BIO Web of Conferences **84**, 02029 (2024) https://doi.org/10.1051/bioconf/20248402029
AQUACULTURE 2023

Simulation of a synchronous hydrodynamic system based on a throttle flow divider of a non-valve type

Alexander Rybak[1], *Besarion* Meskhi[1], *Alexey* Pelipenko[1], *Yuliya* Serdyukova[1], *Svetlana* Teplyakova[1*], and *Maxim* Marchenko[1]

[1]Don State Technical University, Gagarin Sq.1, 344003 Rostov-on-Don, Russia

Abstract. Hydraulic drives, which for the most part is multi-circuit and branched, are widely used in hydraulic drives of modern mobile equipment and technological equipment. To synchronize the operation of mechanisms driven by a single power mechanism, throttle flow dividers are used. The article discusses various options for the functioning of the hydrodynamic system, as well as the device of the throttle flow divider. The dynamics of the synchronous hydraulic system based on throttle dividers is presented and the design scheme is described. The time integration of pressure values at various points, the values of movement and position of the moving element in space, the speed of movement in space and the values of velocities in real time and scale is carried out. The conclusion is made about the influence of system parameters on the selection of parameters of a suitable throttle flow divider.

1 Introduction

The complication of technological processes with increasing requirements for the accuracy of execution of executive movements, their consistency poses additional challenges to the developers of hydraulic drives [1-5]. Synchronization of the movements of the output links in an extensive multi-circuit hydromechanical system is complicated by both variable external loading and elastic properties of the hydraulic system itself. Many factors affecting the stability of the hydromechanical system do not allow at the initial stages of design to give an unambiguous answer about the degree of influence of one or another element included in the system [6-12].

The use of throttle flow dividers is an urgent solution to the issue of ensuring consistent control of hydraulic motors, regardless of the value of the applied load. [13-15].

2 Material and research methods

The dynamics of the synchronous hydraulic system based on the DDP, the design scheme of which is shown in Figure 1, can be described by a number of differential equations [16-19].

* Corresponding author: svet-tpl@yandex.ru

Let's consider the sequence of operation of a hydromechanical system equipped with a throttle flow divider of a non-drone type.

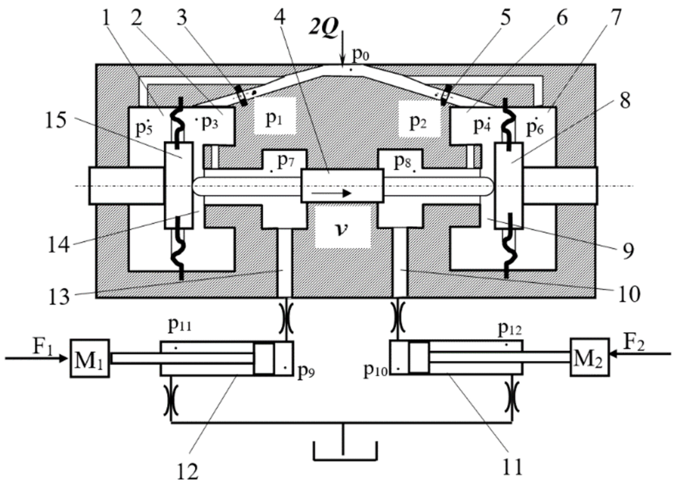


Fig. 1. Scheme of synchronous hydraulic system based on DDP

Let's consider a special case when synchronized hydraulic branches are loaded the same way. That is, the load values are the same, the flow rates of the working fluid passing through the input resistances (3, 5) are also equivalent, and therefore the values of pressure losses in the system will be equal. In this case, the control chambers (2, 6) are subjected to pressures different from the auxiliary chambers (1, 7) for the same difference. The design of the throttle divider includes plungers (14, 9) with the same diameters; therefore, the working areas of the plungers are equal. As a consequence, the forces transmitted to the movable rod with a variable cross-section (4) from the pressure drop on them will also be the same. The regulating element will remain stationary, due to the opposite direction of these efforts. It should be borne in mind that pressure losses on variable hydraulic resistances (13, 10) of the flat valve type remain equal.

Consider the second situation. In the case of an increase in the load on one of the hydraulic motors, for example M1 connected to the inlet (11), and on the other hydraulic motor (M2) connected to the outlet (12), remains unchanged, then such transformations will lead to a decrease in costs both in this branch and through the input resistance (5). Consequently, the value of the pressure drop in this section of the throttle divider will decrease, while the force from the pressure drop on the plungers (9) and (14) will also decrease accordingly. The control element will start moving and will attract a rod with a variable cross section (4) in the direction of closing the variable hydraulic resistance (13). As a result, the costs and resistances in the branches will increase, and the rotational speeds of the shafts of the hydraulic motors will synchronize.

The increment of pressure at point 0 of the design scheme is determined by the equation

$$\frac{dp_0}{dt} = C_0(2Q - Q_1 - Q_2), \tag{1}$$

where dp_0 – the increment of pressure at point 0 of the design scheme over time dt; C_0 – the reduced stiffness of the considered section of the pipe of the design scheme; Q – half of the flow rate at the input to the divider; Q_1 and Q_2 – expenses in the corresponding separated flows of the accident branches at the time under consideration.

$$Q_{1.2} = \mu_{1,2} f_{int} \cdot \sqrt{\frac{2}{\rho}(p_0 - p_{1,2})}, \tag{2}$$

where $\mu_{1,2}$ – the reduced consumption coefficient of the input sections of the branches; f_{int} – the cross–sectional area of the input sections of the branches; p_0 – the value of the pressure at the inlet to the throttle divider (0); $p_{1,2}$ – pressure in front of the sensitive elements of the corresponding branches.

The pressure increment at points 1 and 2 of the circuit is determined by the equation

$$\frac{dp_{1,2}}{dt} = C_{1,2}(Q_{1,2} \mp Q_{rb1,2} - Q_{c1,2}) \quad , \tag{3}$$

where $dp_{1,2}$ – the increment of pressure at the corresponding points over time dt; $C_{1,2}$ – the given hydraulic stiffness of the sections under consideration; $Q_{push1,2}$ – expenses through the bypass channels of the corresponding branches; $Q_{c1,2}$ – expenses through the corresponding sensitive elements.

$$Q_{rb1,2} = (f_{me1,2} - f_{torq})v_r \, , \tag{4}$$

$$Q_{c1,2} = \mu_{c1,2} f_{c1,2} \sqrt{\frac{2}{\rho}(p_{1,2} - p_{3,4})} \, , \tag{5}$$

where $f_{me1,2}$ – the effective areas of the corresponding membrane elements of the regulatory elements of the DDP; f_{push} – the cross-sectional area of the pusher in its expanded part; v_p – the speed of operation of the control element; $\mu_{c1,2}$ and $f_{c1,2}$ – the flow coefficient and the live cross-sectional area of the corresponding sensitive elements; $p_{3,4}$ – the pressure at points 3 and 4 of the design scheme, respectively, which are determined from the ratios.

$$\frac{dp_{3,4}}{dt} = C_{3,4}(Q_{c1,2} \pm Q_{me1,2} - Q_{\text{p}1,2}) \quad , \tag{6}$$

$$Q_{me1,2} = (f_{me1.2} - f_{ot})v_{\text{p}}, \tag{7}$$

$$Q_{\text{p}1,2} = \mu_{\text{p}1,2} f_{\text{p}1,2} \sqrt{\frac{2}{\rho}(p_{3,4} - p_{7,8})}, \tag{8}$$

where $dp_{3,4}$ – increment of pressure at points 3 and 4 of the calculation scheme over time dt; $C_{3,4}$ – the given hydraulic stiffness of sections 3 and 4 of the design scheme; $Q_{me1,2}$ – the flow of hydraulic fluid in the corresponding branches caused by a change in the position of the membrane elements; $Q_{p1,2}$ – the flow of hydraulic fluid in the corresponding branches passing through the variable resistance of the regulator; f_{ot} – the area of the hole of the variable resistance of the regulator; $\mu_{p1,2}$ and $f_{p1,2}$ – the corresponding reduced flow coefficients and the area of the live sections of the drain hydraulic lines; $p_{7,8}$ – pressure at points 7 and 8.

The pressure derivatives at points 7 and 8 of the calculation scheme are determined by the formulas

$$\frac{dp_{7,8}}{dt} = C_{7,8}(Q_{\text{p}1,2} - Q_{p1,2} \mp Q_{torq} \pm Q_{ot}), \tag{9}$$

$$Q_{p1,2} = f_{p1,2} v_{p1,2}, \tag{10}$$

$$Q_{push} = f_{push} v_{\text{p}}, \tag{11}$$

$$Q_{ot} = f_{ot} v_p \quad , \tag{12}$$

where $dp_{7,8}$ – the increment of pressure at points 7 and 8 of the calculation scheme over time dt; $C_{7,8}$ – the reduced hydraulic stiffness of sections 7 and 8 of the design scheme; $Q_{n1,2}$ – expenses in the piston cavities of the cylinders caused by the movement of the pistons; Q_{push} – flow caused by the movement of the regulator pusher; Q_{ot} – the flow rate caused by the movement of the membrane elements of the regulator relative to the corresponding holes; $f_{p1,2}$ and $v_{p1,2}$ – the area of the pistons of the corresponding hydraulic cylinders and the speed of their movement at the time under consideration; f_{push} – cross-sectional area of the remote stem.

The pressure at points 9 and 10 is determined from the conditions:
if

$$v_{rc1,2}=0 \text{ то } p_9 = p_7, \text{ and } p_{10} = p_8 ,$$

otherwise

$$p_9 = p_7 - \Delta p_{7-9} Sign(v_{p1}), \tag{13}$$

$$p_{10} = p_8 - \Delta p_{8-10} Sign(v_{p2}), \tag{14}$$

where $p_{9,10}$ – pressure at points 9 and 10 of the design scheme, respectively, Δp_{7-9} and Δp_{8-10} – pressure losses in the considered areas.

The pressure derivatives in the cavities of hydraulic cylinders with a rod are determined from the expressions

$$\frac{dp_{11,12}}{dt} = C_{11,12}(Q_{st1,2} - Q_{sl1,2}), \tag{15}$$

$$Q_{st.c1,2} = (f_{p1,2} - f_{st1,2})v_{p1,2}, \tag{16}$$

$$Q_{sl1,2} = \mu_{sl1,2} f_{sl1,2} \sqrt{\frac{2}{\rho}(p_{11,12} - p_{atm})}, \tag{17}$$

where $Q_{st1,2}$ – the corresponding expenses in the cavities of hydraulic cylinders with a rod, when moving the pistons; $Q_{sl1,2}$ – the corresponding flow rate in the drain branches of hydraulic cylinders; $f_{st1,2}$ – the cross-sectional areas of the rods of the corresponding cylinders; $\mu_{sl1,2}$ and $f_{sl1,2}$ - the given coefficients of the flow of the drain hydraulic lines and the area of their live sections, respectively; $p_{9,10}$ – pressure in the piston cavities of the corresponding hydraulic cylinders.

The velocities of the moving elements of the system and their movement can be determined by the equation below.

For hydraulic cylinders

$$\mathrm{M}_{1,2} \frac{dv_{1,2}}{dt} = f_{p1,2} p_{9,10} - (f_{p1,2} - f_{st1,2}) p_{11,12} - F_{1,2} \tag{18}$$

where $\mathrm{M}_{1,2}$ - corresponding masses of moving elements, given to the pistons of hydraulic cylinders; $F_{1,2}$ - corresponding forces applied to hydraulic cylinder pistons.

The equation of motion of the regulating element allows you to determine the speed of its movement:

$$m_{\text{per}} \frac{dv_{reg}}{dt} = (f_{me1.2} - f_{push})(p_5 - p_6) + f_{push}(p_7 - p_8) + f_o(p_8 - p_7) +$$
$$+\frac{1}{2}(f_{o.nar} - f_o)(p_4 + p_8 - p_3 - p_7) + (f_{me1.2} - f_{o.nar})(p_4 - p_3) + \frac{\rho}{f_o}(Q^2_{reg1} - Q^2_{reg2}), \tag{19}$$

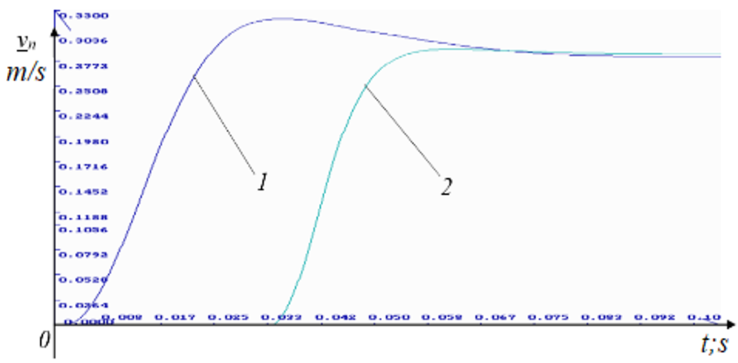
where $f_{me1.2}$ – the effective areas of the membrane elements are assumed to be the same for both branches, when they work in the "sluggish membrane" mode; f_{push}– the cross-sectional area of the pusher in its extended part; f_o – the area of the holes of the seats of variable hydraulic resistances; $f_{o.nar}$ – the area of the saddles by outer diameter.

 https://doi.org/10.1051/bioconf/20248402029

3 Results and discussion

In order to obtain changes in time in the pressure values at various points, the values of the displacement and position of the moving element in space, the velocity of movement in space and the values of velocities in real time and time, the above systems of equations were integrated.

The obtained values of the change in the speed of the shafts of synchronized hydraulic motors are shown in the form of graphs in Figure 2.



1. – $v_1(t)$; 2. – $v_2(t)$;

Fig. 2. Time changes in the speeds of motion of synchronized hydraulic motors

The following initial data were used for the calculations: the reduced mass of the first hydraulic cylinder is 400 kg and a force of 10^3N is applied to it; the reduced mass of the second hydraulic cylinder is 180 kg, and the force applied to it is $30*10^3$ N.

Such characteristics of the system as functional parameters, design features, properties of the working fluid and so on significantly affect the output parameters of the hydrodynamic system. In turn, the proposed mathematical model allows them all to be taken into account, and the data obtained as a result of calculations form the basis of system optimization.

Therefore, for the high-quality operation of the entire system as a whole, it is necessary to take a responsible approach to the selection of structural elements, and specifically to synchronize the system, use throttle flow dividers with appropriate dynamic characteristics.

In general use of multithreaded flow separators allows for the synchronous operation of several parallel hydraulic systems powered by a single power source, with high accuracy, reliability and with minimal costs.

4 Acknowledgement

The study was supported by a grant within the framework of the “Nauka-2030”.

References

1. O. F. Nikitin, Reliability, diagnostics and operation of hydraulic drive of mobile objects (Moscow Publishing House of Bauman Mos-cow State Technical University, 2007)

2. A. Pelipenko, A. Rybak, N. Vyborova, V. Zubtsov, D. Lugantsev, Energy Saving in Hydraulic Testing Systems, Lecture Notes in Networks and Sys-temsthis link is disabled, **510**, 1889–1896 (2023)
3. A.T. Rybak, I.K. Tsybri, A.Yu. Peli-penko, Test bench for hydraulic motors and pumps with energy recovery, Patent for utility model 204153 U1, 05/11/2021. Application No. 2020134672 dated 10/22/2020.0
4. A. T. Rybak, *Investigation of the regenerative hydromechanical system of the test bench of volumetric hydraulic machines*, Bulletin of the Don State Tech-nical University, 11, **9(60)**, 1651-1661 (2011)
5. A.Yu. Pelipenko, A.T. Rybak, *Stand for carrying out life tests of plunger hydraulic cylinders with energy recover*y IOP Conference Series: Materials Science and Engineering, **1001(1)**, 012001 (2020) doi:10.1088/1757-899X/1001/1/012001.
6. A.T. Rybak, I.K. Tsybri, S.V. Nosachev, A.R. Zenin, Theoretical foundations for calculating the control system of the hydraulic drive of the stand for testing piston hydraulic cylinders. Bulletin of the Don State Technical University, **19(3)**, 242-249 (2019) doi.org/10.23947/1992-5980-2019-19-3-242-249
7. A.T. Rybak, A.R. Zenin, A.Yu. Pelipenko, Test bench for reciprocating hydraulic cylinders with energy generation, Patent for invention 2796721 C1, 05/29/2023. Application No. 202213826 dated December 22 (2022)
8. S.A. Nesterov, Mathematical model of contacting the tool and the processed surface in the implementation of SELF-programs, Izvestiya Kabardino-Balkarian State University, **4(5)**, 26-27 (2014)
9. A.E. Zverovshchikov, P.A. Gurin, E.A. Charina, Modeling of contact interaction of working bodies with internal channels of small-sized parts, In the collection: Systems of design, modeling, production preparation and project management CAD/CAM/CAE/PDM. collection of articles of the IX International Scientific and Practical Conference, 34-38 (2015)
10. A. E. Zverovshchikov, Simulation of the kinematics and gas dynamics of the centrifugal dispersion stand, Journal of Physics: Conference Series, Krasnoyarsk, Russia, Krasnoyarsk Science and Technology City Hall of the Russian Union of Scientific and Engineering Associations, Krasnoyarsk, Russia: IOP Publishing Ltd, 42046 (2021) doi 10.1088/1742-6596/2094/4/042046
11. A. T. Rybak, Improving the scientific and methodological foundations of designing drive systems of technological machines, Bulletin of the Don State Technical University, Rostov–on-Don, 10, **2(45)**, 249-257 (2010)
12. A.T. Rybak, M.P. Shishkarev, A.A. Demyanov, V.P. Zharov, *Modeling and calculation of hydromechanical systems dynamics based on the volume rigidity theory*, MATEC Web of Conferences, **226**, 01001 (2018)
13. A. T. Rybak, On the issue of calculating the reduced coefficient of volu-metric rigidity of hydrolines, Col-lection of tr. international scientific and practical conference "Mechanics, equipment, materials and technologies" dedicated to the 100th anniver-sary of the Kuban State University. technol. un-t", Krasnodar 169-175 (2018)
14. A.A. Abalov, S.V. Nosachev, V.P. Zharov, V.A. Minko, *Using the simintech dynamic modeling environment to build and check the operation of automation systems*, MATEC Web of Conferences, **226**, 04003 (2018) doi:10.1051/matecconf/201822604003.
15. B.A. Kartashov, Dynamic mode environment of technical systems SimInTech : manual, 424 (Moscow: DMK-Press, 2017)

16. D. Rudoy, V. Pakhomov, A. Babajanyan, T. Maltseva, E. Rumyantseva, A. Tatarova, *Review and analysis of extrusion technology in the production of feed additives based on probiotic microorganisms*, E3S Web of Conf., **413**, 01014 (2023) doi: 10.1051/e3sconf/202341301014
17. A. Kolinko, S. Kambulov, I. Chervyakov, D. Rudoy, A. Olshevskaya, *Investigation of the uniformity of seed distribution during sowing of winter wheat*, E3S Web of Conf., **413**, 01009 (2023) doi: https://doi.org/10.1051/e3sconf/202341301009
18. A. Rybak, R. Gorbunov, A. Olshevskaya, N. Ugrekhelidze, M. Egyan, Mathematical Model of a Throttle Flow Divider with an Elastic Regulating Element, XIV International Scientific Conference “INTERAGROMASH 2021". Lecture Notes in Networks and Systems, Springer, Cham., **246** (2022) doi.org/10.1007/978-3-030-81619-3_23
19. A. Rybak, B. Meskhi, A. Pelipenko, D. Rudoy, A. Olshevskaya, Y. Serdyukova, M. Odabashyan, E. Alentsov, *Mathematical modelling of the hydromechanical drive of the test bench for plunger hydraulic cylinders with energy recovery*, E3S Web of Conferences, **413**, 02047 (2023) https://doi.org/10.1051/e3sconf/202341302047