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Numerical analysis and experimental studies on solenoid common rail diesel injector with worn control valve

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Abstract. A mathematical model of a solenoid common rail fuel injector was developed. Its difference from existing models is control valve wear simulation. A common rail injector of 0445110376 Series (Cummins ISf 2.8 Diesel engine) produced by Bosch Company was used as a research object. Injector parameters (fuel delivery and back leakage) were determined by calculation and experimental methods. GT-Suite model average R^2 is 0.93 which means that it predicts the injection rate shape very accurately (nominal and marginal technical conditions of an injector). Numerical analysis and experimental studies showed that control valve wear increases back leakage and fuel delivery (especially at 160 MPa). The regression models for determining fuel delivery and back leakage effects on fuel pressure and energizing time were developed (for nominal and marginal technical conditions).

1. Introduction

Most of modern vehicles with diesel engines are equipped with common rail fuel injection systems. They help to meet strict requirements for exhaust gas emission rates. This is possible due to high fuel pressure generation (until 300 MPa) and use of biodiesel fuel [1-4].

Common rail fuel injectors are the most important components of the system [5]. They allow fuel delivery into cylinders with high accuracy depending on the operating engine mode. There are two types of injectors - with piezo actuators and with solenoid actuators [6-8]. Each of them has some advantages and disadvantages [9]. Regardless of the type of an injector used, high accuracy of fuel delivery must be provided for quite a long period.

Technical conditions of a common rail fuel injection system change with an automobile mileage increase [10]. This is typical for injectors. In this case, the fuel delivery process is broken, and the exhaust emission volume increases. The periodicity of soot filters regeneration also increases and its resource decreases. For this reason, injector parameters which change as a result of wear have to be studied. This will allow for an on-board diagnosis to track technical conditions very accurately [11, 12].

Most of the studies aim to optimize common rail construction factors and their effects on response and delivery performance [1, 6, 9, 13-15]. The fuel amount depends on common rail pressure and energizing time. A regression model of an injector of 0445110376 Series (ISf 2.8 Cummins) was developed for nominal conditions [16]. The fuel pressure generation process was studied for moving vehicles [17].

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2. Mathematical simulation

The simulation scheme of a solenoid injector is presented in Figure 1. Complex dynamic hydromechanical processes are typical for the mathematical model of a common rail injector.

Electromagnet processes in the drives of injector control valves have significant effects on fuel injection processes.

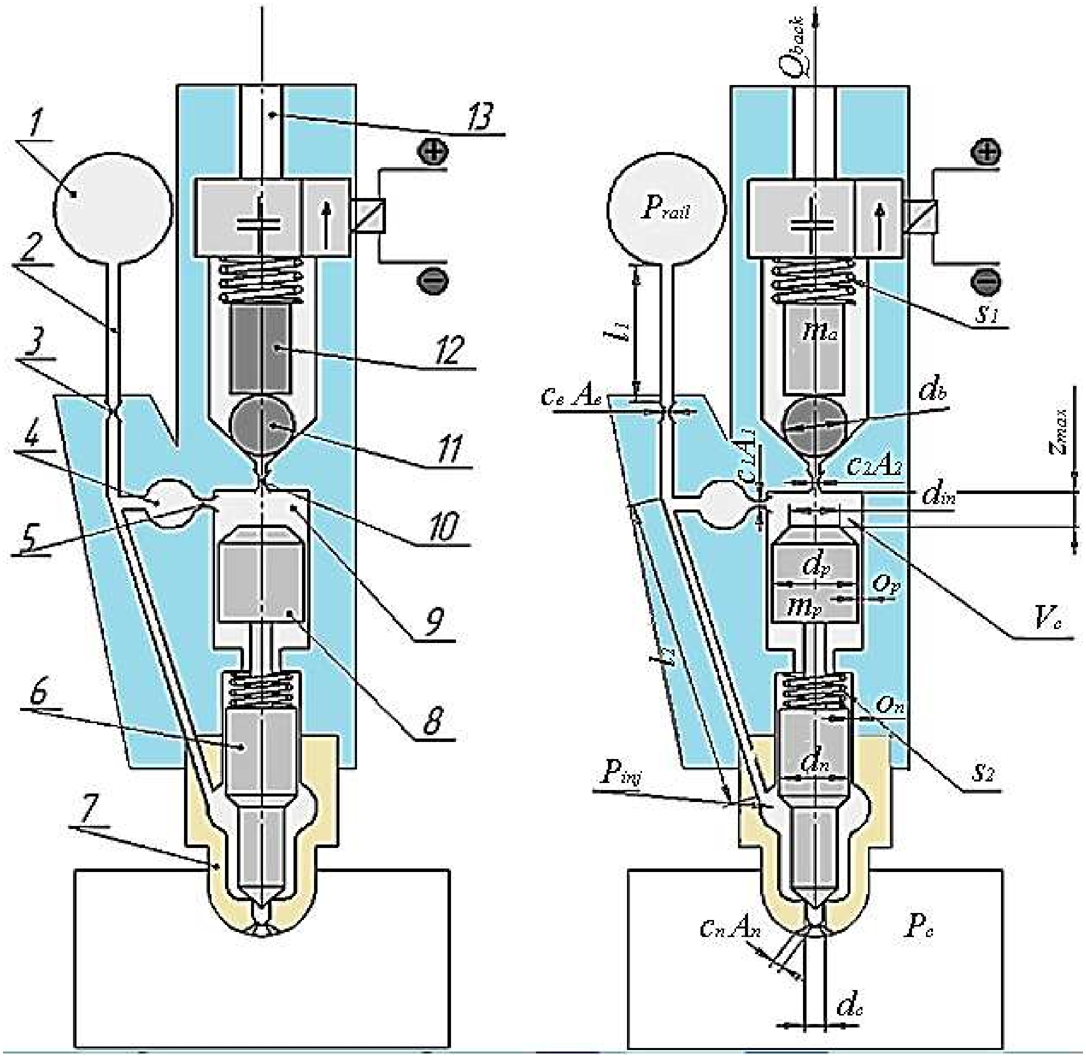

Figure 1. Schematic cross-section of a common rail injector: 1 – rail; 2 – high pressure pipe; 3 – edge filter; 4 – inlet chamber volume; 5 – inlet orifice; 6 - needle; 7 – nozzle body; 8 – plunger; 9 – control chamber; 10 – outlet orifice; 11 – ball; 12 – anchor; 13 – backleak volume.

The equation for determining valve armature movement is:

$$m_a \frac{dv_a}{dt} + k_{d1} v_a = (P_v - P_{bl}) \cdot A_v + \sigma_d \cdot F_m + s_1(x_1 + h) - s_2 \cdot h - \lambda_d \cdot F_1 \qquad (1)$$

where: m_a–anchor mass; h – armature (anchor) lift; v_a – armature velocity; k_{d1} – damping coefficient; P_v – control valve pressure (down); P_{bl} – back leakage volume pressure; A_v– cross-sectional area of the valve; σ_d and λ_d – algorithmic driving functions; F_m – magnet force; s_1- armature spring stiffness; s_2– over-travel spring stiffness; x_1 – preload of spring; F_1 – force of impact.

The equation for determining the algorithmic driving function (these functions are restituting valve armature moving) is:

$$\sigma_d = \begin{cases} 0, if\ t > \tau_e \\ 1, if\ t < \tau_e \end{cases} \tag{2}$$

$$\lambda_d = \begin{cases} 1, if\ h = h_{max} \\ 0, if\ 0 < h \leq h_{max} \\ 1, if h = 0 \end{cases} \tag{3}$$

where: τ_e – valve energizing time.

Let us solve the following first order differential equation (Faraday's law) to determine the current through the inductor [5,9]:

$$U_0 = i \cdot R + L\frac{di}{dt} \tag{4}$$

where L – inductance; i – current; R - coil resistance.

Magnet force was determined based on the Maxwell model [5]:

$$F_m(\delta) = \frac{i^2 \cdot w^2 \cdot \mu \cdot S}{2(\delta_m - h)} \tag{5}$$

where w- number of turns (of the coil); μ- permeability of free space; δ_m – air gap; S – equivalent electromagnet area.

When the valve armature is moving in upper and lower contacts, the impact occurs. Impact force simulated with regard to contact stiffness [1] is:

$$F_1 = k \cdot \delta \tag{6}$$

where k – contact stiffness; δ – penetration.

The body height over time is measured to obtain a restitution coefficient and contact time. For the configuration of a single mass interacting with a fixed ground, it is possible to convert a restitution coefficient and contact time into a damping coefficient and stiffness by the following formula [1]:

$$k = m_a \cdot \left(\frac{\pi}{\Delta t}\right)^2 \left[1 + \left(\frac{\ln k_r}{\pi}\right)^2\right] \tag{7}$$

where: m_a- mass of armature; k_r – restitution coefficient; Δt – contact time.

The restitution coefficient was calculated by formula [1]:

$$k_{\text{в}} = -\frac{\dot{h_{im}}}{\dot{h}} \tag{8}$$

where $\dot{h}_{im}$ – valve velocity after impact; $\dot{h}$ - valve velocity before impact.

The equation for displacement moving elements (control piston and needle) is:

$$m_p\frac{dv_z}{dt} + k_{d2}v_z = \sigma_1\left[\sigma_2\frac{\pi d_{in}^2 P_v}{4} + \frac{\pi(d_p^2 - d_{in}^2)P_v}{4} - P_{inj}A_n + s_3(z + x_2) - P'_{inj}\frac{\pi d_c^2}{4}\right] \tag{9}$$

$$v_z = \frac{dz}{dt} \tag{10}$$

where m_p – control piston and needle mass; k_{d1} – damping coefficient; P_v – control valve pressure; v_z – control piston and needle velocity; d_{in} – inner control piston diameter; d_p – outer control piston diameter; P_{inj} – pressure of injection; A_n - cross-sectional area of the needle; s_3– needle spring stiffness; x_2– preload of a needle spring; d_c – diameter of a needle cone.

Algorithmic driving function of control piston cross-sectional area changing is:

$$\sigma_2 = \begin{cases} 0, if z = z_{max} \\ 1, if\ 0 \leq z < z_{max} \end{cases} \tag{11}$$

The cross-sectional area of needle changes when needle moving is:

$$A_n = \frac{\pi d_n^2}{4}, if\ z > 0{,}005 \tag{12}$$

$$A_n = \frac{\pi(d_n^2 - d_c^2)}{4}, if\ z \le 0{,}005 \tag{13}$$

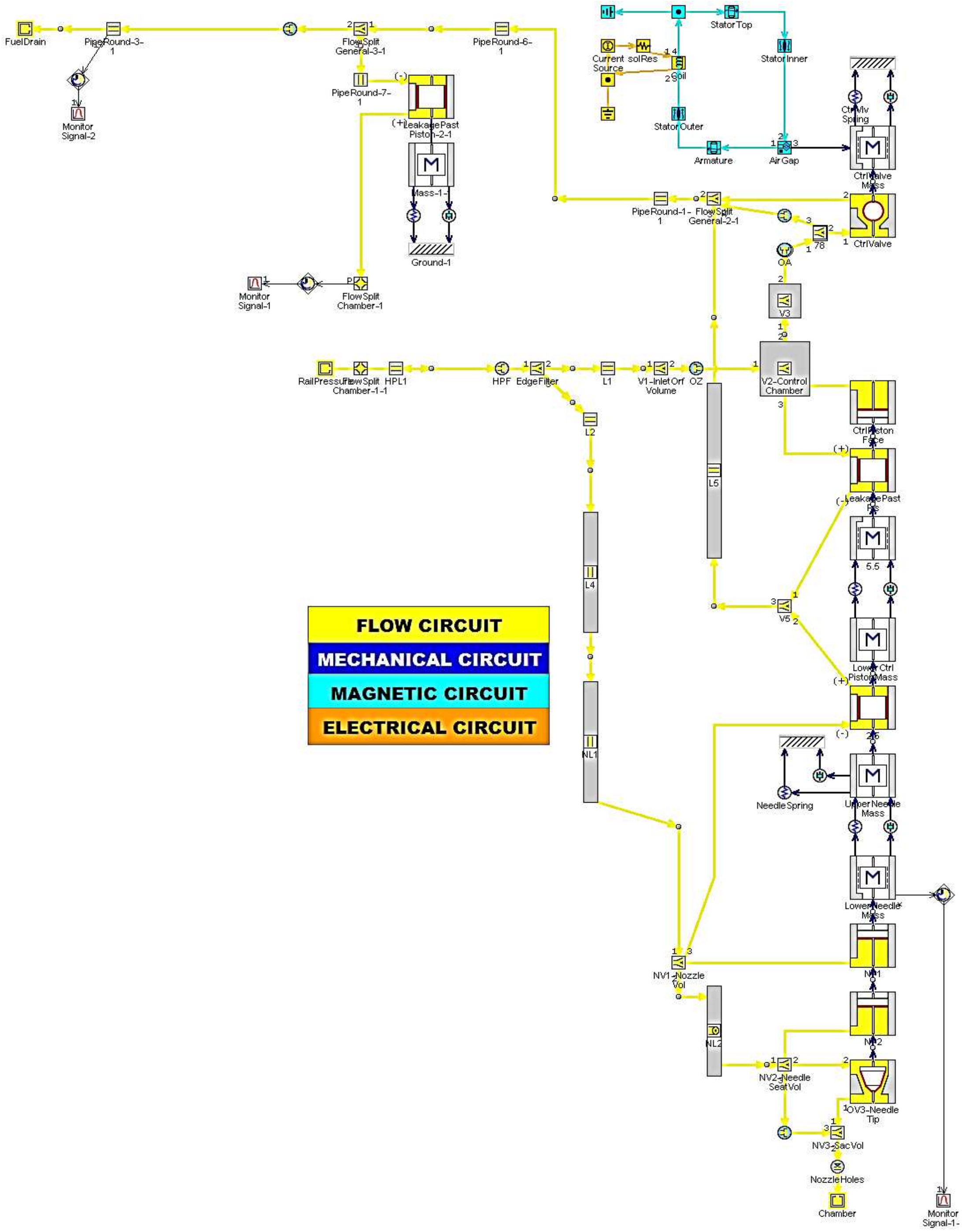

Figure 2. The common rail injector simulated using GT-Suite.

An algorithmic driving function of control piston moving restitution is:

$$\sigma_1 = \begin{cases} 0, if\ z = 0; \\ 0, if z = z_{max} \\ 1, if\ 0 < z < z_{max} \end{cases} \tag{14}$$

The equations are solved under initial conditions (initial time t=0):

$v_z = 0;\ P_v = P_{rail}; v_z = 0; h = 0; z = 0$.(15)

To solve the problem of unsteady fuel flow in a high-pressure line, GT-Suite was used. The model is presented as a scheme (figure 2).

The conditions set forth in [10] were used when determining injector back leakages. The main reason for reducing common rail injector resources is an abrasive wear.

A control valve seat wears out very often. Its wear forms are different. The authors assumed that the control valve wear is an extra orifice between a control chamber and backleak volume (figure 2).

A volume balance equation for the control chamber (figure 1) is:

$$\frac{dP_v}{dt} = \frac{Q_1 - Q_2 - Q_3 - \frac{\pi d_p^2}{4} \cdot \frac{dz}{dt}}{\beta\left(V_c - z \cdot \frac{\pi d_p^2}{4}\right)} \tag{16}$$

where Q_1 – fuel flow through an inlet orifice; Q_2 – fuel flow through an outlet (discharge) orifice; Q_3 – fuel flow through an extra orifice between the control chamber and backleak volume; d_p – control piston diameter; V_c – initial volume of the control chamber; z – control piston displacement; β – bulk modulus (fuel compressibility factor).

Fuel flow through i orifice is:

$$Q_i = c_i \cdot A_i \cdot \sqrt{\frac{2}{\rho} \Delta P} \tag{17}$$

where c_i - discharge coefficient of orifice i; A_i - cross-sectional area of orifice i.

Discharge coefficients of orifices were determined by experiments based on GT-Suite with regard to a flow regime and cavitation phenomenon.

The first order approximation to the change of density as a function of pressure and temperature was used. This model is useful if the bulk modulus and coefficient of thermal expansion are known at a single point or at many points [1,6,7].

3. Results and discussion

The results of the numerical simulation were compared with the results obtained under bench conditions.

The authors found that the control valve seat wear causes fuel delivery and back leakages. It is important to understand which wear is acceptable, and which is the marginal.

The results are shown in figures 3 and 4. The solid line corresponds to the calculated values of nominal technical conditions of the common rail injector of 0445110376 Series produced by Bosch Company. The greatest differences in performance are achieved at maximum fuel pressures (160MPa).

Needle motion results from an imbalance of pressure forces acting on the control piston and needle assembly. By opening the control valve, the control chamber pressure drops below the pressure in the nozzle volume, thus creating a net positive force in the needle assembly.

By closing the control valve, the control chamber pressure increases above the pressure in the nozzle volume thus creating a net negative force in the needle assembly. This force closes the needle until the end of the injection event occurs when the needle seats in the nozzle again.

Depressurization and repressurization of the control chamber is time dependent and introduces a lag in the needle motion relative to the electrical control signal, the motion of the control valve and its wearing.

When back leakage increases threefold, injection quantity increases by 13-40%, and the shorter the energizing time is, the larger the differences are (figure 3). The GT-Suite model average R^2 is 0.93 meaning that it predicts the injection rate shape very accurately.

An increase in fuel delivery when the control valve seat is worn out is due to increasing depressurization and decreasing volume of the control chamber. In this case, the needle lift and opening time increase (figure 4).

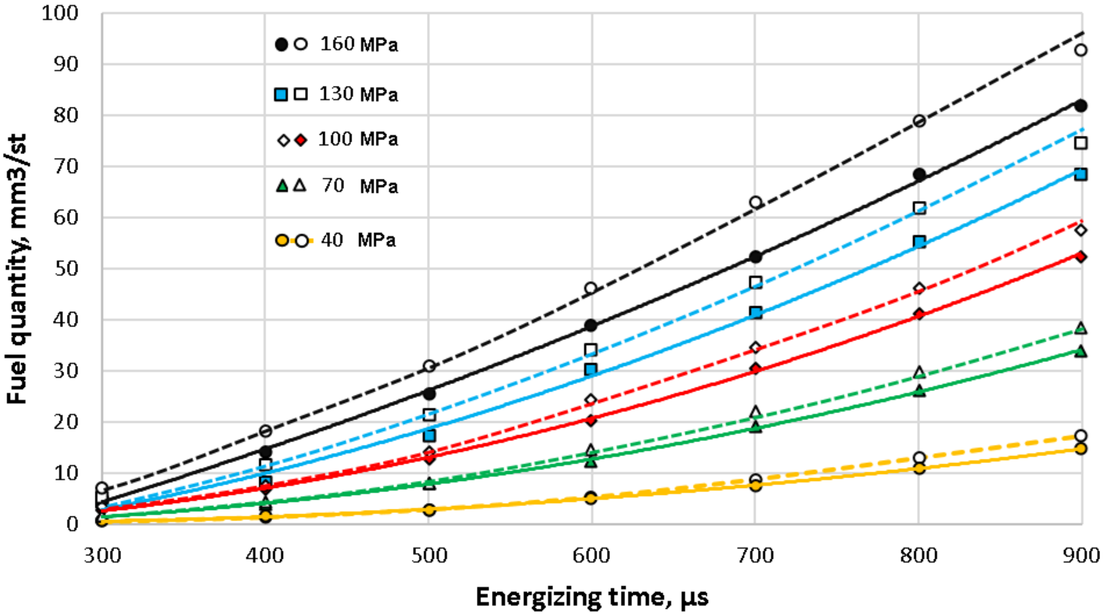

Figure 3. Graphical fuel quantity effects on energizing time and fuel pressures (solid line – injector of nominal technical conditions (calculation); dashed line – injector of marginal technical conditions (calculation); symbol – experiment)

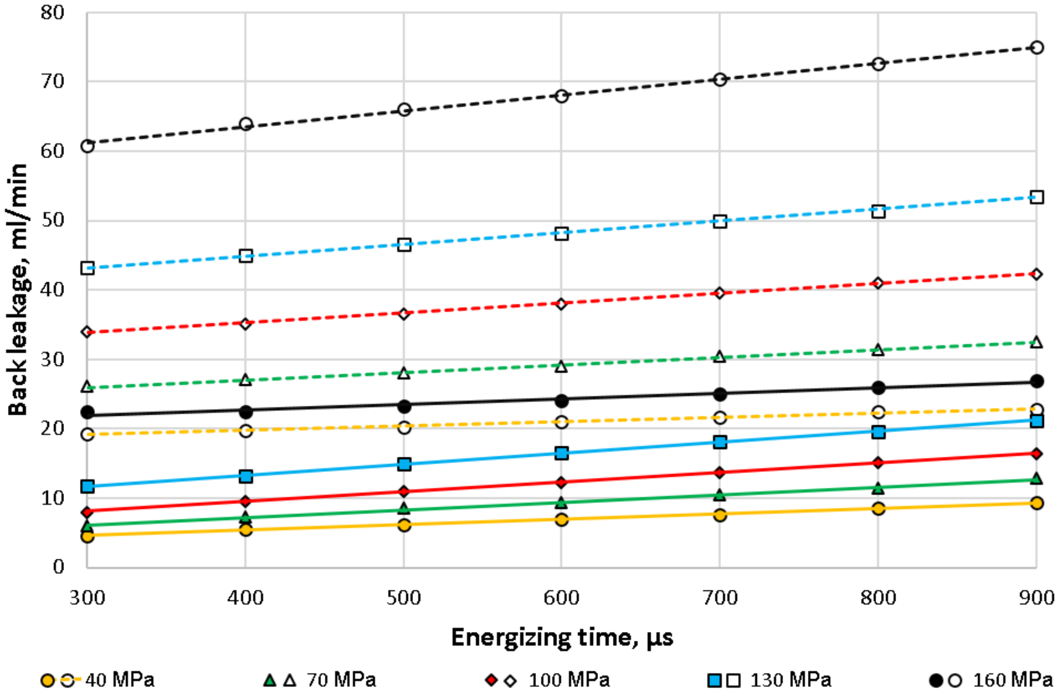

Figure 4. Graphical back leakages influence energizing time and fuel pressures (solid line – injector of nominal technical conditions (calculation), dashed line – injector of marginal technical conditions (calculation), symbol – experiment).

As seen from the calculation, the leakage cross sectional area is equal to 0.013 mm^2. This value is marginal. So, if the cross-sectional area of leakage is more than 0.013 mm^2, the Common Rail injector breaks down and there is a need for repair or replacement.

Numerical analysis and experimental data allowed development of regression models for nominal and marginal technical conditions. The polynomial equation was taken as a basis:

$$g\ (P_{rail}, \tau_e) = b_0 + \cdot P_{rail} + b_2 \cdot \tau_e + b_3 \cdot P_{rail} \cdot \tau_e + b_4 \cdot P_{rail}^2 + b_5 \cdot \tau_e^2 \quad (18)$$

where b_0, b_1, b_2, b_3, b_4, b_5– coefficients; P_{rail} – Common Rail fuel pressure;τ_e – energizing time.

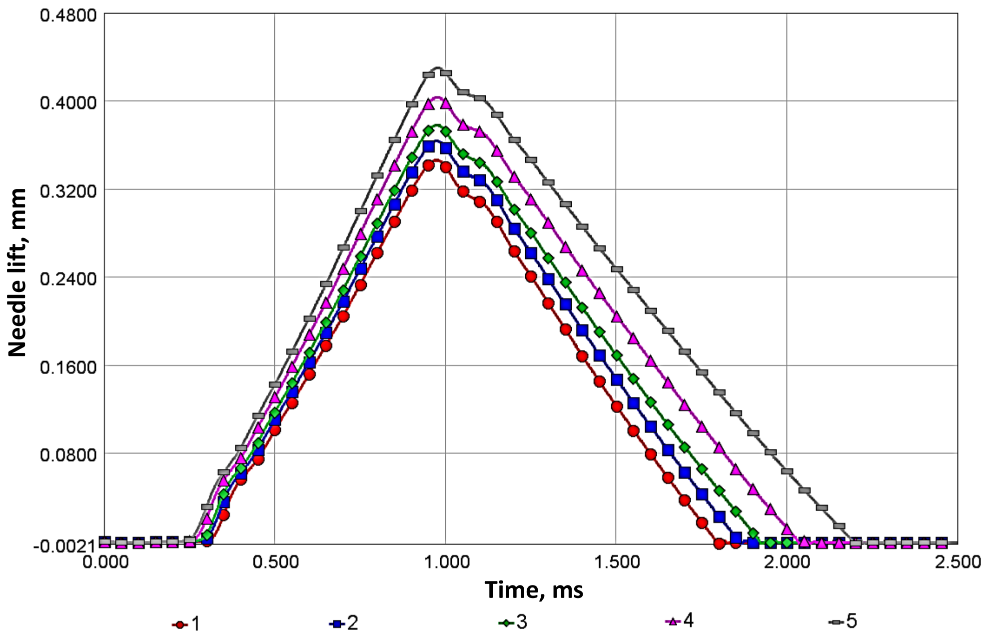

Figure 5. Effects of graphic needle lift of the injector of 0445110376 Series on a conditional control valve leakages area (calculation, pressure is 160 MPa, energizing time is 900 μs): 1 – 0 mm^2; 2 – 0.0095 mm^2; 3 – 0.012 mm^2; 4 – 0.0142 mm^2; 5 – 0.0165 mm^2

Assessment of the statistical significance of equation coefficients made it possible to discard non-significant terms. Polynomial equations are presented below. The equation for determining fuel delivery (nominal condition) is:

$$g\ (P_{rail}, \tau_e) = 19.6 - 0.254 \cdot P_{rail} - 0.0815 \cdot \tau_e + (9.046e - 04) \cdot P_{rail}\tau_e + \\ +(5.986e - 05) \cdot \tau_e^2 \quad (19)$$

The equation for determining back leakage (nominal condition) is:

$$Q\ (P_{rail}, \tau_e) = -6.83 + 0.1395 \cdot P_{rail} + 0.01129 \cdot \tau_e \quad (20)$$

The equation for determining fuel delivery (marginal condition) is:

$$g\ (P_{rail}, \tau_e) = 16.63 - 0.2481 \cdot P_{rail} - 0.07851 \cdot \tau_e + (9.807e - 04) \cdot P_{rail}\tau_e + \\ +(5.874e - 05) \cdot \tau_e^2 \quad (21)$$

The equation for determining back leakage (marginal condition) is:

$$Q\ (P_{rail}, \tau_e) = -5.335 + 0.3774 \cdot P_{rail} + 0.01421 \cdot \tau_e \quad (22)$$

Table 1. Parameters of the statistical assessment of the regression model

Parametric approximation equation		Sum of squares due to error (SSE)	Adjusted R^2	Root mean square error (RMSE)
Nominal technical condition	$g\ (P_{rail}, \tau_e)$	44.06	0.997	1.212
	$Q\ (P_{rail}, \tau_e)$	95.84	0.932	1.731
Marginal technical condition	$g\ (P_{rail}, \tau_e)$	50.89	0.997	1.302
	$Q\ (P_{rail}, \tau_e)$	410	0.955	3.58

4. Conclusion.

A mathematical model of a common rail fuel injector with an electromagnet actuator was developed. Control valve wear simulation makes it different from other existing models. The GT-Suite model average R^2 is 0.93, meaning that it predicts the injection volume and back leakages rate shape very accurately.

Numerical analysis and experimental studies showed that control valve wear increases leakages and fuel deliveries (especially when pressure is 160 MPa). So, when back leakage increases threefold, injection quantity increases by 13-40%, and the shorter the energizing time is, the larger the differences are.

Numerical analysis and experimental data allowed development of regression models for effects of fuel delivery and back leakage on common rail pressure and valve energizing time for nominal and marginal technical conditions (injector of 0445110376 series produced by Bosch company). So, the regression model of fuel delivery influences common rail pressure and valve energizing time, which is a polynomial second-order equation. The back leakage regression model influences common rail pressure, and valve energizing time is a first-order equation.

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