MODELLING AND INVESTIGATION OF A DIESEL ENGINE WITH ETHANOL AND METHANOL ADDITIVES

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Abstract

In this study, the effects of ethanol-diesel (E5, E10 and E15) and methanol-diesel (M5, M10 and M15) fuel blends on the emissions of a diesel engine were investigated. In the simulation study, the single-cylinder, four-stroke, direct injection compression ignition engine was used. The results obtained from the use of ethanol-diesel and methanol-diesel fuel blends were compared to those of pure diesel fuel (E0 and M0). One-dimensional simulation of a single-cylinder diesel engine with the help of the GT-Power software has been done. This software is capable of predicting the exhaust emissions. The simulation is designed to test the characteristics of the diesel engine working with methanol-diesel blends at full load.

Keywords: Ethanol; Methanol; Diesel Engine; Engine Simulation; Engine Emissions

1. Introduction

Diminishing oil reserves and increasing prices, as well as continuously rising concern over energy security, ecological problems and global warming have been identified as the most influential environmental ones. Unconventional fuels derived from resources other than oil provide a means for sustainable development, energy efficiency, energy conservation and environmental protection. Some of the alternative fuels explored are natural gas, propane, ethanol, methanol and hydrogen etc. The use of additives from oxygen-containing fuels (methanol, ethanol, butanol, biodiesel, etc.) improves the environmental performance of vehicles, but their increased share leads to a deterioration in engine power performance [1]. For this reason, they are added in small percentages so far. Some of the major additives to improve fuel efficiency are organic compounds containing oxygen (oxygenates). At present times, ethanol and methanol are promising fuels used in vehicles as an alternative to petroleum fuels [2], [3]. One reason for using ethanol and methanol is that it can be produced from natural products or waste materials, unlike petrol, which is a non-renewable fuel source.

The oxygen content in the ethanol helps to reduction of soot and particulate matter formation during the combustion. The combustion process is also enhanced by the using of ethanol-diesel blended fuels compared to the pure diesel.

Higher oxygen content and low sulphur content of methanol lead to a less pollutant emission. The higher laminar flame propagation speed leads to finishing the combustion earlier, which leads to improving the thermal efficiency of the engine.

The addition of ethanol and methanol to diesel reduces the cetane number of the blended fuel which increases the ignition delay period. Viscosity and energy density are also reduced. The dynamic viscosity of ethanol and methanol (table 2) is much lower than diesel which can cause a lubricity problem.

From the literature review, it was concluded that the emission and performance characteristics of different blends of ethanol and methanol in different engines have not been investigated sufficiently. For this reason, this study investigated the effects of ethanol–diesel and methanol–diesel fuel blends on the performance and emissions characteristics of a compression ignition engine at different engine speed and compared them with those of the pure diesel.

Engine simulation is becoming an increasingly important engineering tool for time and cost efficiency in the development of internal combustion engines (ICEs). Most of the results that are obtained by simulation are rather difficult to be obtained experimentally. Another advantage is that the engine design parameters can easily be changed during the simulation and an optimum design condition may be recommended. The use of Computational Fluid Dynamics (CFD) simulations allow researchers to understand flow behaviour and quantify important flow parameters such as mass flow rates or pressure drops, provided that the CFD tools have been properly validated against experimental results [2].

Engine modelling is a very large subject, in part because of the range of engine configurations possible and the variety of alternative analytical techniques or sub-models, which can be applied in overall engine models. Engine modelling is a fruitful research area and as a result many research laboratories have produced their own engine thermodynamics models with varying degrees of complexity, scope and ease to use

2. Exposition

There are numerous studies on the use of mixtures of ethanol-diesel and methanol-diesel to power compression ignition engines, which show the advantages and some limitations resulting from such a solution. In one of the studies, the author studied the combustion, performance and emission from a four cylinder, naturally aspirated diesel engine using diesel-ethanol blends up to 15% by volume. They observed that maximum pressure, heat release rate and ignition delay period increased with the ethanol addition. Brake specific fuel consumption and brake thermal efficiency were increased with the ethanol addition. The emissions of CO, HC, NO_x and PM were less with the use of ethanol diesel blends [4]. In another report [5], the author studied Euro V diesel fuel, pure biodiesel and biodiesel blended with 5%, 10% and 15% of ethanol and 5%, 10% and 15% of methanol in a four-cylinder naturally-aspirated direct-injection diesel engine. The results showed that compared with Euro V diesel fuel, the blended fuels could lead to the reduction of both NO_x and PM emissions of a diesel engine, with the methanol blends being more effective than the ethanol blends. Another authors [6] has investigated a naturally aspirated, four-stroke, single cylinder diesel engine with methanol-diesel blended fuel. They observed a reduction in CO, HC and smoke emissions. It was reported by them, that NO_x emission increased with the addition of methanol in the blended fuel. Whereas, brake specific fuel consumption was increased and brake thermal efficiency was decreased with the addition of methanol. Also, other interesting research [7] studied the performance and emission characteristics of the engine fueled with biodiesel–methanol–diesel and biodiesel–ethanol–diesel in a compression ignition engine. The results showed that biodiesel–alcohol–diesel blends had higher brake-specific fuel consumption than diesel. As alcohol concentrations in blends increased, CO and HC emissions increased, while NO_x emissions were reduced. Also, methanol blends were more effective than ethanol blends for reducing CO and HC emissions, while NO_x reduction was achieved by ethanol blends. In another study [8] conducted with a 5.9 liter diesel engine, torque was tested with two blends containing 10% and 15% ethanol. Engine power is reduced by about 8% with both fuel blends.

Description	Value
Fuel	Diesel
Engine type	Single cylinder
Piston diameter [mm]	87,5
Piston stroke [mm]	110
Volume [cc]	661
Compression ratio	17,5:1
Power [kW]	3,5
Maximum rotation speed [min ⁻¹]	1500
Opening suction valve before TDC [°]	4,5
Closing suction valve after BDC [°]	35,5
Opening exhaust valve before TDC [°]	35,5
Closing exhaust valve after BDC [°]	4,5

Table 1. Characteristics of the diesel engine

The characteristics of the diesel engine used in the study are shown in Table 1, Figure 2 and Figure 3 shows the variation of the specific fuel consumption at different ethanol and methanol concentrations of 0% Ethanol (E0), 5% Ethanol (E5), 10% Ethanol (E10), 15% Ethanol (E15), 0% Methanol (M0), 5% Methanol (M5), 10% Methanol (M10),

15% Methanol. The study was performed at an engine speed of 1500 min^{-1} . The results obtained show that with an increase in the ethanol and methanol concentration in the mixture (E5 - E15) and (M5 - M15) the specific fuel consumption increases, with a maximum specific consumption being obtained in mixtures with 15% ethanol E15. The lower ethanol calorific value of 26.8 and methanol caloric value of 19.7 compared to pure diesel fuel 43 (Table 1) increases the fuel consumption to obtain the same amount of energy. Also, the low density and viscosity of ethanol and methanol relative to pure diesel contributes to higher specific fuel consumption.

Figure 1 shows a single-cylinder diesel engine, developed using the GT-Power software.

Fig. 1. Single-cylinder diesel engine model

In the table. 2 are given some of the basic properties of diesel fuel ethanol and methanol.

Fuel properties	Unit	Diesel	Ethanol	Methanol
Chemical formula	-	$C_{10}\sim H_{15}$	C_2H_5OH	CH_3OH
Density at 20 °C	kg/m^3	837	788	790
Cetane number	-	50	5-8	3,8
Viscosity at 40 °C	mm^2/s	2.6	1,2	0,44
Surface tension at 20 °C	N/m	0,023	0,015	0,023
Calorific value	MJ/kg	43	26,8	19,7
Specific heat capacity	J/kg°C	1850	2100	2545
Boiling point	°C	180-360	78	65
Oxygen content	% weight	0	34,8	37,5
Latent heat of evaporation	kJ/kg	250	840	1178
Hydrogen content	(wt%)	~14	13	1300
Fuel-air attitude	-	15,0	9,0	6,47
Molecular weight	kg	170	46	32

Table 2. Essential properties of diesel and ethanol

Fig. 2. Specific fuel consumption variation at different concentrations of ethanol

Fig. 3. Specific fuel consumption variation at different concentrations of methanol

Figure 4 and Figure 5 shows the variation of the exhaust gas temperature of the engine at a different concentration of ethanol and methanol. From the results obtained, it becomes clear that with the increase of the load the temperature of the exhausted gas increases. With an increase in the ethanol and methanol content (E10 - E15) and (M10 - M15), a slight increase in the exhaust gas temperature is noticeable and is the highest at E15 and M15. This can be explained by the improvement of the combustion process due to the presence of oxygen in the ethanol mixtures. Higher latent heat of ethanol evaporation is the other factor that influences the reduction of exhaust temperature.

Fig. 4. Exhaust gases temperature variation at different ethanol concentrations

Fig. 5. Exhaust gases temperature variation at different methanol concentrations

The variation of the ignition delay period at different concentrations of ethanol and methanol is shown in Fig. 6. By increasing the ethanol and methanol content in the mixture (E5 - E15) and (M5 - M15) the cetane number of the mixture is reduced resulting in an ignition delay period. At full load, the ignition delay period of pure diesel fuel is 15,620 compared to the E15 mixture, which is 23,120 and M15 mixture, which is 23,120. The longer ignition delay period of ethanol and methanol mixtures results in the formation of poorer mixtures, which in turn leads to a decrease in the combustion temperature and a reduction in the formation of nitrogen oxides.

Fig. 6. Variation of the ignition delay period at different concentrations of ethanol and methanol

The in-cylinder pressure variation, depending on the crankshaft angle, at different ethanol and methanol concentrations, is shown in Fig. 7. From the results obtained, it can be seen that the maximum pressure for pure diesel fuel is 7.96 MPa at 50° after TDC, for the mixture E5 is 7.8 MPa at 50° after TDC, for the mixture E10 is 7.56 MPa at 60° after TDC and for E15 is 7.15 MPa at 70° after TDC. For methanol mixtures: maximum in-cylinder pressure M5 is 7.7 MPa at 8°C after TDC, for the mixture M10 is 7.38 MPa at 6°C after TDC and for M15 is 6.72 MPa at 6°C after TDC. Due to the lower cetane number of mixtures with ethanol (E5 - E15) and methanol (M5 - M15) the start of the combustion process occurs later. Increasing the methanol content of the mixture, the cetane number of the blends gets lowered and causes increase in ignition delay. That could lead to burn more fuel in the premixed phase of combustion, thus the rate of pressure rise increases and peak pressure decreases.

Fig. 7. In-cylinder pressure variation depending on the crankshaft angle with different concentrations of ethanol and methanol

Figure 8 shows the variation of NO_x emissions depending on the ethanol and methanol concentration in the diesel fuel mixture. As the ethanol and methanol concentration increases in the mixture, NO_x decreases due to the low latent heat of evaporation of the mixture. Another possible cause is the formation of a lean mixture, which leads to a decrease in the combustion temperature. The decreasing of NO_x emissions is more with ethanol for the same percentage of the blends.

Fig. 8. Variation of NO_x emissions at different concentrations of ethanol and methanol

Fig. 9. Variation CO₂ emissions at different concentrations of ethanol

The CO₂ variation at different concentrations of ethanol and methanol is shown in Fig. 9 and Fig. 10. The figures show that blending diesel fuel with ethanol and methanol results in CO₂ reduction at low loads. Increasing the ethanol and methanol content of the mixture reduces the C/H ratio and increases the O₂ content in the cylinder. This circumstance creates a condition for a lean mixture and reduces the temperature at the end of the combustion process. Also, the reduction of CO₂ occurs in parallel with a decrease in the C/H ratio.

With low engine load, the combustion temperature is insufficient to initiate full combustion, resulting in an increase in CO₂ emissions for all tested fuels and mixtures. However, at higher loads, combustion temperatures are high enough to achieve fuller combustion, resulting in a reduction in CO₂ for all tested fuels and mixtures. The reduction of CO₂ emissions has also been reported by other researchers [9].

Fig. 10. Variation CO₂ emissions at different concentrations of methanol

3. Conclusion

The main purpose of this study is to investigate the effect of using ethanol-diesel and methanol-diesel fuel blends on the diesel engine characteristics. According to the simulation results, it can be concluded that:

1. By increasing the ethanol and methanol concentration in the mixture (E5-E15) and (M5-M15), the specific fuel consumption increases, with a maximum specific fuel consumption being obtained in mixtures with 15% ethanol E15.
2. With an increase in the ethanol and methanol concentration in the mixture, a slight increase in the exhaust gas temperature is observed in comparison with pure diesel.
3. Increasing the ethanol and methanol content of the mixture (E5 - E15) and (M5 - M15) reduces its cetane number, resulting in a long ignition delay period. At full load, the ignition delay period for pure diesel fuel is 15,620 compared to the E15 mixture, which is 23,120 and for M15 mixture is 23,120.
4. The maximum in-cylinder pressure for pure diesel fuel is 7.96 MPa at 50° after TDC, for the mixture E5 is 7.8 MPa at 50° after TDC, for E10 is 7.56 MPa at 60° after TDC and for E15 is 7.15 MPa at 70° after TDC, for the mixture M5 is 7.7 MPa at 80° after TDC, for M10 is 7.38 MPa at 60° after TDC and for M15 is 6.72 MPa at 80° after TDC. Due to the lower cetane number of mixtures with ethanol (E5 - E15) and methanol (M5 - M15) the start of the combustion process occurs later.
5. With an increase in the ethanol and methanol concentration, the NO_x emissions decrease, due to the low latent heat of evaporation of the mixture. Another possible cause is the formation of lean mixture, which leads to a decrease in the combustion temperature.
6. Increasing the ethanol concentration in the mixture results in CO₂ reduction at low loads.

As the future plan of the experiment team will investigate behaviour of diesel engine model with butanol additives and their impact on exhaust emissions.

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