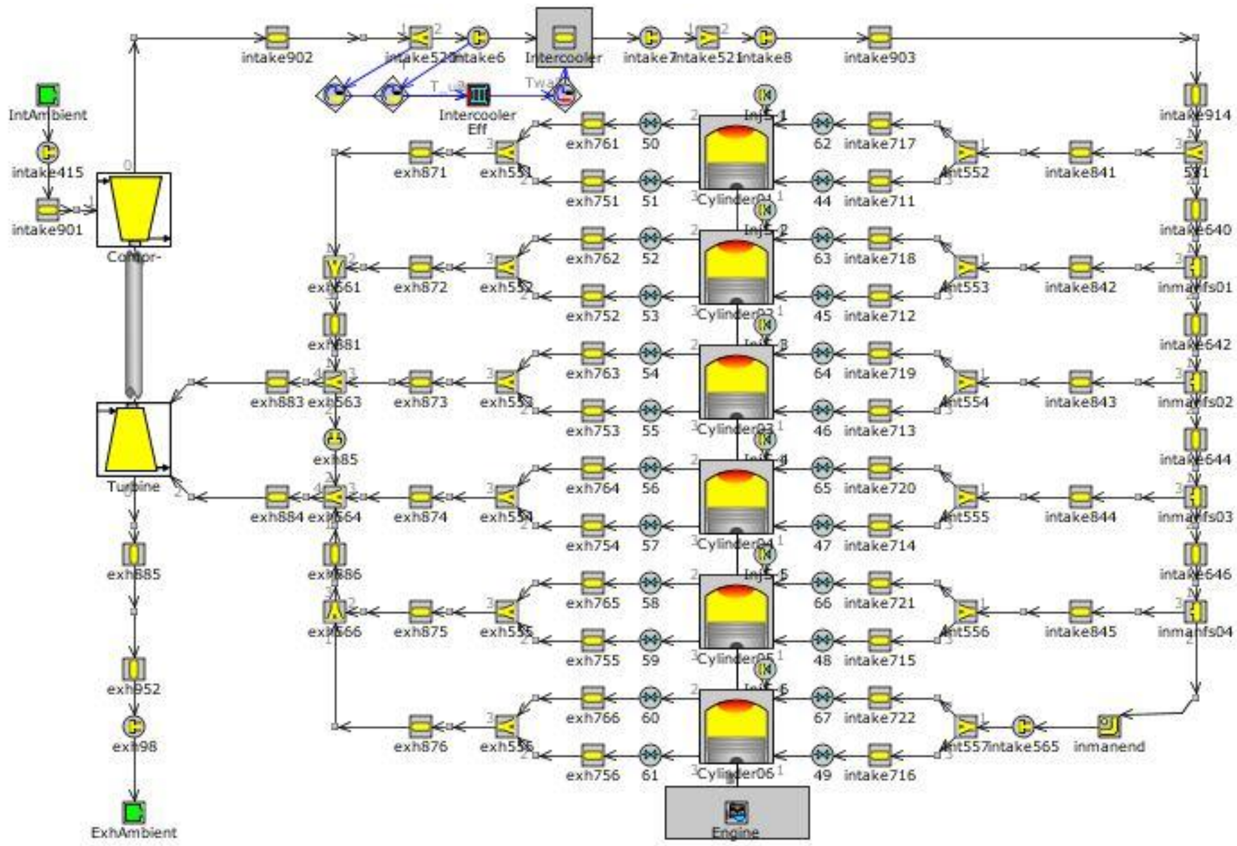

Fig.1. The 1-D model of Diesel engine

$$A_d = A_{in} - \left(A_{exh} + A_{cw} + A_{bp}\right) \quad , \tag{9}$$

where A_{in} is input exergy, A_{exh} is exhaust exergy, A_{cw} is cooling water exergy, and A_{bp} is the shaft power exergy. The input exergy is the exergy that goes into the system by the fuel and includes thermomechanical and thermochemical terms. For the exergy of the fuel, the thermomechanical term is assumed to be zero because the fuel is almost close to the dead state (environment). So, only thermochemical exergy is considered as exergy of the fuel that can be obtained as [24]

$$A_{in} = A_{in}^{tm} + A_{in}^{ch} \quad , \tag{10}$$

$$A_{in}^{tm} = (h - h_0) - T_0(s - s_0) \quad , \tag{11}$$

$$A_{in}^{ch} = \dot{m}_f \times [1.0401 + 0.1728\frac{h}{c} + 0.0432\frac{o}{c} + 0.2169\frac{s}{c}\left(1 - 2.0628\frac{h}{c}\right)] \times LHV \tag{12}$$

where h, c, o and s are mass fraction of hydrogen, carbon, oxygen and Sulphur in fuel; and LHV is the lower heating value of the fuel. As with fuel exergy, the exhaust gas exergy has both thermomechanical and thermochemical terms and could be obtained as [25]

$$A_{exh} = A_{exh}^{tm} + A_{exh}^{ch} \tag{13}$$

$$A_{exh} = \dot{Q}_{eg} - \left[\dot{m}_{eg} \times T_0 \times \left\{C_{pe} \times \ln(T_{exh}/T_0) - R_{eg} \times \ln(P_{exh}/P_0)\right\}\right] + A_{exh}^{ch} \tag{14}$$

$$A_{exh}^{ch} = \bar{R}T_0 \sum_{i=1}^{n} a_i \ln(\frac{Y_i}{Y_i^e}). \tag{15}$$

In the above equation $\dot{Q}_{eg}$ and $\dot{m}_{eg}$ are the heat and mass of exhaust gas rate, respectively. T_0 and P_0 are the temperature and pressure of the reference environment which, in this study, the ambient values are given to them. T_{exh}, P_{exh}, C_{pe} and R_{eg} are temperature, pressure, specific heat and specific heat constant of exhaust gas, respectively. Y_i and Y_i^e are molar ratios of the exhaust gas and the reference environment. The values of Y_i^e are considered as Table 3 [26].

Table 3. The molar ratio of the reference environment

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

The exergy of the engine cooling water is calculated as

$$A_{cw} = \dot{Q}_{cw}\left(1 - {T_0}/{T_{cw}}\right), \tag{16}$$

$$\dot{Q}_{cw} = \dot{m}_{cw}C_{cw}(T_2 - T_1)\ , \tag{17}$$

where $\dot{Q}_{cw}$ is the heat of cooling water rate, C_{cw} is the specific heat of cooling water and $\dot{m}_{cw}$ is the mass flow rate of water that circulated in the cooling system. The exergy of the shaft power is the brake power of the engine that is determined as

$$A_{BP} = (2\pi \times N \times T)/60{,}000\ , \tag{18}$$

where N is the engine speed and T is the engine torque. Therefore, by calculating the destroyed exergy, the exergy efficiency can be calculated as [25]

$$\eta_{II} = (1 - {A_d}/{A_{in}}) \times 100\ . \tag{19}$$

2.3. Model validation

In this study, the experimental results of the brake torque, brake specific fuel consumption, temperature and pressure after intercooler were used to validate the model. In the one-dimensional turbocharged engine models, the performance of the turbine and compressor has the most significant difference from the experimental results and can be the most important cause of error in these models. This is also due to the three-dimensional nature of the phenomena occurring in this set and the inability of the one-dimensional model to accurately simulate them. Heat transfer in turbocharger assemblies is one of the causes of the difference in enthalpy and computation of turbine and compressor power and consequently output pressure. So, to fix it, we considered coefficients to change the system efficiency.

The turbocharger efficiency coefficient is defined to match the experimental data of outlet pressure of intercooler for the model engine in GT-Power software. Also, to match the experimental data of outlet temperature of intercooler, heat transfer rate to the intercooler wall should be calibrated, so the heat transfer rate to the intercooler wall coefficient is also defined in GT-Power [27]. Comparisons of the simulated results and experimental values [28] of the outlet pressure and temperature of the intercooler after calibration of the turbine efficiency coefficient and heat transfer rate to the intercooler wall are shown in Fig. 2.

The one-dimensional model simulation accuracy for the pressure and the output temperature of the intercooler was about 6%, with the highest difference being the output temperature of the intercooler at 1500 rpm. This is due to the physical complexity of the intercooler as a heat exchanger in the internal combustion engine, which is replaced in the one-dimensional model by a tube with a high heat transfer rate. The heat transfer in the cylinder is an important cause of error in one-dimensional models. This is due to the three-dimensional nature and physical complexly of the heat transfer occurrence. Therefore the heat transfer coefficient is defined in GT-Power software to calibrate the Woschny model. In the next step, calibration of the in-cylinder heat transfer rate is performed to match the experimental results of the brake torque for the model engine and results are presented in Fig 3. The maximum error of these two graphs is related to the brake torque at 1900 rpm with a value of 12%**.**

The cause of the difference in the brake torque and subsequently in the brake specific fuel consumption (BSFC) between experimental and simulation can be attributed to the inability of the heat transfer sub-model to accurately predict the heat transfer inside the cylinder from the hot gases to the wall. It is clear from Figures 2, 3 that the engine model has been able to predict the performance of the Weichai WP12.336E40 engine well.

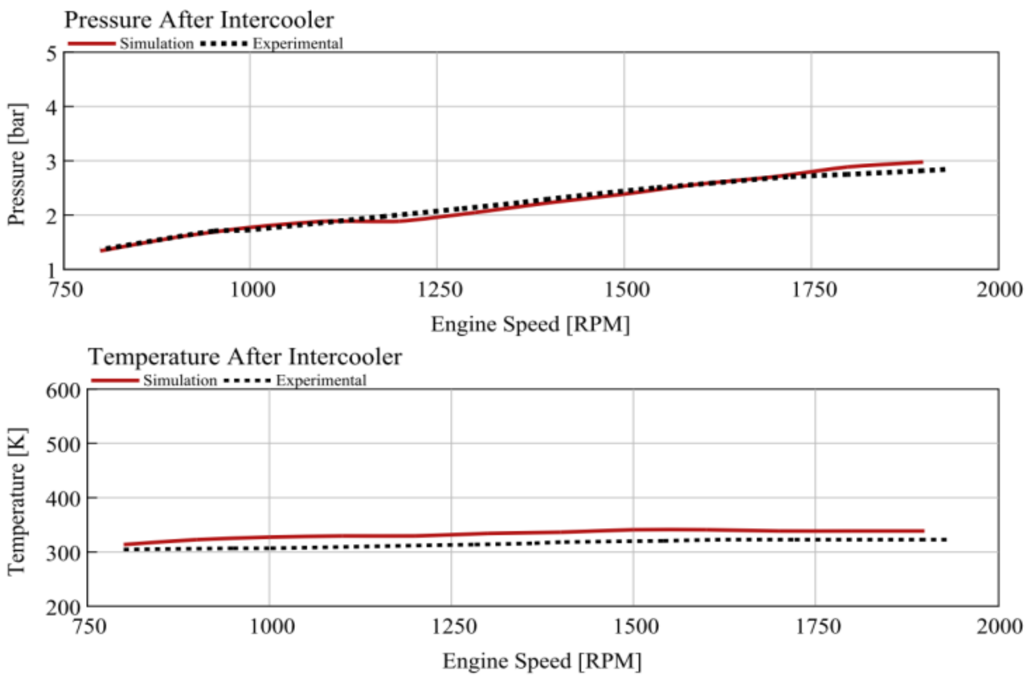

Fig.2: Temperature and pressure after intercooler for experimental and simulation results

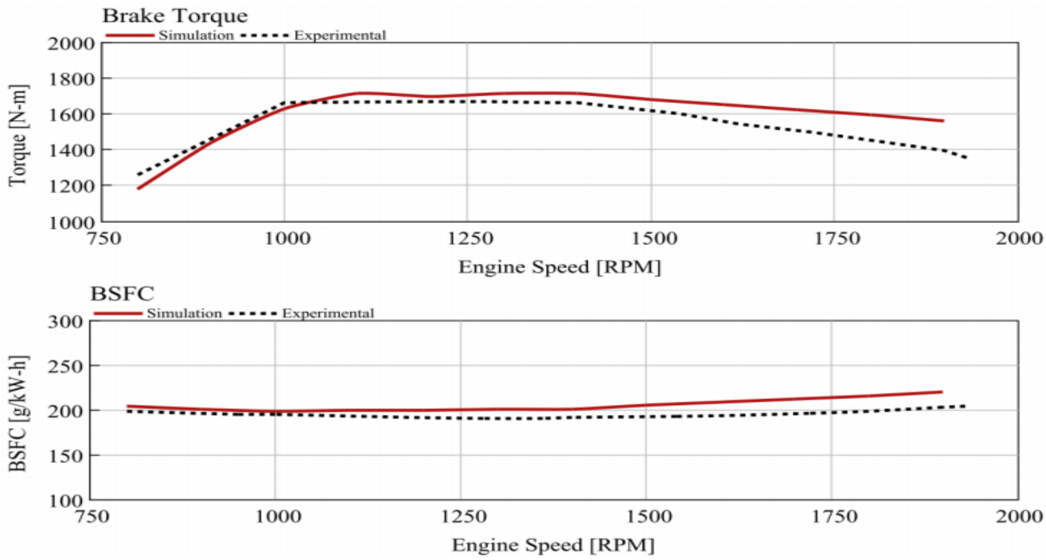

Fig.3. Brake torque and BSFC for experimental and simulation results

3. Results and discussion

After completing the calibration steps of the model and ensuring the model's acceptable accuracy in calculating engine performance parameters, the use of fuels with methanol and ethanol combinations is investigated. According to the studied literature, it seems that the use of fuels with 5 to 10% alcohol additives such as ethanol and methanol with Diesel fuel is a more favorable option for the internal combustion engine and the focus of the researchers has been on these fuels [29]. Consequently, the use of M5, E10, E5 and M10 fuels will be investigated in this study.

Figure 4 shows the results of calculating braking torque and brake specific fuel consumption in simulating the use of Diesel fuel with alcoholic compounds. Addition of alcoholic compounds to Diesel fuel decreases the heating value of the fuel compared to D100 reduces the maximum torque compared to pure diesel. This reduction in torque is increased with the addition of alcoholic compounds so that M10 fuel had about 10% less torque than the D100 case. Due to the similarity of the spray pattern in each of the simulated fuels, the difference in the specific brake fuel consumption diagram is only due to the difference in brake torque.

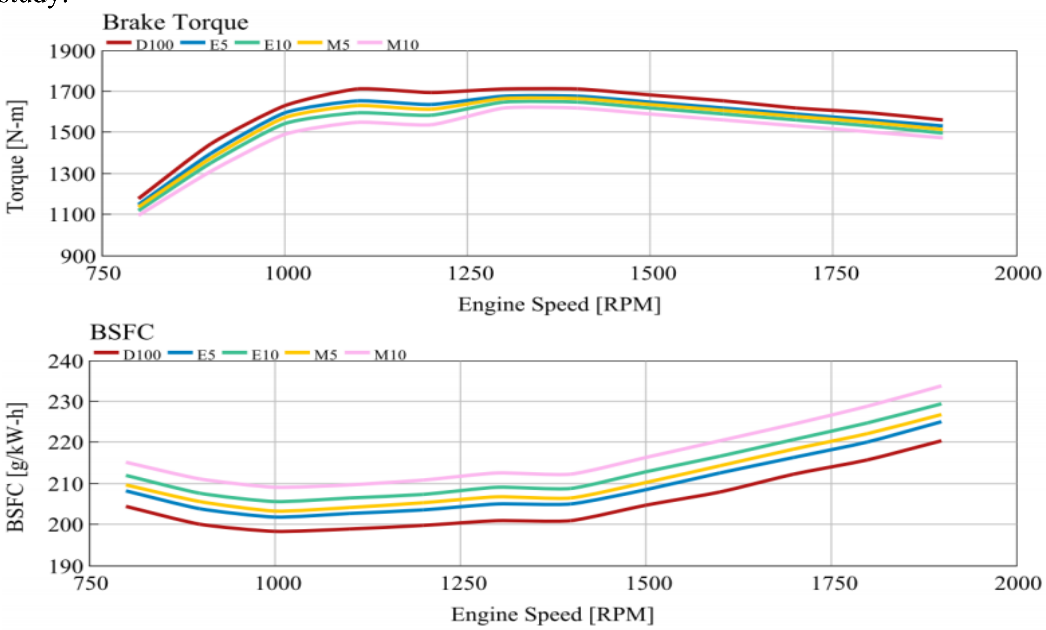

Fig.4. BSFC and torque value for D100, E5, E10, M5 and M10

The addition of alcoholic compounds to Diesel fuel reduces the heating value of the fuel compared to D100. Also, the alcoholic compounds have less reactivity (Cetane Number) compared to D100, so the ignition delay increases. These two reasons cause a reduction in torque. The presence of an oxygen bond in the alcohols increases the oxygen concentration in the fuel, which also reduces the fuel-to-air equivalence ratio and dilutes the inlet mixture to the cylinder, so the combustion process in the alcoholic blends fuel will be more complete than Diesel. For this reason, the brake-specific fuel consumption has increased with the increase in alcohol. As the heating value of Diesel is higher than ethanol and methanol, it also increases the fuel injection for a certain value of power output.

In addition to the performance parameters of the engine, it is also important to consider combustion pollutants. For this purpose, the results of concentrations of CO_2, CO and NO pollutants are presented in Fig. 5 to 7. Since variations in CO and CO_2 concentrations are highly dependent on the air to fuel ratio, the air to fuel ratio is also plotted. Based on these results, it can be concluded that the use of Diesel fuel with alcoholic compounds significantly reduces the emissions of engine emissions. This reduction is at most equal to 15%, which is particularly important for pollutants such as CO and NOx. The poor performance nature of diesel engines in terms of fuel-to-air ratio causes higher CO_2 emissions in these engines.

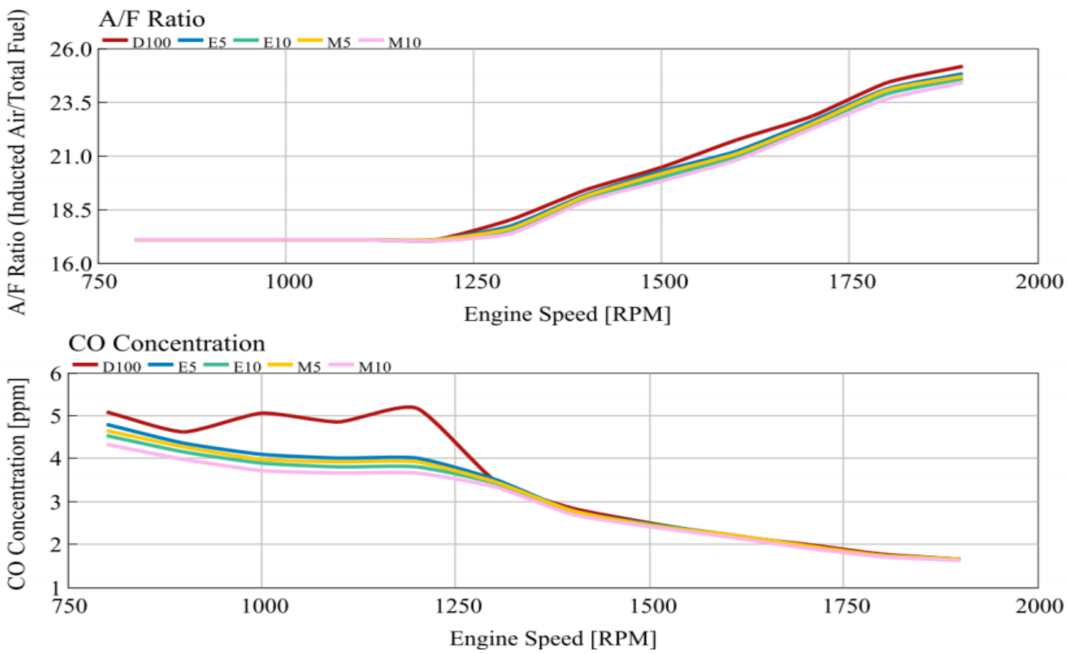

Fig. 5. Air to fuel ratio and concentration of CO emission for different fuels

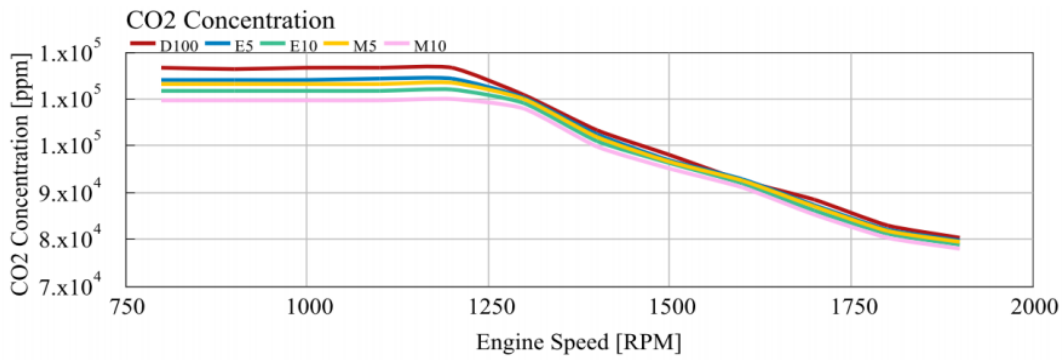

Fig.6. CO_2 emission for different fuels

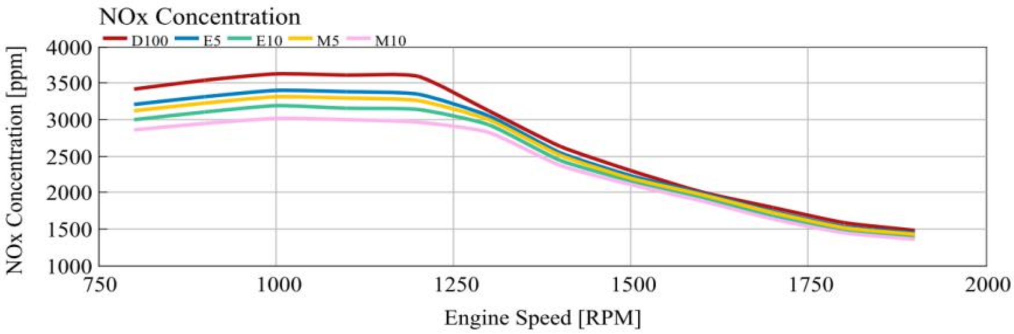

Fig.7. NO_x emission for different fuels

The trend of changes in carbon monoxide emission for ethanol-methanol mixture fuels is illustrated in Figure 5. As the engine speeds up, carbon monoxide emissions decrease. Carbon monoxide concentrations are also decreased with the addition of ethanol and methanol to diesel fuel. The variations in CO concentration are highly dependent on the air-fuel ratio due to more complete combustion resulting from increased oxygen concentration and leaner combustion of base fuel (D100). As shown in Fig. 6, the carbon dioxide concentration decreases with increasing the engine speed, which is indicative of the fact that variations in CO_2 concentration are highly dependent on the air-fuel ratio. The addition of alcoholic fuels makes combustion more complete and leaner than diesel base fuel. Moreover, Fig. 7 shows that the real-time for the formation of NOx emission and consequently the overall NOx pollutant decreases by increasing engine speed.

So far, it has been observed that with the addition of methanol and ethanol to diesel fuel in the CI engine its emissions are reduced.

The engine irreversibilities are identified and evaluated quantitatively using exergy analysis. Fig. 8 illustrates the exergy analysis of diesel base fuel (D100). The destroyed exergy of the system because of the irreversibility is 43% and the brake power exergy is 35%. The exhaust gas and cooling water exergies are 19% and 3%, respectively. This analysis is performed for other fuel blends and the results are presented in Table 4. It should be noted that by adding methanol and ethanol to diesel fuel the brake power is reduced, but on the other hand, the input fuel exergy decreases due to the lower LHV of the ethanol and methanol compared to diesel. So, the exergy analysis of the other fuel blends is similar to D100, as shown in Fig. 8.

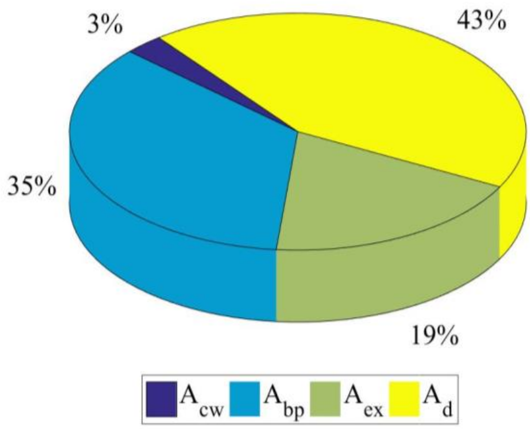

Fig.8. D100 exergy analysis
($\boldsymbol{A_{cw}}$ = exergy rate of cooling water,
$\boldsymbol{A_{bp}}$ = exergy rate of brake power,
$\boldsymbol{A_{ex}}$ = exergy rate of exhaust gases,
$\boldsymbol{A_d}$ = destroyed exergy rate)

The exergy efficiencies of fuel blends; including D100, E5, E10, M5 and M10 are presented in Fig. 9. As can be seen, by adding alcoholic compounds to the diesel fuel the exergy efficiency is reduced. But as mentioned earlier, due to the lower LHV of the ethanol and methanol and lower brake power related to them, the exergy efficiency doesn't change significantly. This shows that with adding methanol and ethanol to diesel fuel, emissions of the engine are reduced while the exergy efficiency remains almost constant.

Table 4. Exergy analysis of the different fuel blends

Fuel type	A_{in} (kW)	A_{ex} (kW)	A_{cw} (kW)	A_{bp} (kW)	A_d (kW)	η_{ex} (%)
D100	878.91	165.19	24.24	310.41	379.06	56.87
E5	861.56	161.03	23.80	304.15	372.57	56.78
E10	847.76	157.81	23.32	298.01	368.61	56.52
M5	853.26	156.14	23.59	301.55	371.96	56.40
M10	831.15	151.68	22.93	292.78	363.75	56.23

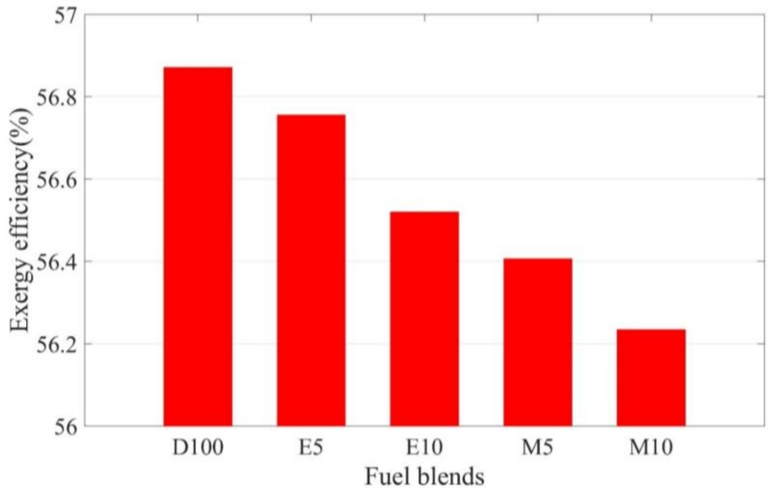

Fig.9. Exergy efficiency for different fuel blends

Figure 10 shows the exergy destruction for engine speed from 900 to 1900 rpm. As seen, by enhancement the engine speed, exergy destruction declines, due to the same input exergy and increasing exergy of the shaft power (brake power). Although adding ethanol and methanol to diesel fuel reduces braking power, it also reduces input exergy, which decreases exergy destruction. So as seen in Fig. 10, exergy destruction of the D100 case is higher than other cases for all engine speeds. This indicates that adding ethanol and methanol to diesel fuel is appropriate to reduce emissions while not greatly altering performance.

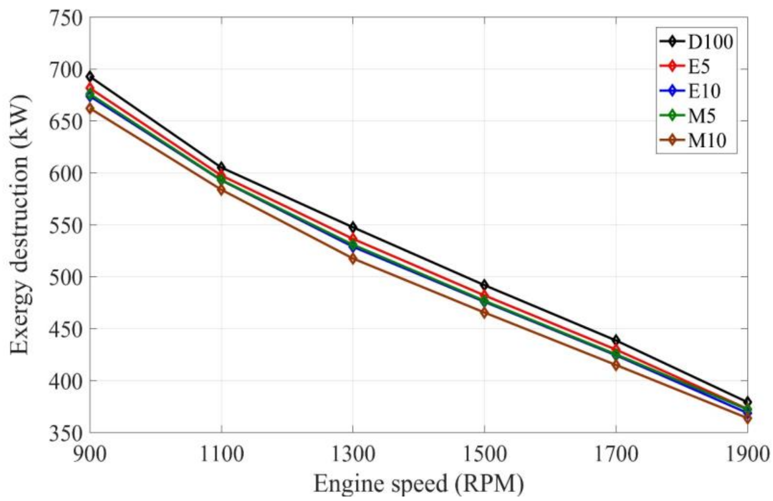

Fig.10. Exergy destruction versus engine speed

4. Conclusions

In this study one-dimensional modeling of a diesel engine in GT-Power was performed to investigate engine performance and emissions with the M5, E10, E5 and M10 alcoholic fuels. The engine used for the current study is a 6-cylinders turbocharged diesel engine of Weichai Company. In the next step, exergy analysis is performed to investigate the efficiency and destruction of desired cases. The main conclusions of the work can be written as:

- Modeling results of these alternative fuels showed that the NOx emission in the engine was reduced by approximately 15%.
- Carbon monoxide concentrations are also decreased with the addition of ethanol and methanol to diesel fuel, due to more complete combustion resulting from increased oxygen concentration and leaner combustion of base fuel.
- The carbon dioxide concentration has reduced with the engine speed, the addition of alcoholic fuels makes combustion more complete and leaner than diesel base fuel.
- Reducing engine emissions has reduced torque output. Therefore, in order to achieve the desired torque, all aspects of using alternative fuels must be considered simultaneously.
- The exergy analysis results show that adding alcoholic fuels to diesel fuels doesn't affect the exergy efficiency significantly.
- Adding ethanol and methanol to diesel fuel decreases exergy destruction.
- Adding ethanol and methanol to diesel fuels up to 10% is an appropriate way to reduce emissions of the diesel engine.

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