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Mikhail ZHILEVICH[1], Sergey ERMILOV[2], Denis KAPSKI[3], Yuriy VOVK[4], Oleg LYASHUK[5], Iryna VOVK[6]

METHOD OF CALCULATING THE DESIGN PARAMETERS OF A MODULATOR ANTI-LOCK BRAKING SYSTEM WITH A HIGH FLOW OF WORKING FLUID

Summary. The design dimensions of the executive hydraulic cylinders of the brake system of heavy-duty mining dump trucks cause high fluid flow during the braking process. Therefore, dimensions of the anti-blocking system modulator spool pair require unique electromagnets or hydraulic amplifiers to control. These solutions do not allow the required modulator performance. Thus, a modulator scheme with a division of the flow of fluid from the source to the brake cylinders was developed. This scheme allows during emergency braking passing,

[1] Department of Hydropneumatic Automatics and Hydropneumatic Drive, Belarusian National Technical University, Nezavisimost Avenue 65, 220013 Minsk, Belarus. Email: mzhilevich@bntu.by. ORCID: https://orcid.org/0000-0002-5600-8250
[2] Department of Hydropneumatic Automatics and Hydropneumatic Drive, Belarusian National Technical University, Nezavisimost Avenue 65, 220013 Minsk, Belarus. Email: gpa_atf@bntu.by. ORCID: https://orcid.org/0000-0003-0861-6478
[3] Department of Transport Systems and Technologies, Belarusian National Technical University, Nezavisimost Avenue 65, 220013 Minsk, Belarus. Email: d.kapsky@bntu.by. ORCID: https://orcid.org/0000-0001-9300-3857
[4] Department of Automobiles, Ternopil Ivan Puluj National Technical University, 56, Rus'ka Str., 46001 Ternopil, Ukraine. Email: vovkyuriy@ukr.net. ORCID: https://orcid.org/0000-0001-8983-2580
[5] Department of Automobiles, Ternopil Ivan Puluj National Technical University, 56, Rus'ka Str., 46001 Ternopil, Ukraine. Email: oleglashuk@ukr.net. ORCID: https://orcid.org/0000-0003-4881-8568
[6] Department of Automobiles, Ternopil Ivan Puluj National Technical University, 56, Rus'ka Str., 46001 Ternopil, Ukraine. Email: vovk.ira.2010@gmail.com. ORCID: https://orcid.org/0000-0002-4617-516X

an additional amount of fluid to cylinders through the valve, installed parallel to the main valve upon pressure increase phase and controlled by the pressure difference. The task is to develop a method for calculating the main structural dimensions of a modulator. The calculation of the valve of the second cascade, installed in parallel to the main stage, is carried out for the emergency braking mode with the maximum flow rate to ensure the required performance of the braking system. The balance of fluid flows equations is compiled at the key points. The flow rate of the fluid through each of the valves is determined by the Torricelli formula, and the pressure difference across the valves is assuned equal. The obtained relations allows building a family of Q-p curves, which can be used to select the diameter and stroke of the additional valve depending on the flow rate in the brake system.

Keywords: anti-lock braking system, hydraulic brake system, modulator, method of calculating, dump truck

1. INTRODUCTION

A mandatory requirement for modern cars is to ensure a minimum braking distance while maintaining a stable and controlled movement during braking. To fulfil this condition, anti-lock braking systems (ABS) are used. The first known ABS patents date back to the late 1920s. However, until the end of the 1940s, such systems were used only on experimental and sports cars. Active research began after the introduction of electronic ABS on Ford cars in 1969.

Currently, ABS is an obligatory structural part of almost the entire range of cars. Pneumatic antilock braking systems were widely used. Hydraulic ones are usually used in light-duty cars and trucks. As a rule, improvement of systems is currently carried out by developing optimal control algorithms for the anti-lock system with an established structure and design of actuators [1].

Following UNECE Regulation No. 13, the use of ABS in heavy-duty vehicles is not regulated [2]. However, international manufacturers are working on the installation of ABS prototypes in mining trucks (BelAZ, Liebherr, Caterpillar) [3-5]. One of the main problems in the implementation of anti-lock systems on mining trucks is the large size of the actuators, and as a result, the high flow rate of the working fluid in the brake drive.

2. LITERATURE REVIEW

ABS is a closed system of automatic control, the main task of which is to change the speed of rotation of the wheels of a vehicle by adjusting the pressure in the brake lines during braking. The ABS actuator that controls the pressure in the brake cylinders is a pressure modulator.

As a rule, the design of the modulator includes actuating elements of the spool or valve type with electromagnetic control. In [6], a comparative analysis of such executive elements is carried out. The calculations showed that for the operation of the spool-type actuators, a 20% lower electromagnet force is required, while valve elements have a higher sensitivity. It further found that the stability of the flow characteristics at equal pressures for spool valves provided in the frequency range 25-40% greater than for valve valves [6]. In this regard, it recommended using valve elements only in multiphase modulators for draining the liquid and relieving pressure. The apparatus with spool type actuators has no application restrictions and can be installed in both the pressure and the drain lines of a hydraulic brake drive.

By the type of duty cycle, modulators are built based on discrete elements and can be two-phase and three-phase ones. The advantage of a two-phase cycle is the simplicity of the design of the ABS modulator. However, a three-phase cycle allows reducing the flow rate of the working fluid and increasing the average braking torque. The presence of a holding phase helps to prevent water hammer and wave processes in pipelines with a sharp change in the direction of fluid flow.

The two-phase algorithm of the open-drive modulator implemented using a three-line on-off valve. For three-phase and multiphase algorithms, combinations of such valves are used.

The use of proportional equipment allows creating a multi-phase non-cyclic algorithm to maintain the braking torque in a given optimal range during braking of the vehicle.

The classification scheme of ABS modulators is shown in Figure 1.

Design features of hydraulic modulators and their work are considered in [6-8].

As noted earlier, in ABS modulators of hydraulic brake drives, disc-type spool valves with electromagnetic control are usually used. The low flow rates of the hydraulic fluid in the hydraulic brake drive allow the use of valves with small spool sizes. In this case, the controls are commercially available electromagnets, since it does not require much effort to move the spools. However, on particularly heavy-duty vehicles, the dimensions of the executive hydraulic cylinders, and as a consequence, the flow rate of the working fluid increase significantly. Accordingly, the structural dimensions of the spool pairs of the modulator and their inertia grow. To control such spools, it is necessary to use either hydraulic boosters, which reduces the speed of the brake drive as a whole and does not allow providing the required frequency of operation of the modulator, or special forced electromagnets that provide the specified frequency and force on the plunger.

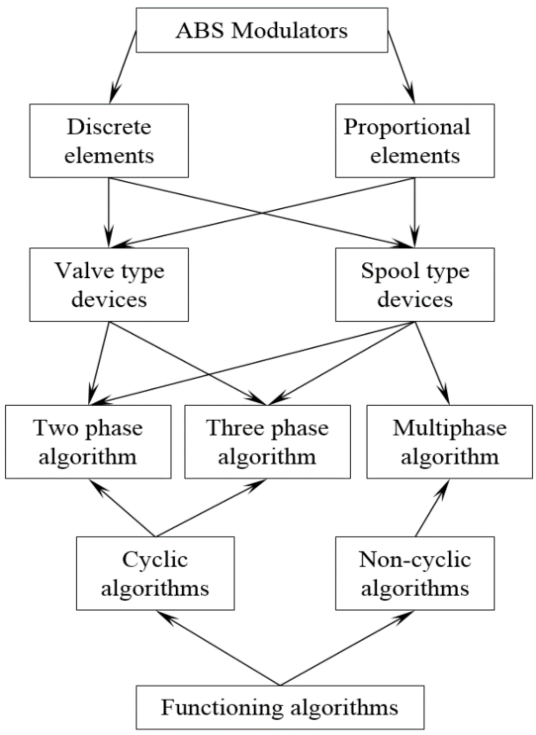


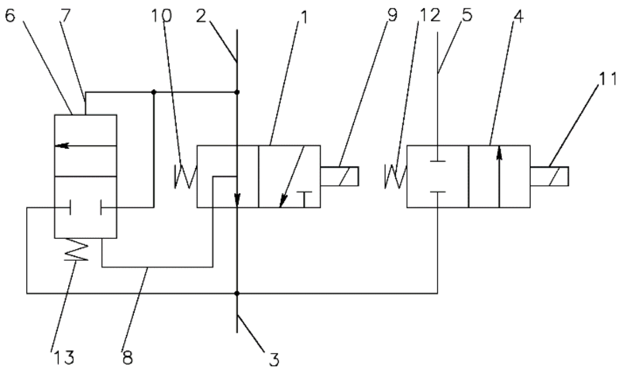
Fig. 1. Classification of ABS modulators

The circuit solution [9], justified in [10], presented in Figure 2, allows to reduce the inertia of the elements of the hydraulic modulator and to ensure the given speed of the anti-lock system at high flow rates of the working fluid. The required actuator response speed is achieved by dividing the flow of the working fluid into the main and auxiliary using an additional normally closed valve installed parallel to the main.

3. METHODOLOGY

The proposed modulator applied when the anti-lock braking system operated using a two- or three-phase algorithm.

During the braking phase, the electromagnets 9 and 11 are disconnected. The fluid flow passes from the pressure source through inlet valve 1 to the brake cylinder. As the conditional passage of the inlet is small, it causes a pressure drop to occur in lines 2 and 3, transmitted via control lines 7 and 8 to the shut-off element of the auxiliary valve 6, which held through a spring 13 in the upper position according to the scheme. When the pressure differential reaches a certain value, the auxiliary valve's spool 6 moves down according to the scheme, passing an additional flow of working fluid to the brake cylinder through line 3.



1 – inlet valve; 2 – supply line of the working fluid; 3 – line for supplying a working fluid to the brake cylinder; 4 – exhaust valve; 5 – line pressure relief; 6 – auxiliary valve; 7, 8 – control lines of the auxiliary valve; 9, – control valve of the intake valve; 10 – intake valve return spring, 11 – exhaust solenoid control valve; 12 – exhaust valve return spring; 13 – additional auxiliary return spring

Fig. 2. Hydraulic modulator

The braking phase carried out when the control signals supplied to the electromagnets 9 and 11. In this case, the pressure drop in lines 7 and 8 becomes equal to zero and the auxiliary valve 6 closes. The fluid from the brake cylinder through valve 4 and line 5 is discharged.

The holding phase (when using the three-phase cyclic algorithm) provided by removing the signal from the electromagnet 11 of the valve 4 with the control signal turned on the electromagnet 9 of the inlet valve 1. The spoon of the exhaust valve 4 shifted to the right according to the scheme and cuts off the hydraulic cylinder from the drain line. The valve 1 is in the left position in the scheme because the pressure in the lines 7 and 8 are equal and the spoon of the valve 6 is in the upper position according to the scheme by the spring 13.

4. RESULTS

4.1. Determination of the modulator main parameters for a hydraulic actuator with a large flow of working fluid

When performing the static calculation, the following assumptions were observed:
- no friction losses;
- all elements of the modulator are absolutely rigid;
- the modulator is absolutely tight.

The design scheme of the modulator for the braking phase is shown in Figure 3.

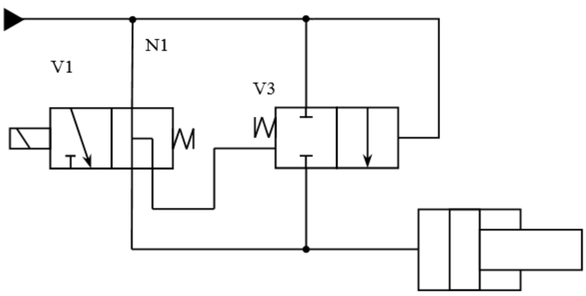


Fig. 3. The design scheme of the modulator (braking phase)

In the calculations, it assumed that the dimensions of the hydraulic valve V1 of the first cascade (the main hydraulic valve of the braking phase) is known in advance or can be set based on the sizes of the spool pairs of serially produced hydraulic valves with electromagnetic control. The calculation of the valve V3 of the second cascade installed parallel to the main one, carried out for emergency braking with the maximum flow rate of the working fluid, which provides the specified quick-action brake system.

The balance of the flow rate of the working fluid in the node N1 represented as the equality of the sums of the flow rates of the fluid at the input Q_{in} and output Q_{out}.

The inlet flow rate Q_{in} is the fluid flow Q coming from the pressure source.

The output flow rate Q_{out} represented by the sum of expenses:

$$Q_{out} = Q_{v1} + Q_{v3}, \tag{1}$$

where Q_{v1}, Q_{v3} are flow rates in valves V1 and V3, respectively.

The flow rate of the working fluid through the hydraulic valve is determined by the formula:

$$Q_i = \mu_i S_i \sqrt{\frac{2\Delta p_i}{\rho}}, i = 1,3, \tag{2}$$

where:
μ_i is the flow coefficient of the i-th valve;
Δp_i is pressure drop of the i-th valve;
ρ is the density of the working fluid;
S_i is the area of the bore of the working window of the i-th valve.

The pressure drops on both valves are taken equal to:

$$\Delta p_1 = \Delta p_3 = \Delta p \,. \tag{3}$$

For a cylindrical spool, the flow area determined by the Eq.:

$$S_i = \pi d_{spi} x_i , i = 1{,}3, \tag{4}$$

where:
d_{spi} is the diameter of the spool pair of the i-th valve;
x_i is movement of the spool of the i-th valve.

Substituting expressions (3) and (4) in (2), we obtain:

$$Q = \mu_1 \pi d_{sp1} x_1 \sqrt{\frac{2\Delta p}{\rho}} + \mu_3 \pi d_{sp3} x_3 \sqrt{\frac{2\Delta p}{\rho}} \,. \tag{5}$$

The flow-differential characteristic of the valve of the second cascade obtained from formula (5) is the dependence of the pressure loss Δp on several variables: the passing flow Q and the ratio of the spool diameter d_{sp3} of the valve of the second cascade and its stroke x_3:

$$\Delta p = \frac{2}{\rho}\left(\frac{1}{\mu_1 \pi d_{sp1} x_1 + \mu_3 \pi d_{sp3} x_3}\right)^2 Q^2 \,, \tag{6}$$

In addition, from equation (5), the dependence of the product of the parameters of the working window of the auxiliary valve P3 $d_3 3 x_3 3$ can be obtained as a function of several variables: pressure loss Δp and flow rate Q:

$$d_{sp3} x_3 = \frac{Q}{\pi \mu_3} \sqrt{\frac{\rho}{2\Delta p}} - \frac{\mu_1}{\mu_3} d_{sp1} x_1 \,, \tag{7}$$

The obtained dependences (6) and (7) allow us to build a family of flow-differential characteristics. According to these characteristics, it is possible to select the diameter and stroke of the valve spool of the second cascade depending on the flow rate in the brake system and allowable pressure losses on the modulator, thus, ultimately design a standard range of modulators for brake systems with different flow rates.

5. RESULTS

The initial data for calculating the flow-differential characteristic is presented in Table 1, while the initial data for calculating the parameters of the working window dsp3x3 of the auxiliary valve P3 is shown in Table 2.

The calculation results are presented in Figures 4 and 5.

The obtained characteristics (Fig. 4 and Fig. 5) show that increasing the area of the working window of the auxiliary valve V3 will significantly reduce the pressure drop across the modulator.

Tab. 1

The initial data for the calculation of flow-differential characteristics

Fluid density	ρ	880 kg/m^3
Fluid flow	Q	40–120 l/min
The coefficient of flow of the main valve V1	μ_1	0.65
Spool diameter of the main valve V1	d_{sp1}	8 mm; 10 mm
Spool stroke of the main valve V1	h_{sp1}	0.5 mm; 1.0 mm
The coefficient of flow of the auxiliary valve V3	μ_3	0.65
Spool diameter of the auxiliary valve V3	d_{sp3}	10 mm; 16 mm; 20 mm; 25 mm; 32 mm
Spool stroke of the auxiliary valve V3	h_{sp3}	1.0 mm; 1.5 mm; 2.0 mm; 2.5 mm; 3.0 mm

Tab. 2

The source data for calculating the parameters of
the worker windows d_{sp3}×x_3 of the auxiliary valve V3

Fluid density	ρ	880 kg/m^3
Fluid flow	Q	40–120 l/min
The coefficient of flow of the main valve V1	μ_1	0.65
Spool diameter of the main valve V1	$d_{зол1}$	8 mm; 10 mm
Spool stroke of the main valve V1	$h_{зол1}$	0.5 mm; 1.0 mm
The coefficient of flow of the auxiliary valve V3	μ_3	0.65
Differential pressure on the auxiliary valve P3	Δp	0.5 MPa; 1.0 MPa; 1.5 MPa; 2,0 MPa; 3.5 MPa

6. CONCLUSIONS

The designed scheme allows the use of hydraulic valves with ordinary electromagnetic control elements as intake and exhaust valves of an ABS modulator of brake systems with a high flow rate of working fluid.

The calculated static characteristics of the modulator according to the quality criteria correspond to the expected ones, which confirms the adequacy of the proposed model.

The selection of the structural dimensions of a normally closed valve installed parallel to the inlet valve makes it possible to create a standard range of modulators for systems with different flow rates of the working fluid based on the use of standard, widely used hydraulic control valves and electromagnetic control devices as inlet and outlet valves.

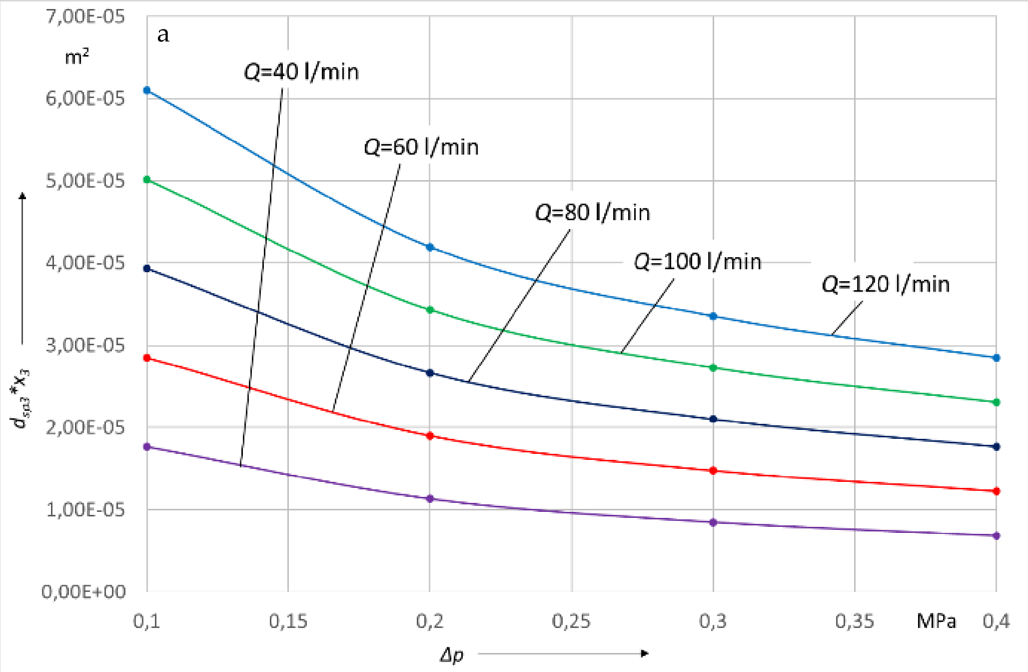


Fig. 4a. Flow-rate characteristic of the proposed modulator with parameters of the working window of the main valve V1; a) $d_{sp1}\times x_1$=8mm×0.5mm; b) $d_{sp1}\times x_1$=8mm×1.0mm; c) $d_{sp1}\times x_1$=10mm×0.5mm; d) $d_{sp1}\times x_1$=10mm×1.0mm

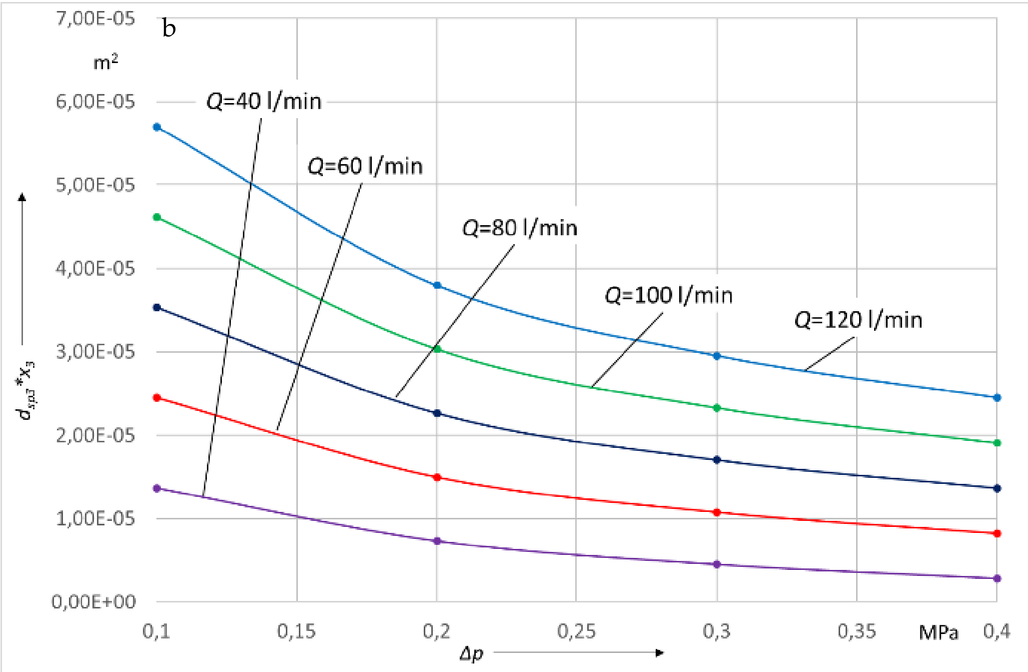


Fig. 4b. Flow-rate characteristic of the proposed modulator with parameters of the working window of the main valve V1; a) $d_{sp1}\times x_1$=8mm×0.5mm; b) $d_{sp1}\times x_1$=8mm×1.0mm; c) $d_{sp1}\times x_1$=10mm×0.5mm; d) $d_{sp1}\times x_1$=10mm×1.0mm

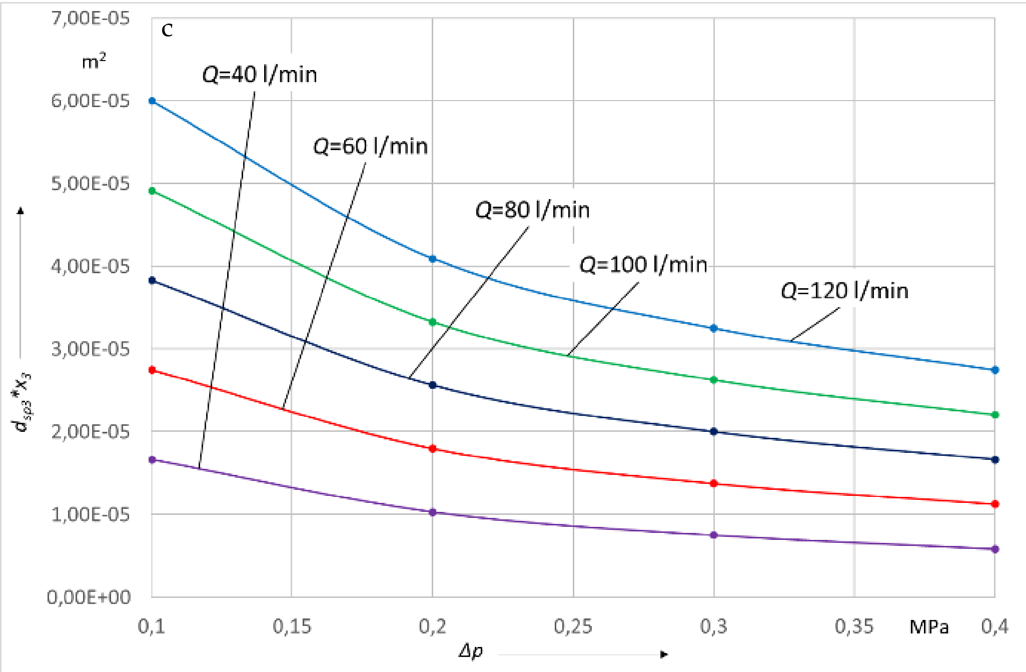


Fig. 4c. Flow-rate characteristic of the proposed modulator with parameters of the working window of the main valve V1; a) $d_{sp1}\times x_1$=8mm×0.5mm; b) $d_{sp1}\times x_1$=8mm×1.0mm; c) $d_{sp1}\times x_1$=10mm×0.5mm; d) $d_{sp1}\times x_1$=10mm×1.0mm

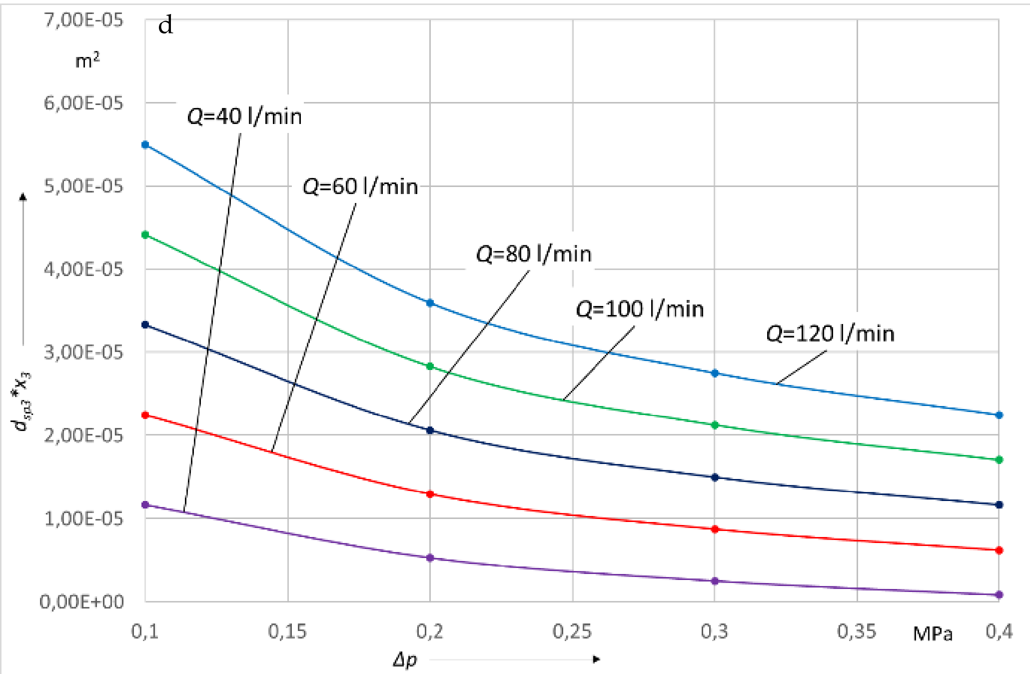


Fig. 4d. Flow-rate characteristic of the proposed modulator with parameters of the working window of the main valve V1; a) $d_{sp1}\times x_1$=8mm×0.5mm; b) $d_{sp1}\times x_1$=8mm×1.0mm; c) $d_{sp1}\times x_1$=10mm×0.5mm; d) $d_{sp1}\times x_1$=10mm×1.0mm

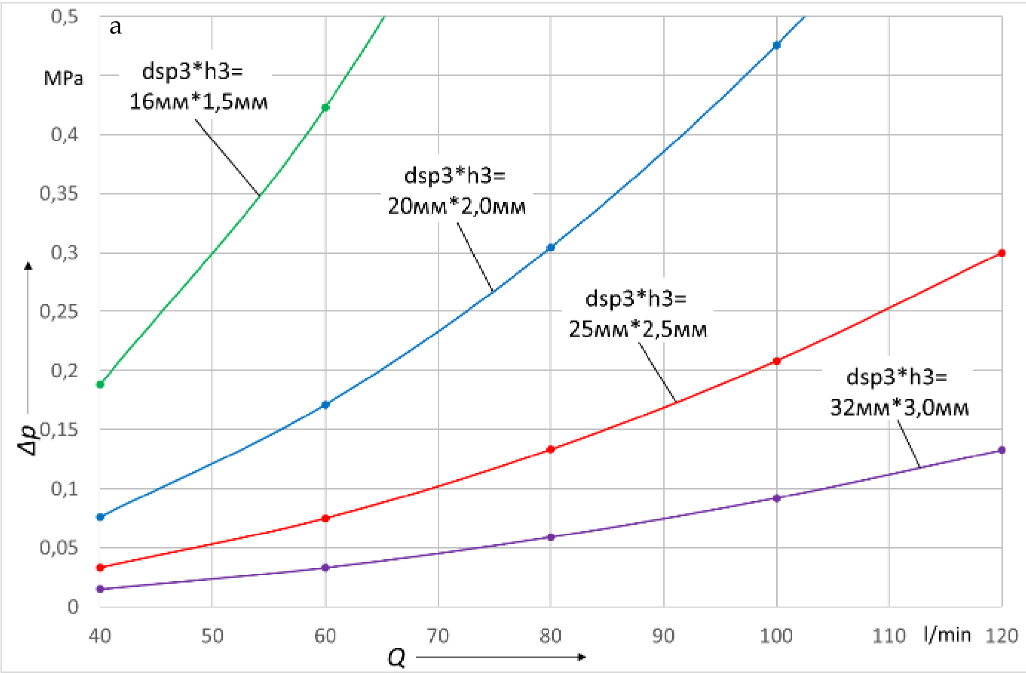


Fig. 5a. The design scheme of the modulator (braking phase)

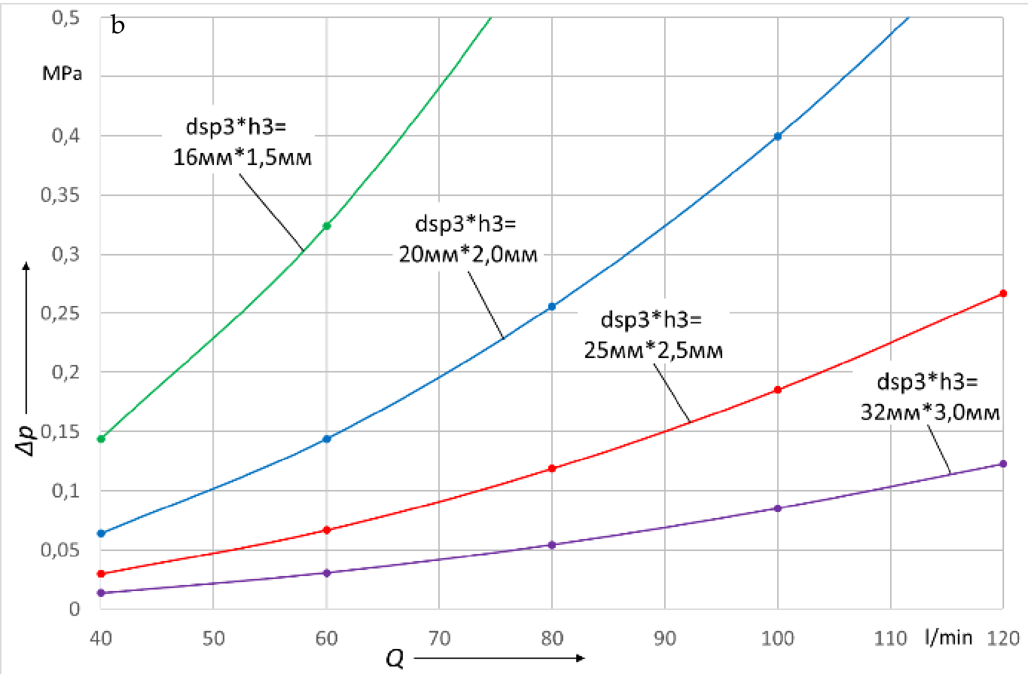


Fig. 5b. The design scheme of the modulator (braking phase)

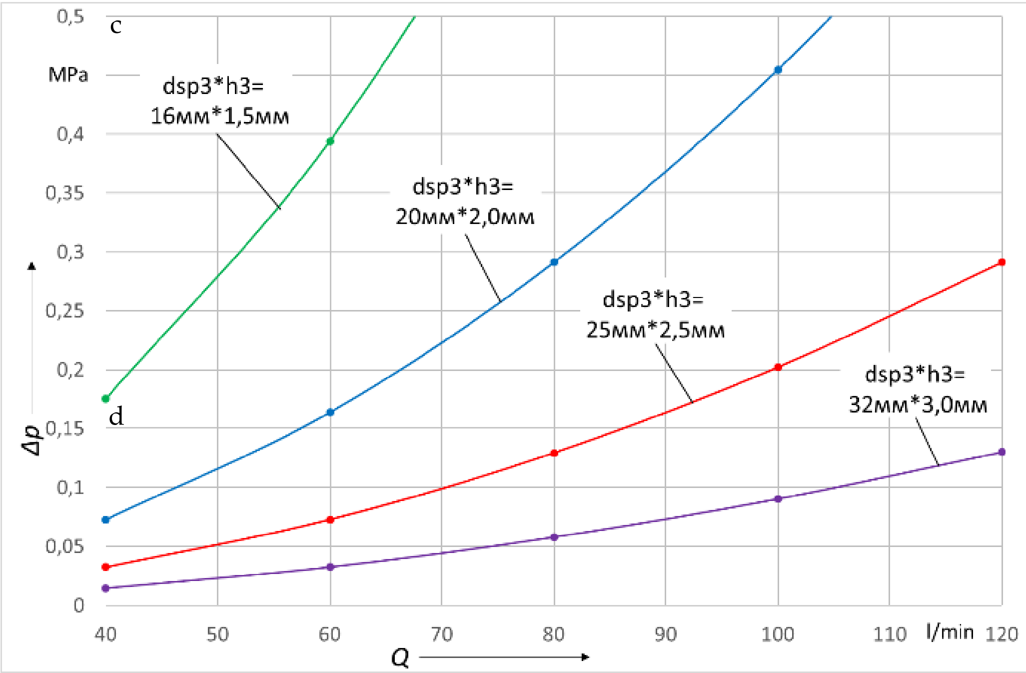


Fig. 5c. The design scheme of the modulator (braking phase)

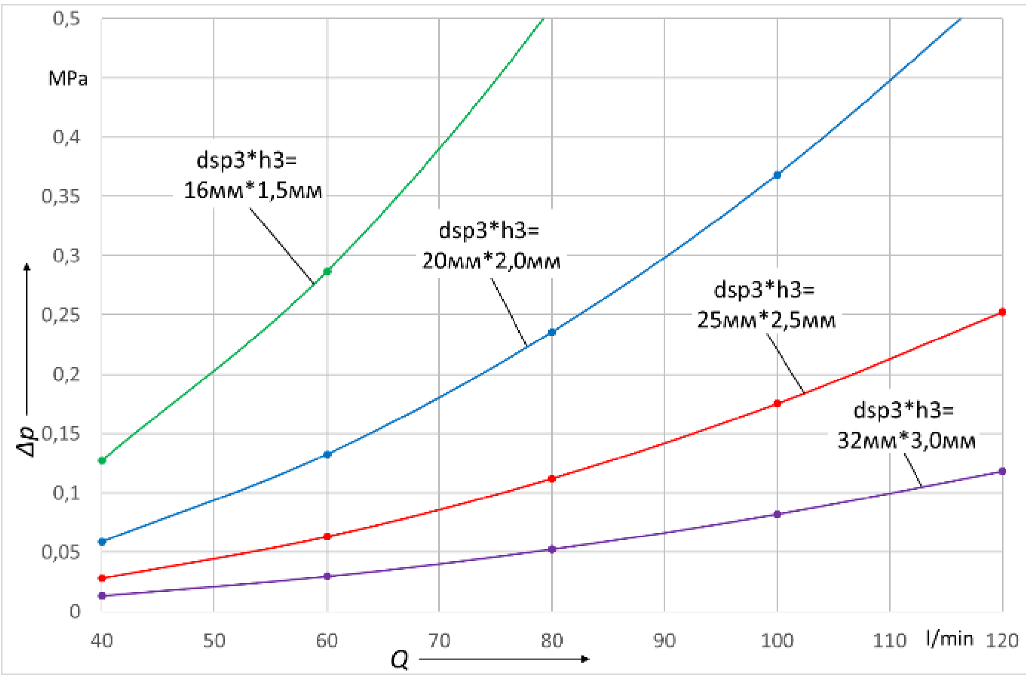


Fig. 5d. The design scheme of the modulator (braking phase)

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