

Experimental Combustion Analysis of a Small Size Diesel Engine Fueled With Tire Derived Fuel/Diesel Fuel Blends

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Abstract

This study aims to analyse the detailed combustion characteristics of tire derived fuel blends (TDF) in small-size DI diesel engines. From this point of view, the refined TDF and reference diesel fuel (No.2 Diesel) were blended in various percentages and tested in a single cylinder naturally aspirated DI diesel engine to clarify its detailed combustion characteristics. The experimental test results point out that ignition delay period is observed to be 4.32 CAD (No.2 Diesel fuel), 5.57 CAD (TDF20), 6.56 CAD (TDF40), 8.48 CAD (TDF60), 11.09 CAD (TDF80), and 14.26 CAD (TDF100) for the low engine speeds (1400 rpm). With the increasing TDF in fuel blends and engine speed, the ignition delay period prolongs more, and the total combustion duration shortens. However, due to the excessively long ignition delay period of TDF100, it is not possible to run the test engine at more than 3000 rpm. TDF blends usually demonstrate a longer premixed combustion period than that of No.2 Diesel, while it exhibits a shorter diffusive combustion period. TDF blends also display a higher maximum value of heat release rate (HRR_{max}), a higher peak value of in-cylinder pressure (P_{max}) and a higher rate of pressure rise (dP/dCAD). In addition, combustion events occur late, and the center of combustion moves away from the top dead center into expansion stroke in the case of TDF blends.

Keywords: Pyrolytic fuel, Diesel fuel, Combustion analysis.

Research Article

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1. Introduction

Scrap tires are one of the very common and important solid wastes all over the world. Scrap tire production shows an increasing trend due to the increasing number of vehicles in developed and underdeveloped countries [1]. However, scrap tires are very resistant to biological degradation. Hence, the scrap tires should be kept under control or disposed of by using suitable methods without damaging the environment. One of the environmentally friendly disposing methods is pyrolysis, which is the thermal degradation of solid wastes at high temperatures (300-900 °C) in the absence of air (oxygen) to produce char, pyrolytic fuel or syngas. The pyrolytic fuel (raw tire-derived fuel-TDF), which is a mixture of various paraffin, olefins and aromatic compounds, is a dark brown liquid. The raw TDF usually has a specific gravity of about 0.90-0.98 kg/l, a sulphur content of about 1-1.5 %, and kinematic viscosity of about 2.5-5.5 centistokes (cSt) which makes it a non-viscous liquid and the calorific value of raw TDF is 41-44 MJ/kg, which is similar to that of diesel and gasoline [2-9].

Producing Tire Derived Fuel (TDF) from scrap tyres not only eliminates the problem of waste disposal but also provides a meaningful potential to minimize the usage of natural sources to obtain useful energy. Tire-derived fuel (TDF) has received increasing attention recently as an alternative fuel for diesel engines due to growing energy demand and finite petroleum fuel sources. Some studies show that TDF can be used as an alternative fuel in CI engines successively [10-18].

Kennedy and Rathinaraj [10] studied the utilization of TDF blends with diesel fuel in a DI diesel engine, and they showed that TDF could be used successively in diesel engines. In the same study, it was reported that the TDF was associated with lower emissions of HC, CO compared to levels associated with diesel fuel. In addition, smoke and noise level was not notably different from neat diesel fuel. Doğan et al. [16, 18] also reported similar results.

Murugan et al. [11-13] investigated the effect of lower and higher concentrations of the TDF/diesel fuel blends on the performance, emission and combustion characteristics of a single

cylinder, 4-stroke, and air-cooled DI diesel engine. They reported that HC, NO_x, CO and smoke emissions usually increased with the increasing TDF content in the test fuels. They also found that the maximum combustion pressure, rate of pressure rise, and ignition delay showed an increasing trend in the case of TDF blends. Murugan et al. [14] studied the influence of the distillation process on raw pyrolytic fuel properties, engine performance and emissions in a DI diesel engine. They reported that the engine performance and emissions can be improved by using distilled TDF in a diesel engine. Recently, İlkılıç and Aydın [17] performed a study to evaluate the influence of TDF on the performance and emission characteristics of a single-cylinder DI diesel engine. They concluded that HC, CO, SO₂ and smoke emissions usually increased, and NO_x emissions decreased with the increasing TDF content in the fuel blends. In addition, engine output power, torque and brake-specific fuel consumption deteriorated with the increasing TDF content in the test fuels.

Despite the studies on the performance and emissions of TDF/diesel fuels in DI diesel engines in the open literature, there is a lack of information on detailed combustion characteristics of TDF/diesel fuel blends in DI diesel engines. The primary objective of this study is to provide a complete understanding and comprehensive combustion analysis of TDF blends in a diesel engine. The secondary objective of this study is to compare the combustion characteristics of TDF blends with those of the No. 2 Diesel fuel at various engine speeds.

2. Experimental Section

2.1 The refining and desulfurization process of raw pyrolytic oil

The scrap tires were collected randomly (car tires, bus tires, truck tires, bike tires, etc.) from the scrap tire sector in Turkey and prepared for the pyrolysis process. The raw TDF was obtained by vacuum pyrolysis at EN-TEK Tire Recycling Company (Turkey). The raw pyrolytic oil (raw TDF) contains impurities like carbon particles, sulphur, and moisture. The raw TDF was subjected to the refining and desulfurization process to remove the impurities. Refining and desulfurization process of the raw TDF contains five stages, 1- Hydro-sulphuric acid (H₂SO₄) treatment, 2- Activated Bentonite-Calcium Oxide (CaO) treatment, 3-Vacuum distillation, 4-Oxidative desulfurization, 5- Washing and drying. All these stages improve the chemical and physical properties of the fuel. The first three stages are known as the hydro-sulphuric acid (H₂SO₄) treatment (acid-clay method), which reduces the impurities and sulphur level of the waste oils. The last two stages are known as oxidative desulfurization for liquid fuels. Details of the refining and desulfurization process can be found in Ref [18]. The final product oil of the refining process was called TDF in this study. The raw TDF and refined TDF properties were presented in Table 1. As shown in Table 1., the refining and desulfurization process improves the fuel properties significantly.

2.2 Fuel characterization

Reference diesel fuel (No.2 Diesel, Ultra Low-Sulphur), which was purchased from a commercial supplier, and the final pyrolytic oil (TDF) properties used in this study were presented in Table 1.

The comparison of fuel properties shows that the TDF has relatively closer fuel properties to that of No.2 Diesel. Hence, no fuel system modifications are required for the utilization of TDF in the existing engine. It was observed that TDF blended easily with reference diesel fuel, and no homogeneity problems were seen with the fuel blends.

Table 1. Test fuel properties.

	EN-590	No.2 Diesel	Raw TDF	TDF	
Density (kg/m ³ , 15 °C)	820–845	829	944.4	904	
Flash point (°C)	55>	67	42.5	60.5	
Kinematic viscosity (cst, 40 °C)	2–4.5	2.96	5.06	2.164	
LHV (MJ/kg)	–	42.9	39.98	40.94	
Cetane number	51	>50	-	<40	
Carbon residue (% m/m)	0.3	0.3	21.51	1	
Water content (mg/kg)	<200	200	2645	140	
Ash (% m/m)	0.01	0.01	0.006	0.0005	
Sulphur (mg/kg)	0–10	≈0	11385	4327	
PAHs (% m/m)	0–11	8	>12	9.5	
Hydrocarbon types (% v/v)	Aromatics	–	20–30	–	69.7
	Olefins	–	–	–	15.3
Distillation	Initial boiling point (°C)	–	144	69.6	78.1
	at 250 °C (% v/v)	<65	36.5	39.5	56.2
	95% (°C)	<360	348.1	324.4	300.1

Variations in fuel properties such as viscosity, density, volatility (distillation characteristics and flash point), cetane number, sulphur content, aromatic content and hydrocarbon composition have all effects on combustion characteristics. Some of these properties are dependent on each other and it is difficult to completely isolate the effect of each one [19].

Fuel viscosity affects the spray characteristics of injected fuel and mixture formation rate. The fuel density has a direct effect on the time for pressure wave propagation and a consequent effect on the dynamic start of fuel injection, which makes the fuel being injected into a lower or higher temperature environment in the combustion chamber. Fuel density also affects the amount of injected fuel mass and peak cylinder pressure. As an example, for the same value of LHV, a higher density fuel provides higher heat input to the combustion chamber and results in higher peak cylinder pressures [19-22]. Flashpoint and distillation characteristics of fuel can be considered as an indicator of volatile and flammable compounds in the fuels. The lower the distillation range and flash point, the higher the fuel volatility. In addition, the higher olefin content of fuel decreases the temperature of the distillation range, makes the fuel more evaporative, leads to

greater heat absorption from the combustion chamber, and results in a delay in the combustion process [23]. The hydrocarbon composition of fuels such as aromatic and olefin content affects the cetane number. The general trend is that the olefins and aromatics have low cetane numbers [19, 24-27].

TDF has a higher density but less energy content than that of No.2 Diesel. A shorter fuel injection delay and advanced fuel injection timing are expected due to the higher density of TDF when compared with those of the No.2 Diesel operation. The energy content of TDF is 95.2% of that of No.2 Diesel, so the engine does not need to be modified when running with TDF blends. The cetane number of TDF is much lower than that of No.2 Diesel, and therefore a longer ignition delay period and a higher rate of pressure rise are expected. The hydrocarbon composition of test fuels shows that TDF is more olefinic and aromatic than that of No.2 Diesel. The distillation range and viscosity of TDF are lower than that of No.2 Diesel, therefore the evaporation process and the fuel-air mixing process are expected to occur faster in the case of TDF blends.

2.3 Experimental apparatus for combustion analysis

The experimental test setup includes a diesel engine coupled to the DC dynamometer and an engine pressure measurement system, which consists of an Oprand AutoPSI-S model fibre-optic based direct sensor (in-cylinder pressure), an Oprand AutoPSI-S2000 sensor (fuel line pressure), and an optical shaft encoder. Oprand pressure sensors are integrated with the signal conditioner in production. Pressure sensors and the shaft encoder were coupled to the diesel engine for monitoring the in-cylinder pressure, fuel line pressure and crank angle data. The signals of pressure sensors and the shaft encoder were recorded by using TFX combustion measurement system, which consists of a sensor interface box, data-acquisition apparatus, PC and TFX software (Fig. 1).

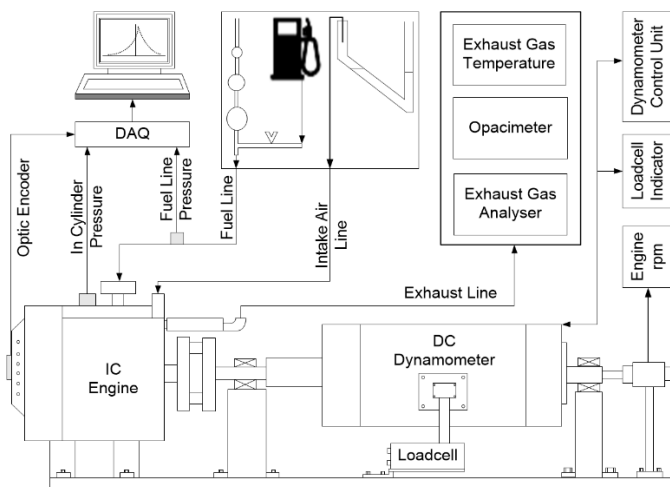


Figure 1. Schematic view of test setup

The engine tests were performed in a single cylinder, four-stroke, air-cooled, naturally aspirated, and high-speed DI diesel

engine under full load conditions and at various engine speeds (1400 rpm, 2000 rpm, 2600 rpm, 3200 rpm). The detailed technical specification of the test engine is presented in Table 2.

The test engine remained unmodified when operating on TDF blends. Engine tests were carried out initially by using reference diesel fuel (No.2 Diesel, Ultra Low-Sulphur). Later on, TDF blends (TDF20 (e.g. TDF20 contains 20% TDF and 80% No.2 Diesel on volume basis), TDF40, TDF60, TDF80, and TDF100) were prepared and tested under same conditions for comparison.

Engine speed, in-cylinder pressure, fuel line pressure and shaft encoder data were recorded by using TFX software during the engine tests. The detailed combustion characteristics and heat release parameters were analysed by using Matlab software after engine tests. To clarify the effect of the TDF blend on combustion characteristics, the combustion characteristics of each fuel blend were compared with those of No.2 Diesel combustion.

Table 2. Test engine details.

Model	Katana KM 178 F
General	Single cylinder, DI, naturally aspirated, and four-stroke.
Diameter (mm) x Stroke (mm)	78x62
Cylinder volume (cm ³)	296
Compression ratio	18/1
Engine power (kW) @ rpm	4 @ 3600
Fuel pump delivery angle (CAD BTDC)	31
Injector opening pressure (bar)	205 ±5
Number of injector nozzle holes	4
Injector nozzle hole diameter (mm)	0.1

2.4 Calculation method

Initially, the recorded cylinder pressure data were averaged by using 100 cycles to reduce the effect of cycle-by-cycle variations, and the final cylinder pressure trace was estimated. Later on, the final cylinder pressure data was filtered by using a forward and backward 4th order butter worth low-pass filter. In the present study, heat release rate (HRR) analysis was performed to evaluate the combustion characteristics by using the filtered cylinder pressure and the cylinder volume data. The HRR analysis is carried out in two parts with the first being the calculation of net HRR (Eq.(1)). The second part is cumulative of the net heat release rate (CHRR) (Eq.(2)) to estimate the end of combustion and center of combustion (CAD50).

The net HRR ($dQ_n/dCAD$) can be calculated by applying the first thermodynamic law as follows [28]:

$$\frac{dQ_n}{dCAD} = \frac{k}{k-1} P \frac{dV}{dCAD} + \frac{1}{k-1} V \frac{dP}{dCAD} \quad (1)$$

where k is the ratio of specific heats, c_p/c_v , for diesel heat release analysis k is 1.35 [28], P is the cylinder pressure, V is the instantaneous cylinder volume, CAD is the crank angle, and Q_n is the net HRR. CHRR (Eq.(2)) is the integration of the net HRR, and it indicates the amount of energy spent for a given output [29].

$$CHRR = \int \left(\frac{k}{k-1} P \frac{dV}{dCAD} + \frac{1}{k-1} V \frac{dP}{dCAD} \right) \quad (2)$$

Ignition delay is the time delay between the start of dynamic fuel injection (SOI) and the start of combustion (SOC) in a diesel engine. Ignition delay period is estimated by analyzing the net HRR diagram and the fuel line pressure trace together. The fuel line pressure history is used to determine the SOI in terms of CAD by considering the point at which the designed nozzle opening pressure is attained.

The SOC is predicted by using the net HRR curve, and it is defined as the CAD at which the net HRR curve crosses the x-axis (from negative to positive value) around the top dead center.

The time between the start of combustion and the end of combustion is specified as the total combustion duration. The end of combustion is estimated from the 90% value of the CHRR. In this study, the premixed and diffusion combustion periods are distinguished by using the beginning of net HRR and the second derivative of the net HRR. The center of combustion (CAD50) is defined as the degree of the crankshaft, and it is estimated from the 50% value of the CHRR.

3. Results and Discussion

3.1 Ignition delay period

The ignition delay period versus TDF proportion of test fuels at various engine speeds is shown in Fig. 2. Increasing TDF proportion in test fuel blends increased the ignition delay period at all engine speeds. Ignition delay period was found to be 4.32 CAD (No.2 Diesel fuel), 5.57 CAD (TDF20), 6.56 CAD (TDF40), 8.48 CAD (TDF60), 11.09 CAD (TDF80), and 14.26 CAD (TDF100) for the low engine speeds (1400 rpm). For the high engine speed (3200 rpm) tests, ignition delay period was observed as 12.83 CAD (No.2 Diesel fuel), 13.92 CAD (TDF20), 14.72 CAD (TDF40), 16.77 CAD (TDF60), and 20.00 CAD (TDF80). However, due to the excessively long ignition delay period of TDF100, it was not possible to run the test engine at more than 3000 rpm.

It is noticed that the ignition delay of TDF and its blends are considerably longer compared to that of No.2 Diesel fuel. As the proportion of TDF in the fuel blend increases, test fuel becomes more volatile and less viscous, which contributes to better fuel atomization and mixture formation, resulting in a short ignition delay. However, the TDF fuel blend causes a higher temperature drop and greater heat absorption in the combustion chamber due to its lower distillation range, better volatility and higher density. This phenomenon overcomes the above-mentioned effect and may lead to a larger physical ignition delay [20, 22, 23, 30]. Increased ignition delay owing to a higher density of TDF blends can be accounted for by the drop in temperature during the delay period due to increasing evaporation as more fuel is injected [31]. Increased

density of TDF blend also reduces the time for pressure wave propagation in the engine fuel line and results in advance in fuel injection timing. The advancing of fuel injection timing makes the fuel to be injected into a lower temperature environment in the combustion chamber and increases the ignition delay [20-22].

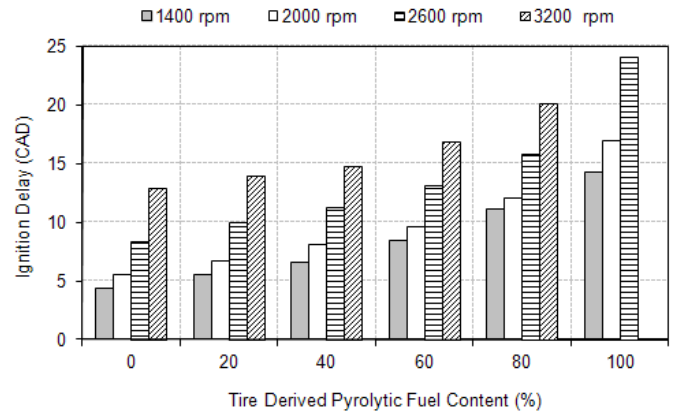


Figure 2. Comparison of ignition delay period

Moreover, the olefin content increase and the cetane number decreases as the proportion of TDF in the fuel blend increases. Increased olefin content reduces the physical ignition delay and increases the chemical ignition delay due to olefin chemistry explained in detail in Ref [23]. The reduced cetane number of TDF blends also prolongs the chemical ignition delay and the total effect of the above-mentioned factors leads to the TDF blend having a longer ignition delay.

3.2 Heat release rate (HRR) analysis

The HRR is the indicator of the heat amount added into the combustion chamber to produce the observed pressure traces in the engine cylinder and HRR analysis is a method used to obtain important combustion information from the cylinder pressure data. It strongly affects the fuel economy, power output and the emissions of the engine [29].

The comparison of heat release rate for TDF blends and for No.2 Diesel at various engine speeds is shown in Fig. 3.

It can be seen that a more negative dip of net HRR is observed at the beginning of HRR during the ignition delay period in the case of TDF blends and this is more obvious at lower engine speeds (Fig. 3). This is because of the heat absorption during the ignition delay period arising from the lower distillation range and higher volatility of TDF blends [32]. The negative dip of net HRR becomes positive after combustion is initiated. However, a delay is observed in the heat release starting position in the case of TDF blends and the end of heat release remains almost the same. The delay in heat release starting position is due to higher olefin content, lower distillation range and better volatility of the TDF blend. A higher olefin content of fuel reduces the temperature of the distillation range and leads to great heat absorption in the combustion chamber and slows down the chemical reaction rate [23]. A higher density of TDF blends provides

more fuel into the combustion chamber and it also leads to more heat energy absorption in the combustion chamber as more fuel is injected [20, 31]. The total effect of the above-mentioned factors leads to the delayed net HRR in the case of TDF blends.

The maximum net heat release rate (HRR_{MAX}) and the location of net HRR_{MAX} versus TDF content of test fuels at various engine speeds are shown in Fig. 4.

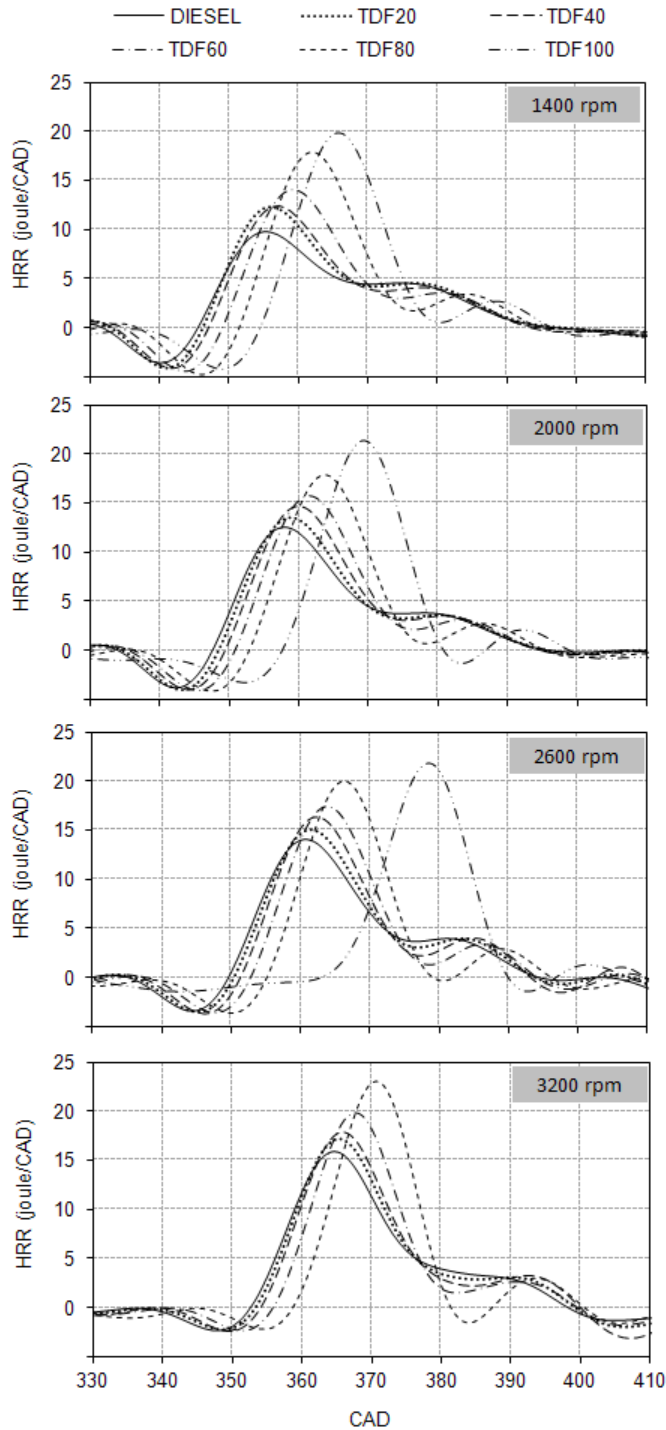


Figure 3. Comparison of net heat release rate

Through Fig.4, it can be seen that the maximum value of net HRR shows an increasing trend in the case of TDF blends for all engine speeds. For the low engine speeds (1400 rpm), maximum value of net HRR showed an increase from 9.79 Joule/CAD (No.2 Diesel fuel) to 12.3 Joule/CAD (TDF20), 12.37 Joule/CAD (TDF40), 14.01 Joule/CAD (TDF60), 17.93 Joule/CAD (TDF80), and 19.83 Joule/CAD (TDF100). For the high engine speed (3200 rpm) tests, net HRR_{MAX} was noticed as 15.95 Joule/CAD (No.2 Diesel fuel), 17.16 Joule/CAD (TDF20), 17.84 Joule/CAD (TDF40), 19.83 Joule/CAD (TDF60), and 23.05 Joule/CAD (TDF80).

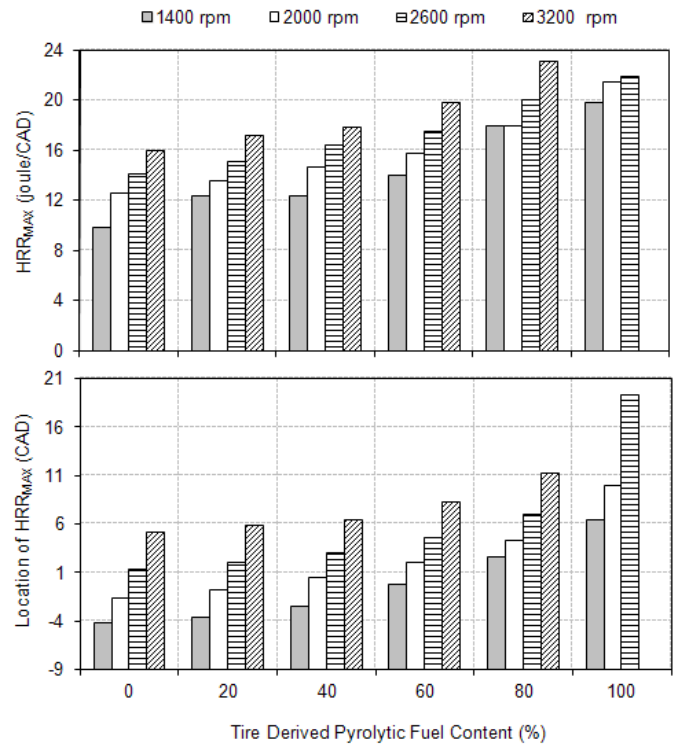


Figure 4. Comparison of HRR_{MAX} and the location of HRR_{MAX}

The greater density of TDF blends provides more fuel into the combustion chamber, hence greater amount of heat is released. The longer ignition delay, better volatility, and lower viscosity of TDF blends lead to a larger amount of fuel accumulation in the combustion chamber at the time of the premixed burning period, leading to a higher net HRR_{MAX} [20, 32, 33]. However, the net HRR_{MAX} of TDF blends occurs at later CAD. The higher temperature drop effect and higher olefin content of TDF fuel blends make the chemical reaction rate slow down at the early stage of net HRR and this can shift the location of net HRR_{MAX} into expansion stroke.

3.3 Duration of combustion stages

Duration of combustion stages consists of premixed, diffusion and total combustion duration. After the ignition delay period, premixed combustion occurs rapidly, followed by diffusion

combustion duration, where the combustion rate is controlled by the fuel/air mixing rate. The premixed combustion duration, the diffusion combustion duration, the total combustion duration, and the center of combustion (CAD50) are introduced in Fig. 5.

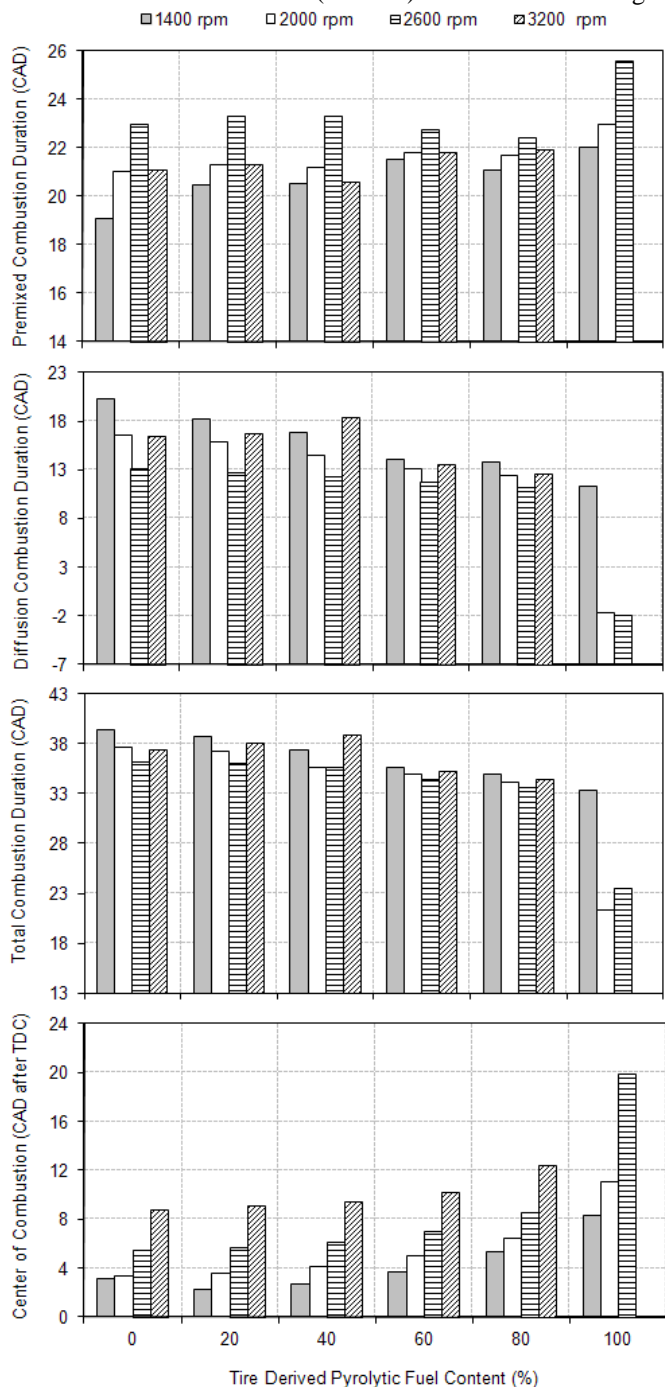


Figure 5. Duration of combustion stages and CAD50.

For the higher engine speeds, the increase in the premixed combustion duration of TDF blends is not significant. However, the premixed combustion duration of TDF blends increases significantly at lower engine speeds due to greater heat absorption at lower engine speeds (see Fig. 3). The longer ignition delay,

higher olefin content and faster evaporation of TDF lead to the larger amount of fuel accumulation in the combustion chamber during premixed part of combustion [20, 34], therefore premixed combustion duration usually increases in case of TDF blends.

The diffusive combustion duration shows a remarkable reduction in the case of TDF blends (Fig. 5) due to a better mixing rate arising from the lower viscosity and better volatility of TDF blends [22, 35-37]. It is well known that a shorter diffusive combustion duration is favorable to the reduction of smoke emission.

It is noticed that the total combustion duration is shorter in the case of TDF blends (Fig. 5). The addition of TDF in the fuel promotes the combustion process, improves diffusive combustion, and reduces the total combustion duration. The center of combustion (CAD50) moves away from the top dead center for TDF blends. This fact can be attributed to a late start of combustion and a short diffusion combustion duration observed in the case of the TDF blend.

3.4 In-cylinder pressure analysis

The in-cylinder pressure in a CI engine is mainly dependent on the fuel-burning rate during the premixed burning phase, which in turn affects the combustion and heat release. The in-cylinder pressure of the engine fueled with TDF/diesel fuel blends versus the crank angle at various engine speeds is indicated in Fig. 6.

Development of cylinder pressure is slightly retarded in the case of TDF blend, due to a longer ignition delay and tardy premixed combustion owing to its lower cetane number. Combustion pressure traces of TDF100 cannot be formed effectively over 2600 rpm and it is not able to run the engine over 3100 rpm due to excessive ignition delay and very late combustion in case of the TDF100 as the cylinder volume is expanding. The cylinder volume expansion prevents pressure rise and this does not permit to the combustion development in the combustion chamber for TDF100.

It is well known that maximum in-cylinder pressure (P_{max}) depends on the combustion rate at the initial stages, which is influenced by the amount of fuel taking part in the premixed combustion phase, which in turn is governed by the ignition delay period. The larger the ignition delay, the more will be fuel accumulation, which finally results in a higher P_{max} . In addition, P_{max} is an indicator of the amount of fuel accumulated during the ignition delay period that takes part in the premixed combustion period [28, 29, 36, 38].

P_{max} and the location of P_{max} versus TDF/diesel fuel blends at various engine speeds can be seen in Fig. 7. For the low engine speeds (1400 rpm), P_{max} tends from 70.1 bar (No.2 Diesel fuel) to 73,8 bar (TDF20), 72.98 bar (TDF40), 73.07 bar (TDF60), 79.36 bar (TDF80), and 72.36 bar, which occurs at 3.92 CAD after TDC, 4.62 CAD after TDC, 1.61 CAD after TDC, 3.62 CAD after TDC, 3.84 CAD after TDC, and 8.06 CAD after TDC respectively. The P_{max} for No.2 Diesel fuel at high engine speed (3200 rpm) is found to be 74.91 bar, which occurs at 7.89 CAD

after TDC, whereas for TDF20, TDF40, TDF60, and TDF80 fuel the P_{max} values are about 76.62 bar, 77.49 bar, 76.25 bar, 72.57 bar, which occurs at 7.89 CAD after TDC, 8.34 CAD after TDC, 8.83 CAD after TDC, 10.88 CAD after TDC, and 13.94 CAD after TDC respectively.

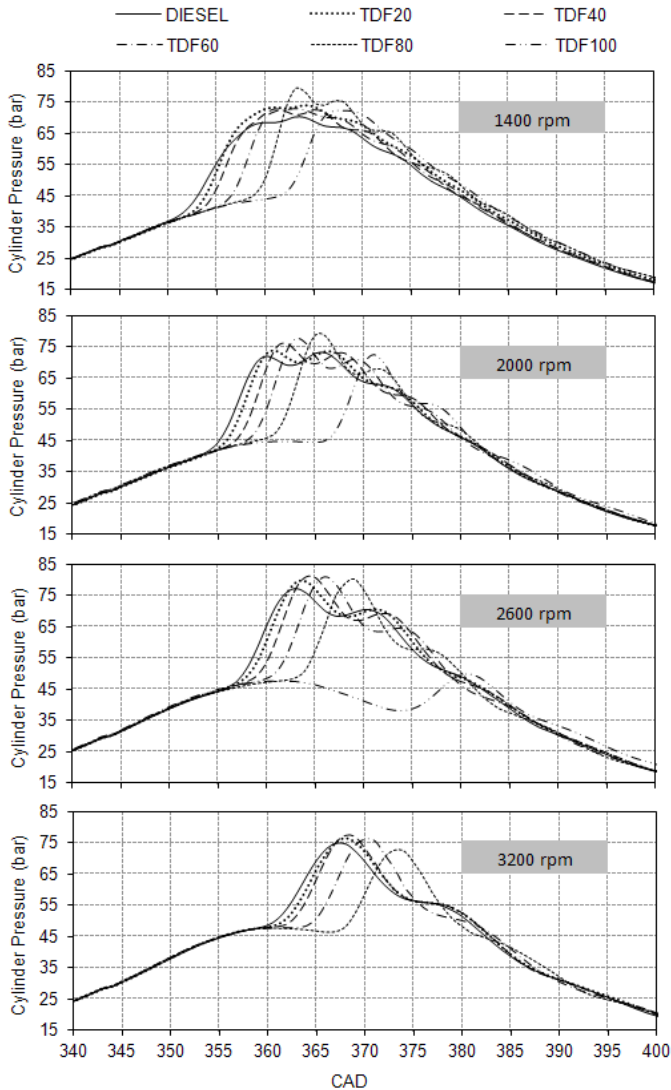


Figure 6. Comparison of in-cylinder pressure histories.

In case of TDF blends, the P_{max} is slightly higher and occurs at a later crank angle. The lower cetane number, higher olefin content and faster evaporation of TDF increase the fuel accumulation during the premixed part of combustion, leading to the higher heat release in premixed combustion duration. The higher heat release during the premixed combustion and longer ignition delay are the main reasons for the higher P_{max} in the case of TDF blends. However, TDF100 test fuel leads to the lowered P_{max} over 2600 rpm. This fact can be attributed to a very late start of combustion and excessive ignition delay in the case of TDF100, hence P_{max} attains a low value and occurs at a later crank angle degrees as the cylinder volume is expanding (Fig. 7).

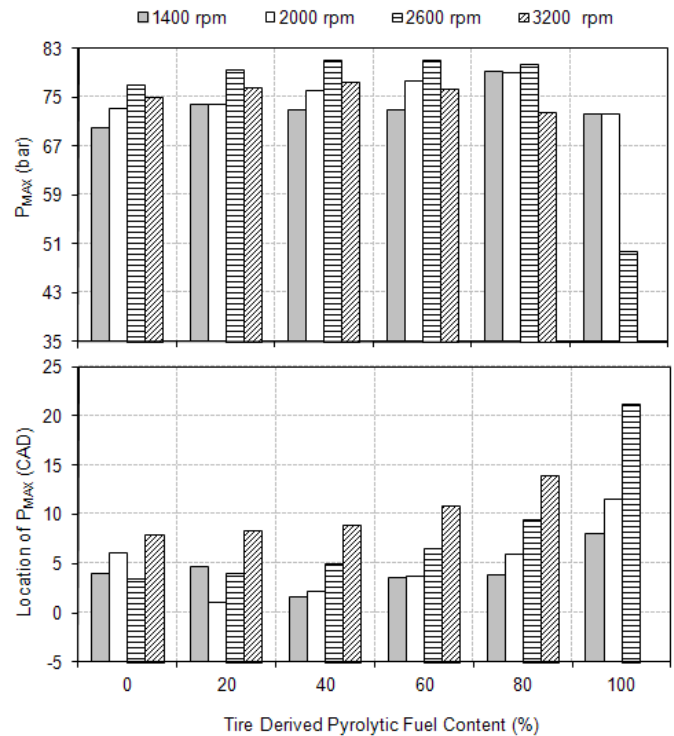


Figure 7. Comparison of P_{max} and location of P_{max}

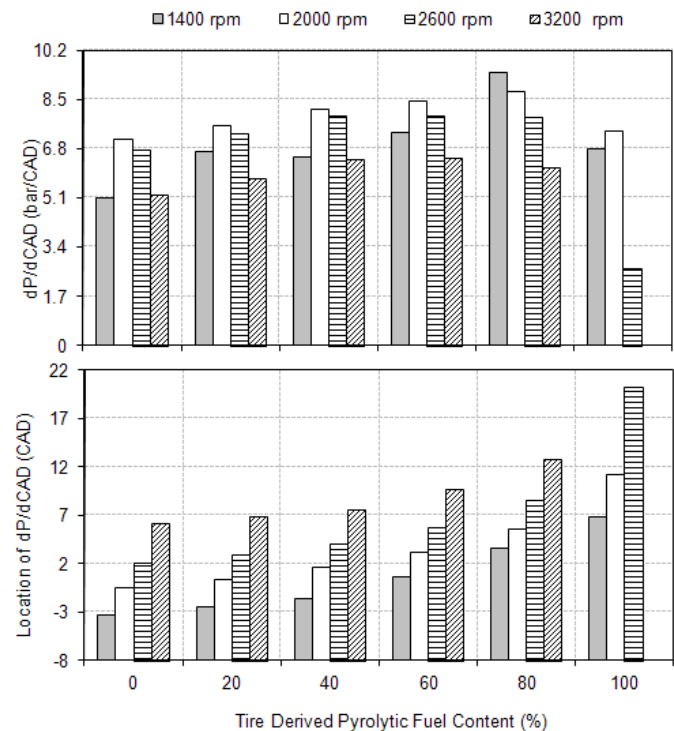


Figure 8. Rate and location of pressure rise ($dP/dCAD$).

The The rate of pressure rise ($dP/dCAD$) and the location of $dP/dCAD$ versus TDF content at various engine speeds are shown in Fig. 8.

The rate of pressure rise ($dP/dCAD$) is the first derivative of cylinder pressure that relates to the smoothness of engine operation. The duration of the ignition delay period, and hence the quantity of fuel injected, and the extent of mixing achieved during the delay period prior to the onset of combustion, contribute to the maximum rate of pressure rise [39].

The rate of pressure rise ($dP/dCAD$) for the low engine speeds (1400 rpm) showed a variation from 5.1 bar/CAD (No.2 Diesel fuel) to 6.7 bar/CAD (TDF20), 6.5 bar/CAD (TDF40), 7.4 bar/CAD (TDF60), 9.4 bar/CAD (TDF80), and 6.8 bar/CAD. For the high engine speed (3200 rpm) tests, $dP/dCAD$ was observed as 5.2 bar/CAD (No.2 Diesel fuel), 5.7 bar/CAD (TDF20), 6.4 bar/CAD (TDF40), 6.45 bar/CAD (TDF60), and 3.75 bar/CAD (TDF80).

The $dP/dCAD$ is higher and occurs at a later crank angle in the case of the TDF blends. This is mainly due to the lower cetane number of the TDF blend, which considerably lengthens the ignition delay and the premixed combustion duration, resulting in higher P_{max} and $dP/dCAD$ in the combustion chamber.

For the higher engine speed, adding more than TDF80 into diesel fuel reduces the $dP/dCAD$. Since the ignition delay becomes excessively longer over TDF80 and cylinder volume expands more during the ignition delay, leading to the loss in combustion energy.

4. Conclusions

In the present study, combustion characteristics of TDF blends were investigated at various engine speeds under full load conditions. Based on the experimental results of this work, the following conclusions can be drawn;

- The DI diesel engine can not run with the TDF100 over 3000 rpm due to excessive ignition delay arising from its fuel characteristics,
- TDF blends exhibit a slightly longer ignition delay period due to their low cetane numbers,
- The heat release occurs at a late crank angle in the case of TDF blends mainly due to the higher olefin content of TDF, and the maximum value of net HRR increases with the increasing TDF fraction in fuel blends,
- TDF blends show usually longer premixed combustion duration, and shorter diffusive and total combustion duration due to better volatility and lower viscosity of TDF blends. In addition, the center of combustion ($CAD50$) moves away from the top dead center to the expansion stroke,
- Increasing TDF fraction in the fuel blends increases the maximum in-cylinder pressure and the rate of pressure rise up to TDF80. In addition, an in-cylinder pressure trace cannot be occurred in the case of TDF100 over 3000 rpm due to excessive ignition delay of TDF.

A general conclusion of this study is that TDF fuel blends can be used in diesel engines without any modification in the fuel system up to TDF40 fuel blends. Adding more than TDF40 requires a cetane booster due to excessive ignition delay, especially for high-speed diesel engines.

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Conflict of Interest Statement

The authors declare that there is no conflict of interest in the study.

CRedit Author Statement

Oğuzhan Doğan: Conceptualization, formal analysis, validation, writing-original draft.

Mustafa Aydın: Formal analysis, validation, writing-original draft.

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