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Investigation of operating range in a methanol fumigated diesel engine

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HIGHLIGHTS

- Operating range is identified on a methanol fumigated diesel engine.
- DMDF range is restricted to partial burn, misfire, roar combustion and knock.
- Systematic analysis of combustion characteristics on each bound of the range.
- DMDF mode worsens the BTE at low load while boosting it at medium and high load.

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ABSTRACT

An experimental study was conducted to investigate the operating range and combustion characteristics in a methanol fumigated diesel engine. The test engine was a six-cylinder, turbocharged direct injection engine with methanol injected into the intake manifold of each cylinder. The experimental results showed that the viable diesel methanol dual fuel (DMDF) operating range in terms of load and methanol substitution percent (MSP) was achieved over a load range from 6% to 100%. The operating range was restricted by four bounds: partial burning, misfire, roar combustion and knock. The lower bound of the operating range was the partial burn bound, which occurred under very low load conditions with high MSP. As the load increased to medium load, MSP reached its maximum value of about 76%, and the onset of misfire provided the right bound for normal operation. At medium to high load, maximum MSP began to decrease. DMDF combustion with excessive MSP was extremely loud with high pressure rise rate, which defined the roar combustion bound. As it increased to nearly full load, measured pressure traces in-cylinder showed strong acoustic oscillations. The appearance of knock provided the upper bound of the operating range. In general, as the load increased, the characters of the combustion changed from partial burn to misfire to roar combustion and to knocking. The range between these four bounds and the neat diesel combustion bound constituted the viable operating range. Over the viable operating range, DMDF combustion worsened the brake thermal efficiency (BTE) at light load while boosted it at medium and high load.

1. Introduction

Compared with spark-ignition engine, researchers are more interested in compression–ignition (CI) engine due to its better fuel economy with high compression ratio and no throttling loss. However, the conventional CI engine sustains with high nitrogen oxides (NO_x) and particulate matter (PM) emissions. In addition, improving the fuel efficiency is always a goal because of the direct connection to carbon dioxide (CO_2) emissions and crude oil usage. Hence, the heavy-duty CI diesel engine has been a topic of research over the last two decades. Moreover, the sources of fossil fuel are dwindling with time's going, which causes the price of petroleum oil becoming higher on a daily basis. These all pose challenges to the availability of fossil fuel. Under these circumstances, the demand of alternative fuels is increasing as a substitute of conventional fossil fuel in transportation sector to address energy security issues. Among the alternative fuels, methanol and ethanol have received considerable attention as suitable diesel fuel replacement. In particular, methanol is readily available from the conversion of

Abbreviations: CI, compression–ignition; NO_x, nitrogen oxides; PM, particulate matter; CO_2, carbon dioxide; DMDF, diesel methanol dual fuel; MSP, methanol substitution percent; BTE, brake thermal efficiency; DMCC, diesel methanol compound combustion; ECU, electronic control unit; CA, crank angle; ATDC, after top dead center; BTDC, before top dead center; AHRR, apparent heat release rate; PPRR, peak pressure rise rate; PCP, peak cylinder pressure; COV, coefficient of variation; IMEP, indicated mean effective pressure.

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biomass, coal and natural gas [1]. The storage, transportation, distribution, and application of methanol are similar to those of traditional gasoline and diesel fuels as a liquid. Therefore, the substitution of diesel fuel with methanol is of significant economic and environmental importance in countries like China which has large coal reserve, and in particular, huge amount of coke-oven gas resources.

The foremost drawback for the utilization of methanol in diesel engines is probably its low cetane number, which, depending on the measurement method, typically ranges from 2 to 12 [2]. The much high latent heat of vaporization also weakens its auto-ignition ability [3]. In spite of these drawbacks, methanol has been used in diesel engines primarily in one of the following ways: blends, neat methanol and dual fuel. Recently, the team of Huang investigated on the emissions and combustion characteristics of a single-cylinder diesel engine running on a stabilized diesel–methanol mixture with up to 18% by weight of methanol. And smoke emission decreases with the increase of the oxygen mass fraction in the blends without increasing the NO_x emission [4,5]. However the blending of methanol with diesel fuel requires additives for stabilizing the mixed fuel and there is a limitation on the amount of methanol that can be premixed with diesel fuel for stable operation [6]. Actually, the diesel–methanol blending has been made possible only by the addition of surfactants in order to form micro-emulsions, rather than real solutions [7]. Moreover, the use of neat methanol in diesel engines usually requires the addition of relatively large amount of expensive ignition-improving compounds and very high compression ratios [8].

In this regard, dual fuel combustion has received renewed interest due to its adaptability for alternative fuels and due to its excellent performance and ultra-low emissions compared to conventional diesel combustion. Dual fuel combustion is an approach that utilizes a high cetane number fuel such as diesel, biodiesel to ignite a low cetane number fuel such as alcohol [9]. Separate fuels direct injection [10], dual fuel injector [11] and fumigation [12] are used for diesel methanol dual fuel (DMDF) operation. However, the use of two separate fuels injection system is more complicated because it involves significant engine modifications as the methanol injector is placed at the top of combustion chamber. Using only one injector to inject two fuels in an engine is only reported by the system developed by Westport Corp., called HPDI [11]. In this regard, fumigation is favored currently, because it requires a minimum of modification to the engine since methanol injectors is placed at the intake manifold. However methanol fumigation is unfavorable for cold start and low load operation. Based on the method of fumigation, Yao et al. [13,14] developed a diesel/methanol compound combustion (DMCC) system. Under DMCC mode, at cold start and low speed conditions, the engine operates on diesel alone to ensure cold starting capability and to avoid aldehydes production under these conditions. At medium to high loads, the engine operates on diesel methanol dual fuel (DMDF) mode, of which methanol is fumigated into intake manifold and the homogeneous air/methanol mixture is ignited by the diesel directly injected [14]. The advantage of DMCC system is that there is no cold start difficulty when the engine operates at dual fuel mode. Furthermore, in case of lacking methanol fuel supply, this engine still runs according to the diesel cycle by switching from dual fuel mode to neat diesel mode [15]. Unlike natural gas dual fuel engine, there is no simultaneous reduction of air supply [16]. Hence, the compression pressure and the mean effective pressure of the engine are not decreased and even boosted with methanol fumigation.

Many previous investigations were performed with a DMCC system. Recently, using a 4-cylinder direct-injection diesel engine with fumigation methanol, Cheng et al. [17] showed that the concentration of nitrogen oxides is significantly reduced except under full load conditions. There is also a reduction in the smoke opacity and the particulate matter mass concentration. With the same engine setup and operating conditions, Zhang et al. [18] found that under low engine loads, the brake thermal efficiency (BTE) decreases with the increase of fumigation methanol; but under high loads, it is slightly boosted with the increase of fumigation methanol. Using the same engine with the present study, Geng et al. [19] observed that the mass and number concentrations of particulate matter significantly decrease at low and medium loads, while they increase when the tested engine operated at high loads. Li et al. [20] developed a multi-dimensional model to investigate the combustion and emission characteristics of a fumigated methanol and diesel reactivity controlled compression ignition engine. They found that methanol addition is an effective way to achieve the efficient and clean combustion and all the emissions are reduced with moderate methanol addition.

The above brief survey of the relevant literatures shows that, though many studies have examined DMDF combustion on single-cylinder naturally aspirated light-duty engines, few researchers have reported DMDF combustion results from multi-cylinder turbocharged heavy duty engines. Furthermore, hardly any researchers have focused on the operating range and combustion characteristics of DMDF combustion. Based on the authors' previous studies on DMDF engines, methanol substitution percent (MSP) and brake thermal efficiency (BTE) of DMDF combustion depended greatly on engine load. In order to further understand the effect of MSP on a DMDF engine, this work concentrated on establishing the operating range with regard to MSP and engine load on a methanol fumigated six-cylinder turbocharged heavy duty diesel engine and investigated the combustion characteristics of conditions at each range bound.

2. Experimental apparatus and methods

2.1. Test engine and fuels

The original engine was an in-line six-cylinder, direct injection, turbocharged diesel engine with an electronically controlled unit injection pump. Technical specifications of the engine are listed in Table 1. Fig. 1 shows the schematic of the engine layout. The engine was modified to run on DMDF mode with introducing the methanol fuel by 6 electronically controlled methanol injectors fixed at the intake manifold of each cylinder. The methanol was injected at a pressure of 0.4 MPa and the mass of methanol injected was controlled by an electronic control unit (ECU) developed by ourselves. The engine was coupled to an electronically controlled hydraulic dynamometer. The engine speed and torque could be controlled by the EMC2020 heavy diesel engine test system, which allowed to changing engine speed and load as required.

The pressure trace in-cylinder was measured with a Kistler 6025C piezoelectric pressure transducer in series with an AVL 612 IndiSmart combustion analyzer, which had a signal amplifier for piezo inputs. For each engine operating point, 100 consecutive cycles of cylinder pressure data were recorded. The collected cycles

Table 1
Parameters of the engine.

Parameters	Value
Number of cylinders	Six in-line
Displacement (L)	7.14
Bore × stoke (mm)	130 × 108
Compression ratio	18:1
Number of nozzle holes	6
Nozzle hole diameter (mm)	0.235
IVC(° ATDC)	−124.5
EVO(° ATDC)	79.6

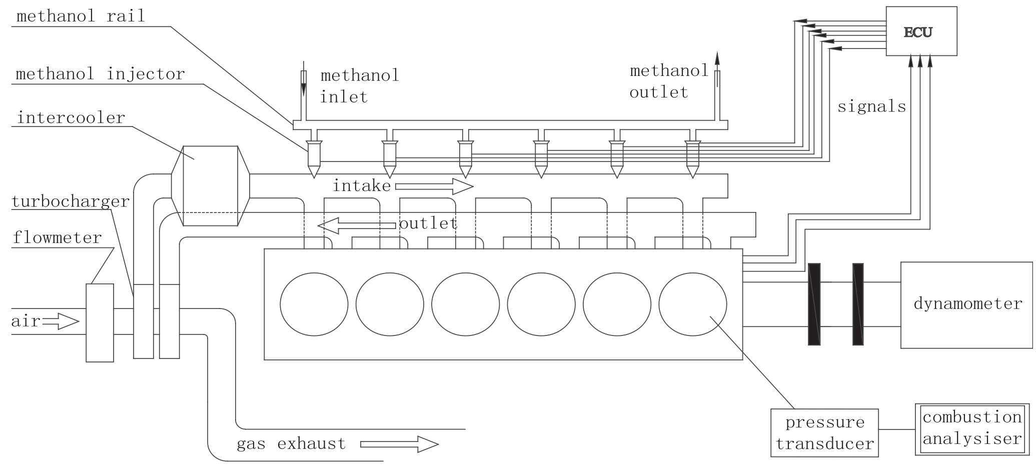

Fig. 1. Schematic of the engine layout.

were ensemble averaged to yield a representative cylinder pressure trace, which was used to calculate the apparent heat release rate (AHRR) by the AVL 612 IndiSmart combustion analyzer. AHRR curve was captured by use of a flexible basis of B-splines whose knots density was automatically selected from the data. A shaft encoder with 720 pulses per revolution was used to send engine speed, which supplied a resolution of 0.5° crank angle (CA). The diesel injection timing was controlled by the ECU of original engine and was unable to be changed. The methanol injection system was independent of the diesel ECU. Although the diesel injection timing controlled by the original engine ECU was unable to be changed, injection timing varied with the change of accelerator positions. The accelerator positions pulls back as the MSP increased, thus injection timing at diesel mode and DMDF mode was different. Special conditions at each bound and its injection timing are shown in Table 2. Diesel and methanol fuel consumption were independently measured gravimetrically using two coriolis meters with a precision of 0.1 g. Engine coolant temperature, exhaust temperature, and inlet air temperature were recorded using *K* type thermocouples with an accuracy of 1 °C. The diesel used in the test was commercial 0# diesel fuel, while the methanol used was industrial grade with a purity of 99.9%. The main fuel properties of diesel and methanol are given in Table 3.

2.2. Engine operating method and test conditions

Actually, the primary objective of the current research was to quantify the limits of methanol substitution percent over a wide range of engine loads (loads from 6% to 100%). Consequently, for the dual fuel experiments, the engine started with 6% of full load with neat diesel, then an effort was made to keep the pilot amount of diesel fuel constant. The power output of the engine was adjusted by the amount of methanol supplied through the intake port until the DMDF engine reached its full load or operating limits, such as knock and misfire. Then same procedure was conducted at different initial loads, ranging from 11% to 88% of the full load with an 11% increment. At each test point, the period of operation was maintained for about 4 min long, and experimental data was the weighted average of the data stream. All experiments were performed at a constant engine speed of 1400 r/min throughout the test, as 1400 r/min was the speed at which the maximum torque was achieved. Peak pressure rise rate (PPRR) of 1.5 MPa/°CA and peak cylinder pressure (PCP) of 15 MPa were set separately as the upper limits of roar combustion and engine knock, which are the tolerable limits for the engine designed. Limit of PPRR is controlled due to engine noises, and limit of PCP is controlled due to mechanical stress. Misfire bound was defined as misfire cycles occurred on combustion analyzer. If the engine output torque stopped increasing while fumigated methanol was still added on, we considered that the DMDF combustion had reached its partial burn limit because that the excess methanol was emitted into the exhaust pipe. Based on the engine load and the mass consumption rates of diesel and methanol, the BTE and MSP can be calculated by using Eqs. (1) and (2) respectively [17].

Brake thermal efficiency (BTE):

$$\mathrm{BTE} = \frac{P_b}{(q_{\mathrm{m'd}} \times Q_{\mathrm{LHV'd}}) + (q_{\mathrm{m'm}} \times Q_{\mathrm{LHV'm}})} \times 100\% \qquad (1)$$

where P_b is the brake power, kW; $q_{\mathrm{m'd}}$ the mass consumption rate of diesel fuel, kg/s; $q_{\mathrm{m'm}}$ the mass consumption rate of methanol, kg/s; $Q_{\mathrm{LHV'd}}$ the lower heating value of diesel fuel, kJ/kg; and $Q_{\mathrm{LHV'm}}$ is the lower heating value of methanol, kJ/kg.

Methanol substitution percent (MSP):

$$\mathrm{MSP} = \frac{q_{\mathrm{m'dd}} - q_{\mathrm{m'dm}}}{q_{\mathrm{m'dd}}} \qquad (2)$$

where $q_{\mathrm{m'dd}}$ is the diesel fuel consumption rate in neat diesel operation mode; $q_{\mathrm{m'dm}}$ is the diesel fuel consumption rate in DMDF mode.

Table 2
Engine test conditions and injection timing.

	Load (%)	MSP (%)	Injection timing	
			Diesel	DMDF
Partial burn	20.04	69.18	0.7°CA ATDC	5.8°CA BTDC
Misfire	35.64	75.56	–	1.7°CA ATDC
Roar combustion	46.48	66.80	6.1°CA BTDC	3.7°CA ATDC
Knock	85.03	22.01	4.7°CA ATDC	4.7°CA ATDC

Table 3
Properties of diesel and methanol.

Properties	Diesel	Methanol
Molecular formula	$C_{10}H_{22}$ to $C_{15}H_{32}$	CH_3OH
Molecular weight (g/mol)	190–220	32
Cetane number	50	<5
Low heating value (MJ/kg)	42.5	19.7
Heat of evaporation (kJ/kg)	260	1178
Stoichiometric air–fuel ratio	14.7	6.45
Auto-ignition temperature (°C)	316	464

3. Results and discussion

3.1. Operating range of DMDF combustion

The investigation of the viable operating range of DMDF combustion was carried out with a range of methanol substitution percent (MSP) and engine loads. A set of data points were recorded in a range between the maximum and minimum MSP without violating the limits of measured engine knock and misfire. The MSP-load matrix results filled with the impact of BTE at 1400 r/min are shown in Fig. 2. The maximum MSP greatly depends on the engine load, reaching its peak of about 76% at 43.5% engine load under this particular speed. Thus the attainable DMDF range is limited by four bounds: partial burn, misfire, roar combustion and knock.

At low engine load, especially below 20%, the increase of MSP results in late and incomplete combustion, which established the partial burn bound. When the engine load is between 20% and 45% of full load and maximum MSP is over 70%, the misfire bound is reached associated with alternating misfire and knock cycle after cycle. At medium to high load, there is a sharp decrease of maximum MSP with the increase of engine load because of roar combustion and measured engine knock. In general, as the load increases, along maximum MSP bound the characters of the combustion change from partial burn to misfire to roar combustion and to knocking. The range between these four bounds and the neat diesel combustion bound is the viable operating range for DMDF combustion.

Fig. 2 also shows the brake thermal efficiency contours of the DMDF engine. It is observed that for all the MSP (containing neat diesel operation, 0 MSP), the BTE is improved with increasing engine load. But as the MSP increases at a given engine load, the change in BTE is not uniform. Compared with neat diesel combustion, the BTE of DMDF is reduced at low engine load when fumigation methanol is increased. At low engine load, the cooling effect, together with the leaner air/methanol mixture, result in poorer combustion and thus reduce the BTE with more fumigation methanol. At medium load, the BTE slightly drops at first and then increases with the increment of MSP. The homogeneous air/methanol mixture burns with a higher rapid rate of heat release, which might increase the BTE at high MSP. At high engine load, the higher in-cylinder gas temperature, the longer the ignition delay and the combustion of the diesel fuel in a richer air/methanol mixture resulted in the improvement of BTE.

3.2. Partial burn

As shown in Fig. 2, the lower bound of the operating range is the partial burn bound. As the engine load increases from 6% to 20%, the maximum MSP increases readily from 0% to about 69% under this particular speed. Under these conditions DMDF combustion gives lower BTE compared to neat diesel mode. Cylinder pressure and AHRR results along partial burn bound are presented in Fig. 3. In Fig. 3, the compression lines of both modes remain the same but the expansion line of DMDF combustion is much higher than that of neat diesel combustion. Only the premixed combustion is observed at the neat diesel combustion, while the DMDF combustion contains a long tail of late combustion caused by flame propagation. Meanwhile DMDF mode showed a longer ignition delay and a slightly lower maximum premixed combustion peak with respect to the corresponding one in neat diesel mode. As a result, the late combustion and long combustion duration of DMDF decrease the efficiency of work extraction since the combustion is in the expansion stroke. Though fumigated methanol is still added on, the engine output torque stopped increasing.

Thus the combustion process is highly worsened at lower load in DMDF mode, and will aggravate further with the increase of MSP. At 30% MSP, the combustion starts about 7°CA earlier than 69% MSP, and there is no long tail of late combustion. First of all, since methanol has higher heat of vaporization, which results in cooling effect in the intake process and the compression stroke, the temperature of intake air at 69% MSP is much lower than neat diesel mode and 30% MSP in DMDF mode. Second, we should keep in mind that methanol has higher octane ratings which does a negative effect on diesel ignition [21,22]. The lean mixture of methanol and air also prolongs the combustion duration. In addition, the mass and duration of injected diesel is too short and the energy given by the diesel injection is not enough to onset multiple propagation flames. Finally, at high MSP conditions some methanol–air

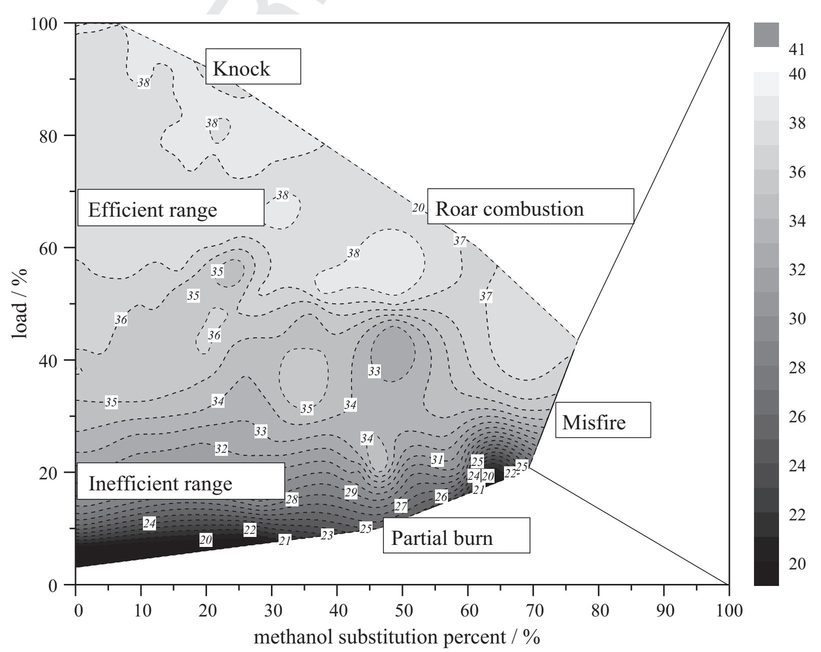

Fig. 2. Experimental operating range and BTE contours for the DMDF engine.

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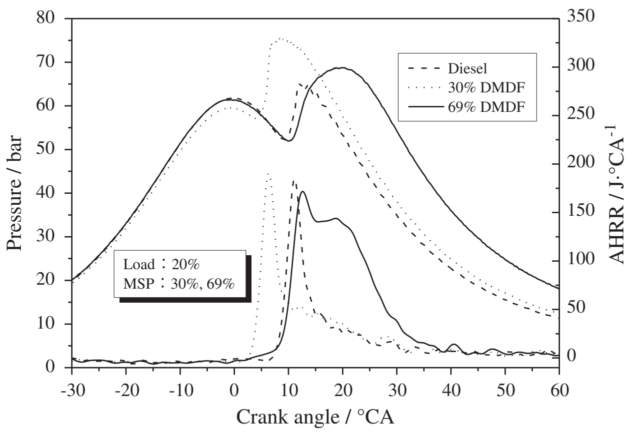

Fig. 3. Comparison of combustion characteristics between diesel and DMDF mode at partial burn condition.

mixture may escape combustion during this phase due to valve overlap, flame quenching on the walls or the effects of crevices. Concerning the above reasons, the worsened combustion process results in the reduction of BTE from 25% to 22%.

3.3. Misfire

As engine load rises from 20% to 44% and MSP increases to about 76%, the misfire bound is reached. With the further increase of MSP and further decrease of pilot diesel injection, the phase of DMDF combustion starts to become unstable. Finally misfire occurs in some cycles shown by Fig. 4. Fig. 4 shows the variation of cylinder pressure with ten consecutive cycles at misfire condition. It is clear that roar combustion and misfire occur alternately cycle after cycle. The misfire combustion in alternative cycles leads to more unburned charge, characterized by high carbon monoxide and unburned hydrocarbons, and the increase in coefficient of variation (COV) of net indicated mean effective pressure (IMEP). Misfire bound is a further step of partial burn and roar combustion, as the bounds between misfire and roar combustion or between misfire and partial burn are not clear. At misfire condition, the small amount of diesel injected may not be compressed to ignite due to the large plenty of methanol mixture in combustion chamber and then misfire cycle occurs. Then in the cycle after misfire the unburned diesel and methanol from the last misfire cycle together with the new injected fuel formed high reactivity premixed mixtures, which is more easily to ignite and more likely to cause knock. Subsequent cycles show an alternating occurrence of HCCI-like knock and misfire throughout the relatively short experiment. The engine speed varies over 300 r/min and combustion variation is more than 10% COV of net IMEP. Clearly, under 40% of full load, this would impose a maximum MSP of about 65% in order to remain within tolerable limits under these conditions.

3.4. Roar combustion

As the load continues to increase, maximum MSP is reduced in order to avoid roar combustion or combustion noise. Along the roar combustion bound, maximum MSP is limited by its PPRR of approximately 1.5 MPa/°CA. Fig. 5 illustrates the cylinder pressure and AHRR under a roar combustion condition. First it can be seen, as expected, that the compression pressure of DMDF mode decreases significantly compared with the neat diesel combustion, which is mainly due to the higher heat of vaporization of methanol fuel. The results also show that at DMDF mode almost all the diesel fuel and the entrained air–methanol mixture are burned in premixed stage with a sharp peak on the AHRR trend. As the injection timing of pilot diesel, as is shown in Fig. 5, is much more advanced compared with that of neat diesel mode, start of ignition under DMDF mode is much more advanced. Increasing the methanol fuel at constant speed and load results in an increase in the mass of premixed fuel admitted to the engine and a decrease of the pilot diesel fuel mass injected. This increase in the mass of methanol then causes an increase in the ignition delay period of pilot diesel which then auto-ignites and starts burning the premixed methanol fuel at a higher rate of pressure rise. This is also shown by Nielsen et al. [16] on dual fuel engine where natural gas is admitted in the inlet air manifold. Furthermore, compared with misfire bound, the shorter diesel injection event is able to provide enough energy to lead multi-site combustion. Meanwhile, compared with lower load, the amount of methanol is large enough to form a richer homogeneous mixture. In addition, because of the enhanced charge temperature, the combustion is phased to near thermodynamically ideal conditions. The rapid burn of DMDF mode also shortens the combustion duration and has a positive effect on BTE.

3.5. Knock

As the engine load continues to rise, engine knock occurs, which is limited by its PCP of 15 MPa, and the maximum MSP continues

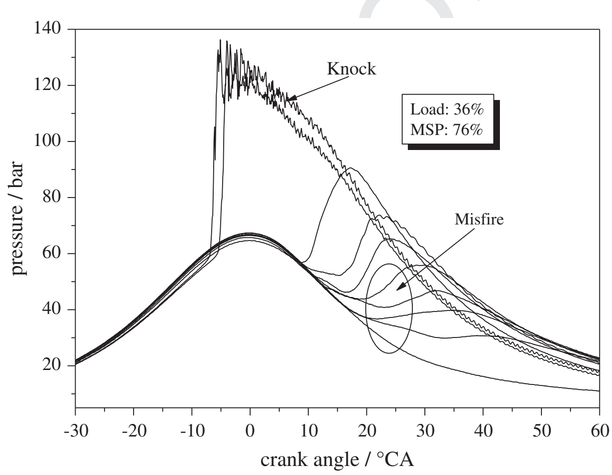

Fig. 4. Variation of cylinder pressure at misfire condition with ten consecutive cycles.

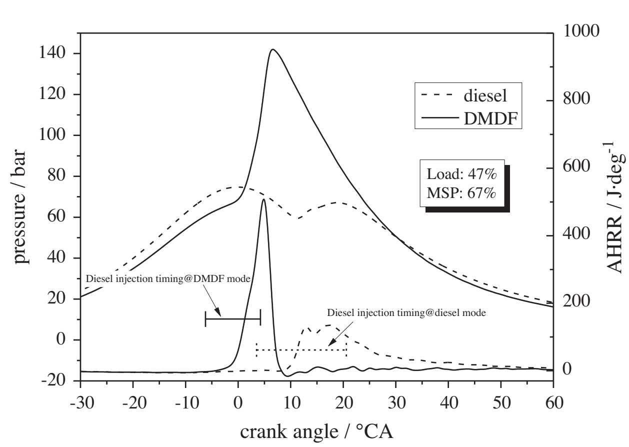

Fig. 5. Comparison of combustion characteristics between diesel and DMDF mode at roar combustion condition.

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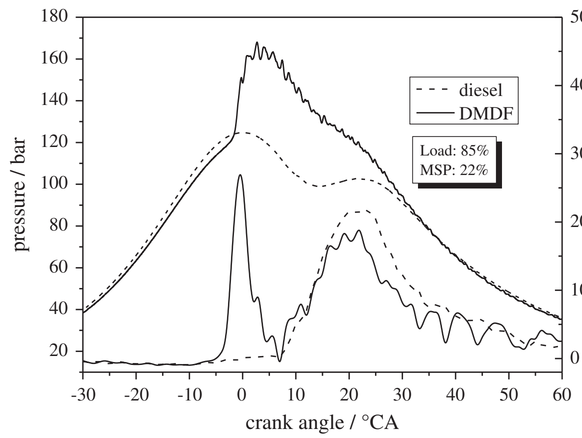

Fig. 6. Comparison of combustion characteristics between diesel and DMDF mode at knock condition.

to fall down to about 10% at full load. And an interesting phenomenon occurs, as shown in Fig. 6, that the premixed methanol with low reactivity auto-ignites before the diesel injects. This phenomenon differs from most of the current literatures stating that methanol fumigation increases ignition delay. The high load condition together with high intake temperature provides more heat for methanol to undergo pre-combustion heat addition. And the advanced methanol injection through the intake port gives the methanol in the charge air more time to carry on pre-combustion chemical reaction [23]. Even though the methanol in the charge air has already commenced combustion when diesel fuel is introduced, the effect on diesel combustion process is not very significant. From Fig. 6 we can see that methanol and diesel fuel almost combust separately, and there is only little change on the combustion process of diesel fuel compared to neat diesel condition. But the auto-ignition of methanol results in greatly increased inter-cycle variability in in-cylinder pressure parameters such as peak pressure and maximum rate of pressure rise with relatively minimal effects on the engine work output. At knock conditions, peak cylinder pressure reaches 17 MPa at 7°CA ATDC, limiting the further increase of MSP.

4. Conclusions

In this paper, the operating range and combustion characteristics of methanol fumigation on a heavy duty diesel engine were investigated. A series of tests were conducted separately under neat diesel mode and diesel methanol dual fuel (DMDF) mode, with the engine running at ten different loads and fixed engine speed of 1400 r/min. During each test, mass flow rates of the diesel and methanol were measured separately as well as the cylinder pressure. From the analysis results, plots of the pressure and the apparent heat release rate in the combustion chamber reveal four operating bounds, which shed light into the combustion mechanism when using methanol fumigation on heavy duty diesel engines. The investigation results are summarized as follows:

(1) The viable DMDF operating range in terms of load and methanol substitution percent (MSP) has been achieved over a load range from 6% to 100% of full load. The operating range is restricted by four bounds: partial burn, misfire, roar combustion and knock respectively.
(2) The lower bound of the operating range is limited by partial burn, which occurs at 20% load conditions with high MSP. At the partial burn bound, the combustion process is highly worsened due to the cooling effect of premixed methanol in the intake process and compression stroke.
(3) The onset of misfire provides the right bound for normal operation when the load increases to medium load. The decrease of pilot diesel injection and extremely low intake temperature cause the fuel/air mixture unable to ignite or reach HCCI-like knock in some cycles.
(4) The extremely loud engine noise defines the roar combustion bound at medium to high load. The increase in the mass of methanol causes an increase in the ignition delay period of pilot diesel which then auto-ignites and starts burning the gaseous fuel at a higher rate of pressure rise.
(5) At high load, the appearance of knock provides the upper bound of the operating range. DMDF combustion with excessive MSP is extremely loud and measured in-cylinder pressure traces show strong acoustic oscillations.
(6) Over the viable operating range, the maximum MSP is quite load dependent, reaching its peak of about 76% at 43.5% engine load under this particular speed. DMDF combustion worsens the BTE at low load while boosts it at medium and high load.

5. Uncited references

[24,25]. Q3

Acknowledgements

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