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Effects of exhaust gas recirculation at various loads on diesel engine performance and exhaust particle size distribution using four blends with a research octane number of 70 and diesel

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ABSTRACT

Partially premixed combustion using gasoline-like fuels on compression ignition engines shows great potentials to break the soot-nitrogen oxides trade off and reduce both emissions simultaneously. By simply adjusting the dilution strategies and injection events, the control of partially premixed combustion is relatively easier compared to other low-temperature combustion concepts. However despite these advantages, recent research shows this concept tends to emit ultra-fine particles. Most previous work on partially premixed combustion only focuses on the soot emissions while the particulate matter in terms of number concentration and size distribution are not well investigated. Ultra-fine particles are dangerous to human health and are getting increasing attentions. Thus the detailed particulate matter emission from partially premixed combustion needs to be further investigated. In this work four gasoline-like ternary fuel blends are designed and experimentally tested under partially premixed combustion. The test blends all share the same two base fuels and blended with different additives. The fuel composition is varied to have the same research octane number. Tests are conducted under different engine loads and dilution strategies since the temperature and oxygen concentration are the key factors in the formation and oxidation of soot. Standard diesel is also tested under the same conditions as a comparison. It is found that these blends are capable of running under partially premixed combustion at low and medium loads and they produce near zero soot emissions when using high exhaust gas recirculation rate. However, these blends do emit smaller particles than diesel under all test loads. Besides, blends with oxygen content yield less soot emissions and smaller particles compared to non-oxygen blends.

1. Introduction

Heavy-duty (HD) diesel engines are vital to the modern society due to their high torque capability, reliability, as well as good fuel economy. However, conventional diesel engines suffer from high nitrogen oxides (NOx) and soot emissions due to a wide range of local in-cylinder equivalence ratios and temperatures [1]. These emissions cannot meet the strict emission standards if not controlled by exhaust after-treatment technologies. However, after-treatments often require maintenance and decrease fuel economy. Many advanced in-cylinder measures have been proposed and investigated, including new combustion concepts such as homogenous charge compression ignition (HCCI), reactivity controlled compression ignition (RCCI), and partially premixed charge compression ignition (PPC). The latter two often combined with high exhaust gas recirculation (EGR) rates. The significant advantages of these low-temperature combustion (LTC) strategies are low NOx, soot emissions and high efficiency.

Onishi et al. [2] proposed a combustion concept "Active Thermo-Atmosphere Combustion" (ATAC) in 1979, which differs from conventional gasoline and diesel combustion processes. With ATAC the fuel consumption and exhaust emissions of two-stoke cycle spark-ignition engines are remarkably improved, and there is the possibility of employing this combustion process in other types of engines. Najt and Foster [3] showed an initial investigation of HCCI in 1983, their research concentrates on producing an overall understanding of the basic mechanisms involved in HCCI combustion. After several decades of research on HCCI, the lack of efficient control of combustion process particularly under transient conditions and vary narrow running range are still its major draw-

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Nomenclature

AD	aerodynamic diameter
ATDC	after top dead centre
ATAC	active thermo-atmosphere combustion
BTDC	before top dead centre
CR	compression ratio
CN	cetane number
CAD	crank angle degree
CA10	crank angle where 10% of the heat has been released
CA50	crank angle where 50% of the heat has been released
CO	carbon monoxide
EGR	exhaust gas recirculation
EEPS	engine exhaust particle sizer
FSN	filter smoke number
HD	heavy duty
HC	hydrocarbons
HCCI	homogeneous charge compression ignition
ID	ignition delay
IMEP	indicated mean effective pressure
IMEPg	gross indicated mean effective pressure
LTC	low-temperature combustion
LHV	lower heating value
MON	motor octane number
NOx	nitrogen oxides
PF	premixed fraction
PPC	partially premixed charge compression ignition
RCCI	reactivity controlled compression ignition
RON	research octane number
SOI	start of injection
SMPS	scanning mobility particle sizer

backs. Unlike HCCI and PPC engines which only use a single fuel, RCCI is a dual-fuel principle [4] that uses direct injection of high-reactivity fuels to trigger the ignition and combustion of low-reactivity fuels typically supplied through a port-fuel injection system. Reitz et al. [5] have done many research on RCCI. RCCI provides more efficient control over the combustion process and is able to increase fuel economy and reduce pollutant emissions. The difference with a conventional dual fuel engine is the early timings of the direct injection event. PPC uses a single fuel and it effectively correlates the ignition timing with injection timing to make a good control of combustion process [6]. Zhang et al. [7] indicated that PPC consists of a two-stage combustion process: in the first stage the stratified fuel/air mixture auto-ignites, which leads to partial oxidation of fuel in the fuel-rich zone and a mixture of radicals and hot products in the fuel-lean region. In the second stage the partially oxidized fuel/air mixture is further oxidized in a thin diffusion flame where the diffusion and chemical reaction both play an important role. According to the experimental research from Lund University [8] on a Scania D12 HD engine with a compression ratio (CR) of 12.4, diesel PPC can produce low NOx and soot emissions simultaneously at 8 bar indicated mean effective pressure (IMEP) if more than 70% EGR is applied. Unfortunately the high EGR level and low CR resulted in a combustion efficiency below 90%. Kalghatigi introduced the concept of gasoline PPC in 2006 [9], inspired by the fact that a longer ignition delay can be achieved with a fuel that is harder to ignite even without using high EGR rates, too early injection timings or too low compression ratios. Fuels with a research octane number (RON) of 70 are identified to be close to optimal for PPC [10].

In our previous work [11], blends with a RON of 70 were experimentally investigated using an injector which has 7 holes of 207 μm and its spray cone angle is 150°. The results showed that the three blends are capable of running under PPC mode when IMEP is below 8 bar. When the IMEP continues to increase, the three blends start to present short ignition delay and produce high soot emissions (1–3 Filter Smoke Number (FSN)) due to the reduced premixedness. The effect of dilution strategies on soot emissions using blends of RON 70 under PPC mode was also investigated [12]. Previous research [11–13] only focused on soot emissions in g/kW h or in FSN. However, recent studies correlate fine particulates and ultra-fine particulates to adverse human health effects [14]. The porous particulate agglomerations could deposit in the deep lung and cause diseases like asthma, pneumonectasis and nasopharyngeal darcinoma [15] and [16]. PPC mode may contribute to more fine and ultra-fine particles. Load, dilution and fuels can be the major influences. Therefore, it is necessary to acquire knowledge about the particulate matter in terms of number concentration and size distribution when using blends of RON 70.

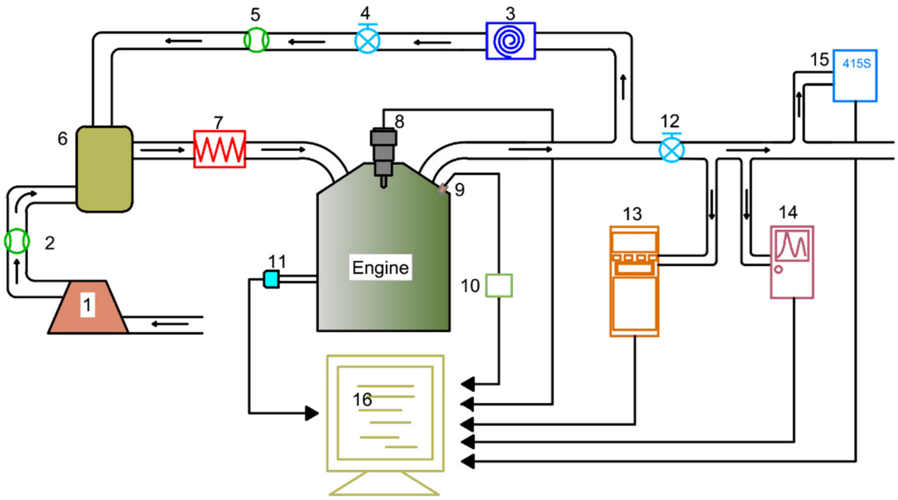
In this study, four blends with a RON of 70 have been designed. The first blend only consists of *n*-heptane ($CH_3(CH_2)_5CH_3$) and *iso*-octane ($(CH_3)_3CCH_2CH(CH_3)_2$) and is denoted as PRF. The other three blends use *n*-heptane and *iso*-octane as base fuels and blended with ethanol (CH_3CH_2OH) or *n*-butanol ($CH_3(CH_2)_3OH$) or toluene ($C_6H_5CH_3$) separately. They are denoted as ERF, BRF and TRF, respectively. The fuel composition is varied to have the same RON of 70. The limited number of components makes that the fuels can be handled in numerical simulations, in addition, the chemical and spectral purity makes them suitable for use in optically accessible engines as well. Moreover, given this desired reactivity, the influence of fuel structure and oxygen content on particulate emissions can also be investigated. Diesel is also tested under the same experimental conditions for reference. A latest type of injector which matches the bowl shape better is used to optimize combustion and extend the upper load range of PPC. Moreover, the combustion characteristics and particulate emission characteristics including particle number concentration and size distribution using the fuels with a RON of 70 are also investigated. The effects of EGR on combustion parameters and particulate emissions are discussed as well.

2. Experimental details

In this section, firstly the test setup including engine and emission testing equipment are introduced. Secondly, how to design the test blends are talked about and their properties are given. Thirdly, related parameters used in this paper are defined. Lastly, the experiment conditions are discussed.

2.1. Test setup

Experiments were conducted on a modified 12.6 L in-line six cylinder DAF XE355c engine. Cylinder 1 with a CR of 15.7 was isolated as a test cylinder, configuration is depicted in Fig. 1. The high pressure fuel system is composed of a Resato high pressure pump and a pressure regulator. Intake pressure is provided by the air compressor and up to 5 bar can be achieved. Exhaust pressure is also adjustable to ensure sufficient EGR rate, and it was set to 0.3 bar higher than intake pressure during the experiments. A heater was used to heat up the intake charge to a desired temperature.



1. Air compressor 2&5.Micromotion flow meter 3.EGR cooler 4.EGR valve 6.Mixing tank 7.Heater 8.Injector 9.Pressure transducer 10.Charge amplifier 11.Encoder 12.Backpressure valve 13.Exhaust analyzer 14.Engine exhaust particle sizer 15.Smoke meter 16.Electric control unit & data acquisition.

Fig. 1. Schematic layout of the experimental set-up.

The injector used in this work has 7 holes of 180 μm in diameter and the spray cone angle is 139°.

The concentration of gaseous emissions, for example, hydrocarbons (HC), carbon monoxide (CO), carbon dioxide (CO_2) and NOx, are measured using a Horiba Mexa 7100 DEGR exhaust analysis system, which measures HC by flame ionization method, CO and CO2 by nondispersive infrared method, and NOx by chemiluminescent method. Smoke emissions (in FSN units) are monitored using an AVL 415S smoke meter. The engine exhaust particle sizer (TSI EEPS 3090) is used in the experiments to obtain detailed information about particle concentration and size distribution. It was designed specifically for measuring engine exhaust and uses a measurement based on electrical mobility classification. It is equipped with a rotating disk diluter MD19-3E, an air supply/evaporation tube ASET15-1 and a data acquisition unit CU-2. The EEPS can measure the size distribution of engine-exhaust particle emissions in the range from 5.6 to 560 nm with the fastest time resolution available (10 Hz). In the experiments, real time sampling rate is 1 Hz and sampling period is 1 min. During all the experiments primary dilution temperature is set at 150 °C with a dilution factor of 50 and the secondary dilution temperature is set at 300 °C with a dilution factor of 6.7, thus the overall dilution ratio is 335. In this work the EEPS spectrometer data is processed with the soot matrix that matches more closely to the data from a TSI scanning mobility particle sizer (SMPS) spectrometer. Software version 3.2.5 and firmware version 3.1.2 are used when using EEPS.

Fast changing parameters, such as in-cylinder pressure and common rail fuel pressure, are recorded using a SMETEC Combi crank angle resolved system. An in-house data acquisition system is used to measure slow changing parameters, like intake air flow and exhaust gas temperature. These parameters are measured during a period of 40 s at a frequency of 20 Hz.

2.2. Test fuels

Fuels with a RON of 70 are identified close to optimal for PPC [10]. In this study *iso*-octane and *n*-heptane are selected as base fuels, ethanol and *n*-butanol are chosen as oxygenated additives while toluene is used as an aromatic additive. Four blends (PRF, ERF, BRF, and TRF) with a RON around 70 were tested, EN590 diesel was added to the test-matrix as a comparison. Relevant physical properties for the neat compounds can be found in Table 1. Table 2 depicts the properties of the four blends.

2.3. Definition of related parameters

(1) Gross indicated mean effective pressure (IMEPg). Due to the layout of the engine, the brake specific mean effective pressure (i.e. with respect to the power output at the crankshaft) cannot be used. Therefore in this paper, the IMEP is calculated based on in-cylinder pressure signal. To be able to evaluate the combustion performance at different intake pressures and exhaust pressures, the IMEPg which excludes the gas exchange stroke is used to present engine load and calculate the indicated emissions.
(2) Ignition delay (ID). The crank angle where 10% and 50% of the heat has been released is referred as CA10 and CA50 respectively. ID in this paper is defined as the crank angle difference between start-of-injection (SOI) and CA10: ID = CA10-SOI (CAD).
(3) Combustion efficiency. Combustion efficiency is a measure of how efficiently fuel energy is converted into useful energy. When calculating the combustion efficiency, only combustible species like HC and CO in the exhaust are considered. This is a common practice. Possible effects of other components are ignored.
(4) EGR rate. In this study, the EGR rate is experimentally determined from the ratio of the CO_2 mole concentration in the intake gas to the CO_2 mole concentration in the exhaust gas.
(5) Premixed fraction (PF). Premixed combustion phase is controlled by chemical kinetics and its combustion speed is faster than diffusion combustion phase. The PF is introduced in the previous study [13]. It is defined as a ratio of heat release from premixed combustion to the total heat release. Premixed- and diffusive- heat release rates are normally not separated clearly, a Gaussian profile as presented in Eq. (1) is fitted to the heat release rate to calculate the premixed heat release. The Gaussian profile is a mathematical representation of the premixed reaction phase. Still it provides a convincing measure of the premixing degree.

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Table 1
Properties of fuel compositions.

Composition	*n*-Heptane	*iso*-Octane	Ethanol	*n*-Butanol	Toluene	EN590
Density at 20 °C (kg/m^3)	679.5[a]	692[a]	788[a]	810[a]	870[a]	820–845
RON	0	100	109[b]	98[b]	129[c]	–
MON	0	100	90[b]	85[b]	109[c]	–
LHV (MJ/kg)	44.6[d]	44.4[d]	26.8[b]	33.1[b]	40.6[d]	42.9
Boiling point (°C)	98[a]	99.3[a]	78.2[a]	117.6[a]	111[a]	149–371
Viscosity at 20 °C (mPa·s)	0.42[e]	0.5[e]	1.074[a]	2.9[f]	0.59[e]	2–4.5

[a] [20].
[b] [21].
[c] [22].
[d] [23].
[e] [24].
[f] [25].

Table 2
Properties of the four blends.

Blends	PRF	ERF	BRF	TRF
Vol-% *n*-heptane	30	52	35	36
Vol-% *iso*-octane	70	23	40	39
Vol-% additive	0	25	25	25
Additive	–	Ethanol	Butanol	Toluene
RON	70	70	72.9	70.8
DCN	27	25.8	26.1	27.4
LHV (MJ/kg)	44.47	40.18	41.27	43.33
LHV of stoichiometric mixture (MJ/kg)	2.76	2.79	2.75	2.77
Stoichiometric air-fuel ratio	15.11	13.42	13.99	14.63
H/C ratio	2.26	2.41	2.32	1.91
O/C ratio	0	0.0961	0.0582	0

$$G(x) = h \cdot e^{\frac{-(x-x_0)^2}{2\alpha^2}} \quad (1)$$

Here x_0 is the central position of the peak, h and α represent the height and width of the Gaussian profile respectively. Fig. 2 shows the heat release rates of PRF at three different loads with 0% EGR and the fitted Gaussian profiles.

2.4. Test conditions

Before the tests start, the engine is warmed up until the lubrication oil and coolant fluid temperature reach 85 °C. All the tests are conducted under steady state conditions and single injection strategy is used. The engine speed is set to 1200 rpm and intake temperature is controlled at 40 °C. Combustion and emissions characteristics at different loads (5 bar, 10 bar, and 15 bar IMEPg) are investigated and an EGR sweep is performed at each load. In order to achieve different loads and also keep injection duration short, various injection pressures are applied: 600 bar at low load, 1200 bar at medium load, and 1800 bar at high load. Intake pressure is set at 1.2 bar instead of ambient pressure at low load to ensure the combustion of these test blends, while at medium and high loads intake pressures are chosen to be 1.8 bar and 2.5 bar respectively. This is done to ensure that the air/fuel ratio λ is higher than 1.2 even when 45% EGR is used to avoid excessive soot emissions. The SOI timing is the same for all test fuels at low load, thus the combustion phasing varies as shown in Fig. 3. At low load, the combustion phasing of the four blends is very sensitive to the increase of EGR, even a misfire occurred for BRF when 30% EGR is used. Therefore at low load the maximum EGR rate used for these blends are reduced. At medium and high loads, all test fuels share a same SOI timing but are injected later into the cylinder compared with low load to reduce cyclic variations and achieve acceptable maximum pressure rise rates.

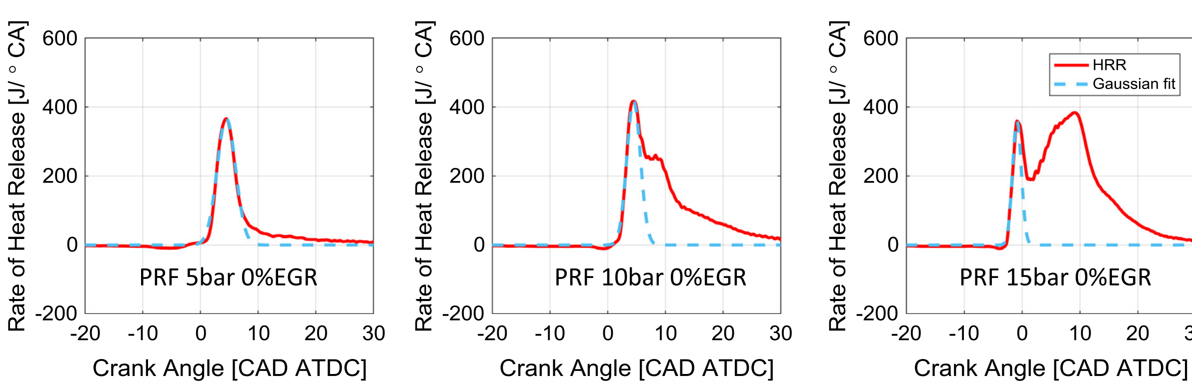

Fig. 2. Heat release rates (red line) and Gaussian profiles (blue dashed) of a PRF experiment at three loads with no EGR. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

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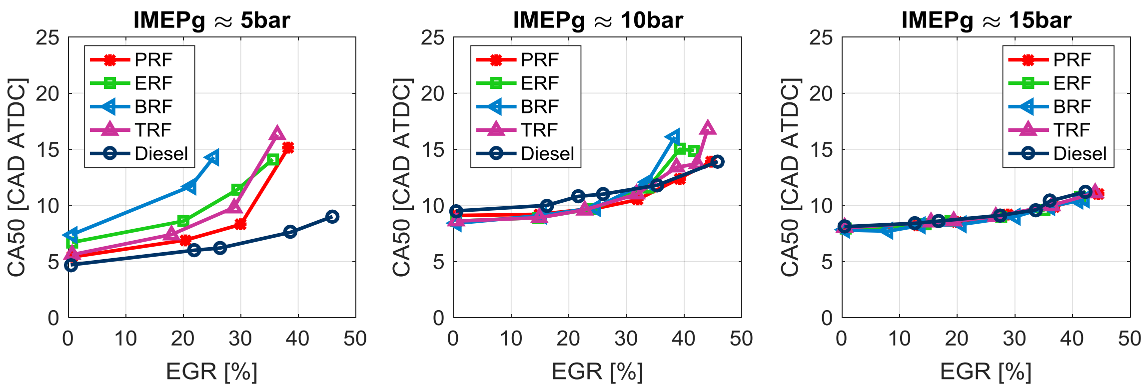

Fig. 3. Combustion phasing vs EGR at various loads.

3. Results and discussion

This section discusses the main results of this research. The first subsection discusses the combustion characteristics of the test fuels at various loads. The second subsection analyses the trade-off relationship between NOx and soot emissions for the test fuels at different test conditions. Finally, the third subsection shows the particulate size distribution analysis.

3.1. Combustion characteristics

The influences of EGR on combustion phasing, heat release rate, ignition delay and premixed fraction, and combustion efficiency using four blends of RON 70 and diesel at different operating loads are discussed here.

3.1.1. Combustion phasing CA50

As shown in Fig. 3, the four blends are very sensitive to the increase of EGR rate at low load. Less fuel is burned at low load, the combustion temperature is lower compared with medium and high loads. The four blends ignite later than diesel, therefore experience a longer mixing process and are consequently more prone to so-called overleaning. When EGR rate is increased, the in-cylinder combustion temperature reduces as does the oxygen concentration. Hence the ignition delay time increases and the blends start to burn unstably and fail to ignite finally. The ignitability of an air/fuel mixture highly depends on the intake air temperature and the boost pressure. At medium and high loads, higher boost reduces the mixing period and leads to more diffusion combustion. More energy is released and the combustion temperature is also sufficient to maintain sustainable combustion for the blends with higher RON when using high EGR levels. Obviously, the combustion phasing is retarded for all the fuels with the increase of EGR, since SOI timing is maintained the same for each load. Adding EGR reduces reaction rates leading to longer burn rates hence even more retarded CA50.

3.1.2. Heat release rate

Fig. 4 depicts the heat release rates at low load, medium load, and high load with 0% EGR. At low load, the four blends provide almost pure premixed combustion while diesel shows both premixed and diffusion combustion and thus a longer burn duration. At medium load, the four blends are still controlled by premixed combustion mainly but start to show more diffusion combustion. Diesel presents a typical two-stage combustion process, a small amount of premixed combustion followed by large amount of diffusion combustion. At high load, all fuels burn in a conventional diesel combustion mode and are dominated by diffusion combustion. Note that the heat release rate graphs illustrate that the combustion duration of the four blends is shorter than that of EN590 diesel leading to more expansion works.

3.1.3. Ignition delay and premixed fraction

Fig. 5 illustrates the effects of EGR on ignition delay times for the test fuels at various loads. The increase of the EGR rate results in longer ignition delay and the overall combustion shifts to a later stage. The differences in ignition delay between blends of RON70 and diesel become less as load increases, and even converge at high load. ERF and BRF show slightly longer ignition delay compared to PRF and TRF, which correlates with their lower cetane numbers.

It is seen from Fig. 6 that the four blends are dominated by premixed combustion and provide the largest premixed fraction at low load. While at high load the four blends show almost no difference compared with diesel, they all burn in a conventional diesel combustion mode and have the smallest premixed fraction. Higher boost and in-cylinder temperature allow blends of RON 70 to burn shortly after the SOI at high load, thereby reducing ignition delay. Medium load can be regarded as a transition combustion mode between PPC and conventional diesel combustion mode. At medium load, all fuels show a two-stage combustion mode starting with premixed combustion and followed by diffusion combustion. At low load EGR has little influence on the premixed fraction of the four blends although the increased EGR rate results in a longer ignition delay. The increased ignition delay should promote more premixed combustion or higher premixed fraction, but the selected blends already show almost pure premixed combustion when EGR is not applied. Compared with the blends with higher RON, diesel can tolerate higher EGR rate at low load. Fig. 6 nicely shows that a considerable premixed fraction is only reached at low load and high EGR rates for diesel, as expected.

3.1.4. Combustion efficiency

Fig. 7 demonstrates that at low load the four blends have a lower combustion efficiency than diesel especially when more EGR is applied. At low load, fuels are injected earlier into the cylinder and the four blends provide a longer time allowing fuel to premix with air. The typical long ignition delays will produce more regions where fuel is already mixed beyond the lean limit of combustion (overleaning) and produces more unburned HC emissions. With the increase of EGR, both the in-cylinder temperature and oxygen concentration are reduced, in addition, the retarded combustion phasing reduce the combustion temperature further. Therefore generating more HC and CO emissions because of the incomplete combustion and thus low combustion efficiency. When load increases, more fuel is burned and more energy is released, thus a higher combustion temperature and a more complete combustion hence less HC and CO emissions. Moreover, as load increases less over-leaning occurs due to the reduced ignition

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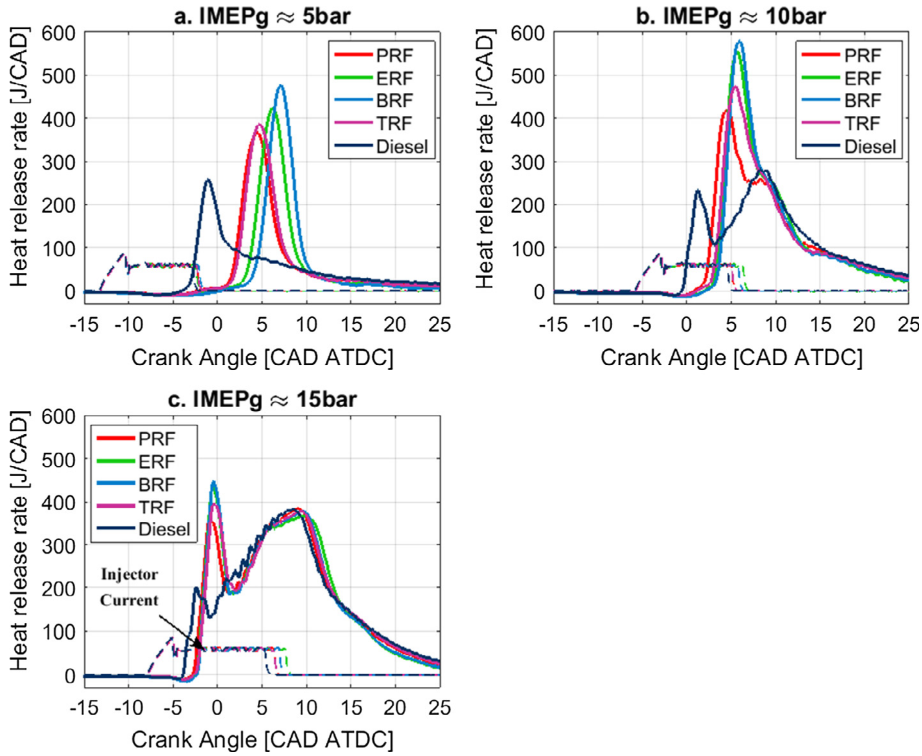

Fig. 4. Heat release rates at various loads with 0% EGR.

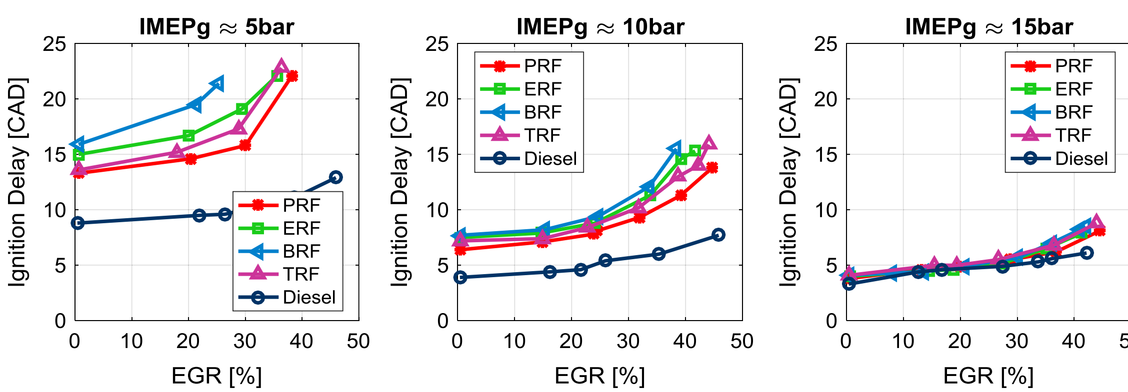

Fig. 5. Ignition delay vs EGR at various loads.

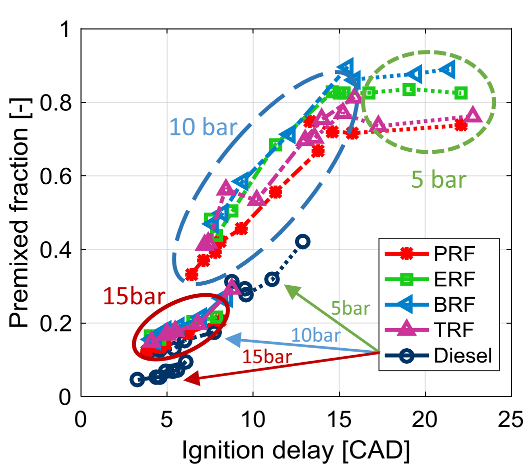

Fig. 6. Premixed fraction vs ignition delay at various loads.

delay times of the blends. Consequently, less unburned HC emissions are produced and a higher combustion efficiency is obtained.

3.2. Trade-off relationship between NOx and soot emissions

Fig. 8 illustrates the trade-off relationship between NOx and soot emissions for the test fuels at various loads. At low load, the NOx emissions of the four blends are reduced when more EGR is applied, and their soot emissions remain at zero. But their NOx emissions are slightly higher than 0.4 g/kW h (EURO VI) since less than 40% EGR is used. This problem can be solved by using more EGR while adjusting the SOI to advance the combustion phasing to maintain the stability of combustion. At medium load when more than 40% EGR is applied, all blends with RON 70 produce near-zero soot and the NOx emissions are below 0.4 g/kW h (EU VI). While at high load, all fuels show a NOx and soot trade-off relationship, but oxygenated fuels (ERF and BRF) still produce less soot emissions. At low and medium loads, the four blends of RON70 are

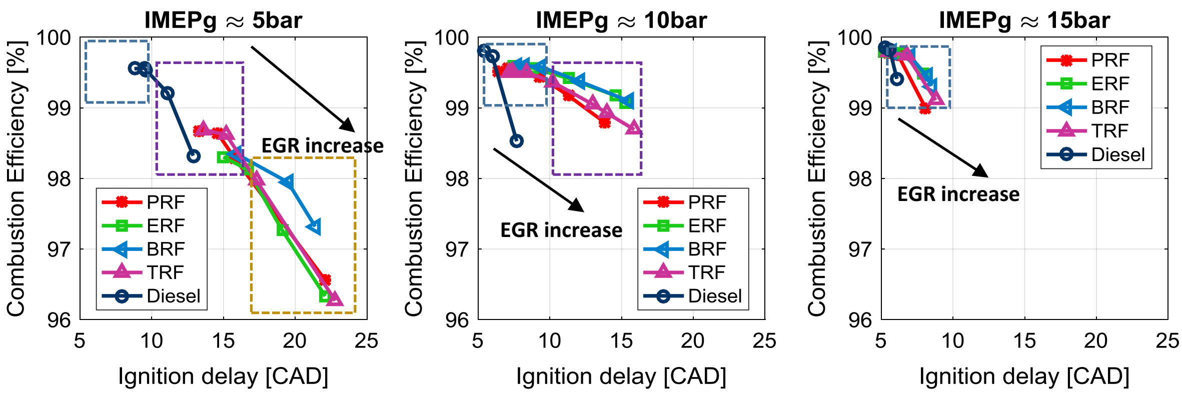

Fig. 7. Combustion efficiency vs ID at various loads.

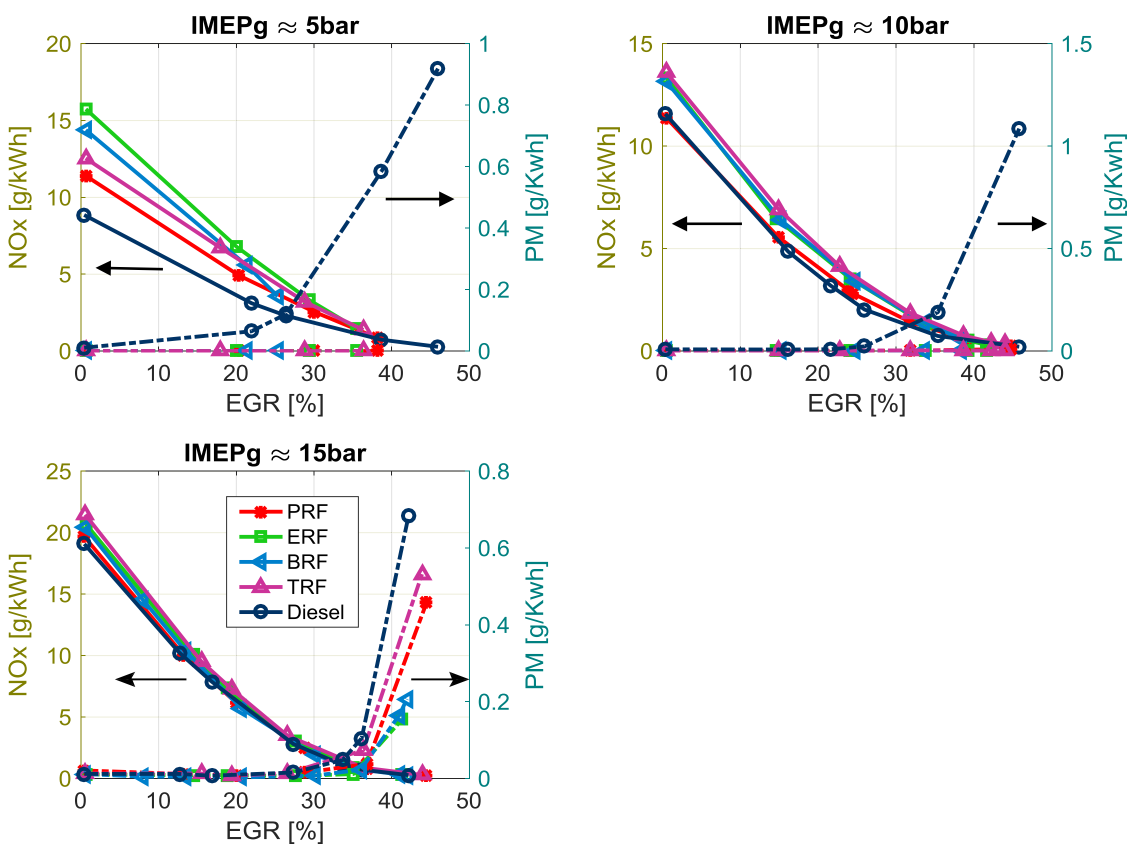

Fig. 8. NOx and PM vs EGR at various loads.

dominated by premixed combustion. At medium load using a high level of EGR rate, the four blends provide higher premixed fractions which are comparable to that at low load. However, at high load all fuels are dominated by diffusion combustion and run under conventional diesel combustion mode with a premixed fraction smaller than 0.3. Therefore more research work should be done to expand the proportion of premixed combustion and alleviate the trade-off relationship between NOx and soot emissions at high load. It is also noticed that overall NOx emission goes up as the load increases, which is attributed to the high temperature caused by the increased diffusion combustion.

3.3. Particulate emissions

Particle number size distributions exhibit unimodal or bimodal distributions depending on operating conditions. Particles smaller than 20 nm are commonly referred to as nucleation mode while particles larger than 20 nm are typical of accumulation mode. Nuclei particles are mainly due to semi-volatile components originating from unburned fuel, lubricant oil and partial combustion products [17]. The accumulation mode is composed primarily of carbonaceous agglomerates and adsorbed material, they are formed in locally fuel-rich regions of the flame [17].

At higher-pressure conditions and the anoxic environment when a high level of EGR is used, soot nucleation are promoted, which contributes to the growth of the existing soot nuclei [18]. The soot oxidation rate declines and particles in the nucleation mode with reactive surfaces tend to grow and agglomerate to generate large particles. With an increase in the number of particles, coagulation rate increases and therefore larger particles are formed, resulting in an increase of particulate aerodynamic diameter.

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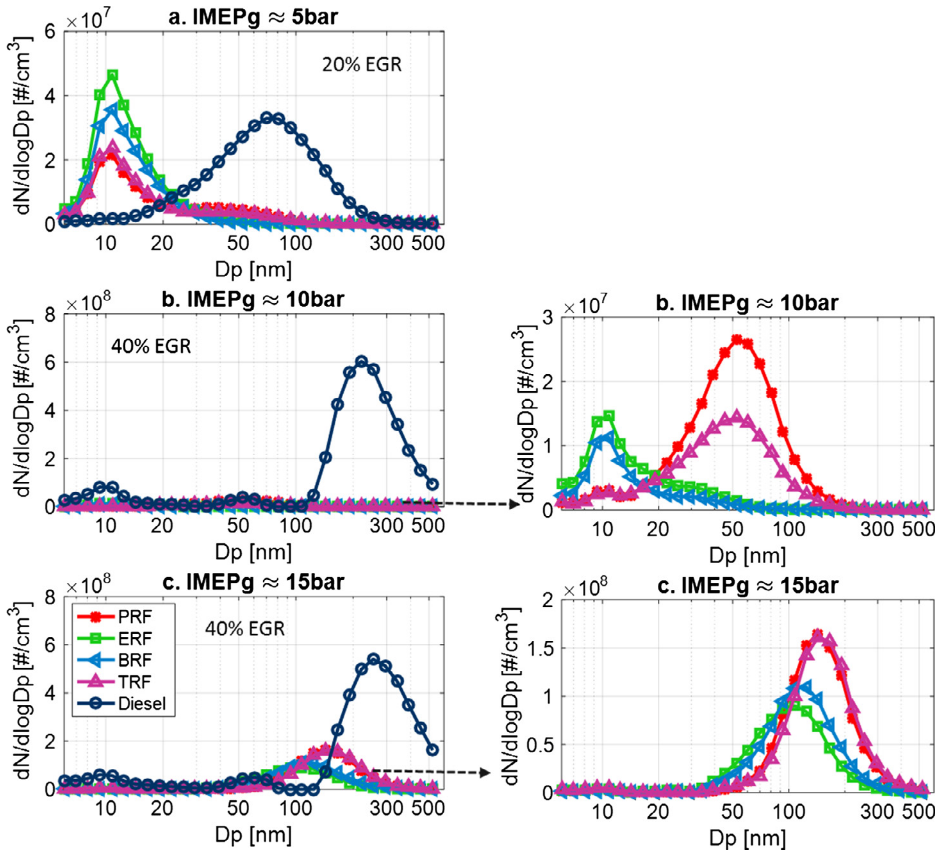

Fig. 9. Particulate number concentration and size distribution at various loads, right column is a zoomed plot to distinguish the results of the four blends individually.

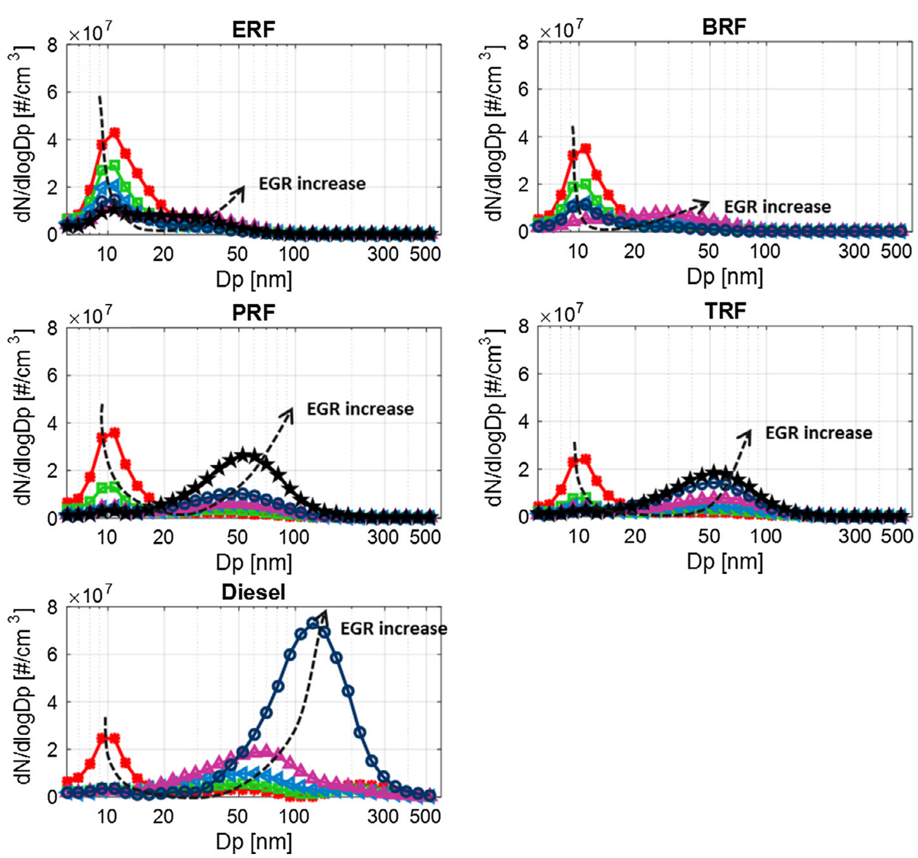

Fig. 10. Particle number concentration and size distribution vs EGR at medium load.

3.3.1. Particulate emissions from the selected fuels at low, medium and high loads

In this study, the particle number concentration and size distribution are measured and compared. At low load with 20% EGR (the highest stable common EGR level at low load) the four blends show an increase in the number of particles at aerodynamic diameter (AD) below 20 nm compared to diesel as depicted in Fig. 9. At low load, the particle size distribution of diesel is unimodal. Most particles are produced in the accumulation mode with the peak diameters of 78–82 nm. For each fuel, in general the particle number shifts upwards and its distribution curves move towards larger size when engine load increases. At medium load the oxygenated blends (ERF and BRF) still produce most particles in the nucleation mode with the peak diameters of 8–12 nm, while the size distribution profiles of PRF and TRF shift towards larger size and produce most particles in the accumulation mode with the peak diameters of 50 nm. Still the four blends produce much less particles in both nucleation mode and accumulation mode compared to diesel.

3.3.2. Effects of EGR on particulate emissions for the selected fuels

Here only the particulate emissions at 10 bar IMEPg is discussed. A similar trend can be also found at 5 bar and 15 bar IMEPg. As shown in Fig. 10, the size distribution curves of all test fuels shift towards larger particles as the EGR increases. There is a noticeable increase of the particles (AD > 50 nm) for PRF, TRF and diesel when more than 30% EGR is used, while ERF and BRF still produce most particles with an AD smaller than 40 nm even when 40% EGR is used. This must be associated with the local equivalence ratio since the ignition delays of all blends are comparable. With similar injection velocities (i.e. injection pressures) the air entrainment is the same for all, while the local equivalence ratio of ERF and BRF in the spray is lower because they contain hydroxyl groups. This might explain the observed phenomenon. Without the use of EGR, the peak concentration occurs at the diameter range of 10 nm for all selected fuels, but when 30% EGR is used, the peak concentrations of the particles are around 11 nm, 11 nm, 50 nm, 52 nm, and 60 nm for ERF, BRF, PRF, TRF, and diesel, respectively. At medium load, the increased EGR rate reduces the number of particles in the nucleation mode but doesn't have significant influences on the concentration of particles in the accumulation mode for ERF and BRF. For non-oxygenated fuels, the increased EGR rate reduces the number of particles with small aerodynamic diameters but increases the number of particles with large aerodynamic diameters, hence the total particle mass increases. However, the increase in the total particle mass is more significant for diesel with the increased EGR rate, which is in agreement with the steeper rise of smoke emissions (PM) presented in Fig. 8. Diesel produces most particles with an AD larger than 100 nm when more than 35% EGR is used, and these particles dominate the mass size distribution due to the cubic dependency of the particle diameter. The cooled EGR is a well-known technique to reduce NOx emissions. The increase of EGR rate results in a lower oxygen concentration which favours the formation of soot, in addition, the increased EGR reduces peak in-cylinder temperatures and leads to incomplete soot oxidation. The use of high EGR rate leads to higher amount of soot emitted in a given exhaust volume where coagulation, accumulation, condensation of volatile fractions on the particles and surface growth are likely to happen [19]. All these factors result in the increasing concentration of larger diameter particles in the accumulation mode.

4. Conclusion

The present study explored combustion characteristics and particulate characteristics in terms of number concentration and size distribution of the selected fuels at various loads and EGR levels. From the experimental results aforementioned several conclusions could be drawn:

- At low load, all the test blends show almost pure premixed combustion, while diesel provides a premixed fraction around 0.3. The degree of premixing of the four blends is not sensitive to the change of EGR rate as it is already approaching 0.9, while diesel presents more premixed combustion as the increase of EGR. The four blends have a lower combustion efficiency compared to diesel, but they can produce near zero soot.
- At medium load, all the test blends start to present a diffusive burning phase. Their premixing degree increases with EGR percentage. The four blends are still dominated by premixed combustion ($0.4 < PF < 0.9$), while diesel shows a much lower premixed fraction ($PF < 0.2$) which is dominated by diffusion combustion. The four blends can alleviate the trade-off relationship between NOx and soot emissions, in addition, when more than 40% EGR is applied their soot and NOx emissions are below EURO VI emission standards.
- At high load, the engine operates mostly under convention diesel combustion mode for all the test fuels. There is a small difference in ignition delay and premixed fraction among all the fuels. All fuels show a NOx-soot trade-off relationship, but oxygenated fuels (ERF and BRF) still produce less soot.
- Compared with other fuels, ERF and BRF yield less soot emissions and produce considerably more particles in the nucleation mode. This is probably due to their lower local equivalence ratios as the air entrainment rates are similar for all fuels. In general the particle size distribution curves shift upwards and towards bigger sizes as the engine load is increased for all the fuels.
- EGR technology is an effective way to control NOx emissions but also influences the smoke and particulate matter emissions. When more EGR is applied, in general the particulate number concentration in nucleation mode decreases, while more particles in accumulation mode are generated. The combination of fuels with higher RON and a suitable EGR rate, apart from reducing NOx emissions, also leads to a significant reduction in particle mass or smoke. Especially for the oxygenated fuels, the size distribution remains dominated by small particles even at high load where all fuels mainly burn in the diffusion combustion. Further research work is required in order to determine how much EGR is optimal for various fuels under different combustion modes.

In summary, the four tested blends with a RON 70 are suitable for heavy-duty diesel engines in terms of soot and NOx emissions. ERF and BRF can alleviate the NOx-soot trade-off relationship even at high load. In the future work, binary blends will be tested instead of ternary blends to further investigate the influence of different chemical structures on soot and particulate emissions.

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