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MODEL-BASED ANALYSIS OF INJECTION PROCESS PARAMETERS IN A COMMON RAIL FUEL SUPPLY SYSTEM

MODELOWA ANALIZA PARAMETRÓW PROCESU WTRYSKU W UKŁADZIE ZASILANIA TYPU COMMON RAIL*

In this work, a simplified model of a rail-based fuel supply system in a compression ignition engine is presented. High pressure hoses were not taken into consideration and an empirical model was developed to simulate the injectors. The basic equations of the model are presented. Phenomena were described by 17 first-order ordinary differential equations. This work also contains an evaluation of the impact of the rail's geometrical parameters on the injection process. The evaluation was carried out via a program calculating the injection process, using a balance model of the injection system. A method for making preliminary decisions on the geometrical parameters of the rail is proposed.

Keywords: *common rail injection system, simulation calculations, injection process, fuel supply system operating parameters.*

W pracy przedstawiono uproszczony model zasobnikowego układu zasilania w paliwo silnika o zapłonie samoczynnym. W rozważaniach nie uwzględniono przewodów wysokiego ciśnienia, a do symulacji pracy wtryskiwaczy opracowano empiryczny podmodel. Przedstawiono podstawowe równania modelu. Zjawiska zostały opisane układem 17 równań różniczkowych zwyczajnych, pierwszego rzędu. W pracy również zawarto ocenę wpływu parametrów geometrycznych zasobnika na proces wtrysku. Ocenę przeprowadzono za pomocą programu obliczającego proces wtrysku, wykorzystującego model rozważanego układu wtryskowego. Zaproponowano sposób wstępnego doboru parametrów geometrycznych zasobnika.

Słowa kluczowe: *układ wtryskowy common rail, obliczenia symulacyjne, proces wtrysku, parametry eksploatacyjne układu zasilania*

1. Introduction

The development of contemporary high-speed compression-ignition engines is linked to the development of their fuel supply systems. Currently, common rail fuel injection systems dominate the supply of fuel to such engines. When choosing a fuel supply system for an engine, it is essential to consider many operational and control factors. Making use of simulation research for analysis of those parameters significantly eases and accelerates development work.

In the rail of a fuel supply system, the achievement of high pressure occurs in a piston-based high-pressure pump, from where a high-pressure hose leads to the fuel rail, after which relatively short hoses lead to the injectors.

Existing models describing common rail injection systems were created by research teams aiming predominantly at comparisons of their operational parameters and performance with those of other injection systems. They contained analyses and discussions of basic parameters connected to the injection event [3], yet they also allowed the determination of quantities which are difficult to measure, such as (for example) the effective cross-sectional area of the flow. Considerations can also be found dealing with compression-ignition fuel supply systems and problems associated with their control. One of the most fundamental works in this area is [2], in which the authors – as some of the first to do so – dealt with the fuel flow and control of injectors. In turn, in [4] a model of the overflow valve controlling the fuel pressure was developed, as well as a model of the fuel throttle valve in the high-pressure pump, which also considered a sub-model of that pump. The latter was controlled by a pseudo-random binary sequence. Analytical simulations of the properties of materials used in the construction of solenoid valves used in injectors and the operations of those values have also been carried out [8]. In turn, the authors of [10] focused on the stiffness of the needle-controlling rod assembly in their considerations, the values of the flow coefficients through the nozzle outlets and determination of the non-dimensional cavitation number. Determining the impact of wave phenomena in the rail on the injection event was the main topic addressed in [1], carried out by Daimler Chrysler AG, as well as [6], in which the impact of fuel properties and pressure, injection duration and the length and diameter of the injection hose on pressure changes in the system was investigated. Evaluations and selection criteria for injection systems' geometric parameters of can be found in [1, 9]. Another group of studies contains attempts to describe models aiming at control of pressure in a common rail injection system. Here, the modulus of compressibility, pressure, fuel temperature, and engine rotational speed were all taken into consideration and the results were linearized control models and initial design for controllers and regulators used for approval of systems controlling rail pressure [5, 7]. Work continues [11] on fourth-generation rail-based injection systems and on-line models adjusting the flow of fuel from the nozzle. This has served as inspiration to elaborate a simplified model of the fuel supply system.

A tendency towards the use of computer software for model ling and analysis of one-dimensional, multi-domain and mechatronic systems (interface, static and dynamic analysis) can be noted. In the majority of cases, the AMESim and Matlab/Simulink packages are used

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– although this is not a rule. Often, initial development is conducted via those packages and more precise efforts dedicated to a specific purpose are carried out using conventional programming languages. Commonly, these are one-dimensional models, describing the non-steady, elastic flow of fuel in the system. However, simulation work is conducted based on computational models of varying degrees of complexity. Technical analyses often employ simplified models, paying attention to a reduced set of factors impacting the trace of the processes under analysis. In such cases, it is important to always determine the impact of the simplifications on the accuracy of the parameterisation of the phenomena under consideration.

After conducting initial analyses of the available literature, it was concluded that models using fluid mechanics equations models in connection with results from test benches are lacking. Thus, it was decided to undertake work on a theoretical-empirical model of the system, by means of conventional programming language.

The work presented here concerns a simplified system model, in which fuel from the three-piston high-pressure pump is fed to the inlet chamber and from there to the pressure vessel (rail).

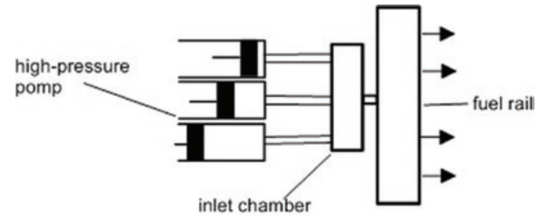

Fig. 1. Schematic of the modelled fuel supply system

In these considerations, the high pressure hose, the injector hoses and the injectors were not considered. The injection (fuel flow) proceeds through four openings, directly from the pressure vessel (Fig. 1).

2. Differential equations describing the system

Differential equations considering the system can be show in the form presented below.

2.1. Equations determining the pressure in the pump chambers

Mindful of the diminutive size of the chambers, it was taken that the change in pressure $p_p^{(i)}$ as a function of time can be can be determined from simplified continuity equations:

$$\frac{dp_p^{(i)}}{dt} = \frac{E_p^{(i)}}{V_p^{(i)}}\Big[\; A_p^{(i)}\frac{dh_p^{(i)}}{dt} - \mathrm{sgn}\,(p_p^{(i)} - p_d)\,\varepsilon_A^{(i)}\,\mu_d F_d^{(i)}\sqrt{\frac{2}{\rho_p^{(i)}}\left|p_p^{(i)} - p_d\right|} - \mathrm{sgn}\,(p_p^{(i)} - p_L)\,\varepsilon_B^{(i)}\,\mu_w F_w^{(i)}\sqrt{\frac{2}{\rho_p^{(i)}}\left|p_p^{(i)} - p_L\right|} - \varepsilon_u^{(i)} F_u^{(i)}\sqrt{\frac{2}{\rho_z^{(i)}}\left|p_p^{(i)} - p_z\right|}\;\Big]\,\eta_p\,\eta_t \tag{1}$$

for $i = 1, 2, 3$.

where: $p_p^{(i)}$ – pressure in the chamber of the ith pump piston,

$V_p^{(i)}$ – volume of the chamber of the ith pump piston,

$E_p^{(i)} = E_p^{(i)}(p_p^{(i)}, T)$ - modulus of elasticity of the fuel in the chamber of the ith pump piston,

$A_p^{(i)}$ – surface area of the ith pump piston,

$\frac{dh_p^{(i)}}{dt}$ – speed of the ith pump piston,

p_d – feed pressure,

μ_d – inlet opening flow coefficient,

$\varepsilon_A^{(i)}$ – control indicator,

$F_d^{(i)}$ – surface area of the inlet opening to the chamber of the ith pump piston,

$\rho_p^{(i)} = \rho(\rho_\rho^{(i)}, T)$ – density of the fuel in the chamber of the ith pump piston,

$\varepsilon_B^{(i)}$ – control indicator,

p_L – pressure in the inlet chamber,

μ_w – outlet opening flow coefficient,

$F_w^{(i)}$ – surface area of the outlet opening of the chamber the ith pump piston,

$\varepsilon_u^{(i)}$ – control indicator,

$F_u^{(i)}$ – surface area of the relief aperture,

p_z – rail pressure,

η_p – pump efficiency, dependent on rotational speed and fuel pressure,

η_t – correction factor considering changes in the effectiveness of the ith piston in the high-pressure pump in response to changes in the fuel temperature.

Controlling coefficients were inserted into the equations above; their interpretations are as follows:

$\varepsilon_A^{(i)}$ – volumetric outflow element, dependent on the pressure difference $p_p^{(i)}$ and p_d active only where the lift of the inlet valve $h_g^{(i)} \rangle 0$,

$\varepsilon_B^{(i)}$ – volumetric outflow element, dependent on the pressure difference $p_p^{(i)}$ and p_L active only where the lift of the valve ball connecting the pump chamber to the inlet chamber zero $h_k^{(i)}$ is greater than zero;

$\varepsilon_u^{(i)}$ – the third outflow element is active in equation (1) where the rail pressure p_z exceeds a given boundary value $p_z^{(gr)}$, simultaneously the pump piston moves upwards ($\dot{h}_p^{(i)} \rangle 0$) and the relative error of deviation of p_z from $p_z^{(gr)}$ exceeds the permissible level ε.

Moreover, it was accepted that the vent cross-sectional area $F_u^{(i)}$ changes in response to the pressure difference p_z and $p_z^{(gr)}$, according to the formula:

$$F_u^{(i)} = F_{u0}^{(i)}\sqrt{\frac{2}{\rho_z}\left|p_z - p_z^{gr}\right|} \tag{2}$$

The equations that make up (1) are non-linear first-order ordinary differential equations.

2.2. Equations of movement of the inlet valves

From Newton's second law of motion, it follows that motion of the inlet valves in a straight line is described by:

$$\frac{d^2h_g^{(i)}}{dt^2} = \frac{\varepsilon_g^{(i)}}{m_g^{(i)}} f_g^{(i)} \tag{3}$$

where: $f_g^{(i)} = -\left(h_g^{(i)} + h_{gl0}^{(i)}\right) k_g^{(i)} - \left(p_p^{(i)} + p_d^{(i)}\right) F_g^{(i)}$

$F_g^{(i)} = \frac{\pi \left[g_1^{(i)}\right]^2}{4}$ for i = 1, 2, 3,

$h_g^{(i)}$ – inlet valve lift,

$m_g^{(i)}$ – valve mass,

$h_{gl0}^{(i)}$ – basic tension of the spring,

$k_g^{(i)}$ – spring constant,

$F_g^{(i)}$ – valve surface area.

The control indicator ε_g present in the equations above takes values:

$$\varepsilon_g^{(i)} = \begin{cases} 1 & \text{where } wG^{(i)} = 1 \\ 0 & \text{in all other cases} \end{cases}$$

In turn, indicator $wG^{(i)}$ reflects whether the valve head has returned to the valve seat ($wG^{(i)}$ = 0), or is in motion ($wG^{(i)}$ = 1), and also whether it reached its maximum lift value $h_{g_{\max}}^{(i)}$ ($wG^{(i)}$ = 2), for which $wG^{(i)} \in \{0,\ 1,\ 2\}$.

Equation (3) constitutes a system of three second-order ordinary differential equations.

2.3. Equations of the motion of the outlet ball valves

The equations of motion of the outlet valves have the following form:

$$\frac{d^2h_k^{(i)}}{dt^2} = \frac{\varepsilon_k^{(i)}}{m_k^{(i)}} f_k^{(i)} \tag{4}$$

where: $f_k^{(i)} = -\left(h_k^{(i)} + h_{kl0}^{(i)}\right) k_k^{(i)} - \left(p_p^{(i)} + p_o^{(i)}\right) F_w^{(i)}$ for i = 1, 2, 3,

$h_k^{(i)}$ – outlet valve ball lift,

$m_k^{(i)}$ – ball mass,

$h_{kl0}^{(i)}$ – initial spring tension,

$k_k^{(i)}$ – spring constant,

$F_w^{(i)}$ – surface area of the outflow aperture.

The control indicator εk present in those equations takes values:

$$\varepsilon_g^{(i)} = \begin{cases} 1 & \text{where } wG^{(i)} = 1 \\ 0 & \text{in all other cases} \end{cases} \qquad wK^{(i)} \in \{0,\ 1,\ 2\}$$

In turn, $wK^{(i)}$, as with indicator $wG^{(i)}$, indicates the position of the ball valve: 0 where no flow between the pump chamber and the inlet chamber occurs, 1 if the ball is in motion, 2 if the ball achieved its maximum lift value.

The equations of (4) constitute a system of three second-order ordinary differential equations.

2.4. Equations describing the pressure in the inlet chamber

As with the case of the high-pressure pump chamber, it was taken that changes in the inlet chamber pressure can be described by a simplified continuity equation:

$$\frac{dp_L^{(i)}}{dt} = \frac{E_L^{(i)}}{V_L^{(i)}}\Big[\ \text{sgn}\,(p_p^{(i)} - p_L)\, \varepsilon_B^{(i)}\, \mu_w^{(i)} F_w^{(i)} \sqrt{\frac{2}{\rho_p^{(i)}}\left|p_p^{(i)} - p_L\right|} + \\ - \text{sgn}\,(p_L - p_Z)\, \mu_L F_L \sqrt{\frac{2}{\rho_0}\left|p_L - p_Z\right|}\ \Big] \tag{5}$$

for i = 1, 2, 3,

where: $V_L^{(i)}$ – inlet chamber volume,

$\mu_w^{(i)}$ – flow coefficient for the rail inlet aperture,

F_L – area of the inflow opening to the rail, equal to the cross-sectional area of the aperture connecting the inflow chamber with the rail;

all other symbols as defined in section 2.1.

It is worth noting that volume V_L must be increased by the volume of the hose connecting the inlet chamber with the rail:

$$V_L := V_L + \frac{\pi d^2}{4} L \tag{6}$$

where: d – hose diameter, L – length of the hose connecting the inlet chamber with the rail.

2.5. Rail pressure equations

A continuity equation was used to describe the pressure differential:

$$\frac{dp_z}{dt} = \frac{E_z}{V_z}\Big[\ -\sum_{i=1}^{4} \text{sgn}\,(p_z - p_k)\, \varepsilon_z^{(i)}\, \mu_z^{(i)} A_z^{(i)} \sqrt{\frac{2}{\rho_z}\left|p_z - p_k\right|} + \\ \text{sgn}\,(p_L - p_Z)\, \mu_L F_L \sqrt{\frac{2}{\rho_L}\left|p_L - p_Z\right|}\ \Big] \tag{7}$$

where: $A_z^{(i)}$ – variable surface area of fuel outflow through the outlet aperture,

V_L – volume of the inlet chamber,

p_k – pressure in the combustion chamber (back-pressure).

$$\varepsilon_z^{(i)} = \begin{cases} 1 & \text{where } \ t \in \left\langle t_A^{(i)},\ t_B^{(i)} \right\rangle \ \ i \ \ p_z > p_k \\ 0 & \text{in all other cases} \end{cases}$$

$t_A^{(i)}, t_B^{(i)}$ – aperture opening times,

other symbols as previously defined in parts 2.1 and 2.4.

Equation (7) is a non-linear first-order ordinary differential equation.

Modelling of hydrodynamic phenomena in the injector presents a number of difficulties. An essential aspect for a correct model of the rail is the definition of a rule describing the flow of fuel from the rail to the combustion chamber. In the algorithm presented here, it was assumed that further apertures are opened at intervals equal to 180° of pump shaft rotation. It is also important to choose the values for a number of coefficients necessary to conduct a proper quantitative assessment of the phenomena taking place. Here, coefficients of hydraulic resistance, flow rate coefficients and coefficients of resistance to movement of moving parts can all be mentioned. The values of those quantities vary in response to fuel pressure, which complicates their determination. Furthermore, when model ling electronically-controlled injection systems, solenoid control valves must be considered, which requires familiarity with further quantities, especially the properties of the materials used. The values of certain quantities are sometimes difficult to estimate, and thus it was decided to develop and empirical model for the flow of fuel from the atomizer, based on function $A_z^{(i)}$. Values of characteristic times obtained from the analysis of injection events were used.

A basic observation made during the experiments was that the real trace of the needle lift, as well as function $A_z^{(i)}$ deviate from theory, in which the following are defined:

$t_B^{(i)}$ – given opening time, $t_0^{(i)}$ – given pause time.

Most importantly, it can be stated that the real opening time $\overline{t_B}$ is greater than the given $t_B^{(i)}$ by an approximately constant magnitude, denoted as $t_d^{(i)}$ – injection delay time. The delay time considers differences between the given and realized injection time; the value was determined experimentally. The trace of function $A_z^{(i)}$ also took forms more closely approximation to a parabola than to the theoretical profile in the form of a rectangular function.

Depending on the value of $t_0^{(i)}$ and $t_d^{(i)}$, the following two cases can be obtained:

$$t_d^{(i)} < t_0^{(i)} \qquad \text{and} \qquad t_d^{(i)} > t_0^{(i)}$$

taken into consideration in the computer programme developed in this work.

In the programme calculating the injection process, the possibility of entering a pressure value below the injection should not commence was included (this is equivalent to the pressure opening the injector). This is protection against calculating injection parameters in cases where the quality of the atomisation process (not analysed via this model) could prove to be unsatisfactory.

3. Numerical integration of the differential equation system

The majority of the methods for integration of systems of ordinary differential equations require the insertion of higher-order equations to first-order equations. Thus, equations (3) and (4) were inserted into the appropriate two first-order equations. Equations (1), (3), (4) and (7) were then saved in the form of a system of first-order equations, of form:

$$\dot{X} = F(t, X) \tag{8}$$

where F is a vector function and X:

$$X = \left[p_L, p_p^{(1)}, ..., p_p^{(3)}, h_g^{(1)}, \dot{h}_g^{(1)}, ..., h_g^{(3)}, \dot{h}_g^{(3)}, h_k^{(1)}, \dot{h}_k^{(1)}, ..., h_k^{(3)}, \dot{h}_k^{(3)}, p_z \right]^T$$

is a vector with $m = 17$ elements.

It is therefore necessary to integrate the system m = 17 first-order ordinary differential equations. To that end, the Runge-Kutte 4th order method was employed, using constant integration steps.

4. Initial conditions, further points

Calculations were conducted assuming that for the first instant (t = 0) all pressures are equal to the pressure of the inflowing (feed) fuel p_d and all lift values and speeds are zero, i.e.:

$$X_{t=0} = \left[p_d, p_d, p_d, p_d, 0, 0, 0, 0, 0, 0, 0, 0, 0, 0, 0, 0, p_d \right]^T \tag{9}$$

It was also assumed that the first section of the high-pressure pump is the first to start working, the other two sections commencing after (respectively) 120 and 240 degrees of pump shaft rotation. It was therefore assumed that angles $\alpha^{(i)}$ are described by the dependency:

$$\alpha^{(i)} = \begin{cases} 0 & \text{where } t < t_i \\ \omega(t - t_i) & \text{where } t > t_i \end{cases} \tag{10}$$

where: ω - angular velocity,

$$t_i = 0 \quad \text{dla } i = 1, \qquad t_i = \frac{2\pi}{3\omega} \quad \text{dla } i = 2, \qquad t_i = \frac{4\pi}{3\omega} \quad \text{dla } i = 3.$$

As empirical research was conducted for the assumed operating conditions for the analysis of the fuel supply system, results obtained from the computer simulation can be considered only after multiple work cycles (φ > 720°), since during the initial phase the computation is too sensitive to the impact of initial conditions (9) and shifts t_i in equation (10).

As previously mentioned, the outlet apertures on the rail are further activated (cyclically) every:

$$\Delta T = t_B^{(i)} - t_A^{(i)} = \frac{\pi}{\omega} \tag{11}$$

The programme for modelling operation of a common rail injection system developed based on the dependencies presented above, facilitates the calculation of the pressure trace in the pump chambers, the inflow chamber and the fuel rail, as well as the lift of pistons and movable valve elements. The complete fuel dose in the injection and the fuel flow rate through particular injection apertures are computed. Injection event traces can be designated for both non-split and split doses and for varying pause time values.

Verification calculations were conducted for a fuel supply system with a cylindrical high pressure rail (rail). Comparisons were carried out for the following: split injection – pilot event 450 μs, pause 600 μs, main event 450 μs, at a set rail pressure of 700 bar, pump rotational speed 695 rpm, injection order 1 – 2 – 3 – 4. Differences between the calculated and measured values ranged from 2.4% to 7.7%, depending on the injector selection group. Such differences mainly result from the simplifications adopted in the model, as high pressure hoses and injector assemblies are not included. The dose delay time and its dependence on fuel pressure have a significant impact on the dose.

5. Impact of rail geometric parameters on the injection process

Using the considered injection system model, using the program calculating the injection event, calculations were performer for various setpoints of the injector control signal. A split (two-part) fuel dosing strategy was considered. The calculations aimed to provide qualitative and quantitative evaluations of the influence of test quantities on the injection parameters. The considerations regarding the influence of geometrical parameters of the fuel container on the injection process are presented here.

The rail is a relatively simple element in terms of its design, yet it plays a key role in the limitation of the propagation of pressure waves. A correctly chosen capacity ensures continuity of dosing during abrupt changes in engine operating parameters. As previously mentioned, a cylindrical high pressure rail used on the injection system of a compression-ignition engine (of swept volume 1700 cm^3) was adopted for carrying out model calculations.

Using the model, reviews of the influence of the length, diameter and volume of the rail on the fuel supply parameters were conducted. Calculations were conducted: for fixed rail diameter and variable rail length and fixed rail length and variable rail diameter. Changes to the trace of the injection process, the fuel dose, the angle of the start of injection and the angle of injection duration were analysed.

Evaluation of the impact of rail length

Figures 2 and 3 present the traces of injections from an injector calculated for constant diameter and various rail lengths. The dashed purple line presents results for the basic rail length used by the engine manufacturer (201.4 mm). Here, the injection angle is 11° and this remains the same for all cases. However, the angle of start of injection changes; for the considered range of lengths the range of changes amounts to 8° of pump shaft rotation. This is a relatively important change in an important injection parameter, which must be considered in the design of algorithms controlling engine operation. Such changes result – above all – from the means of controlling the injector in the model, which enables it to open at a given pressure value.

Together with the increase in rail length, the mean value of rail pressure barely changes (0.02%) and such changes are practically unnoticeable. Similarly insignificant changes result from extreme flow rates of fuel from the atomizer.

Conversely, there are differences between the maximum and minimum pressure values. For a rail of length 160 mm, the difference amounts to 77 bar, for a rail of length 201.4 mm 62.6 bar, and for 250 mm as little as 51.9 bar. Such changes affect the behaviour of fuel in the rail.

The changes in injection process parameters presented here result – above all – from increases in the volume of the element under consideration. As the calculations were made with control system settings unchanged, increases in the volume chase progressively later

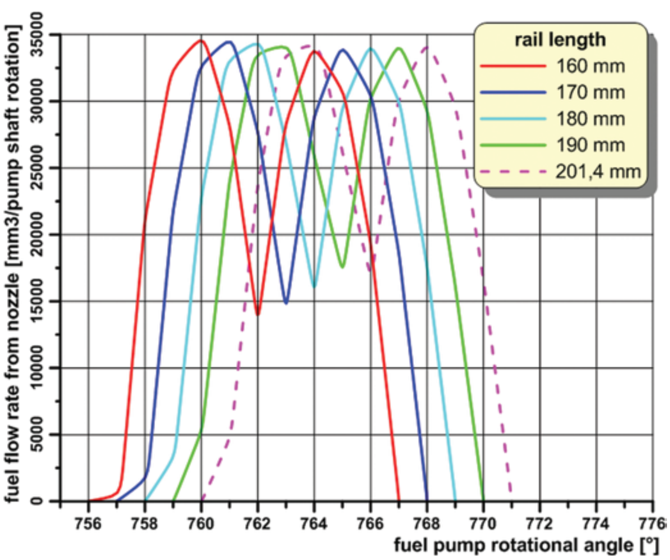

Fig. 2. Calculated injection traces for rail lengths 160 mm – 201.4 mm

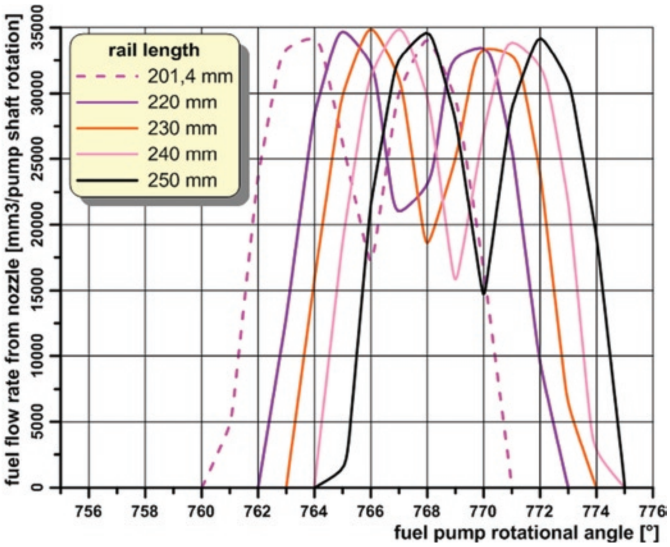

Fig. 3. Calculated injection traces for rail lengths 201.4 mm – 250 mm

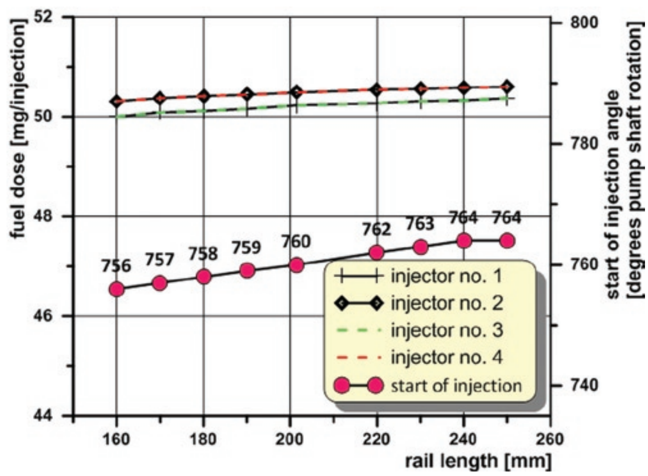

Fig. 4. Calculated fuel doses and angles of start of injection for various rail lengths

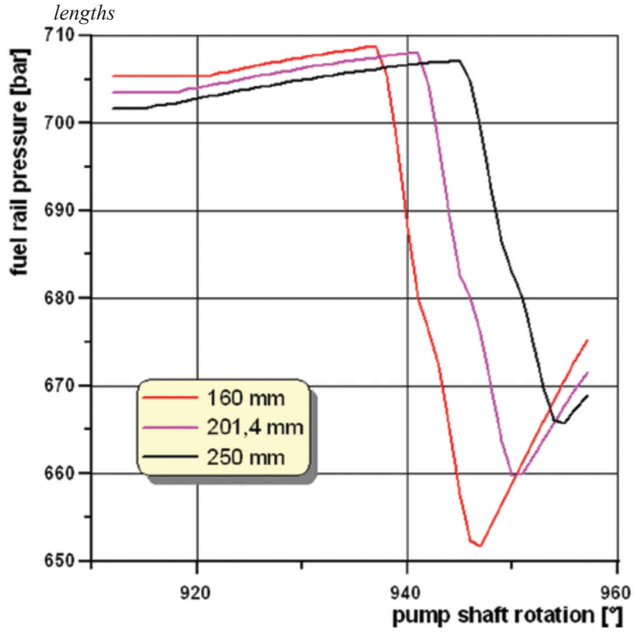

Fig. 5. Calculated pressure changes in the rail during the injection event, together with angles of start of injection for selected rail lengths

achievement of the required pressure level. This is the delay of start of injection (Fig. 4). As the injection duration time does not change, the end of the injection event also occurs later and later, which leads to ever smaller differences in the pressure between the start and end of injection (Fig. 5). Thus, a small (0.7%) increase in the fuel dose was noted.

Evaluations of the influence of rail diameter

The changes in the high-pressure rail length analysed above had a linear character and changed the injection parameters in the same way. The situation is somewhat different where injections are considered for constant rail length and variable rail diameter. In such considerations the direction of change is similar, yet changes in the volume are significant and occur non-linearly, in proportion to the given diameter raised to the second power. In order to obtain a full picture of changes in injection parameters, a wide range of diameter values was used, from the smallest (corresponding to the diameter of the injection hose) up to 20 mm – thus being values greater than those used in the majority of fuel rails used in passenger car engines. With regard to changes in rail diameter, it can be seen that a greater proportion of the energy delivered to the rail is used in the process of compressing the fluid. The greater quantity of fuel in the rail absorbing the delivered energy causes significant delays in the start of injection – by as much as 64° (Fig. 6, also Fig. 8), at a constant injection angle value of 11°. However, in this case an important role is played by the control of

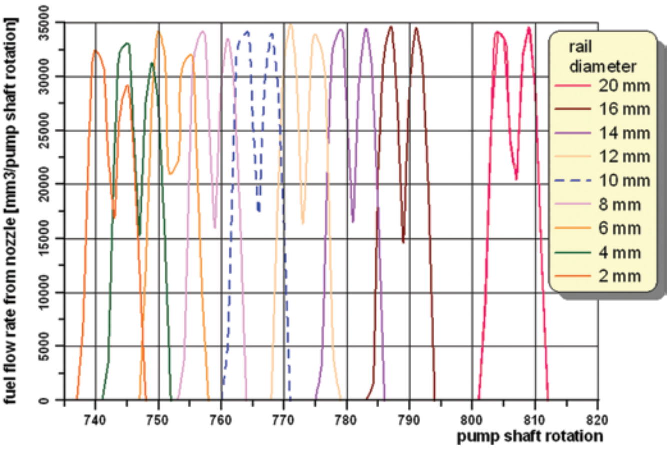

Fig. 6. Calculated injection traces for various rail diameters

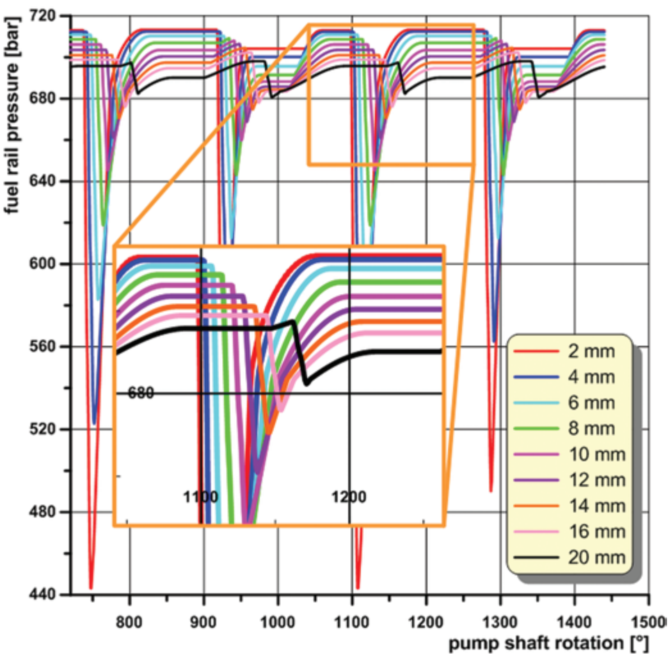

Fig. 7. Calculated pressure traces for various rail diameters

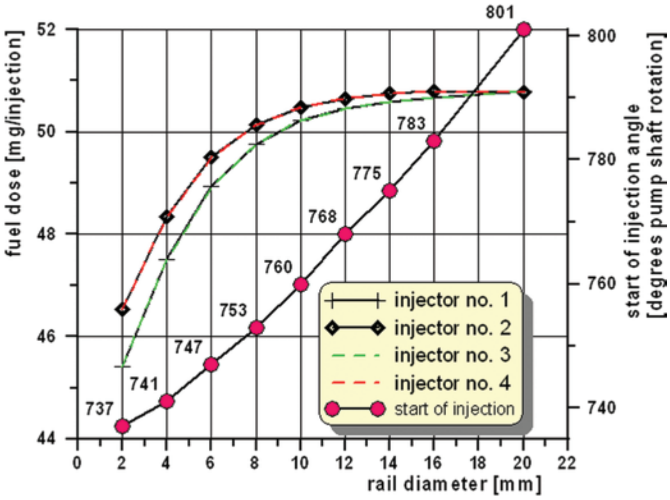

Fig. 8. Calculated dose and angle of start of injection values for various rail diameters

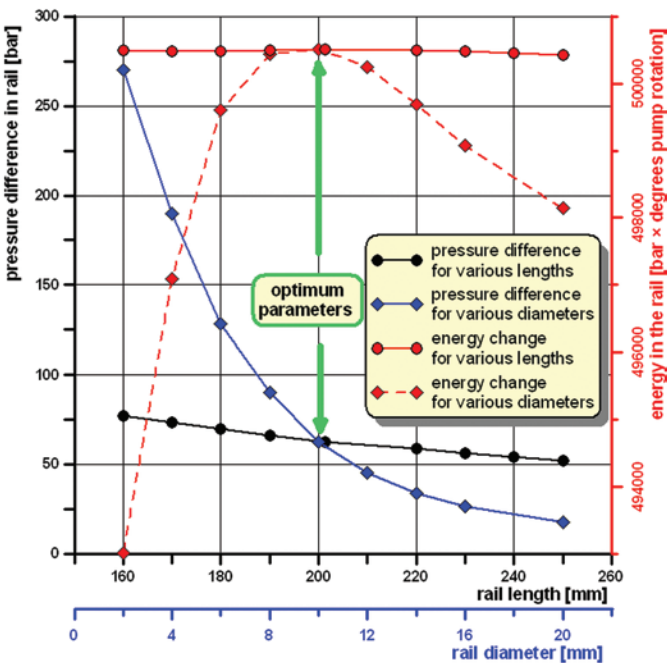

Fig. 9. Calculated pressure differences and energy changes for various rail dimensions

injection opening. Mean rail pressure values change by some 1.5% and do not reflect changes in the rail during the injection process, especially in the case of the smallest diameters, where large pressure changes occur (Fig. 7). This causes greater changes in the fuel dose and angle of start of injection. With an increase in rail diameter, significant differentiation in the flow rate of fuel from the atomizer is in fact not observed (Fig. 6). Such changes can be observed only for the smallest rail diameters – i.e. where the greatest pressure drops occur. The shape of the injection event does not change. Pressure differences (Fig. 7) have an impact on the fuel dose delivered.

For large pressure drops, part of the process occurs at low pressure values, leading to a lower quantity of fuel (Fig. 8). Over the entire range considered the change in the dose was significant, as it was 11.8%.

The modelling analysis results presented above do not fully resolve all questions. Additional calculations were made, the results of which were taken into consideration in a qualitative analysis of the influence of the aforementioned fuel rail operating parameters on the injection process (Table 1).

Table 1. Qualitative evaluations of investigated parameters

Parameter	Value range	Impact of test parameter on:		
		fuel dose [mg]	injection duration angle [°]	angle of start of injection [°]
Pause time	200 μs-900 μs	+	+ + +	- -
Injection delay time	100 μs-900 μs	+ + +	+ + +	- -
Rail length (with spigot spacing changed proportionally)	160 mm-250 mm	+ -	- -	+
Rail diameter	2 mm-20 mm	+ + +	- -	+ + +

Legend: + + + high impact
+ significant impact
+ - insignificant impact
- - no impact

The aforementioned differences between the maximum and minimum rail pressure values are presented in Fig. 9. Differences for various lengths are shown in black and compared with differences occurring as a result of changes in rail length (blue line). It can be seen that as the length and diameter of the rail increase, the pressure differences resulting from the injection process are reduced, and the impact of changes in the rail diameter is significantly greater.

If it may be assumed that a measure of the available energy in the fuel before the injection is the area under the rail pressure trace curve, then that quantity changes for various rail lengths to a degree equal to changes in the mean pressure value – that is to say, very little (Fig. 9, solid red line). The same figure shows changes in the energy of the fuel in the rail (red line). While changes in rail length (within the considered range) do not cause significant changes in this quantity, the situation is completely different for changes in diameter (Fig. 9, dashed red line). The trace reaches its maximum for a rail of diameter 10 mm and length 201.4 mm. For such a dimensional configuration the fuel energy before the injection is the greatest and can be property used to prepare the fuel-air mixture. The values given here as optimal were taken by the manufacturer of the fuel system under analysis and employed in engines of swept volume 1700 cm^3.

It should, however, be underlined that the rail was modelled in a simplified manner, without considering wave phenomena. A real fuel rail is subject to the laws of wave propagation; local areas of pressure higher or lower than the given value occur, which can influence the dosing process to a great degree. Following consideration of such effects, changes in the quantitative evaluations of the dependencies presented here may occur.

6. Summary

Modelling plays a significant role in the design and selection of machine parts. It allows the implementation time of the designed system to be significantly shortened, as well as to adapt it to the building stage in parallel with the design process. In order for the mathematical model to best reflect real phenomena, a correct physical model of the test system should be prepared. It is obvious that the degree of simplification of the modelled system will affect the accuracy of the results of calculations, but in many cases the use of simplifications is necessary, due to the complexity of the mathematical model, which would increase the required calculation time and thus reduce the efficiency of the program.

The developed injection process model, applicable to the commonly used fuel injection system in common rail compression-ignition engines, allowed determination of the relationships between the parameters of the test system. From the obtained test results, it is possible to distinguish the factors which have the greatest impact on the fuel dose, the injection trace, the start angle and the duration of injection. The impacts of the analyzed quantities on injection parameters varied; they can be summarized as follows.

- The pressure trace has a significant impact on the en tire injection process and fuel dose. Increased rail pressure causes changes in the flow rate of fuel leasing the atomizer, which translates into an increase in fuel expenditure.
- For a given length within the range of considered values, the diameter of the rail Has a significant influence on the angle of the start of injection. Increases in this value cause increases in the start angle and decreases in the quantity of fuel dosed. This results from the volume and compressibility of the fuel, since greater volume cause extended reaction times to the signal controlling the rail pressure.
- Changes in rail length for constant rail diameter have an insignificant influence on the angle of the start of injection. Greater rail length increases the angle of the start of injection, but to a lesser degree than changes in diameter. This results from the lesser increase in fuel volume. This quantity does not affect the injection duration angle.
- Changes in rail diameter and length at constant rail volume do not have Any influence on the injection parameters analysed here.

Taking into account the simulation results obtained, as well as differences between the results of calculations and the values measured on the test bench, it has been confirmed that the values are very similar. However, as always, there are discrepancies between the real system and the model. The values of these discrepancies permit assessment of the quality of the model and its susceptibility to changes in the set values. Despite the considerable complexity of the algorithms and the large number of parameters that can be changed, the model is relatively predictable in terms of the results it generates. This feature allows for quick model tests and selection of suitable initial parameters that will allow the desired trace of the injection process and fuel dosage to be obtained. The computer calculation program, developed on the basis of the physical model presented in this paper can be qualified as reflecting the studied parameters of the injection system well. Due to some simplifications adopted in the mathematical model, there are differences in the results of calculations and results from measurements, but they do not significantly change the results. The evident imperfection is the limited split of the fuel dose into parts.

In the current version of the program, only two parts can be made. In the trace of further work, the program and model should be adapted to current requirements and, using experimental results, empirical and computational models should be developed, taking into account both the possibility of dividing the injection into a larger number of parts and including a larger number of control parameters.

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