Combustion Characteristics of an Indirect Injection (IDI) Diesel Engine Fueled with Ethanol/Diesel and Methanol/Diesel Blends at Different Injection Timings

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Abstract: In this study, the influence of methanol/diesel and ethanol/diesel fuel blends on the combustion characteristic of an IDI diesel engine was investigated at different injection timings by using five different fuel blends (diesel, M5, M10, E5 and E10). The tests were conducted at three different start of injection {25^o, 20^o (original injection timing) and 15^o CA before top dead center (BTDC)} under the same operating condition. The experimental results show that maximum cylinder gas pressure (P_{max}) and maximum heat release rate $(dQ/d\theta)_{max}$ increased with advanced fuel delivery timing for all test fuels. Although the values of P_{max} and $(dQ/d\theta)_{max}$ of E10 and M10 type fuels were observed at original injection and retarded injection (15^o CA BTDC) timings, those of the diesel fuel were obtained at advanced injection (25^o CA BTDC) timing. From the combustion characteristics of the test fuels, it was observed that ignition delay (ID), total combustion duration (TCD) and maximum pressure rise rate $(dP/d\theta)_{max}$ increased with advanced fuel delivery timing. The ID increased at original and advanced injection timings for ethanol/diesel and methanol/diesel fuel blends when compared to the diesel fuel. It was also found that increasing methanol or ethanol amount in the fuel blends caused to increase in ID and to decrease in TCD at all injection timings. At original injection timing, the $(dP/d\theta)_{max}$ increased with increasing methanol or ethanol amount in the fuel blends. To see the cycle to cycle variation, the fifty cycles of each fuel were also investigated at the different injection timings. It was found that, at the advanced injection timing, cyclic variability of the test fuels was higher when compared to the original and retarded injection timings. The maximum cyclic variability was observed with the M10 at the advanced injection timing.

Keywords: *Ethanol, Methanol, IDI diesel engine, Injection timing, Combustion characteristics*

1. Introduction

Compression ignition (CI) or diesel engines are widely used for transportation, automotive, agricultural applications and industrial sectors because of their high fuel economy and thermal efficiency. The existing CI engines operate with conventional diesel fuel derived from crude oil. It is well known that the world petroleum resources are limited and the production of crude oil is becoming more difficult and expensive. At the same time, with the increasing concern about environmental protection and more stringent government regulation, the researches on the decreasing of exhaust emissions and improving fuel economy have become a major research issue in the engine combustion and development. A lot of research related to the emissions reduction has been performed by using different injection parameters such as injection time and injection pressure, exhaust gas recirculation and oxygenated alternative fuels. In the recent years, methanol and ethanol are attractive oxygenated alternative fuels for diesel engines. The oxygenated alternative fuels such as methanol and ethanol have provided more oxygen during combustion. Therefore, the oxygenated alternative fuels and blends with gasoline and diesel fuel are more clean combustion processes than that of diesel and gasoline fuels [1-7]. The studies related to the alternative fuels should be enhanced for diesel engines especially for indirect injection (IDI) diesel engines. Because, they have a simple fuel injection system and lower injection pressure level. They do not depend upon the fuel quality and have lower ignition delay (ID) and faster combustion than direct injection (DI) diesel engines.

For a diesel engine, the fuel injection timing is a major parameter that affects the combustion and exhaust emissions. If the start of fuel injection timing is earlier, the initial air temperature and pressure will be lower, so that the ID will increase. The increase in the ID period causes to increase in the premixed burning phase, the cylinder gas temperature and the NO_x emissions. However, this trend decreases PM emissions. If the start of fuel injection timing is later (when piston is closer to TDC), the temperature and pressure will be slightly higher, therefore the ID will decrease. For this reason, injection timing variation has a strong effect on the combustion characteristics and exhaust emissions, because of changing maximum pressure and temperature in the cylinder.

Canakci et al. [2] experimentally investigated the combustion and exhaust emissions of a single cylinder diesel engine at three (25, 20 original injection timing and 15^o CA BTDC) different injection timings when methanol/diesel fuel blends were used from 0 to 15%, with an increment of 5%. The results indicated that the P_{max} decreased and the ID increased with the increase of methanol mass fraction at all injection timings. The increment in the ID caused to the deteriorating combustion thereby reduced the P_{max}. Also advanced injection timing boosted the P_{max} and the rate of heat release because of the increase in ID. Huang et al. [8, 9] used the diesel/methanol blend and combustion characteristics and heat release analysis in a CI engine. According to the experimental results, the increase in methanol mass fraction in the diesel/methanol blends resulted in an increase in the heat release rate at the premixed burning phase and shortened the combustion duration of the diffusive burning phase. The ID increased with increasing of the methanol mass fraction. This trend was more obvious at low engine load and high engine speed. TCD and P_{max} increased by advancing fuel delivery timing. The P_{max}, the $(dP/d\theta)_{max}$ and the $(dQ/d\theta)_{max}$ of the diesel/methanol blends obtained a higher value than that of diesel fuel. Yao et al. [10] researched the effect of diesel/methanol compound combustion (DMCC) fuel injection method on combustion characteristics. In this fuel injection method, the methanol was injected into the air intake of each cylinder. The diesel fuel was injected into the cylinder to ignite a methanol/air mixture. This system was tested on naturally aspired diesel engine. The test results showed that the ID increased and the cylinder gas temperature reduced with the DMCC fuel injection method due to the high latent heat of methanol.

Xing-cai et al. [11] conducted research on the heat release and emissions of a high speed diesel engine fuelled with ethanol/diesel blend. They found that the ID increased and TCD shortened for ethanol/diesel fuels when compared to diesel fuel. It was observed that the maximum heat release rate of ethanol/diesel blends were lower than that of diesel fuel. In the other studies, Rakopoulos C.D. et al. [12] investigated the effect of ethanol/diesel blends with 5%, 10% and 15% (by vol.) ethanol on the combustion and emissions characteristics of a high speed direct injection diesel engine. According to the experimental results, the ID for the E15 blend was higher than pure diesel fuel; also there was no significant difference among the P_{max} for each load conditions.

The combustion characteristics of IDI diesel engines are different from the DI diesel engines, because of greater heat-transfer losses in the swirl chamber. This handicap causes the brake-specific fuel consumption (bsfc) of the IDI engine to increase and the total engine efficiency to decrease compared to that of a DI diesel engine. Because of these disadvantages of the IDI diesel engines, most engine research has focused on the DI diesel engines. However, IDI diesel engines have a simple fuel injection system and lower injection pressure level because of higher air velocity and rapidly occurring air-fuel mixture formation in both combustion chambers of the IDI diesel engines. In addition, they do not depend upon the fuel quality and produce lower exhaust emissions than DI diesel engines [13].

From the literature review, it was concluded that the combustion characteristics of an IDI diesel engine have not been clearly investigated when using methanol/diesel and ethanol/diesel fuel blends at different injection timings. For this reason, this study experimentally investigated the effects of methanol/diesel and ethanol/diesel fuel blends on the combustion characteristics of an IDI diesel engine and compared them with those of diesel fuel.

2. Materials and method

In this study, a naturally aspirated, water-cooled, four cylinders IDI diesel engine was used as a test engine. The test engine specifications are compression ratio: 21.47, the maximum brake torque (95 Nm) was obtained at 2000 rpm and the maximum power 38 kW at 4200 rpm, start of injection timing: 20^{o} CA BTDC and injector opening pressure: 130 bar. A hydraulic dynamometer was directly coupled to the engine output shaft. Fig. 1 shows the schematic diagram of the experimental setup. The following parameters were recorded during the each test: engine speed, load, fuel consumption, air flow rate, and ambient, cooling water inlet-outlet, and oil and exhaust temperatures. Conventional diesel fuel, methanol and ethanol were used, and their properties are shown in Table 1. To obtain cylinder gas pressure and fuel line pressure data, piezoelectric-type sensors were used. The cylinder gas pressure sensor was installed on the first cylinder of the engine head. The cylinder gas pressure was obtained by using a Kistler water-cooled piezoelectric sensor type 6061B. An AVL quartz pressure sensor 8QP500c was mounted on the fuel line of the first cylinder to measure the fuel line pressure. The outputs of the pressure sensors were amplified by a Kistler charge amplifier 5015A type. The output of the charge amplifier and a signal from the magnetic pick-up were converted to digital signals and recorded by an Advantech PCI 1716A data acquisition card, which has a 16-bit converter and 250 kS/s sample rate. The pressure and crank angle data were stored in a computer. A computer program was written to collect the pressure data, with a resolution of 0.25° of crankshaft angle. To analyze the cylinder gas pressure, a combustion analysis program was written. To eliminate cycle-cycle variation, the cylinder gas pressure data of 50 cycles were averaged using a computer program. Then, the pressure data was used to calculate the heat-release rate. Experiments were performed after the test engine reached to the steady-state conditions. The steady-state conditions were determined with the engine oil temperature (~70 °C). The test engine was run at least 5 min after the test engine was loaded, and then data was collected for each test. The test procedure was repeated 3 times to verify the each engine test condition, and the results were averaged.

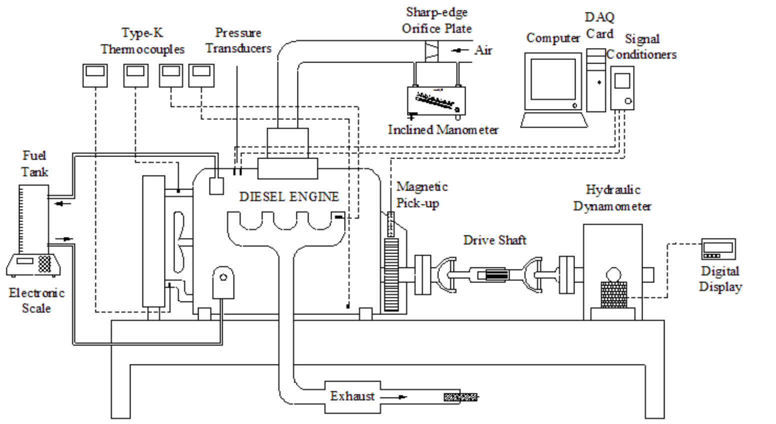

Fig. 1: Schematic diagram of the experimental set-up

Table 1. Properties of the test fuels

	Methanol	Ethanol	Diesel
Formula	CH_3OH	C_2H_5OH	$C_{10.8}H_{18.7}$
Molecular weight (kg/kmol)	32	46	170
Boiling temperature (^oC)	64.7	78	180 – 330
Density (g/cm^3, at 20 ^oC)	0.79	0.78	0.83
Auto-ignition temperature (^oC)	470	423	235
Lower heating value (MJ/kg)	20.27	26.8	43
Cetane number	4	5-8	50
Viscosity (mm^2/s, at 25^oC)	0.59	1.2	2.6
Heat of vaporization (MJ/kg)	1.11	0.856	0.280

3. Heat release analysis

The heat release analysis was based on the changes of the cylinder gas pressure and cylinder volume during the cycle. Therefore, some assumptions were made to calculate the heat release rate. It was assumed that no passage throttling losses exist between both chambers. Large temperature gradients, pressure waves, leakage through the piston rings, fuel vaporization and charge mixtures were ignored. Hence the intake and exhaust valves assumed to be closed. After using these assumptions, the heat release rate is calculated by using the following formula:

$$\frac{dQ}{d\theta} = \left[\frac{k}{k-1}\right] P \frac{dV}{d\theta} + \left[\frac{1}{k-1}\right] V \frac{dP}{d\theta}$$

Where: *(dQ/dθ)* is the combination of heat-release rate, *P* is cylinder gas pressure, *V* is cylinder volume, θ is the crank angle, k is the ratio of specific heats.

The parameters of combustion characteristics are ID, start of combustion, TCD which are obtained from heat release curve. The heat release curve in a diesel engine examines ID and TCD. The ID is defined as the time between the start of injection and the start of combustion. The start of injection time is determined by the fuel line pressure reached the injector nozzle opening pressure. The start of combustion is defined as the point where the heat release rate turns from negative to zero. The TCD is defined as the time from the start of combustion to the end of the heat release.

4. Results and discussion

In this study, the engine test was conducted at three different start of injection {25^o (advanced), 20^o (original) and 15^o (retarded) CA BTDC} under 1400 rpm and 40 Nm. The maximum fuel/air ratio was observed at 1400 rpm for diesel fuel, therefore the test condition was chosen as 1400 rpm. The relationship between the combustion characteristics and injection timings were focused by using conventional diesel fuel (D), E5, E10, M5 and M10. These fuel blends content of methanol or ethanol in different mass ratios (e.g., E5 contains 5% ethanol and 95% diesel fuel by mass). In this study, the combustion characteristics defined as the cylinder gas pressure and heat release rate were analyzed as shown in Fig. 2. The ID, TCD, $(dQ/d\theta)_{max}$, $(dP/d\theta)_{max}$, and the variation of the fifty consecutive P_{max} were also investigated as shown in Figures 3 and 4.

Fig. 2 illustrates the cylinder gas pressure and heat release rate of test fuels at three different injection timings under the same engine operating conditions. As shown in Fig. 2, it can be clearly seen that the cylinder gas pressure and heat release rate increased by advancing fuel injection timings for all test fuels. This behavior was such that, as injection started earlier, the cylinder gas pressure and the heat release rate become higher due to more fuel injected during the ID period. In addition, the location of P_{max} and the start of combustion points occurred early with advanced fuel injection timing. Therefore the premixed combustion phases occurred earlier and also this phase finished before TDC at 25^o and 20^o CA injection timing. Diffusion or controlled combustion phase of the M10 and E10 formed lower burning than that of other test fuels at original injection timing. The lower viscosity and density of M10 and E10 led to high atomization and vaporization, so the lower burning was observed in the diffusion combustion phase. At the same time, the fraction of the heat release in the premixed or uncontrolled burning phase of the E10 and M10 blends decreased and the peak of premixed combustion phase of these blends increased at original injection timing. These results can be explained by increasing ethanol and methanol mass fraction in the blends.

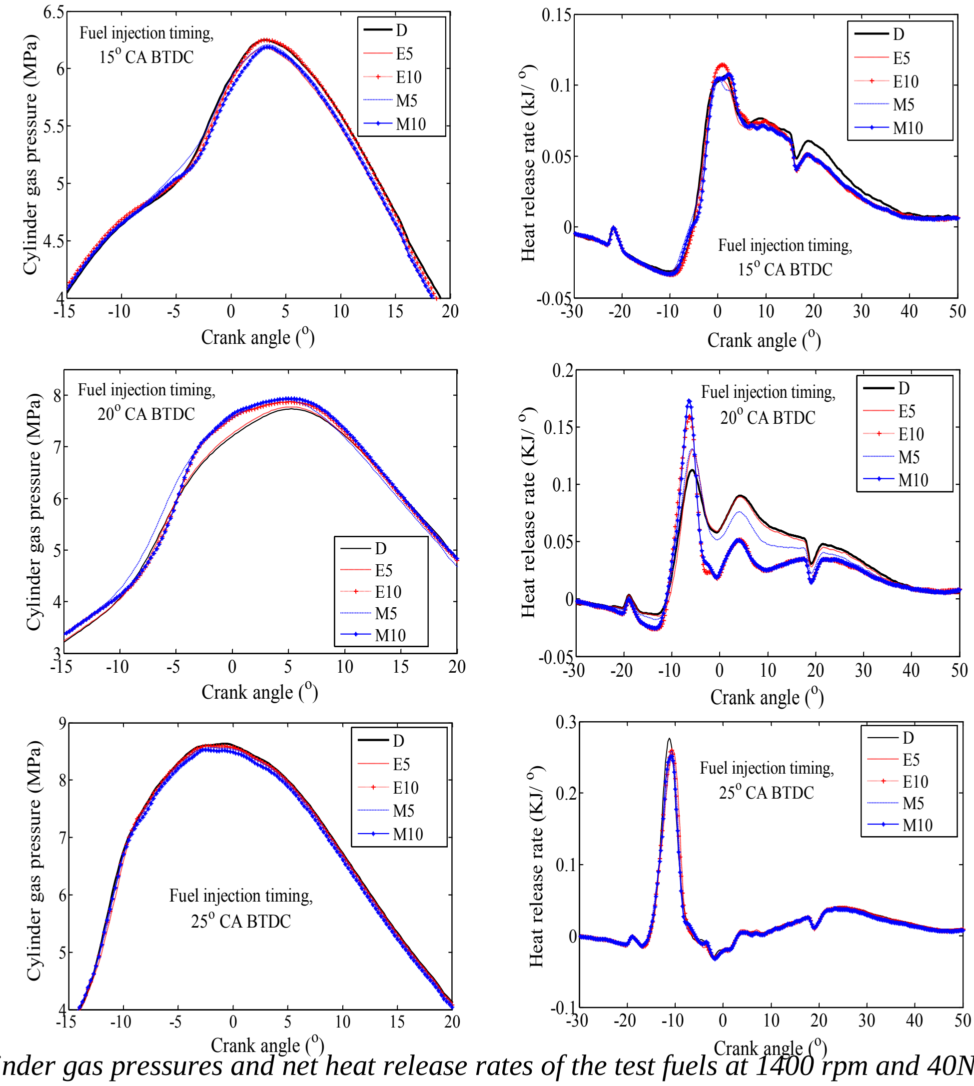

Fig.2 Cylinder gas pressures and net heat release rates of the test fuels at 1400 rpm and 40Nm

Fig. 3 shows the variation of ID, TCD, $(dQ/d\theta)_{max}$ and $(dP/d\theta)_{max}$ under three different injection timings. It was observed that the ID decreased with retarded injection timing for all test fuels. This behavior can be explained by the pressure, temperature and vaporization in the cylinder increased with retarded injection timings. It was found that, at advanced and original injection timings, the IDs of the blends are longer than that of conventional diesel fuel. This effect was interpreted by two different reasons. The first reason is that cetane numbers of the blends which are lower than that of conventional diesel due to the cetane number decreased with the increase in methanol and ethanol mass fraction in the fuel blends. The second reason is that the methanol and ethanol have higher heat of vaporization than that of conventional diesel fuel. It was observed that the IDs of the E5 and M5 were shorter than that of E10 and M10 due to lower cetane number of the E10 and M10 blends. The TCD decreased with retarded fuel injection timing for all test fuels. The reason for the decrease in TCD is the increase in the premixed or uncontrolled combustion phase due to long ID and decrease in the diffusion or controlled combustion phase. It was revealed that, at all injection timings, TCD with blends was longer than that of conventional diesel fuel. This result can be explained by the increasing amount of the oxygen in the blends. It is known that the increase in amount of the oxygen enhances the combustion and causes to the diffusion combustion phase which becomes shorter.

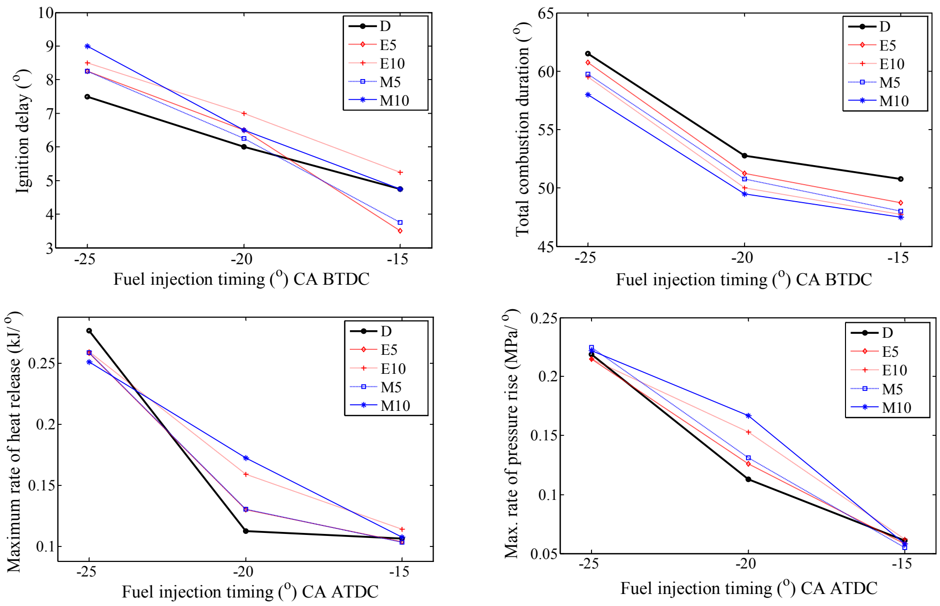

Fig.3 Effect of injection timing on the ID, TCD, $(dQ/d\theta)_{max}$ and $(dP/d\theta)_{max}$ at 1400 rpm, 40 Nm for each test fuel

As shown Fig. 2, the net heat-release profile has a slight negative dip during the ID period, which is mainly heat loss from the cylinder during the fuel vaporizing phase. It is more obvious at retarded injection timings. Because of the temperature in the cylinder increasing with retarded injection timing, the injected fuel during the ID period causes an increase in the evaporation heat. Therefore, the $(dQ/d\theta)_{max}$ decreased with retarded injection timings for all test fuels. The $(dP/d\theta)_{max}$ increased with the advancing injection timing as shown in the Fig. 3. This can be attributed to the increase in the injected fuel into the engine cylinder during the ID period, and so that produced higher the $(dP/d\theta)_{max}$ and the cylinder gas pressure. Also, there is no significant difference among the $(dQ/d\theta)_{max}$ and the $(dP/d\theta)_{max}$ of the test fuels at advanced and retarded injection timing, while at original injection timing, the $(dQ/d\theta)_{max}$ and the $(dP/d\theta)_{max}$ of the blends were higher than that of conventional diesel fuel. The main reason for this situation is that in order to obtain the same bmep from the blends, more fuel was injected into engine cylinders due to the blends have lower heating value than that of conventional diesel fuel. At the same time, it w as observed that the $(dQ/d\theta)_{max}$ and the$(dP/d\theta)_{max}$ increased with the increase in the mass fraction methanol and ethanol in the blends at original injection timing. This was caused by E10 and M10 fuel blends which have more oxygen rate than E5, M5 and conventional diesel fuel. Thereby, the combustion became better and the $(dQ/d\theta)_{max}$ and the $(dP/d\theta)_{max}$ increased.

Fig. 4 shows the average of the P_{max} achieved from 50 consecutive cycles for all test fuels and all injection timings. It was observed that the cyclic variability decreased with the retarding fuel injection timings. Specially, at 25° CA injection timing, the cyclic variability of the M10 test fuel was higher than those of other injection timings. As shown in Fig.4, similar cyclic variability and the smooth operation of the engine can be achieved by using E5, E10, M5 and M10 blends when compared the conventional diesel fuel.

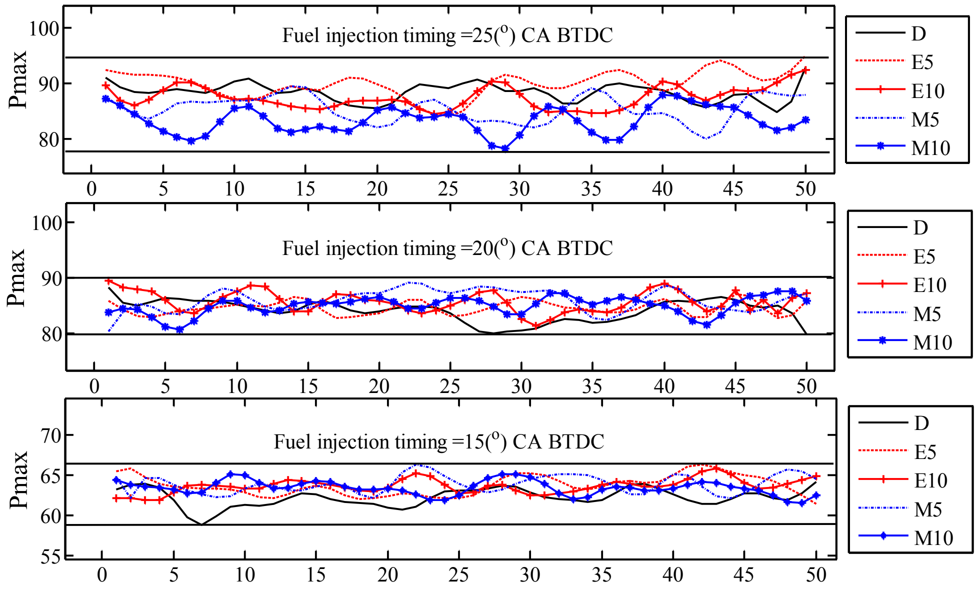

Fig.4 Effects of blends and fuel injection timing on the variation of maximum cylinder gas pressure

5. Conclusion

The paper presented the results of experimental research on the effects of injection timing on the combustion characteristics of an IDI diesel engine using the ethanol and methanol blends with diesel fuel. The following conclusions can be drawn from the current paper:

(1) The P_{max} and premixed combustion rate increased with advanced fuel injection timings for all test fuels.
(2) The location of P_{max} and the start of combustion points occurred early with advanced fuel injection timing.
(3) The ID and TCD decreased with retarded injection timing for all test fuels.
(4) It was determined that the IDs of the blends were longer than that of conventional diesel fuel at originally and advanced injection timings.
(5) An increase in the mass fraction of the methanol and ethanol in the fuel blends generally caused to increase in ID, but it decreased TCD.
(6) The retarding of injection timing decreased the $(dQ/d\theta)_{max}$ and the $(dP/d\theta)_{max}$ for all test fuels.
(7) It was found that the characteristics of $(dQ/d\theta)_{max}$ and $(dP/d\theta)_{max}$ of the blends are higher than that of conventional diesel fuel. These characteristics increased with the increase of methanol and ethanol mass fraction in the fuel blends at original injection timing.
(8) It was observed that the cyclic variability decreased with the retarding fuel injection timings. Also, the maximum cyclic variability was observed with the M10 at the advanced injection timing. The fuel blends used in the current study may replace with

conventional diesel fuel in terms of the combustion characteristics, cycle to cycle variation and smoothness of the engine operation.

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