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Preliminary analysis of a innovative type of low pressure valves

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Analysis and modelling

ABSTRACT

Purpose: Presentation and analysis of the innovative type of a direct operated low pressure relief valves.
Design/methodology/approach: In this paper was presented a proposal of a improvement of a new low pressure relief valve. The advantages and disadvantages of this valve was estimated by numerical simulation with MATLAB/SIMULINK aid.
Findings: Main advantages of presented relief valve are: high precision of the operating pressure and short time responses, the disadvantage of the modelled valve is the high value of the peak pressures.
Research limitations/implications: The expected next stage of valve research is the experimental tests. For that purpose one should the prototype of valve build.
Practical implications: Presented low pressure relief valve can be used in low pressure hydraulic system, for example for feeding of the hydrostatic or hydrodynamic bearings.
Originality/value: Original value of this paper is a idea of low pressure relief valve.
Keywords: Constructional design; Numerical techniques; Hydraulics; Relief valve

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1. Introduction

In this paper was presented a innovative type of a direct operated low pressure relief valves. The idea of this type of a low pressure control valves based on untypical method of the cracking pressure setting (Fig. 1). On the basis of this idea was created a mathematical model of the valve and next, a computer simulation in MATLAB/SIMULINK environment. The results of the simulation indication that new type of the low pressure relief valve characterized:

- almost stable operating pressure in flow-function,
- short setting times,
- acceptable pressure overshoot.

In the paper the model analysis of the low pressure relief valve for high flow rate was taking into consideration. The construction of the prototype pressure relief valve was based on the 2-way cartridge valves for pressure control functions type LC…DB… produced by Bosch Rexroth [1].

Innovation of presented low pressure valve (Fig. 1) consist in this new valve is a direct operated pressure relief valve. In typical direct operated pressure relief valve (Fig. 2) the cracking pressure setting is made by variation of the compression of spring 2, which forces on operating element 5 (spool or poppet). In pilot operated pressure valve the cracking pressure setting is made otherwise.

Typical pilot operated pressure relief valve consists of a pilot valve and main valve. Pressure in the hydraulic system affect the bottom side of the main spool and at the same time the pilot valve generated pressure affect the upper side of the main spool. At standstill, the pressure is equal on both sides of the main spool. The characteristic feature of the pilot operated valves is that the pressure of the pilot valve direct affect the main spool (poppet).

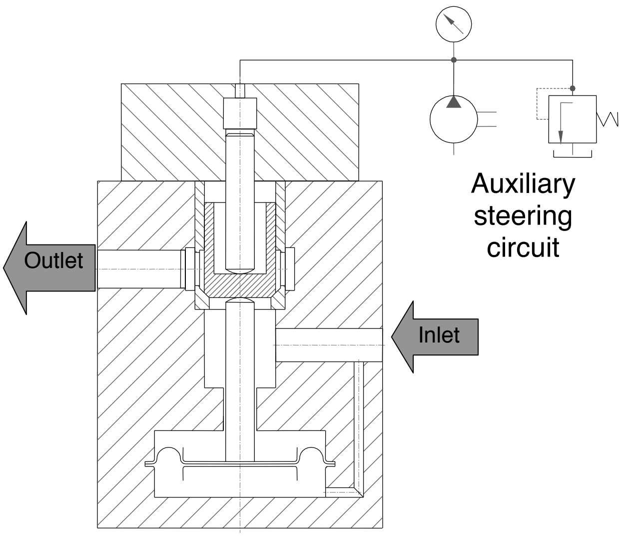

Fig. 1. Low pressure relief valve conception

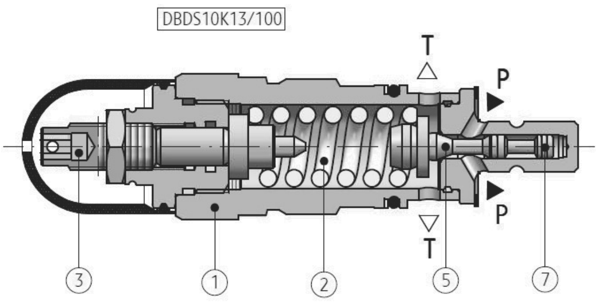

Fig. 2. Typical direct operated pressure relief valve [2]

The new low pressure valve is hydraulically operated [3-8], but the control pressure affect the spool indirectly. The control pressure, generated by auxiliary hydraulic system, affect the steering plunger, that pushed on upper side of the spool of cartridge type LC. On bottom side of the spool affect the sum of the hydrostatic forces originated from the system pressure affect the bottom side of the spool and the diaphragm.

2. Low pressure relief valve simulation

In first simulation stage made the assumptions:

- parameters of the valve model are discrete,
- viscosity of the working liquid is constant,
- movable valve elements that have the same degree of freedom one mass substituted,
- working area of the rolling diaphragm is constant,
- gauge pressures were computed.
- For the physical model the followings data were accepted:
- movable mass (spool, plunger, membrane with slide) m = 1.63 kg
- mass density of the working liquid $\rho = 900\ kg/m^3$
- kinematic viscosity of the working liquid $\nu = 46$ cSt
- compression modulus of the working liquid $B = 1.4\cdot10^3$ MPa
- radial clearance of the movable couples $\varepsilon = 3\ \mu m$
- inlet volume $V_{WE} = 8\cdot10^{-4}\ m^3$
- volume of the auxiliary system $V_{ST} = 2.8\cdot10^{-5}\ m^3$

The volumes V_{WE} and V_{ST} were assumed so as to corresponded to hydraulic pipes 1 m length.

In the next stage was the mathematical model made. The mathematical model was consisted of [9-12]:

- the equation of the movable mass motion:

$$m\cdot\ddot{x} + k\cdot\dot{x} + T\cdot sign(\dot{x}) + F_{HD} + F_{spr\ M} + F_{ster} + F_{stat} = 0 \qquad (1)$$

- equations for balances of the flow rates:

$$Q_P = Q_{ZP} + Q_M + Q_{sc} + Q_{\dot{x}}$$
$$Q_{PS} = Q_{ZS} + Q_{sc\ S} - Q_{\dot{x}\ S} \qquad (2)$$

T - Coulomb friction force (assumed as 1 N),
F_{HD} - hydrodynamic force,
$F_{spr\ M}$ - membrane elasticity force,
F_{ster} - steering plunger force,
F_{stat} - sum of the hydrostatic forces of system pressure,
Q_P - system pump output flow rate,
Q_{ZP} - LC cartridge flow rate,
Q_M - flow rate affected diaphragm deformations,
Q_{sc} - flow rate affected compressibility of the working liquid in main hydraulic circuit,
Q - flow rate affect the spool motion,
Q_{PS} - steering pump output flow rate,
Q_{ZS} - steering relief valve overflow flow rate,
$Q_{sc\ S}$ - flow rate affected compressibility of the working liquid in auxiliary hydraulic circuit,
$Q_{\ S}$ - flow rate in auxiliary hydraulic circuit affect the steering plunger motion

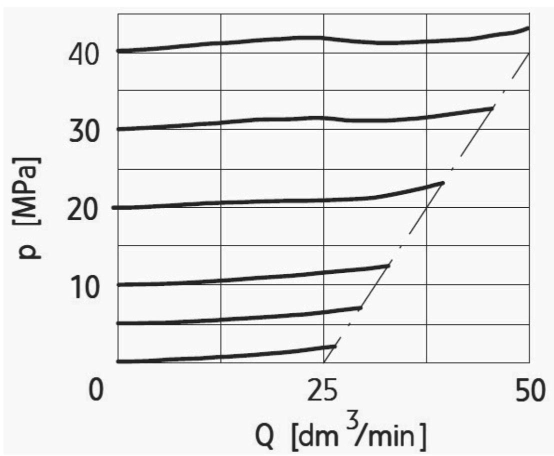

Fig. 3. Performance curves of the direct operated pressure relief valves type DBD 6 [2]

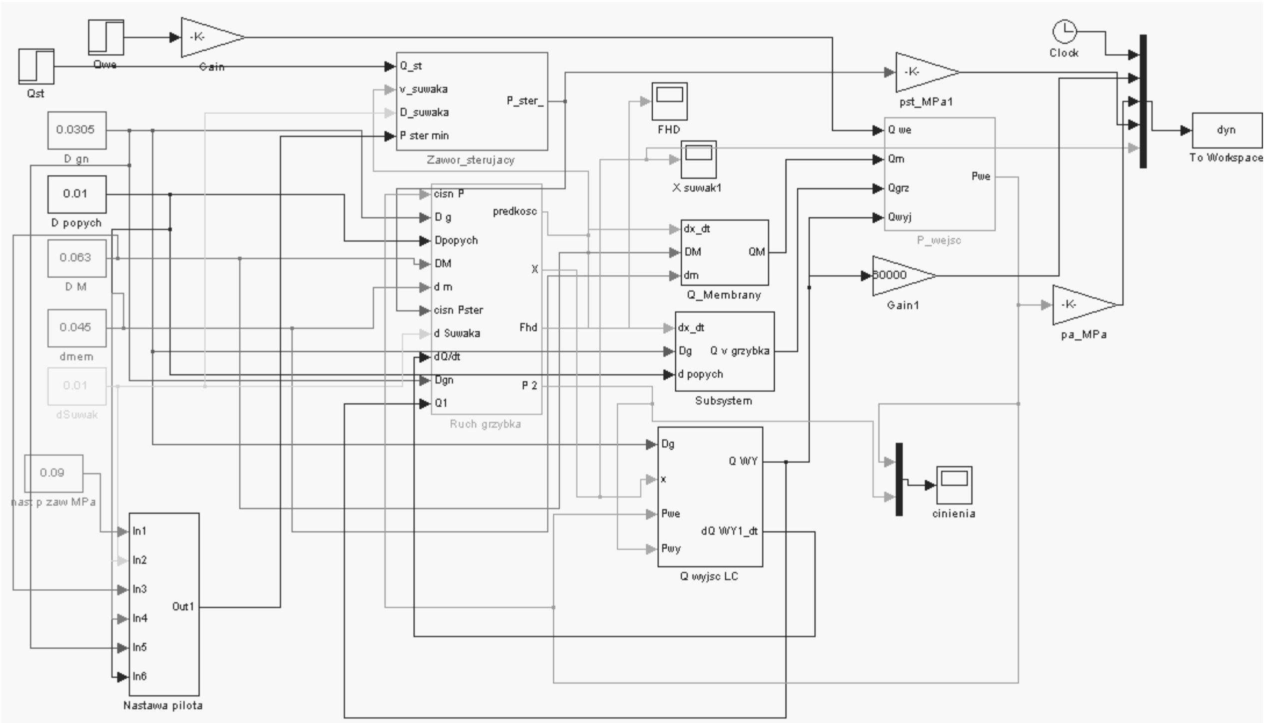

Fig. 4. Computer model of the low pressure valve

In auxiliary hydraulic circuit model was assumed the direct operated pressure relief valve type DBD 6 made of the PONAR S.A. characterized by operating pressure range up to 5 MPa and maximum allowable flow rate up to 25 dm^3/min [2]. The time-constant of the DBD 6 valve as 1 ms was accepted and the performance curves was presented in Fig. 3.

On the basis of the mathematical model of the low pressure valve presented as equations (1) and (2), was created the numerical model using the MATLAB/SIMULINK software [13-15]. Graphical form of the computer valve model was presented in Fig. 4.

3. Results of the model testing

3.1. Identification of the static properties of the low pressure valve

During the model testing of the low pressure relief valve were obtained the performance curves, presented in Figs. 5-7.

On the performance curves basis were the inequality factor of the pressure as:

$$\delta_p = \frac{p_{\max} - p_o}{p_o} \cdot 100\% \qquad (3)$$

The values of the δ_p were presented in Fig. 8.

Computed performance curves were characterized by the pressure hysteresis. The hysteresis values were computed for 60 dm^3/min flow rate and presented in Fig. 9.

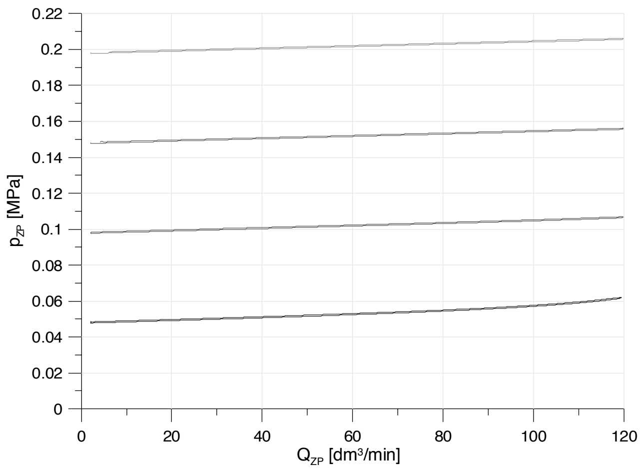

Fig. 5. Characteristic curves of the low pressure relief valve with ∅63/45 diaphragm

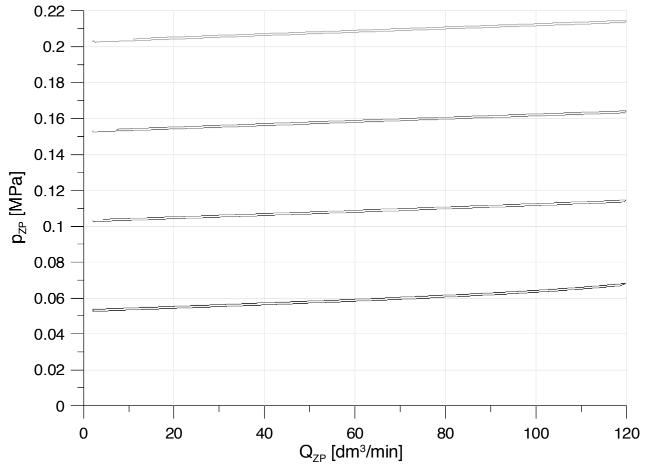

Fig. 6. Characteristic curves of the low pressure relief valve with ∅45/32 diaphragm

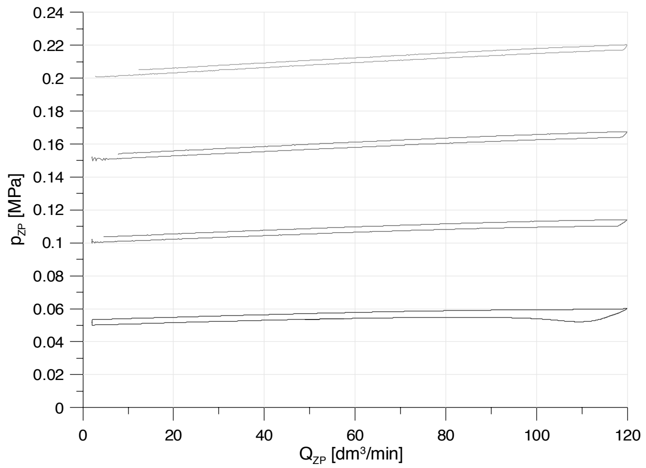

Fig. 7. Characteristic curves of the low pressure relief valve without diaphragm

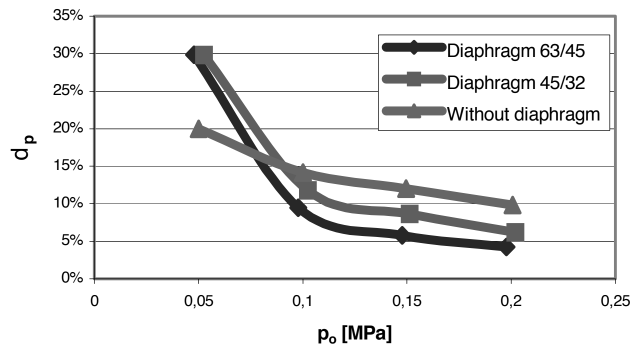

Fig. 8. Influence the diaphragm dimensions and the pressure setting on the inequality factor of the pressure δ_p

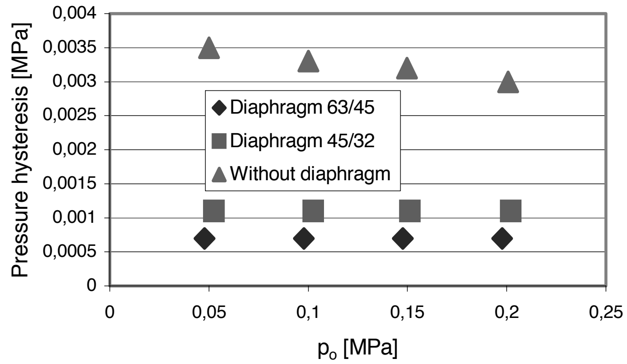

Fig. 9. Influence the diaphragm dimensions and the pressure setting on the pressure hysteresis

Characteristics presented in Figs. 8 and 9 indicated that main spool opening assisted by diaphragm increased the static properties of the low pressure relief valve (additional area increased the sensitivity of the valve). Only for the lowest pressure setting as 0.05 MPa the smaller inequality factor of the pressure for the valve without the diaphragm was obtained. Whereas the pressure hysteresis were 3-4 times more for the valves without the diaphragm.

3.2. Analysis of the dynamic properties of the low pressure relief valve

The relief valves usually have working with steady conditions. Seemingly, the dynamic properties of this kind of hydraulic valves are not important. Also, many producers of the hydraulic elements have not given this information in catalogues. The failure possibility the omitting of the dynamic properties of the valves caused. During transient states, the pressure jumps can considerably exceed the nominal pressure. The pressure overshoot depend on the dynamic parameters of the pressure relief valve, the compression modulus of the liquid and volume of the compressed liquid.

Within the framework of dynamic analysis of the valve model the pressure response on flow rate step were determined (Figs. 10, 11). The flow rate step was realized between 10 and 120 dm^3/min and vice versa at t = 0.2 s.

For this flow rate step function the high pressure overshoot was received (peak pressure nine times exceed the steady state pressure).

The pressure overshoot is irrespective of the steering pump output flow rate Q_{PS}. The high pressure overshoot the low system inlet volume V_{WE} was caused. The value of the peak pressure not exceed the 1 MPa therefore the pressure overshoots are not dangerous for hydraulic system.

Next, the pressure setting times T_R (valve response times) were determined for the pressure error ≤ ±5% steady pressure. The pressure setting times, showed in Fig. 12, were determined for the following steering pump output flow rate: 0.2; 1; 5 and 25 dm^3/min. The results, showed in Fig. 12, indicated that the steering pump output flow rate weakly affected the pressure setting times. For example, the increase 125 times of the steering pump output flow rate caused 22% reducing of the pressure setting times T_R.

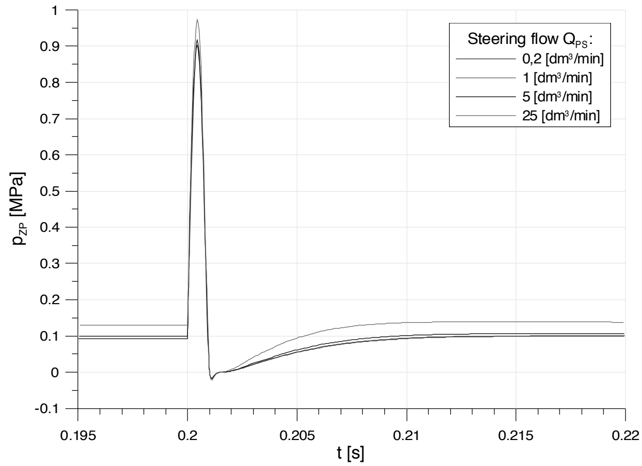

Fig. 10. Pressure response for flow rate step from 10 to 120 dm^3/min

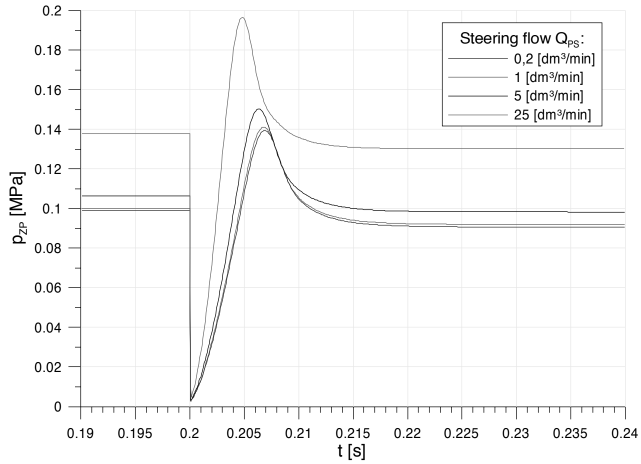

Fig. 11. Pressure response for flow rate step from 120 to 10 dm^3/min

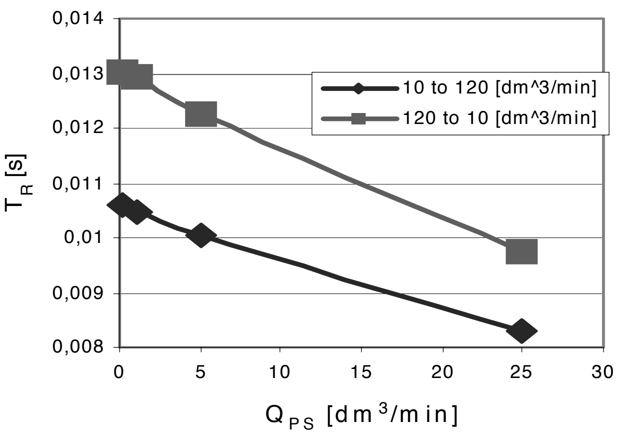

Fig. 12. Pressure setting times T_R for flow rate step

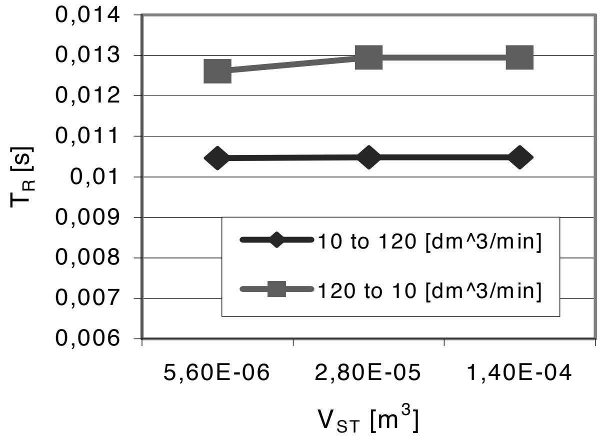

Fig. 13. The influence of auxiliary system volume on the pressure setting times

In Fig. 13 the auxiliary system volume influence on the pressure setting times was presented. This simulation with constant steering pump output flow rate Q_{PS} = 1 dm^3/min was realized.

The results indicated that the pressure setting times T_R is insensitive to the auxiliary system volume V_{ST}. Simultaneous, the valve stability is depended on the auxiliary system volume. The characteristic showed on Fig. 14 this indicated.

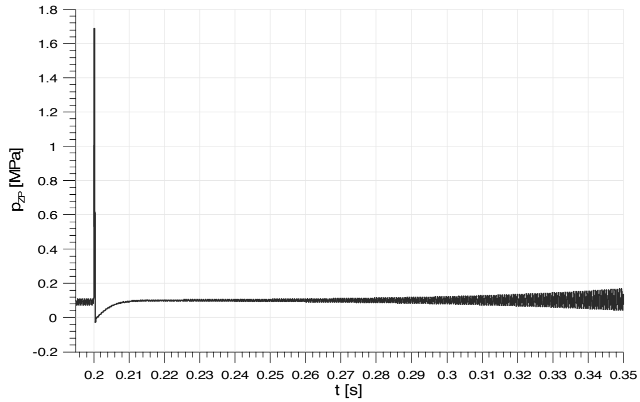

Fig. 14. The valve response on flow step 10 to 120 dm^3/min for $V_{ST} = 3.73 \cdot 10^{-6}$ m^3

Dynamics of the hydraulic systems is depended on dynamic properties of the hydraulic elements and the volume of the pressure hydraulic circuits. Additionally, the dynamic properties of the hydraulic system was influenced by changes of properties of the working liquid (produced by liquid aeration).

In Figs. 15 and 16 was presented the simulation results, received for inlet volume bigger 5 times and smaller 5 times than original V_{WE}.

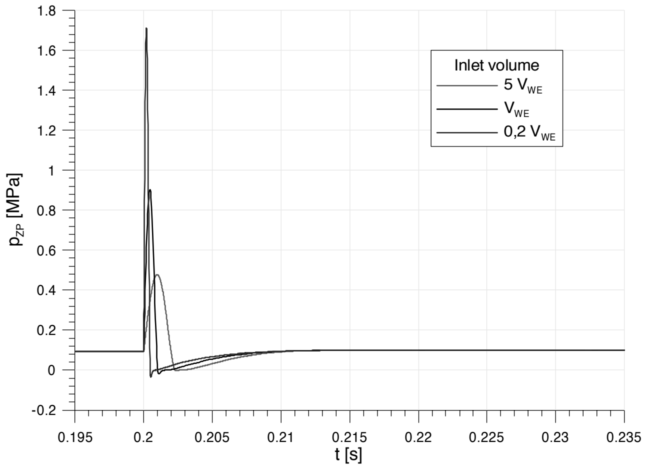

Fig. 15. The valve response on flow step 10 to 120 dm^3/min for different inlet volumes

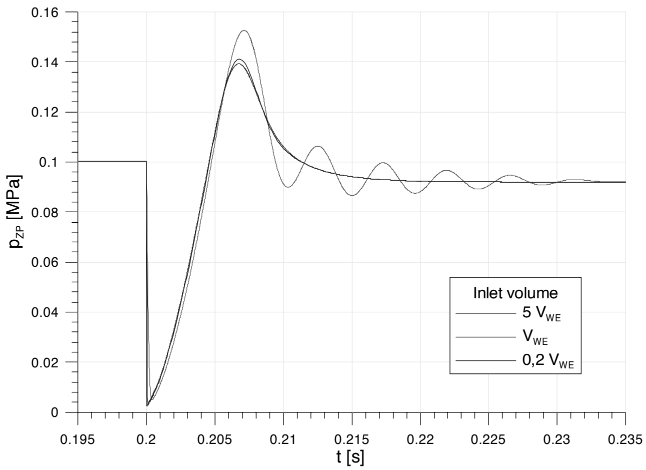

Fig. 16. The valve response on flow step 120 to 10 dm^3/min for different inlet volumes

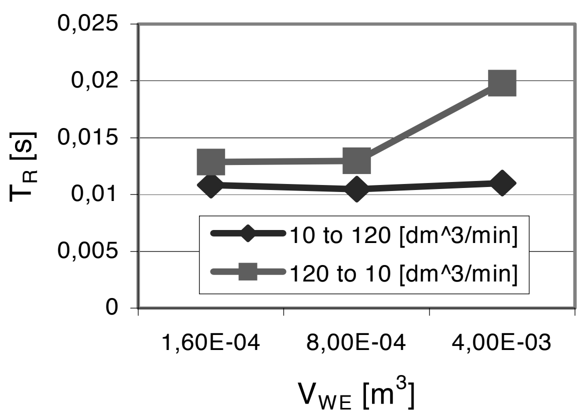

Fig. 17. The influence of inlet volume V_{WE} on the pressure setting times T_R for flow step

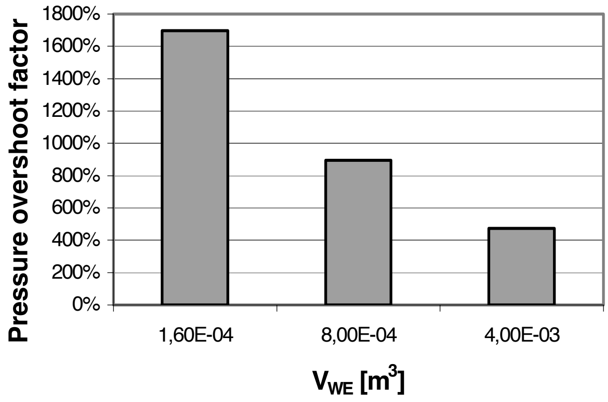

Fig. 18. The influence of inlet volume V_{WE} on the peak pressures for flow step

Expected effect of the increasing inlet volume was the decreasing the peak pressures for step of valve inlet flow rate. Additionally, with increasing the inlet volume, the increasing of the valve response time for flow step with increasing was observed.

3.3. Analysis of the dynamic properties of the low pressure relief valve with proportional control

The share of proportional elements in hydraulic elements market continuously improved. The presented low pressure relief valve can be performed as the electrically and proportionally operated. In that order, in the auxiliary hydraulic system using of the proportional pressure relief valve, for example type WZPSE6 produced by PONAR Wadowice, is necessary.

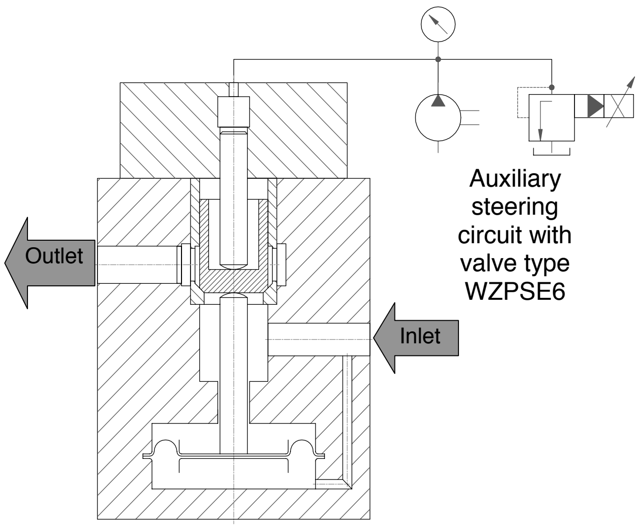

Fig. 19. Proportionally operated low pressure relief valve

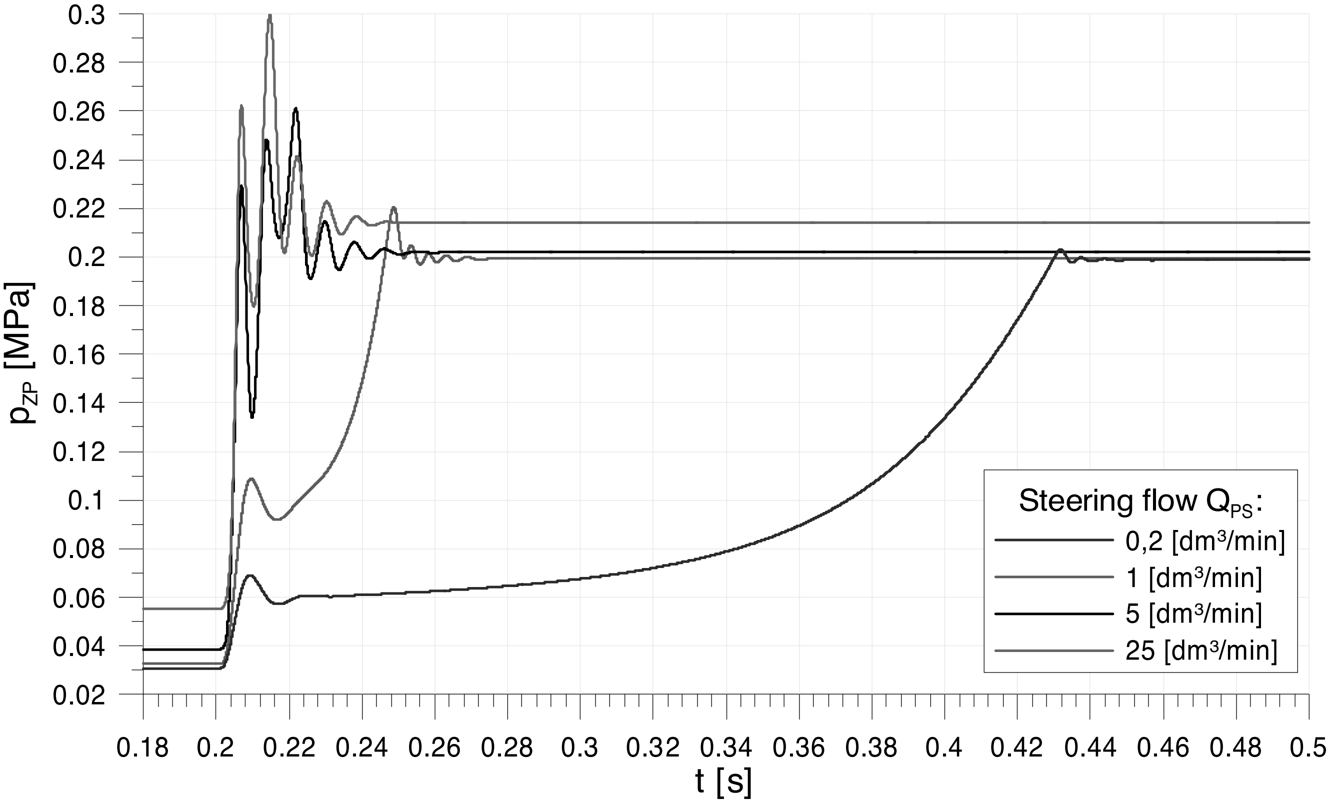

Fig. 20. Pressure response for current step from 0 to rated current at 0.2 s

The predesign of proportional low pressure relief valve is presented in Fig. 19.

The static properties of the valves not depended on the operating method. Both the proportionally operated version of the low pressure valve and the basic version are characterized by performance curves presented in Figs. 5-7.

The characteristic feature of the proportional valves is the possibility of the sudden or step change of the pressure (or flow) setting.

The basic dynamical feature of the proportional valves is the response time for the step of current of the solenoid. The simulations were realized for the following assumptions:

- variation of solenoid current: from 0 to rated current at 0.2 s,
- diameter of the diaphragm: 63 mm,
- diameter of piston: 45 mm,
- flow rate in main low pressure circuit: 120 dm^3/min,
- flow rate in auxiliary circuit: 0.2 dm^3/min; 1 dm^3/min; 5 dm^3/min; 25 dm^3/min.

The valve response on current step for different flow rate in auxiliary circuit is presented in Fig. 20.

Next, the valve response times T_R were determined for the pressure error $\leq \pm5\%$ steady pressure. The pressure setting times are showed in Fig. 21. The results, indicated that under 5 dm^3/min, the steering pump output flow rate strongly affected the pressure setting times.

For example, the increasing of the steering pump output flow rate from 0.2 dm^3/min to 1 dm^3/min caused 77.8 % reducing of the pressure setting times T_R.

Whereas, the increasing of the steering pump output flow rate from 1 dm^3/min to 5 dm^3/min caused 38.8 % reducing of the pressure setting times T_R. The increasing of the steering pump output flow rate above 5 dm^3/min is profitless.

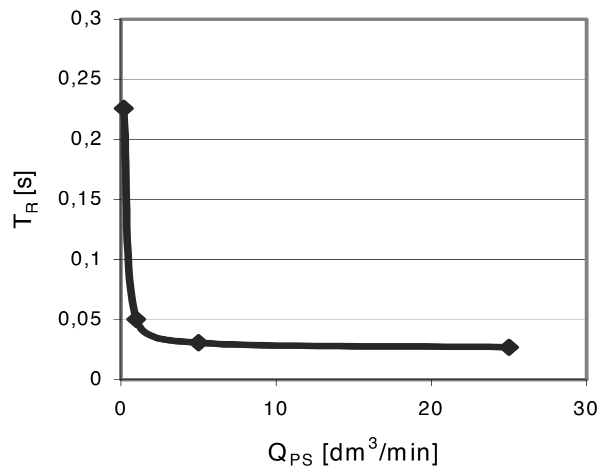

Fig. 21. Pressure setting times T_R for current step

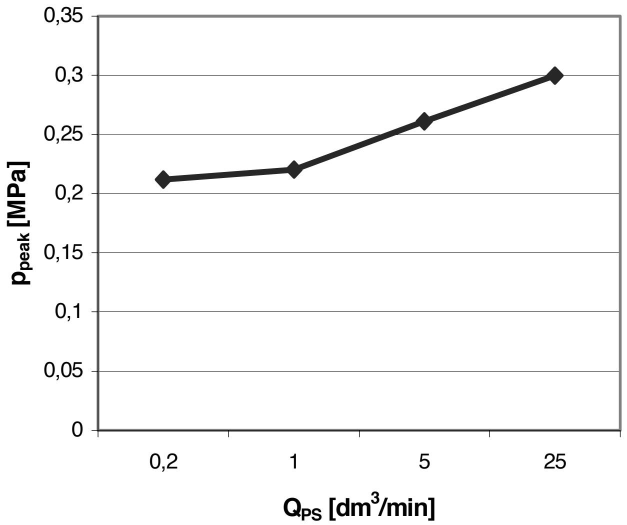

Fig. 22. The peak pressures produced by current step for different inlet volumes

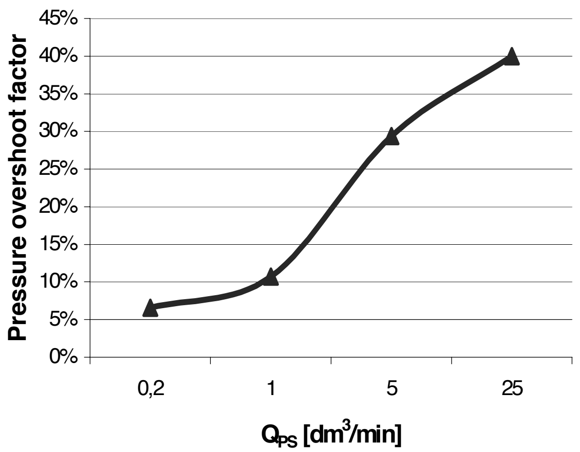

Fig. 23. The pressure overshoot factor for current step

The results, showed in Figs. 20 and 22, indicated that the growth of the steering pump output flow rate increased the peak pressure at the instant of current step.

In Fig. 23 the pressure overshoot factor is presented.

4. Conclusions

The simulation results, presented in this paper, indicated that presented new low pressure relief valve characterized by high useful properties. For accepted construction parameters of the modelled valve the high precision of the operating pressure (pressure error below 5%) and short time responses (0.01-0.02 s) were received. As the disadvantage of the modelled valve is the high value of the peak pressures. However, the peak pressure can be reduced by the pressure circuit volume increasing or the hydraulic accumulators using as well as shock absorbers. Also, the increasing of the circuit pipes flexibility (by the hoses using) the resulted in the reducing of the peak pressures.

The dynamic simulation with the ideal step excitation were realized. In real hydraulic systems the immediate flow steps are impossible. The finite rise time of flow resulted in the decreased of the real peak pressures. For example, 5 ms rise time of flow from 10 to 120 dm^3/min five times reduced the pressure overshoot (by comparison with ideal flow step).

Results of the testing of the proportionally operated version of the valve indicated that the optimum value of the steering pump output flow rate is 1 dm^3/min.

Recapitulated, the prototype constructing of the new low pressure relief valve and the experimental researching are arguable.

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