

Experimental Investigation on Combustion and Performance Characteristics of DI Diesel Engine Using Waste Cooking Oil Biodiesel Blends



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Abstract The hefty prices of raw materials are the greatest hurdle to biodiesel's adoption as an alternate fuel. While waste cooking oil offers a cost-effective solution, it poses challenges in disposal and the potential for contamination. These factors lead waste cooking oil to a promising alternative. Waste cooking oil biodiesel blends have therefore been developed and employed to evaluate their impact on engine combustion and performance parameters. In the present experimental study, a catalytic transesterification process was used to produce biodiesel (waste cooking oil). Biodiesel was blended with diesel in three proportions (W10–10% waste cooking oil blend 90% diesel, W20–20% waste cooking oil blend 80% diesel, and W30–30% waste cooking oil blend 70% diesel). In-cylinder pressure, rate of pressure rise, net heat release, cumulative heat release, mean gas temperature, ignition delay, brake specific fuel consumption (BSFC), and brake thermal efficiency (BTE) were among the combustion and performance parameters that were analysed on a diesel engine at five engine loads (20%, 40%, 60%, 80%, and 100%). The percentage of biodiesel in the blend was increased, which improved the values of combustion and performance parameters. The optimal blend in terms of combustion and performance parameters was found to be W30. In comparison to diesel, W30 had a 13% shorter ignition delay, 22% lower net heat release, 10% slower rate of pressure rise, 2% higher BTE, and 1% lower BSFC. Therefore, biodiesel blends with a high concentration may prove to be a viable substitute for diesel fuel. 034.

Keywords Biodiesel · Waste cooking oil · Diesel engine · Combustion · Performance

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Nomenclature

ATDC	After Top Dead Centre
BSFC	Brake Specific Fuel Consumption
BTE	Brake Thermal Efficiency
CA	Crank Angle
CI	Compression Ignition
CR	Compression Ratio
DI	Direct Injection
EOC	End Of Combustion
ICE	Internal Combustion Engines
NHR	Net Heat Release
TDC	Top Dead Centre
ROPR	Rate of Pressure Rise
SOC	Start Of Combustion
SOI	Start Of Injection
WCO	Waste Cooking Oil

1 Introduction

Since the turn of the twentieth century, the industrial revolution, demographic expansion, population growth, and a multiplied rise in energy demand have all contributed to a sharp increase in the use of Energy worldwide. The development of ICEs and ongoing improvements in engine technology have also increased our dependency on petroleum sources. In every area of the world, even contemporary transportation and its dependence on conventional energy sources like fossil fuels exacerbate the depletion of those resources, as stated by Gupta et al. [1]. This enormous demand for Energy derived from the combustion of fossil fuels has led to the widespread exploitation of fuel supply. The consequence of continuous use of fossil fuels in response to the increase in global energy demand has led scientists and researchers to search for innovative and efficient alternative fuels that are both renewable and non-polluting, with the potential to mitigate global warming and pollution [2, 3]. Several studies indicate that biodiesel is the most effective and viable alternative Fuel for diesel engines, as it is non-toxic, biodegradable, and environmentally friendly. In addition, it is composed of fatty acids and may be generated from vegetable oils, animal fats, and waste cooking oil through the catalytic transesterification process. Moreover, it is devoid of sulphur and minerals, and its high oxygen content results in low engine emissions [4, 5].

Numerous earlier studies have demonstrated that biodiesel gives the same or minimally enhanced thermal efficiency when compared to diesel, as well as lower or equivalent BSFC values for larger loads. Siddarth and Naveen [6] examined the performance and emissions of ternary fuel mixes of diesel, biodiesel, and octanol

and found that the blend has 10% octanol, 10% biodiesel, and 80% diesel had better BTE and BSEC than diesel. In addition, the mixtures produced fewer emissions (CO, unburned hydrocarbon, NO, and smoke opacity) and improved combustion. Furthermore, Devaraj et al. [7] examined the impact of two oxygenated additives, namely DMC (Dimethyl-carbonate) and Pentanol (n-P), on the ignition patterns of biodiesel/diesel blends in a diesel engine. Adding DMC and Pentanol to base gasoline raised peak pressure by 2.3 and 3.1 bar, decreased BSFC by 0.4 and 0.5 g/kW while increasing BTE by 0.3 and 0.6 percent. Moreover, the addition of DMC and Pentanol to NBD50D50 resulted in an increased heat release rate of 3.9 and 2.1 J/CA. Abed et al. [8] investigated the effect of WCO on the performance of a single-cylinder diesel engine and created three mixtures with concentrations of 10, 20, and 30%. The thermal efficiency of waste cooking oil biodiesel blends was found to be lower than that of diesel fuel; however, the BSFC was higher. In addition, as the mixture's concentration increases, thermal efficiency also increases, and BSFC decreases simultaneously. Di et al. [9] studied waste cooking oil biodiesel blends combined with diesel fuel at an engine speed of 1800 rpm for 5 engine loads of 28, 70, 130, 190, and 230 N meters. The usage of biodiesel boosted BSFC and BTE while reducing the temperature of the exhaust gases. Annisa et al. [10] studies performance and emissions of WCO water emulsions, Four loads were applied to a four-cylinder turbocharged Yanmar diesel engine throughout the testing. The experiment results demonstrated that the BTE of BD5W10 at low load was up to 8.92 percent greater than diesel. At medium load, the BSFC of BD5W10 increased by 11.66 percent compared to diesel. BD5W10 has higher CO emissions than diesel and B10. Compared to diesel under heavy load, BDSW10 CO emissions were reduced by 12.8%. Pasupathy and Mohan [11] investigated the effects on performance, combustion, and emissions of a turbocharged engine by changing compression ratios for the blends and comparing them with the standard diesel. Among compression ratios tested (16:1, 17:1, and 18:1), 18:1 yielded the best performance. Increasing the compression ratio without a turbocharger increases NO_x and CO₂ emissions while reducing CO and HC emissions. Without a turbocharger, B10 has lower NO_x and HC emissions than B20 and diesel but somewhat greater CO emissions. The turbocharger reduces emissions overall. With greater compression ratios, the rate of pressure rises, and heat release is accelerated considerably. Kumar and Jaikumar [12] examined the efficiency and combustion characteristics of a single-cylinder WCO-powered diesel engine. According to the findings, the WCO mixture enhanced the in-cylinder pressure and the delay period. Compared to pure diesel, pure WCO demonstrated lower peak pressure under all operating circumstances. As engine power rose, the BTE of every investigated Fuel increased (BMEP). Clean diesel has a lower BTE content than WCO. At a maximum BMEP of 4.4 bar, the maximum BTE was determined to be 26%, whereas it was 30% for clean diesel. Although multiple research studies on performance characteristics and some combustion parameters for biodiesel blends of waste cooking oil have been undertaken in the past, the literature lacks a comprehensive combustion and performance assessment of these blends. Consequently, with a constant CR of 17.5 and an engine speed of 1500 rpm, this article presents the impact of load fluctuation and mix modification on a variety of combustion parameters,

including cylinder pressure, Net Heat Release, Rate of Pressure Rise, Cumulative Heat Release Mean Gas Temperature and Ignition Delay period. Under differing engine loads, variations in these properties have been noted. Finally, performance parameters such as BTE and BSFC are calculated and compared with diesel.

2 Experimental Setup

Figure 1 depicts the block diagram of the experimental arrangement [13]. Table 1 provides the technical characteristics of the engine test bed and the combustion parameters, respectively. In this study, a twin-cylinder, four-stroke, water-cooled, constant-speed DI diesel engine linked to a hydraulic dynamometer was used. The apparatus consists of a diesel engine, an airflow assembly, a pressure transducer, an orifice meter, manometers, a K-type thermocouple, a hydraulic dynamometer, a CA sensor, a piezo-powered unit, and a computer. Using an orifice meter, the rate of air flowing into the engine cylinder through the airbox was determined, and the inclined manometer was used to determine the density of the upstream air. During combustion, a pressure transducer was employed to keep track of the pressure within the cylinder. A temperature sensor was used to gauge the engine's air inflow temperature, and a K-type thermocouple was employed to gauge the cylinder gas temperature. A crank encoder was used to measure the crank angle in degrees, and data was gathered using a data-collecting card. The experimental setup utilized a hydraulic dynamometer to alter the engine load in conjunction with a transparent glass tube fuel meter that was directly linked to the fuel tank, enabling the accurate determination of the mass flow rate of Fuel entering the engine cylinder. All combustion parameters were measured for 1° CA for a mean cycle of 05. Table 2 shows the accuracy of the several types of equipment employed.

2.1 Experimental Methodology

In this experiment, waste cooking oil biodiesel blends were employed as diesel engine fuel. Biodiesel was created in the Combustion and Pollution Control Laboratory (Department of Mechanical Engineering, Zakir Husain College of Engineering and Technology, Aligarh Muslim University) using the conventional catalytic transesterification process. W10 contains 10% WCO and 90% diesel, W20 contains 20% WCO and 80% diesel, and W30 has 30% WCO 30% WCO and 70 percent diesel. The engine was initially run without any load for 10 min before any measurements could be taken so that it could run in a steady state. The following loads were applied: 20%, 40%, 60%, 80%, and 100% at a steady engine speed of 1500 rpm. At a given load on the engine, a pressure sensor in one of the engine cylinders transmitted signals to the computer, enabling the creation of graphs and tables illustrating various combustion

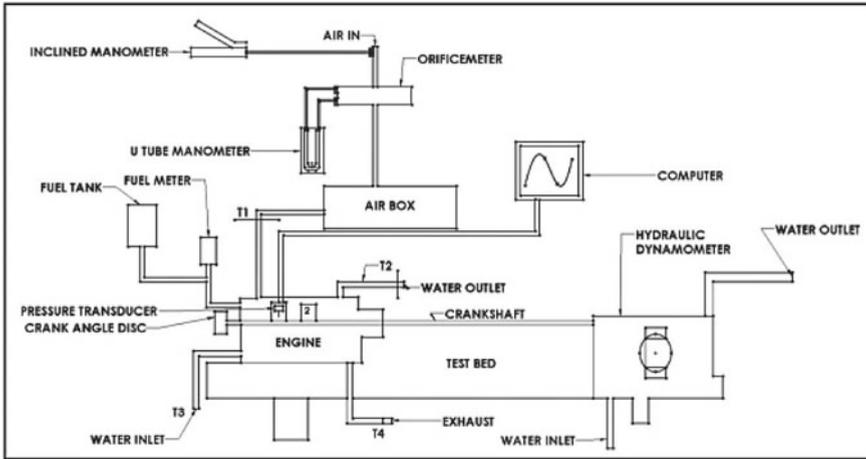


Fig. 1 Block diagram of a DI diesel engine test bed (taken from Ref. [13])

Table 1 Engine test bed details and combustion parameters [13]

Engine type	Kirloskar, TV2, 2-Cylinder, 4-Stroke, water-cooled, Diesel engine
Power	10 kW @ 1500 rpm
Cylinder bore	87.50 mm
Stroke length	110.00 mm
Compression ratio	17.50
Swept volume	661.45 cc
Injection timing	27° bTDC
Orifice diameter	25 mm
Fuel pipe diameter	34.50 mm
Ambient temperature	27 °C
Fuel injection pressure	200 bar
Specific gas constant	1.00 kJ/kg.K
Air density	1.17 kg/m ³
Adiabatic index	1.41
Polytrophic index	1.13
Number of cycles	05

Table 2 Accuracy of instruments used [13]

Instrument	Accuracy
Pressure sensor (Model S111A22)	± 0.1 mV/psi
K-type thermocouple	± 2.2%
Hydraulic dynamometer	± 0.1 kg
CA sensor	1 degree (resolution)

characteristics. The combustion graphs seen in the text's later portion were created using a program called IC Engine Soft.

2.2 Biodiesel Production Process and Fuel Properties

The biodiesel fuels used in this experiment were made via catalytic transesterification of waste cooking oil with ethanol and catalysed with sodium hydroxide (NaOH) as the base. The concentration of the catalyst was determined to be 1% by the weight of the oil. The ratio of alcohol to oil was 1:6 by weight of oil. The temperature of the reaction was 60 °C, and the speed of the stirrer was set at 1000 rpm. To accelerate the process, the samples were heated between 50 °C and 70 °C in an electric heater. The determined amount of ethanol was mixed with sodium hydroxide (NaOH) and stirred until the entire catalyst reacted with the ethanol while the oil was heated. The alcohol–catalyst combination was then added to the oil, which was heated to 60 °C and stirred continuously for one hour. This resulted in the separation of the mixture into two distinct layers: golden liquid ester on top and light brown denser glycerine on the bottom. This immiscible solution was then poured through a transparent funnel to separate glycerine and ester. Finally, the waste cooking oil esters were thoroughly washed many times with room-temperature distilled water until all FFA was removed. Ultimately, this biodiesel was heated for 15 min at 100 °C to remove any leftover water.

Prepared biodiesel characteristics, such as kinematic Viscosity, dynamic Viscosity, and density, were evaluated in the Combustion and Pollution Control Laboratory using a kinematic viscosity bath and hydrometer. The kinematic Viscosity was tested at 40 °C. The apparatus was filled to the appropriate level with 25L of distilled water. Then, a 25 ml sample of Fuel was collected in a clean, dry, and calibrated viscometer. The test fuel was suspended in the viscosity bath for 30 min until it achieved the required temperature of 40 °C. Later, suction was applied to the viscometer's thicker arm, test fuel was made to flow from mark (C) to mark (G), and the flow time was recorded.

Likewise, a hydrometer with a range of 800–850 kg/m³ was selected to calculate the density. A 500 ml sample of test fuel was poured into a beaker, and the hydrometer was immersed in it for two minutes before settling. Later, the density measurements were noted. Table 3 lists the properties of all test fuels.

Table 3 Properties of all test fuels

Parameters/test fuels	Diesel	W10	W20	W30
Density (kg/m ³) ^a	820	830	839	850
Kinematic viscosity (cSt) ^a	3.65	5.37	6.93	9.77
Dynamic viscosity (mPa-s) ^a	2.993	4.45	5.81	8.30
Calorific value (kJ) [14]	42,200	41,800	41,204	40,954
Cetane number [14]	48	49.5	51	52.5
Flashpoint (°C) [14]	74	84.2	94.4	104.6

A measured property

3 Results and Discussion

3.1 Combustion Characteristics

3.1.1 Cylinder Pressure

The premixed burning phase in a diesel engine determines the peak cylinder pressure or pressure generated at the beginning of combustion, which is directly related to the amount of Fuel burned during this phase. Consequently, cylinder pressure indicates how efficiently the air/fuel combination burns in the combustion chamber. Figure 2 shows the variation in cylinder pressure with load for diesel and biodiesel blends of waste cooking oil (W10, W20, and W30). Figure 2a illustrates how the cylinder pressure of a diesel engine rises as the engine load increases from 20 to 100%. Among all the fuels tested, Diesel has a maximum cylinder pressure of 63.6 (bar), which is approximately 12° aTDC (after Top dead centre). Pressure rises with the load as a result of increased fuel flow to the engine [15].

Figure 2b–d illustrates how the cylinder pressure changes with load under the blend comparison and rises as a result of a greater energy release. Diesel has a peak cylinder pressure of 63.6 bar, which is higher than the cylinder pressures of all WCO mixtures (63.13 bar for W10, 62.31 bar for W20, and 62.52 bar for W30). Despite the fact that the W10 blend had pressure values comparable to those of diesel, the other two tested fuels had in-cylinder pressure curves that were identical, suggesting that they had similar combustion processes in the engine, specifically premixed combustion followed by diffusion combustion. WCO blending has improved combustion at all engine loads, with peak pressures for blends approaching 10° aTDC. Furthermore, faster ignition of the blends after injection into the combustion chamber and larger pressure peaks have been achieved by WCO blends because of higher cetane numbers [16]. Ignition delay is the time difference between SOI and SOC, and it comprises physical and chemical delay phases. Physical delay refers to the time it takes for Fuel to evaporate and atomize, while chemical delay specifies the time it takes for Fuel to ignite and a flame to form [17].

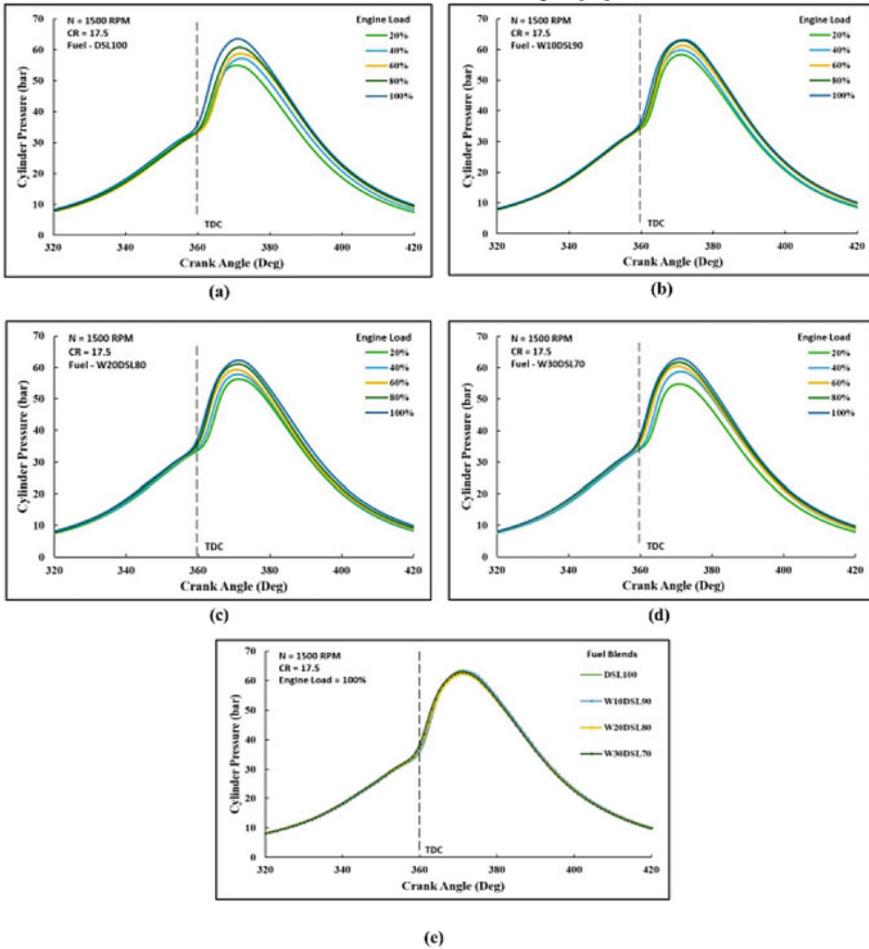


Fig. 2 Variation of cylinder pressure versus crank angle for diesel and waste cooking oil blends

Figure 3 demonstrates that diesel's ignition delay period for increasing load is larger than that of most biodiesel mixes. Diesel fuel has a longer ignition delay period than biodiesel mixes due to its lower cetane number. Since biodiesel is not extracted from crude oil, the high density and Viscosity of biodiesel fuel result in poor atomization, slower mixing, more penetration, and a lowered cone angle. That is why biodiesel has lower volatility and a shorter ignition delay than diesel fuel. This is the result of a complicated, rapid pre-flame chemical process that occurs at high temperatures, which produces lighter compounds and may ignite more rapidly, resulting in a shorter ignition delay.

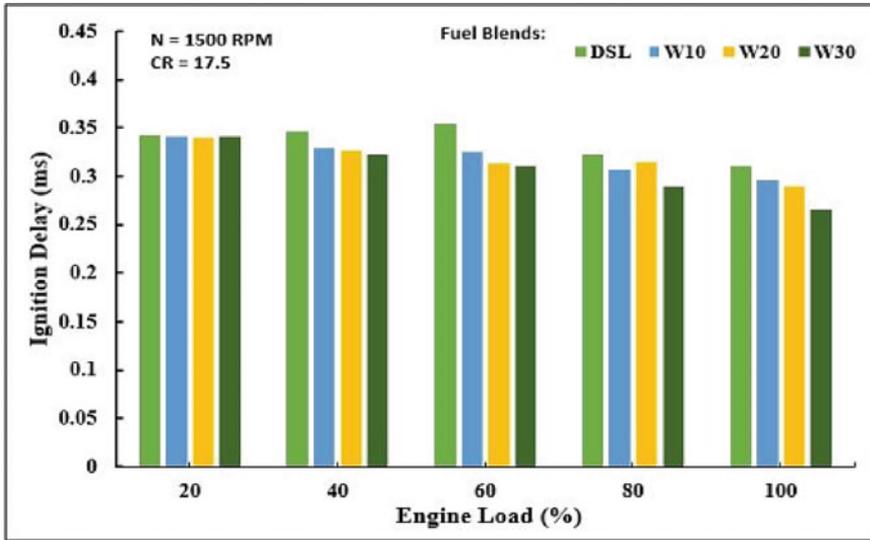


Fig. 3 Variation of ignition delay period for various fuel blends with varying loads

3.1.2 Net Heat Release

NHR is caused by the fast combustion of air–fuel mixture accumulated during the ignition delay interval in the combustion chamber [18]. Figure 4 illustrates the variation in net heat release vs. crank angle for diesel and waste cooking oil biodiesel blends under varied engine loads. For all mixes with an initially rising load, it is likely to see an increase in NHR. A greater NHR value is the result of a greater amount of heat produced by the Energy transformation of the Fuel during combustion. Thus, increasing engine loads result in improved fuel combustion due to the release of more heat energy as cylinder pressures increase. Longer ignition delay periods cause more air–fuel mixture to be collected, leading to the rapid ignition of all the charge present and, thus, a greater NHR at engine loads of 80 to 100 percent. The rate of increase in the NHR is found to remain constant, which is a result of typically constant delay times with even greater loads, as demonstrated [19].

Figure 4b–d illustrate a negative heat release at the start of the compression stage. The negative heat is due to the endothermic chemical and physical reactions that happen during the ignition delay period. In addition, the concentration of Fuel encourages vaporization, which is the phenomenon of heat absorption and subsequent negative heat release. Diesel and biodiesel combinations experience combustion stages that are similar. At the conclusion of the delay phase, the air/fuel mixture ignites rapidly during the premixed combustion period, and heat release becomes positive. Furthermore, the quantity and velocity of air/fuel mixing may be controlled at this stage to manage the burn rate. For a 100% engine load, Fig. 4e illustrates the fluctuation of NHR vs. crank angle for diesel, and WCO blends at 100 percent engine

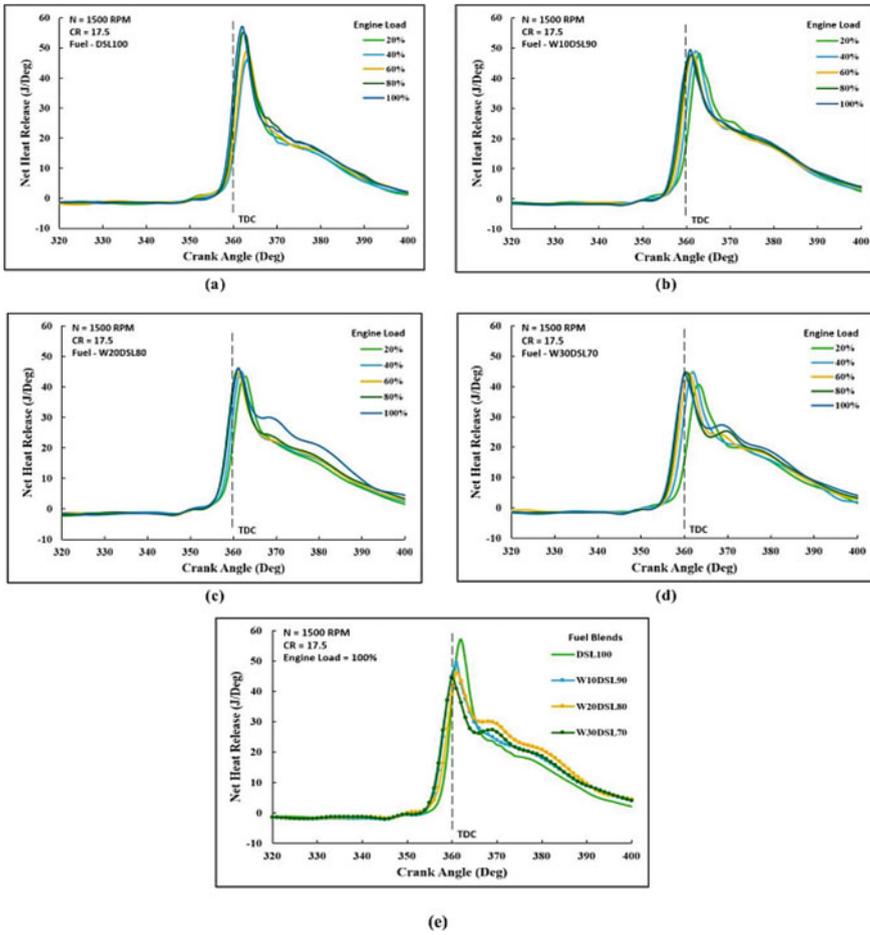


Fig. 4 Variation of net heat release versus crank angle for diesel and waste cooking oil blends

load. Peak values of heat release at full engine load for diesel, W10, W20, and W30 blends are 57.12 J/deg and 362 CA; 49.37 J/deg and 361 CA; 46.12 J/deg and 361 CA; and 44.33 J/deg and 360 CA, respectively. Diesel has a significantly higher peak NHR than its mixes because of its stronger volatility and superior ability to mix with air. Increased accumulation of fuel mixture during the longer delay periods is also a significant cause for the bigger NHR peak, and its most distant position for diesel is 362 CA. These observations were also referenced in [20, 21]. WCO blends provide an appropriate progression of increasing and decreasing NHR. Due to the shortest delay period, as noted in Fig. 3, and the highest cetane number of the W30 blend, the NHR value measured for it is the lowest and is located at TDC, at 360 CA. In addition, the NHR of W20 and W10 mixes increases as the delay duration increases [1, 22]. In terms of heat release and combustion acceleration, biodiesel blends also

beat diesel fuel. This is mostly due to biodiesel's higher oxygen concentration, which allows it to burn more swiftly and completely than diesel fuel and causes a greater proportion of biodiesel to ignite more slowly than diesel fuel [1, 23].

3.1.3 Rate of Pressure Rise

The rate of pressure rise during engine operation indicates the roughness of the combustion, the peak cylinder pressure, the operation's smoothness, and the produced power. ROPR is determined by the rate of combustion of the collected air/fuel mixture in the engine cylinder during the premixed combustion phase. Figure 5 illustrates how the rate of pressure rise changes with the crank angle for diesel and waste cooking oil blends operating under various loads. Diesel and all blend records reveal that ROPR increases non-linearly with load and tends to stabilize between 80 and 100 percent engine load. The combustion of a substantial volume of injected Fuel gathered during the ignition delay interval is indicated by growing ROPR measurements as the load increases. Due to the quick and fast combustion that happens after the first flame kernel, longer delay durations result in a faster rate of pressure rise. Consequently, for diesel and all WCO blends, the charges accumulated during the initially longer delay periods burn, resulting in a high heat release, an increase in cylinder pressure, and subsequently, an increase in ROPR.

Figure 5a–d illustrate the influence of combining waste cooking oil with diesel on ROPR for varying engine loads. When the load is raised for all fuel mixes, the ROPR peak switches to TDC as compared to diesel. This is likewise attributable to the earlier onset of combustion, before TDC, for the mixes in comparison to diesel. Moreover, the higher the cetane number of the mixtures, the quicker the air/fuel mixture burns, which advances the ROPR peak towards TDC [24]. According to Fig. 5e, the ROPR value and position for diesel, W10, W20, and W30 under full load circumstances are 4.75 dP/dθ at 361, 4.63 dP/dθ at 360, 4.21 dP/dθ at 360, and 4.25 dP/dθ at 360, respectively. Full-load diesel ROPR is the highest reported value due to the excellent mixing of air and fuel during the time of premixed combustion. In addition, diesel's premixed combustion NHR is larger, which may account for its quicker ROPR [25]. As the blending fraction of a WCO blend grows, its impact on ROPR and location changes proportionally. Lastly, as the blending % grows, the magnitude of ROPR for the blends decreases, which may be ascribed to the lowered energy content, poor flowability, and decreased volatility of biodiesel in the blends.

3.1.4 Cumulative Heat Release

The entire heat produced from the chemical energy of fuel after combustion is referred to as cumulative heat release. CHR depicts information regarding the engine's combustion progress. Figure 6 shows the variance in CHR versus crank angle for diesel, and WCO blends at various engine loads. CHR rises with increasing engine load, as seen in the graph for all blends, and follows a modest nonlinear pattern.

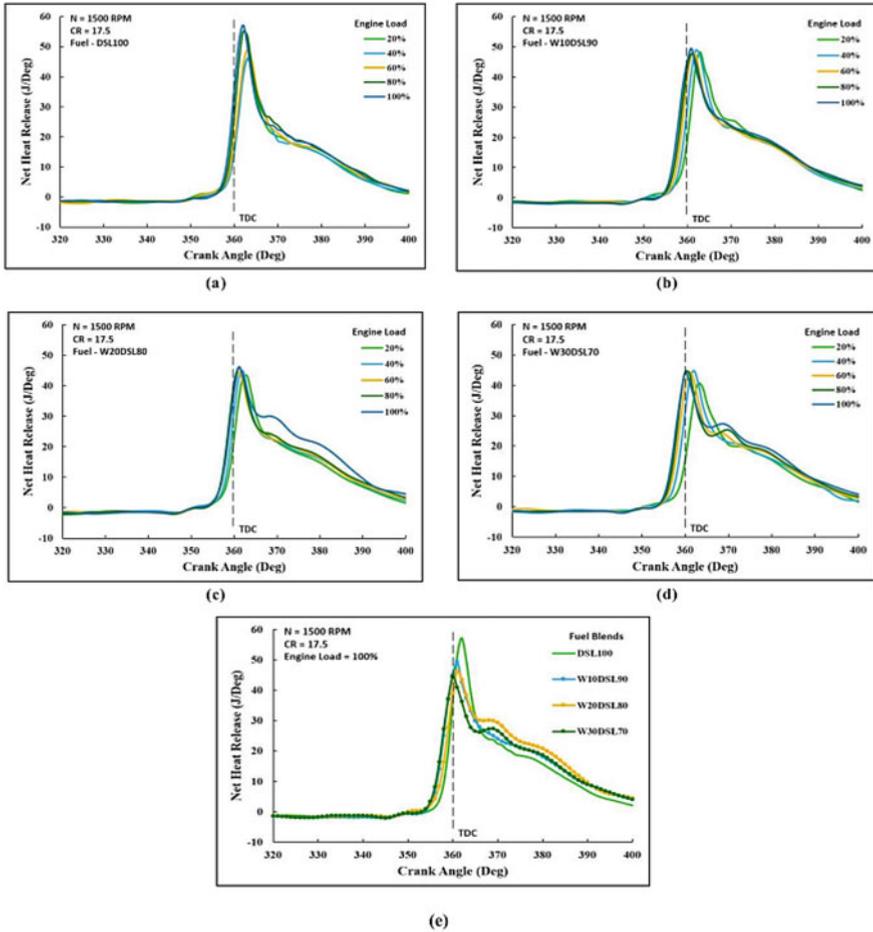


Fig. 5 Variation of the rate of pressure rise versus crank angle for diesel and waste cooking oil blends

Figure 6a–d illustrate the influence of combining waste cooking oil with diesel on CHR for varying engine loads. From Fig. 6e, the maximum value of CHR for diesel, W10, W20, and W30 is 0.95 kJ, 0.99 kJ, 1 kJ, and 0.97 kJ, respectively. When compared to WCO, CHR for diesel fuel is lower during the early stages of combustion. This might be attributed to the fact that diesel takes longer to start igniting than biodiesel mixes owing to the greater delay time. WCO biodiesel also enables early combustion because of its greater cetane number and intrinsic oxygen. Also, even after delayed combustion of diesel compared to WCO blends, the CHR of diesel approaches the W10, W20, and W30 values as the combustion progresses due to the better ignition and mixing property, greater energy content, and altogether a better combustion property of diesel fuel.

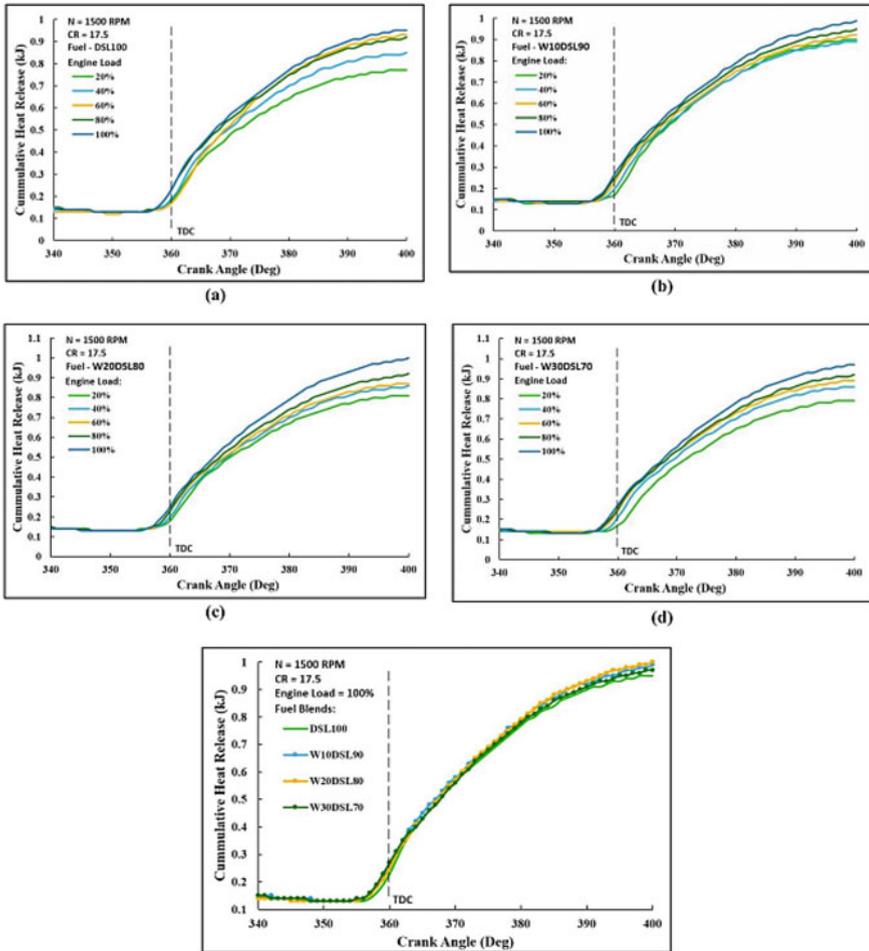


Fig. 6 Variation of cumulative heat release versus crank angle for diesel and waste cooking oil blends(e)

3.1.5 Mean Gas Temperature

The temperature of the gas generated in the engine cylinder following the power stroke is referred to as the mean gas temperature. MGT is a tool for determining combustion efficiency. Figure 7 shows the effect of WCO blends on MGT under various load conditions. The MGT value increases with increased engine load for every operating condition, as shown by the figures for all blends. On increasing the engine load, as the power demand of the engine is increased, more Fuel is injected into the cylinder to compensate, resulting in increased heat release at higher engine loads. This increase in heat release causes the charge temperature during the exhaust

stroke to rise, which in turn increases the MGT for any fuel. Figure 7a–d illustrate the influence of combining waste cooking oil with diesel on MGT for varying engine loads.

From Fig. 7e, variation in MGT for WCO blends under full engine load conditions is presented. During the early stage of combustion, the gas temperature of diesel is recorded to be lowest as compared to WCO blends. It is due to delay in the start combustion for diesel because of low cetane number along with longer delay period. All WCO blends, on the other hand, have demonstrated the fastest increase in gas temperature, owing to advances in combustion due to the lowest delay periods for full load conditions and the highest cetane number. The MGT of diesel approaches the W10, W20, and W30 values in the diffusion combustion phase, owing to improved

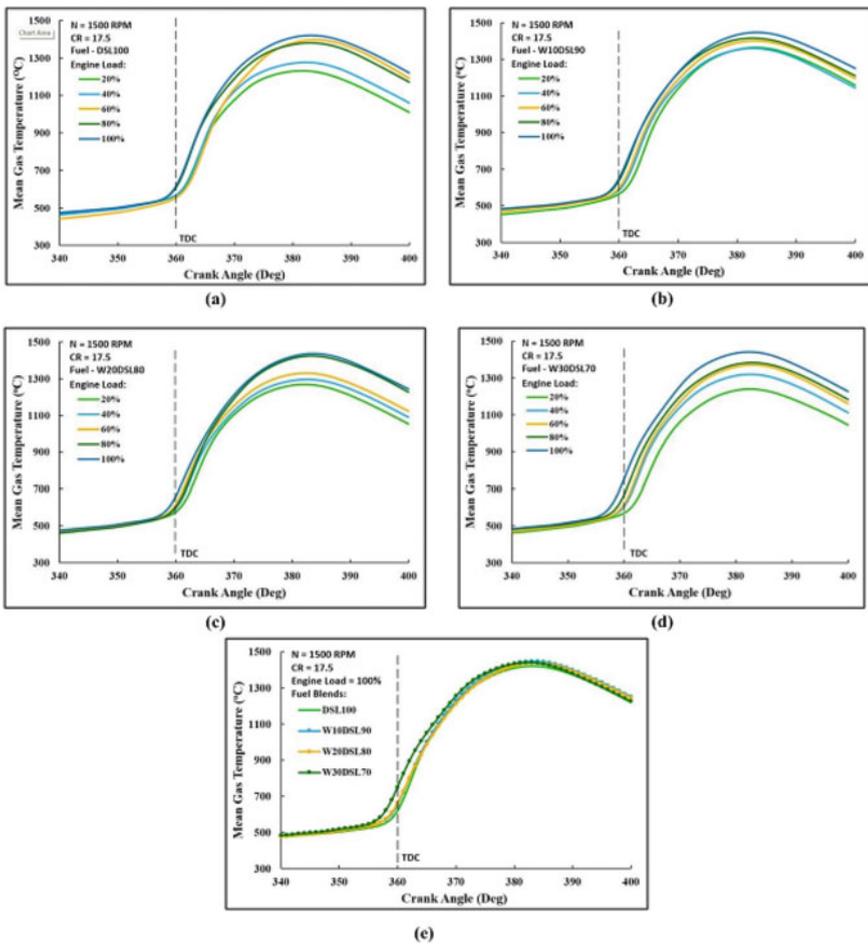


Fig. 7 Variation of mean gas temperature versus crank angle for diesel and waste cooking oil blends

mixing of diesel fuel with the air and higher heat content, all of which contribute to efficient combustion. The MGT values recorded for diesel, W10, W20, and W30 for full engine load conditions are 1421 °C, 1448 °C, 1437 °C, and 1440 °C, respectively.

3.2 Performance Parameters

3.2.1 Brake Thermal Efficiency

Brake thermal efficiency is a statistic that measures the efficiency with which an engine transforms heat from the Fuel into mechanical Energy. It refers to the conversion of fuel energy into usable work that is produced at the crankshaft. Figure 8 illustrates the variance in experimental BTE for diesel and various biodiesel blends at W10, W20, and W30 concentrations under increasing engine loads. The BTE bar graph demonstrates that as loads have grown, the BTE has risen for every blend in every operating condition. At the greatest loads, improved spray characteristics, air/fuel mixing, appropriate cylinder pressure, temperature, heat release, and reduced delay periods have led to successful combustion. The BTE for diesel is 8.82% at 20% load and 25.4% at 100% load. For WCO fuels, the best efficiency is attained at 100% engine load when considering the blend changes. However, compared to pure diesel, the efficiency of WCO blends is somewhat lower at lower loads, except for the W30 blend, which has demonstrated about the same efficiency as diesel at lower loads while surpassing it at higher loads. This improvement in BTE for the W30 blends as engine load increases is primarily due to more combustion reactions taking place as combustion pressure and temperature rise. In contrast, the drop in BTE found for other mixes is a result of incomplete combustion caused by a high air-to-fuel ratio. BTE recorded for an 80% engine load indicates values that are nearly equal for all mixes. Similar results are also studied by [1, 16, 25].

3.2.2 Brake Specific Fuel Consumption

The Brake Specific Fuel Consumption (BSFC) is the quantity of fuel required to produce one unit of power per hour by an internal combustion engine. It is a crucial performance parameter that assists in the evaluation of an engine's efficiency and performance. Figure 9 illustrates the fluctuation in BSFC values for diesel and WCO blends under various loads. The results indicate that when load increases, the BSFC value of all blends decreases dramatically, which may be related to the proportionate increase in brake power with load compared to the increase in fuel consumption [26]. When the load is elevated, the cylinder pressure and temperature rise, resulting in an increase in combustion temperature and improved combustion at peak loads. Consequently, when loads increase, the quantity of Fuel required to deliver a unit of brake power decreases, resulting in a lower BSFC. It has been noted that, for lighter loads, WCO mixes provide a greater fuel flow rate to the engine than diesel, also stated

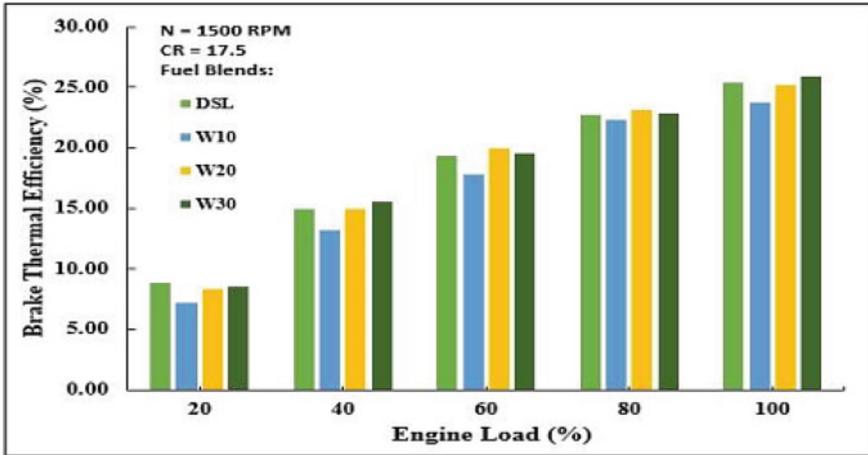


Fig. 8 Brake thermal efficiency for various fuel blends with varying loads

by [27]. When considering 100 percent load, it is noted that the BSFC of all mixes has decreased to the minimum value and is relatively comparable. Nevertheless, across all the blends and engine loads, the W30 blend has demonstrated the closest BSFC value to that of diesel, which is 0.348 kg/kWh (0.337 kg/kWh for diesel). Therefore, higher load values for biodiesels are quite favourable since the BSFC is very close to that of diesel, and there is a rise in diffusion combustion and thermal efficiency [16].

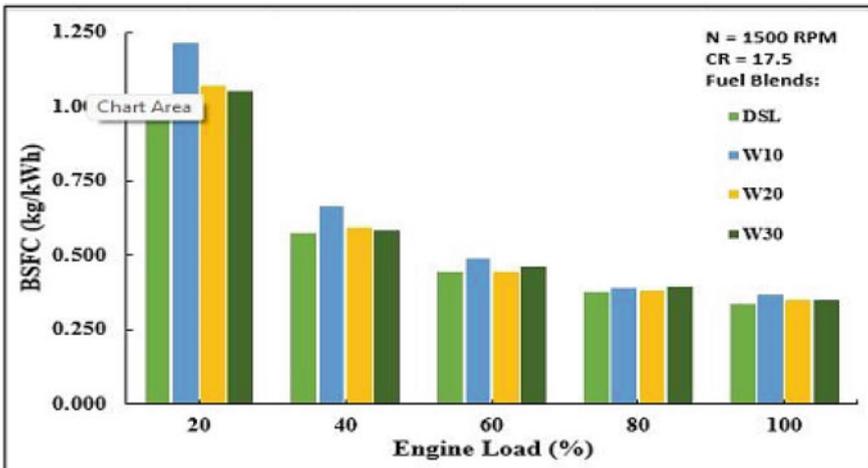


Fig. 9 Brake specific fuel consumption for various fuel blends with varying loads

4 Conclusions

In the current experimental research, different combustion and performance measures were compared between the WCO blends and diesel fuel. At full load, combustion characteristics such as cylinder pressure CHR and MGT were measured to be the greatest for all test fuels. The variation in cylinder pressure for WCO mixes was minor and near the diesel value. In contrast, ROPR and NHR exhibited an initial rise with increasing load up to 60 percent, followed by stagnation at 80 percent and 100 percent load. Also, combustion was seen to advance for higher blending ratio due to their lower cetane number as by the early increase of mean gas temperature in the MGT graph. Performance metrics such as BTE and BSFC were enhanced at full load. At full load conditions, the BSFC is found to be the lowest, whereas the BTE peaked at full load (25%). Ignition delay for all biodiesel blends was lower than that of diesel. In contrast, parameters such as NHR and ROPR decreased in magnitude as the blending fraction of WCO increased in comparison to diesel due to less intense premixed combustion as a result of shorter delay periods. According to the evaluated combustion and performance criteria, W30 proved to be the optimal blend. At maximum engine load, the cylinder pressure for the W30 blend was 62.52 bar compared to 63.6 bar for diesel. At maximum engine load, an NHR of 44.33 J/deg was found, compared to diesel's 57.12 J/deg.

Likewise, the lowest ROPR of 4.25 dP/d θ for W30 as compared to diesel's 4.75 dP/d θ was recorded. In addition, W30 had a 13% reduced delay period than diesel at maximum load. A little improvement in performance characteristics was also recorded for W30, with a 2% rise in BTE and a 1% decrease in BSFC compared to diesel at maximum load.

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