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Nghiên cứu đánh giá ảnh hưởng của các thông số kết cấu đến chất lượng phun nhiên liệu trên động cơ Diesel Kubota D1703-M-DI

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Ngày nhận bài: 27/11/2023; Ngày sửa bài: 21/02/2024;
Ngày nhận đăng: 08/03/2024; Ngày xuất bản: 28/06/2024

TÓM TẮT

Trên cơ sở nghiên cứu lý thuyết quá trình cung cấp nhiên liệu, quá trình hình thành hỗn hợp trong động cơ Diesel, cơ sở lý thuyết tính toán trong phần mềm Hydsim, kết hợp với quá trình khảo sát, đo đạc và phân tích đặc điểm kết cấu hệ thống cung cấp nhiên liệu trên động cơ Diesel Kubota D1703-M-DI nhóm tác giả đã ứng dụng phần mềm Hydsim để mô phỏng quá trình cung cấp nhiên liệu động cơ Diesel Kubota D1703-M-DI với điều kiện mô phỏng ở tải định mức, tải trung bình, không tải và có sự thay đổi giá trị các thông số kết cấu: biên dạng cam, tỉ số D/h_i, đường kính lỗ phun, chiều dài lỗ phun, độ cứng lò xo vòi phun,... Dựa trên kết quả mô phỏng thu được, nhóm tác giả đã phân tích, đánh giá kết quả mô phỏng và đưa ra các kết luận liên quan về ảnh hưởng của các thông số kết cấu đến chất lượng phun nhiên liệu trên động cơ Diesel Kubota D1703-M-DI, từ đó đưa ra các đề xuất cải tiến về mặt thông số kết cấu phù hợp hơn. Việc làm này có ý nghĩa quan trọng trong việc giảm tiêu hao nhiên liệu, giảm tiếng ồn, giảm hàm lượng các chất ô nhiễm trong khí xả động cơ.

Từ khóa: *Mô phỏng, hệ thống cung cấp nhiên liệu động cơ Diesel, máy cày Kubota L3804VN, bơm cao áp tập trung, vòi phun cao áp.*

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https://doi.org/10.52111/qnjs.2024.18310

QUY NHON UNIVERSITY
JOURNAL OF SCIENCE

Research to evaluate the influence of structural parameters on fuel injection quality on Kubota D1703-M-DI Diesel engine

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Received: 27/11/2023; Revised: 21/02/2024;
Accepted: 08/03/2024; Published: 28/06/2024

ABSTRACT

Based on theoretical research of the fuel supply process, mixture formation process in Diesel engines, the theoretical basis of calculation in Hydsim software, combined with the process of surveying, measuring, and analyzing structural characteristics of the fuel supply system on the Kubota D1703-M-DI Diesel engine, the authors have applied Hydsim software to simulate the fuel supply process of the Kubota D1703-M-DI Diesel engine under simulated conditions at rated load, average load and no load and with changes in the values of structural parameters: cam profile, D/h_i ratio, nozzle diameter, nozzle length, nozzle spring stiffness spray, etc. Based on the simulation results obtained, the authors analyzed and evaluated the results and made relevant conclusions about the influence of structural parameters on the quality of fuel injection in the Kubota D1703-M-DI Diesel engine, thereby providing suggestions for more suitable structural parameters. This is important in reducing fuel consumption, noise, and the content of pollutants in engine exhaust.

Keywords: *Simulation, Diesel engine fuel supply system, Kubota L3804VN tractor, centralized high-pressure pump, high-pressure injector.*

1. INTRODUCTION

In the period of industrialization and modernization of the country, especially in the current mechanization of rural agriculture, internal combustion engines, in general, and Diesel engines, in particular, are indispensable. Diesel engines are widely used today, and they have advantages such as large capacity, high efficiency, and cheaper fuel costs than gasoline. In particular, small-sized Diesel engines, with a power range of 5 ÷ 40 HP (Horsepower), are highly economical and meet the needs of users, so they are increasingly approaching rural life and construction sites built and used as the primary power source for services such as plows, cultivators, harrows, water pumps, harvesters, etc.; milling machine, rice screening machine, etc.; agricultural vehicles, motorboats, etc.; small capacity concrete mixer; air compressors with small and medium capacity, etc.

The number and scale of our country's engine manufacturers still need to be improved. They mainly produce small single-cylinder diesel engines for agriculture and construction. Due to product price conditions, these engines mostly use old-style fuel supply systems (high-pressure

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https://doi.org/10.52111/qnjs.2024.18310

mechanical pumps) to ensure competitiveness with engines produced in other countries.

Among them, we must mention engines with capacity from 5 to 25 HP from Vikyno company, with characteristics of low fuel consumption, durable use, and production process requiring high precision, such as VIKYNO RV-70, RV-105, RV-125, RV-195, KND5B, D9. These products are used on agricultural machinery such as plows, generators, milling machines, water pumps, etc., and are machines that serve the construction industry favored by domestic and foreign consumers. Most of these products are transferred production technologies under copyright from KUBOTA (Japan) with an increasing rate of details localization, from 40% in 1992 to 70 - 80% in 2004 and currently about 90%.

However, these engines have a small working capacity range, are noisy, and do not meet consumer needs. The problem is that it is necessary to quickly research and produce multi-cylinder engines with a broader power range and higher efficiency and economy.

To meet consumer needs, agricultural machinery companies have imported many products, such as tractors, combine harvesters, etc., from brands such as KUBOTA, YANGMA, MITSUBISI, ISUZU, etc. This product has a 3, and 4-cylinder Diesel engine with a 20 ÷ 40 horsepower capacity. In addition, during engine operation at high load and high speed, the fuel injection process is poor, producing much black smoke due to falling, burning on the expansion line, and consuming much fuel. When running at idle speed, the engine design is unstable, causing environmental pollution and reducing the engine's economy. One of the reasons is the poor quality of the fuel injection process.

The problem is that it is necessary to improve or research new designs of engines in general and fuel systems of these types of engines, in particular, to suit the conditions of use in our country, bringing the highest economic benefit for users. But calculating, designing, manufacturing, renovating, installing new things, and doing experiments are complicated and take much time. Therefore, researching and exploiting the application of information technology software in calculating design, simulating engines in general, and the process of supplying fuel to internal combustion engines, in particular, will help significantly shorten the time for designing, testing, and optimizing the fuel supply process of the engine fuel system. This will significantly reduce design, manufacturing, and testing costs, thereby reducing product costs.[1]

Currently, in the world, there appears a lot of information technology software related to simulating engines in general and hydrodynamic processes, in particular, such as KIVA software is used to calculate and simulate the thermodynamic process of the engine); PROMO software (German) is used to calculate the thermodynamics of the engine's working process based on CFD (computational Fluit Dynamics) computational fluid dynamics theory; The software BOOST, FIRE, HYDSIM, EXCITE, GLIDE, TYCON, BRICKS from AVL (Austria) are very powerful toolkits in calculating and simulating kinetic, dynamic, thermodynamic processes, hydrodynamics, etc. of structures and systems in internal combustion engines; AUTOMATION STUDIO software is used to estimate and simulate hydraulic and pneumatic systems; Festo's FluidSIM 5.0 software is used to calculate and simulate hydraulic and pneumatic systems; Fluent software is software capable of modeling cylinder engines, ballistics, turbine engines and equipment, and multiphase systems; etc. This software can be used for in-depth research on engine work cycles and can design models, test theoretical models, etc. In Vietnam, this software has just been used in recent years, so it is currently in the research stage. In addition to those specialised software, there are some very commonly used software such as MATLAB SIMULINK software. The software

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can handle most mathematical operations based on the available command set; moreover, it can simulate systems in mechanics and electronics.

In short, each software has its advantages in a specific area. In particular, Hydsim software is software in the AVL Workspace software suite of the Republic of Austria, an in-depth software for calculating and simulating the fuel supply process of Diesel engines. In addition, Hydsim software is also suitable for calculating and simulating the fuel supply process for gasoline engines and engines using other alternative fuels such as alcohol, biogas, LPG, etc. The software has been widely applied in developed countries and modern automobile companies such as Audi, Fiat, etc. Several officials, lecturers, students, and research and application students use the software in Vietnam.

Therefore, the authors applied Hydsim software to simulate the fuel supply process of the Kubota D1703-M-DI Diesel engine based on simulation conditions with rated load, average load, and no load and with a change in the value of structural parameters.[2,3] Based on the simulation results obtained, the authors will analyze and evaluate the simulation results and draw relevant conclusions about the influence of structural parameters on the quality of fuel injection on the Kubota Diesel engine D1703-M-DI. From there, proposals for improvements in more appropriate structural parameters are made. This is important in reducing fuel consumption, noise, and the content of pollutants in engine exhaust gases.[1,4]

2. ANALYSIS OF STRUCTURAL CHARACTERISTICS AND CONSTRUCTION OF A SIMULATION MODEL OF THE FUEL SUPPLY SYSTEM OF KUBOTA DIESEL ENGINE D1703-M-DI

2.1. Analyze structural features

2.1.1. Kubota Diesel Engine D1703-M-DI

The Kubota D1703-M-DI engine is a diesel engine with a 4-stroke, 3-cylinder, 1-2-3 firing order, placed vertically, compression ratio of 20:1, displacement of 1.647 liters, and maximum capacity of 22.7 kW power, etc. The engine uses a unified combustion chamber, has a ω-shaped piston top cutout, and injects fuel directly onto the piston top, creating a swirling movement of the gas flow, optimizing the process. Mix the mixture, thus reducing up to 50% of PM particles compared to level 2 of the EPA standard. The MoS_2 plating layer on the valve body and piston reduces noise by 1-2 dBA compared to a conventional engine. It is manufactured by the Japanese company Kubota and is used on tractors, such as L3408VN (Figure 1), in KUBOTA, Japan. The D1703-M-DI engine complies with temporary regulations on EPA (US) tier 4 exhaust standards and EU (European) stage 3A standards passed in 2012 and has been recognized by the European market.

Figure 1. KUBOTA L3408VN tractor.

2.1.2. Kubota D1703-M-DI Diesel engine fuel system

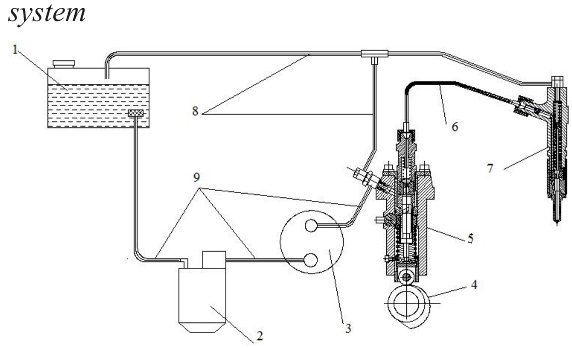


Figure 2. Structure diagram of Kubota D1703-M-DI Diesel engine fuel system. 1. Oil tank, 2. Oil filter, 3. Low-pressure pump, 4. Camshaft, 5. High-pressure pump, 6. High pressure pipe, 7. Injector, 8. Oil return pipe, 9. Supply oil pipe.

Kubota D1703-M-DI Diesel engine uses a Diesel engine fuel system with a 3-cylinder, in-line (5) centralized high-pressure pump of the

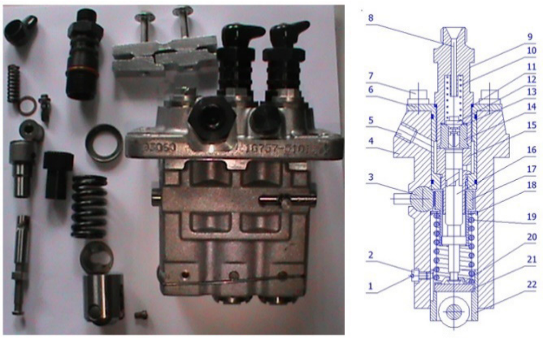
Bosch "K" Type Mini Pump. Each pump unit supplies fuel for one machine (Figure 3). Fuel is injected directly into the combustion chamber through closed nozzles (7) of the Bosch "P" Type Hole Nozzle type with five spray holes (Figure 11). The oil tank is stamped from a steel plate and has a capacity of 34 liters. Oil filters use a pleated paper-type filter element, which increases filtration efficiency and compact structure. A diaphragm-type low-pressure pump is located on the side of the engine body and is driven by the high-pressure pump camshaft.

The low-pressure pump sucks fuel from the tank through the filter into the pump and then is pumped to the high-pressure pump. Filters filter out dirt mixed in fuel. The high-pressure pump compresses the fuel further into the high-pressure line and then to the nozzle. Fuel is injected into the engine combustion chamber at the end of the compression process. Then, the injected fuel beam is shredded, heated, evaporated, and mixed evenly with air to create a mixture that spontaneously ignites. Excess fuel in the injector and high-pressure pump passes through the return valve to the tank.

a. High-pressure pump

The Kubota D1703-M-DI Diesel engine uses a centralized high-pressure pump with three machines in line and a fuel supply order 1-2-3. The removable camshaft is installed in the engine body. The high-pressure pump consists of 3 pump groups. Each pump group includes a pair of high-pressure pump piston-cylinders. The control of changing the amount of fuel supplied for each cycle is thanks to the oblique groove on the piston, which is controlled to rotate by the rack and pinion mechanism. The unique feature of this type of pump is adjusting the fuel quantity between pump units; we change it by rotating the adjustment tube (6) back or forth.

*** Structure (Figure 3):** This is a high-pressure pump that adjusts the amount of fuel supplied to the cycle with a piston valve, changing the amount of the fuel provided by changing the proper stroke of the piston. The main details of the high-pressure pump are the pair of pistons (15) and cylinders (16). This is a super precise pair requiring very high precision manufacturing. It has been chosen to be installed together, and when replaced, the entire pair must be replaced. The cylinder is inserted into the hole in the adjusting tube body, positioned by the pin on the adjusting tube (6). The space inside the cylinder is connected to the fuel chamber in the pump body by a port and to the high-pressure fuel line when the high-pressure valve is open.



(a) (b)

Figure 3. Structure of high-pressure pump of Kubota D1703-M-DI Diesel engine. (a) Actual image of high-pressure pump, (b) longitudinal section of a pump unit. 1. Locking ring, 2. Locating pin, 3. Pinion rack, 4. High-pressure pump body, 5. Oil inlet line, 6. Adjusting tube, 7. Retainer bolt, 8. Fuel to injector, 9. High-pressure hose connector, 10. High-pressure valve spring, 11. Oil seal, 12. High pressure valve, 13. Copper gasket, 14. High-pressure valve seat, 15. Cylinder, 16. Piston, 17. Pipe teeth, 18. Upper spring stop plate, 19. Piston return spring, 20. Lower spring stop plate, 21. Adjusting pad, 22. Roller head.

The high-pressure pump has another super-precise pair: the high-pressure valve (12) and valve seat (14). The screw (9) is screwed tightly onto the adjusting pipe body (the adjusting pipe is screwed tightly on the pump body) to press the high-pressure valve seat tightly onto the cylinder head (16) so the contact surface between the valve seat (14) and the cylinder (16) consistently tight. Thanks to the high-pressure valve spring (10), the high-pressure valve (12) is pressed tightly against the conical surface of the valve seat, separating the space above the piston of the pump unit from the high-pressure pipe. The tooth rack (3) and tooth tube (17) are connected to the piston tail, and the amount of fuel injection supplied to the cycle is adjusted.

*** Working principle (Figure 4):**

Fuel suction process: When the cam drives, the high-pressure pump rotates, and the roller rolls on the cam profile. When the rocker roller goes down, thanks to the force of the piston return spring, the high-pressure valve closes thanks to the high-pressure valve spring. At this time, under the influence of the high-pressure pump spring, it will push the piston (1) down, creating a vacuum in the cylinder chamber (5). When the piston opens the port (4), the fuel from the fuel chamber (3) will be loaded into the cylinder chamber until the piston is at its lowest position.

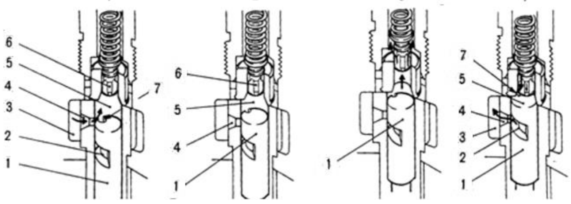


Figure 4. Fuel suction and ejection stroke of the high-pressure pump. 1. High-pressure pump piston, 2. Groove on piston, 3. Fuel compartment, 4. Diaphragm port, 5. Cylinder cavity, 6. High pressure valve, 7. Groove on high-pressure valve.

Fuel pushing process: When the rotating cam pushes up the piston (1), the fuel is initially pushed out through the hole (4). When the piston covers the hole, the pressure fuel supply process begins. In the high-pressure pump, the effect on the high-pressure valve continues to increase until the tension of the high-pressure valve spring and residual pressure on the high-pressure pipe are overcome, the high-pressure valve opens, and fuel enters the high-pressure line to the nozzle. The fuel supply process continues until the piston's inclined groove opens the port (4), ending the fuel supply. Leading to a sudden decrease in fuel pressure in the pump chamber, the high-pressure valve closes tightly on the valve seat (under the Influence of the high-pressure valve spring and fuel pressure on the high-pressure pipe). The fuel injection ends even though the piston continues to move up. Complete a fueling cycle and then repeat the cycle as above. Due to the throttling phenomenon of the port (4) and the compression phenomenon of the fuel, the actual supply start and end times are different from the geometric supply start and end times.

Controlling the amount of fuel supplied: To change the fuel provided to an engine's working cycle, we move the rack, and the gear tube rotates, making the piston rotate. This changes the proper stroke of the pump piston.

*** Structure of main details in high-pressure pump:**

- Ultra-precise piston and cylinder duo:

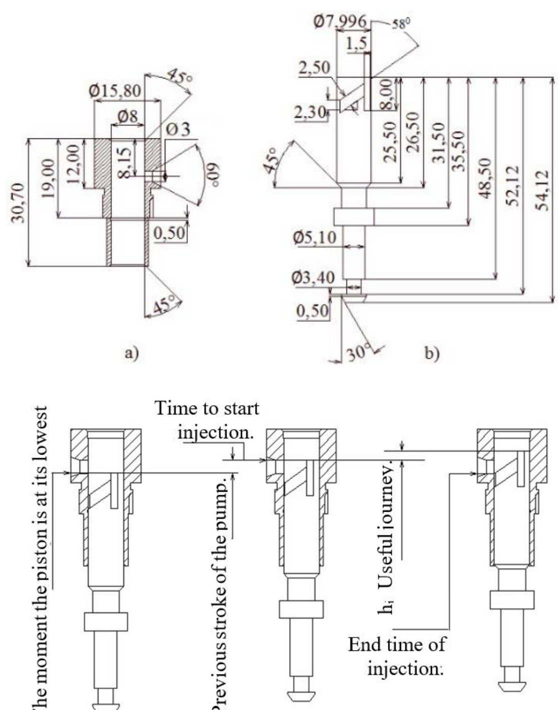


Figure 5. Measurement parameters of the high-pressure pump piston-cylinder duo. (a) Cylinder, (b) piston, (c) Pump working parameters.

High-pressure pump pistons and cylinders have precise geometric shapes and good wear resistance. The manufacturing material is Cr15 steel, which has a stable microstructure and more

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stable geometric dimensions. The part is heat-treated to meet the requirements of the friction surfaces of the piston and cylinder pair having a hardness not less than 58 HRC, and the end faces having a hardness not less than 55 HRC.

Main parameters of piston and cylinder (Figure 5):

+ The mass of the piston is m_p = 15.7 g.

+ Piston diameter: d_p = 7.996 mm.

+ Suction hole diameter: d_{lh} = 3.0 mm.

+ Spiral groove elevation angle (oblique): α = 32 degrees.

+ Spiral groove width (oblique): b = 2.5 mm.

+ Vertical chamfer width (vertical): b_1 = 1.5 mm.

+ Piston front stroke: h_t = 2.8 mm.

- Piston return spring: responsible for returning the pump piston during the cam lowering stroke. Make sure the roller is always in contact with the cam surface. The parameters of the piston return spring are shown in Figure 6:

+ Mass of piston return spring: m_{lx} = 16.6 g.

+ Number of twist steps: 5 steps.

+ Initial pressure of plunger piston spring: F_0 = 185 N.

+ Hardness: k = 32000 N/m.

+ Damping degree: C = 10 N.s/m.

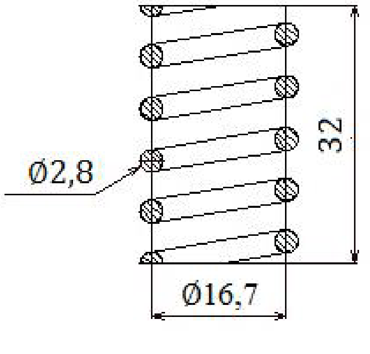

Figure 6. Measurement parameters of piston return spring.

- Roller jack:

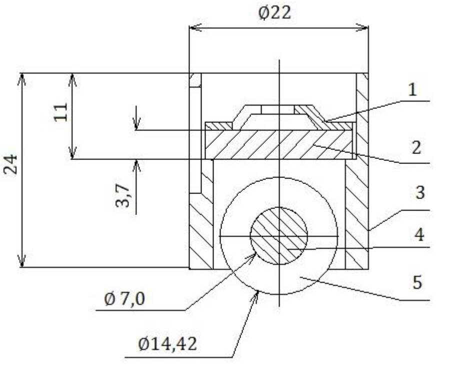

Figure 7. Measurement parameters of the roller handle. 1. Spring stop disc, 2. Adjusting pad, 3. Jack body, 4. Roller shaft, 5. Roller.

Roller jack: This helps reduce friction during contact between the roller and the cam. Thanks to that, the cam can rotate easily, avoiding cam jamming. It has a total mass of 46.5 g (including the spring stop disc and adjusting pad).

- Pair of high-pressure valves and high-pressure valve seats: Each pump unit is equipped with a high-pressure valve cluster, which has the following tasks: Prevent gas from the engine cylinder from entering the high-pressure pump cylinder; Prevent fuel on the high-pressure pipe from flowing back to the high-pressure pump cylinder; Complete the fuel supply process decisively, avoiding the phenomenon of dropped spray.

The high-pressure valve pair is a precision pair made of Cr15 alloy steel. The valve has a hardness after heat treatment of about HRC 56÷62, and the valve seat is HRC 60÷64. The valve and valve seat must be ground together. The tightness of a high-pressure valve is usually checked by using compressed air with a residual pressure of 0.4 ÷0.5 MN/m^2, immersing the valve in a barrel of kerosene; there must be no air bubbles. Main parameters of high-pressure valve assembly (Figure 8):

+ Mass of high-pressure valve spring: m_{lxv} = 1.9 g.

+ Mass of high-pressure valve: m_v = 2.2 g.

+ Valve spring hardness: k_{lxv} = 13500 N/m.

+ Valve spring damping degree: C_{lxv} = 5 N.s/m.

+ Valve seat hardness: k_{dv} = 50000000 N/m.

+ Valve seat damping degree: C_{dv} = 50 N.s/m.

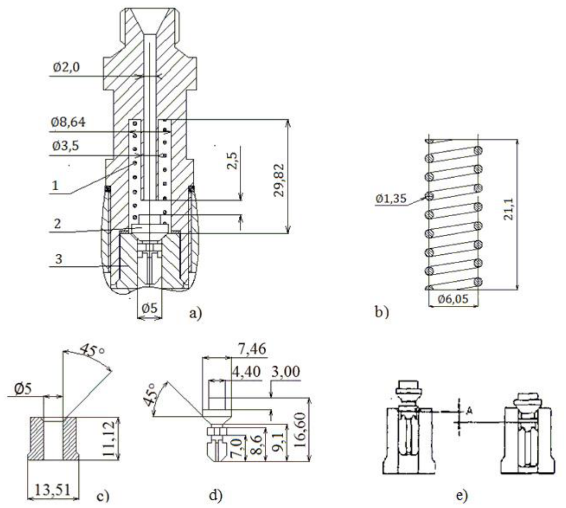

Figure 8. Measurement parameters of the valve duo and high-pressure valve seat. (a) High-pressure valve assembly, (b) high-pressure valve spring, (c) High-pressure valve seat, (d) High-pressure valve, (e) High-pressure valve lift stroke A, 1. High-pressure valve spring, 2. High pressure valve, 3. High-pressure valve seat.

- High-pressure pump camshaft: *The camshaft is cast in one piece and designed with an almost straight beveled cam face.* As the cam lift stroke increases, the piston's movement speed increases, rapidly increasing fuel pressure.

Main parameters:

+ Base circle radius of cam: R = 17.04 mm.

+ Roller radius: r_{roll} = 7.21 mm.

+ Effective width of roller: b_{roll} = 8.5 mm.

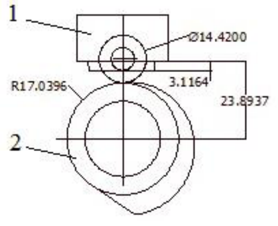

Figure 9. High-pressure pump cam profile of Kubota D1703-M-DI engine. 1. Roller shaft, 2. High-pressure pump camshaft.

b. High-pressure hose

A high-pressure steel pipe with high hardness is used to carry high-pressure fuel from the high-pressure pump to the high-pressure injector.

Figure 10. D1703-M-DI engine high-pressure hose.

Main parameters:

+ Overall length: l = 325 mm.

+ High-pressure oil hole diameter: d = 1.5 mm.

+ High-pressure pipe wall thickness: δ = 1.5 mm.

c. High-pressure nozzle

The Kubota D1703-M-DI engine uses a two-stage nozzle. The nozzle head is arranged with five spray holes with a diameter of about 0.2 mm distributed around with angles about 75 degrees apart to suit the Combustion chamber structure to create the best mixture. This injector will inject more fuel in the 2nd stage as the fuel pressure increases. Using a 2-stage nozzle reduces injection pressure to lift the injector, thereby improving low-speed injection stability and unloading capability. On the other hand, because the initial amount of fuel injection is small, it improves typing and smoothness of motion.

Two springs (No. 8 and No. 12) and a push rod (No. 14 and No. 13) are inside the nozzle. In this two-gap stage, a gap between pin 14 and pin 13 for fuel injection is called initial lift.

The initial lift, the tension of spring No. 8 (stage 1 fuel pressure), and the tension of spring No. 12 (stage 2 fuel pressure) are adjusted by replacing the corresponding adjusting pads. Nominate them.

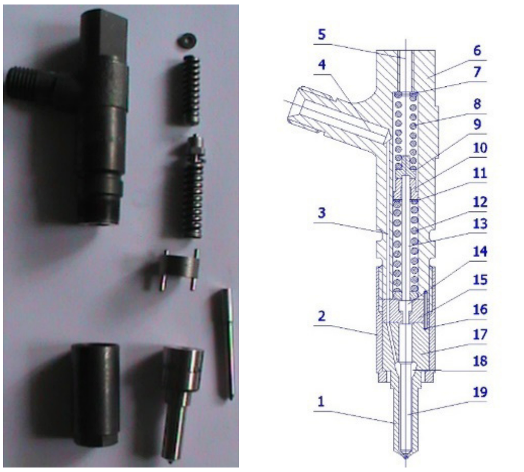

Figure 11. Structure of D1703-M-DI engine nozzle. 1. Injector body, 2. Locking nut, 3. Body longitudinal fuel pipe, 4. High-pressure hose connection, 5. Oil return line connection, 6. Injector body, 7. Adjusting gasket, 8. 1st stage spring, 9. Spring stop plate, 10. Guide tube, 11. Adjusting pad, 12. 2nd stage spring, 13. Push rod, 14. Intermediate connecting pin, 15. Setting base maximum injector lift stroke position, 16. Locating pin, 17. Nozzle body, 18. High-pressure chamber, 19. Injector.

Stage 1 (Figure 12): When the fuel pressure increases due to the operation of the high-pressure pump and reaches about 190 KG/cm^2, it overcomes the tension of spring number 8, pushing the injector upward and causing injection to begin. After pin 14 comes into contact with spring base 12, the lift of the injector only changes once the pressure increases to about 230 KG/cm^2.

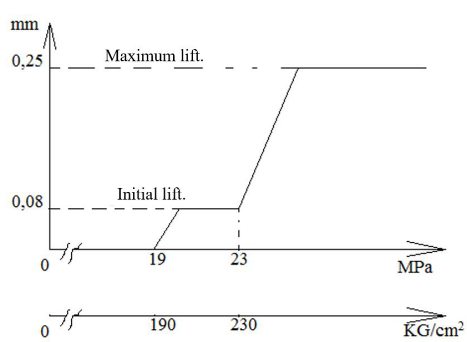

Figure 12. Graph of nozzle operating pressure evolution.

Stage 2 (Figure 12): when the fuel pressure reaches about 230 KG/cm^2, it overcomes the tension of springs No. 8 and No. 12 and lifts the injector higher. Once the injector contacts the spacer (15), the needle lift will not change anymore, even if the fuel pressure increases.

Therefore, when the engine is under a light load, only a tiny amount of fuel is injected at the low lift stage. On the other hand, under heavy load, a small amount of fuel is injected at the initial lift stage, and then large amounts of fuel are injected at a more advanced stage. The main parameters of the nozzle are shown in Figures 13 and 14.

+ Injector mass: m_{kim}= 2.9 g.

+ Spray hole length: l_{hole} = 0,8 mm.

+ Nozzle cavity diameter at hole: d = 1,0 mm.

+ Injector base angle: α_{seat} = 45^0.

+ Diameter of injector spring wire 1: d_{lx1} = 1.86 mm.

+ Average diameter of injector spring 1: d_{tb} = 4.52 mm.

+ Mass of injector spring 1: m_{lx1} = 3.0 g.

+ Diameter of injector spring wire 2: d_{lx2} = 2.05 mm.

+ Average diameter of injector spring 2: d_{tb} = 5.25 mm.

+ Mass of injector spring 2: m_{lx2} = 4.4 g.

+ Number of injector spring steps 1,2: 10 steps.

+ Push rod mass: m_{td} = 1.2 g.

+ Mass of connecting pin: m_{cn} = 0.4 g.

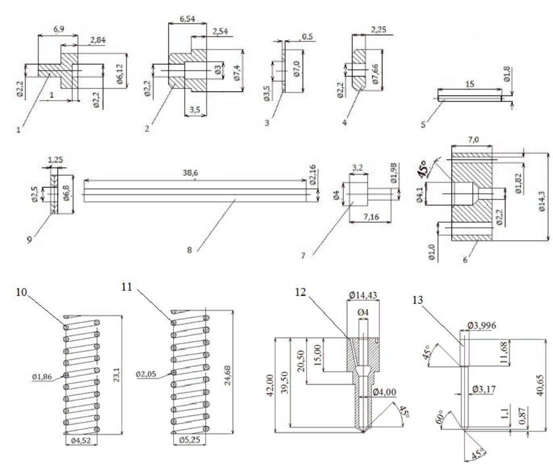

Figure 13. Parameters of the high-pressure nozzle of Diesel engine D1703-M-DI. 1. 1st stage spray spring stopper disc, 2. Guide tube, 3. 2nd stage spray spring adjustment pad, 4. 2nd stage spray spring stopper disc, 5. Locating pin, 6. Fixing base maximum injector lift stroke position, 7. Intermediate connection pin, 8. Push rod, 9. Stage 1 injection spring adjustment pad, 10. Stage 1 injection spring, 11. Stage 2 injection spring, 12. Nozzle body, 13. Injector.

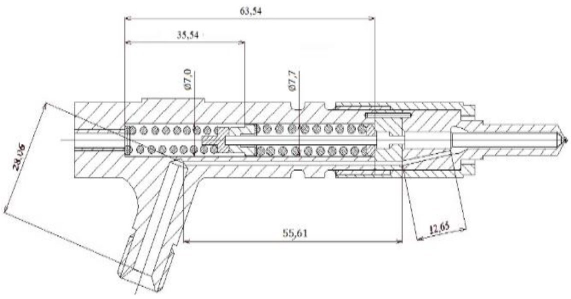

Figure 14. Basic parameters of the nozzle.

d. Combustion chamber shape

The Kubota D1703-M-DI Diesel engine uses a unified combustion chamber and ω-shaped piston top to create an airflow vortex, improving the quality of mixture formation.

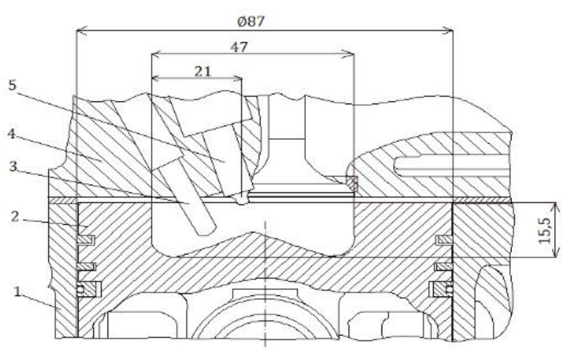

Figure 15. Combustion chamber parameters of Kubota D1703-M-DI Diesel engine. 1. Cylinder, 2. Piston, 3. Dryer shaft, 4. Engine cover, 5. Nozzle.

Conclusion: The quality of fuel injection in a Diesel engine (average fuel particle diameter, spray beam taper angle, and spray beam length) will determine the quality of mixture formation and combustion. Many parameters must be considered to achieve the best fuel injection quality, such as fuel system structural parameters, operating conditions, engine type, fuel properties used, etc. We must research, calculate, and choose to make these parameters optimal.

2.2. Build a model and simulate the fuel supply system of the Kubota D1703-M-DI Diesel engine

2.2.1. Algorithm diagram

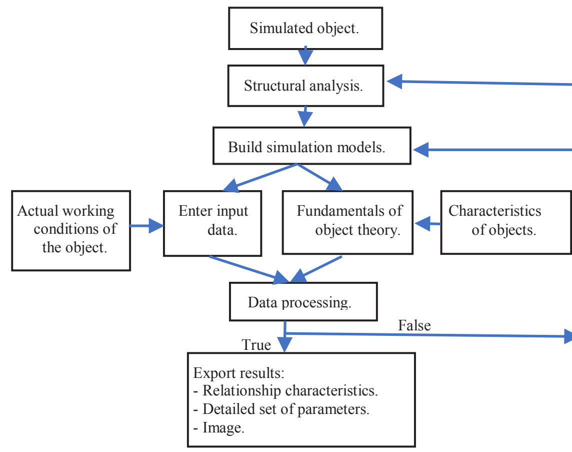

Figure 16. Algorithm diagram of the simulation program.

Based on simulation theory,[4] calculations on Hydsim software, and the structure and actual working conditions of the Kubota D1703-M-DI Diesel engine fuel system, we have created a schematic diagram of the Kubota D1703-M-DI Diesel engine fuel supply system simulation program (Figure 16).

2.2.2. Build simulation models

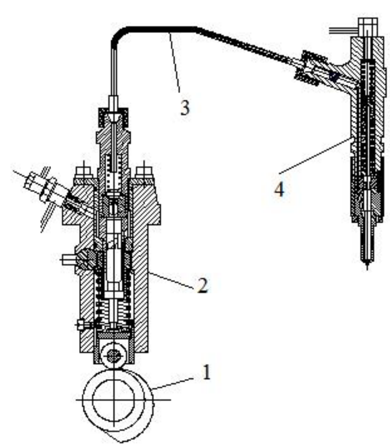

Figure 17. Main assemblies used for simulation. 1. High-pressure pump camshaft, 2. High-pressure pump assembly, 3. High pressure pipe, 4. Injector nozzle.

The detailed assemblies used to simulate the Kubota D1703-M-DI Diesel engine fuel system (Figure 17) structurally combine three main elements: a high-pressure pump, pipes, and a nozzle.

The in-line high-pressure pump consists of 3 pump groups. Because the pump groups have the same structure, the simulation is only performed for one pump group.

The structure of a pump unit includes Cam (4) (convex cam), which rotates thanks to the drive shaft, causing the pump piston to move up and down (piston plunger). The cam (4) has a convex cam lobe, so the plunger piston goes up and down once during its one revolution. The cavity above the plunger piston is the pressure chamber (chamber before the high-pressure valve). Fuel from the low-pressure pump enters the common intake chamber of the high-pressure pump (Pressure margin), then through the fuel inlet (inline inlet/overflow type) and into the high-pressure pump chamber. Here, the plunger piston compresses the fuel, which is pushed through the high-pressure valve (which has a pressure-reducing rim) to the chamber behind the high-pressure valve (this chamber has residual pressure). From here, high-pressure fuel follows the high-pressure pipe to the high-pressure injector (high-pressure oil chamber that lifts the injector). Part of the fuel leaks through the gap between the plunger piston and the cylinder through the port to the fuel tank.

The high-pressure pipeline transports high-pressure fuel from the high-pressure pump to each injector. During the working process, high-pressure pipes expand and contract.

High-pressure fuel from the high-pressure pipe follows the pipeline along the nozzle body to the high-pressure chamber in the nozzle body. When the oil pressure applied to the injector cone surface is enough to overcome the injector spring tension (2 springs), the injector is lifted (2 stages), and high-pressure fuel is injected into the engine combustion chamber through the spray hole on the nozzle head.

When the inclined groove on the pump piston head (metering groove) opens the inlet/pressure chamber in the high pump chamber before the high-pressure valve suddenly decreases, the high-pressure valve closes to reduce the pressure and the return force. The injector closes by pressing the high-pressure valve spring, ending the fuel injection.

From analyzing the structural characteristics of the fuel supply system of the Kubota D1703-M-DI Diesel engine, we select the corresponding elements in Hydsim software to perform the simulation. Once the corresponding elements have been identified, we create a block model of the equipment in the fuel system.

a. Create a high-pressure pump block model

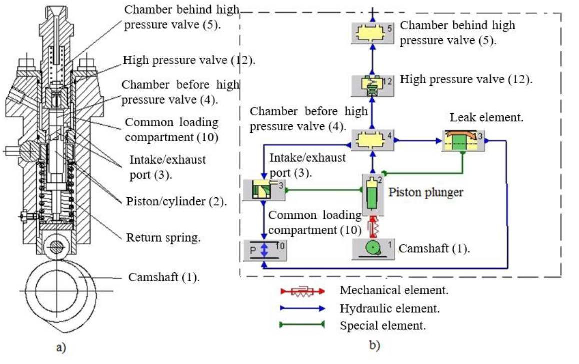

Figure 18. Model of Kubota D1703-M-DI Diesel engine high-pressure pump block. (a) Actual high-pressure pump structure, (b) High-pressure pump block model.

b. Create a high-pressure pipeline block model

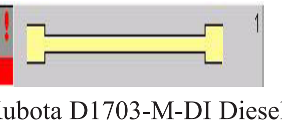

Figure 19. Kubota D1703-M-DI Diesel engine high-pressure pipe block model.

c. Create a nozzle block model

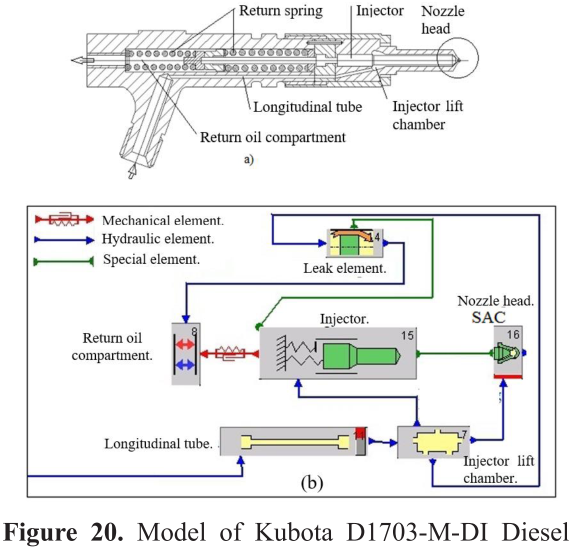

Figure 20. Model of Kubota D1703-M-DI Diesel engine high-pressure injector block. (a) Fundamental structure of high-pressure nozzle, (b) block model of high-pressure nozzle.

d. Create a block model of the combustion chamber

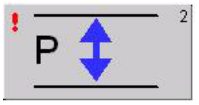

Figure 21. Model of the combustion chamber block of the Kubota D1703-M-DI Diesel engine.

2.2.3. Simulation model of Kubota D1703-M-DI engine fuel system

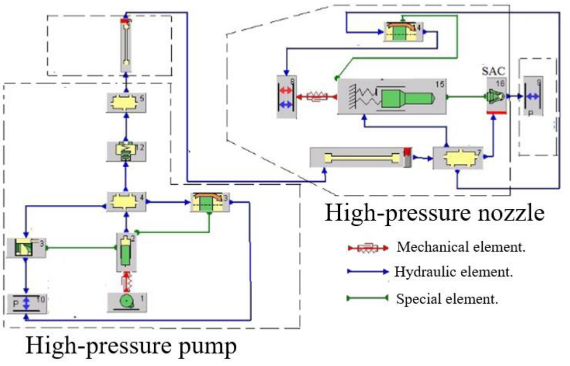

Figure 22. Simulation model of D1703-M-DI Diesel engine fuel system. 1. Camshaft (1), 2. Piston, 3. Intake/exhaust port, 4. Chamber before the high-pressure valve, 5. Chamber behind the high-pressure valve, 6. High pressure pipe, 7. Injector lift chamber, 8. Return oil compartment, 9. Combustion chamber, 10. Joint loading compartment, 11. Longitudinal tube, 12. High-pressure valve, 13,14. Leak element, 15. Injector, 16. Nozzle head SAC.

2.2.4. Analysis of simulation calculation mode of Kubota D1703-M-DI engine fuel supply system

The parameters of the fuel equipment, combustion chamber structure, charging mechanism, and engine rotation are designed with the rated working mode (revolution and rated load) to ensure good atomization and mixture quality. Changing the engine's working mode causes the quality of the mist and mixture to deteriorate, affecting the engine's economy, reliability, and longevity.

a. Rated load mode

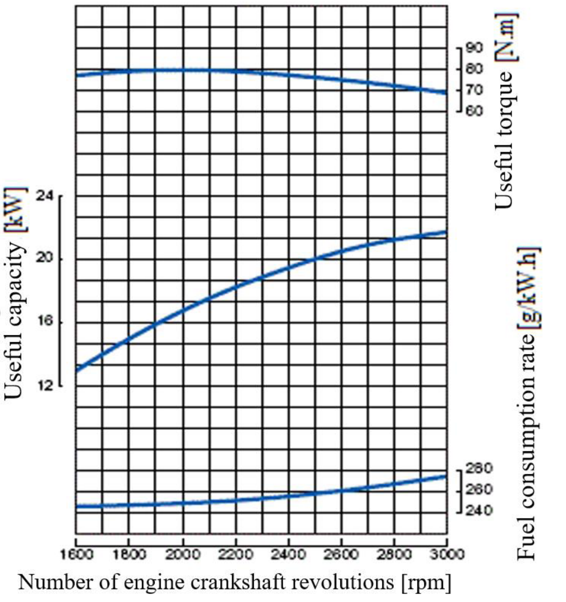

Figure 23. External speed characteristic graph of D1703-M-DI engine.

The characteristic parameter for the rated load mode at the rated number of revolutions and rated load is (h_{imax}: maximum helpful stroke of the high-pressure pump piston). Based on the external speed characteristic graph of the D1703-M-DI engine (Figure 23) and the technical specifications of the D1703-M-DI engine, we can determine that the rated speed of the engine is 2800 rpm, rated power N_e= 22.7 kW. The remaining thing is to determine the h_{imax}. The amount of fuel supplied to a cylinder during a working cycle is calculated according to the following formula:[2-4]

$$V_x = \frac{N_e.g_e.\tau.10^{-3}}{120.n.i.\rho_{nl}} \text{ [mm}^3\text{]} \quad (1)$$

We have: τ = 4; i = 3;

The D1703-M-DI engine uses Diesel fuel, so it has ρ_{nl} = 0.82 g/cm^3.

Considering the speed of 2800 rpm, look at the graph in Figure 23. We get Ne = 22.7 kW and g_e = 265 g/kW.h.

Substitute numbers into equation (1). We get V_x = 29,111 mm^3.

From there, we can calculate the proper stroke of the pump piston: h_{imax} ≈ 0.85 mm.

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b. Idle mode

Idle mode is when the engine operates stably at the lowest speed without external load. The no-load mode corresponds to the minor helpful stroke of the high-pressure pump piston (h_{imin}).

Figure 24. Measuring fuel consumption of Kubota D1703-M-DI engine.

Table 1. Engine fuel consumption without external load.

Accelerator pedal displacement (%)	0	20	40	60	80	100
Amount of fuel consumed in 60 seconds (g)	4	8.5	12	16	27	30.3
Number of crankshaft revolutions (rpm)	1000	1600	2000	2400	2650	2800
Proper stroke of H_{imin} pump piston (mm)	0.08	0.11	0.12	0.13	0.21	0.22

From the results of measuring actual fuel consumption (Figure 24) according to the accelerator pedal stroke from 0% to 100% when the engine is not carrying a load (Table 1), with the electronic weighing device's error being 0.1 g, we can determine the minimum helpful stroke of the high-pressure pump piston corresponding to the no-load mode $h_{imin} \approx 0.08$ mm.

c. Intermediate loading mode

Based on the actual working conditions of the Kubota D1703-M-DI engine on the L3408VN tractor, we see that when the tractor is operating (plowing), the engine almost operates in the area of 80 ÷ 100% of the table travel. Step on the accelerator (the tractor runs in gear 2) at a speed of 1600 ÷ 2000 [rpm]. However, when moving, changing direction, etc, the engine usually runs at an average load of 40 ÷ 70% of the accelerator pedal stroke, with a speed of about 2000 ÷ 2800 rpm. Therefore, determining the valuable stroke in local load modes is very complicated. For simplicity, we select the proper stroke at representative local load modes for general simulation as h_{itb}= 0.42 mm.

Conclusion: To see the influence of structural parameters of the fuel system and engine operating mode (speed, load, proper stroke of high-pressure pump) on the quality of the fuel supply process, we choose the simulation mode corresponding to 4 engine crankshaft speed positions: 1600, 2000, 2400, 2800 rpm. We conduct a thin tissue for each simulated speed position corresponding to the three high-pressure pump rack positions: h_{imax} = 0.85 mm, h_{itb} = 0.42 mm, and h_{imin} (Table 1).

2.2.5. Declare input data for elements

Declare input and output data for elements. I am declaring boundary conditions and properties of Diesel fuel. Run the simulation and export the results.

3. ANALYSIS AND EVALUATION OF THE INFLUENCE OF STRUCTURAL PARAMETERS ON THE QUALITY OF THE FUEL SUPPLY PROCESS OF THE KUBOTA D1703-M-DI ENGINE

3.1. Influence of cam profile

The Kubota D1703-M-DI engine high-pressure pump camshaft is designed with high rigidity and a sudden growth profile (almost straight bevel).

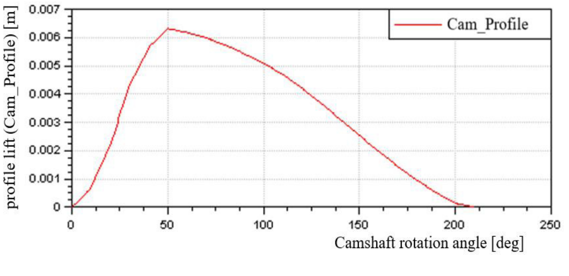

Figure 25. Lift graph of the cam profile.

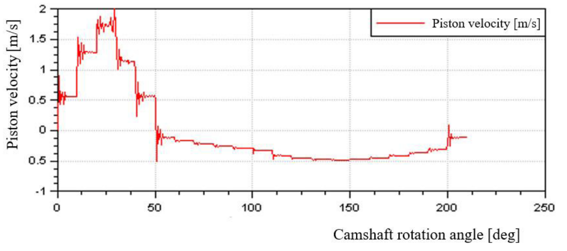

Figure 26. Graph of variation of pump piston displacement speed.

The lift of the cam profile is in the x direction (Figure 25). During the cam lift stroke, the piston is raised at high speed (Figure 26), and the fuel pressure in the pump chamber increases rapidly, leading to high chamber pressure behind the valve. Pressure, high-pressure pipe, and injector lift chamber increase quickly (Figure 27). When this pressure overcomes injector spring tension, the injector lifts fuel into the engine combustion chamber. The process of injecting fuel into the combustion chamber lasts until the inclined groove on the piston opens, and the injection process ends.

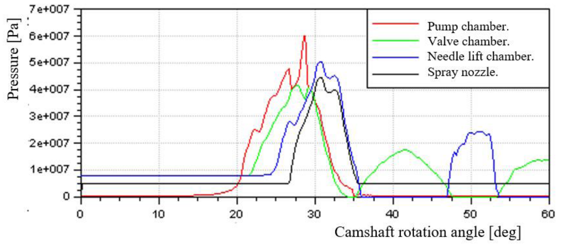

Figure 27. Graph of fuel pressure variation in the fuel system.

Due to the cam profile's sudden growth, the system's fuel pressure increases rapidly. The compression stroke of the piston is turbulence. This turbulence is in the form of a pressure wave that propagates in the fuel line in an elastic medium with the speed of sound; this speed depends on the compressibility and fuel density. The pressure wave reaches the injector later than the fuel supply period. High injection pressure and fuel compressibility not only cause a phase shift in the fuel supply in the pump and nozzle but also cause complex oscillatory motion of the fuel layer in the high-pressure pipeline, thus causing the circulation process to change. The movement of fuel through the injector orifice sometimes has a pulsating character. The source of pressure fluctuation interference is the movement of pistons, injectors, and high-pressure valves. Due to strong fluctuations in fuel pressure, the injector can open repeatedly after closing, causing a spray drop. Fuel injection on the expansion line has poor atomization quality due to low injection pressure. Drop injection increases the combustion period on the expansion path and reduces engine economy.

The fuel injection quality is quite good in the delay and main injection stages due to the high fuel injection pressure. The average fuel particle diameter is small and uniform (Figure 28), and the spray beam taper angle (Figure 29) and the spray beam length (Figure 30) gradually increase. However, in the free-flow phase (fuel supply has stopped), the fuel injection occurs thanks to the high-pressure pipeline's fuel compression energy and elasticity. Hence, the injection pressure gradually decreases, the diameter of the average fuel particle increases, and the spray beam taper angle and the spray beam length decrease progressively.[2-5]

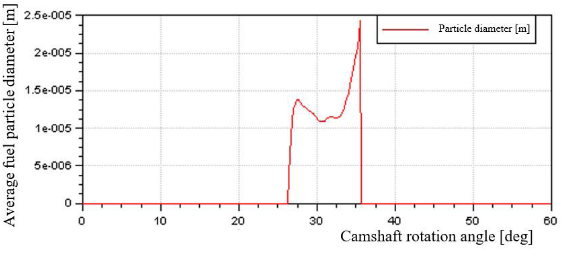

Figure 28. Graph of average fuel particle diameter variation.

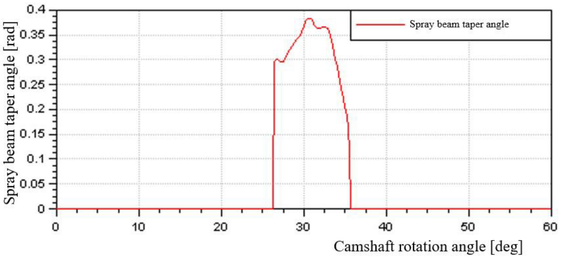

Figure 29. Graph of fuel injection beam cone angle variation.

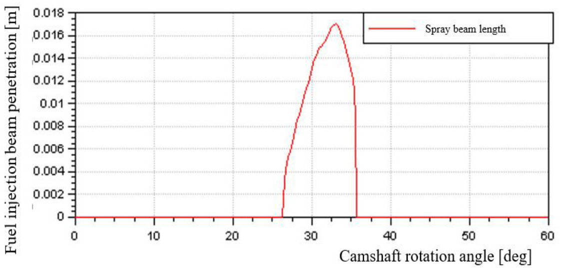

Figure 30. Graph of fuel injection beam length variation.

In general, the quality of fuel injection at the beginning and end of injection is not good, negatively affecting the quality of mixture formation and combustion, reducing engine economy, and causing environmental pollution. The quality of fuel injection is relatively good at about 30 ÷ 33^0 camshaft rotation angle (average fuel particle diameter is small, and even spray beam taper angle is significant, and beam length is large enough. [2-5]

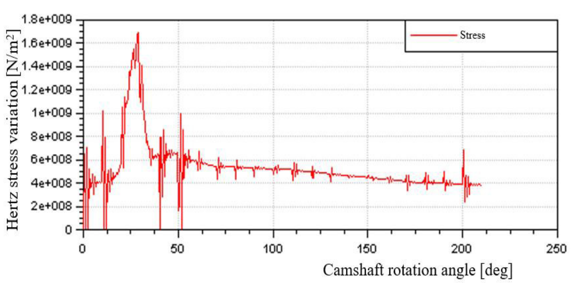

Figure 31. Hertz stress variation graph.

Value and variation of contact stress between roller file and convex cam during the working process (Figure 31). The most significant stress at the position where the plunger piston goes up compresses the fuel with the highest pressure at about 28-29^0 camshaft rotation angle (GQTC). Then the stress decreases rapidly corresponding to the moment the inclined groove on the piston head opens the exhaust port; the fuel pressure in the pump chamber suddenly decreases along with the elasticity of the high-pressure pump spring, causing the stress to fluctuate strongly, then The stress gradually decreases when only the elastic force of the spring remains, causing the cam lowering stroke. [2-5]

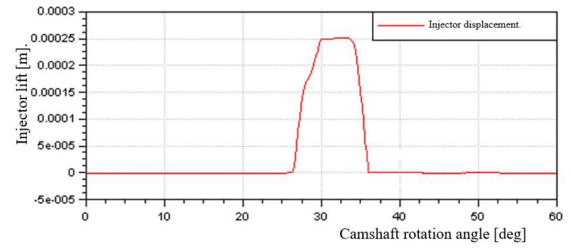

Figure 32. Graph of injector lift variation.

Due to the sudden lift of the cam, the pressure in the injector lift chamber increases rapidly. The hydraulic force acting on the needle is vast, overcoming the tension of both injector springs, so the injector is lifted to inject fuel into the engine combustion chamber. When fuel begins to be injected into the combustion chamber, the pressure in the injector lift chamber decreases slightly. Then, it continues to increase, causing a slow increase in the injector lift stroke.

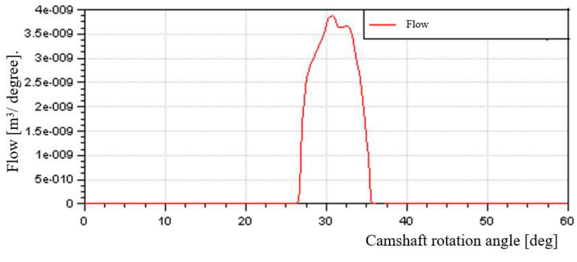

Figure 33. Graph of fuel injection flow variation.

The amount of fuel supplied in one working cycle of a pumping unit corresponding to the rated mode is about 26 mm^3 (Figure 34). Due to low injection pressure, the flow is small at the beginning and end of the injection process. The most enormous injection flow is about 3.8 mm^3/degree (Figure 33). The injection time is about 0.00119 s.

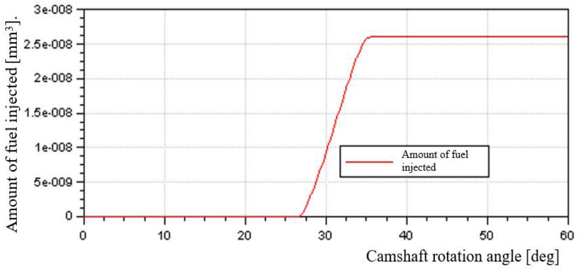

Figure 34. Graph of fuel injection volume variation in one cycle.

3.2. Influence of D/h_i ratio

To study the influence of the ratio between piston diameter and proper stroke (D/h_i) on fuel injection quality under the same conditions. Corresponding to the same amount of fuel supplied for one engine cycle at rated load mode is 29.11 mm^3. We choose the piston diameters to be 8, 7.5, 7, 6.5, and 6 mm. We can calculate the maximum beneficial journey: 0.85, 0.96, 1.11, 1.28, 1.50 mm.

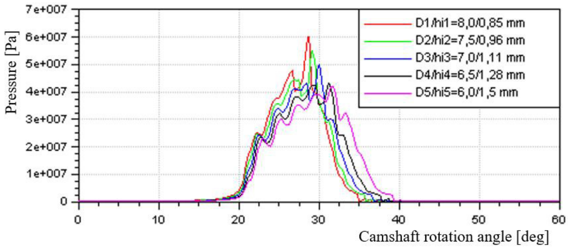

Figure 35. Influence of D/h_i ratio on pump chamber fuel pressure.

Simulation results show that the more minor the piston diameter, the larger the piston's helpful stroke, the lower the pressure in the system, and the larger the fluctuations (Figure 35). However, the pressure pulse at the end of the injection process has a gradually decreasing amplitude (Figure 36), meaning that the spray drop phenomenon is slowly overcome. As the pressure in the injector lift chamber decreases, the hydraulic force acting on the injector also decreases (Figure 37), so the injector lift gradually decreases (Figure 38). Therefore, the fuel circulation cross-section through the injector base also decreases (Figure 39). As a result, the fuel flow through the nozzle decreases (Figure 40), prolongs the fuel injection completion time, and negatively affects the mixture formation and combustion process.[2-5]

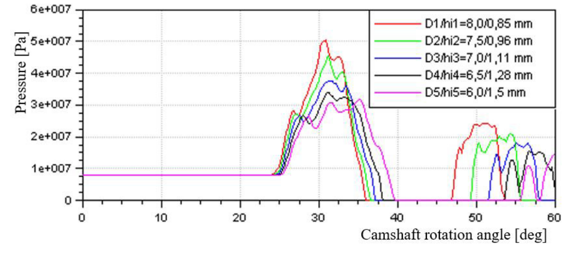

Figure 36. Influence of D/h_i ratio on injector lift chamber pressure.

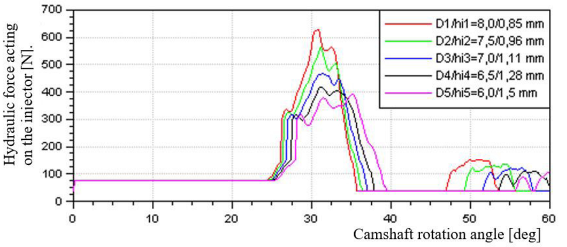

Figure 37. Influence of D/h_i ratio on the hydraulic force acting on the injector.

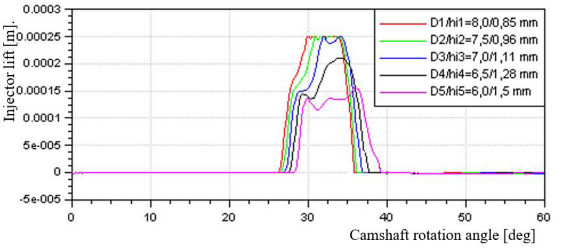

Figure 38. Influence of D/h_i ratio on injector lift.

As the piston diameter becomes smaller, the proper stroke of the piston becomes larger. We can see that the injection start time is gradually delayed, the average particle diameter gradually increases (Figure 41), the spray beam taper angle decreases (Figure 42), and the direction of the spray beam decreases (Figure 42). The spray beam length gradually decreases (Figure 43), and the injection end time is slowly delayed. The average fuel particle diameter gradually decreases, the spray beam taper angle decreases (Figure 42), and the spray beam length gradually increases (Figure 43).

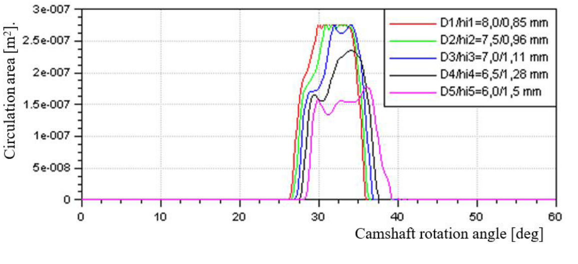

Figure 39. Influence of D/h_i ratio on the flow cross-section to the injector.

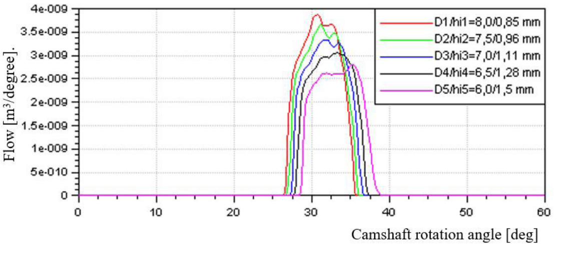

Figure 40. Influence of D/h_i ratio on fuel injection flow.

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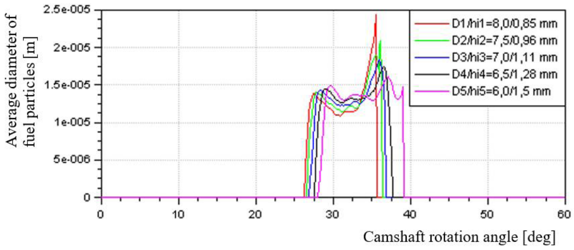

Figure 41. Influence of D/h_i ratio on average fuel particle diameter.

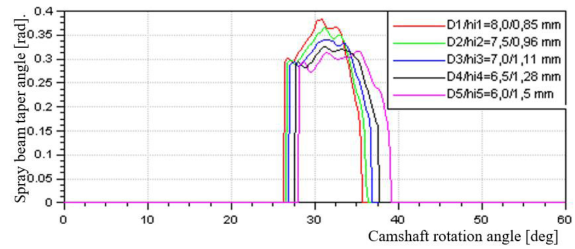

Figure 42. Influence of D/h_i ratio on fuel injection beam taper angle.

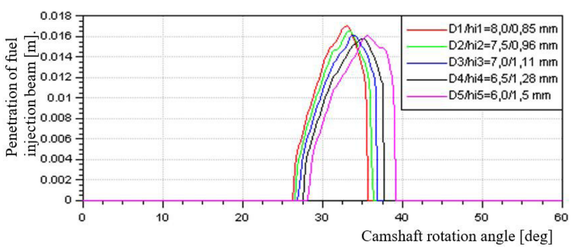

Figure 43. Influence of D/h_i ratio on fuel injection beam length.

Conclusion: When changing the D/h_i ratio, we see a massive change in the system's fuel injection quality. With the same amount of fuel supplied to a cycle, the more we reduce the piston diameter, the more helpful stroke the pump piston will have, so the injection time will be longer. Due to the decrease in piston diameter, the instantaneous compressed fuel flow through the high-pressure valve decreases, and the pressure in the system decreases. The force lifting the injector decreases, the fuel flow through the injector decreases, and the fuel flow leaking through the injector decreases. The average fuel particle diameter is generally small and uniform throughout the injection process. The spray beam taper angle and spray length are reduced but not significantly.

3.3. Influence of nozzle diameter

When other conditions are equal, we change the nozzle hole diameter by 0.18, 0.22, 0.25 mm, and 0.30 mm and simulate rated load mode. Consider the impact on fuel injection quality compared to when the nozzle hole diameter is 0.20 mm. The results show that when the nozzle hole diameter is smaller (0.18 mm), the fuel flow through the nozzle is more minimum (Figure 47), and the spray beam quality is uneven. In the delay and main injection stages, the average fuel particle diameter is the smallest (0.1÷0.13) µm. Still, in the free injection stage at the end of the injection process, the average fuel particle diameter is 0.38 µm (Figure 44), the spray beam taper angle (Figure 45) and the spray beam length are too small (Figure 46), and when spraying starts at the nozzle mouth, mist often condenses into mist, and at the end of the spraying process. There is a phenomenon of spray drop due to pressure pulsation in the system. Adversely affects the quality of the mixture formation and combustion process, reduces engine economy, and causes environmental pollution.[2-5]

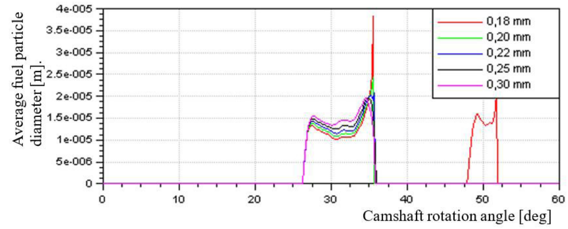

Figure 44. Influence of nozzle diameter on average fuel particle diameter.

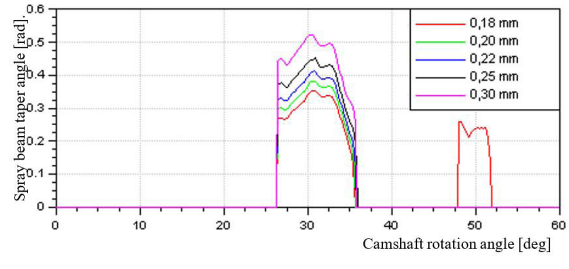

Figure 45. Influence of nozzle diameter on fuel injection beam taper angle.

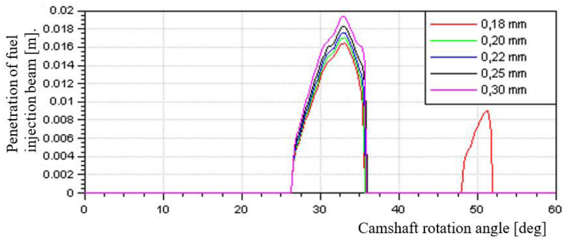

Figure 46. Influence of nozzle diameter on fuel injection beam length.

As the nozzle hole diameter grows, the fuel flow through the nozzle increases (Figure 47), so the fuel injection pressure gradually decreases (Figure 48). This results in the average fuel particle diameter increasing progressively (Figure 44), the spray beam taper angle gradually increasing (Figure 45), the spray beam length increasing progressively (Figure 46), and the average final fuel particle diameter being too high. The spraying process gradually decreases. At the same time, it overcomes the phenomenon of creating pressure pulses in the system, avoiding the phenomenon of spray drops. However, the diameter of the nozzle hole must be manageable, leading to the average diameter of the fuel particles being too large, rough, complex to tear apart, and evaporation slow, leading to poor mixture formation quality. [2-5]

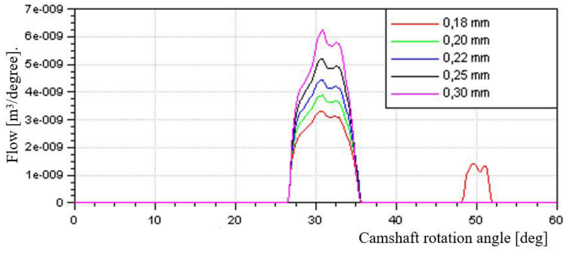

Figure 47. Influence of nozzle diameter on fuel injection flow.

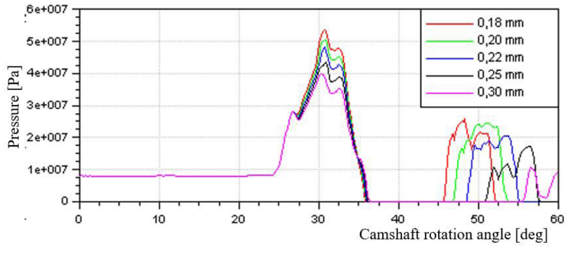

Figure 48. Influence of injection hole diameter on injector lift chamber pressure.

Conclusion: When increasing the nozzle diameter from 0.20 mm to 0.22, 0.25, and 0.30 mm, the spray circulation area will increase, the amount of fuel injected will increase, and the fuel flow will increase significantly. Therefore, the oil pressure in the injector lift chamber decreases, increasing the average fuel particle diameter, spray taper angle, and spray length (poor tearing level). In this case, we see a decrease in fuel flow leaking through the injector.

When the nozzle diameter is reduced from 0.20 mm to 0.18 mm, the spray circulation cross-section will be reduced, and the amount of fuel injected and the fuel injection flow will decrease. Therefore, the oil pressure in the injector lift chamber increases, and the injector opens large, decreasing the average fuel particle diameter, spray taper angle, and spray length (due to too strong tearing). In this case, we see an increase in fuel leaking through the injector. However, in this case, pressure pulses appear, causing spray drops that negatively affect the quality of the mixture formation process.[2-5]

3.4. Influence of nozzle length

Increasing the nozzle length from 0.8 mm to 1.0 mm decreases the spray cone angle and vice versa. Reducing the nozzle length from 0.8 mm to 0.6 mm decreases the beam taper angle and the spray increases.

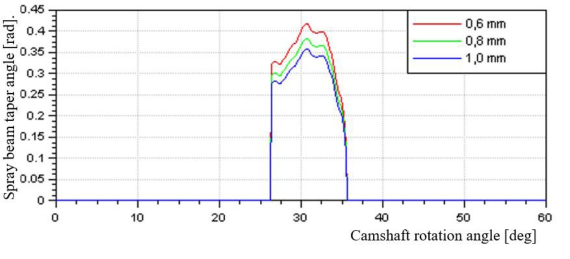

Figure 49. Influence of nozzle length on spray beam taper angle.

https://doi.org/10.52111/qnjs.2024.18310

3.5. Influence of nozzle spring stiffness

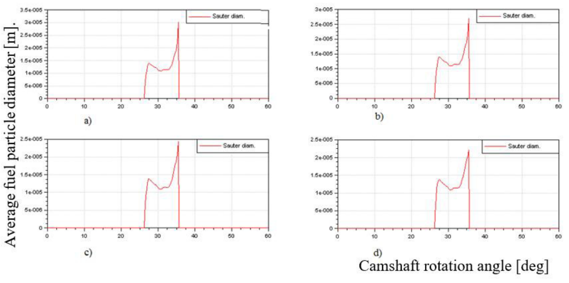

Figure 50. Influence of injector spring stiffness on average fuel particle diameter. (a) k_1=200000 N/m and k_2=300000 N/m, (b) k_1=225000 N/m and k_2=325000 N/m, (c) k_1=250000 N/m and k_2=350000 N/m, (d) k_1=275000 N/m and k_2=375000 N/m.

It remains constant when the stiffness of the two injector springs is changed with other conditions. We can see that the greater the injector spring stiffness, the better the fuel injection quality the smaller and more uniform the average diameter of fuel particles at the end of the injection process. However, the hardness of the two injector springs must be manageable, which will affect the injector lift and fuel circulation cross-section through the injector base. [2-5]

4. CONCLUSIONS

Through the process of researching and applying HydSim software to simulate, analyze, and evaluate the influence of structural parameters on the quality of the fuel supply process of the D1703-M-DI engine, we see that Hydsim is a software used to simulate and calculate the fuel system is quite powerful. The simulation parameters achieved are close to the actual parameters of the engine, and the simulation results accurately reflect the influence of structural parameters, operating conditions, and combustion chamber pressure on fuel injection quality.

During the implementation of the project, with many runs and tests, the desired analytical results were achieved. In general, the quality of the fuel injection beam is satisfactory. Still, there are some limitations, such as at high speeds and large loads, the average diameter of fuel particles at the end of the extensive injection process, there is a phenomenon of spray drop; the fuel injection beam taper angle is small, the spray penetration is slight (L/S<1.05). To improve fuel injection quality, we can proceed by:

- Because the cam profile is not reasonable, the pressure in the system increases suddenly and fluctuates strongly. When the engine operates at high speed and high load at the end of the injection process, because the pressure in the system is still significant and fluctuates strongly, the injector does not close tightly, leading to the injection falling.

- The diameter and helpful stroke of the pump piston are not suitable. Combined with the sudden growth of the cam profile and high operating speed, the time for fuel injection into the combustion chamber needs to be longer. The free flow injection period is extended. The pressure drops sharply, so the average diameter of fuel particles at this stage is extensive, negatively affecting the quality of mixture formation and combustion. According to the simulation results of Section 3.2, the best spray quality is selected with a piston diameter of 7 mm and a piston stroke of 1.11 mm.

- The nozzle hole diameter is small, so when the engine operates at high speed and load, the injector opens repeatedly, causing spray to drop or spray on the expansion line. The simulation results in Section 3.3 show that the best spray hole diameter is from 0.22 ÷ 0.25 mm.

- The length of the spray hole only affects the spray beam taper angle. Due to the considerable design nozzle length, the spray beam taper angle is slight, and the ability to fill the combustion chamber space could be improved. Therefore, the mixing quality could be better. The larger the spray beam, the better. According to the simulation results of Section 3.4, we need to reduce the spray hole length to 0.6 mm.

- The injector hardness is not suitable and small, causing the injector to close slowly and not decisively, leading to dropped injection, spraying on the expansion path, and the average size of fuel particles at the end of the injection process is significant. According to the simulation results in Section 3.5, it is necessary to increase the injector hardness (k_1 = 275000 N/m and k_2 = 375000 N/m).

In addition, other structural parameters of the fuel system, such as pump spring stiffness, high-pressure pipe length, the viscosity of fuel used, etc., and the influence of operating parameters such as speed, the degree of the high-pressure pump camshaft, the proper stroke of the high-pressure pump (h_i), and the compressed air pressure in the combustion chamber at the time of fuel injection also need to be thoroughly studied as a whole. From there, we propose solutions to improve this engine fuel system to become more complete and more suitable for operating conditions in our country.

Acknowledgment

This study is conducted within the framework of science and technology projects at Quy Nhon University's institutional level under project code T2023.808.18.

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