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# THE EFFECT OF INTAKE PRESSURE AND INJECTION TIMING ON LOW LOAD COMBUSTION PROCESS OF DIRECT INJECTION NATURAL GAS ENGINE

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**ABSTRACT**

Taking diesel-ignited direct injection natural gas (NG) engine as the research object, the synergistic control of intake pressure and NG injection timing on the organization of in-cylinder mixture and combustion process was analyzed under the condition of low load (1300rpm) by numerical simulation, in order to achieve low HC emission and high thermal efficiency. The results show that by improving intake pressure, the NG jet momentum, squeeze flow and turbulent motion in cylinder are enhanced, which is beneficial to the stratification of mixture, gradual ignition and combustion rate in cylinder. In addition, too prior injection results in more combustible mixture entering the cylinder clearance, leading to incomplete combustion; too late injection causes too late combustion, reducing the power capability and thermal efficiency. By optimal matching the intake pressure and NG injection timing, a higher indicated thermal efficiency (ITEg) of 49.03% can be obtained when the intake pressure is 0.2MPa and the injection timing is -10°CA ATDC with a lower NOx emission level without using EGR technology.

**Keywords:** natural gas direct injection engine, diesel ignition, injection strategy, intake pressure, combustion process

## 1. INTRODUCTION

With the intensification of energy and environmental issues, energy conservation and emission reduction have become an urgent task for internal combustion engines. Compared with diesel fuel, natural gas (NG) has the advantages of low calorific value and abundant reserves. By studying high-efficiency clean combustion technology and applying it to diesel/NG dual-fuel engines, it will effectively alleviate pressures such as large energy gaps and environmental pollution.

However, due to the low laminar flow flame propagation velocity of the NG, the way in which diesel-ignited injection NG in the intake port has the problem of low NG substitution rate and incomplete combustion. Therefore, diesel-ignited direct injection NG engines are receiving a lot of attention. Since the diesel is ignited at the same time in multiple points in cylinder, the distribution of NG has an important influence on the combustion and emission characteristics. Through the control of the combustion parameters, the reasonable organization of the mixture in the combustion process is realized, which is the key means to achieve efficient and clean combustion.

Su WH et al.[1] found that the direct injection technology makes the distribution of fuel and temperature in the mixing process at the microscopic level uneven, which has a significant impact on the ignition and combustion process. In addition, Zeng K et al.[2] found that NG direct injection can effectively improve volumetric efficiency, reduce UHC emissions and the possibility of knocking during combustion. Florea R et al. [3] used a co-direct injection of diesel fuel and NG ($DI^2$) at 1000rpm on a six-cylinder machine with 2.5L displacement, 17 compression ratio, 30MPa injection pressure produced by Westport Company. Under the condition of 1000rpm and Indicated Mean Effective Pressure (IMEP) of 1.2MPa, the effective thermal efficiency is increased from 42.5% (High Pressure Direct Injection, HPDI) to 44.5%($DI^2$).

In summary, through the control of the stratification of the in-cylinder mixture and the optimization of the NG injection strategy, it is helpful to achieve efficient combustion under low load conditions. However, due to the low NG equivalent in the low-load conditions, it is difficult to form a mixture of reliable self-ignition.

Therefore, the intake pressure is used to enhance the NG jet momentum, squeeze flow and turbulent motion in cylinder. By controlling the NG injection timing to improve the stratified state of the in-cylinder mixture, a more stratified combustion is achieved.

## 2. MODEL ESTABLISHMENT AND RATIONALITY VERIFICATION

### *2.1 Model establishment*

This article is based on Westport Company's dual-fuel HPDI engine with a diesel/NG coaxial needle injector. Table 1 shows the main technical parameters of the engine prototype. The engine was modeled using CONVERGE software. Since the number of nozzles of the original injector is 9 with geometrical symmetry, in order to improve the calculation efficiency, the 1/9 model of the combustion chamber is taken for numerical simulation, as shown in Fig 1. The numerical simulation starts from the intake valve closing time (-150°CA ATDC) and ends at the exhaust valve opening time (140°CA ATDC).

Table 1 Main technical parameters of engine

| Parameter | Value |
|---|---|
| Bore / mm | 137 |
| Stroke / mm | 169 |
| Displacement / L | 14.9 |
| Compression ratio | 17 |
| Fuel injection cone angle / ° | 72 |
| Number of nozzles × nozzle diameter / mm | 9×0.17 |

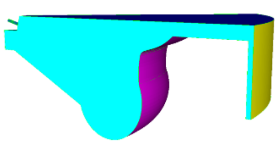
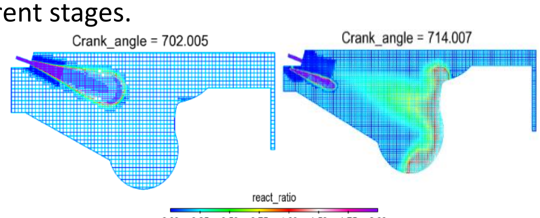



Fig 1 Combustor geometry model (at top dead center, TDC)

### *2.2 Calculation grid*

CONVERGE adopts Adaptive Mesh Refinement (AMR) technology to automatically make full use of the mesh in different calculation areas with high calculation fidelity and efficiency. In this study, the maximum and minimum grid numbers are 407485 and 13651 respectively. Fig 2 shows the grid distributions at different stages.




(a) NG injection stage (b)diesel fuel injection stage

Fig 2 Grid distributions of combustion chamber

### *2.3 Calculation sub-models*

Table 2 shows the sub-models selected for the turbulence, spray, combustion and emission modules in the CFD numerical simulation.

Table 2 CFD numerical simulation sub-models

| Module | | Model |
|---|---|---|
| Turbulence | | Standard $k$-$\varepsilon$[4] |
| Spray | break | KH-RT[5] |
| | evaporation | Frossling Correlation |
| | collision | NTC、Wall film |
| Combustion | combustion | SAGE chemical reaction solver |
| | wall heat transfer | Han and Reitz[6] |
| Emission | NOx | Extended Zel'dovich NOx |
| | SOOT | Hiroy SOOT |

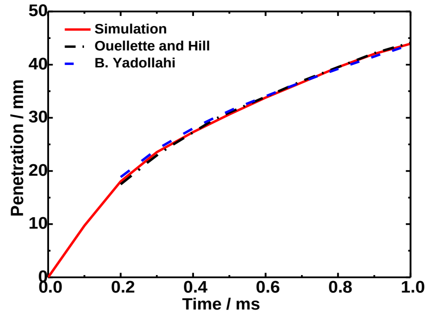
Since the empirical model related to gas jet flow has not been developed in the CONVERGE software, this study used the method of Baratta M et al.[7] to establish a gas nozzle model and set initial boundary conditions such as gas temperature, injection pressure, and the proportion of the components of gas mixture to complete the numerical simulation calibration of the gas jet process in the cylinder.

### *2.4 Model rationality verification*

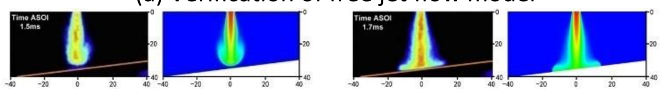
In order to verify the accuracy of the jet flow process, this paper separately calibrates the free jet flow process and the wall collision jet morphology. For the free jet flow process, this paper referred to the experimental data of Ouellette P [8] and the numerical simulation results of Yadollahi B [9] to calibrate the jet flow penetration distance; for the wall collision jet process, the wall jet flow test of Yu JZ et al.[10] was referred to calibrate the shape of the wall collision jet flow of NG in cylinder. Table 3 and Fig 3 show the calibration parameters and results of the jet model respectively, which verify the validity of the NG jet flow in cylinder.

Table 3 Validation of relevant parameters for jet model

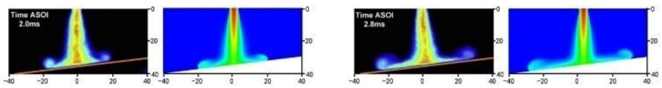
| Parameter | Value |
|---|---|
| Chamber (Radius × Length) / mm×mm | 20×90 |
| Injection pressure / MPa | 15 |
| Injection temperature / K | 350 |
| Chamber pressure/ MPa | 5 |
| Chamber temperature/ K | 850 |
| Wall temperature/ K | 450 |
| Nozzle diameter/ mm | 0.5 |
| Turbulence kinetic energy/ $m^2 \cdot s^{-2}$ | 1.5 |




(a) Verification of free jet flow model



（1） 1.5 ms ASOI （2） 1.7 ms ASOI



（3） 2.0 ms ASOI （4） 2.8 ms ASOI

(b) Comparison of wall collision jet model experiment (left) and simulation (right)

Fig 3 Calibration of NG jet process

This study also combined the test data[3]of South-west Research Institute to calibrate the numerical results under 1000 rpm and BMEP≈1.2MPa conditions. Table 4 and Fig 4 give the main calibration technical parameters and the calibration results of the numerical simulation and test data respectively. It shows that the numerical simulation results agree well with the experimental data.

Table 4 Calibration test related parameters

| Parameter | Value |
|---|---|
| Engine type | Westport HD 15L HPDI |
| EGR | 31% |
| NG injection timing /°CA ATDC | -30 |
| Diesel injection timing /°CA ATDC | -7 |
| Injection pressure/MPa | 0.2 |

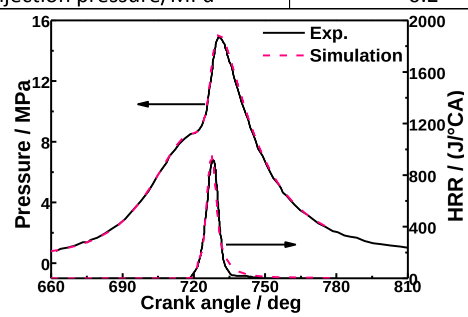



Fig 4 Comparison of numerical simulation results with experimental results

### *2.5 Numerical simulation scheme*

In this paper, the injection method of co-direct injection of diesel fuel and NG was adopted, and the combustion process under low load condition was numerically simulated without using EGR technology. The influences of combustion control parameters such as intake pressure ($P_{in}$) and NG injection timing ($SOI_{(NG)}$) on the stratified state, combustion process and emission characteristics of the in-cylinder mixture were studied. The setting of the numerical simulation parameters is shown in Table 5.

Table 5 Numerical simulation parameters

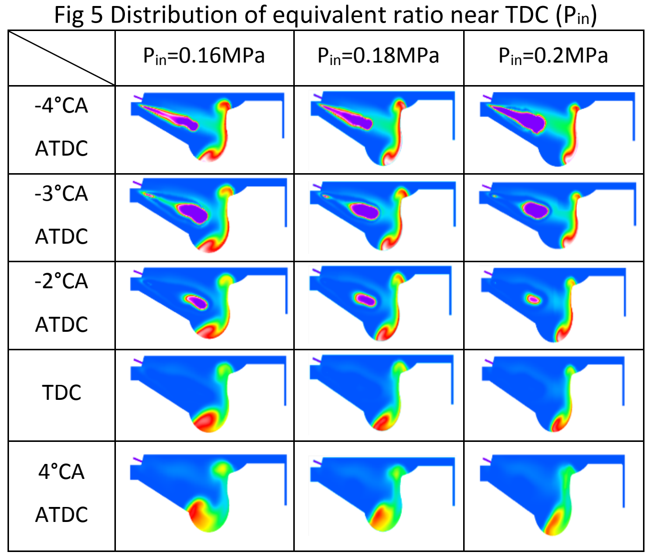
| Parameter | Value |
|---|---|
| Speed /rpm | 1300 |
| IMEP /MPa | 0.5~0.6 |
| equivalence ratio | 0.206~0.257 |
| Intake pressure /MPa | 0.16/0.18/0.2 |
| Intake temperature /K | 333 |
| NG energy proportion | 92% |
| NG injection per cycle /mg | 58.45 |
| Diesel fuel per cycle /mg | 5.63 |
| Diesel injection pressure /MPa | 40 |
| Diesel injection timing /°CA ATDC | -7 |
| NG injection pressure /MPa | 10 |
| NG injection timing /°CA ATDC | -25/-20/-15/-10/-5 |

## 3. RESULTS AND DISCUSSION

### *3.1 Influence of intake pressure and $SOI_{(NG)}$ on the distribution of mixture*

Fig 5 is the effect of intake pressure (injection timing $SOI_{(NG)}$ = -20°CA ATDC) on the microscopic level variation of the in-cylinder mixture near TDC under the conditions of Table 5, showing that NG is injected into the cylinder before the pilot diesel fuel is injected. When all NG has been injected into the cylinder, in-cylinder mixture has not been evenly mixed yet. The diesel fuel is ignited after the NG in the cylinder is stratified. Increasing intake pressure causes a reduction in cylinder mixture concentration zone, and results in a shorter jet flow penetration distance in cylinder at the same crank angle (eg: -4°CA ATDC) because the resistance of the jet flow motion in cylinder becomes larger. The mixing degree of the mixture also lags behind.

Fig 5 Distribution of equivalent ratio near TDC ($P_{in}$)



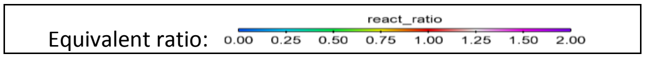
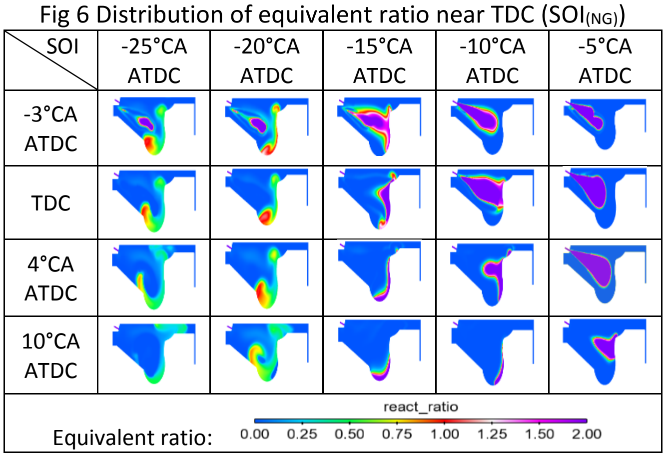



Fig 6 is the effect of injection timing ($P_{in}$= 0.16 MPa) on the microscopic level variation of the in-cylinder mixture near the TDC under the conditions of Table 5. It can be found that the too prior injection (-25~-20°CA ATDC) makes more combustible mixture enter the cylinder clearance while too late injection (-5°CA ATDC) may cause the late combustion and the power capability is reduced, both of which lead to incomplete combustion. The NG equivalence ratio($\varphi < 0.3$)is relatively low under the low load conditions. At the injection timing of -15~-10°CA ATDC, a concentration zone of NG can be formed near the TDC, which is superimposed with that of the diesel fuel. The superposition of the two zones is easier for the pilot diesel fuel to ignite, which can drive the rest zone to achieve better combustion effect.

Fig 6 Distribution of equivalent ratio near TDC ($SOI_{(NG)}$)




*3.2 Influence of intake pressure and $SOI_{(NG)}$ on combustion process*

Fig 7 shows the effect of intake pressure on Turbulent Kinetic Energy (TKE). It can be found that when the intake pressure is increased, the squeez flow motion and the turbulent motion in the cylinder under low load conditions are enhanced, the jet momentum of NG and the activity of charge in cylinder are also increased, leading to the acceleration of the diesel spray breaking. As a consequence, the mixing of the diesel fuel becomes better, which can advance the combustion.

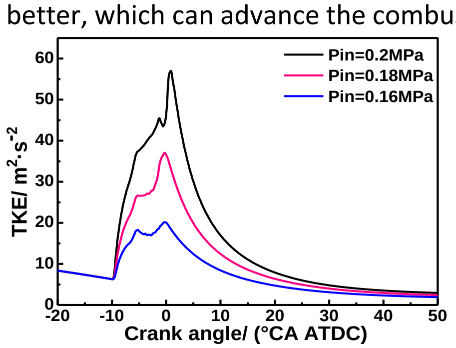



Fig 7 The effect of intake pressure on TKE in cylinder

Fig 8 shows the effect of injection timing on the combustion phase for different intake pressure. In combination with Fig 5, the higher intake pressure is, the more uneven stratification near the TDC will be, which is beneficial to the combustion reaction under low load conditions, and the combustion phase is advanced.

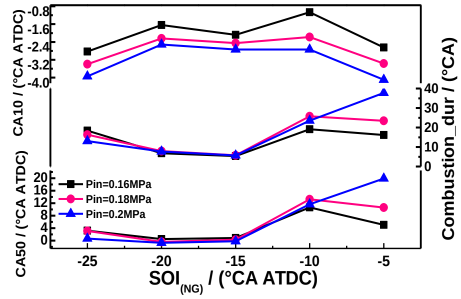



Fig 8 The effect of injection timing on combustion phases under different intake pressure

Fig 9 shows the microscopic level changes in the cylinder temperature near the TDC at different NG injection timings. Delaying the NG injection timing will put off the combustion process. However, due to the existence of the concentration zone near the TDC under the NG injection timing of -15~-10°CA ATDC, the oxidation reaction of NG under low equivalence ratio conditions is strengthened, which speeds up the flame propagation speed, promoting the complete combustion reaction and combustion efficiency.

Fig 9 Temperature distribution near TDC($SOI_{(NG)}$)

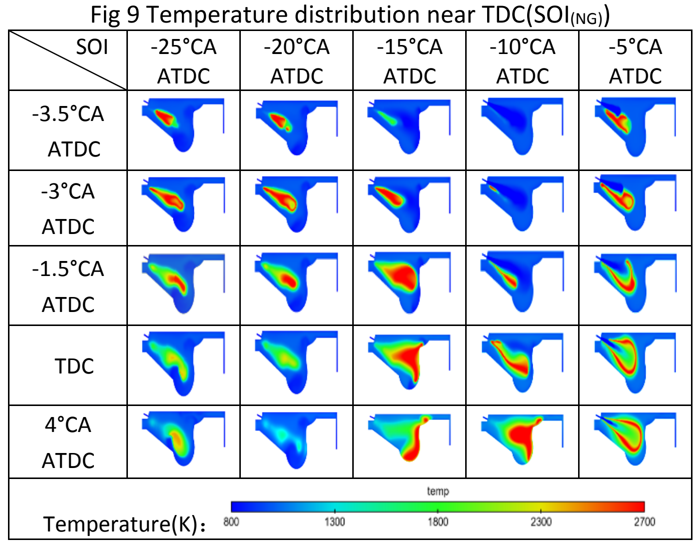



Fig 10 shows the effect of NG injection timing on temperature and pressure ($P_{in}$=0.2MPa). It indicates that the temperature and cylinder pressure are at high levels when the injection timing is -15~-10°CA ATDC. Combined with Fig 8, both cases maintain the injection and combustion processes simultaneously after ignition, promoting the combustion reaction.

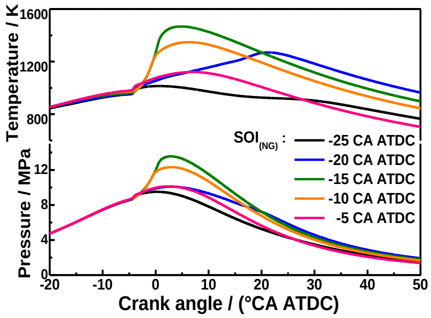


Fig 10 The effect of injection timing on pressure and temperature ($P_{in}$=0.2MPa)

### 3.3 *Influence of intake pressure and SOI$_{(NG)}$ on combustion efficiency*

Fig 11 shows the effect of intake pressure and NG injection timing on the indicated thermal efficiency (ITEg) of the combustion process. Too prior injection may cause more in-cylinder mixture enter the clearance, resulting in a sharp drop of the combustion efficiency. With the delay of injection timing, the overall ITEg is on the rise and remains at 45%~49%. When the intake pressure is increased, the mixing efficiency, the heat release rate, and ITEg are all improved.

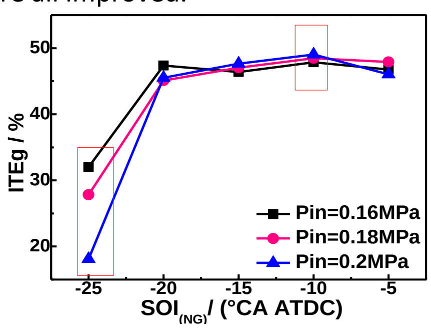


Fig 11 The effect of injection timing on ITEg under different intake pressure

### 3.4 *Influence of intake pressure and SOI$_{(NG)}$ on emission characteristics*

Fig 12 shows the effect of intake pressure and injection timing on four main emissions. Both CO and UHC maintain high emission levels when NG injection timing is too prior. At NG injection timing of -15~-10°CA ATDC, however, NOx emission comes to the peak because of the high combustion temperature required for NOx formation. Since the proportion of pilot diesel fuel in this study is small and the maximum combustion temperature in cylinder is lower than that of the traditional diesel engine, the overall SOOT emission level is low.

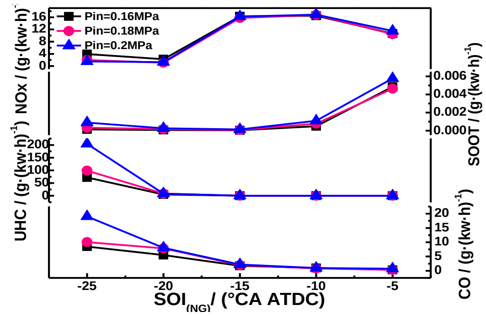


Fig 12 The effect of injection timing on main emissions under different intake pressure

## 4. CONCLUSIONS

(1) Under low-load conditions, the design of co-direct injection of diesel fuel and NG will superimpose the NG and pilot diesel concentration zones, which is beneficial to igniting the diesel fuel near the ignition point and achieving better combustion results.

(2) Advancing the intake pressure causes an increased jet flow motion resistance in cylinder and a shorter jet flow penetration distance, but the diesel fuel mixing is fuller, and the combustion is advanced.

(3) Too prior or late NG injection timing causes incomplete combustion and a decrease in ITEg. Therefore, a proper delay of the injection timing (-15~-10°CA ATDC) will promote local preferential and global complete combustion, resulting in the reduction of UHC and CO emissions, and the increase of NOx emissions.

(4) Without using EGR technology, the higher indicated thermal efficiency operating point (ITEg=49.03%) is obtained when the NG injection timing is -10°CA ATDC and the intake pressure is 0.2MPa on the condition of low levels of four main emissions.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