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Research paper

TRANSIENT PERFORMANCE OF A THREE-WAY PRESSURE REDUCING VALVE

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Abstract: *The article presents experimental and theoretically simulated transient processes with serially produced three-way reduction valve with direct control NG6 of the company BoschRexroth. This class of valves is used to maintain a reduced pressure of constant level in a secondary circuit of the hydraulic system, independent of change the pressure of primary circuit or change of the flow through the valve. This paper deals with modeling of the transient processes of threeway pressure reducing valves by changes of the flow through the valve. The validity of the proposed model is assessed experimentally. The study aims to point out the transient change of the clamping force in hydraulic systems with three-way pressure valve.*

Key words: *three-way reducing valve, dynamic characteristics, transient performance.*

1. INTRODUCTION

Pressure reducing valves are applied to control the tightening force of robots grippers, weight balance of moving parts, clamping devices of machine tools, energy saving in the hydraulic systems of road transport machine and many other hydraulic systems.

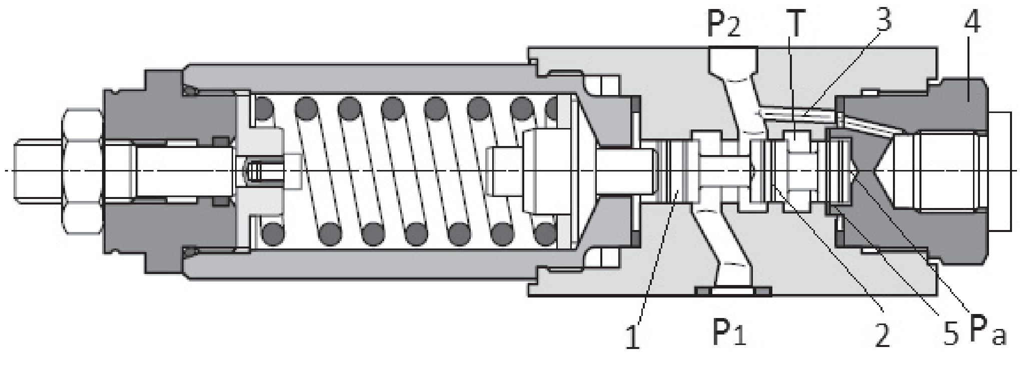
Few publications are found to deal with the performance of this class of valves. Osama Gad [1] developed a mathematical model and assessed experimentally in the steady-state and transient modes of three-way reducing valve by changes of the pressure of primary circuit in hydraulic system. Dieter Will and Norbert Gebhardt [2] developed a mathematical model of two-way reducing valve by changes of the flow through the valve, without validated experimentally. Storr [3] and Zimmerman [4] in their dissertations developed a dynamic model for a pilot operated pressure reducing valves.

The aim of the present study is to determine the variation of the reduced pressure in the transient processes in hydraulic systems of clamping devices in case of a sudden change of the flow through the reducing valve.

By the moving of hydraulic cylinder in the tightening direction, the flow rates passing through the reducing valve sudden changes its value to zero, when the stroke of the hydraulic cylinder reaches the clamped part. The change of the flow rates causes an increase of the pressure in the volume V of the hydraulic cylinder and the tightening force, respectively.

2. VALVE COMPONENTS DESCRIPTION AND SCHEMATIC DIAGRAM FOR THE INVESTIGATION OF TRANSIENT PROCESSES

Fig. 1a shows the design of the studied three-way pressure reducing valve of company BoschRexroth, size NG6 (type 0811150233). The middle spool land 2 of the control spool 1 (Fig.1b) has a positive overlap of the two control edges l_0 ($l_0 \approx 0{,}1\ mm$), one between the inlet pressure p_1 and the reduced pressure p_2, the second between the reduced pressure p_2 and the tank T. There are control notches at the edges of the control spool with circle segments with a radius of $r_0 = 3\ mm$ and width of $a = 0.75\ mm$ and $b = 0.35\ mm$ (Fig.1b). The movement of the control spool 1 is damping by hydraulic resistance of the line 3 and the gap 5 between the closure nut 4 and the cylindrical surface of the spool 1 (Fig.1a).



a)

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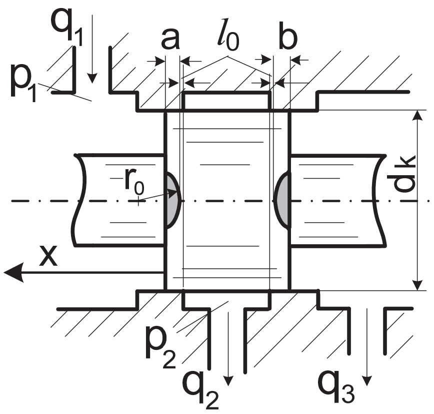


b)

Fig.1. Three-way pressure reducing valve (BoschRexroth)
a) design; b) control spool land

A schematic diagram for determining the transient processes is shown in Fig.2. The pump transmits flow of q_0 = *40 l /min* to the pressure relief valve 1 and the pressure reducing valve 2. The pressure relief valve 1, type BoschRexroth NG6 is set to p_{10}= *80 bar*. The volume of fluid in the pipelines between the two valves is V_1 = *95* cm^3. The pressure reducing valve 2 is set to the p_{20}= *20 bar*, with a flowrate of q_{20}= *10 l/min*. The flow q_2 passes through the pipeline with the linear resistance r_p and the inertial resistance L_p to the volume *V* and to the directional control valve 3. The outlet of the directional control valve 3 is connected to the tank *T* by a two-way flow regulator 4. The volume *V* corresponds to the chamber of piston side of the hydraulic cylinder in which the reduced pressure acts during the clamping. The researches were carried out at three levels of the fluid volume *V = 105, 250* and *700* cm^3.

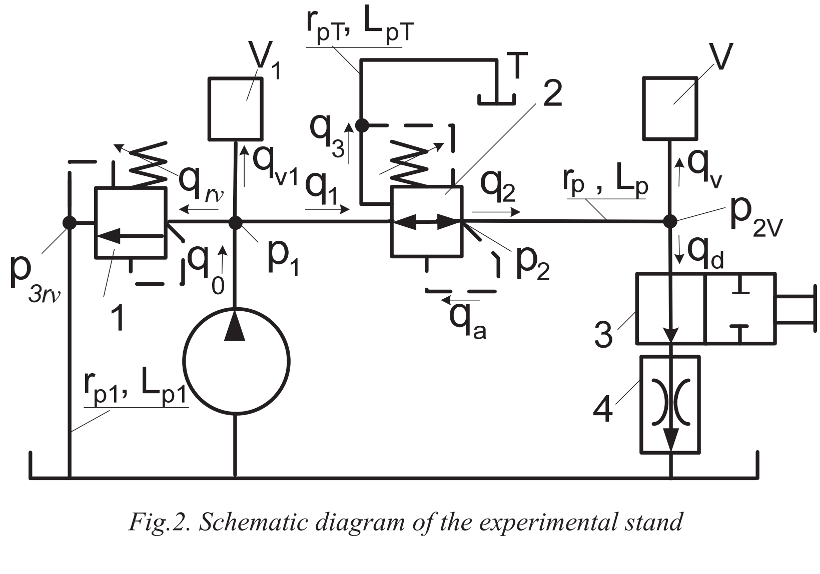


Fig.2. Schematic diagram of the experimental stand

A transient response process in the system on Fig.2 is created with the closing of the directional control valve with mechanical control 3, by means of a knock. In this way, a sudden reduction to zero of the velocity of the piston of the clamping hydraulic cylinder at the end of its stroke is simulated. The closing time of the directional control valve 3 (Fig.2) in this study is about t_1 = *3 ms*.

With sudden closing of the directional control valve, the flow q_d is reduced to zero and the transient process begins in the system. The flow q_2 from the pressure reducing valve keeps entering through the pipeline in the volume *V*, and increases the pressure $p_{2v,max}$ to its maximum peak. Increasing the pressure p_2, the control plunger 1 of the pressure reducing valve (Fig.1) shifts to the closing position thus the flows q_1 and q_2 are reduced. This leads to change of the flow q_{rv} through the pressure relief valve, which increases to the flow of the pump q_0. After the transient process the inlet pressure is stabilized to a value of p_{1e} = *88 bar* [5].

The control plunger 1 of the three-way pressure reducing valve exceeds the position $x = 0,\ q_1 = 0$ and the control edge (Fig.1a) opens to the tank T. The pressure reducing valve now works as a pressure relief valve, through it beginning to pass a flow rate of q_3 from the volume V and the pressure p_2 quickly decreases. After the transient response process passes, the stationary values of the reduced pressure p_{2e} *=30 bar*, of the displacement of the plunger *x=0* and of the flow q_2*=0* have been established. Thus the error of the static characteristics of the pressure reducing valve forms $\Delta p_{2,st} = p_{2e} - p_{20} = 10\ bar$.

3. MATHEMATICAL MODELLING OF THE TRANSIENT RESPONSE

The equations describing the dynamic processes of the three-way pressure reducing valves vary depending on the displacement *x* and the corresponding changes in the cross section of the throttle edges. When the valve operates as a pressure reducing, the displacement of the control plunger is negative and changes from a starting value x_0 to the positive overlap area $-l_0 \le x < l_0$. When the valve operates as a pressure relief valve, the displacement has a positive value of $x \ge l_0$.

3.1. Mathematical model in reducing mode of operation of the valve

■ *The equation of flowing through throttle areas*

$$\begin{aligned}
&q_2 = C_{d1}.A_1.\sqrt{\frac{2}{\rho}p_{1,2}} - A_k.\frac{dx}{dt}, && \text{if } x < -l_0 \\
&q_2 = A_k.\frac{dx}{dt}, && \text{if } -l_0 \le x < l_0 \qquad (1)\\
&q_2 = -C_{d2}.A_2.\sqrt{\frac{2}{\rho}p_{2,3}} - A_k.\frac{dx}{dt}, && \text{if } x \ge l_0
\end{aligned}$$

where: $A_1(x)$,$A_2(x)$ the throttling areas; C_{d1}, C_{d2} – the discharge coefficients; $A_k = \pi d_k^2/4$ – the area of the control plunger with the diameter $d_k = 10\ mm$.

■ *The equation of motion of the control plunger*

$$\begin{aligned}
&m\frac{d^2x}{dt^2} + c(s_0 + x) - F_{J1} = A_k p_a \pm F_f, && if\ x < -l_0 \\
&m\frac{d^2x}{dt^2} + c(s_0 + x) = A_k p_a \pm F_f, && if\ l_0 \le x < l_0 \qquad (2)\\
&m\frac{d^2x}{dt^2} + c(s_0 + x) + F_{J2} = A_k p_a \pm F_f && if\ x \ge l_0
\end{aligned}$$

where: $m = m_b + \frac{1}{3}m_s$ – the mass of the plunger m_b and of the spring m_s; $c = 83{,}65\ N/mm$ - the spring stiffness; $s_0 = \frac{A_k p_{2e}}{c}$ – the deformation of the spring when $x = 0$, $q_2 = 0$ and the value of the reduced pressure of $p_2 = p_{2e}$; $p_a = p_2 - r_d.A_k.\frac{dx}{dt} - L_d.A_k.\frac{d^2x}{dt^2}$ – the pressure in the control cavity of the control plunger 1, determined by the pressure p_2 and the pressure losses from the linear and the inertial resistance in the damping channels 3 and 5 (Fig.1a).

$$\begin{aligned}
&F_{J1} = 2C_{d1}A_1 cos\theta_1.p_{1,2} = A_{h1}.p_{1,2}, && if\ x < -l_0 \\
&F_{J1} = 0;\ F_{J2} = 0, && if\ l_0 \le x < l_0 \qquad (3)\\
&F_{J2} = 2C_{d2}A_2 cos\theta_2.p_{2,3} = A_{h2}.p_{2,3}, && if\ x \ge l_0
\end{aligned}$$

-the jet forces in reducing and relieving mode, expressed by the respective areas $A_{h1}(x)$, $A_{h2}(x)$ and the pressure drops $p_{1,2}$ and $p_{2,3}$; θ - the flow streamline angle; F_f- the friction force that due to its low value can be neglected.
The values of the discharge coefficient C_d and the flow streamline angle θ in (1) and (3) depend on the design of the pressure reducing valve. They can be determined by the methodologies set out in the works [6], [7] and [8].
Another approximate method for determining q_1, q_3 and $A_{h1}(x), A_{h2}(x)$ is the use of the experimental static characteristics $p_2 = f_2(q_2)$ and $x = f_3(q_2)$ of the studied valve, shown on Fig.3.
The characteristics of Fig.3 can be considered as being composed of three parts, assuming constant values of the discharge coefficient C_d and the flow streamline angle θ in each section.

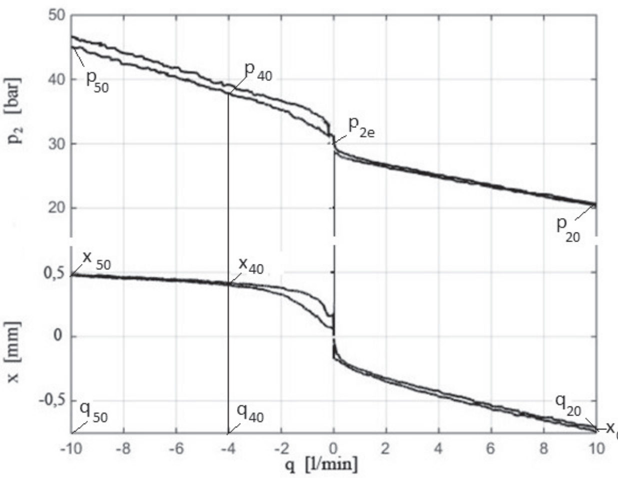


Fig.3. Valve Steady-State Characteristics $p_2 = f_2(q_2)$ *and* $x = f_3(q_2)$

In the *first section* $x_0 \le x < -l_0$ the throttling area is a segment with maximum area rate $A_a = 2.1{,}33.\sqrt{2r_0}.a^{3/2}$ [9]. For the displacement x_0, the area rate is $A_1(x_0) = A_a.\left|\frac{x_0 - l_0}{a}\right|^{3/2}$. Fig.3 shows the starting point values of $q_{20} = C_{d1}.A_1(x_0).\sqrt{\frac{2}{\rho}(p_{1,2})_0}$.
For a given displacement x at constant discharge coefficient C_{d1} in the section, the flow q_2, which is multiplied and divided by q_{20}, is $q_2 = C_{d1}.A_1(x).\sqrt{\frac{2}{\rho}p_{1,2}} = q_{20}.\left|\frac{x - l_0}{x_0 - l_0}\right|^{3/2}.\sqrt{\frac{p_1 - p_2}{(p_{1,2})_0}}$. The area of the jet force for the value of x_0 is defined by the expression $A_{h0} = \frac{c(s_0 + x_0) - A_k.p_{20}}{p_{10} - p_{20}}$. At constant value of the discharge coefficients C_d and the flow streamline angle θ, the area of the jet force is $A_{h1}(x) = A_{h0}.\left|\frac{x - l_0}{x_0 - l_0}\right|^{\frac{3}{2}}$.
In the *second section* $l_0 \le x < b + l_0 = x_{40}$ which is also a segment throttling area with height *b*, we similarly get

$A_{2,1}(x) = A_b.\left|\frac{x-l_0}{b}\right|^{\frac{3}{2}}$, $q_2 = -q_{40}.\left|\frac{x-l_0}{b}\right|^{3/2}\sqrt{\frac{p_2-p_3}{p_{40}-p_{3,10}}}$,

$A_{h40} = \frac{A_k.p_{40}-c(h_o+x_{40})}{p_{40}}$ and $A_{h2,1}(x) = A_{h40}.\left|\frac{x-l_0}{b}\right|^{3/2}$,

where $p_{3,10}$ – the pressure value at the beginning of the pipe connecting the pressure reducing valve and the tank at a flow rate of q_{40}.

In the *third section* $x \geq x_{40}$, the throttling area is opening across the whole periphery $\pi.d_k$ and the area is changing linearly of the displacement *x*. The corresponding expressions are as follows: $A_{2,2}(x) = A_b + \pi.d_k.(x - x40,$

$q_2 = -q_{50}.\frac{A_b+\pi.d_k.(x-x_{40})}{A_b+\pi.d_k.(x_{50}-x_{40})}.\sqrt{\frac{p_2-p_3}{p_{50}-p_{3,20}}}$, $A_{h50} = \frac{A_k.p_{50}-c(h_o+x_{50})}{p_{50}}$ and $A_{h2,2}(x) = A_{h40} + (A_{h50} - A_{h40}).\frac{x-x_{40}}{x_{50}-x_{40}}$. The pressure $p_{3,20}$ is determined for the flow rate of q_{50} through the pipeline.

- *The equation of the flowing process in the pipeline connecting the valve and the volume V [9]*

$$p_{2v} = p_2 - r_{p,l}q_2 - r_{p,m}q_2^2 - L_p\frac{dq_2}{dt}, \quad (4)$$

where: $r_{p,l}$ – the coefficient of linear laminar friction loss; $r_{p,m}$ – the coefficient of local loss; L_p – the inertia resistance of the *8 mm* diameter pipeline and *0.85 m* long.

- *The continuity equation of the volume V*

$$\frac{V}{K}\frac{dp_{2v}}{dt} = q_1 - A_k\frac{dx}{dt} - q_d, \quad (5)$$

where: $q_d = q_{20}(1-\frac{t}{t1})$ for $t \leq t_1$ and $q_d = 0$ for $t > t_1$ - the approximate variation of the flow through the directional control valve 3 (Fig.3) in the transition process; *K* – the bulk module of the oil.

- *The equation of the flowing process in the pipeline connecting the valve and the tank T*

$$p_{3T} = r_{pT,l}.q_3 + r_{pT,m}.q_3^2 + L_{pT}\frac{dq_3}{dt}, \quad (6)$$

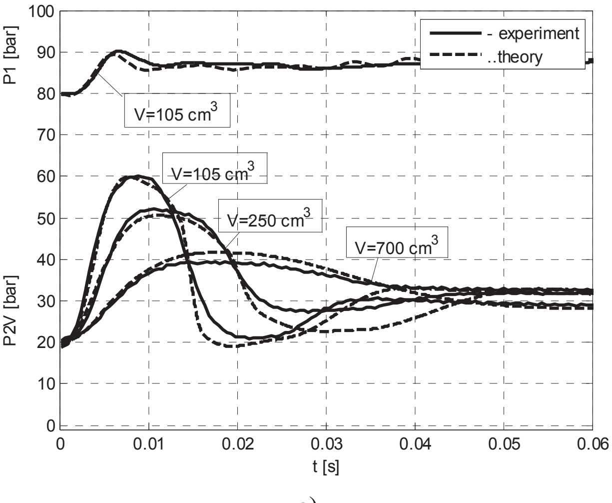
where: $r_{pT,l}$ – the coefficient of linear laminar friction loss in the section; $r_{pT,m}$ – the coefficient of local loss in the section; L_{pT} – the inertia resistance of the *8 mm* diameter pipeline and *0.9 m* long.

3.2. Dynamics of the pressure relief valve

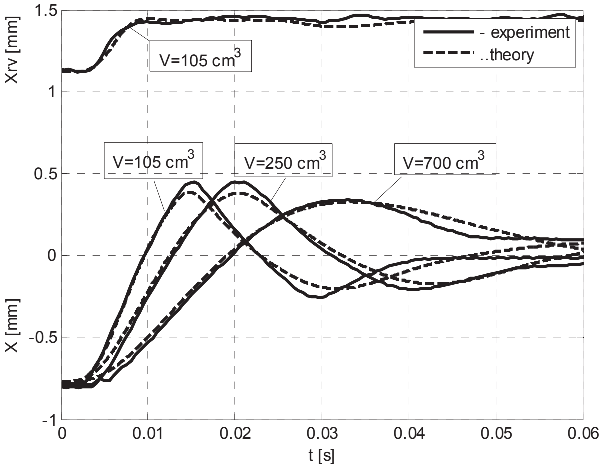
The simulation of the transient process of the pressure relief valve, size NG6 (BoschRexroth) with a volume of oil at inlet port V_1 and a pipeline at outlet port was carried out according to the method presented in [5].

4. EXPERIMENTAL AND THEORETICAL TRANSIENT RESPONSE

The experimental research of Bosch Rexroth's three-way direct-acting pressure reducing valve NG6 is carried out according to the diagram shown in Fig.2. The pressures , p_{2V} and the displacements of the control spools of the pressure reducing valve x and the pressure relief valve x_{rv} with pressure transducers and displacement transducers, products of Bosch, were measured. The flows through the pressure reducing valve and pressure relief valve have been measured in static mode of operation by means of a rolling vane hydraulic motor type Indramat SM110.1. The experimental data were recorded on a computer by means of the National Instrument Data Acquisition Card NI USB-6009 and are plotted in Fig.4.
It can be noticed well matching of the theoretically simulated transient processes with the experimental ones.
In the transient process the pressure p_{2V} with the volume $V = 105\ cm^3$ is increasing from the starting value of $p_{20} = 20\ bar$ to the maximum value of $p_{2V,max} = 60\ bar$ and then it is established at the stationary value of $p_{2e} = 30\ bar$ (Fig.4a). This double rise of the pressure in the transition process would also correspond to a greater clamping force of the clamped part. An increase in the value of the pressure p_{2V} occurs also at the other volume values *V*. Increasing the volume *V*, the maximum pressure value $p_{2V,max}$ decreases.
Fig.4a also shows the pressure change at the inlet port p_1 only for the volume of *V=105 cm³* in the cylinder 3. At the other volumes of oil *V = 250 cm³* and *700 cm³*, the transient processes of the inlet pressure are very close to these with volume of oil *V=105 cm³* and are not shown on Fig.4.
It can be seen from Fig.4 that the maximum pressure values $p_{2V,max}$ are obtained when the values of the displacement of the pressure reducing valve is *x = 0*. At this point the connection to the inlet pressure p_1 is interrupted.



a)



b)

Fig.4. The experimental and theoretical transient response processes at different volumes V

The maximum values of the displacement in the relief mode of operation x_{max} are approximately the same, close to the segment height of the throttling area $b = 0.35\ mm$ (Fig.1b).

The relatively large pressure variation at the inlet port p_1 of *10 bar*, determined by the transient process of the pressure relief valve, has little impact on the changes of the reduced pressure p_{2V}. This is illustrated in Fig.5, which shows the comparison of the experimental transient processes of volume $V = 105\ cm^3$ from Fig.4a and the theoretical simulation of the transition processes at constant inlet pressure $p_1 = const$.

As can be seen from Fig.4b, the displacement of the control spool of the pressure reducing valve x at a volume $V = 105\ cm^3$ is performed rapidly, from $x = -0.75\ mm$ to the value of $x = 0$ for only *0.008 s*. Beginning from this point up to *0.0225 s* the pressure reducing valve works as a pressure relief valve, whereby the pressure connection with the inlet p_1 is interrupted. This fact explains the very small variation of the maximum pressure value $p_{2V,max} = 60\ bar$ in the theoretical transient process with constant inlet pressure shown in Fig.5.

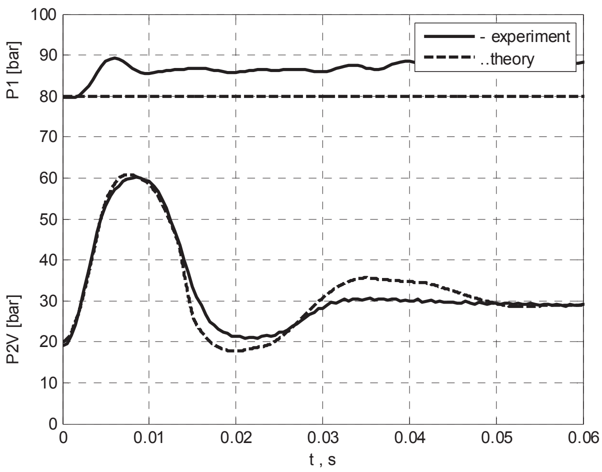


Fig.5. The transient process at variable and constant pressure p_1

The time of the transient process at $V = 105\ \text{cm}^3$ of the studied three-way pressure reducing valve is approximately $t_0 = 0.035\ \text{s}$ (Fig.4) to achieve the set value of $p_{2e} = 30$ bar.

An approximate estimation of the variation of this time can be done when two-way pressure reducing valve is used with the outlet port connected with the tank by the throttle with constant resistance. If it is assumed the flow rate through this throttle is $q_T = 20\ \frac{cm^3}{min}$ at constant pressure $p_2 = 60\ bar$, the coefficient of the local pressure loss will be $r_{dr} = \frac{p_2}{q_T} = \frac{60.10^5.60}{20.10^{-6}} = 18.10^{12}\ \frac{Ns}{m^5}$. The time for the decreasing of the pressure from $p_{2V,max} = 63\ bar$ to $p_{2e} = 30\ bar$ will be $t_{10} = r.\frac{V}{K}.ln\frac{p_{2V,max}}{p_{2e}} = 0.95\ s$, i.e. around 20 times higher than using three-way pressure reducing valve.

A transient response at the double lower flow of $q_{20} = 5\ l/min$ through the pressure reducing valve in presented at Fig.6. The maximum value of the reduced pressure decreasing from 60 bar to 42 bar.

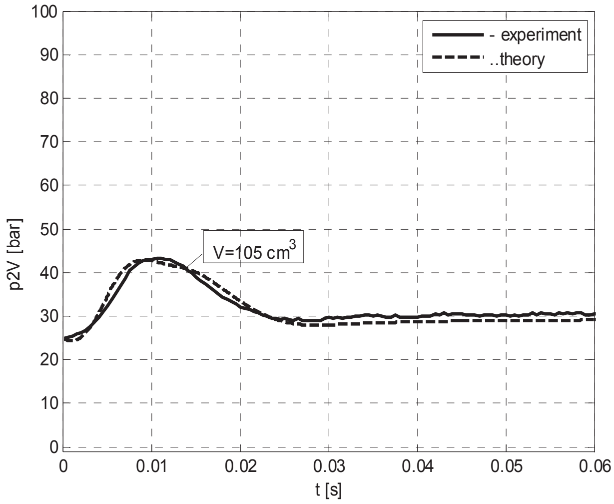


Fig.6. The transient process at $q_{20} = 5\ l/min$

5. CONCLUSION

The transient processes with change of the flow rate through the three-way direct-acting pressure reducing valves show a relatively high momentum increase of the reduced pressure and respectively of the clamping force in the transition processes.

In the simulation of the transient processes of the three-way pressure reducing valves with throttle area on the control spool with complex shape, the experimental steady-state characteristics of the pressure and the displacement of the control spool in dependence on the flow can be used.

Reduction of the maximum value of the reduced pressure in the transient processes can be achieved by reducing the flow rate respectively the velocity speed of the clamping piston.

Input pressure variation in the transition process has little effect on the maximum value of the reduced outlet pressure.
Decreasing of the variation of the reduced pressure in the considered transient processes can also be achieved introducing pilot operated pressure reducing valves. However, this leads to an increase in energy losses in the system due to the continuous flow of pilot oil through the valve towards the tank.

REFERENCES

[1] Osama G. (2016). *Modeling and Simulation of the Steady-State and Transient Performance of a Three-Way Pressure Reducing Valve*, Journal of Dynamic system, Measurement, and Control. Vol. 138

[2] Will D., Gebhard N. (2011). *Hydraulik*, Springer-Verlag, Berlin

[3] Storr A. (1967). *Beitrag zur Klärung des dynamischen Verhaltens vorgesteuerter, ölhydraulischer Druckregelventile*, Dissertation, Technische Hochschule Stutgart

[4] Zimmerman M.L. (1984). *Untersuchung des statischen und dynamischen Verhaltens vorgesteuerter Druckminderventile*, Dissertation, RWTH Aachen

[5] Dimitrov S., Komitovski M. (2013). *Static and Dynamic Characteristics of direct operated Pressure Relief Valves*, Machine Design, Vol.5

[6] Stone J.A. (1960). Discharge Coefficients and Steady-State Flow Forces for Hydraulic Poppet Valve, Transactions of the ASME

[7] Woben, D. (1978). *Strömungskräfte an 2-Wege-Einbauventilen*, Indusrie Anzeiger, Nr.22, v.17.3.

[8] Altare G., Rundo M., Olivetti M. (2016). *3D Dynamic Simulation of a Flow Force compensated Pressure Relief Valve*, Proceedings of the ASME, November 11-17.

[9] Komitovski M. (1985). *Hydraulic and Pneumatic Elements (in Bulgarien)*, Verlag Technika. Sofia