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Experimental research characteristics of counterbalance valve for hydraulic drive control system of mobile machine

***Abstract**. Experimental studies of counterbalance valve for the hydraulic drive control systems of mobile machines are presented. According to the results of theoretical research experimental prototype of the counterbalance valve is developed. Experimental stands are elaborated, where leak tightness degree was studied and static characteristics were obtained. Dependences of operating fluids supply on the parameters of the pressure difference at its basic spool and load pressure are plotted. Transient processes in the hydraulic drive control system are analyzed, adequacy of the theoretical calculations to the elaborated mathematical models for the control system of mobile machine hydraulic drive is proved.*

***Streszczenie.** Przedstawiono badania eksperymentalne zaworu równoważącego w układach sterowania napędem hydraulicznym maszyn mobilnych. Zgodnie z wynikami badań teoretycznych opracowano eksperymentalny prototyp zaworu równoważącego. Opracowano stanowiska eksperymentalne, na których badano stopień szczelności oraz uzyskano charakterystyki statyczne. Narysowano wpływ zasilania płynów eksploatacyjnych na różnicę ciśnień na szpuli podstawowej i ciśnienie obciążenia. Przeanalizowano procesy przejściowe w układzie sterowania napędem hydraulicznym, udowodniono zgodność obliczeń teoretycznych z opracowanymi modelami matematycznymi dla układu sterowania napędem hydraulicznym maszyny mobilnej. **(Charakterystyki eksperymentalne zaworu równoważącego układu sterowania napędem hydraulicznym maszyny mobilnej)**.*

Keywords: counterbalance valve, experimental stand, leak tightness degree, static characteristics, transient processes.
Słowa kluczowe: zawór równoważący, stanowisko doświadczalne, stopień szczelności, charakterystyki statyczne, procesy przejściowe.

Introduction

Mobile machines with hydraulic drive are widely used in industry, transport branch, agriculture [1-3]. Such machines, as a rule, are equipped with high variety of removable operating tools. It enables to use them during greater part of calendar year for performing various operations, such as digging, materials handling, loading operations with bulky materials. Systems of hydraulic drives control of such machines for energy saving, must provide the possibility of regulation of operation organs motion speed and working pressure of pumps in wide ranges. It is important to provide the reliable motion control of the executive device in case of alternating load. Hydraulic drives, based on spool hydraulic separators are widely used in mobile machines. In the process of operation such spools wear, this reduces the degree of leak tightness of hydraulic lines and complicates the operator's control over the position of the executive device with the load. To provide the possibility of the control over the motion of the executive device and to decrease nonproductive power expenditures in case of the alternative loading counterbalance valves (CBV) are used in the hydraulic drives of mobile machinery. At the same time to provide high degree of leak tightness of operating hydraulic lines controlled check valves are used. Available CBV and controlled check valves increase the cost of the system of hydraulic drive control, its dimensions, mass and operation cost of the mobile machinery [1-7].

The authors elaborated the construction of CBV that simultaneously provides the possibility of the motion speed control of the executive device at the alternative loading and increases the leak tightness degree of the operating hydraulic lines of the hydraulic drive. The following problems are solved to realize the elaborated CBV construction:

- provision of high degree of leak tightness;
- statistic characteristics study;
- validation of the adequacy of the developed mathematical model to real physical processes, occurring in the device.

Analysis of the studies of the hydraulic drives control systems of mobile machinery

Introduction of CBV in the system of mobile machines hydraulic drive control enables to increase the leak tightness degree of operating hydraulic lines. According to the catalogues of such well-known companies as Hidromek, Ponar Wadowice, Bosch Rexroth, Eaton, Oleostar the leak tightness degree of their CBV is 0.25–0.4 ml/min. If we take into account, that the leak tightness degree of spool hydraulic separator is up to 15 ml/min, then the application of CBV will decrease considerably the sagging of the executive device with the load in the process of operation.

Counterbalance valves are installed at various mobile machines, where the alternative load (as a rule, it is accompanying load) must be controlled. In [4] the characteristics of CBV, installed in the system of hydraulic drives control for offshore knuckle boom crane are studied. Mathematical model that includes both the crane's mechanical system and the electro-hydraulic motion control system is developed. A novel black-box model for counterbalance valves is presented, which uses two different pressure ratios to compute the flow through the valve. By means of the experiments mathematical model of the crane is calibrated and verified. The mathematical model allows the engineers to design similar installations if they do not have the complete access to the data of the control system components.

The effect of oscillations decrease in the hydraulic drive control system due to CBV application is shown [5]. The mathematical model for the novel concept of single boom actuated by a cylinder is elaborated. Experimental research, carried out, showed the adequacy of the developed mathematical model to real acting processes it describes. The flexibility of the mechanical system at the expense of continuous opening of the CBV was achieved. Friction in the hydraulic cylinder and proper characteristics of the directional control valve are determined.

Oscillations decrease impact on energy characteristics of the hydraulic drive control system was revealed in [6]. Construction and mathematical model of CBV is presented, experimental studies are carried out. The obtained results showed considerable energy losses during the provision of qualitative transient processes as compared with the transient processes, that have poor dynamic characteristics. However, installation of the servo-spool on CBV enables to decrease energy losses at main spool during the operation. Such principle is implemented in our model of CBV.

The advantages of CBV over valve-controlled cylinders for hydraulic drives control system at tote dumpers and lifters are shown in [7]. Experimental stand is developed, studies, showing its efficiency are carried out. Qualitative process of executive device control, energy losses decrease and operating fluid overheating is achieved. However, the hydraulic drive control system, based on CBV does not recycle energy [8,15,16,20-23].

Installation of CBV on the winch enables to avoid problems dealing with emergence of cavitation processes during its braking [8]. In particular, circuit, mathematical model and experimental stand for studying CBV operation in the system of the winch hydraulic drive control is developed. As the experimental prototype, real construction of CBV, manufactured by Bosch Rexroth Company is chosen. The results obtained, showed the efficiency of hydraulic drive control system, based on CBV and the discrepancy between theoretical and experimental calculations of 9%.

Simulation modeling [9-12,18,19,24-26] of hydraulic drive control system of mobile machines by means of program products MAPLE, MATLAB Simulink, ANSYS, Solidworks Flow Simulation, Autodesk Simulation CFD enables to solve complex engineering problems of hydraulic equipment design. Such method of the design is economically efficient and accurate. But simulation models require experimental verification of the results obtained to provide the adequacy
of decisions made and assumptions.

The analysis, carried out, shows that the greater part of CBV studies are performed outside the borders of Ukraine. For the domestic mobile machines either throttles with relief valve or relief valves are used. That is why, development and study of the CBV of Ukrainian manufacture with the improved technical and economic indexes will have demand not only at the Ukrainian market, but also at the world market.

Materials and methods of research of the counterbalance valve

On the base of the useful model patent of Ukraine [13] and theoretical research [14,15] circuit diagram of CBV (fig. 1) was developed, according to this circuit diagram experimental model (Fig. 2) was manufactured for studying in the system of hydraulic drive control.

Characteristic feature of CBV as compared with other valves of such type is the available servo-spool. It provides the operation of CBV with the functions of the relief valve in the positions of hydraulic separator which provide neutral position of the executive device and its lifting. When the hydraulic separator is switched in the position of lowering of the executive device, servo-spool is also switched and as a result, main spool operates performing the functions of CBV.

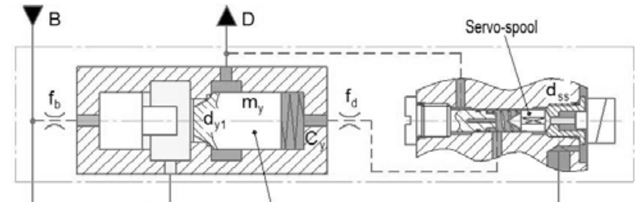

Fig. 1. Circuit diagram of CBV with servo-spool

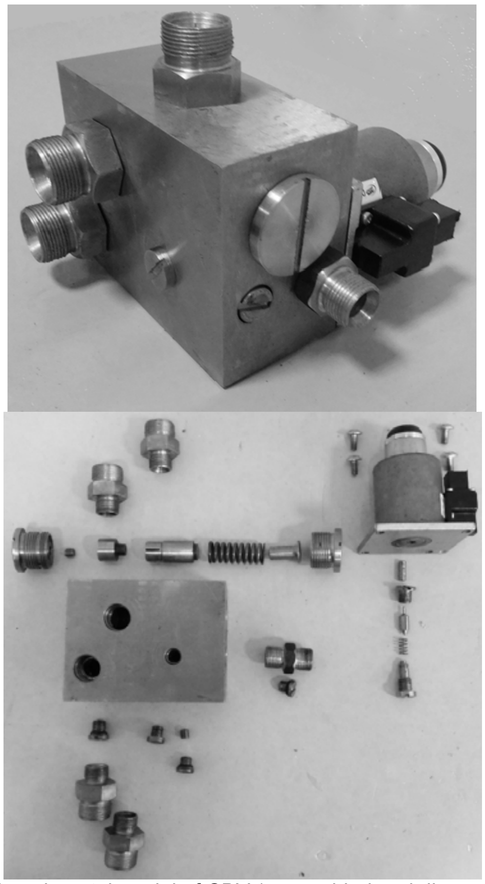
Fig. 2. Experimental model of CBV (assembled and disassembled)

For studying the leak tightness degree of the main spool and servo-spool of CBV, the scheme of the experimental stand I is developed (fig. 3). Experimental stand I contains the pump station 1, CBV 2, main spool 3, servo-spool 4 and measuring cylinders 5.

Experimental stand I is developed on the base of pumping station Г48-1, that contains constant delivery pump FVP* of НШ32-2 type with the working volume of $32 \cdot 10^{-6}$ m^3, hydraulic reservoir R*, asynchronous motor M with rotation rate of 1490 RPM, two coarse mesh filter F1* and fine mesh filter F2* with nominal filtration fineness of 25 and 10 μm, correspondingly, relief valves RV1* and RV2*. Industrial oil I-30A is used as the working fluid.

Losses of the working fluid were determined in the following manner. Working fluid was supplied from the constant delivery pump FVP* across the coarse mesh filter F1* to CBV 2. Pressure regulation in the hydraulic drive was carried out by means of relief valve RV1* and pressure gauge PG2*, with the accuracy of 0.16 and $p_{\text{мах}}$=2500 PSI. Working fluid, which was flowing across the main spool 3 and servo-spool 4, was fixed in measuring cylinders 5 (measuring accuracy is 0.02 ml). Amounts of working fluid losses ΔQ_{ms} and ΔQ_{ss} determined the degree of leak tightness of the main spool and servo spool, correspondingly.

For studying statistical and dynamic characteristics the experimental stand II (fig. 4) is developed [16]. Circuit of the experimental stand II consists of such main components: experimental model of CBV, supply system, loading system, measuring and registration system.

Characteristics of the experimental stand II elements are given in Table 1. The table also contains the accuracy indices of measuring and registration equipment, that is an important aspect of the efficient realization of the experimental research.

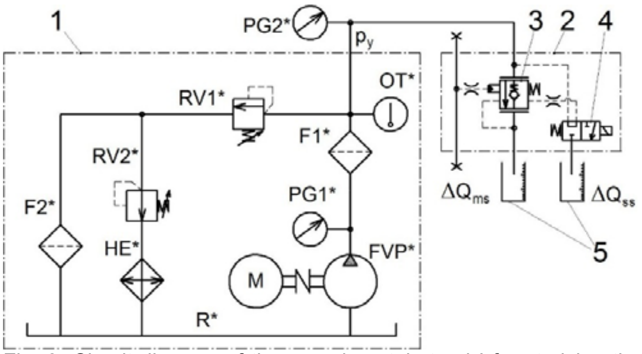

Fig. 3. Circuit diagram of the experimental stand I for studying the leak tightness degree of CBV spools

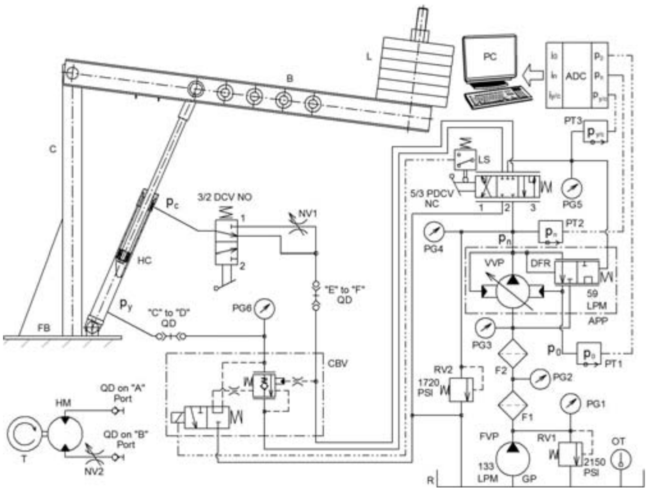

Fig. 4. Circuit diagram of the experimental stand II for studying statistical and dynamic characteristics of the hydraulic drive control system on the base of CBV

Determination of the statistical characteristics was performed by means of connection of hydraulic motor HM with the throttle needle valve NV2 and tachometer T. The experiment was carried out at least 3 times for measured value of rotation frequency of hydraulic motor HM shaft at various parameters of pressure p_y from loading to supply across hydraulic separator 5/3 PDCV NC. Value of pressure p_y from loading in the working hydraulic line was registered by the pressure gauge PG6. The feed value across the hydraulic separator 5/3 PDCV NC changed as a result of the displacement of its spool and determination of the corresponding area of the working window f of the hydraulic separator. The value of the rotation frequency of the hydraulic motor HM shaft and its characteristic working volume are presented by the feed value Q_y.

The dependence of the flow stabilization error δ on the area of the working window f of the hydraulic separator was calculated analytically by the formula:

$$(1)\quad \delta = \frac{\left|Q_{y\max} - Q_{y\min}\right|}{Q_{y\min}} \cdot 100\%,$$

where: $Q_{y\ max}$ and $Q_{y\ min}$ – are maximum and minimum values of the working fluid supply Q_y across CBV for one established position of the hydraulic separator 5/3 PDCV NC spool, but at different values of technological loading.

Also, by means of the installation with the hydraulic motor HM the change of the feed value Q_y across CBV on the value of pressure p_y and pressures difference Δp_y on the main spool was recorded. Pressure values were changed by the needle valve NV2 and were recorded on pressure gauges, installed before and after CBV. Rotation frequency of the hydraulic motor HM shaft corresponded to the feed value Q_y across CBV. The obtained table of measurement results was downloaded in the program product DataFit. The approximation was carried out to obtain the formula and graph of $Q_y=f(p_y, \Delta p_y)$ dependence.

Table 1. Experimental stand II parameters

Supply system	
Unregulated FVP gear pump GP	Flow up to 133LPM
Adjustable VVP axial-piston pump APP	Flow up to 59LMP
Relief valves RV1, RV2	Adjusted for pressure 2150 and 1720PSI accordingly
Pressure filters F1, F2	Nominal filtration fineness up to 25mkm
Industrial oil I-30A	Density 890kh/m^3, kinematic viscosity 41−51sSt at 40°C
Loading system	
Hydraulic cylinder HC	Stroke of the cylinder rod 40 cm
Hydraulic motor HM	Flow up to 50LMP
Load L	One cargo weighs 15 kg
Measuring and registration system	
Pressure gauges PG1...PG6	Measuring range 0…2500 PSI, accuracy 0,3 PSI
Pressure transducers RT1...RT3	Measuring range 50…2500 PSI, accuracy ±1,6%
Thermometer OT	Measuring range 0…150°C, accuracy 0,5°C
Tachometer T	Measuring range 0…1000 RPM, accuracy ±1%

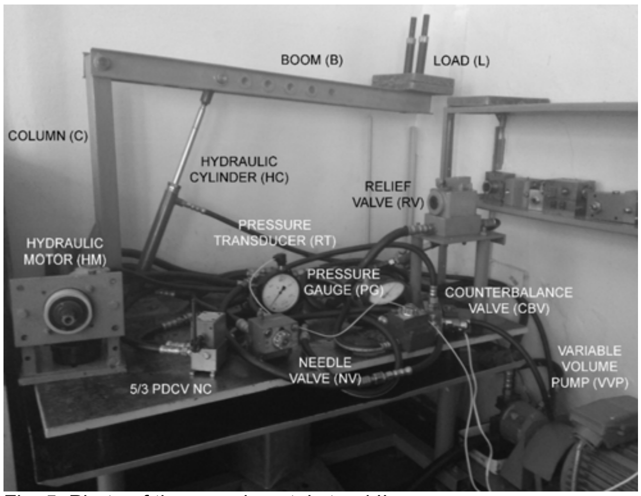

Fig. 5. Photo of the experimental stand II

Determination of the dynamic characteristics was performed by means of switching on of the hydraulic cylinder HC with the manipulator. Load L on the boom of the manipulator changed by adding or subtraction of the cargo. System of hydraulic drive control was switched on at various loads L, pressure oscillations in the working hydraulic lines were recorded by means of pressure transducers RT1...RT3. The obtained values were compared with the results of mathematical modelling for the developed mathematical model [14] to determine the adequacy of the developed mathematical model to real physical processes, which it describes.

Fig. 5 shows the external view of the experimental stand II during research, main elements are denoted. The research was carried out at the temperature of mineral oil 35-45°C, except the determination of leak tightness degree of CBV.

Results and discussion

Due to valve construction of CBV spools their leak tightness must be maximally high. This is provided by self-pressurizing under load pressure. Experimental research showed that the losses of working fluid ΔQ_{ss} across the servo-spool at the temperature T=30°C were not observed. Losses were not also observed when the temperature of mineral oil increased to T=55°C. Valve construction, qualitatively machined surfaces and small diameter of working window d_{ss}= 2.2 mm provided zero losses of working fluid.

For the main spool with the diameter of working part d_{y1}= 18 mm losses of the working fluid ΔQ_{ms} changed, depending on the value of pressure p_y, load and temperature T (fig. 6). For the pressure value p_y=5 MPa losses are maximal, and further they decrease due to self-pressurizing. Increase of oil temperature leads to the increase of working fluid losses across the main spool.

Thus, it is expedient to use the developed CBV on mobile machines, such as front side forklift while transporting of the technological load. Under the pressure in the hydraulic drive of 8–20 MPa it will provide high degree of leak tightness and prevent from load sagging. In the process of studying static characteristics, the impact of the pressure value p_y of the technological load on the feed value Q_y across CBV is determined (fig. 7).

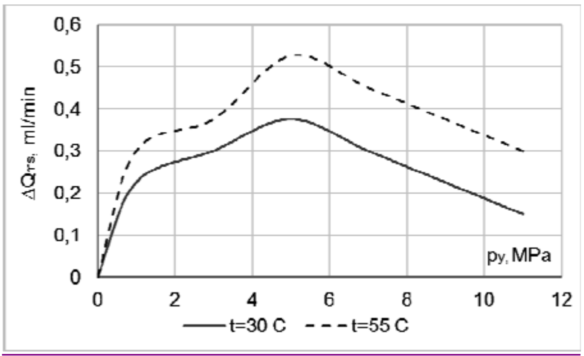

Fig. 6. Dependence of working fluid loss ΔQ_{ms} across the main spool on the pressure p_y, load and temperature

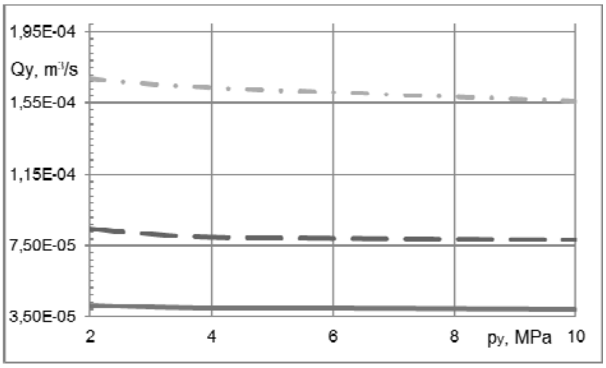

Fig. 7. Impact of pressure p_y value of the technological load on the feed value Q_y

By the results of the research the dependence of flow stabilization error δ on the area of the working window f of the hydraulic separator is calculated (fig. 8). Error of flow stabilization is δ=6-8%, it is acceptable index for the proportional hydraulic equipment.

The dependence of feed value Q_y=f(p_y, Δp_y) across CBV on the pressure value p_y and pressure difference Δp_y on its main spool is approximated. Determination factor for the dependence of Q_y=f(p_y, Δp_y) is R^2=0,989. Approximated dependence Q_y=f(p_y, Δp_y) has the form:

$$Q_y = c_0 + \frac{c_1}{\Delta p_y} + c_2 \times p_y + \frac{c_3}{\Delta p_y^2} + c_4 \times p_y^2 + \frac{c_5 \times p_y}{\Delta p_y}, \quad (2)$$

where: c_0, c_1, c_2, c_3, c_4, c_5 – are the coefficients.

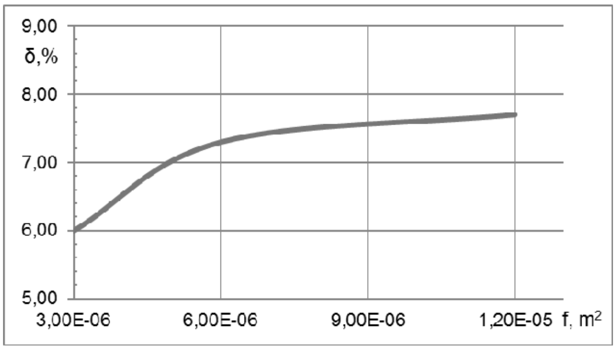

Fig. 8. Dependence of flow stabilization error δ on the area of the working window f of the hydraulic separator

Due to the complex profile of the working window of the main spool of CBV and in order to simplify the mathematical model [15], the feed value Q_y of CBV at various operation modes of the experimental stand II is measured. The obtained results are shown in fig. 9.

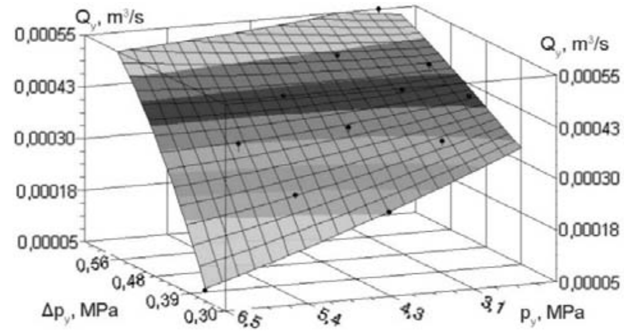

Fig. 9. Dependence of feed value Q_y=f(p_y, Δp_y) across counterbalance valve

Transient processes in the system of hydraulic drive control of the experimental stand II on the base of the manipulator are investigated. Fig. 10 shows the oscillograms of pressure value p_y(t) change in the working hydraulic line between the piston of the hydraulic cylinder and main spool of CBV. Switching of the hydraulic separator 5/3 PDCV NC was a disturbing factor. Research was carried out for the load on the executive device of the manipulator of 300 and 450 N. Dynamic characteristics are calculated: the value of overregulation σ and time of transient process t_p, shown in fig. 10. Load increase of 1.5 time resulted in the increase of the overregulation value σ – 1.68 time, and transient process time t_p – 1.14 time. In the process of the experimental research the value of overregulation σ did not exceed 30%, and the time of transient process t_p < 0.4c.

Theoretical graphs of transient processes of pressure value p_y'(t) change in working hydraulic line between the piston of the hydraulic cylinder and main spool of CBV are obtained by means of non-linear mathematical model [14]. The initial parameters of mathematical model and of experimental stand II are identical.

For validation of the mathematical model adequacy the error of the experimental was determined. Experimental oscillogram (fig. 10, b) was compared with the calculated one [14] in five points N=5. Besides, experimented oscillogram was obtained under the same conditions three times r=3. For assessment of pressure value p_y(t) deviations form its average value p_y(t) the dispersion of parallel experiments S_j^2 was calculated. The results of the experiments and their calculation parameters are given in Table 2.

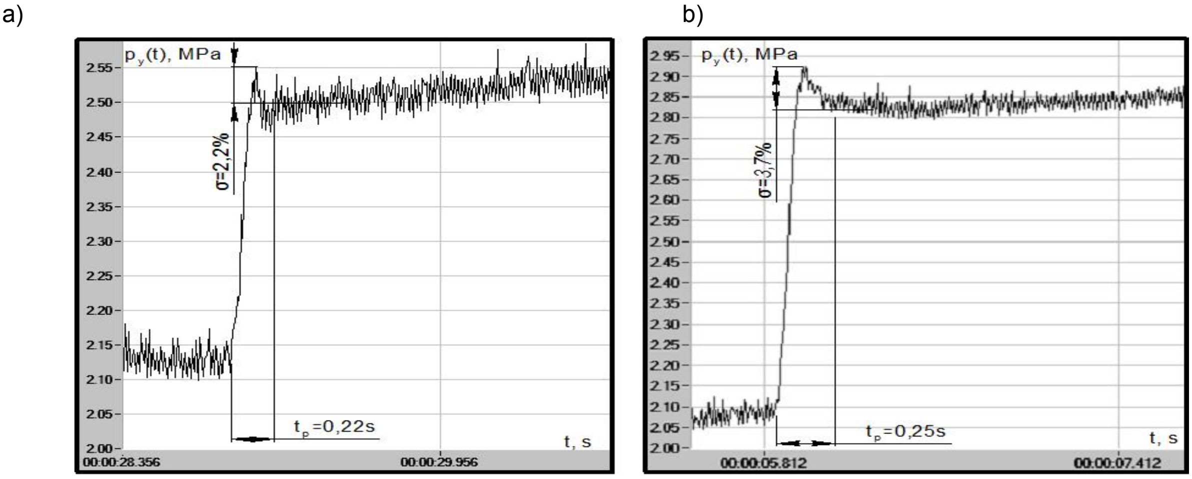

Fig. 10. Dynamic characteristics of CBV: a – the load of the boom of the manipulator is 300 N; b – the load of the boom of the manipulator is 450 N

Table 2. Results of the oscillograms experiences at the load of the boom of the manipulator of 450 N

Number	Value of pressure $p_y(t)\times10^6$ MPa				$y_1-\bar{y}$	$y_2-\bar{y}$	$y_3-\bar{y}$	$(y_1-\bar{y})^2$	$(y_2-\bar{y})^2$	$(y_3-\bar{y})^2$	S_j^2
	y_1	y_2	y_3	$\bar{y}$							
1	2.92	3.1	2.95	2.99	-0.07	0.11	-0.04	0.005	0.012	0.002	0.009
2	2.81	2.93	2.82	2.85	-0.04	0.08	-0.03	0.002	0.006	0.001	0.004
3	2.84	2.95	2.85	2.88	-0.04	0.07	-0.03	0.002	0.005	0.001	0.004
4	2.85	2.97	2.87	2.90	-0.05	0.07	-0.03	0.002	0.005	0.001	0.004
5	2.87	2.98	2.89	2.91	-0.04	0.07	-0.02	0.002	0.004	0.001	0.003
Σ				14.53	Σ						0.025

Table 3. Results of the calculation of adequacy dispersion

Number	$\bar{y}$	y'	$\bar{y}-y'$	$(\bar{y}-y')^2$
1	2.99	3.15	-0.16	0.026
2	2,85	2.76	0.09	0.008
3	2,88	2.82	0.06	0.004
4	2,90	2.86	0.04	0.002
5	2,91	2.9	0.01	0.0001
Σ				0.04

Uniformity of the parallel experiments dispersion was verified by Cochran's G-criterion:

(3) $G_{py}=S^2_{jmax}\Big/\sum_{j=1}^{N}S_j^2=0.009/0.025=0.372$.

Dispersion uniformity hypothesis is proved if the calculated value of the criterion does not exceed the table value [17]. Significance level of all the considered criteria is α=0.05. The probability P of the valid answer is:

(4) $P=1-\alpha=1-0.05=0.95$ or 95%.

Calculated value of the criterion was compared with the table value for the degrees of freedom of the numerator $f_1=r-1=3-1=2$ and denominator $f_2=N=5$. As $G_{tab}=0.684$ > $G_{py}=0.372$ [17], then the uniformity of parallel studies hypothesis is accepted. Thus, the dispersion of the reproducibility will be equal:

(5) $P=1-\alpha=1-0.05=0.95$ or 95%.

The error of the experiment is:

(6) $S(y)=\sqrt{S^2(y)}=\sqrt{0.005}=0.071$.

Table 3 contains calculated values for determination of adequacy dispersion, where $\bar{y}$ – arithmetic mean of the pressure $\bar{p}_y(t)$ value in the operating hydraulic line, obtained in the process of experimental studies; y' – is the value of pressure $p_y'(t)$ value in the operating hydraulic line, obtained in the process of mathematical modeling.

Adequacy of the mathematical model was evaluated applying Fisher test:

(7) $F_{py}=S^2_{ag}\big/S^2(y)$.

Then the adequacy dispersion is:

(8) $S^2_{ag}=\dfrac{1}{N-\beta}\sum_{j=1}^{N}(\bar{y}-y')^2=\dfrac{1}{5-1}0.04=0.01$.

Calculated Fisher's criterion by the formula 7:

(9) $F_{py}=0.01/0.005=2$.

By [17] critical (table) value of F-criterion is determined of the degrees of freedom of $f_1^*=N-\beta=5-1=4$ and $f_2^*=N(r-1)=5(3-1)=10$. As $F_{py}=2$ < $F_{tab}=3.5$, then mathematical model [14] is adequate.

Conclusions

1. Experimental model of CBV with servo-spool is investigated. The degree of leak tightness is ΔQ_{ms}=0-0.38 ml/min at the operation of the system of hydraulic drive control with the working pressure py up to 5 MPa and mineral oil temperature T=30°C, this corresponds the indices of CBV of the foreign models. When pressure value p_y is more than 5 MPa, leak tightness degree increases at the expense of self-pressurization. Self-pressurization is observed at the temperature of mineral oil of T=55°C. Working fluid losses are maximal ΔQ_{ms}=0.52 ml/min at the pressure value of 5 MPa. That is why, it is recommended to apply the developed CBV for mobile machinery such as front side forklift, where the working pressure in the process of cargo transportation can reach 8–20 MPa.

2. Flow stabilization error is δ=6-8% and corresponds to foreign analogs of CBV. By the results of the research of static characteristics the dependence of the feed value $Q_y=f(p_y, \Delta p_y)$ across CBV is approximated for the simplification of the mathematical model.
3. Transient processes during experimental research are analyzed: load change from 300 to 450 N leads to the increase of the overregulation value σ – 1.68 times, and the time of transient process t_p – 1.14 times. During the experimental research the overregulation value σ did not exceed 30%, and the time of transient process $t_p < 0.4$s. The adequacy of the developed mathematical model on the base CBV by Fisher test was proved.

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