Vestnik of Don State Technical University. 2019. Vol. 19, no. 3, pp. 242–249. ISSN 1992-5980 eISSN 1992-6006
Вестник Донского государственного технического университета. 2019. Т. 19, № 3. С. 242–249. ISSN 1992-5980 eISSN 1992-6006

MACHINE BUILDING AND MACHINE SCIENCE
МАШИНОСТРОЕНИЕ И МАШИНОВЕДЕНИЕ

UDC 62-82 https://doi.org/10.23947/1992-5980-2019-19-3-242-249

Theoretical background of hydraulic drive control system analysis for testing piston hydraulic cylinders ***

A. T. Rybak[1], I. K. Tsybriy[2], S. V. Nosachev[3], A. R. Zenin[4]**
[1, 2, 3, 4] Don State Technical University, Rostov-on-Don, Russian Federation

Теоретические основы расчета системы управления гидравлического привода стенда для испытаний поршневых гидравлических цилиндров*

А. Т. Рыбак[1], И. К. Цыбрий[2], С. В. Носачёв[3], А. Р. Зенин[4]**
[1, 2, 3, 4] Донской государственный технический университет, г. Ростов-на-Дону, Российская Федерация

Introduction. The durability and performance of hydraulic machines is determined through life tests. At that, various braking devices (mechanical, electric, hydraulic, etc.) are used for strength loading of the hydraulic motor, as a result of which a significant amount of energy is lost. This can be avoided if the method of rotational motion with energy recovery is used during life tests. This approach is applicable for hydraulic pumps, motors, and hydraulic cylinders.
Materials and Methods. A test bench is presented, the design of which provides recreation of the conditions most appropriate for the field operation of hydraulic cylinders. In this case, energy recovery is possible. To solve the research problems, methods of mathematical modeling were used, the basic functional parameters of the proposed design were calculated. The determination of the pressure increment at various points in the hydraulic system is based on the theory of volumetric rigidity. When modeling the motion of the moving elements of the bench hydraulic system, the laws of rotor motion are used.
Research Results. In the structure of the test bench, the cylinders in question are located in the pressure main between the hydraulic pump and the hydraulic motor. This enables to significantly reduce the bench itself and to save a significant amount of energy due to its recovery. A basic hydraulic diagram of the test bench for piston hydraulic cylinders is presented, in which the operation of the moving elements of the system is shown. A mathematical modeling of the hydraulic system of the bench is performed. A kinematic diagram of the mechanism for transmitting motion between test cylinders is shown.
Discussion and Conclusions. The system of equations present-

Введение. Долговечность и работоспособность гидравлических машин определяется в результате ресурсных испытаний. При этом для силового нагружения гидравлического двигателя применяются различные тормозные устройства (механические, электрические, гидравлические и др.), в результате чего теряется значительное количество энергии. Этого можно избежать, если при ресурсных испытаниях использовать метод вращательного движения с рекуперацией энергии. Такой подход применим для гидравлических насосов, моторов, а также гидравлических цилиндров.
Материалы и методы. Представлен испытательный стенд, конструкция которого позволяет воссоздать условия, максимально соответствующие реальной эксплуатации гидравлических цилиндров. При этом возможна рекуперация энергии. Для решения задач исследования использованы методы математического моделирования, рассчитаны основные функциональные параметры предлагаемой конструкции. Определение приращения давления в различных точках гидравлической системы базируется на теории объемной жесткости. При моделировании движения подвижных элементов гидравлической системы стенда использованы законы движения ротора.
Результаты исследования. В структуре испытательного стенда рассматриваемые гидроцилиндры размещены в напорной магистрали между гидронасосом и гидромотором. Это позволяет существенно уменьшить сам стенд и сэкономить значительное количество энергии за счет ее рекуперации. Приведена принципиальная гидравлическая схема стенда для испытаний поршневых гидроцилиндров, в рамках которой показана работа подвижных элементов системы. Выполнено математическое моделирование гидравлической системы стенда. Показана кинематическая схема механизма передачи движения между испытуемыми цилиндрами.
Обсуждение и заключения. Представленная в статье система уравнений показывает, каким образом определяется

* The research is done within the frame of the independent R&D.
** E-mail: 2130373@mail.ru, irconst@mail.ru, nosachev-s@yandex.ru, azenin@donstu.ru
*** Работа выполнена в рамках инициативной НИР.

ed in the paper shows how the increment of pressure at the selected nodal points of the energy recovery system is determined (in particular, how the increment depends on time, reduced coefficient of volumetric rigidity, operating fluid consumption, and piston areas). The velocities of the hydraulic pistons are determined according to the kinematic scheme of the mechanical transmission of the bench. Thus it can be argued that, thanks to the solution presented in the paper, the life test results of hydraulic cylinders will adequately reflect their operation under rated duties.

приращение давления в выбранных узловых точках системы рекуперации энергии (в частности, как приращение зависит от времени, приведенного коэффициента объемной жесткости, расхода рабочей жидкости, площади поршней). Величины скоростей перемещения плунжеров гидравлических цилиндров определены согласно кинематической схеме механической передачи стенда. Итоги исследования позволяют утверждать, что, благодаря представленному в статье решению, результаты ресурсных испытаний гидравлических цилиндров будут адекватно отражать их работу при номи нальном режиме эксплуатации.

Keywords*:* piston hydraulic cylinders, test bench, testing, energy recovery, math modeling, kinematic motion transmission scheme.

Ключевые слова: поршневые гидравлические цилиндры, стенд, испытания, рекуперация энергии, математическое моделирование, кинематическая схема передачи движения.

For citation: A.T. Rybak, et al. Theoretical background of hydraulic drive control system analysis for testing piston hydraulic cylinders. Vestnik of DSTU, 2019, vol. 19, no. 3, pp. 242‒249. https://doi.org/10.23947/1992-5980-2019-19-3-242-249

***Образец для цитирования*:** Теоретические основы расчета системы управления гидравлического привода стенда для испытаний поршневых гидравлических цилиндров / А. Т. Рыбак [и др.] // Вестник Дон. гос. техн. ун-та. — 2019. — Т. 19, № 3. — С. 242‒249. https://doi.org/10.23947/1992-5980-2019-19-3-242-249

Introduction. One of the important stages of the machine-building production including the production of hydraulic machines is the end product qualification test [1].

The most significant (at the same time labor and energy consuming) tests are life time tests. They determine the durability and long-term intended operability. Life tests should be performed in the mode most closely approximate to the nominal conditions of hydraulic cylinders. In this case, various braking devices (mechanical, electrical, hydraulic and others) are used for power loading of the hydraulic motor, as a result of which a significant amount of energy is converted into heat. This is especially true for testing medium-duty and high-energy-rate hydraulic machines.

As a result of an active search for a solution to the indicated problem, a life time test for rotary hydraulic machines with energy recovery is developed [2–4]. This approach provides significant savings when testing hydraulic pumps and hydraulic motors. Test methods with energy recovery are also developed for hydraulic cylinders [5–10]. The schemes described in [9, 10] make it possible to produce a bench tester for hydraulic cylinders operating mode fully compatible with their field work.

Materials and Methods

Problem formulation. A stand design is proposed that provides recreating conditions most closely resembling the field work of hydraulic cylinders. At that, energy recovery is possible, which reduces significantly its costs, especially during resource tests. To solve the research problems, mathematical simulation techniques were used, the basic functional parameters of the proposed design were calculated.

Description of the test bench. The test bench for life time tests of piston hydraulic cylinders with energy recovery was developed on the basis of the previously proposed test method with energy recovery of rotary hydraulic volumetric machines [3, 4]. This solution implies that the hydraulic motor returns energy to the hydraulic pump shaft through the mechanical drive system. The tested hydraulic cylinders are located in the pressure line between the hydraulic pump and the hydraulic motor [5, 6]. Such tests can reduce drastically the stand itself and save a significant amount of energy due to its recovery.

The hydraulic circuit diagram of the test bench for piston hydraulic cylinders is shown in Fig. 1.

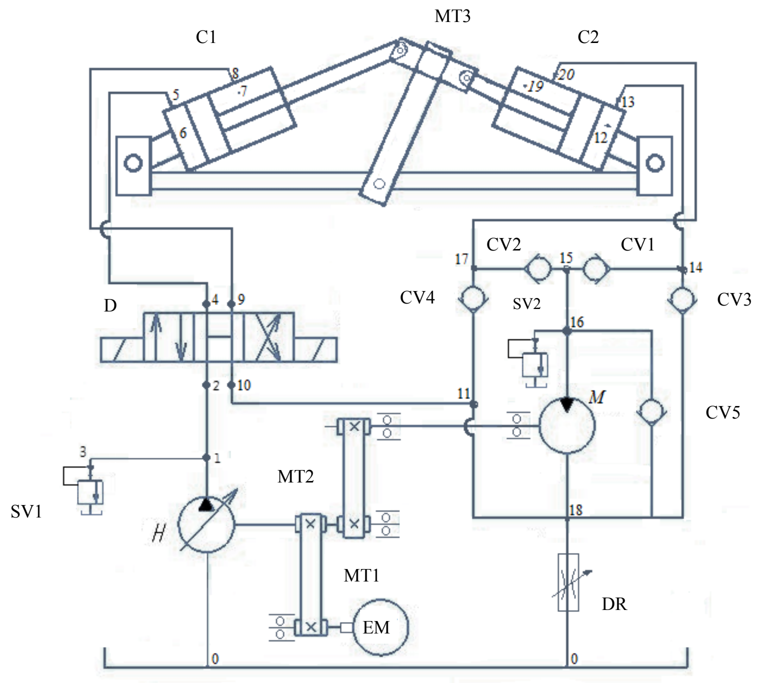

Fig. 1. Hydraulic circuit diagram of the stand for resource tests of piston hydraulic cylinders with energy recuperation

Fig. 1 shows that the tested hydraulic cylinders *C1* and *C2* interconnected by a mechanical transmission *MT3* are installed between a hydraulic pump and a hydraulic motor.

The stand operates as follows. The electric motor *EM* through the mechanical transmission *MT1* drives the shaft of the hydraulic pump *P*. The energy transported by the actuation fluid is transferred by it along the hydraulic line 1-2 to the input of the hydraulic distributor *D*, which directs it, for example, along the highway 4-5-6 to the piston cavity of the hydraulic cylinder *C1*.

The hydraulic cylinder *C1*, through the mechanical transmission *MT3*, transfers the energy received from the actuation fluid to the rod of the hydraulic-cylinder *C2*, which, in this case, acts as a pump and transfers the energy of the actuation fluid located in its piston cavity.

From the piston cavity of the hydraulic cylinder *C2*, the actuation fluid is fed through hydraulic line 12–13–14–15–16 to the input of the hydraulic motor *M*, which converts the energy obtained from the actuation fluid into the shaft rotation energy. The rotation of the shaft of the hydraulic motor *M* through a mechanical transmission *MT2* is transported to the shaft of the hydraulic pump *P*. *MP2* transmission is designed so that the speed transmitted to the shaft of the hydraulic pump *P* from the shaft of the hydraulic motor *M* is slightly higher than the frequency with which the primary energy source (electric motor *EM*) rotates the hydraulic pump shaft. This helps to slow down the rotation of the shaft of the hydraulic motor *M*. As a result, the inlet pressure rises, and the safety valve *SV2* opens. Pressure also increases in the cylinders *C1* and *C2*, which determines their operation in the corresponding mode.

When the hydraulic cylinder (*C1*) rod is extended at full travel, a position change command is sent to the distributor *D,* and the actuation fluid supplied to the input of the distributor *D* from the hydraulic pump *P* is sent along the hydraulic line 9–8–7 to the rod cavity of the hydraulic cylinder *C1*. This causes a reverse movement of its piston, but the energy recovery system operates as in the forward stroke of the piston.

Research Results

Mathematical modeling of the hydraulic system of the stand. We are developing a mathematical model of the proposed recuperative system for testing piston hydraulic cylinders. As a basis, we use the theory of volumetric rigidity [11–13] considering the given coefficients of volumetric s rigidity of hydraulic elements. This approach more accurately simulates a system resembling field conditions of hydraulic cylinders [14–19]. Under modeling hydraulic drives, special attention should be given to determining the reduced volumetric rigidity coefficient of hydraulic lines. Its value for metal pipelines is calculated according to well-known dependencies, and for high pressure hoses (HPH), it is determined experimentally [20, 21].

In accordance with the theory of volumetric rigidity, the equation of pressure increment at any point of the hydraulic system can be determined from the equation:

$$dp = C_i \left(\sum Q_{\text{вх}i} - \sum Q_{\text{вых}i} \right) dt \,,$$

where $\sum Q_{\text{вх}i}$ and $\sum Q_{\text{вых}i}$ are total input and output flow rate of the actuation fluid during the time dt coming out of the considered (i-th) volume of the system; C_i is the reduced volumetric rigidity factor of the selected area of the hydraulic system.

Conventionally, we will divide the hydraulic system of the bench (see Fig. 1) by nodal points. We assume volume of the hydraulic tank with atmospheric pressure as point 0, and write the equations for determining pressure at the selected nodal points.

$$dp_1 = C_1(Q_{\text{Н}} - Q_{1-2} - Q_{1-3})dt\,,$$

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

$$dp_3 = C_3(Q_{1-3} - Q_{\text{КП}1})dt\,,$$

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

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

$$dp_6 = C_{\text{п}1}(Q_{5-6} - v_{\text{п}1} f_{\text{п}})dt\,,$$

$$dp_7 = C_{\text{шт}1}(v_{\text{п}1} f_{\text{п.шт}} - Q_{7-8})dt\,,$$

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

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

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

$$dp_{11} = C_{11}(Q_{10-11} - Q_{11-18} - Q_{\text{ОК}4})dt\,,$$

$$dp_{17} = C_{17}(Q_{\text{ОК}4} - Q_{17-20} - Q_{\text{ОК}2})dt\,,$$

$$dp_{20} = C_{20}(Q_{17-20} - Q_{20-19})dt\,,$$

$$dp_{19} = C_{\text{шт}2}(Q_{20-19} - v_{\text{п}2} f_{\text{п.шт}})dt\,,$$

$$dp_{12} = C_{\text{п}2}(v_{\text{п}2} f_{\text{п}} - Q_{12-13})dt\,,$$

$$dp_{13} = C_{13}(Q_{12-13} - Q_{13-14})dt\,,$$

$$dp_{14} = C_{14}(Q_{13-14} + Q_{\text{ОК}3} - Q_{\text{ОК}1})dt\,,$$

$$dp_{15} = C_{15}(Q_{\text{ОК}1} + Q_{\text{ОК}2} - Q_{15-16})dt\,,$$

$$dp_{16} = C_{16}(Q_{15-16} + Q_{\text{ОК}5} - Q_{\text{КП}2} - Q_{\text{М}})dt\,,$$

$$dp_{18} = C_{18}\big(Q_{11-18} + Q_{\text{М}} - Q_{\text{ДР}} - Q_{\text{ОК}3} - Q_{\text{ОК}5}\big)dt.$$

Here, $dp_1 ... dp_5$ are pressure increments at characteristic points of the pressure hydraulic line of cylinder *C1* in the time dt; *dp7 ... dp10* are pressure increments at characteristic points of the drain line of cylinder *C1* in the time *dt*; *dp11 ... dp18* are pressure increments at the characteristic points of the hydraulic energy recovery system including the hydraulic cylinder *C2* and hydraulic motor *M*, during the time *dt*; *dp6 and dp7* are pressure increments in the piston and rod cavities of the hydraulic cylinder *C1* during the time *dt*; *dp12* and *dp19* are pressure increments in the piston and rod cavities of the hydraulic cylinder *C2* during *dt*; *dp20* is pressure increment at the outlet of the rod cavity of the cylinder *C2* during *dt*; *C1 ... C5, C8 ... C11, C13 ... C18* and *C20* are the reduced volumetric rigidity factors at the characteristic points of the bench hydraulic system; $C_{\text{п}1}$ and $C_{\text{п}2}$ are the volumetric rigidity factors of the piston cavities of the hydraulic cylinders *C1* and *C2* [11–13]; $C_{\text{шт}1}$ and $C_{\text{шт}2}$ are the reduced volumetric rigidity factors of rod cavities of hydraulic cylinders *C1* and *C2* [11–13]; $Q_{\text{Н}}$ is the hydraulic pump (*P*) capacity; $Q_{\text{М}}$ is the flow rate of the actuation fluid through the hydraulic motor *M*; $Q_{\text{ОК}1} \dots Q_{\text{ОК}5}$ are flow rates of the actuation fluid through the check valves *CV1 ... CV5*;
$Q_{\text{КП}1}$ and $Q_{\text{КП}2}$ are the flow rate of the actuation fluid through the safety valves *SV1* and *SV2*; Q_{1_3}, Q_{1_2}, Q_{2_4}, Q_{4_5}, Q_{5_6}, Q_{7_8}, Q_{8_9}, Q_{9_10}, Q_{10_11}, Q_{11_18}, Q_{19_20}, Q_{13_14} are fluid flows in the corresponding sections of the bench hydraulic system; ; $v_{\text{п}1}$ and $v_{\text{п}2}$ are piston velocities of the hydraulic cylinders *C1* and *C2*, respectively; $f_{\text{п}}$ are piston areas of the tested hydraulic cylinders *C1* and *C2*; $f_{\text{п.шт}}$ are piston areas of the tested hydraulic cylinders *C1* and *C2*from the side of the rod cavities.

Flow rate values of the actuation fluid required to calculate the increment of pressure are determined from the formula:

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

Here, p_i and p_{i+1} are pressure values at the inlet and outlet of hydraulic resistances; f is a clear area of the corresponding resistance; μ is the drag coefficient; ρ is the fluid density.

For sections of the hydraulic lines (linear hydraulic resistances), the reduced flow coefficient is determined from the formula:

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

where d_l and l_l are the inner diameter and length of the corresponding section of the pipeline; λ_l is the pipe friction number of the pipeline section determined with account for the flow regime of the actuation fluid and the properties of the pipeline.

The given volumetric rigidity factors of metal pipe wires are determined from the formula [11–13]:

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

where d and l are the diameter of the pipeline under study and its length; δ is the pipe wall thickness; E_{fl} and E_l are the elastic modulus values of the liquid and material of the pipeline wall.

The reduced volumetric rigidity factor of the HPH and pipelines made of elastic materials should be determined experimentally [20, 21].

The pump performance is determined by its volumetric efficiency:

$$Q_\text{Н} = \frac{q_\text{Н} \omega_\text{Н}}{2\pi} \eta_0 ,$$

where $q_\text{Н}$ is the hydraulic pump displacement; $\omega_\text{Н}$ is the rotational speed of the hydraulic pump shaft; η_0 is the momentary value of the pump volumetric efficiency.

$$\eta_0 = 1 - (1 - \eta_{0.nom}) \cdot \frac{p_\text{Н}}{p_{nom}}.$$

Here, η_0 is the rated value of the pump volumetric efficiency (taken equal to the volumetric efficiency at the nominal pressure (PN) of the pump); p_{nom} is rated working pressure (RWP) of the hydraulic pump; p_H is the current pressure value at the pump outlet (pressure at point 1 of the hydraulic system).

Simulation of the movement of moving elements of the stand hydraulic system. Values of operating gaps of the check valves are determined from the equation of motion of their gates:

$$\frac{dv_\text{кл}}{dt} = \frac{1}{m_\text{кл}} \left[\frac{d_\text{кл}^2}{4} (p_{1\text{кл}} - p_{2\text{кл}}) - F_\text{пр}\right], \qquad \frac{dh_\text{кл}}{dt} = v_\text{кл},$$

where $v_\text{кл}$ is the valve gate speed; $m_\text{кл}$ is the reduced valve gate mass; $h_\text{кл}$ is motion (size of the operating gap) of the valve gate; $d_\text{кл}$ is the valve bore diameter; $F_\text{пр}$ is the spring impact on the valve gate; t is time.

The law of rotor float of the hydraulic pump P and the hydraulic motor M is described by their motion equation:

$$\frac{d\omega_\text{М}}{dt} = \frac{1}{\text{J}_\text{М}} [w_\text{М}(p_{1\text{М}} - p_{2\text{М}}) - M_\text{М}],$$

$$\frac{d\omega_\text{Н}}{dt} = \frac{1}{\text{J}_P} (M_\text{ЭД} i_{\text{МП}1} + M_M i_{\text{МП}2} - w_\text{Н}(p_1 - p_\text{ат})).$$

Here, $w_\text{М}$, $w_\text{Н}$ are characteristic volumes of the motor M and pump P, respectively; $\omega_\text{М}$ and $\omega_\text{Н}$ are angular rotational velocities of the shafts of the hydraulic motor M and pump P; $\text{J}_\text{Н}$, $\text{J}_\text{М}$ are central moments of inertia of the rotors of the hydraulic pump and hydraulic motor; $p_{1\text{М}}$ is the pressure at the inlet of the hydraulic motor M; $p_{2\text{М}}$ is the pressure at the outlet of the hydraulic motor M; $p_\text{ат}$ is the pressure at the inlet of the pump P (pressure in the tank taken equal to the atmospheric pressure); p_1 is the pressure at the outlet of the hydraulic pump P; $M_\text{ЭД}$ and $M_\text{М}$ are torques created by the electric motor and hydraulic motor, respectively; $i_{\text{МП}1}$ and $i_{\text{МП}2}$ are gearing ratios of mechanical transmissions *MT2* (from the shaft of the hydraulic motor to the shaft of the hydraulic pump H) and *MT1* (from the shaft of the electric motor to the shaft of the hydraulic pump P), respectively.

The mechanical transmission *MT2* provides the ratio of the shaft speeds of the hydraulic motor M and the hydraulic pump P described by the formula:

$$\omega_{\text{М}} = \omega_{\text{Н}} i_{\text{МП2}}.$$

Simulation of the stand mechanical schematic. To determine the hydraulic piston velocities, we consider the kinematic scheme of the mechanical transmission of the *MT3* bench (Fig. 2).

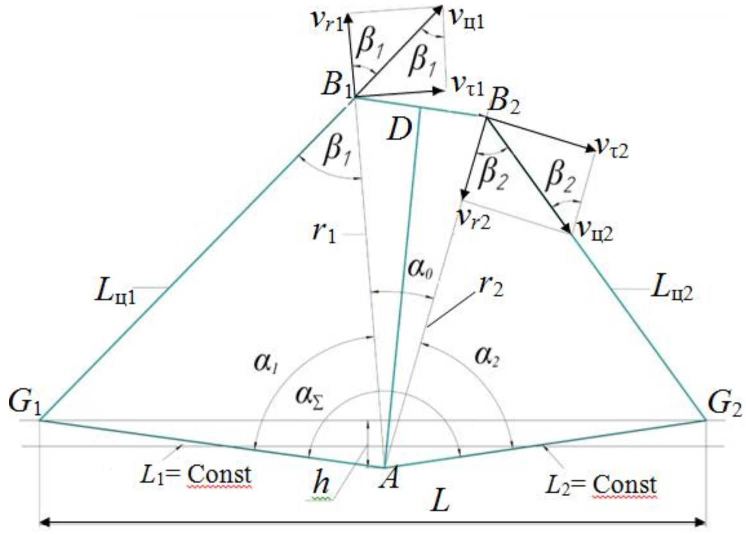

Fig. 2. Kinematic scheme of motion transmission mechanism between the tested cylinders

The transmission works as follows. Bodies of the hydraulic cylinders *C1* ($L_{ц1}$) and *C2* ($L_{ц2}$) are pivotally mounted at the points *G*1 and *G*2, respectively, and their pistons are pivotally connected at the points *B*1 and *B*2 to the rocker arm with a rotation axis at the point *A*.

Assume that the leading hydraulic cylinder (hydraulic engine) is the hydraulic cylinder $L_{ц1}$, and the working cylinder (hydraulic pump) is the hydraulic cylinder $L_{ц2}$. The hydraulic piston speed $L_{ц1}$ is set according to the flow rate Q_{5_6} of the actuation fluid entering its piston cavity. Then, its piston speed can be determined from the formula:

$$v_{ц1} = \frac{Q_{5\_6}}{f_{п}}.$$

The hydraulic piston motion $L_{Ц1}$ is transmitted through the pivot B_1 to the rocker arm *AD* which rotates around the point *A*. Having decomposed the joint speed B_1 into radial v_{r1} and tangential v_{r1}, we determine the value of the tangential component:

$$v_{\tau1} = v_{ц1} \cdot \text{Sin}\beta_1 .$$

Then the angular velocity ω_{AD} of rotation of the rocker arm *AD* can be determined from the expression:

$$\omega_{AD} = \frac{v_{\tau1}}{r_1},$$

where r_1 is the length of the radius connecting the point *A* of the rocker arm rotation with the pivot *B1*.

The motion of the piston of the hydraulic cylinder $L_{ц1}$ is transmitted through the rocker arm *AD* to the piston of the hydraulic cylinder $L_{ц2}$ connected to the rocker arm *AD* through the pivot B_2. In this case, the tangential velocity of the pivot B_2 is determined from the formula:

$$v_{\tau2} = r_2\, \omega_{AD} ,$$

where r_2 is the length of the radius connecting the point *A* of the rocker arm rotation with the pivot *B2*.

Projecting the tangential velocity of the pivot *B2*on the direction of piston motion of the hydraulic cylinder $L_{ц2}$, we determine the speed of its movement:

$$v_{ц2} = \frac{v_{\tau2}}{Sin\beta_2}.$$

The angles β_1 and β_2 are determined according to the law of cosines from the triangles AG_1B_1 and AG_2B_2, respectively:

$$\text{Cos}\beta_1 = \frac{L_{ц1}^2 + r_1^2 - L_1^2}{2L_{ц1}r_1},$$

$$\text{Cos}\beta_2 = \frac{L_{ц2}^2 + r_2^2 - L_2^2}{2L_{ц2}r_2}.$$

Here, $L_{ц1}$ and $L_{ц2}$ are distances between axes of attachment of the respective hydraulic cylinders to the stand frame and to the rocker arm considering the degree of extension of their pistons; L_1 and L_2 are distances between the axis of rotation of the rocker arm and the attachment points of the pivots of the hydraulic cylinders bodies to the stand frame.

Discussion and Conclusions. The system of equations presented in the paper shows how the pressure increment at the selected nodal points of the energy recovery system is determined (in particular, how the increment depends on the time, the reduced volumetric rigidity factor, the flow rate of the actuation fluid, and the area of pistons). The motion velocities of the hydraulic pistons are determined according to the kinematic scheme of the stand mechanical transmission. The study findings show that, thanks to the solution presented in the paper, the results of the life tests of hydraulic cylinders will adequately reflect their operation under the rated duties.

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Submitted 25.02.2019
Scheduled in the issue 05.04.2019

Authors:

Rybak, Alexander T.,
professor of the Instrument Making and Biomedical Engineering Department, Don State Technical University (1, Gagarin Square, Rostov-on-Don, 344000, RF), Dr.Sci. (Eng.), professor,
ORCID: https://orcid.org/0000-0001-9950-3377
2130373@mail.ru

Tsybriy, Irina K.,
Head of the Instrument Making and Biomedical Engineering Department, Don State Technical University (1, Gagarin Square, Rostov-on-Don, 344000, RF), Cand.Sci. (Eng.), associate professor,
ORCID:https://orcid.org/0000-0002-6281-1832,
irconst@mail.ru

Nosachev, Sergey V.,
senior lecturer of the Production Automation Department, Don State Technical University (1, Gagarin Square, Rostov-on-Don, 344000, RF), Cand.Sci. (Eng.),
ORCID: http://orcid.org/0000-0003-0302-2937
nosachev-s@yandex.ru

Zenin, Alexander R.,
postgraduate, Don State Technical University (1, Gagarin Square, Rostov-on-Don, 344000, RF),
ORCID:https://orcid.org/0000-0003-0336-315X
azenin@donstu.ru