Operation-related features of diesel fuel injection systems at pressures up to 400 MPa

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Abstract. The interest in the high pressure injection in diesel engines continues to grow. Optimal injection pressure is a function of the engine parameters and its application and is open to question. The study investigates the injection features of diesel fuel at pressures above 300-400 MPa. The injection pressure level, at which the fuel flow through the nozzle ceases to grow, was experimentally determined, and an improved simulation method for the heavy fuel injection modelling was proposed.

1. Introduction
High fuel injection pressure remains the most important way to reduce emissions of harmful components and fuel consumption in engines. Recommendations for the optimal injection pressure are found in fundamental publications [1] and many journals.

Industrial fuel systems now provide injection pressures of 180-250 MPa. Some publications mention that there are plans to develop fuel equipment with injection pressure up to 280-300 MPa. Fuel injection systems that ensure ultrahigh injection pressure are being developed [2, 3].

Numerous experimental studies of jet motion patterns involve high pressure air chambers. These studies have been demonstrated that the jet evolution depends on the direction and position of the injector holes [4]. All these results showed a close dependence of the length of the fuel jet in the chamber on the injection pressure. It is important that this dependence differs from the known dependences that were obtained at outdated injection pressure levels [5]. These features of jet characteristics also apply to diesel fuels of vegetable origin, as well as various mixtures [6].

The research in respect of fuels being injected into the constant volume box aims at identifying the nature and quantitative indicators of such injection at elevated pressures [7]. However, no anomalies are detected about 300 MPa. The Paper [8] investigates shock waves generated during intense injection into the air chamber. To visualize the shock waves, the author employs the method of Schlieren imaging. However, the shock waves developed during the experiment did not exceed 350 m/s, and the pressure remained between 200 and 400 MPa. The only difference is that such a limit will be achieved more quickly at a pressure of 400 MPa.

The form and pattern of the fuel jet motion changes at ultrahigh (elevated) injection pressures. At the same time, the question remains unclear whether the fuel consumption in the injector holes changes.
The well-known direction of development of fuel equipment for intensification of injection is out of question; at the same time, the optimal value of the pressure still remains unclear. The only scientifically proven method for selecting an injection pressure value exists, namely, optimization of atomization, evaporation and combustion in the engine's combustion chamber.

Previously, the search for optimal solutions was sought by experimental methods. However, this is resource and time consuming. Methods of mathematical modeling and the corresponding software allow solving such problems [9].

To simulate fuel atomization and combustion successfully, it is necessary to determine the patterns of the velocity and fuel consumption through the nozzle holes at ultrahigh pressures.

2. Issues related to fuel injection at ultrahigh pressures

Ultrahigh pressures in Common Rail systems pose new issues related to design aspects, combustion arrangement, and operation of engines, creating new conditions for their running and setting new tasks. So, our expertise in designing high-pressure fuel pumps allows to recognize several factors that confine their operability range (Figure 1). These include: 1 – performance capability of plain bearings; 2 – ultimate deformations and loss of tightness; 3 – bearing overheating; 4 – long-run serviceability of valves; 5 - filling the volume above the plunger; 6 - the appearance of a dynamic clearance in the plunger drive; 7 - range of injection pressures for optimal combustion conditions. Thus, the operating conditions of a high-pressure fuel pump have many limitations, but taking them into account allows us to design more advanced pumps for ultrahigh discharge pressures.

![Figure 1. Limitations of the operability area of the common rail high pressure fuel pump [10].](image)

The common rail electro-hydraulic injector has its own optimization criteria at high injection pressures: minimum fuel consumption for control, maximum injection pressure at a given rail pressure, minimum cycle fuel mass, monotonic load characteristic, optimized injection profile.

Increased fuel pressures can cause the following defects in the operation of the Common Rail injector: loss of hydraulic balance of the control valve, loss of tightness, unacceptable deformations, loss of strength, loss of valve and needle mobility. High pressures increase the cost of the systems and reduce the resource.

There are also other difficulties, including new aspects of mathematical modeling of processes at elevated pressures, new dependencies for describing the motion and atomization of fuel jets [11]. In addition to the above, many other problems exist when creating fuel systems with ultra-high injection pressure, they lead to additional costs and problems [12].

3. Experimental set-up

Two types of equipment were used for different tests. The installation for static fuel pouring at ultrahigh pressures was created on the basis of a conventional KI71157 fuel test bench with a hydraulic shaft
drive with a power of 18 kW. The unit was equipped with a system with a calibrated diaphragm, a load controller, and the system for measuring fuel before and behind the diaphragm. Installation and loading device are shown in Figure 2.

**Figure 2.** The installation for static fuel pouring and load controller: 1-high-pressure pumps, 2-hydraulic accumulator (Rail), 3-pressure sensor, 4-beakers, 5-potentiometer; 6-orifice, 7-loading device, 8-consumption regulator, 9-electronic control unit, 10–potential amplifier

It is technically challenging to ensure the pressure up to 400 MPa. For the purposes of this paper, high-pressure pumps were manufactured according to original schemes (Figure 3). These pumps used eccentrics with an intermediate sleeve to drive the plungers, fluorine-bronze plain bearings, and rolling bearings. Over the course of several years, the models of high-pressure pumps were improved, as a result, pumps operating at pressures of 380-400 MPa were built.

**Figure 3.** Original high-pressure pumps for high-pressure fuel supply.

The fuel was passed through the orifice with a diameter of 0.12 mm. Chromel-copel thermocouples were located at the inlet and outlet of the chamber with the orifice. The thermocouples were calibrated, the measurement accuracy is no worse than 1.5% at inlet temperatures of 0-200°C. The loading device ensured stable fuel pressure and installation safety. Stationary pressure was measured with a pressure gauge with an accuracy of 1.0 and an amplitude of 400 MPa.

The unsteady pressure in Rail was measured with a 5QP6002 type high pressure piezoelectric sensor (AVL) and a Russian-made T6000 sensor. The high-sensitivity amplifier brand 3056A01 (AVL) had a sensitivity of 0.01-100 V/C and a signal amplification factor of up to 10^5.

Tests of Common Rail electro-hydraulic injectors were carried out using another test bench, which was retrofitted with a special electronic control system with increased capabilities for controlling the parameters of the electrical signal (Figure 4).

Increased frequency of injector firing and a high-pressure pump with a high consumption was used. This reduced the errors associated with neglecting radiative heat transfer.
Original high-pressure pumps and industrial pumps were used in testing various configurations of fuel equipment at elevated pressures of diesel fuel and alternative fuels. The high-pressure pump CP3.4 (R. Bosch) was lubricated with special hydraulic oils with additives.

**Figure 4.** Open system-dieselland 12 PSB+Bench for Common Rail injector testing with high-pressure pump CP3.4.

In this work, the authors tested specially designed electro-hydraulic Common Rail nozzles, as well as several industrial nozzles at a rail pressure of 200-400 MPa. Industrial injectors were prepared: new body parts were manufactured, the pre-tension of the spring of the hydraulically unbalanced control valve was increased, the electric power of the solenoid and piezo actuator increased.

Six Common Rail injectors underwent the testing process, namely: experimental injector with an internal accumulator for a medium-speed engine manufactured by AZPI (Russia); experimental nozzle for a medium-speed engine manufactured by NZTA (Russia); ready-to-use fuel injector DELPHI, model CSD 28237259; ready-to-use injector BOSCH, model CR12-16; ready-to-use injector BOSCH, model CR13-18; ready-to-use injector BOSCH, model CR12-22.

4. **Test results**

The loading characteristics of the AZPI Plant Common Rail injector at various pressures in the Rail in the range of 50-400 MPa are shown in Figure 5. It can be noted that at elevated pressures the load characteristic becomes less favorable (less smooth), but this phenomenon will be considered less important and depends on the characteristics of the given injector.

**Figure 5.** Loading characteristics of the Common Rail injector at various pressures in the accumulator in the range of 50-400 MPa (AZPI Plant injector tests).
More importantly, it has been found that cycle fuel mass stops increasing when rail pressure continues to rise after reaching 280-300 MPa. If we consider that these results are not accidental, then an unexpected conclusion follows from them, that the known trend in the continuous increase in injection pressure is not absolutely correct in terms of increasing fuel consumption and shortening the injection duration. In addition, an increase in pressure above 280-300 MPa leads to the opposite effect - a slight decrease in the mass cycle fuel through the injector. The importance of this phenomenon is confirmed by its repetition for all six tested injectors.

When testing six different new experimental and well-known industrial injectors from different manufacturers and intended for different engines, the common features of their operation were revealed. This attests to the fact that there is non-randomness as to the results obtained.

As shown below, a new effect occurs when the fuel gains the sound velocity in the nozzle throat and in the minimum section of the control valve. The cessation of the increase in fuel consumption when the pressure rises above 300 MPa requires a revision of the mathematical models of fuel supply at increased injection pressures.

To elucidate the physical nature of the phenomenon and find quantitative relationships, tests were carried out at ultrahigh pressures, which showed distinctive features of physical modeling. For this purpose, the fuel spillage was used. This method can be considered a physical simulation of the real injection process. Figure 6 shows typical results obtained with the unit depicted in Figure 2.

![Figure 6. Mass fuel consumption through the orifice of d=0.12 mm during static pouring as a function of supply pressure.](image)

It can be observed that when flowing into the atmosphere, the fuel flow ceases to grow when the pressure exceeds 290-300 MPa.

5. Mathematical model of fuel flow through nozzle holes

The mass consumption $G_m$ is calculated as follows:

$$G_m = (c_a A)_{inject} \sqrt{2\rho(P_{inject} - P_{cham})}.$$  \hspace{1cm} (1)

Here $(c_a A)_{inject}$ means the effective cross-sectional area of the injectors, $\rho$ is the fuel density, $P_{inject}$ is the pressure at the inlet to the injector, $P_{cham}$ is the pressure in the chamber behind the injector (or in the engine cylinder).

It is well known that the flow coefficient $c_a$ decreases as the difference between the injection pressure $P_{inject}$ and the pressure in the injection chamber $P_{cham}$ increases [6]. On the other hand, as follows from Equation (1), the mass consumption $G_m$ should increase with increasing $P_{inject}$. This factor usually prevails.

This reasoning contradicts the experimental results obtained at elevated pressure.
It is proposed to abandon the flow equation for an incompressible fluid (1), which is obtained from the Bernoulli equation for an incompressible fluid. To obtain an equation for the flow rate of an incompressible fluid, an equation of state is required.

The equation of state with the following form demonstrates good results [10]:

$$
\left( \frac{\rho}{\rho_{0t}} \right)^{\kappa} = \frac{B + P}{B}.
$$

(2)

In this equation, $B$ and $\kappa$ are constants for a given type of fluid, $\rho_{0t}$ denotes the density value at atmospheric pressure and initial operating temperature $t$.

Constants $B$ and $\kappa$ can be determined based on specific tests for each fuel type. For diesel fuel, in the simplest case, the following ratios can be used [10]:

$$B = 10^6 \left[ 184 - 0.95(t - 20) + 0.51(\rho_{20} - 825) \right],$$

$$\kappa = 8.0 + 0.004(t - 20).$$

In this equations, $B$ and $P$ are expressed in units of Pa, temperature $t$ is in °C, density $\rho_{20}$, $\rho_{0t}$ are in kg / m$^3$, and $\rho_{20}$ is the fuel density at $t = 20$ °C.

Using equation of state (2), the mass flow equation for fuel as a compressible fluid, can be represented as follows:

$$G_m = (c_{a} A)_{inj} \frac{2\kappa \rho_{0j} \left( P_{inj} + B \right)}{\kappa - 1} \left[ \left( \frac{P_{tham} + B}{P_{inj} + B} \right)^{\kappa} - \left( \frac{P_{tham} + B}{P_{inj} + B} \right)^{\kappa - 1} \right].$$

(3)

In the case of using the flow Equation (3), the calculated fuel consumption at increased injection pressures will be lower than when calculated by Formula (1). However, the flow continues to rise at any pressure upstream of the injector.

Along the way, we present an expression for calculating the sound velocity $\alpha$, which can be obtained using Formula (2):

$$\alpha = \sqrt{\frac{\kappa}{\rho_{0}}} \left( \frac{1}{B^{\kappa} (P + B)^{\kappa - 1}} \right).$$

(4)

The heating of the fuel that flows through the throttle can be calculated using an expression that is derived from the thermal energy balance equation written for an open thermodynamic system:

$$\frac{dI}{dt} = f_i \alpha_{w} (T_{wi} - T_i) + \frac{dQ_v}{dt} + \frac{k}{\rho_i} \frac{dP}{d\tau} + \sum_{k=1}^{K} U_{i,k} f_{i,k} C_p (T_{i,k} - T_i) + \sum_{k=1}^{K} \xi_{i,k} \frac{U_{i,k}^3}{2}.$$

(5)

In this equation:

- $\alpha_{w}$, $f_i$, $V_i$ are heat transfer coefficient, surface, volume of the $i$-th cavity;
- $\rho_i$, $C_p$, $I$, $T_i$ - denote fuel density and heat capacity, enthalpy, temperature, in the $i$-th cavity;
- $f_{i,k}$, $T_{i,k}$, $U_{i,k}$ denote the section for the passage of the fuel, the temperature and speed of the fuel that flows into the $i$-th cavity through the $k$-th section;
- $Q_v$ is the release of heat that occurs in the gas phase of the gas-fuel mixture when its parameters change in the $i$-th cavity;
- $\xi_{i,k}$ is a local hydraulic resistance coefficient; coefficient $\xi_{i,k}$ is adjusted depending on the direction of flow;
- $\tau$ stands for the time value.

6. Discussion

The speed of sound, calculated according to the theoretical Formula (4) for injection pressures up to 400 MPa, can reach values up to 2200 m / s. This value significantly exceeds the adiabatic fuel velocity $U_{ad}$, which can reach 1000 m / s. For these reasons, with the help of popular calculation formulas for the flow
of droplet liquids, it is not possible to describe the situation when the velocity averaged over the consumption in the nozzle holes approaches the local sound velocity.

To estimate the sound velocity value and the actual flow velocity in the holes, and the fuel consumption correctly, it is required to specify the fuel temperature locally. Thermal effects of fuel injection become significant at feed pressure levels above 80-120 MPa. It is possible to estimate the amount of fuel heating $\Delta T$ under conditions of adiabatic compression using Equations (2) and (3). Figure 7 demonstrates these calculation results.

![Figure 7](image)

**Figure 7.** Dependence of heating $\Delta T$ and heating rate $dT/dp$ of diesel fuel at adiabatic compression on the compression pressure.

However, the nature of heating in common rail fuel system cells is more complex. Figure 8 shows the experimental results on fuel heating values obtained during tests on the above-mentioned units of both types.

To solve the Equation (5), one needs to address conjugate problems, hence experimental studies do not reduce in their importance.

On the other hand, as the injection pressure rises, the fuel temperature rises at the injector inlet and even more at the injector outlet (Figure 8). The increased fuel temperature leads to a decrease in the local sound velocity $\alpha$ in the nozzle holes.

This is a very important and new result for fuel equipment. Previously, the prevailing opinion was that with increasing pressure, the density and the modulus of elasticity, and hence the sound velocity, increase. This decrease in the sound velocity can be calculated using Equation (4). Figure 9 shows such results.

Taking into account the fact of fuel heating, it becomes clear that at some high-pressure values, the adiabatic flow rate $U_{\text{adiabatic}}$ increases to the value of the local sound velocity $\alpha$ in the injector holes (Figure 9). The injector orifices are not specially profiled nozzles for supersonic flows. For this reason, when the sound velocity is reached, energy losses increase sharply and the achievement of supersonic flows becomes impossible.

In the section of the injector hole, the fuel velocity changes from maximum values in the flow core near the center of the section to negative values in the boundary layer separation zones at the walls of the hole. Therefore, the consumption coefficient $c_a$ in Formulas (1), (3) usually has values of 0.65-0.7 for drilled holes, and the average consumption velocity $U_{\text{actual}}$ is lower than the adiabatic fuel velocity $U_{\text{adiabatic}}$.

The described picture of the phenomenon contributes to the understanding of the result obtained in Figure 5. The fuel consumption and the speed averaged over the consumption through the nozzle holes increase with an increase in the injection pressure according to relations (1) and (3). This dependence is illustrated on the left side of the graph (Figure 9). When the pressure reaches a level of 290-310 MPa, the local flow velocity into the nozzle holes reaches the sound velocity and the increase in fuel consumption stops. Moreover, as a result of increasing losses and further heating of the fuel,
the sound speed and the average fuel consumption speed begin to decrease (the right side of the graph in Figure 9).

![Figure 8](image1.png)

**Figure 8.** Dependence of diesel fuel heating $dT$ on the pressure: calculated approximation of the experimental data, experiments with an injector CR BOSCH, experiments with orifices.

![Figure 9](image2.png)

**Figure 9.** Local sound velocity $c_{\text{sound}}$, adiabatic flow velocity $U_{\text{adiabatic}}$ and actual consumption-average velocity $U_{\text{actual}}$ as a function of discharge pressure.

Thus, in order to increase the instant fuel consumption and reduce the duration of the injection, increasing the injection pressure more than 290 ... 310 MPa is irrational [13]. This does not apply to the conclusion about the possibility of further reducing the droplet diameter and improving other parameters of the fuel jet; additional research is required in this issue.

In mathematical modeling of the processes in the fuel equipment, it is recommended to use Equations (1) or (2) with control by the sound speed determined by Equation (4). It is necessary to abandon the previously used concept of the isothermal process of fuel flow in the channels of the fuel equipment. It is required to take into account the heating of the fuel according to the dependences of adiabatic compression (Figure 7) and heating from the flow through the throttling sections, for example, using relation (5).

Incorporating the terms of reduction of mass fuel flow through the relevant nozzle makes the calculations of fuel injection more correct. Figure 10 presents field fuel cycle mass of the considered injector AZPI as a function of the rail pressure and the control signal duration. The experimentally
determined value of fuel cycle mass is in figure as well. Error calculations for the range of pressures above 300 MPa has the same level as for the traditional, lower pressure injection.

Figure 10. The fuel cycle mass of the considered injector AZPI as function of the rail pressure and the control signal duration.

7. Conclusions
The average diesel fuel consumption speed reaches 560-580 m / s at an injection pressure above 290-300 MPa and stops growing with a further increase in pressure. The duration of the injection can no longer be shortened with a further increase in the injection pressure.

It is necessary to take into account that the fuel is heated in the channels of the fuel equipment. Heating the fuel decreases the sound velocity and leads to the achievement of the injection rate limitation. Mathematical modeling of hydrodynamic processes in fuel equipment should take into account fuel heating during flow in channels, cavities and throttles.

Injection intensification has well-known capabilities, but results in increased power consumption of the high pressure fuel pump, leakage problems, and increased complexity and cost of fuel systems. Therefore, an increase in the injection pressure above 300 MPa requires additional justification.

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