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CALCULATION OF VOLUMETRIC HYDRAULIC DRIVES AND JUSTIFICATION OF THEIR DESIGN

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Abstract
A number of machines and equipment are now widely used in mining, construction and agriculture with hydraulic systems. In order to increase the stable and reliable operation of hydraulic systems of these machines, it is important to calculate their volumetric hydraulic drives and justify their design, this article considers the basics of calculating volumetric hydraulic drives and designing them.

Keywords: project, hydraulic drive, distribution device, static, dynamic, loading, salt performance, nominal.

Introduction

The design of hydraulic drives begins with the analysis of the initial data[1].
The following are used as preliminary data:
1) Complete characteristics of the load, consisting of static (M_{st}, F_{st}), inertial (Min, Fin) downloads and friction (M_{tr}, F_{tr}) loading. Static loading can be given as a function of time or as a function of movement when moving back and forth;
2) operating mode and operating conditions;
3) the number, type and nature of the movement of working mechanisms for controlling them with hydraulic motors, the interposition and interconnection of these mechanisms, the need to fix them in order or application, trussing highways;
4) in some cases, it is necessary to structure the moments in the hydraulic engine, which determine the speed of cycloramic loads on the hydraulic cylinder, the output joint or the speed of rotation of the Acting Member;

LITERATURE ANALYSIS AND METHODOLOGY

Before calculating and selecting hydraulic drive elements, a schematic diagram is drawn up. At the same time, at the same time as the hydraulic scheme, it is recommended to draw up a kinematic (and possibly electrical) scheme of the mechanisms of the designed product. The scheme is determined by the initial data, on the basis of which a decision is made on the circulation system of the working fluid, pumps, hydraulic motors, auxiliary hydropower equipment, their role in the scheme. As a rule, you should focus on mass-produced equipment[2,3,4].

If system regulation is necessary, it is important to choose the right method of regulation. Throttle control, as a rule, is used in hydraulic cylinder systems, volumetric – hydraulic motor systems. With successive launches of the throttle and hydraulic motor, throttle regulation is less cost-effective than parallel launches. With sequential throttle regulation at low loads in the hydraulic cylinder, parallel start-up regulation has a large limit at large loads. Volumetric regulation, as a rule, is carried out only by changing the working volume of the pump[5,6,7,8].

If the speed of the mechanism in the mode of operation is significantly greater than the operating speed (of the order or more), then an additional pump should be included in the hydraulic circuit. In this case, it is necessary to ensure that this additional pump is turned off from the circuit during the main operation of the hydraulic system. This is often done by draining the fluid into the hydraulic tank and moving the pump to idle. Similarly, it should be done with short-term stoppage of hydraulic motors [9,10].

Hydraulic accumulators are required with frequent short-term restarts of hydraulic motors in cycles.

A mandatory element of volumetric hydraulic systems is the maintenance valves. As a rule, they are installed behind the pump, sometimes it is included in the pump structure. During the sequential adjustment of the throttle and the hydraulic motor, in the circuits with throttle control, as well as in the suction pumps, instead of the check valve, excess (pressure) valves are installed and used.

In a circuit with different pressures and several hydraulic motors working with one pump, pressure-reducing valves must be installed and flow dividers must be installed to synchronize the movement of the output joints. According to the necessary requirements, filters are placed in the hydraulic system.

The maximum value of the calculated load is determined by taking into account all available forces R and moments M[11,12]:

$$M_{\text{max}} = M_{\text{st max}} + M_{\text{tr}} + M_{\text{in max}};$$

$$R_{\text{max}} = R_{\text{st max}} + R_{\text{tr}} + R_{\text{in max}}.$$

Inertial loads are found by the following relations:

$$M_{\text{in}} = I\frac{d^2\varphi}{dt^2}; \qquad R_{\text{in}} = m\frac{d^2 s}{dt^2}.$$

If there is no data on the maximum speed and accelerations, then the calculated load is determined by increasing the static load, taking into account the reserve factor:

$$M_{\max} = K_{zn}\, M_{\text{st max}}; \qquad R_{\max} = K_{zn}\, R_{\text{st max}}.$$

The working pressure in the hydraulic system is selected from the standard pressure series for hydraulic systems:

2,5; 6,3; 10; 16; 20; 25; 32; 40; 50; 63 MPa.

This pressure is determined, on the one hand, by the load and working volume of the hydraulic motor, and on the other hand, by the characteristics of the pump. When choosing the pressure, it is necessary to follow the technical characteristics of mass-produced hydraulic machines and devices, which are given in the technical description of the manufactured hydraulic drive. Pressure values of 20, 25 and 32 MPa are recommended for hydraulic operation of high-powered machines and mechanisms, and pressure values of 6.3 and 10 MPa are recommended for operation of auxiliary equipment.

At high pressures, the system becomes compact, but in this case, the requirements for the production quality of the hydraulic control elements increase.

Taking into account the losses in the pipes, the value of the working pressure may be slightly lower than the nominal pressure in the pump [13,14]:

$$p_{ish} = (0{,}85 \div 0{,}9)p_{nn}\,.$$

The pressure in the pouring pipes can be up to p_k = 0.3÷0.6 MPa.

RESULTS

If a hydraulic cylinder is chosen as a hydraulic motor, then the given force P_d and the pressure assumed from the above equation are:

$$D = 1{,}13\sqrt{\frac{P_d}{\eta_d\; p_{ish}}}$$

The diameter of the piston is D, according to the equation (2.9) - the diameter of the rod is determined:

$$\frac{d}{D} = \sqrt{1 - \frac{\upsilon_1}{\upsilon_2}}\,.$$

The calculated diameters are rounded up to the standard, and the really required pressure is determined by reverse calculation according to equation:

$$P_d = p\frac{\pi}{4}\left(D^2 - d^2\right).$$

The amount of pressure in the working space of the hydraulic motor p_{nagr} and in the counter-pressure space rpr depends on the method of regulating the hydraulic drive. When installing the throttle on the supply pipe of the hydraulic motor, p_{nagr}=2/3 pnn, because in this case the efficiency of the useful work coefficient of the hydraulic drive is maximum. In all other cases,

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webofjournals.com/index.php/4

we can take $p_{hay} = p_{ish}$. For the same reasons, p_{pr}=1/3 is chosen equal to p_{ish} when installing a throttle on the pipeline, in other cases, you can take the value $p_{pr} = p_{sl}$, or p_{pr}=0.5 when there is an overpressure valve at the exit of the hydraulic system You can choose a value of –1.0 MPa. For a given stock speed and standard dimensions, the equation determines the consumption of a hydraulic cylinder:

$$v = \frac{Q\eta_0}{F}$$

If a hydraulic motor is used as a hydraulic motor, then according to the given torque M and the assumed pressure p_d, the value of the working volume q_{dn} is found:

$$p = \frac{2\pi M}{q_{dn}}.$$

The resulting working volume is rounded up according to standards or pd i qdn is selected according to the technical specifications of the mass-produced hydraulic motor. Then, according to the given frequency of rotation, the consumption amount of the hydraulic motor is determined:

$$Q_{dqs} = q_{dn}\, n_d\ .$$

As a rule, radial-piston hydraulic motors are recommended with a large given torque (M_d >200 N•m) and a small rotation frequency of the shaft (n_d < 200 min –1). Axial-piston hydraulic motors with smaller torques and variable shaft rotation frequency (n_d = 20÷1800 min−1) are recommended. It is recommended to use plate and pinion hydraulic motors at a rotation speed of at least 300 min–1 and for a relatively short time (M_d < 200 N•m).

The pump is selected depending on the required supply transfer and the accepted value of the nominal pressure.

The delivery of the pump must be determined in terms of providing the required speed of the output connection of the hydraulic system. We remind you that the speed for different types of hydraulic motors is determined as follows:

For a hydromotor - rotation frequency n_m;

- for power hydraulic cylinders - displacement speed:

$$v = \frac{s}{t}$$

- for a rotary hydraulic motor (torque hydraulic cylinder):

$$w = \frac{\varphi}{t} \cdot \frac{2\pi}{360}$$

Maintenance cost of hydraulic motors:

- for hydraulic motor:

$$Q_{dsq} = q_{dn}\, n_m$$

- for power hydraulic cylinders:

$$Q_{dsq} = F\, \upsilon$$

- for a rotary hydraulic motor (torque hydraulic cylinder):

$$Q_{dsq} = \frac{\omega}{2\pi} \cdot q_0 z$$

Taking into account volumetric losses in the hydraulic motor and pump, the required flow rate in the hydraulic system is found by the following:

$$Q_{tiz} = \frac{Q_{dsq}}{\eta_{0.N}\eta_{0.D}}$$

Taking into account the losses in the hydraulic pipe, the pump pressure is equal to p_n=(1.05÷1.1)p_d.

Based on Q_{tiz} and p_n value indicators, the pump is selected and its description is determined: nominal efficiency of the system $Q_n > Q_{tiz}$, nominal working volume q_{nn}, nominal rotation frequency n_{nn}, volumetric efficiency of the pump and full useful work coefficient η_n. The installed power of the selected drive motor is:

$$N = \frac{Q_n \ p_{nn}}{\eta_n}.$$

If high transmission and pressure of the pump is required, there is no need to regulate it, and there are no restrictions on the size of the machine, it is recommended to use eccentric piston pumps. If the regulation of delivery meets the specified requirements, then it is advisable to use radial-piston pumps. Axial-piston pumps are more compact than radial-piston pumps, create high pressure (up to 32 MPa), but require better oil cleaning. Plate and gear pumps produce smaller displacements and pressures than piston pumps. Because gear pumps are not regulated, they are often used in auxiliary operations.

DISCUSSION

If the scheme uses several hydraulic motors that do not work at the same time, then before choosing the pumps, it is necessary to make a histogram of costs, determine the number of pumps working in parallel and determine the need to use accumulators. Then the power of the pump is determined and the drive motor is selected.

Hydraulic calculation includes a check description, its purpose is to determine the pressure losses in the main pipelines and compare them with the accepted pressure limit:

$$\Delta p_{zax} = p_{nn} - p_{ish} .$$

Pressure losses are calculated separately for pressure, discharge and suction pipes and consist of friction losses and local losses. Pressure loss in pipes is calculated by Darcy-Weissbach formula.

$$\Delta h_{yo`q} = \lambda_{ishq} \frac{l}{d} \frac{v^2}{2g}.$$

CONCLUSION

When calculating losses due to local resistance, only the hydraulic resistance of hydraulic devices is taken into account. The total hydraulic losses in the pressure and discharge pipelines should not exceed the pressure reserve Dpzakhira of other loads. On the contrary, it is necessary to correct the calculation of the operating parameters of the hydraulic motor and pump.

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In addition, in order to determine the need to use heat exchangers, it is necessary to perform a thermal calculation of hydraulic processes, as well as to check the pump under cavitation-free operating conditions.

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