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Analysis of Hydro Motor Velocity Stabilization

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Introduction

The progress of production technologies is closely related to the highly qualified control of machinery. The proper control of power flows used for the drive of technological implements is of essential importance. In most cases, the maintenance of stable speed at variable loads becomes the underlying guarantee of quality of the production process. In the most processing equipment and portable machines the power distribution to peripheral devices is carried out by using hydrostatic drives where the rotation speed of a hydraulic motor needs to be stabilized. The various known engineering solutions of this problem are mostly based on the control of a hydraulic fluid flow supplied to a hydraulic motor and the latter's displacement by using a rotation speed sensor [4]. Also the solutions where flow regulators or flow dividers [1,5] are used for the stabilization of the rotation speed are known. In this study, two systems of the rotation speed stabilization of a hydraulic engine by using flow dividers on the inlet of the latter have been suggested.

The work‘s object and method

This study deals with two types of hydraulic drives with hydraulic fluid flow dividers, which also carry out the functions of pressure converters. For such purpose the sections of flow dividers can be connected in series or in parallel. Each connection type provides a flow converter with different functional features. Besides that the variable displacement hydraulic machines are used in the flow divider sections.

The schemes under consideration are shown in figures 1 and 2. Hereinafter they are conditionally referred to as follows: the drive shown in figure 1 as "A" type drive, the drive shown in figure 2 as "B" type drive.

Two variable displacement hydraulic machines 1 and 2 comprising the fluid flow divider are used in both types of drives. In the "A" type drive the inlets of both hydraulic machines 1 and 2 are connected to the hydraulic fluid line where a fluid pressure p=const, i.e. both sections of the flow divider are connected in parallel.

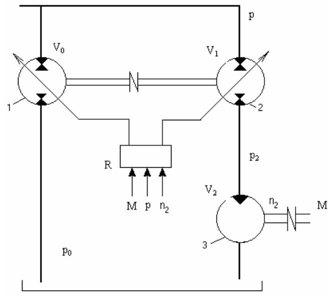

Fig. 1. The transmission with parallel connection of stream divider sections

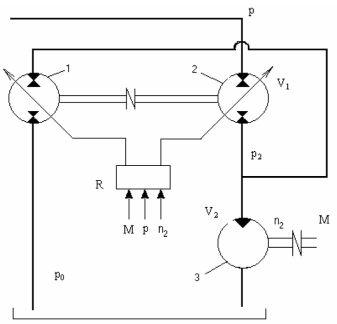

Fig. 2. The transmission with consecutive connection of stream divider sections

In the "B" type drive only the feeding line of the hydraulic machine 2 is connected to the hydraulic fluid line, whereas the feeding line of the hydraulic machine 1 is connected to the return line of the hydraulic machine 2, i.e. the sections of the flow divider are connected in series. In both drives the return lines of the flow divider section 2 are connected to the feeding line of the drive hydraulic motor 3. The pressure of fluid getting into the hydraulic motor 3 is $p_2 \geq p$. The hydraulic motor 3 is loaded with a resistive torque M=$M(t)$. The rotors of both hydraulic machines 1 and 2 of the flow divider are interconnected and the return line of the hydraulic machine 1 is connected to the reservoir, where a pressure is p_0=0.

The signals for the control of displacements of the hydraulic machines V_0 and V_1 are formed by the controller R according to the corresponding algorithm [1], by using the signal M of the torque sensor; the pressure signal p at the hydraulic motor inlet; the signal n_2 from the sensor of rotation speed of a hydraulic motor shaft.

The signal n_2 from the sensor of rotation speed of a hydraulic motor shaft is used only when putting the system into the stabilization mode where the control signal is formed by using the external signals of torque M and pressure p.

In the study [1, 4], the analysis of such systems in the stationary operation mode is provided.

After assessment of the possibilities of analysis methods, the simulation modeling system *Simulink* [2, 3] has been chosen. Mathematical models and computational schemes were developed for the diagrams shown in figures 1 and 2. The speed of rotation of a hydraulic motor shaft is described by the following differential equation:

$$p_2 V_2 = M + I_2 \dot{n}_2 \quad n_2(0) = n_0, \tag{1}$$

where p_2 is a hydraulic fluid pressure at the hydraulic motor inlet (Pa); V_2 is a cyclical displacement of the hydraulic motor (m^3); M is a resistive torque of the hydraulic motor (N·m); I_2 is a reduced mass moment of inertia of the moving masses ($N \cdot m \cdot s^2$); n_2 is a rate of revolution of the hydraulic motor shaft (s^{-1}).

The hydraulic fluid pressure p_2 in the "A" system is described by the following equation:

$$p_2 = p \frac{V_1 + V_0}{V_1}. \tag{2}$$

The pressure p_2 in the "B" system is equal to:

$$p_2 = p \frac{V_1}{V_1 - V_0}, \tag{3}$$

where p is a hydraulic fluid pressure at the inlet of the system (Pa); V_0 and V_1 are displacements of the flow divider sections (m^3).

Where there is no flow divider in the system $p_2 = p$.

The following control signals should be formed by the controller for the stabilization of rotation speed of the hydraulic motor:

- in system A0 (Fig. 1) controlling of displacement V_0 of the 1st hydraulic machine

$$V_{A0} = \begin{cases} V_1 \dfrac{M - p_1 V_2}{p V_2}, & if \quad n_{st} - \Delta n_2 \le n_2 \le n_{st} + \Delta n_2, \\ V_{0\max}, & if \quad n_2 \le n_{st} - \Delta n_2, \\ V_{0\min}, & if \quad n_2 \ge n_{st} + \Delta n_2; \end{cases} \tag{4}$$

- in system A1 (Fig. 1) controlling of displacement V_1 of the 2nd hydraulic machine

$$V_{A1} = \begin{cases} V_0 \dfrac{p V_2}{M - p_2 V_2}, & if \quad n_{st} - \Delta n_2 \le n_2 \le n_{st} + \Delta n_2, \\ V_{1\min}, & if \quad n_2 \le n_{st} - \Delta n_2, \\ V_{1\max}, & if \quad n_2 \ge n_{st} + \Delta n_2; \end{cases} \tag{5}$$

- in system B0 (Fig. 2) controlling of displacement V_0 of the 1st hydraulic machine

$$V_{B0} = \begin{cases} V_1 \dfrac{M - p V_2}{p V_2}, & if \quad n_{st} - \Delta n_2 \le n_2 \le n_{st} + \Delta n_2, \\ V_{0\max}, & if \quad n_2 \le n_{st} - \Delta n_2, \\ V_{0\min}, & if \quad n_2 \ge n_{st} + \Delta n_2; \end{cases} \tag{6}$$

- in system B1 (Fig. 2) controlling of displacement V_1 of the 2nd hydraulic machine

$$V_{B1} = \begin{cases} V_0 \dfrac{M}{M - p_2 V_2}, & if \quad n_{st} - \Delta n_2 \le n_2 \le n_{st} + \Delta n_2, \\ V_{1\min}, & if \quad n_2 \le n_{st} - \Delta n_2, \\ V_{1\max}, & if \quad n_2 \ge n_{st} + \Delta n_2; \end{cases} \tag{7}$$

where V_0, V_1, V_2 – the displacements of the hydraulic machines; p, M, n_2 – sensor signals; $V_{0\min}$, $V_{0\max}$, $V_{1\min}$, $V_{1\max}$ – the minimum and maximum displacement values of the hydraulic machines; n_{st} – a stabilized value of the rate of revolutions of the hydraulic motor; Δn_2 – a stabilization tolerance of the rate of revolutions of the hydraulic motor.

If the stabilization of the rate of revolutions of the hydraulic motor is carried out directly by changing its displacement "SINGLE MOTOR", then the following control signal should be formed:

$$V_2 = \begin{cases} \dfrac{M}{p}, & if \quad n_{st} - \Delta n_2 \le n_2 \le n_{st} + \Delta n_2, \\ V_{2\max}, & if \quad n_2 \le n_{st} - \Delta n_2, \\ V_{2\min}, & if \quad n_2 \ge n_{st} + \Delta n_2; \end{cases} \tag{8}$$

where $V_{2\max}$ and $V_{2\min}$ – the minimum and maximum displacement values of the hydraulic machines.

In the real system, a displacement regulator changes the displacement of the regulated hydraulic machine at a finite speed S_V according to the control signal (4–8). In S-models of the systems under consideration developed for such simulation of the regulator operation we use the Simulink Library Browser environment code "Rate Limited" [3] specialized for this function. S-models of the systems under consideration are shown in Fig. 3.

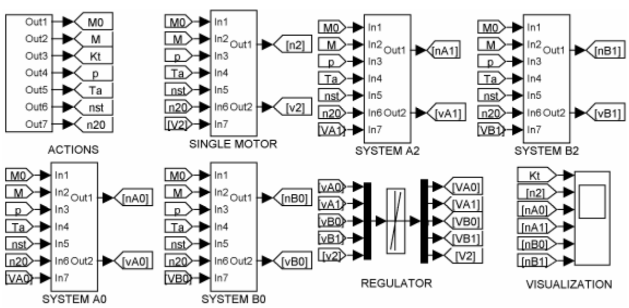

Fig. 3. S-models of the investigated hydro systems

They are based on the lower hierarchy subsystems "SYSTEM A0", "SYSTEM A1", „SYSTEM B0", „SYSTEM B1" and „SINGLE MOTOR" of S-models. The structure of these subsystems is shown in Fig. 4. The value of the mass moment of inertia I_2 going into the differential equation (1) is calculated by the subsystem blocks "[I2]":

$$I_2 = T_a \frac{M_0}{n_{st}}, \tag{9}$$

where T_a – the time (s) needed for the increase of rotation speed of the hydraulic motor up to the value n_{st} under the action of the torque M_0 (N·m).

The values of the arguments in the equations (1–9) are generated in the general subsystem "ACTIONS" of the S-model (Fig. 5).

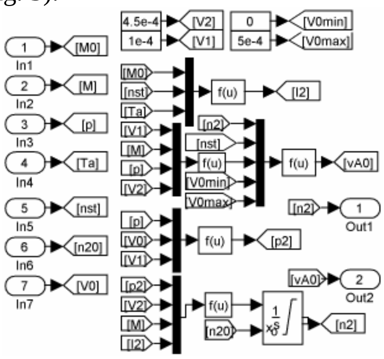

Fig. 4. The structure of subsystem "SYSTEM A0"

Also, in the S-model developed, the common regulator block "REGULATOR" simulating the control of all displacements of hydraulic machines has been used. In the S-model developed for the monitoring of the results of these tests the block "VISUALIZATION" has been assigned.

Results of Experimental Test

Using the developed S-models, we have carried out the tests of the hydraulic drives under consideration (Fig. 1 and 2). For the tests of the systems under consideration the external actions and structural parameters common for all schemes have been selected: $p = 2{\cdot}10^6(1{+}0.1\sin 0.1t)$ Pa, $M_0{=}3{\cdot}10^3$ N·m, $M_t{=}M_0(1{+}0.6\cos ft)$ N·m, $f = (0.01;\ 0.1;\ 1.0)$ Hz, $T_a = (0.1;\ 1.0;\ 10.0;\ 100.0)$ s, $n_{st} = 10$ Hz, $\Delta n_2 = 0.5$ Hz, $V_2 = 4.5{\cdot}10^{-4}$ m^3 (excluding the scheme "SINGLE MOTOR" with the direct control of the hydraulic motor displacement).

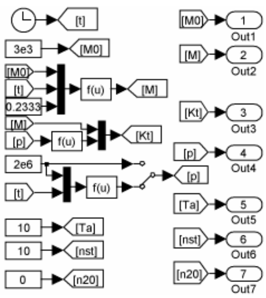

Fig. 5. The structure of subsystem "ACTIONS"

The speed of the displacement control of the hydraulic machines s_v simulated by the block "REGULATOR" was chosen in the course of the testing.

The following structural parameters of the hydraulic machines have been selected by assessing the conditions for the stabilization of the rotation speed of the hydraulic motor 3: for system A0: $V_1 = 10^{-4}$ m^3, $V_{0\min} = 0$, $V_{0\max} = 5.4{\cdot}10^{-4}$ m^3; for system A1: $V_0 = 10^{-4}$ m^3, $V_{1\min} = 4{\cdot}10^{-6}$ m^3, $V_{1\max} = 4{\cdot}10^{-4}$ m^3; for system B0: $V_1 = 10^{-4}$ m^3, $V_{0\min} = 0$, $V_{0\max} = 4{\cdot}10^{-5}$ m^3; for system B1: $V_0 = 10^{-4}$ m^3, $V_{1\min} = 1.1{\cdot}10^{-4}$ m^3, $V_{0\max} = 5{\cdot}10^{-4}$ m^3; for hydraulic motor 3 $V_{2\min} = 0$, $V_{2\max} = 4{\cdot}10^{-3}$ m^3.

The stability of the hydro motor rotor depends on transmission type. In Fig. 6 there is shown rotor rotation diagrams in case when regulation velocity of displacement $S_V = 3{\cdot}10^{-4}$, and external impacts p (Pa) and M (N·m) as shown in 1st diagram of Fig. 6.

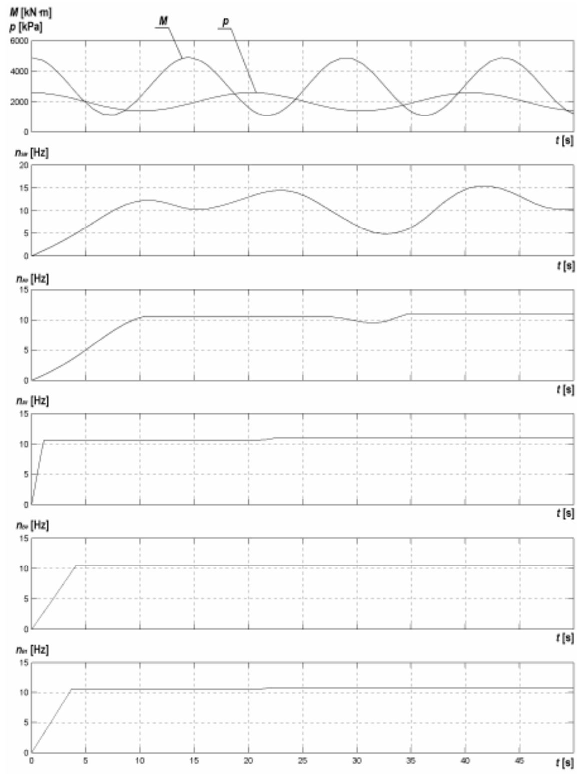

Fig. 6. Diagrams of hydro volume transmission tests results. M – resistance torque [kN·m], p – oil pressure in transmission input [kPa], n_{SM}, n_{A0}, n_{A1}, n_{B0}, n_{B1} – rotation of hydro motor [Hz]

Table 1. The summary of the test results

Coefficient T_a, [s] of I_2	M alter frequency [Hz]	Minimal required regulation velocity $S_{V\min}$ [dm^3/s] of displacement			
		SINGLE MOTOR	SYSTEM A0	SYSTEM B0	SYSTEM A1 and SYSTEM B1
0.1	0.01	$3.5{\cdot}10^{-2}$	$2.0{\cdot}10^{-2}$	$3.8{\cdot}10^{-3}$	$3.2{\cdot}10^{-2}$
	0.1	$1.9{\cdot}10^{-1}$	$4.5{\cdot}10^{-2}$	$7.9{\cdot}10^{-3}$	$7.0{\cdot}10^{-2}$
	1	1.2	$2.9{\cdot}10^{-1}$	$4.3{\cdot}10^{-2}$	$4.1{\cdot}10^{-1}$
1	0.01	$3.5{\cdot}10^{-2}$	$1.9{\cdot}10^{-2}$	$3.7{\cdot}10^{-3}$	$3.0{\cdot}10^{-2}$
	0.1	$1.9{\cdot}10^{-1}$	$4.3{\cdot}10^{-2}$	$7.1{\cdot}10^{-3}$	$6.7{\cdot}10^{-2}$
	1	1.1	$2.8{\cdot}10^{-1}$	$4.2{\cdot}10^{-2}$	$3.9{\cdot}10^{-1}$
10	0.01	$3.4{\cdot}10^{-2}$	$1.7{\cdot}10^{-2}$	$3.5{\cdot}10^{-3}$	$2.7{\cdot}10^{-2}$
	0.1	$1.8{\cdot}10^{-1}$	$4.0{\cdot}10^{-2}$	$6.9{\cdot}10^{-3}$	$6.2{\cdot}10^{-2}$
	1	1.0	$2.6{\cdot}10^{-1}$	$4.0{\cdot}10^{-2}$	$3.7{\cdot}10^{-2}$
100	0,01	$3.3{\cdot}10^{-2}$	$1.6{\cdot}10^{-2}$	$3.2{\cdot}10^{-3}$	$2.4{\cdot}10^{-2}$
	0,1	$1.8{\cdot}10^{-1}$	$3.8{\cdot}10^{-2}$	$6.6{\cdot}10^{-3}$	$5.7{\cdot}10^{-2}$
	1	1.0	$2.5{\cdot}10^{-1}$	$3.8{\cdot}10^{-2}$	$3.5{\cdot}10^{-1}$

How we can see from diagrams in Fig. 6 picture, the rotation speed of hydro motor shaft is stable only in systems A1, B0 and B1. Also we notice, that there is dependency between transmission type and time $T_{0,nst}$, in which shaft rotation speed changes from rest speed ($n_2 = n_0$) to stabilization speed ($n_2 = n_{st}$).

The conditions under which the stable speed of rotation of the hydraulic motor in the drives under consideration is ensured were investigated by carrying out the tests. The summary of the test results is provided in Table 1. The test results allow to state that in all drives the minimum required speed of the displacement control of the hydraulic machine $S_{V\min}$:

- practically does not depend on the mass moment of inertia of the rotor of hydraulic motor (coefficient T_a of I_2);
- depends on the change rate of the resistive torque M of a hydraulic motor. With the increase of this rate the control speed $S_{V\min}$ should be higher;
- under the same external actions the least $S_{V\min}$ is obtained in a drive with the direct control of a hydraulic motor.

Also the specific speed of the drives under consideration has been determined. The specific speed of the drives is estimated by the value

$$K_g = \frac{T_{0,nst}}{T_a}, \tag{10}$$

where $T_{0,nst}$ – the values(s) of the acceleration of the hydraulic motor rotor, loaded with resistive torque $M_0 = 3 \cdot 10^3$ N·m from the idle position (n_2 =0) to the stabilization rotation speed ($n_2 = n_{st}$), T_a – the value(s) of the acceleration of the hydraulic motor rotor in the system "SINGLE MOTOR". Results of these tests are summarized in Table 2.

Table 2. Results of speed tests

System Type	Single Motor	System A0	System B0	System A1	System B1
K_g	1	1.2	0.45	0.14	0.42

Conclusions

1. Various drives allowing to retain a stable speed of various processing equipment can be developed by using hydraulic machines with a variable displacement. The stable rotation speed of the hydraulic motor rotor is achieved when the displacement of one hydraulic machine in the drive is controlled at the minimum by the individual drive structure speed. In the drive comprised of one hydraulic motor the maximum control speed of its displacement is required and it must be higher than 1 dm^3/s.
2. For the stabilization of speed in a hydraulic drive with a constant displacement hydraulic motor the displacement control speed can be decreased down to $4 \cdot 10^{-2}$ dm^3/s by using a positive displacement flow divider where the displacement of one section is regulated.
3. In a hydraulic drive having two controlled flow divider sections, the specific drive speed can be increased up to 7 times.

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A. Lapinskas, A. Kirka, R. Lapinskas. Analysis of Hydro Motor Velocity Stabilization // Electronics and Electrical Engineering. – Kaunas: Technologija, 2008. – No. 1(81). – P. 57–60.

Various means such as throttles, flow regulators, variable displacement hydraulic machines, etc. are used for the flow regulation and control in the complex hydraulic systems. One of the functions, which often fall on a hydraulic drive, is speed stabilization of operating parts at a variable load. For this purpose flow regulators or hydraulic drives with variable displacement pumps and hydraulic motors may be used. This study deals with a task concerning the regulatory control of the hydraulic motor speed by using a positive displacement flow divider with variable displacement sections. Two different ways used for the connection of the flow divider into the hydraulic system and the possible methods for the control of the variable displacement of its sections have been considered and the control algorithms and operating factors have been developed. Ill. 6, bibl. 5 (in Lithuanian; summaries in English, Russian, Lithuanian).

А. Лапинскас, А. Кирка, Р. Лапинскас. Исследование стабилизации скорости вращения гидрообъемного мотора // Электроника и электротехника. – Каунас: Технология, 2008. – № 1(81) – С. 57–60.

Исследуется гидрообъемная передача, в которой на входе ее гидрообъемного мотора для стабилизации скорости его вращения включены две гидрообъемные машины. Проведен совместный анализ четырех схем включения и управления вспомогательных гидрообъемных машин в исследуемых гидрообъемных передачах. Исследования проведены в системе Simulink. Получены количественные оценки стабильности вращения гидрообъемных моторов и скорости их разгона в исследуемых гидрообъемных передачах. Ил. 6, библ. 5 (на литовском языке; рефераты на английском, русском и литовском яз.).

A. Lapinskas, A. Kirka, R. Lapinskas. Hidraulinio variklio sukimosi greičio stabilizavimo tyrimas // Elektronika ir elektrotechnika. – Kaunas: Technologija, 2008. – Nr. 1(81). – P. 57–60.

Tiriama hidrostatinė pavara, kurios hidraulinio variklio sukimosi greičiui stabilizuoti jo įėjime panaudotos dvi keičiamo darbinio tūrio hidraulinės mašinos. Tarpusavyje lyginamos keturios šių mašinų jungimo ir valdymo schemos. Šios pavaros lyginamos su pavara, sudaryta iš atskiro keičiamo darbinio tūrio hidraulinio variklio. Tyrimai atliekami imitaciniais bandymais Simulink sistemoje. Ištirti nagrinėjamų pavarų hidraulinio variklio sukimosi greičio stabilumo ir greičio pokyčių greitaeigiškumo kiekybiniai parametrai. Il. 6, bibl. 5 (lietuvių kalba; santraukos anglų, rusų, ir lietuvių k.).