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Investigations into High-Pressure Diesel Spray-Wall Interaction on Reduction of Exhaust Emission from DI Diesel Engine

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Abstract

The current paper is a continuation of research on fuel atomization presented in SAE 2012-01-1662. The influence of varied position of the injector inside the combustion chamber on combustion, toxic compounds formation and exhaust emission were investigated. The simulation research (injection and combustion with NO formation) was supported with the model using the FIRE 2010 software by AVL. Modelling studies of toxic compounds formation were compared with the results of measurements on single-cylinder AVL 5804 engine. There thermodynamic evaluation indicators and exhaust emission were made.

Introduction

The tightening of the exhaust emissions limits forces the engineers to seek new solutions in combustion or improve the existing designs. Such attempts are related to motor vehicles [13, 15], special vehicles [14, 20] or the engines alone. Currently, engineers investigate the possibilities of optimization of the combustion systems [11, 12, 23] of both gasoline and diesel engines. There are a variety of works treating on new combustion systems [8, 19]. These works are however, still in the phase of development. Much attention is devoted to combustion systems, particularly in the aspect of the fuel spray interaction with the walls of the combustion chamber in a diesel engine [1, 9, 16, 17]. In gasoline engines the research is conducted mainly on the fuel injection and atomization in a spray-guided system [21, 24, 25]. The majority of investigations is related to the exhaust emissions generated in the combustion chamber. The authors [18] performed research on the identification of important mechanisms affecting soot formation after the time of wall interaction. It is expected to be useful for understanding these processes in more complex and realistic diesel engine geometries. The results of this work suggest that soot formation is lower for a plane wall jet compared to a free jet. Two possible explanations for this reduction include: (1) an increase in mixing with ambient air caused by wall impingement, which enhances soot oxidation and reduces soot formation. (2) Thermal interaction that cools the jet, slowing the rate of soot formation. It was also observed that although the wall-jet geometries are highly simplified, the basic processes governing the soot production when wall impingement is present are expected to be of significant interest to the diesel engine community.

Investigation of the influence the effect of spray targeting on exhaust emissions, especially soot and carbon monoxide (CO) formation, were made by Lee and Reitz in a single-cylinder, diesel engine and presented in [9]. In this research for PCCI system was found, that require different spray targeting on the piston to reduce CO and soot emission. In the paper [5] authors investigated the influence of bowl geometry, spray targeting and swirl ratio on pre-and post-combustion mixing processes under a low-temperature diesel combustion regime. The research was conducted only computationally. Only simplify piston bowl geometry were used for this study. An optimal targeting point was found to exist along the bowl edge. In the paper [4] three common spray-targeting strategies are examined: conventional piston-bowl-wall targeting; narrow-angle floor targeting; and wide-angle piston-bowl-lip targeting included angle. For this three strategies authors using injectors with different spray-targeting. Spray-wall and flame-wall interactions were investigated [7] in a combustion chamber with diesel engine temperature and pressure conditions. The authors concluded that all investigated variations: different fuels, gas and wall temperatures have a significant influence on the spray-wall and the flame-wall interactions respectively.

In this work authors investigated the influence of varied position of the injector inside the combustion chamber on combustion process and emission.

Results of the Model Research

The evaluation of the interaction of the fuel spray and piston was performed in the aspect of different piston positions against the injector, which resulted in the fuel spray propagation onto different areas of the combustion chamber.

Based on the experimental research presented in SAE 2012-01-1662 [26] the authors have drawn the following conclusions:

The conditions of the fuel injection determined by the fuel pressure and the air backpressure do not significantly influence the area of the combustion chamber covered by the fuel spray (assuming constancy of the start of the injection - in the research it meant a constant position of the piston against the injector). The changes in the size of the areas covered by the fuel spray are within the range of 10%.

The change in the position of the piston (the change of the start of the injection against the current piston position) results in great differences in the areas of the combustion chamber covered by the fuel spray. The changes are from 60 to 30% in the case of the area above the piston (reduction of the value when the piston is higher). The second area covers the values from 15% to 30% depending on the position of the piston (increase in the value when the piston is higher). This share is the lowest of all the three analyzed areas. The share of the third area of the chamber inside the piston that the fuel spray reaches changes from 30 to 60% (as the position of the piston gets higher, the values grow).

Taking the above conclusions into account the author observed that different injector positions may significantly modify the fuel atomization and, as a consequence, its combustion. For this reason in the experimental research the author focused on the analysis of the fast-varying processes and exhaust emissions tests.

Methodology of the Experimental Research

The model research related to the fuel injection and atomization was conducted in a constant volume chamber. Full investigations of these phenomena and the interaction of the fuel with the piston walls have been presented in [26]. Current investigations allowed the assessment of the influence of the injector position and fuel atomization on the fuel combustion and exhaust emissions. The investigations were conducted in two stages. The first stage was experimental research on a diesel single cylinder research engine (Fig. 1). These investigations were validated using computer simulation in FIRE by AVL. To this end, a simulation research was carried out of the processes of fuel injection and atomization. The conclusion of these works was the determination of the influence of the position of the injector on the formation of exhaust gas components inside the cylinder. The simulation research allowed an evaluation of these processes and combustion indexes that were impossible to evaluate in the experimental research.

The experimental research was carried out for an example engine work point with a varied position of the injector inside the combustion chamber. The technical specifications of the engine and its operating conditions have been shown in Table 1. From the data it results that the engine is representative of a 4-cylinder system of the capacity of approx. 2 dm^3 - an engine system widely applied in passenger vehicles. For this reason the presented investigations partially should be of universal nature and at the same time represent the mixture formation processes of a wider population of diesel engines. In modern passenger vehicle engines we may encounter flatter combustion chambers, yet, in MDV (medium duty vehicle) vehicles the presented chambers still have a significant share in direct injection engines.

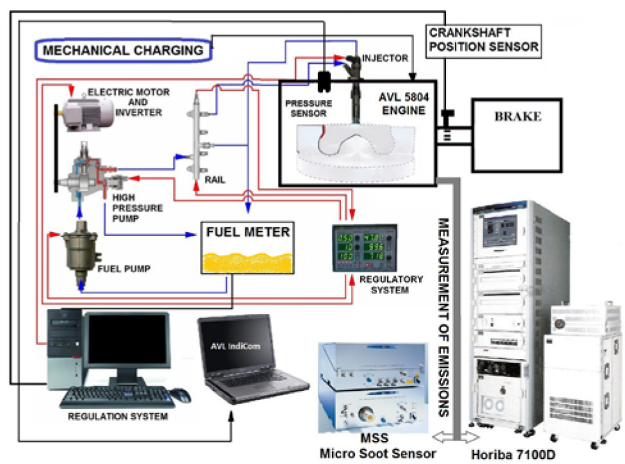


Fig. 1. Test stand.

Cylinder pressure was measured with a AVL GH11D piezo-electric pressure transducer and the measured signal was amplified with a AVL IFEM charge amplifier. The output signal from the charge amplifier was sampled with a AVL IndiSmart 621 data acquisition board (resolution of crank angle degree was 0.1).

For the exhaust emission tests the following were used:

a). Exhaust emission analyzer Horiba 7100D by AVL (Tab. 2);

b). Particulate matter concentration measurement device fitted with a sample conditioning system - exhaust dilution (Micro Soot Sensor by AVL - Tab. 3).

Tab. 1. Technical specifications of the AVL 5804 engine and its operating conditions during the experimental research.

Displaced volume	510.7 cm^3		
Stroke	90 mm		
Bore	85 mm		
Compression ratio	16:1		
Number of valves	4		
Exhaust valve open	57.5° BBDC		
Exhaust valve close	18° ATDC		
Inlet valve open	10° BTDC		
Inlet valve close	468° ABDC		
Type of injector	electromagnetic, 6-hole, d = 0.17 mm, α = 166°		
Injection system	Bosch common rail system		
Top clearance	0.8 mm		
Head gasket	2 mm (need to obtain an appropriate compression ratio)		
Engine work conditions (experimental tasks)			
Speed	1500 rpm		
Injector position	1	2	3
Distance between end of injector and engine head	2 mm (5 mm from the piston molding)	5 mm (3 mm from the piston molding)	8 mm (1 mm from the piston molding)
Torque	5 Nm	6 Nm	5 Nm

Tab. 2. Technical specifications of the Horiba 7100D analyzer.

Parameter	Measurement method	Accuracy
Emission		
CO	NDIR; range 0–10%	±0.5% of range
THC	FID; range 0–50,000 ppm	±0.5% of range
NOx	CLA; range 0–10000 ppm	±0.5% of range
CO_2	NDIR; range 0–10%	±0.5% of range
O_2	MPA; range 0–25%	±0.5% of range

Tab 3. Specification of analyzer Micro Soot Sensor by AVL.

Parameter	Value
Measured value	Concentration of soot (mg/m³, μg/m³)
Measuring range	0.005–50 mg/m³
Detection limit	~ 5 μg/m³
Dilution ratio (DR)	Adjustable from 2–10 and from 10–20
Exhaust gas temperature	Up to 1000°C

The simulation research was conducted using the FIRE software where a model of combustion chamber was implemented, identical to that of the AVL 5804 engine (Fig. 2). The experimental combustion chamber was selected because of its identical shape in the research model and the actual research engine.

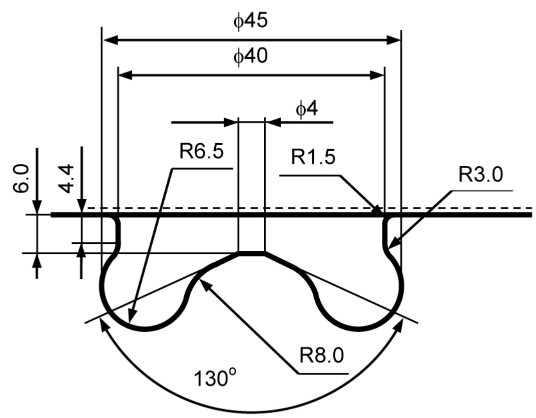


Fig. 2. Shape and size of the research combustion chamber used in the simulation.

Analysis of the Simulation Research

Simulation research was conducted for the same injector positions as in the model research. They were marked as Position 1, Position 2 and Position 3. Thermodynamic analysis of the system inside the cylinder was performed and a simulation of combustion carried out analyzing the exhaust emissions. The injection and combustion analyses were realized in the range from 180 to 540°CA. with the resolution of 1°CA. In the interval when intense processes inside the cylinder occur i.e. from 350 to 400°C.A. the resolution was increased to 0.1°CA. The cylinder and the combustion chamber models used have been shown in Fig. 2. These models were in line with the experimental research.

The simulation research was conducted at the engine speed n = 1500 rpm. The injection duration was set at t_{inj} = 0.3 ms - the same as in the experimental research (fuel injection dose was q_o = 7.7 mg). The model mesh contained 23,600 cells - this enabled a detailed reproduction of the combustion chamber.

In the research the authors used:

a). the k-zeta-f model of turbulence; this model was developed by Hanjalic, Popovac and Hadziabdic [6]; the advantages of this model were important in the liquid region, where the wall effects were very strong
b). the Dukowicz model of droplet evaporation
c). the Wave model of droplet disintegration
d). Naber and Reitz model of spray-wall impingement
e). Eddy break-up model of combustion; eddy-break-up type model (in combination with a suitable auto-ignition model) is applicable to conventional diesel combustion [2, 3, 22]
f). NO_x model: Extended Zeldovich

The thermodynamic analysis performed during the model research indicates differences in the tracings of the combustion pressure in its part after self-ignition (Fig. 3). For all the cases the self-ignition delay is constant, no differences were observed of the start of the combustion. This course indicates that the disintegration of the fuel spray on the edge of the combustion chamber (Position 1) leads only to a faster increment of the pressure after the self-ignition. The self-ignition delay is approx. 5°CA (start of combustion is 356 CA, start of injection 352 CA, i.e. four degrees of self-ignition delay). This means that the disintegration of the fuel spray on the edge of the piston leads to a much better mixture formation, thus allowing an obtainment of greater indexes of this process.

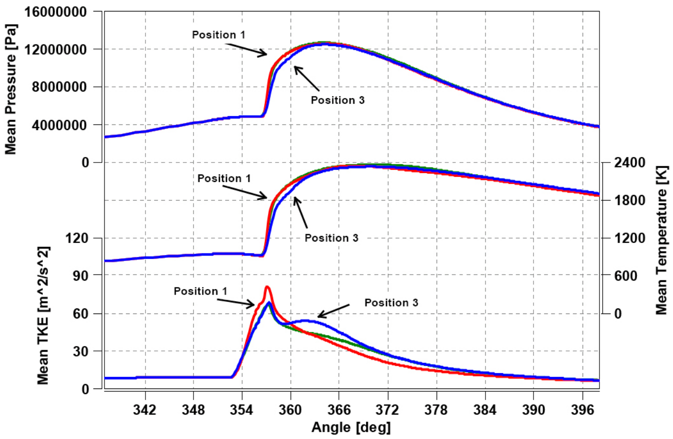


Fig. 3. Thermodynamic combustion indexes determined during the fuel injection simulation research (simulation AVL FIRE).

The average temperature inside the cylinder depends on the moment of the start of combustion, yet after 366°CA it assumes identical values. Assuming that the formation of NO occurs in the initial part of the combustion process, we should expect high values of this exhaust component at the piston position (Position 1). The advantage of the highest position of the of the injector is high value of the turbulence kinetic energy (TKE) that is 10% higher than it is in the case of Position 2 and Position 3. Increasing the turbulence kinetic energy in the second (diffusive) part of the combustion is characteristic of the third position of the injector (Position 3). This means that in this case an additional turbulent motion is possible. Yet, the effects of this phenomenon are not observed on the tracings of the thermodynamic quantities.

Further investigations were related to the pre-flame period-atomization and evaporation of fuel. The amount of the evaporated fuel is almost constant (Fig. 4 - upper data). Position 3 indicates the worst fuel evaporation, which results from the fact that fuel reaches the wall of the combustion chamber in the piston. In Position 1 we may observe the lowest fuels pray penetration, yet the differences are not significant. Changes in the shares of the mass of the combusted fuel as a function of crank angle should be deemed significant in the initial phase of the combustion. The highest combustion rate is obtained when the fuel is injected via the injector in Position 1. The total share of the combusted fuel is the same for all three analyzed cases.

The analysis of the mean heat transfer coefficient - HTC indicates its highest values during the injection and combustion when the injector is in Position 1. This is a consequence of a large amount of fuel reaching the area above the piston. The heat release coefficient assumes values of approx. 7500 W/m^{2}K. For other injector positions the maximum values of this index amount to approx. 4500 W/m^{2}K. The amount of combusted fuel shown in Fig. 4 corresponds to the rate of heat release - Fig. 5. The rate is the greatest for the highest position of the injector (Position 1). A similar rate of heat release was observed during a simulation of the combustion with the injector in Position 2. The analysis of the accumulated value of the rate of heat release does not show significant differences in the value from Position 1 and Position 2 of the injector. Consequently, the formation of nitric oxides will be the lowest for Position 3 of the injector.

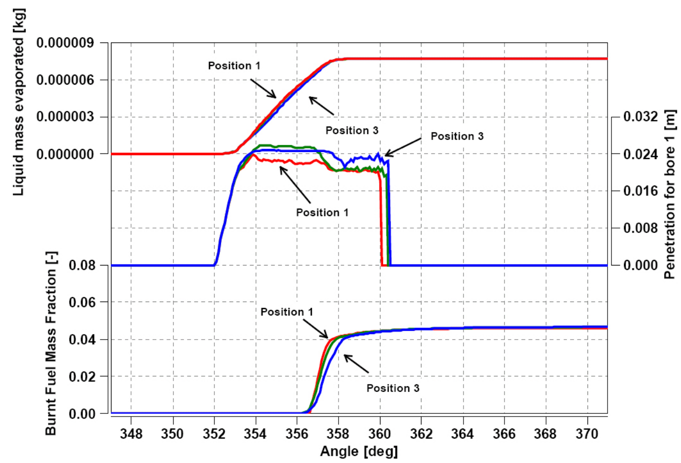


Fig. 4. Fuel spray indexes (evaporated mass and penetration) and the share of the mass of the combusted fuel for three injector positions (simulation AVL FIRE).

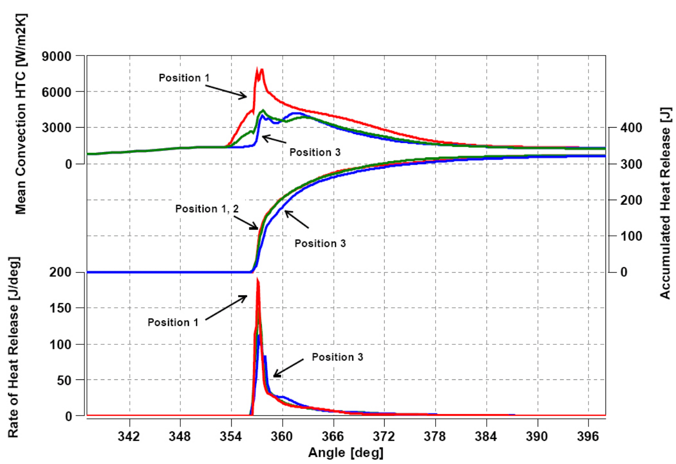


Fig. 5. Heat convection towards the cylinder head walls, rate of heat release and accumulated heat release during simulation research (simulation AVL FIRE).

The most intense combustion process occurs for Position 1 of the injector (Fig. 6). A high initial share of carbon dioxide content confirms this. For Position 3 the rate of carbon dioxide formation is the lowest. This indicates deterioration in the process of fuel oxidation. The temperature inside the combustion chamber presented before is not directly responsible for the NOx formation. It is the temperature of the flame that is accountable for that, yet, high values of average temperatures correlate with intense NO production (Fig. 6). For the combustion with the injector in Position 1 the share of NO is 50% greater than it is in the case of the combustion with the injector in Position 3. The initial changes of these shares are also greater, which confirms great rate of formation of this exhaust component. Similarly, a great share of the OH group for combustion of large amounts of fuel injected above the piston (Position 1) may indicate high temperatures of the flame and the formation of the OH radicals.

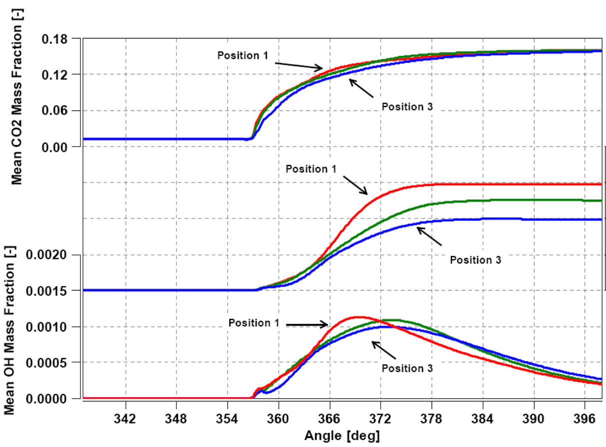


Fig. 6. In-cylinder emissions determined based on the simulation research (simulation AVL FIRE).

The above-indicated phenomena influence the formation of nitric oxides. The example observations confirm advantageous fuel atomization and high NO production (Fig. 7).

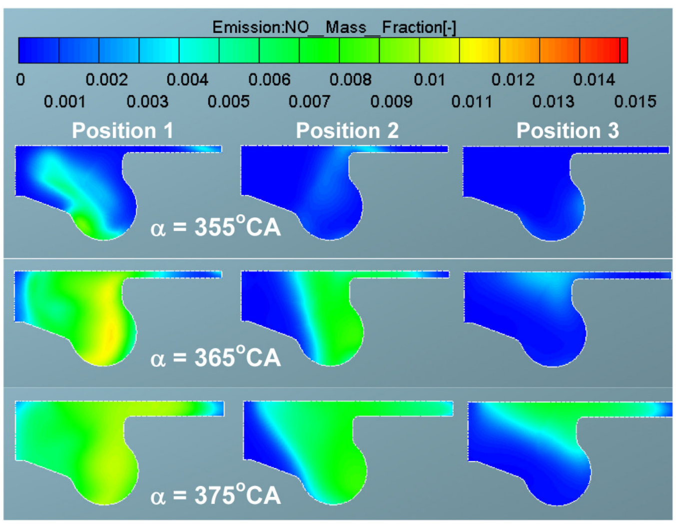


Fig. 7. The sequence of NO formation for different injector positions (simulation AVL FIRE).

The presented simulation research was confronted with the experimental one. On this basis the author ascertained the compatibility of the models contained in the simulation research with the actual course of the combustion process. The author also answered the question whether simulation research confirms the experiments. The above will potentially enable further model works in this matter.

Analysis of the Experimental Research on the Single Cylinder Engine

The experimental research was conducted according to Table 1. The aim was an analysis at a selected engine work point (single cylinder AVL 5804 engine). The selection was made in the medium range of engine operation with a small load. The selection of this point was made based on typical engine operating conditions, where the emission of NO_x is not at its peak and when the fuel dose is injected in the vicinity of the upper edge of the piston. An additional obstacle was the fact that the injected fuel dose was undivided. This results from the fact that the author determined only the influence of a single fuel dose on the fuel spray development and its further combustion.

The analysis of the fast-varying processes pertained mainly to the pressure inside the cylinder of a combustion engine (Fig. 8). Varied injector locations result in different maximum combustion pressures. In the case of an injection onto the edge of the piston (Position 1) and on the vertical edge of the piston, similar $P_{cyl\text{-}max}$ were obtained. The way in which it was obtained, however, is different. In the first case the ignition delay is the smallest. As the injector position lowers the self-ignition delay increases. It amounts to: 4.7; 5.5 and 6.4°CA respectively (self-ignition delay was determined as difference between start of combustion and start of injection - not presented in figures). The rate of heat release is the same for the first two positions of the injector. The only difference is a delayed start of the combustion for Position 2. The combustion with the injector in Position 3 resulted in a much lower rate of heat release. Similar results were obtained during the simulation research (see Fig. 5). This confirms the compliance of the simulation results with the experimental ones. The simulation studies and the engine investigation gave different self-ignition delay and the related start of heat release rate. This demonstrates the need to adapt the simulation model.

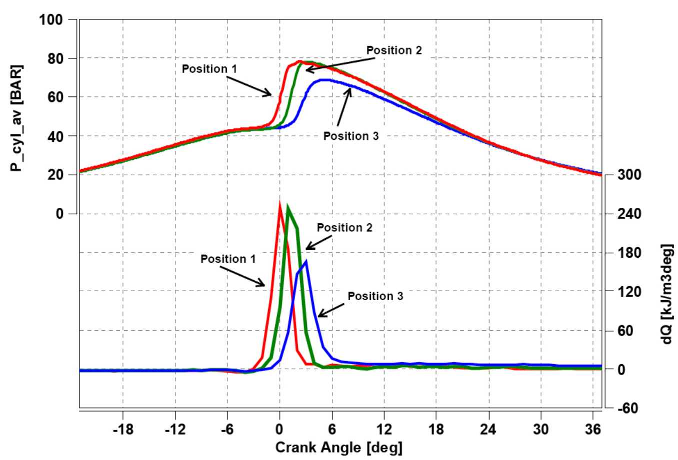


Fig. 8. The tracing of the cylinder pressure and the rate of heat release determined on this basis (AVL 5804 engine).

The accumulated rate of heat release indicates a changing trend in the total released heat under different injector positions. In the case of Position 1 faster combustion was obtained and the intensity of combustion was similar to that of Position 2 of the injector (Fig. 9). Yet, after approx. 2°CA, past the TDC the total heat released is greater than that of Position 2 of the injector. We must note the lower amount of release heat during combustion in Position 3 of the injector. After approx. 30°CA, past the TDC an equalization of the total released heat takes place. This means that despite different courses of the initial phase of combustion similar amounts of heat are obtained, thus we have a similar value of the engine indicated efficiency.

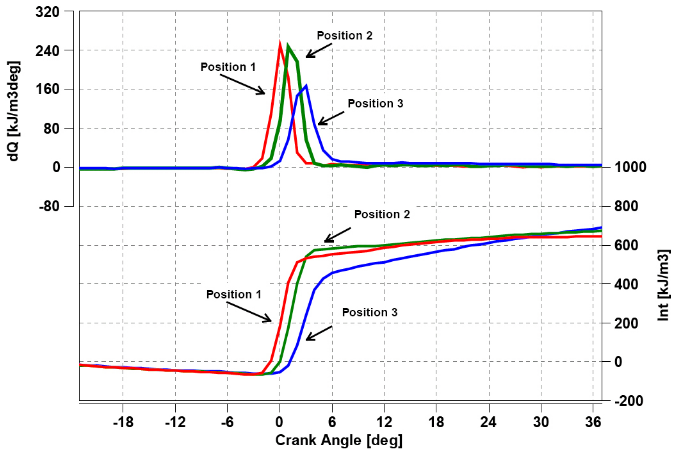
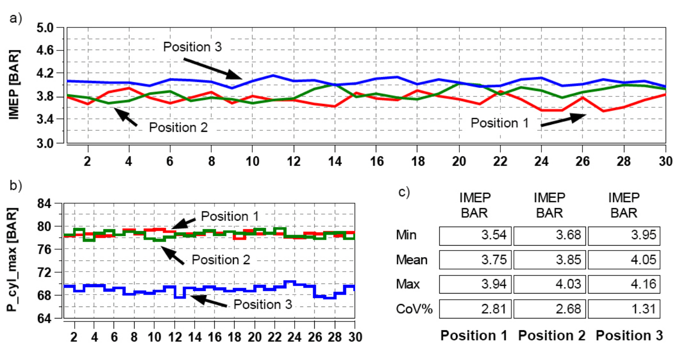


Fig. 9. The rate of heat release and accumulated value of the released heat for different injector positions (AVL 5804 engine).

Despite delayed combustion in the case of injector Position 3 the IMEP values are the highest (Fig. 10). These values result from a higher losses pumping in Position 1 (Fig. 11). Despite the highest values of $P_{cyl-max}$ for Position 1 of the injector in Position 3 we achieve the smallest CoV (Coefficient of Variation) from the mean indicated pressure. The value of CoV is in this case 1.31% for the sample of 30 test cycles.



	IMEP BAR	IMEP BAR	IMEP BAR
Min	3.54	3.68	3.95
Mean	3.75	3.85	4.05
Max	3.94	4.03	4.16
CoV%	2.81	2.68	1.31
	Position 1	Position 2	Position 3

Fig. 10. The analysis of the indexes for different injector positions (AVL 5804 engine).

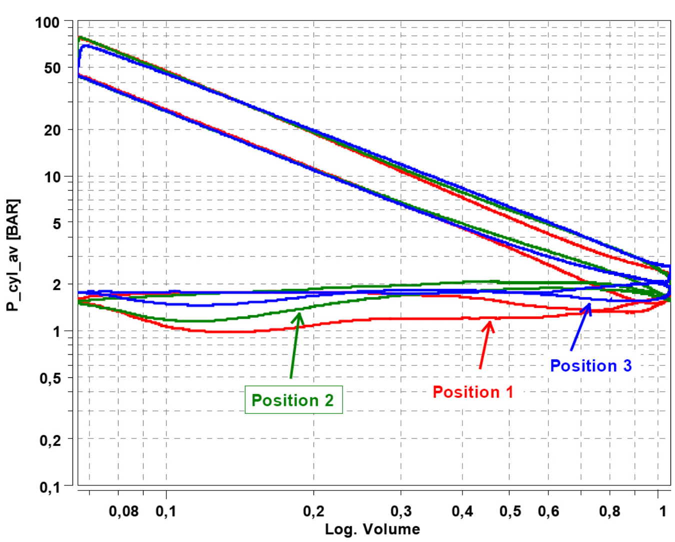


Fig. 11. The samples of the cylinder pressure (to determine the IMEP value) for different injector positions (AVL 5804 engine).

The discussed combustion methods lead to the obtainment of different values of the exhaust components concentration. For the first position of the injector the smallest values of CO and HC were obtained. Low values were also obtained for soot (Fig. 12). However, the values of the latter were not minimal. High concentrations of NO_x were expected and such were obtained. The obtained values of the exhaust component concentrations are predictable based on the optical research and simulations of injections of diesel fuel onto the piston walls. High soot values during the injection when the injector is in Position 3 indicate that this kind of fuel atomization is undesired.

Additionally, considering the identical consumption and the obtainment of the IMEP results allowed determining of the unit exhaust emissions. Their nature has been shown in Fig. 13. From the figure results that the application of Positions 1 reduces the emission of CO by more than 50% as compared to Position 3. The reduction of the emission of HC is also possible by over 70% between positions 1 and 3. The emission of NO_x in the case of Position 1 of the injector is the highest. For Position 2 it is lower by 16% and for Position 3 by approx. 68%. The reduction of the emission of soot is significant. In the first two positions of the injector the reduction of soot is on the level of 81% and 94% respectively.

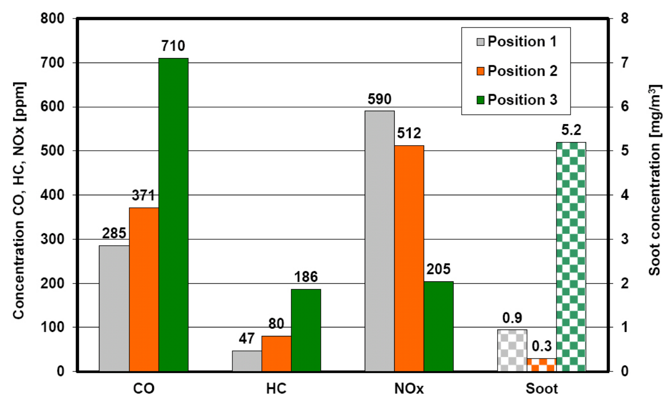


Fig. 12. Exhaust component concentration during the tests on the AVL 5804 engine (n = 1500 rpm, part load Mo = 5 Nm; q_o = 7.7 mg, P_{inj} = 100 MPa).

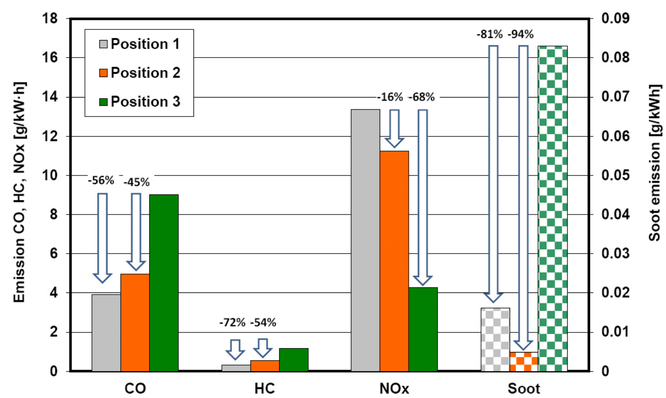


Fig. 13. Specific exhaust emissions for different injector positions (AVL 5804 engine).

Summary

The experimental and simulation research on the fuel atomization inside the combustion chamber allowing for the interactions of the fuel spray with the combustion chamber walls indicate a significance of the location of the fuel injection. This is of particular importance in terms of thermodynamic analysis of the engine operation and exhaust emissions. The conducted simulation research indicates the following:

Position of the injector in the combustion chamber is as important as the chamber shape (not recognized in this work). Location injector affects the atomization of fuel and the consequent effects of combustion.

The most intense combustion process takes place when the injector is in Position 1. High values of the combustion temperature, rate of heat release and a high share of NO_x in the combustion products confirm this.

Thermodynamically, the least proper is the combustion when the injector is in Position 3 where the worst combustion indexes are obtained.

The experimental research allowed determining of the following regularities:

1. Fuel injection (spray targeting) on a vertical wall of the combustion chamber (position 2) is a compromise due to the thermodynamic indicators and emission of harmful compounds. This is confirmed by both simulation studies and on the engine.
2. For Position 1 the highest indexes of maximum pressure inside the cylinder and the highest rates of heat release were achieved.
3. In Position 1 the authors observed the lowest unit emission of CO, HC, except NO_x, the emission of which was maximum.
4. Simulation research partially confirmed the experimental research and the applied models properly describe the phenomena occurring during the injection of fuel. The combustion model is not fitted to engine because ignition delay is not depending on the spray targeting. In this case, the optical studies are needed to clarify the simultaneous reduction of NO_x and soot emissions.

References

1. Andreassi, L., Ubertini, S. and Allocca, L., "Experimental and Numerical Analysis of High Pressure Diesel Spray-Wall Interaction," *International Journal of Multiphase Flow* 33 (7):742-765, 2007, doi:10.1016/j.ijmultiphaseflow.2007.01.003.
2. Cipolla, G., Vassallo, A., Catania, A., Spessa, E. et al., "Combined application of CFD modeling and pressure-based combustion diagnostics for the development of a low compression ratio high-performance diesel engine," SAE Technical Paper 2007-24-0034, 2007, doi:10.4271/2007-24-0034.
3. Dahlén, L. and Larsson, A., "CFD Studies of Combustion and In-Cylinder Soot Trends in a DI Diesel Engine - Comparison to Direct Photography Studies," SAE Technical Paper 2000-01-1889, 2000, doi:10.4271/2000-01-1889.
4. Genzale, C., Reitz, R., and Musculus, M., "Optical Diagnostics and Multi-Dimensional Modeling of Spray Targeting Effects in Late-Injection Low-Temperature Diesel Combustion," *SAE Int. J. Engines* 2(2):150-172, 2009, doi:10.4271/2009-01-2699.
5. Genzale, C., Reitz, R., and Wickman, D., "A Computational Investigation into the Effects of Spray Targeting, Bowl Geometry and Swirl Ratio for Low-Temperature Combustion in a Heavy-Duty Diesel Engine," SAE Technical Paper 2007-01-0119, 2007, doi:10.4271/2007-01-0119.
6. Hanjalić, K., Popovac, M. and Hadžiabdić, M., "A Robust Near-Wall Elliptic-Relaxation Eddy-Viscosity Turbulence Model for CFD," Int. J. Heat Fluid Flow 25: 1047-1051, 2004, doi:10.1016/j.ijheatfluidflow.2004.07.005.
7. Held, F., Bornschlegel, S., Riess, S., Vogel, T. et al., "Time and Spatially Resolved Measurements of the Interaction between Liquid and Combusting Diesel Spray and Walls in Modern Diesel Engine Conditions," SAE Technical Paper 2013-24-0063, 2013, doi:10.4271/2013-24-0063.
8. Hofbauer, P. and Brehob, D., "Piston Boxer Engine for Trucks," MTZ 4 2102 316-323, doi:10.1007/s35146-012-0437-8.
9. Joelsson T., "PDPA Measurements of Diesel Fuel Droplets Impinging on a Temperature Controlled Wall," Master's Thesis, Chalmers University of Technology, Gothenburg, 2007.
10. Lee, S. and Reitz, R., "Spray Targeting to Minimize Soot and CO Formation in Premixed Charge Compression Ignition (PCCI) Combustion with a HSDI Diesel Engine," SAE Technical Paper 2006-01-0918, 2006, doi:10.4271/2006-01-0918.
11. Liu, Y. and Reitz, R., "Optimizing HSDI Diesel Combustion and Emissions Using Multiple Injection Strategies," SAE Technical Paper 2005-01-0212, 2005, doi:10.4271/2005-01-0212.
12. Lucchini, T., D'Errico, G., Ettorre, D., Brusiani, F. et al., "Experimental and Numerical Investigation of High-Pressure Diesel Sprays with Multiple Injections at Engine Conditions," SAE Technical Paper 2010-01-0179, 2010, doi:10.4271/2010-01-0179.
13. Merkisz, J., Pielecha, I., Pielecha, J. and Brudnicki, K., "On-Road Exhaust Emissions from Passenger Cars Fitted with a Start-Stop System," *Archives of Transport* 23(1): 37-46, 2011, doi:10.2478/v10174-011-0003-z.
14. Merkisz, J., Pielecha, I., Pielecha, J. and Szukalski, M., "The Analysis of the Exhaust Emission Level in Combat Vehicles Under Real Operating Conditions," SAE Technical Paper 2011-26-0041, 2011, doi:10.4271/2011-26-0041.
15. Merkisz, J., Pielecha, J. and Pielecha, I., "Road Test Emissions Using On-board Measuring Method for Light Duty Diesel Vehicles," *Jordan Journal of Mechanical and*

Industrial Engineering, Vol. 5, No. 1, February 2011, ISSN 1995-6665, s. 89-96.

16. Montorsi, L., Magnusson, A. and Andersson, S., "A Numerical and Experimental Study of Diesel Fuel Sprays Impinging on a Temperature Controlled Wall," SAE Technical Paper 2006-01-3333, 2006, doi:10.4271/2006-01-3333.
17. Montorsi, L., Magnusson, A., Andersson, S., and Jedrzejowski, S., "Numerical and Experimental Analysis of the Wall Film Thickness for Diesel Fuel Sprays Impinging on a Temperature-Controlled Wall," SAE Technical Paper 2007-01-0486, 2007, doi:10.4271/2007-01-0486.
18. Pickett, L. and López, J., "Jet-Wall Interaction Effects on Diesel Combustion and Soot Formation," SAE Technical Paper 2005-01-0921, 2005, doi:10.4271/2005-01-0921.
19. Pielecha, I., Borowski, P., Czajka, J. and Wislocki, K., "Spray Analysis Carried Out With the Use of Two Angularly Arranged Outward-Opening Injectors," ILASS Americas 25th Annual Conference on Liquid Atomization and Spray Systems, Pittsburgh, PA, May 5-8, 2013. p. 7.
20. Pielecha, I., Pielecha, J. and Merkisz, J., "Gaseous and PM Emission from Combat Vehicle Engines during Start and Warm-Up," SAE Technical Paper 2010-01-2283, 2010, doi:10.4271/2010-01-2283.
21. Pielecha, I., Maslennikov, D. and Wislocki, K., "Optical Research of Spray Development of E85 Fuel in High Pressure Gasoline Direct Injection System," SAE Technical Paper 2010-01-2285, 2010, doi:10.4271/2010-01-2285.
22. Tatschl, R., Pachler, K. and Winklhofer, E., "A Comprehensive DI Diesel Combustion Model for Multidimensional Engine Simulation," In Proceedings of COMODIA 1998. Kyoto, pp 141-148.
23. Wickman, D., Yun, H. and Reitz, R., "Split-Spray Piston Geometry Optimized for HSDI Diesel Engine Combustion," SAE Technical Paper 2003-01-0348, 2003, doi:10.4271/2003-01-0348.
24. Wislocki, K., Pielecha, I., Czajka, J. and Maslennikov, D., "Evaluation of the Liquid Fuel Spray Parameters Based on Experimental Research and Numerical Simulation for the Piezoelectric Injectors," Archives of Transport 23(1): 111-121, 2011, doi:10.2478/v10174-011-0008-7.
25. Wislocki, K., Pielecha, I., Czajka, J. and Maslennikov, D., "The Influence of Fuel Injection Parameters on the Indexes of Fuel Atomization Quality for a High Pressure Injection," SAE Technical Paper 2010-01-1499, 2010, doi:10.4271/2010-01-1499.
26. Wislocki, K., Pielecha, I., Czajka, J. and Stobnicki, P., "Experimental and Numerical Investigations into Diesel High-Pressure Spray - Wall Interaction under Various Ambient Conditions," SAE Technical Paper 2012-01-1662, 2012, doi:10.4271/2012-01-1662.

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Definitions/Abbreviations

BBDC - before bottom dead center

BTDC - before top dead center

CA - crank angle

CO - carbon monoxide

CoV - coefficient of variation

HC - hydrocarbons

HTC - heat transfer coefficient

IMEP - indicated mean effective pressure

Mo - torque

n - engine speed

$\mathbf{NO_x}$ - nitrogen oxides

OH - hydrogen oxide group

P - pressure

$\mathbf{P_{air}}$ - air pressure

Pcyl - cylinder pressure

$\mathbf{P_{cyl\text{-}max}}$ - maximum of cylinder pressure

$\mathbf{P_{inj}}$ - fuel injection pressure

$\mathbf{q_o}$ - fuel quantity

SOI - start of injection

$\mathbf{t_{inj}}$ - time of fuel injection

TKE - turbulence kinetic energy

VCR - variable compression ratio

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