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A new future fuel injection system for CPGC four-stroke engine family in the 32 to 45cm bore segment

Fuel Injection & Gas Admission and Engine Components

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ABSTRACT

With the increasing concern from IMO about ship carbon emissions, mandatory regulations have been issued to limit and restrict the carbon emissions of ships. Therefore, seeking low-carbon and zero-carbon alternative fuels to replace traditional fossil fuels has become a very hot topic. At present, various new fuels such as methanol, ethanol, LPG, LNG, ammonia, hydrogen, DME, etc. have been technically proven to be available for the internal combustion engines, and which could not only replace traditional diesel fuel for engine combustion, but also reduce the carbon emissions of ships to varying degrees according to the type of fuel. However, compared with the traditional diesel fuel, the new low-carbon fuels have the common characteristics of low viscosity, low flash point, low calorific value, strong corrosion, strong toxicity and other properties. As a result, the fuel injection system needs to be customized individually.

CPGC (CSSC Power Group Co., Ltd) has been working on cleaner and greener marine power system developments since 2022. Based on 32cm, 39cm and 45cm large-bore dual-fuel engines, a new retrofit solution that focuses on a methanol conversion in the first stage has been designed and investigated, considering both PFI and HPDI technology. For the PFI solution, the complete multi-cylinder engine test was completed and AIP certified by CCS. For the HPDI solution, a patent new flexible fuel injector based on common rail injection has been introduced, which takes full advantage of traditional diesel fuel to control, lubricate, seal and cool the methanol injection system, so that no extra hydraulic system was introduced. The dual-fuel engine is characterized by only diesel and methanol circulation lines. In addition, due to the continuous nozzle cooling, the thermal load of the nozzle under both diesel and methanol combustion modes has been improved. Meanwhile, a special safety and tailored purging concept was investigated to make sure the new future fuel system meets regulation requirement and keeps the engine as safe as before. Additionally, this state-of-the-art solution makes an easy way to convert from dual-fuel gas engines to methanol engines.

This paper will introduce CPGC activities in future fuel development, which includes the future fuel system layout, 1D hydraulic performance, CAE & CFD simulation results, and the test results under the test rig and platform.

1 INTRODUCTION

With the gradual intensification of the requirements for ship carbon emissions by the International Maritime Organization, laws and regulations will be introduced in the future to restrict and constrain ship carbon emissions. Thus, seeking low-carbon and zero-carbon clean fuels as substitutes for traditional fuels has emerged as a highly critical and effective approach to address ship carbon emissions [1]. Methanol, being a low-carbon fuel, can effectively reduce carbon emissions and NO_x emissions. Furthermore, the advent of green methanol refining technology has greatly enhanced the accessibility and environmental friendliness of methanol throughout its entire life cycle. Methanol has already emerged as a highly promising clean alternative fuel for internal combustion engines and has been applied on a large scale in commercial vehicles [2]. In recent years, along with the continuous vigor of the methanol market, the research and development, as well as the promotion of marine methanol engines, have been progressing steadily. Nevertheless, due to the relatively low viscosity, flash point, and calorific value of methanol fuel, coupled with its corrosive and slightly toxic characteristics, higher demands have been imposed on the design of marine methanol fuel injection systems [3]. Conducting research on key technologies, integration, and application technologies of methanol power systems hold significant importance for enabling ship power to comply with emission regulations and the dual carbon goals and for enhancing market competitiveness.

The CPGC M320DM methanol engine is a green and low-carbon fuel engine launched based on the independently developed CPGC M320 series marine medium-speed engine platform. It is capable of using multiple fuels such as methanol, MGO, and HFO, and supports the operation in both methanol and diesel fuel modes. Under the methanol mode, the engine can reduce carbon dioxide emissions by 40% - 90% through the utilization of "green" methanol produced in a carbon-neutral manner, and simultaneously can also reduce nitrogen oxide emissions by 20% - 30%. In accordance with the demands of users' usage, two types of products can be provided: inlet port fuel injection (PFI) and in-cylinder high pressure direct injection (HPDI). The inlet port fuel injection methanol engine (as shown in Figure 1) is characterized by the installation of methanol nozzles on the intake manifold. The injection quantity and timing of methanol are controlled by an electronic control unit to achieve the mixture of methanol and air and the subsequent entry into the cylinder for combustion. Currently, the inlet port fuel injection methanol engine has been successfully developed and achieved the first order breakthrough. The high pressure in-cylinder direct injection methanol engine (as shown in Figure 2) is characterized by the direct injection of methanol fuel into the cylinder through a self-developed high pressure injection system and injector. After mixing with high-temperature and high-pressure air, it is ignited through either spark ignition or compression ignition to generate power and drive the piston movement. This type of product is still in the stage of experimental iteration. Among them, the inlet port fuel injection technology is relatively mature, has a simpler structure, and generates less carbon deposits; while the in-cylinder direct injection product has more complex technology and higher maintenance costs, but offers a higher methanol substitution rate, faster power response, and higher fuel efficiency. Both types of products have their respective advantages and disadvantages, and the selection should be based on specific application scenarios and requirements.

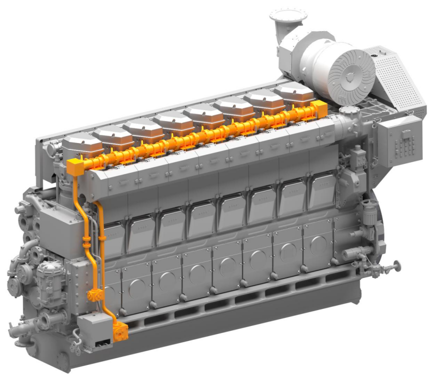


Figure 1. CPGC M320DM low pressure inlet port fuel Injection methanol-diesel dual fuel main engine

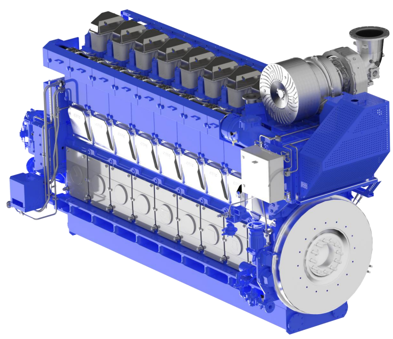


Figure 2. CPGC M320DM high pressure direct injection methanol-diesel dual fuel main engine

2 CHALLENGES AND SOLUTIONS

2.1 PFI System

2.1.1 Introduction to the PFI System

The methanol injection system for the CPGC low pressure inlet port dual-fuel main engine (as shown in Figure 3) is mainly composed of a low-pressure methanol common rail and electronically controlled methanol injectors. The working pressure of the low-pressure methanol common rail ranges from 4 to 10 bar, and it adopts a double-wall design. The inner tube is filled with low-pressure methanol during the methanol operation mode. The annular space between the inner and outer tubes is maintained under negative pressure ventilation to ensure an inert environment. Pressure and temperature sensors are installed in the inner tube of the methanol common rail, and negative pressure sensors and gas concentration detectors are placed between the inner and outer tubes, enabling real-time monitoring of the methanol supply and leakage status. In the event of an anomaly detection, it can switch to the diesel mode immediately. Unlike the mainstream low-pressure methanol supply systems, the CPGC low-pressure methanol system incorporates bellows suitable for the methanol medium. This design enhances the flexibility of the methanol supply pipeline and effectively reduces the risk of methanol leakage caused by the vibration of the main engine.

Figure 3. Inlet port fuel Injection system of methanol

The low pressure electronically controlled methanol injectors of the CPGC low pressure inlet port dual-fuel main engine are procured from domestic suppliers, as depicted in Figure 4. This low-pressure methanol injector is primarily constituted by a filter, a solenoid valve, a needle mating component, a needle spring, a nozzle assembly, a shell and corresponding sealing elements. The moving pairs within the injector employ the DLC coating process, which can effectively reduce the friction coefficient of the moving components during the injection process and obviate the introduction of an additional lubricating oil system.

Figure 4. Inlet port methanol injector

2.1.2 Application of PFI System in Machine Matching

The CPGC low pressure inlet port methanol injection system has been utilized on the CPGC M320DM main engine (Figure 5). Based on the test data of the main engine, this system can achieve stable operation with a maximum methanol energy ratio of 56% and a maximum methanol mass ratio of 73% within the range of 20% to 100% of the engine. The methanol mode can reach the same rotational speed and power as the diesel mode, and the fuel switching is smooth and rapid. Nitrogen oxide emissions comply with the IMO Tier 2 standard. Additionally, the dual-injector design expands the flow range of methanol injection, shortens the methanol injection time, effectively enhances the quality of methanol spray, and the injectors are placed in the intake manifold close to the valve position, which can reduce the wall-adhering phenomenon of methanol in the intake manifold, improve the stability of in-cylinder combustion, reduce methanol tail gas escape, and effectively improve the transient response characteristics of the engine.

Figure 5. CPGC M320DM low pressure inlet port fuel Injection methanol-diesel dual fuel engine real shot

2.2 HPDI System

2.2.1 Introduction to the HPDI System

The design and research of the CPGC in-cylinder high pressure direct injection fuel injection system commenced in 2023. This system employs an integrated dual-electronic control injection technology route for diesel and methanol. The injection pressure of methanol is 60 MPa, and that of diesel can reach up to 160 MPa. The overall layout of the system on the main engine is presented as follows in Figure 6.

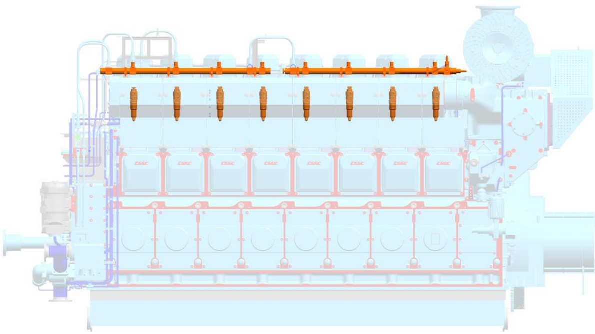


Figure 6. Design scheme of high pressure direct injection system of methanol in cylinder

This four-stroke methanol-diesel dual-fuel in-cylinder high pressure direct injection system is primarily composed of a high-pressure diesel common rail, a low-pressure methanol common rail, a methanol booster pump, and an integrated dual-fuel injector. The working pressure range of the high-pressure diesel common rail is 800 to 1600 bar, which can concurrently serve as both a pressure-stabilizing source for diesel injection and a driving and control oil source for methanol injection. The working pressure of the low-pressure methanol common rail is 4 to 10 bar, adopting the same supply pressure as PFI. This ensures that shipowners do not need to add additional fuel supply or drive systems regardless of whether they opt for PFI dual-fuel main engines or HPDI dual-fuel main engines.

Further, both the high-pressure diesel common rail and the low-pressure methanol common rail adopt the structural form of inner and outer double walls (Figure 7). Specifically, the inner tube of the methanol rail is filled with low-pressure methanol in the methanol mode, and the annular space between the inner and outer tubes is purged and inactivated in the form of negative pressure ventilation. In the heavy oil or diesel mode, pure water is introduced into the methanol rail, and the circulating pure water can be employed to inactivate and cooling the dual-fuel injectors.

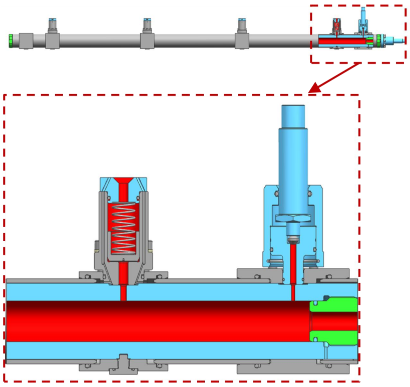


Figure 7. Schematic diagram of methanol common rail structure

The transformation from low-pressure methanol (4-10 bar) to medium-pressure methanol (500-600 bar) is achieved using a liquid booster pump. This pump design, inspired by CPGC's 2016 diesel injection system, features a fast 3/2-way valve and uses high-pressure diesel as the driving medium for methanol pressurization and injection. To address methanol's low carbon content and viscosity, the pump has a double-wall structure and independent lubrication/sealing oil circuits. Simultaneously, to achieve rapid pressure build-up during methanol injection, avoid the secondary lifting of the needle at the end of injection, and prevent the residual methanol within the injector during non-injection intervals from vaporizing in the high-temperature environment of the cylinder head, the outlet valve on the booster pump adopts an equal capacity structure design, which can ensure that the injector and pipeline maintain a methanol pressure of 80 - 150 bar during non-injection intervals. The overall structural schematic diagram of the booster pump is presented in Figure 8.

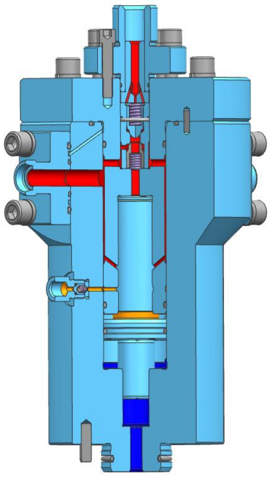


Figure 8. Structure diagram of methanol booster pump

2.2.2 Integrated Fuel Injector

Constrained by the space of the engine cylinder head, the design scheme of arranging one or more larger methanol fuel injectors beside the original diesel injector is scarcely acceptable for dual-fuel main engines. It is essential to use a dual-fuel injection system that allows the same injector to switch between diesel and alternative fuel modes. CPGC has developed an integrated methanol-diesel dual-fuel injector. The injector adopts an inner and outer double-needle nested structure (Figure 9), the inner needle is employed for the injection of igniting and main-burning diesel, while the outer needle is for the injection of methanol. This nested needle pair structure enables the methanol and diesel injection holes to be positioned at the center of the combustion chamber, ensuring that either fuel can achieve an evenly distributed effect along the central axis during injection, and achieving uniform combustion and higher combustion efficiency in diesel ignition, diesel main combustion, and low-carbon fuel combustion modes.

The solenoid valve for diesel injection control is installed within the injector and adopts a classic balanced control valve structure, whereas the solenoid valve for methanol injection control is installed on the aforesaid methanol booster pump and employs a conventional spool valve structure. Both sets of control valves are driven by diesel from the high-pressure common rail. In the low-carbon fuel operating mode, the diesel injected from the annular cavity between the inner and outer needles and the control diesel discharged from the damping hole of the balanced control valve can also perform a cooling function. Furthermore, since high-pressure diesel fills the spaces between the inner and outer needles and between the outer needle and the needle body, it can also offer lubrication for the moving pairs and prevent the low-carbon fuel from overflowing through the gaps, providing sealing protection.

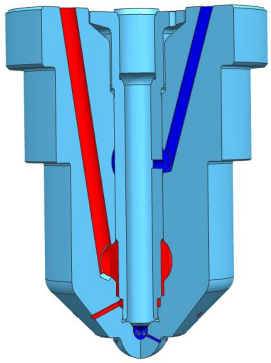


Figure 9. Schematic diagram of the nozzle coupling structure of the injector

Distinct from traditional diesel injectors, the proposed dual-fuel injector has also incorporated the functions of discharge and purging of methanol fuel in failure modes. The discharge valve is situated at the top of the injector and adopts a control strategy of being normally open when power is off, with the driving source derived from the starting air of the main engine. The purging valve is positioned opposite the inlet of the methanol oil passage. When the top discharge valve is opened and the methanol pressure within the methanol channel is released to below 5 bar, N_2 enters the adjacent methanol channel via the purging valve and circulates from the methanol storage tank to the distal methanol channel, ultimately being discharged from the top discharge valve. This enables a simple, convenient, efficient and reliable switch from the methanol to the diesel mode. After purging and inactivating the methanol channel within the injector, pure water at 6 - 8 bar can be introduced at the inlet of the purging valve. The pure water circulates within the methanol channel and flows out from the outlet of the discharge valve. This can cool the injector in non-methanol working modes, especially in heavy oil mode, and play a role in protecting the solenoid valve and nozzle. The overall structure of the injector is depicted in Figure 10.

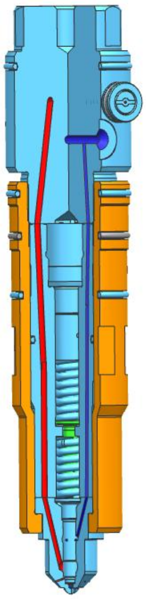


Figure 10. Schematic diagram of the injector structure

3 NUMERICAL SIMULATION DESIGN OF NEW DUAL-FUEL INJECTOR

3.1 One-dimensional Transient Hydraulic Injection Characteristics of the Injector

During the development of the methanol injection system, in order to verify the feasibility of the system scheme, the hydraulic and injection performance of the system were predicted and analyzed via one-dimensional simulation software AMESim. As depicted in Figure 11, the simulation model of the HPDI dual-fuel methanol injector encompasses three parts: the diesel injection circuit, the methanol booster pump, and the methanol injection circuit. In the modeling process, the detailed motions of each valve component and their interaction with the hydraulic circuit were taken into account, and the pressure fluctuations of the methanol fuel injection system were modeled elaborately by considering the fluid inertia.

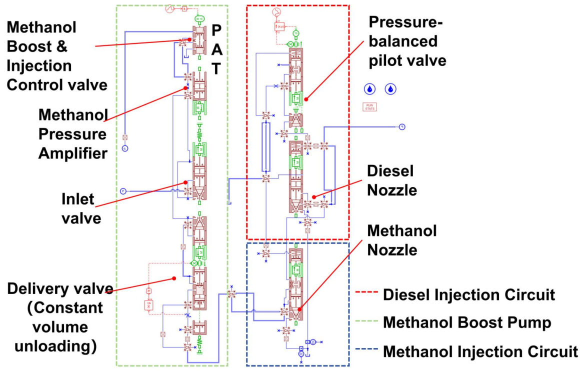


Figure 11. Simulation model of HPDI methanol dual fuel injector

Figures 12-14 present the methanol injection rate, injection pressure and injection quantity under various loads. It can be observed that the designed fuel injector is capable of achieving an excellent rectangular injection rate, and the injection pressure is stable near the target value. The injector is able to satisfy the demands of injection pressure and injection quantity of the main engine under different loads.

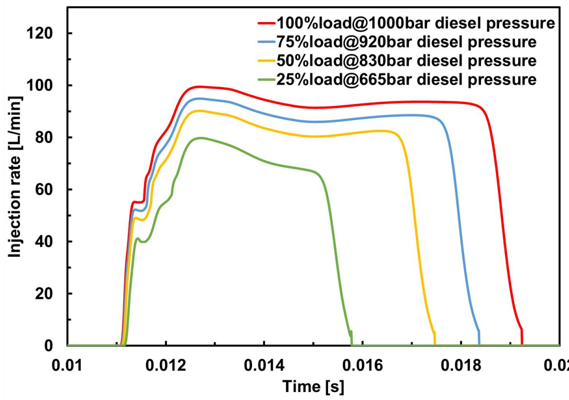


Figure 12. Methanol injection rate in varying engine load

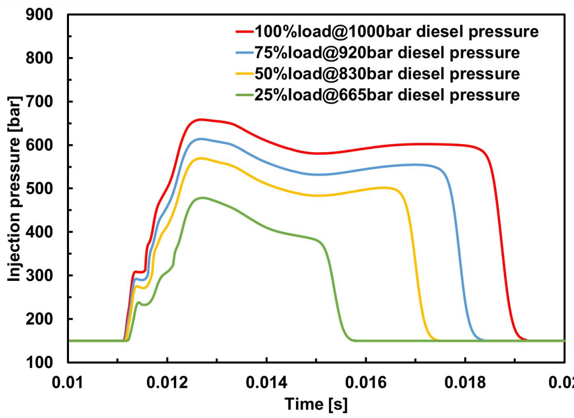


Figure 13. Methanol injection pressure in varying engine load

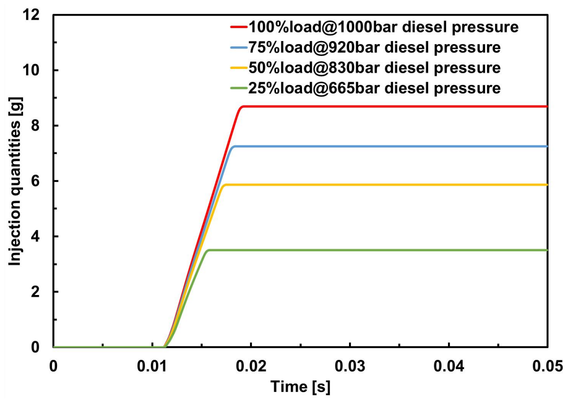
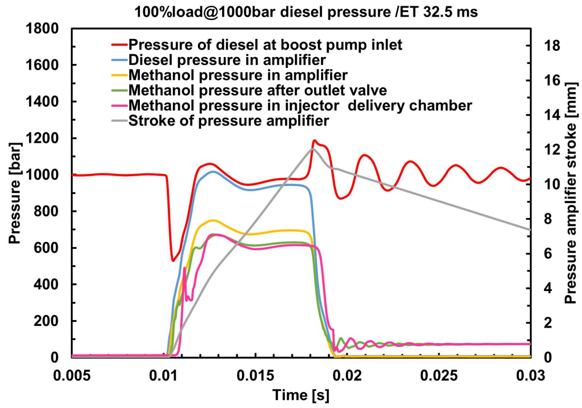


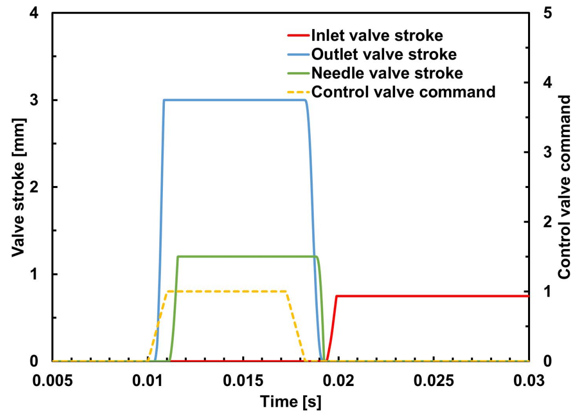
Figure 14. Methanol cycle injection quantities in varying engine load

Figure 15 illustrates the intricate dynamic characteristics of the high-pressure methanol injection process within the methanol HPDI system. As depicted, upon receiving the control signal, the control valve allows the high-pressure diesel at the inlet of the methanol booster pump to traverse through it and engage with the booster piston, thereby pressurizing the low-pressure methanol. The low-pressure methanol achieves its designated injection pressure in accordance with the meticulously pre-designed pressure ratio, subsequently navigating through the outlet valve, the high-pressure methanol conduit, and finally entering the HPDI injector to facilitate methanol fuel injection. Upon cessation of the control signal, the diesel pressure within the boost chamber diminishes precipitously, leading to a corresponding reduction in methanol pressure. Consequently, the needle closes, marking the termination of the methanol injection cycle.

It can be observed from Figure 15 that the injection response delay of the system is 1.5 ms, and the response delay for the end of injection is 1 ms, fulfilling the requirements of the high response characteristics of medium-speed engines. This is the outcome of minimizing the dead volume in the system as much as possible during the design process of the methanol HPDI system, thereby shortening the hydraulic delay during the injection process.



(a)

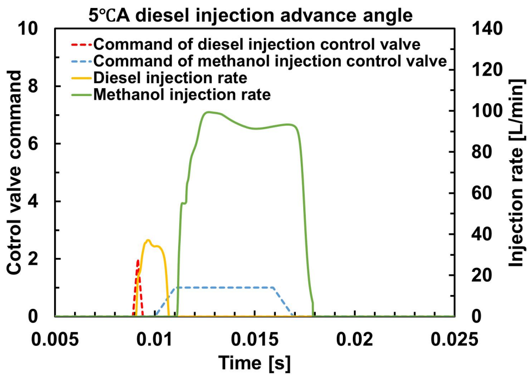


(b)

Figure 15. Dynamic injection characteristics of methanol fuel

During system design, installing a damping hole at the diesel common rail outlet significantly reduces post-injection pressure fluctuations. As shown in the figure 15, pressure oscillations decay rapidly during injection, ensuring stable methanol pressure. After injection, the outlet valve closes quickly, maintaining high methanol pressure in the pipeline. This reduces the high-pressure methanol needed for the next cycle and enhances system response. Additionally, this pressure prevents methanol vaporization due to increased system temperature.

Figure 16 presents the injection characteristics of the methanol HPDI injector under various control timings of the diesel and methanol injection control valves. As can be observed from the figure, this injector is capable of realizing flexible injection timing combinations of diesel and methanol fuels. Moreover, the injection processes of diesel and methanol are mutually independent, and the injection law of the fuel is not influenced by the injection timing. This offers the potential for optimizing the combustion performance of the engine through different combustion organization forms.



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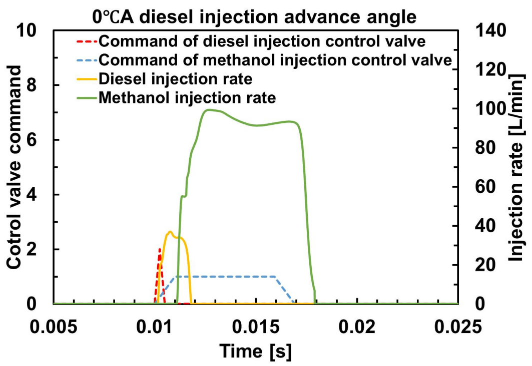

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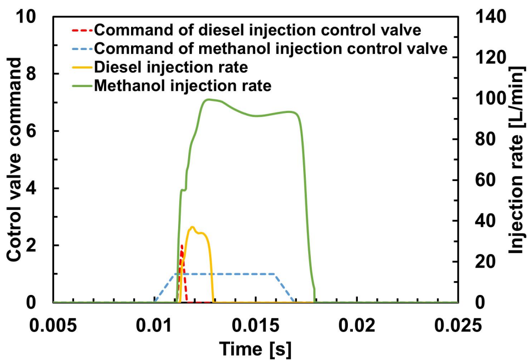

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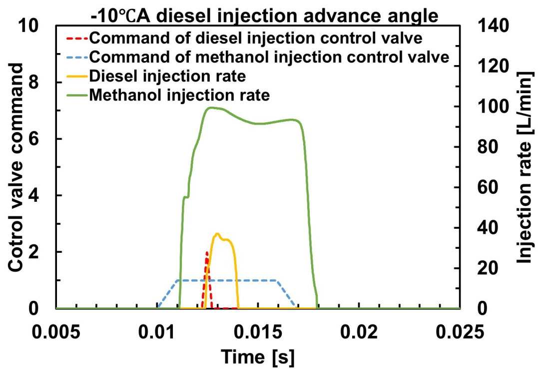


(d)

Figure 16. Flexible dual fuel injection dwell time

This system enables the injection of methanol fuel at diverse pressures through the adjustment of diesel pressure. Simultaneously, by modifying the injection pulse width, it can fulfill the fuel requirements of the main engine within an extensive range of operating conditions. Figure 17 presents the cycle injection quantity Map of methanol high-pressure injection. It can be discerned from this figure that the injector exhibits excellent linearity within a broad injection quantity range. By regulating different injection pressures and pulse widths, the injection quantity of methanol fuel can be precisely controlled.

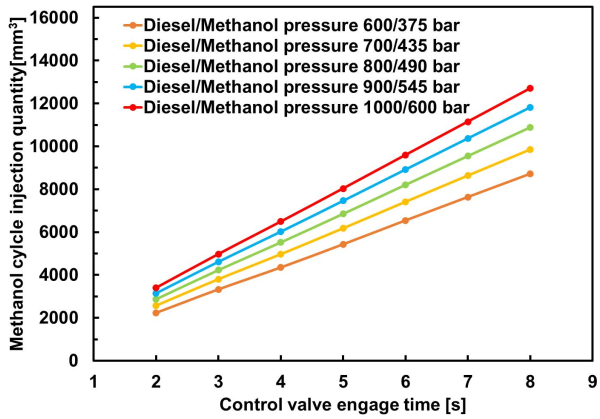


Figure 17. Injection quantities Map of methanol

Figure 18 presents the cycle injection quantity Map of diesel injection for this integrated injector. The diesel injection adheres to the principle of a typical common rail electromagnetic valve-controlled electronic injector. Through optimizing the internal oil passage layout, the structure of the needle sealing surface, and the design of the pressure-balanced pilot valve, stable regulation of the injection quantity within a wide range can be achieved, thereby enabling stable injection of a large amount of diesel in the diesel mode and micro-injection of diesel for ignition in the dual-fuel mode of diesel and methanol. Among them, the minimum stable injection quantity of diesel micro-injection can reach 5% of the maximum flow rate, which can satisfy the injection requirements of diesel micro-injection for ignition when the methanol substitution rate is high.

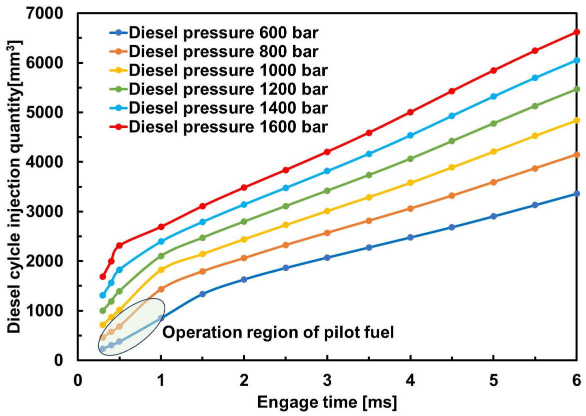


Figure 18. Injection quantities Map of diesel

3.2 Three-dimensional Transient Flow Characteristics Inside the Injector

3.2.1 Transient Flow of Diesel in the Balance Control Valve

The control oil circuit of a typical common rail injector is constituted by the OZ orifice, the control chamber, the OA orifice, and the control ball valve. This patented novel flexible injector based on common rail injection follows the control oil circuit mode of previous injectors, while improving the internal oil passage layout. It makes full use of traditional diesel to control the injection of methanol fuel, avoiding the potential hazards of corrosive methanol to the injector control oil circuit, greatly simplifies the internal fuel circulation pipeline of dual-fuel engines, and enhances the safety and reliability of the operation of the entire system.

In the control oil circuit of this injector, a balanced control valve is employed to replace the traditional control ball valve, thereby reducing the hydraulic pressure-bearing area. Even within the confined space inside the injector, the small electromagnetic force generated by the small-sized solenoid valve can successfully open the oil OA orifice, guaranteeing the rapid dynamic response of the needle. Nevertheless, during the flow of high-pressure fuel from the oil inlet hole through the balanced control valve, the transient high-pressure conditions result in an extremely high fuel flow velocity, inevitably giving rise to cavitation. Cavitation can alter the fuel flow characteristics at the balanced control valve, generating local choked flow and causing cavitation erosion at the valve and valve seat, which affects the sealing reliability between the valve and the valve seat. Additionally, the cavitation in the balanced control valve has a direct impact on the response characteristics, injection rate, injection pulse width and injection quantity of the injector, and subsequently influences the efficiency and emission performance of the dual-fuel engine.

In order to analyze the transient cavitation flow characteristics of diesel within the balance control valve of the injector and mitigate the fuel cavitation effect at the balance control valve, the flow domain within the balance control valve of the injector was established and meshed. Taking into account the complexity of the flow field in the OZ and OA orifice, more intensive processing was carried out. Concurrently, boundary layer meshes were set at the cone-cone seat, the wall of orifice, and the inlet and outlet of the flow domain to capture the minute variations of the wall flow, as depicted in Figure 19. The boundary conditions of the model are presented in Table 1.

Table 1. The computational boundary conditions and initial conditions

Parameter	Unit	Value
Fuel temperature	K	300
Inlet pressure	MPa	65
Outlet pressure	MPa	0.1

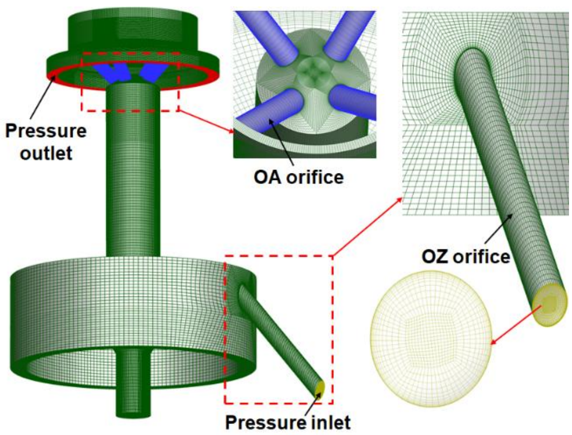
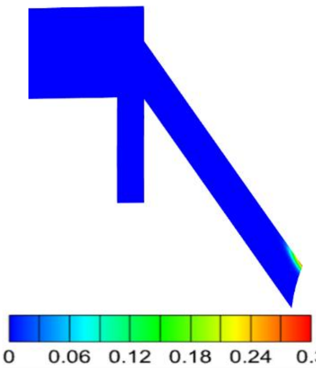
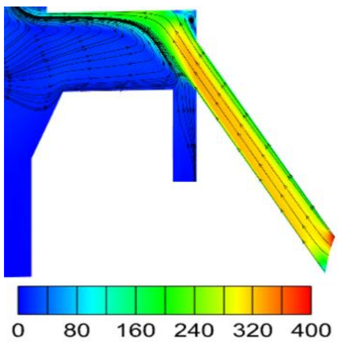


Figure 19. Mesh model of injector poppet valve

The transient cavitation flow characteristics of diesel within the balance control valve were analyzed via CFD simulation. Regarding the transient flow inside the OZ orifice (Figure 20), as there is a certain curvature at the entrance of the OZ orifice, the fuel flow velocity near the upper wall of the OZ orifice entrance is higher than that near the lower wall. In the later stage of the movement of the balance control valve, a negative pressure zone emerges at the entrance. When the pressure drops below the saturated vapor pressure of the fuel, cavitation takes place. Additionally, since the angle between the axis of the OZ orifice and the axis of the balance control valve is relatively small at the design stage, the fuel flowing out of the OZ orifice directly rushes towards the wall of the control chamber, resulting in an uneven velocity distribution within the control chamber and the formation of vortices in the right-angle area, where pressure concentration and increase are prone to occur. Hence, when enhancing the flow field distribution characteristics in this area, measures such as optimizing the angle between the axis of the OZ orifice and the axis of the balance control valve, the structure of the OZ orifice entrance, and the fillet radius need to receive extra attention.



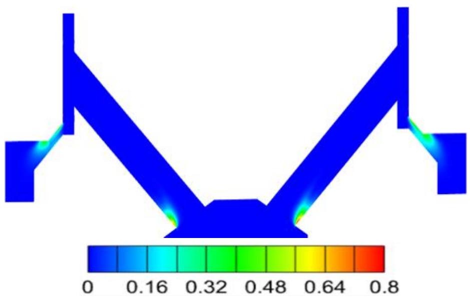
(a) Vapor fraction/-



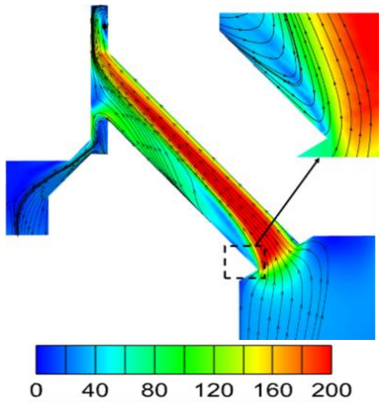
(b) Velocity/(m/s)

Figure 20. Transient flow characteristics in OZ orifice at maximum lift of poppet valve

Regarding the transient flow in the OA orifice and the balance control valve-seat region (Figure 21), owing to the guiding effect of the OA orifice on the fuel, the fuel flow velocity near the upper wall of the OA orifice is higher. The mainstream zone of the fuel forms an angle with the lower wall of the inlet, giving rise to a flow separation phenomenon. A portion of the fuel flows back to the vicinity of the lower wall of the OA orifice inlet, where the increase in fuel flow velocity leads to the emergence of a low-pressure zone, thereby causing cavitation. Once the fuel exits the OA orifice, a part of it continues to advance along the direction of the OA orifice and rushes into the upper cavity to form a vortex, resulting in the loss of some kinetic energy. Another part of the fuel flows along the lower flow channel into the balance control valve-seat region. Due to the extremely small flow area in this region, the fuel flow velocity increases rapidly, resulting in the appearance of a low-pressure zone. When the pressure is lower than the saturation vapor pressure, cavitation occurs. Furthermore, the fuel does not enter the balance control valve-seat channel smoothly from the OA orifice. Hence, adjusting the axis direction of the OA orifice, adding fillets to the inlet and outlet of the OA orifice, and optimizing the shape and volume of the cavity formed by the valve and the valve seat are all crucial measures for reducing cavitation and improving the fuel flow distribution.



(a) Vapor fraction/-



(b) Velocity/(m/s)

Figure 21. Transient flow characteristics in OA orifice and balance control valve-seat region at maximum lift of poppet valve

3.2.2 Transient Flow of Methanol within the Nozzle

As one of the most complex parts in terms of flow structure, the injector nozzle, owing to its small injection hole and high-pressure gradient, finds it challenging to mitigate the negative effects brought about by cavitation pitting during the opening and closing processes of the nozzle. This not only may give rise to cavitation damage at the nozzle orifice but also closely influences the mixing quality of the fuel spray with air through the changes in internal flow characteristics it induces. To conduct a detailed analysis of the transient flow characteristics of the fuel within the nozzle, reveal the formation mechanism of cavitation within it, and optimize the situation, a flow domain within the injector nozzle was established and meshed. Meanwhile, considering the complexity of the flow in the pressure chamber and at the nozzle inlet, and with the study target being cavitation within the nozzle, therefore, during the meshing process, the area near the inlet of the nozzle orifice was meshed more intensively to capture more flow details. The mesh model is depicted in Figure 22. The initial boundary conditions are presented in Table 2.

Table 2. The computational boundary conditions and initial conditions of spray simulation

Parameter	Unit	Value
Diesel Inlet pressure	MPa	120
Methanol Inlet pressure	MPa	65
Both Outlet pressure	MPa	20

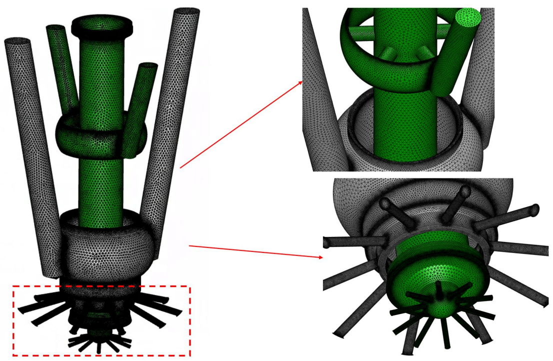


Figure 22. Mesh model of injector nozzle

CFD studies were carried out on the flow within the nozzle, where methanol was located in the outer channel and diesel in the inner channel. To minimize the leakage of methanol droplets, the volume of the pressure chamber in the methanol channel was kept very small, while the diesel channel still adopted the conventional pressure chamber structure. As illustrated in Figure 23(a), this figure presents the cavitation within the nozzle. At the maximum lift position of the needle, diesel exhibits a higher cavitation intensity and a broader influence range compared to methanol. Specifically, super cavitation is observed on the upper wall of the injection hole for diesel, whereas cavitation initiates at the entrance of the lower wall of the injection hole for methanol. As depicted in Figure 23(b), this figure shows the fuel velocity and streamline map inside the nozzle. Owing to the higher injection pressure of diesel, its average velocity within the injection hole is nearly double that of methanol. Additionally, vortices are highly likely to form at the turning and sharp corner positions of the flow channel of the needle mate part.

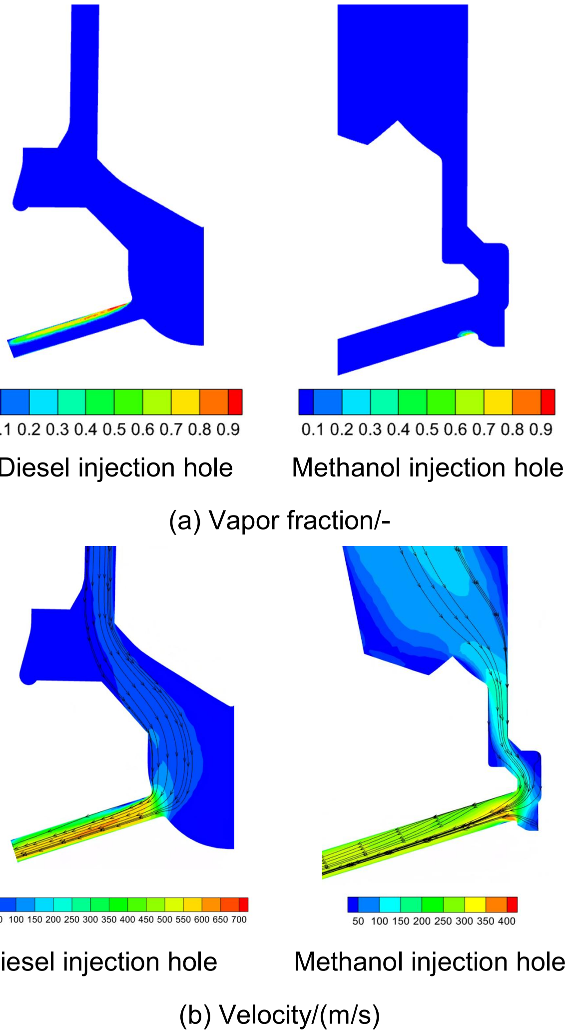


Figure 23. Transient flow characteristics in nozzle region at maximum lift of needle valve

3.3 Atomization Characteristics of Methanol Spray from the Injector

To guarantee an efficient and clean combustion process within the engine cylinder, it is requisite to undertake more elaborate simulation analyses on the macroscopic and microscopic features of the fuel spray injected from the injection holes, with the aim of achieving a homogeneous and superior fuel-air mixture, enhancing the combustion efficiency of the engine, and reducing emissions. Consequently, a simulation model for methanol spray of the injector was established, as depicted in Figure 24. The Lagrangian method was employed for the discretization simulation of methanol spray. Simultaneously, to better analyze the spray characteristics of each injection hole, the full model form was utilized for spray simulation. The boundary conditions and initial conditions during the numerical simulation calculation process are presented in Table 3.

Table 3. The computational boundary conditions and initial conditions of spray simulation

Parameter	Unit	Value
Inlet pressure	bar	600
Background pressure	bar	200
Background temperature	K	1073
Model wall temperature	K	1073
Methanol temperature	K	300
Injection pulse width	ms	11.0

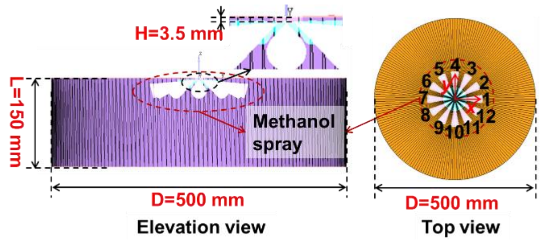


Figure 24. Simulation model of methanol spray

As shown in Figure 25, the liquid-phase velocities of the methanol sprays from each injection hole are presented. Under the effect of pressure differential, methanol is injected from the nozzle hole at a speed exceeding 50 m/s. In the vicinity of the nozzle hole, the velocity of methanol spray reduces to approximately 30 m/s. As the spray progresses gradually, momentum exchange occurs between the methanol spray and the external environment, and the spray velocity decreases gradually along the nozzle axis outward. The velocities of different methanol sprays show a high degree of agreement in temporal sequence and spatial distribution.

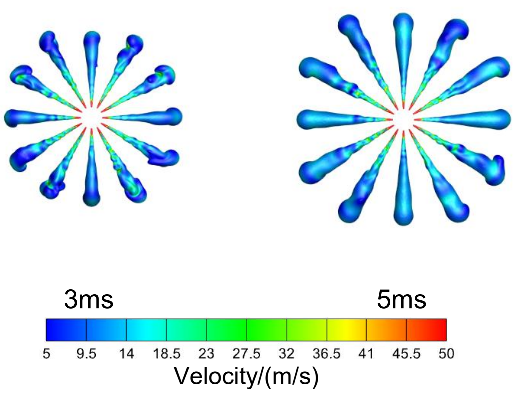


Figure 25. Velocity of liquid phase methanol spray

As depicted in Figure 26, at the 5.0 ms instant, the methanol mass fraction at the spray head commences to diffuse towards all directions. Owing to the fragmentation and evaporation of the liquid, the front end of the spray is high-temperature methanol vapor. Due to the uncertainties of turbulent disturbance and the liquid droplet fragmentation process, there exist certain disparities in the temperature distribution of the methanol vapor at the front end of the spray from each injection hole. At the 12.5 ms instant, the injection has concluded, and the spray center no longer contains methanol in the form of high-mass-fraction and low-temperature liquid. The methanol spray has entirely evaporated into vapor due to heat transfer, but the spray will continue to expand forward due to kinetic energy and inertia.

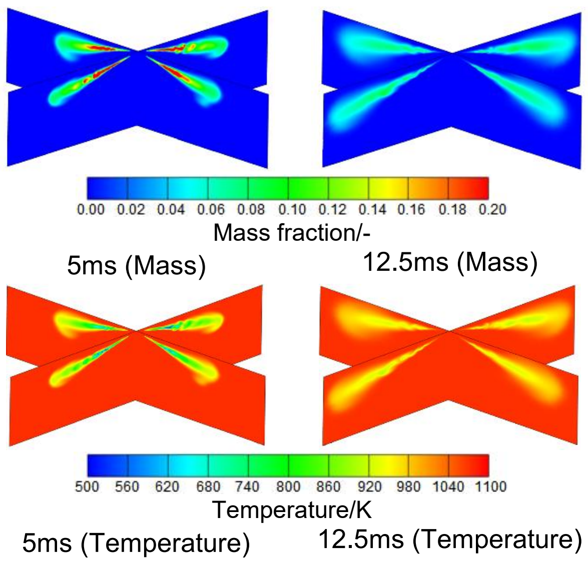


Figure 26. Mass fraction and temperature distribution of methanol spray

From the variation trend of the Sauter mean diameter in Figure 27, it can be observed that in the initial stage of the spray, the average diameter of the droplets rises significantly due to droplet coalescence, and then decreases as a result of the initial droplet breakup, with a "mushroom head" structure emerging at the spray head. During the continuous methanol injection stage, the average diameter of the droplets remains at a relatively low level, and the broken "mushroom head" transforms into methanol vapor, and the spray begins to stabilize gradually. Nevertheless, upon the conclusion of the fuel injection, the stable state of the spray is disrupted, the momentum of the spray commences to decay, and the velocity of the droplet movement starts to decline. Under such circumstances, the probability of large droplet fragmentation under the influence of surface tension begins to decrease, and small droplets commence to coalesce, leading to an increase in the Sauter mean diameter of the droplets.

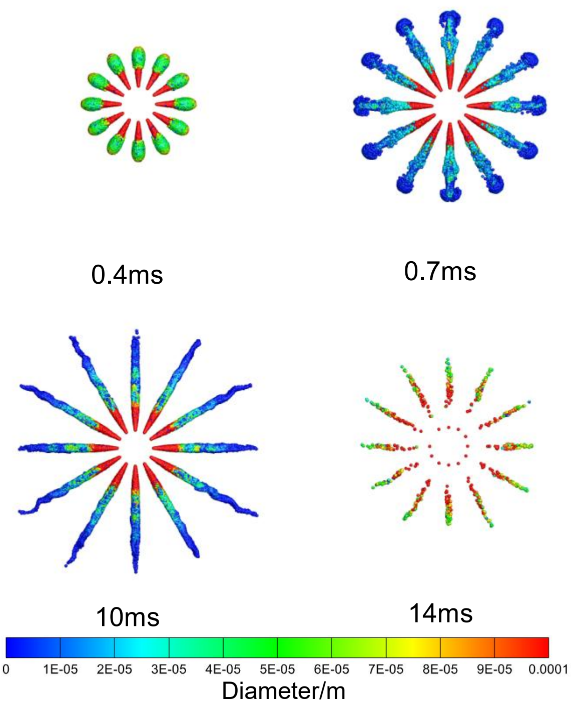


Figure 27. Droplet size distribution of methanol spray

The identification of the presence and size of vortices in turbulent motion enables the analysis of the rotational flow regions in methanol spray, facilitating the comprehension of the complex structure and dynamic characteristics of the spray motion.

As depicted in Figure 28, the screening results of the vortex distribution based on the Q criterion demonstrate the existence and size of vortices in turbulent motion. At the early stage of injection (0.5 ms), vortices commence to form and gradually intensify, indicating that the flow in the spray area starts to become complex and vortex development initiates. At 0.8 ms, the vortices at the front end of the spray exhibit a shedding phenomenon, suggesting that the spray enters the secondary breakup stage, with droplet breakup and shedding behavior occurring at the spray head. The formation and detachment process of vortices substantiates the secondary breakup of the spray. With the passage of time (4.5 ms), the distribution of vortices becomes more widespread, and the vortices in the spray area significantly expand. In the later stage of injection (20 ms), the distribution of vortices indicates that the flow in the spray area becomes more intricate, with methanol vapor presenting larger vortices, while the vortex structure in the liquid phase is smaller.

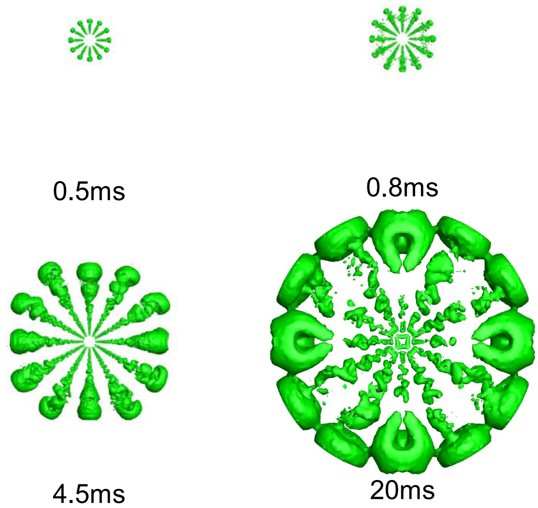


Figure 28. Vortex distribution of methanol spray (isosurface Ω=100)

4 FUNCTIONAL TESTS AND APPLICATION OUTLOOKS OF THE INJECTOR

4.1 Test Background

Currently, the first batch of injector and booster pump prototypes suitable for the M320DM in-cylinder direct injection methanol-diesel dual-fuel main engine have been fabricated (as depicted in Figure 29). Since the test bench for the methanol injection system is still under construction, to validate the working characteristics and performance parameters of the methanol-diesel dual-fuel injectors, we have conducted a retrofit design on the existing relatively mature fuel injection test bench. Through the retrofitted fuel injection test bench, the objective of conducting functional test verification on the first batch of methanol injector prototypes can be achieved.

Figure 29. The sample of the methanol-diesel dual-fuel injector and booster pump

4.2 Brief Introduction to the Test Apparatus

The functional tests of the dual-fuel injector and the booster pump for methanol were carried out on a simple diesel high-pressure common rail test bench. The high-pressure fuel pump and the high-pressure common rail of this test bench could concurrently act as the driving oil sources for both diesel injection and methanol supercharging. Meanwhile, to simulate the intake and supercharging process of low-pressure methanol, we employed the low-pressure fuel pump of the dual-electromagnetic valve controlled diesel injection system test bench of CPGC in 2016 as the supply source of low-pressure diesel for the integrated methanol injector. With the assistance of a simply modified and designed bracket capable of accommodating the injector and the fuel pump, a functional test bench for the integrated testing of the methanol booster pump and the injector was jointly constructed. Figure 30 presents the schematic diagram of the temporary experimental test bench, and Figure 31 shows the physical image of the temporary experimental test bench.

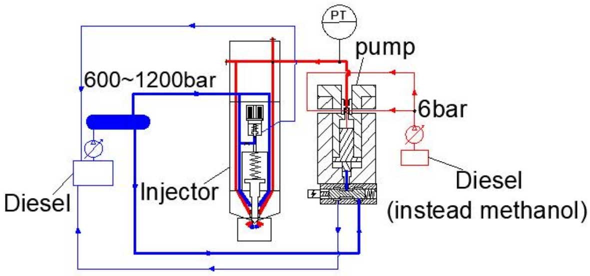


Figure 30. Schematic diagram of the temporary experimental test bench

Figure 31. Physical drawing of the temporary experimental test bench

To verify and test the basic injection function of the booster pump and the injector, a pressure sensor was installed at the outlet of the booster pump. The layout of the measurement points is shown in Figure 32 as follows.

Figure 32. Pressure sensor measurement point diagram

To sum up, the parameters and instrumentation information of the entire temporary test bench are presented in Table 4.

Table 4. Main parameters of test equipment and test instrument

Parameter	Value	Remarks
High pressure pump flow	12L/min	From diesel high pressure common rail test bed.
Diesel common rail pressure	600~1200bar	
High pressure common rail volume	3.54L	
Low diesel supply pressure	6bar	From the CPGC double solenoid valve-controlled diesel injection system test bench.
Low pressure diesel supply flow	30L/min	
Seal oil pressure	10bar	
Seal oil supply flow	30L/min	
Booster pump outlet pressure sensor	0~1600bar	Trafag
Data acquisition instrument	GEN3i	HBM

4.3 Analysis of Test Results

With the assistance of the temporary experimental test bench, we performed injection pressure tests on the self-developed methanol booster pump and integrated dual-fuel injector. The tests revealed that when the driving pressure of diesel ranged from 600 to 1200 bar, the injection pressure data at the outlet of the booster pump are illustrated in Figure 33. Due to the distinct physical properties of diesel and methanol fuel, the flow-limiting effect of the high-pressure hose, and the constraints of the test bench conditions, discrepancies exist between the experimental data and the simulation results. Nevertheless, based on the aforementioned measured data, it can be concluded that the independently developed and designed diesel-boosted methanol-diesel in-cylinder direct injection dual-fuel injector by CPGC has achieved the anticipated injection performance.

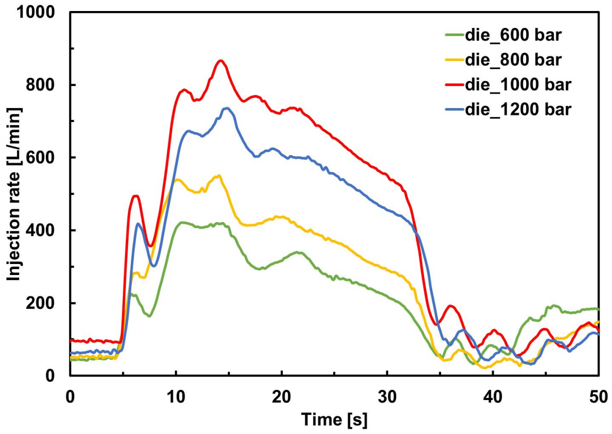


Figure 33. Test value of booster pump outlet pressure when driving pressure is 600~1200bar

4.4 Next Work Plan

Limited by the test conditions of the existing temporary experimental test bench and due to the significant differences in the fuel physical parameters between diesel and methanol, the test data obtained from the functional test bench at present is relatively small, and there are certain differences in the test results compared with the methanol mode. Therefore, in order to accurately feedback the working performance of the independently developed and designed methanol booster pump and integrated methanol injector in the methanol mode, the main research directions in the future include:

(1) Based on the subsequently completed methanol-specific test bench, carry out the test and measurement work of the special test bench in the methanol mode, and test the injection pressure, injection flow rate and other test data of the injector under different injection pressures and different injection pulse widths. This will provide guidance for the optimization and improvement of the internal structure of the injector and the combustion in the engine cylinder.

(2) Carry out the flexible switching function test of methanol-diesel dual fuel and the performance test of the injector under different modes after the switch, providing data support for the subsequent design and optimization of the flexible fuel injector.

(3) Based on the data tested by the special test bench, calibrate the simulation model, and carry out iterative optimization design work according to the possible problems in the test process.

(4) Carry out the durability test of the methanol-diesel in-cylinder direct injection dual fuel injector to evaluate the material selection design status and structural design status of the methanol-diesel dual fuel injector.

(5) Based on the optimized and designed diesel-boosted methanol-diesel in-cylinder direct injection dual fuel injector assembled on the CPGC M320DM main engine, conduct the main engine performance load and durability reliability test to provide support for the production path.

5 CONCLUSIONS

CPGC provides innovative solutions for the PFI and HPDI technologies of green and low-carbon fuel engines. CPGC low-pressure intake port dual-fuel methanol injection system applied to the CPGC M320DM main engine has completed a complete engine test, which can achieve stable operation with a maximum methanol energy ratio of 56% and a methanol mass ratio of 73%. The power performance, emission performance, safety performance and transient response performance of the engine have been comprehensively improved.

The CPGC in-cylinder high-pressure direct injection fuel system adopts the diesel-methanol integrated dual-electronic control injection technology route. The designed and developed combination of the methanol booster pump and the low-pressure methanol common rail enables both the PFI dual-fuel main engine and the HPDI dual-fuel main engine to operate without the need to add additional fuel supply or drive systems. At the same time, a new type of integrated dual-fuel injector has been developed, which makes full use of traditional diesel to control, lubricate, seal and cool the methanol injection system, thus not introducing an additional hydraulic system. Through a large number of one-dimensional/three-dimensional simulations and experimental studies, multiple rounds of technological iteration and optimization have been completed, verifying its superior practicability in the limited cylinder head space. Through this technology, a clean dual-fuel engine with equivalent power density and dynamic performance to the current diesel engine can be designed. The engine can freely switch between single fuel and dual-fuel combinations to meet different usage requirements and meet increasingly strict emission regulation standards in the future. In general, the new integrated dual-fuel injector has the following characteristics:

1. The integrated design significantly enhances space utilization and markedly reduces the size of the injector.
2. By adopting dual-solenoid valve structure, this design enables independent and precise control of methanol and diesel injection, ensuring that the fuel injection rate remains unaffected by the injection sequence.
3. An internally nested needle coupling design is utilized, positioning both the methanol and diesel injection holes at the center of the combustion chamber.
4. Methanol injection pressure can reach up to 60 MPa.
5. Diesel injection pressure can reach up to 160 MPa, with stable injection of a minimum of 5% of the maximum flow rate.
6. Both methanol and diesel fuels support multiple injections.
7. The internal oil passage layout is improved, making full use of traditional diesel to realize injection and simplifying the internal fuel circulation pipeline of the dual-fuel engine.
8. The discharge and purging functions of methanol fuel in the fault mode are added, improving safety and reliability.

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