Simulation of the testing process of reciprocating hydraulic cylinders with energy recovery

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Abstract. One of the most important problems of modern mechanical engineering is the energy efficiency of technological equipment and manufactured products. A separate issue in this series is the technology and means of testing finished product samples, including hydraulic cylinders, which can be carried out with passive or active energy recovery. The purpose of the work is to develop a test bench for piston hydraulic cylinders that provides recovery of part of the energy during the test, and at the same time the tests would take place in a mode as close as possible to the real conditions of their operation during their tests, as well as solving the problem of its modeling and calculation of the main functional parameters. In this article, a design scheme of a test bench for reciprocating hydraulic cylinders with active energy recovery and a method for mathematical modeling of the process of its functioning based on the application of the "theory of volumetric rigidity of hydraulic systems" is proposed, which makes it possible to describe with a high degree of accuracy and reliability the processes occurring in the test system during its operation. A scheme of a test bench for piston hydraulic cylinders with energy recovery is proposed. A mathematical model of the stand has been developed and an example of a preliminary calculation of its functioning is given. The proposed modeling method simplifies the modeling process and allows the use of numerical integration methods in calculations. The resulting mathematical model makes it possible to obtain rational parameters of the stand already at the design stage, without resorting to the need for expensive and labor-intensive field studies.

1 Introduction

At the current level of industrial production development, energy consumption is growing rapidly, both in the production process and at the stage of operation of technological equipment. One of the most expensive technological processes in the chain of development, research, design, production and operation of technological equipment, including the production of hydraulic drive systems and their elements, is the process of testing the finished

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product, or its experimental sample, for compliance with technical requirements. Especially costly is the process of resource tests, which are carried out at nominal loading of the equipment for a long time due to the standard service life of the equipment. In the process of testing hydraulic equipment when it is loaded, various types of braking devices (electrical, mechanical, hydraulic, etc.) are usually used, while all the energy supplied to the test system is irretrievably lost, in addition, a large amount of "harmful" heat is released into the environment as a result of the test. Naturally, the greatest energy losses (decrease in the energy efficiency of the testing process) occur when testing hydraulic machines and devices of high and medium power.

For a long time, scientists from all over the world, including Russian ones, have been looking for ways to solve this problem. As a result of these searches, a test method with energy recovery of hydraulic machines (pumps and motors) of rotational motion was proposed [1-3]. The proposed method has shown its effectiveness, and therefore stands have been developed on its basis for conducting tests with energy recovery and hydraulic reciprocating machines (hydraulic cylinders) [4-8].

2 Test bench for reciprocating hydraulic cylinders with energy recovery

The purpose of this work is to develop a test bench for piston hydraulic cylinders, which provides recovery of part of the energy during the test, and at the same time the tests should be carried out in a mode as close as possible to the real conditions of their operation, as well as solving the problem of its modeling and calculation of the main functional parameters.

Figure 1 shows the design scheme of the test bench for piston hydraulic cylinders, the test process on which has a high energy efficiency due to the recovery of part of the energy spent on the test. The essence of energy recovery in the process of testing hydraulic cylinders is based on the repeated conversion of energy supplied to the test system from the primary energy source – an electric motor.

The stand includes an electric motor 1, the shaft of which 2 is connected to the shaft 4 of the hydraulic pump 5 by means of a mechanical transmission 3. The tested hydraulic cylinders 9 and 12 are mounted on a frame 11 and connected kinematically by means of a mechanical transmission 10. The hydraulic distributor 7 controls the operation of the driving hydraulic cylinder 7, and the hydraulic distributor 13 directs the working fluid from the driven hydraulic cylinder 12 to the input of the hydraulic motor 14, whose shaft 17 is kinematically connected to the shaft of the hydraulic pump 4 by means of a mechanical transmission 18. The output of the hydraulic motor 14 is connected to its input through the check valve 16, and the input to the output through the pressure valve 15. The stand also contains a safety valve 6 and a hydraulic tank 8.

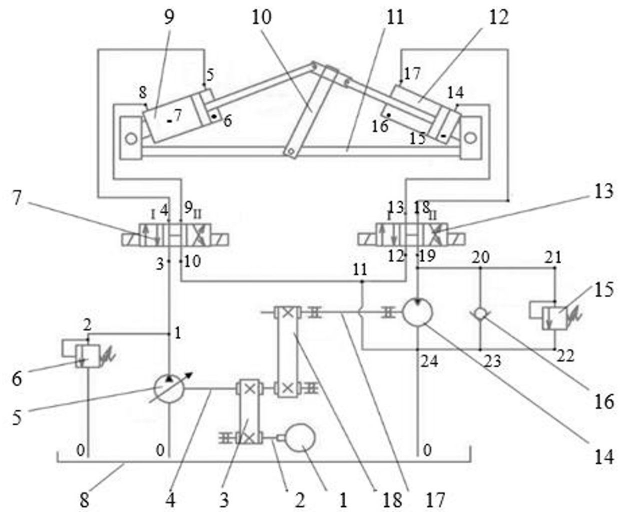

Fig. 1. Calculated hydrokinematic scheme of the test bench for piston hydraulic cylinders with energy recovery.

The energy supplied to the shaft of the hydraulic pump is converted into hydraulic energy and directed to the first hydraulic cylinder, which in this case performs the functions of a hydraulic motor, is converted into mechanical energy and, by means of mechanical transmission, is transmitted to the rod of the second hydraulic cylinder, which performs the functions of a hydraulic pump, in which it is again converted into hydraulic energy and directed to the input of the hydraulic motor, where it is again converted into mechanical energy of rotation of the shaft of the hydraulic motor and by means of a mechanical transmission (in this case belt) is again supplied to the shaft of the hydraulic pump, where it is combined with the energy supplied from the primary source and converted back into hydraulic…

Thus, the energy of testing hydraulic cylinders is not spent on creating "harmful heat", but circulates inside the test system, the electric motor 1 only compensates for the energy losses inside the test system caused by the need to overcome hydraulic and mechanical friction with this system.

3 Simulation of the testing process of piston hydraulic cylinders with energy recovery

It is well known that mathematical modeling of a technical system followed by its theoretical studies aimed at identifying the most important elements of the system that have the greatest impact on quality indicators is the most effective way of preliminary analysis when creating new and original technical solutions [9-11].

3.1 Simulation of the hydraulic system of the test bench

The mathematical model of the hydraulic system of the proposed stand, the design scheme of which is shown in Figure 1, was developed using the fundamentals of the theory of volumetric stiffness [12-14], according to which the pressure increment at any point of the hydraulic system can be determined by the equation

$$dp = C_{pri}(\Sigma Q_{vhi} - \Sigma Q_{ishi})dt\,,$$

where $\Sigma Q_{\mathrm{vh}i}$ is the total amount of working fluid flow entering the considered section of the hydraulic system; $\Sigma Q_{\mathrm{ish}i}$ is the total amount of working fluid flow coming from the considered section of the hydraulic system; $C_{\mathrm{pr}i}$ is the reduced coefficient of volumetric stiffness of the considered section of the hydraulic system [16, 17].

After dividing the hydraulic system of the stand, the design scheme of which is shown in Figure 1, by conditional nodal points into sections, we compile a mathematical model of the hydraulic system in the form of a system of equations describing the pressure change at the nodal points of the system during its operation

$$\begin{gathered}
dp_1 = C_1(Q_{\mathrm{H}} - \mathrm{Q}_{1-2} - \mathrm{Q}_{1-3})dt\,, \\
dp_2 = C_2(Q_{1-2} - Q_{KP6})dt\,, \\
dp_3 = C_3(Q_{1-3} - Q_{1-4})dt\,, \\
dp_4 = C_4(Q_{3-4} - Q_{4-5})dt\,, \\
dp_5 = C_5(Q_{4-5} - Q_{5-6})dt\,, \\
dp_6 = C_{c9.sht}(Q_{5-6} - v_{p9}f_{p.sht})dt\,, \\
dp_7 = C_{c9.p}\left(v_{p9}f_p - Q_{7-8}\right)dt\,, \\
dp_8 = C_8(Q_{7-8} - Q_{8-9})dt\,, \\
dp_9 = C_9(Q_{8-9} - Q_{9-10})dt\,, \\
dp_{10} = C_{10}(Q_{9-10} - Q_{10-11})dt\,, \\
dp_{11} = C_{11}(Q_{10-11} - Q_{11-12} - Q_{11-24})dt\,, \\
dp_{12} = C_{12}(Q_{11-12} - Q_{12-13})dt\,, \\
dp_{13} = C_{13}(Q_{12-13} - Q_{13-14})dt\,, \\
dp_{14} = C_{14}(Q_{13-14} - Q_{14-15})dt\,, \\
dp_{15} = C_{c12.p}\left(Q_{14-15} - v_{p12}f_p\right)dt\,, \\
dp_{16} = C_{c12.sht}\left(v_{p12}f_{p.sht} - Q_{16-17}\right)dt\,, \\
dp_{17} = C_{17}(Q_{16-17} - Q_{17-18})dt\,, \\
dp_{18} = C_{18}(Q_{17-18} - Q_{18-19})dt\,, \\
dp_{19} = C_{19}(Q_{18-19} - Q_{19-20} - Q_{\mathrm{M}})dt\,, \\
dp_{20} = C_{20}(Q_{19-20} + Q_{\mathrm{OK}} - Q_{20-21})dt\,, \\
dp_{21} = C_{21}(Q_{20-21} - Q_{KP15})dt\,, \\
dp_{22} = C_{22}(Q_{KP15} - Q_{22-23})dt\,, \\
dp_{23} = C_{23}(Q_{22-23} - Q_{23-24} - Q_{\mathrm{OK}})dt\,, \\
dp_{24} = C_{24}(Q_{23-24} + Q_{\mathrm{M}} - Q_{11-24} - Q_{24-0})dt\,,
\end{gathered}$$

where $dp_1 \ldots dp_{24}$ are pressure increments at the corresponding points of the hydraulic system of the stand during dt; $C_1 \ldots C_5$, $C_8 \ldots C_{14}$ $C_{17} \ldots C_{24}$ are the reduced volumetric stiffness coefficients at the corresponding points of the hydraulic system of the stand; $C_{\mathrm{c9.p}}$ is the reduced volumetric stiffness coefficient of the piston cavity of the hydraulic cylinder 9; $C_{\mathrm{c12.p}}$ is the reduced volumetric stiffness coefficient piston cavity of the hydraulic cylinder 12; $C_{\mathrm{c9.sht}}$ – the reduced coefficient of volumetric stiffness of the rod cavity of the hydraulic cylinder 9; $C_{\mathrm{c12.sht}}$ – the reduced coefficient of volumetric stiffness; Q_{H} – the actual performance of the hydraulic pump; Q_{M} – flow of working fluid through the hydraulic motor; Q_{OK} – flow of working fluid through the bypass check valve; Q_{KP6} – flow of working fluid through the safety valve 6; Q_{KP15} – flow of working fluid through the pressure valve 15; Q_{1_2}, Q_{1_3}, Q_{4_5}, Q_{8_9}, Q_{10_11}, Q_{11_12}, Q_{13_14}, Q_{17_18}, Q_{19_20}, Q_{20_21}, Q_{22_23}, Q_{23_24}, $Q_{11\text{-}24}$ and Q_{24_0} – the flow of working fluid in the corresponding sections of the hydraulic system of the stand; Q_{3_4} and Q_{9_10} – the flow of working fluid through the corresponding hydraulic lines of the distributor 7; Q_{12_13} and Q_{18_19} – the flow of working fluid through the corresponding hydraulic lines of the distributor 13; Q_{5_6} and Q_{7_8} – the flow of working fluid through the inlet fittings of the hydraulic cylinder 9; Q_{14_15} and Q_{16_17} – the flow of working fluid through the inlet fittings of the hydraulic cylinder 12; v_{p9} – the speed of movement of the piston of the hydraulic cylinder 9; v_{p12} – the speed of movement of the piston of the hydraulic cylinder 12; f_{p} – the working area of the pistons of the tested hydraulic cylinders; $f_{\text{p.sht}}$ – the working area of the pistons of the tested hydraulic cylinders from the side of the stem cavities.

The flow rates of the working fluid for use in the pressure increment equations are determined by the flow formula

$$Q_i = \mu f \sqrt{\frac{2}{\rho}|p_i - p_{i+1}|} \cdot \text{sign}(p_i - p_{i+1}),$$

p_i – the instantaneous value of the inlet pressure of the corresponding system resistance; p_{i+1} – instantaneous pressure value at the outlet of the corresponding system resistance; f – the area of the passage section of the corresponding resistance; μ - the flow rate of the corresponding hydraulic resistance; ρ – the current value of the working fluid density.

When calculating linear hydraulic resistances, the reduced flow rate of the corresponding hydraulic line is used, determined taking into account the flow regime of the fluid at the time under consideration by the formula

$$\mu = \mu_l = \frac{1}{\sqrt{\lambda_l \frac{l_l}{d_l}}},$$

where d_l – diameter of the live section of the corresponding hydraulic line; l_l – the length of the considered section of the hydraulic line; λ_l – the coefficient of hydraulic friction of the working fluid on the considered section of the hydraulic line, which is calculated taking into account the flow mode of the fluid.

The given coefficients of volumetric rigidity of metal pipelines are determined by analytical dependencies [12...14]

$$C_l = \frac{4}{\pi d^2 l} \cdot \frac{E_{fl}}{1 + \frac{d}{\delta}\frac{E_{fl}}{E_l}},$$

where d – inner diamet

er of the pipeline section; l – the length of the pipeline section under consideration; δ – wall thickness of the pipeline section under consideration; E_{fl} – the current value of the elastic modulus of the working fluid; E_l – the value of the elastic modulus of the wall material of the pipeline section under consideration.

The given coefficients of volumetric rigidity of the RVD (high-pressure hoses) are determined experimentally [15., 16].

The theoretical supply of the hydraulic pump is determined by the formula

$$Q_{teor} = q_{\text{H}} n_{\text{H}} ,$$

where q_{H} – hydraulic pump working volume; n_{H} – speed of rotation of the hydraulic pump shaft;

The actual instantaneous value of the hydraulic pump performance is determined taking into account the current value of its volumetric efficiency according to the dependencies:

$$Q_{\text{H}} = Q_{teor} \eta_{\text{o}} ,$$

where η_o – the current value of the volumetric efficiency of the hydraulic pump, which is determined by the formula

$$\eta_{\text{Ho}} = 1 - (1 - \eta_{\text{o}.nom}) \cdot \frac{p_{\text{H}}}{p_{\text{H}.nom}} ,$$

where $\eta_{o.nov}$ – the value of the nominal value of the volumetric efficiency of a hydraulic pump, which is assumed to be equal to the volumetric efficiency of a pump operating at nominal pressure; $p_{\text{H}.nov}$ – the value of the nominal value of the working pressure of the hydraulic pump; p_{H} – the current value of the operating pressure at the outlet of the hydraulic pump.

3.2 Simulation of the movement of mechanical elements of the hydraulic system of the stand

The area of the passage section of the shunt check valve 16 is determined from the equation of motion of the shutter angle

$$\frac{dv_{\text{ок}}}{dt} = \frac{1}{m_{\text{ок}}} \left[\frac{\pi d_{\text{ок}}^2}{4} (p_{23} - p_{20}) - F_{\text{прОК}} \right] ,$$

$$\frac{dh_{\text{ок}}}{dt} = dv_{\text{ок}} ,$$

where $v_{\text{ок}}$ – The area of the passage section of the shunt check valve 16 is determined from the equation of motion of the shutter angle; $m_{\text{ок}}$ – its reduced mass; $h_{\text{ок}}$ – displacement (cross-sectional area) of the gate of the shunt check valve 16; $d_{\text{ок}}$ – diameter of the seat of the shunt check valve 16; $F_{\text{прОК}}$ – the force of the spring action on the gate of the shunt check valve 16. Rotation of the rotor of the hydraulic motor 14

$$\frac{d\omega_{\text{M}}}{dt} = \frac{1}{\text{J}_{\text{M}}} [w_{\text{M}}(p_{19} - p_{24}) - M_{\text{M}}],$$

$$\frac{d\omega_{\text{H}}}{dt} = \frac{1}{\text{J}_{\text{H}}} (M_{EM} i_{2\_4} + M_{\text{M}} i_{17\_4} - w_{\text{H}}(p_1 - p_{\text{ат}})),$$

where w_{M} – characteristic volume of the hydraulic motor 14; w_{H} – characteristic volume of the hydraulic pump 5; ω_{M} – angular rotation speed of the hydraulic motor shaft 14; ω_{H} – angular rotation speed of the hydraulic pump shaft 5; J_{M} – the central moment of inertia of the rotor of the hydraulic motor 14; J_{H} – the central moment of inertia of the rotor of the hydraulic pump 5; p_1 – pressure at the outlet of the hydraulic pump 5; $p_{\text{ат}}$ – pressure at the inlet of the hydraulic pump 5 (assumed to be atmospheric); p_{19} – pressure at the inlet of the hydraulic motor 14; p_{24} – pressure at the outlet of the hydraulic motor 14; M_{EM} – torque on the shaft 2 of the electric motor 1; M_{M} – torque on the shaft 17 of the hydraulic motor 14; i_{2_4} – mechanical transmission ratio 3; i_{17_4} – the gear ratio of the mechanical transmission 18.

The rotational speeds of the hydraulic motor 14 and the hydraulic pump 5 are rigidly connected by the equation

$$\omega_{\text{M}} = \omega_{\text{H}} \cdot i_{17\_4}.$$

The evaluation of the effectiveness of the testing process was carried out on the basis of calculating the efficiency coefficient of the testing process, which can be used in two types - the instantaneous efficiency coefficient and the average efficiency coefficient, which are determined by the formulas:

$$k_{efi} = \frac{N_{isp}}{N_{ist.}}$$

where k_{efi} – instantaneous value of the efficiency coefficient of the test process; N_{isp} – power on the tested hydraulic cylinder at the i moment of time; $N_{ist.еи}$ is the power supplied to the input of the primary energy source at the time under consideration;

$$k_{ef.sr} = \frac{W_{isp}}{W_{ist.}}$$

where $k_{ef.sr.}$ – the average value of the efficiency coefficient of the test process; W_{isp} – the energy that has passed through the tested hydraulic machine from the moment of the beginning of the tests to the time under consideration; W_{ist} – the energy consumed by the primary energy source from the start of the tests to the time under consideration.

4 Calculation of the main functional characteristics of the stand

To study the obtained mathematical model, a special program was compiled using the SimInTech differential equation solution unit [17-22].

As a result of preliminary calculations, the functional characteristics of the stand operation were obtained, showing the relationship of the system parameters and the influence of the design parameters of the stand on them.

Figure 2 shows the results of calculating the functional parameters of the test system during its operation.

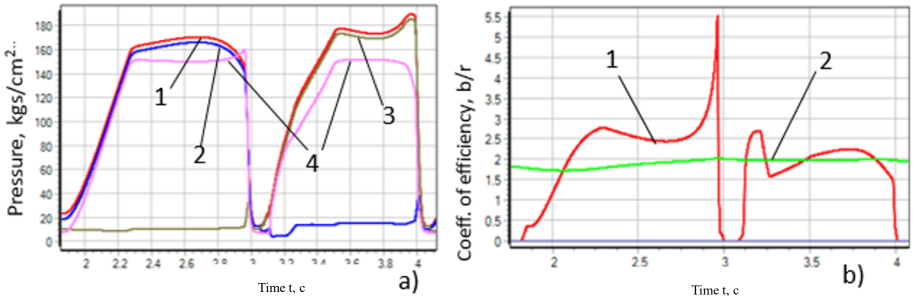

Fig. 2. The results of preliminary calculations of testing piston hydraulic cylinders: a) pressure: 1 at the outlet of the hydraulic pump 5, 2 – in the piston cavity of the hydraulic cylinder 9, 3 – in the stem cavity of the hydraulic cylinder 9, 4 – at the inlet of the hydraulic motor 14; b) the test efficiency coefficient 1 – instantaneous, 2 – average per cycle.

5 Conclusion

Thus, the proposed mathematical model makes it possible to conduct a numerical experiment to study the functioning of the proposed stand. The numerical experiment makes it possible to identify the influence of the main design and functional parameters of the test system on the qualitative characteristics of the test process and, ultimately, to obtain rational parameters of the stand, without resorting to the need for expensive and labor-intensive field studies.

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