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Extension of the Lower Load Limit in Dieseline Compression Ignition Mode

Buyu Wang, Zhi Wang, Shijin Shuai*, Linjun Yu, Jianxin Wang

State Key Laboratory of Automotive Safety and Energy, Tsinghua University, Beijing 100084, China

Abstract

A study to extend the low load limit of the mixture of commercial gasoline and diesel in the compression mode is performed on a single cylinder diesel engine. The additional measures, like intake heating, rebreathing, negative valve overlap, are not employed. By adopting boosting, sweeping the injection pressure and varying the fuel octane number, the minimum fuelling rate and the minimum IMEP gained is compared. Besides, the thermal efficiency and emission results at these operation points are also discussed.

The results illustrate that the high intake pressure, the low injection pressure and the low fuel octane number are very effective to extend low load limit. With these strategies, gasoline-type fuels can get the lowest load 0.07MPa IMEP (0.14MPa intake pressure and 20MPa injection pressure) and successfully replace diesel at low load operation points in the compression mode. Increasing the intake pressure and reducing the injection pressure can significantly reduce the minimum fuelling rate and then the minimum IMEP. The minimum IMEP (0.34MPa) of the calibration point on the original engine at test speed (1600rpm) can be achieved by G80 without boosting.

The combustion efficiency is influenced by the intake pressure and the injection pressure, however, the ISFC is more dependent on the engine load rather than other factors. If there is more over-lean mixture in cylinder when adjusting the experimental conditions, CO and HC emissions are higher. To satisfy the Euro VI regulation on NO_x (<0.4g/kWh), a small amount of EGR is needed to control NO_x emission.

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Keywords: Dieseline; Low octane fuel; Intake pressure; Injection pressure

1. Introduction

As a promising combustion mode, gasoline compression ignition is widely studied in recent years. High volatility and low reactivity provide the gasoline-type fuels sufficient time to form pre-mixture to achieve high efficiency and low emissions [1]. In 1970s, Homogeneous Charge Compression Ignition

* Corresponding author. Tel.: +86-10-62772515; fax: +86-10-62772515.
E-mail address: sjshuai@tsinghua.edu.cn (S.Shuai).

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(HCCI) was firstly investigated as a gasoline compression ignition mode, however, the rough combustion at high loads and the difficulties in controlling the combustion phase limit the application of HCCI in commercial engines [2,3]. To extend the high loads limit, Partially Premixed Compression Ignition (PPCI) [1,4] and Multiple Premixed Compression Igntion (MPCI) [5,6] were proposed by using single or multiple injections to form the stratified mixture or achieve multi-stage combustion, respectively. Compared with high octane gasoline, the low octane gasoline is more suitable for the compression mode due to its high reactivity [7,8]. Among the low octane fuels, blending the commercial diesel and gasoline is the most convenient method to reduce the fuel octane number. Therefore, this blending fuel is used in the test. Xu et al. named this kind of fuel as "dieseline" [9,10]. With the parametric studies of the blending ratio, injection timing and EGR ratio, they concluded that dieseline offered high fuel conversion efficiency and greatly reduced the total concentration and mean diameter of particulate matter.

However, stable combustion at very low load is more difficult for gasoline-type fuel than that for diesel due to the differences in fuel reactivity. Some previous studies have been performed to extend the low load limit. But most of them needed to have special modifications on the diesel engine, such as Negative Valve Overlap (NVO) [11], rebreathing [12], intake heating [13] and narrow-angle injector [14]. The goal of this study is to investigate the low load limit of dieseline in the compression mode without any additional equipment on a diesel engine. In order to increase the fuel reactivity and avoid over-lean mixture at low load conditions, three methods are adopted: boosting, reducing the injection pressure and decreasing the fuel octane number. When evaluating the effect of different methods, the thermal efficiency and emission results are also discussed.

2. Experimental setup and methodology

The engine used in this study is a single-cylinder diesel engine modified from a four-cylinder production light-duty diesel engine. The high pressure common rail injection system is equipped, which can provide the injection pressure as low as 10MPa. The original 5-hole injector is employed, of which the spray angle is 158^{o} and the nozzle diameter is 0.19mm. Engine specifications are shown in Table 1 in [15]. The measurement of cylinder pressure data is accomplished by an AVL GH14P cylinder pressure sensor. The exhaust gas emissions, including CO, CO_2, NO_x and THC, are extracted downstream from the exhaust tank by an AVL CEB2 gas analyzer. The schematic of the experimental apparatus is shown in Figure 1 in [15]. To investigate the effect of octane number, two blending ratios of the commercial gasoline and diesel are used: G90 and G80 (90vol% and 80vol% gasoline, respectively). The gasoline and diesel properties are listed respectively in Table 2 in [15]. According to the equation 6 in [16], the research octane number of the blending fuels can be estimated, which are 84.1(G90) and 74.3(G80). It can also be inferred that the volatility of the blending fuels are similar with gasoline-like initial boiling point and diesel-like final boiling point [17].

Table 1. Experimental test matrix

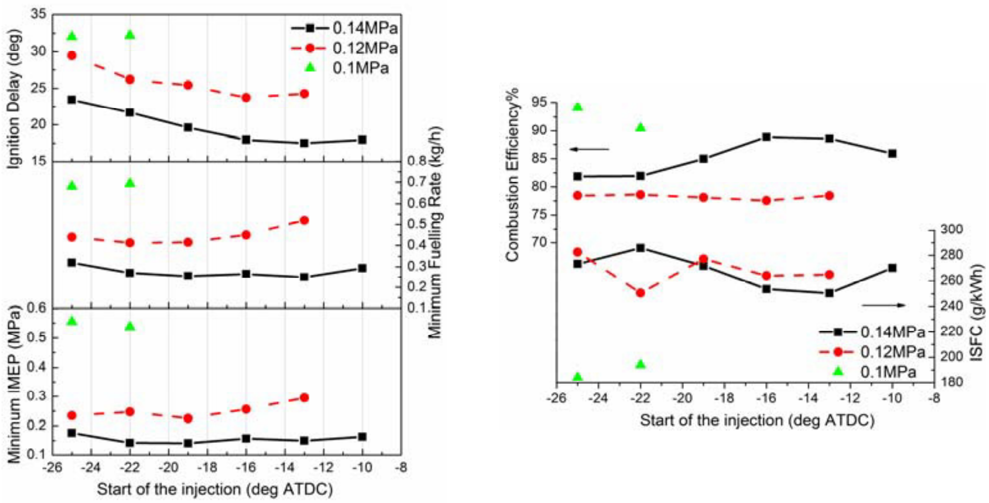
Experiment	1	2
Engine speed (rpm)	1600	1600
Intake pressure (MPa)	0.1-0.14	0.1-0.14
Injection timing (CAD ATDC)	-25 - -10	-13 or -19
Injection pressure (MPa)	50	20-80
Test fuel	G90 and G80	G90 and G80
Coolant temperature (K)	352±2	352±2

The engine speed is fixed at 1600 rpm, which is frequently used by the original engine. When extending the low load limit, a "minimum fueling" method is employed for the injection timing and pressure sweep [14]. The fuelling rate is reduced at each operation point until the COV of IMEP gets 5% and this load is the lowest one to be achieved with stable combustion. Three intake pressures (0.1, 0.12, and 0.14MPa) are tested with sweeping the injection timing from -25 CAD ATDC to -10 CAD ATDC and the injection pressure from 20MPa to 80MPa (Table 1). No EGR and intake air heating are used, so the intake temperature is just the ambient temperature. The coolant temperature is controlled at 352±2K.

3. Results and discussion

3.1. Effect of the intake pressure

Increasing the intake pressure significantly improves the reactivity of fuel-air mixture and reduces the ignition delay (Fig.1a). As a result, the stability of combustion is increased and lower load can be gained at high intake pressure conditions. The lowest load attained in 0.14MPa intake pressure is 0.14MPa IMEP, while it is 0.54MPa IMEP without boosting. Even though the ignition delay is shortened by retarding the injection timing, the lowest load is not gained by the last injection attributed to the different fuelling rates at different injection timings. Previous studies proved that very late or early injection prolonged the ignition delay by influencing the status of the mixture in cylinder [18,19]. But in this study, more fuel is injected in the late injection point leading to shorter ignition delay and higher load. Another interesting finding is that the lowest load is not gained by the minimum fuelling rate, which may be the result of variation in the combustion efficiency (Fig. 1b). The combustion efficiency is calculated according to the Equation 2 in [15].

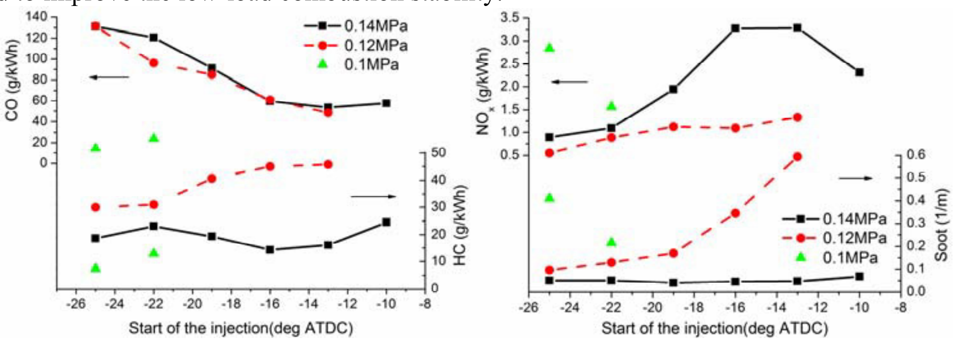


(a) (b)

Fig. 1. (a) Minimum fuelling rate, IMEP, ignition delay and (b) combustion efficiency and ISFC of G90 at different intake pressures

Obviously, the combustion efficiency and thermal efficiency are the highest without boosting due to the high IMEP and the ISFC is high at the operation point with the lowest load (Fig. 1b). In boosting conditions, the combustion efficiency is less than 90%. Comparing with the effect of injection timing and intake pressure, ISFC is more relied on the load. The high combustion temperature at high loads reduces the incomplete combustion and improves the fuel conversion efficiency.

CO emission increases with advancing the injection timing attributed to the lean mixture (Fig.2). However, HC emission is not sensitive to the injection timing, since the effect of quenching near the wall or in the crevice is similar in such short sweep range of injection timing here. Low CO and HC emission is expected due to the high IMEP. Despite of the high load in the 0.12MPa intake pressure cases, HC emission is higher than that in 0.14MPa intake pressure cases. Longer injection duration of 0.12MPa cases increases the spray penetration and then the impingement on the piston wall to form more wall film, which is transformed to local rich regions. As a result, there are more incomplete combustion and high HC emissions. This could also explain the less NO_x emission and higher soot emission in 0.12MPa intake pressure cases. At high intake pressure, retarding the injection timing increases NO_x emission, but has minor effect on soot emission due to the sufficient oxygen. However, the soot emission increases significantly with retarding the injection at small intake pressure because of the less mixing time. The traditional trade-off relationship between NO_x and soot cannot be found at the small intake pressure case attributed to the increase of the local temperature and equivalence ratio by more fuel amount and higher load simultaneously when retarding the injection timing. In the test, no EGR is used. It can be expected, if uncooled EGR is employed, the NO_x emission can be reduced and additionally the intake temperature is increased to improve the low-load combustion stability.



(a) (b)

Fig. 2. (a) CO and HC, (b) NO_x and soot emission results of G90 at different intake pressures

3.2. *Effect of the injection pressure*

The curves in Fig. 3 with different colors represent different intake pressure conditions. Reducing the injection pressure can greatly extend the low load limit, especially for the high intake pressure condition (0.07MPa minimum IMEP at 20MPa injection pressure). The high injection pressure contributes to the spray break-up and form over-lean mixture, which is not good for the stable combustion. Thus, low load cannot be easily attained by the high injection pressure. In fact, at the medium and high loads, the required injection pressure is also lower for gasoline-type fuels than diesel [20]. Considering the whole operation region, gasoline-type fuel favors the low injection pressure, which is an advantage of gasoline compression ignition. Different from the previous study [21], the ignition delay is extended by reducing the injection pressure, because the high fuelling rate at high injection pressure conditions.

Even though the high combustion efficiency is gained by the low injection pressure when the intake pressure is 0.14MPa, the ISFC is still high as a result of the low combustion temperature at very low load. It should be noted that, although very low load can be attained when injection pressure is 20MPa at 0.14MPa intake pressure conditions, the ISFC is quite high and might not be directly employed as a real engine operation point without any optimization.

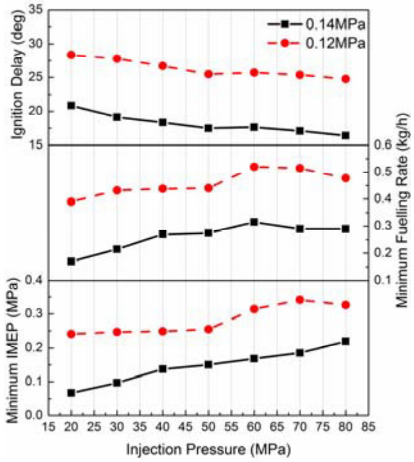
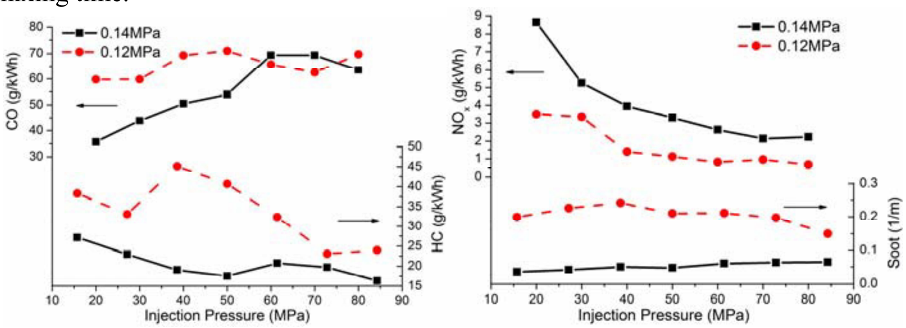


Fig. 3. Minimum fuelling rate, minimum IMEP and ignition delay of G90 at different injection pressures

CO emission decreases with reducing the injection pressure by avoiding over-lean mixture, while HC emission increases with reducing the intake pressure by forming more local rich zones near the piston wall during the spray impingement and wall film evaporation (Fig. 4). When the intake pressure is 0.14MPa, more combustion takes place with the local equivalence ratio close to stoichiometric concentration leading to the high NOx emission and the low soot emission. The NOx emission gets worse by reducing the injection pressure, while soot emission is not sensitive to the injection pressure due to the sufficient mixing time.

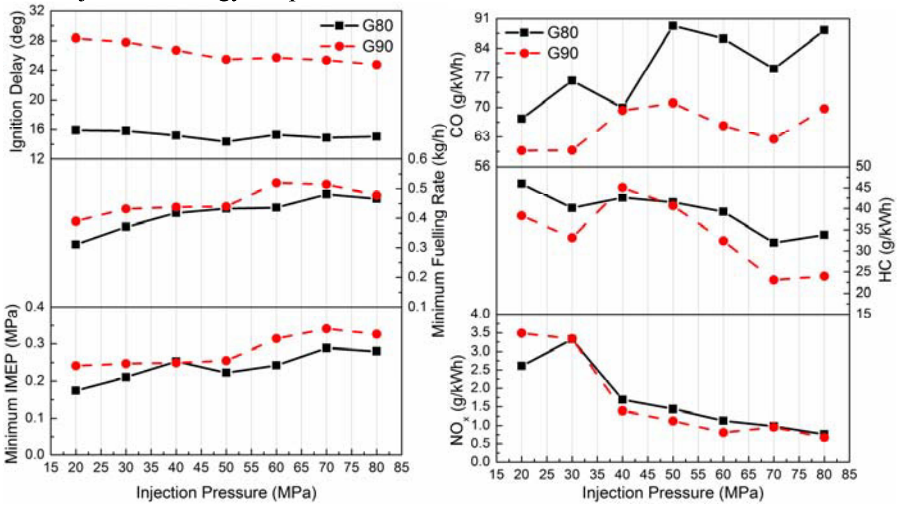


(a) (b)

Fig. 4. (a) CO and HC, (b) NO_x and soot emission results of G90 at different injection pressures

3.3. Effect of the octane number

The test is carried out at 0.12MPa intake pressure and the injection starts at -13 CAD ATDC. Not surprisingly, G80 has shorter ignition delay than G90 (Fig. 5a). Attributed to its high reactivity, G80 can operate with lower fuelling rate and have lower IMEP. When the injection pressure is 20MPa, the minimum IMEP gained is less than 0.2MPa. Even without boosting, the minimum IMEP of G80 is 0.34MPa, which is the lowest calibration point of original engine fuelling diesel at this speed. That is to say, G80 can successfully replace diesel to achieve most of the operation points. It can be expected that further reducing the octane number can offer lower load limit, however, high octane fuel have better performance at high loads than low octane fuel because of the lower pressure rise rate and lower soot emissions [20]. Thus, G80 might be more proper than other blending ratios considering the load range, the demand of boost and the emission results. The emission results of G80 and G90 are compared in Fig. 5b. G80 has higher CO and HC emissions than G90 due to the lower temperature and then more incomplete combustion, while the NO_x emission for both fuels is similar. The low load limit benefits from the low injection pressure, but at the moment NO_x emission is high for G80. In terms of the load limit and emission results, both 20MPa injection pressure and a small amount of EGR are necessary. As a result of high CO and HC emissions, the combustion efficiency of G80 is a little smaller than that of G90. Also, the ISFC of G80 is a little higher than that of G90. But in this study the injection timing for both G80 and G90 are the same, which is the optimized one for G90. So the thermal efficiency of G80 might be improved, if the injection strategy is optimized.



(a) (b)

Fig. 5. (a) Minimum fuelling rate, minimum IMEP, ignition delay and (b) CO, HC, NO_x emissions of G80 and G90 at different injection pressures

4. Conclusions

This study investigates the low load limit of dieseline in the compression mode without any additional equipment on the diesel engine. Three methods are employed to extend the low load: boosting, reducing the injection pressure and decreasing the fuel octane number. The conclusions drawn from the experiments are summarized as follows.

1. The high intake pressure, the low injection pressure and the low fuel octane number are effective ways to extend low load limit. With these strategies, gasoline-type fuels can replace diesel at low load operation points in the compression mode and get the lowest load 0.07MPa IMEP.

2. Increasing the intake pressure and reducing the injection pressure can significantly reduce the minimum fuelling rate and then the IMEP attained. The minimum IMEP (0.34MPa) of the calibration point on the original engine at test speed (1600rpm) can be achieved by G80 without boosting.

3. The ISFC is more related to the engine load rather than other factors, even though the combustion efficiency is influenced by the intake pressure and the injection pressure. If there is more over-lean mixture in cylinder when adjusting the experimental conditions, CO and HC emissions are higher. A small amount of EGR is needed to control NO_x emission in order to satisfy the EURO VI regulation.

Acknowledgements

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Biography

Dr. Shijin Shuai is a professor in the State Key Laboratory of Automotive Safety and Energy, Tsinghua University, China. He has 20+ years' fundamental research and development experience ranging from GDI & HCCI combustion, emission control and alternative fuels with 100+ technical papers published in journals and international conferences.