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Numerical analysis of the impact of injection rate shape at injection pressures up to 350 MPa on the performance and emissions of a diesel engine

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The fuel system of a diesel engine is a crucial component that significantly impacts the economic and technical performance, as well as the quality of exhaust emissions released into the environment. Improving fuel injection pressure, combined with changes in fuel injection rules, the number of fuel injections per cycle, and the geometric dimensions of the injectors, helps improve efficiency, reduce harmful emissions, and lower fuel consumption. This study evaluated the effect of the shape of the fuel injection characteristic at injection pressures up to 350 MPa, with the intake air pressure maintained at 0.25 MPa, using a 3D model in AVL Fire. The results showed that at ultra-high fuel injection pressures, the amount of particulate matter from the diesel engine was very small, ranging from 0.0012 to 0.0062 g/kWh. By changing the distribution rules, the minimum amount of NO_x was formed, corresponding to the case where the injection velocity was highest in the initial phase (case 1 and a value of 9.51 g/kWh) and gradually decreased in the latter phase of the injection process (case 5 and a value of 21.01 g/kWh). In addition, engine power decreased by 1.9%, from 23.32 to 22.88 kW, and fuel consumption increased by 1.8%, from 216 to 220 g/kWh. Due to inadequate fuel mixing between zones and fuel being mixed leaner than the combustion limit when increased injection pressure is used with a large injection volume in the initial phase. As a result, the maximum HC amounts increase by 4.8 times and the maximum CO amounts increase by 1.6 times compared to other injection cases.

Key words: *injection rate shape, diesel engine, 3D simulation, exhaust emission.*

1. Introduction

To date, fossil fuel engines continue to play an extremely important role in powering vehicles, agricultural machinery and construction machinery. Although the number of electric vehicles is increasing rapidly (reaching 20% of vehicles sold by 2024), fossil-fuel vehicles still play a dominant role in meeting people's transportation and travel needs [7]. Along with the issue of global warming caused by greenhouse gases and exhaust emissions from internal combustion engines, emission gas requirements for these engines are becoming increasingly stringent. Besides, technical improvements to enhance performance are also a focus of research. Research directions that have been and are being considered as follows: 1) research on the use of alternative fuel sources that do not create greenhouse gases [5, 9, 21, 23]; 2) research on optimizing engine operation to increase efficiency and reduce fuel consumption [6, 20]; 3) research on improving fuel injection process combined with the use of exhaust aftertreatment solutions remove harmful pollutants [2, 8, 16] and 4) research on calculating and redesigning engine parts [1, 12].

Solutions to improve fuel injection processes, combined with exhaust aftertreatment that removes harmful pollutants, are receiving widespread research attention, especially as materials and manufacturing technologies improve. Replacing mechanical fuel injection systems with electronic ones allows the injection process in diesel engines to be controlled and adjusted more precisely. Studies have been conducted, including increasing injection pressure, increasing the number of injections per cycle, adjusting the distribution rules in the combustion chamber, and adjusting the geometric dimensions of the injectors [10, 14, 17]. The fuel spray is determined experimentally using high-resolution optical devices. This allows for the study of fuel spray development, atomization, and the visual observation of the fuel spray's geometric parameters [18, 22].

It is possible to use needle-seat throttling or adjust the injection pressure to control the fuel distribution into the combustion chamber when using an electronic fuel injection system. In particular, the injection pressure control method helps the engine balance the NO_x-soot trade-off and reduce fuel consumption compared with the needle-seat throttling solution [19]. Using boost-shaped produced significant changes in the diffusion combustion phase, and the NO_x amount reduced [3, 4]. The injection velocity significantly affects spray penetration, heat release rate, the NO_x-soot trade-off, and engine noise during combustion. Studies have been conducted with injection pressures reaching 1700 bars. The results demonstrate that a slow initial injection rate and a gradual increase in the later stages will help reduce NO_x and increase soot [11]. Variations in injection shape were performed with B20 biofuel and 5 different injection shapes. The results showed that the optimal injection shape could help reduce NO_x by 11% and soot by 4%. However, the study was conducted only at an injection pressure of 350 bar [15].

Modern diesel engines use fuel systems that achieve 170 MPa and can reach up to 240 MPa. However, studies have shown that increasing injection pressures combined with adjustments to nozzle diameter, boost pressure, and exhaust gas recirculation is an effective solution or achieving Euro V emission standards without the need for additional exhaust treatment systems [24]. Case studies with pressures up to 300 MPa have yielded very good results for both traditional and alternative fuels. With the use of common-rail fuel systems and improved component manufacturing technology, increasing the pressure beyond 300 MPa is a promising direction.

Although increasing fuel injection pressure improves engine performance and emissions, this solution also has some limitations that need to be addressed in the future. Firstly, it increases the power loss in driving the fuel pump. Secondly, fuel leakage increases due to the direct impact of injection control and the manufacturing precision of the system's components. Furthermore, high injection pressure makes it difficult to control the fuel injectors, especially in modes with low injection volumes. In addition, the cost of an ultra-high-pressure fuel system, along with the necessary turbochargers, increases the engine's overall cost.

Within the scope of this study, an evaluation was conducted to assess the effects of increasing the injection pressure to 350 MPa and changing the injection rate shape in the combustion chamber on the engine operating parameters and harmful emissions. This foundational research helps shape the mechanical design and explore the limits of injection pressure ranges for next-generation fuel systems. The results obtained provide a basis for controlling the injection process of modern diesel engines.

2. Research subject and research model

In this study, a 3D model of a single-cylinder turbocharged diesel engine was established using the AVL Fire package. The engine has a compression ratio of 15.4 and uses an electronic fuel-injection system with injection pressures up to 350 MPa. The engine used for the study has the parameters shown in Table 1.

Table 1. Basic parameters of the engine under study [12]

Item	Descriptions
Engine type	4-stroke, single-cylinder CRDI
Number of cylinder	1
Bore × Stroke	120 mm × 130 mm
Connecting rod length	224 mm
Displacement	1.47 L
Compression ratio	15.4:1
Injection pressure	Up to 3500 bar
Intake air pressure	0.25 MPa
Number of nozzle holes	8 (–)
Nozzle hole diameter	0.15 mm
Intake valve: Open Close	 10°CA bTDC 15°CA aTDC
Exhaust valves: Open Close	 50°CA bTDC 6°CA aTDC

Important parameters when establishing the model are the fuel injection quantity and its characteristics. These values need to be determined experimentally and correspond to the fuel system used. Figure 1 shows the experimental results for the fuel injection quantity at injection pressures ranging from 100 to 350 MPa. To determine the injection amount according to the control signal pulse, an experiment with an ultra-high-pressure fuel system was performed as the experimental scheme presented in previous studies [13]. The injection amount value was averaged over 100 injection cycles and performed 3 times consecutively. From the experimental results, we select and input into the 3D model the injection quantities corresponding to different engine load conditions.

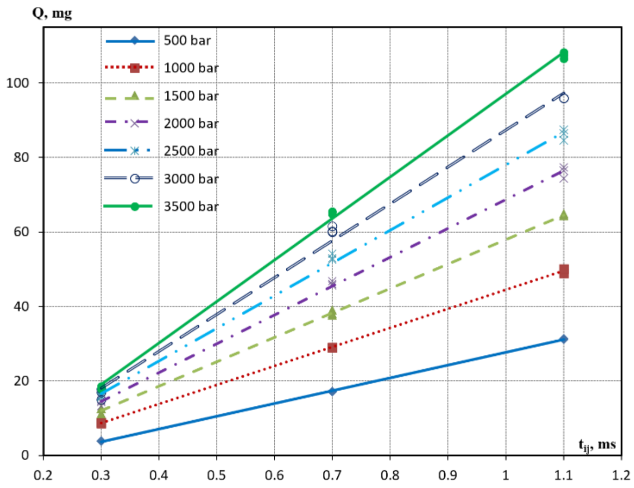


Fig. 1. Experimental results of fuel injection quantity according to injection pressure and control signal pulse

In addition to fuel injection quantity, engine operating parameters are determined by actual operating conditions during testing. The basic parameters are collected and entered into the 3D model as shown in Table 2.

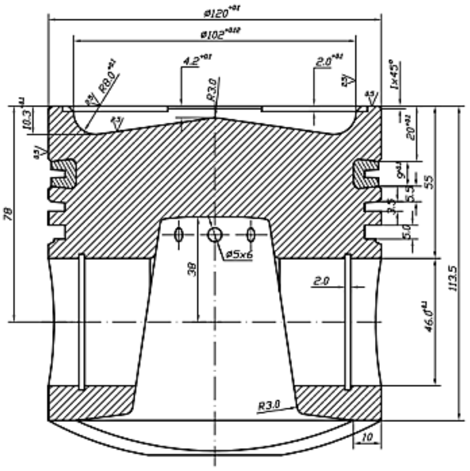
Table 2. Boundary conditions of the 3D model in AVL Fire

Conditions	Descriptions
For diesel engine	
Engine speed	1400 rpm
Boost pressure	0.25 MPa
Intake air temperature	307 K
Cylinder head temperature	550
Cylinder wall temperature	475 K
Piston top temperature	330
Fuel injection temperature	330
Fuel mass per cycle	120 mg
Start of injection	17°CA bTDC
Environmental standards	EURO 4
For 3D model	
Spray model	Wave
Spray wall interaction model	Walljet1
Droplet Evaporation model	Dukowicz
Turbulence model	k-zeta-f
Ignition model	Auto-ignition
NO formation	Extended Zeldovich
Soot formation	Kinetic

In addition to the operating parameters, to establish the 3D engine model for the study, we selected the ECFM-3Z combustion model, the k-zeta-f turbulent model, and the automatic ignition model. In addition, the Dukowicz fuel droplet evaporation model and the Wave spray model were chosen. Finally, to determine NO and soot formation in the engine, we selected the Zeldovich extended models and their corresponding kinetic models.

Figure 2a shows the detailed drawing of the actual dimensions of the experimental engine piston. After importing the data into the model and meshing, we obtained the results shown in Fig. 2b, with each cell having a size of 1 mm. This value helps reduce computation time and provides sufficient results to assess changes in engine performance [12].

a)



b)

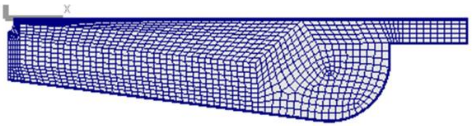
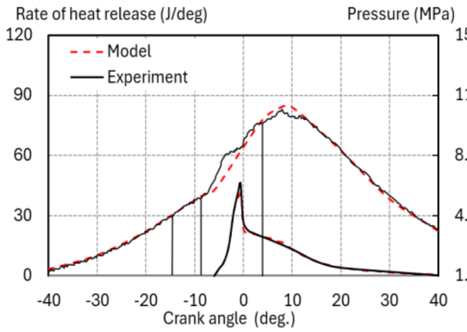


Fig. 2. Piston shape of the experimental engine (a) and after meshing in the model (b)

To evaluate the model's accuracy, several test results from the actual experimental engine were compared with those obtained from the simulation. The results are shown in Fig. 3. This is the position corresponding to the engine speed at which maximum torque is reached.

a)



b)

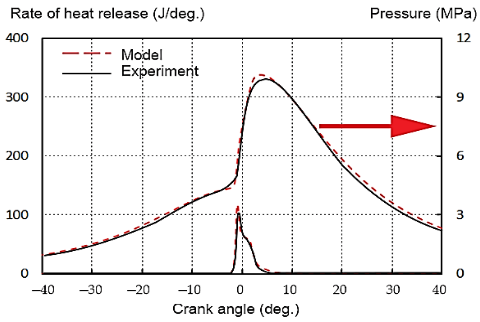


Fig. 3. Comparison of simulation and experimental results: a) engine speed at 1400 rpm, fuel injection pressure of 150 MPa, injection mass per cycle of 60 mg, intake pressure of 1 bar (absolute pressure); b) engine speed at 1400 rpm, fuel injection pressure of 300 MPa, injection mass per cycle of 60 mg, intake pressure of 1.5 bar (absolute pressure)

The results obtained from the model and the experiment are relatively close. Therefore, the established model can be used for further research to evaluate and improve the engine.

3. Results and discussion

Simulations were performed on a diesel engine with a fuel injection rate of 120 mg/cycle, a fuel injection pressure of 350 MPa, and a boost pressure of 0.25 MPa, using a 5-injection-rate shape as shown in Fig. 4. The fuel injection characteristics were selected based on a typical injection rate shape for diesel engines. In these cases, case 1 is the early-peak/front-heavy profile, with fuel distribution concentrated primarily at the start of the injection process. Next, case 2 is the rapid-decay/front-heavy with tail profile. With this injection pattern, a large amount of fuel is delivered at the beginning, the flow is reduced rapidly in the middle of the cycle, and a small, sustained injection volume is maintained at the end. Case 3 is the symmetrical/parabolic profile. The fuel is evenly distributed, with most of it concentrated in the middle of the injection process, with smaller injection volumes at the beginning and end. Case 4 is the late-peak/rear-heavy profile, in which fuel is retained and supplied primarily during the final stage of the injection stroke. Finally, case 5 is the boot-shaped profile with the spraying process divided into a low-pressure phase to control the fire delay and a high-pressure phase for the main spray. The change in fuel delivery shape, corresponding to an injection volume and an injection pressure of 350 MPa, is supported by the Diesel RK software.

Cases 1 and 2 show the change in injection speed during the latter stages of the fuel injection process. Cases 4 and 5 show the difference during the initial stages of the injection process. These spray patterns are imported into the nozzle submodel in AVL Fire.

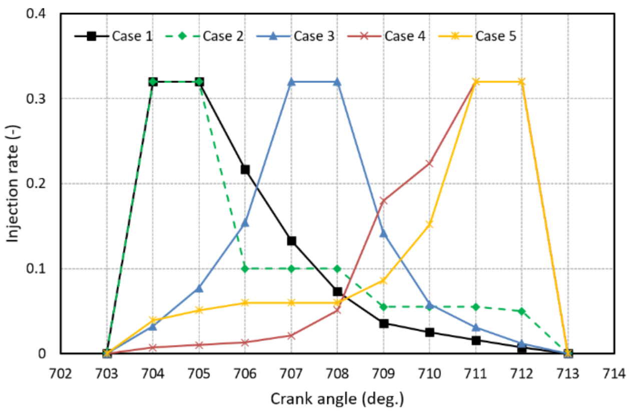


Fig. 4. Injection rate conditions for the surveyed cases

Figure 5 illustrates the engine's mean total pressure and the rate of heat release in the combustion chamber. It can be seen that in cases 1 and 2, the amount of fuel in the initial phase of the injection process is large, causing the combustion process to occur quickly.

a) b)

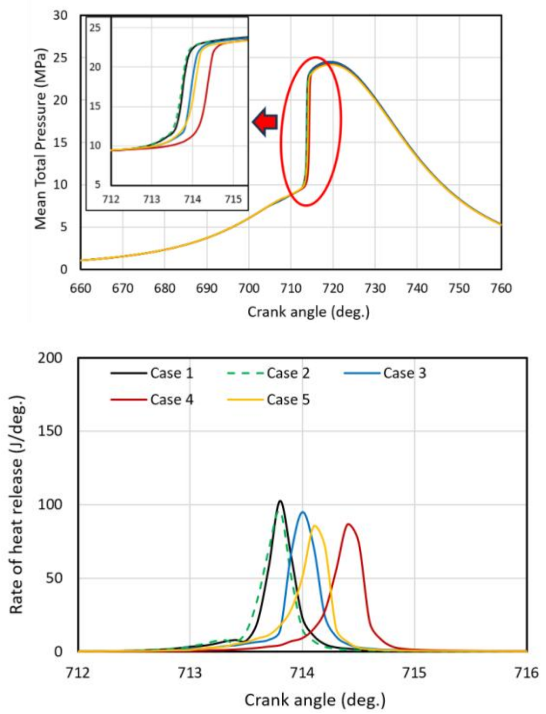


Fig. 5. Mean total pressure (a) and rate of heat release (b) in cylinder

The mean total pressure and the rate of heat release in the cylinder increase rapidly. Since the fuel injection amount is 120 mg/cycle and only the fuel injection shape changes, the maximum mean total pressure in the combustion chamber remains constant, but the slopes differ clearly. The rate of heat release in case 1 is the highest, so the engine tends to be noisier than in the other cases. In cases 4 and 5, because the initial fuel injection volume is low and gradually increases, the maximum rate of heat release is lower and the rate of heat release curve is flatter compared to cases 1, 2, and 3. From Fig. 5a, it is also evident that the rate of pressure rise for the simulated engine at an injection pressure of 350 MPa is exceptionally high. This is attributed to the simulation being conducted with a single injection per cycle. Furthermore, with an injection quantity of up to 120 mg/cycle and high injection pressure, the fuel atomizes and mixes thoroughly with air, resulting in rapid combustion. This leads to a peak pressure-rise rate of (12–18) MPa/deg. CA, exceeding the permissible limit for diesel engines. Consequently, the engine risks severe diesel knock, which imposes high thermal and mechanical stresses that could lead to structural damage. In practical applications involving modern electronic high-pressure fuel injection systems, multiple pilot injections must be implemented to maintain the rate of pressure rise below 0.4 MPa/°CA.

Figure 6 shows the indicated power and indicated fuel consumption of the engine. It can be seen that cases 1, 2, and 3 show almost constant indicated power at 23.3 kW, but the engine power decreases when fuel injection is performed according to rules 4 and 5. The power value in case 5 is 22.8 kW, a decrease of 2.15% from case 1. Similarly, the indicated fuel consumption rate changes. Cases 1, 2, and 3 show an indicated fuel consumption rate of 216 g/kWh, but in case 5, it increases to 220 g/kWh. Thus, cases 1, 2, and 3 show the highest indicated power and the lowest indicated fuel consumption.

a) b)

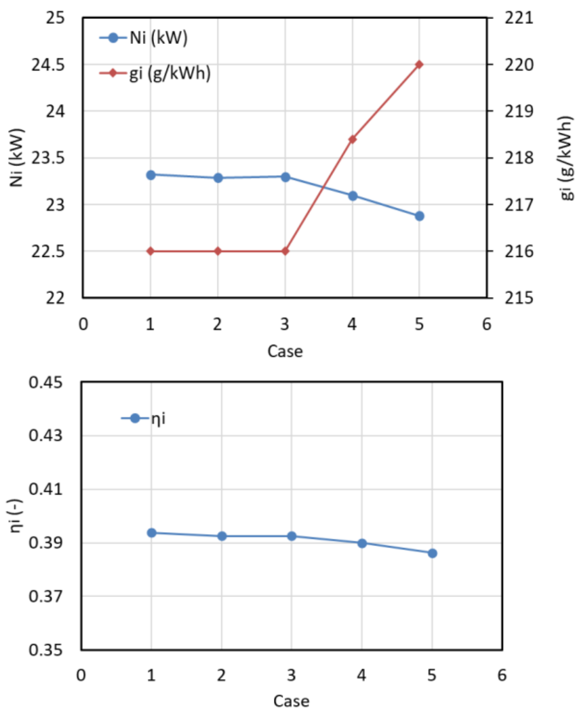


Fig. 6. Engine performance indicators: a) Indicated power (N_i) and indicated fuel consumption (g_i); b) Indicated efficiency (η_i)

In parallel with the change in indicated power, the change in the engine's indicated efficiency is shown in Fig. 6b.

The emissions from the surveyed engine correspond to 5 variations, as shown in Fig. 7. The lowest NO_x emissions were recorded in case 1. The decrease in NO_x is due to the large amount of fuel supplied during the initial phase of fuel injection. The combustion process is rapid and efficient, thus reducing the rate of NO_x formation. If the fuel injection rate increases gradually and reaches its maximum near the end (as in case 5), then the amount of NO_x formed also reaches its maximum. The value obtained is 21.01 g/kWh, an increase of almost 1.2 times compared to case 1 (Fig. 7a). In contrast to the variation in NO, the soot value is shown in Fig. 7a. The lowest soot value was obtained in case 5, reaching 0.0012 g/kWh, four times lower than in case 1. In all cases surveyed, the soot value was very low. This is explained by the fact that the fuel injection pressure of the model under study reached 350 MPa, and the injector had a nozzle diameter of 0.15 mm. The ultra-high injection pressure combined with the small nozzle diameter resulted in finely dispersed fuel, rapid combustion, and reduced soot production.

Figure 7b shows the CO and HC emission results for the surveyed cases. The amounts of CO and HC are calculated in the model run report, which shows the final CO and HC formation from the simulation cycle. High levels of CO and HC were observed in cases 1 and 3, while lower levels were observed in case 4. The high CO and HC levels in case 1 can be attributed to the large amount of fuel injected in the initial phase, which reduced the number of fuel particles in full contact with the air, thus limiting the complete oxidation to CO_2. Along with that, due to inadequate fuel mixing between zones and fuel being mixed leaner than the combustion limit. In addition, due to the high injection pressure, the spray develops very quickly (Fig. 8) and is well dispersed, causing some of the fuel to evaporate and be drawn into the dead space between the piston crown and the cylinder head (Fig. 9a). The result was an increase in CO and HC of 1.6 and 4.8 times, respectively. In case 4, because the amount of fuel injected increases gradually, the fuel particles are more readily exposed to fresh air, leading to complete fuel oxidation. As a result, the amount of CO and HC decreases.

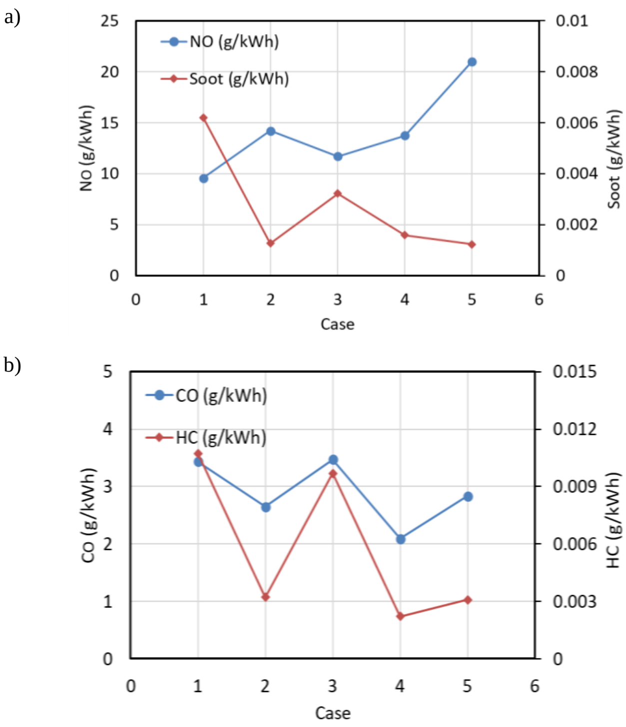


Fig. 7. Variation of the diesel emissions: a) NO and soot; b) CO and HC

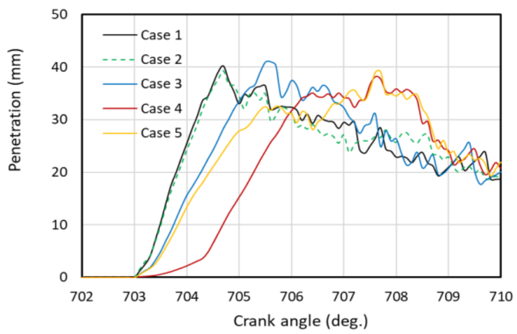


Fig. 8. Fuel penetration in the studied cases

At the end of the fuel injection process in case 1, a very small amount of fuel was injected, resulting in high CO and HC levels. By addressing fuel leakage in the final stage, HC and CO levels in case 2 decreased compared to case 1.

The development of the fuel spray as well as the fuel penetration is shown in Fig. 8 and Fig. 9. The results were obtained from the AVL Fire model run report.

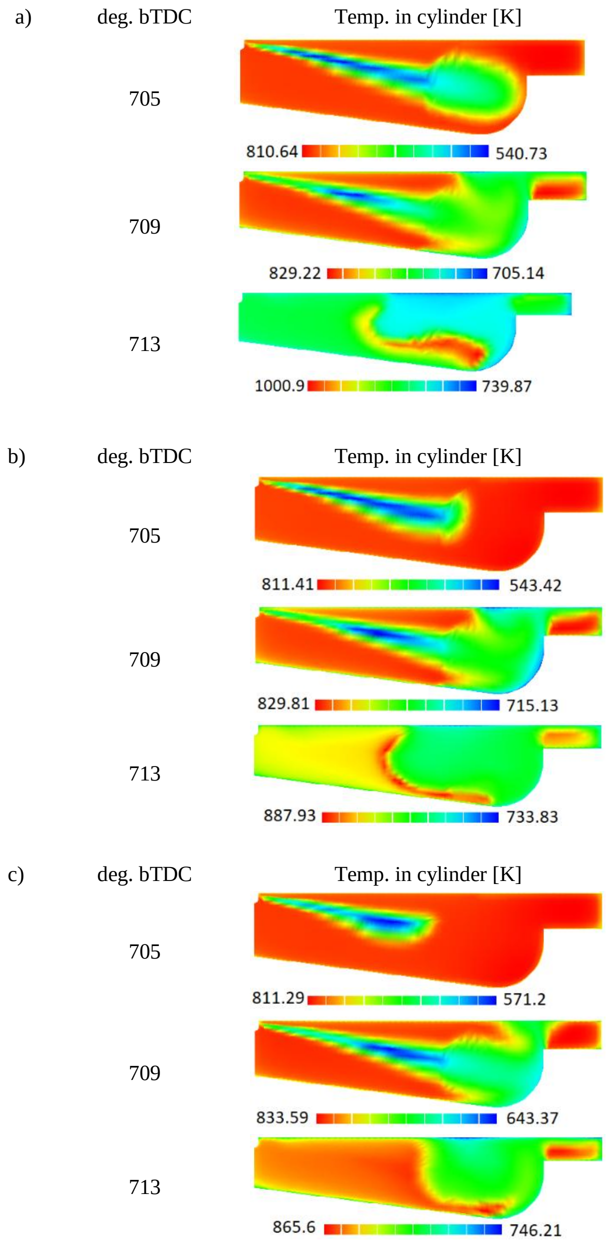


Fig. 9. Temperature field in the combustion chamber: a) case 1; b) case 3; c) case 5

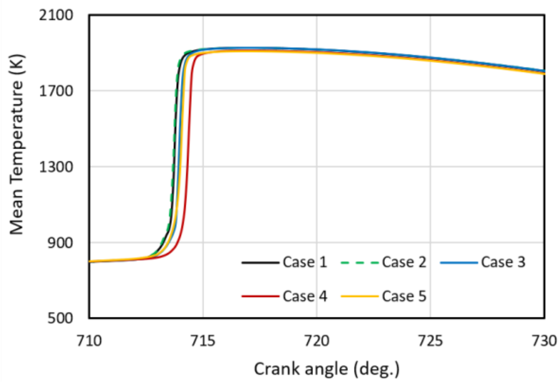


Fig. 10. Mean temperature in cylinder

In case 1, the large amount of fuel injected in the initial phase creates a cone-shaped spray (Fig. 9a), and the spray develops rapidly, causing the fuel penetration to reach its maximum quickly. Subsequently, the fuel spray rapidly contacts the combustion chamber walls and begins to burn at an angle of 713°CA bTDC. Conversely, in cases 3 and 5, because the amount of fuel injected in the initial stage is insufficient, the fuel spray develops gradually, and the fuel penetration changes accordingly. Combustion occurs later than in case 1, starting at 714°CA bTDC. Also, because the amount of fuel per cycle is constant, the injection timing is fixed, so the maximum temperature in the combustion chamber remains constant (see Fig. 10).

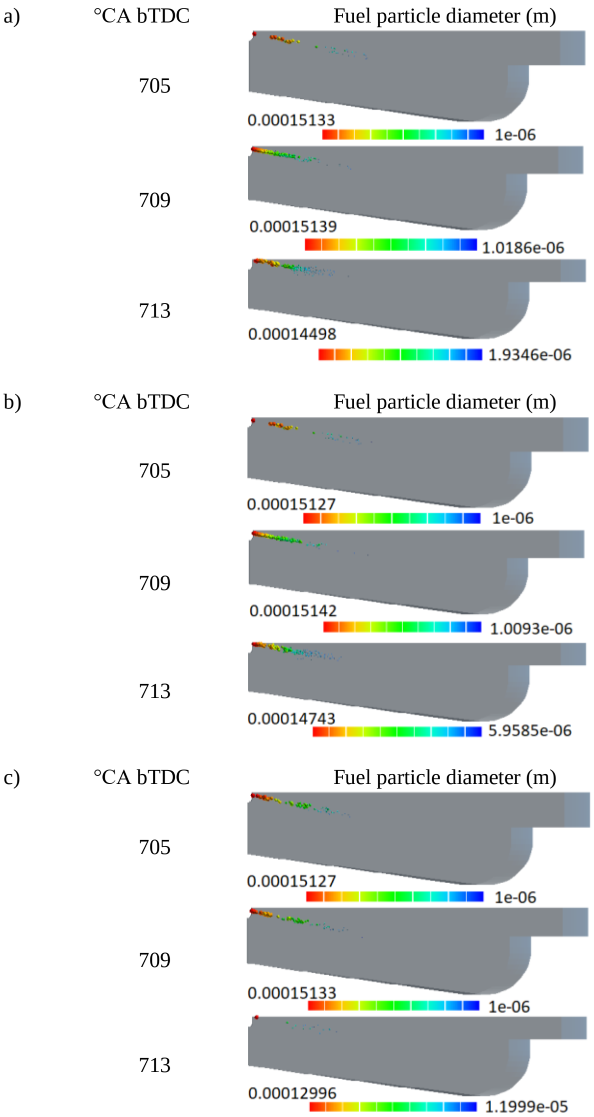


Fig. 11. Fuel particle size in combustion chamber: a) case 1; b) case 3; c) case 5

The development of the fuel spray as a function of fuel particle size is shown in Fig. 11. The results are similar to those for fuel penetration under an injection pressure of 350 MPa.

The results show that fuel injection according to rule 1 yields the best performance in indicated power, indicated fuel consumption, and NO reduction. Although the soot value is large compared to other fuel injection shapes, when combined with a small nozzle diameter and injection pressure of 350 MPa, the amount of soot formed is only 0.0062 g/kWh.

4. Conclusion

The study presented the influence of the shape of the fuel injection characteristic, combined with ultra-high injection pressure and turbocharging, on the performance and emissions of diesel engines. Within the scope of this study, the following results were achieved:

A reliable model of a turbocharged single-cylinder diesel engine was established using the AVL Fire package to evaluate operating and emission parameters during engine improvement studies.

– Conduct a survey on the changes in economic, technical, and environmental indicators of a single-cylinder diesel engine when the injection rate shape is changed. It can be seen that injecting a large amount of fuel in the initial phase resulted in a 1.9% increase in power, a 1.8% reduction in fuel consumption, and a 2.2 times reduction in NO emissions compared to the other case studied.
– Due to the large amount of fuel injected in the initial phase, the number of fuel particles in full contact with the air is decreased in the fuel spray area, and therefore, complete oxidation to CO_2 is limited. Additionally, due to inadequate fuel mixing between zones and fuel being mixed leaner than the combustion limit, CO and HC increased by factors of 1.6 and 4.8, respectively.
– Although the soot value in case 1 is large compared to other fuel injection shaping, when combined with a small nozzle diameter and an injection pressure of of 350 MPa, the amount of soot produced is only 0.0062 g/kWh. However, the engine releases heat at a high rate, making it difficult to operate if fuel distribution is implemented according to this rule.

Based on the results of this study, injection strategies are considered alongside factors such as injection pressure, boost pressure, and injector geometry to reduce emissions in diesel engines.

Nomenclature

aTDC after top dead center
bTDC before top dead center
CO carbon monoxides
CO_2 carbon dioxides
CRDI common rail diesel injection
HC hydrocarbons
NO_x nitrogen oxides

Bibliography

[1] Balz R, Bernardasci G, von Rotz B, Sedarsky D. Influence of nozzle geometry on spray and combustion characteristics related to large two-stroke engine fuel injection systems. Fuel. 2021;294:120455. https://doi.org/10.1016/j.fuel.2021.120455

[2] Boccardo G, Millo F, Piano A, Arnone L, Manelli S, Fagg S et al. Experimental investigation on a 3000 bar fuel injection system for a SCR-free non-road diesel engine. Fuel. 2019; 243:342-351. https://doi.org/10.1016/j.fuel.2019.01.122

[3] Desantes JM, Benajes J, Molina S, González CA. The modification of the fuel injection rate in heavy-duty diesel engines. Part 1: effects on engine performance and emissions. Appl Therm Eng. 2004;24(17-18):2701-2714. https://doi.org/10.1016/j.applthermaleng.2004.05.003

[4] Desantes JM, Benajes J, Molina S, González CA. The modification of the fuel injection rate in heavy-duty diesel engines: Part 2: effects on combustion. Appl Therm Eng. 2004;24(17-18):2715-2726. https://doi.org/10.1016/j.applthermaleng.2004.05.004

[5] Dimitriou P, Javaid R. A review of ammonia as a compression ignition engine fuel. Int J Hydrog Energy. 2020; 45(11):7098-7118. https://doi.org/10.1016/j.ijhydene.2019.12.209

[6] He S, Liu Y, Wang S, Hu L, Xiao F, Li C. Optimization of engine control strategies for low fuel consumption in heavy-duty commercial vehicles. CMES – Comput Model Eng Sci. 2023;137(3):2693-2714. https://doi.org/10.32604/cmes.2023.028631

[7] International Energy Agency. Global EV Outlook 2025 Expanding sales in diverse markets. 2025. https://www.iea.org/reports/global-ev-outlook-2025

[8] Ko A, Woo Y, Jang J, Jung Y, Pyo Y, Jo H et al. Complementary effects between NO oxidation of DPF and NO_2 decomposition of SCR in light-duty diesel engine. J Ind Eng Chem. 2019;80:160-170. https://doi.org/10.1016/j.jiec.2019.07.045

[9] Longwic R, Tatarynow D, Kuszneruk M, Wozniak-Borawska G. Preliminary tests of a Diesel engine powered by diesel and hydrogen. Combustion Engines. 2023;195(4): 35-39. https://doi.org/10.19206/CE-169485

[10] Millo F, Boccardo G, Piano A, Arnone L, Manelli S, Tutore G et al. Numerical simulation of the combustion process of a high EGR, high injection pressure, heavy duty diesel engine. SAE Technical Papers. 2017-24-0009. 2017. https://doi.org/10.4271/2017-24-0009

[11] Mohan B, Yang W, Yu W, Tay KL, Chou SK. Numerical investigation on the effects of injection rate shaping on combustion and emission characteristics of biodiesel fueled CI engine. Appl Energy. 2015;160:737-745. https://doi.org/10.1016/j.apenergy.2015.08.034

[12] Nguyen TQ, Dunin AY. Investigation into the impact of piston bowl size on diesel engine characteristics with changes in fuel injection pressure and boost pressure. Appl Sci. 2024;14(10):4334. https://doi.org/10.3390/app14104334

[13] Nguyen TQ, Dunin AY, Shatrov MG. An experimental approach and a signal processing method with the common rail injection system of a diesel engine. Int J Online Biomed Eng. 2021;17(14):19-31. https://doi.org/10.3991/IJOE.V17I14.27401

[14] Nguyen Thin Q, Le Hoai D, Nguyen Cao V. Experimentally determine the effect of pressure on injection mass and development of the diesel fuel spray. Transp Commun Sci J. 2024;75(6):1934-1947. https://doi.org/10.47869/tcsj.75.6.1

[15] Niculae AL, Chiriac R, Racovitza A. Effects of injection rate shape on performance and emissions of a diesel engine fuelled by diesel and biodiesel B20. Appl Sci. 2022;12(3): 1333. https://doi.org/10.3390/app12031333

[16] Palanisamy M, Lorch J, Truemner RJ, Baldwin B. Combustion characteristics of a 3000 bar diesel fuel system on a single cylinder research engine. SAE Int J Commer Veh. 2015; 8(2):479-490. https://doi.org/10.4271/2015-01-2798

[17] Payri R, Martí-Aldavarí P, Montiel T, Viera A. Influence of aging of a diesel injector on multiple injection strategies. Appl Therm Eng. 2020;181:115891. https://doi.org/10.1016/j.applthermaleng.2020.115891

[18] Porras Perucho HA, Moreira CM V, Boruc Ł, Grochowalska J, Kapusta ŁJ. Spray structures formed by a multi-nozzle injector during the injection of a multi-component surrogate synthetic fuel under flash-boiling conditions. Combustion Engines. 2026;205(2):170-176. https://doi.org/10.19206/CE-214424.

[19] Seebode J. Injection strategies under the influence of pressure modulation and free rate shaping in modern DI-diesel engines. CIMAC Congr. 2004.

[20] Soudagar MEM, Shelare S, Marghade D, Belkhode P, Nur-E-Alam M, Kiong TS et al. Optimizing IC engine efficiency: A comprehensive review on biodiesel, nanofluid, and the role of artificial intelligence and machine learning. Energy Convers Manag. 2024;307:118337. https://doi.org/10.1016/j.enconman.2024.118337

[21] Sun X, Jiang YC, Zhao P, Jing G, Ma T. Effect of ammonia/hydrogen blending and injection modes on combustion emission and performance of marine engine. Fuel. 2024; 371:131894. https://doi.org/10.1016/j.fuel.2024.131894

[22] Vera-Tudela W, Haefeli R, Barro C, Schneider B, Boulouchos K. An experimental study of a very high-pressure diesel injector (up to 5000 bar) by means of optical diagnostics. Fuel. 2020;275:117933. https://doi.org/10.1016/j.fuel.2020.117933

[23] Wang B, Yang C, Wang H, Hu D, Duan B, Wang Y. Study on injection strategy of ammonia/hydrogen dual fuel engine under different compression ratios. Fuel. 2023;334:126666. https://doi.org/10.1016/j.fuel.2022.126666

[24] Wloka JA, Pflaum S, Wachtmeister G. Potential and Challenges of a 3000 bar common- rail injection system considering engine behavior and emission level. SAE Int J Engines. 2018;3(1):801-813. https://doi.org/10.4271/2010-01-1131

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