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# Calculation of a three-stage electro-hydraulic amplifier

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**Annotation.**On this paper is analyzed the research state of three-stage electro-hydraulic amplifiers, the modeling features of such amplifiers are described. The parameters of the electro-hydraulic amplifier included in the servo hydraulic drive for the control system of the hovercraft are calculated. A mathematical model is compiled, the static and dynamic characteristics of the amplifier are obtained. Modeling is carried out in the MATLAB / Simulink software package. The amplifier was optimized for speed and dynamic transient error by varying the gain of the proportional controller in the direct circuit of the device.

## Introduction

The electro-hydraulic amplifier is an integral part of the servo hydraulic drive, it connects the executive and information subsystems, contains various mechanical, electrical, hydraulic, control devices and converts the input electrical signal into a hydraulic signal to control the load of any object. Electro-hydraulic amplifiers are widely used in various hydraulic systems [1]–[5]. For example, an electro-hydraulic amplifier plays an important role in controlling an hovercraft. Hovercraft can be used in various difficult terrestrial conditions [6], and with the development of such vessels, requirements both to the control systems themselves and directly to the characteristics of hydraulic drives used in the systems increase.

The three-stage electro-hydraulic amplifier has the advantages of high flow rate and speed. It has a small size, compact structure, high sensitivity, small input signal, large output power and high reliability [7]–[9]. Currently, many engineers are studying three-stage electro-hydraulic amplifiers. So, through modeling, it was found that the rigidity of the feedback rod and the area of the spool of the output stage have a significant impact on the dynamic characteristics of a three-stage electro-hydraulic amplifier. Reasonable rigidity of the feedback rod and correctly selected spool diameter can not only improve the performance of the amplifier, but also increase the stability of its operation. Under the same conditions, the rectangular feedback rod improves the dynamic characteristics of the amplifier to a greater extent [10]–[12]. Van Dongway developed a simulation model of a three-stage electro-hydraulic amplifier using the AMESIM software package and obtained graphs reflecting its characteristics, which confirmed the accuracy of the developed model. Are particularly interesting the studies of typical failure phenomena during obliteration of the spool throttle distributor of a three-stage electro-hydraulic amplifier, as well as in the presence of a large amount of air in the working fluid, which leads to a deterioration in the performance of the amplifier [13]. At the same time, the researchers analyzed the optimal design of the amplifier [14]–[16]. When analyzing the operation of the spool of a three-stage electro-hydraulic amplifier, factors that affect the spool performance, such as hydrodynamic force and friction force, are also taken into account.

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The three-stage electro-hydraulic amplifier consists of an electromechanical control signal converter, a hydraulic signal pre-amplification stage (nozzle-damper type device), a spool throttle valve of the first spool (second stage), a spool throttle valve of the second spool (third stage) and a spool position sensor. The combination of an electromechanical converter, a hydraulic signal pre-amplification stage and a first spool valve is essentially a two-stage electro-hydraulic amplifier. A schematic diagram of a three-stage electro-hydraulic amplifier is shown in Fig. 1, where the number 1 shows the electric signal amplifier; 2 — electromechanical converter; 3 — the first spool; 4 — second spool; 5 — spool position sensor (feedback sensor); 6 — two-stage electro-hydraulic amplifier.

A two-stage electro-hydraulic amplifier converts an input electrical signal into an output fluid flow to control a second spool. The second spool for positioning uses electric feedback on the position of the spool, in addition, the feedback coefficient is easy to adjust. The role of feedback is to determine the position of the second spool using a sensor, the output signal from which, after transmission and amplification, is converted into a feedback signal proportional to the position of the second spool and is directed to an electric signal amplifier, where it is subtracted from the input (control) electric signal.

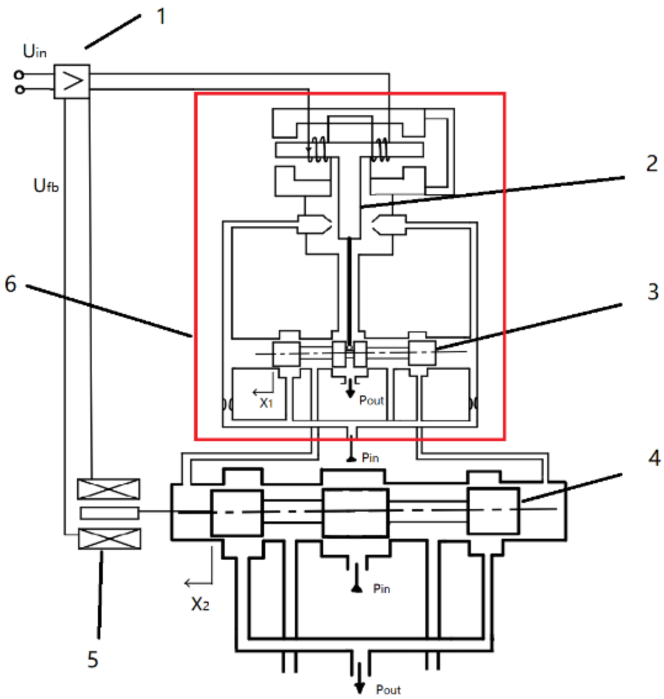


**Fig. 1.**Schematic diagram of a three-stage electro-hydraulic amplifier

**The mathematical model of an electro-hydraulic amplifier**

Two-stage electro-hydraulic amplifier is widely used at the present, its research results of are very developed, so in this paper it does not paid to it much attention. Here is discussed mainly about the calculation of the second spool and the characteristics of the entire three-stage electro-hydraulic amplifier [17]–[20].

Maximum flow at maximum displacement of the second spool and zero load

$$Q_z^{max} = \pi \cdot d_z \cdot k_p \cdot \mu_z \cdot x_z^{max} \cdot \sqrt{\frac{P_{in} - P_{out}}{\rho}}$$

Where $x_z^{max}$ — maximum spool offset relative to neutral; $d_z$ — spool diameter; $k_p$ — spool perimeter utilization factor.

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In the case of study, the flow rate through the spool valve at idle follow-up hydraulic actuator for the control system of the hovercraft is $Q_z^{max} = 5,8 \cdot 10^{-3} \dfrac{m^3}{s}$.

The spool perimeter utilization factor is taken equal to $k_p = 0,6$.

It's taken $x_z^{max} = 0,1 d_z$,

$$d_z = \sqrt{\frac{Q_z^{max}}{\pi \cdot k_p \cdot \mu_z \cdot 0,1 \cdot \sqrt{\dfrac{P_{in} - P_{out}}{\rho}}}} = \sqrt{\frac{5,8 \times 10^{-3}}{\pi \times 0,6 \times 0,65 \times 0,1 \times \sqrt{\dfrac{(32 - 0,3) \times 10^6}{850}}}} = 0,0157\, m,$$

Where $\rho$=850 $\dfrac{\text{kg}}{\text{m}^3}$— fluid density

And $d_z = 0,016\, m$,

Then the perimeter utilization factor of the second spool is

$$k_p = \frac{Q_z^{max}}{\pi \cdot d_z^2 \cdot \mu_z \cdot 0,1 \cdot \sqrt{\dfrac{P_{in} - P_{out}}{\rho}}} = \frac{5,8 \times 10^{-3}}{\pi \times 0,016^2 \times 0,65 \times 0,1 \times \sqrt{\dfrac{(32 - 0,3) \times 10^6}{850}}} = 0,5773$$

The maximum possible displacement of the second spool from the neutral position is assumed to be equal to

$$x_z^{max} = 0,1 \cdot d_z = 0,1 \times 0,016 = 0,0016\, m$$

The total width of the window casings

$$b_{ok\,\Sigma} = k_p \cdot \pi \cdot d_z = 0,5773 \times 3,14 \times 0,016 = 0,0290\, m.$$

Width of each window of the sleeve (there are 4 windows in the sleeve)

$$b_{okna} = \frac{b_{ok\,\Sigma}}{4} = \frac{0,0290}{4} = 0,00725\, m$$

*Then* $b_{okna} = 7,3\, mm.$

And $b_{ok\,\Sigma} = 4 \cdot b_{okna} = 4 \times 7,3 = 29,2\, mm.$

$$k_p = \frac{b_{ok\,\Sigma}}{\pi \cdot d_z} = \frac{29,2}{\pi \cdot 0,016} = 0,5809$$

Conductivity of the spool valve

$$k_z^{'} = \pi \cdot d_z \cdot k_p \cdot \mu_z \cdot \sqrt{\frac{2}{\rho}} = \pi \times 0,016 \times 0,5809 \times 0,65 \times \sqrt{\frac{2}{850}} = 9,2067 \cdot 10^{-4}$$

We specify the diameter of the spool

$$d_z = \sqrt{\frac{Q_z^{max}}{\pi \cdot k_p \cdot \mu_z \cdot 0,1 \cdot \sqrt{\dfrac{P_{in} - P_{out}}{\rho}}}} = \sqrt{\frac{5,8 \times 10^{-3}}{\pi \times 0,5809 \times 0,65 \times 0,0016 \times \sqrt{\dfrac{(32 - 0,3) \cdot 10^6}{850}}}} = 0,016\, m.$$

The spool diameter is finally taken equal to $d_z = 0,016\, m.$

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Since the second spool works by receiving an output hydraulic signal from a two-stage electro-hydraulic amplifier, it can be considered as a load on a two-stage electro-hydraulic amplifier. For the second spool, the input signal represents the flow rate of the first spool, and the output signal represents the offset of the second spool.

(1) Flow equation for controlling the second spool:

$$Q_y = F_z \frac{dx_z}{dt} + \frac{V_y}{2B_r}\frac{dP_y}{dt},$$

*Where* $F_z$ – second spool area $\left(m^2\right)$, $x_z$ – the displacement of the second spool $(m)$,

$B_r$ – volumetric elasticity modulus of the liquid $(Pa)$,

$V_y$ – volume for controlling the second spool $\left(m^3\right)$,

$p_y$ – differential pressure for controlling the second spool $(Pa)$.

(2) Motion equation of the second spool

$$m_z \frac{d{x_z}^2}{dt^2} + k_{tp}\frac{dx_z}{dt} + 2C_{gd}x_z = F_z P_y$$

$m_z$ – weight of the second spool $(kg)$, $k_{tp}$ – coefficient of friction,

$C_{gd}$ – the rigidity of the hydrodynamic spring.

The block diagram of the amplifier in MATLAB / Simulink in accordance with the equations described above is shown in Fig. 2.

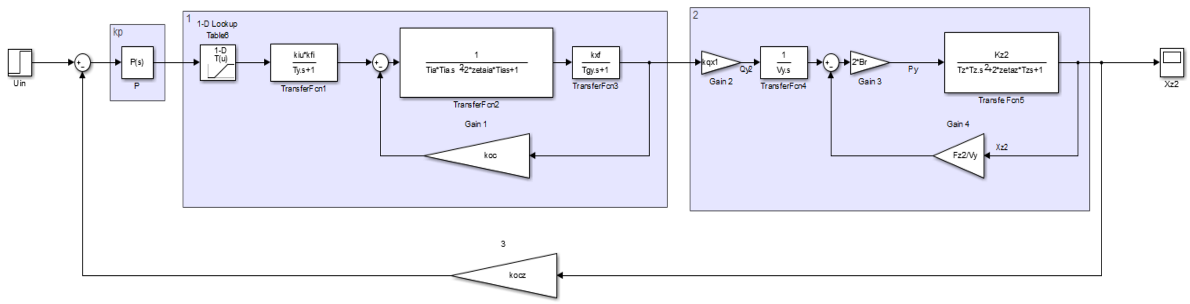


**Fig. 2.**Structural diagram of a three-stage electro-hydraulic amplifier: 1 — two-stage electro-hydraulic amplifier, 2 — second spool, 3 — feedback sensor.

The numerical values of the coefficients are accepted as follows: *$k_{iu}$=4e-3 ;$k_{fi}$=0.7750;Ty=4.8e-3; Tia=2.839e-4; zetaia=0.2676; Tgy=0.4502; $k_{xf}$=35.1613;koc=14.4351; $k_{qx1}$=0.1666; Vy=1.6085e-6; Br=1.4e9; Tz=3.0996e-4; zetaz=0.3; Kz2=1.865e-9; $F_{z2}$=8.0425e-4;kocz=1875.*

**Simulation results**

The simulation results are presented by the static and dynamic characteristics of the amplifier. Fig. 3 shows the static characteristic of the second spool (input electric current in the control winding of the electromechanical converter — moving the spool)

To calculate the flow-differential characteristics of the second spool valve in a three-stage electro-hydraulic amplifier, the following equation is used:

$$Q_z\left(p_h, x_{z2}\right) = k_z^{'} * x_{z2} * \sqrt{\frac{p_{in} - p_{out} - p_h * sign\left(x_{z2}\right)}{2}}.$$

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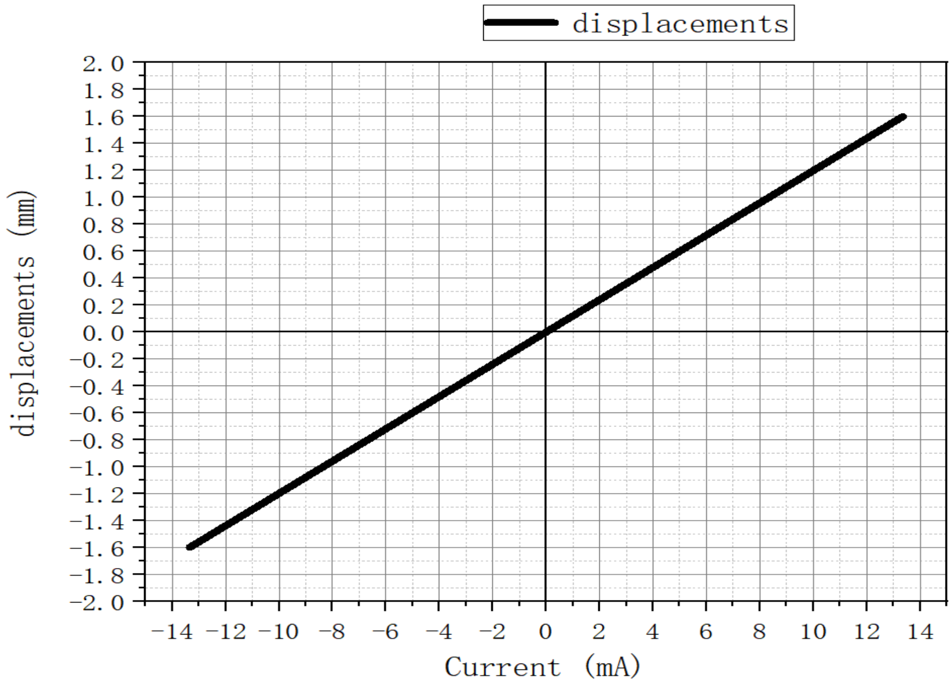


**Fig. 3.**Static characteristic of the second spool

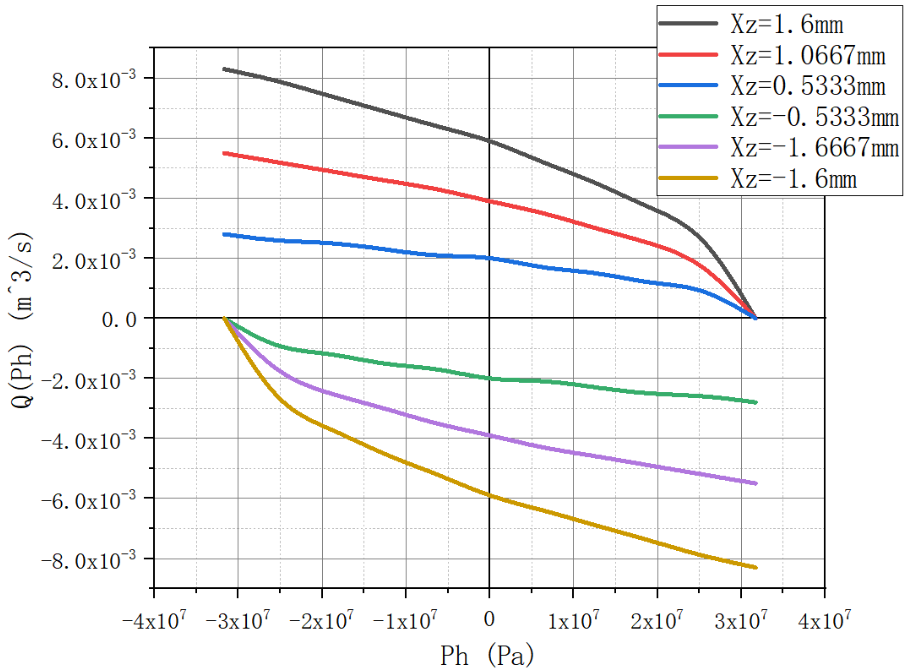


**Fig. 4.**Flow-rate characteristic of the second spool valve

Figure 4 shows the flow-rate characteristic of the second spool valve for various movements of the spool.

According to the above model, the transients are calculated according to the position of the second spool with an input signal of 3 V (Fig. 5), different results were obtained by setting different values of the proportional controller Kp ($Kp_1=0,9$; $Kp_2=0,75$; $Kp_3=0,5$).

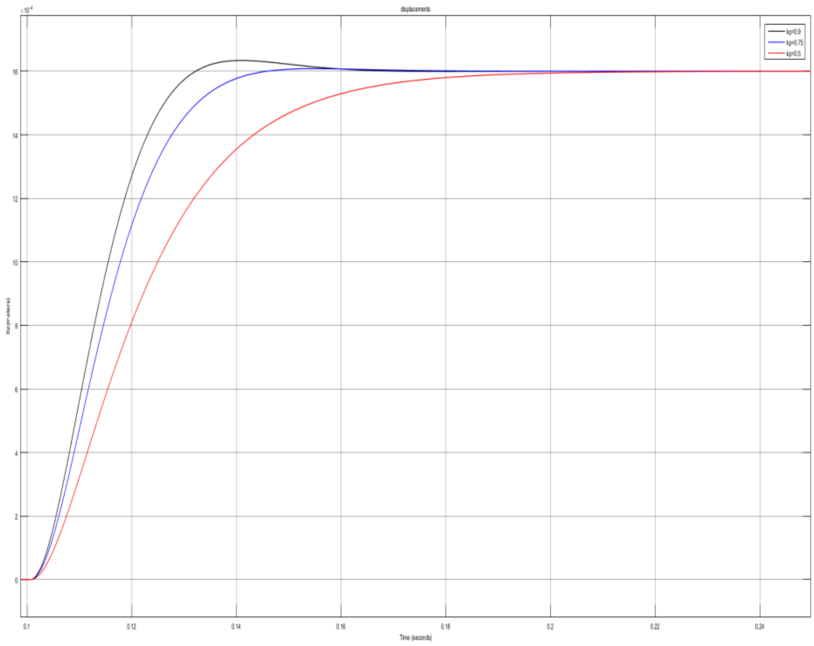

**Fig.5.**Transient process of the second spool

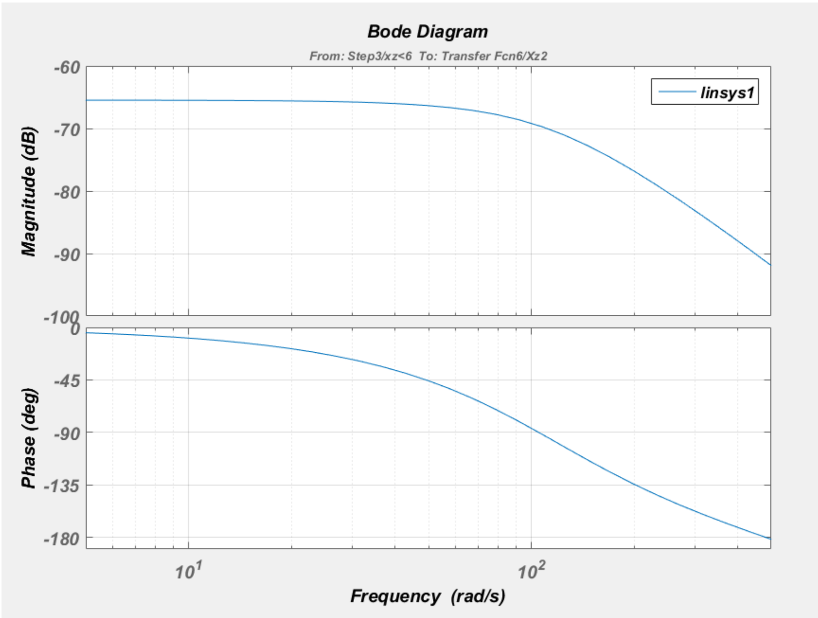


**Fig. 6.**Bode Chart

Figure 6 shows the calculated Bode diagram of the electro-hydraulic amplifier at $Kp$ = 0,75. The frequency characteristics of the amplifier turned out to be quite satisfactory.

When varying the gain of the proportional controller in the direct circuit of the device, the amplifier was optimized for the quality of the transition process. At Kp = 0.75, a fairly fast transition process was obtained, which has a small overshoot (less than 2%).

## Conclusions

In this work, the parameters of a three-stage electro-hydraulic amplifier included in the servo hydraulic drive for the control system of a hovercraft are calculated and a mathematical model of such an amplifier is created. Using the MATLAB / Simulink software package, modeling was performed, and the static and dynamic characteristics of the amplifier were obtained. The amplifier was optimized for speed and dynamic transient error by varying the proportional controller in the direct circuit of the device to obtain better system dynamics. It can be seen from the results that the mathematical model of the electro-hydraulic amplifier meets the requirements of the system and can be used for further studies of servo hydraulic drives.

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