High-torque axial-plunger hydraulic motor with phase control of the shaft rotation speed

A I Nizhegorodov¹⁺, A N Gavrilin² and B B Moyzes³

¹Department of Construction and Road-Making Machines and Hydraulic Systems, Irkutsk National Research Technical University, Irkutsk, 664074, Russia
²Division for Materials Science, National Research Tomsk Polytechnic University, Tomsk, 634050, Russia
³Division for Testing and Diagnostics National Research Tomsk Polytechnic University, Tomsk, 634050, Russia

*nastromo_irkutsk@mail.ru

Abstract. The article reviews the design and parameters of high-torque axial-plunger hydraulic motor with phase control of the shaft rotation speed. This hydraulic motor is distinguished by a two-row eighteen-cylinder block and a rotary distribution disk with two rows of sickle-shaped distribution windows for supplying and discharging the working fluid. The hydraulic motor differs by a tripled displacement and torque from a similar hydraulic machine, which classifies it a high-torque hydraulic engines. Its power volume increases 1.6 times, and power response more than four times. At the same time, the fluctuation of the shaft rotation speed is measured by the corresponding value of the fluctuation coefficient equal to 0.0154. The phase control provides the hydromotor with a control range of 150...200 and more, while hydraulic motors with variable working volume range maximum 5. Therefore, the new hydraulic machine has a high potential for operating in hydraulic drives of machines.

1. Introduction

The hydraulic drives for machinery and equipment become more widely used in different industries. Many designs of rotary hydraulic machines have been developed, but research in this field continues throughout the world [1-4].

Among two- and multi-row rotary hydraulic machines, only radial-piston hydraulic motors and pumps [5, 6] were further developed. They are equipped with two or more sets of working chambers and they are mainly used as a high-torque hydraulic motors [7-9].

The present work reviews the design and parameters of high-torque axial-plunger hydraulic motor with phase control of the shaft rotation speed, which is distinguished by a two-row cylinder block, a significantly higher volume capacity, increased specific power, increased torque, increased power response and a large control range.

The considered hydraulic machine with phase control remains reversible; it can operate as a pump and be used in hydraulic drives of all types of machines and equipment.

The aim of the research is to develop a high-torque axial-plunger hydraulic machines with phase control of the flow and speed of shaft rotation, which provides a wide range of these parameters control.
2. The axial-plunger hydraulic motor design

Figure 1 shows a long cross section and a view A of an axial-plunger hydraulic motor, which includes a housing 1, a drive shaft 2 mounted in bearings 3 and connected by a splined joint 4 to the cylinder block 5, which left spherical flank is attached to the spherical surface of the distribution disc 6 (hereinafter distributor) by spring 7 through the ring in the cylinder block. The spherical conjugation of these surfaces allows the unit 5 to self-adjust during rotation. The distributor is an integral with the shaft 8, which is capable of being rotated by an angle $\varphi$ from 0 to 90° in thrust 9 and needle 3 bearings. The seal 10 provides hermiticity of the distributor and the motor housing conjugation, and fittings 11 – supply and discharge of the working fluid.

The bearing 9 in the end wall 12, which is tightened to the body by bolts 13, takes the pressure of the block 5 to the distributor. The pressure provides the tightness of the connection .

There are 14 main and 15 additional cylinders in block 5. The channels of the cylinders go to its end of the block. The cylinders are connected with sickle-shaped windows (Figure 2). Plungers 16 and 17 with hydrostatic bearings (thrust bearings) 18 and 19 are placed in the block cylinders. The plungers are designed to reduce the contact frictions between the plungers and the inclined swash plate 20. There are valve channels connected by the fittings 21 through the tubes 22 with the fittings 22 for supplying and draining the service fluid in the distributor 6.

Windows 1 and 3 – are used to supply the service fluid to the hydraulic motor from the pump of the hydraulic system. The windows are connected with the hole 5 of the supply pipe 11 (Figure 1). Windows 2 and 4 are used for discharge of the fluid in the hydraulic system tank and these windows are connected with the hole 6 of the drain pipe 11 (Figure 1). Figure 2 a shows the distributor 6 located on the side of the cylinder block 5 (Figure 1) and enlarged fragments of the valve channels (Figure 2 b, section I-I). Sickle-shaped windows 1 and 2, corresponding to the main block cylinders, and sickle-shaped windows 3 and 4 for additional cylinders are cut in it.

Between themselves, sickle -shaped windows are connected by slots 7. Sickle -shaped windows, like in other axial-piston hydraulic machines [6, 7], are separated by jumpers 8 and 9, in which outlets 10 are made. The outlets connect to valve distributor channels, which are shown in figure 2 b. There are check valves 12 in valve channels 11 of upper jumpers. They allow the service fluid to flow towards the main and additional block cylinders 5 (Figure 1). They are connected to a common channel 13, which leads to the supply fitting 21 (Figure 1). There are check valves 15 in the channels 14 of the lower jumpers. They allow the service fluid to flow from the main and additional block cylinders 5 through the common channel 16 to the drain fitting 21 (Figure 1).

![Figure 1. The axial-plunger hydraulic motor: a – view A; b – longitudinal section.](image-url)
Figure 2. Distribution disk: $a$ – channel scheme; $b$ – long cross section with check valves.

Figure 3 shows a two-row cylinder block of a hydraulic machine. A typical seven-cylinder block from a series 310 machine [10] with an axial diameter of cylinders $D_1=0.07$ m, an external diameter $D_3=0.11$ m with a diameter and cylinder area $d=0.025$ m and $s=0.5 \cdot 10^{-3}$ m$^2$ was used as the basis. But in our case, because of the phase control of the rotational speed, which will be discussed below, nine cylinders are used.

For a meaningful comparison of the indicators, we take the condition that the total area of the nine cylinders of the hydraulic machine is equal to the area of all the cylinders of the analogue. It follows that the diameter of the cylinder block (Figure 3) should be less and equal to (m):

$$d = \sqrt{\frac{7 \cdot 0.025^2}{9}} = 0.02205.$$  \hspace{1cm} (1)
To fulfill parts commonality of the pumping unit, the diameters of the cylinders of the first 1 and second 2 rows (Figure 3) are the same (0.022 mm).

Calculate the plunger stroke according to the following formula [6]:

\[ h = D \cdot \tan \gamma_{\text{max}}. \]  

(2)

Calculate the plungers travel \( h_1 \) of the first row (2) (m):

\[ h_1 = D_1 \cdot \tan \gamma_{\text{max}} = 0.07 \cdot 0.466 = 0.0327, \]  

(3)

where \( D_1 \) is the diameter along the axes of the cylinders, 0.07 mm; \( \gamma \) is the tilt angle of the washer, 25°.

Working volume of a single-row hydraulic machine with nine cylinders (m³):

\[ V_0 = 9 \cdot h_1 \cdot s_1 = 9 \cdot 0.0327 \cdot 0.00038 = 0.111834 \times 10^{-3}, \]  

(4)

where \( s_1 \) is the area of one cylinder with a diameter of 0.022 m.

The diameter along the axes of the cylinders of the second row is \( D_2 = 0.107 \) m (the size is obtained from the proportions according to the diagram in Figure 3), and the plunger travel \( h_2 \) is determined by the expression (2) (m):

\[ H_2 = D_2 \cdot \tan \gamma_{\text{max}} = 0.107 \cdot 0.466 = 0.0499. \]  

(5)

Since the diameters of all cylinders are equal and their number is equal too, the volume capacity of the new axial-plunger hydraulic machine (hydraulic motor) is determined (m³):

\[ V_{02} = \frac{2 \cdot h_2 \cdot V_0}{h_1} = \frac{2 \cdot 0.0499 \cdot 0.000111834}{0.0327} = 0.00033997536, \]  

(6)

which is 3.04 times more.

The outer diameter \( D_4 \) is equal to 0.150 m (the size is obtained from the proportions according to the scheme in Figure 3), which is only 1.36 times more than the outer diameter of the seven-cylinder block \( D_3 \) (0.11 m), taken as a analogue.

3. Performance analysis of axial piston hydraulic machines

Let’s see how the specific power \( N/m \) of the axial-plunger hydraulic machine under consideration will change. Define the masses of the double row \( m_1 \) and single row \( m_2 \) blocks, taking into account that the drive shafts of the compared machines are the same:

\[ m_1 = 1.05 \cdot \rho \cdot 0.25 \cdot \pi \cdot D_1^2 \cdot l, \]  

(7)

\[ m_2 = 1.025 \cdot \rho \cdot 0.25 \cdot \pi \cdot D_2^2 \cdot l, \]  

(8)

where 1.05 and 1.025 are the coefficients taking into account the masses of hydrostatic supports of additional plungers, \( \rho \) is the steel density, \( l \) is the length of the cylinder block.

Substituting the equations (7, 8) into the equations of the power volume, we obtain the relation [11]:

\[ \frac{N_1}{m_1} = \frac{V_{02} n}{1.05 \rho \cdot 0.25 \cdot \pi \cdot D_2^2 \cdot l} = \frac{V_{02}}{0.105 \cdot D_1^2} = \frac{0.00033997536}{0.15 \cdot 0.15^2} = 1.6. \]  

(9)

Here \( N_1/m_1, N_2/m_2 \) are the power volume of the new hydraulic machine and single-row hydraulic machine, W/kg.

As can be seen from the equation (9), the power volume of the high-torque hydraulic motor increases by approximately 60%. Approximately, because the result is obtained for the power volume of the cylinder block, and not the entire machine as a whole.
Torque on the shaft without friction losses is determined by the formula [7] (N·m):

\[ M = \frac{V_o \Delta P}{2\pi}, \]  

(10)

where \( \Delta P \) is the pressure difference at the inlet and outlet of the hydraulic machine, N/m².

The ratio of moments is equal to the ratio of working volumes:

\[ \frac{M_2}{M_1} = \frac{V_o}{V_0} = 3.04, \]  

(11)

therefore, the hydraulic motor under consideration, which has three times higher torque, can be categorized as high-torque hydraulic machines.

Another important property of hydromotors is power response, which characterizes the ability to accelerate the inertial loads, which is estimated by the coefficient \( k [11] \):

\[ k = \frac{M}{\sqrt{J}} = \frac{M}{\sqrt{0.5m r^2}}, \]  

(12)

where \( J \) is the moment of inertia of the cylinder block and all its rotating masses, kg·m², \( m \) is the mass of the block with plungers, kg, \( r \) is the outer radius of the block, m.

To estimate the power response of the new hydraulic motor, we correlate the coefficients for the modified \( k_2 \) and the initial \( k_1 \) of cylinder blocks:

\[ \frac{k_2}{k_1} = \frac{3.04 \cdot M}{\sqrt{0.5m \cdot (1.36r)^2}} \times \frac{M}{\sqrt{0.5m \cdot r^2}} = 4.13, \]  

(13)

where 3.04 is the coefficient for increasing the torque, 1.36 is the coefficient for increasing the external radius of the modified cylinders block.

Let's see how the pulsation of the service fluid supply changes when passing through the hydraulic motor and the angular velocity of rotation of its shaft.

The pulsation of the supply of rotary hydraulic machines is checked by the coefficient of uneven feed \( \sigma_Q \), determined with an odd number of plungers according to the formula [6]:

\[ \sigma_Q = 2 \cdot \tan(0.25 \cdot \pi / z), \]  

(14)

where \( z \) is the number of cylinders. For single-row nine-cylinder hydraulic machines, it is equal to 0.0154.

In the considered two-row hydraulic motor with eighteen cylinders, the coefficient of unevenness is determined by a different formula [6]:

\[ \sigma_Q = 2 \cdot \tan(0.5 \cdot \pi / z), \]  

(15)

which gives the same value 0.0154, since for hydraulic machines with an even number of cylinders, \( \sigma_Q \) doubles. This is considered to be a good indicator [6].

4. Phase control of shaft speed

Volumetric regulation of axial-plunger hydromotors is carried out by changing the angle of the washer \( \gamma \) (Figure 1). In this case, the angular velocity \( n \) of its shaft without taking into account leaks is determined by the formula [6]:

\[ n = \frac{Q}{V_0}, \]  

(16)

where \( Q \) is the flow rate of the working fluid passing through the working chambers of the machine, m³/s.
With a constant pressure drop $\Delta P = \text{const}$ and a decrease in the volume capacity $V_0$ from a conventional unit to zero, the shaft rotation speed $n$ will increase according to a hyperbolic law (Figure 4, section 1 and 2). At speed $n$, which corresponds to point a, the internal friction torque becomes equal to the torque, the hydraulic motor starts to work unstable (section 3), and then stops (point b). Because of that, the range of regulation $D$, describing the change in the speed of the shaft rotation is limited: $D < 5$ [6].

![Figure 4. Graphs shows the regulation of the speed of rotation of the hydromotor shaft with volumetric and phase methods of controlling the volume capacity](image)

When rotation of the distributor disk 6 is phase regulated (Figure 1), the graph of the rotational speed of the hydromotor shaft becomes a straight line 4 (Figure 4).

The process of phase regulation of a new reversible hydraulic machine will be considered on the example of the pump mode operation. Figure 5 shows a diagram of the overlay of the contours of the sickle-shaped windows of the distributor on the vents in the channels in the block. Let the cylinder block rotate with an angular velocity $n$ counterclockwise, and the distribution disc rotates clockwise by an angle $\varphi$ from 0 to 90°.

When $\varphi = 0$, the distributor is in the position shown in figure 2 a, when its jumpers are located vertically. In this case (Figure 5), the plunger of cylinder 1 is at the bottom dead center (b.d.s.), and the plunger of cylinder 2 is at the upper dead center (u.d.s.), and their speed is zero, so both are in reverse (both cylinders are highlighted in black).

Cylinders 3, highlighted in white shown in figure 5 and numbered from 1 to 6, as well as with letters c and d, suck in the service fluid and fill the chambers through windows 4 and 5. They are connected with the suction pipe through the vent, which is shown in figure 5 by black dot. Cylinders 6, highlighted with gray, marked with letters a and b and numbered from 1 to 6, displace service fluid with plungers through windows 7 and 8, which are connected to the discharge pipe through the vent shown by another black dot.

In this initial state, the hydraulic machine operates in the maximum feed mode and all its process chambers are filled and emptied completely.

With the distributor turned at an angle $\varphi$ (Figure 5), only six «white» cylinders (1...6) suck in new fluid through windows 4 and 5 from the hydraulic system tank, and cylinders c and d suck it from windows 7 and 8. Therefore, one turn of the block, six of the eight cylinders will fill their process chambers with a new liquid, which is 75% of the full volume. And only six of the eight «gray» cylinders...
displace the liquid through the windows 7 and 8 into the pressure line, and the cylinders \( a \) and \( b \) displace it into the sickle-shaped windows 4 and 5, through which the working chambers are filled. This means that 75% of the maximum possible volume will be displaced.

**Figure 5.** Conventional scheme of imposing contours of sickle-shaped windows on the holes of the channels of the block

Plungers in cylinders \( c \) and \( d \), which suck fluid in from windows 7 and 8, under pressure, moving from the upper dead center to the bottom dead center, will reduce the torque on the pump shaft. And the plungers in cylinders \( a \) and \( b \), displacing fluid in windows 4 and 5, where it is under pressure below atmospheric, will not effect the torque.

Increasing \( \phi \) will lead to reduced volume of filling and emptying of operation chambers, torque will decrease more and more, and at \( \phi=90^\circ \) the distributor will be in a position where its jumpers will lie horizontally. In this case, the suction and flow will stop, and the torque will be determined only by the forces of friction.

The considered hydraulic machine was originally thought to be used as a hydraulic motor in order to obtain a linear adjustment (theoretical) characteristic (Figure 4, graph 4), and has the property of reversibility [7, 11]. And if the fluid from the hydraulic system is brought to its sickle-shaped windows 7 and 8, the hydraulic motor shaft at the rated torque will rotate at the nominal speed \( n_n \) corresponding to the point c (Figure 4). From windows 4 and 5, the fluid will be displaced into the discharge line of the hydraulic system. When the distributor rotates at an angle \( \phi \) from 0 to 90\(^\circ\), the productive torque will decrease along with the angular velocity \( n \), and at \( \phi=90^\circ \), the shaft will stop completely. But in contrast to the volumetric control, the phase control of the hydro-machine increases the control range \( D \) to values of 150...200 and more.

When the phase control of the motor is applied, the angle \( \phi \) is in the range from 0 to 90\(^\circ\), the block cylinders pass through the jumpers, having a certain speed.

If the cylinder is filled with liquid under the pressure of a hydraulic system pump, for example cylinder \( b \) (Figure 5), and is under a jumper, then the pressure in its operation chamber starts dropping (the pressed liquid expands). As soon as it decreases by 0.8...1.2 bar, the backflow valves 12 will open (Figure 2), and the service fluid from the supply fitting 21 (Figure 1) will flow through channel 13 into the operation chamber. At the same time there will be no discharging or cavitation.

If the cylinder displaces the service fluid into the drain line of the hydraulic system (for example, cylinder \( d \), Figure 5), where there is always a light pressure, and is under the jumper, then the pressure starts to rise in its operation chamber (compression of the pressed volume). At that moment the backflow
valves 15 will open (Figure 2) and the service fluid of the pressed volume through channel 16 through drain fitting 22 (Figure 1) will go to the drain and there will be no compression.

5. Conclusion

The study of the properties and parameters of a new high-torque reversible axial-plunger hydraulic machine, designed for operation in the hydraulic motor mode with phase rotation speed control, showed its advantages over typical designs of axial-piston machines.

The hydraulic motor with a two-row cylinder block has a three-fold greater operation efficiency and a torque, which allows it to be qualified as the type of high-torque hydraulic machines. Its power density increases 1.6 times, and power response more than four times. In this case, the pulsation of the flow rate of the fluid supplied to it (and the shaft rotation speed) is described by more than an acceptable value of the pulsation coefficient and equals to 0.0154.

The use of phase control provides the hydraulic motor with a control range of up to 150...200 or more, whereas traditional hydraulic motors have a control range of no more than 5, and this limits their implementation in hydraulic drives as adjustable hydraulic motors.

Therefore, the new axial-piston hydraulic motor with phase control has a high potential for practical implementation in hydraulic drives of machines as a type of motor that is specified as a high-torque and wide control range one.

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