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Attainment and Load Extension of High-Efficiency Premixed Low-Temperature Combustion with Dieseline in a Compression Ignition Engine

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A study of high-efficiency premixed low-temperature combustion (LTC) with dieseline (a blend of diesel and gasoline) was conducted on a high-speed compression ignition engine and strategies to broaden the operational range of this dieseline LTC were investigated. Increased ignition delay and higher volatility of dieseline relative to diesel fuel contribute to produce a well-mixed charge, and a simultaneous reduction in NO_x and smoke emissions with a moderate exhaust gas recirculation (EGR) rate can be achieved across a relatively narrow load window. An intake boost strategy was employed to broaden the LTC operational window. Higher intake pressure increases air charge and allows for higher fuelling, so the upper load limit of this LTC mode is extended. An increase in NO_x emissions is observed at light to mid loads, but a slight increase in EGR rate can reduce NO_x emissions to an acceptable level. Because dieseline requires only a relatively light use of EGR, combustion efficiency across the entire dieseline LTC operational range is maintained at a high level.

1. Introduction

The trade-off between NO_x ($NO + NO_2$) and particulate matter (PM) emissions from diesel engines is a well-known drawback of classical diesel combustion. NO_x and PM formation are influenced by in-cylinder local equivalence ratios and combustion temperatures.[1] Therefore, if combustion can be shifted outside the regions where the local equivalence ratios and combustion temperatures are favorable for NO_x and PM formation, the simultaneous reduction of NO_x and PM emissions can be achieved. This is the concept of low-temperature combustion (LTC).[1,2] One major challenge of LTC is to prepare a fully premixed charge before the start of combustion (SOC). Utilizing large amounts of exhaust gas recirculation (EGR) and high fuel injection pressure are common strategies used to promote the fuel and air mixing process. High EGR reduces in-cylinder temperatures and extends ignition delay while high injection pressure promotes fuel vaporization and atomization, both of which are beneficial for the mixing process.[3,4] As a result, conventional diffusion combustion can be largely eliminated and combustion proceeds in a more premixed and cooler mode.

However, the application of high EGR reduces in-cylinder oxygen content, causing deterioration of combustion efficiency that leads to increased hydrocarbon (HC) and carbon monoxide (CO) emissions.[5] Utilizing high EGR or cooling the EGR to lower temperatures can also worsen EGR cooler fouling.[6] To avoid these disadvantages of high EGR rates, the fuel with a lower cetane number and higher volatility is considered in this study as an alternative path to LTC. The resistance to autoignition of the low-cetane fuel may provide a sufficient ignition delay for mixing and faster vaporization by high volatility can increase mixing rate. Therefore, thorough mixing can be achieved without high EGR and deteriorated combustion efficiency.

Several researchers have investigated the relationship between LTC operational range and cetane number. A moderate range of cetane numbers (40−63) was investigated with an early injection combustion mode.[7] The low-cetane fuel produced less smoke emissions at high loads that could extend the engine operation to an upper load level. However, the combustion noise of the low-cetane fuel increased at high loads, limiting the operational range in that study. Shimazaki et al. tested a series of fuels with cetane numbers ranging from 19 to 61 using a late injection LTC strategy and found the CN 30 fuel to be optimal for load extension. Any deviation from this cetane number would either aggravate knocking at heavy loads or lead to increased combustion instability at light loads.[8] In a study by Kusaka et al., three different cetane number fuels (CN 58, 31, 22) were tested in LTC mode on a light-duty diesel engine.[9] At high loads, the low-cetane fuel was superior, forming a homogeneous lean mixture and improving the NO_x, smoke emissions, and thermal efficiency as well.

Kalghatgi et al. explored an LTC strategy with gasoline-like fuels and found ultralow soot and NO_x emissions could be

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achieved at loads up to 15.95 bar IMEP.[10,11] Moreover, Weall et al. studied the plausibility of pump gasoline on a light-duty LTC engine at low loads and low speeds.[12] In this gasoline LTC combustion mode, smoke emissions were always near zero and NO_x emissions were controlled at a relatively low level even without EGR. However, because of gasoline's resistance to autoignition, strategies like intake boost, intake heat or fuel stratification had to be used for stable operation in this LTC combustion mode.[12] Therefore, a fuel with a moderately reduced octane rating, such as a blend of diesel and gasoline, recently termed "dieseline",[13] has been suggested for LTC, in particular at light loads.[12] In an LTC strategy with 1500 bar injection pressure and an EGR rate higher than 40%, the blend of 50% diesel and 50% gasoline could maintain stable operation at idling conditions and the benefits in emissions were also observed at heavy loads.[14]

Following Weall et al's work,[14] in this paper dieseline was prepared by adding a moderate amount of gasoline (20% by volume) into diesel fuel and was tested using a previously developed[15,16] late-injection premixed LTC mode. Fuels with gasoline proportion higher than 20% was not investigated due to the increased combustion instability. Different from prior work, the EGR rate was limited to less than 40% to avoid the production of high concentrations of incomplete combustion products and decreased combustion efficiencies found in previous LTC research.[5] A novel high-efficiency and low-emission dieseline LTC mode and the extension of its operational range are the focus of this work.

This paper is organized in the following order: First, experimental apparatus, fuels, and test methods are introduced. Next, both diesel and dieseline fuels are tested across a load sweep in a naturally aspirated LTC mode with 35% EGR. The superiority of dieseline in reducing smoke and NO_x emissions are demonstrated. Finally, intake pressure and EGR are adjusted to extend the heavy and light load limits of dieseline LTC, so simultaneous ultralow NO_x and smoke emissions are achieved across a broader operational range than that obtained in the second step.

2. Experimental Methods

2.1. Experimental Setup. The experimental study was carried out on a single-cylinder engine based on a four-cylinder direct-injection production diesel engine sold by General Motors in Europe. The compression ratio of this engine was decreased from 19:1 to 15:1 by replacing the original piston with one featuring a larger volume combustion bowl. The test engine had four valves per cylinder and one centrally mounted common rail fuel injector. A summary of the engine's geometry is listed in Table 1, and more information about the experimental apparatus can be found in colleagues' prior work.[17]

Cylinder pressure was acquired by a water-cooled Kistler 6041A piezoelectric pressure transducer with a resolution of 0.2° crank angle for 200 consecutive engine cycles at each operation point.

Table 1. Summary of Engine Specification

displacement	425 cm^3
bore	79.0 mm
stroke	86.0 mm
connecting rod length	160.0 mm
compression ratio	15:1
injector nozzle hole number	6
injector nozzle spray angle	150°

Table 2. General Properties of the Test Fuels

general properties	diesel	dieseline
cetane number	42.1	36.7
sulfur (ppm)	8	11.5
density at 15.56 °C (g/mL)	0.85	0.83
lower heat value (MJ/kg)	42.8	42.7
T50 (°C)	257	239
viscosity (mm^2/s)	2.403	1.803
carbon (wt %)	86.6	86
hydrogen (wt %)	13.4	13.4
oxygen (wt %)	0	0.6

The signal from the pressure transducer was sent to a charge amplifier and a high-speed data acquisition system. The analysis of heat release, which includes the calculation of heat release rate, bulk gas temperature, and burned mass fraction, was based on a zero-dimensional ideal-gas combustion model.[18] Hohenberg's correlation was used for the heat transfer calculation,[19] and the thermodynamic properties of the trapped gas in cylinder are estimated using the correlation described by Gatowski et al.[18] The amount of heat loss was corrected using a scaling coefficient so that the sum of apparent heat release and heat loss is equal to the amount of chemical energy in the fuel minus the chemical energy in incomplete products of combustion found in the exhaust.

Gaseous emissions, which include NO_x, CO, CO_2, and HC were measured by a Horiba series 23 emissions bench and converted to fuel-specific emissions, called emission index in units of grams per kilogram of fuel. Smoke emissions were sampled with an AVL 415S smoke meter and expressed in units of filter smoke number (FSN). Combustion noise was estimated by an AVL 450S combustion noise meter and reported in decibels.

2.2. Test Fuels. Two fuels were chosen for this work. The baseline fuel was an ultralow-sulfur certification diesel fuel, which had a cetane number of 42. The comparison fuel, dieseline, was prepared by blending 80% of the baseline diesel and 20% commercial 93 octane, (RON + MON)/2, unleaded gasoline by volume. The major differences between dieseline and diesel are that dieseline has a lower cetane number and higher volatility. The detailed specifications of two fuels are listed in Table 2, and their distillation curves are illustrated in Figure 1.

2.3. Test Method. Throughout this experimental study, engine speed was maintained at 1500 rpm and injection pressure was held constant at 1000 bar. Injection timing was adjusted to keep combustion noise, as estimated by the AVL 450S combustion noise meter, below 90 dB. Exhaust pressure was adjusted to be 10 kPa higher than intake pressure to guarantee that sufficient exhaust gas could be inducted into the intake manifold. Coolant, oil, and EGR temperatures were maintained at 85 °C. The intake pressures investigated in the study were 1.0, 1.2, and 1.5 bar, and a 35% EGR rate was used at all three intake pressures. The intake oxygen concentrations at these three intake pressures were 15.5%, 16.3%, and 16.8%, respectively. A 40% EGR rate was also tested at 1.5 bar intake pressure with an intake oxygen concentration of 16.3%. The engine load range was analyzed as the test fuel, intake pressure, and EGR rate were changed. The light load boundary was dictated by combustion instability, indicated by COV of IMEP exceeding 5% or engine misfire occurring. On the other hand, the heavy load boundary was

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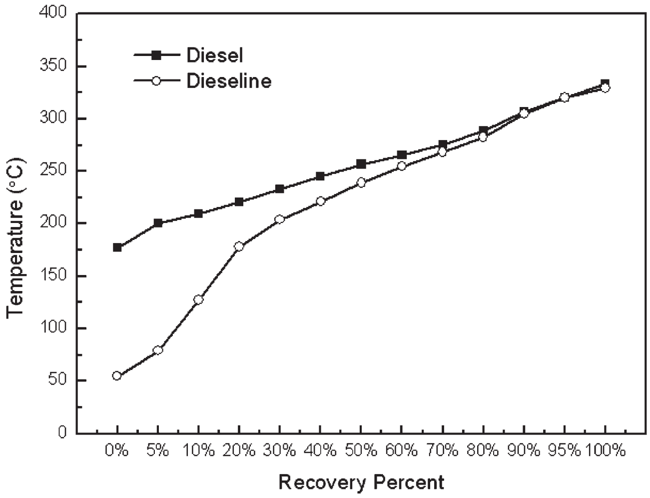

Figure 1. Distillation curves of diesel and dieseline fuels.

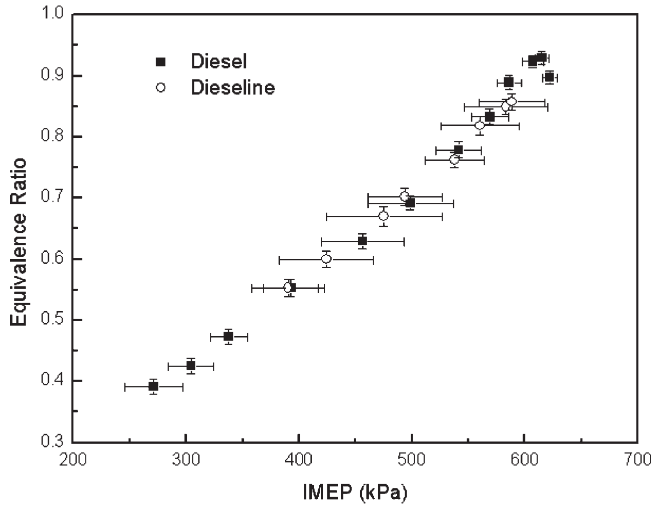

Figure 2. Equivalence ratio versus load for diesel and dieseline.

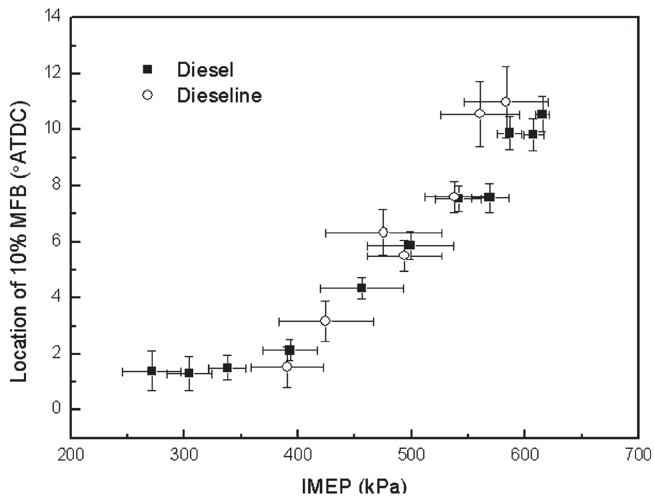

Figure 3. Matched combustion phasing for diesel and dieseline fuels in the LTC load test.

defined as either of the following conditions occurs (1) an increase in fuelling not raising IMEP and (2) smoke emissions reaching a value of 2 FSN.

2.4. Uncertainty Analysis. Experimental error was estimated by combining measurement uncertainty with repeatability uncertainty. Measurement uncertainty refers to instrument systematic uncertainty (i.e., error in the average reading of an instrument). Repeatability uncertainty is the variation in a measured variable represented by two standard deviations. These errors were combined by means of the root sum square method to represent a 95% confidence interval. For those parameters calculated from multiple individual variables, the sequential perturbation method was used for the error propagation.[20]

3. Results and Discussion

3.1. Potential of Dieseline to Reduce NO_x and Smoke Simultaneously at Naturally Aspirated Conditions. The first phase of this study investigates the performance of the two test fuels at the naturally aspirated LTC condition. The load was increased by extending the injection duration, and injection timing was retarded to maintain combustion noise below 90 dB. As can be seen in Figure 2, the equivalence ratios of both fuels approach stoichiometric at the heavy load limit. Combustion phasing strongly affects NO_x emissions, as illustrated in the study of Ickes et al.[21] Therefore, to eliminate the influence of combustion phasing, the 10% mass fraction burned (MFB) locations of the two test fuels were kept consistent, as shown in Figure 3.

One requirement of this LTC mode is that fuel injection ends before SOC in order to allow fuel and air mixing prior to combustion.[22] Figure 4 visually displays the relationship between the injection and combustion processes of both diesel and dieseline in this LTC mode at heavy load (equivalence ratio near 0.9). To yield the same combustion phasing, dieseline requires earlier injection timing. The lower cetane number of dieseline causes a longer ignition delay, so if the same mixing rate is assumed, dieseline has more time for mixing and produces a more well-mixed charge. Besides, the higher volatility of dieseline also increases the mixing rate and contributes to a more well-mixed charge.

Figure 5 compares the ignition delay of both fuels across the entire load sweep. In this study, ignition delay is defined as the duration of crank angle between the start of the injection event and the location of 10% mass fraction burned. The ignition delay for dieseline is longer than pure diesel at any given operational condition. The ignition delay of both fuels increases at light and heavy loads. Ignition delay should increase at light loads because the leaner fuel air mixture reduces the possibility of autoignition.[23] In contrast, the increased ignition delay at heavy load is probably due to the effect of retarded combustion phasing, as demonstrated by Kimura et al.[24]

Figure 6 illustrates the smoke and gaseous emissions trends versus load for both test fuels. Diesel fuel leads to a sharp increase of smoke emissions at the heavy load limit, where the global in-cylinder equivalence ratio is near stoichiometric and soot easily forms in local fuel-rich areas.

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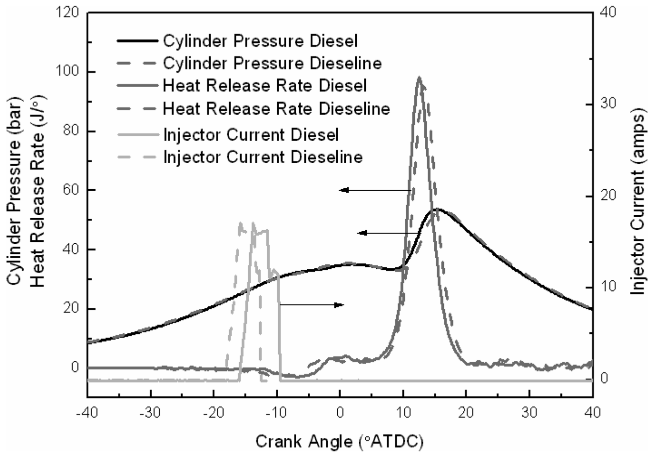

Figure 4. Cylinder pressure, heat release rate, and injector current versus crank angle for diesel and dieseline LTC condition at heavy load. Indicated mean effective pressure (IMEP), 585 kPa; 11° after top dead center (ATDC) 10% mean fraction burned (MFB); injection timing, 14.2° before top dead center (BTDC) (diesel), 16.4° BTDC (dieseline).

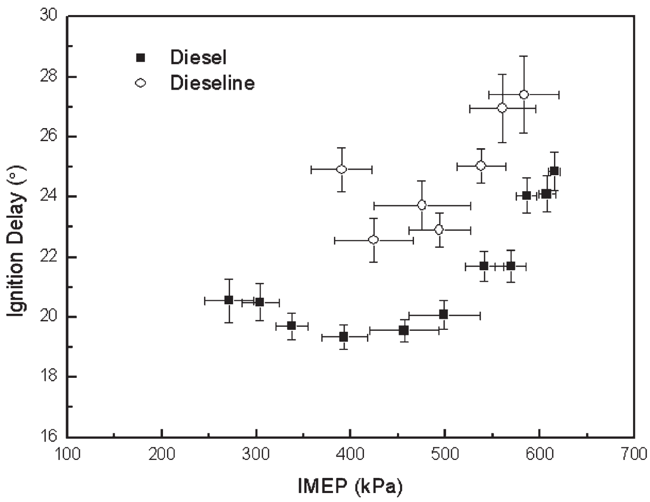

Figure 5. Ignition delay comparison between diesel and dieseline fuels with load sweep.

In contrast, the extended ignition delay of dieseline provides a longer time to form a well-mixed charge and alleviates the effect of high local equivalence ratios, greatly reducing the soot generation. In addition to the extended ignition delay, dieseline's higher volatility may also pose an effect on the smoke emissions. In Kitano et al.'s study, higher fuel volatility, for a fixed cetane number, reduced soot emissions.[25] As is shown in Figure 1, the distillation temperature curve for dieseline is lower than for diesel fuel. Therefore, the vaporization and atomization process of dieseline is quicker than diesel, so the mixing process is likely improved. As a result, with joint action of extended ignition delay and improved fuel volatility, nearly smokeless combustion can be achieved with dieseline throughout the load sweep.

The NO_x emissions trends of the two fuels are similar: a peak appears at a medium load. Moving to either lighter or heavier loads causes a decrease in NO_x emissions. At heavy loads, the quasi stoichiometric mixture reduces the possibility of local fuel-lean regions, and retarded combustion phasing results in lower combustion temperatures. At light loads, in spite of the overall fuel lean environment, decreased combustion temperatures reduce NO_x formation. It is notable that NO_x trends of the two fuels correlate well with ignition delay trends. The operational condition yielding peak NO_x emissions also has the shortest ignition delay, suggesting the mixture at this condition may not be as uniform as in other conditions. Nevertheless, since dieseline always has a longer ignition delay than diesel, even at the condition with the minimum ignition delay, the maximum NO_x from dieseline is still lower than that from the diesel fuel.

The incomplete combustion products of LTC, such as HC and CO emissions, show a "U curve" across the load sweep. At loads ranging from 400 to 600 kPa, the incomplete combustion products present a flat trend and remain at a low level. However, when IMEP approaches the heavy and light load limits, linked to the under-mixed and overmixed mechanisms respectively, incomplete combustion products increase sharply.[26] Because of the relatively narrow operational range of dieseline, the steep increase of these incomplete combustion products at heavy and light load limits is not observed. However, within the common operational range between the two test fuels, both fuels show comparable CO and HC emissions.

Figure 7 illustrates the operational range for the dieseline LTC mode at the naturally aspirated condition. Requirements of premixed LTC for this study are ultralow NO_x, smoke emissions and high combustion efficiency. The limits for allowable smoke and EI NO_x emissions in this development are defined as 0.5 FSN and 1.0 g/kg fuel, respectively. The smoke emissions standard in previous premixed diesel LTC research was defined as 2.0 FSN (visible smoke limit), based on the consideration that diesel particulate filters (DPF) are employed for further particulates reduction.[27,28] However, the smoke emissions standard in this study is limited to 0.5 FSN as done by Northrop et al.,[22] as to achieve an ultralow smoke LTC operation without strong reliance on DPF. The limit of NO_x emissions is estimated based on the requirements to meet Tier 2, Bin 5 emissions standards.[27] The defeat of the NO_x and smoke trade-off in prior LTC research is usually accompanied by the sacrifice of combustion efficiency due to the heavy use of EGR. Therefore, although not a limiting factor, combustion efficiency is also important and monitored within the dieseline LTC operational window. As shown in Figure 7, the combustion efficiency is always maintained at a high level (above 97%), but the engine operational range qualified for simultaneous ultralow NO_x and smoke emissions is very narrow (around 550 to 600 kPa IMEP). Smokeless combustion is achieved throughout the load sweep so smoke does not serve as a limiter in this LTC operation. However, the high NO_x emissions at low to mid loads limit the LTC operation extended to the light load end. The next sections explore two ways of expanding this narrow load window.

3.2. Effect of Intake Boost on Load Extension of Dieseline Premixed LTC. Since the dieseline LTC operational range at the naturally aspirated condition is narrow, as a second step,

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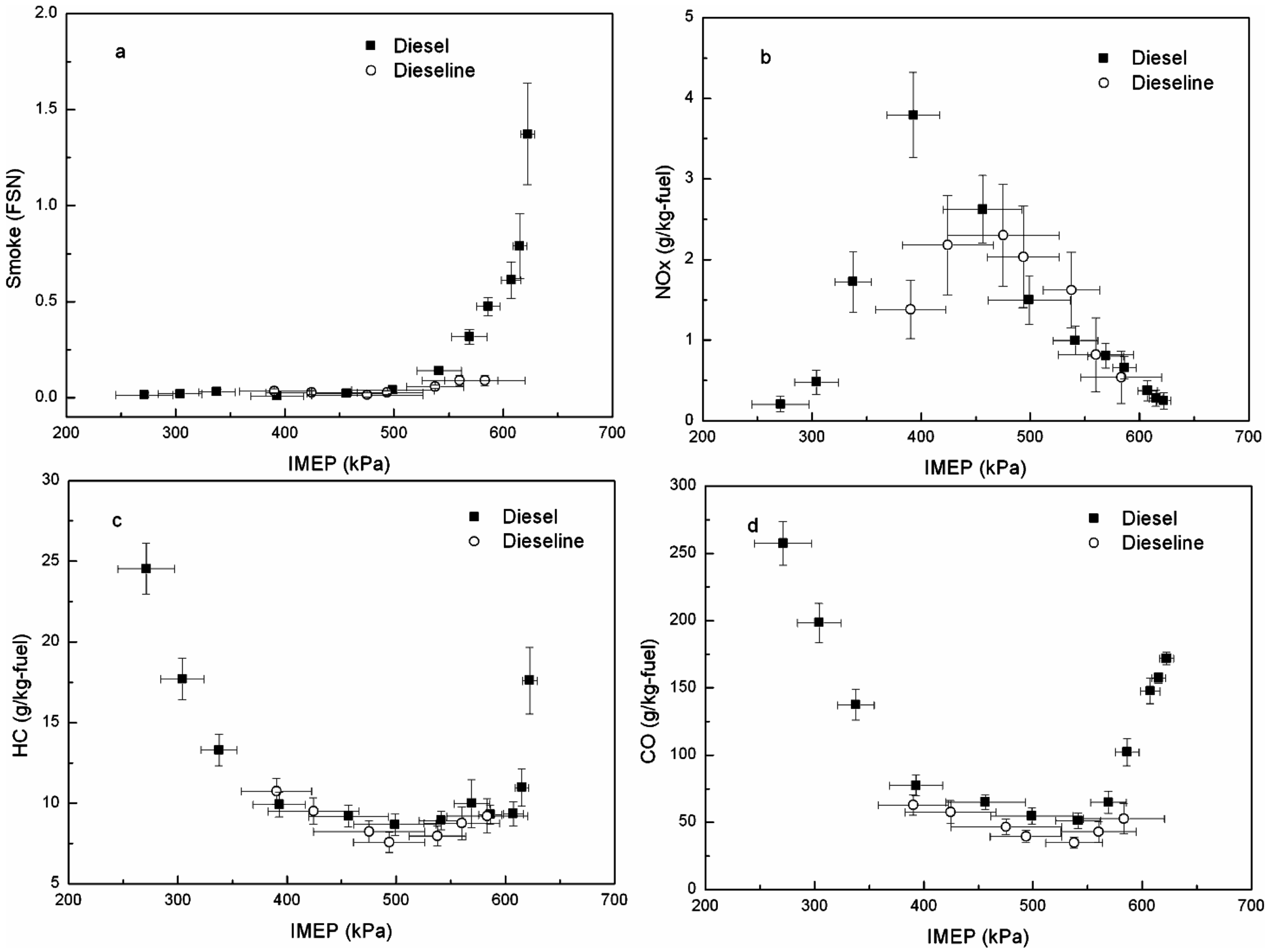

Figure 6. Emissions comparison between diesel and dieseline with load sweep: (a) smoke, (b) NOx, (c) HC, and (d) CO.

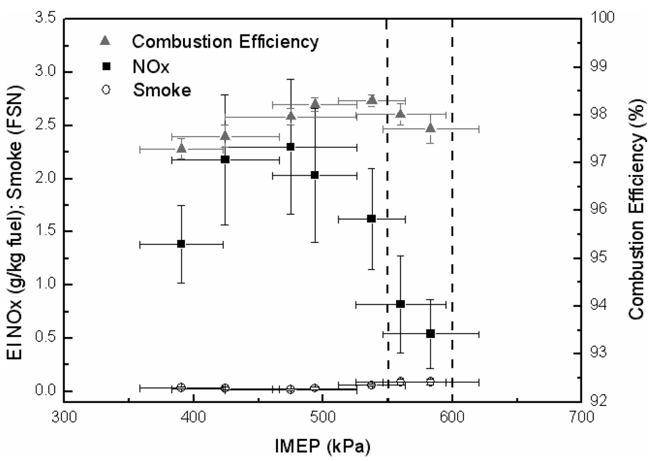

Figure 7. Applicable operation window of dieseline LTC at naturally aspirated conditions.

an intake boosting strategy is investigated as a method of extending the LTC mode to higher loads. With the same in-cylinder equivalence ratio, intake boost leads to higher volumetric efficiency, allowing increased fuelling to produce a higher load output. In this portion of the study, three different intake pressures (1.0, 1.2, and 1.5 bar) are selected to explore their effects on the LTC load extension. Increased intake density has been found to reduce ignition delay;[10] therefore, intake boost advances combustion phasing at a given injection timing. In the combustion strategy employed for this study, advanced combustion phasing increases the amount of heat release around TDC and thus leads to a higher pressure rise rate and combustion noise. To maintain the combustion noise below 90 dB, the injection timing is retarded with increased intake pressure. For the three intake pressures (1.0, 1.2, and 1.5 bar) investigated in this section, constant injection timings of 17° BTDC, 12° BTDC, and 7° BTDC were chosen, respectively. Figure 8 shows the effect of intake boost on load extension of the dieseline LTC mode. Increasing the intake pressure from 1.0 to 1.5 bar extends the heavy load boundary to about 900 kPa IMEP. The light load boundary also increases, however, because higher intake pressures at light loads cause leaner charges and higher combustion instability.

Figure 9 illustrates the emissions trends versus load for dieseline LTC at various intake pressures. In the smoke plot, as shown in Figure 9a, higher intake pressure (1.5 bar) extends smokeless LTC to around 800 kPa IMEP, much higher than under the naturally aspirated condition (around 600 kPa). At each load, intake boost results in a lower equivalence ratio and more oxygen, which is beneficial for the soot oxidation process. The increased intake pressure also promotes fuel atomization and air entrainment, thus

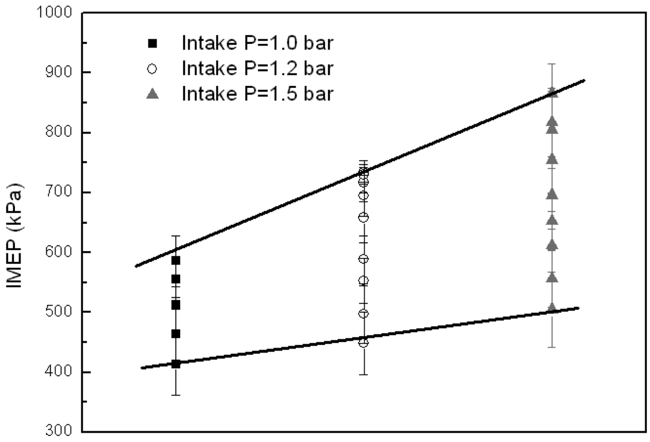

Figure 8. Effect of intake boost on load range of dieseline premixed LTC.

creating a well-mixed charge and reducing local fuel rich regions.[5,29,30]

Trends of NO_x emissions versus load at different intake pressures are shown in Figure 9b. In the mid to heavy load range (IMEP > 500 kPa), NO_x emissions gradually decrease with load for all intake pressures. The decreasing trend of NO_x emissions likely results from lower combustion temperatures. Although increased fuelling at high loads releases more heat, extended fuel injection duration retards combustion phasing away from TDC and the overall effect is reduced combustion temperatures. The effect of retarded combustion phasing overwhelms the effect of increased fuelling so cylinder temperatures and NO_x emissions are decreased, as demonstrated by Han et al.[31] At a given load, intake boost brings more air mass into the cylinder, resulting in slightly higher NO_x emissions. Similar to the results shown in Figure 6b, below 500 kPa IMEP the NO_x emissions at 1.0 bar intake pressure decrease as load decreases because fuel-lean mixing causes reduced combustion temperatures. At 1.2 bar intake pressure, however, NO_x emissions continue to increase as the load decreases below 500 kPa IMEP because combustion phasing continues to be advanced. If load is decreased enough, NO_x emissions under intake boost conditions eventually decrease as temperatures are sufficiently reduced due to the leaner charge. However, at these conditions, COV exceeds 5%, and the data are consequently not shown.

At most intake pressures except 1.5 bar intake pressure, HC and CO emissions trends follow a "U curve" with engine load, which agrees with results in colleagues' prior research.[28] Similar to what was observed in the naturally aspirated condition, incomplete combustion products increase at both light and heavy load limits. As discussed in the previous section, the increase of HC and CO emissions at heavy and light load limits is due to the under-mixing and overmixing mechanisms, respectively. However, the increase in HC emissions at high loads was not observed at 1.5 bar intake pressure because the smoke emissions have already reached the defined limit. Furthermore, the minimum HC emissions at the bottom of the "U curve" are lower, while the minimum CO emissions are higher at boosted conditions. Also, the minimum of HC and CO emissions curves are located at different IMEP. It may not be easy to find which variable, intake pressure or IMEP, leads to the different minimum HC and CO emissions. It is noted that the minimum of HC and CO emissions at increased intake pressures moves to higher IMEP, where more fuel is injected. Increased fuelling at higher intake pressure perhaps produces similar equivalence ratios, so a translation from IMEP to equivalence ratio may eliminate the effect of IMEP and better explain the difference in minimum HC and CO emissions with different intake pressures.

Plotting HC and CO emissions versus equivalence ratio can help explain the effect of intake boost on these incomplete combustion products. This is because the fuel and air proportion is fixed at a given equivalence ratio and intake boost serves as the primary variable. As shown in Figure 10, increased intake pressure causes a moderate decrease in HC emissions throughout the load sweep. Reduced liquid fuel dispersion[32] and shorter ignition delay due to the increased intake pressure are possible reasons because they decrease the possibility of fuel reaching crevice areas and reduce HC emissions stemming from cold boundary zones.

In contrast, CO emissions are found to slightly increase with higher intake pressure at high loads while do not show obvious difference at light loads. The trends of CO emissions with intake pressure at high loads are consistent with the results from Liu et al. who studied the effect of intake boost on the emissions of HCCI combustion with several gasoline-like fuels.[33] Bulk gas temperature is believed to affect CO emissions because CO formation is facilitated within a range of gas temperatures between 1000 K and 1400 K.[34] Trends of peak bulk cylinder temperatures, calculated based on the heat release model, versus equivalence ratio at different intake pressures are illustrated in Figure 11. Peak bulk cylinder temperatures for different intake pressures are similar at lower equivalence ratios (φ < 0.6), while at higher equivalence ratios (0.6 < φ < 0.9), intake boost reduces peak bulk cylinder temperatures. Lower gas temperatures from intake boost are probably the source of the increased CO emissions at high equivalence ratios. Since bulk gas temperatures at low equivalence ratios do not show much variance, CO emissions at low equivalence ratios are also similar for different intake pressures.

It is worth noting that as equivalence ratios approach stoichiometric, peak bulk cylinder temperatures decrease rapidly, as indicated by the data in Figure 11 for 1.2 bar intake pressure (the trend is not shown for 1.0 and 1.5 bar intake pressure conditions because the heavy load limits have been reached for these two intake pressures before the this phenomenon becomes visible). Generally, as the overall equivalence ratio is below stoichiometric, increased equivalence ratios by increased fuelling result in more heat release and higher peak bulk temperatures, as shown in Figure 11. However, further increasing fuelling beyond stoichiometric is unable to increase peak bulk temperatures, because the inadequate in-cylinder oxygen is unable to burn the additional fuel completely, leading to deteriorated combustion quality and reduced heat release. The increase in incomplete

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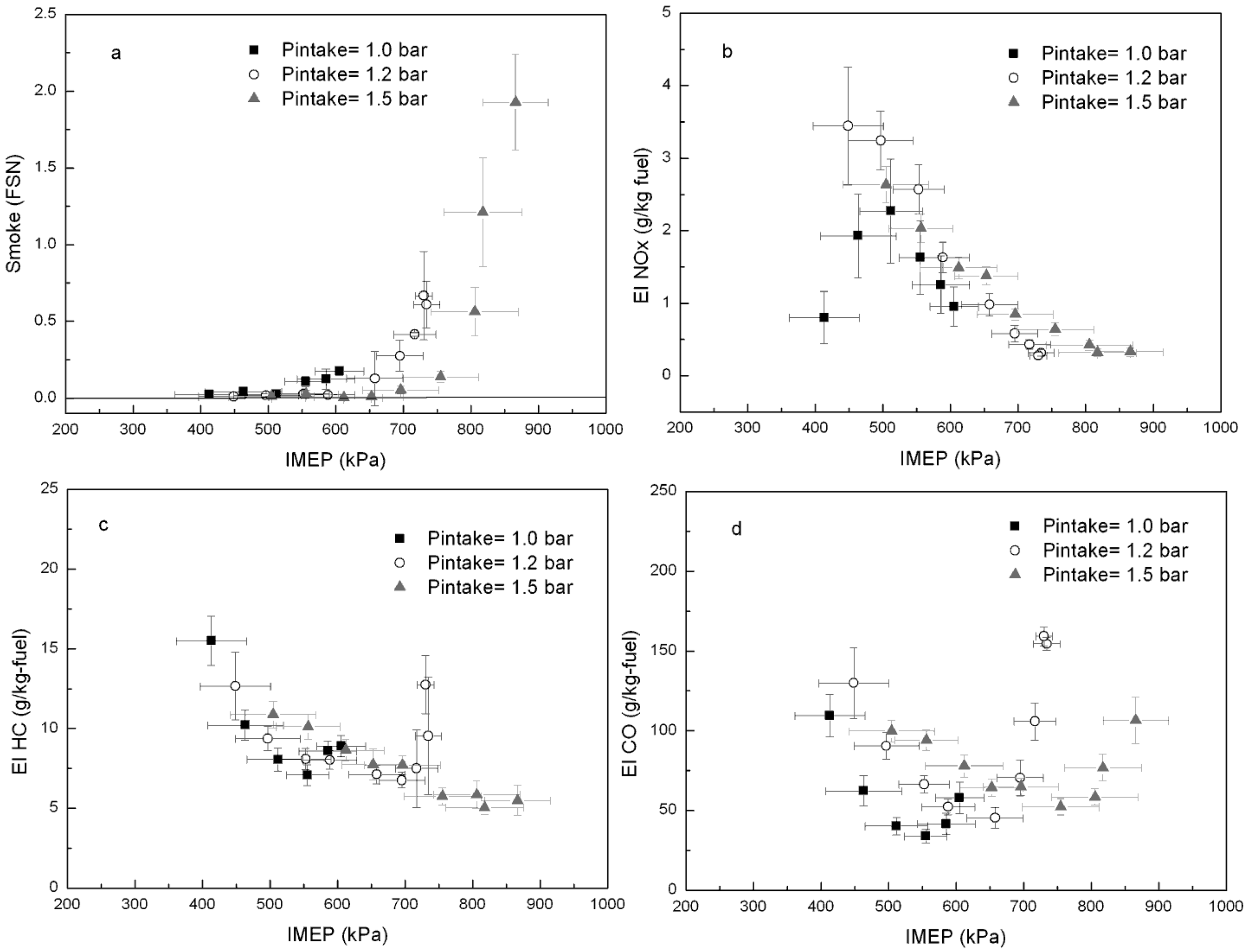

Figure 9. Relationship between emissions and load of dieseline LTC under different intake pressures: (a) smoke, (b) NOx, (c) HC, and (d) CO.

combustion products at this area also demonstrates the lower combustion quality, as shown in Figure 10.

The trends of combustion efficiency versus equivalence ratio at these intake pressures shown in Figure 12 are to the inverse of the trends of HC and CO emissions because combustion efficiency is calculated based on the concentrations of incomplete combustion products. Combustion efficiency decreases at both the light load limit and the heavy load limit, where HC and CO emissions rapidly increase. However, intake boost demonstrates little effect on combustion efficiency at a fixed equivalence ratio, possibly because the opposing trends of HC and CO emissions mitigate any change in combustion efficiency. As intake pressure increases at a given equivalence ratio, HC emissions increase while CO emissions decrease, and the total chemical energy carried by the incomplete combustion products in the exhaust gases does not vary greatly.

The LTC operational range with 1.5 bar intake pressure is shown in Figure 13. The criteria to be satisfied are the same as in the previous section. Dieseline LTC can be applied to a broader range of engine loads at 1.5 bar intake pressure than under the naturally aspirated condition, from 680 to 800 kPa IMEP. Both the light load and heavy load limits move higher compared to the naturally aspirated condition. Increased volumetric efficiency brought by higher intake pressure allows for more air and fuel at a given equivalence ratio. As a result, the stoichiometric condition is brought to a higher load condition and the smokeless LTC operation is also extended to a higher load level. Meanwhile, higher intake pressure produces higher NO_x emissions at light to mid loads, so the light load limit of LTC also moves toward higher load. The combustion efficiency within the defined operational window is still sufficiently high, although a decrease in combustion efficiency is observed near the heavy and light load limits.

3.3. Higher EGR Rate to Control NO_x Emissions at Light Load. Intake boost effectively extends the allowable dieseline LTC operation to high loads but plays a negative role in its extension to light loads by increasing NO_x emissions. This section aims to reduce NO_x emissions at light to mid loads to below 1.0 g/kg fuel, thus applying the LTC operation with simultaneous ultralow NO_x and smoke emissions to a broader range. The effectiveness of EGR in reducing NO_x emissions has been widely reported,[35,36] so a moderately increased EGR rate (40%) is utilized. Throughout the entire load sweep with 40% EGR, injection timing is held constant at 8° BTDC.

The operational window of dieseline LTC with 40% EGR and 1.5 bar intake pressure is shown in Figure 14. The

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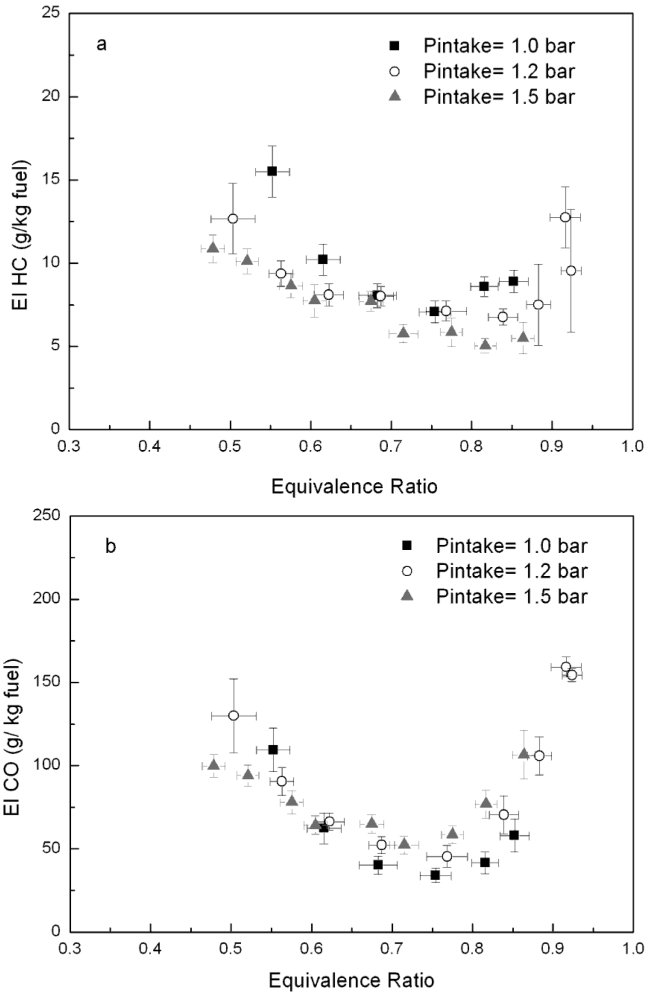

Figure 10. HC (a) and CO (b) emissions versus equivalence ratio under different intake pressures for dieseline.

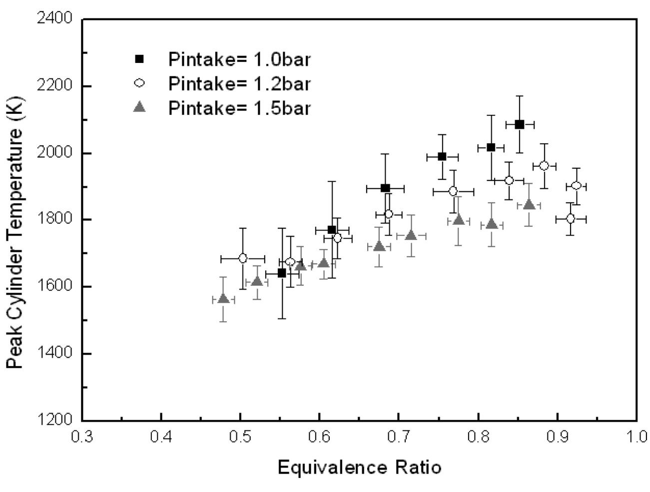

Figure 11. In-cylinder peak bulk cylinder temperature versus equivalence ratio at different intake pressures for dieseline.

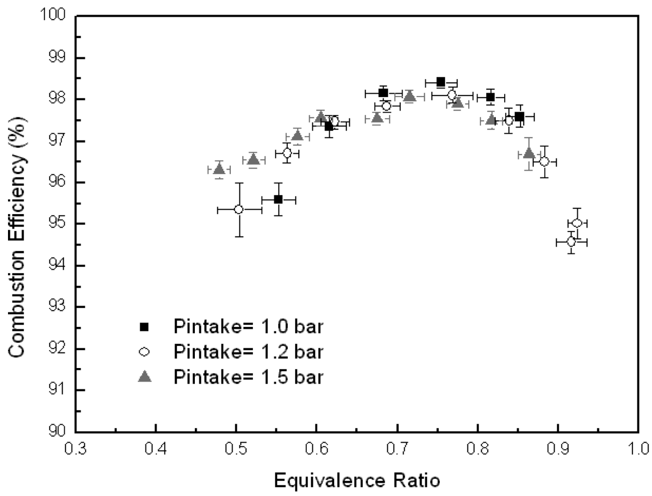

Figure 12. Combustion efficiency versus equivalence ratio of dieseline LTC at various intake pressures.

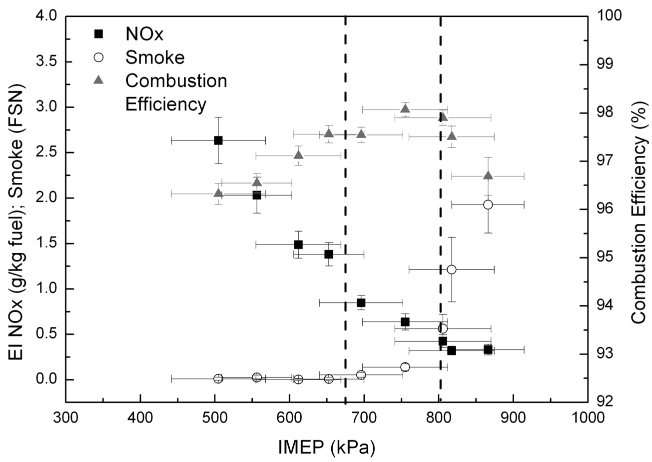

Figure 13. Operational window of dieseline LTC at 1.5 bar intake pressure.

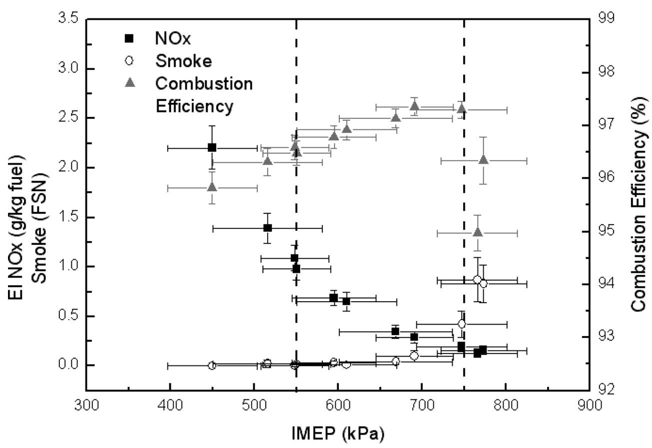

Figure 14. Operational window of dieseline LTC at 40% EGR and 1.5 bar intake pressure.

introduction of a higher EGR rate reduces combustion rate and thus combustion temperatures; therefore, NO_x emissions were lowered throughout the load sweep with the most apparent effect at light load conditions. The light load boundary of dieseline LTC operation with simultaneous ultralow NO_x and smoke emissions is extended to 550 kPa. On the other hand, the utilization of higher EGR rates reduces the amount of fresh air into the cylinder. The load limit at the heavy end with 40% EGR decreases compared to the 35% EGR condition because of insufficient air for complete combustion. Smoke emissions show a steep increase around 750 kPa IMEP, 50 kPa lower than the turning point with the 35% EGR condition, shown in Figure 9a.

Nevertheless, the overall effect of 40% EGR brings about a broader LTC operational window than 35% EGR, because the NO_x emissions at light loads are greatly restricted. The combustion efficiency in the operational window shown in Figure 14 slightly decreases compared to the 35% EGR condition, as shown in Figure 13, but it is still maintained at a high level (higher than 96%), which means the benefits of high combustion efficiency in this dieseline LTC strategy are maintained.

4. Conclusions

Dieseline, a blend of diesel and gasoline, was used in a premixed LTC mode featuring light use of EGR and high combustion efficiency in this study. The potential of simultaneous reduction of NO_x and smoke emissions by this dieseline LTC mode and the extension of the low-emission and high-efficiency operational range were investigated.

The longer ignition delay and higher volatility of dieseline produce a more homogeneous fuel air mixture before the start of combustion. The more premixed charge extends smokeless combustion to higher loads and produces lower NO_x emissions across the load sweep, with relatively high combustion efficiency maintained by light use of EGR.

Intake boost shifts the entire dieseline LTC operation range to higher loads. Increased air quantity as a result of intake boost allows more fuelling so the high load limit is increased and the light load limit also moves upward due to the increased NO_x emissions at light to mid loads. Slightly increased EGR reduces NO_x emissions, extending dieseline LTC to lighter loads without heavily deteriorated combustion efficiency.

Within the context of this study, intake boost poses opposing effects on HC and CO emissions. Given an equivalence ratio, increased intake pressure results in reduced HC emissions because reduced ignition delay with intake boost restricts the possibility of unburned fuel to enter the cold boundary zones. On the contrary, CO emissions increase with intake pressure at high equivalence ratios because the decreased bulk cylinder temperatures contribute to CO formation.

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Nomenclature

LTC = low temperature combustion
NO_x = $NO + NO_2$
PM = particulate matter
HC = hydrocarbon
CO = carbon monoxide
MFB = mass fraction burned
IMEP = indicated mean effective pressure
EGR = exhaust gas recirculation
CN = cetane number
FSN = filter smoke number
COV = coefficient of variance
HCCI = homogeneous charge compression ignition
ATDC = after top dead center
BTDC = before top dead center
SOC = start of combustion