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Hydrodynamics of the Injection Equipment

Abstract: An additional expansion of fuel from the area under the nozzle needle seat into the combustion space considerably influences HC, CO and PM formation. The mentioned expansion can be limited by the injection process optimization. One element of the fuel system playing a significant role in the injection process - its hydrodynamics, is a delivery valve of the injection pump fulfilling also the role of relieving the high-pressure area.

The paper deals with hydrodynamics of the injection equipment with a multi-cylinder injection pump. Achieved results serve as input parameters for a subsequent model and experimental solution of the influence of observed parameters on economic and ecological parameters of diesel engines.

Key words: diesel engine, multi-cylinder injection pump, isovolumetric and equal pressure delivery valves, fuel delivery timing

1. Introduction

The environment includes a wide range of influences having impact on human beings. The question of disease occurrence as a consequence of combustion engines operation is also significant. The harmful impact of combustion engines operation lies not only in chemical pollution of the environment but also in physical effects (noise, vibrations). The injection equipment of combustion engines contributes considerably to both chemical and physical pollution of the environment.

From the point of view of formation and preparation of fuel mixture for direct injection diesel engines, the decisive influence is attributed to fuel delivery timing and to equal and fine fuel spray.

The atomization of fuel spray changes during injection. The arithmetic mean of fuel droplets characterizing the fineness of atomization changes with the needle lift. It is advantageous for the gradient of the nozzle needle opening and closing to be maximum. An additional expansion of fuel from the area under the nozzle needle seat – Fig. 1 – to the combustion space is greatly significant for HC, CO and PM formation. The additional fuel expansion in the mentioned area can be limited by the injection process optimization.

Time course of fuel delivery, fineness and equality of fuel atomization depend on constructional parameters of the engine and individual parts of its injection equipment, its hydraulic characteristics and operational factors.

One element of the fuel system which plays an important role in the injection process, namely its hydrodynamic, is the injection pump delivery valve fulfilling also the function of the high-pressure area relief.

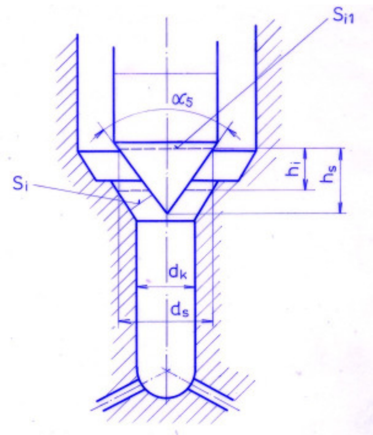

Fig. 1 Area under the nozzle needle seat

The paper deals with hydrodynamics of the injection equipment with a multi-cylinder injection pump focused on the delivery valve motion. Achieved results serve as input parameters for subsequent model and experimental solution of the influence of observed parameters on economic and ecological parameters of diesel engines.

2. Definition of the mathematical model

The set of further presented non-linear differential equations, a partial differential equation of second order and algebraic equations describing hydrodynamics and dynamics of phenomena of the system – Fig. 2, were solved by the iteration method. Losses due to fuel leak, changes and permitted error in individual magnitudes calculation were taken into consideration at solution.

If the elementary amount of fuel delivered to the engine cylinder results from the relation

$$dQ = \alpha_v S_v \rho \left(\frac{2}{\rho} \left| p_{t1} - p_{pl} \right| \right)^{0.5} dt \qquad (1)$$

and the relation between fuel delivery cross sections is given by the relation

$$\alpha_i S_i = \frac{\alpha_v S_v \alpha S}{\left[(\alpha_v S_v)^2 - (\alpha S)^2\right]^{0.5}} \quad , \qquad (2)$$

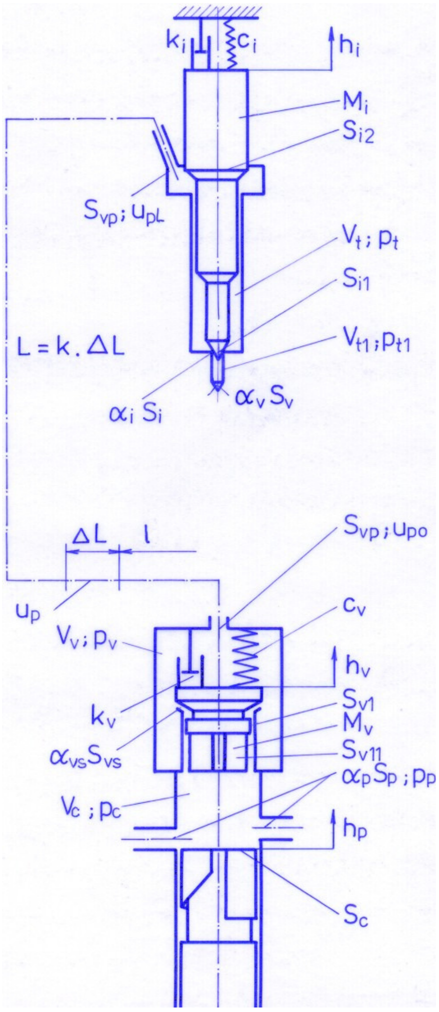

Fig. 2 Scheme of system with used signs

then the relation between the fuel pressure in the area under the needle seat and above it – which determins a possible additional expansion of fuel from the area under the needle seat into the combustion space results from the equation of motion of the nozzle needle - equation (3)

$$M_i \ddot{h}_i + k_i \dot{h}_i + c_i h_i - p_{t1} S_{i1} - p_t (S_{i2} - S_{i1}) = 0 \, .$$

Individual members of this equation express:

- inertia force of moving fuel injection parts,
- friction force in the needle guide,
- force from the injector spring pre-stress,
- force excited by pressing the spring at the needle motion,
- force from fuel pressure p_{t1},
- force from fuel pressure p_t.

The elementary amount of fuel delivered to the engine valve depends on (members of the right hand side of equation 4):

- amount of fuel delivered through the needle seat,
- volume released by the needle cone,
- changes in fuel volume due to its compressibility

$$\alpha_v S_v \left(\frac{2}{\rho} \left| p_{t1} - p_{pl} \right| \right)^{0.5} = \qquad (4)$$
$$= \alpha_i S_i \left(\frac{2}{\rho} \left| p_t - p_{p1} \right| \right)^{0.5} - \dot{V}_i - \frac{V_{t1}}{E} \dot{p}_{t1} \, .$$

Fuel leak due to untightness in the needle guidance is given by the relation

$$U_i = \pi d_i \left[(p_t - p_a) s_i^3 (12 \nu l_i)^{-1} + 0.5 \dot{h}_i s_i \right] . \qquad (5)$$

Fuel flow from the high pressure manifold to the nozzle chamber influences the pressure in this space, which can be expressed by dependence (6)

$$S_{vp} u_{pL} = \frac{V_t}{E} \dot{p}_t + S_{i2} \dot{h}_i + \qquad (6)$$
$$+ \alpha_i S_i \left(\frac{2}{\rho} \left| p_t - p_{p1} \right| \right)^{0.5} - \dot{V}_i + U_i \quad ,$$

in which the individual members refer to:

- volume of fuel delivered through the manifold,
- change in volume due to compressibility,
- volume released by the needle motion,
- volume delivered through the needle seat,
- volume released by the needle cone,
- fuel leakage due to untightness.

Nonstationary fuel flow in the high pressure manifold is described with the partial differential equation

$$\frac{\partial^2 u_p}{\partial l^2} - a^{-2} \frac{\partial^2 u_p}{\partial t^2} - \frac{2K_p}{a^2} \frac{\partial u_p}{\partial t} = 0 \; , \qquad (7)$$

in which the coefficient of hydraulic resistance of the manifold K_p depends on the fact whether the flow is laminar or turbulent.

Fuel volume at the high pressure manifold intake is given by the relation

$$S_{vp} u_{p0} = -\frac{V_v}{E} \dot{p}_v + S_{v1} \dot{h}_v + \qquad (8)$$
$$+ (\alpha_{vs} S_{vs} + \alpha_r S_r) \left(\frac{2}{\rho} \left| p_c - p_v \right| \right)^{0.5} ,$$

in which the members on the right hand side refer to:

- influence of fuel compressibility in the delivery valve chamber,
- volume taken by the delivery valve,
- volume delivered through the valve seat – Fig. 3,
- volume delivered through the relief hole (for equal pressure valve).

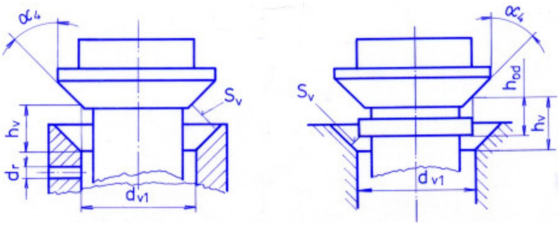

Fig. 3 Equal pressure and isovolumetric valves

When fuel leakage due to untightness in the pump (through the charging and relief holes and radial slot in the piston guide) is considered, then the non-linear differential equation (9) holds for the flow of fuel through the delivery valve seat and relief hole

$$(\alpha_{vs}S_{vs}+\alpha_r S_r)\left(\frac{2}{\rho}|p_c-p_v|\right)^{0.5} = S_c\dot{h}_p - S_{v1}\dot{h}_v - \frac{V_v}{E}\dot{p}_v - \alpha_p S_p\left(\frac{2}{\rho}|p_c-p_v|\right)^{0.5} - U_c$$

in which the members on the right hand side refer to:

- volume delivered by the pump valve,
- volume released by the delivery valve motion,
- change in the volume due to compressibility,
- volume flown through the charging and relief holes,
- fuel leakage due to untightness.

The connection between pressure in the pump space and delivery valve (which consequently influences the pressure of fuel in the area under the needle seat, and thus influences a possible additional expansion of fuel from the area under the needle seat into the combustion space) results also from the equation of motion of the delivery valve

$$M_v\ddot{h}_v + k_v\dot{h}_v + c_v h_v + c_v h_0 = S_{v1}(p_c - p_v) \quad (10)$$

3. Definition of the experiment

The experiments were carried out on a test bench of the injection equipment with a multi-cylinder injection pump equipped with isovolumetric and equal pressure delivery valves (Fig. 3) and a hydraulically controlled multi-hole injector.

During the experiment the following quantities were recorded: lift of the delivery valve and nozzle needle, pressures of fuel in the areas observed for a specific amount of fuel in dependence on pre-stress of the delivery valve spring and pump revolutions.

For the needs of experiment some special sensors enabling to scan observed parameters had to be designed and manufactured. To mention at least one of them: a multiple sensor to detect the delivery valve lift and fuel pressure in the delivery valve chamber – Fig. 4. In this case the delivery valve was adjusted as well. Due to the adjustment its weight increased approx. by 15 %. The consequent increase of inertia force (approx. by 5 N) does not influence the course of lift as other forces acting on the valve are considerably higher (from fuel pressure approx. 13.5 N and from the spring compression approx. 20 N).

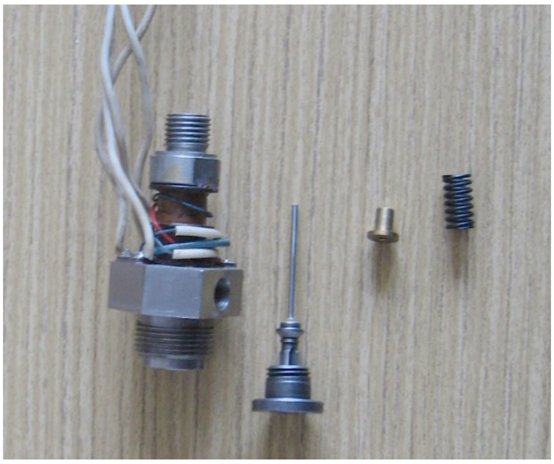

Fig. 4 Adjustment of delivery valve and its body for scanning its motion and fuel pressure in its chamber

Fig. 5 presents one recording of the course of measured quantities. From such recordings the experiment was subsequently assessed and achieved results were compared with results of mathematical modeling.

4. Obtained results

An objective of mathematical modeling and calculation was to judge the influence of considered phenomena on the operation of the isovolumetric and equal pressure valves. Consequently, the obtained results will be used for solution of injection process, fuel mixture preparation, burning and emission formation.

From results obtained from calculation and experiment the following can be seen:

- differences in the course of pressure changes in the charging chamber of the pump with isovolumetric and equal pressure valves are not expressive,

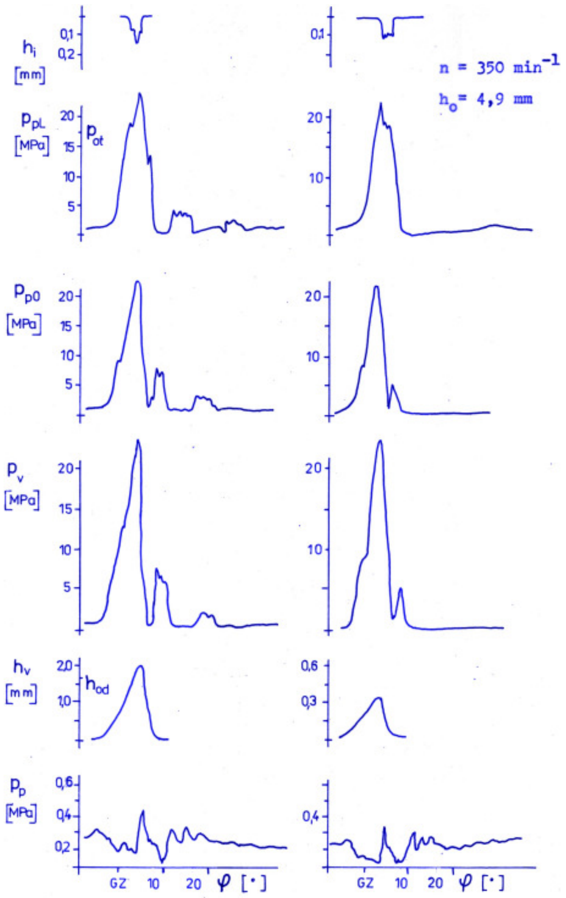

Fig. 5 Record of measurements for delivery valves: isovolumetric - left and equal pressure – right

- level of pressure in the delivery valve chamber decreases with decreasing revolutions. The pressure level for an equal pressure valve is on the average by 1 – 2 MPa lower that the one for an isovolumetric valve. Differences between measured and calculated values of maximum pressure are 1.5 – 2 MPa,
- in the high pressure part of the injection equipment with an equal pressure delivery valve more time is needed for silencing pressure waves after fuel relieving is over than for the system with an equal pressure delivery valve,
- initial compression – pre-stress of the delivery valve spring causes an inexpressive change in pressure and secondary pressure wave in the delivery valve chamber,
- in the area of idle run the value of maximum injection pressures is relatively low 1.25 – 1.5 multiple of the opening pressure,
- maximum value of the delivery valve lift increases with increasing revolutions, while this increase is approx. by 25 % higher for the equal pressure valve than for the isovolumetric valve,
- amount of pre-stress of the delivery valve spring does not significantly influence the value of its maximum lift,

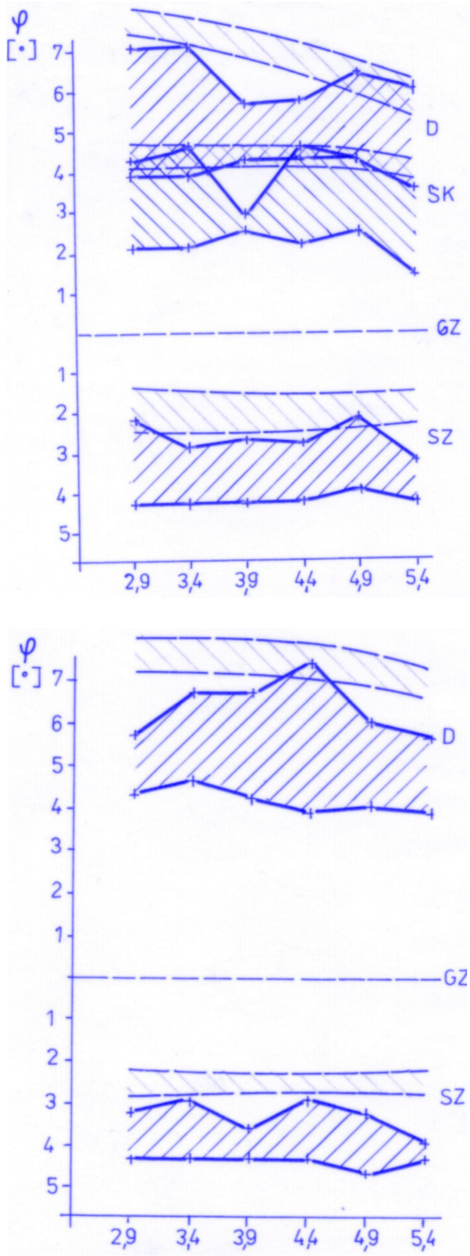

Fig. 6 Delivery timing for injection equipment with an isovolumetric delivery valve - upper part and the equal pressure valve – lower part in dependence on initial compression of valve spring (dashed line –calculation, solid line – experiment)

- shape of lift curve of the isovolumetric valve changes significantly neither with the change of revolutions nor with the change of pre-stress of the spring,

- shape of the lift curve of the equal pressure valve changes with the change of revolutions,
- lift of the nozzle needle is for both systems (equal pressure and isovolumetric relief) approximately equal, although there are differences in its course. For the system with the equal pressure delivery valve the needle does not show such oscillation as for the system with the isovolumetric valve. This fact is subsequently manifested in the change of time needed for fuel mixture preparation,
- change – increase in advance of real fuel delivery beginning at increasing revolutions compared with geometric advance is less expressive for the equal pressure valve system than for the isovolumetric valve system,
- change of the initial pre-stress of the spring does not substantially influence the advance of real fuel delivery beginning – Fig. 6; the calculated advance delivery beginning compared with the geometrical one is smaller than the one obtained experimentally,
- change of revolutions and change of the spring pre-stress do not significantly influence the real end of fuel delivery,
- seating of the valve into the seat with increasing revolutions will occur for both types of delivery valves at greater angles of the camshaft,
- increase of the spring pre-stress causes a partial decrease of angle of valve seating into the seat,
- change of revolutions and change of the spring pre-stress will be manifested more expressively in delivery timing for the pump equipped with the isovolumetric delivery valve.

4. Conclusion

Increasing requirements put on mostly ecological parameters of diesel engines are to a great extent reflected also in requirements put on fuel mixture preparation, i. e. also on parameters of injection equipment. Realization of such requirements requires a detailed knowledge of processes taking place in the injection equipment.

The article is a contribution to the solution of the above mentioned problem because it deals with it not only in a way of modeling but it also describes the behavior of the delivery valve during the experiment.

Nomenclature

D	valve seating into the seat
GZ	geometric beginning of fuel delivery
SZ	real beginning of fuel delivery
SK	real end of fuel delivery

other signs can be understood from figures

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