

## NUMERICAL AND EXPERIMENTAL ANALYSIS OF INJECTION PROCESS IN DIESEL ENGINES FOR SUPPLY SYSTEM EQUIPPED WITH IN-LINE INJECTION PUMP

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**Summary.** In the article the mathematical simulation of the injection process run with in-line injection pump is presented. The verification of the model was done in correlation with the results obtained on a real engine. An analysis of represented results of the mathematical simulation gives good reason to ascertain that results correctly reproduce character of courses.

**Key words:** mathematical modelling, injection process run, in-line injection pump, experimental verification

### INTRODUCTION

The utility and ecological parameters are largely determined by the combustion rate, characterized by the process of heat release and pressure change. The combustion course causes some specific results of both physical and chemical character. The physical effects are, first of all, the quality of fuel energy release and economy of its utilization, quantity of dynamic loads of crank mechanism and work noise level. Chemical results of the combustion refer mostly to the composition of exhaust gases, especially to smokiness and contents of toxic matter. The significance of the chemical results in the development of compression-ignition engines is very high at the moment, with regard to the environmental protection requirements.

The technical, economic and ecological parameters of a diesel engine, as it was mentioned above, depend on the combustion process course. The combustion process efficiency is mostly conditioned by the quality of fuel-air mixture preparation. From among many factors influencing these processes, and connected immediately with the injection system, the following ones should be specified:

- beginning, duration and rate of injection fuel dose,
- pressure and speed of fuel injection,
- location of fuel jet in the combustion chamber.

One of the set of basic parameters describing the fuel injection course is the characteristic injection time. An analysis of the injection course as time function (or crank

angle function) permits for a precise recognition of the phenomena proceeded in a high-pressure apparatus, so it can be properly designed or made-up from the already existing units and elements for a chosen engine. The so far existing methods of preparation of the injection apparatus for a given engine were most often connected with prolonged and expensive experimental investigations. The shortening of the optimum solutions obtainment time can be achieved by the creation of mathematical models of injection systems and calculation of interesting parameters in the numerical simulation way.

#### A DESCRIPTION OF A HYDRAULICALLY CONTROLLED INJECTION

The main problem in a diesel engine is an achievement of a compromise among exhaust cleanness, values of effective work parameters and fuel-efficiency. Besides, the mechanical load and work noise level should be also considered. The obtainment of the complete combustion depends first of all on the efficiency of the used system in the forming of a flammable mixture. The characteristic injection course time is a decisive factor here. From the injection characteristic it is required above all that:

- injection velocity at the beginning of the process be at once properly high (making theoretical assumption that wave influences don't occur and the fuel pipes are not deformable, then the injection characteristic would have a rectangular form),
- the rate of fuel injection do not have a pulsating character (pulsating feed of fuel induces pressure fluctuation in the combustion chamber, causing „hard” combustion and stopping fuel inflow to the injector in consequence of vapour-locks arising as a result of indentation),
- after the main injection additional injections of fuel do not appear (out-of-control trickled fuel from sprayer does not burn completely causing an increase of concentration of toxic components and also carbon deposit settles in sprayer holes which deteriorates the spraying quality).

In the so far generally used conventional fuel systems the course of injection process is controlled in a very narrow range. As regulation parameters there are the fuel dose and, in small size, injection advance angle changed by centrifugal governor. However, the requirements determined for modern compression-ignition engines are higher and higher. Assurance of the optimum engine parameters causes the necessity of elaborating a new fuel system. A relatively simple and so far effective way of reaching an improvement of these parameters is an application of hydraulically controlled injection. Modification of injection characteristic is obtained mostly by constructional changes in units and elements of conventional injection apparatus. The following practical methods belong to the most frequent in this range:

- graded injection,
- double fuel supply,
- two-phase fuel injection.

A graded injection is the subject of the present sub-chapter. This method is grounded on the fact that dynamic coefficients of the combustion process immediately depend on the volume of fuel injected to the cylinder during the time of self-ignition

delay. For a given duration of this period the fuel volume should be diminished, so that the speed of heat release in the first phase of combustion does not exceed permissible limits. It was fixed, so that in the average operating conditions of an engine, the injected fuel dose in the period of self-ignition delay should not exceed  $6 \text{ mm}^3$  on 1 litre of engine capacity. Then enough gentle growing of pressure in the first combustion phase is received, and its further course depends on the fuel injection speed.

The practical realization of the graded injection can be received by many solutions. In the engines with direct injection the pump camshaft with two-lobe cams is the most often used. In the high-speed engines (with divided combustion chamber) this effect is obtained by injecting of the fuel with a special nozzle. Thanks to the suitable shaping of the needle tip during the initial lift, a choking of fuel outflow takes place, which diminishes the fuel dose injected into the cylinder in the period of self-ignition delay. However, with regard to short duration of the needle lift time, the influence of a so controlled injection on the combustion is too light, if an engine's operation at different rotary speeds and loads is considered.

#### MATHEMATICAL MODEL – BASIC EQUATIONS

A fuel injection system equipped with an in-line injection pump was analysed. The draft of the analysed system is presented in Fig. 1. Calculations are based on method of „moving volumetric element”. The essence of this method is, that an introduction of the new notion allows to use the same equation for describing the phenomena occurring in the injection system in all conditions – both for compressed fuel and fuel being under vacuum. In this manner the possibility was obtained of an immediate determination of indentation spaces, which is very significant in high-pressure systems. Simulation works of different fuel injection systems with the „moving volumetric element” method were carried out in the Department of Automotive Vehicles and Combustion Engines at the Rzeszów University of Technology. This method is still being improved by introducing other variables into the model, so that the actual phenomena are described more exactly. Numerous papers have been published in the Department concerning this subject, presenting the results on many domestic and foreign conferences. For an elaboration of theme being the subject of the present paper, i.e. the following works [Lejda 1992, 1994, 1996, 1997] are used.

While analysing the fuel flow in whole system, the derivation of the flow formulas should be begun from the section pressure (Fig. 1). Fuel volume corresponding to time unit, pumped from the pressure section by moving plunger  $F_k C_k$  is equal to the sum of time derivative of the „moving volumetric element”  $dY_k/dt$  and flow flux from pressure section  $\mu_z F_z C_z$ :

$$F_k C_k = \frac{dY_k}{dt} + \mu_z \cdot F_z \cdot C_z \quad (1)$$

After exchanging the finite differences and transformations we receive an equation for the determination of a „moving volumetric element” in the pumping chamber of the following form:

$$Y_k = D_k - k_z \cdot \text{sign}(P_k, P_z) \sqrt{|P_k - P_z|} + F_k (h_k - h_{k1}) \quad (2)$$

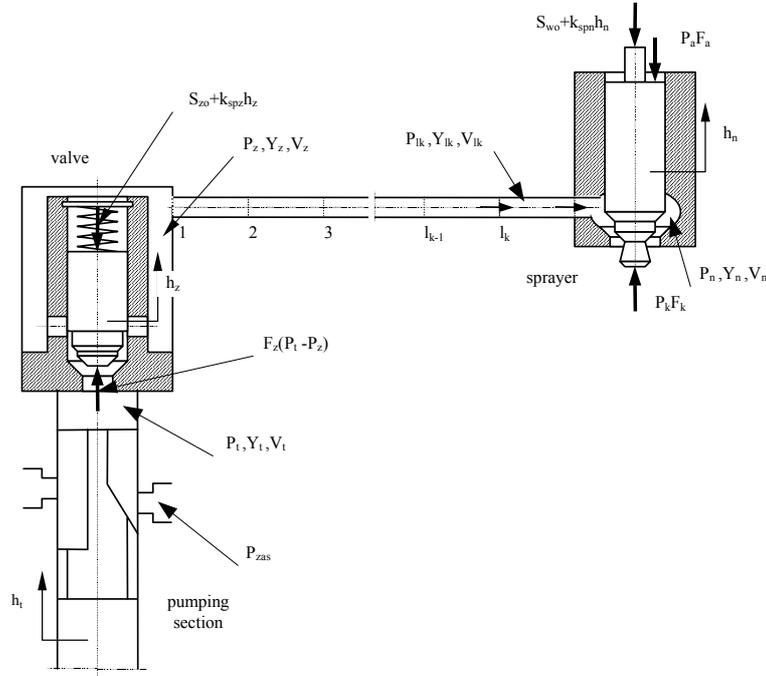


Fig. 1. A draft of the analysed injection system

Unload valve by reason of occurrence in the construction of oscillatory system, is mathematically described by equations of flow continuity and motion equations. The value of „moving volumetric element” in a valve chamber, after an analogous procedure of reasoning and transformations as for the pumping section chamber, has the following form:

$$Y_z = k_{zp} \left[ k_z \cdot \text{sign}(P_k, P_z) \sqrt{|P_k - P_z|} + D_z + D_{zp} \right] \quad (3)$$

An equation of valve head motion, after allowing for all the forces working in the system, can be written as follows:

$$F_z (P_k - P_z) = M_z \frac{d^2 h_z}{dt^2} + \lambda_z \frac{dh_z}{dt} + \delta_z (h_z + h_{z0}) \quad (4)$$

Similarly as occurrences in an unloaded valve, there are ones modelled in an injector. Computational equations include fuel flow and needle motion. An equation on the value of the „moving volumetric element” is constructed in the same way as for the pumping section and unloaded valve:

$$Y_w = k_{wp} \left[ D_w - k_w \cdot \text{sign}(P_w, P_c) \sqrt{|P_w - P_c|} - F_w \cdot h_w + F_{w1} \cdot h_{w1} \right] \quad (5)$$

The needle motion coming from equilibrium of forces is described with the following formula:

$$F_w \cdot P_w + F_{w1} \cdot P_c - F_{w2} \cdot P_a = M_w \frac{d^2 h_w}{dt^2} + \lambda_w \frac{dh_w}{dt} + \delta_w (h_w + h_{w0}) \quad (6)$$

Equations defining a „moving volumetric element” in an injection pipe are derived from equations of flow continuity, with analysing the displacement of fuel in segments [Lejda 1996]. For both ends of the pipeline the formulas are as follows:

$$Y_{r(1)} = \frac{F_r \cdot \Delta L_r}{V_z} \cdot Y_z \quad (7)$$

$$Y_{r(n)} = \frac{F_r \cdot \Delta L_r}{V_w} \cdot Y_w \quad (8)$$

Knowing of values  $Y_k, Y_z, Y_w, Y_r$  we can determine the pressures in each chamber of the pumping system, taking advantage of the formula for defining the “moving volumetric element”:

– pumping section chamber  $P_k = \frac{Y_k}{\beta \cdot V_k}$  for  $Y_k > 0$  (9)

– unload valve chamber  $P_z = \frac{Y_z}{\beta \cdot V_z}$  for  $Y_z > 0$  (10)

– chamber of sprayer  $P_w = \frac{Y_w}{\beta \cdot V_w}$  for  $Y_w > 0$  (11)

– section of pipeline  $P_r = \frac{Y_r}{\beta \cdot V_r}$  for  $Y_r > 0$  (12)

Denotations:

$C_k$  – plunger speed,

$C_z$  – speed of fuel flow in valve chamber,

$$D_k = Y_k - Y_z \cdot \text{sign}(P_k, P_z) \sqrt{|P_k - P_z|}$$

$$D_w = k_w \cdot \text{sign}(P_w, P_c) \sqrt{|P_w - P_c|}$$

$$D_{zp} = k_z \cdot \text{sign}(P_k, P_z) \sqrt{|P_k - P_z|}$$

$F_k$  – area of cross-section of the plunger,

$F_r$  – internal area of injection line,

$F_w$  – changing area of flow between seat and needle,

$F_{w1}$  – section that is impacted by the pressure in the cylinder,

$F_{w2}$  – section that is impacted by the atmospheric pressure,

$F_z$  – area of cross-section of the in-valve chamber,

$g$  – gravitational acceleration,

$H_k$  – jump of section plunger,  
 $H_w$  – lift of sprayer needle,  
 $H_z$  – jump of valve-head,

$$k_{wp} = \frac{1}{1 + \frac{F_r \cdot \Delta L_r}{V_w}}$$

$$k_z = \frac{\mu_z \cdot F_z \cdot \Delta t}{2} \sqrt{\frac{2g}{\gamma}}$$

$$k_{zp} = \frac{1}{1 + \frac{F_r \cdot \Delta L_r}{V_z}}$$

$\Delta L_r$  – length of section of injection line,  
 $M_w$  – mass of moving elements in the injector,  
 $M_z$  – mass of moving elements in the valve,  
 $P_a$  – atmospheric pressure,  
 $P_c$  – back pressure in engine cylinder,  
 $P_k$  – pressure in chamber of pumping section,  
 $P_r$  – pressure in section of pipeline,  
 $P_w$  – pressure in chamber of the sprayer,  
 $P_z$  – pressure in chamber of the valve,  
 $V_k$  – changing volume of the chamber of pumping section,  
 $V_r = F_r \cdot \Delta L_r$  – volume of section of pipeline,  
 $V_w$  – changing volume of sprayer chamber,  
 $V_z$  – changing volume of valve chamber,  
 $Y_k$  – „moving volumetric element” in pumping section,  
 $Y_r$  – „moving volumetric element” in defined section of pipeline,  
 $Y_w$  – „moving volumetric element” in sprayer chamber,  
 $Y_z$  – „moving volumetric element” in valve chamber,  
 $\beta$  – compressibility of the fuel,  
 $\delta_w$  – scale of an injector spring,  
 $\delta_z$  – scale of a valve spring,  
 $\gamma$  – density of the fuel,  
 $\lambda_w$  – resistance coefficient of motion for sprayer needle,  
 $\lambda_z$  – resistance coefficient of motion for valve head,  
 $\mu_z$  – flow coefficient.

## CALCULATION RESULTS AND THEIR VERIFICATION

Simulating calculations were executed for an injection system composed of an in-line injection pump PWM4 type of WAW-MIELEC production, of injector with a special nozzle DNOSD193 type and of high-pressure line at lengths  $L_r = 35.75$  cm and flow diameter  $d_r = 0.2$  cm. All the equations used for calculations were written in a form of finite differences. Thanks to this, the sought variables would be calculated via iterative method, which consists in assuming the unknown values at the beginning of calculations, calculating and making comparison between the received values and the assumed ones.

The precision of the calculations performed in this way depend on an error being the difference of the calculated value from the assumed one for the same iteration. The accepted values of the errors were properly equal to the following:  $\Delta B_1 = 10^{-4}$  cm for the calculation of needle lift,  $\Delta B_2 = 10^{-6}$  cm<sup>3</sup> for the calculation of „moving volumetric elements”. A test of enlargement of calculation precision causes considerable enlargement of calculation time without significant differences in the obtained results. Computational step is changeable and comes from the computed each time average acoustic velocity in the fuel, the taken rotational engine speeds and the accepted lengths of digitising. The number of digitisation points of the pipeline (the accepted length of segment was  $\Delta L_r = 10$  mm) is in programme equal to  $n = 36$ . the programme makes it possible to trace and register free selected courses as the function of time or crank-shaft angle for the parameters required by a user.

The selected results of the calculations and their verification are introduced in Fig. 2, 3, 4, 5, 6 and 7, according to the rotational speeds of the pumps  $n = 1600$  rpm and  $n = 2000$  rpm.

On every picture there was drawn the course of pressure in the pumping section chamber, the course of pressure before the injector and the fuel dose distribution.

The established mathematical model ought to map correctly the courses of real phenomena considering that it was based on the existing computational models, which were done in Automobiles and Internal Combustion Engines Department of the Rzeszów University of Technology for different configuration of injection systems. These models were experimentally verified.

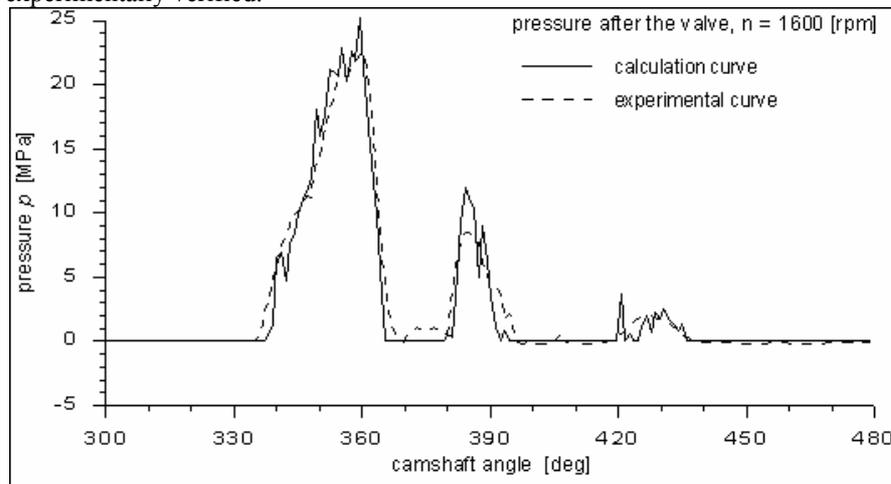


Fig. 2. A comparison between the pressure rates in an injection pipeline after the injection pump (the measured fuel dose  $Q_{zm} = 96,7$  mm<sup>3</sup>/inj)

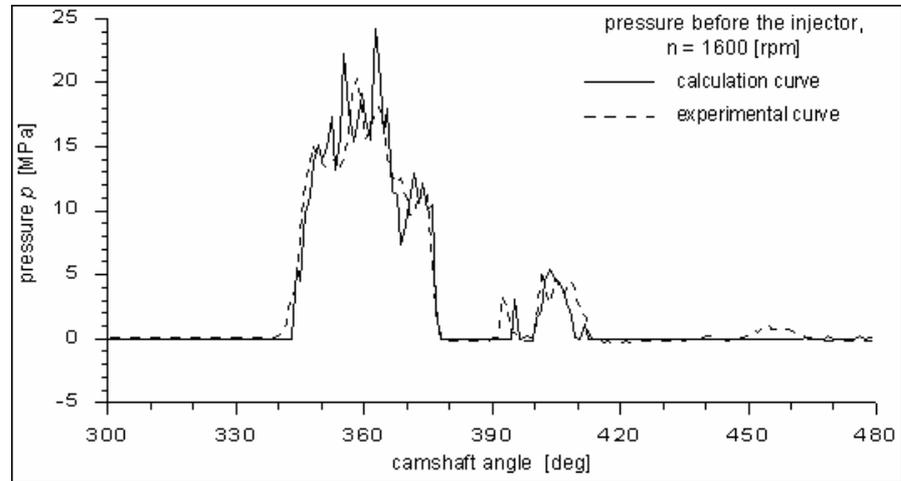


Fig. 3. A comparison between the pressure rates in an injection pipeline before the injector (measured fuel dose  $Q_{zm} = 96,7 \text{ mm}^3/\text{inj}$ )

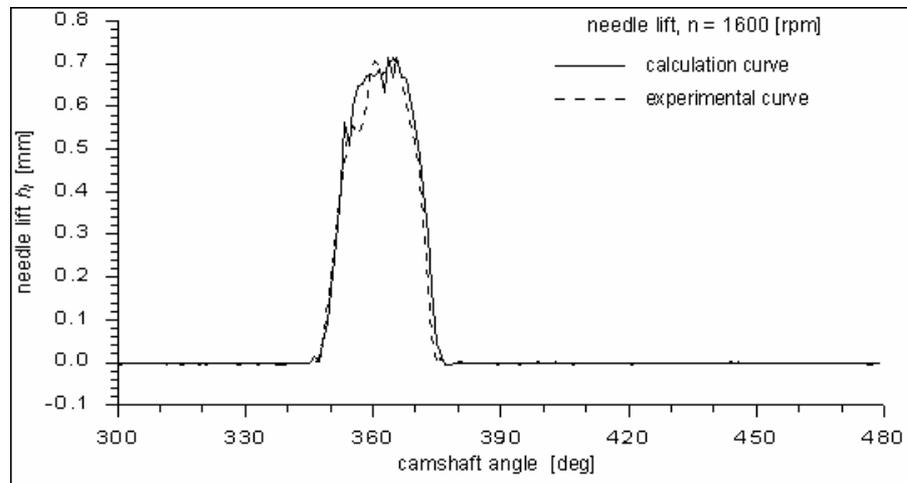


Fig. 4. A comparison between the injector needle lift rates (measured fuel dose  $Q_{zm} = 96,7 \text{ mm}^3/\text{inj}$ )

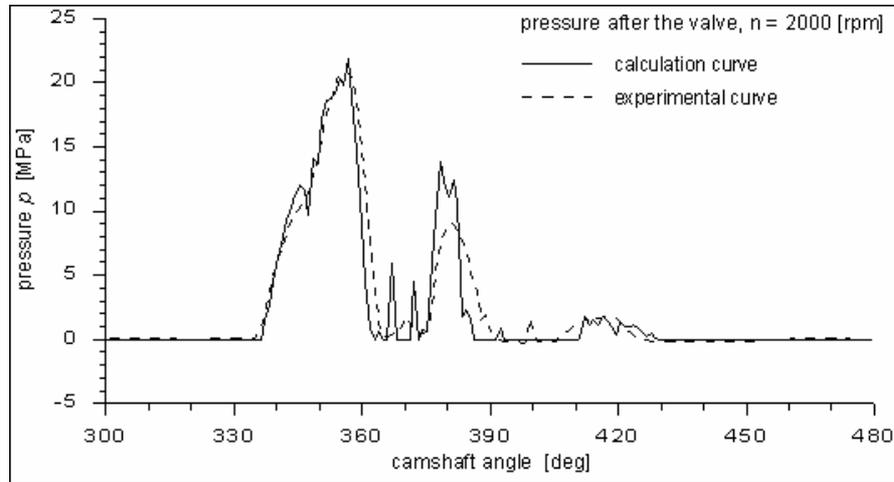


Fig. 5. A comparison between the pressure rates in an injection pipeline after the injection pump (the measured fuel dose  $Q_{zm} = 72,7 \text{ mm}^3/\text{inj}$ )

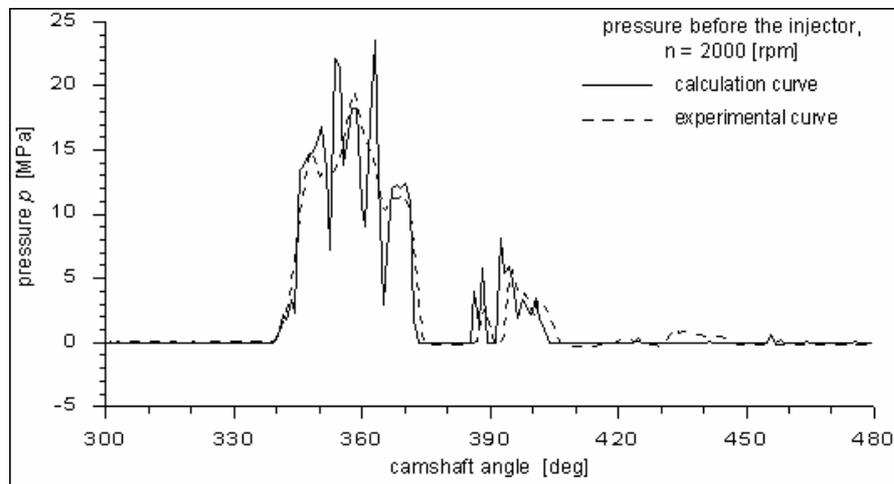


Fig. 6. A comparison between the pressure rates in an injection pipeline before the injector (the measured fuel dose  $Q_{zm} = 72,7 \text{ mm}^3/\text{inj}$ )

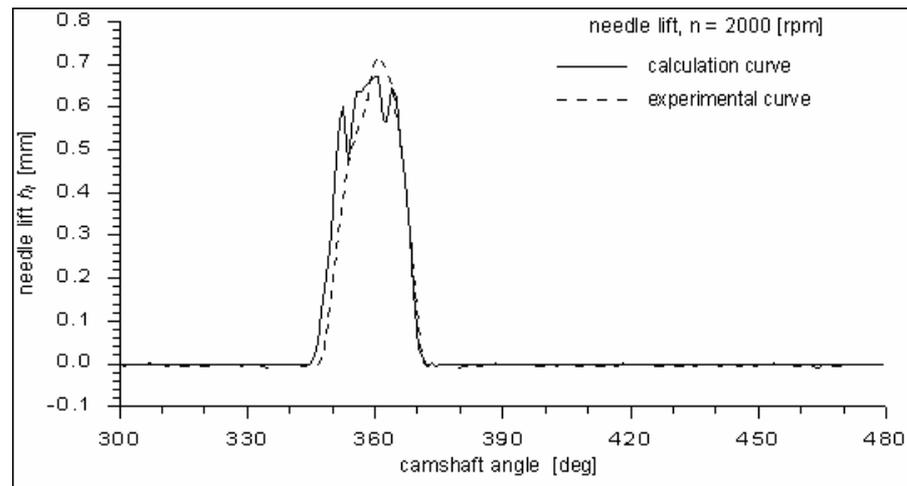


Fig. 7. A comparison between the injector needle lift rates (the measured fuel dose  $Q_{zm} = 72,7 \text{ mm}^3/\text{inj}$ )

## CONCLUSION

Mastery of correct injection characteristics is indispensable for the proper design or selection of an injection system for a diesel engine. The development of engines nowadays is conditioned mostly by a realization of stricter and stricter ecological standards, but for obvious reasons also fuel economy and movement parameters are taken into consideration. An improvement of all these parameters is a difficult matter, all the more that in a lot of cases it should reach a compromise. An improvement of one of the operating coefficients makes worse another one. It seems, that the future of injection systems aims inevitably in the direction of electronics. Certainly in this way it will be possible to assure a more precise dosage and injection angles with reference to all the rotational speeds and loads of an engine. The development of electronic technology in the field of fuel dose steering involves an advent of completely new fuel systems constructions, as e.g. COMMON-RAIL, HEUI, CELECT [Lejda 1996, Prosp. and catal. 2000-2004]. This progress however involves huge costs, which makes such engines considerably more expensive.

At present, the widespread conventional injection systems may be also optimised from the point of view of useful engine parameters, i.e. through the hydraulic steering of the injection process. The costs of the obtainment of desirable effects are many times lower. That's why it appears advisable to take interest in the methods of hydraulic steering. The worked out programme makes possible a quick simulation of this idea in

different variants, without the construction of prototypes or time-consuming experimental investigations.

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