14 AUTOMOBILE HYDRAULICS

Effective and safe operation of automotive equipment in many cases is ensured by the wide use of hydraulic devices in its constructions: gear, piston and electro-hydraulic pumps; hydraulic motors (hydraulic motor); valves; hydraulic cylinders, hydraulic accumulators and volume drives; other auxiliary devices.

A specific element of hydraulic devices is the working fluid, the patterns of its flow in pipelines and devices, the energy characteristics of these processes determine the features of the design and operation of hydraulic systems.

14.1 Elements of applied hydraulics

14.1.1 Properties of working fluids

The liquid has a unique property - easy mobility of its particles. This property - fluidity - determines the wide possibilities of using liquid in hydraulic drives.

Working fluids are characterized by physical parameters (specific weight, density, viscosity, etc.), which significantly affect the operation of the hydraulic drive, and operational properties, which are also a consequence of certain physical and chemical characteristics, but are manifested in certain indicators of convenience, economy, safety operation, durability, etc.

To number *basic physical parameters* (parameter) of working fluids should be attributed to the specific gravity γ, density ρ, dynamic μ and kinematic some viscosity coefficients, as well as viscosity index IV, compression ratio β_ρ.

Specific gravity is the weight of a unit volume of liquid, N/m^3

$$\gamma = \frac{G}{V}, \tag{14.1}$$

where G is the weight of the volume V liquid

Density ρ is called the mass of a unit volume of liquid, kg/m^3

$$\rho = \frac{m}{V}, \tag{14.2}$$

where m is the mass of the volume V liquid Ratio between specific gravity and density

$$\gamma = \rho \cdot g, \tag{14.3}$$

where g - acceleration of free fall.

*Viscosity*is called the property of a liquid to resist relative movement of its particles. During the movement of liquid in the pipeline due to viscosity, the layers located close to the axis of the pipe will have the highest speed. Layers that are adjacent to the walls will be inhibited.

*According to Newton's hypothesis*the force of internal friction that occurs between the dry in deep layers of moving liquid is directly proportional to the speed of relative movement and the area of the contact surface and does not depend on pressure.

*In practice, the so-called is used*kinematic coefficient viscosity (viscosity factor),*which characterizes both viscosity and inertia- the presence of liquid*

$$=\mu/\rho. \qquad (14.4)$$

It is measured in the city2/with. There are units in the literature , which is called by Stokes (St) - see2/s and centistokes (cSt) – mm2/with.

The viscosity of the working fluid significantly affects the characteristics of the hydraulic drive. At high values of viscosity, significant forces of internal and external friction (against the walls of the pipeline) cause significant energy losses, which leads to heating of the liquid. The efficiency of the hydraulic drive decreases, and the liquid often has to be cooled, spending additional energy on it. At low viscosities, energy losses due to friction and liquid cooling are significantly reduced, but at the same time, leaks due to gaps and leaks in pumps, hydraulic motors and other hydraulic units increase sharply. This means not only increased energy losses, but also a violation of the kinematic accuracy of the hydraulic drive.

The choice of the optimal viscosity allows obtaining minimal leaks with small frictional energy losses and is an important technical and economic task. Usually, the value of the optimal viscosity depends on the working pressure in the hydraulic drive and increases as the latter increases (Table 14.1).

Table 14.1 - Recommended values of kinematic viscosity

Pressure range, MPa	to 7.0	to 20.0	up to 60.0
Recommended kinematic viscosity *, cSt*	20 - 40	60 - 110	100 - 170

In addition to pressure, the type of pumps and hydraulic motors for which the working fluid is selected significantly affects the optimal viscosity value.

Fluids used in hydraulic drives can be divided into viscosity classes, indicated in table 14.2.

Table 14.2 – Working fluid viscosity classes

Viscosity class	5	7	10	15	22
Kinematic viscosity, mm2/s, at 40°C	4.14 5.16	6,12 7.48	9.0 11.0	13.5 16.5	19.8 24.2
Viscosity class	32	46	68	100	150
Kinematic viscosity, mm2/s, at 40°C	28.8 35.2	41.6 50.6	61.2 74.8	90.0 110.0	135.0 165.0

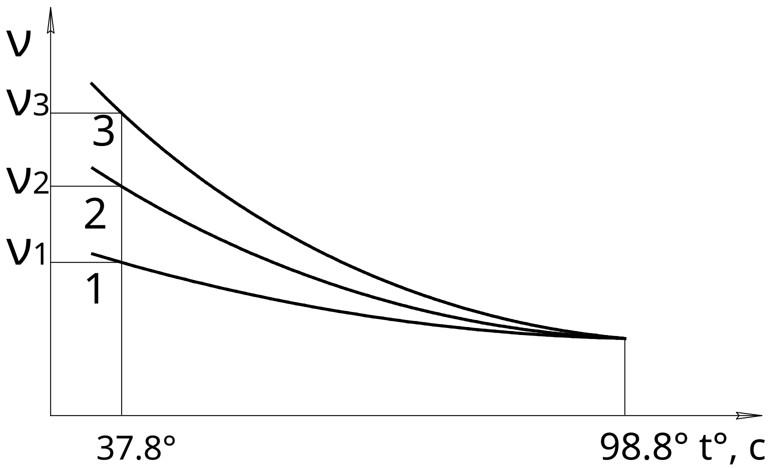
An important parameter of the working fluid is the viscosity index (IV), which evaluates the intensity of viscosity change depending on temperature.

$$IB=\frac{\nu_2-\nu_1}{\nu_3-\nu_1}\cdot 100, \qquad (14.5)$$

where$_3$– the viscosity of the reference liquid with IV = 100, that is, the viscosity of the liquid which significantly depends on the temperature t;

$_1$is the kinematic viscosity of the reference liquid with IV = 0, that is, the liquid the viscosity of which practically does not depend on the temperature t;

$_2$is the viscosity of the real liquid for which the IV is determined, and chen $_1$, $_2$and$_3$are determined at t = 37.8°C, and att = 98.8°C should be $_1$= $_2$= $_3$(Fig. 14.1).



1 – reference liquid with IV = 0; 2 – tested liquid; 3 – reference liquid with IV = 100

Figure 14.1 – Scheme for determining the IV viscosity index

Practically, the IV of working fluids is determined according to tables, for which it is necessary to know the kinematic viscosity at a temperature of 50°C and100°C.

The dynamic properties of the hydraulic drive - stability, self-oscillating processes, adjustment accuracy - are significantly affected by the compression effect of the working fluid, which is estimated by the volumetric modulus of elasticity$E_p$or coefficient compressionβ_p, moreover

$$\beta_p=\frac{1}{E_p}\text{, MPa}^{-1}. \qquad (14.6)$$

The compression coefficient shows the relative change in the volume of the liquid, related to the unit of pressure increase, i.e.

$$\beta_p=-\frac{\Delta V}{V_0\cdot\Delta p}, \qquad (14.7)$$

where ΔV – change of initial volume V_0 of liquid caused by pressure increase Δp.

It is known that working fluids contain dissolved and undissolved (in the form of an independent phase) gas. Most often it is air. With increasing pressure, undissolved air turns into a solution and vice versa.

Undissolved gas impurities significantly affect the compression ratio β_p. Operating conditions of hydraulic drives, as a rule, contribute to both microvolumes of air by streams of working fluid, as well as the release of air, which was previously in a dissolved form, as a result of the passage of the fluid flow through various openings - the working windows of hydraulic equipment elements, and the intensity of gas release is influenced by the shape of the working window and the speed of the liquid flow. The more undissolved gas in the liquid, the smaller its bulk modulus, especially in the area of low pressures.

The operational properties of the working fluid, which complement the physical parameters, include:

- the purity of the liquid, i.e. the characteristics of impurities present in it;
- the presence of various additives in the liquid, which give it additional expluation properties;
- stability of chemical and physical properties in a certain range temperatures, as well as a low solidification temperature, which should be below the maximum working temperature range by 10...15°C;
- high lubricating and anti-corrosion quality;
- compatibility with the materials of structural elements of the hydraulic system;
- high antifoam resistance;
- fire protection, environmental neutrality and compliance sanitary standards;
- durability, economy and non-scarcity.

It is practically impossible to provide all the listed properties at the same time, so the optimal option that solves a specific problem is chosen.

The presence of impurities in the working fluid - mechanical impurities - is regulated by the state standard (GOST 17216), which provides for 19 classes of fluid purity. Sizes of pollution particles (particles of metal, ceramics, resin formations, organic particles, etc.) and their number in a volume of 100±5 cm ^{3} form the dispersed composition, and the mass content - the pollution limit.

In practice, for each design of the hydraulic unit, its developers indicate the nominal fineness of filtration, i.e. the maximum allowable size of particles of pollution. This requirement is fulfilled by installing filters in the hydraulic system - devices that retain all particles larger than the maximum permissible.

Approximate recommendations regarding the fineness of filtration of the working fluid for various hydraulic units are indicated in the table. 14.3.

Table 14.3 - Recommended fineness of filtration

The name of the hydraulic unit		The nominal fineness of the filter walkie-talkie, micron
1. Pumps and hydraulic motors:		
	gear	25
	piston	10; 25
	hydraulic cylinders	40; 63
2. Distributors		10; 25; 40; 63
3. Valves:		
	reverse	25; 40; 63
	pressure	10; 25; 40; 63
4. Pressure switch		63

According to the presence of additives in working fluids, one of the most common types of fluids – hydraulic oils – is divided into categories HH-HG (Table 14.4).

Table 14.4 – Classification of lubricants of group H for hydraulic systems under hydrostatic conditions

Category product	Product characteristics	Field of application
NN	Refined mineral oils without additives	
HL	Purified mineral oils with improved anti-corrosion and antioxidant properties	
NM	HL type oils with improved anti-wear properties	hydraulic systems, which turn on under a very high load dumb
HR	HL type oils with improved viscosity-temperature properties	
HV	HM type oils with improved viscosity-temperature properties	Construction and marine machinery
HS	Synthetic liquids that do not have special fire-resistant characteristics	
HG	HM type oils with anti-seize properties	Hydraulic drives with a single system of circulation culation

All oils are products of oil processing and have high lubricating properties.

Their significant drawback is the dependence of viscosity on temperature (*IV* =85...90), which requires stabilization of the working temperature at the level *t*= 40°C...50°C.

14.1.2 Flow of working fluid in pipelines

If in the flow of liquid moving in a pipe, we draw a line so that the tangents to it at each point coincide with the directions of the velocities of the liquid particles located at these points at the given time, then such a line (straight or in the more general case, the curve) is called*by the line whose*.

In fig. 14.2 shows liquid particles 1...6, velocity vectors of these particles$\bar{\upsilon}_1$...$\bar{\upsilon}_6$and flow line*T*.

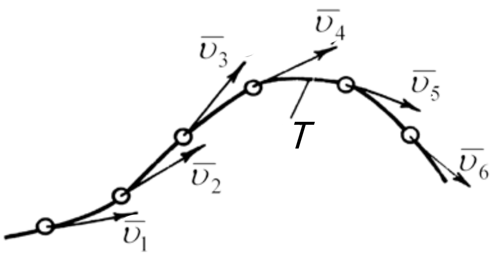


Figure 14.2 – Flow line

Let's select an elementary plane*dS*in a moving fluid is normal to the velocity vectors of fluid particles (Fig. 14.3), and through all points of this plane we will draw flow lines for any instant of time. We will get, as it were, a bundle of flow lines, which is called*elementary flow*, and its side surface is*flow tube*.

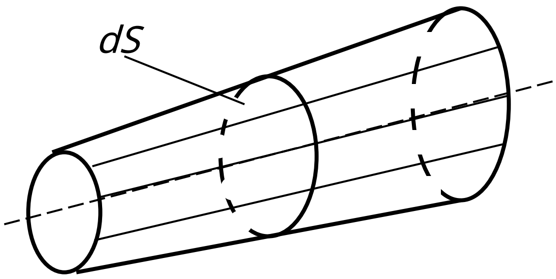


Figure 14.3 – Flow pipe

If the movement is constant, that is, the parameters do not depend on time, then the shape of the flow pipe remains unchanged over time, and the flow lines through the flow pipe do not go from the inside to the outside and vice versa. Velocities at all points of the cross section*dS*flow tubes are the same due to smallness*dS*. It can be assumed that the speedυ_1,υ_2, -,$\upsilon_i$are the values of velocities in one different cross-sectionsdS_1,dS_2, ...,dS_{and}, and the total fluid flow in the pipe is a set of elementary fluid flows.

The surface drawn through a given point within the flow perpendicular to the flow lines is called a live section.

If the curvature of the currents is small (the angle of separation between individual currents is small), then the live sections of the flow are planes perpendicular to the axis of the flow. So in fig. 14.3 a live section is a circle perpendicular to the axis of the would, area S which is equal to $S=\pi/4\cdot d^2_{tr}$, where d_{tr} is the diameter of the pipe opening.

Such a movement of a liquid is called a smoothly variable movement.

Through the live section of the stream (in our case, through the transverse section of the pipe opening) *a volume flows per unit of time, which is called a waste*. The term is also used *innings*.

Expenditure size Q - m^3/with. This is a volume expense.

The total flow through the live section of finite dimensions will equalize

$$Q=\int_S dQ_i = \int_S \upsilon_i \cdot dS_i. \tag{14.8}$$

In practice, they use the so-called *average speed.* *Average by the flow rate υ_c in this live cross-section is called the conditional density, the same for all particles located at the points of this section, which forms the same flow rate as the valid distribution of velocities*. Then

$$Q=\upsilon_c S=\int_S \upsilon_i \cdot dS_i. \tag{14.9}$$

The actual distribution of the velocities of the fluid particles belonging to the live flat cross-section at each moment of time depends on the mode of fluid motion.

There are two well-defined flow regimes:

– *laminar mode*, in which the liquid flows in separate layers, without stirring;

– *turbulent regime*, in which the speed distribution is chaotic, and the liquid is mixed throughout the volume of the flow.

It was established that the transition from the laminar to the turbulent regime and vice versa is determined by the dimensionless criterion Re, which is called *by number Reynolds*, moreover

$$\mathrm{Re} = \frac{\upsilon_c d_{tr}}{\nu}. \tag{14.10}$$

When Re < 2000...2320, the flow is laminar, and when Re > 2320, it is turbulent. There are a number of factors (roughness of the walls of the pipe opening, conditions of flow entering the pipe, etc.) that can change the above critical values.

The flow mode of the liquid in the pipeline affects the energy losses during the movement of the liquid in the pipe. There are two types of energy losses of a moving fluid – *travel losses* and *losses in local obstacles.* *Road losses* are called energy losses to overcome frictional forces on a certain length of a pipeline of constant cross-section.

Pressure losses Δp_{ter} on friction, taking into account the mode of movement of the liquid, which is characterized by the Reynolds number Re, can be determined by the formula

$$\Delta p_{ter} = \xi_c \rho \frac{\upsilon^2}{2} \qquad (14.11)$$

where ξ is the loss coefficient, which in the case of path losses is equal to friction only

$$\xi = 16 \cdot a_t \frac{l}{d_{tr} \cdot \mathrm{Re}}. \qquad (14.12)$$

where a_t is the correction factor.

Formula (14.11) is universal. It follows that energy losses to overcome both road and local resistances are proportional to the average flow speed. Therefore, the determination of the optimal value υ_c is responsible technical and economic task.

It was indicated above that, in addition to travel losses, there may also be so-called *E- direct losses*. They arise when the liquid flows through local supports as a result of various changes in the configuration of the flow, that is, when it expands or narrows by transitioning from a straight section to a curved section.

In local resistances, the energy of the liquid is spent on the redistribution of velocities, changing the direction of movement, vortex formation and mixing in the flow.

Hydraulic drives are often used to regulate the flow parameters of the working fluid and, accordingly, to control the operating modes *throttle slow (throttling) elements*, which are specially created local supports.

A simplified diagram of a throttle element is a hole in a membrane installed in a pipeline or channel of a certain hydraulic device (Fig. 14.4). If the diameter d_{Dr} hole is larger than its length l, then the wall is considered thin. In this case, frictional losses along the throttle opening are practically zero (in formula (14.12) $l \to 0$) and, as follows from the diagram in Fig. 14.4, energy is lost due to the sudden narrowing of the flow before entering the hole and its expansion at the exit.

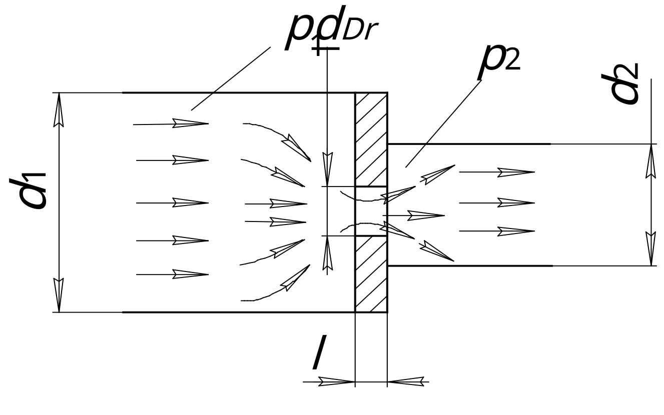


Figure 14.4 – Diagram of a throttle element

We apply the formula (14.11) to calculate the characteristics of the throttle device according to fig. 14.4, considering that υ_c is the average speed of the flow of liquid in the orifice of the throttle with a diameter d_{Dr} with liquid consumption Q_{Dr}. Then

$$\upsilon_c = \frac{Q_{dp}}{f_{dp}} = \frac{Q_{dp}}{0.25\pi \cdot d^2_{Dr}}. \qquad (14.13)$$

Pressure losses

$$\Delta p_{ter} = p_1 - p_2 = \xi \cdot \frac{\rho}{2} \cdot \left[\frac{Q^2_{dp}}{(\pi^2/16) \cdot d^4_{dp}}\right], \qquad (14.14)$$

from where the fluid flow through the throttle at a given pressure drop is equal to

$$Q_{dp} = f_{dp} \cdot \sqrt{\frac{1}{\xi}} \cdot \sqrt{\frac{2}{\rho}\left(p_1 - p_2\right)}. \qquad (14.15)$$

Size $\sqrt{1/\xi}$ is called the flow rate and is determined value ξ, which, as mentioned above, depends on the structural and geometric parameters of the throttle device.

For example, for the design, the scheme of which is shown in fig. 14.4, value ξ depends on the ratio of diameters d_1, d_{Dr}, d_2 and is determined by tab- Lyceum 14.5.

Table 14.5 – Value of the coefficient ξ

d_{Dr}/d_1	d_1/d_2				
	0.45	0.65	0.80	0.90	0.95
0.45	1.46	1.15	0.88	0.67	0.58
0.65	1.20	0.92	0.67	0.47	0.39
0.80	0.94	0.67	0.45	0.27	0.20
0.90	0.74	0.50	0.30	0.16	0.13
0.95	0.60	0.40	0.23	0.10	0.08

For the ratios of the geometric parameters of the throttle, which are most often found in real designs, when $d_1/d_2 \to 1$, a $d_{Dr}/d_1 \ll 1$, size rank $1/\sqrt{\xi} = 0.65 \ldots 0.70$.

In throttle devices in which l >> d_{Dr} (see fig.14.4), become significant them friction losses in the throttle opening. Since friction depends on the viscosity of the working fluid, and the viscosity changes with temperature, the resistance of such a throttle depends significantly on the type of working fluid and its temperature. In many cases, this is an undesirable factor.

Throttle with $l \ll d_{Dr}$ free from the specified defect. All by design Throttle devices are divided into *unregulated*, that is, with d_{Dr}=const, and *re- played*, that is, with a variable value of the area of the live section of the hole, which is called *working window*.

14.1.3 Pipelines

Fixed and non-stationary processes in pipelines significantly affect the technological characteristics of the hydraulic drive as a whole. The choice of the optimal value of the live section of the pipeline and the assessment of wave processes in it are of practical importance, because the pipeline with moving liquid is a dynamic system with distributed parameters. An understanding of the constructions of pipelines used in cars allows us to assess their influence on the operation of hydraulic drives.

The main calculation parameter of the pipeline is *living area section,* that is, the area S_{tr} of the cross-section of its opening, because at given cost Q_{tr} through the pipe size S_{tr} determines the average speed υ_c flow

$$\upsilon_c = Q_{tr} / S_{tr}. \qquad (14.16)$$

It follows from formula (14.11) that pressure losses in the pipe are proportional to the square of the average velocity, that is, inversely proportional to the square of the cross-sectional area of the opening. Reducing S_{tr}, i.e. reducing the transverse ones dimensions and weight of the pipeline, we dramatically increase energy losses, that is, reduce the efficiency of the hydraulic drive. Therefore, the calculation comes down to choosing the optimal practical value υ_c by recommendations, for example table 14.6.

Table 14.6 - Recommended average pipe velocities

Purpose of the pipeline	Approximate values of the maximum average velocities of the liquid in the pipe, m/s
Absorbent	1,2
Drainable	2.0
Discharge at pressures, MPa	
up to 2.5	2.0
to 6.3	3.2
to 16.0	4.0
to 32.0	5.0
over 63.0	6.3 - 10.0
Short channels in hydraulic units	to 20.0

Then from (14.16)

$$S_{tr} = \frac{\pi}{4} \cdot d_{tr}^2 = \frac{Q_{tr}}{\upsilon_c}. \qquad (14.17)$$

Where

$$d_{tr} = \sqrt{\frac{4}{\pi} \cdot \frac{Q_{tr}}{\upsilon_c}}. \qquad (14.18)$$

Since the second geometric parameter of the pipeline - the length - is determined by the hydraulic drive layout, it is necessary to check the amount of pressure losses in the pipeline (both road and local).

Total pressure losses along the length of the pipeline and in local supports should not exceed 5...10% of the pressure provided by the pump. In some cases, losses of up to 20% of the pump pressure are allowed.

In general, along the pipeline, which includes, in addition to straight sections, several local resistances, the total losses are equal to

$$\Delta p_{mep\Sigma} = \sum_{i=1}^{i=n} \xi_i \rho \cdot \frac{\upsilon_{ci}^2}{2} \qquad (14.19)$$

where ξ_i – coefficients of losses along the length and in local supports, respectively;

υ_{ci} – average flow velocities in these sections.

The liquid in the pipeline moves in waves. Its movement can be accompanied by the phenomenon of hydraulic shock, which occurs when local resistances suddenly act. The result of this process is often the destruction of pipes, hydraulic cylinders, violation of the necessary modes of operation of the hydraulic drive.

To estimate the value Δp_{min} pressure increase caused by hydraulic with a sharp blow, you can use Eq

$$\Delta p_{min} = with \cdot \rho \cdot \Delta \upsilon, \qquad (14.20)$$

where *with* – speed of sound in the flow of liquid in the pipe;

ρ – liquid density;

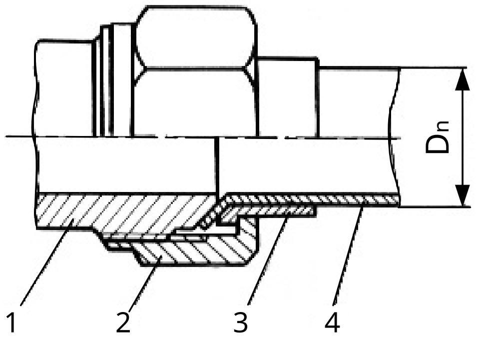
$\Delta \upsilon$ – the increase in speed caused by the closing (opening) of damper (or a device similar to the damper).

Construction of pipelines depends on the mutual location of the guides snail aggregates and features of their operation. All pipelines connecting hydraulic units, the mutual location of which is constant, are made of rigid metal pipes. Such pipes are connected to each other by welding (non-removable connection) or by means of fittings (removable connection).

For pressures of 15 MPa and above, seamless steel pipes are used. Steel electrowelded pipes are used at pressures up to 7 MPa, aluminum alloy pipes - for pressures up to 15 MPa, copper pipes - up to 3 MPa.

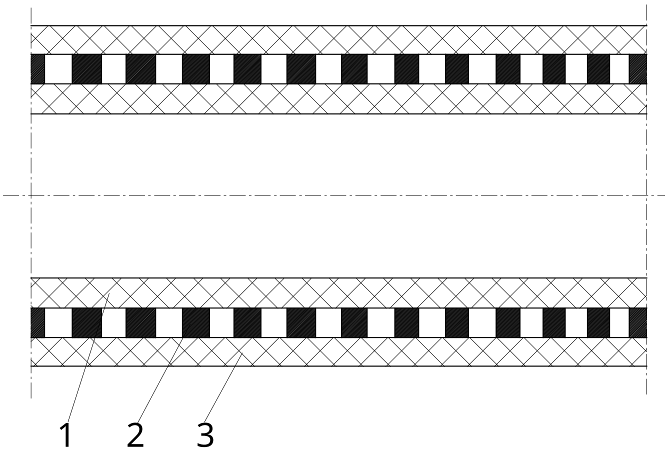
The design of the fitting connection is shown in Fig. 14.5.

Flexible pipelines - high-pressure hoses are used to connect hydraulic units, the mutual location of which changes during the operation of the technological machine. In fig. 14.6 shows the structural diagram of the high-pressure hose. The inner tube 1 made of oil-resistant rubber is wrapped on the outside with a metal mesh 2, which is a power element that absorbs the pressure of the liquid in the sleeve. From above, the entire structure is protected by a rubber layer 3. In addition to rubber and metal layers, windings made of strong cord threads can also be used, which ensure the monolithicity of the entire structure.



1 – a pipe with a welded threaded part of the fitting; 2 – overhead nut;
3 – nipple; 4 – a pipe with a flared end

Figure 14.5 – Fitting connection of pipes



1 – internal rubber tube; 2 – metal mesh; 3 – outer rubber layer

Figure 14.6 – Structural diagram of a high-pressure hose

One or three metal frames are used in the construction of high-pressure hoses, which are separated by rubber-cord layers. Of course, the more metal frames, the higher the permissible working pressure in the sleeve, but the strength properties of only the inner frame, which perceives about 60% of the pressure in the sleeve in a multi-frame design, are fully used. The least effective is the third frame, which perceives only about 10% of the pressure.

Two-frame sleeves are most often used.

High-pressure hoses are sensitive to the quality of manufacture and operation, namely:

− it is not possible to twist the sleeve relative to its axis when installing in hydraulic system;

- sleeve bending radii less than permissible should not be allowed;

- the resource of the sleeve is mainly determined by the shape, amplitude and frequency by the percentage of pressure pulsation in it, provided that the value of the working pressure is not exceeded;

- the responsible element of the sleeve design is the place of attachment of the tal tips - fitting elements.

The ability of a high-pressure sleeve to resist fluid pressure is much lower than that of steel pipelines, that is, the deformations of its cavity are much greater. Therefore, as follows from equation (14.20), the speed of sound in it is much lower, and the pressure increase during hydraulic shocks is also lower.

14.2 Hydraulic volume drives

Hydraulic drives belong to the group of the most used energy flow converters (PE).

All energy converters can be attributed to one of the following groups:

− PEs that transform the parameters of the energy flow without changing its output do These are mechanical gearboxes, on the input shaft of which there is a moment M_1 and angular speed ω_1, on the original − M_2 and ω_2, respectively; mechanical type transmissions "screw-nut" or "gear-rail" converting rotary motion with energy parameters M and ω in translational, which is characterized by strength F and speed U; electric transformers, on the input winding of which railway line U_1 and current I_1, on the day off − U_2 and I_2, respectively, and others;

- PEs that convert types of energy. These are electric motors that restore the tension U and current *AND* at the engine input, i.e. electrical energy gium, into mechanical energy at the output, i.e. moment M and angular velocity ω on the shaft; electrical generators that convert mechanical energy, i.e. moment M and angular velocity ω on the shaft, in electric, that is, voltage U, current *AND*; hydraulic pumps that convert mechanical energy with parameters M, ω on the flow of energy, the parameters of which are volume or mass supply Q fluids and pressure p at the pump outlet;

- PE of the combined type, in which the conversion is carried out as energy flow parameters and its type.

It should be noted that multiple transformations of the type of energy can occur in PE of combined action, the ultimate goal of which is the most effective transformation of the type and parameters of the energy flow at the input of the PE into the corresponding energy characteristics at the output of the PE.

PE efficiency criteria can be: efficiency of the transformation process (regulation); mass (metal capacity) of the device and its dimensions; reliability; the possibility of constructive placement of PE nodes in a technological machine, for example, a car.

Hydraulic actuators, and this manual considers the so-called *voluminous hydraulic drives*, are one of the most common types of PE of the combined type.

14.2.1 Principle of operation of the simplest three-dimensional hydraulic drive The scheme of the simplest three-dimensional hydraulic drive (HP) is shown in fig. 14.7 and contains almost all the main elements of such devices.

On the input shaft 5.1 of this hydraulic drive, the mechanical energy is characterized by a moment M and angular velocity ω. The first energy transformation is carried out in this scheme by the crank-connecting mechanism (marked in Fig. 14.7 by positions 5.1, 5.2, 5.3), at the output of which the force develops F_1, attached to the piston 4.3, which moves with speed υ_1.

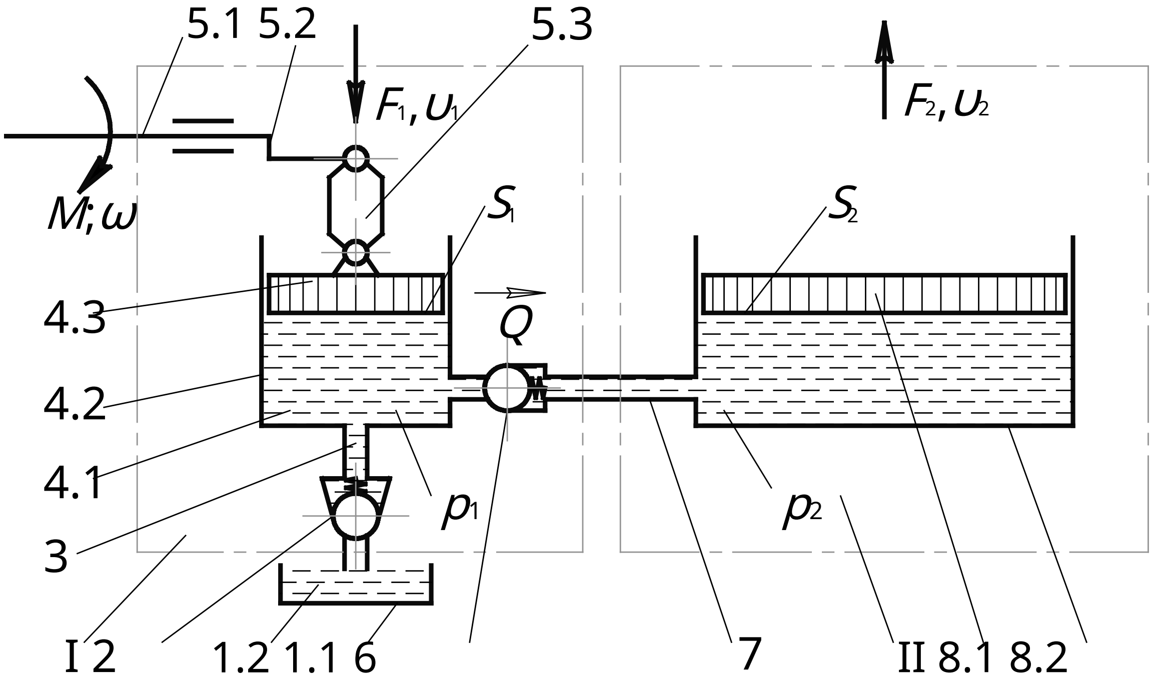


Figure 14.7 – Scheme of volumetric hydraulic drive

The presence of a primary converter of mechanical energy at the input of the hydraulic drive is not mandatory. For example, it is absent in GPs with gear-type pumps. In the presence of such a converter, it is not always a crank mechanism. The design variety of such converters is quite large.

If the direction of movement of piston 4.3 corresponds to the arrow in fig. 14.7, then the liquid 4.1, which is in the cylinder 4.2, is squeezed by the piston 4.3 through the valve 6 and the pipe 7 into the cylinder 8.2, forcing the piston 8.1 to move up, that is, in the direction indicated by the arrow. At the same time, the mechanism, which consists of a cylinder 4.2 and a piston 4.3, converts mechanical energy (F_1, υ_1) on hydraulic (pressure p_1, submission Q), and in the device, which consists of a cylinder 8.2 and a piston 8.1, hydraulic energy is converted into mechanical energy of translational movement at the output of this PE, i.e. force F_2 and speed υ_2.

As indicated above, the hydraulic energy flow parameters are pressure p and supply *Q* liquid

By pressure *p* on the surface area is called the pressure force ratio *F*, which applies to this plot, to the area of the plot *S*, i.e $p = F/S$.

In the SI system, pressure is measured in Pascals (Pa), 1Pa = N/m^2.

In practice, a derived unit of pressure is used - *megapascal* (MPa = 10^6 Pa), as well as the non-system unit of pressure - *the atmosphere* (at).

$$\text{at} = \frac{\text{kHz}}{\text{see}^2} = 10^{-1}\ \text{MPa} = 10^5\ \text{Pa}.$$

Liquid supply *Q* is called the volume of liquid that flows through the river section of the pipeline per unit of time. This is the so-called volumetric supply, that is $Q = V/t$, m^3/with.

Liquid supply can also be called consumption. In the SI system, volume supply is measured in m^3/s. The power of the flow of hydraulic energy is equal to $P_{Mr} = p \cdot Q$.

Part of the device shown in fig. 14.7, which is within frame I and converts mechanical energy into hydraulic energy, is a pump, and within frame II is a hydraulic motor that converts hydraulic energy into mechanical energy.

The movement of the piston 4.3 with the help of the crank-connecting mechanism 5.1 - 5.3 is carried out so that after the piston reaches the bottom "dead" point, it begins to move up. At the same time, valve 6 closes under pressure p_2, and valve 2 also opens into cylinder 4.2 liquid 1.2 is sucked from tank 1.1. During the upward movement of piston 4.3, piston 8.1 of the hydraulic motor is motionless. If it is necessary to obtain continuous movement of the hydraulic motor, a pump with several cylinders is used, the periods of liquid supply of which are phase-shifted.

Valves 2 and 6 are hydraulic diodes. They are called non-return valves and are organically included in the pump device 1, automatically controlling fluid flows. Fluid flow control systems at the inlet (suction) and outlet (discharge) of the pump do not necessarily consist of non-return valves and differ in design diversity.

If the radius of the crank mechanism is equal to *r*, and the area of the pump piston S_1, then for each stroke of the piston (one revolution of the drive shaft) volume is squeezed out of cylinder 4.2 V_1 liquid, where

$$V_1 = 2 \cdot r \cdot S_1. \tag{14.21}$$

If the pump has one piston, then V_1 is called the working volume if *Z* of pistons, then the working volume V is equal to

$$V = V_1 \cdot Z = 2 \cdot r \cdot S_1 \cdot Z. \tag{14.22}$$

The volume of the cylinder 4.2, located under the piston 4.3, is called the working chamber of the pump, and the liquid supply is carried out by squeezing it out of the working chamber by reducing the volume of the latter by ΔV_H.

The movement of the piston 8.1 of the hydraulic motor II is carried out by supplying liquid to the cylinder 8.2, while the volume of the cylinder 8.2, located under the piston 8.1, that is, the volume of the working chamber of the hydraulic motor, increases byΔV_{two}. At the same time

$$\Delta V_{two}=\Delta V_H. \qquad (14.23)$$

Analyzing the principle of operation of this hydraulic drive described above, we can conclude that a volumetric hydraulic drive is an energy converter in which, in order to control the parameters of the energy flow, there is a simultaneous transformation of mechanical energy into hydraulic energy and vice versa, carried out by changing the working volumes pump and hydraulic pump chambers.

It is known that the pressure generated by surface forces is transmitted unchanged to every point of the liquid and does not depend on the orientation of the plane on which it is measured, that is, it is the same in any direction. This is Pascal's law. During the movement of liquid, for example, through pipeline 7 (see Fig. 14.7), as well as through check valve 6, energy losses occur and$p_2<p_1$. It can be taken into account using a coefficientη_{Mr}– hydraulic efficiency,

$$p_2=p_1\cdot\eta_{Mr}. \qquad (14.24)$$

If the pipeline 7 is of short length and the valve 6 has a slight resistance to the flow of liquid, then with some error it can be assumed$\eta_{Mr}\approx1$.

In this case

$$p_2\approx p_1=p. \qquad (14.25)$$

EffortF_1, applied to piston 4.3 by connecting rod 5.3, is equal to

$$F_1=p_1\cdot S_1+F_{mep1}, \qquad (14.26)$$

whereF_{mep1}– friction force of piston 4.3 in cylinder 4.2.

In turn, effortF_2, which develops on the piston 8.1 pressurep_2, is equal to

$$F_2=p_2\cdot S_2-F_{mep2}=p_1\cdot\eta_{Mr}\cdot S_2-F_{mep2}, \qquad (14.27)$$

whereF_{mep2}– friction force of piston 8.1 in cylinder 8.2.

If we consider frictional forcesF_{mep1}andF_{mep2}insignificant, and$\eta_{Mr}\approx1$, then in such an ideal case

$$F_1=p\cdot S_1,\ \text{and}\ F_2=p\cdot S_2. \qquad (14.28)$$

Coefficientn_{F_i}transformation of the flow of mechanical energy by effort in in the ideal case will be equal to

$$n_{F_i}=\frac{F_2}{F_1}=\frac{p\cdot S_2}{p\cdot S_1}=\frac{D_2^2}{D_1^2}. \qquad (14.29)$$

Taking into account the energy losses due to the movement of liquid through the pipeline and friction in the piston pairs, that is, taking into account the relationship (14.24), (14.26) and (14.27), we obtain the formula for the real coefficient n_{F_p} re-flow of mechanical energy by effort

$$n_{F_p} = \frac{p_1 \cdot \eta_{Mr} \cdot S_2 - F_{mep2}}{p_1 \cdot S_1 + F_{mep1}} = \frac{p_1 \cdot \eta_{Mr} \cdot S_2 \cdot \left(1 - \frac{F_{mep2}}{p_1 \cdot \eta_{Mr} \cdot S_2}\right)}{p_1 \cdot S_1 \cdot \left(1 + \frac{F_{mep1}}{p_1 \cdot S_1}\right)}$$

or

$$n_{F_p} = n_{F_i} \cdot \eta_{Mr} \cdot \frac{\left(1 - \eta_{fur2}\right)}{\left(1 + \eta_{fur1}\right)} \tag{14.30}$$

where η_{fur1} and η_{fur2} – mechanical efficiency, characterizing energy losses on friction in pump I and hydraulic motor II, respectively.

In turn, the expression

$$\frac{1 - \eta_{fur2}}{1 + \eta_{fur1}} = \eta_{fur\Sigma} \tag{14.31}$$

is the mechanical efficiency of the hydraulic drive, which characterizes the total friction losses in the pump and hydraulic motor.

Then it follows from (14.29), (14.30) and (14.31) that

$$n_{F_p} = n_{F_i} \cdot \eta_{Mr} \cdot \eta_{fur\Sigma} = \frac{D_2^2}{D_1^2} \cdot \eta_{Mr} \cdot \eta_{fur\Sigma}. \tag{14.32}$$

Let's evaluate the numerical values n_{F_p} at $\eta_{Mr} = \eta_{fur\Sigma} = 1$, that is, the value n_{F_i}. Suppose that D_1 = 10 mm, a D_2 = 100 mm.

These are the most commonly used average values for hydraulic drives of various technological machines. Then

$$n_{F_i} = \frac{D_2^2}{D_1^2} = \frac{100^2}{10^2} = 10^2,$$

that is, the force on the piston of the hydraulic motor will be 10^2 times more force on the pump piston. This property is widely used in the simplest, but highly effective devices, such as hydraulic jacks.

In real hydraulic drives, the value of the coefficient n_{F_i} can reach a larvae 10^3 and more

Let's consider the transformation of another energy flow parameter - speed υ_i, and therefore the displacement of the pistons. When moving y_1 piston 4.3 from the pump (cylinder 4.2) volume is squeezed out V_1 liquid

$$V_1 = y_1 \cdot S_1. \tag{14.33}$$

At the same time, we did not take into account that some volume of liquid can leak out of the cylinder due to the gap between the piston and the cylinder and other loose connections (so-called leaks).

Due to the fact that we consider liquids as a continuous medium without gaps and voids, the same volume $V_2 = V_1$ will reach the hydraulic motor (v in the ideal case with no leaks) and the piston 8.1 will move by an amount y_2

$$y_2 = \frac{V_2}{S_2} = y_1 \cdot \frac{S_1}{S_2},$$

where

$$\frac{y_2}{y_1} = \frac{S_1}{S_2} = \frac{D_1^2}{D_2^2}. \tag{14.34}$$

Moving on to speeds $\upsilon_1 = \frac{dy_1}{dt}$ and $\upsilon_2 = \frac{dy_2}{dt}$ pthe ear pistons, receives mo coefficient $n_{\upsilon i}$ conversion of mechanical energy by speed into an ideal in the absence of leaks.

$$n_{\upsilon i} = \frac{\upsilon_2}{\upsilon_1} = \frac{S_1}{S_2} = \frac{D_1^2}{D_2^2}. \tag{14.35}$$

The presence of leaks can be taken into account with a coefficient η_0 volumetric efficiency. It is easy to make sure that the real coefficient $n_{\upsilon p}$ is equal to

$$n_{\upsilon p} = n_{\upsilon i} \cdot \eta_0 \tag{14.36}$$

In modern hydraulic drives η_0 quite close to 1. Yes, for hydraulic motors of translational movement (hydraulic cylinders) $\eta_0 = 1$, for piston-type pumps and hydraulic motors $\eta_0 = 0.95...0.98$.

For the numerical example considered above, we obtain

$$n_{\upsilon i} = \frac{10^2}{100^2} = 10^{-2},$$

that is, displacement and speed are reduced by a factor of 100. In order to obtain such a reduction in the mechanical PE, it is necessary to turn on the gear and worm gears sequentially.

It is easy to make sure that the result of all transformations is power $P_{entrance}$ energy flow at the entrance to PE and power P_{exit} are different at the output only by the value determined by the total efficiency $-\eta_\Sigma$.

Indeed, the power at the entrance

$$P_{entrance} = F_1 \cdot \upsilon_1. \tag{14.37}$$

Output power

$$P_{exit} = F_2 \cdot \upsilon_2, \qquad (14.38)$$

or, taking into account (14.30), (14.32), (14.35) and (14.36)

$$P_{exit} = F_1 \cdot \upsilon_1 \frac{D_2^2}{D_1^2} \cdot \eta_{Mr} \cdot \eta_{fur_\Sigma} \cdot \frac{D_1^2}{D_2^2} \eta_0 = F_1 \cdot \upsilon_1 \eta_{Mr} \eta_{fur_\Sigma} \cdot \eta_0 . \qquad (14.39)$$

Coefficient η_P the transformation of the energy flow is equal to

$$\eta_P = \frac{P_{exit}}{P_{entrance}} = \eta_{Mr} \cdot \eta_{fur_\Sigma} \cdot \eta_0 = \eta_\Sigma, \qquad (14.40)$$

i.e. is the total efficiency η_Σ of this PE.

Summarizing the consideration of the simplest hydraulic drive shown in Fig. 14.7, we can assess its positive qualities and shortcomings.

Positive qualities include:

− possibility of deep transformation of energy parameters, i.e. knowledge calculation of coefficients η_{F_i} and η_{υ_i} can reach values $10^{-3} \ldots 10^{+3}$;

- the possibility of obtaining significant efforts F at the output of the forward engines motion (hydraulic cylinders) or moments M at the output of rotary motion engines (hydromotors) due to the use of relatively high working pressures;

− significantly smaller dimensions and weight compared to PE of other types, e.g. warehouse, electric;

- high speed of action, which is a consequence of insignificant masses of moving elements nts of the hydraulic drive, i.e. low inertia;

- high operational reliability;

- the possibility of any mutual spatial arrangement of individual units hydraulic drive units, for example, a pump and a hydraulic motor, which greatly facilitates the design of machines in which they are used;

- the lubricating property of the working fluid, which, acting as a link in the kinematic chain, reduces the force of friction and thus ensures better performance of the hydraulic drive.

All these advantages are achieved thanks to the use of liquid as the main element of the hydraulic drive. However, this same factor is also the main disadvantage, because the use of a working fluid requires careful measures to prevent internal leaks, does not allow energy to be transmitted through pipelines over significant distances due to frictional pressure losses, and requires periodic replacement of the fluid, which quickly loses some of its properties (ages), and in cases of accidental destruction of pipelines leads to environmental pollution.

These shortcomings can be mostly eliminated, but they require a high culture of design, production and operation of hydraulic drives.

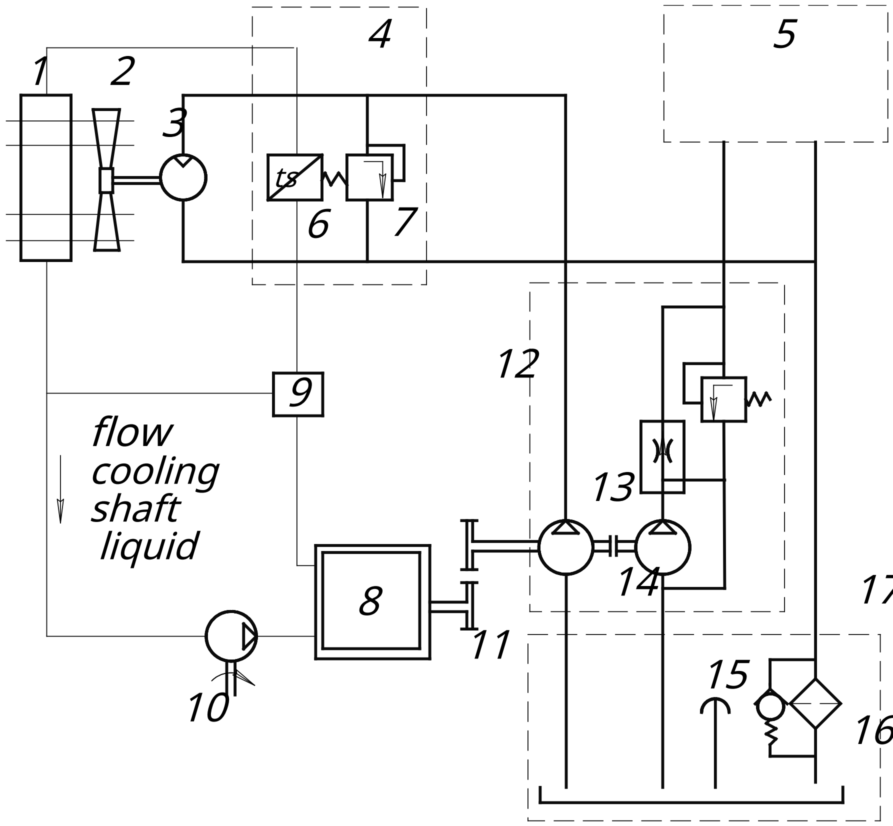
However, the advantages of the hydraulic drive are so obvious that, in addition to traditional use in metal cutting machines and forging and press equipment, it is widely used in agricultural machinery, road, transport and mining equipment, aviation, space and defense equipment. It found its wide use in the construction of cars.

14.2.2 Hydraulic volume drives of fans

The degree of influence of the air flow on the temperature of the coolant in the car engine is regulated by means of thermostatic control of the speed of rotation of the fan (fan). Hydraulic volumetric fan drives are often used on high-power engines (buses, trucks, construction and agricultural machines, stationary power plants), i.e., where special requirements are placed on the placement of the radiator (limited space, encapsulation of the engine). In addition to providing flexibility in the placement of the radiator, such a drive has the advantages of achieving a high power-to-weight ratio (low mass, small dimensions of individual elements), simple control and regulation, reliable operation and reduced wear of parts.

The main elements of the hydraulic drive of the fan (Fig. 14.8) are a pump, a hydraulic motor and a valve, the operation of which depends on the temperature of the cooling liquid and which serves to control the speed of rotation of the fan. The hydraulic motor drives the fan directly or through a belt transmission with a defined gear ratio. The frequency of rotation of the hydraulic motor depends on the characteristics of the specific fan and the effective pressure difference. If losses associated with power transmission are not taken into account, then it can be assumed that the speed of rotation is directly proportional to the pressure in the system.

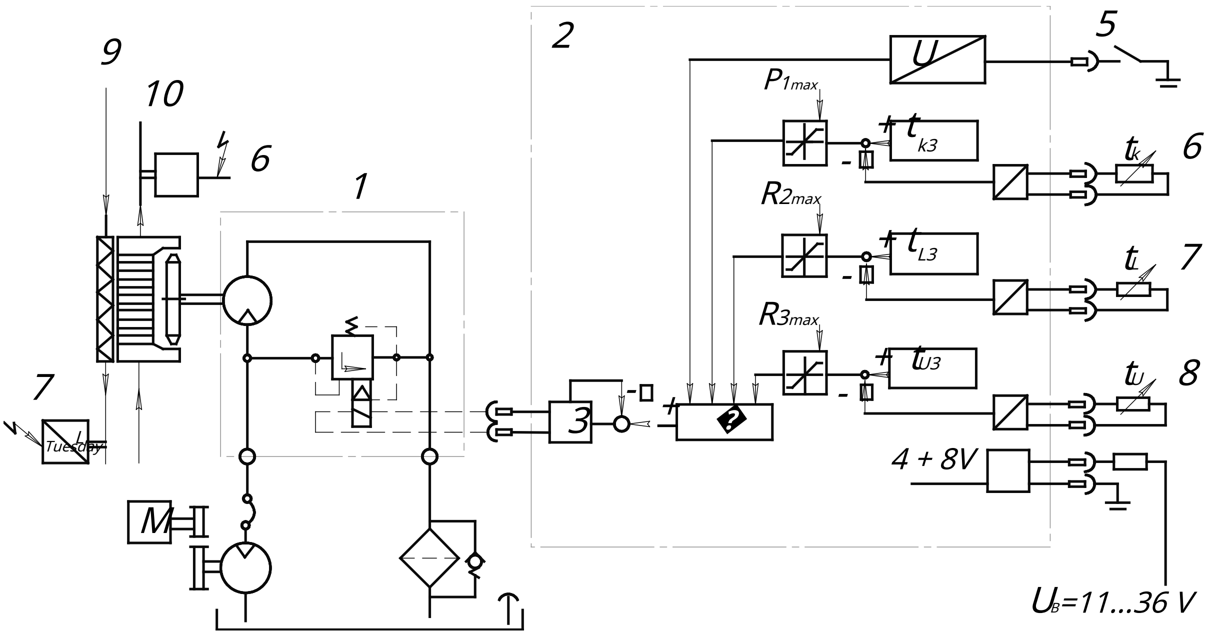
The principles of continuous and intermittent control can be used to regulate the temperature of the coolant in the engine. With two-position (intermittent) control, the bypass valve is used in the form of a distributor with electrical activation, controlled by a thermal switch in the engine cooling system. The discharge valve determines the maximum speed of rotation of the fan. The accuracy of fan speed control is a function of the pressure in the system to which the valve is set (usually 200 bar). A feature of continuous control is the presence of a discharge or throttle valve in the system, supplemented with a bypass pressure valve to limit pressure in the system. The operation of the drive is regulated by a mechanism that responds proportionally to changes in temperature.



1– fan; *2*– radiator; *3*– hydraulic motor; *4*– flow valve; *5*– power steering hydraulic drive; *6*– control element; *7*– discharge valve; *8*– car engine; *9*– thermostat in the cooling system;
10– a pump in the cooling system of a car engine; *11*– pump drive; *12*– pump (double block); *13*- pump *1* cooling system fan drive;
14- pump *2*; *15*– air filter; *16*– filter (cleaner); *17*- liquid tank

Figure 14.8 – Combined hydraulic fan drive and power steering

Electro-hydraulic systems in which the operation of the valve is regulated by a solenoid (proportional solenoid or solenoid with pulse-modulated activation) are becoming more and more common. This type of solenoid is controlled by an output signal from a temperature sensor located in the coolant line. Continuous control of the fan rotation speed is carried out to maintain the temperature with a deviation of no more than 5°C. Losses of the air pressure of the fan with this method of regulation are a maximum of 15%. In order to obtain the necessary ventilation of the engine when the speed of rotation of the fan is reduced, a limit of minimum pressure in the system is provided. Electronic control devices can be used to process additional signals (for example, temperatures inside and outside). They can serve as a basis for additional adjustment of the fan rotation speed. As an example, we can cite the combined use of the fan drive to regulate the temperature of the coolant, air in the heating system blowing and air in the engine compartment, as well as for turning on the maximum fan rotation speed during engine braking (Fig. 14.9).



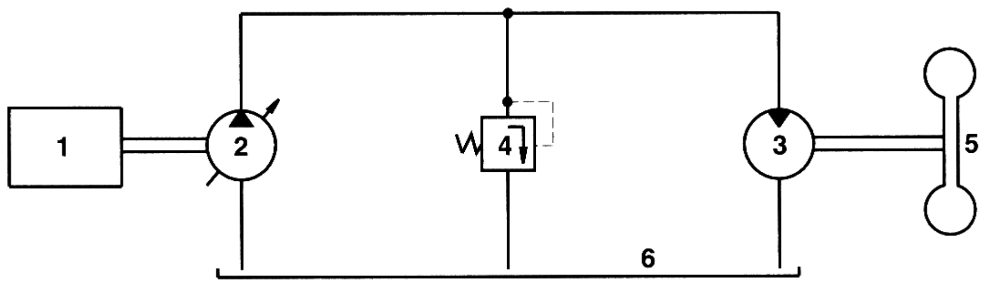
1 – gear hydraulic motor with pressure ratio valve; 2 – control unit; 3 – current regulator; 4 – voltage regulator; 5 – switch of the retarder of the car near; 6 – coolant temperature sensor;
7 – charge air temperature sensor; 8 – external temperature sensor;
9 – air inlet for inflation; 10 – coolant;
U_B– battery voltage; $t_{with}$is the set temperature value

Figure 14.9 – Electrohydraulic engine fan drive

Electrohydraulic systems can be integrated into the engine control system. The fan hydraulic drive can be used with other equipment and provide the operation of such devices as the clutch, transmission, compressors, water pump, generator, power steering, rear wheel steering, hydraulic dump mechanism. Systems with the necessary set of control functions and the use of a large number of pumps can provide priorities in work and safety requirements.

14.2.3 Hydraulic volume transmission drives

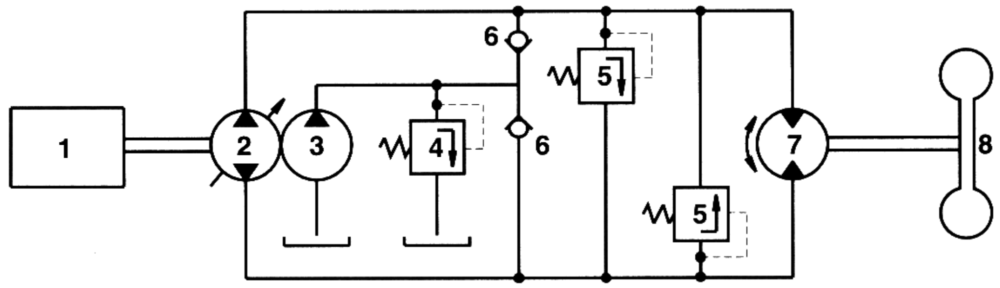
If the discharge line of an adjustable pump is connected to a hydraulic motor with a constant or adjustable frequency of rotation, then you can get a transmission with stepless adjustment of the gear ratio. The gear ratio is defined as the ratio of the working volumes of the pump and hydraulic motor. It is also possible to use schemes with parallel connection of several hydraulic motors or with their serial connection. However, the main one a drive with an open circuit (Fig. 14.10) cannot change the direction of its rotation or ensure the application of braking force without the help of auxiliary mechanisms. This type of circuit is suitable for use in conjunction with adjustable drives such as fan drives, etc. p.



1 – car engine; 2 – regulating pump; 3 – hydraulic motor; 4 – safety valve; 5 – driving wheels; 6 – tank with liquid

Figure 14.10 – Open circuit hydraulic drive

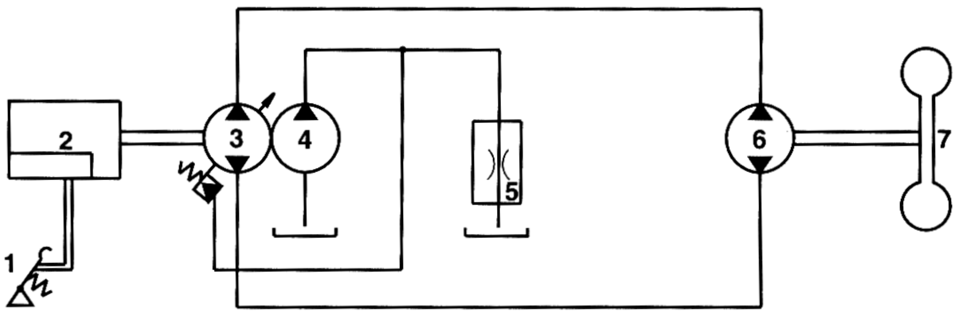
The main reason. Hydraulic systems in cars must ensure the operation of power steering and brake systems. For this reason, systems with a closed loop are preferred (Fig. 14.11). The main (reversible) pump is combined with the feed pump. The supply of liquid from the feed pump to the low-pressure line makes it possible to compensate for all leaks. Since there is always pressure from the feed pump at the inlet of the main pump, the maximum allowable speed of its rotation is higher than when operating in the suction mode. At a constant value of the gear ratio, this type of transmission provides almost the same stiffness of the drive as in a mechanical transmission.



1 – engine; 2 – regulating pump; 3 – feed pump; 4 – safety valve of the freeway; 5 – safety valve of the high pressure main; 6 – non-return valve; 7 – reversible hydraulic motor; 8 – driving wheels

Figure 14.11 – Closed loop hydraulic drive

"Automotive" control was developed to obtain characteristics similar to those required for automobiles. The most famous are schemes when the driver uses the pedal to control only the engine crankshaft speed (Fig. 14.12). Part of the power from the engine is sent through an additional pump to the control circuit, which contains a throttle to create control pressure, the value of which corresponds to the frequency of rotation of the crankshaft of the engine. This pressure, in turn, determines the flow of liquid passing through the main pump with the help of a control mechanism with a characteristic proportional to the pressure. This control concept is not complicated and prevents the motor from stalling because the pump reacts to a decrease in the rotation frequency by switching to lower flow values, which requires lower torque values. However, in order to meet stricter requirements for regulating engine power and fuel efficiency, it is necessary to use more complex schemes.



1 – control pedal; 2 – car engine with speed controller;
3 – hydraulic pump with pressure regulation; 4 – additional pump; 5 – throttle;
6 – hydraulic motor; 7 – drive wheels

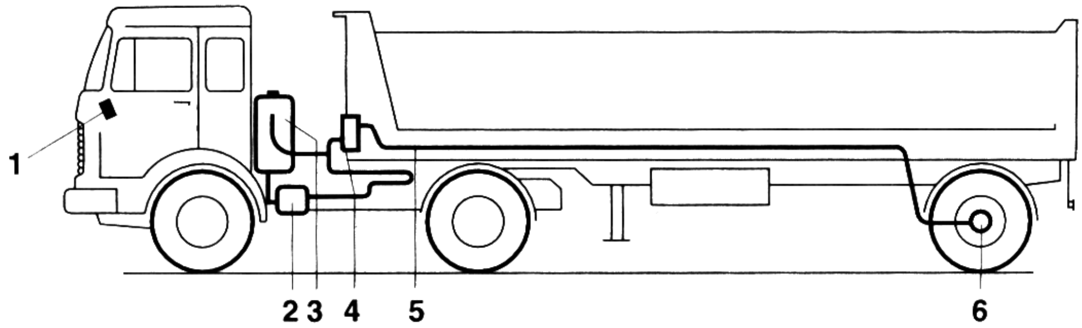
Figure 14.12 – Diagram of a hydraulic volume transmission for a car

Auxiliary drive. Another option for using a hydraulic volume drive is auxiliary drive devices on the non-driving axles of trucks (Fig. 14.13), which work only when moving at low speed on heavy soils. During normal driving on the road, this drive is switched off to reduce power losses.

14.3 Pumps and hydraulic motors

Executive outline(VC) is the main mandatory part volumetric hydraulic drive of any purpose and complexity. Energy flow conversion processes take place in VC.

The VC includes a pump or a group of pumps, a hydraulic motor and pipelines connecting them.



1 – electrical control of activation and deactivation of the auxiliary drive; 2 – pump of constant productivity with variable operating speed and the possibility of turning off nanny; 3 – tank with working fluid; 4 – removable connection of hoses; 5 – hydraulic mains; 6 – hydraulic motors in wheel hubs

Figure 14.13 – Hydraulic volumetric auxiliary drive

The rest of the hydraulic units, which are part of the hydraulic drive, are divided into two groups:

− hydro units that do not affect the energy conversion process and, accordingly, on the accuracy and parameters of the operating modes of the hydraulic drive; they include reservoirs for working fluid -*tanks*, devices for cleaning liquid from pollution -*filters*and others. Further, these hydraulic units will be called service units (SO);

- hydraulic units that manage the process of energy conversion and thus they significantly affect the accuracy and parameters of the operating modes of the hydraulic drive; they belong to various types*pressure regulators (valves)*,*regulation costs (speeds)*,*proportional distributors*and so on These hydraulic units are included in the group*hydraulic equipment*(HA).

*Pump*is a converter of mechanical energy into hydraulic energy. If the coefficient cents n_{Fi} and $n_{υi}$ conversion of energy flow by force and speed for values of this pump are constant, then the pump is called unregulated, i.e. its working volume V=const. If the pump is regulated, then V=var and, accordingly, the variables are n_{Fi} and $n_{υi}$.

Thus, the pump can be non-regulated, regulated non-reversible, that is, with a constant direction of flow, regulated reversible.

The general characteristics of the pump include:

- the main pump parameter -*working volume* V, m³/ob, that is, the volume of the dyna, which is supplied by the pump for 1 revolution of its drive shaft, or*character- r volume* $V' = V/2\pi$, m³/rad, that is, the volume of liquid supplied by the pump for one revolution of the drive shaft per radian;

- working pressure p_p, that is, the pressure provided by the pump during long-term work;

- maximum pressure p_{max}, in which short-term ro-
pump bot;

- consumption Q_n pump at operating pressure; $Q_n = V \cdot n_n \cdot \sigma_{in}$, where n_n - but-
variable frequency of rotation of the drive shaft of the pump;

– σ_{in} – *cost factor*, which is equal to the ratio

$$\sigma_{in} = \frac{Q_n}{Q_t}, \qquad (14.41)$$

where $Q_t = V \cdot n_n$ – theoretical flow rate of the pump;

- Pump efficiency $\eta_n = \eta_0 \cdot \eta_{Mr} \cdot \eta_m$, where η_0 – volumetric efficiency, which characterizes reduces energy losses due to liquid leaks; η_{Mr} – hydraulic efficiency, which is characteristic causes energy losses on local supports, channels and working windows of the pump; η_m - mechanical efficiency, which characterizes energy losses due to mechanical friction in the pump. The hydromechanical efficiency indicator is also used $\eta_{Ahem} = \eta_{Mr} \cdot \eta_m$;

- pump output power $P_n = Q_n \cdot p_p$. Drive power of the pump shaft

$$P_{entrance} = M_{Cr} \cdot 2 \cdot \pi \cdot p_n = \frac{Q_n \cdot p_p}{\eta_n}. \qquad (14.42)$$

Hydraulic motor together with the pump is the main unit included in the composition of the VC and is intended to convert the flow of hydraulic energy generated by the pump into mechanical energy with the parameters necessary for the efficient functioning of the technological machine.

The main part of hydraulic motors belongs to one of two types:

- hydraulic motors of translational movement - *hydraulic cylinders*;
- rotary hydraulic motors - *hydraulic motors*.

The so-called are used much less *rotary engines* – simplified hydraulic motors with a limited angle ϕ rotation of the shaft, that is $\phi \leq 360°$
Of course $\phi = 270°$.

Hydromotors are power units, which according to their purpose opposite to pumps. At the same time, both pumps and hydraulic motors make up a group of hydraulic machines that can be transformed into each other, that is, if a flow of mechanical energy is applied to the shaft of such a hydraulic machine, it will work in the pump mode and vice versa, if the hydraulic input (output) of the hydroma - the tires supply a flow of hydraulic energy, then the hydraulic machine will work in the hydraulic motor mode.

Thus, the scheme of a non-regulated plate pump is in principle identical to the scheme of a non-regulated plate hydraulic motor, the scheme of an axial-piston regulated pump with an inclined disk is identical to the scheme of the same hydraulic motor, the scheme of a gear pump is the scheme of a gear hydraulic motor, etc. others

Despite their fundamental identity, hydraulic motors and pumps of the same type have structural differences that are due to the peculiarities of the processes that occur in them.

Let's consider two features of the characteristics of hydraulic motors. The first of them is due to the fact that the power of the flow of hydraulic energy at the entrance to the hydraulic motor is limited both by the parameters of the hydraulic motor itself and the pump. Therefore, energy losses in the hydraulic motor, especially due to mechanical and hydraulic friction, lead to a decrease in the driving torque on the shaft, in particular, the starting torque. There are problems with starting the hydraulic motor under load, because due to the increased forces of friction at rest, the starting torques of hydraulic motors in most cases are higher than the nominal torques that develop on the moving shaft. The second feature of hydraulic motors is the difference in volumetric consumption at the inlet and outlet, which is especially noticeable in the presence of pressure drops, $\Delta p = p_1 - p_2$, between pressure p_1 at the entrance and pressure p_2 at the output of the hydraulic motor and the presence of undissolved gas in the working fluid.

To determine the torque M_{Cr} on the shaft of the hydraulic motor consider mo ratio

$$M_{Cr} = R_m / \omega, \tag{14.43}$$

where R_m – power on the hydraulic motor shaft;

ω is the angular speed of the hydraulic motor. In turn

$$P_m = R_{Mr} \cdot \eta_0 \cdot \eta_m = \Delta p \cdot Q_m \cdot \eta_0 \cdot \eta_m = (p_1 - p_2) \cdot Q_m \cdot \eta_0 \cdot \eta_m, \tag{14.44}$$

where R_{Mr} – the power of the hydraulic flow supplied to the hydraulic motor.

$$Q_m = \frac{V \cdot \omega}{\eta_0}. \tag{14.45}$$

Then, taking into account (14.43) – (14.45)

$$M_{Cr} = (p_1 - p_2) \cdot V \cdot \eta_m, \tag{14.46}$$

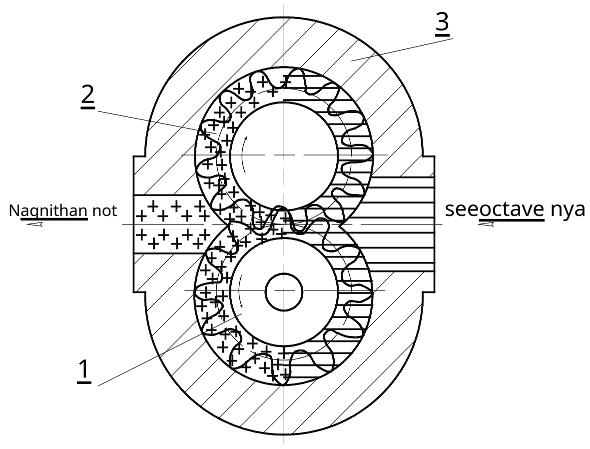
Coefficient η_m takes into account friction moment losses, which in total are equal to eat

$$M_{ter\Sigma} = M_{Cr} (1 - \eta_m) = b_m \cdot \omega + M_{ter}(\omega, p_1, p_2, \tau, M_{ter0}), \tag{14.47}$$

In equation (14.47) b_m is the coefficient of viscous friction between the parts of the hydro- motor (assuming that the viscous component $b_m \cdot \omega$ total moment $M_{ter\Sigma}$ friction is proportional to the angular speed of the shaft (hydraulic motor rotor); $M_{ter}(\omega, p_1, p_2, \tau, M_{ter0})$ is a non-linear component that depends, in general, on the magnitude and direction of the angular velocity ω shaft, pressure p_1 and p_2 in cavities, initial value M_{ter0} moment of friction, which forms is caused by various pretensions of the elastic elements of the hydraulic motor (springs, rubber seals, etc.), as well as by the duration τ preliminary contact of parts subject to mutual friction, preceding the start of the hydromotor.

14.3.1 Gear pumps and hydraulic motors

The rotating gears (Fig. 14.14), which are engaged, take the liquid into the space between the teeth and transfer it from the low-pressure zone to the high-pressure zone, injecting pressure into the pipeline. In the pump, there is almost no gap between the body and the tops of the teeth; in this way, good radial sealing is ensured. In the axial direction, sealing is provided by plates or bushings, which are pressed against the gears by pressure. These plates or bushings also act as bearings for the gears. This design makes it possible to obtain high-pressure pumps with high efficiency (Fig. 14.15). The rotation frequency of the pump gears reaches 4000 rpm^{-1}, maximum pressure – 300 bar, specific power – 6 kW/kg; these parameters make it possible to use gear pumps in automotive hydraulic systems. The performance range from 0.5 to 300 l/min has 4-5 standard sizes of pumps.



1 and 2 – spur gears with an involute profile, which are engaged and disengaged are hidden in the borings-wells of the body 3

Figure 14.14 – Diagram of a gear pump or hydraulic motor (direction fluid flow and gear rotation are shown during pump operation)

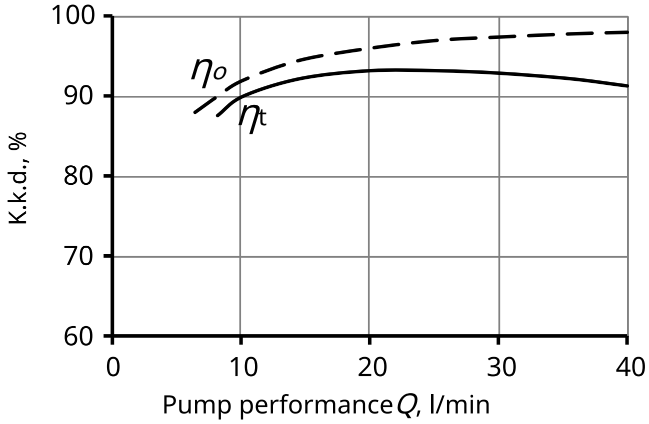


Figure 14.15 – Bosch high-pressure gear pump: dependence of volumetric efficiency η_o and full efficiency η_t from productivity at Δp=210 bar

The simplest gear hydraulic motors are similar to gear pumps: the gears rotate only in one direction, opposite to the direction in which they rotated when operating in pump mode. Hydraulic motors intended for use in automobile drives, i.e. hydraulic motors whose gears rotate in both directions and are loaded when reversing, were also designed on the basis of gear pumps with corresponding changes in axial pressure zones and a number of channels for passing the working fluid. The advantages of gear pumps with gears of external engagement (high specific power, small necessary area for placement and low cost of production) are also characteristic of gear hydraulic motors. Therefore, they are often chosen for use on automobiles, construction and agricultural machinery, street sweepers, graders, vibrators, etc.

14.3.2 Piston pumps and hydraulic motors

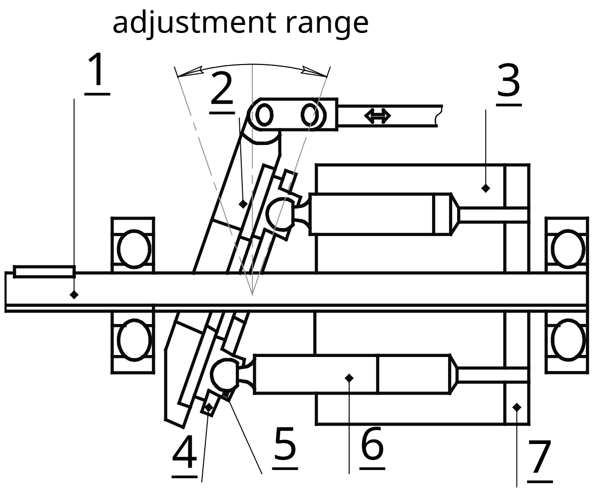
Piston pumps and hydraulic motors are significantly different from piston machines of the classical type. High pressure values (350-400 bar) lead to high forces on the piston. The good lubricating and cooling qualities of the liquid used contribute to the creation of small-sized devices. Hydraulic piston mechanisms make it possible to obtain a maximum specific power of more than 5 kW/kg.

To obtain a uniform fluid flow, hydraulic piston mechanisms are designed using an additional number of piston elements. There are mechanisms with radial and axial movement of pistons, which depends on the drive. Both types of such mechanisms are used as pumps and hydraulic motors with a constant or variable working volume, used in open and closed circuits. The working volumes of these hydraulic motors and pumps change due to changes in the stroke length of the piston. Phase adjustment of hydraulic mechanisms is not used, because the crankshaft is not suitable as a means of adjusting the stroke of the piston. An eccentric or crankshaft and an inclined washer are not suitable for adjusting the amount of stroke and are used only in some hydraulic machines with constant productivity.

Hydraulic machines with variable performance contain a rotating block of cylinders. Together with a stationary distribution disk or pin, the cylinder block forms a rotating spool distribution, which ensures the alternating release of liquid from the cylinder and its filling with this liquid. Axial-piston hydraulic machines are devices with an inclined washer (Fig. 14.16).

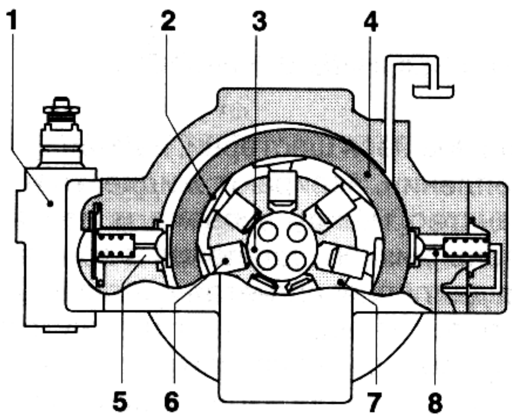
Radial-piston hydraulic machines are devices with a moving ring (Fig. 14.17). By changing the eccentricity of the movable ring with respect to the trunnion, the stroke of the piston changes. In pumps where the feed direction can be reversed, the movable ring can be moved in both directions by the control plungers. All control elements are located in the stator, which allows

allows you to quickly and accurately adjust the flow using hydraulic or electronic servo devices and regulators. The regulating piston mechanism can be incorporated into the electronic control circuit using pressure ratio valves or a servo regulator that operates depending on the fluid flow.



1 – drive shaft; 2 – inclined puck; 3 – cylinder block (rotates); 4 – retaining plate; 5 – slider; 6 – piston; 7 – distribution disk

Figure 14.16 – Axial-piston hydraulic machine (inclined washer)



1 – a control device with an open or closed circuit; 2 – sliders; 3 – distribution pin; 4 – moving ring; 5 – control plunger 1; 6 – piston with radial movement; 7 – cylinder block (rotates); 8 – control plunger 2

Figure 14.17 – Radial-piston hydraulic machine

14.3.3 Electrohydraulic pumps and small mechanisms An electrohydraulic pump is a combination of a conventional hydraulic pump and an electric motor. A direct current electric motor is used in most self-propelled machines, and an alternating current motor is used in stationary installations.

Gear pumps are characterized by a low degree of pulsations and noiseless operation. For these reasons, these hydraulic machines can be used in electro-hydraulic pumps. Such pumps of standard sizes B and F with a productivity of 1–22.5 cm^3 for 1 revolution are used to obtain pressures up to 280 bar, working together with electric motors with a nominal power of up to 8 kW; these pumps are used in a number of mobile hydraulic devices in lifting mechanisms and vehicle control systems.

On passenger cars, electro-hydraulic pumps are used in the suspension to maintain a constant level of the car body and in power steering. A special field of application of such pumps is ABS and ASR systems. The main element of the anti-lock braking system (ABS) and the traction control system (ASR) is the hydraulic device for creating pressure in these systems.

With electro-hydraulic pumps on cars, different valves must be used to perform a wide range of control functions. All this calls for the development of a small and compact device with an output power of up to 4 kW.

In such devices, the electric motor and pump are supplemented with a valve body, a tank for liquid and filters for air and liquid. The designs of these devices allow their modification to perform specific control functions. Cylindrical spool valves and valve seats can be housed in a compact valve block or combined into a system. Miniature versions of these devices are used where it is necessary to obtain high power, despite the limited space for placing the device, for example, in municipal vehicles (street sweepers, rotary snow plows, vans and trucks operating in industrial zones; special cars and passenger cars cars equipped with lifts and turning devices to facilitate the boarding of disabled people).

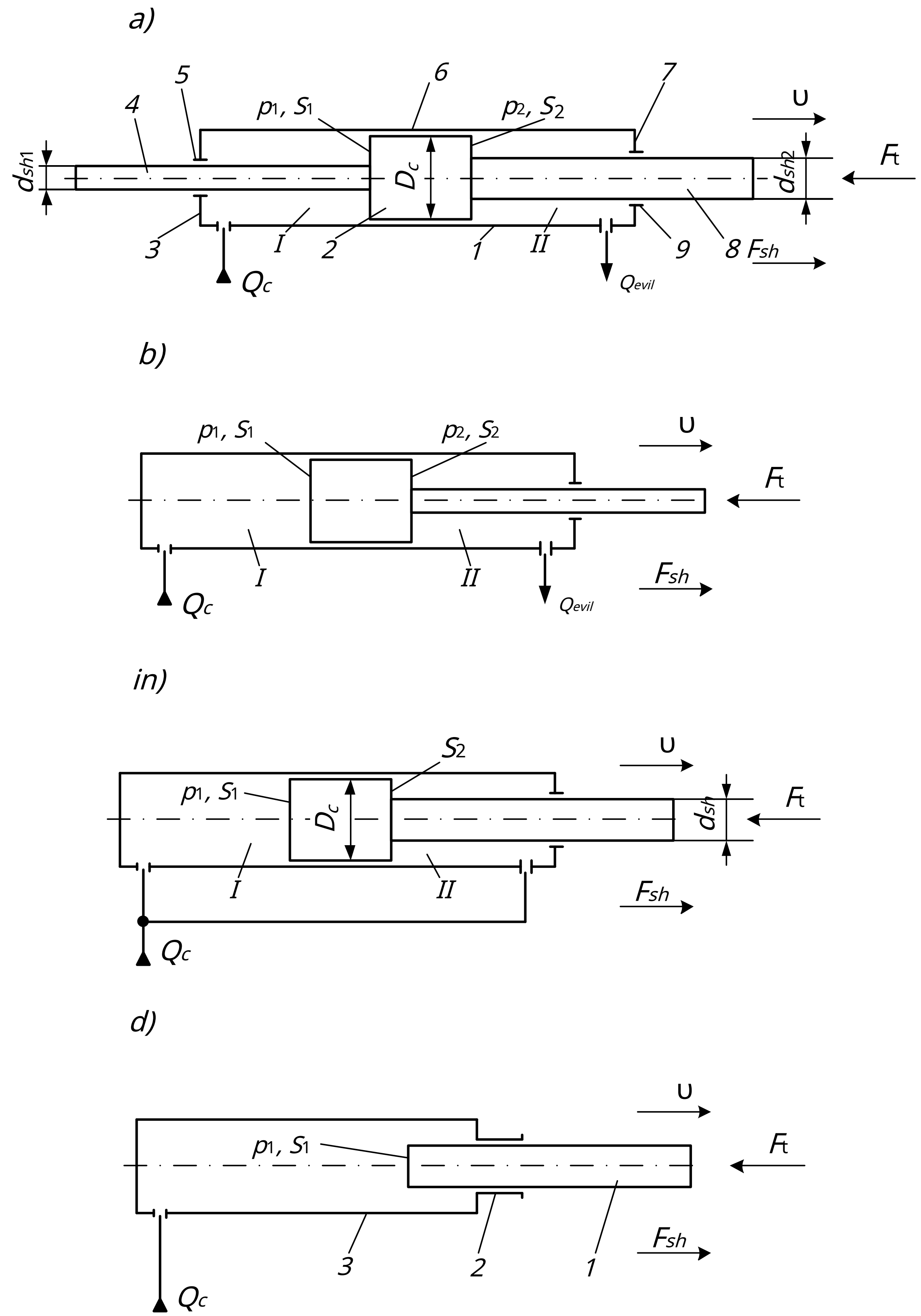
A new field of their application was found on passenger cars. In most bodies with a folding top, hydraulic mechanisms are used to ensure the movement of the roof, its folding, folding into a certain position and installation in the initial position.

14.3.4 Hydraulic cylinders

Hydraulic cylinders are translational hydraulic motors (Fig. 14.18) that convert hydraulic energy with parameters Q_c, p in mechanical with pa-

by meters F, υ.

By energy saturation (power per unit mass), the ability to develop significant forces F and speeds υ, compactness, simplicity of design, reliability, as well as due to low cost, hydraulic cylinders have no analogues among all PEs.



and– asymmetrical with a double-sided stem; *b*– asymmetrical with one-sided stem;
in– differential hydraulic cylinder; *Mr*- plunger

Figure 14.18 - Schemes of hydraulic cylinders

Hydraulic cylinders are characterized by high indicators of specific power and relative simplicity of design. The efficiency of the cylinder is determined by seals, working pressure and the quality of the piston surface. In addition to such parameters as force and speed, the longitudinal bending strength of the cylinder is also an important design criterion used to determine the dimensions of the cylinder and the length of the piston rod.

The schematic diagram of the hydraulic cylinder is extremely simple (for example, in Fig. 14.7, hydraulic motor II is a simple hydraulic cylinder, which consists of its own cylinder 8.2 and piston 8.1).

Most of the known hydraulic cylinders can be attributed to one of the schemes:

- an asymmetrical hydraulic cylinder with a double-sided rod (Fig.14.18, *and*), that is, the diameters of the rods d_{sh1} and d_{sh2} unequal ($d_{sh1} \neq d_{sh2}$); such a hydrocy-the cylinder consists of a cylinder *1*, piston *2*, flanges *3* and *7*, rods *4* and *8*, seals *5*, *6* and *9* in the places of movable landings of the rods in the holes of the flanges and pistons in the cylinder bore;

- a symmetrical hydraulic cylinder with a double-sided rod, that is, a hydraulic cylinder according to the scheme of fig. 14.18, and at $d_{sh1} = d_{sh2}$;

- an asymmetric hydraulic cylinder with a one-sided rod (Fig.14.18, b);

- differential hydraulic cylinder, similar to the diagram in fig.14.18, b, but with the ratio $d_{sh}/D_c = 0.5$ (Fig.$\sqrt{14.}$18, c);

- plunger hydraulic cylinder (Fig.14.18, d), which consists of plun-of the rod 1, which slides in the guide sleeve with a seal 2, and the cylinder 3.

Other schemes of hydraulic cylinders are also used in various branches of mechanical engineering, for example, telescopic, double, etc., many of which have found their application in the design of various devices installed, including on specialized cars.

Considering the direction of effort F_{sh} and speed υ pistons shown on rice. 14.18, it is possible to write down the equations of forces that are applied to the "piston-rod" system in steady uniform motion

$$F_{sh} - F_t - b_c\,\upsilon - F_{ter}\left(\upsilon, p_1, p_2, \tau, F_{ter0}\right) - G\cos a, \qquad (14.48)$$

where F_{sh} – the force developed by the hydraulic cylinder;

F_t – technological load on the rod;

$b_c\,\upsilon$ is the component force of viscous friction, which is proportional to the coefficient b_c viscous friction in seals and speed υ displacement of the piston;

G is the weight of nodes that are moved by a hydraulic cylinder at an angle α to the horizon

$F_{ter}\left(\upsilon, p_1, p_2, \tau, F_{ter0}\right)$ is a non-linear component of the friction forces, which depends on magnitude and direction of speed υ, pressures p_1 and p_2 in the cavities of the hydraulic cylinder, the time τ, the height of the piston before the start of movement, as well as from the chattel force of friction F_{ter0}, which is formed, for example, by tension during the data of sealing elements with piston, rod, etc. p.

Given that the force developed by the hydraulic cylinder is equal to

$$F = p_1 \cdot S_1 - p_2 \cdot S_2, \tag{14.49}$$

where S_1 and S_2 – effective areas of the piston in the first and second cavities of the guide rocylindra, moreover $S_1 = (\pi/4)\cdot\left(D_c^2 - d_{sh1}^2\right)$, a $S_2 = (\pi/4)\cdot\left(D_c^2 - d_{sh2}^2\right)$, found we have from (14.48) and (14.49) the equation for the useful effort F_{sh} on the rod

$$F_{sh} = F_t = p_1 \cdot S_1 - p_2 \cdot S_2 - b_c \cdot \upsilon - F_{ter}\left(\upsilon, p_1, p_2, \tau, F_{ter0}\right) - G\cos a. \tag{14.50}$$

Equation (14.50) allows you to determine the useful force on the rod and for all other schemes of hydraulic cylinders in Fig. 14.18.

At the same time, for the scheme in fig. 14.18, *in* it should be taken into account that the pressure in the two cavities is the same and equal p_1, and for the scheme in fig. 14.18, *Mr* greatness-on $p_2 \cdot S_2 = 0$.

Taking into account the direction (signs) of speed υ and weights G, as well as the difference sign $p_1 \cdot S_1 - p_2 \cdot S_2$ gives a sign, that is, the direction of effort F_{sh}.

The speed of the piston for the schemes in Fig. 14.18, *and* and 14.18, *b*, if the working fluid is fed into the cavity *AND*, is equal to

$$\upsilon_1 = \left[Q_c - p_1 \cdot \sigma_1 - \sigma_{1-2} \cdot (p_1 - p_2)\right] S_1^{-1}, \tag{14.51}$$

where Q_c – fluid flow at the exit to the cavity *AND*;

σ_1 – leaks of liquid into the atmosphere from the cavity *AND*;

σ_{1-2} is the coefficient of liquid flow from the first cavity of the hydraulic cylinder ra in the second

When supplying liquid into the cavity *II* the speed is equal to

$$\upsilon_2 = Q_c \cdot S_2^{-1}, \tag{14.52}$$

not taking into account the above relative value σ, for the scheme in fig. 14.18, *Mr* the speed is equal to

$$\upsilon_{sq} = Q_c \cdot S_1^{-1}. \tag{14.53}$$

For differential connection according to the diagram in fig. 14.18, *in*, taking into account that

$$d_{sh} = D_c \cdot \sqrt{0.5}, \text{ that is } d_{sh}^2 = 0.5 \cdot (\pi/4) \cdot D_c^2$$

or

$$S_2 = 0.5 \cdot S_1 \text{ and } S_2 / S_1 = k = 0.5,$$

we will get

$$\upsilon_{diff} = 2 \cdot \left(Q_c / S_1\right) = Q_c / S_2 = \upsilon_2 = 2 \cdot \upsilon_1, \tag{14.54}$$

where υ_2 is calculated according to (14.52), a υ_1 – according to (14.51).

Thus, with the differential inclusion of the hydraulic cylinder with $k = S_2 / S_1 = 0.5$, you can get three different in magnitude or direction speed of movement at the same flow of liquid at the entrance to the cylinder:

− movement with speed $+\upsilon_1$ in this (see Fig. 14.18, *in*) directions;

− movement at speed $-\upsilon_2 = -2\cdot\upsilon_1$;

− movement with speed $+\upsilon_{diff} = +2\cdot\upsilon_1$.

Brought to the cavity *AND* hydraulic power of the cylinder is equal to

$$P_{c_{entrance}} = Q_{c1}\, p_1 \tag{14.55}$$

The hydraulic power of the liquid squeezed out of the hydraulic cylinder equalizes

$$P_{c_{exit}} = Q_{c2}\, p_2 \tag{14.56}$$

The power of the flow of hydraulic energy, which is realized on the piston of the hydraulic cylinder, is equal to

$$P_{c_{Mr}} = Q_{c1}\cdot p_1 - Q_{c2}\cdot p_2. \tag{14.57}$$

The power of the flow of mechanical energy on the rod

$$P_{c_{fhe}} = F_{sh}\cdot\upsilon \,. \tag{14.58}$$

Efficiency of the driving mode η_{dir} piston, which takes into account energy losses, caused by due to the presence of back pressure p_2 (at $p_1 > p_2$),

$$\eta_{dir} = \frac{P_{c_{Mr}}}{P_{c_{entrance}}} = 1 - \frac{p_2}{p_1}\cdot\frac{Q_{c2}}{Q_{c1}} = 1 - \frac{p_2}{p_1}\cdot\frac{S_2}{S_1}. \tag{14.59}$$

Full cylinder efficiency η_c is equal to

$$\eta_c = \frac{P_{c_{fur}}}{P_{c_{entrance}}} = P_{c_{fur}} = \frac{F_{sh}\cdot\upsilon}{p_1 Q_{1c}} = \frac{F_{sh}\cdot\upsilon}{p_2 Q_{c2}}\cdot\left(1-\eta_{dir}\right), \tag{14.60}$$

given that it follows from (14.59). $p_1\cdot Q_{c1} = \left(p_2\cdot Q_{c2}\right)/\left(1-\eta_{dir}\right)$.

14.4 Hydraulic units serving the executive circuit

Hydraulic aggregates, which ensure the functioning of the VC, but do not control its mode of operation, belong to service devices.

An example of such service devices is *filters*, which ensure cleaning of the working fluid from contamination. The main characteristic of the filter is the accuracy of filtration, that is, the maximum size of particles in micrometers that the filter element passes. Equally important is the amount of pressure loss in the filter and the maximum allowable flow of liquid at the same time.

In the characteristics of the filter, there is almost never an indication of its capacity, that is, the total volume or mass of pollution particles that can linger on the filter element without increasing the pressure loss beyond the permissible level.

The main connections of the filter are shown in the hydraulic diagrams of fig. 14.19.

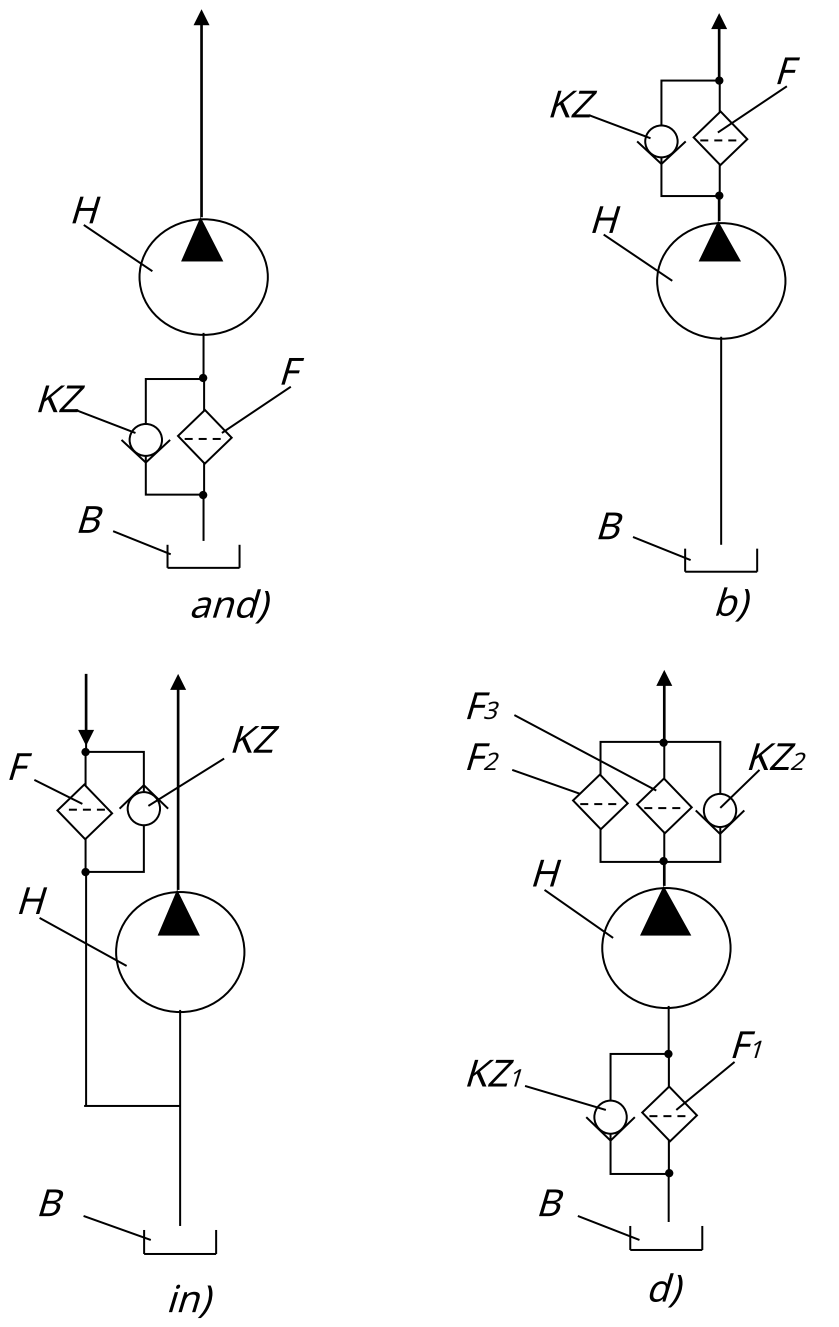


Figure 14.19 – Diagram of connecting filters to the hydraulic system

In the diagram in fig. 14.19,*and*liquid from the tank*B*through the receiving filter *F* absorbed by the pump*N*and is then pumped into the hydraulic system. Valve*KZ* ensures the passage of part of the liquid flow into the system when the filter is clogged and limits the pressure drop on it. The advantage is the capture of particles before they can get into the pump, which increases its service life. The disadvantage is the formation of additional resistance on the suction line. Due to the fact that the liquid moves through the suction pipeline due to the difference between atmospheric pressure and the vacuum created by the pump, the allowable pressure drop between the suction pipeline and the receiving filter is insignificant, no more than 0.020...0.025 MPa. Therefore, working windows of the filter element through which the liquid passes must be large enough and the fineness of filtration at the pump inlet cannot be less than 80...160 microns.

In the diagram in fig. 14.19,*b*liquid from the tank*B*and the pump*N*through the filter*F* enters the hydraulic system, i.e. the filter is installed in the injection line, the valve *KZ*limits the pressure drop across the filter.

Since the pressure reserve on the injection line is quite large, it is possible to allow a greater amount of pressure loss on the filter than in the case of installing a filter on the suction and, accordingly, to ensure a significant fineness of cleaning.

The disadvantage is that the housing and all elements of the filter must be designed for the maximum pressure in the hydraulic system, which causes an increase in their mass.

The following are used as filter elements in pressure filters: a set of plates, into the gaps between which liquid is squeezed and particles whose size is equal to or larger than the size of the gap are retained (fineness of filtration 80...12.5 μm); filter elements - from corrugated cardboard (fineness of filtration 10, 25, 40 microns); magnetic catchers, as well as various combinations of the listed elements.

Installation of the filter on the drain is shown in fig. 14.19,*in*, where the liquid supplied by the pump*N*from the tank*B*, passes through the hydraulic system, and on the drain through the filter*F*, parallel to which the bypass valve is installed*KZ*. Drain filters also have clogging alarm systems and ensure a filtration fineness of 25-40 microns.

The schemes for connecting filters are far from being exhausted by the above. It is possible, in particular, to use the combined proportional cleaning scheme shown in Fig. 14.19,*Mr*. Pump*N*sucks liquid from the tank*B*through the receiving filterF_1, parallel to which the valve is installedKZ_1, and feeds it into the hydraulic system through two filters installed in parallelF_2andF_3
coarse and fine cleaning with a valveKZ_2.

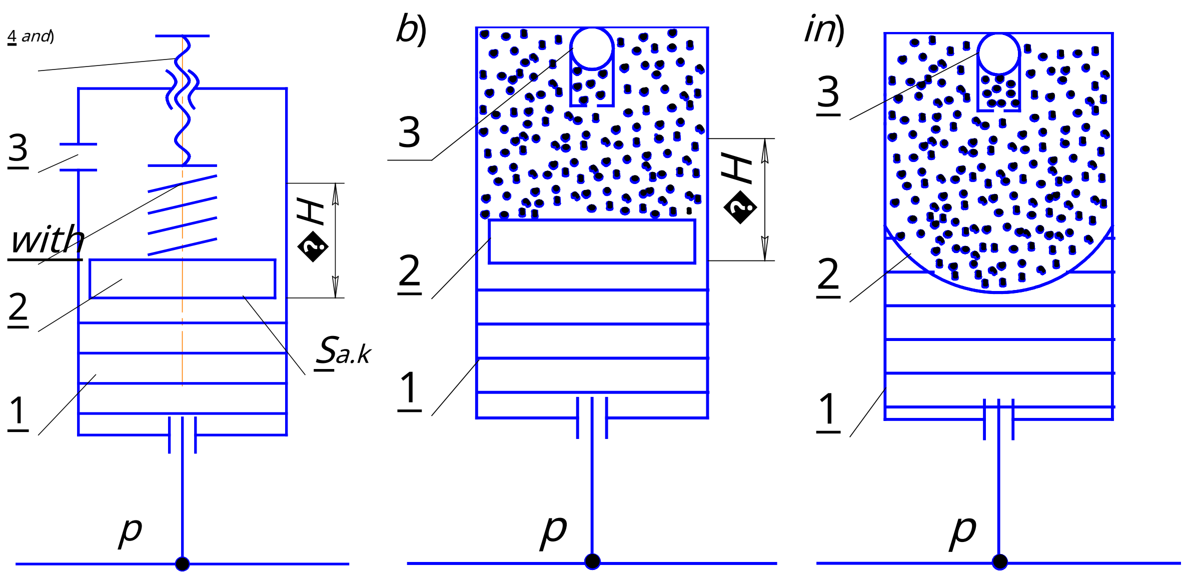
Another example of service devices are tanks - containers that contain the working fluid and provide a number of important conditions for the normal operation of the VC. In particular, the capacity of the tank should take into account changes in the volume of liquid contained in the hydraulic system in different phases of its operation. Thus, when using asymmetric or plunger hydraulic cylinders, the volume of liquid contained in its cavities can change significantly, which is compensated by the corresponding capacity of the tank.

The capacity of the tank significantly affects the temperature regime of the hydraulic drive. If there is no thermoregulation system, stabilization of the temperature regime is possible only by choosing a rational volume of the tank and the working liquid contained in it.

In addition, the design of the tank should provide a rational direction of liquid flow, which will contribute to the mixing of volumes with different temperatures, removal of undissolved air from the free surface, suppression of foaming and ease of cleaning and draining the tank.

14.5 Hydraulic accumulators

Hydraulic accumulators(rice.14.20) are devices that accumulate a part of the energy of the fluid flow at the pump outlet, which is not used by the hydraulic motor in certain modes of its operation, and then give this energy to the hydraulic motor in those modes when the nominal power of the pump becomes insufficient. For example, at low speeds of the hydraulic motor, when the fluid consumption from the pump is significantly less than its consumption, the excess supply can be fed into the hydraulic accumulator. During fast movements, when the pump supply is not enough to ensure high speeds of the hydraulic motor, the accumulator releases energy, increasing the flow of working fluid supplied to the hydraulic motor.



and– elastic piston;*b*– piston pneumohydraulic accumulator;
in– membrane pneumohydraulic accumulator

Figure 14.20 - Schemes of hydraulic accumulators

The purpose of hydroaccumulators is to store energy, dampen fluid pulsations, and perform the functions of an elastic element.

In VC hydraulic drives, accumulators of potential energy are used, which is stored in compressed springs, compressed gas, or in raising the mass to a certain height.

In fact, the accumulation and subsequent release of energy also occurs in the working fluid due to its compressibility, especially in the presence of undissolved gas in it, as well as in pipelines due to their deformations under pressure, primarily with high-pressure hoses. This accumulation significantly affects the dynamic processes in the VC.

Hydraulic accumulators are mainly divided into two large groups: spring (diagram in Fig. 14.20,*and*) and pneumohydraulic (schemes in Fig. 14.20,*b*and 14.20,*in*). The latter can be pistons (Fig. 14.20,*b*), when gas and liquid are separated by a piston, and membrane (Fig. 14.20,*in*), when the separating element is a membrane.

During the working stroke of the pistonΔHpressure$p$in the hydraulic accumulator changes by size

$$\Delta p_{ac} = \frac{with \cdot \Delta N}{S_{ac}}, \tag{14.61}$$

where*with*– stiffness of the spring;

S_{ac}– area of the accumulator piston.

Thus, if we denote the initial pressurep_0, after which begins-charging, the maximum pressure will be equal to

$$\Delta p_{\max} = p_0 + \Delta p_{ac} = p_0 + \frac{with \cdot \Delta N}{S_{ac}}. \tag{14.62}$$

The magnitudeΔp_{ac}adjusted by compressing the spring*with*by means of screw 4. In most hydraulic accumulators, the pre-compression of the spring is a constant value. Hole 3 in housing 1 exists for connection with the atmosphere of the spring cavity. Piston 2 is sealed as in hydraulic cylinders.

In pneumohydroaccumulators, piston 2 separates liquid and gas, which is supplied during charging through check valve 3 in housing 1. Initial pressurep_0 nitrogen and liquid is the same. When moving the piston by an amount $\Delta H$the initial volume of the gas decreases by an amount$\Delta V = \Delta H \cdot S_{ac}$. If this movement is slow, that is, such that the processes in the gas can be considered isothermal, that is

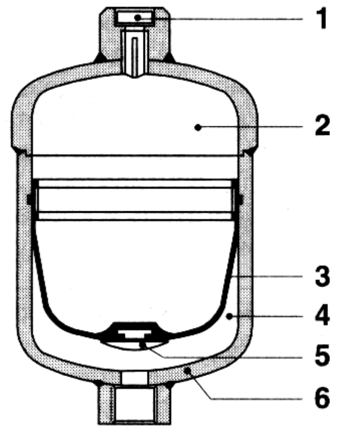
$$p_0 \cdot V_0 = (p_0 + \Delta p_{ac}) \cdot (V_0 - \Delta V) \tag{14.63}$$

where

$$\Delta p_{ac} = p_0 \cdot \frac{\Delta V}{V_0 - \Delta V} = p_0 \cdot \frac{1}{\dfrac{V_0}{\Delta H \cdot S_{ac}} - 1}. \tag{14.64}$$

In pneumohydroaccumulators with membrane 2 (see Fig. 14.20,*in*) gas and liquid in housing 1 interact in the same way as in the design with a piston, and gas charging occurs through valve 3. The design of such a pneumohydroaccumulator is shown in Fig. 14.21.

Nitrogen is used as a gaseous medium in hydraulic accumulators. During operation, the liquid pressure is transferred to the gas, compressing it. Minimum working pressure$p_1$should be at least 10% higher than the initial gas pressure p_2. The ratio of the initial gas pressure to the maximum should not change exceed 1.4 for a battery with an elastic cylinder or diaphragm; 1:10 – for piston.

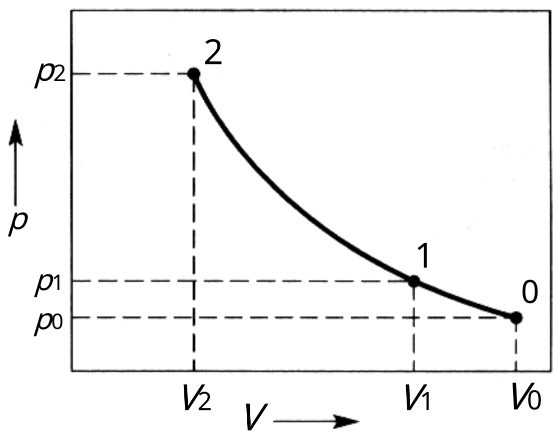


1 – screw cap; 2 – gas chamber; 3 – diaphragm;
4 – liquid chamber; 5 – plug; 6 – steel case

Figure 14.21 – Diaphragm pneumohydraulic accumulator

The three operating states shown in the diagram (Fig. 14.22) are determined by the laws of polytropic changes:

$$p_0 \cdot V_0^n = p_1 \cdot V_1^n = p_2 \cdot V_2^n. \qquad (14.65)$$



p_0 – initial gas pressure; p_1 – minimum working pressure; p_2 – maximum working pressure; V_0 – volume of gas at initial pressure; V_1 – volume at minimum working pressure; V_2 – volume at maximum working pressure

Figure 14.22 – Diagram of working states of the hydraulic accumulator

When using nitrogen, the polytropic exponent p = 1 for isothermal changes of state and p = 1.4 for adiabatic changes.

14.6 Hydraulic equipment

Hydraulic equipment provides control of energy flow conversion modes in volumetric hydraulic drives. At the same time, the parameters of the flow of mechanical energy at the output of the hydraulic drive are adjusted.

The principle of regulation is that special working windows (active hydraulic supports) are installed in certain live cross-sections of the mains of the hydraulic system, the size of the live cross-section of which is adjusted discretely or continuously with the help of hydraulic apparatus mechanisms. In this regard, all hydraulic devices with discrete adjustment of the area of the working windows belong to the group*guide hydraulic devices*(NGA), and with non-interruptible - to the group*regulating hydraulic devices*(RGA). Among the latter, we also include those hydraulic devices whose working windows are permanent hydraulic supports.

In NGA, as a rule, the area of working windows varies discretely from zero to the maximum possible, that is, the corresponding passage for the working fluid is either closed or open. Such NGAs include*hydraulic distribution who*of various designs, non-return valves, etc. Their main purpose is carry out switching of pipelines according to the scheme.

According to their design features, all hydraulic distributors can belong to one of the groups:

- spool distributors, in which closing and opening of working valves kon is carried out by the translational movement of the locking element - spool; the combination of open and closed windows in each fixed position (position) of the spool provides the necessary connection (disconnection) scheme of the pipelines that are connected to the distributor channels;

- valve distributors, in which closing and opening of working windows is carried out by angular movement (rotation) of the closing element of the faucet relative to its axis, while some holes are connected to each other, and others are disconnected;

- valve distributors, in which the opening and closing of working windows carried out by valves.

Various designs belong to RGA*pressure regulators*(safety, safety-overflow and reduction valves),*flow regulators, propor- tion hydraulic distributors, throttle flow synchronizers*.

Pressure regulation in hydraulic drives basically boils down to the following modes:

− protection of the hydraulic system from the action of pressure that exceeds the permissible value chenie, it is carried out*safety valves*, which operate episodically, in emergency modes, and*pressure regulators*general purpose;

- stabilization of a constant pressure value in the pressure line, i.e at the output of the pump, due to the operation of the pressure regulator in the mode of constant draining of the part of the pump supply to the tank; it is carried out*overflow valves*, a due to the fact that they simultaneously perform the functions of safety valves, it is better to call them*safety and overflow valves*;

− pressure reduction in branches from the pressure main, i.e. sta- stabilization of a constant value of pressure in some parts of the hydraulic system, which is lower than the pressure at the pump outlet; it is carried out*reducing valves*;

- a combination of the functions listed above with various modifications of al- pressure control algorithms.

When considering pressure regulators as automatic control systems, they can be divided into 2 groups:

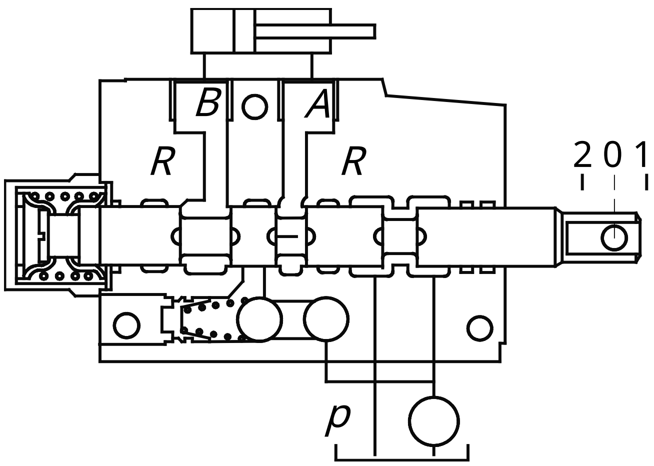
− pressure regulators with one stage of amplification, which are called*class gentlemen of direct action*or*pressure valves*;

- pressure regulators with two stages, when the first stage is small of power controls a powerful second cascade that regulates the main flow of hydraulic power; they are called*valves of indirect action*.

14.6.1 Distribution valves

Distribution valve with open center. In the neutral position, the flow of liquid from the pump passes freely to other valves and to the drain. Fluid consumption is limited by moving the valve spool before the line leading to the servo is opened (Fig. 14.23).

Disadvantages: high pressure losses; adjustment accuracy is achieved due to load pressure.

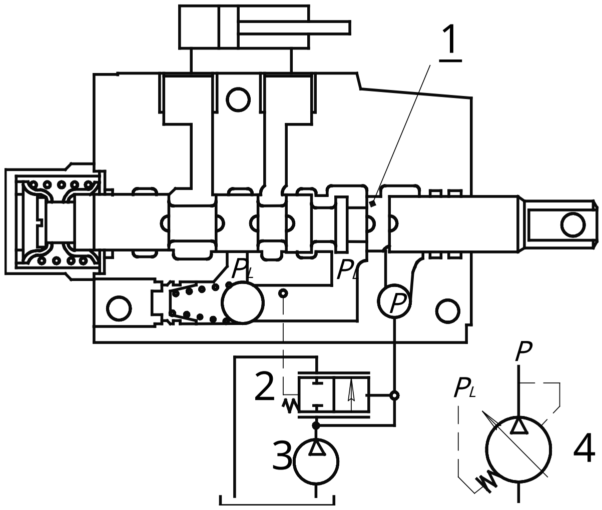


1 – fluid flow in neutral position

Figure 14.23 – Open-center distribution valve

Distribution valves sensitive to load changes. In the neutral position of the valve, the control line allows you to remove pressure from the pump and the pressure compensator. The pump controller and pressure compensator maintain the pressure difference near the cylindrical spool at a constant level (Fig. 14.24).

Advantages: minimal pressure loss in neutral position; increased accuracy of control, which does not depend on the load pressure.



1 – measuring diaphragm; 2 – pressure compensator; 3 – pump with constant flow; 4 – pump with variable flow rate

Figure 14.24 - Load-sensitive diverter valve

14.6.2 Flow regulators

*Throttle valves*are used to regulate fluid flow by changing the cross-sectional area of the flow. This function depends on pressure (Fig. 14.25). Regulation independent of fluid pressure requires the use of control valves.

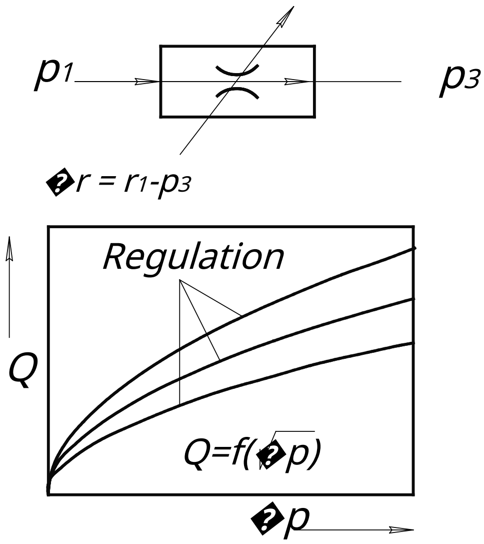
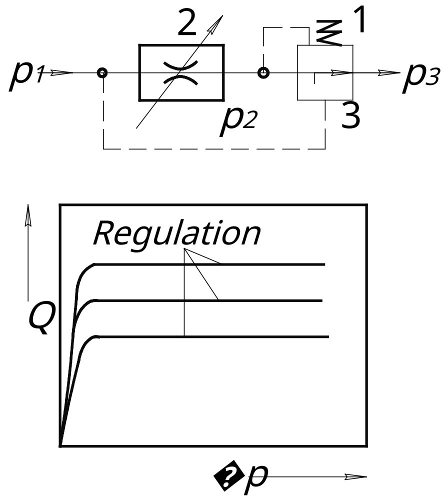


Figure 14.25 – Regulation of liquid flow by a throttle valve

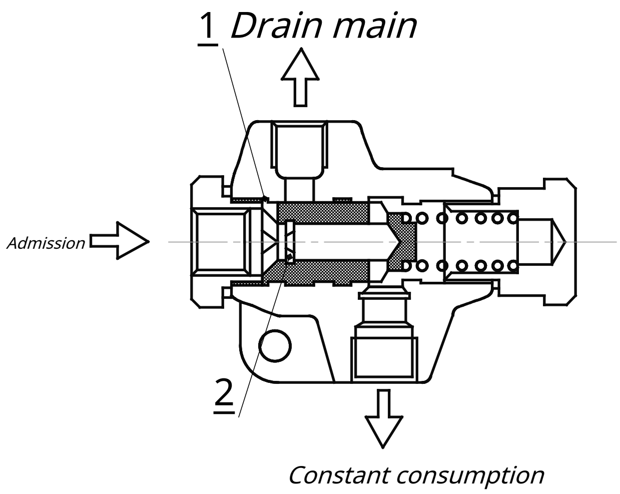
Flow regulators with pressure compensators(rice.14.26). In order to be able to determine the flow of liquid Q regardless of pressure load p_3 when using a two-way regulator valve consumption, pressure difference in the dosing hole $p_1 - p_2$ is kept on the post at the same level with a variable cross-section (pressure reduction). With this type of control, an excess amount of liquid is fed into the system through a safety valve.



1 – spring; 2 – dosing hole; 3 – adjusting throttle (pressure compensator)

Figure 14.26 - Two-way flow regulator

Costs can be reduced by using three-way valves-flow regulators (Fig. 14.27), in which additional outlets are provided, through which the excess amount of liquid falls back into the tank.



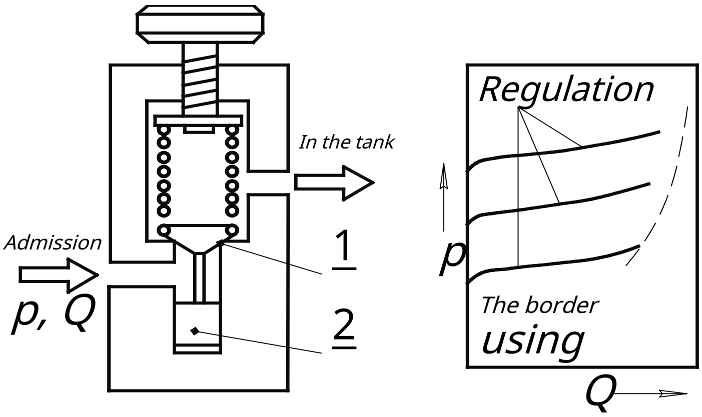
1 – pressure compensator; 2 – measuring diaphragm

Figure 14.27 – Three-way flow control valve

14.6.3 Pressure regulators

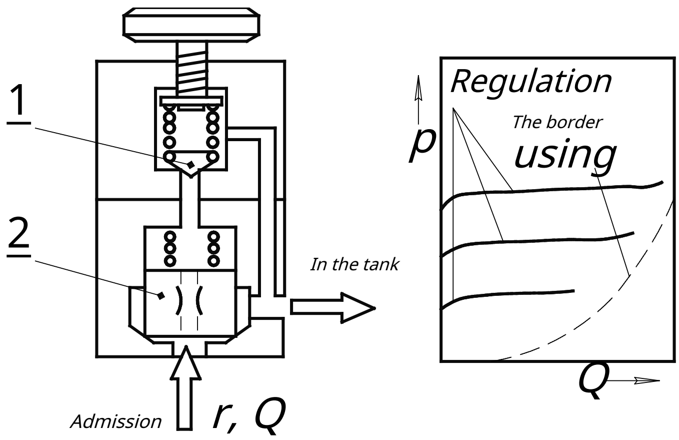
Safety valve(rice.14.28). If the pressure acting on the locking element on the seating diameter creates a force equal to the force of the pre-compressed spring acting on this element, then the latter rises from its seat and the liquid flows into the tank. Safety valves with control valves (Fig. 14.29) are used to obtain large flows.

Reducing valve. It allows to reduce the pressure, which ensures the calculated load in the system.



1 – valve seat; 2 – damping piston

Figure 14.28 - Safety valve



1 – control valve; 2 – the main valve

Figure 14.29 - Safety valve with control valve

Sequence valve. Ensures the passage of working fluid into the main circuit until its pressure reaches a certain value, after which the valve passes this fluid into the auxiliary circuit.

14.6.4 Check valves

Check valves allow you to maintain pressure in the system by counteracting external forces and the force of weight.

Shuttle valve(rice.14.30). With the help of a shuttle valve, a selection of two values of the highest pressure and the supply of liquid under this pressure to the outlet are ensured. The valve closes the supply to the outlet of the liquid under low pressure.

Check valve(rice.14.31). A simple non-return valve ensures the passage of liquid in one direction only and blocks its passage in the opposite direction.

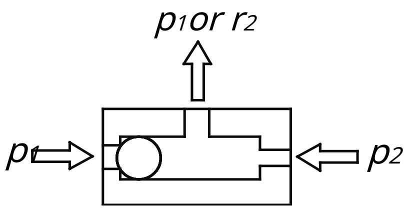
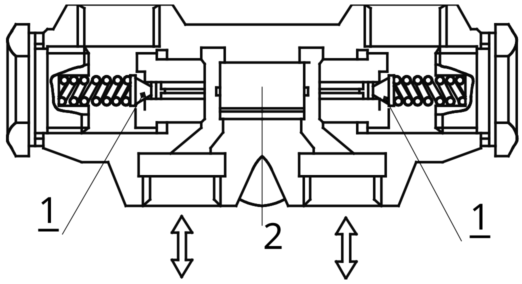


Figure 14.30 – Shuttle valve



1 – valve seat; 2 – piston of disconnection

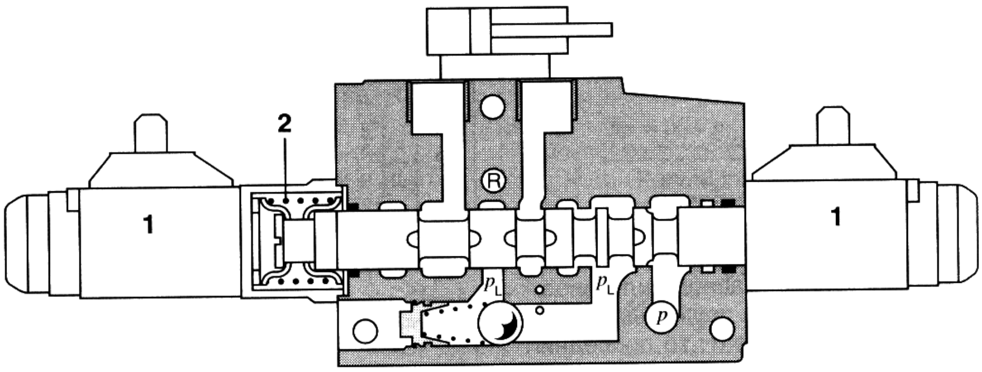
Figure 14.31 - Double check valve with hydraulic control

When performing load-holding functions, the non-return valve must be opened to reduce pressure. In hydraulic locks, this is achieved by using a mechanical, hydraulic or electric drive that lifts the locking element from its seat.

14.6.5 Electric distribution valves

The advantages of such valves are their ease of installation and use.

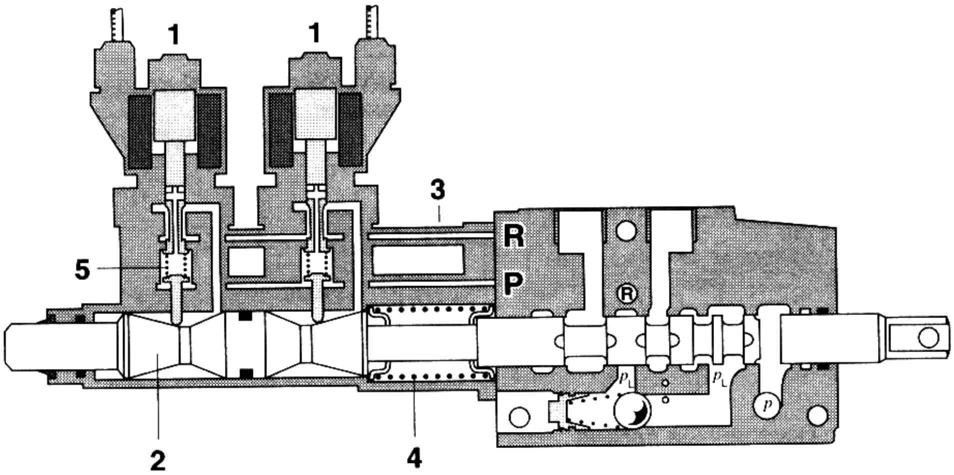
Direct electromagnetic control(rice.14.32). When moving, the traction electromagnet moves the valve spool to the appropriate distance, compressing the spring. The limit conditions of use are determined by the solenoid effort and correspond to, for example, 30 l/min and 200 bar. With a small stroke of the spool, the application of this principle is limited to three-position valves.



1 – traction electromagnet; 2 – rotary (centering) spring

Figure 14.32 – Direct acting solenoid valve

Electrohydraulic control of the distribution valve(rice.14.33). The electro-hydraulic servo mechanism creates large forces to obtain high switching power. The valve plunger is controlled by two type 2/3 solenoid valves. The operation of the valve begins when the current is applied to the control valve and the spool acts on the spring; at the same time, the valve plunger is activated. The copying finger presses the spring through the turning ball. When the spring and solenoid forces are equal, the control spool returns to the locking position and the plunger stops moving. The balancing effect completely eliminates the influence of obstacles, for example, forces created by the fluid flow. The flow of liquid with a flow rate proportional to the current in the solenoid has a hysteresis of less than 3%, while the positioning time of the plunger is less than 0.1 s.



1 – control valve; 2 – plunger of the main valve; 3 - leading and reverse master-li for liquid; 4 – rotary (centering) spring; 5 – position spring

Figure 14.33 – Distribution valve with an electro-hydraulic drive

14.7 Calculation of the parameters of the executive circuit of the hydraulic drive

The selection of the normalized value is part of the calculation of the VC parameters of the hydraulic drive*working pressure*in accordance with the effort or moment that the hydraulic motor must provide, as well as the definition*head-engine parameter*(of the effective area of the hydraulic cylinder piston or ro- side volume of the hydraulic motor) and the choice of pump size. After the calculation of the pipelines and the final selection of the standard sizes of all hydraulic units, both main (hydraulic motor and pump) and maintenance (filters, tanks, etc.), the dynamic characteristics of the hydraulic system can be calculated, if necessary.

14.7.1 Output data for calculation

The technical task (TOR) for design may specify:

− the nature of the movement of the executive hydraulic motor (progressive or rotary linear, continuous or discrete, vibrating, synchronized or tracking), as well as the direction of working and idle strokes;

− operating cycle (cyclogram) of the hydraulic drive being designed;

- characteristics of the technological machine for which the VC is designed hydraulic drive (general contours and dimensions; location of the VC on the machine; possible places of installation of the pumping unit; distance intervals between the pumping unit and the hydraulic motor; permissible dimensions and weight of the hydraulic tank with working fluid);

− energy source for driving the pump; characteristics of the energy source giyas;

- state, industry or special regulatory materials (meaning working pressures; standard sizes of hydraulic motors, pumps; standard sizes of hydraulic units and equipment).

The analysis of input data allows forming the structure of VC, using, for example, typical diagrams of hydraulic drives.

For VC with a hydraulic motor of translational movement - a hydraulic cylinder - in the case of the most common mode of operation - uniform movements of the piston during working and idle movements, the intervals of working efforts must be indicated F_{sh_i} and required speeds υ_i piston at these intervals. These data can to be specified by a table or a graph. In addition, speeds are indicated υ_{xx} idling, as well as distance F_{p_i} workers and F_{x_i} bachelors small town

For VC with a hydraulic motor of rotational movement - a hydraulic motor - similarly to the above for a hydraulic cylinder, the intervals are given $M_{kp_i}...M_{kp_{i+n}}$, as well as corresponding to them $\omega_i...\omega_{i+n}$.

In the case of calculating the drive for slow movements or very small discrete movements, which imposes additional specific requirements on the design of the drive, the parameters of these operating modes are specified.

In those cases when, as a result of the calculation, it is necessary to obtain an extreme value of any VC quality indicator, optimization is carried out. At the same time, the input data for calculating the VC should include an optimization criterion, for example, the minimum weight of the VC, the minimum cost of the VC, the maximum overall efficiency of the VC, as well as other conditions that are related to the specifics of the operation of the technological machine for which a hydraulic drive is designed. In particular, in the process of further calculations, which include the analysis and synthesis of dynamic characteristics, the tasks of dynamic stability, oscillation, speed, etc. can be solved.

14.7.2 Calculation of VC with a hydraulic cylinder

When calculating the VC with a hydraulic cylinder, the hydraulic cylinder scheme is selected depending on the given operating cycle. Thus, with a symmetric working cycle, the diagram shown in Fig. 14.18,*and*, with an asymmetric cycle the scheme according to fig. 14.18,*b*, with an asymmetric cycle with a load in one direction - the diagram in Fig. 14.18,*Mr*.

Depending on the adopted scheme, the effective area of the piston of the hydraulic cylinder is determined. For this, the equation of the forces applied to the piston of the hydraulic cylinder is drawn up. For example, for the scheme of the hydraulic cylinder in Fig. 14.18,*b* equation (14.50) can be used.

Let us present equation (14.50) in the form

$$p_1 S_1\left(1-\frac{p_2\cdot S_2}{p_1\cdot S_1}\right)=F_{sh\max}\cdot\left(1+\frac{b_c\cdot \upsilon+F_{ter0}}{F_{sh\max}}\right)+G\cos\alpha, \qquad (14.66)$$

where $F_{sh\max}$ – maximum effort on the rod.

Let's mark $\frac{S_2}{S_1}=k_c$; $\frac{p_2}{p_1}=k_p$; $\frac{b_c\upsilon+F_{ter0}}{F_{sh\max}}=k_b$.

WARNING! Coefficient k_c determines the asymmetry of the hydraulic cylinder and the use of is drawn to determine the cross-sectional area of the rod $S_{sh}=S_1-S_2=S_1\cdot(1-k_c)$; $k_c=0.5...0.85$.

Coefficient k_p determines the pressure p_2 as part of the pressure p_1, $p_2=k_p\cdot p_1$. Pressure at the entrance to the hydraulic cylinder differs from the pressure at the pump outlet, because there are frictional pressure losses in the discharge line, as well as pressure losses in local supports, which include hydraulic units located in the discharge line (filters, hydraulic distributors, throttles etc.).

In turn, pressure losses in the drainage main (both on the road and in local supports) form a back pressure p_2.

Determining the coefficient k_p, we are the same *previously* we determine which one part of the pressure p_1 will be lost in the VC hydraulic system in both the injection and drain lines; in the first approximation $k_p=0.10...0.25$.

Coefficient k_b determines the forces of viscous and dry friction as part of the force $F_{sh\max}$ payload; for preliminary calculation, you can use to take $k_b=0.08...0.15$.

Taking into account the accepted notations from equation (14.66) at $G\cos a=0$ we will get

$$S_1=\frac{F_{sh\max}\cdot(1+k_b)}{p_{1p}\cdot(1-k_c\cdot k_p)}. \qquad (14.67)$$

To define S_1 according to equation (14.67), it is necessary to select the pressure p_{1p}, using the table for this. 14.7 (pressure *depends on the value* $F_{sh\max}$).

Similarly, calculation equations for hydraulic cylinders according to other schemes of Figure 14.18 are made.

Table 14.7 – Determination of parameters of hydraulic cylinders

F_{sh}, N	10^3 - 10^4	10^4 - 10^5	10^5 - 10^6	10^6 - 10^7
p_{1p}, MPa	10	21	32	50
D_c, mm	40 - 100	80 - 250	200 - 600*	160 – 500*

* two or more hydraulic cylinders operating in parallel can be used by summing up these efforts and correspondingly reducing the diameters D_c each of them.

WARNING! The above calculation is performed for the navan mode pulling the hydraulic cylinder with the maximum useful effort $F_{sh\,max}$. This means tea that for all other values $F_{sh} < F_{sh\,max}$ mode of operation is provided appropriate adjustment of the regulation equipment.

Next, the value is determined $D_{c\,dis}$ diameter of the hydraulic cylinder. At the same time value k_c can be established as a result of a previous calculation rod in tension, compression or longitudinal bending.

Then

$$D_{Cdis} = \sqrt{\frac{4}{\pi} \cdot S_1}. \qquad (14.68)$$

The resulting value D_{Cdis} is rounded to the standard value D_{Cnorms} by a number of standard values, or, in the case of using a serial design, according to the data of the industry catalog.

Next, the diameter of the rod is calculated

$$d_{sh} = \sqrt{\frac{4}{\pi} \cdot S_{sh}}, \qquad (14.69)$$

where $S_{sh} = S_1 \cdot (1 - k_c)$

and its standard (normalized) size is set $d_{sh\,norms}$. Go away the actual value of the effective area is calculated $S_{1\,fact}$ piston

$$S_{1\,fact} = \frac{\pi}{4} \cdot \left(D^2_{Cnorms} - d^2_{shnorms}\right). \qquad (14.70)$$

In the stemless cavity they accept $d_{sh} = 0$. When getting the values D_{Cdis}, far from standard $D_{c\,norms}$, which are listed in industry catalogs, and also technologically difficult to implement or structurally unacceptable, the value is revised p_{1p}, or constructive hydraulic cylinder diagrams and repeated calculations according to formulas (14.67)-(14.70).

14.7.3 Calculation of VC with a hydraulic motor

In the preliminary calculation, the characteristic volume is determined $V_p{'}_{lake}$, or same working volume V_{dis}=2·π· $V_p{'}_{lake}$ hydraulic motor. It is used for this equation (14.46) of moments on the shaft of the hydraulic motor taking into account the losses according to equation (14.47) and is reduced to a form similar to equation (14.66). Then the value V_{dis} is equal to

$$V_{dis} = 2\cdot\pi\cdot \frac{M_{Cr\max}\cdot\left(1+k_{bm}\right)}{p_{1p}\cdot\left(1-k_p\right)}. \tag{14.71}$$

Size p_{1p} working pressure is selected from the normal series of for this type of hydraulic motor. Table 14.8 shows approximate values p_{1p}, V' and V, corresponding to different intervals of values M_{Cr}.

Table 14.8 – Determination of hydraulic motor parameters

M_{Cr}, N·m	10 to 10^2*	10^2- 10^3	10^3- 10^4**
p_{1p} MPa	5	10	32
V', see^3/ glad	2 - 20	10 - 100	30 - 300
V, see^3/about	15 - 150	65 - 660	200 - 2000

* Low-torque hydraulic motors. ** High torque hydraulic motors.

Considerations regarding the assignment of the coefficient k_p=p_2/p_1, as well as the calculation of the value $k_b=\left(\omega_{\max}\cdot b_m+M_{ter}\right)_0/M_{Cr\max}$ set out above.

According to the calculated value V_{dis} value is selected V_{norms}, which from refers to the serial standard size of the hydraulic motor for this industry. At the same time, as a rule, $V_{norms}\geq V_{dis}$, but so that $V_{norms}/V_{dis}\leq 1.15$. Otherwise replacement is being considered p_{1p} and the calculation is repeated.

14.7.4 Calculation of VC pump parameters

When choosing the structure of the pump installation, one of the following design solutions is possible: the use of one regulating pump in the VC; the use of a regulating pump for working movements and an additional pump for idle (fast) movements in the VC; use of a group of pumps, the combination of which provides all ranges of working and idle movements; the use of one pump, which provides all working and idle movements.

To solve this problem, it is necessary to calculate the value of the pump q_{nand} supply for each mode of movement.

For VK with a hydraulic cylinder

$$Q_n = \upsilon_{C\,and} \cdot S_{1\,Norwegian\,m} + \sigma_p (p_1 - p_2), \qquad (14.72)$$

where σ_p is the coefficient of flow in the hydraulic cylinder. For VK with a hydraulic motor

$$Q_{n\,and} = V_{Q\,rm} \cdot p_{and} + \sigma_1 \cdot p_1 + \sigma_p \cdot (p_1 - p_2), \qquad (14.73)$$

where σ_1 and σ_p – coefficients of leaks from the pressure cavity and overflow, in accordance.

The selection of the standard size of the pump is carried out according to the previously defined catalogs p_{1p} and V_n.

A special case of pump selection is the need to ensure a fairly accurate value of the piston speed υ_c with the safety valve closed VK, that is, when $\upsilon_c = Q_n \cdot \eta_0 / S_{1\,norms}$. In this case, simultaneously with selection the standard size of the pump is adjusted $S_{1\,norms}$ to the value $S_{1\,cor}$. At this changes D_c and p_{and}, but so as to ensure $S_{1\,cor} \geq S_{1\,dis}$.

14.7.5 Calculation of pipelines

The main parameter of the pipeline to be calculated is the diameter d_{tr} pipeline. At the same time, an important technical economic task, because the choice of minimum sizes d_{tr} will lead to reducing the mass and dimensions of the hydraulic system, which is a positive point, but at the same time, pressure losses and, accordingly, power losses will increase due to an increase in the flow rate in the pipes. That is, the efficiency of the hydraulic drive will decrease.

The engineering approach to this calculation consists in choosing the optimal, practically tested average speed values υ_{with} currents ro- of barrel fluid depending on the purpose of the pipeline, and then determine the diameter of the pipe opening using the formula

$$d_{tr} = \sqrt{\frac{4}{\pi} \cdot \frac{Q_{tr,}}{\upsilon_{with}}} \qquad (14.74)$$

where υ_{with} – the average speed of the liquid flow in the pipe; Q_{tr} – the maximum flow of liquid through the pipeline.

Recommended values υ_{with} given in the table.14.9.

Table 14.9 - Recommended values of the average speed of movement of the working fluid

Appointment pipeline	Suction-abundant	Drainable	Discharge (at pressure, MPa)		
			to 5.0	to 10.0	more than 15.0

Average speed flow rate, m/s; (not pain- what)	0.8 - 1.2	2.0	3.4 - 4.0	5.0	8.0 - 10.0

As a rule, pressure losses in the pipeline, taking into account losses in local resistances, do not exceed 5...10% of the pressure at the pump outlet. However, with longer pipelines (10...30) m after determination d_{tr} necessary check the total amount of pressure loss. It should be noted that the definition is simpler d_{tr} as a conditional pass D_y by value Q_{tr} [20], but this method is less accurate.

Pressure losses in a round pipe Δp_{tr} in length l_{tr} in accordance with (14.11) are equal

$$\Delta p_{mp} = \rho \cdot \lambda \cdot \frac{l_{mp}}{2 \cdot d_{mp}} \cdot \upsilon^2_{with} \qquad (14.75)$$

where λ – the resistance coefficient, which is the same for metal pipes and straight sections of high-pressure hoses; (λ = 75/Re).

In turn, Re is the Reynolds number, which according to (14.10) for a round pipe is equal to

$$\mathrm{Re} = \frac{Q_{mp} \cdot 4}{d_{mp} \cdot \pi \cdot \nu}. \qquad (14.76)$$

It should be noted that the resistance coefficient λ depends on the roughness of the pipe (with turbulent flow). At the same time, in pipelines of hydraulic drives, as a rule, the flow velocities are such that the turbulence of the flow is insignificant. At the same time, the pipe can be considered practically smooth, and pressure losses can be estimated by nomograms.

Obtained values Δp_{tr}, as well as the value of pressure losses in local supports are used to adjust the values k_p. If necessary, calculations according to formulas (14.67)-(14.76) are repeated.

The algorithmic scheme for calculating VC is shown in fig. 14.34. The scheme corresponds to the sequence of the calculation outlined above and provides for repeated cycles of calculations that ensure the optimal selection of the standard parameters of the VC (diameter D_c cylinder, working volume V hydraulic motor, you- according to the size of the pump, etc.) according to the catalogs of serial products of the industry.

Hydraulic drives are characterized by fast action, which is achieved by an optimal combination of parameters. The problems of speed of action, accuracy and quality of transient processes, as well as dynamic stability are solved by means of dynamic calculations. The latter can be carried out analytically in the case of the use of compact and sometimes simplified mathematical models taking into account some nonlinearities or through simulation studies of the mathematical model on electronic computers.

In any case, the basis for calculation or simulation research is a rationally constructed mathematical model of a hydraulic drive – a dynamic object.

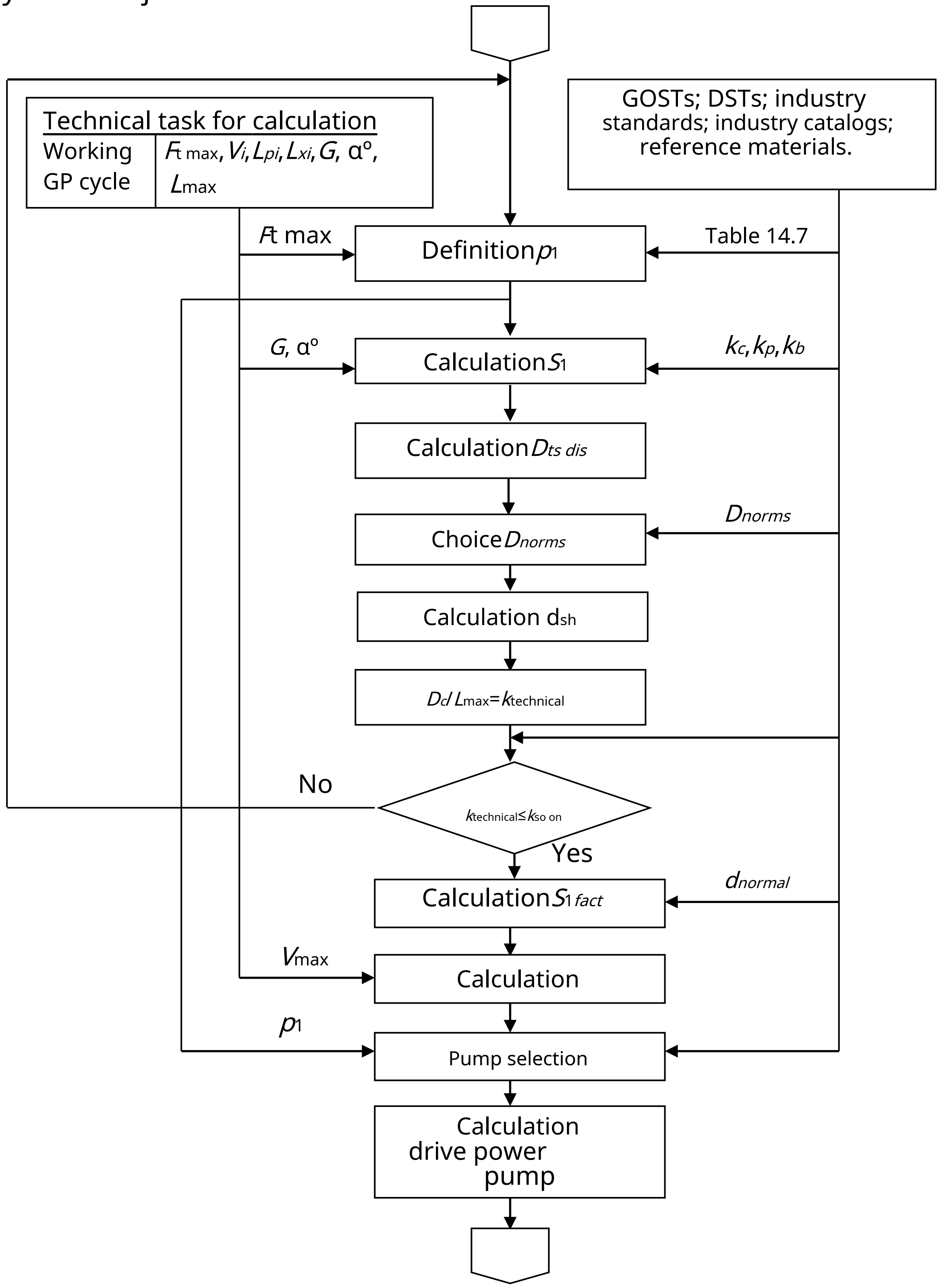


Figure 14.34 – Algorithmic scheme for calculating the executive circuit

Questions for self-control

1. What is automotive hydraulics?
2. What are the main properties of working fluids?
3. What do you know about the flow of working fluid in pipelines?
4. Name the types and advantages of pipelines.
5. Explain the principle of action of the simplest volumetric hydraulic drive.
6. What do you know about hydraulic volumetric fan drives?
7. The purpose and principle of operation of hydraulic volumetric actuators tra-missions
8. What are pumps and hydraulic motors?
9. Describe the design features and principle of operation of the six-rennet pumps.
10. Explain the principle of operation and structure of a piston pump and hydraulic motor
11. What are electro-hydraulic pumps?
12. What are the types of hydraulic cylinders, their advantages and disadvantages?
13. Principle of operation of the hydraulic accumulator.
14. What kind of hydraulic equipment do you know?
15. Purpose of hydraulic distribution valves.
16. Why are liquid flow regulators needed?
17. Principle of operation of pressure regulators.
18. Types and advantages of check valves.
19. Purpose and principle of operation of electric switchgear valves
20. How are the parameters of the executive circuit of the hydraulic drive calculated du?
21. What is the sequence of calculation of the executive circuit from hydrocili-ndrome?
22. What is the sequence of calculating the executive circuit from a hydromotor rum?
23. The principle of calculating the pump parameters of the executive circuit.
24. Pipeline calculation sequence.

15 CAR TIRES

Pneumatic systems as a source of energy are used on cars for:

- opening, closing and blocking of doors, hatches, radiator blinds and sub.;
– activating and controlling the operation of the brakes (see section 12);
– control of suspension properties and floor height adjustment vehicle above the road surface (see section 9);
- regulation of air pressure in tires.

15.1 Control of opening and closing of doors and hatches, blinds and diator

Double-action power cylinders are used in the drive of the bus doors. Three pneumodrive systems are used:
– the piston rod coming out of the power cylinder is connected to the jelly fixed on the axis on which the door is hung;
– the power cylinder is axially fastened with the help of a flange to the axis of the door (the reciprocating movement of the piston in the cylinder turns into a rotational movement of the axis of the door);
– a power cylinder of the rotary type is a combination of a cylinder and a oral axis
To obtain a smooth movement of the door when it is closed or opened, a damping device can be installed in the power cylinder, which is sensitive to the pressure and movement of the door and allows to reduce the speed of the door movement immediately before approaching its extreme position.

As an example, we can cite the 4/2 solenoid valve, which can be used to reverse the movement of the door in this way. The bus driver presses the door control button, which provides current to the system; this forces the solenoid armature through the rod to move the spool to the extreme position. The spool closes the feed line and opens the outlet to the atmosphere for one cylinder cavity while simultaneously closing the outlet and opening the feed line for the other cylinder cavity.

If there are any obstacles to the movement of the door to the closed position, the drive system must be turned off or the direction of movement of the door must be reversed. The force acting on the door when opening it should be limited to 150 N or completely removed when there is resistance to its movement. After the emergency shut-off valve is activated, the door must be moved manually. After the

the return of the emergency valve to its normal position, the movement of the door cannot be initiated until a separate button located on the instrument panel in front of the driver or in the box behind the door is pressed, or the conditions of safe operation of the door are restored.

Systems in city buses

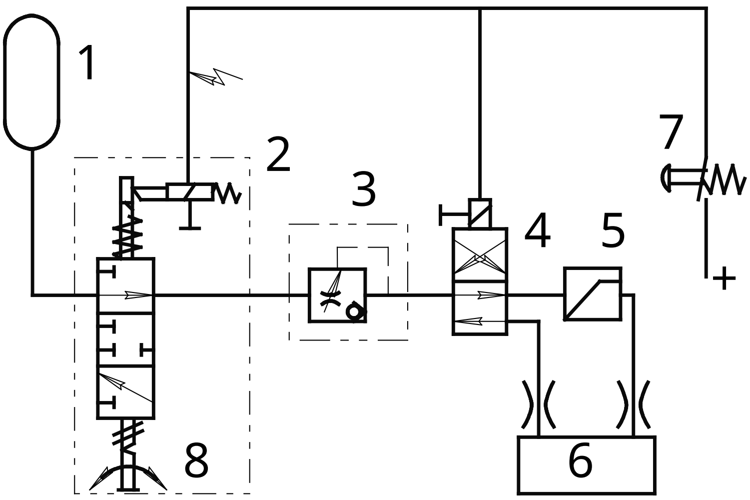
To change the direction of movement of doors in city buses, pressure devices, hermetically sealed pressure drop relays, photocells, flexible door axes and potentiometers are used. When a passenger touches the door, the system reacts to this by generating an electrical signal, which ensures the reversal of the operation of the valve in the door drive system.

On buses with more than two doors, the rear third door must be operated automatically. The driver only releases the door for work.

It is often necessary to open only half of the front door. This is achieved by using a 2/2 solenoid valve located in the closing line of the cylinder for the second half of the door.

Systems for tourist buses

Where the emergency shut-off valve is located to the door actuator valve (Fig. 15.1), the door control button activates the solenoid to release the push rod of the emergency shut-off valve.



1 – air receiver; 2 – emergency shut-off valve with a solenoid disconnection mechanism; 3 – flow limiter; 4 – door drive valve; 5 – reduction valve; 6 – control cylinder; 7 – button; 8 – sound signal electric switch

Figure 15.1 - Door opening system

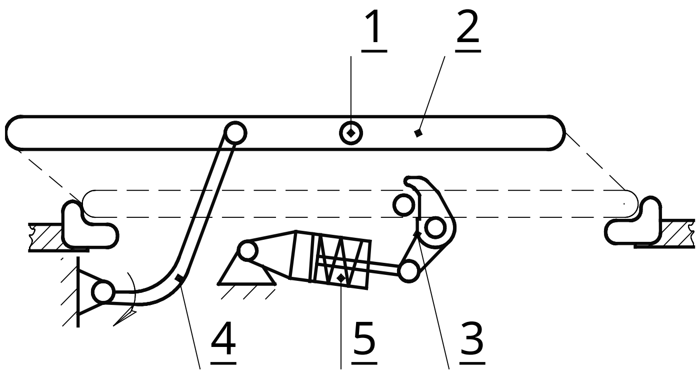
When the drive is turned on, sudden closing of the door is prevented by using a flow limiter.

If the emergency shut-off valve is located behind the door drive valve in the main line leading to the cavity of the control cylinder closure

, the emergency shut-off valve lock is actuated pneumatically. When the pressure in the system reaches an average value, the control rod is pneumatically released and the valve returns to its original position.

Locking doors and hatches

If the tour bus has large doors that can be turned outwards, it is necessary that they remain closed while driving. This is achieved by raising the door immediately after closing or by using additional locking devices with single-action drive cylinders (Fig. 15.2). In the final stage of closing the door, these devices begin to work due to the action of the door itself, for example, through the reversing valve 3/2, thus helping the drive door to complete this process. Devices for closing and blocking doors (gate, lock, shutter) are installed in such a way that they release the door when the pressure in the system is released; at the same time, the door remains locked only with a standard lock, which allows you to open it by hand in an emergency situation.



1 – constipation; 2 – revolving door; 3 – safe grip; 4 – backstage; 5 – drive cylinder

Figure 15.2 – Device for closing and locking doors tourist bus

Contrary to what has been said, the springs activate the locking mechanisms for the hatches of the luggage compartments when the pressure in the pneumatic drive system is released.

Radiator blinds

They are driven by a single-acting power cylinder controlled by a thermostat with a wax filler that expands at a temperature above 80°C and provides reversal of the 3/2 valve.

15.2 Regulation of air pressure in tires

The tire pressure control system is used on all-wheel drive trucks designed for systematic operation in adverse road conditions.

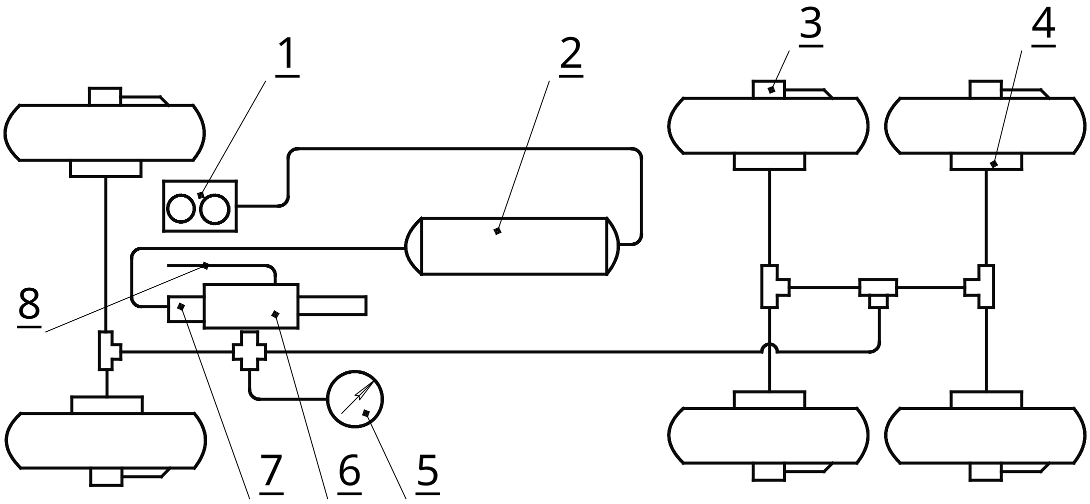
The presence of an air pressure control system allows:

- reducing the pressure in the tires, reducing the specific load on soil, thus increasing the vehicle's passability;

– in case of puncture of the camera, continue driving without changing the wheel;

- constantly maintain the necessary air pressure in the tires. The scheme

of the pressure control system is shown in fig. 15.3. The system is powered by compressed air from the pneumatic brake drive system. The tire pressure control system consists of a pressure control valve, a pressure reduction limiter valve combined with it, packing devices that provide compressed air supply to the rotating tire, and shut-off valves that allow you to disconnect a damaged tire from the system , as well as a pressure gauge and air ducts made of tubes and hoses.

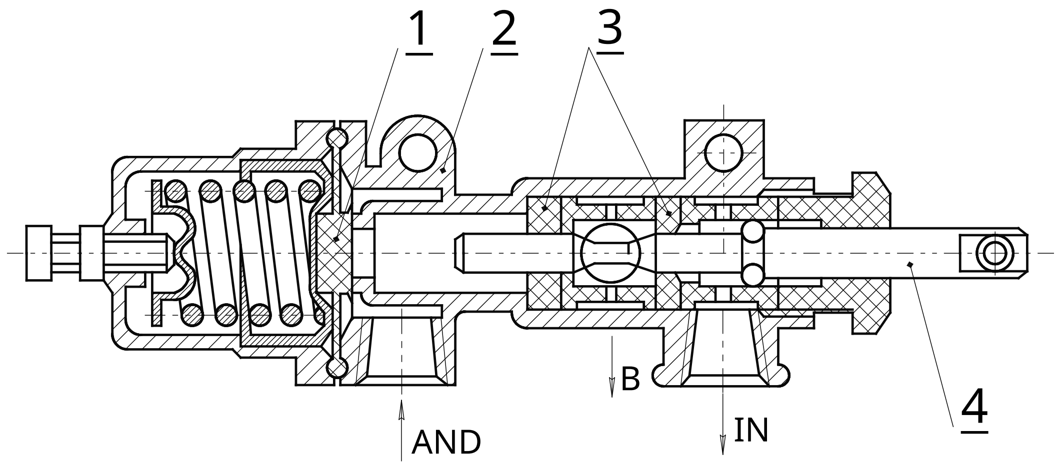


1 – compressor; 2 – cylinder; 3 – shut-off valve; 4 – packing device; 5 – manometer; 6 – tire pressure control valve; 7 – pressure reduction limiter valve; 8 – release into the atmosphere

Figure 15.3 – Scheme of the tire pressure regulation system

Pressure control valve (Fig. 15.4) of spool type. The faucet body has three openings: for supplying compressed air from the pneumatic system, for connecting to tires, and for releasing compressed air into the atmosphere.

The spool, connected by a rod to the lever located in the cabin, can move in the axial direction; it has an annular groove and is sealed with two oil seals. The spool can be in three working positions: left, right and middle. In the left position, the groove of the spool is against the left oil seal, while compressed air enters the tires from the air cylinder. In the right position, the groove of the spool is placed against the right oil seal, while the compressed air from the tires is released into the atmosphere. In the middle position, the groove of the spool is located between the oil seals, blocking the inflow of air from the pneumatic system into the tires and the release of air from the tires into the atmosphere.



A – from the balloon; B - to tires; B – into the atmosphere;
1 – limiter valve; 2 – crane body; 3 – oil seals; 4 – spool

Figure 15.4 – Pressure control valve with relief valve

A limiter valve installed on the faucet disconnects the tire pressure control system from the pneumatic brake drive system when the pressure in it drops below a certain value to ensure a sufficient reserve of compressed air for reliable braking.

When using the air pressure control system in tires, special tires with a large profile width and a reduced number of cord layers should be installed to increase the flexibility of the tire frame.

Questions for self-control

1. What is car pneumatics for?
2. How are doors and hatches controlled?
3. What are the control systems with a pneumatic drive in buses?
4. Explain how doors and hatches are locked using pneumatics
5. How is tire pressure regulated?
6. Describe the design of the pressure control valve.