A Simulation Study of Water Injection Position and Pressure on the Knock, Combustion, and Emissions of a Direct Injection Gasoline Engine

Aqian Li, Zhaolei Zheng,* and Yukun Song

ABSTRACT: A turbocharged downsizing spark ignition (SI) engine combined with direct injection technology has the potential to improve the power and fuel economy and reduce emissions. However, gasoline engines are prone to knocking under low-speed and high-load conditions, which limits the application and development of downsizing SI engines. In this study, numerical simulation methods are used to explore the feasibility of water injection in the intake port to reduce the knock tendency of gasoline direct injection (GDI) engines and to explore the effects of different water injection pressures on combustion and emissions. First, the GDI engine is induced to knock by increasing the compression ratio and advancing the spark timing. Then, the influences of low position and no angle (LPNA) and high position and angled water injector arrangements on engine combustion are explored. When the water injector arrangement is LPNA, the turbulent kinetic energy near the spark plug is higher, the equivalence ratio is more evenly distributed, and the engine knock intensity is smaller. Finally, when the arrangement of the water injector is LPNA, the effects of water injection pressure on the knock, combustion, and emissions of the GDI engine are explored. The results show that when the water injection pressure is 5 bar, the knock intensity of the engine is the smallest, the cycle work is the highest, and the emissions of NOx and unburned hydrocarbon are the lowest.

1. INTRODUCTION

At present, the main purpose of gasoline engine research is to improve power performance and reduce emissions. To optimize the performance of gasoline engines, scholars have adopted a variety of methods. For example, Yusuf et al. added other blends to gasoline to improve engine performance.1−3 A turbocharged downsizing spark ignition (SI) engine cooperating with in-cylinder direct injection technology is one of the most effective ways to solve the above challenges. The gasoline direct injection (GDI) engine has the advantages of high efficiency, low fuel consumption, and low emissions.4−7 However, increasing the compression ratio of the turbocharged downsizing SI engine will increase the heat load and pressure in the cylinder, which may cause abnormal combustion called knock. Knock will cause the pressure in the cylinder to increase sharply, and the pressure will not be balanced enough to generate a high-pressure shock wave. Excessive pressure and high-frequency pressure changes will cause cracks and damage to the piston and other components. In severe cases, the entire gasoline engine may have to be scrapped. If the gasoline engine is operated under knock conditions for a long time, local overheating will cause some alloyed parts to soften or even melt, such as cylinder heads and pistons, so the knock phenomenon has greatly restricted the development of downsizing SI engines.8−10

Water injection is a simple and effective method to suppress knock. The specific heat capacity of water is large. Liquid water absorbs the heat in the cylinder to evaporate into superheated steam, which can effectively reduce the temperature and pressure and thus can effectively reduce the knock tendency.11,12 There are three types of NOx: thermal NOx, prompt NOx, and fuel NOx. Almost all NOx in gasoline engines is thermal NOx. The three elements of thermal NOx production are high temperature (including local high temperature), oxygen enrichment, and the duration of high temperature. Water injection can effectively reduce the temperature in the cylinder and thus reduce the peak temperature. In addition, the superheated steam fills the combustion chamber, diluting the air in the cylinder and reducing the possibility of partial oxygen enrichment. Therefore,
water injection can significantly reduce NO\textsubscript{x} emissions\textsuperscript{13−16}. The lower temperature in the cylinder also reduces the heat transfer from the cylinder wall to the exterior environment. If water is injected during the compression stroke, the heat absorbed by the water can reduce the compression work. Steam can also work with the working fluid on the piston during the expansion stroke. The reduction in knock strength allows for further improvements in the compression ratio and spark timing, which helps improve the power performance and fuel economy of gasoline engines\textsuperscript{17−19}.

There are three main water injection methods: intake port water injection, direct water injection in the cylinder, and fuel-water emulsion\textsuperscript{20−23}. Although the fuel-water emulsion can be directly injected into the cylinder by the fuel injection system, the fuel-water emulsion has some disadvantages: the emulsification process is complicated, the emulsifier is expensive, and the ratio of water to fuel is fixed and cannot be adjusted according to the working conditions\textsuperscript{24−26}. The advantage of direct water injection in the cylinder is that the water injection volume and water injection time can be flexibly adjusted. The disadvantage is that it is more complicated to add a separate water injection system to the cylinder. Intake port water injection occupies a dominant position in the current research. Its biggest advantages compared with the other two water injection methods are that it hardly changes the structure of the engine and the water injection time and quantity can be adjusted\textsuperscript{27,28}. In addition, for intake port water injection, the time between the water injection process and combustion is relatively long, and the water droplets can evaporate well and can also be mixed with the mixture. Intake port water injection also reduces the possibility of water droplets colliding with the cylinder, thereby reducing the problem of oil dilution. Zhuang et al.\textsuperscript{16} found that installing a water injector near the intake valve can reduce wall film formation. Combining water injection and proper advance ignition timing can improve the thermal efficiency of the engine without knock. When the engine speed is 4850 rpm, the thermal efficiency of the engine is increased by 6%; when the engine speed is 1500 rpm, the thermal efficiency of the engine is increased by 3.8%. Due to the low cost and easy installation of the intake port water injection system, the intake port water injection is considered to be the most practical solution for the implementation of water injection in SI engines. Therefore, this paper studies the intake port water injection method.

Tsao et al.\textsuperscript{29} tested the effects of water-fuel emulsions with different water contents on the power output, fuel consumption, thermal efficiency, and emissions of an SI engine. During the experiments, the engine speed was 2000 rpm, and the compression ratio was 7.5. When fuel with a water content of 5−15% was used, the power output increased by 3.8−14%, indicating that water-fuel emulsions are better than pure gasoline fuels in terms of fuel consumption. Zhang et al.\textsuperscript{30} explored the effects of different ratios of water-biodiesel emulsions on the combustion and emissions of medium-speed diesel engines through three-dimensional simulation. The results showed that adding water to fuel plays a very important role in the combustion process. Due to the micro-explosion, the mixing of fuel and air is improved. Adding a small amount of water to biodiesel can reduce NO\textsubscript{x}, CO, and CO\textsubscript{2} emissions.

Wang et al.\textsuperscript{31} studied the effects of intake port water injection on NO\textsubscript{x} emissions through simulation. The numerical prediction was consistent with the experimental measurement results. Liquid water enters the cylinder to absorb the heat released by combustion and the specific heat capacity of water is large, which reduces the combustion temperature. When the engine speed was 4000 rpm and the water-fuel mass ratio was 1, more than 80% NO\textsubscript{x} was removed, but the torque of the gasoline engine was reduced by 25%. Under low-speed and high-load operating conditions, intake port water injection has the potential to effectively reduce NO\textsubscript{x} emissions without severely sacrificing the torque output. Idris et al.\textsuperscript{32} explored the effect of steam injection from the intake port on the combustion and emissions of gasoline engines through experimental methods. It was found that when the quality of steam was 20% of the fuel, the performance and emissions of the engine were the best. The increase in the effective power reached 4.65% at the engine speed of 3200 rpm, the reduction in specific fuel consumption reached 6.44% at the engine speed of 2000 rpm, and the reduction in NO emissions reached 40% at the engine speed of 2800 rpm. Zhuang et al.\textsuperscript{33} explored the effect of the ratio of water injected in the intake port on engine knock and combustion through experiments. The results showed that intake port water injection effectively reduced the combustion temperature and reduced the knock tendency. The combination of inlet water injection and advancing spark timing prevented the engine from knocking, increased the thermal efficiency by 1.5%, and reduced the fuel consumption by 10 g/kWh. During the experiment, the best water injection ratio was 30%. The efficiency of the gasoline engine was improved and the emissions were relatively low.

Emre et al.\textsuperscript{34} investigated the effects of direct water injection in the cylinder on the performance and exhaust emissions of a six-stroke engine. The experimental results showed that the power output after water injection increased by 10% and the thermal efficiency increased by 8.72%. Furthermore, NO\textsubscript{x} and CO emissions decreased significantly. But the fuel consumption increased by 2%. Kahnooji and Yazdani\textsuperscript{35} explored the influence of the time and ratio of direct water injection in the cylinder on the RCCI engine through three-dimensional simulation. The results showed that direct water injection in the cylinder increased the maximum affordable load by reducing the maximum pressure rise rate (MPRR). When the water injection time was −10°CA and the ratio of water to fuel was 3, the engine worked best. The maximum pressure rise rate was reduced by 29%, but the engine power was reduced by 1.4% and the total emissions increased by 1%. By adjusting the injection angle and time, excessive emissions could be prevented.

In summary, the application of water injection technology to GDI engines has the potential to reduce the knock tendency, improve the fuel economy, and improve engine performance. Compared with direct water injection in the cylinder, the advantage of intake port water injection is that the structure of the engine stays almost the same. Compared with water-fuel emulsion, the advantage of intake port water injection is that the ratio, time, and angle of the water injection can be flexibly adjusted. For these reasons, intake port water injection is investigated in this study. Some research has been conducted on intake port water injection, but the effects of intake port water injection technology on knock, combustion, and emissions have not been unified. Three specific topics lacking in the current literature are as follows: (1) the position, pressure, and frequency of intake port water injection affect the combustion of the engine, but their specific impacts have not yet been unified. (2) The mechanisms of the physical and chemical reactions between water and fuel in the cylinder are not yet clear. (3) Water has an impact on engine emissions, but there is no uniform conclusion on its specific impact. Therefore, further
research on intake port water injection is needed. With the continuous development of computer technology, computational fluid dynamics, and combustion theory, numerical simulation has become as important as experimentation for studying internal combustion engines. The biggest advantage of numerical simulation is that all the microscopically effective information of the flow field in the cylinder can be obtained. In this paper, a turbocharged downsizing GDI engine is used as a numerical simulation object. First, two water injector arrangements (LPNA and HPA) are designed. Comparing the distribution of water in the cylinder and the distribution of each field before spark timing under the two working conditions, the effects of different water injector arrangements on combustion and knock are explored. Then, choose the arrangement of the water injector as LPNA to explore the influence of water injection pressure on each field in the cylinder and explore the influence of water injection pressure on combustion, knock, and emission.

2. RESULTS AND DISCUSSION

2.1. Verification of Numerical Simulation. 2.1.1. Geometric Model and Calculation Model of the GDI Engine. The

![Figure 1. Geometry model of the GDI engine.](image1)

Table 1. Basic Parameters of the GDI Engine

| parameter                  | numerical value |
|----------------------------|-----------------|
| number of cylinders        | 1               |
| bore (mm)                  | 76              |
| stroke (mm)                | 82.6            |
| compression ratio          | 9.5             |
| connecting rod length (mm) | 139.3           |
| displacement (L)           | 0.5             |

Table 2. Operating Conditions of the GDI Engine

| parameter                  | numerical value |
|----------------------------|-----------------|
| speed (r/min)              | 2000            |
| fuel injection time (°CA)  | 390             |
| fuel injection duration (°CA) | 140        |
| injection quantity (kg)    | 6.256 × 10⁻³    |
| spark time (°CA)           | 725.7           |

Table 3. Initial Conditions

| parameter                        | numerical value |
|----------------------------------|-----------------|
| cylinder temperature (K)         | 1182.44         |
| cylinder pressure (Pa)           | 273,871.0       |
| composition of various substances in the cylinder | N₂, O₂ (quantity score is 0.77, 0.23) |
| inlet temperature (K)            | 326.936         |
| inlet pressure (Pa)              | 173,196.74      |
| exhaust temperature (K)          | 1029.0          |
| exhaust pressure (Pa)            | 220,310.74      |
| turbulent kinetic energy (m²/s²) | 1               |
| turbulent energy dissipation (m²/s²) | 100       |

Figure 2. Number of calculation model meshes.

Table 4. Selection of Submodels

| model                        | setting |
|------------------------------|---------|
| turbulence model             | RNG K-ε model |
| breakup model                | RT-KH (Rayleigh Taylor–Kelvin Helmholtz) breakup model |
| collision model              | NTC (no time counter collision) model |
| fuel wall model              | wall film model |
| combustion model             | SAGE model |
| nitrogen oxide model         | extended Zeldovich model |
| soot model                   | Hiroyasu model |

Figure 3. Comparison of calculation results of different grid sizes.
geometric model, basic parameters, and operating conditions of the GDI engine are shown in Figure 1 and Tables 1 and 2, respectively.

Boundary conditions mainly include the boundary form (moving/fixed wall, intake port, or exhaust port), pressure, temperature, and other parameters of each part of the engine. For example, the piston top temperature, combustion chamber wall temperature, inlet wall temperature, and exhaust wall temperature at the beginning of the calculation are 565.9, 565.0, 350.0, and 1064.65 K, respectively. The initial conditions mainly refer to parameters such as pressure, temperature, turbulent kinetic energy, and turbulent kinetic energy dissipation in the cylinder, intake port, exhaust port, and other components. The piston and valve are set as moving boundaries, and the rest are set as fixed walls. The calculation start time is 331°CA, at which time the intake valve is about to open and the exhaust valve is about to close. The initial conditions of the model are shown in Table 3.

The grid of the intake and exhaust channels is slightly thicker. The meshes of pistons, intake valves, exhaust valves, spark plugs, cylinder heads, and other parts are thinner. Adaptive mesh refinement is adopted for the temperature and pressure in the entire cylinder. The grid can be coarsened or encrypted at a specified time and space, which can save time while ensuring accurate calculations. When the pressure or temperature gradient in the cylinder exceeds the set value, this area will automatically encrypt the grid. Under comprehensive consideration, the basic size of the grid in this paper is 0.1875−4 mm.

In the numerical simulation, the top dead center of the exhaust stroke is defined as 360°CA, and the top dead center of the compression stroke is defined as 720°CA. Figure 2 displays the change in the grid number of the model. As can be seen from Figure 2, the peak value of the grid number is about 1.7 million.

When a numerical model is established, various submodels need to be determined, such as the combustion model, spray model, emission model, and turbulence model. There are a variety of fluid interactions in the cylinder, including not only fluid flow but also chemical reactions, flame propagation, and complex phenomena such as heat and mass transfer. To accurately describe the changes in the cylinder, it is necessary to select appropriate submodels. The combustion model selected is the SAGE model. It can couple various chemical reaction mechanisms well and also has a fast calculation speed. The composition of gasoline is complex, and a large number of components will lead to a large number of reactions, products, and their kinetic parameters. At present, it is impossible to express the complex chemical reaction of full-blended gasoline in a detailed chemical kinetic model, so it is unrealistic to directly develop the chemical kinetic mechanism of gasoline. Gasoline usually contains linear alkanes, branched alkanes, cyclic alkanes, aromatics, olefins, and oxygen-containing components, and it has corresponding representative components such as n-heptane, isooctane, and toluene. Therefore, a mixture of one or several representative components can be regarded as a gasoline alternative fuel. This paper chooses the chemical reaction kinetic mechanism constructed by Song et al. This mechanism contains 113 components and 201 reactions, which can describe the combustion process in the engine cylinder well. The volume ratios of n-heptane, isooctane, and toluene are 17, 69, and 14%, respectively. The selection of submodels is shown in Table 4.

2.1.2. Model Verification. To determine whether the numerical simulation model established in this paper can
accurately reflect the actual operating conditions of the GDI engine, it is necessary to verify the model. Under low-speed and high-load conditions, the gasoline engine is prone to knocking, so the throttle valve is fully opened in numerical simulation and experiments.

According to the geometric model of the engine, the basic grid sizes are selected as 0.25−8 mm, 0.1875−4 mm, and 0.125−2 mm and named as case 1, case 2, and case 3, respectively. Numerical calculation of cold flow has been performed for three cases (no fuel injection model and combustion model). The initial conditions and boundary conditions are as shown above. The calculation result of the average pressure in the cylinder is shown in Figure 3. The average pressures of case 1, case 2, and case 3 are quite different. But the difference between the average pressures of case 2 and case 3 is very small. It can be considered that when the basic grid size is less than or equal to 0.1875−4 mm, the result of the numerical simulation has nothing to do with the grid size. Considering the accuracy and time of the calculation, the basic size of the grid is selected as 0.1875−4 mm.

To verify the accuracy of the established model, the model was run under experimental conditions. The start time of the calculation is 331°CA and the end time is 1051°CA. Figure 4 is the comparison of the experimental and numerical simulation of the average in-cylinder pressure profiles. It can be seen that the experimental results are very close to the simulation results, and the peak pressure difference is less than 5%. The ignition delay time is defined as the time from the moment of spark timing to the moment when the pressure in the cylinder starts to depart from the pure compression line. The difference in ignition delay time between the experimental result and the simulation result is only 1.3°CA. It can be considered that the model can effectively simulate the operation of a GDI engine.

To better verify the accuracy of the model, a set of 5600 rpm operating condition data has been added. The specific operating conditions are shown in Table 5. It can be seen from Figure 5 that the results of the simulation calculation are consistent with the experimental results. The model can simulate the actual operating conditions of the GDI engine well.

2.2. Various Fields under No Water Injection Conditions. According to some literature, when the equivalence ratio in the cylinder is 0.9 to 1.1, the knock trend is the most obvious. The equivalence ratio under the experimental operating conditions is 1.1, so the equivalence ratio in the simulation is consistent with the experimental condition and is also selected as 1.1. The compression ratio and spark timing have a direct impact on knock. Increasing the compression ratio and advancing spark timing can improve the power performance and fuel economy of the gasoline engine but also increase the knock tendency. Compared with other working conditions, low-speed and high-load working conditions are more prone to knocking. To study the knock conditions, under the experimental conditions, increase the compression ratio and advance the spark timing to induce knock. Then, explore the specific location and influencing factors of knock and lay the foundation for the subsequent research on the suppression effect of intake port water injection on knock. Therefore, under the condition that the given initial, boundary conditions, fuel injection quantity, and other parameters remain unchanged, only the compression ratio is adjusted from 9.5 to 10.2, and the spark timing advances to 704°CA. The other conditions are consistent with the experimental operating conditions, and the specific parameters are the same as above.

In this paper, the knock index (KI) is used to evaluate the intensity of knock. The expression of this parameter is: \[ KI = \frac{1}{N} \sum_{i}^{N} PP_{\text{max,n}} \]
In the above formula, KI represents the intensity of knock, N represents the number of monitoring points, and $PP_{max,n}$ represents the absolute value of the difference between the average pressure in the cylinder and the peak pressure of the
monitoring points. When KI is greater than 2, the gasoline engine knocks; when KI is less than or equal to 2, the gasoline engine does not knock. To evaluate the knock intensity of the engine, it is necessary to set monitoring points in the cylinder to monitor the temperature, pressure, and other parameters of each monitoring point. Combining the pressure at each monitoring point and the average pressure in the cylinder, the knock intensity can be calculated using the above formula. The cause of the knock is that the terminal mixture spontaneously ignites before the flame spreads. Therefore, the area near the wall surface of the combustion chamber far away from the spark plug is more prone to knocking. The monitoring points are all set in the area near the wall of the combustion chamber, and the specific position is shown in Figure 6.

2.2.1. Analysis of the Various Field Diagrams in the Cylinder. Figure 7 shows the in-cylinder pressure temperature, equivalence ratio, and turbulent kinetic energy distribution before spark timing. It can be seen from Figure 7a,b that the temperature and pressure distribution in the cylinder before the spark timing is relatively uniform, and it can be considered that the pressure and temperature field before the spark timing has little effect on the knock. As shown in Figure 7c, the equivalence ratios at most regions are about 1.1, while the equivalence ratio at the center of the cylinder along the Y-axis direction is concentrated at 1.3–1.5. Compared with other monitoring points, the equivalent ratio near the monitoring point 7 is the largest, while the equivalent ratio in the monitoring point 2 and 8 areas is lower than 0.9. When the equivalence ratio is relatively small, the flame propagation speed first increases as the equivalence ratio increases; when the equivalence ratio exceeds 1.1, the flame propagation speed decreases as the equivalence ratio increases. If the equivalence ratio is too large or too small, then the flame propagation speed will decrease, resulting in an increase in the time for the flame to propagate to the terminal mixture, thereby increasing the possibility of knock. It was preliminarily judged that the area near monitoring points 2, 7, and 8 was prone to knocking.

Figure 7d shows that the turbulent kinetic energy of the combustion chamber wall before the spark timing is significantly lower than the turbulent kinetic energy of the cylinder center. The reason for this phenomenon is that the mixture in the central area is more interfered by the upward movement of the piston during the compression stroke. The greater the turbulent kinetic energy, the greater the intensity of turbulence and the more uniform the mixing of fuel and air.

2.2.2. Pressure at Different Monitoring Points. Figure 8 shows the average in-cylinder pressure profiles and the pressure at each monitoring point under knock conditions. According to the formula of KI, the knock index is 7.87, which is greater than 2 and indicates that knock has occurred in the cylinder. It can be seen from the figure that the peaks of monitoring points 2, 7, and 8 are significantly larger than the peak of the average pressure, and the fluctuation of these three monitoring points is more intense. In particular, the pressure peak at monitoring point 8 is the largest, and the fluctuation is also the most intense. Combined with Figure 7, the reason why the pressure fluctuations of these three monitoring points are more severe is as follows. The equivalent ratio near the monitoring points 2

![Figure 9. Schematic diagram of the arrangement of the water injector.](image)

![Figure 6. Water Injection Parameters](image)

| parameter                     | numerical value |
|-------------------------------|-----------------|
| water injection temperature (K) | 298             |
| water injection time (°CA)     | 360             |
| water injector position        | low position and no angle (LPNA)/high position and angled (HPA) |
| water injection pressure (bar) | 5.0             |
| water injection quantity (mg)  | 19.53           |

![Figure 10. Influence of the position of the water injector on the movement of water droplets](image)
and 8 is too small and the equivalent ratio near the monitoring point 7 is too large. If the equivalent ratio is too large or too small, then it is not conducive to flame propagation. Therefore, whether water injection can improve the equivalence ratio distribution in the cylinder to suppress knock is the focus of the following research.

2.3. The Influence of Water Injector Position on Knock Suppression. The position of the water injector in the intake port and the water injection port will affect the knock, combustion, and emissions of the GDI engine. The most direct effect is that water entering the cylinder will affect the temperature field and flow field. In this study, considering the structure of the intake port and the shape of the spray, two water injector arrangement schemes are designed. One arrangement of the water injector is high position and angled (HPA). The water injector is away from the intake valve, and the injection direction has a certain deflection angle toward the intake valve. Another arrangement of the water injector is low position and no angle (LPNA). The position of the water injector is close to the intake valve, the direction of the water injector is horizontal, and there is no tilt angle toward the intake valve. The schematic diagram of the location of different water injectors is shown in Figure 9.

The parameters of the water injector are shown in Table 6.

2.3.1. The Influence of Water Injector Position on Water Movement. As shown in Figure 10, when the water injection pressure is 5.0 bar, the water droplet motion diagram is under the two arrangements of low position and no angle (LPNA-5.0) and high position and angled (HPA-5.0). The blue substance represents water, and the red substance represents fuel. It can be seen that the closer the water injector is to the cylinder, the farther the water enters the cylinder and the less water is left outside the cylinder. The position of the water injector is far away from the cylinder, and there is more room for water droplets to develop. As a result, the water droplets are more likely to collide with the intake port. At 480°CCA, it can be seen that most of water has been injected into the cylinder when the arrangement of the water injector is low position and no angle (LPNA-5.0).

2.3.2. Influence of the Position of the Water Injector on the Equivalent Ratio and Turbulent Kinetic Energy. Figure 11 shows the distribution of turbulent kinetic energy and equivalent ratio at different injection positions before spark timing. From Figure 11a, it can be seen that the turbulent kinetic energy of the vicinity of the spark plug is higher when the arrangement of the water injector is LPNA, which means that the intensity of turbulence is greater, and the mixing of air and fuel is more adequate. It can be seen from Figure 11b that when the arrangement of the water injector is HPA, there is a part of the cylinder in which the equivalent ratio value exceeds 1.6. It is not conducive to the flame propagation, which leads to an increased possibility of spontaneous combustion of the terminal mixture. The LPNA of the water injector arrangement scheme effectively improves the distribution of the equivalent ratio in the cylinder before spark timing.

2.3.3. Influence of Water Injector Position on the Knock Tendency. Figure 12 shows the pressure of each monitoring point under different water injector positions. It can be seen from the figure that the peak pressure of the monitoring point under LPNA-5.0 is significantly lower than that of HPA-5.0 and the fluctuation trend of each monitoring point under LPNA-5.0 is obviously smaller (except for the monitoring point 6). This shows that when the arrangement of the water injector is LPNA, the knock tendency of the GDI engine is smaller. Figure 13a shows the peak pressure of each monitoring point under different working conditions. It can be clearly seen that the peak pressure under LPNA-5.0 is smaller (except for the no. 6 monitor). The knock intensity under different working conditions can be calculated from the previous equation, as shown in Figure 13b. The KI value under LPNA-5.0 is significantly lower than that under HPA-5.0. From the previous analysis, it can be seen that compared with HPA-5.0, the turbulent kinetic energy in the cylinder under LPNA-5.0 is greater, and the equivalent ratio distribution is more uniform. Large turbulent kinetic energy and uniform equivalence ratio distribution are beneficial to increase the flame propagation speed, which effectively suppresses knock.

2.3.4. The Influence of Water Injector Position on Cyclic Power. Figure 14 shows the $P-V$ diagram under different water injector positions. The area of the closed curve in the $P-V$ diagram represents the cyclic work. Integrating the closed curve in the $P-V$ diagram can obtain the cyclic work under different working conditions. The cyclic work under HPA is 1087.0 J; the cyclic work under LPNA is 1102.26 J. The cyclic work under LPNA-5.0 is slightly higher than that under HPA-5.0, but the difference is not obvious. Combined with the previous analysis, it can be seen that the turbulent kinetic energy near the spark plug under LPNA-5.0 is greater and the equivalent ratio...
distribution is more uniform, which all contribute to flame propagation. In addition, the knock intensity of LPNA-5.0 is significantly lower than that of HPA-5.0, which also helps improve the power performance of the GDI engine. The main

Figure 12. Pressure at monitoring points of different water injector positions ($n = 2000$ rpm; spark timing, $704^\circ$CA; compression ratio, 10.2).
reasons are as follows: when knock occurs, part of the chemical energy of the fuel is not converted into useful work to push the piston but is consumed in the collision of the pressure shock wave with the cylinder wall, piston, and other components. Due to the impact of the strong pressure wave, the gas surface layer (a kind of gas surface layer formed on the walls of the combustion chamber, the top of the piston, and the cylinder wall) is destroyed, and the heat transfer of the high-temperature gas to these walls greatly increases. Therefore, the cyclic work under LPNA-5.0 is slightly higher than that under HPA-5.0.

2.4. The Effect of Water Injection Pressure on the Knock, Combustion, and Emissions of the GDI Engine.

Comparing the parameters of the two working conditions of LPNA-5.0 and HPA-5.0, it can be seen that when the water injector arrangement is LPNA, the distribution of the equivalence ratio before spark timing is well improved. The knock trend of the GDI engine has also decreased significantly. In addition, the cyclic power of LPNA-5.0 is also larger than that
of HPA-5.0. Therefore, when the water injector arrangement is LPNA, the influence of water injection pressure on water droplet movement, velocity field, and equivalence ratio distribution is explored. Then, explore the influence of water injection pressure on suppressing the knock tendency. Under the LPNA water injection arrangement scheme, the water injection pressures are selected to be 2.5, 5, and 7.5 bar and these three working conditions with different water injection pressures are named LPNA-2.5, LPNA-5, and LPNA-7.5, respectively.

2.4.1. Water Movement and Various Fields under Different Water Injector Pressures. The most direct result of different injection pressures is the different speeds of water droplets. When water droplets are injected into the cylinder, its own speed is very large. The movement of water will affect the flow field in the cylinder. Due to the limited penetration length of water droplets, the speed of water droplets gradually decreases after entering the cylinder. Then, the turbulent flow in the cylinder dominates.

Figure 15 shows the movement of water and fuel in the intake port and cylinder during 400–610°CA under different pressures. It can be seen that as the injection pressure increases, the time required for water to enter the cylinder decreases. The intake valve is about to close at 530 °C, and almost no more water will enter the cylinder. Under the conditions of different water injection pressures, there is some water at the top of the valve and at the bottom of the intake port. This is because water collides with the intake port and fails to enter the cylinder. Figure 16a shows the in-cylinder Sauter mean diameter (SMD) under different water injection pressures. It can be seen from Figure 16a that increasing the injection pressure can reduce the SMD in the cylinder, thereby improving the atomization of the droplets. Under LPNA-2.5, the atomization of water is the worst, so the evaporation is the slowest. Water and air are injected into the cylinder from the intake port together, which not only interacts with the fuel in the cylinder but also forms a convective heat exchange with the cylinder wall. Increasing the low velocity and turbulence intensity of the mixed gas in the cylinder can make the heat transfer effect better. When the water injection pressure is 7.5 bar, the speed of water is the largest and the atomization is the best, so the evaporation of water is the fastest.

Figure 16b shows the quantity of liquid water entering the cylinder. It can be seen that the quantity of liquid water entering the cylinder under different water injection pressures is slightly different. The quantity of liquid water first increases with the increase in water injection pressure, and then the quantity of liquid water decreases with the increase in water injection pressure. When the water injection pressure is small (2.5 bar), increasing the injection pressure can speed up the movement of water in the intake port, thereby reducing the evaporation quantity of water in the intake port and increasing the quantity of liquid water entering the cylinder. When the water injection pressure is large (5.0 bar), increasing the injection pressure increases the spray cone angle, which increases the quantity of water colliding with the surface of the intake port and reduces the quantity of liquid water entering the cylinder.

The velocity field in the cylinder at 719–723°CA under different water injection pressures is shown in Figure 17. At 719°CA, the flow velocity is basically below 70 m/s and is basically symmetrical about the Y axis. When the water injection pressure is 7.5 bar (LPNA-7.5), the fluid velocity is maximum. At 720°CA, the LPNA-7.5 working conditions of the terminal mixture is the cylinder first spontaneously ignited, and the fluid velocity field changed drastically. At 721°CA, the terminal mixture also spontaneously ignited under LPNA-5.0. Finally, at 722°CA, the terminal mixture spontaneously ignited under LPNA-2.5. The fluid velocity field close to the cylinder wall under different water injection pressures has changed very drastically, but the timing and position of spontaneous combustion of the terminal mixture are different. It can be seen that the injection pressure has a relatively large influence on the flame propagation and the state of the terminal mixture.

It can be seen from Figure 18 that different injection pressures have a significant effect on the equivalence ratio. At 703°CA, under LPNA-2.5, the equivalent ratio near the spark plug is the largest and the equivalent ratio exceeds 1.5. At 720°CA, the equivalent ratio of LPNA-5.0 and LPNA-7.5 is mostly between 1.1 and 1.3. Under LPNA-2.5 working conditions, the equivalence ratio of a large part of the area exceeds 1.3. At 703 and 720°CA, the distribution of equivalence ratios in the LPNA-5.0 and LPNA-7.5 working conditions is more uniform than that under LPNA-2.5 working conditions. Therefore, the flame propagation speed under LPNA-2.5 will be significantly slower than the flame propagation speed under LPNA-5.0 and LPNA-7.5. Water injection pressure can affect the perturbation...
intensity of water to the mixture in the cylinder, which in turn affects the distribution of the equivalence ratio.

2.4.2. Effect of Water Injection Pressure on Combustion. Figure 19a shows the average in-cylinder pressure profiles under different water injection pressures. Because the flame spreads the fastest under LPNA-7.5, the average pressure in the cylinder rises the fastest. The pressure increase rate and pressure peak under LPNA-5.0 are relatively close to those under LPNA-7.5. The flame propagation speed is the slowest under LPNA-2.5, so the rate of pressure rise is the slowest. Figure 19b shows the average in-cylinder temperature profiles under different water injection pressures. It can be seen that the trend of the temperature under different water injection pressures is basically the same, but the temperature peak and the moment of reaching the peak are slightly different. This is caused by the difference in the heat release rate of the combustion under different water injection pressures. The cumulative heat release under different pressures is shown in Figure 20a. The sooner the cumulative heat release reaches the peak, the sooner the temperature reaches the peak. It can be seen from Figures 19b and 20a that the change trend of the temperature is roughly consistent with the change trend of the cumulative heat release. During the period of $700^\circ$-$720^\circ$CA, the cumulative heat release is in the order LPNA-7.5 > LPNA-5.0 > LPNA-2.5. The average temperature during the corresponding period has the same conclusion.

Define the combustion duration as follows: The starting point of combustion is the crank angle corresponding to 10% of the cumulative heat release, and the crank angle corresponding to 90% of the cumulative heat release is the end point. The time interval between two crank angles is the combustion duration. The cumulative heat release curve under different water injection pressures is shown in Figure 20a. It can be found that the cumulative heat release at $725^\circ$CA is basically 2280 J, which means that the different water injection pressures have little effect on the cumulative heat release. From the above definition, the combustion duration under different water injection pressures is about $6^\circ$CA. This shows that the influence of water injection pressure on the combustion duration is also not obvious.
Figure 20b shows the instantaneous heat release rate under different water injection pressures. It can be seen from the figure that the peak value of the instantaneous heat release rate under LPNA-2.5 is 1800 J/°CA and the peak heat release time is relatively lagging compared to other conditions. This is because the effect of water atomization under LPNA-2.5 is relatively poor, the diameter of water droplets is relatively large, and the mixing with fuel is not uniform. The relatively large water droplets block the flame propagation speed at the initial stage of combustion. However, under some special circumstances (related to the diameter, mass, and speed of the water droplets), the water droplets can accelerate the combustion of the mixture and strengthen the combustion flame. When the water droplets are in contact with the flame, the flame propagation speed will be accelerated within a certain time. When the mass of water droplets is not enough to extinguish the flame, the flame will continue to propagate until the end of the combustion. Some studies have shown that the main reason for flame strengthening is the “azeotropy” phenomenon caused by the water droplets reaching the fuel surface in the early stages of combustion. For two incompatible liquids, the boiling point of the system is always lower than that of any single component. The phenomenon where two or more incompatible liquids start to boil below the boiling point of each single-phase component is called azeotropy. The reason why the fine water droplets enhance flame combustion is the overpressure generated by the sharp boiling after the fine water droplets reach the fuel surface. Overpressure greatly increases the boiling rate of the fuel, which intensifies the combustion. When the amount of water droplets is small, the temperature gradient on the fuel surface is relatively small. The cooling effect of fine water droplets hitting the fuel surface and entering the interior is very small, and the boiling rate is still very high, so there is a flame strengthening effect.
The flame propagation process under different water injection pressures is shown in Figure 21. The flame propagation speed is the slowest under LPNA-2.5. It can be seen from Figure 15 that the water distribution under LPNA-2.5 is relatively concentrated. Moreover, the diameter of the water droplets is larger under LPNA-2.5. Therefore, the presence of water under LPNA-2.5 hinders the flame propagation speed in the initial stage of combustion. However, under LPNA-5.0 and LPNA-7.5 conditions, the water distribution is relatively uniform and the diameter of the water droplets is relatively small. During the propagation of the flame, fine water droplets approach the surface of the fuel and evaporate in the high-temperature and high-pressure environment in the cylinder to form an overpressure environment on the fuel surface, thereby enhancing the boiling of the fuel and enhancing the flame combustion. The flame propagation speed from large to small is in the order LPNA-7.5 > LPNA-5.0 > LPNA-2.5.

Figure 22a shows the $P-V$ diagram under different water injection pressures. Integrate the closed curve of Figure 22a to obtain the cyclic power under different water injection pressures, as shown in Figure 22b. The cyclic power of LPNA-5.0 is the largest, and the cyclic power is 1102.26 J. The cyclic power under LPNA-2.5 is the smallest, and the cyclic power is 1091.1. The difference of the cyclic power under different water injection pressures is not obvious.

2.4.3. Influence of Water Injection Pressure on Suppressing the Knock Tendency. Figure 23 shows the pressure profiles of the eight monitoring points set in the cylinder under different water injection pressures. Combined with the pressure of the eight monitoring points when water is not injected in Figure 8, it can be seen that under the LPNA-5.0 and LPNA-7.5 conditions, the pressure peaks are significantly reduced. Figure 24 shows the KI under different water injection pressures. The KI is 7.87 when water is not injected, and the KI values are 2.9 and 3.9 under LPNA-5.0 and LPNA-7.5 conditions, respectively. Therefore, LPNA-5.0 and LPNA-7.5 conditions can effectively suppress the knock tendency. However, LPNA-5.0 conditions have a stronger effect on suppressing knock than LPNA-7.5. This is because when the water injection pressure is 7.5 bar, the collision of water with the intake port during the movement causes a
considerable part of water to be left outside the cylinder, resulting in the amount of water entering the cylinder being less than that when the water injection pressure is 5 bar (LPNA-5.0). Under LPNA-2.5, the pressure fluctuations of monitoring points 2 and 8 are still obvious, even the peak pressure of monitoring point 2 exceeds 35 MPa, and monitoring point 8 pressure exceeds 30 MPa. The KI under LPNA-2.5 is 6.5. Compared with the case without water injection, although the knock tendency is weakened under LPNA-2.5, the suppression effect of water injection on knock is not obvious. This is mainly because the water distribution under LPNA-2.5 is uneven and the diameter of the water droplets is relatively large. Water in the cylinder suppresses the flame propagation, resulting in uneven flame propagation.

2.4.4. Emission Performance under Different Working Conditions. Figure 25a shows the quantity of NOx in the process of combustion under different water injection pressures. It can be seen from the figure that the trend of the NOx generation quantity in the cylinder under different water injection pressures is basically the same. NOx increases rapidly and then decreases slowly until it stabilizes. This is determined by the NOx generation mechanism. Nitrogen oxides are mainly NO during the combustion of gasoline engines. High-temperature NO is the main source of nitrogen oxide emissions from the GDI engine. In this paper, the emission mechanism of nitrogen oxides is selected as the extended Zeldovich mechanism. The three elements of NO production are high temperature (including local high temperature), duration of high temperature, and oxygen enrichment (including local oxygen enrichment), all of which are indispensable. In particular, the temperature in the cylinder has the most obvious influence on NOx emissions. The movement of water in the cylinder affects the temperature field in the cylinder, which in turn affects the emission of NOx.42,43 Among them, eqs 2–4 are all strong endothermic reactions, which can be carried out only at high temperatures greater than 1800 K. When the temperature is higher than 1800 K, the
Figure 23. Pressure profiles of each monitoring point under different water injection pressures ($n = 2000$ rpm; spark timing, $704^\circ$CA; compression ratio, 10.2).
reaction rate of NO will increase 6−7 times for every 100 K increase in temperature. Therefore, a high temperature and local high temperature create a good environment for NO\(_x\) generation. After the spark timing, the temperature increases sharply during the upward movement of the piston, so the quantity of NO generated increases rapidly. Since eqs 2−4 are reversible, the temperature will decrease during the downward movement of the piston, and the oxygen concentration will continue to decrease, so the reverse reaction dominates and the mass of NO decreases.

\[
\begin{align*}
N_2 + O_N + NO, & \quad \Delta H = 75 \text{ kcal/mol} \\
N + O_2O + NO, & \quad \Delta H = -31.4 \text{ kcal/mol} \\
N + O_2H + NO, & \quad \Delta H = 40.8 \text{ kcal/mol}
\end{align*}
\] (2-4)

Figure 25b shows the quantity of NO\(_x\) under different water injection pressures. It can be seen that the quantity of NO\(_x\) under LPNA-5.0 is the smallest. Temperature has the greatest influence on the formation of NO, so the temperature field in the cylinder is analyzed. Figure 26 shows the temperature distribution under different water injection pressures. It can be seen that under LPNA-2.5, the high temperature area is wider and the high temperature lasts longer. The reason is that compared with LPNA-5.0 and LPNA-7.5, when the water injection pressure is 2.5 bar (LPNA-2.5), the mixing of water and fuel is not uniform enough. Moreover, under LPNA-2.5, water has a relatively large particle size, which not only cannot intensify flame combustion but also hinders flame propagation. It can be seen from Figure 24 that the KI value of LPNA-2.5 is as high as 6.5. Compared with LPNA-5.0 and LPNA-7.5, the knock intensity under LPNA-2.5 is significantly greater. Water in the cylinder hinders the spread of the flame and easily induces knock in a local area of the cylinder, which makes the temperature of the local area increase sharply, which creates good conditions for the generation of NO. There is little difference in local high temperature distribution under LPNA-5.0 and LPNA-7.5. The specific reason why the quantity of NO\(_x\) under LPNA-5.0 is lower than that of LPNA-7.5 needs to be further explored. Figure 27 shows the O\(_2\) concentration distribution in the cylinder under different water injection pressures. It can be seen that during 704−717°CA, the oxygen-enriched area under
Figure 26. Temperature distribution in the cylinder under different water injection pressures ($n = 2000$ rpm; spark timing, $704^\circ$CA; compression ratio, 10.2).
LPNA-2.5 and LPNA-7.5 is wider than that under LPNA-5.0. Therefore, the NO\textsubscript{x} quantity under LPNA-7.5 is also higher than that under LPNA-5.0.

Figure 27. \textit{O}_\textsubscript{2} concentration distribution under different water injection pressures (n = 2000 rpm; spark timing, 704°CA; compression ratio, 10.2).

(a) The quantity of UHC under different crank angles

(b) Quantity of UHC

Figure 28. Quantity of UHC under different water injection pressures (n = 2000 rpm; spark timing, 704°CA; compression ratio, 10.2).

\textit{Figure 28a} shows the generated quantity of UHC in the process of combustion. The reduction in the quantity of hydrocarbons means that the fuel is consumed. The changing
trend of hydrocarbons in the cylinders under different water injection pressures is basically the same: first slowly decreasing, then rapidly decreasing, and finally stabilizing. Figure 8b shows the quantity of UHC in the cylinder at 740 °CA. Comparing the different water injection pressures, it can be seen that the quantity of UHC is the lowest under LPNA-5.0 and the greatest under LPNA-2.5. The main causes of UHC are incomplete combustion, so the quantity of UHC is the largest. In the two working conditions of LPNA-5.0 and LPNA-7.5, the particle size and combustion, wall quenching, and wall oil film. Especially, incomplete combustion increases UHC emissions.44,45 From the previous analysis, it can be concluded that the presence of water under LPNA-2.5 is not conducive to the propagation, and the uneven distribution is not conducive to flame combustion, so the quantity of UHC is the largest. In the two working conditions of LPNA-5.0 and LPNA-7.5, the particle size of water is small and the distribution is relatively uniform, so the presence of water enhances flame propagation.

3. CONCLUSIONS

1. Comparing the LPNA and HPA water injector arrangements, when the water injector arrangement is LPNA, the knock intensity of the GDI gasoline engine is smaller and the cycle work is greater.
2. The influence of water injection pressure on knock and combustion: when the water injection pressure is 5 bar, the knock intensity is the smallest and the cycle work is the largest.
3. The influence of water injection pressure on emissions: when the water injection pressure is 5 bar, NOx and UHC emissions are the lowest.

**AUTHOR INFORMATION**

**Corresponding Author**
Zhaolei Zheng — Key Laboratory of Low-grade Energy Utilization Technologies and System, Ministry of Education, Chongqing University, Chongqing 400044, China; orcid.org/0000-0003-2905-0549; Phone: +86-02-6510 2473; Email: zhengzhaolei@cu.edu.cn; Fax: +86-023-6510 2473

**Authors**
Aqian Li — Key Laboratory of Low-grade Energy Utilization Technologies and System, Ministry of Education, Chongqing University, Chongqing 400044, China
Yukun Song — AECC Sichuan Gas Turbine Establishment, Chengdu 610500, China

Complete contact information is available at: https://pubs.acs.org/10.1021/acsomega.1c01792

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