# Micro-pilot-induced Ignition Diesel/ Natural Gas Engine Control System Development and Engine Performance /Emission Optimization

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**Synopsis**

Diesel/natural gas dual fuel engine is acquiring more and more attention due to its potential to reduce $NO_X$ and soot emission simultaneously. Micro-pilot-induced diesel ignition natural gas engine is a popular manner to further improve the emission reduction capability of dual fuel engine. A six cylinder, four stroke, common-rail diesel engine is converted into dual fuel engine. Natural gas is injected into the intake manifold after the throttle. Five gas injection valves are used to control natural gas flow rate. Based to the established fuel supply system, a dual fuel control system is developed by using MS9S12XEP100 MCU. Voltage boosting circuit, fuel injector driving circuit, gas injection valve driving circuit and MeUn driving circuit are integrated on the platform of MCU hardware. Two ECU is connected to each other by CAN bus and several I/O ports to fulfil the fuel injection functional requirement. A software framework involves gas injection timing synchronization, fuel mode managing, multi-time injection. A MAP based fresh air mass flow rate and intake charge efficiency model is integrated in the MCU to calculate the fresh air quality in cylinder. The last part is performance optimization research at low load. Ignition diesel is divided into two stages, and the first injection timing, first injection ratio and injection pressure are used as controllable parameter to reduce $NO_X$ and HC emission. Experimental result reveal that by dividing ignition injection into two stage and advancing first injection to 60° CA BTDC $CH_4$ emission can be reduced by 77% while $NO_X$ remains unchanged. Increasing the first injection ratio and injection pressure can also reduce THC emission. If injection pressure is higher than 75MPa, the effect of HC reduction effect is not that obvious. Experimental results shows that developed control system can accomplish the functional requirements of dual fuel engine management. Emission test results demonstrate that IMO TierII can be satisfied at diesel mode. DF mode emission performance can meet the requirement of IMO TierIII. Furthermore, as the first domestic product dual fuel dedicated control system, which has passed through the CCS authentication in China, the engine emission level can meet the current and upcoming China's emission standard on non-road engine on the premise of guaranteeing engine power and economy.

*Keywords:* Micro-pilot; Dual fuel engine; Emission; Model-based calibration.

## 1. Introduction

In the context of both energy crisis and environment problem, high efficiency and clean internal combustion engine become a hot topic of research. The substantial deposits, clean combustion process and favourable anti-knock property of the natural gas (NG) made it becomes an ideal substitution of diesel. One of the advantages of NG engine is the simultaneously reduction of $NO_X$ and PM (Zhou, L. et al. 2013). The fuel flexibility of DF engine makes it compatible with Chinese national condition. DF engine uses high pressure common rail diesel injection system to ignite NG in cylinder. Combined with the advantage of multi-jet and flexible injection pressure and injection timing of common rail diesel injection system, emission and economical potential of DF engine can be more sufficiently explored. The combustion process of MPI DF engine is started with premixed combustion then followed with diffusion combustion of NG. At low speed and low load condition, because of the high specific heat capacity and slow flame propagation velocity of NG, incomplete combustion phenomena is obvious. Unburned NG, on the one hand, will deteriorate the fuel economy; On the other hand, $CH_4$ will cause green house effect which will be much stronger than $CO_2$. While at low and medium speed and high power condition, because of the reduced charging coefficient caused by NG injection, Knocking margin is narrowed down（Papagiannakis, R. G. 2010）. These characteristics of NG combustion lead to bad load performance (Srinivasan, K. K. 2006). Based on all questions above, proper MPI DF engine performance can not be reached without the comprehensive control of diesel injection, NG injection, and in-cylinder charging condition.

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MPI DF engine control system is developed aiming at a six-cylinder four stroke high pressure common rail diesel engine transformed MPI DF engine. NG is injected at intake manifold. Two ECU-based hardware framework is established. Based on the self-developed control system, low load characteristic of MPI DF engine is studied.

## 2. Bench and test device

The engine used in the research is transformed from a six-cylinder high pressure common rail engine. CR16 HPCR of Bosch is used as the diesel injection system. Engine parameters are as listed in table 1. Throttle valve is installed on intake manifold, and NG is injected into the intake manifold after the throttle. AMA-i60-R1DI of AVL Company is used to measure $NO_X$, HC, CO, $CO_2$ emission. 6125B cylinder pressure sensor of KISLER and AVL 621 combustion analyzer are used to measure cylinder pressure. Experimental equipment layout is as shown in figure 1.

Table 1 parameters of engine

| ***Item*** | ***Parameters*** |
|---|---|
| Engine type | YC6K420LN-C31 |
| Bore/mm×stroke/mm | 129×155 |
| Displacement/L | 10.338 |
| Combustor type | ω |
| Compress ratio | 16.5:1 |
| Rated power/kW | 309（1800r/min） |
| Maximum torque/N·m | 1640 |

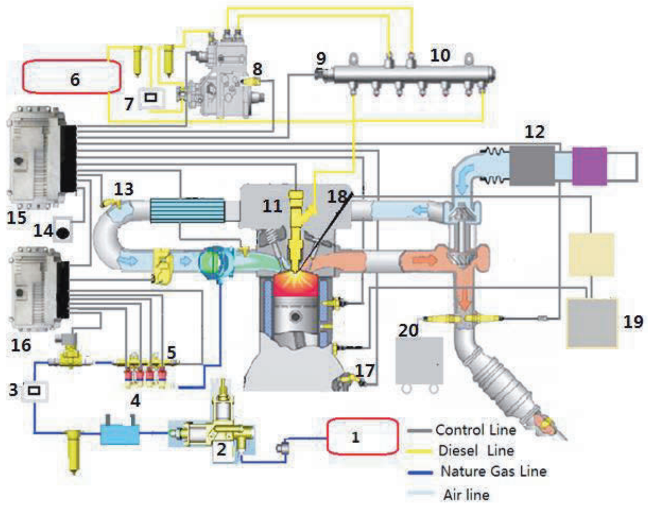


Figure1: experimental equipment layout of the test bench

1 NG tank 2 high pressure reduce valve 3 flow meter 4 NG rail 5 NG pressure and temperature sensor 6 oil tank 7 oil meter 8 high pressure pump 9 rail pressure sensor 10 rail 11 injector 12 ABB air flow rate meter. 13 intake air pressure sensor 14 DF mode operating switch 15 diesel injection controller 16 Gas injection controller 17 crankshaft 18 cylinder presser 19 combustion analyzer 20 emission analyzer

### 3. Control system development

The main function of marine DF engine control system is to efficiently and cleanly and safely operate the engine at the setting speed. MPI DF engine control system include following function connotation. (1)HPCR engine control system. It is that MPI DF engine that can operate at diesel mode in the hole operating condition. There is nearly no difference with traditional HPCR engine at this point. (2)Fuel mode management. DF engine control system can switch between different fuel mode automatically according to the engine operating condition, NG supply status, and manual operation. (3)Intake charging system control. It is indispensable to achieve high efficient operation of DF engine intake charging system, especially as a manner to optimize the low


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load emission performance. This is the most obvious different with traditional diesel engine. (4)NG injection control. Inject NG into intake manifold according to the demand of engine management modular. On the basis of MPI DF engine control system requirement analysis, following function should be included in the control system. In terms of engineering compromise of fuel mode flexibility of DF engine, MCU hardware source configuration, and control system availability at partial fault, two ECU included MPI DF engine control system framework is designed. System configuration and dependencies are as illustrated as figure 2.

Table 2: MPI DF engine control system requirement

| *function requirement* | *Input function requirement* | *output function requirement* |
|---|---|---|
| High pressure common rail system | 1rail pressure acquisition<br>2intake temperature and pressure acquisition<br>3coolant temperature acquisition<br>4 lubrication oil temperature and pressure acquisition<br>5 camshaft and crankshaft acquisition<br>6 MeUn valve drive current acquisition | 1 injector driving of six cylinder<br>2 MeUn valve driving |
| Fuel mode management | 1 NG temperature and pressure acquisition<br>2 fuel mode switch operating input | No |
| Intake air management | 1throttle position acquisition<br>2 intake pressure and temperature acquisition | Throttle driving |
| Gas injection | 1 NG temperature and pressure acquisition | NG injector driving of six cylinders |

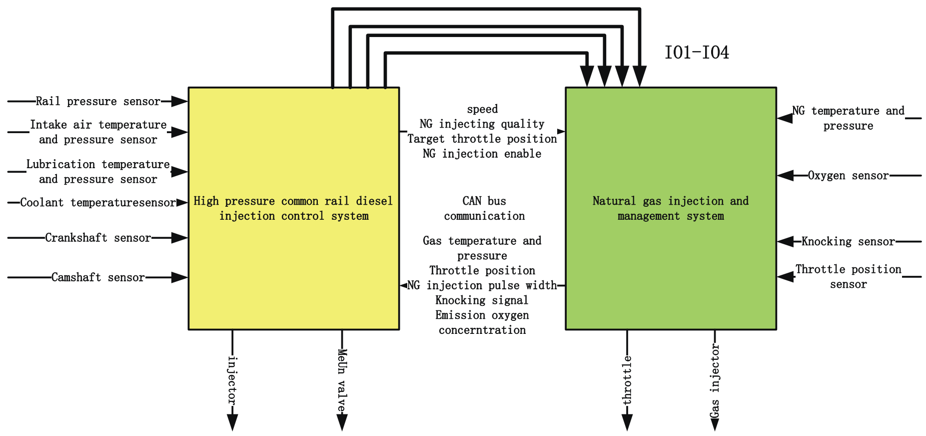


Figure 2: System configuration and dependencies of MPI DF engine control system

### *3.1. CAN bus and I/O based dual ECU synchronization*

For the dual ECU case, if camshaft and crankshaft signal are divided simply and connected into two ECUs, it is likely to cause signal absent and then lead to starting failure. CAN bus communication and I/O trigger is combined to achieve gas injection timing control logic. The basic idea of proposed method is to communicate injection width by CAN bus and trigger injection by logic coding. Gas injection cylinder number and gas injection delay time are calculated at gas injection calculation tooth before compress stroke TDC. At the same time sent the gas injection cylinder number to gas injection board by I/O1-I/O3 generous purpose input and output ports coding. When injection delay timing crankshaft edge interrupt comes I/O4 output a falling edge to trigger gas injection function to inject a pulse width as received from CAN bus communication.


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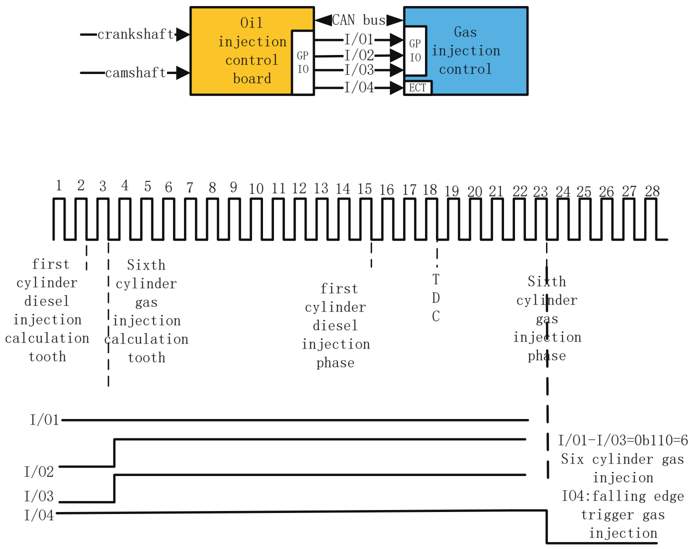


Figure 3: CAN bus and IO based dual ECU synchronization theory diagram

## 4. Dual fuel engine control strategy

### *4.1. Fuel mode management of dual fuel engine*

Different control function is needed at different fuel mode. Fuel mode management modular(FMM) is needed to manage the engine running at different fuel mode and switching between different fuel modes. The function of FMM: based on the engine operating condition, gas supply system status and man-machine interaction status confirming current fuel mode and providing mode basis for engine management and fuel injection. Four fuel modes are distinguished in DF control system. They are diesel mode, diesel switching to dual fuel mode, dual fuel mode and dual fuel switching to diesel mode. When system runs in diesel mode there is nearly no difference with traditional diesel engine. At DF mode gas injection and diesel injection is distributed by fuel distribution strategy. The other two fuel modes are transition mode.

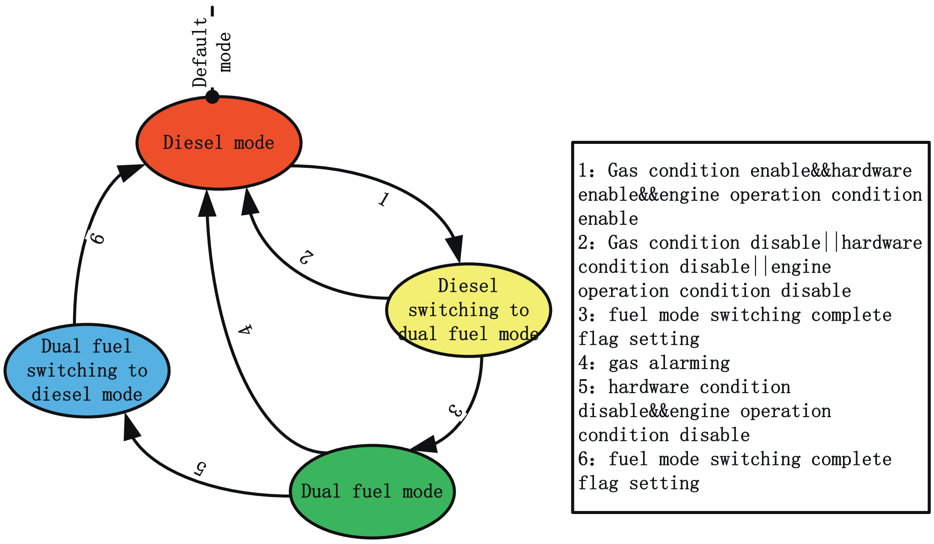


Figure 4: fuel mode management theory diagram

During fuel mode switching process, the key task is to stabilize engine speed and the same load change should be allowed in some degree. The second requirement of fuel mode switch is to ensure safely operation, knocking should be actively avoided. Fuel mode switching should be accomplished as soon as possible. NG injection amount is increased with a calibrated step size, while diesel injection amount is equal to speed closed loop calculation amount. Due to the fact that NG combustion torque is not included in the speed closed loop control, so the diesel injection amount will be decreased as the increase of NG injection amount. When the diesel


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injection amount decrease to the amount smaller or equal to ignition diesel amount, it means that fuel mode switching process is accomplished. When fuel mode switch is accomplished to keep speed stable and consistent integral result will be assigned a value equals to diesel ignition amount adding gas injection amount. Under dual fuel mode the amount of diesel injection equal to that of calibrated ignition, and NG injection equals that speed closed loop calculate fuel amount minus ignition diesel amount.

Speed closed loop fuel amount is actively divided during dual fuel mode switching to diesel mode. Diesel is increased step by step, and NG injection amount is the amount speed closed loop fuel amount subtracted by diesel injection amount. As the increase of diesel amount NG amount will be decreased gradually. When NG decreased to the minimum allowed NG injection amount means the accomplishment of fuel mode switching. Minimum allowed NG injection amount should be bigger than minimum injection amount of injection valve. Fuel mode switching process theory diagram and bench test results are as illustrated in figure 5 and figure 6.

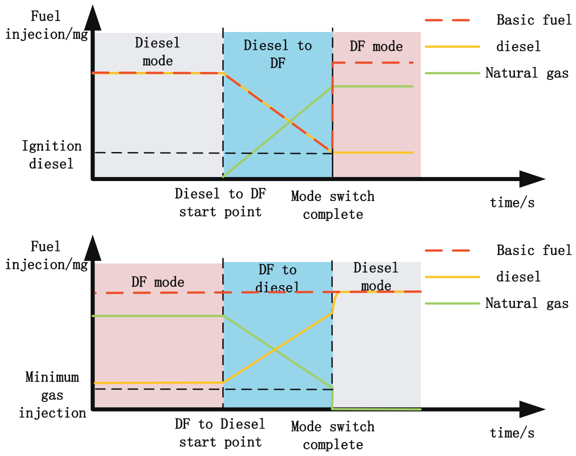


Figure 5: fuel mode switching process

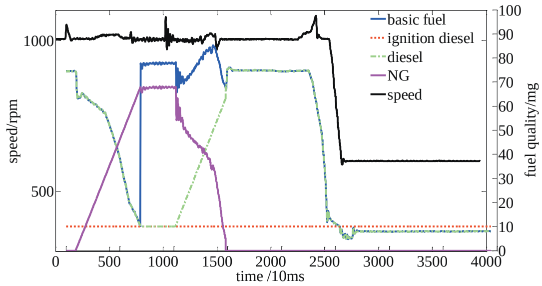


Figure 6: fuel mode switching bench test

### *4.2. Closed loop speed control strategy*

The basic function of marine MPI DF engine control system is speed close loop control no matter in diesel mode or in dual fuel mode. Comparing to tradition HPCR diesel engine control system two extra modular should be added into control flow chart. The first modular is fuel mode management (FMM) designed above. FMM modular is inserted after engine management (EM). The other modular is intake air management (IAM). The function of IAM is to predict the in cylinder air quality according to operating condition. IAM is installed after FMM and before injection mode management (IMM).To achieve proper operating performance in all fuel modes and whole operating conditions, two small changes are made in EM and IMM modular. Two sets of PID calculation MAP is configured in speed closed loop control according to different fuel mode in EM modular. As to IMM modular injection mode pilot injection quality can be calculated according to different MAP at different fuel mode.

EM modular calculates the demanded fuel quality to balance load torque. The output of EM is so called basic demand fuel quality. Basic demand fuel quality is distributed by FMM modular according to different fuel mode. In diesel mode entire basic demand fuel will be delivered to IMM modular. In DF mode FMM modular calculate basic diesel quality and basic gas quality according to diesel ignition quality. To avoid too small AFR leaded knocking actively, basic NG quality is limited by in-cylinder air quality. Research shows that DF engine will be more liable to knocking when AFR is smaller than 1.4. AFR 1.5 is selected as the lower limit in this research so as to leave enough knocking margin (Vávra, J. 2017). Final diesel quality and gas quality is determine after FMM and ITM calculation and limitation. NG injection quality is transferred to gas injection control board by CAN bus. Diesel injection quality is transferred to IMM modular to calculate the pilot injection quality, main injection quality and interval between two injections. Diesel injection control board is responsible for diesel injection. Figure 8 shows the closed loop speed performance of designed control strategy.


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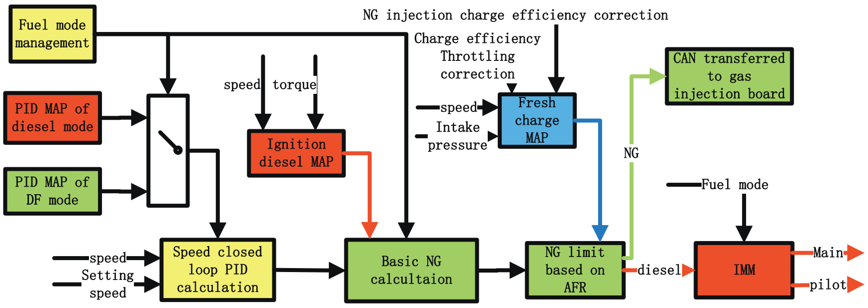


Figure 7: closed loop speed control diagram

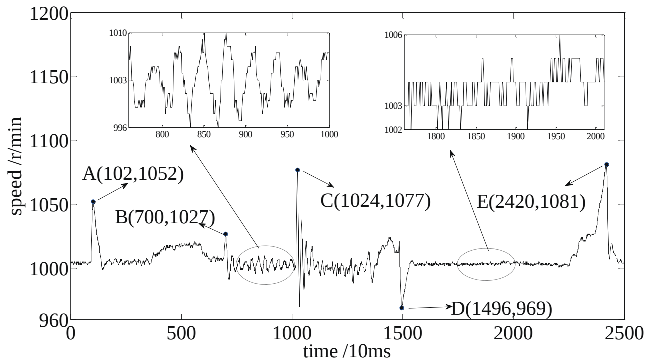


Figure 8: closed loop speed control at different fuel mode and load sudden load sudden take off

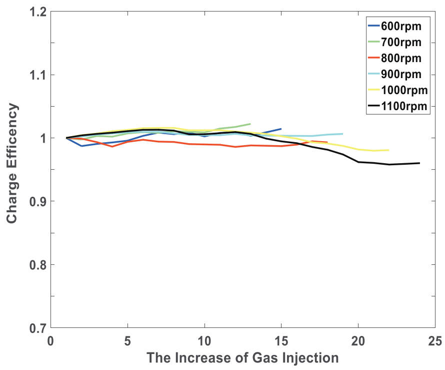


Figure 9: charge efficiency impact of NG injection

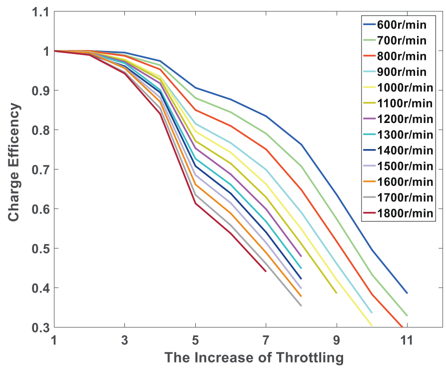


Figure 10: throttle charge efficiency

### *4.3. The effect of throttle and NG injection on charge efficiency*

A throttle is installed on intake manifold to control AFR at low and part load by decreasing the fresh air in cylinder. Throttling is an effective method to reduce incomplete combustion at low load condition (Lounici, Mohand S. et al. 2014). The effect of throttling is studied at diesel mode. Diesel engine runs at specific speed and torque, and changes throttle valve position. At the same time record intake air mass flow rate, fuel injection quality, and intake air pressure after intercooler. By changing NG injection pulse width manually at diesel model at specific speed and torque, intake air flow rate at different substitute rate is measured. Mole flow rate analysis is conducted during the analysis of NG injection impact factor. NG injected into intake manifold will occupy a part of cylinder volume this will decrease cylinder fresh air charge efficiency. This part of effect can be calculated according to Avogadro law. However injecting NG at intake manifold will inevitably impose extra flow resistance. The impact of flow resistance is got by experiment test. Throttling charge coefficient is considered by introduce a throttling factor in control system. Throttling factor is the ratio of measured throttled air mass flow rate and the air mass flow rate with throttle valve fully opened. Scavenging process and throttling caused exhaust temperature increase are ignored.


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## 5. Middle and low load characteristic study of DF engine

Incomplete combustion of DF engine at low and medium leading to HC emission overtop is a main question faced with DF engine. By splitting ignition diesel into two stages and inducing in-cylinder combustion transforming to HCCI like combustion process is an effective HC cutoff method (Xu, M.et al. 2016). The research focus on the study of the effect of interval of two injection stages, diesel injection quality ratio of two injection events and injection pressure on combustion and emission characteristic. Engine speed is kept constant at 1000r/min, 725Nm of torque. During experiment ignition diesel quality is kept constant as 10mg per cylinder per cycle. Second stage injection timing is 10° CA BTDC. Diesel injection is 80Mpa. NG injection pressure is 0.65Mpa.

Cylinder pressure at different first stage injection timing is as illustrated in figure 11. It is obvious that the effect of first injection timing on in-cylinder pressure can be divided into two stages. When first injection timing is about 15-30° CA BTDC, maximum pressure will increase with the ahead of first injection timing. This is because the combustion first stage injected diesel will have positive effect on the second injection (Aksu, Cagdas et al. 2016). Violent second stage oil combustion leads to faster NG flame spread velocity, and then leads to higher maximum in-cylinder pressure and combustion phase shift ahead. However, if first injection is further advanced to 30-60° CA BTDC, in-cylinder maximum pressure starts to decrease, combustion phase is retarded. This can be explained by following theory. When first injection is fully advanced, the diesel injected at first stage wills not combust immediately because of lower in-cylinder temperature. Whereas, firstly injected diesel will mix and evaporate with and within NG-air mixture which means the more diesel will be burned at premixed combustion stage. Combustion phase will still be controlled by second injection like single injection (Wang, Z.S. et al. 2016). Maximum in-cylinder pressure will be lower than later first injection. While because of better mixture status of firstly injected diesel, multipoint ignition source is more likely to happen during second injection. On the one hand this will increase combustion velocity of NG, decrease the flame propagation distance, on the other hand it will extend the ignition point distribution field, and more NG will probably to be ignited, and improve fuel economy.

Cylinder pressure at different diesel quality ratio of two stage injection is as shown in figure 12. It is can be seen that in-cylinder maximum pressure increase with increase of first stage diesel injection ratio. This is mainly because that the more diesel injected into cylinder at 60° CA BTDC the more sufficient ignition source by the time of second injection stage. At the same time the increase of first stage injection ratio means the increase of premix combustion of ignition diesel, this will leads to more violent combustion and higher in-cylinder maximum pressure.

The effect of different first injection timings on exhaust emission is shown in figure 13 and figure 14. As the advance of first injection timing $NO_X$ emission presents increasing trend at first. While as the first injection timing further advanced to 30-60° CA BTDC $NO_X$ emission will gradually decrease. This phenomenon is in accordance with the change of maximum pressure. $CH_4$, THC and CO emission keep go down as the enlarge of two injection interval. This is because increased maximum in-cylinder provide favourable environment for $NO_X$ generation (Ishiyama, et.al2011). The maximum $NO_X$ emission 13.7 g/kW • h condition comes when first injection timing is 27-30° CA BTDC. The minimum $NO_X$ emission 7 g/kW • h comes when first injection timing is 60° CA BTDC and when there is only second injection. $CH_4$ emission decrease to 1.8 g/kW • h from nearly 7.8g/kW • h. Emission data reveal that by dividing ignition diesel into two stages, $CH_4$ emission can be cut off by 77% while $NO_X$ emission kept unchanged. HC is decreased about 73%, and CO is decreased by 70%.This is mainly because early injected diesel provide more ignition source so as to enable more NG will take part in combustion(Guerry, E Scott et al. 2016). Unburned $CH_4$ and incomplete THC and CO decreased dramatically.

The influence of first injection ratio on emission is as shown in figure 15 and figure 16. $NO_X$ emission is about 6.8 g/kW • h when there is only single ignition injection. If first injection ratio is 30% $NO_X$ will decrease to 6.2 g/kW • h. With first injection ratio increasing to 70%, $NO_X$ emission goes to about 7.7 g/kW • h. $NO_X$ emission increases about 13% while as $CH_4$ emission is decreased by 85% from 6.5 g/kW • h to 1g/KWh. Because of the increase of first injection ratio more diesel is burned during premix combustion stage. This will increase in-cylinder pressure on the one hand and then lead to $NO_X$ emission increase, on the other hand more ignition source and higher in-cylinder temperature lead to the decrease of unburned $CH_4$ and incomplete combustion of THC and CO.

The effect of diesel injection pressure is also studied in this research. It is can be seen that with the increase of injection pressure, in-cylinder maximum pressure intends to increase. Peak cylinder pressure is about 10Mpa at 40Mpa injection pressure. While if injection pressure increase to 120Mpa, peak cylinder pressure is increased 30% and reaches 13MPa. During this process $NO_X$ emission is doubled. While THC emission is decreased from 3.75 g/kW • h to 2.6 g/kW • h. Attention should be paid on the phenomenon that minimum HC emission comes


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when injection pressure is 75Mpa. Further growth of injection pressure will have little effect on THC emission reduction. This can be explain by the theory that increase injection pressure will improve diesel atomization and evaporation, and increase the ratio of premix combustion (Yousefi, A., 2017). Increase injection pressure will shorten ignition delay period which lead to peak cylinder pressure shift to TDC.

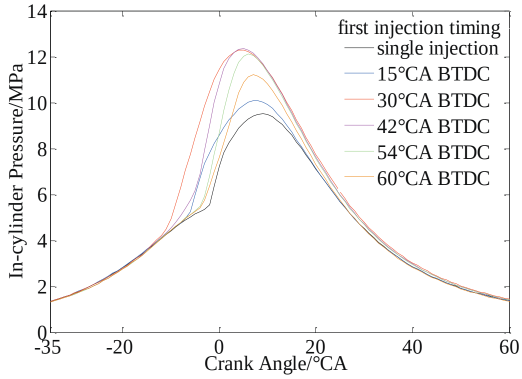


Figure 11: in-cylinder pressure at different first injection timing

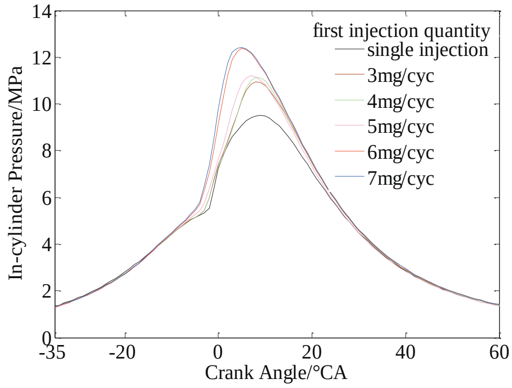


Figure 12: in-cylinder pressure at different first injection ratio

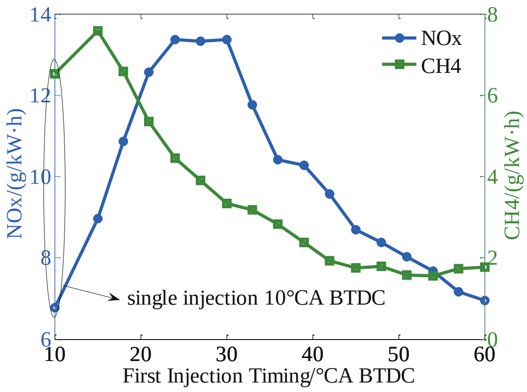


Figure 13: $NO_X$ and $CH_4$ emission at different first injection timing

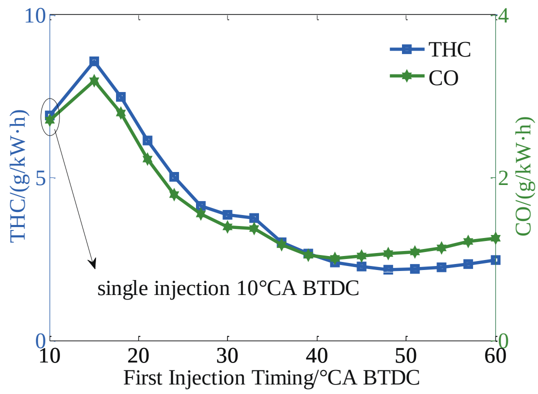


Figure 14: THC and CO emission at different first injection timing

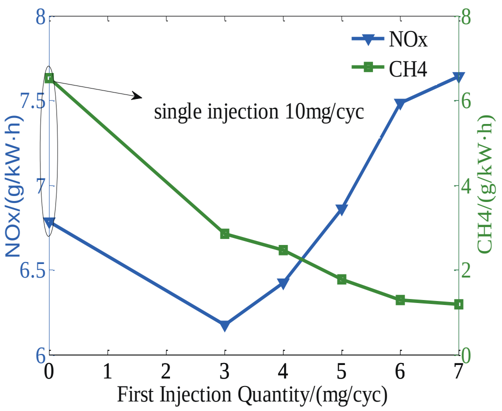


Figure 15: $NO_X$ and $CH_4$ emission at different first injection ratio

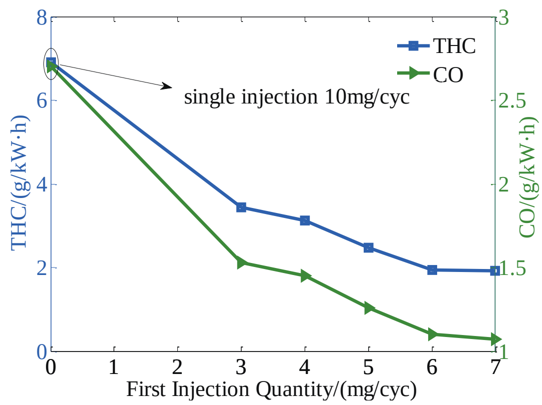


Figure 16 THC and CO emission at different first injection ratio


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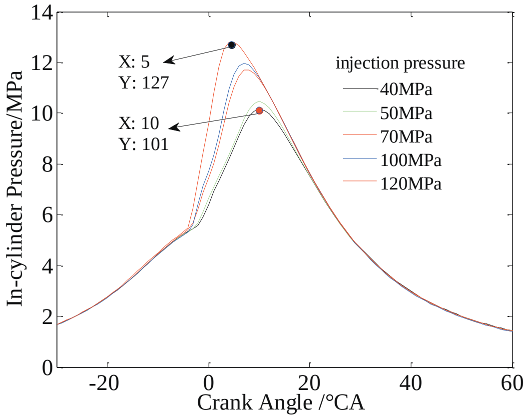


Figure 17: in-cylinder pressure at different injection pressures

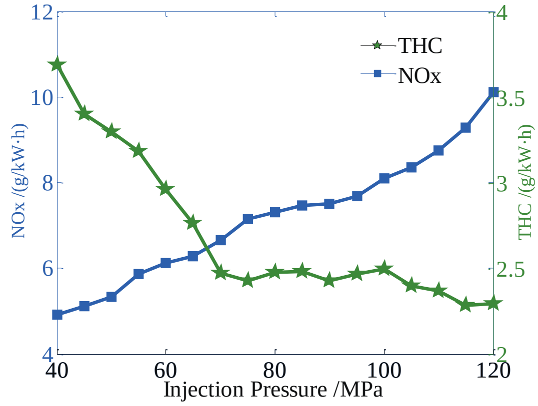


Figure 18: $NO_X$ and THC emission at different injection pressure

### 5.1. *Engine emission performance test*

Table 3 and table 4 shows the emission performance test results at emission regulation specified operation condition. In diesel mode $NO_X$ weighted calculation result is 5.08 g/kW • h. It means IMO TierII is satisfied. While at DF mode $NO_X$ weighted calculation result is 1.207 g/kW • h. This can totally meet the requirement of TierIII 2.0 g/kW • h. Emission performance test result reveal that diesel ignition NG dual fuel engine is capable of satisfying the severest emission regulation. The designed DF engine control system can fully explore the emission reduction potential.

Table 3: Emission statistics at diesel mode

| ***Speed r/min*** | ***Torque Nm*** | ***$NO_X$ (PPM)*** | ***$NO_X$ g/kW•h*** | ***weighting coefficient*** | ***IMO TierII $NO_X$ limitation 7.84 g/kW•h*** |
|---|---|---|---|---|---|
| 1800 | 1607 | 632 | 4.95 | 0.2 | |
| 1638 | 1325 | 606 | 5.26 | 0.5 | $NO_X$ weighted calculation result 5.08g/kW•h |
| 1440 | 1005 | 500 | 4.56 | 0.15 | |
| 1134 | 638 | 500 | 5.16 | 0.15 | |

Table 4: Emission statistics at DF mode

| ***Speed r/min*** | ***Torque Nm*** | ***$NO_X$ (PPM)*** | ***$NO_X$ g/kW•h*** | ***weighting coefficient*** | ***IMO TierIII $NO_X$ limitation 2.0 g/kW•h*** |
|---|---|---|---|---|---|
| 1800 | 1607 | 280 | 2.06 | 0.2 | $NO_X$ |
| 1638 | 1325 | 144 | 1.15 | 0.5 | weighted calculation result |
| 1440 | 1005 | 104 | 0.89 | 0.15 | 1.207 g/kW • h |
| 1134 | 638 | 57 | 0.58 | 0.15 | |

## 6. Conclusions

Control system of a 6-cylinder HPCR DF engine is developed and designed. Based on the newly designed DF engine control system, the effects of split ignition diesel fuel and injection pressure on low load characteristics of MPI DF engine are studied. The experimental results reveal that by dividing the ignited diesel fuel into two stages and advancing the first-stage injection timing to 30-60° CA BTDC, $CH_4$ emission can be decreased by 77% while NOX emission keeps unchanged. At the same time, THC emission decreased by 73% and CO emission decreased by 70%. Increasing the first-stage injection ratio can also improve THC emission performance. And THC emission reduction is finite by changing the injection pressure. When the injection pressure is higher than 75MPa, not only HC emission reduction effect is not that obvious, but also NOX emission performance will be deteriorated.


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## 8. Glossary of terms

CCS
MPI: Micro Pilot ignition
HPCR High Pressure Common Rail
AFR: Ari Fuel Ratio
EM: Engine Management
IAM: Intake Air Management
FMM: Fuel Mode Management
IMM: Injection Mode Management

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