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COMBUSTION ANALYSIS OF JATROPHA METHYL ESTER AND ITS ETHANOL AND ACETONE BLENDS IN A DIESEL ENGINE

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ABSTRACT

The present study analyzes the heat release and combustion characteristics of diesel, pure biodiesel (B100) obtained from Jatropha oil and its 80% blending with Ethanol (BD80E20) and Acetone (BD80AC20) as fuels in a single cylinder four stroke direct injection (DI) diesel engine. The analyses of experimental results indicate that the pressure of the in-cylinder gases rise early in case of B100, BD80E20 and BD80AC20. Early rate of pressure rise causes the cylinder pressure to rise early in case of the alternate fuels with a resulting low peak rate of pressure rise and peak pressure. The Net Heat Release Rate (NHRR) analysis shows lower rate of heat release with B100, BD80AC20 during the premixed phase of combustion. However, the rate of heat release is higher for these fuels in the diffusion combustion phase. As a result, the cumulative heat release (CHR) is higher for B100 and BD80AC20 as compared to that of normal diesel. The pressure in the fuel line was remarkably higher for B100, while the biodiesel ethanol blend shows almost the same trend and the fuel pressure is slightly lower for BD80AC20 when compared with diesel fuel pressure. Ignition delay in case of B100, BD80E20 and BD80AC20 is less compared to normal diesel; however, there was slight increase in delay period with addition of ethanol and acetone in B100. Combustion durations are longer for all these alternate fuels as compared to that of conventional diesel.

Keywords: Biodiesel, diesel engine, combustion, heat release

1. INTRODUCTION

In India, lot of research activities are going on in the field of production of biodiesel, mainly from non edible sources and subsequently testing their suitability in diesel engine either in the form of blending with conventional diesel or the neat biodiesel. The advantages with biodiesel fuel are well known. Biodiesel is non toxic, renewable and environment friendly with minimum sulpher and aromatics content. However, there are certain disadvantages as well. Biodiesel has lower heating value, it is comparatively less volatile. It has higher viscosity compared to conventional diesel fuel. Combination of all these lead to problem of higher fuel consumption, poor atomization of fuel spray, incomplete combustion, coking of the injector tips, oil ring sticking and thickening and gelling of the lubricant oil. Unsaturation in the fatty acid chain, oxidation, gum formation, water absorption and microbial activity during storage are some of the issues that require research attention with biodiesel. However, if standard biodiesel specification is maintained, these are normally not recognized as major problems.

Many reports are available on engine performance, fuel combustion and emission evaluation with non edible biodiesel. Some of these are listed in Table 1. Among these, the works in the references [6, 8, 10-12] specifically analyse the combustion characteristics of bio-diesel fuel. Besides these, some other analyses done on combustion of biodiesel fuel include the works of Shaheed and swain [14], Canakci [15], Qi et al.[16], Radu et al.[17], Saravanan et al. [18] etc. Rao et al. [6] observed lower combustion duration for Jatropha methyl ester and its diesel blended fuels compared to diesel. The decrease in combustion duration was mainly due to the early start of combustion that causes a higher pressure and temperature rise in the combustion chamber, thereby completing the combustion process at a faster rate. Banapurmath et al. [8] however, observed higher combustion duration with Honge, Jatropha and sesame oil methyl esters which they attributed to the longer diffusion combustion phase of the esters. Sahoo and Das [10] made a combustion analysis using neat biodiesel and their blends (B20 and B50) from Jatropha, Karanja and Polanga oil in a single cylinder air cooled diesel engine at varying loads (0, 50 and 100%). The ignition delays were found consistently shorter for the neat biodiesels when compared with diesel. Shaheed and swain [14] presented a comparative combustion analysis of coconut oil (COIL), coconut oil methyl esters (COME) and diesel in a naturally aspirated single-cylinder diesel engine. Their results showed that the ignition delay and the magnitude of the heat release during premixed and diffusion combustion for COME

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and COIL were less than that of diesel. Canakci [15] compared the combustion characteristics of soybean based biodiesel with two different variety of diesel fuel (No. 1 and No.2 diesel) in a four cylinder turbocharged DI diesel engine at full engine load at 1400 rpm. He observed that the fuel injection timing was advanced and ignition delay was shorter for biodiesel. Qi et al. [16] found early start of combustion with biodiesel from soybean. They found that at lower engine loads, the peak cylinder pressure, the peak rate of pressure rise and the peak of heat release rate during premixed combustion were higher for biodiesel. However at higher engine loads, the peak cylinder pressure of biodiesel was almost similar to that of diesel, but the peak rate of pressure rise and the peak of heat release rate were lower than diesel. Similar observations were found with Koroch seed oil methyl ester blends [13]. Radu et al. [17] observed lower injection rate, higher injection duration, lower combustion pressure using waste oil biodiesel blend. They found that lower pressure wave propagation time and ignition delay of the biodiesel caused earlier start of combustion and a lower heat release rate. Saravana et al. [18] analyzed the combustion characteristics of diesel and B20 obtained from crude rice bran oil methyl ester (CRBME) in a stationary small duty DI diesel engine. Their observation was that the delay period and the maximum rate of pressure rise for CRBME blend were lower than those for diesel. The magnitude of maximum heat release rate for the CRBME blend was less and it occurred in advance when compared to diesel. Most of the combustion analyses reveal lower ignition delay, early heat release although biodiesel has slightly higher viscosity and lower volatility. However, results vary considerably depending upon the type of biodiesel used, engine configurations, test conditions.

Table 1: Survey of past literature on engine performance with non edible biodiesel

Type of biodiesel	Engine Configuration	Reference
Jatropha curcus oil and its diesel blends.	Single cylinder four stroke DI diesel engine, rated power 3.68 kW@1500 rpm	Pramanik, 2003 [1]
Karanja methyl ester and its blends	Single cylinder four stroke DI water cooled diesel, Rated power 7.5 kW @ 3000 rpm	Raheman and Phadatare, 2004 [2]
Rubber seed oil and its diesel blend	Single cylinder four stroke DI diesel engine, Rated output 5.5 kW @ 1500 rpm	Ramadhas et al., 2005 [3]
Tobacco seed oil methyl ester blends	Four cylinder, four stroke turbocharged indirect injection (IDI) variable speed diesel engine	Usta, 2005 [4]
Mahua seed oil methyl and ethyl ester	Single cylinder four stroke Ricardo E6 engine, Rated output 9 kW @ 1500 rpm	Raheman and Ghadge, 2007 [5]
Jatropha methyl ester blends	single cylinder, four stroke, naturally aspirated, air cooled DI diesel engine, rated power 4.4 kW@ 1500 rpm	Rao et al., 2007 [6]
Jatropha oil (preheated and blends)	Single cylinder four stroke DI diesel engine, 7.4 kW @ 1500 rpm	Agarwal and Agarwal, 2007 [7]
Honge, Jatropha and sesame oil methyl esters	Single cylinder four stroke DI diesel engine, rated power 5.2 kW@1500 rpm	Banapurmath et al., 2008 [8]
Jatropha, Karanja and Polanga based biodiesel	3-cylinder 4-stroke diesel engine, rated power 44.1 kW @2200 rpm	Sahoo et al., 2009 [9]
Jatropha, Karanja and Polanga based biodiesel	Single cylinder 4-stroke air cooled diesel engine, rated power 6 kW	Sahoo and Das, 2009 [10]
Jatropha methyl ester and Karanja methyl ester	Single cylinder four stroke DI diesel engine, rated power 3.5 kW @ 1500 rpm	Jindal et al., 2010 [11]
Jatropha Oil	Single cylinder four stroke direct injection (DI) diesel engine, rated power 3.68 kW@1500 rpm	Kumar et al.,2010 [12]
Koroch seed oil methyl ester blends	Single cylinder four stroke DI diesel engine, rated power 3.5 kW @ 1500 rpm	Gogoi and Baruah, 2011[13]

From the literature review on engine performance analysis, it was found that higher fuel consumption and reduced brake thermal efficiency are most common observation with biodiesel fuel. B100 is usually viscous and less volatile compared to conventional diesel fuel. When it is used in unmodified diesel engine, engine consumes more fuel due to its lower heating value and this together with its higher viscosity and lower volatility leads to poor atomization of the fuel spray resulting in incomplete fuel combustion. Ethanol and acetone have comparatively lower density and viscosity and are relatively volatile, particularly the acetone. Therefore the objective of the present study is to find how addition of ethanol and acetone in B100 affect its combustion behavior. Ethanol and acetone was mixed with B100 and all these fuel including normal diesel and B100 were tested in a single cylinder four stroke naturally aspirated DI diesel engine to make a comparative study of heat release and combustion. The properties of the various fuels are given in Table 2.

2. METHODOLOGY

Specifications of test engine (Fig. 1) are given in Table 3. The engine is provided with necessary instruments for combustion pressure, fuel pressure and crank-angle (CA) measurements. The in-cylinder and the fuel pressure are sensed by two piezo sensors. Signals from these pressure transducers are fed to a charge amplifier. A high precision CA encoder is used to give signals for top dead centre (TDC) and the CA. The signals from the charge amplifier and the CA encoder are supplied to a data acquisition system which is interfaced to a computer through engine indicator for obtaining pressure CA diagram. There are provisions in set up also for interfacing airflow, fuel flow and load measurement. The engine is coupled with an eddy current dynamometer for controlling the engine torque through computer. A Lab view based engine performance analysis software package evaluates the on line engine performance. The tests were conducted at steady state and full load at average engine speed of 1,535 rpm where the average engine torque was 21.85 Nm. This yielded an average brake power (BP) of 3.5 kW in each fuel test. Three test runs were performed under identical conditions to check for the repeatability of all the results. The repeatability of the results was found to be within an acceptable limit. The test results were then averaged and the average test results have been reported.

Table 2: Properties of diesel, B100 and its ethanol and acetone blends

Property	Diesel	B100	BD80E20	BD80AC20
Sp. gravity	0.83	0.89	0.87	0.87
Kinematic viscosity at 40°C (cSt)	2.7	6.4	4.3	2.8
Pour point (°C)	-6	-3	-6	-9
Saponification value (mg KOH/gm)	0.04	220	172	146
Calorific value	45220	36800	35962	37056

Table 3: Engine Specifications

Make and Model	Kirloskar –TV1
Rated power and speed	3.5 kW and 1500 rpm
Type of Engine	1cylinder, DI type, 4Stroke
Compression ratio (CR)	12-18:1
IV Opening	4.5° before TDC
IV Closing	35.5° after BDC
EV Opening	35.5° before BDC
EV Closing	4.5° after TDC
Bore & Stroke	87.5mm and 110mm
Nozzle opening pressure	220 bar
Cooling Medium	Water cooled

Fig. 1: Test engine set up

3. RESULTS AND DISCUSSION

3.1 Combustion analysis

The combustion characteristics of an engine is usually defined by parameters such as pressure crank angle diagram, peak pressure, rate of pressure rise, NHRR, CHR, ignition delay and combustion duration etc. Analysis of fuel combustion characteristics is important because it provides the above important information which in turn helps in interpreting engine performance and exhaust emissions. Besides, combustion characteristics sometimes can also be used to explain the effects of engine-operating conditions on the engine performance or to compare the alternative fuels under the same operating conditions. Hence, this has been done to assess the combustion characteristics of B100, BD80E20 and BD80AC20 and to compare with the combustion behavior of normal diesel.

3.1.1 Pressure crank angle variation and peak pressure

The cylinder pressure variation with crank angle at full load is shown in Fig. 2. Slight variation in pressure was observed with B100, BD80E20, and BD80AC20. Early pressure rise is distinct in case of the alternate fuels. B100 showed the earliest pressure rise followed by the ethanol and the acetone blend with close matching between the two. Pressure rise was slightly late for diesel; however there was a steep rise for diesel compared to B100 and its ethanol and acetone blends.

The peak pressure as can be seen from Fig. 3 was the maximum for diesel and it was the least for B100 at full load. Peak pressure for BD80E20 and BD80AC20 was between these two and it was marginally lower for BD80E20. The peak pressure values for B100, BD80E20, and BD80AC20 are 55.54 bar, 56.51 bar and 57.66 bar as compared to diesel peak pressure value of 58.33bar. The corresponding CA for peak pressure of B100, BD80E20, BD80AC20 and diesel are 372, 370, 370 and 369° CA respectively. Peak pressure depends on the amount of fuel taking part in the premixed combustion phase, which is governed by the delay period and spray envelope of the injected fuel. Larger the ignition delay more will be the fuel accumulation and higher is the peak pressure.

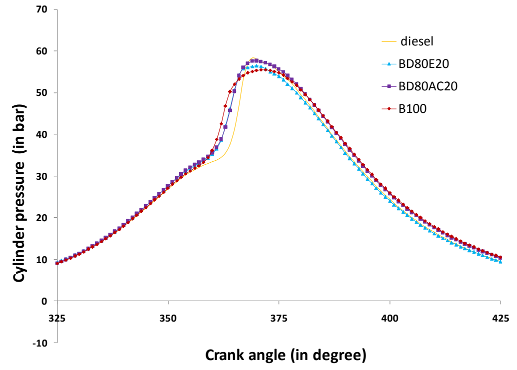

Fig. 2: Cylinder pressure variation for the tested fuels

3.1.2 Rate of pressure rise

The rate of pressure rise for the tested fuels at full load is presented in Fig. 4. Analysis of rate of pressure rise is

important in engine study because it is possible to determine how smoothly the combustion progresses in the combustion chamber from the observation of rate of pressure rise. It is necessary that the maximum rate of pressure rise should be as low as possible for reduced engine noise and increased engine life. It was observed that the pressure rise rate first decreased during the delay period for all the fuels and then it increased before the start of combustion (SOC) with a sharp rate of rise after SOC. The minimum value to which the pressure rise rate decreased during delay period was less for diesel compared to the other fuel blends. However the peak of the rate of pressure rise was less for the alternate fuel blends and the CA at which this occurred also advanced for the biodiesel blends. Among the alternate fuels, the peak rate of pressure rise value was the least for B100 and the blends ‘BD80E20’ and ‘BD80AC20’ showed almost the same trend. The early peaking characteristics of an engine with biodiesel fuel needs careful attention because a peak pressure occurring very

close to TDC or before may cause severe engine knock affecting the engine durability. However in the present case, the peak pressure at full load occurred well after TDC for all the fuels.

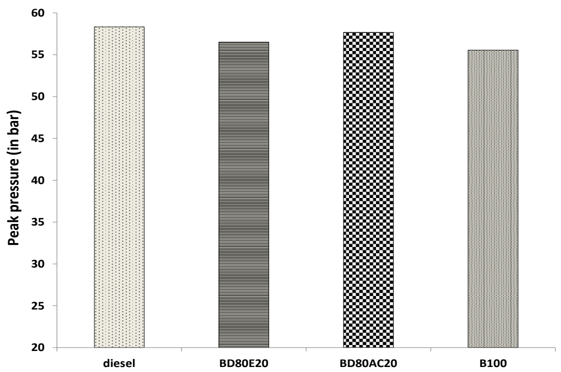

Fig. 3: Peak pressure for the tested fuels

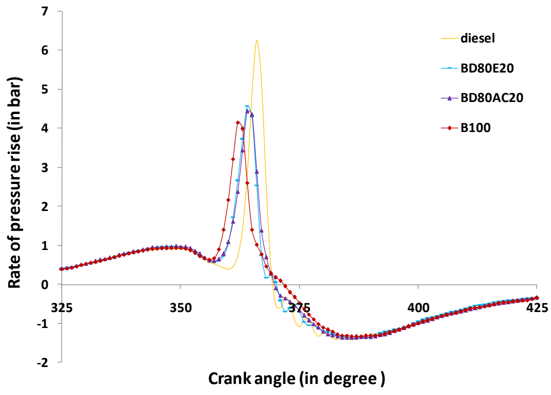

Fig. 4: Rate of pressure rise for the tested fuels.

3.2 Net heat release rate

The heat release rate was calculated by first law analysis of the pressure CA data. The apparent net heat release rate which is the difference between the apparent gross heat release rate and the heat transfer rate to the walls is given by Equation (1) as given below.

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \quad (1)$$

Value of γ for all the fuels were taken as 1.35. Fig. 5 shows the NHRR for the tested fuels at full load. Lower heat release rate was observed in case of B100 and its ethanol and acetone blends at full load. It was seen that the NHRR characteristics of the fuels also followed the same trend of the rate of pressure rise. As can be seen from the figure, the NHRR is low during premixed combustion while the heat release rate is more for the alternate fuel blends during the mixing controlled diffusion phase of combustion. B100 showed the earliest heat release rate followed by the ethanol and acetone blends. The heat release rate during premixed combustion was the minimum for B100 and the same was the maximum for B100 during diffusion combustion. For the blends BD80E20 and BD80AC20, the peak of NHRR and heat release rate during premixed combustion was higher than B100. The NHRR during diffusion combustion was more for BD80AC20 than BD80E20. Early premixed combustion associated with early heat release rate in case of the alternate fuels signify shorter ignition delay period for the blends. Due to heat loss from the cylinder and the cooling effect of the fuel vaporizing as it is injected into the cylinder, the heat release rate is slightly negative during the ignition delay period.

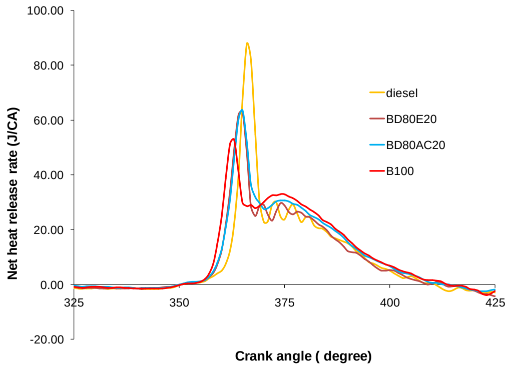

Fig. 5: NHRR vs. CA for the tested fuels at full load

3.3 Cumulative heat release

CHR for the tested fuels is shown in Fig. 6. CHR is the integration of the NHRR results. Further, it indicates the amount of energy spent for a given output. It was observed that CHR increases towards the later part of the combustion process for all the fuel. It was higher for B100 and BD80AC20 compared to diesel and BD80E20. This is due higher heat release during diffusion and late combustion process in case of B100 and BD80AC20 compared to that of diesel and BD80E20. We observed similar CHR pattern for BD80E20 and diesel towards later part of combustion. In fact, the trend of cylinder pressure, rate of pressure rise and NHRR variation with CA was same for these two fuels during the expansion process.

3.4 Determination of start of injection

The CA corresponding to start of injection (SOI) can be determined from Fig. 7 which shows the fuel pressure variation with the crank angle. In Fig. 7, the CA at which the fuel pressure in the fuel line reached its maximum value followed by a sudden drop in pressure was considered as the SOI. The fuel pressure varies during fuel pumping operation with fuels having different physical properties (such as viscosity, density and compressibility). The CA corresponding to SOI for all the fuels being about same, but the corresponding values of maximum pressure are different for various fuels. In case of B100, the peak fuel pressure at which the fuel is injected to the cylinder was more compared to the other fuels. It could possibly be due to higher bulk modulus of the biodiesel. Diesel and BD80E20 showed the same variation of fuel pressure and the pressure at which these fuels were injected was lower than B100. In case of BD80AC20, the fuel injection pressure was lowest.

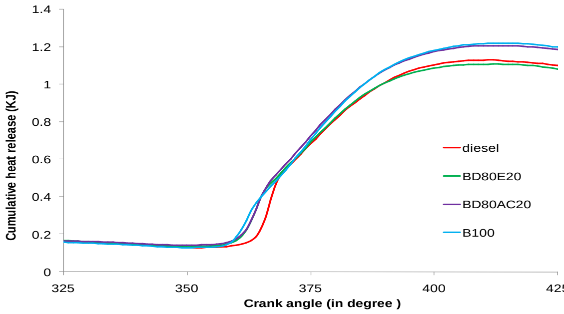

Fig. 6: CHR vs. CA for the tested fuels at full load

3.5 Determination of start of combustion

The determination of start of combustion (SOC) in diesel engine is necessary to evaluate the point of auto ignition, ignition delay and the combustion duration. Actually there exists no exact single point of SOC if one takes into consideration the effect of combined physical and chemical delay period and the interlacing of the exothermic pre-flame reactions by the endothermic fuel droplet evaporation reactions during the delay period.

Criterion based on max ($\frac{d^2p}{d\theta^2}$) presented in [19, 20] is widely used as it predicts SOC with delay and is used in the present study to determine SOC.

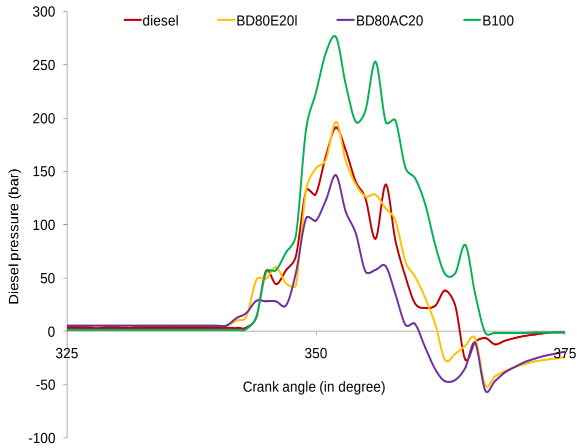

Fig. 7: Fuel line pressure variation with CA for the tested fuels

3.6 Ignition delay

Ignition delay is defined as the time period between SOI and SOC. It is a significant parameter in determining the knocking characteristics of C.I. engines. The cetane number of a fuel, which indicates the self-igniting capability, has a direct impact on ignition delay. The higher the cetane number, the shorter the ignition delay, and vice versa. Fig. 8 shows the ignition delay for pure diesel, BD80E20, BD80AC20 and B100. It was observed that the ignition delay was the maximum (12° CA) for pure diesel. Ignition delay with B100 was the least (7° CA). It was 10° CA in case of BD80E20 and BD80AC20. Biodiesel typically contains unsaturated fatty acid which gets oxidized when exposed to an oxygen environment. May be due to presence of higher oxygen content, biodiesel gets ignited earlier than that of diesel. Addition of ethanol and acetone in B100 however, increased the delay period by small degrees and this was evident from cylinder pressure, rate of pressure rise, NHRR variation. This was that the peak pressure was maximum for diesel (due to higher ignition delay) and it was minimum for B100 (lower ignition delay).

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3.7 Combustion duration

The variation of combustion duration for the tested fuels is shown in Fig. 9. The combustion duration is the minimum (46˚CA) for diesel may be due to lower fuel consumption with diesel. Combustion duration for BD80E20, BD80AC20 and B100 was 50, 51 and 53˚CA respectively. The engine consumed more fuel during B100 operation and hence combustion also continued for a longer period of time. With addition of ethanol and acetone there was slight reduction in fuel consumption and as a result, the combustion duration showed a marginal reduction with these fuels.

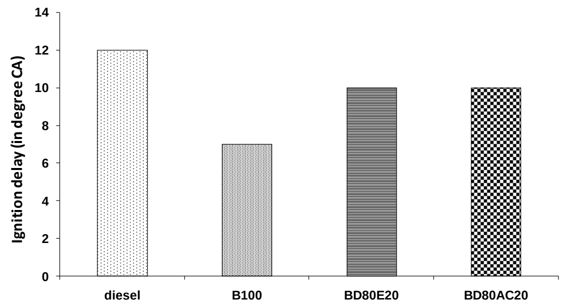

Fig.8: Ignition delay of various fuels at full load

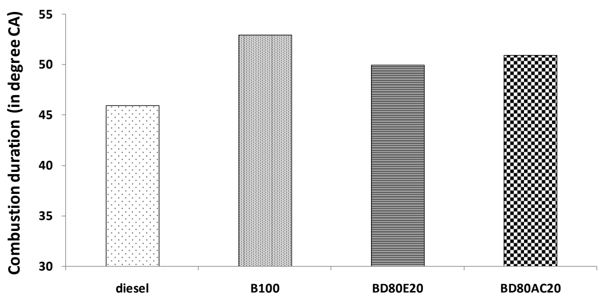

Fig. 9: Combustion duration for the tested fuels at full load

4. CONCLUSIONS

The combustion characteristics of biodiesel obtained from Jatropha curcus oil and its ethanol and acetone blends was evaluated in a single cylinder four stroke DI diesel engine and compared with normal diesel. Early pressure rise was observed in case of B100, BD80E20 and BD80AC20 as compared to pure diesel. The maximum rate of pressure rise was less for the B100 while the same was the maximum for diesel fuel. The rate of pressure rise for the ethanol and the acetone blends was intermediate between B100 and diesel. NHRR characteristics showed direct proportionality with the rate of pressure rise. Early heat release was observed with B100 and its ethanol and acetone blends. It was observed that the rate of heat release during premixed combustion was less for all the alternate fuels while this was more during diffusion combustion for B100 and BD80AC20. The CHR values were higher for B100 and BD80AC20 during the period from SOC till the end of combustion while the same for BD80E20 was higher initially during SOC but the values remained almost the same with diesel towards the later part of combustion. The fuel injection pressure was found to be the maximum for B100 compared to the other fuel samples. This maximum fuel pressure was nearly the same for diesel and BD80E20 and was the minimum for BD80AC20. The problem of poor fuel atomization with lower injection pressure however is not an issue with BD80AC20 because addition of acetone increases the volatility of the blend. Early pressure rise and heat release was an indication of lower ignition delay for the alternate fuel. It was found that the ignition delay for B100, BD80E20 and BD80AC20 was less than that of diesel. Ignition delay for B100 being the lowest of them, because of oxygenated nature of the fuel. More the ignition delay, higher will be the fuel accumulation and resulting peak pressure. This supports the results of maximum peak pressure with diesel fuel and the lowest peak pressure in case of B100. The combustion duration of the alternate fuels was more than that of diesel which may be due to higher fuel consumption in respect of these fuels. Fuel consumption was more for B100 compared to the other fuels. Reduction in fuel consumption was witnessed with addition of acetone and ethanol in B100. It can be recommended that acetone and ethanol can be added with biodiesel to use in an unmodified diesel engine. Blending of biodiesel (in higher proportions) with diesel along with acetone and ethanol could provide a best possible blend for use in diesel engine.

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Dr. Tapan Kumar Gogoi is currently an Associate Professor of Mechanical Engineering at Tezpur University, Assam, India. He has more than 18 year of teaching experience including 5 years experience in research in the field IC engine, alternate fuel technology. Engine performance, combustion and emission analyses with alternate fuels, numerical simulation of engine processes, energy and exergy analyses of advanced hybrid thermal power systems, computational heat transfer and fluid flow, study of fuel spray and its atomization characteristics, spray combustion modeling including modeling of fuel injection system, droplet heating and evaporation are some of his current areas of research.