

# Modeling and calculation of the technological process of testing piston 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 machines. In this article, an original scheme of a test bench for piston hydraulic cylinders and a method of 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 high accuracy and reliability the processes occurring in the test system during its operation. The test efficiency coefficient is proposed, which makes it possible to evaluate and compare various test processes by the criterion of their energy efficiency. An example of a preliminary calculation of the functioning of the proposed stand is given. **Keywords:** mathematical modeling, piston hydraulic cylinders, test bench, energy recovery, test efficiency coefficient.

## 1 Introduction

At the current level of development of industry and society as a whole, energy consumption is growing rapidly, both at the production stage and at the stage of using technological equipment. This circumstance forces the scientific community to pay special attention to the energy efficiency of the equipment used and manufactured and designed [1...8]. The greatest effect in terms of saving energy during the operation of the equipment is given by the method of its recovery and regeneration during the operation of the equipment [9...16]. At the same time, it should be noted that the most expensive and at the same time one of the most important technological processes in the production chain of creating new industrial equipment, including hydraulic machines, are its tests for compliance with technical requirements. At the same time, various types of braking devices (mechanical, electrical, hydraulic, and others) are used for power loading of a hydraulic engine, as a result of which a significant amount of energy is lost, which turns into “harmful” heat. This is especially true when testing medium- and high-power hydraulic machines.

In recent years, an active search for a solution to this problem has been conducted, as a result of which a test method for hydraulic rotary motion machines with energy recovery has been developed [17...20]. This method provides significant savings when testing hydraulic

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pumps and hydraulic motors. Methods of testing with energy recovery have also been developed for hydraulic cylinders [21...23]. It should be noted that when testing hydraulic machines, their functioning should be close to functioning in real operating conditions.

## **2 Problem Statement**

This paper presents an original scheme of the test bench for piston hydraulic cylinders with energy recovery. The stand allows saving energy due to the hydro-mechanical system of its recovery [19, 20]. The work is devoted to the development of methods of mathematical modeling and calculation of the main functional parameters of the proposed design of the stand, and its purpose is to develop a stand for resource testing of piston hydraulic cylinders, providing maximum approximation of the tested hydraulic cylinders to real operating conditions and minimal energy losses during their tests, as well as solving the problem of its modeling and calculation of the main functional parameters.

## **3 Test Bench Description**

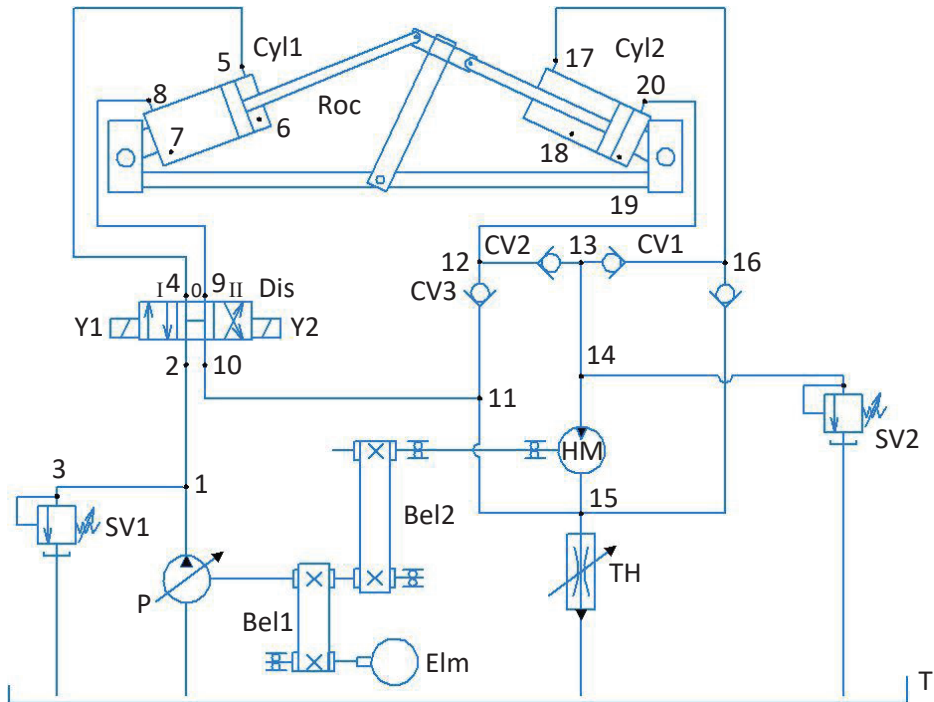
The development of a stand for resource testing of hydraulic cylinders with energy recovery was carried out based on the use of the previously proposed method of testing with energy recovery of volumetric hydraulic machines of rotational action. In this case, the load for one tested hydraulic machine (for example, for a hydraulic pump) is another tested hydraulic machine (hydraulic motor), and vice versa. This method significantly reduces the dimensions and simplifies the scheme of the test bench, since there are no special loading systems. This method increases the efficiency of the test system and, as a result, reduces energy consumption. In this paper, it is proposed to use the pressure line of the stand circuit for testing hydraulic rotary motion machines for resource testing of piston hydraulic cylinders [21, 22].

The basic hydraulic scheme of the test bench for piston hydraulic cylinders is shown in Figure 1. The essence of this hydraulic cylinder test scheme is that a pair of piston hydraulic cylinders Cyl1 and Cyl2 is installed in the pressure hydraulic line of the test scheme for rotary hydraulic machines between the hydraulic pump and the hydraulic motor.

The work of the stand is as follows. The Elm electric motor rotates the shaft of the adjustable hydraulic pump P. by means of a mechanical transmission Bel1. The mechanical energy received from the shaft of the electric motor is converted by the hydraulic pump into the hydraulic energy of the working fluid, which through the main 1-2 enters the inlet of the hydraulic distributor Dis, which directs it, for example, along the main 4-5 into the stem cavity of the tested hydraulic cylinder Cyl1 (which in this case performs the functions of a hydraulic motor).

The hydraulic energy of the working fluid due to the operation of the hydraulic cylinder Cyl1 is converted into the mechanical energy of the movement of its piston, which is mechanically connected to the rod of the second test cylinder Cyl2 by means of a mechanical transmission Roc.

The mechanical energy of movement of the piston of the Cyl2 hydraulic cylinder, which in this case performs the function of a hydraulic pump, is converted into the hydraulic energy of the working fluid located in its stem cavity. Along the highway 17-16-CV1-13-14, the liquid is directed to the inlet of the HM hydraulic motor.



**Fig. 1.** Schematic hydraulic diagram of a test bench for piston hydraulic cylinders with energy recovery.

In the hydraulic motor, the hydraulic energy of the working fluid is converted into the mechanical energy of rotation of the motor shaft, which is transmitted by means of a mechanical transmission Bel2 to the hydraulic pump shaft P, where it is combined with the rotational energy of the electric motor shaft and converted again by the hydraulic pump into the energy of the working fluid. The operation of the stand when the cylinder Cyl1 moves from the leftmost to the rightmost position is carried out in the same way, while the distributor Dis directs the liquid entering its inlet along the 9-8 line into the piston cavity of the cylinder Cyl1.

In the proposed scheme, the Bel2 mechanical transmission creates a condition under which the hydraulic pump supply will slightly exceed the required flow rate of the hydraulic motor at a given speed of rotation of the hydraulic pump shaft corresponding to the rotation frequency of the Elm electric motor. This will lead to an increase in pressure in the pressure line connecting the cylinder Cyl2 with the hydraulic motor, up to the pressure setting of the safety valve SV2. Excess working fluid will be discharged from this line into the hydraulic tank through a safety valve. Thus, by adjusting the setting of the safety valve SV2, it is possible to change the pressure in the line, thereby adjusting the amount of loading of the tested hydraulic cylinders.

## 4 Modeling and Calculation of the Hydraulic System of the Test Bench

The most effective method of preliminary research of the created drives of technological equipment is its mathematical modeling and calculation [24...34]. The mathematical model of the present regenerative system was developed based on the use of the theory of volumetric

rigidity [35...37]. Particular attention should be paid to determining the reduced coefficient of volumetric rigidity of high-pressure hoses, as well as pipelines made of other elastic materials when modeling hydraulic drives [35, 36].

In accordance with the theory of volumetric rigidity, the equation of pressure increment at any point of the hydraulic system has the following form:

$$dp = C_{pri}(\sum Q_{inti} - \sum Q_{outi})dt, \quad (1)$$

where  $Q_{inti}$  and  $\sum Q_{outi}$  are, respectively, the sums of all the working fluid flows entering and exiting the system ( $i$ -th) volume under consideration during  $dt$ ;  $Q_{inti}$  is the reduced volumetric rigidity coefficient of the hydraulic system ( $i$ -th) section under consideration.

Then conditionally breaking the hydraulic system of the test bench (see Fig. 1) taking the pressure in the hydraulic tank equal to atmospheric pressure, a series of equations can be written describing the pressure change during the operation of the stand at all points of interest to us. The number of equations will correspond to the number of selected "nodal" points.

$$dp_1 = C_1(Q_P - Q_{1-2} - Q_{1-3})dt,$$

$$dp_2 = C_2(Q_{1-2} - Q_{2-4})dt,$$

$$dp_3 = C_3(Q_{1-3} - Q_{SV1})dt,$$

$$dp_4 = C_4(Q_{2-4} - Q_{4-5})dt,$$

$$dp_5 = C_5(Q_{4-5} - Q_{5-6})dt,$$

$$dp_6 = C_{Cyl1.st}(Q_{5-6} - v_{p1}f_{Cyl.st})dt,$$

$$dp_7 = C_{Cyl1.p}(v_{Cyl1}f_{Cyl.p} - Q_{7-8})dt,$$

$$dp_8 = C_8(Q_{7-8} - Q_{8-9})dt,$$

$$dp_9 = C_9(Q_{8-9} - Q_{9-10})dt,$$

$$dp_{10} = C_{10}(Q_{9-10} - Q_{10-11})dt,$$

$$dp_{11} = C_{11}(Q_{10-11} - Q_{11-15} - Q_{CV3})dt,$$

$$dp_{12} = C_{12}(Q_{CV3} - Q_{12-20} - Q_{CV2})dt,$$

$$dp_{13} = C_{13}(Q_{CV2} + Q_{CV1} - Q_{13-14})dt,$$

$$dp_{14} = C_{p2}(Q_{13-14} - Q_{HM} - Q_{SV2})dt,$$

$$dp_{15} = C_{15}(Q_{11-15} + Q_{HM} - Q_{TH} - Q_{CV4})dt,$$

$$dp_{18} = C_{Cyl2.st}(v_{p2}f_{Cyl.st} - Q_{18-17})dt,$$

$$dp_{17} = C_{17}(Q_{18-17} - Q_{17-16})dt,$$

$$dp_{16} = C_{16}(Q_{17-16} + Q_{CV4} - Q_{CV1})dt,$$

$$dp_{20} = C_{20}(Q_{12-20} - Q_{20-19})dt,$$

$$dp_{19} = C_{Cyl2,p}(Q_{20-19} - v_{p2}f_{Cyl,p})dt,$$

where  $dp_1 \dots dp_{20}$  are the pressure increments at the corresponding points of the hydraulic system of the stand during time  $dt$ ;  $C_1 \dots C_5$ ,  $C_8 \dots C_{13}$ ,  $C_{15} \dots C_{17}$  and  $C_{20}$  are the reduced volumetric rigidity coefficients at the corresponding points of the hydraulic system of the stand;  $C_{Cyl1,p}$ ,  $C_{Cyl1,st}$  and  $C_{Cyl2,p}$ ,  $C_{Cyl2,st}$  are the given coefficients of volumetric rigidity of the corresponding working cavities of the hydraulic cylinders Cyl1 and Cyl2 [11...13];  $Q_P$  is the hydraulic pump supply;  $Q_{HM}$  is the flow of the working fluid through the hydraulic motor;  $Q_{CV1} \dots Q_{CV4}$  are the flow of the working fluid through the corresponding check valves;  $Q_{SV1}$  and  $Q_{SV2}$  are the flow of the working fluid through the corresponding safety valves;  $Q_{1-2}$ ,  $Q_{1-3}$ ,  $Q_{2-4}$ ,  $Q_{4-5}$ ,  $Q_{5-6}$ ,  $Q_{7-8}$ ,  $Q_{8-9}$ ,  $Q_{9-10}$ ,  $Q_{10-11}$ ,  $Q_{11-15}$ ,  $Q_{12-20}$ ,  $Q_{20-19}$ ,  $Q_{18-17}$ ,  $Q_{17-16}$  and  $Q_{13-14}$  are the flow rates of the working fluid in the corresponding sections of the hydraulic system of the stand;  $v_{p1}$  and  $v_{p2}$  are the speed of movement of the pistons of the hydraulic cylinders Cyl1 and Cyl2, respectively;  $f_p$  is the working area of the piston of the tested hydraulic cylinders;  $f_{p,st}$  is the working area of the piston of the tested hydraulic cylinders from the side of the rod cavities.

The flow rates of the working fluid required to calculate the pressure increment are determined by the formula:

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

where  $p_i$  and  $p_{i+1}$  are the pressure values at the inlet and outlet of the corresponding hydraulic resistance;  $f$  is the area of the flow section of the resistance;  $\mu$  is the flow coefficient of the corresponding hydraulic resistance;  $\rho$  is the density of the working fluid.

For hydraulic resistances in the form of sections of hydraulic lines, the value of the reduced flow coefficient of the corresponding section of the hydraulic line is substituted into the flow formula. This value is determined by the formula:

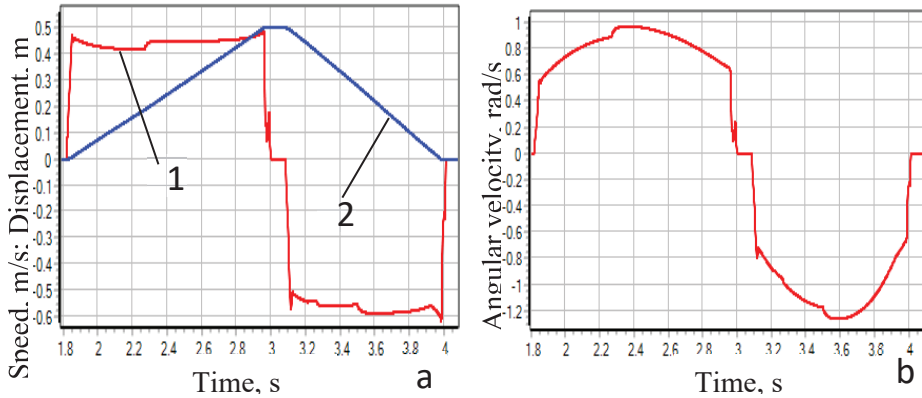
$$\mu = \mu_l = \frac{1}{\sqrt{\lambda_l \frac{l_l}{d_l}}} \quad (3)$$

where  $d_l$  and  $l_l$  are, respectively, the diameter and length of the pipeline section under consideration;  $\lambda_l$  is the hydraulic friction coefficient of the pipeline, which is determined taking into account the flow regime of the liquid and the properties of the pipeline.

The values of the reduced coefficients of volumetric rigidity of sections of metal pipelines are determined by the formula [11...13]

$$C_l = \frac{4}{\pi d^2 l} \frac{E_{fl}}{1 + \frac{d E_{fl}}{\delta E_l}} \quad (4)$$

where  $d$  and  $l$  are, respectively, the inner diameter of the pipeline section under consideration and its length;  $\delta$  is the thickness of the pipeline wall;  $E_f$  and  $E_l$  are the elastic modulus of the working fluid and the wall material of the hydraulic line, respectively.



**Fig. 2.** The results of preliminary calculations of the test of piston hydraulic cylinders: a) the parameters of the movement of the piston of the first hydraulic cylinder: 1 – speed, 2 - displacement; b) the angular velocity of the rocker arm rotation.

The above coefficients of volumetric rigidity of high-pressure hoses and pipelines made of other elastic materials are determined experimentally [35, 36].

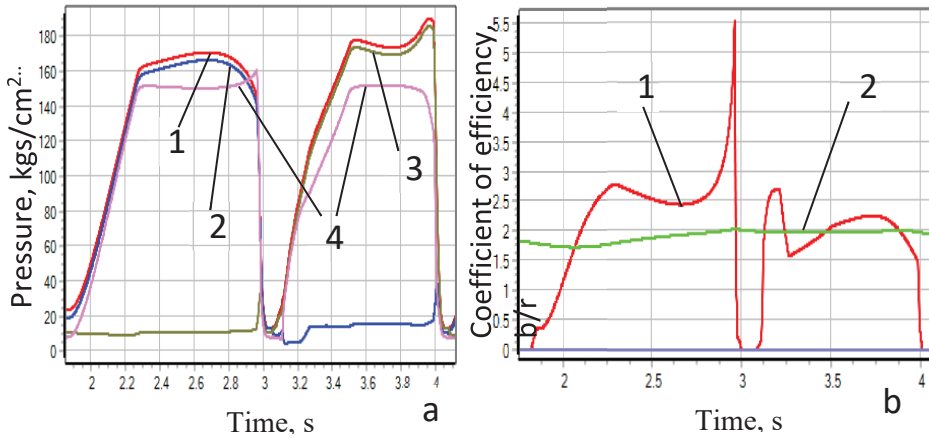
To study the obtained mathematical model, a special program has been developed using a block of solutions of differential equations of the SimInTech software package [39, 40], which, in combination with the principle of partial synthesis [41], allows obtaining calculation results close to the real operating conditions of the equipment [42...44].

Figures 2 and 3 show the results of the preliminary calculation of the operation of the test bench for piston hydraulic cylinders.

The efficiency of the test bench, in terms of energy costs for testing, is determined by the test efficiency coefficient [38], which is the ratio of the power on the tested  $N_{test}$  machine (in this case, hydraulic cylinders) to the power consumed by the electric motor  $N_{Elm}$ :

$$k_{eff} = \frac{N_{test}}{N_{Elm}}. \quad (5)$$

The higher the efficiency coefficient, the more efficient, that is, more economical, the test system.



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

From Figure 3 (b) it can be seen that the efficiency coefficient of the testing process of piston hydraulic cylinders on the proposed stand is close to 2, which means that the tests of hydraulic cylinders are carried out at twice the power consumed from the primary energy source.

## 5 Conclusion

The proposed mathematical model makes it possible to conduct a numerical experiment to study the functioning of the test bench for piston hydraulic cylinders, during which the influence of the design and functional parameters of the stand on its energy characteristics, as well as the effectiveness of the testing process is determined, which will allow obtaining rational parameters of the stand already at the design stage, without resorting to the need for expensive and labor-intensive field studies.

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