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DIESEL FUEL INJECTION BY PRESSURE UP TO 400 MPa

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Abstract

High injection pressure is one of the resources for diesel engines characteristics improvement. Required injection pressure depends on the engine's parameters and application and is still debatable. Peculiarities of diesel fuel injection under the pressure above 300 MPa are investigated in this research. Rational level of injection pressure increase has been defined and a new advanced method for hydrodynamic simulation of ultra-high pressure fuel injection has been proposed.

Keywords: Low-emission diesel engine, Fuel system, Common rail, Injection pressure, Fuel flow, Fuel heating.

Introduction

Intensification of fuel injection is an efficient method of reducing harmful emissions and fuel consumption in diesel engines. Some authors even propose averaged recommendations on how injection pressure should be changed to achieve better diesel performance [1].

Nowadays in serial fuel systems the injection pressure has achieved 250 MPa (Scania XPI, Bosch CRSN3-25); there are a few publications with plans to create fuel systems with 300 MPa injection pressure (Delphi, Denso).

Experimental studies suggest that it is possible to reduce harmful emissions with increasing injection pressure up to 320 MPa [2, p. 6]. However the findings are ambiguous for different nozzle configurations as well as for reduction of NOx emission.

The current trend of increasing injection pressure does not give rise to doubt; however, the injection pressure target values remain a disputable matter. There is only one scientifically reasonable method of injection pressure selection it is optimization of the engine working process. The experimental solution of this problem is expensive and time-consuming. The authors of this research have developed a software to solve these problems by methods of mathematical modeling [3, 4].

Attempting to reduce the harmful emissions in exhaust gases and the fuel consumption by increasing fuel injection pressure, one would face a number of limitations.

Our experience in designing the fuel injection pumps allows recognition of several factors which bound the pump's workability region (Figure 1). These include: 1 – sliding bearing workability; 2 – ultimate strains and lost of leaktight ness; 3 – overheat of the bearing; 4 – long term workability of valves; 5 – above plunger (in-barrel chamber) volume filling; 6 – breaking kinematic links in the plunger drive; 7 – bounds of the optimal injection pressures required by engine working process. So the pump design is becoming more sophisticated with improved performance and increased injection pressure.

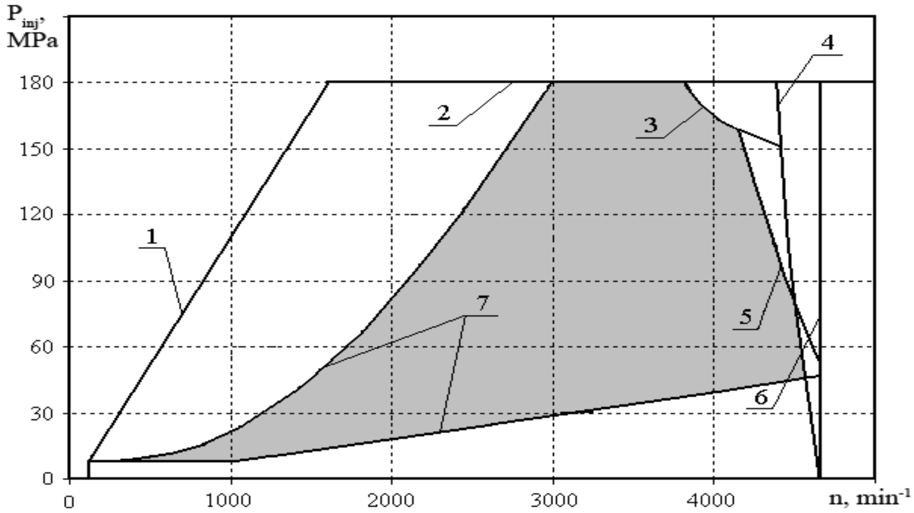

Figure 1: Common Rail fuel pump operability range [5, p.105].

Fuel injection through the Common Rail Electrohydraulic Injector (EHI) under high pressures entails higher requirements to EHI perfection: minimized fuel consumption to control, maximum p_{inj}/p_{acc}, fast acting control valve, and optimized injection profile.

Operation of the injector at high fuel pressures may be disrupted due to the following reasons: loss of operability of unbalanced control valve, lost of leaktightness, unacceptable deformations, loss of strength. The cost of these systems is higher and service life is shorter.

There are other complications, such as personnel safety problems, difficulty in assuring complete leak tightness of the fuel system fittings (seals in joints), unknown mathematical relationships for simulation of fuel flow and atomization parameters. The list of problems with creating fuel systems with higher injection pressures can be supplemented with extra cost of these systems [6, p.2].

In current research there were tested at rail pressure up to 400 MPa specially developed experimental EHIs as well as some commercially available injectors [7]. The latter had to be modified by increasing the spring pretension of the hydraulically unbalanced control valve, forcing the power input of the electromagnetic and piezoelectric drive, and manufacturing new body parts. Figure 2 illustrates typical test results.

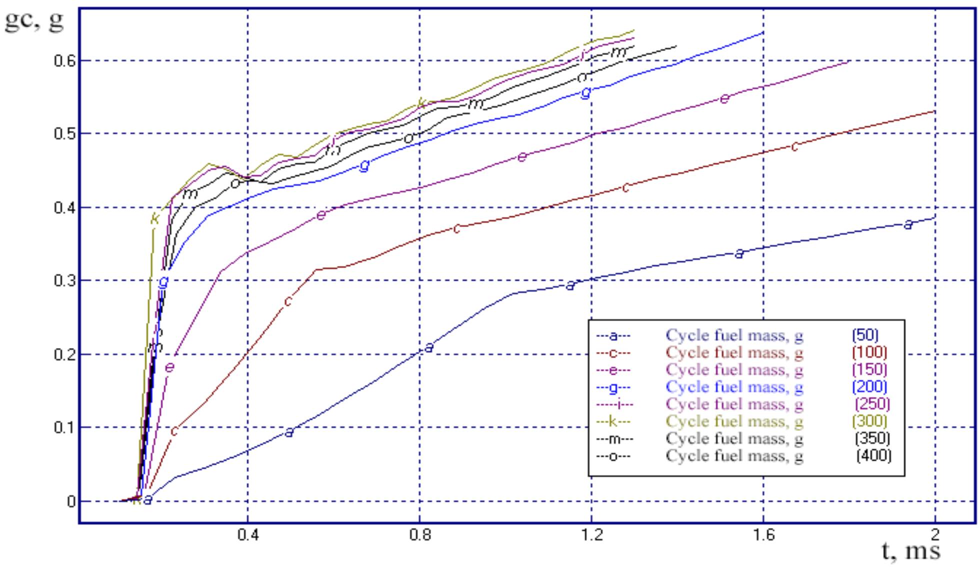

Figure 2: Cycle fuel mass as a function of control pulse time at rail pressures of 50 … 400 MPa.

It was determined that the cycle fuel mass stops to increase when the rail pressure keeps rising after reaching 280…300 MPa. Therefore, any further increase of injection pressure can no longer reduce the injection time and is most probably unpractical. Moreover, increasing the pressure above 280 MPa leads to a certain reduction of the fuel flow rate of the nozzle. Additionally, with higher pressures, the fuel consumption being necessary for control increases, but this phenomenon is more dependent on the injector's design, i.e. on presence and dimensions of additional orifices of control valve.

As we will demonstrate below, the determined effect appears when the fuel achieves the speed of sound in the minimum section of spray holes and control valve. It means a new mathematical model is needed to simulate the processes and design nozzles for injection pressures above 300 MPa. The process analysis and the new mathematical model for simulation are presented below.

Mathematical model

The mass injection rate is given by:

$$\dot{m}_{inj} = (\mu\, A)_{nozl} \sqrt{2\rho(p_{inj} - p_{cyl})}\,, \qquad (1)$$

where $(\mu\, A)_{nozl}$ denotes the actual (effective) cross section area of the injector nozzles, ρ denotes the fuel density, p_{inj} is the pressure in the injector chamber (nozzle sac), and p_{cyl} is the pressure in the chamber into which the fuel is injected (engine cylinder).

It is known: the more injection pressure (or the more pressure drop in orifice), the less discharge coefficient. However, corresponding with equation (1) which is derived from Bernoulli's equation for incompressible fluid, the $\dot{m}_{inj}$ grows as p_{inj} increases.

This does not agree with the experimental results obtained at very high pressure. If we use equation for mass flow written for fuel as compressible fluid, the $\dot{m}_{inj}$ can be presented as follows:

$$\dot{m}_{inj} = (\mu\ A)_{nozl}\sqrt{\frac{2\aleph\rho_{inj}\left(p_{inj}+B\right)}{\aleph-1}\left[\left(\frac{p_{cyl}+B}{p_{inj}+B}\right)^{\aleph}-\left(\frac{p_{cyl}+B}{p_{inj}+B}\right)^{\frac{\aleph+1}{\aleph}}\right]}, \qquad (2)$$

where B and $\aleph$ are the constants in equation of state of compressible fuel [5, 7]:

$$\left(\frac{\rho}{\rho_{0t}}\right)^{\aleph} = \frac{B+p}{B}, \qquad (3)$$

where ρ_{0t} is a fuel density at atmospheric pressure but at actual temperature t.

For example, in simplest calculations the expressions for constants B and $\aleph$ can be presented as follows (the pressure p is in Pa, the density in kg/m^3, and temperature t in °C) [5, 8]:

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

$$\aleph = 8.0 + 0.004\left(t-20\right),$$

where ρ_{20} is fuel density at t = 20^{0}C.

In this case the value of the calculated fuel flow at high pressures will be lower. But there is no cessation in the increase of flow at increasing pressure.

Using (3), the speed of sound α can be defined as:

$$\alpha = \sqrt{\frac{\aleph}{\rho_0}B^{\frac{1}{\aleph}}\left(p+B\right)^{\frac{\aleph-1}{\aleph}}} \qquad (4)$$

where ρ_0 is fuel density at atmospheric conditions in kg/m^3.

The speed of sound calculated from (4) for the pressure range of 0 up to 400 MPa is varied between 1400 and 2200 m/s. This is considerably higher than even the adiabatic (theoretic) fuel velocity U_{ad} in the orifice. The U_{ad} ranges between 0 and 1000 m/s. Therefore, within the classical methods of fuel flow simulation there is no possibility to describe situation when the flow velocity in the nozzle hole will achieve local speed of sound.

On another hand, when the fuel is injected under pressures above 100 MPa a non-isothermicity of the process becomes evident. The authors of this research have done estimation of fuel heating ΔT in adiabatic compression conditions with equations (2) and (3) [5]; the results are presented in Figure 3. This simulation (theory) data are confirmed by the measurements of instantaneous temperatures in non-stationary fuel injection process [7].

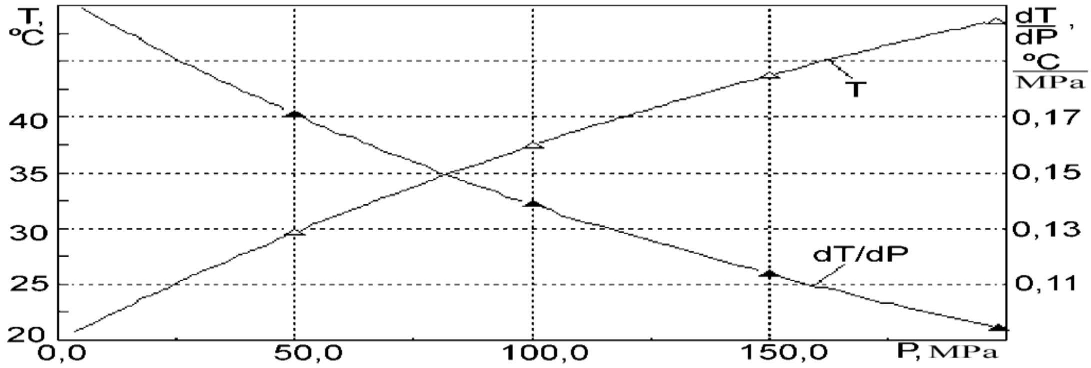

Figure 3: Diesel fuel heating ΔT and its rate *dT/dp* due to adiabatic compression in fuel injection process

Results and Discussion

However, considerably greater fuel heating was experimentally determined at pressures up to 350 MPa. The tests were conducted to determine the temperature of the fuel flowing out of orifice having bore 0.1 mm as well as for fuel flowing out of the experimental Common Rail nozzles. In both cases, the injection was into the atmosphere. The experimentally obtained data of fuel heating vs. injection pressure p_{inj} are presented in Figure 4; latter these data were used in simulations.

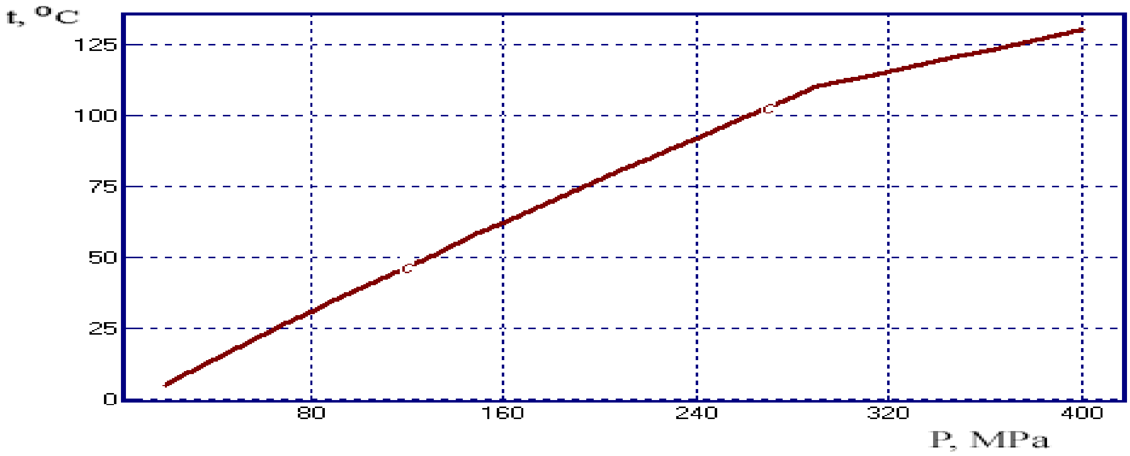

Figure 4: Approximation of experimental data for the heating of diesel fuel ΔT flowing out of nozzle orifices to the atmosphere.

Heating of the fuel supplied through a nozzle can be calculated from the expression, which was derived from the energy balance equation for an open thermodynamic system:

$$\frac{dI_i}{d\tau} = \frac{f_i \alpha_{wi}\left(T_{wi} - T_i\right) + \frac{dQ_v}{d\tau}}{\rho V_i} + \frac{1}{\rho_i}\frac{dp_i}{d\tau} + \frac{\sum_{k=1}^{k=K} U_{i,k} f_{i,k} C_{Pi}\left(T_{i,k} - T_i\right) + \sum_{k=1}^{k=K} k_{dir}^{i,k} \xi_{i,k} \left|U_{i,k}^3\right| \frac{f_{i,k}}{2}}{V_i}, \quad (5)$$

where: V_i, f_i, α_{wi} are the volume, surface area and heat transfer coefficient of *i*-volume; I_i, T_i, C_{pi}, ρ_i are enthalpy, temperature, heat capacity at *p=Const* and density of fuel in *i*-volume; $f_{i,k}$,$U_{i,k}$, $T_{i,k}$ are flow area, velocity and temperature at fuel inflow into *i*-volume trough *k*-orifice; Q_v is a heat of processes taking place in gas phases in fuel in *i*-volume; $\xi_{i,k}$ is a factor of local hydraulic losses; $k_{dir}^{i,k}$ is a coefficient accounting direction of flow; □□ is a time.

Solution of equation (5) requires solutions of conjugate problems, so importance of experimental researches is not reduced.

Taking into account the rise of fuel temperature due to the rise of injection pressure (fig 4), it turns out that the change of fuel state leads to a reduction of the local speed of sound Alpha. This is an important result because□□ is usually expected to rise if pressure increases. Such a reduction is described in equation (4) and plotted in Figure 5.

In this case, it becomes obvious that at certain values of high pressure the adiabatic flow velocity U_{ad} may reach the local speed of sound *U* in the nozzle orifice (Figure 5). Since the nozzle hole is not profiled to enable supersonic flows, the flow velocity is limited to the speed of sound due to great losses.

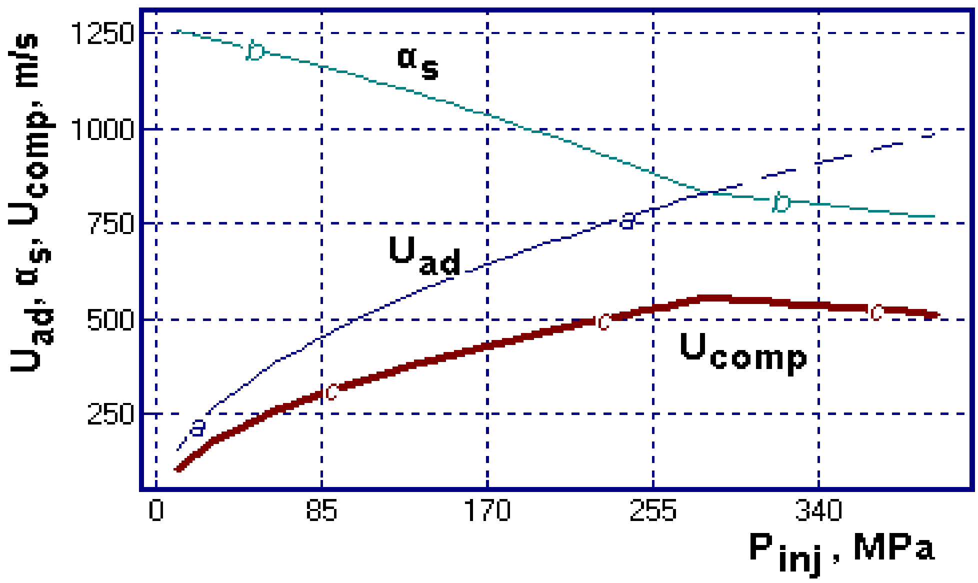

Figure 5: Dependency of adiabatic flow velocity U_{ad}, local speed of sound *U*, and actual mean flow velocity U_{act} vs. injection pressure

In the nozzle hole section the velocities vary widely: from negative values in the zones of boundary-layer separation to maximum values in the flow core. As a result, the discharge coefficient μ in equations (1) and (2) is below 1 (generally 0.65 to 0.7 for drilled spray nozzles with injection pressures over 5 MPa) and the average fuel flow velocity U_{act} is below the adiabatic rate.

These findings help to understand the key final result: at the high injection pressure the average flow velocity and the fuel flow rate through the nozzle hole stop to be grown vs. injection pressure (Figure 5). They explain why the throughput capacity of Common Rail nozzles ceases to grow as the accumulator fuel pressure increases (Figure 2).

With injection pressures over 280…300 MPa the design flow velocity calculated with the incompressible fluid formulae will reach the local speed of sound. Increasing pressure above these values would be unnecessary from the standpoint of flow velocity. It remains unclear whether a further increase of pressure can facilitate atomization.

In hydrodynamic simulation of fuel systems it is advisable to use equations (1) or (2) with reference to the speed of sound determined with equation (4). It is important to remember that the physical parameters of fuel have to be determined for a non-isothermal process, e.g. using the trend in Figure 4.

Conclusions

- Average flow velocity of diesel fuel achieving 560…580 m/s at pressure above 280 MPa, ceases to increase. Further increase of injection pressure is unpractical from the hydraulic standpoint.
- It is necessary to consider the thermal conditions of fuel injection for atomization and delay of ignition prediction. Hydrodynamic simulation of a fuel system should be carried out with account the non-isothermicity of the process.
- Increasing pressure leads to higher power outlay of the fuel pump drive, technical difficulties, unreliability, and higher costs of fuel injection system.
- From the standpoint of engine working process optimization and harmful emissions there is no evidence of practicality of increase in injection pressure above 280…320 MPa.
- Further research is necessary to study the peculiarities of fuel atomization at ultra-high injection pressures.

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