

# THE THEORETICAL STUDY ON INFLUENCE OF FUEL INJECTION PRESSURE ON COMBUSTION PARAMETERS OF THE MARINE 4-STROKE ENGINE

Jerzy Kowalski

Gdynia Maritime University  
Department of Engineering Sciences  
Morska Street 81-87, 81-225 Gdynia, Poland  
tel.: +48 58 5586434, fax: +48 58 5586399  
e-mail: jerzy95@am.gdynia.pl

## Abstract

*The manuscript presents the analysis of influence of fuel injection pressure on the combustion parameters of marine 4-stroke diesel engine. Analysis is based on computational fluid dynamic (CFD) model designed on the basis of the motion mesh of combustion chamber of the marine engine cylinder and air inlet and exhaust gas ducts. Presented model consists of models of fuel injection into combustion chamber, breaking-up and evaporation of the fuel, mixing with air and turbulent combustion with heat transfer to construction elements of the engine cylinder. Mentioned CFD model is validated according to boundary and initial conditions taken from direct measurements. The chosen research object is laboratory 4-stroke turbocharged Diesel engine with direct injection of the fuel and mechanically controlled of both cylinder valves and the injector. During the calculation the fuel dose, delivered into the engine cylinder was changed without any other changes in the initial and boundary conditions. This approach to the problem allows to the cause-effect analysis. The results of presented study are as follows: The increase of the fuel injection pressure causes the increase of fuel dose, delivered into the engine cylinder and the increase of intensity of both kinetic and diffusion stage of the combustion process. The result of this is the increase of pressure and temperature of the combustion and significant increase of the NO<sub>x</sub> fraction despite the decrease of the O<sub>2</sub> content in the combustion chamber of the engine.*

**Keywords:** marine engine, multidimensional model, combustion model, fuel injection pressure, CFD

## 1. Introduction

The most popular machines to produce the mechanical energy on ships are piston, Diesel engines with a direct fuel injection to the combustion chamber and auto ignition. Mentioned engines are produced in both versions: 4-stroke engines to electric power generation and main propulsion and 2-stroke engines, which are operated as main propulsion engines. The requirements of environmental protection impose on shipping companies makes the producers of engines continuous improvement of its design. Obtaining reduced fuel consumption and, as required by the law, reduced greenhouse gas emissions, requires a modification of the processes occurring in the combustion chambers and air-exhaust gas exchange systems of the engine. It should be noted, that properties of fuel injected to the combustion chamber of the marine engine plays a significant role in the control of the combustion process. Parlak et al. [17] observe that injection timing is a good solution to the decrease of the nitric oxides (NO<sub>x</sub>) emission from relatively small engines. The second important observation is that is possible to find the optimal injection timing for minimal specific fuel consumption and minimal NO<sub>x</sub> emission for all considered engine rotational speeds. Similar investigations are presented in [1]. The fuel injection strategy is commonly known method to reduce both, the NO<sub>x</sub> emission and fuel consumption during low loads of the engine. The analysis of this phenomenon is presented in [7] and [16]. The changes in the combustion process are possible to do by modification of the fuel nozzle parameters. The influence of the shape of fuel spray on the emission and combustion parameters is experimentally [18, 19] and theoretically [5, 13] analysed.

Desantes et al. [3, 4] present the laboratory study on the heavy-duty engine. They observe changes of NO<sub>x</sub> and soot emission and specific fuel consumption with the changes in the fuel rate. The used method of changing the fuel rate is the changes in the fuel injection pressure. Mentioned observations show that modification of the fuel injection pressure is simple solution to modification of the combustion process parameters in the marine engines. According to this statement, the aim of presented study is theoretical analysis of the influence of fuel injection pressure on the parameters of the combustion process in the marine, 4-stroke Diesel engine. Mentioned theoretical analysis is prepared with use of 3-dimensional and multi-zone computational fluid dynamic (CFD) model, which is validated on the basis of values, obtained during laboratory investigations.

## 2. Laboratory research

The initial and boundary conditions and the validation data are taken from direct measurements. The selected research object was the marine 4-stroke, 3-cylinder, direct injected and turbocharged Diesel engine. The parameters of the engine are presented on the Tab.1 and the detailed description of the laboratory research is presented in [9-11].

Tab. 1. The laboratory engine parameters

Parameter	Value	Unit
Max. electric power	250	kW
Rotational speed	750	rpm
Cylinder number	3	–
Cylinder bore	250	mm
Stroke	300	mm
Compression ratio	12.7	–
Injector nozzle		
Holes number	9	–
Holes diameter	0.325	mm
Holes position diameter	7	mm
Holes position angle	150	deg.
Spray cone angle	6	deg.
Opening pressure	25	MPa

## 3. Model description

The CFD model is based on the moving mesh, which was prepared on the basis of the technical documentation of the research object. The detailed parameters of the mesh preparing and selection are presented in [8]. The fuel injection model is based on geometrical dimensions of the injector nozzle, which are presented in Tab.1 also. Measured injection timing equals to 18 degrees before top dead center of the piston position. The delay between pressure signal in the indicator valve and the combustion pressure is neglected. The initial value of the droplet diameter of the fuel injection is taken as the diameter of fuel nozzle holes. A further break-up of fuel droplets has been described by the Lagrange description [21] with TAB model application [15]. This model specifies the conditions for breaking-up of fuel droplets as a dimensionless factor that depends on the density of fuel and surrounded air, the viscosity of the fuel droplet, the relative velocity and the diameter of droplets. Generally, if the value of the mentioned factor is greater than 1, the droplet breaks up. Distribution of the mean droplet diameter, determined by the Sauter's method [21], is assumed as the  $\chi^2$  distribution.

Simultaneously with the fuel atomization process the process of evaporation is beginning. This process results from the heating of fuel droplets. The Dukowicz's model [6] is used to modeling

the heat flow from air to fuel droplets and the mass flow of fuel vapors from droplets to air. The spherical shape of fuel droplets (microgravity conditions) and a constant temperature and heat transfer conditions on the surface of the droplet is assumed.

Evaporated fuel is mixed with air in the engine cylinder. The modeling of turbulent mixture flow was prepared according to the k-zeta-f model [20]. The combustion process was described by the ECFM-3Z model [2]. It is a model developed for modeling the combustion in diesel engines and it belongs to the CFM (Coherent Flame Model) class of models. This model assumes that the chemical reactions take place in the relatively thin layer of the flame. The flame progresses to the direction of fresh mixture of air and fuel. Mentioned flame layer is defined as homogeneous mixture of fuel and air and its shape and size is defined by diffusion phenomena. In the present model, the auto-ignition delay is determined by air temperature, the density of the mixture and the molar concentrations of oxygen (O<sub>2</sub>) and fuel. Chemical kinetic calculations are prepared for assumed substitute fuel composition in the form of C<sub>13</sub>H<sub>23</sub> hydrocarbon. The detailed description of the model is presented in [12]. Presented model was validated according to maximum combustion pressure, mean combustion pressure (MIP), NO<sub>x</sub> and O<sub>2</sub> fractions in the cylinder during the combustion process. Detailed validation results are presented in [14].

#### 4. Results and discussion

The increase of fuel pressure before fuel nozzle causes the increase of the fuel dose delivered to the combustion chamber. Assuming the flow coefficient as constant the flow characteristic of fuel to the combustion chamber is proportional. The presented analysis assumes the increase and the decrease of fuel dose in relation to values, measured during the laboratory tests. The considered parameters of the analysis are presented in Tab. 2.

Tab. 2. The laboratory engine parameters

Fuel dose at max load per cylinder [g]	0.652	0.734	0.815	0.896	0.978
Changes [%]	-20	-10	0	10	20
Mean fuel pressure changes [%]	-33.1	-17.5	0	19.4	40.5

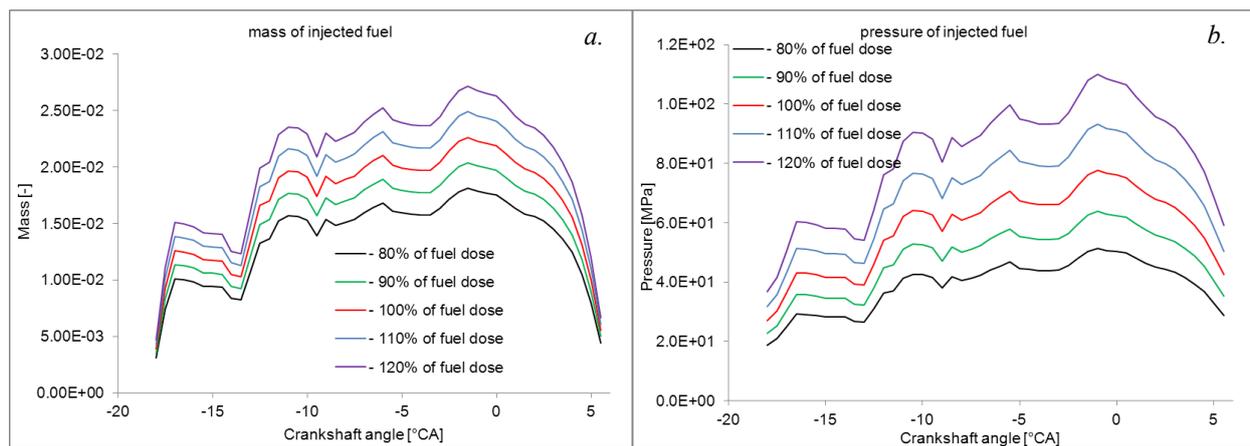


Fig. 1. Assumed characteristics of: a.) injected fuel mass and b.) fuel injection pressure

The assumed mass characteristics of fuel injection into engine cylinder are presented on the Fig. 1a. Fig. 1b presents characteristics of injected fuel pressures for all considered points of calculations. All parameters are presented for the maximum electric power and rotational speed of the engine, presented in Tab. 1.

According to results presented in Fig. 1, the increase of fuel pressure causes the increase of fuel dose injected into engine cylinder at the same time. The time of injection was assumed as constant,

and it equals  $24^\circ$  of crankshaft angle (CA). The increase of fuel injected dose increases of the fuel mass evaporation. According to results, presented on the Fig. 2a and Fig. 2b the evaporated mass of fuel is proportional to the injected fuel dose. It should be noted, that the quantity of fuel injected into the cylinder not significant influences of the auto-ignition time. The extreme of considered fuel doses cause changes in the auto-ignition position by only  $0.7\text{CA}$ . The Fig. 2c and the Fig. 2d present characteristics of the heat release and the cumulative heat release during combustion process. The increase of fuel dose causes the intensification of both kinetic and diffusion stages of the combustion process. The increase of the fuel pressure causes the increase of maximum value of the heat release in both kinetic and diffusion stages of the combustion.

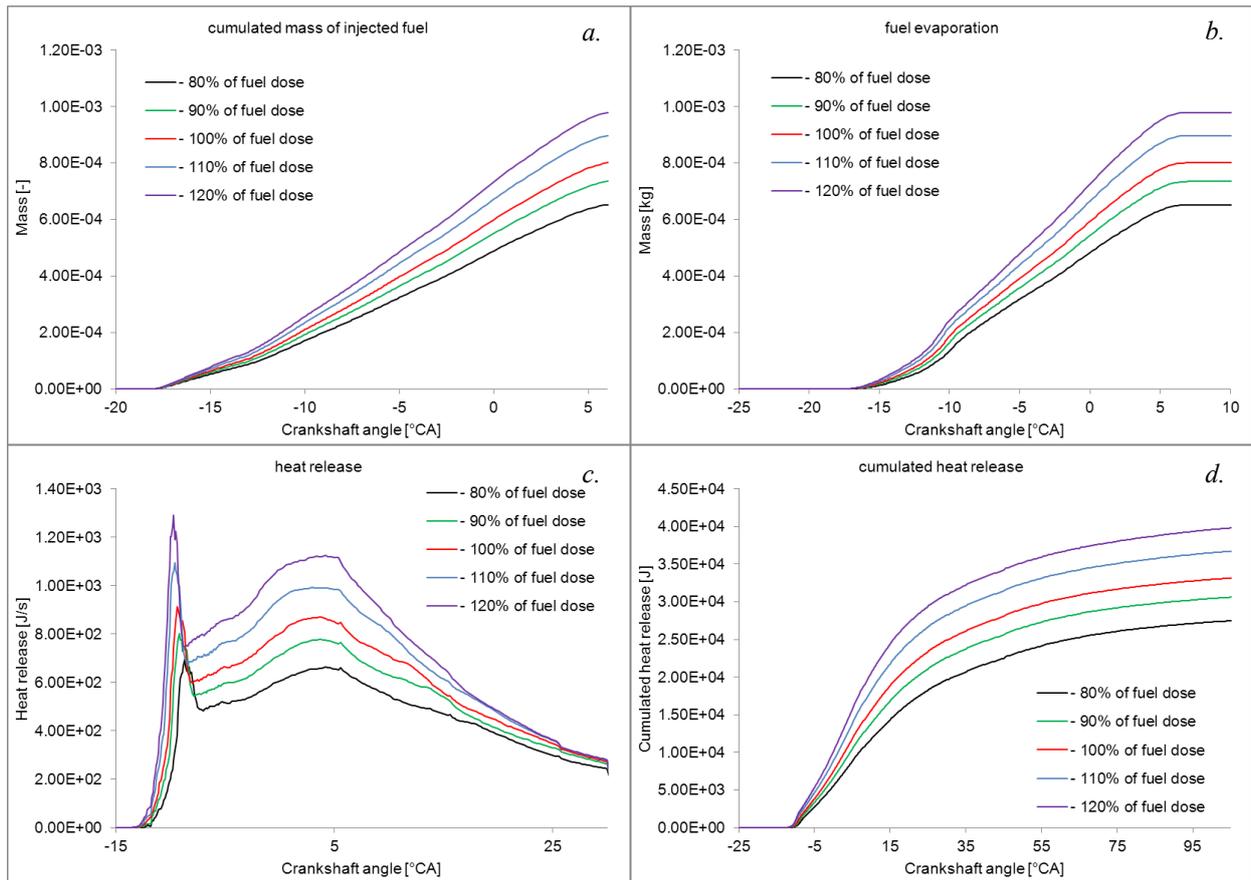


Fig. 2. Characteristics of: a.) cumulated mass of injected fuel, b.) fuel evaporation, c.) heat release and d.) cumulated heat release

The speed of heat release increases with the increase of the fuel pressure. The increase of the fuel pressure causes the changes of the maximum value of the kinetic stage of the heat release from  $-8.3\text{CA}$  to  $-9.7\text{CA}$  and maximum value of diffusion stage of the heat release from  $4.2\text{CA}$  to  $3.0\text{CA}$  accordingly. The reason of this is the increase of fuel molar fraction in the combustion chamber with the increase of fuel dose.

Presented changes in the combustion process cause the changes in the thermodynamic parameters in the engine cylinder. The Fig. 3a and Fig. 3b present the changes in the pressure and temperature in the combustion chamber. Obvious is fact, that the increase of the fuel injection pressure causes the increase of both temperature and pressure of the combustion. It should be noted that presented characteristics are qualitatively similar to each other. The result of this is the increase of temperature and pressure of exhaust gas after exhaust valve opening. Mentioned dependence is presented on the Fig. 3d. On the other hand is interesting that, according to results presented in the Fig. 3c, the increase of maximum combustion pressure causes the increase of the MIP. Observed differences for extreme considered values of injection pressure equals 17.5% for

the maximum combustion pressure and 13.3% for MIP. The cause of this effect is the increase of the combustion efficiency with the increase of the fuel injection pressure. This statement is indirect presented in the Fig. 4. The mean values for overall cylinder volume of mass fractions of chosen chemical compounds are presented in the Fig. 4. According to presented results the increase of injected fuel dose causes the increase of the carbon oxide (CO) and the carbon dioxide (CO<sub>2</sub>) fraction in the combustion chamber.

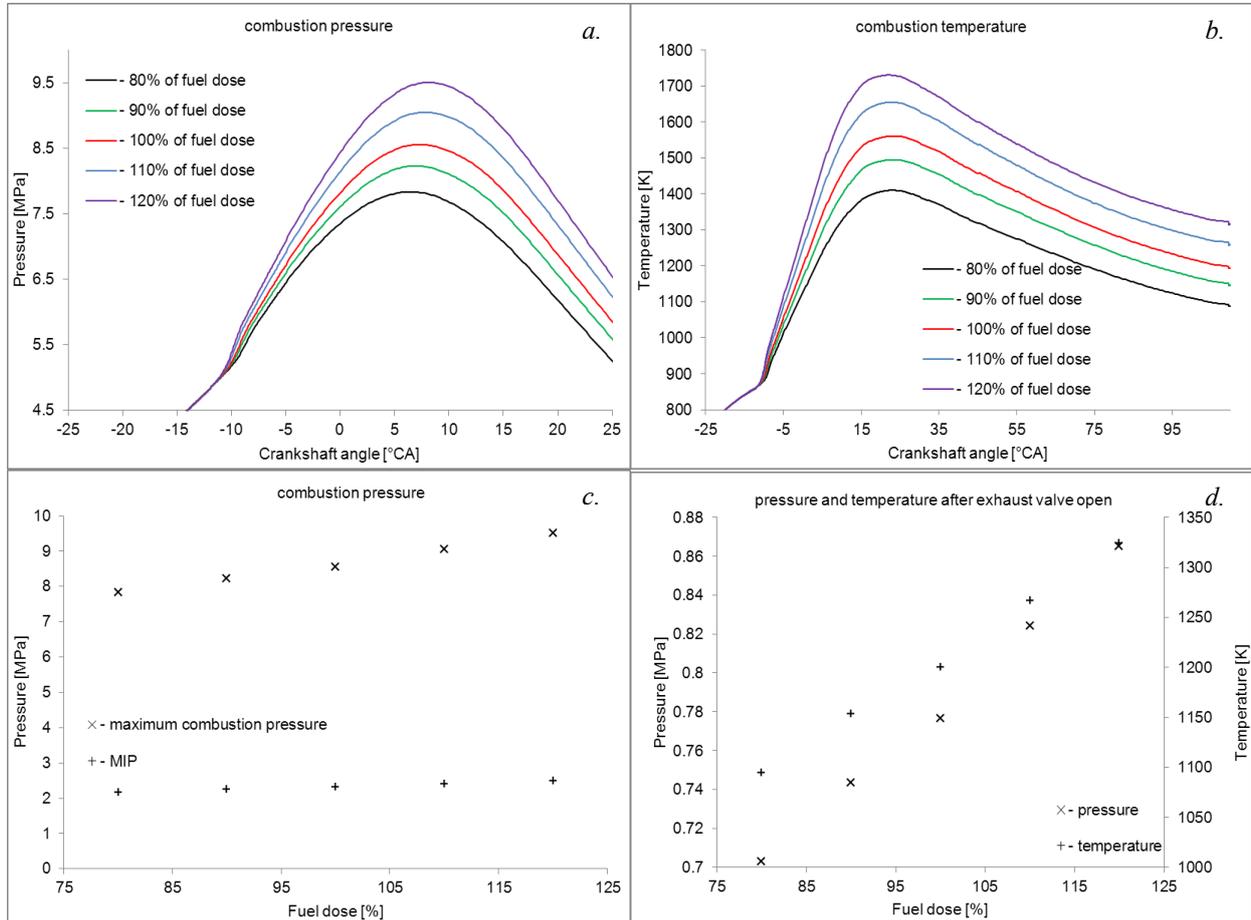


Fig. 3. Characteristics of: a.) combustion pressure and b.) temperature c.) maximum and MIP combustion pressure and d.) temperature and pressure after exhaust valve opening

Naturally, the increase of the CO and the CO<sub>2</sub> fractions is the effect of the increase of fuel mass in the same volume of the cylinder, but it should be noted, that the increase of the CO mass fraction is observed at the end of the combustion process also. According to results, presented in the Fig. 4c the difference for considered extreme values of fuel dose equals only 9.7% for the CO fraction and even 41.1% for the CO<sub>2</sub> fraction. It means that the increase of quantity of fuel delivered into the engine cylinder causes the increase of combustion process efficiency due to i.e. the increase of intensification of the combustion process. Mentioned intensification is the main reason of the increase of the NO<sub>x</sub> fraction in the combustion chamber. The changes of the NO<sub>x</sub> fraction in the combustion chamber are presented in the Fig. 4b. According to presented results, the extreme values of fuel pressure cause changes in the NO<sub>x</sub> fraction from 333 ppm for 80% of fuel dose to 1663 ppm for 120% of fuel dose. Mentioned results are observed at the exhaust valve opening time (right side of the Fig. 4b). This rapid increase of the NO<sub>x</sub> fraction is observed despite the decrease of the O<sub>2</sub> fraction in the cylinder. According to presented results, the statement should be formulated that in the marine Diesel engines, the dominant role in the NO<sub>x</sub> composition is the speed and temperature of combustion, the O<sub>2</sub> content in the combustion chamber is not so important. The characteristic of the O<sub>2</sub> fraction in the combustion chamber is presented in the

Fig. 4a. According to presented results, the increase of the fuel injection pressure and the fuel dose cause the decrease of the O<sub>2</sub> content in the combustion chamber. Observed result is expected due to the increase of fuel content in the engine cylinder. It should be remembered, that presented results are true only for the theoretical point of view. During the real engine operation, the increase of the fuel dose causes the increase of the turbocharger efficiency. The effect of this are changes in the air (and oxygen) content in the engine cylinder.

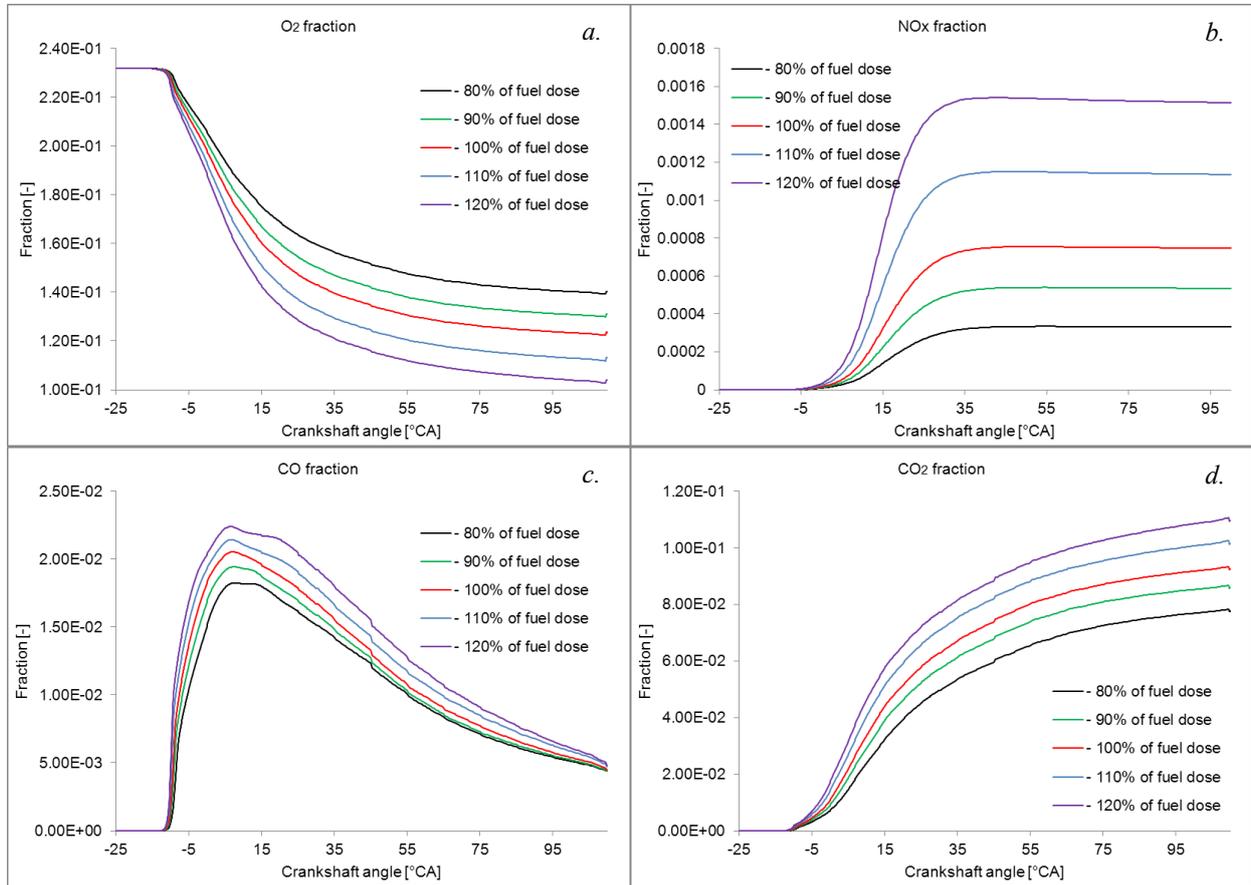


Fig. 4. In-cylinder gas composition. Fractions of: a.) oxygen b.) nitric oxides c.) carbon oxide and d.) carbon dioxide

## 5. Conclusions

The article presents the theoretical investigations on the influence of the fuel injection pressure on the parameters of the combustion process and the composition of the exhaust gas. Presented analysis is based on the CFD model of the combustion process. During the calculation the fuel dose, delivered into the engine cylinder was changed without any other changes in the initial and boundary conditions. This approach to the problem allows to the cause-effect analysis. According to presented results, the following statements should be formulated:

- the increase of the fuel injection pressure causes the proportional increase of the fuel dose injected into the engine cylinder,
- the increase of fuel dose causes increase of the speed of both kinetic and diffusion stages of the combustion process,
- the result of increase of the intensity of the combustion process is the increase of the pressure and temperature in the engine cylinder,
- the increase of the injected fuel dose causes the increase of the combustion process efficiency,
- the dominant role in the NO<sub>x</sub> composition is the speed and temperature of combustion, the O<sub>2</sub> content in the combustion chamber is not so important.

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## References

- [1] Buyukkaya, E, Cerit, M. *Experimental study of NOx emissions and injection timing of a low heat rejection diesel engine*, Int J Therm Sci. 2008, 47(8):1096-1106.
- [2] Colin, O., Benkenida, A., *The 3-Zones Extended Coherent Flame Model (ECFM3Z) for Computing Premixed/Diffusion Combustion*, Oil & Gas Science and Technology – Rev., Vol. 59, No. 6, pp. 593-609, 2004.
- [3] Desantes, J, M, Benajes, J, Molina, S, González, C, A., *The modification of the fuel injection rate in heavy-duty diesel engines*, Appl Therm Eng., 24(17-18):2715-2726, 2004.
- [4] Desantes, J, M, Benajes, J, Molina, S, González, C, A., *The modification of the fuel injection rate in heavy-duty diesel engines. Part I: Effects on engine performance and emissions*, Appl Therm Eng., 24(17-18): 2701-2714, 2004.
- [5] Desantes, J, M, Payri, R, Salvador, F, J, Gil, A., *Development and validation of a theoretical model for diesel spray penetration*, Fuel, 85 (7-8):910-917, 2006.
- [6] Dukowicz, J, K., *Quasi-steady droplet change in the presence of convection*, Informal report Los Alamos Scientific Laboratory, LA7997-MS, Los Alamos 1979.
- [7] Han, Z, Uludogan, A, Hampson, G, J, Reitz R, D., *Mechanism of Soot and NOx Emission Reduction Using Multiple-injection in a Diesel Engine*, 1996.
- [8] Jaworski, P., Kowalski, J., *3D mesh model for RANS numerical research on marine 4-stroke engine*, Journal of Polish CIMAC, Vol. 9-1, pp. 87-94, 2014.
- [9] Kowalski, J., *An experimental study of emission and combustion characteristics of marine diesel engine with fuel pump malfunctions*, Applied Thermal Engineering, Vol. 65, pp. 469-476, 2014.
- [10] Kowalski, J., *An experimental study of emission and combustion characteristics of marine diesel engine in case of cylinder valves leakage*, Polish Maritime Research, Vol. 22, No. 3, pp. 90-98, 2015.
- [11] Kowalski, J., *An experimental study of emission and combustion Characteristics of marine diesel engine with fuel Injector malfunctions*, Polish Maritime Research, Vol. 23, No. 1, pp. 77-84, 2016.
- [12] Kowalski, J., *Model procesu spalania w 4-suwowym silniku okrętowym*, Mechanik, Vol. 10, pp. 49-58, 2015.
- [13] Kowalski, J., *The analysis of influence of fuel spray angle on NOx fraction in the exhaust gas from marine 4-stroke diesel engine*, Journal of KONES, Vol. 22, No. 3, pp. 309-316, 2015.
- [14] Kowalski, J., *The analysis of influence of fuel spray angle on NOx fraction in the exhaust gas from marine 4-stroke diesel engine*, Journal of KONES, Vol. 22, No. 3, pp. 309-316, 2015.
- [15] O'Rourke, P, J. Amsden, A, A., *The Tab Method for Numerical Calculation of Spray Droplet Break-up*, SAE Paper Technical 872089, SAE International 1987.
- [16] Özkan, M, Özkan, D, B, Özener, O, Yılmaz, H., *Experimental study on energy and exergy analyses of a diesel engine performed with multiple injection strategies: Effect of pre-injection timing*, Appl Therm Eng., 53(1):21-30, 2013.
- [17] Parlak, A, Yaşar, H, Haşimoglu, C, Kolip, A., *The effects of injection timing on NOx emissions of a low heat rejection indirect diesel injection engine*, Appl. Therm. Eng., 25 (17-18): 3042-3052, 2005.

- [18] Payri, F, Bermúdez, V, Payri, R, Salvador, F, J., *The influence of cavitation on the internal flow and the spray characteristics in diesel injection nozzles*, Fuel, 83 (4-5): 419-431, 2004.
- [19] Payri, R, Salvador, F, J, Gimeno, J, Zapata, L, D., *Diesel nozzle geometry influence on spray liquid-phase fuel penetration in evaporative conditions*, Fuel, 87 (7): 1165-1176, 2008.
- [20] Poinso, T., Veynante, D., *Theoretical and numerical combustion*, Edwards 2005.
- [21] Zienkiewicz, O, C., Taylor, R, L., *Finite Element Method*, Vol. 3, Fluid Dynamics, Fifth Edition, Butterworth-Heinemann, Oxford 2000.