SELECTED PROBLEMS OF MODELLING THE WORKING OF CONTAINER INJECTION SYSTEMS OF COMMON RAIL TYPE

Mirosław Walkowski

Naval Academy, Institute of Vessel Construction and Operation
81 – 103 Gdynia, ul. Śmidowicza 69
tel.: +48 (58) 6262653, 6262665
email: mwal@interia.eu

Abstract

In the work an attempt was made to replace the conventional system of controlling the dose and angle of advance of fuel injection by an electronically controlled system, which was implemented by attaching a fuel container to the conventional high-pressure fuel system, and an electronically controlled, electro-hydraulic fuel-dosage valve.

It was assumed that by controlling the current impulse of the electro-hydraulic valve it would be possible to model the dose of fuel injected into the combustion chamber and to control injection time.

The kinematics of the valve slide has been presented taking account of fuel parameters, hydrodynamic phenomena and axial leakage occurring during valve overdrive.

Keywords: *Common rail, electro-hydraulic valve electronically controlling the fuel dose*

1. Introduction

In conventional and distributional injection pumps, as well as in pump injectors, the combining of the fuel stamping and dosing process with angular location of the shaft or cam ring brings in undesirable changes in the course of fuel injection parameters, with the change of rotational speed of the pump shaft.

Widening of the combustion engine's optimal work range became possible due to an electronic control system, which enables controlling the course of basic parameters of fuel dosage and injection in the whole work range of the engine, in various conditions of the surroundings, also taking into account fuel properties.

The replacement of a conventional control system (by fuel dose and injection angle of advance) with an electronically controlled system was implemented in that the conventional high-pressure fuel system was supplemented with a fuel tank and an electronically controllable electro-hydraulic fuel dosage valve.

This work presents the course of kinematic parameters (shift and speed) of the control valve slide and a calculation fragment from the second phase of the slider movement, i.e. when the slide is shifted upwards up to the moment when the upper piston of the slide exposes the bore chamber in the cylinder at the outlet orifice to the injector.

2. Diagram of a CR JCB – 1 laboratory engine feeding system

A diagram of a fuel system with attached fuel tank and electronically controllable electro-hydraulic fuel-dosage valve has been presented in Fig. 1. A similar feeding system has been applied in Sulzer RT – flex60C engine. By controlling the current impulse of the electro-hydraulic valve it becomes possible to freely model the size of the fuel dose flowing into the combustion chamber in the engine cylinder and to control injection time.

Below is a diagram of a Common Rail (CR) feeding system of a one-cylinder research engine (JSB), which will replace the conventional fuel system.

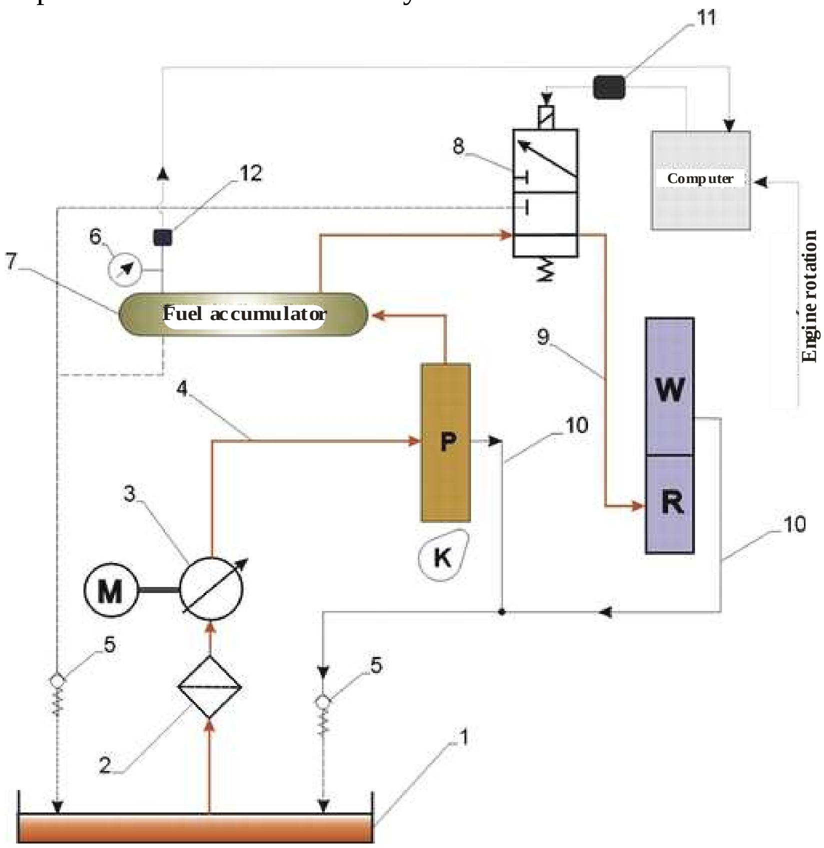

Fig. 1. Diagram of CR injection system of a JSB engine: 1 – fuel tank; 2 – filter; 3 – pump; 4 – low-pressure fuel conduit; 5 – overflow valve; 6 – manometer; 7 – high-pressure fuel tank; 8 – electro-hydraulic valve electronically controlling the fuel dose; 9 – high-pressure fuel conduit; 10 – low-pressure (overflow) fuel conduit; 11 – signal amplifier; 12 – pressure sensor; K – cam; P – high-pressure fuel pump; R – sprayer; W – injector

2.1. Fuel-injection controlling valve

A fuel-injection controlling valve was singled out in the feeding system, designed on the base of JSB engine work parameters and on the required unit of fuel dose, control parameters and the injection angle of advance.

Below there is a figure of the injection-control valve.

The work of the electro-hydraulic valve was divided into nine characteristic phases of the control slide movements [6]: in the first phase the slide is in the bottom position; then it is shifted upwards up to the moment when the upper piston of the slide exposes the bore chamber in the cylinder at the outlet orifice to the injector; the next phase is further movement upwards with two open channels: the overflow channel and the channel delivering fuel to the sprayer; the control slide continues its upward movement – the upper piston is above the bore chamber edge at the orifice supplying fuel to the injector, the overflow orifice being closed. In the fifth movement phase the control slide leans against the upper fender (maximum piston travel); 6^{th} phase – the control slide moves downwards to the moment when the overflow orifice is opened; the next phase is the slide moving downwards with open channels, the overflow channel and the one supplying fuel to the injector; in the eighth phase the movement of the control slide is downwards and the fuel is pumped to the overflow channel; in the final ninth phase, the control slide rests on the lower fender.

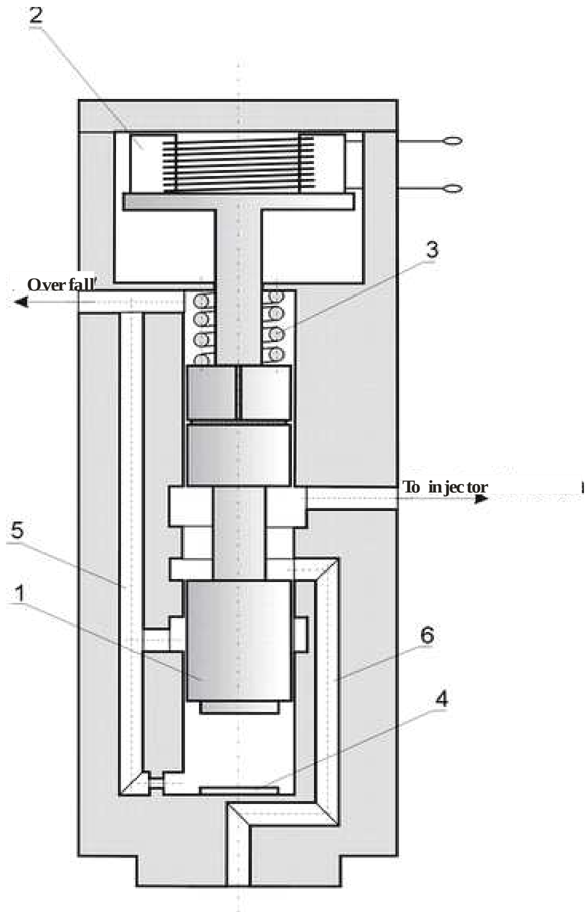

Fig. 2. Electro-hydraulic valve electronically controlling the fuel dose:
1– control slide, 2–coil, 3–rebound spring, 4–fender, 5– overflow channel, 6– main fuel channel

3. Dampened slide movement of the electromagnetic valve

In calculating operation parameters of the electromagnetic valve the following simplified assumptions were made: fuel is a compressible fluid, with longitudinal modulus of elasticity E and is subject to Hooke's law; what was omitted were the elastic deformations of the fuel injection conduit caused by its pressure changes; fuel flow in the high-pressure conduit was treated as an isothermic one-dimension flow (flow characteristic parameters change along the conduit axis, but are constant in cross section); friction effect was taken account of with the assumption that losses caused by its effect during stationary and non-stationary flow are equal (in dependencies describing fuel flow in the conduit they are expressed by the hydraulic resistance coefficient); omitted were the undulatory phenomena in the injection system; when calculating the mean flow speed of fuel in the fuel conduit the time required for opening and closing the injector valve was omitted; injection pressure change does not affect fuel density (ρ= const.).

When analysing the slide movement of the electromagnetic valve the following were taken account of [5, 6]:

- Energy losses when the fuel flows through constant and variable valve section areas;
- Hydraulic losses between pistons and the control valve cylinder;
- Hydrodynamic forces in successive phases of the valve slide movement;
- Viscous resistance forces in extreme positions of the control valve [5];
- Spring resistance force (including its initial tension);
- Inertial force from the return spring mass, electromagnet armature mass, control slide mass (including fuel shifted with the slide).

4. Selected problems of modelling the work of an electro-hydraulic control valve

This work presents the course of kinematic parameters (shift and speed) of the control valve slide and a calculation fragment from the second phase of the slider movement, (i.e. when the slide is shifted upwards up to the moment when the upper piston of the slide exposes the bore chamber in the cylinder at the outlet orifice to the injector) and the share of axial leaks in particular phases of the slide movement, and also the effect of hydrodynamic forces of the flowing fuel on the slide movement.

4.1. Hydrodynamic force in successive phases of the valve slide movement

The designation of hydrodynamic force variable in time and dependent on the position of the hydrodynamic force slide $\mathbf{F_h}$ [4, 6].

$$AB(t)=\sqrt{s_p^2+e_p^2(t)}=e_p(t)\sqrt{1+\frac{s_p^2}{e_p^2(t)}}\,;$$

$$CD(t)=\sqrt{s_w^2+e_w^2(t)}=e_w(t)\sqrt{1+\frac{s_w^2}{e_w^2(t)}}\,;$$

$$A_{AB}(t)=\pi D_c AB(t); \qquad A_{CD}(t)=\pi D_c CD(t);$$

$$\cos\Theta_p(t)=\frac{s_p}{AB(t)}\;; \qquad \cos\Theta_w(t)=\frac{s_w}{CD(t)}\;;$$

$$F_{hp}(t)=A_{AB}\Delta P_{1,5}(t)\cos\Theta_p(t); \qquad F_{hw}(t)=A_{CD}\Delta P_{1,8}(t)\cos\Theta_w(t);$$

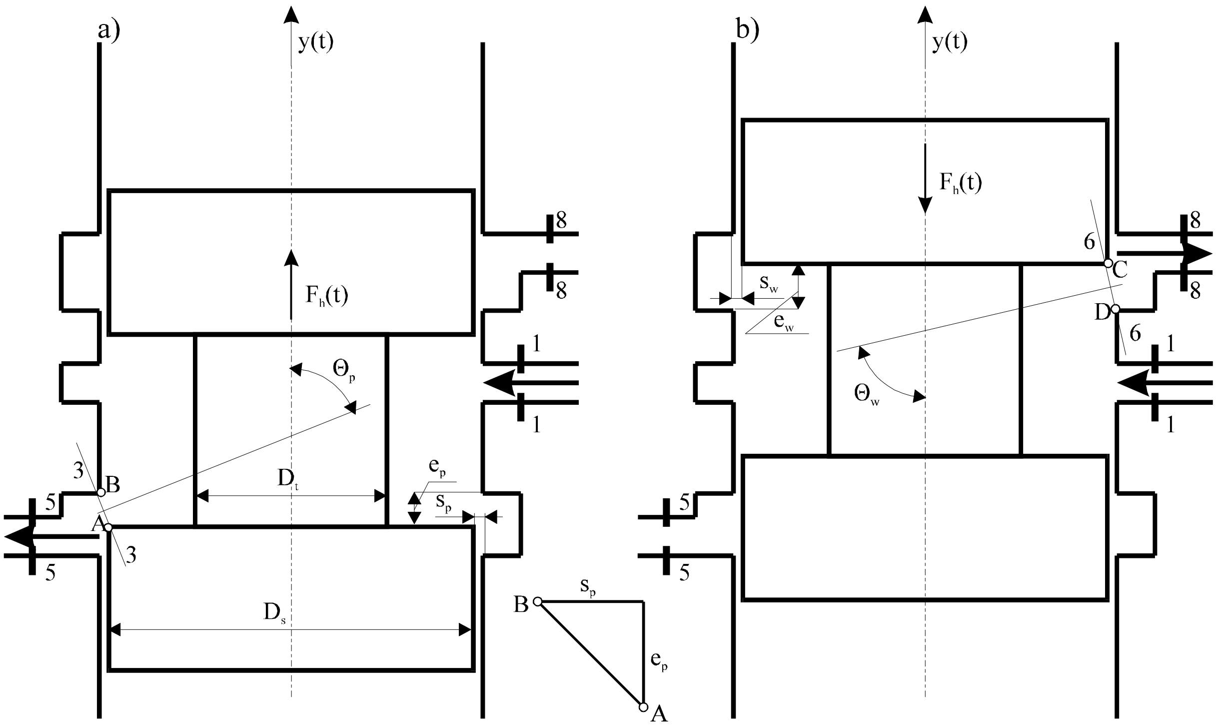

Fig. 3. Sketch for indicating the effect of fuel flow direction change on hydrodynamic force F_h.

$$\vec{F}_h=\vec{F}_{hp}+\vec{F}_{hw}$$

AB(t), CD(t) - distance between the control slide edge from the edge of cylinder bore chamber of the control valve (Fig. 9.), m,
e_p, e_w - respectively: $e_p = b_p - y(t)$; $e_w = b_w - y(t)$; (Fig. 3), m,
s_p, s_w - size of slot between control slide and valve cylinder, m,
Θ_p, Θ_w - inclination stream of fuel stream in relation to slide axis, [^{o}],
$F_{hp}(t)$, $F_{hw}(t)$ - hydrodynamic forces: when flowing through overflow orifice or when flowing through orifice supplying fuel to the injector, N,
F_h - resultant hydrodynamic force, N,
B_p, b_w - bore chamber height, m.

Flow losses resulting from axial leaks in annular gaps

The gap between piston and cylinder was assumed to be concentric.
$s_0 = (D_2 - D_1) / 2$ – gap between cylinder and piston,
$D = (D_1 + D_2) /2$ – mean gap diameter with the assumption that $\mathbf{s_0 << D}$,
μ - dynamic viscosity coefficient, [kg/m·s],
ν - kinematic viscosity coefficient, [m^2/s], $\boldsymbol{\nu = \mu / \rho}$.

The expense through the gap with piston immobile in relation to the cylinder:

$$Q = \frac{\Delta p\, s_0^3 \pi D}{12\mu l} = \frac{\Delta p\, s_0^3 \pi D}{12\nu\rho l}$$

For a piston moving in relation to the cylinder with constant speed $\mathbf{v_0}$, the flow expense through the gap will be equal to

$$Q = \left[\frac{\Delta p\, s_0^3}{12\mu l} \pm \frac{v_0 s}{2}\right]\pi D = \left[\frac{\Delta p\, s_0^3}{12\nu\rho l} \pm \frac{v_0 s}{2}\right]\pi D$$

sign **„ + ”** – corresponds to the state when the piston is moving towards higher pressure,
sign **„ - ”** – corresponds to the movement of the piston towards lower pressures.

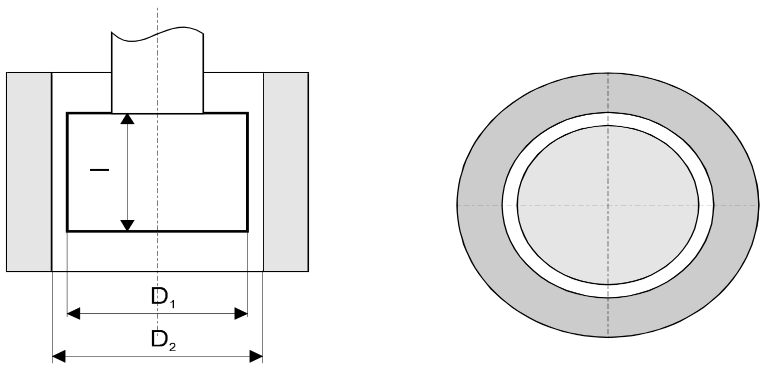

Fig. 4. Concentric annular gap

5. Kinematic equations of control valve slide

In the selection of equations of the 2nd phase of the control valve slide movement the limitation was made to general kinematic dependencies, the limitation resulting from general framework assumed for this study. The recording of equations given does not infringe on the principles of model adequacy for the object analysed, although its more detailed solutions are possible too.

Shifts, speeds and accelerations of the slide caused by forces in the valve (excluding rebounds from movement limiters) are described by the dependencies [6]:

2nd phase; $t_1 \le t < t_4$,

$$y(t) = Y\left[1 - e^{h_i(t-t_1)}\cos\lambda_i(t-t_1)\right],$$
$$\dot{y}(t) = Y\,\lambda_i\,e^{-h_i(t-t_1)}\,sin\,\lambda_i\,(t-t_1),$$
$$\ddot{y}(t) = Y\lambda_i^2 e^{-h_i(t-t_1)}\left[\cos\lambda_i(t-t_1) - \frac{h_i}{\lambda_i}\sin\lambda_i(t-t_1)\right]$$

Substitutions:

$$h_i(t) = \frac{c_z(t)}{2m_z}, \qquad h_j(t) = \frac{c_z(t)}{2m_z}$$

$$\lambda_i(t) = \left[\frac{k_z}{m_z} - h_i^2(t)\right]^{0.5} = \frac{\pi}{2T_{14}}, \qquad \lambda_j(t) = \left[\frac{k_z}{m_z} - h_j^2(t)\right]^{0.5} = \frac{\pi}{2T_{69}},$$

$c_z(t)$ – coefficient of viscous dampening for respective time intervals, [Ns/m],
k_z – coefficient of spring stiffness, [N/m],
$\lambda_i(t)$,$\lambda_j(t)$ – angular frequency of the slide and the armature in respective time intervals, [s^{-1}].

The angular frequency of free vibration of the slide and the armature are equal to:

when: $0 < t < t_4$ $$\omega_i = \left[\lambda_i^2(t) + h_i^2(t)\right]^{0.5},$$

when: $t_6 < t < t_9$, $$\omega_j = \left[\lambda_j^2(t) + h_j^2(t)\right]^{0.5}.$$

Exciting force frequency of valve slide movement

$$f_i = 2\pi n_i, \qquad f_i = 2\pi n_w.$$

ni , nw – frequencies of electric impulse and fuel injection.

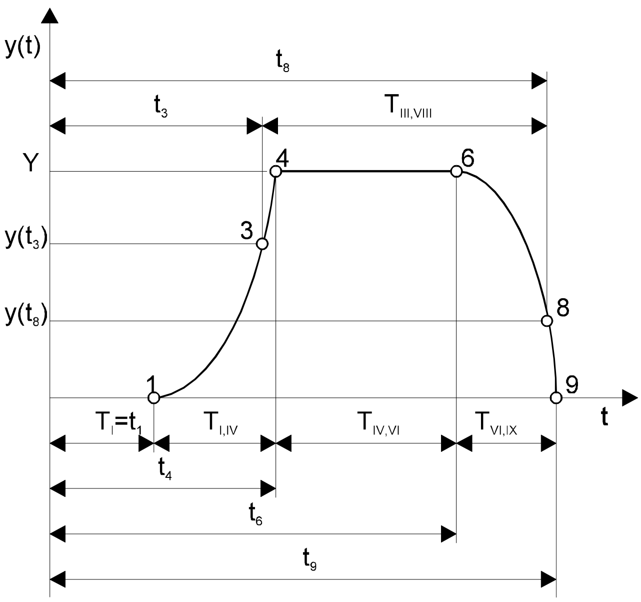

Fig. 5. Diagram for describing the course of slide movement of a viscously dampened valve [8]

Fig. 6 presents the time courses of the electro-hydraulic valve slide shift and speed.

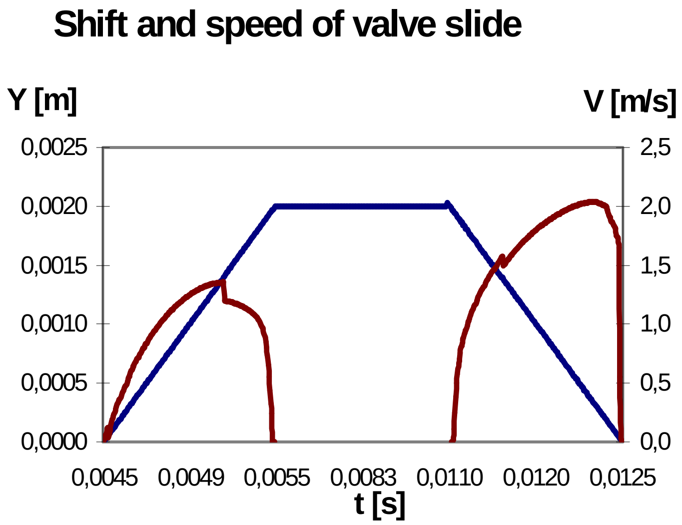

Fig.6. Electro-hydraulic valve slide movement and speed courses

After making initial assumptions, determining valve work parameters and introducing input data, there were determined the kinematic parameters of slide movement, i.e. the time course of slide movement, the change of its speed and acceleration at a single overdrive caused by forces active in the valve.

The graph in Fig. 4 presents the time course of the slide movement, with visible main movement phases:

Preparation for starting the slide, $y = 0$, $t = 0$,

Starting the slide, $y = 0$ whereas $t = t_1$,

Slide movement under the effect of electromagnet force, $0<y<Y$, $t_1<t<t_4$,

Stopping and holding down the slide at the upper fender, $y = y_4 = Y$, $t_4 \leq t \leq t_6$

Return movement of the slide under the effect of spring forces, $t_6<t<t_9$, $0<y<Y$,

Holding down the slide at the lower fender, $t>t_9$, $y = 0$.

6. Résumé

Adapting particular elements of the electronic control system, that is, an electronically controlled fuel-dosing valve and fuel tank into the conventional high-pressure injection system of a JSB engine, it is possible to obtain a completely controllable injection process.

The results obtained, that is the calculation of slide movement kinematics of the valve controlling the fuel dosage, constitute very important information enabling to analyse the particular movement phases of the valve slide and to shape fuel injection course during the engine's work cycle.

When controlling the current impulse it is possible to control the fuel dose size, and also to model multiple injection, characteristic for electronically controlled injection systems.

Simulation calculations were conducted based on theoretic and empirical dependencies without verification and research on a real valve model, which is why they should be treated as estimation results.

In order to verify the model assumed experimental research should be conducted on a real valve and the results obtained should be used for correcting the assumed calculation model.

References

[1] *Elektroniczne sterowanie silników wysokoprężnych. Układ wtryskowy Common Rail*, WKŁ, Warszawa 2000.
[2] *Elektroniczne sterowanie silników wysokoprężnych. Układy wtryskowe UIP/UPS*, WKŁ, Warszawa 2000.
[3] Gałąska, M., Kaczmarczyk, J., Maruszkiewicz, J., *Hydromechanika stosowana*, Wojskowa Akademia Techniczna im. J. Dąbrowskiego, Warszawa 1972.
[4] Guillon, M., *Teoria i obliczanie układów hydraulicznych*, WNT Warszawa.
[5] Landau, L. D., Lifszyc, E. M., *Hydrodynamika*. PWN Warszawa 1994.
[6] Ochocki, W., *Numerycznie sterowane systemy wtrysku paliwa silników wysokoprężnych*, Wydawnictwo Poznańskiego Towarzystwa Przyjaciół Nauk, Poznań 1994.
[7] Sobieszczański, M., *Modelowanie procesów zasilania w silnikach spalinowych*, WKŁ Warszawa 2000.
[8] *Zasobnikowe układy wtryskowe Common Rail*, BOSCH. WKŁ Warszawa 2005.
[9] Zbierski, K., *Układy wtryskowe Common Rail*, MiW Łódź 2001.