ALMALYK BRANCH OF FEDERAL STATE AUTONOMOUS EDUCATIONAL INSTITUTION OF HIGHER EDUCATION

"National Research Technological University "MISIS" in Almalyk city

Mining Engineering Department

EDUCATIONAL AND METHODOLOGICAL COMPLEX ON THE SUBJECT "HYDRAULIC DRIVE OF MINING MASHINES"

(for students in the field of education: specialisation

21.05.04 - **"Mining Engineering")**

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ALMALYK BRANCH OF FEDERAL STATE AUTONOMOUS EDUCATIONAL INSTITUTION OF HIGHER EDUCATION

"National Research Technological University "MISIS" in Almalyk

COLLECTION OF LECTURES

on the subject

" HYDRAULIC DRIVE OF MINING MASHINES "

Area of study and history of the subject
HYDRAULIC DRIVES
Lecture 1

Fluid and gas mechanics, basic concepts. Static processes of hydraulic and pneumatic drives of mining machines

Definition of liquid. A *liquid* is a physical body that easily changes its shape under the action of any magnitude of forces. A liquid differs from a solid in that it has a great mobility of particles (fluidity) and takes the shape of the vessel in which it is placed.

There are two types of liquids: dripping and gaseous.

Droplet liquids include, for example, water, oil, oil, mercury, etc. A droplet liquid is characterised by very low compressibility and low resistance to tensile forces.

Gaseous liquids include all gases under normal conditions which are characterised by great compressibility and lack of resistance to tensile forces.

In hydraulics it is customary to unite liquids, gases and vapours under a single name - liquids. This is explained by the fact that the laws of motion of liquids and gases (vapours) are practically the same, if their speeds are much lower than the speed of sound. Therefore, in the future, liquids will be referred to as all substances that have fluidity when the smallest shear forces are applied to them.

The general laws of equilibrium and motion of liquids are usually expressed in the form of differential equations obtained by considering the liquid as a continuous homogeneous medium. It is neglected that the elementary volume or particle of liquid is a set of molecules located at some distances one from another. This assumption is possible because the particle sizes are always much larger than the mean free path length of molecules.

When deducing the basic regularities in hydraulics, the concept of hypothetical ideal fluid is introduced, which, unlike real (viscous) fluid, is absolutely incompressible under pressure, does not change density with temperature change and does not possess viscosity.

Real liquids are divided into *droplets* and *elastic* liquids (gases or vapours).
Droplet liquids are virtually incompressible and have a very small coefficient of volume expansion. The volume of elastic liquids varies greatly with changes in temperature or pressure.

The following basic properties of liquids are distinguished:

Density. The mass of a unit volume of a liquid, i.e. the ratio of mass m to its volume V is called density and is denoted by ; ρ

$$\rho = \frac{m}{v} \tag{1.1}$$

where: ***m-mass of*** liquid;

V ***is the*** volume of the liquid.

The unit of density is taken as kilogram per cubic metre (kG/m$^{3)}$, which corresponds to the density of a homogeneous substance with a mass of one kilogram per cubic metre.

In hydraulics is also widely used the concept of *relative density, which* is the ratio of the density of the liquid in question to the density of water at t = + 3.98 ° C and atmospheric pressure. Relative density is denoted by d.

Consequently, the relative density of water d is the ratio of the density of water at a given temperature to the highest density of water corresponding to t = + 3.98 °C. Then the dependence of relative density of water on temperature at atmospheric pressure is characterised by the data given in Table 1.

Table 1.

t^0C	d	t^0C	D	t^0C	d	t^0C	d
0	0,99987	10	0,99750	30	0,99576	70	0,97794
3	0,99999	15	0.99915	40	0,99235	80	0,97194
3,98	1,00000	20	0,99826	50	0.98820	90	0,96556
5	0,99999	25	0.99712	60	0,98338	100	0,95865

Specific gravity is the weight per unit volume of a liquid and is denoted by γ , i.e.:

$$\gamma = \frac{G}{V} \tag{1.2}$$

Where: G - weight; V - volume of liquid
Table 2 shows the specific gravities of some liquids.

Table 2.

Name of liquids	Specific weight (kG/m^3)	t^0C	Name of liquids	Specific Weight (kG/m^3)	$t\ ^0C$
Clean fresh water	1000	4	Petrol	700-750	15
Ordinary sea water	1020- 1030	4	ordinary Solar oil	880 - 890	15
Light crude oil	860 - 880		Lubricating oils	890 - 920	15
Crude oil medium	880-900	15	Fuel oil	890-940	15
Oil heavy	920 930	15	tar	930-950	15
Paraffin	790-820	15	Alcohol anhydrous	790 - 800	15
Aviation petrol	650	15	Glycerine	1260	0
		15	Mercury	13600	0

Compressibility is characterised by the volumetric compression ratio β_v , which is the relative change in volume for a 1 Pa change in pressure:

$$\beta_v = \frac{(V_1 - V_2)}{(W_1(p_2 - p_1))} \tag{1.3}$$

Where: V- initial volume; W_2 ~ final volume; P_1 and P_2 are initial and final pressures.

Surface tension (capillarity) is a property of liquid, which is caused by mutual attraction forces arising between the particles of the surface layer and causing its stressed state. Under the action of these forces, the surface of the liquid appears as if covered by a uniformly stretched thin film, which tends to give the liquid volume the shape with the smallest surface area.

Surface tension forces exert an additional pressure on a liquid normal to its surface. This pressure is measured in Newtons per square metre (N/m^2) and can be determined by Laplace's formula:

$$p = \sigma\left[\frac{1}{r_1} + \frac{1}{r_2}\right] \tag{1.4}$$

where: p - surface tension coefficient;
r_1 and r_2 are the radii of curvature of the curves obtained by intersection of the liquid surface by any two mutually perpendicular planes drawn through the normal to this surface at any point.

The average values for some liquids at the interface with air are as follows:

Water .... 0, 073 Oil.............. 0, 025
Alcohol.......... 0, 0225 Glycerin 0, 065
Benzene.......... 0, 029 Mercury................. 0, 490

Generally, the surface tension of liquids decreases as the temperature increases.

Surface tension is particularly strong in tubes of very small diameter (capillary tubes), where, due to the additional pressure caused by this tension, the position of the surface changes from its normal level (capillarity).

For capillary tubes the formula (1.4) takes the form:

$$p = \frac{2\sigma}{r} \tag{1. 5}$$

where: r is the radius of the tube.

There are two possible cases of level change: rising if the liquid wets the walls (e.g. water) and falling if the liquid does not wet the walls (mercury).

For water at t = 20^{0}C the height of capillary rise (in mm) in a glass tube is determined by the formula:

$$h = \frac{29{,}8}{d}$$

where: d *is the* inner diameter of the tube.

For mercury under the same conditions, the lowering of the level (in mm):

$$h = \frac{10{,}15}{d}$$

Temperature expansion is characterised by the coefficient of thermal expansion of liquids, which expresses the relative increase in volume with a 1°C increase in temperature:

$$\beta_t = \frac{(V_1 - V_2)}{(W_1 \Delta t)},$$

where: Δ t is the change in temperature.

The coefficient of thermal expansion for water increases with increasing pressure, but for most other droplet liquids this coefficient decreases with increasing pressure. Table 3. shows the values of the coefficient of thermal expansion for water.

Table.3.

Pressure am	At temperature t,^{0}C				
	4-10	10-20	40-50	60-70	90-100
1	0,000 014	0,000150	0,000 422	0,000556	0,000719
100	0,000 043	0,000165	0,000 422	0,000548	-
500	0.000 149	0,000236	0,000 429	0.000523	0,000523

The coefficients of thermal expansion for droplet liquids are much higher than their volume compression coefficients, yet they are also very small.

1.4 Newton's Law for friction in liquids. Viscosity.

Viscosity is the property of a fluid to resist shear tangential forces. This property cannot be detected when the fluid is at rest, as it is only manifested when it is in motion.

To clarify the physical essence of the concept of viscosity, consider the following diagram. Let there are two parallel plates A and B (Fig. 1.1). A liquid is enclosed in the space between them. Let the lower plate be stationary, the upper plate moves progressively with some constant velocity ϑ_1.

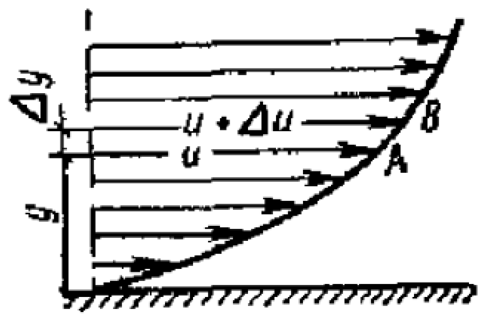


Figure 1.1.

In this case, as experience shows, the liquid layers adjacent directly to the plates (stuck), will have the same velocity with them, i.e. the layer adjacent to the upper plate B will move with the velocity ϑ_1, and adjacent to the lower plate A, will be at rest. The intermediate layers will slide one over the other with a velocity proportional to their distance from the bottom plate. If the distance between the plates is denoted by *n*, then the velocity $\vartheta_{(y)}$ of the liquid layer located at a distance *y* from this plate will be equal to

$$v_{(y)} = v_1 \frac{y}{n}$$

Newton suggested (later confirmed by experience) that the drag forces arising from such sliding of layers are proportional to the area of contact between the layers and the sliding velocity. Then, taking the area of contact equal to one, we can write:

$$\tau = \mu \, (dv/dy) \qquad (1.6)$$

Where: τ - drag force per unit area, or friction stress: μ is a coefficient of proportionality that depends on the kind of fluid, or the dynamic viscosity of the fluid.

Thus, viscosity is a physical property of a fluid that characterises its resistance to sliding or shear.

1.5 Ideal and real gases.
Boyle-Marriott's law, Gay-Lussac's law.

Insert a piston into a tube, the opposite end of which is sealed. By pushing the piston in, we reduce the volume of gas in the tube. The gas pressure increases: if the piston is not held down, the gas lifts the piston to the same height and occupies the original volume.

Pulling the piston out of the tube increases the volume of the gas, at the same time its pressure decreases, because under the action of atmospheric pressure the piston returns to its previous position as soon as we release it.

Precise experiments allowed the English scientist Robert Boyle (1627-1691) and the French scientist Edmus Marriott (1620-1684) to establish the following regularity, which is **called the Boyle-Marriott law**: *at constant temperature, the pressure of a constant mass of gas is inversely proportional to the volume of the gas.*

Gaseous liquids, compared to droplets, have a much lower density, which is subject to large changes with pressure and temperature.

For perfect (ideal gases) obeying the ***Boyle-Marriott*** *and* ***Gay-Lussac laws****,* there is the following relationship between pressure p, density ρ and temperature t:

$$\frac{p}{\rho} = Rt \tag{1.7}$$

known as the equation of state of ideal gases, where R is the specific gas constant, it is equal to the work of expansion of 1kg of gas when heated by 1K at constant pressure. The gas constant is measured in joules per kilogram and kelvin [J/(kg K)].

The density (at t = 0°C, p = 101325 Pa) and gas constant are given in Table .4.

Table 4.

Gas	$\rho, кг/м^3$	R J/(kg K)	Gas	ρ, kg/m^3	R J/(kg K)
Air	1,293	287,0	Argon	I,783	208,2
Oxygen	1,429	259,8	Helium	0,179	2078,0
Nitrogen	1,251	296,8	Methane	0,717	518,8
Hydrogen	0,090	4124,0	Ethylene	1,251	296,6
Carbon dioxide	1,977	188,0	Ammonia	0,771	488,3

Real gases do not obey the equation of state (1.7). Deviations of their properties from this equation increase with increasing pressure and decreasing temperature, and at high pressures are accounted for by introducing correction factors of compressibility established by experience.

The saturated vapour pressure of a liquid, or vapour elasticity, is the pressure at which the vapour of a liquid is in equilibrium with the liquid and the number of molecules passing from liquid to vapour is equal to the number of molecules making the reverse transition.

The saturated vapour pressure of various liquids depends largely on temperature and, as a rule, increases with its increase (Table 5.).

Table 5.

Saturated vapour pressure (Pa)

Liquid	Liquid temperature t, °C											
	0	5	10	20	30	40	50	60	70	80	90	100
Water	613	872	1225	2332	4214	7350	1234	1989	3 1 16	4733	7007	
Light	343 0	-		7840		1372	8	4	4	4	0	-
oil						0	_	3724	_	8526	-	
Petrol	646	-	7938	1068	1656			0		0		

Clay solution	8		1764	1	2	2253	3 194				-	-
				3136	5390	8	8	-	-	-	-	
						8320	1372	-		-		

The saturated vapour pressure can be defined in the same way as the pressure corresponding to the boiling point of a liquid at a given temperature. Therefore, for example, if a liquid is in a vessel (tank, pipeline), the absolute pressure $p_{абс}$, which is equal to the saturated vapour pressure $p_{n.p.}$ ($p_{abs}=p_{n.p.}$), the liquid will boil and the vessel will be filled with its vapour.

1.6 Hydrostatic pressure.

Forces acting on a fluid. When a fluid is at rest, it exhibits viscous forces. Consequently, real fluids at rest will be characterised by properties very close to those of an ideal fluid. Therefore, all hydrostatic problems considered using the concept of an ideal fluid are solved with great accuracy.

A resting fluid is subject to two categories of external forces: mass and surface forces. The mass forces are forces proportional to the mass of the liquid - gravity forces, as well as inertia forces, the latter acting,
e.g. when the liquid is at relative rest, being placed, for example, in a moving tank, etc. *Surface forces* are forces acting on the surface of the fluid volumes under investigation, e.g. the forces of piston pressure on the surface of the fluid. External forces result in stresses inside the fluid, measured in kilograms per square metre (kG/m^2), etc.

The compressive stress developed within a resting fluid is called *hydrostatic pressure or hydrostatic pressure stress.*

Let us establish the basic provisions related to the concept of hydrostatic pressure. Consider a volume of a liquid body in equilibrium (Fig. 1.2). Let us divide this volume of liquid into two parts by the plane AB. The liquid enclosed in part 1 of the investigated volume will act on part 2 along the plane of partition AB. Let's denote the area of the interface plane by ω (Fig. 1.2), mentally discarding the right part 1. Then, in order to preserve the equilibrium of the remaining left part, we will replace the influence of the discarded right part on it by the force *P*, called the force of hydrostatic pressure acting on the area . ω

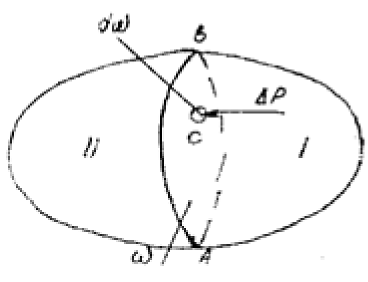


Figure 1.2.

Dividing the hydrostatic pressure force *P* by the area ω , we obtain the average hydrostatic pressure:

$$p_{cp} = \frac{p}{\omega} \qquad (1.8)$$

Let us take an arbitrary point C in the plane AB and select a small area $d\omega$ around it (Fig. 1.3). Some force ΔP will fall on this area. If we reduce the area $d\omega$ in such a way that it tends to zero, we will obtain the limit of the ratio of force to area , called hydrostatic pressure at this point C:

$$p = \lim_{d\omega \to 0} \left(\frac{\Delta P}{d\omega} \right) \tag{1.9}$$

Lecture 2

2.1 Basic properties of liquid at rest

Let's consider the equilibrium of the liquid. For this purpose, we choose a system of coordinate axes *x, y, z* with the centre in the point O and fix an arbitrary point A with coordinates *x, y, z* (Fig. 2.1). Then about the point A we will allocate an infinitesimal parallelepiped 1-2-3-4-5-6-7-8 with infinitesimal sides *dx, dy, dz* so that the point A was in the centre of this parallelepiped. The hydrostatic pressure arising in the point A under the action of external forces is denoted by *p.* The selected parallelepiped under the action of external forces will be in equilibrium if the sum of projections of all acting forces on any of the coordinate axes is equal to 0.

Let us establish the external forces acting on the liquid parallelepiped we are studying. The external forces here are: 1) volumetric forces proportional to the mass of the parallelepiped; 2) hydrostatic pressure forces acting on the faces of the parallelepiped from the side of the surrounding liquid, expressed by differential equations of Euler equilibrium.

Let us denote by *X, Y* and *Z the projections of all mass forces* (gravity and inertia forces) per unit mass on the coordinate axes *x, y*, *z*. Then the projection of the volume forces dQ_x on the *x-axis* is equal to:

$$dQ_x = XdM,$$
$$dM = dxdydz\rho.$$

Hence,

$$dQ_x = Xdxdydz\rho.$$

The projections of mass forces on the *x, y* and *z* axes are determined in the same way:

$$dQ_y = Xdxdydz\rho \text{ и } dQ_z = Zdxdydz\rho$$

Let's proceed to the establishment of hydrostatic pressure forces acting on the faces of the parallelepiped. Let us consider the forces acting on the vertical faces 1-2-3-4 and 5-6-7-8. According to the first property of hydrostatic pressure, these forces act normal to the specified sites, i.e. they are directed along the *x-axis*. Let's draw a horizontal line BC through point A, which will intersect the face of the parallelepiped 1-2-3-4 in point B and the face of 5-6-7-8 in point C. We denote the hydrostatic pressure at point B by p_c, and at point C by p_c. Since in a liquid medium the hydrostatic pressure varies continuously according to a linear law, the hydrostatic pressures at points B and C will be expressed as:

$$p_B = p - \frac{dx}{2}\frac{\partial p}{\partial x} \text{ и } p_C = p + \frac{dx}{2}\frac{\partial p}{\partial x},$$

where the partial derivative $\frac{\partial p}{\partial x}$ is called the hydrostatic pressure gradient.

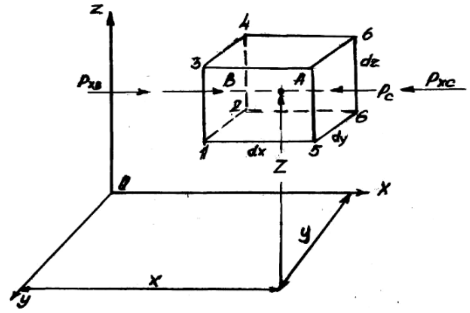


Figure 2.1.

Sites 1-2-3-4 and 5-6-7-8 are infinitesimal, so hydrostatic pressures at points B and C can be considered as average hydrostatic pressures for these sites. Consequently, it is possible to establish the values of hydrostatic pressure forces on the considered sites P_{xB} and P_{xC} (Fig. 2.1):

$$P_{xB} = (пл.1-2-3-4)\,p_B = dydz\left(p - \frac{\partial p}{\partial x}dx\right);$$

$$P_{xC} = (пл.5-6-7-8)\,p_C = dydz\left(p - \frac{\partial p}{\partial x}dx\right) \qquad (2.1)$$

Let's make the equation of equilibrium of the liquid parallelepiped 1-2-3-4-4-5-6-6-7-8 under study with respect to the x-axis. Projecting all external forces acting on the parallelepiped onto the x-axis, we obtain:

$$P_{xB} - P_{xC} + dQ_x = 0 \qquad (2.2)$$

Here, the hydrostatic pressure forces P_{xB} and P_{xC}, being normal to faces 1-2-3-4 and 5-6-7-8, are projected on the *x-axis* in full size. The projections of all other hydrostatic pressure forces acting on other faces will be equal to zero, and therefore will not be included in equation (2.2). Equation (2.2) can be rewritten as follows:

$$dydz\left(p - \frac{1}{2}\frac{\partial p}{\partial x}dx\right) - dydz\left(p + \frac{1}{2}\frac{\partial p}{\partial x}dx\right) + Xdxdydz\rho = 0.$$

After simple transformations we obtain:

$$-\frac{\partial p}{\partial x}dxdydz + Xdxdydz\rho = 0.$$

FINALLY:

$$-\frac{\partial p}{\partial x} + \rho X = 0. \qquad (2.3)$$

The equations of equilibrium with respect to the *y* and *z* axes can be derived in a similar way:

$$-\frac{\partial p}{\partial x} + \rho Y = 0. \qquad (2.3^{/})$$

$$-\frac{\partial p}{\partial x} + \rho Z = 0. \qquad (2.3^{//})$$

The resulting equations (2.3), $(2.3^{/)}$ and $(2.3^{//})$ are differential equations of fluid equilibrium (Euler):

$$\left.\begin{aligned} -\frac{\partial p}{\partial x}+\rho X &= 0. \\ -\frac{\partial p}{\partial y}+\rho Y &= 0. \\ -\frac{\partial p}{\partial z}+\rho Z &= 0. \end{aligned}\right\} \quad (2.4)$$

For further investigation, we transform the system of differential equations (2.4). Multiplying each of the equations (2.4) by *dx, dy* and *dz*, respectively, we obtain:

$$\left.\begin{aligned} -\frac{\partial p}{\partial x}dx+\rho Xdx &= 0. \\ -\frac{\partial p}{\partial y}dy+\rho Ydy &= 0. \\ -\frac{\partial p}{\partial z}dz+\rho Zdz &= 0. \end{aligned}\right\} \quad (2.4^{/})$$

Let's add up this system of equations:

$$\frac{\partial p}{\partial x}dx+\frac{\partial p}{\partial y}dy+\frac{\partial p}{\partial z}dz=\rho(Xdx+Ydy+Zdz) \quad (2.5)$$

Since hydrostatic pressure is a function only of the coordinates of the point $x = f\,(x,y,z)$, the left side of the equation is the total differential of the pressure:

$$dp=\frac{\partial p}{\partial x}dx+\frac{\partial p}{\partial y}dy+\frac{\partial p}{\partial z}dz \quad (2.6)$$

$$dp=\rho(Xdx+Ydy+Zdz). \quad (2.7)$$

Since the density of the fluid under consideration ρ is constant, equation (2.7) can make sense only if the right part of this equation is also a full differential. For this purpose it is necessary that there exists such a function $U=f(x,y,z)$, *the* partial derivatives of which for *x, y* and *z* would be equal:

$$\frac{\partial U}{\partial x}=X;\frac{\partial U}{\partial y}=Y;\frac{\partial U}{\partial z}=Z. \quad (2.8)$$

Such a function is called a potential, or force function, and the forces that are expressed by this function are called *forces having a potential.*

Consequently, a fluid can be in equilibrium only when the system of mass forces acting on it has a potential. Many forces with potential are known from mechanics, the most important of them are gravity and inertia forces.

2.2 Free surface of a liquid.

Let us consider the most important for practice case of equilibrium of a liquid under the action of gravity forces only.

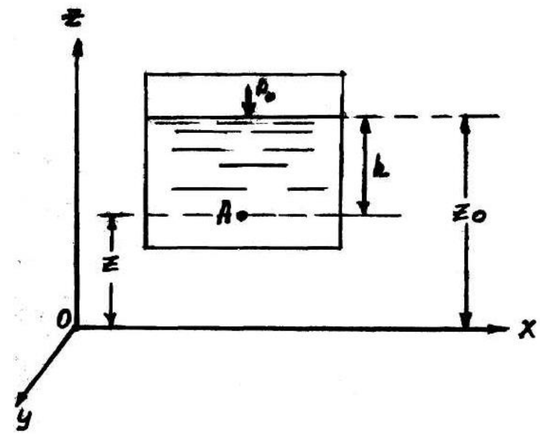


Fig.2.2.

Assume that the liquid is in a closed vessel, as shown in Fig. 2.2. Let us also assume that the known pressure p_0, different from atmospheric pressure, acts on the surface of the liquid. Then the projections of volume forces (in this case gravity forces) on the xnu axes will be equal to zero:

$$X = \frac{\partial U}{\partial x} = 0 \text{ и } Y = \frac{\partial U}{\partial y} = 0.$$

The projection of the gravity force on the *z-axis* per unit mass will be equal to:

$$Z = \frac{\partial U}{\partial z} = -g,$$

since the *z-axis* has a direction opposite to the direction of gravity. To solve a number of practical problems, let us rewrite Euler's differential equations in this form:

$$dp = \rho\left(\frac{\partial U}{\partial x}dx + \frac{\partial U}{\partial y}dy + \frac{\partial U}{\partial z}dz\right)$$

or

$$dp = \rho dU \qquad (2.9)$$

I.e., the differential equation for the considered case will take the following form:

$$dp = -\rho g dz = -\gamma dz \qquad (2.10)$$

$$\frac{dp}{\gamma} + dz = 0 \qquad (2.10')$$

The obtained equation (2.10$'$) is a *differential equation of equilibrium of a fluid* under the action of gravity alone.

As a result of integration of equation (2.10$'$) we have:

$$z + \frac{p}{\gamma} = C \qquad (2.11)$$

We know the boundary conditions on the liquid surface: at $z=z_0$ the pressure $p=p_0$.
Hence,

$$z_0 + \frac{p_0}{\gamma} = C \qquad (2.12)$$

Let's substitute the obtained expression for the integration constant into the dependence (2.12)

$$z + \frac{p}{\gamma} = z_0 + \frac{p_0}{\gamma},$$

or definitively:

$$p = p_0 + \gamma(z_0 - z) \tag{2.13}$$

2.3 The basic equation of hydrostatics.

From equation (2.14)

$$\left.\begin{aligned} -\frac{\partial p}{\partial x} + \rho X = 0. \\ -\frac{\partial p}{\partial y} + \rho Y = 0. \\ -\frac{\partial p}{\partial z} + \rho Z = 0. \end{aligned}\right\} \tag{2.14}$$

it follows that the pressure in a resting fluid varies only vertically (along the *z-axis,* Fig. 2.3), remaining the same at all points of any horizontal plane, since the pressure variation along the *x* and *y* axes is zero. Due to the fact that in this system of equations the partial derivatives $\frac{\partial p}{\partial x} u \frac{\partial p}{\partial \gamma}$ are equal to zero, the partial derivative of $\frac{\partial p}{\partial z}$ can be replaced by $\frac{dp}{dz}$ and hence:

$$-\rho g - \frac{dp}{dz} = 0$$

Hence.

$$-dp - \rho g dz = 0 \tag{2.15}$$

Dividing the left and right parts of the last expression by ρ g and changing signs, we present this equation in the form:

$$dz + d\frac{1}{\rho g}dp = 0$$

For an incompressible homogeneous fluid the density is constant and hence

$$dz + d\left(\frac{p}{\rho g}\right) = 0$$

or $$d\left(z + \frac{p}{\rho g}\right) = 0$$

whence after integration we obtain

$$z + \frac{p}{\rho g} = const \tag{2.16}$$

For two arbitrary horizontal planes 1 and 2, equation (2.16) is expressed in the form:

$$z + \frac{p_1}{\rho g} = z_0 + \frac{p_2}{\rho g}, \tag{2.17}$$

Equation (2.16) or (2.17) is the ***basic equation of*** of ***hydrostatics***. In equation (2.17): z_1 and z_2 are heights of location of two points, inside a resting homogeneous droplet liquid above an arbitrarily chosen horizontal plane of reference (plane of comparison), and p_1 and p_2 are hydrostatic pressures at these points.

Consider, for example, two fluid particles, of which one is located at point 7 inside the fluid volume (Fig. 2.3) at height z from an arbitrarily chosen *0-0* comparison plane, and the other is located at point 2 on the fluid surface at height z_0 from the same plane. Let p and p_o be the pressures at points 1 and 2, respectively. With these notations according to the equation (2.17):

$$z + \frac{p}{\rho g} = z_0 + \frac{p_0}{\rho g}, \tag{2.17.a}$$

Or

$$\frac{p - p_0}{\rho g} = z_0 - z, \tag{2.17.б}$$

Lecture 3. Kinematics of pressurised fluid flows, energy losses

3.1 Fluid kinematics and dynamics.

Classification of motions, local fluid velocity.

Fluid motion is determined by the velocities of particles at individual points in the fluid flow, pressures occurring at different depths, depths, as well as the overall shape of the flow. In this case, the depth of the fluid flow, velocities, accelerations and pressures at the points of the flow depend on the position of the points determined by the coordinates *x, y, z.* Consequently, these quantities are functions of coordinates. In addition, the quantities characterising the fluid motion can change in time, being also a function of time *t*. In this connection, two types of motion are distinguished: steady and unsteady.

Steady motion is a type of motion in which velocities, accelerations, pressures, depths do not change with time, and depend only on the position in the fluid flow of the point under consideration, being a function of coordinates:

$$u = f(x,y,z);\ p = f_1(x,y,z);\ h = f_2(x,y,z).$$

Here *and* is the velocity of the fluid; .

p - hydrodynamic pressure at the point under consideration;

h is the depth of the flow.

Unsteady motion is a type of motion in which all the above components are a function not only of coordinates but also of time:

$u = f(x,y,z,\ t)\!:\ p = f_{(1)}(x,y,z,t)\ h = f_{(2)}(x,y,z,t).$

Let us illustrate the above types of fluid motion by the example of liquid flowing out of a tank. Suppose that there is a tap in the tank to release water. The water is supplied to the tank by a water pipe equipped with a gate valve. If we simultaneously open the outlet tap and the gate valve in the pipe and adjust their positions so that the amount of water flowing out is equal to the amount of of incoming water, then we will observe in the reservoir a steady motion. Indeed, the depth of water in the reservoir *H* will be constant, not changing with time; therefore, at any point of the fluid the hydrodynamic pressure *p, the* immersion depth *h of* the considered point and the velocity will also not change with time.

Close the water pipe gate valve and leave the outlet tap open. The tank will be emptied. At the same time we will observe an unsteady motion of the liquid. In fact, the depth of water in the tank *H* decreases with time. Because of this, the depth *h* of immersion of the point in question in the liquid, the pressure and the velocity of flow at this point decrease. As a result, there will come a moment when the reservoir is empty and all components of motion *{i, p, h)* will be equal to zero.

Steady-state motion is divided into uniform and non-uniform *motion*. ***Uniform motion*** is a type of steady motion in which all components of motion - velocity, pressure, channel shape, depth - do not change along the length (*x-axis*) of the flow. The cross-section of the flow in uniform motion is constant along its length.

An example of uniform motion is the motion in a channel of regular shape with constant filling depth (Fig. 3.1). The motion of a flow with constant velocity in a cylindrical pipe of constant cross-section will also be uniform.

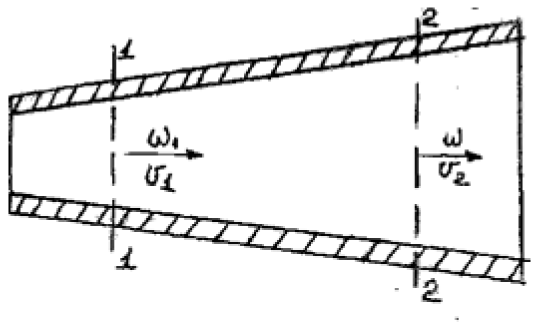


Figure 3.1.

Uneven motion can be observed in a conical tube in which the flow cross-sections and hence velocity (Fig. 3.1), pressure and depth vary along its length.

Depending on the causes and general conditions under which the movement takes place, a distinction is made between pressurised and unpressurised movement. ***Pressure*** motion is the movement of fluid in a flow without a free surface; it is usually observed in closed pipelines or other hydraulic systems. In pressurised flow, the fluid completely fills the cross-section formed by the solid walls restricting the flow. Pressurised flow occurs due to the presence of head differences along the length of the flow, e.g. created by a water tower, a feed tank of a gravity fuel system, a mains-connected pump, etc.

Motion when the flow is not bounded on all sides by solid walls but has a free surface is called ***unpressurised*** or free surface motion. In most cases, the free surface is in contact with the atmosphere, and therefore, in unpressurised flow, the pressure at the surface of the flow is almost always equal to the atmospheric pressure. The cause of unpressurised flow is the action of gravity.

The current line and its direction.

When solving many problems of practical hydrodynamics, an assumption is made that the flow of a moving fluid consists of separate (Fig. 3.2) elementary jets that do not change their shape. Thus, the flow is mentally divided into a number of elementary jets-tubes, as schematically shown in Fig. 32.2, and will be considered by us as a set of moving elementary jets. Let us define the concept of an elementary jet and give its properties.

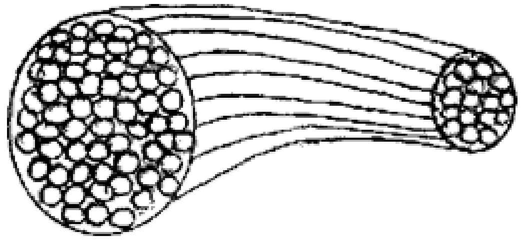


Figure 3.2.

Consider the flow of a fluid in steady motion (Fig. 3.3). Take point 1 in this flow and plot in it the velocity vector and_1 expressing its magnitude *and* direction.

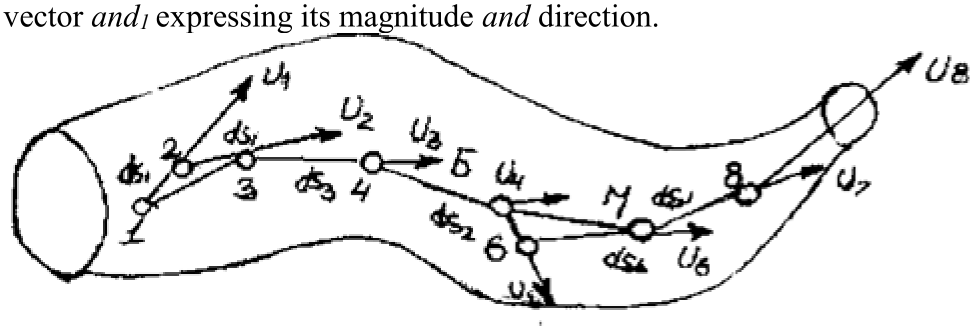


Fig.3.3.

On this vector we take point 2 at infinitesimal distance ds_1 from point 1. At point 2, we construct a velocity vector i_2, on which we take point 3 at infinitesimal distance ds_2 from point 2, etc. If we reduce the distances between points ds_1, ds_2, etc. to zero, then instead of a broken line 1-2-3-4-4-5-6-6-7-8 we will in the limit obtain a curve starting at point 1 and called a current line. A *current line* is a line at each point of which at a given instant the vector of fluid velocity coincides with the direction of the tangent to this line. In steady motion, current lines coincide with the trajectories of fluid particles. In this case, the fluid particle moves along the current line. Therefore, in steady motion, the current lines coincide with the trajectories of the moving particles.

Let us construct a closed contour around point 1, forming an infinitesimal area *dw*, and draw current lines through all points of the contour (Fig. 3.4). We obtain the so-called current tube. If we draw current lines through all points of the infinitesimal area *dw*, we obtain an elementary jet filled with a "bundle" of current lines.

On the basis of all the above, it is assumed that the elementary jet has the following properties:
1. The shape of the elementary jet remains unchanged over time because the appearance of the current lines that make up the jet does not change over time in steady-state motion.

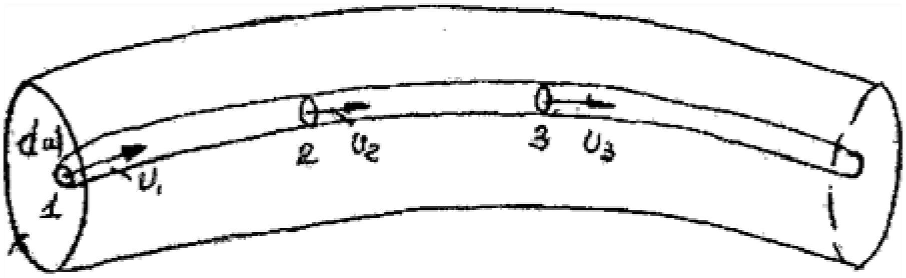


Figure 3.4.

1. The elementary jet formed by the current lines is as if impermeable for the liquid particles moving in the neighbouring jets. Particles of liquid from neighbouring jets, sliding on the surface of the jet, cannot penetrate inside it.
2. Due to the smallness of the cross-section of the elementary jet, the velocities at all points of its cross-section are the same.
3. A fluid flow consisting of elementary jets possessing the above properties is sometimes referred to as a "jet model of fluid motion". Such a flow can be, for example, represented by the motion of liquid in a model consisting of a tube filled with thin glass tubes.

Turbulent mode of fluid motion.

In industrial practice, turbulent motion of liquids is the most common. In turbulent motion due to chaotic motion of particles there is an equalisation of velocities in the bulk of the flow and their distribution over the pipe cross-section is characterised by a curve that differs in shape from a parabola, and the curve has a much wider apex (Fig. 3.5.).

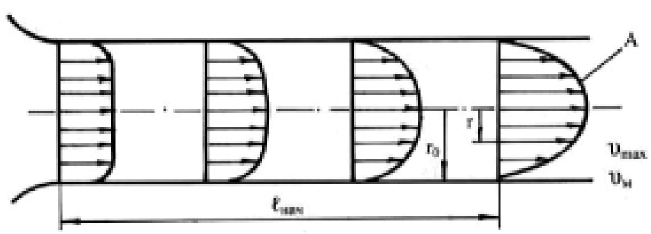


Fig.3.5.

Experience shows that the average velocity *v* in turbulent motion is not equal to half of the maximum velocity (as' for laminar motion), but is much greater than this value, and: $w/w_{(max)}=f$ (Re). For example, at Re = 108 the value of velocity $v \approx 0.9\ w_{max}$.

Pulsation and mean flow velocity in turbulent fluid flow regime.

Due to the complex nature of turbulent motion, it is not possible to obtain the velocity distribution profile and the value of w/w_{max} strictly theoretically. In addition, in turbulent flow, the velocity profile expresses the distribution of time-averaged velocities rather than the true velocities.

At each point in turbulent flow, the true velocity does not remain constant in time due to the chaotic nature of particle motion. Its instantaneous values experience fluctuations, or irregular pulsations, which are chaotic in nature.
A typical picture of the variation of the component of the true instantaneous velocity wx (along the x-axis of the flow) for some point as a function of time t is shown in Fig. 6.4. 6.4. The true velocity itself is practically impossible to measure because of chaotic movement of particles in all directions. As can be seen from Figure 6.4, the velocities pulsate around some time-averaged value, becoming greater or lesser than it. For a given point, the value of the time-averaged m\. velocity can be found from the relation:

$$w_x = \frac{\int_0^{\tau} w_x d\tau}{\tau}$$

through Δ w (we omit the index x hereinafter):

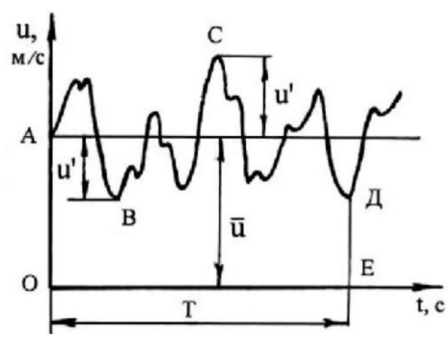


Figure 3.6.

Thus, the value of wx is equal to the height of a rectangle equal to the area enclosed between the pulsation curve and the abscissa axis within the time variation from 0 to τ (Fig. 3.6.).
The difference between the true and averaged velocities is called the instantaneous pulsation velocity and denoted by
$u - u_{cp} = \Delta u$.

The role of the hydraulic machine in mechanical engineering.

Hydraulic machines are used to convert the mechanical energy of a motor into the energy of the fluid being moved (pumps) or the hydraulic energy of a fluid flow into mechanical energy (hydraulic motors). The first hydraulic motor was the water wheel, which utilised the energy of flowing water, and the first pump was the piston pump. Water wheels began to be used more than 300 years ago in China, Egypt and India as a source of energy to lift water into irrigation canals and to turn millstones. Water mills, piston pumps driven by human and animal muscle power and pumps driven by water wheels have been known for a long time.

In the middle of the 18th century, Leonhard Euler (1707-1783) created the famous theory of vane hydraulic machines, published in his work "A More Complete Theory of Machines Driven by the Action of Water". Academician Euler derived dependencies characterising the work of hydraulic vane machines, being ahead of the technology by almost 100 years. Only in the middle of the XIX century, when in1835 г . A.A. Sablukov invented the centrifugal pump, Euler's equations began to find application in the design of hydraulic turbines and centrifugal pumps. The use of Euler's work began in the late 19th century, when sufficiently fast engines for pumps were created and hydropower began to be more widely developed.

A large number of centrifugal pumps of various types are used in mechanical engineering for equipment of feed systems of steam boilers of thermal power plants and ship installations, for pumping oil, fuel oil, oil, pumps for cracking process, in fuel supply systems of aeroplanes. Volumetric pumps are necessary equipment for hydraulic presses and similar installations. In addition, rotary pumps of special types (vane, rotary knuckle, screw, etc.) are widely used in mechanical engineering to supply lubricating oil to lubricating circulation systems, for operation of hydraulic drives in machine tools, etc.

3.1 Classification of hydraulic machines.

Hydraulic machines are divided into *vane* hydraulic machines (centrifugal and axial pumps, hydraulic turbines, etc.) and *volumetric* machines operating on the principle of fluid displacement (piston, rotary and other pumps).

Hydraulic machines also include some special devices that, like pumps, are used to move liquids:

1) ***hydraulic rams***, based on the principle of utilising the pressure generated by hydraulic shock;
2) ***ejectors*** (water jet pumps), in which the liquid is lifted by utilising the kinetic energy of the jet;
3) ***erlifts*** - devices for lifting water from wells by compressed air.

1.3 Basic definitions used in pump theory.

Pump theory uses a number of terms and definitions that apply to all types of pumps. Let's consider the scheme of operation of a pump included in a system supplying water from a water supply source to a pressure tank (Fig. 3.7). When the pump is operating, a vacuum is created in the suction pipe and suction chamber, which ensures that water is lifted through the suction pipe from the intake well to the pump. This vacuum must be sufficient to lift the water from the well to a height h_{BC} from the water level in the well to the centre of the pump), to overcome the energy losses in the suction line h_w, and to create a velocity in the suction pipe.

The vertical distance from the water level in the well to the centre of the pump $h_{(wc)}$ is called the geodetic suction height; the energy loss in the suction line h_{wc} is called the suction loss.

The liquid entering the pump is supplied with energy (mainly in the form of pressure energy), which is used to overcome the resistance in the pressure pipe through which the liquid flows and to lift the liquid into the tank.

The vertical distance h_{wH}" from the centre of the pump to the water level in the tank is called the geodetic discharge height; the energy loss in the discharge line is called the discharge loss h_{wH}.

The total head to be generated by the pump can be defined as the difference in specific energy of the liquid flow at the cross-sections corresponding to the beginning of the discharge line and the end of the suction pipe. Manometers and vacuum gauges are usually installed at these cross-sections. Let's determine the values of specific energy of the flow in section I-I, where the vacuum gauge is installed, and in section II-II, where the pressure gauge is installed. Let's assume that the pressures on the surface of the water level in the well and on the tank are the same and equal to atmospheric pressure.

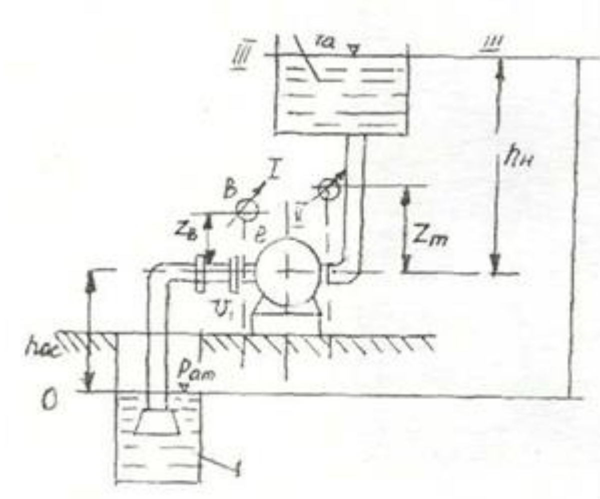


Fig 3.7.

Taking as a plane of comparison the level of the free surface in the receiving well 0-0 (Fig. 1.1.), we obtain an expression for determining the specific energies:

$$E_1 = (h_{BC} + z_B) + \frac{p_1}{\gamma} + \frac{\upsilon_1^2}{2g}$$

$$E_H = (h_{BC} + z_H) + \frac{p_2}{\gamma} + \frac{\upsilon_2^2}{2g}$$

here: z_B - vertical distances from the vacuum gauge and pressure gauge mounting points to the pump axis;

P_1 *and* P_2 are absolute pressures at the instrument installation locations;

V_1 and V_2 are the velocities in the suction and discharge pipes.

Therefore, the total pump head H is equal to:

$$H = E_n - E_1,$$

or

$$H = \frac{p_2 - p_1}{\gamma} + \Delta h + \frac{\upsilon_2^2 - \upsilon_1^2}{2g}, \qquad (3.1)$$

Where: . $\Delta h = z_H - z_B$

The vacuum gauge indicates the amount of vacuum (vacuum) $H_{вак}$ in the suction pipe, therefore its value:

$$H_{вак} = \frac{P_{am}}{\gamma} - \frac{P_1}{\gamma}, или \frac{P_1}{\gamma} = \frac{P_{am}}{\gamma} - H_{вак},$$

pressure gauge shows the overpressure in the discharge line

$$H_{ман} = \frac{P_2}{\gamma} - \frac{P_{am}}{\gamma}, или \frac{P_2}{\gamma} = H_{ман} + \frac{P_{am}}{\gamma},$$

substituting these values into the dependence (3.1), we obtain:

$$H = H_{ман} + H_{вак} + \Delta h + \frac{\upsilon_2^2 - \upsilon_1^2}{2g}, \qquad (3.2)$$

It should be noted that if the pressure gauge is located below the vacuum gauge, the value of Δh will be negative. The sum of three values

$$H_{ман} + H_{вак} \pm \Delta h = H_м. \qquad (3.3)$$

namely: the pressure gauge and vacuum gauge readings expressed in metres of water column and the vertical distance between the connection points of the instruments, is called the gauge head of the pump. The total head of a pump can be expressed by this equality:

$$H = H_{м} + \frac{V_2^2 - V_1^2}{2g} \tag{3.4}$$

Thus, the pump head is the energy delivered by the pump to each kilogram of liquid pumped. If the diameters of the suction and discharge pipes are equal ($v_{(1)} = v_2$), the total pump head is equal to the gauge head. If the surface pressure of the liquid in the connected tanks is different, the pump must overcome the pressure difference $\Delta p = p_2 - p_1$, where: $p_{(1)}$ and p_2 are the surface pressures in tanks 1 and 2. Then the total head developed by the pump should be
Equals:

$$H = h_{BC} + h_{H} + h_{wB} \left(\frac{p_2 - p_1}{\gamma}\right). \tag{3.5}$$

The volume of liquid delivered by the pump into the pipeline per unit of time is called the pump capacity and is denoted by *Q*. The capacity dimension is *l/sec* or *m^3/sec.*

The *useful power* produced by the pump to lift and move the liquid at head H is equal to:

$N_H = \gamma Q H,\ kGm$

and the usable power of the pump will be the value:

$$N = \frac{\gamma Q H}{75}\ л.с. = \frac{\gamma Q H}{102}\ кВт \tag{3.6}$$

Due to the presence of losses in the pump (hydraulic, mechanical, volumetric), which can be evaluated by the total efficiency of the pump η, the motor power N consumed by the pump will be greater than the useful power of the pump:

$$N = \frac{N_H}{\eta} = \frac{\gamma Q H}{75\eta}\ л.с. = \frac{\gamma Q H}{102\eta}\ кВт \tag{3.7}$$

Here the total efficiency of the pump is the product of three partial coefficients

$\eta = \eta_{Г} \cdot \eta_{O} \cdot \eta_{м}$,

Where:

η_H - hydraulic efficiency, which takes into account hydraulic energy losses arising during the movement of liquid through the pump (impact losses at inlet and outlet, friction and vortex losses);

η_m - mechanical efficiency taking into account losses related to overcoming mechanical friction in pump elements (friction in bearings, in glands, friction of impeller against liquid);

η_0 - volumetric efficiency, taking into account water leakage through gaps and seals bypassing the impeller.

The efficiency of a pump depends on many factors: on the size and types of pumps, thoroughness of manufacturing and assembly of its individual parts, operating conditions, etc. Total efficiency of vane pumps varies in the range: 0.70 - 0.90. At that, the highest coefficients correspond to machines of large sizes

Supervisory Questions:

1.What is the role of hydraulic machines in mechanical engineering?
2. What is the principle of operation of vane machines?
3. Derive the basic equation of vane machines.
4. Into which types are hydraulic machines divided ?
5. What is meant by the term hydraulic machine?
6. Into which types are vane machines divided ?

Lecture 4

Vane pumps

4.1 Schematic representation of the pump.

In principle, any positive displacement pump can be operated as an electric motor if pressurised liquid is supplied into the cavity.
In centrifugal pumps, the liquid is drawn in and discharged uniformly and continuously by the centrifugal force generated by the rotation of an impeller with vanes enclosed in a spiral-shaped casing.

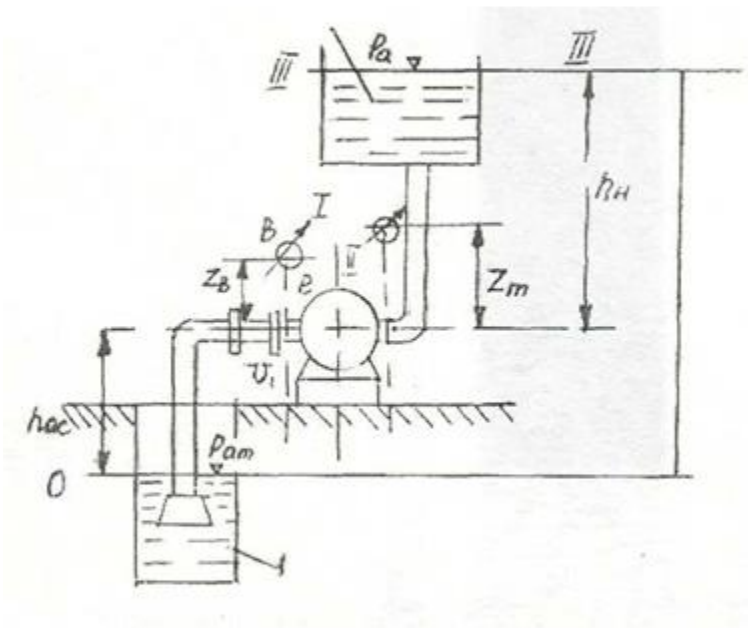


Figure 4.1. Schematic diagram of a centrifugal pump:

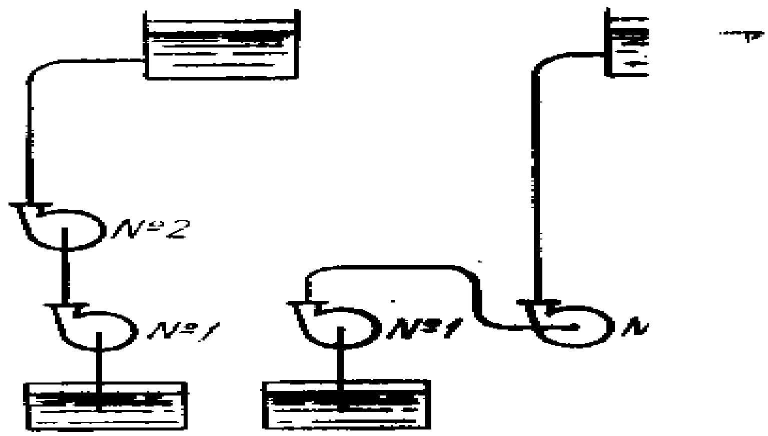
In a single-stage centrifugal pump (Fig. 4.1) the liquid from the suction pipe 1 enters along the axis of the impeller 2 into the pump casing 3 and, falling on the vanes 4, acquires rotational motion. The centrifugal force throws the liquid into a channel of variable cross-section between the casing and the impeller, in which the liquid velocity decreases to a value equal to the velocity in the discharge pipe 5.

1 - suction pipe; 2 - impeller; 3 - housing; 4 - blades; 5 - discharge pipe.

As follows from Bernoulli's equation, the kinetic energy of the liquid flow is converted into statistical head, thus increasing the pressure of the liquid. At the inlet of the wheel a reduced pressure is created and the liquid flows continuously from the inlet tank into the pump.

The pressure developed by a centrifugal pump depends on the rotational speed of the impeller. Due to the large clearances between the impeller and the pump casing, the vacuum generated by the rotation of the impeller is not sufficient to lift the liquid through the suction pipe, unless the suction pipe and the pump casing are filled with liquid. The centrifugal pump is therefore primed with the liquid to be pumped before starting. To prevent the liquid from pouring out of the pump and the suction pipe when

the pump is primed or during short stops, a check valve with a strainer is installed at the end of the suction pipe immersed in the liquid (not shown).
The head of single-stage centrifugal pumps (with one impeller) is limited and does not exceed50 м .
To create higher heads, multistage pumps (Fig. 4.2) are used, having several impellers 1 in a common casing 2, in a common casing 2, arranged in series on one shaft 3. The liquid leaving the first impeller flows through a special outlet channel 4 in the pump casing into the second impeller (where additional energy is imparted to it), from the second impeller through the outlet channel into the third impeller, etc. Thus, without taking into account the number of losses, it can be assumed that the head of a multistage pump is equal to the head of one impeller multiplied by the number of impellers. The number of impellers in a multistage pump usually does not exceed five.



4.2 Connecting the pumps in series.

If the pumps are connected in series, the overall characteristic is obtained by adding the pump heads for each capacity value.

With this type of pump connection it is possible to increase the head considerably if the characteristic of the network is steep enough.

$$H_{1+2} = H_1 + H_2$$

$$N_{1+2} = N_1 + N_2 = \frac{\gamma Q H_1}{102\eta_1} + \frac{\gamma Q H_2}{102\eta_2}$$

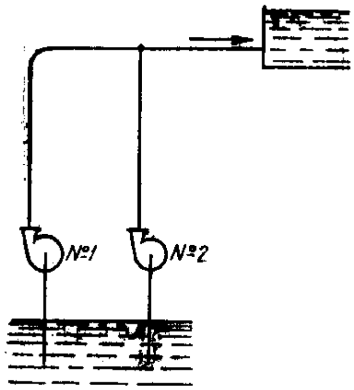
$$N_{1+2} = \frac{\gamma Q (H_2 + H_2)}{102\eta_{cp}}$$

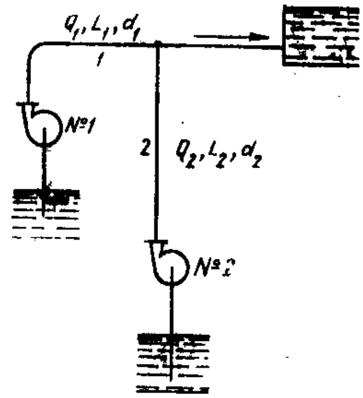
$$\eta_{cp} = \frac{H_1 + H_2}{\frac{H_1}{\eta_1} + \frac{H_2}{\eta_2}}$$

Where H is head; N is power;
Q - productivity

4.3 Parallel connection of pumps.

In parallel connection, the total pump characteristic is obtained by adding the abscissa of the characteristics of each pump for a given head.
This is why parallel pumping is used to increase the capacity of the pumping unit when the network characteristics are sufficiently gentle. The head increase is insignificant.





$Q_{1+2} = Q_1 + Q_2 = 2Q_{(1)}$
Where Q is productivity.

4.3 Cavitation in vane pumps.

Cavitation occurs at high rotational speeds of impellers of vane (centrifugal) pumps and when pumping hot liquids under conditions where there is intense vapour formation in the liquid in the pump. The vapour bubbles travel with the liquid into the higher pressure region where they instantly condense. The liquid rapidly fills the cavities where the condensed vapour had been, which is accompanied by hydraulic shocks, noise and shaking of the pump. Cavitation leads to rapid destruction due to hydraulic shocks and increased corrosion during the vapourisation period. The pump performance and head are drastically reduced during cavitation.

Supervisory Questions:

1. Which hydraulic machines are called pumps
2. Write the basic equations of a centrifugal pump.
3. Give the main characteristics of the pumps.
4. What are centrifugal pumps designed for ?
5. Into which types are centrifugal pumps coassified?

5.1 General information on positive displacement hydraulic machines.

Positive displacement pumps are used to deliver pressurised liquid and *hydraulic motors* are used to convert the pressure energy of the liquid into mechanical energy. Any positive displacement pump can operate as an electric motor if pressurised liquid is delivered into its cavity and mechanical power is removed from the shaft. In spite of the fact that in a hydraulic motor there is an inverse process compared to a pump, the principle of operation is the same and is based on the displacement of a low-compressible liquid from a closed volume. A distinctive feature of volumetric hydraulic machines is the reciprocating or rotary motion of the displacer made in the form of a sliding or rotating piston. Unlike vane machines, in the volumetric hydraulic machine under the direct influence of the piston changes the potential energy of pressure with virtually unchanged kinetic energy of the fluid. This is why volumetric hydraulic machines are sometimes referred to as hydrostatic machines.

5.2 Principle of operation and purpose of volumetric hydraulic machines.

By purpose, positive displacement pumps can be divided into pumps for water and other liquids and pumps for hydraulic drives. Hydraulic motors are used as an integral part of hydraulic drives. By design, positive displacement pumps are divided into *piston, rotary piston, rotary vane and rotary gear pumps.* Various schemes of piston pumps are predominantly used as pumps for water and other liquids. In positive displacement hydraulic transmissions and drives, all of the above mentioned types of basic design varieties of positive displacement hydraulic machines are used.

Since the droplet liquid compresses very little when displaced from a confined volume, the pressure in the volume increases to significant values:

$$\Delta p = E_{ж} \frac{\Delta W}{W}. \tag{5.1}$$

where: Δp - pressure change at change of closed volume W by the value ΔW; $E_{(w)}$ - volume modulus of elasticity of liquid compression.

Schematic diagrams of volumetric hydraulic machines

The operating cycle of a piston pump is characterised by an indicator diagram, which graphically shows the change in cylinder pressure over one complete revolution of the crank.

Fig. 5.1 shows an indicator diagram of a perfect piston pump. In this case there is no leakage of liquid through valves and piston, valves work without overlapping and do not create hydraulic resistance. The line se/ corresponds to the process

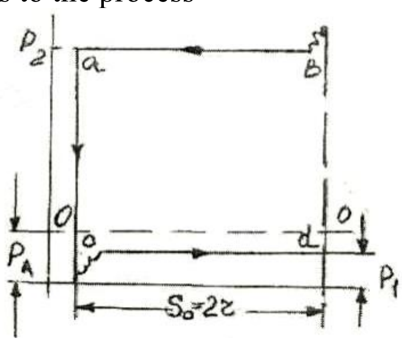


Fig.5.1.

suction, line cd the discharge process. Since the compressibility of the fluid is small, lines ac and ab are vertical. Some oscillation at the beginning of suction (point c) and at the beginning of discharge (point b) is due to the opening of valves.(Fig. 5.1) .

If there are faults in the pump, the display diagrams may be different from the

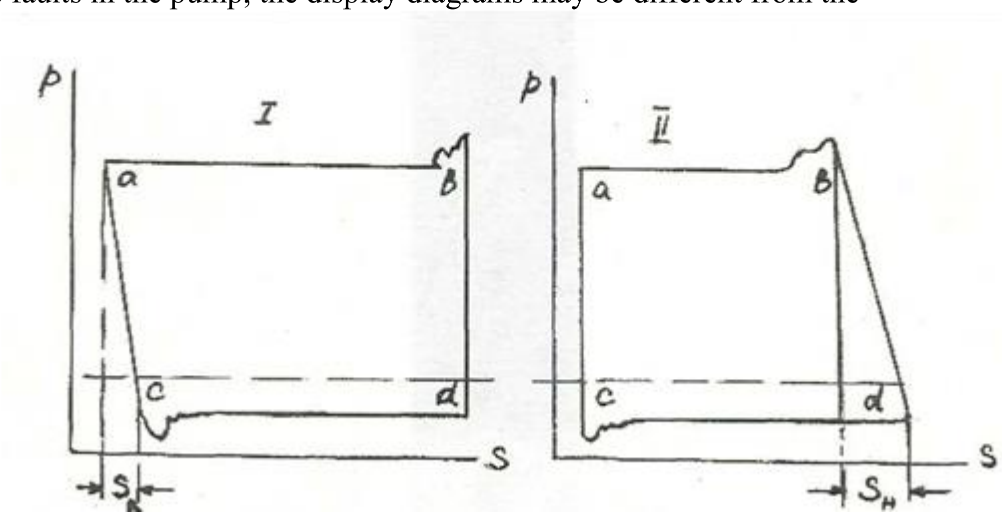


Figure 5.2.

of the diagram shown in Fig. 5.2.

Thus, diagram I (Fig. 5.2) is characteristic of a pump with a delayed closing of the discharge valve and diagram II is characteristic of a pump with a delayed closing of the suction valve.

Changes in indicator diagrams can also occur in other faults or when air enters the cylinder. The area of the indicator diagram on a known scale is equal to the work communicated to the fluid by the piston for one revolution of the crank. If the area of the indicator diagram is divided by the piston stroke s_0, we obtain the average indicator pressure p_i. The power p_i corresponding to the pressure is called the indicator power:

$$N = \frac{p_i \Omega_n s_0 n}{60 * 102}. \tag{5.2}$$

where: Ω_n - piston area, m^2;
S_o - piston stroke m; n - number of working strokes

Supervisory Questions:

1. What is the principle of operation and purpose of volumetric hydraulic machines
2. General properties of positive displacement pumps.
3. What are the different types of volumetric machines depending on their construction?
4. State the advantages and disadvantages of steam and direct-displacement pumps.
5. Principle of operation of volumetric hydraulic machines.
6. Throttling exit velocity and volumetric equation.
7. Characterisation and efficiency of volumetric hydraulic drives.

6.1. Schematic diagram and principle of operation of piston and plunger pump.

Let us consider the design and principle of operation of piston pumps (Fig. 6.1). The figure shows the circuit diagram of a simple-acting pump driven by rotary motion machines, e.g. an electric motor.

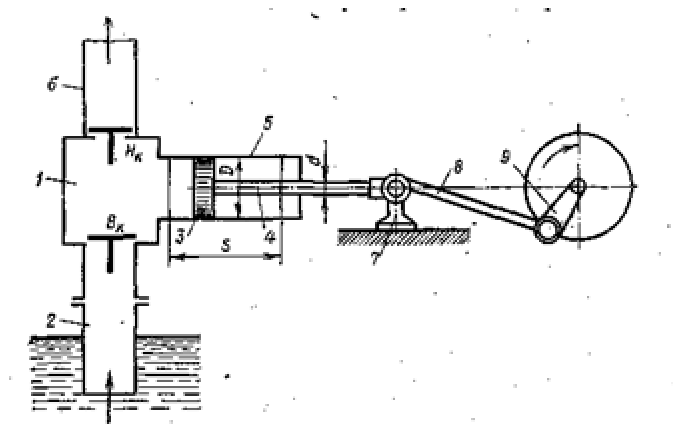


Figure 6.1.

The piston pump consists of a working chamber, inside which there are suction B_k and discharge I_a valves; cylinder-5, piston-3, making reciprocating motion inside the cylinder; suction 2 and pressure 6 connections. To convert the rotary motion of the crank 9 into reciprocating motion of the piston, the rod 4, slider 7 and connecting rod 8 are used.

The most common piston pump designs are described below. Piston pumps are classified according to their purpose, operating conditions and design features as follows.

By occupation action: 1) simple-acting pumps

2) double-acting pumps

Double-acting pumps have working chambers 1 and 2 on either side of the cylinder, each with discharge 5 and suction valves 4. Therefore, during the stroke of the piston 10, driven by the

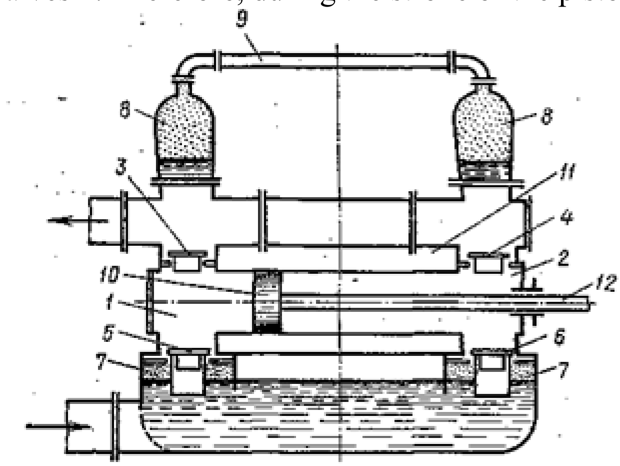


Figure 6.2.

when the piston rod 12 moves both to the left and to the right in the cylinder 11, both suction and discharge take place at the same time. For example, when the piston moves to the right, suction valve 5 is opened in chamber 1 and liquid is sucked in, and discharge valve 4 is opened in chamber 2 and liquid is supplied to the pressure pipe. In this way, almost double the volume of liquid is pumped per piston stroke (left and right movement) compared to a simple-acting pump.

Air hoods 7 on suction and on discharge, connected by tube 9, serve to reduce the pulsation of the pumped liquid;

3) built-up pumps (Fig. 6.3). They consist of three simple-acting cylinders, the pistons of which are mounted on a common crankshaft, the cranks being at an angle of 120° to each other. In this way, one portion of water is sucked in and discharged for every third revolution of the shaft, thus achieving a more uniform operation;

4) twin double-acting pumps. The pump consists of two double-acting pumps with common suction and discharge connections;

5) differential pumps (Fig. 6.4). In the differential pump the liquid supply is carried out more evenly, in two steps; during the stroke of the piston 2 to the left part of the liquid enters the right cavity of the cylinder 1, and during the stroke of the piston to the right it is supplied to the pipeline in the presence of only two valves 4 - suction and 5 - discharge

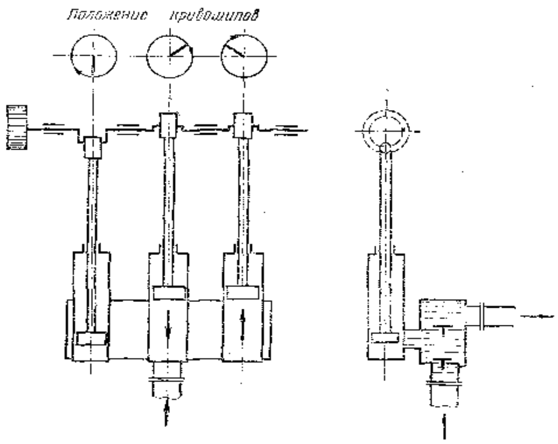


Рис. XI.3

Fig. 6.4 shows: 3 - suction air cap, 2 - discharge air cap, 7 - pressure connection, 8 - stem. The dimensions of the differential pump are almost the same as those of the simple pump. The rod 8 of the differential pump is made with a cross-sectional area equal to half the area of the piston; equal volumes are then delivered per stroke.

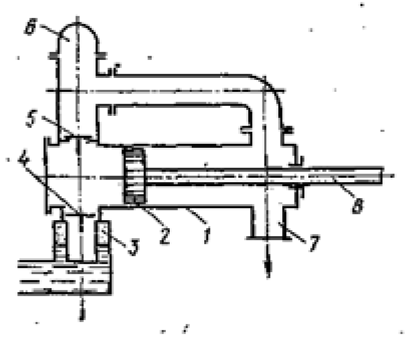


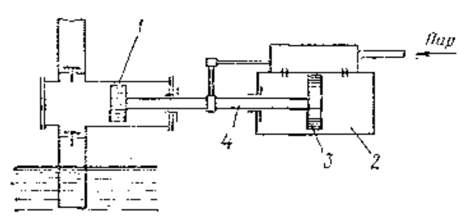
Рис. XI.4

By method of activation:

1) driven, operating from a separately located motor connected to the pump by a crank mechanism (Fig. 6.5) or other transmission;

2) steam - direct-acting; they have a common rod 4 (Fig. 6.6);) manual, manually actuated. These pumps of BKF type are widely used

Fig.6.6



By construction of the working body:

1) piston pistons proper, in which a disc piston moves in a bored cylinder. The piston is sealed with sealing rings or oil seals;

2) plunger (scalloped) (Fig. 6.7), in which the working organ is a plunger in the form of a hollow cup, which moves in the sealing gland without touching the inner walls of the cylinder, in operation these pumps are simpler, as they do not have replaceable piston rings, collars, etc.;

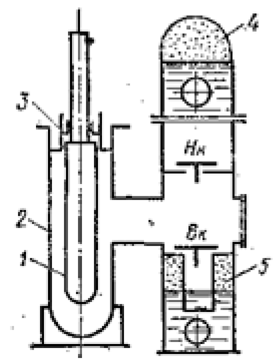


Fig.6.7

Fig.4.6 shows the scheme of such pump, where 1 - rolling pin; 2 - cylinder; 3 - gland; 4 - discharge air cap; 5 - suction air chamber; B_k and I_k - suction and discharge valves; 3) diaphragm diaphragms with flexible diaphragm made of rubberised fabric or leather;
4) deep-water pumps with through piston.
By purpose:
1) water; 2) sewage; 3) acid and alkaline; 4) oil, etc.

6.2 Piston pump delivery schedule.
When the piston enters the cylinder to the right, a volume of liquid is sucked into the cylinder

$$V = Fs,$$

where P - piston area, m^2; s - piston stroke, m.
When the piston is moved to the left, the same volume is pushed into the pressure pipe, i.e. one suction and one discharge occur during one stroke (one crank revolution) . If the crank makes n revolutions per minute, then theoretically the capacity is

$$Q = Fsn.$$

(in m^3/min) is equal to

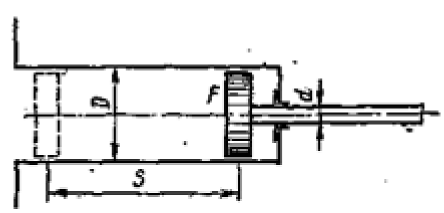


Рис. XI.7

The actual capacity will be less due to leakage of liquid through the glands, delayed opening and closing of valves, air separation from the liquid, all of which are taken into account by the volumetric efficiency of the pump or in terms of m^3/h.

$$Q = \eta_0 Fsn$$

$$Q = 60\eta_0 F s n.$$

The volumetric efficiency for different pumps varies within the range of
0,85-0,97.

Thus, the performance of a reciprocating pump is independent of head.

The capacity of the double-acting pump will be slightly less than double the capacity of the simple pump, due to the fact that in one of the chambers part of the volume is occupied by a rod with diameter A and cross-sectional area 1.

Therefore, for a double-acting pump

$$Q = 60\eta_0 (2F - f) s n. \tag{6.1}$$

Triple-acting pump capacity in m^3/h

$$Q = 60\eta_0 3 F s n; \tag{6.2}$$

dual dual action :

$$Q = 60\eta_0 2 (2F - f) s n. \tag{6.3}$$

Control Questions:

1. What does the design of piston pumps look like and the definition of their main dimensions?
2. What are the applications of reciprocating pumps?
3. Plunger pump design.
4. Scope of application of plunger pumps.
5. What does the indicator diagram express?
6. Efficiency of the reciprocating pump.
7. What does the shift graph express?

Reference words: valve bore diameter, piston pumps , thousands of atmospheres, mating with any motor, allowable voltage, given by the number, $D = \sqrt{\frac{4Q}{60K\pi 30 v_{cp}\lambda}}$, equation allows to calculate, suction air chamber, pumps can develop, head.

Volumetric rotary pumps and compressors.

7.1 Design and principle of operation of rotary piston machines.

The delivery of single-cylinder piston pumps is very uneven. A more uniform fluid delivery can be obtained by using multi-cylinder piston machines, the cylinders of which are combined into a common block. In multi-cylinder machines, the liquid is displaced in series by several pistons driven directly from a rotary motion motor. Such multi-cylinder piston hydraulic machines are called rotary piston machines. Depending on the way the pistons are driven, rotary piston machines are distinguished between rotating and stationary units. The cylinders can be arranged radially and axially in relation to the axis of the block. If the cylinders in the block are arranged radially, such hydraulic machines are called radial piston machines. If the cylinders are axially arranged in the block, the hydraulic machines are called axial piston hydraulic machines.

A characteristic feature of most rotary piston machines is the absence of suction and pressure valves. This feature allows them to be used at high speeds. In rotary piston hydraulic machines there is no usual crank-rod mechanism, but their kinematic basis is inversions of the crank-rod mechanism.

Rotary piston hydraulic machines are widely used in volumetric hydraulic transmissions and drives. They are used as constant and variable displacement pumps and rotary hydraulic motors with constant and variable torque.

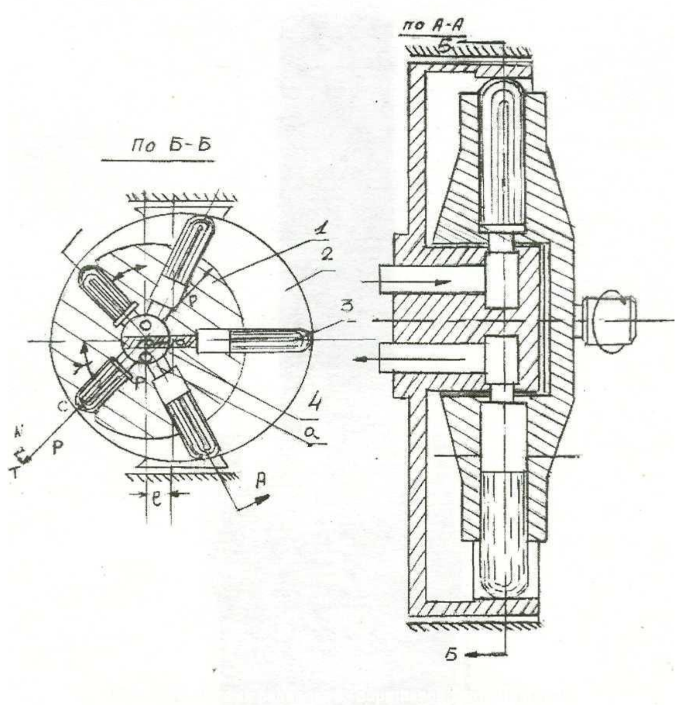


Figure 7.1.

Schematic diagram of a radial piston hydraulic machine.

In radial piston hydraulic machines (Fig. 7.1) the rotor *1* is located eccentrically relative to the stator *2*. Radial cylindrical holes (cylinders) are drilled in the rotor. Pistons *3* during rotation of the rotor perform reciprocating motion in the cylinders, sliding their spherical heads on the inner surface of the stator. The cylinder bottoms have through radial holes a, which communicate with the upper or lower segmental cut-out in the distributor journal *4.* The segmental cutouts of the journal are separated by a partition and form two chambers: when the rotor rotation is clockwise, the upper cutout is the suction chamber and the lower one is the discharge chamber. Pistons, connected at the moment with the upper cutout in the journal *4,* moving from the axis of rotation, suck liquid from the upper cutout into their cylinders and together with the rotating block, having passed the sealing partition of the journal, pass to the lower half of the machine. Here the pistons, moving towards the axis of rotation of the block, displace the liquid under pressure into the lower segmental cut-out of the trunnion, i.e. into the discharge chamber. Thus, with continuous rotation of the rotor, the liquid from the suction chamber to the discharge chamber is supplied. When pressurised fluid is supplied to the upper trunnion cutout, the rotor rotates, discharging fluid through the lower cutout of the distributor trunnion, i.e. the radial piston hydraulic machine operates as a hydraulic motor.

7.2 Radial piston pumps.

Radial piston pumps. From the diagram of radial piston hydraulic machines, it can be seen that the delivery of a radial piston pump depends on the value of the eccentricity e. In adjustable pumps, the eccentricity can be varied in magnitude by shifting the stator in the housing guides. Fig. 7.1 shows the construction diagram of a radial piston pump with nine cylinders. In the casing *1* is installed stator *2*, in which eccentrically located rotor *3,* rotating on a fixed distributor trunnion *4.* In this journal are cut out distribution grooves and channels through which the liquid is supplied and discharged. The stator is mounted on the frame *5*. The pistons *6* are connected with the stator by their rollers *7*, in which the corresponding grooves are made for this purpose. The frame *5* can be moved, changing the eccentricity *e* by means of the mechanism *8. The* rotor shaft *9* is connected to the motor. In the distributor journal, the suction cavity is labelled *11* and the discharge cavity *10.* Openings *12 and 13 are* connected to the cavity of the distributor slots by axial drilled holes in the journal *4* and serve to connect the suction and pressure pipes. Stator is mounted on the frame on ball bearings *14.* The motor shaft *16* is connected to the rotor shaft *9 by* a cam coupling *15*. When adjusting the pump, the frame *5* moves in guides *17.*

The average delivery of a radial piston pump Q_o can be determined using the general formula:

$$Q_0 = q_T n$$

Expressing the stroke of the piston through the eccentricity *e:*

$$Q_q = qn = \eta_0 \tfrac{\pi d^4}{4} szn = \eta_0 \tfrac{\pi d^4}{4} ezn, \tag{7.1}$$

where *Z is the* number of pistons;

d - piston diameter

e - eccentricity;

S = *2e* - piston stroke across the cylinder.

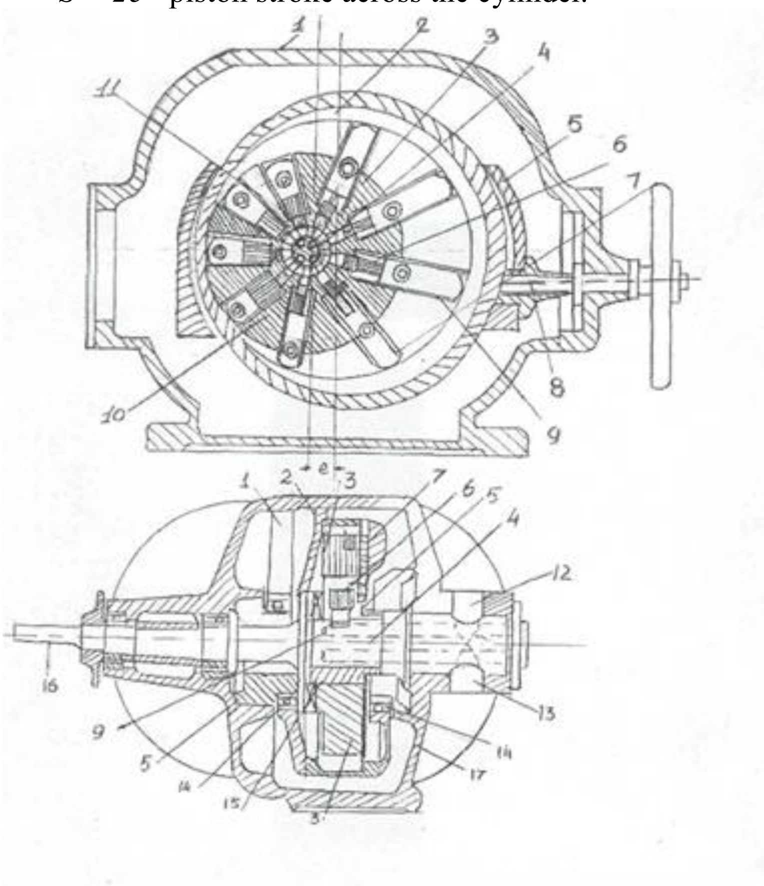


Fig.7.1 ***Design diagram of the radial piston pump***

7.3 Axial piston pumps.

Axial piston pumps. Axial piston pumps and hydraulic motors are high-speed variable displacement hydraulic machines. In these hydraulic machines, the axes of the cylinders (pistons) are arranged parallel to the axis of the block uniformly along the circumference r_0. This makes the machine more compact, reduces the rotor's moment of inertia and allows the rotational speed to be increased.

Fig. 7.3. shows an axial piston hydraulic machine with an inclined disc (washer), in which the block has cylindrical holes (cylinders) parallel to its axis of rotation.

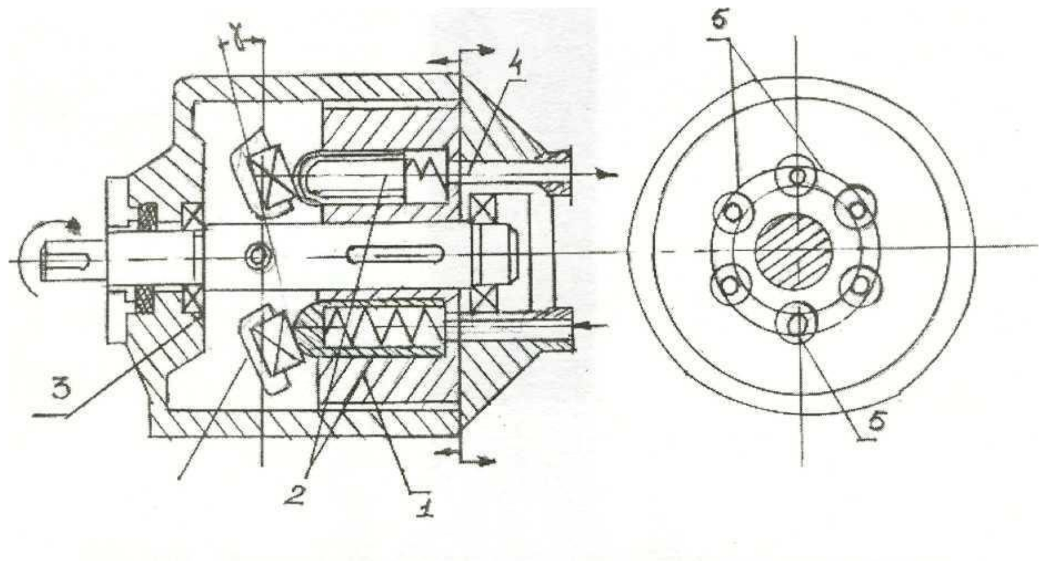


Fig. 7.3. ***Principle diagram of axial piston hydraulic machine.***

Pistons 2, pushed out of the cylinders by springs, rest their spherical heads on a fixed inclined disc 3. When the block rotates, the pistons resting on the disc make reciprocating motion relative to the cylinders. In the cover 4, to which the rotor tightly fits its end, there are two arc-shaped grooves 5, separated by a sealing partition 6. One of the grooves communicates with the suction line and the other with the pressure line.

During the rotation of the unit, the bottom holes of the cylinders, moving along the arc-shaped grooves, connect the cylinder cavities with the suction line and with the pressure line. At the moment when the bottom hole crosses the partition 6, the liquid filling this cylinder is transferred from the suction cavity to the discharge cavity.

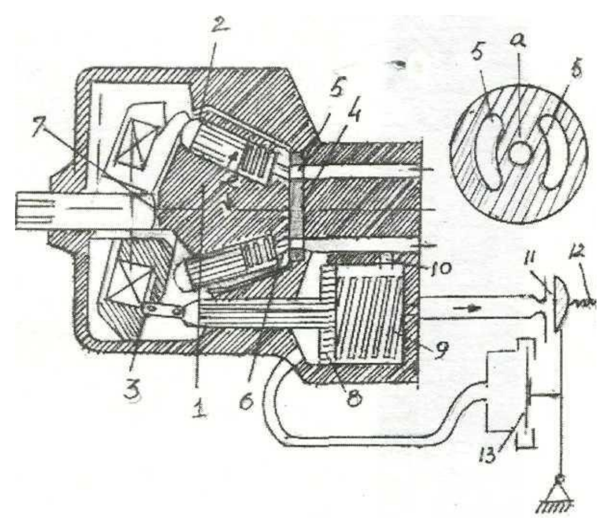


Fig. 7.4**.** ***Structural diagram of the adjustable axial-axial-axial of the piston pump.***

The principle diagram of axial piston hydraulic machines shows that the delivery of an axial piston pump depends on the angle γ of the inclined disc (washer). In adjustable pumps, the angle can be changed by rotating the disc about an axis perpendicular to the axis of rotation of the unit. Fig. 7.4 shows the structural diagram of an adjustable axial piston pump. The pump consists of a cylinder block 1. having usually 7 or 9 parallel cylinders. In each cylinder moves a piston 2, resting on an inclined disc 3, fixed with a thrust bearing on the cage 7, which is connected to the pump casing. The cage together with the disc is inclined to a plane perpendicular to the axis of the block.

As the rotor shaft rotates, the pistons will make a reciprocating movement relative to the cylinders in the block. In one revolution of the shaft, each piston will make two strokes on the cylinder: a discharge stroke and a suction stroke. The right end part of the cylinder block in rotation slides on the plane of the fixed distributor head 4, which has two arc-shaped grooves 5, which are separated by sealing bridges a. One of the arc-shaped grooves serves as a suction chamber for supplying fluid to the suction ports of the cylinders, and the other is a discharge chamber for the discharge of fluid supplied by the pistons through these holes under pressure. The centre of the lintels coincides with the centre of the lintel, the piston is in the neutral position (dead centre) and its velocity relative to the cylinder is zero.

The pump delivery is regulated by changing the angle γ by turning the cage and with it the inclined disc. Rotation of the cage is carried out by a pull rod when liquid is supplied from the pressure pipe under the piston 8 as a result of pressure increase above the set pressure due to reduction of flow rate in the pressure pipe. At the same time, the liquid from the pressure pipeline flows to the diaphragm 13, through which affects the valve 11, providing free release of liquid from the cavity of the spring 9 through the opened valve 11. At the same time the rod together with the piston 8 will go to the right, reducing the angle γ , and hence the supply Q. After the feed rate has decreased to the set value, the movement of the piston 8 will stop due to the equalisation of the forces acting on it from the left and from the side of the spring 9. In the cavity of spring 9 with the help of jets 10 and valve 11 the pressure is kept lower than in the pressure pipe, due to hydraulic losses during continuous movement of liquid from the pressure chamber through the jets to the cavity of spring 9 and further through valve 11 to drain into the receiving tank of the pump. If the pressure in the pressure chamber changes as a result of a change in the flow rate in the system, the pump delivery will automatically change due to the piston 8 taking a different position in its cylinder.

7.4 Rotary vane pumps.

Rotary vane pumps. Rotary-plate (gate) hydraulic machines are known in mechanical engineering as "vane" and are the simplest of the existing types of volumetric hydraulic machines. The name "vane" does not correspond to the principle of operation and design of this type of hydraulic machines, and according to the new terminology they are called rotary-plate pumps.

The scheme of the simplest rotary vane pump is shown in Fig. 7.5. The rotor 1 is placed in the pump casing between two tightly pressed to the

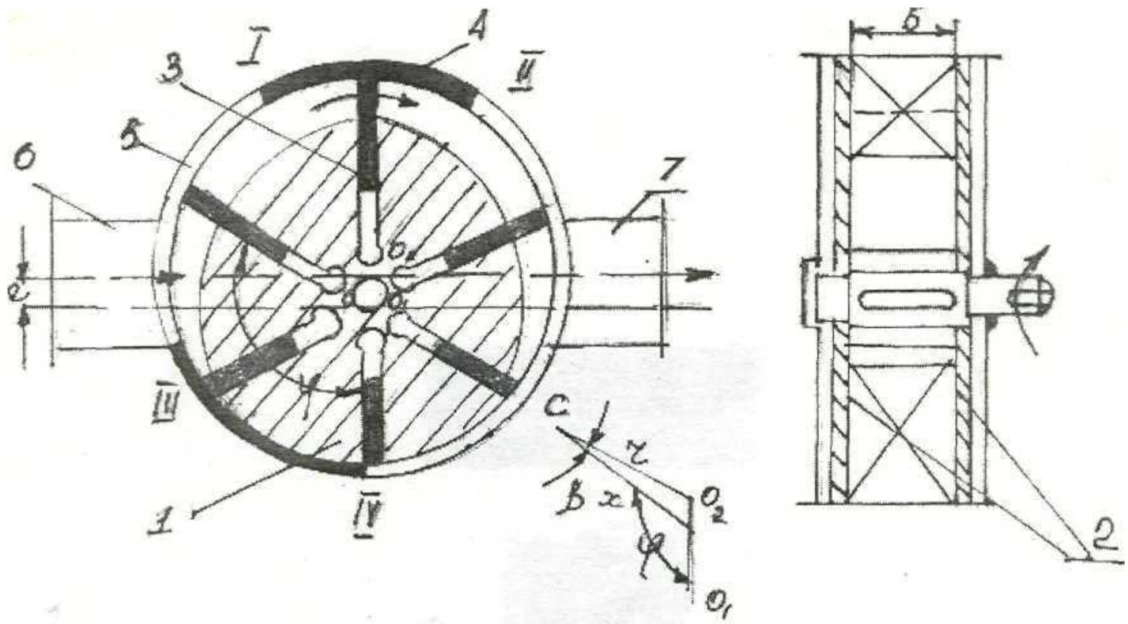


Fig. 7.5. ***Schematic diagram of the simplest rotary vane pump***.

to it by end discs 2. In the radial or slightly inclined to the rotor radius grooves there are plates (screens) 3. The rotor rotation axis is located eccentrically in relation to the stator 4. Pressed against the stator and rotating together with the rotor, the plates slide on the inner cylindrical surface of the stator, simultaneously reciprocating movement relative to the rotor in its slots.

Due to the eccentricity of the rotor, as the plates move away from the point where the distance between rotor and stator is minimum, the volume of the cavity between the plates increases. As a result, the pressure decreases and the cavity is filled with oil entering through window 5 located on the periphery of the stator and communicating with the suction connection 6 of the pump. The oil entering the suction cavity is carried by the plates in the direction of rotation of the rotor. Then, when the plates pass the point of maximum distance between the rotor and the stator, the volume of space between the plates starts to reduce and the oil is forced into the discharge cavity through the opposite window into the discharge connection 7 of the pump. Vane pumps are manufactured with constant flow and with adjustable flow. Adjustment of vane pumps is carried out by changing the eccentricity e. The delivery of vane pumps is pulsating.

The minimum delivery occurs when the plate is at the position corresponding to the largest distance between stator and rotor. Thereafter, the pump delivery decreases again and reaches its minimum at the moment the plate is out of operation.

It is recommended to use from 4 to 12 plates to reduce the pulsation of the liquid supply. To eliminate the possibility of connecting the discharge cavity with the suction cavity, sealing projections I - II and III - IV are provided. The length of sealing protrusion I - II is made in such a way that at the moment when one plate enters the sealing protrusion, the previous plate goes beyond it. In order to eliminate the pinching of oil in the closed volume, the protrusion III - IV in front of the suction chamber is shorter than the protrusion I - II in front of the discharge chamber. In the pump under consideration, each plate takes part once in the suction of oil and once in the discharge during one revolution of the rotor. These machines are called single-acting rotary vane pumps.

Rotary vane pumps are characterised by high flow rates with relatively small pump dimensions.

To determine the theoretical delivery of a single-acting rotary vane pump, assume that the pump has an infinite number of infinitely thin plates. Fig. 15.5 shows the calculation diagram of such a pump. From ΔO_1O_2C it follows that:

$$x = r\cos\beta - e\cos(180^0 - \varphi) = r\cos\beta - e\cos\varphi \tag{7.2}$$

The working part of the plate *h is* equal to:

$$h = x-(r-e) \tag{7.3}$$

Substituting in (7.3) the value of (7.2), we obtain the formula for determining h:

$$h = r\cos\beta - e\cos\varphi - (r-e) = e(1-\cos\varphi) + r(\cos\beta - 1).$$

Since the value of the ratio $\frac{e}{r}$ in rotary-plate pumps is small enough, angle β ≈0 and cos β≈ 1, the size of the working part of the plate can be determined by the formula:

$$h = e(1-\cos\varphi) \tag{7.4}$$

The elementary volume dq_T, transferred from the suction cavity to the discharge cavity, when the rotor is rotated by an angle dφ is equal to :

$$dq_T = hbrd\varphi \tag{7.5}$$

where b is the rotor width; g r is the rotor radius.

Integrating expression (7.5) within the range 0-2π, we obtain the specific theoretical delivery of the rotary vane pump for one revolution of the rotor:

$$q_T = \int_0^{2\pi} hbrd\varphi = erb\int_0^{2\pi}(1-\cos\varphi)d\varphi = 4\pi erb. \tag{7.6}$$

The average theoretical pump delivery per unit time will be:

$$Q_{0T} = q_T n = 4\pi erbn \tag{7.7}$$

Taking into account a finite number of plates z and their finite thickness δ, as well as oil leakage through seals, estimated by volumetric efficiency, we obtain a formula for determining the average delivery of the rotary vane pump:

$$Q_0 = \eta k_z Q_{OT} = 2\eta_0 be(2\pi r - z\delta)n \tag{7.8}$$

The displacement of a double-acting pump will be equal to twice the displacement determined by formula (5.7). In double-acting rotary vane pumps, in the section between the suction and discharge windows, and the guide is profiled on a circle circumscribed from the centre of the rotor, and in the sections occupied by the windows - on Archimedes' spiral. The average flow rate of a double-acting pump is equal to:

$$Q_0 = 2\eta_0 b\left[\pi(r_2^2 - r_1^2) - \frac{(r_2 - r_1)\delta z}{\cos\alpha}\right]n$$

where: r_1 and r_2 are the large and small stator radii; b is the rotor width (plate length);
δ - plate thickness; z - number of plates;- plate inclination angle (for radial plates cos a= 1).

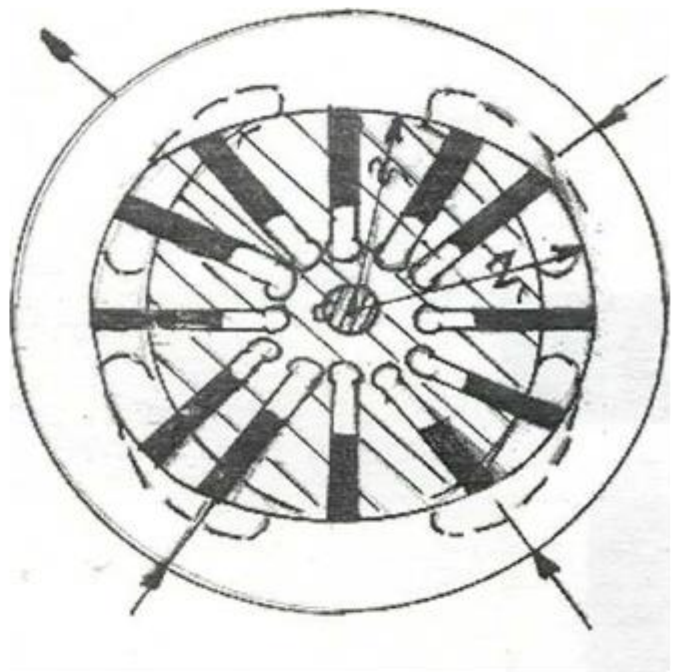


Fig.7.6 **Schematic diagram of double-acting rotary vane pump**.

7.5 Gear pumps.

Gear pumps. The casing 4 of such a pump contains

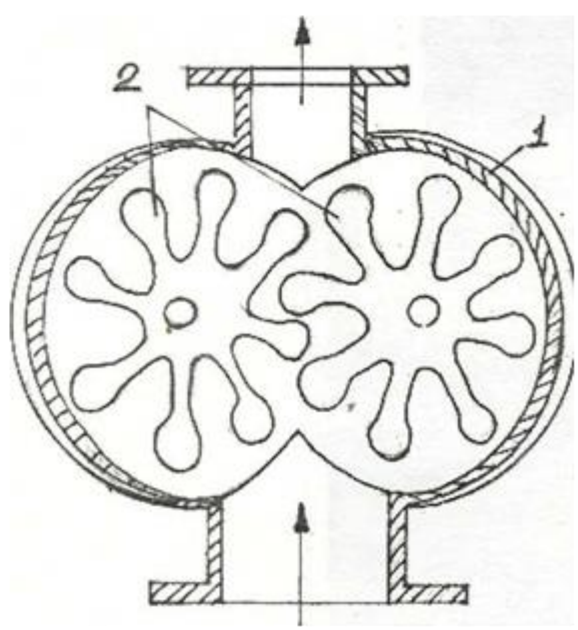


Fig.7.7 **Schematic diagram of the gear pump:**

two gears 2, one of which (driving gear) is driven by an electric motor. When the gear teeth are out of mesh, a vacuum is created, which causes the liquid to be sucked in. It enters the casing, is caught by the gear teeth and moves along the

1- housing; 2- gears

The liquid is displaced and flows into the pressure line. In the area where the teeth engage again, the liquid is displaced and flows into the pressure line.

7.6 Screw Pumps.

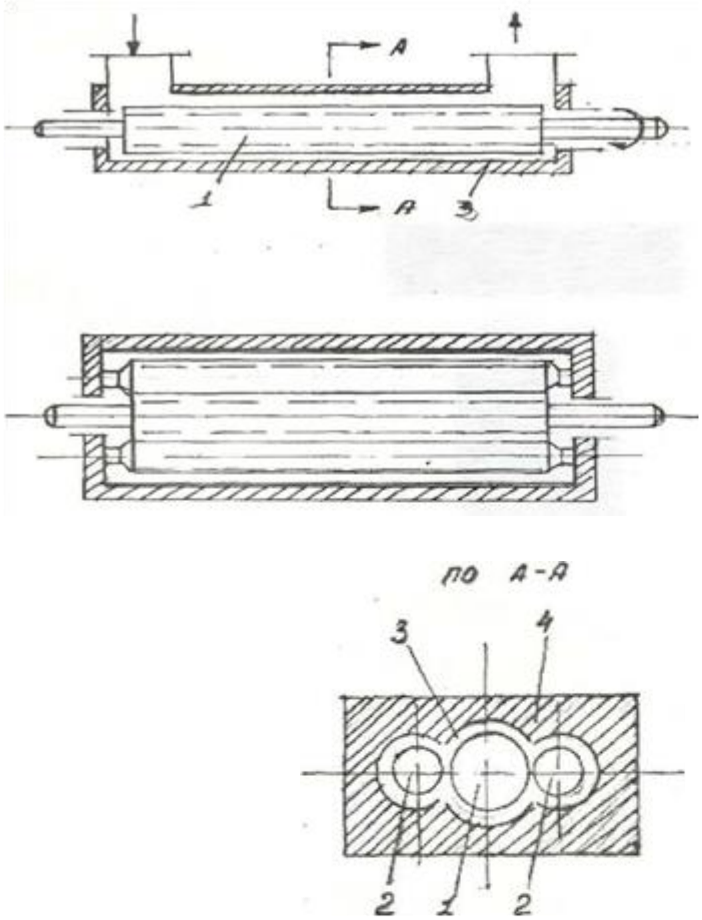


Figure 7.8.

Schematic diagram of a screw pump: 1 - master screw; 2 - slave screw; 3 - casing; 4-casing, is a master screw 1 and several slave screws 2, enclosed in a casing 3, located inside the casing 4. Pumps with three screws - one master screw and two slave screws (as shown in Fig. 7.8) are predominantly used in industry. The casing 3 consists of three interlocking cylindrical cavities within which the screws rotate: the middle one - the master screw - and two identical slave screws of smaller outside diameter. The screws are meshed. The screw threads have a special shape and form hermetic seals at the points where the screws touch each other, which divide the pump lengthwise into a series of closed cavities. The threading direction of each driven screw is opposite to that of the master screw. For example, if the master screw has right-hand threading, the slave screws have left-hand threading. All screws are usually double-riveted.

Supervisory Questions:

1. What do you know about positive displacement rotary pumps and their applications?
2. Which pumps are called gear pumps? Principle of operation.
3. Define rotary vane pumps.
4. Screw pumps and their principle of operation.
5. What is the principle of rotary piston machines
6. Tell us about axial piston pumps.

Reference words: pumps, positive displacement rotary pumps and compressors, multi-cylinder, radial piston, constant and variable torque rotary hydraulic motors, $Q_0 = q_T n$ **, axial** piston **pumps,** flow reduction in discharge pipework, screw touching, hermetic seals, gear pumps, binder pumps, rotary vane pumps, teeth, jumper centre, two-stage screw compressors.

8.1 Rotary vane pumps and hydraulic motors.

Rotary-plate (gate) hydraulic machines are known in mechanical engineering as "vane" and are the simplest of the existing types of volumetric hydraulic machines. The name "vane" does not correspond to the principle of operation and design of this type of hydraulic machines, and according to the new terminology they are called rotor-plate hydraulic machines.

The scheme of the simplest rotary vane pump is shown in Fig. 8.1. The rotor 1 is placed in the pump casing between two tightly pressed to the

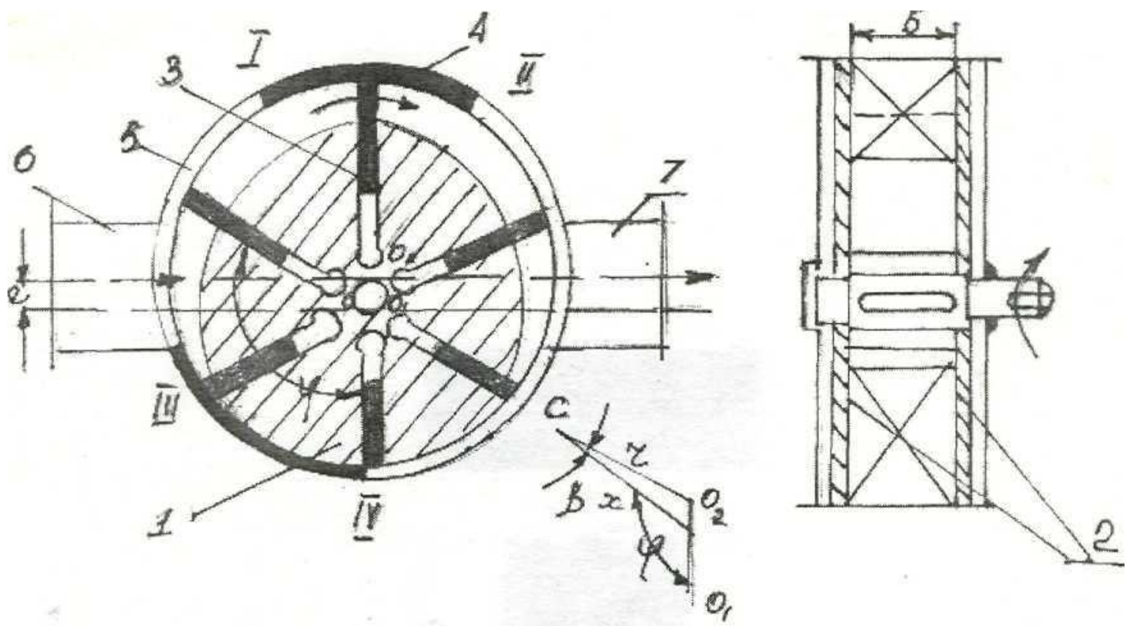


Fig. 8.1. **Schematic diagram of the simplest rotary vane pump**.

to it by end discs 2. In the radial or slightly inclined to the rotor radius grooves there are plates (screens) 3. The rotor rotation axis is located eccentrically in relation to the stator 4. Pressed against the stator and rotating together with the rotor, the plates slide on the inner cylindrical surface of the stator, making simultaneously reciprocating movement relative to the rotor in its slots.

Due to the eccentricity of the rotor, as the plates move away from the point where the distance between rotor and stator is minimum, the volume of the cavity between the plates increases. As a result, the pressure decreases and the cavity is filled with oil entering through the window 5 located on the periphery of the stator and communicating with the suction connection 6 of the pump. The oil entering the suction cavity is carried by the plates in the direction of rotation of the rotor. Then, when the plates pass the point of maximum distance between the rotor and the stator, the volume of space between the plates starts to reduce and the oil is forced into the discharge cavity through the opposite window into the discharge connection 7 of the pump. Vane pumps are manufactured with constant flow and with adjustable flow. Adjustment of vane pumps is carried out by changing the eccentricity e. The delivery of vane pumps is pulsating.

The minimum delivery occurs when the plate is at the position corresponding to the largest distance between stator and rotor. Thereafter, the pump delivery decreases again and reaches its minimum at the moment the plate is out of operation.
It is recommended to use 4 to 12 plates to minimise pulsation of the liquid supply. To eliminate the possibility of connecting the discharge cavity with the suction cavity, provides sealing projections I - II and III - IV. The length of sealing protrusion I - II is made in such a way that at the moment when one plate enters the sealing protrusion the previous plate goes beyond it. In order to eliminate the pinching of oil in the closed volume, the protrusion III - IV in front of the suction chamber is shorter than the protrusion I - II in front of the discharge chamber. In the pump under consideration, each plate takes part once in the suction of oil and once in the discharge during one revolution of the rotor. These machines are called single-acting rotary vane pumps.
Rotary vane pumps are characterised by high flow rates with relatively small pump dimensions.
To determine the theoretical delivery of a single-acting rotary vane pump, assume that the pump has an infinite number of infinitely thin plates. Fig. 18.5 shows the calculation diagram of such a pump. From ΔO_1O_2C it follows that:

$$x = r\cos\beta - e\cos(180^0 - \varphi) = r\cos\beta - e\cos\varphi \quad (8.1)$$

The working part of the plate *h is* equal to:

$$h = x-(r-e) \quad (8.2)$$

Substituting in (8.2) the value of *x* in (8.1), we obtain the formula for determining h:

$$h = r\cos\beta - e\cos\varphi - (r-e) = e(1-\cos\varphi) + r(\cos\beta - 1).$$

Since the value of the ratio $\frac{e}{r}$ in rotary-plate pumps is small enough, angle β ≈0 and cos β≈ 1, the size of the working part of the plate can be determined by the formula:

$$h = e(1-\cos\varphi) \quad (8.3)$$

The elementary volume transferred from the suction cavity to the discharge cavity when the rotor is rotated by an angle dφ is equal to:

$$dq_T = hbrd\varphi \quad (8.4)$$

where b is the width of the rotor;
r- radius of the rotor.

Integrating expression (8.4) within the range 0-2π, we obtain the specific theoretical flow rate of the rotary vane pump for one rotor revolution:

$$q_T = \int_0^{2\pi} hbrd\varphi = erb\int_0^{2\pi}(1-\cos\varphi)d\varphi = 4\pi erb. \quad (8.5)$$

The average theoretical pump delivery per unit time will be:

$$Q0T = qTn = 4\pi erbn \quad (8.6)$$

Taking into account a finite number of plates z and their finite thickness δ, as well as oil leakage through seals, estimated by volumetric efficiency, we obtain a formula for determining the average delivery of the rotary vane pump:

$$Q_0 = \eta k_z Q_{OT} = 2\eta_0 be(2\pi r - z\delta)n \quad (8.7)$$

The displacement of a double-acting pump will be equal to twice the displacement determined by formula (8.7). In double-acting rotary vane pumps, in the section between the suction and discharge windows, and the guide is profiled on a circle circumscribed from the centre of the rotor, and in the sections occupied by the windows - on Archimedes' spiral. The average flow rate of a double-acting pump is equal to:

$$Q_0 = 2\eta_0 b\left[\pi(r_2^2 - r_1^2) - \frac{(r_2 - r_1)\delta z}{\cos\alpha}\right]n$$

where: r_1 and r_2 are the large and small stator radii; b is the rotor width (plate length);
δ is the thickness of the plate;
z is the number of plates;
α - angle of inclination of plates (for radial plates cos a= 1).

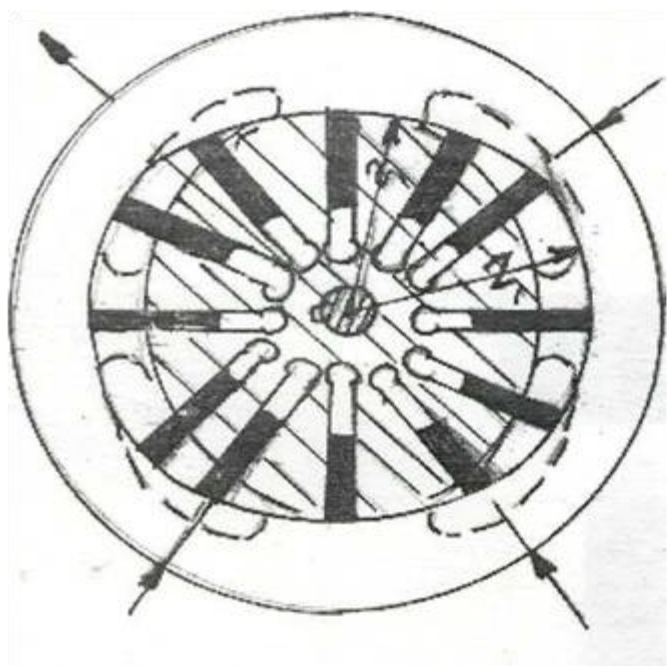


Fig.8.2 ***Schematic diagram of double-acting rotary vane pump***.

Supervisory Questions:
1. List the basic properties of a liquid at rest?
2. Write the differential equation of equilibrium of a fluid (Euler's equation)?
3. What is called the free surface of a liquid?
4. the basic equation of hydrostatics?
5. Concept of piezometer, piezometric height and vacuum?
6. Instruments and methods of measurement, pressure?

Reference words: rotary piston hydraulic machines, radial piston pumps and hydraulic motors, axial piston pumps and hydraulic motors, URS regulator, pistons, oil.

9.1 Tasks of hydrodynamic transmissions and their field of application.

In machines, it is often necessary to transfer mechanical energy between shafts rotating at different and variable angular velocities during operation. Such a task can be solved by means of hydrodynamic feeding, where there is no direct contact between the driving and driven links, but the motion is transmitted through an intermediate medium. The intermediate medium is a droplet liquid.

A hydrodynamic transmission is a mechanism consisting of two vane machines (a centrifugal pump and a vane turbine) extremely close together in one housing, the connection between which is carried out by a closed fluid flow. The simplest hydrodynamic transmission is a hydraulic coupling, which serves for elastic connection of shafts.

Often the characteristics of the machines between which mechanical energy is to be transferred do not match. As a result, machines operate at uneconomical modes, with high overloads or underloads, etc. Matching of machine characteristics can be achieved by using hydrodynamic transmissions.

Hydraulic couplings and torque converters are used in diesel locomotives, automobiles, drives of powerful fans and pumps, in ship and drilling rigs, in earthmoving and road building machines

9.2 Hydrodynamic transformers.

In contrast to a fluid coupling, a torque converter transfers mechanical energy between coaxial shafts with a change in torque. As a rule, torque converters are used to increase the torque on the driven shaft. In terms of their function they correspond to a variator. With automatic, stepless variation of the speed of the driven shaft. The torque converter housing is externally supported to absorb the reactive torque generated by the reactor blades, which are connected to the housing. Hydrotransformers can be designed as three-, four-, and multi-wheeled with a single-stage pump, single-, two-, and three-stage turbine with one or more reactors.

9.3 Devices and principle of operation of the hydrodynamic coupling.

Hydraulic clutches are divided into unregulated and regulated clutches. Non-adjustable clutches are those in which the speed of the driven shaft depends only on the load torque on the driven shaft at a constant drive shaft speed. In an adjustable fluid coupling, the speed of the driven shaft also depends on the position of an externally controlled regulating device. Both non-adjustable and adjustable fluid couplings can be of constant or variable speed. The increase of the transmitted torque by 20-25 times compared to the calculated one at slip variation within 0.03's'1 is a big disadvantage of constant-fill couplings and in most cases is completely inadmissible, as such a coupling cannot be a reliable safety device.

The hydraulic coupling can be used for power absorption as a hydraulic brake, if the driving shaft of the coupling is connected to the tested engine, and the braking torque is created on the driven (turbine) shaft of the hydraulic coupling. In this case it is necessary to intensively remove heat from the flow cavity of the hydraulic coupling, creating circulation of the working fluid between the flow cavity and the cooler.

The value of meridional velocity c_m depends on the difference of heads created by the wheels of the hydraulic coupling in the interwheel (axial) gap δ. At small slips the pressure field in the rotating cavity of the hydraulic coupling is very close to the pressure distribution at relative equilibrium of liquid in rotating vessels. Therefore, the pressure drop in the interwheel gap and its corresponding head can be determined by the equation for the relative equilibrium of the fluid. Experimental studies and calculations show that the angular velocity of the fluid ω_w between the rotating wheel and the stationary wall parallel to it can be assumed to be equal to half of the angular velocity of the wheel ω:

$$\omega_{1ж} = \frac{\omega_1}{2} \quad \omega_{2ж} = \frac{\omega_2}{2}$$

where $\omega_{1(g)}$ and ω_{2g} are the angular velocities of the fluid caused by the rotation of the pump and turbine wheel;
ω_1, ω_2-the angular velocities of the pump and turbine wheel.
The head developed by the pump wheel at the centre of the axial clearance δ, according to the equation for relative equilibrium, is equal to

$$H_{1\infty} = \frac{r_2^2 - r_1^2}{2g} \frac{\omega_1^2}{4}$$

The head developed by the turbine wheel:

$$H_{2\infty} = \frac{r_2^2 - r_1^2}{2g} \frac{\omega_2^2}{4}$$

The resulting head in the gap δ produced by the two wheels will be equal to the difference of the heads $H_{1\infty}$ and $H_{2\infty}$:

$$H_{12\infty} = H_{1\infty} - H_{2\infty} = \frac{(r_2^2 - r_1^2)(\omega_1^2 - \omega_2^2)}{8g}$$

Head H(12∞) (causes annular motion of the fluid with meridional velocity) (c)m, therefore

$$H_{12\infty} = \frac{C_m^2}{2g}$$

And the value of meridional velocity

$$C_m = \sqrt{2gH_{12\infty}}$$

Substituting the value of H12∞ (from) equation

$$H_{12\infty} = H_{1\infty} - H_{2\infty} = \frac{(r_2^2 - r_1^2)(\omega_1^2 - \omega_2^2)}{8g}$$

into the formula $C_m = \sqrt{2gH_{12\infty}}$, get the value of meridional velocity

$$C_m = \sqrt{(r_2^2 - r_1^2)\frac{\omega_1^2 - \omega_2^2}{4}}.$$

9.4 Hydrodynamic couplings.

The fluid coupling transmits the torque from the drive shaft to the driven shaft unchanged because the housing, which forms a solid cavity, rotates freely with the pump wheel.

For fluid couplings, the moment equations take the form:
$M_1+M_2=0$.

The difference between the number of revolutions of the pump wheel and the turbine wheel of the fluid coupling, divided by the number of revolutions of the pump wheel, is called the slip of the fluid coupling:

$$s = \frac{n_1 - n_2}{n_1}$$

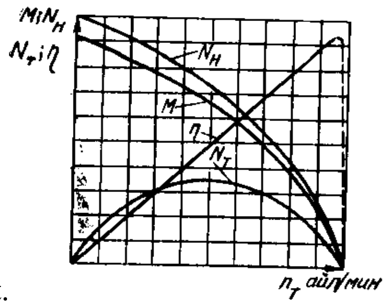
The hydraulic power of the hydraulic coupling is equal to

$$N = \lambda_{N\infty} D^2 n_1 \gamma$$

where $\lambda_{N\infty} = \frac{\lambda_{M\infty}}{974{,}5}$ is the hydraulic power factor of the hydraulic coupling.

9.5 Characteristics of the hydrodynamic coupling.

The external characteristic of the hydraulic coupling is the dependence of torque M and efficiency of the hydraulic coupling on the number of revolutionsn2 of the turbine wheel. The external characteristic of the hydraulic coupling (Fig.17.1.) is built according to the results of its tests. Characteristics of the hydraulic coupling are taken at its full and partial filling with working fluid. Reducing the filling of the working cavity of the hydraulic coupling with fluid equivalent to an increase in the bushing ratio $\bar{r}_0 = \frac{r_0}{k}$ and reduces the transmitted



moment.

The universal characteristic of a fluid coupling is the dependence of its torques on the number of revolutions of the turbine wheel at different revolutions of the pump wheel.

9.6 Hydraulic coupling and motor working together.

Hydraulic couplings are used to protect engines from dangerous overloads and to change the speed of various machines. Non-adjustable hydraulic couplings are used to protect machines from dangerous overloads while achieving smooth torque transmission. Adjustable hydraulic couplings are safety couplings and at the same time allow to adjust the speed of different machines. They are particularly useful for changing the speed of machines driven by an unregulated AC motor. Adjustable hydraulic couplings are usually used to change the number of revolutions of vane machines, the power of which is proportional to the cube of the number of revolutions (draught blowers fans, centrifugal pumps), since the nature of the change in the torque transmitted by the hydraulic coupling $M_1=-M_{(2)}=\lambda_M \gamma n_1^2 D^5$ corresponds to the characteristics of these machines. Selection of hydraulic couplings is made according to the characteristics using the formula $N=\lambda_{N\infty} D^5 n_{(1)\gamma}$ and the subsequent construction of joint characteristics of the motor - hydraulic coupling.

Supervisory Questions:

1. The field of application of hydrodynamic gears and their tasks.
2. Explain hydrodynamic couplings.
3. Principle of operation of a hydrodynamic transformer.
4. Devices and principle of operation of hydrodynamic coupling.
5. Main parameters of the hydraulic coupling.
6. Characterisation of the fluid coupling .
7. Joint operation of the hydraulic clutch and the motor.
8. Regulation of the hydraulic clutch operation.

Reference words: switchgear, servo-operated universal safety relief valves, throttling devices, head losses in the damper, over-valve cavity, hydraulic accumulator, filter materials, auxiliary devices, tanks, filters, hydraulic multipliers and accumulators.

SWITCHGEAR.

10.1 Main types of switchgear: spool, crane, valve.

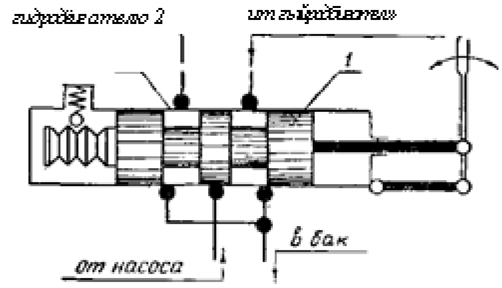


Fig.10.1.

Distributors are designed to distribute and change the direction of fluid flow between units and elements of hydraulic drive. According to design features, distributors are divided into crane, spool and valve distributors.

Two-, three- and multi-position valves are distinguished by the number of fixed positions.

Spool valves (spool valves) are the most widely used in volumetric hydraulic drives. Spool valves are controllable elements of hydraulic equipment, with the help of which fluid distribution, reversing of movement and switching of pipelines are carried out. The movable link of the spool valve (Fig.10.1) is made in the form of a plunger 1 with grooves for liquid passage and a cylindrical body *2* with holes for liquid inlet and outlet. By displacement of the plunger 1 relative to the housing *2* of the spool during operation of the hydraulic actuator it is possible to change the direction of fluid movement due to the corresponding overlapping of the working windows of the spool pair.

Spool movement control can be manual, cam, electromagnetic and hydraulic.

Fig. 10.2 shows the scheme of reversing spool valve of G-72 type with hydraulic control, which consists of body 1, plunger *2*, covers 1, throttles *4,* ball valves *3.* Plunger *2* of the spool can occupy two extreme positions - right and left, corresponding to two directions of hydraulic motor movement. The plunger *2* is moved from one position to another by the pressure *p* of the fluid, which is supplied to the plunger

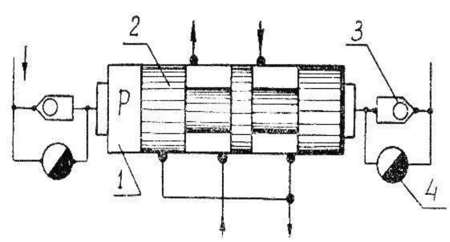


Fig.10.2.

under the plug ends. The ram travelling speed is regulated by means of throttles *4,* check valves *3* provide Fig. 10.2 dependent regulation. Valve distributors have advantages over slide valve distributors at low flow rates but high pressures due to high tightness, compactness and ease of operation.

Thus, in hydraulic drives for which tightness is not crucial and oil consumption is high, it is advisable to use spool valves.

In hydraulic drives with low oil flow rates at high pressures, where reliable sealing plays a decisive role, valve manifolds are used.

10.2 Valves

Valves are the most common elements of hydraulic drives. They are used to protect hydraulic drive units from overloads, set a certain sequence of work of units, create a certain direction of flow, set a given pressure, divide the flow into parts, create a constant pressure drop, etc. Valves are grouped by purpose, operating principle and design. Often one and the same valve can fulfil different functions depending on its connection in the system and its setting.

There are three groups of valves: check valves, safety valves (overflow and underflow valves) and pressure reducing valves.

Check valves are designed to allow fluid to flow in one direction *only.* If the direction of flow is changed, the check valve closes and the flow of liquid is stopped. When open, check valves must have minimum resistance and when closed, they must provide the specified tightness.

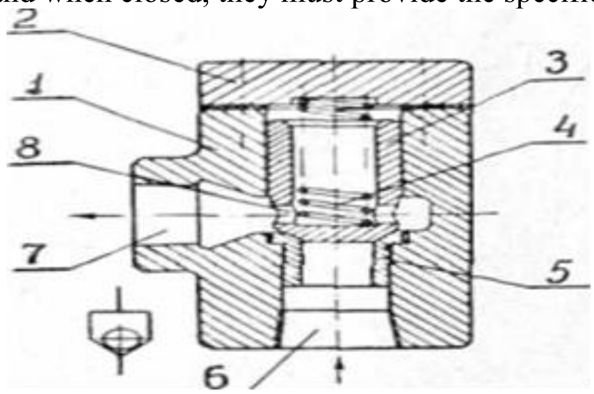


Fig.10.3.

The spring force of the check valve must therefore be kept to a minimum, sufficient only to seat the valve securely, as the valve is opened and closed by the force of the liquid pressure.

Fig.10.3. shows a plug check valve type G-51, which consists of body 1, cover *2,* plug *3,* spring *4* and seat *5.* Plunger *3* with its conical end is pressed against seat *5,* cylindrical side surface of the plunger enters the guide hole of the body.

When the valve is operating, the liquid supplied to orifice *6* lifts the plug off the seat and opens the passage to orifice 7.

As the flow direction changes, the plug *3* is pressed firmly against the seat *5 by* the fluid pressure. This pressure acts over the entire valve cross-section as oil enters *the above-valve cavity* through plug *3* through bore *8.* The pressure on the plug increases as the pressure rises, which stops the flow of liquid in the opposite direction. Spring *4* only serves to overcome the friction force of the plug against the body.

Safety valves, unlike check valves, have a spring with a high pressure force. The principle of operation of valves in this group is based on balancing the pressure force of the liquid by the spring force or back pressure of the liquid. They can be divided into direct acting, differential and servo-operated valves.

When it is necessary to protect the hydraulic drive against excessive pressure rises, safety valves are used that open occasionally in the event of a pressure rise above a set limit value. Direct-acting safety valves can be ball, cone or plug valves.

The simplest safety relief valve is the ball valve. The scope of application of the ball safety relief valve is limited,

It is used at low pressures and flow rates in hydraulic systems with small and infrequent overloads. Reliable sealing between the ball

and seat are difficult to realise, so there is fluid leakage through the ball valve in the closed state.

In addition, the ball vibrates and periodically strikes the seat when the fluid is bypassed.

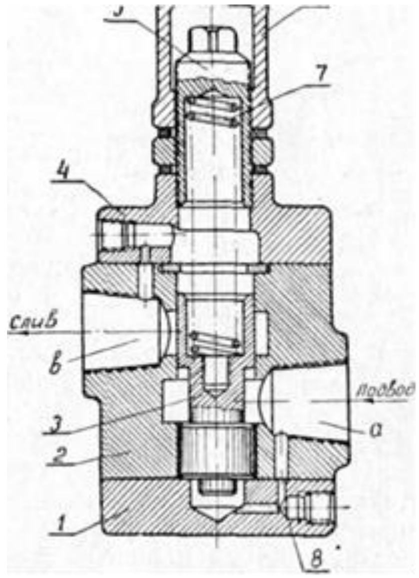


Fig.10.4.

By replacing the ball with a cone head, we obtain a cone relief valve. A prerequisite for ensuring the tightness of such a valve is the strict alignment of the cone and cylindrical part of the valve of the body guide cylinder and the cone seat for the valve. Otherwise, the cone valve quickly loses its tightness. Plunger valves are the most common in hydraulic drives. Plunger valves can be used to protect the hydraulic actuator from overloading and to maintain a certain constant pressure, i.e. depending on the system connection and setting, the same valve can be used as a safety, overflow or back pressure valve.

Fig. 10.4. shows a section of the plug valve type G-54. The latter consists of body *2,* lower cover 1, upper cover *4,* plug Z_y adjusting screw *5* and spring 7.

Spring *7* pushes plunger *3* to its lowest position, separating chamber *a*, which is connected to the pump, and chamber c, which is connected to the drain line. At the same time, pressure is transmitted through the calibrated orifice *8* to the lower end of plunger *3.* When the pressure in the system increases sufficiently to overcome the force of spring 7, plunger *3* moves upwards. Chambers *a* and *c* are connected and the liquid is diverted to the drain.

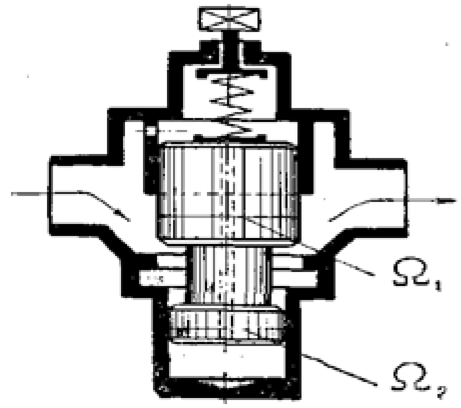


Fig.10.5.

A calibrated orifice *8 is* used to stabilise the valve, i.e. to damp plug oscillations. The valve is adjusted to the set pressure by screw 5, which is used to change the spring force 7.

At higher fluid pressures, the spring stiffness must be increased. Differential *valves are* used to avoid the use of high spring stiffness and thereby reduce the frequency of free oscillation of *the* plug. In differential valves, part of the fluid pressure force acting on the plug is hydraulically balanced (Fig.10.5) due to the difference in the areas of the working $\Omega(_1)$ and balancing $\Omega(_2)$parts of the plug.

Despite the use of damping devices, the valve plug oscillates, causing periodic pressure changes in the system. All direct acting valves suffer from this disadvantage. To stabilise the pressure, universal servo-operated safety valves are used, which can also be used as overflow or pressure relief valves with a certain system connection and appropriate spring setting.

Servo-operated valves can be used to protect the hydraulic actuator against overloads and maintain a defined constant pressure regardless of the fluid flow rate.

By design, the servo-actuated valve is a combination valve, which combines a main plug valve *2* with damper and a ball servo valve *4*.

Let us consider the principle of operation of the valve with servo action on the example of the most common valve type G-52 (Fig.10.5). The valve consists of the following parts: body *1,* plug *2,* springs *3,* 5, ball servo valve *4* and cover *6.* The working fluid from the pump is fed into cavity *a* and discharged from the valve into the tank through cavity *c.* Plunger *2* is loaded by a weak spring *3* and is held in the down position. A damper *8* (small diameter calibrated bore) is screwed into the centre hole of plug *2*, by means of which chamber *b is* permanently connected to chamber *a. In addition*, chamber a is permanently connected to chamber *b.* In addition, chamber a communicates with chamber *2.* Through the centre hole *9* the liquid is supplied from the chamber *e to the* chamber *b* and under the ball *4.* The ball *4* is pressed to the seat by spring 5, the spring compression force can be adjusted by means of screw 7.

As long as the fluid pressure force acting on the ball *4 does* not exceed the force to which the spring 5 is adjusted, the ball is pressed against the seat and the pressure in chamber *b is* equal to the system pressure. In this case, the plunger is in the lower position under the action of spring *3,* because the pressure forces on the plunger *2* from the cavity *b* side are balanced by the pressure forces from the side

At this position of the plunger*, the aiv* cavities are disconnected, so the fluid passage from the system to the tank is closed. tank is closed. As soon as the force of liquid pressure overcomes the force of spring *5,* ball *4* moves away from its seat, and liquid in a small amount from chamber *b* through the ball valve enters chamber *c* and from there to drain.

From chamber *e*, the liquid flows through damper *8* into chamber b. The calibrated orifice of the damper *8* creates resistance, *causing* pressure loss during the flow of liquid, so the pressure in chamber *b* will be lower than the pressure in chambers *a* and *e* by the amount of head loss in the damper. As a result of the resulting pressure difference, the equilibrium is disturbed and the plunger rises upwards under the action of the high pressure in chambers *e* and *d.* As the plunger *2* rises, the cavities *aiv* communicate and the pressurised liquid flows from cavity *a* to cavity c and then to the tank. The plunger rises until equilibrium is reached, i.e. until the total force of the fluid pressure in chambers *e* and *d* is so reduced that it becomes equal to the total force of spring *3* and the force of the fluid pressure in chamber *b.*
After the plunger has reached equilibrium, the fluid pressure in cavity *a is* kept constant, and a small amount of fluid flows continuously through the damper and open ball servo valve from cavity *a* to cavity *c.*

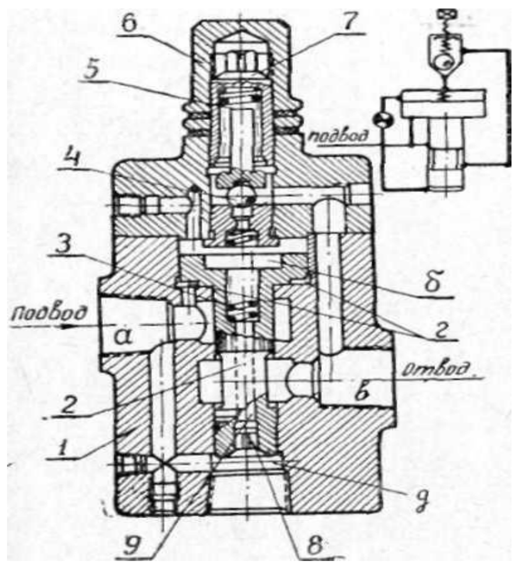


Fig.10.6.

If for some reason the pressure in cavity *a* starts to rise again, the equilibrium of forces is disturbed because the pressure force on the plunger from chambers *e and d* increases. The cross-section of the gap between the body and the plunger edge grows, which increases the fluid flow in cavity c and leads to a decrease in the pressure in cavity *a. The* pressure decreases until the pressure in cavity a is reduced. The pressure decreases until equilibrium is again established. In case of pressure decrease in chamber *a,* spring *5* closes the ball servo valve and stops draining liquid from cavity b, after which the pressure in chambers b, d, *e is* equalised, spring *3* lowers plunger *2* and the valve closes. *The valve is* regulated by changing the tension of spring *5* by turning screw 7. The valve type G-52 is characterised by high sensitivity, it works stably without oscillations and noise, as the damper brakes the plug movement by absorbing the energy of the compressed spring.
Having considered the operation of safety valves, it can be concluded that they are two-way, directly acting valves: closed at normal pressure and open when the inlet pressure to the valve rises. Pressure reducing valves are used to reduce the pressure. In contrast to a safety valve, the controlling influence of a pressure reducing valve is the outlet pressure , i.e. it is actuated when the pressure at the valve outlet changes. The plug of a pressure reducing valve is held open by a spring until actuation.

If the pressure at the outlet of the pressure reducing valve exceeds the set value, the force of the liquid pressure on the valve will compress the spring and the plug will move towards closing, making it difficult for the liquid to pass through the valve. The plug will continue to move in the closing direction until the set reduced pressure is reached at the valve outlet. Plug pressure reducing valves are used in hydraulic actuators.

Fig. 8.6 shows a servo-operated pressure reducing valve type G-57, which is designed to reduce and maintain pressure reduced compared to the pressure developed by the pump. The valve consists of body *1,* plug 2, springs *3* and *5,* ball servo valve *4* and cover *6.* The working fluid is fed into cavity *a* and discharged through chamber *c.* The plug is loaded by a weak spring *3,* which holds it in the lower position.

A damper *8 is* screwed into the centre hole of the plunger, through which chamber *c is* permanently connected to chamber b. Chamber *d* communicates with chamber *c* via damper *10.* Ball *4* is pressed against the seat by spring *5.*

The compression force of the spring *5* can be adjusted by means of the screw 7. As long as the fluid pressure acting on the ball *4 does* not exceed the force to which the spring *5 is adjusted, the* ball *4* is pressed against the seat. At the same time the plunger *2* is in the lower position under the action of spring *3.*

In the lower position of the plunger*, the aiv* cavities are connected, so that the liquid from the system flows freely through the pressure reducing valve and the pressure in cavity *c* is equal to the pressure developed by the pump.

When the pressure force at the valve outlet overcomes the force of spring *5,* ball valve *4* opens and oil from chamber *e* starts to flow through damper *8* into chamber *b,* from where it *flows* through the ball valve to the drain. In the orifice of the damper *8* during the flow of liquid there is a loss of pressure, so the pressure in chamber *b* will be lower than in chambers *e* and d, by the amount of pressure loss in the damper *8. This* causes the plunger *2* to rise.
As the plunger rises upwards, it impedes the passage of fluid from cavity *a* to chamber c, resulting in an increase in pressure in *cavity a* compared to the pressure in cavity *c.* The pressure in the guide chambers balances the pressure in chamber b *and the force of spring 3.* When the pressure in the *guide* chambers balances the pressure force in chamber *b* and the force of spring *3, the* plunger will be in equilibrium.
If for some reason the pressure in chamber *c* starts to drop, the equilibrium of forces acting on plunger *2 is* disturbed, because the pressure forces on the plunger from chambers *d* and *e* communicating with chamber *c* are reduced. Spring *3* pushes plug *2* down, increasing the cross-section of the gap between the body and the edge of the plug, which increases the inflow of liquid into the chamber, and hence the pressure in it until equilibrium is again established. When the G-57 valve is in operation, a small amount of liquid flows continuously from its drain port, and a constant pressure is maintained in the line downstream of the valve, which is less than that of the pump pressure line.

10.3 Application device and characterisation throttle devices.

Throttling devices in hydraulic drives are used to limit or control the flow of fluid and are hydraulic resistances. Throttling devices can be unregulated hydraulic resistances, or hydraulic dampers, or regulated (throttles).

Hydraulic dampers are used in various elements and devices of hydraulic equipment for braking (throttling) of liquid at oscillations and other non-stationary processes, i.e. for stabilisation of operation of equipment and mechanisms of hydraulic drives.

Throttles (Fig.18.7) are designed to regulate fluid flow rate by changing the value of the slit cross-section. Throttle regulation of hydraulic drives is one of the most common methods of speed regulation of low-power hydraulic motors.

When the fluid passes through the throttle slit, part of the fluid energy is lost to overcome the resistance of the slit, which leads to a reduction in the speed of the hydraulic motor. With throttle control, the energy received from the pump must always exceed the energy required to move the pump

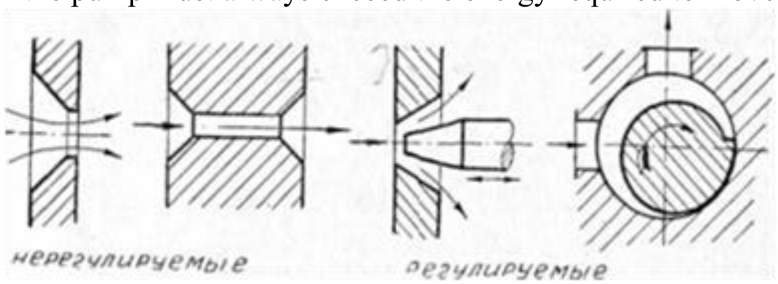


Fig.10.7

of the hydraulic motor at the set speed. According to the shape of adjustable slots, throttles are divided into slotted and grooved throttles (see Fig. 10.7).
Fig. 10.8 shows throttle of G-77 type, which consists of body 1, front cover *2,* back cover *3,* throttle *4,* limb *5*, seal b, scale 7, nut *8.* Liquid is supplied to the throttle through opening 9 and, having passed through slit *10, is* discharged through opening *11.*

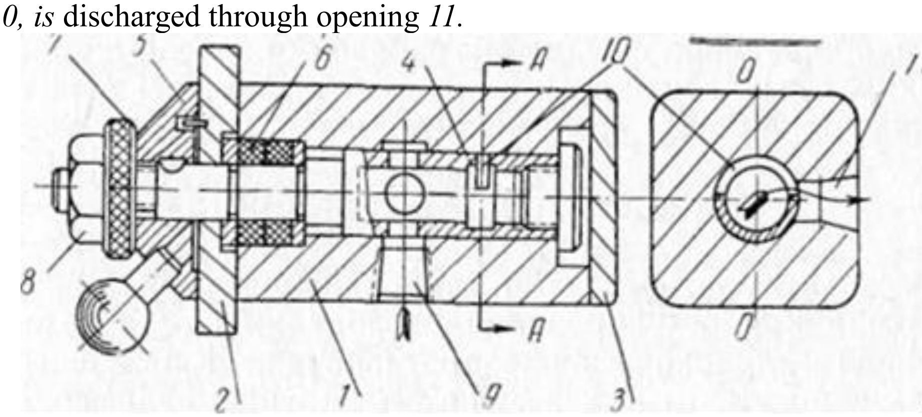


Fig.10.8

Depending on the angular position of the throttle slit *4* relative to the O-O axis, the passage cross-section of the slit changes, which accordingly increases or decreases the flow rate of liquid passing through the throttle. During adjustment, the nut *8* is pressed off for free rotation of the throttle *4.* Adjusted and set required cross-section of the slit is fixed by nut *8,* which is pressed against limb *5.*

As throttling devices are also used special control throttling spools (Fig. 10.8), allowing to change smoothly the liquid velocity in pipelines due to changing the area of the operating window.
The liquid is double throttled in the spool *valve 2*. It flows from pump *1* under pressure into the spool valve. When the spool is displaced from the neutral position, two passage windows are formed in it: at the inlet to the hydraulic motor *3* and at its outlet. Throttling of liquid through these windows is accompanied by energy loss, which causes pressure loss.
In an ideal spool valve, the width
The width of the plunger belt should be equal to the width of the throttle window. However, in practice to increase sensitivity often make spools with liquid flow (Fig.10.9). The width of the plunger belt of these spools is less than the width of the window by some microns. Control spool valves are used and with an overlap of a few microns.

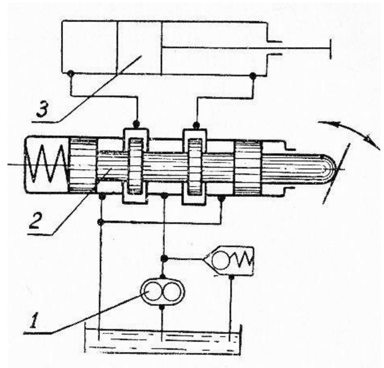


Fig.10.9

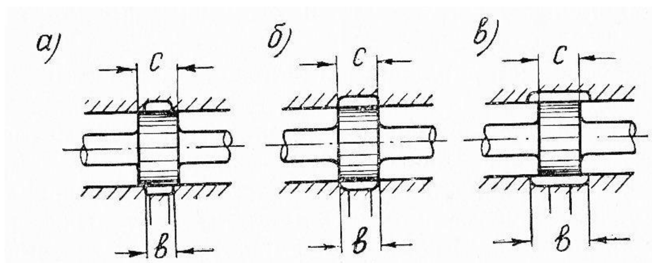


Fig.8.10

Slides with overlapping (Fig.10.10, b) in the neutral position have considerably less leakage, but the insensitive zone of such a slide increases.

11.1 Auxiliary devices (pipes, filters, accumulators)

Auxiliary devices of hydraulic drives include pipelines and their connections, tanks, filters, hydraulic multipliers and accumulators.

Pipelines serve as conduits through which the energy from the pumps is supplied to the hydraulic motors. Depending on the operating conditions, rigid and flexible pipework is used. Most often round steel seamless pipes and sometimes aluminium *alloy and* cast iron pipes are used as pipelines for hydraulic drives. Hydraulic calculation of pipelines is performed according to hydraulic formulas for viscous fluid flow. Pipe joints and their connection to elements and units of hydraulic drives must be strong and tight. When connecting steel pipes, welding and flange connections are used. Connection of pipes of small diameter is made by cap nuts with flared joints of the pipe ends; for high and ultra-high pressures nipple connection is used.

Recently, hinged and spring-loaded steel pipe joints have been used.

If there is mutual movement of hydraulic drive units during operation, they are connected by flexible pipelines (rubber-fabric hoses and metal hoses). Modern hydraulic drives use pipes made of reinforced plastics based on polyester or epoxy resins with glass fibre reinforcement, especially in those cases when metal pipes are inapplicable due to high weight or insufficient chemical resistance.

During operation of the hydraulic drive, the working fluid is continuously contaminated. The fluid is contaminated due to foreign bodies penetrating from outside and due to destruction and wear of rubbing surfaces. Therefore, in the scheme of the hydraulic drive must be provided constantly operating filtering devices. In hydraulic drives most often used filters of mechanical fluid purification.

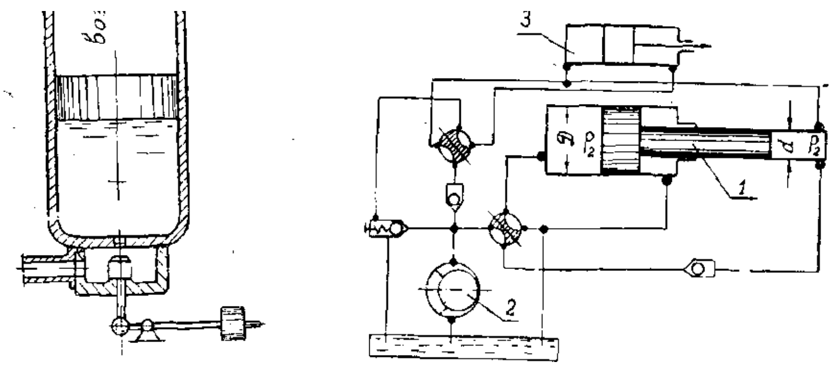
Various filtering materials are used in filters: metal meshes, metal plates, cloth, felt, paper impregnated with oil-resistant resins, plastics, metal ceramics, porous metal powders. In addition to mechanical filters, centrifugal cleaners, magnetic and electrostatic filters are used. Filters in hydraulic drive are installed depending on their purpose and working conditions either in series or in parallel, and mainly in pressure lines, in easily accessible places.

Special tanks are provided for replenishing and supplying the hydraulic drive system with working fluid. In order to improve oil settling and heat dissipation, the minimum capacity of the tank should be 2-3 times higher than the minute pump delivery. Partitions are made in the tanks to separate the drain line from the suction line. The height of the partition should be 2/3 *h*, where ***h is*** the depth of the tank filled with liquid. Location and design of the tank should provide easy filling and easy control of the liquid level. Often tanks are closed and filled with air or other pressurised gas. The use of closed tanks improves the operation of suction lines, pumps are better filled with liquid during operation.

The flow rate of the fluid consumed by the hydraulic motor can vary over time at a constant pump flow rate.

At certain intervals it may be greater or less than the average pump flow. To ensure normal operation of the hydraulic drive, it is necessary to provide either a pump with a delivery equal to the maximum flow rate of the hydraulic motors or a hydraulic accumulator.

Obviously, the use of a pump with a high flow rate would only be justified for a short period of time, during the *rest of the time the* excess liquid would have to be discharged into the tank.



a) б)

Fig.11.1.

If the pump delivery is greater than the flow rate through the hydraulic motors, the pressurised fluid is accumulated in the volume of the accumulator; if it is less than the maximum flow rate of the hydraulic motors, the accumulator returns the pressurised fluid to the system, ensuring the maximum fluid flow rate together with the pump. Application of hydraulic accumulator allows increasing the pump power utilisation factor, providing more uniform operation of hydraulic motors with elimination of pressure pulsations created by pump operation.

Hydraulic accumulators can be pneumatic, load and spring accumulators. Pneumatic hydraulic accumulators are the most widespread (Fig. 11.1). Pressure multipliers are used to obtain high pressures, which cannot be provided by a pump. The simplest scheme of multiplier 1 is shown in Fig. 11.1 a. A large diameter cylinder is supplied with fluid pressurised by a pump *2.* As the multiplier plunger moves in the small cylinder, the pressure increases in proportion to the ratio of cylinder areas.

Thus, the pressure $p_{(2)}$ *of* the fluid after the multiplier will be equal to

$$p_2 = p_1 \frac{D^2}{d^2},$$

where $p_{(1)}$ is the pressure generated by the pump;

D-- diameter of the large cylinder;

d-diameter of the smaller cylinder.

The multiplier is mounted between the pump and the high-pressure power cylinder *3*.

11.2 The development of GPPs in the mining engineering industry.

The hydraulic drive of mining machines is ahead of the hydraulic drives used in other branches of technology in terms of operating parameters. For example, in hydraulic drives of mechanised mounts the fluid energy is transmitted at a distance of up to200 м at a pressure in hydraulic stands of up to 60 MPa. The installed power of single motors exceeds 100 kW. The mining industry was the first to use non-flammable water emulsions for hydraulic drives.

Hydraulic drive is widely used in mining machines in underground and open-pit mining operations. Application of hydraulic drive allows to create progressive designs of machines to reduce their overall dimensions, increase durability, expand the possibilities of control automation. Hydraulic drive provides the possibility of creating multi-drive systems, the realisation of high power in the limited dimensions of the mining machine, high starting torques with reliable protection against overload, precise control of movements and speeds of mechanisms, autonomous power supply, low reliability. Efficient and simple designs of progressive motion mechanisms are created on the basis of hydraulic drive.

The use of hydraulic drive in mining machines largely determines the safety of labour of workers, which is one of the main criteria determining the possibility of introducing new mining equipment.

The scope of application of hydraulic drive in mining machines is growing continuously. Development of complex mechanisation of mining operations requires more and more perfect hydraulic mechanisms, machines and hydraulic systems for complex conditions of mining enterprises. Therefore, knowledge of hydraulic drive, its technical and production capabilities, rules and norms of operation is a necessary condition for the creation of high-performance mining machines, complexes and aggregates that ensure the effective operation of mining enterprises.

The purpose of this book is to present a systematic presentation of the current understanding of the hydraulic drive of mining machines on the basis of the current course curriculum. To reflect the general level of development of hydraulic drives of mining machines, the existing and possible variety of their schematic and constructive solutions.

11.3 Safety precautions in work with hydraulic pneumatic actuators.

Ensuring safety during the operation of mining equipment. Labour protection measures are carried out by the administration of enterprises under the control of trade union bodies. Supervision over labour safety is also carried out by specialised state bodies: Gosgortechnadzor, Energy Supervision, Sanitary Supervision.

Working conditions in underground workings have specific features: equipment is placed in cramped conditions, despite the fact that machines and mechanisms are compact, overall dimensions of workplaces and passages are small, people move along narrow passages and vertical workings; equipment, workplaces and workers are constantly moving as the face is moved. Increased humidity and dustiness of the mine atmosphere and its aggressiveness contribute to intensive wear of equipment, which increases the number of machine and mechanism failures, worsens sanitary and hygienic labour conditions. In underground workings, the risk of rock collapse and flooding, sudden rock and gas emissions, rock impacts, and underground fires cannot be ruled out.

All this requires in-depth knowledge and excellence in mining operations and the strictest adherence to the rules and regulations of work safety and safe operation of equipment.

Basic safety measures during the operation of underground equipment. The main safety measures during the operation of underground equipment, which are provided by the operating personnel, are:

- Work should only be carried out with faultless equipment and all safety equipment and interlocks provided for;
- carrying out inspection, repair, lubrication and other maintenance work when the equipment is stopped (and, if necessary, blocked);
- Do not carry out inspection and repair work on electrical equipment and installations, energised power lines;
-Avoiding uncoordinated actions of people and the conscientious use of warning signals and prohibition signs;
- qualitative fulfilment of all production, maintenance and repair works stipulated by the operating instructions for this type of equipment.

Various forms and methods of training and professional development and all types of instruction play a major role in ensuring labour safety and compliance with the Safety Rules - primary, conducted when sending a worker to a new place of work or after a long break in work; permanent, conducted every day when issuing a work order; periodic, conducted under a special programme and within the time limits established by the Safety Rules and Regulations or Rules and Technical Operation for certain professions, as well as extraordinary and additional.

One of the basic measures to ensure safe operation of equipment is to conduct a thorough technical inspection, test assembly and testing of the machine or equipment on the surface or in the machine shop before it is lowered into the mine or installed for its intended use.

All machines and equipment must have instructions and technological passports for operation, maintenance and repair, regulating the composition of mandatory preventive maintenance, safety requirements for their performance of machine control operations, lubrication map, etc. The study and observance of all rules and regulations of operation and maintenance is one of the basic, main conditions for safe operation of equipment.

On the other hand, safety measures during the operation of underground equipment are regulated and established by regulations on the safety of downhole machines and complexes, the fulfilment of which is mandatory when creating new and modernised downhole machines and complexes.

Supervisory Questions:

1. What are throttling devices for?
2. How do throttling devices work?
3. What equipment is referred to as an auxiliary device?
4. Why use filters?
5. What is a hydraulic accumulator and what is it for?

Supporting words. throttling devices, hydraulic accumulators, filters, pipes, accumulators.

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Laboratory work
on the subject

" HYDRAULIC DRIVE OF MINING MASHINES "

LABORATORY WORK №1
STUDY OF FLUID FLOW REGIMES
ON A REYNOLDS DEVICE

PURPOSE OF THE WORK:

1. Visual observation of laminar and turbulent modes of motion.
2. Determination of Reynolds number values in laminar and turbulent regimes of motion.

BRIEF THEORETICAL BACKGROUND

Two flow regimes are possible when a liquid moves in a pipeline (channel) - laminar and turbulent.

Laminar regime is characterised by parallel-jet motion, in which separate layers of liquid move without mixing with each other. Such motion occurs at low velocities and small live sections of liquid flow, at movement through capillaries, at movement of viscous liquids (oil, fuel oil, oils), at movement in soil pores, etc.

The turbulent regime is characterised by disordered, chaotic motion, when fluid particles move along complex, constantly changing, trajectories. Due to the presence of velocity components transverse to the direction of motion in turbulent flow, intensive mixing takes place in the liquid. In engineering practice, the turbulent regime is most often observed in the movement of water and other liquids of low viscosity (paraffin, petrol, alcohol, etc.), in heating, ventilation, gas supply, heat supply, water supply systems.

The existence of two modes of fluid motion was clearly shown by the English physicist O. Reynolds. Reynolds' experiments, confirmed later by other scientists, showed that the criterion for determining the mode of fluid motion in a circular tube is the expression:

$$Re = \frac{V \cdot d}{\nu},$$

where Re is a dimensionless criterion called the Reynolds number:

V - average speed of liquid movement, cm/s;
d - pipe diameter, cm;
ν - kinematic viscosity coefficient, cm^2/s.

The value of the Reynolds number at which the transition from the laminar regime to the turbulent regime occurs is called the critical Reynolds number - $\mathbf{Re_{(kr}}$

At **Re**< $\mathbf{Re_{kr}}$ the mode of motion is laminar, at **Re**> $\mathbf{Re_{(kr)}}$) turbulent.

In a certain range of **Re** numbers there is an unstable region where both regimes are possible depending on the character of velocity changes. The value of the critical number Re_{kr} depends on a number of circumstances: conditions of the pipe entrance, roughness of the pipe walls, absence or presence of initial disturbances, etc. and can take different values in each individual case.

For circular pipes $\mathbf{Re_{kr}=2320}$ is usually taken. The velocity at which the turbulent regime transitions to the laminar regime of fluid motion is called the critical velocity.

At **Re≤2320**, the regime is laminar.

When **Re>2320**, the regime is turbulent.

PILOT PLANT DESCRIPTION

The pilot plant (Fig. II) consists of a pressure tank /I/, into which water from the water supply network is supplied via a pipeline /2/. To maintain a constant water level in the tank there is a weir /4/. Inside the tank there is a grate /5/, which serves for calming the water coming into it, and a thermometer /8/ for measuring the water temperature.

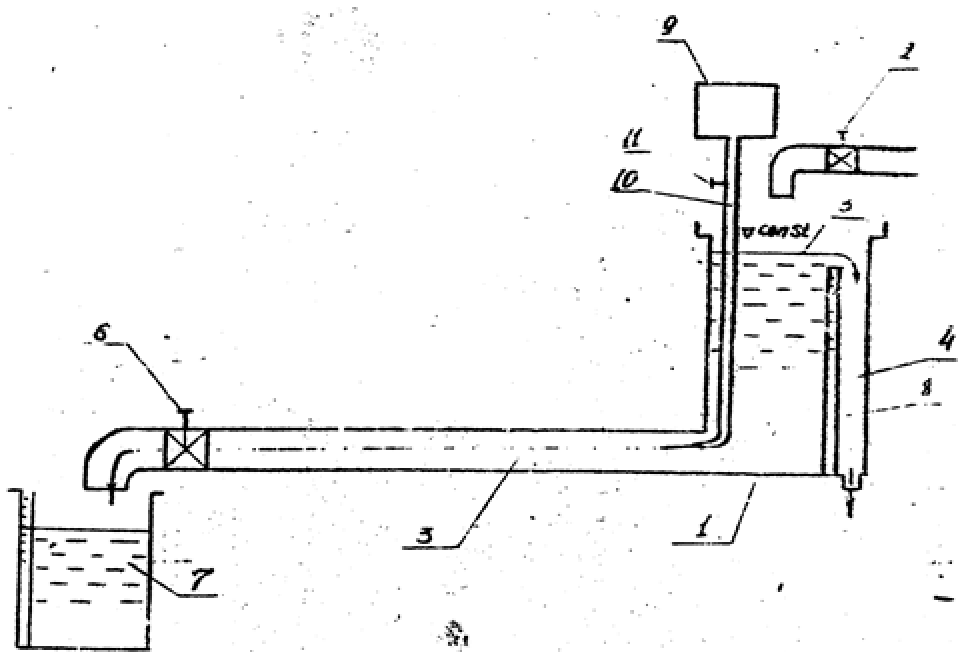


Fig.1.1 Schematic diagram of the pilot plant

A glass tube /3/ is connected to the tank /I/, at the end of which a tap /6/ is installed to regulate the water flow rate. The water flow rate is determined by means of a measuring tank /7/. The unit has a small tank /9/ for the dye with a tube /10/ and a tap /11/.

In the experiment, the mode of motion is observed in the main glass tube /3/ when a dye is introduced into the main flow. Change of the mode is achieved by controlling the liquid flow rate through the tube with the help of a tap /6/.

EXPERIMENTAL PROCEDURE

1. With closed taps /6/ and /11/, fill pressure tank /1/ with water.
2. By slightly opening the valve /6/ set in the pipe /3/ the liquid flow rate at which there is a slow flow.
3. By opening the tap /11/, the dye is introduced into the main flow. Observe the character of liquid movement in the glass tube. The jet motion of the dye will indicate the presence of the laminar regime. Gradually increase the opening of the tap /6/ and observe the change in the mode of motion with increasing speed. At first, the coloured jet acquires a wavy character and the laminar regime becomes unstable. At further increase of speed the coloured jet disappears, the whole liquid is uniformly coloured - the laminar mode of motion has changed into turbulent one.
4. The flow rate of water in the pipe is determined at steady flow. For each flow regime, the volume of water ***W*** entering the measuring tank for time ***t is*** determined, and the water temperature is recorded at the same time.

EXPERIMENTAL DATA PROCESSING

1. The kinematic viscosity coefficient υ is determined from the table.

Table 1.1.

Water temperature in deg. C(0)	0	5	10	15	20	25
Kinematic viscosity coefficient υ, cm^2/sec	0,0173	0,015	0,0131	0,0114	0,0102	0,0090

1. Water consumption:

$$Q = \frac{W}{t} \quad (cm^3/s),$$

where W is the volume of water in the measuring tank, cm^3.
t - time of tank filling, s.

The average velocity of the liquid in the pipe is $V = \frac{Q}{\omega}$ (cm/s);

where ω - living section area, pipe, cm^2 $\omega = \frac{\pi d^2}{4}$

d- diameter of the glass tube, cm.

2. Using the known d, v, υ, the Reynolds number value is calculated for each experiment

$$\mathrm{Re} = \frac{v \cdot d}{\nu}$$

The results of measurements and calculations are recorded in a table.

Table 1.2.

№№	Measurement data		Calculation data			Movement mode	Constant values
	W	t	Q	V	Re		d=2.0 cm
	cm^3	c	cm^3/s	cm/s			
1	2	3	4	5	6	7	8

Control questions
1. Name the basic elements of fluid motion?
2. What is a trajectory, current line, elementary jet, flow?
3. Name the hydraulic elements of flow?
4. What is fluid flow rate?
5. What is the average flow velocity?
6. What modes are observed in the movement of fluid in pipelines?
7. How is the laminar flow regime characterised?
8. How is the turbulent flow regime characterised?
9. Draw the graphs of velocity distribution along the flow cross-section at different cross-sectional regimes?
10. What should be the appearance of the tinted jet at laminar regime?
11. What is the appearance of an underpainted jet in turbulent regime?

Laboratory work
CENTRIFUGAL PUMP CHARACTERISTIC

Purpose of the work:

Pumps are used in many sectors of the economy. All pumps are divided into two main groups: dynamic and positive displacement pumps.

Dynamic pumps are pumps in which the energy of the liquid is communicated by the action of hydrodynamic forces on an unenclosed volume of liquid at constant pump inlet and outlet communication.

Volumetric pumps are pumps in which the energy of the liquid is communicated periodically by changing the enclosed volume in alternating communication with the pump inlet and outlet.

Vane pumps are pumps in which the energy of the liquid is communicated by flowing around the vanes of the working class. Vane pumps combine two groups of pumps: centrifugal pumps and axial flow pumps.

Centrifugal pumps are vane pumps with the liquid moving through the impeller from the centre to the periphery and axial vane pumps with the liquid moving through the impeller in the direction of its axis.

Friction and inertia pumps are a group of dynamic pumps in which the movement of liquid is carried out by friction and inertia forces. This group includes vortex, screw, labyrinth, worm and jet pumps.

The positive displacement pump group includes piston, plunger, diaphragm, rotary, gear, and screw pumps.

Consider the operating diagram of a centrifugal pump (Fig. 2)

The advantages of centrifugal pumps are compactness, relatively light weight, small size with high performance, possibility of direct connection to the electric motor, smooth and continuous flow of liquid, easy start-up and adjustment.

Disadvantages: unstable head - as the capacity increases (at n= const) the head generated by the pump decreases, low efficiency for small capacities.

Inside the fixed casing 1 there is an impeller 2 fixed on a shaft 3. The pump casing is connected to the suction and pressure pipes by sockets 4 and 5.

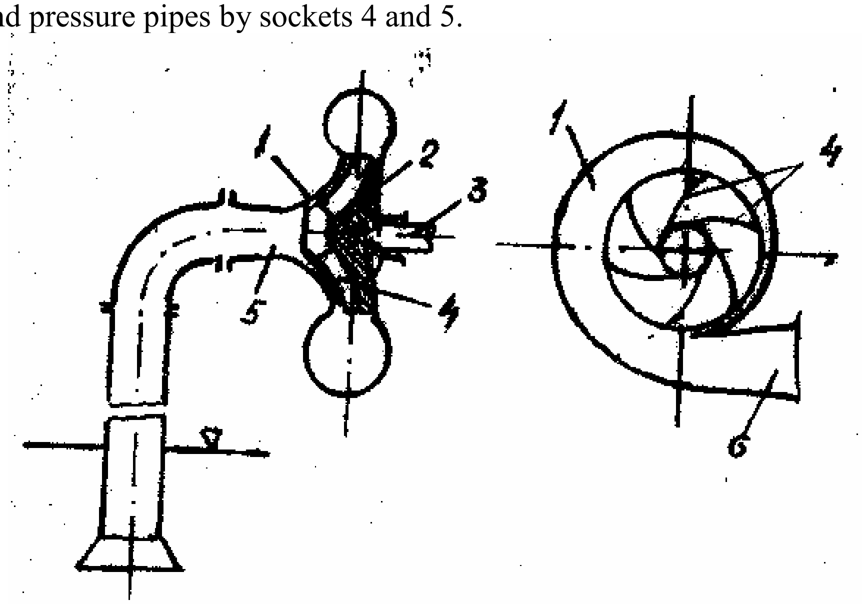


Fig. 2

If the suction pipe and the pump casing are filled with liquid and the impeller is rotated, the liquid filling the channels between the vanes will be pushed from the centre of the impeller to the periphery by the central force. After leaving the wheel, the liquid enters the spiral chamber and then into the discharge pipe. At the same time, a vacuum is created before the liquid enters the impeller, under the action of which the liquid from the receiving tank flows through the suction pipework into the pump. Centrifugal pumps can be not only single-stage, but also multistage, but the principle of their operation in all cases remains the same - the movement of liquid is carried out under the action of the centrifugal force developed by the rotating impeller.

In the case of a centrifugal pump, other operating parameters such as head and power also change as the capacity changes,

Characteristics of a centrifugal pump.

Before start-up, centrifugal pumps are primed with the pumped liquid. If the speed "K" of the centrifugal pump, its delivery Q, head "H" and power consumption "N" are varied within small limits, the following relationships apply:

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2};\; ; \qquad \frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2 \frac{N_1}{N_2} = \left(\frac{n_1}{n_2}\right)^3$$

The relationships Q-H; Q-N; Q-η are called pump characteristics and are established experimentally.

Installation Description.

The centrifugal pump (3) is mounted on the same shaft as the AC motor. The number of revolutions is measured. Water is sucked by the pump from the flow tank (5). The suction pipe is fitted with a suction valve (6), which prevents water from escaping when the pump is primed through the suction tube (7). On the discharge pipe (8) there is a pressure gauge (9) and a valve for regulating the flow rate (supply) of water (10). Water from the discharge pipe flows into one of the measuring tanks (11). Each of the tanks has a water measuring scale (4), which is calibrated in units of volume (litres), and also in the tank there is a drain pipe to avoid overfilling. In the bottom of the tanks there are spigots with valves (1), through which water from the measuring tank is drained (1) into the consumption tank, from where it is sucked again by the pump.

Methodology for carrying out the work.

When testing the pump of the plant, the values necessary for the construction of the pump characteristics are determined: Q-H, Q-N, Q-η . The tests are carried out at constant speed but at different, increasing flow rates (delivery rates) Q of the pump. The flow rate Q is changed by gradually opening the valve(12). The first observation is carried out with the valve fully closed, the subsequent ones with gradual opening by a quarter of a turn. At the same time it is necessary to measure: pump delivery, vacuum in the suction pipe, pressure on the discharge pipe, voltage of electric current on the motor.

Measurement of the pumping unit performance is carried out as follows.

Supply: closes the drain valve in one of the measuring tanks and starts the stopwatch. The quantity of water measured on the water measuring scale and the time of measurement are recorded in the table. Head expressed in.m. column of the supplied liquid (water) is determined as follows:

$$H = P_m + P_b + \frac{v_H^2 - v_b^2}{2g} + h$$

P_m and P_b - reading of pressure gauge and vacuum gauge in m.column of the supplied liquid;

v_m *and* $v_{(}v)$ - water velocity at the connection points of manometer and vacuum gauge tubes.

h - Distance between vacuum gauge and pressure gauge connection levels.

The suction and discharge pipes are the same diameter, so v_m *and* v_{in} are the same and then $H = P_{(m)} + P_{(in)} + h$.

Experimental data processing and report writing.

Pump capacity (flow rate) (m^3/s); $Q = \dfrac{Q^1}{1000\tau}$ where

Q^1 - water volume determined from the water meter glass, dm (or litres);
τ - duration of measurement, s.
Power consumption of the pumps by the installation

$$N = \frac{VJ}{1000}$$

V - current voltages, V.
J - current strength, A.
The efficiency of the pump is determined from the formula

$N = \frac{QHg\rho}{1000\eta}$ from where $\eta = \frac{QHg\rho}{1000N}$

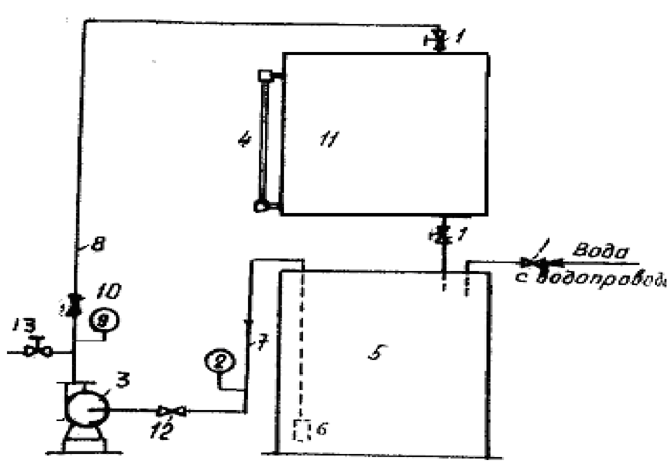
where Q - capacity (pump flow rate), m^3/sec
ρ - density of the liquid, kg/m^3;
g - acceleration of free fall, m/s^2;
H is the total head generated by the pump, in metres of liquid column.
The work is completed by plotting Q-N, Q-η, Q-H.

Table

№	Number of revolutions rpm	Measurement duration, s	Packs of water, dm^3	Pressure P_M		Resolution P_V		Full head H	Power kW	EFFICIENCY %
				$\frac{кгс}{см^2}$ or mmHg.	m. v. st	$\frac{кгс}{см^2}$	m.water st			
	n	τ	Q	R_M	H_M	P_C	H_B	H	N	η

Take 3 measurements for each feed (Q). Enter the average of the 3 measurements in the table.

Installation diagram



1.Valves 2. Vacuum gauge. 3. Pump 4.Water measuring glass 5. Consumption tank. 6. Check valve 7. Suction pipework 8. Pressure pipework 9. Pressure gauge 10., 12, Regulating valves 11. Measuring tanks. 13 Drain valve.

Supervisory Questions:

1. Design of a centrifugal pump.
2. Characteristics of a centrifugal pump.
3. Suction height and cavitation phenomenon.
4. The head developed by the pump.
5. Dependence of main parameters of centrifugal pump operation on impeller speed.

Types and designs of centrifugal and piston pumps

Practical exercises on the discipline:

" HYDRAULIC ACTUATORS "

Solving problems on the topic: "Basic properties of liquids".

The main physical properties of liquids are: *density, volume (specific gravity), compressibility* and *viscosity.*

The density ρ *, kg/m³*, is the mass of liquid contained in a unit volume.

$$\rho = \frac{m}{V} \tag{1-1}$$

where: m - mass of liquid, *kg*;
ρ - density of the liquid, kg/m^3;
V - volume of liquid, m^3.

The volumetric (specific) weight of a liquid γ *,* N/m^3 is the weight of a unit volume of this liquid

$$\gamma = \frac{G}{V} \tag{1-2}$$

where: *G is the* weight of the liquid (the force of attraction of the liquid to the Earth), *N;*

Density and volume (specific gravity) are related by a relationship:

$$\gamma = \rho\, g \tag{1-3}$$

where: *g is the* acceleration of free fall, m^2/s;

The density and volume weight of a liquid vary with pressure and temperature.

Compressibility is the property of a liquid to change its volume with a change in pressure. The compressibility of liquids is characterised by the coefficient of isothermal volume compression m^2/H :

$$X = \frac{\Delta V}{V_o \Delta P} \tag{1-4}$$

where: V_o - initial volume, m^3;
ΔV - change of liquid volume, m^3.
ΔP - pressure change, *Pa*.

The inverse of the isometric volume compression coefficient is called the modulus of elasticity of the liquid. For water under normal conditions it is possible to take $E = 2.0 \cdot 10^9$ Pa.

When a liquid is heated, the increase in volume is estimated by the temperature coefficient of volume expansion X_t:

$$X_t = \frac{\Delta V}{Y_o \Delta t} \tag{1-5}$$

Viscosity is the property of a fluid to resist the relative motion (shear) of fluid particles.
The viscosity of a liquid is characterised by the coefficient of kinematic ν , *m^2/s,* and dynamic viscosity μ , *N. s/m⁽²⁾,* which are related by the following relationship:

$$\nu = \frac{\mu}{\rho} \tag{1-6}$$

The temperature dependence of the kinematic viscosity coefficient of water is determined by the following formula:

$$\nu = \frac{0,017}{(1+0,00337\ t^2 + 0,000221\ t^2)\cdot 10^4} \tag{1-7}$$

The viscosity of a liquid in Engler degree conditions (viscosity conditional - VU) is determined by the formula:

$$BY = \frac{\tau_{ж}}{\tau_B}$$

where: τ_g - time of flow of 200 *cm⁽³⁾ of* the test through the calibrated opening of the viscometer at a given temperature, *s*;

τ_B - flow time of 200 cm^3 of distilled water at 20^oC (water number of the viscometer), *s*.

Determination of the kinematic viscosity coefficient from the conditional viscosity given in Engler degrees is made by the formula:

$$\nu = \left(0,0731\ {}^oBY - \frac{0,0631}{{}^oBY}\right)\cdot 10^{-4} \tag{1-8}$$

(1-8)

For conditional viscosity (CV) greater than 16^oC, the formula should be used:

$$\nu = 7,4\cdot 10^{-6}\,BY$$

EXAMPLES.

Example 1.

A vessel is filled with water occupying a volume $V_1 = 2$ м³ . How much will this volume decrease and what will be equal to when the pressure increases by the value Δ p= 200 kG/cm²?

Solution.

The change in liquid volume with increasing pressure is determined by the equation

$$\beta_v = -\frac{1}{V}\cdot\frac{\Delta V}{\Delta p}$$

where V is the initial volume of the liquid;

Δ V- change of this volume with pressure increase by value Δp .

$$\Delta V = -\beta_v V \Delta p\ .$$

Since the average value of the compressibility coefficient for water is

$$\beta_v = 47.5 \cdot 10^{-10}\, м^2 / кГ = 0{,}0000475 см^2 / кГ\ ,$$

in the considered case we obtain

$$\Delta V = 0.0000475 \cdot 2 \cdot 200 = 0.019 м^2$$

Therefore, the required volume will be equal to

$$V_2 = V_1 + \Delta V = 2 - 0.019 = 1.981 см^2$$

Response: $V_2 = 1.981 см^2$

Source: Rabinovich E.Z. Rabinovich. "Hydraulics", p. 25

Example 2.

Determine the density of water at temperature t = 100 ºC in the physical system of units.

Solution.

Find the specific gravity of water at temperature t = 100 ºC in the technical system of units

$$\gamma = 0{,}958 \frac{Г}{см^3} = 958 \frac{кГ}{см^3}$$

Then by equation (1-5) we find the density of water in the same system

$$\rho = \frac{\gamma}{g} = \frac{958}{9.81} = 97{,}6 \frac{кГ \cdot сек^2}{м^4}$$

and, bearing in mind that there is a relation between the quantities of density units in both systems

$$\rho_ф = \frac{\rho_T}{102},$$

determine the density in the physical system of units

$$\rho = \frac{97{,}6}{102} = 0{,}958 \frac{Г}{см^3}$$

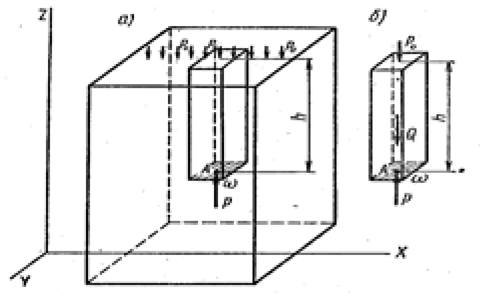
Response: $\rho = 0{,}958 \frac{Г}{см^3}$

Source: Rabinovich E.Z. Rabinovich. "Hydraulics", p. 26

Tasks for self-solving.

Task 1.

Determine all types of hydrostatic pressure at point A of a vessel of water at depth: $h = 4 м$; $p_0 = p_a = 10^5 H / м^2$; $\gamma = 10^4 H / м^3$

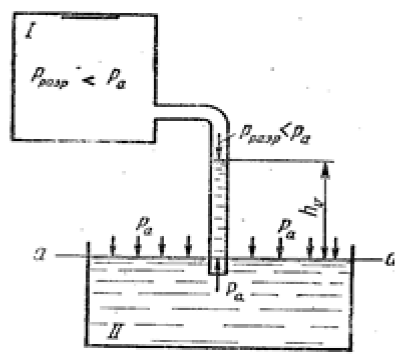


The answer is: $p = 14 \cdot 10^4 Н/м^2$;

$p_{изб} = 4Н/см^2$

Task 2.

Let vessel I be pressurised $p_{разр} = 4Н/см^2$. In the open vessel II the liquid is water with specific gravity γ = 0.01 $[Н/см^3]$. Determine the vacuum value.



The answer is: $p_v = 6Н/см^2; h_v = 600 м.вод.ст$

Solving problems on the topic: "Basic equations of hydrostatics".

The forces acting on fluid particles are subdivided into *surface* and *mass* forces.

Surface forces, for example, include pressure forces directed normal to the area on which they act and internal friction forces that are tangential.

Mass forces include gravity and inertia forces. Mass forces are characterised by the accelerations they impart to a unit of mass.

The force acting per unit area along the normal to the surface that confines an infinitesimal object within a resting fluid is called *hydrostatic pressure*.

The hydrostatic pressure at any point of a liquid is the sum of the pressure on its free surface and the pressure of the liquid column, the height of which is equal to the distance from this point to the free surface:

$$p = p_o + \rho g h \qquad (1\text{-}1)$$

where: p - hydrostatic pressure, *Pa*;

p_o - pressure on the free surface of the liquid, *Pa*;

ρ - density of the liquid, *kg/m^3*;
g is the acceleration of free fall, *m/s^2;*
h - height of the liquid column above this point, *m.*

Expression (2-1) is called the *basic equation of hydrostatics.* It follows from this equation that the external pressure p_o on the free surface of the liquid is transmitted to any point of the liquid uniformly (Pascal's law).

Hydrostatic pressure is called *total* or *absolute* p_{abs}, and the valueρ gh is called *relative* (or, if atmospheric pressure acts on the free surface of the liquid - *overpressure*) *pressure*. Thus, if the pressure on the free surface of the liquid is equal to atmospheric pressure, then:

$$p_{abs} = p_{at} + p_{eq} \tag{1-2}$$

When the absolute pressure is less than atmospheric pressure, the gauge indicates vacuum (vacuum):

$$p_{abs1} = p_{at} - p_{razr} \tag{1-3}$$

Negative overpressure is called *vacuum pressure*.

When calculating the strength of various hydromechanical structures, it is necessary to determine the pressure of the liquid on the wall and bottom of these structures.

The excess fluid pressure per unit area of a flat wall is equal to:

$$p_{(isb)\ gh} = \rho \tag{1-4}$$

The total force acting on a flat wall is equal to the product of the wetted area of the wall and the hydrostatic pressure at its centre of gravity:

$$P = (P_o + \rho\ g\ h^{)\ (.)}\ F_{st} \tag{1-5}$$

In an open vessel at $P_o = 0$ the total pressure force:

$$P = \rho g\ h_{\ ts.t(.)\ (}{}^{)\ (.)}\ F_{st} \tag{1-6}$$

where: $h_{ts.t.}$ - depth of immersion of the centre of gravity of the area, *m*;
F_{st} - wetted area of the wall, *m^2*.

The point of application of force *P is* called the *centre of pressure.* The centre of pressure usually lies below the centre of gravity of the wall. For a rectangular wall, for example, the centre of gravity is half the height from the base and the centre of pressure is one third of the height.

A special case of curved wall is the walls of cylindrical tanks, boilers, pipes, etc.

Total pressure force acting on a cylindrical surface:

$$P = \sqrt{P_x^{\ 2} + P_y^{\ 2}} \tag{1-7}$$

where: P_x - horizontal component equal to the fluid pressure force on the vertical projection of the cylindrical surface, *N:*

$$P_x = \rho \cdot g \cdot h_{y.m.} \cdot F_{верт.} \tag{1-8}$$

where: P_y *is the* vertical component of the pressure force *P* equal to the force of gravity acting in the volume of the pressure body V_t:

$$P_y = \rho g\ V_t \tag{1-9}$$

The *volume of a pressure body* V_t is the volume of fluid bounded from above by a free surface and from the sides by a vertical surface drawn through the perimeter bounding the wall.

The direction of the total pressure force *P is* determined by the angle formed by the vector *P* with the horizontal plane:

$$tg\ \beta = \frac{P_y}{P_x} \tag{1-10}$$

For a cylindrical tank with a vertical axis, the vertical component of P_y is zero, so the total pressure force on the side surface is P_x:

$$P = P_x \qquad (1\text{-}11)$$

Any body immersed in a liquid is subject to an expulsive force equal to the gravity of the liquid displaced by that body (Archimedes' law):

$$P_c = \rho g V \qquad (1\text{-}12)$$

where: P_c - expulsive (Archimedean) force, *N;*

ρ - density of the liquid, kg/m^3;

g is the acceleration of free fall, m/s^2;

V_t - volume of the immersed part of the body, m^3.

The product$\rho \cdot V$ *is* called *displacement.*

Depending on the relationship between the force of gravity of a body and the force of gravity displaced by its fluid, three states of a body:

1. The force of gravity of the body is greater than the force of gravity of the displaced fluid:

$$G > \rho g V_t$$

Such a body will sink.

2. The force of gravity of a body is equal to the force of gravity of the displaced liquid:

$$G = \rho g V_t$$

In this case, the body will float.

3. The force of gravity of the body is less than the force of gravity of the displaced liquid:

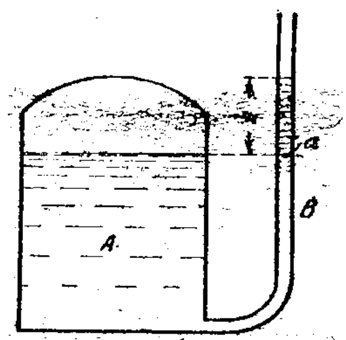
$$G < \rho g V_t$$

At this ratio, the body will float.

EXAMPLES.

Example 1.

A piezometric tube B is inserted into the side wall of the vessel A filled with water



Determine the absolute pressure p on the free surface of the liquid in the vessel, if under the action of this pressure the water in the tube rose to a height h = 1.5m.

Solution.

The pressure at the free surface of the liquid in the vessel is equal to the pressure in the section a of the piezometer and is determined by the basic equation of hydrostatics

$$p = p + \gamma h \ .$$

Hence, taking into account that the pressure on the free surface

$p_0 = p_{atm} = 1$ kG/cm^2,and specific gravity of water

$$\gamma = 1000\frac{\kappa\Gamma}{c\text{м}^3} = 0{,}001\frac{\kappa\Gamma}{\text{м}^3},$$

we get it:

$$p = 1 + 0.001 - 150 = 1.15 \text{ atm (kG/cm}^2)$$

Answer: p = 1.15 atm (kG/cm^2)

Tasks for self-solving.

Task 1.
Determine the hydrostatic pressure force and the centre of water pressure on a circular shield of diameter d= 2 м , closing an opening in a vertical wall that maintains the water level in a channel of depth H. = 4 м
The answer is: $P = 94.2Tc$;
Task 2.
Given H = 6m, V = 2m, the liquid is water ($\gamma = 10 кH/м^3$). Determine P and the position of the centre of pressure.
Answer: P = 360 $кH$; $h_D = 4м$

Solving problems on the topic: "Hydraulic resistance".
A uniformly moving fluid flow in a pipe loses some energy due to friction on the pipe surface as well as internal friction in the fluid itself. These losses are called frictional head losses along the length of the flow.

The main task of fluid dynamics is to determine friction losses.

Energy losses are usually determined using the Bernoulli equation. The left-hand side of the equation usually has one unknown, and in many cases there are initial pressures.

The head losses along the length are determined using the Darcy-Weisbach equation:

$$h_c = \lambda \frac{L}{d} \cdot \frac{V^2}{2g},$$

where L is the length of the pipeline section under consideration;
d is the diameter of this section of the pipeline;
V- average velocity of the fluid;
λ - coefficient of hydraulic friction

The coefficient λ is a function of the Reynolds number (Re) and the relative roughness of the pipeline walls (Δ /d):

$$\lambda = f(\mathrm{Re}, \Delta/d)$$

At laminar motion of a liquid the coefficient λ is defined as follows:

$$\lambda = \frac{64}{\mathrm{Re}}.$$

Due to the fact that turbulent flow has its own complex structure. The coefficient λ is found experimentally.

According to the widespread Prandtl hypothesis in turbulent flow at the walls of the pipeline a thin layer of liquid (thickness δ) with laminar regime is formed. The thickness of the laminar layer - "δ " is very small (hundredths and sometimes thousandths of mm).

Depending on the relative roughness (Δ/δ) and the laminar layer (i.e. the value of the Re number), three flow zones are distinguished in the turbulent regime.

The zone of hydraulically smooth pipeline wall - here the thickness of the laminar layer is several times the absolute size of the roughness protrusions (Δ), i.e. $\delta >> \Delta$. In this condition, the rough non-smooth wall will be completely covered by the laminar layer, i.e. the fluid will flow smoothly and the roughness of the walls will not affect the head loss. In this case, the coefficient of hydraulic friction will depend only on the Reynolds number (Re):

$$\lambda = f(\mathrm{Re})$$

As a result of his experiments, Blasius found it possible to determine the coefficient of hydraulic friction for hydraulically smooth pipes:

$$\lambda = \frac{0,3164}{\sqrt[4]{\text{Re}}}$$

This equation is valid for Re < 105, otherwise: $2300 < \text{Re} < 10\frac{d}{\Delta}$.

In the mixed friction zone, as the velocity increases, i.e. as the Re number increases, the thickness of the laminar layer δ will be smaller than the absolute size of the roughness protrusions, i.e. $\delta < \Delta$. In this condition, the coefficient α will depend on both the Re number and the absolute roughness : Δ: $\lambda = f(\text{Re}, \frac{d}{\Delta})$.

In this case, λ is determined using the Altschul equation:

$$\lambda = 0,11\ (\frac{d}{\Delta} + \frac{68}{\text{Re}})$$

Here:

$$10\frac{d}{\Delta} < \text{Re} < 500\frac{d}{\Delta}.$$

The zone of complete roughness or quadratic zone. In this case, as a result of the rapidly increasing Re number, the thickness of the laminar layer δ becomes several times smaller than the absolute size of the roughness protrusions, i.e. $\delta << \Delta$.. In this state, the coefficient λ depends only on the absolute roughness: $\lambda = f\ (\frac{d}{\Delta})$. In this zone, the coefficient λ is determined using the Nikuradze equation:

$$\lambda = \frac{L}{\left(21g\frac{d}{\Delta} + 1.14\right)^2}.$$

For this third zone: $\text{Re} > 500\frac{d}{\Delta}$.

Example 1.

Determine the absolute viscosity of water in the technical system of units at temperature t = 100 C.

Solution.

Preliminary by formula:

$$v = \frac{0.0178}{1 + 0.0337 \cdot t + 0.000221 \cdot t^2},$$

where v - kinematic viscosity in stokes;

t is the temperature in ^{0}C,

find the kinematic viscosity of water at temperature 100 ^{0}C in the physical system:

$$v=\frac{0.0178}{1+0.0337\cdot100+0.000221\cdot100}=0.0028 \text{ stokes cm2/s}$$

As the density of water at t = 100 ^{0}C in the same system of units

ρ = 0.958 g/cm^3, then the absolute viscosity in the physical system according to Eq.

$$v=\frac{\mu}{\rho}=\frac{\mu\cdot g}{\gamma}$$

will be:

$$\mu=v\cdot\rho=0.028\cdot0.958=0.268 \text{ poise } \frac{г}{см\cdot сек}$$

For the transition to the technical system of units, we take into account that there is a relationship between the quantities of absolute viscosity units in different systems

$$\mu_{ф}=\frac{1}{98,1}\mu_{T},$$

Therefore, the absolute viscosity in the technical system of units will be equal to

μ = 98.1 - 0.268 =26,30 кГ - sec/m^2.

Answer: μ = 26,30 кГ - sec/m^2.

Example 2.

Oil of specific gravity γ = 900 kG/m^2 has viscosity in Engler degrees ^{0}E = 5. Determine the kinematic and absolute viscosities in the physical system of units.

Solution.

The kinematic viscosity in the physical system of units is found as follows directly from the formula:

$$v=0.0731°E-\frac{0.0631}{°E},$$

where v- kinematic viscosity coefficient in stokes,

$°E$ - Engler's degrees.

$$V=0.0731\cdot \text{E} - \frac{0,0631}{E}=0.0731\cdot 5-\frac{0,0631}{5}=0.353 \text{ stokes}$$

Next, we determine the specific gravity and density in the physical system. Since

$$\gamma_{ф}=\frac{\gamma_{T}}{1,02},$$

then

γ = 900 / 1.02 = 882 g/cm^2 - sec^2,

And hence,

$$\rho=\frac{\gamma}{g}=\frac{822}{981}=0.900 г/см^3$$

After that, according to the equation

$$v=\frac{\mu}{\rho}=\frac{\mu\cdot g}{\gamma}$$

we find the absolute viscosity in the physical system of units:

$$\mu = \nu \cdot \rho = 0{,}353 \cdot 0{,}900 = 0{,}318 пуаза (г / см \cdot сек)$$

Answer: V = 0.353 stokes, $\mu = 0{,}318$ poise $\frac{г}{см \cdot сек}$

Ex. lit. litt.: Rabinovich E.Z. "Hydraulics" , c.223

Tasks for self-solving.

Task 1**.**

Determine the mode of water flow in a pipe of diameter d=100mm

flow velocity v = 1.5 m/sec; kinematic viscosity coefficient $\upsilon = 0.01 см^2 / сек.$

Answer: the mode of motion is turbulent

Task 2.

Determine the mode of oil movement along the flume of rectangular section with the base150 мм and layer height 100mm, flow velocity v=0,2m/sec; kinematic viscosity coefficient . $\nu = 0.5 см / сек$

Answer: The mode of motion is turbulent

Problem solving on the topics Hydraulic drives

Example 1.

The pump at speed n_0 = 1450 rpm develops head H_0 = 100m at $Q_{(0)}$ = 50l/sec and η_0 = 0.70. Suppose that the pump is to operate at the same capacity but with an increased head up to 130m. The required speed and power requirement must be determined.

Solution.

According to the task Q/Q_0 = 1. Find the head ratio

N/N_0 = 130/100 = 1.30.

It follows from this relation that

n/n_0 = 1.10, or n = 1450 - 1.10 = 1595 rpm.

The efficiency of the pump in the new mode will be equal to

$\eta = 0{,}67$

Power requirement

$$N = \frac{\gamma QH}{\eta 102} = \frac{50 \cdot 100}{0{,}67 \cdot 102} = 73{,}7 кВт$$

The answer is: . $N = 73{,}7 кВт$

Example 2.

The pump is built for optimum speed n_0 = 2520 rpm, optimum head H_0 = 26.0m and optimum capacity

Q_0 = 16l/sec at efficiency $\eta_{макс} = 0{,}81$. Pump capacity in optimum mode

$$N_0 = \frac{\gamma Q_0 H_0}{\eta 102} = \frac{16 \cdot 26}{0{,}81 \cdot 102} = 5{,}04 кВт.$$

It is required to determine the operating condition of the pump if the pump capacity is to be increased to 22 l/s without changing the speed.

Solution.

According to the curve $H = f(Q)$ in the figure at n = 2520 rpm we find that the capacity Q = 22 l/sec corresponds to the head H= 21 м and efficiency. $\eta \approx 0{,}8.$ Power requirement

$$N = \frac{\gamma QH}{\eta 102} = \frac{22 \cdot 21}{0{,}8 \cdot 102} = 5{,}66\kappa Bm$$

The answer is: . $N = 5{,}66\kappa Bm$

Example 3.

The centrifugal pump model was tested in scale $\lambda = 5$ at speed n = 590 rpm. Thus productivity of model pump Q_M = 8,9m/sec at head H_M = 0,68m. Considering the efficiency of the model and nature are the same, we determine:

1. pump capacity and head at the same speed

n_n = 590 rpm;

2. pump capacity and head at speed

n_n = 730 rpm.

Solution.

Using dependencies

$$\frac{Q_H}{Q_M} = \frac{\omega_H \cdot c_{2H} \cdot \sin\alpha_2}{\omega_M \cdot c_{2M} \cdot \sin\alpha_2} = \frac{l_H^2}{l_M^2}\lambda\frac{n_H}{n_M} = \lambda^3\frac{n_H}{n_M}$$

и

$$\frac{H_H}{H_M} = \frac{\varepsilon_{ZH}\eta_{\Gamma H}H_{TH}}{\varepsilon_{ZM}\eta_{\Gamma M}H_{TM}} = \lambda^2\left(\frac{n_N}{n_M}\right)^2 \cdot \frac{\eta_{\Gamma H}}{\eta_{\Gamma M}}$$

we get

$$Q_H = Q_M\lambda^3\frac{n_H}{n_M} = 8{,}9 \cdot 5^3\frac{590}{590} = 1112\,\text{л/сек},$$

$$H_H = H_M\lambda^2\left(\frac{n_N}{n_M}\right)^2 = 0{,}68 \cdot 5^2\left(\frac{590}{590}\right)^2 = 17\,\text{м}$$

The required data are obtained in the same way:

$$Q_H = Q_M\lambda^3\frac{n_H}{n_M} = 8{,}9 \cdot 5^3\frac{730}{590} = 1372\,\text{л/сек},$$

$$H_H = H_M\lambda^2\left(\frac{n_N}{n_M}\right)^2 = 0{,}68 \cdot 5^2\left(\frac{730}{590}\right)^2 = 25{,}9\,\text{м}$$

In our example, for ease of calculation, we have assumed the efficiency of the pump to be constant and independent of the model scale and speed. In fact, the efficiency varies depending on the machine size and speed, which is taken into account in practice.

Response: $H_H = 17\,\text{м}$, . $H_H = 25{,}9\,\text{м}$

Example 4.

Determine the type of impeller by the coefficient of rapidity, if the number of pump revolutions n = 1450 rpm, capacity

Q = 0.05m^3/sec and head H = 25.0m.

Solution.

Using the formula

$$n_s = \frac{3{,}65n\sqrt{Q}}{H^{3/4}}$$
we get
$$n_s = \frac{3{,}65n\sqrt{Q}}{H^{3/4}} = \frac{3{,}65 \cdot 1450\sqrt{0{,}05}}{25^{3/4}} = 106об/мин.$$
In the case under consideration, a wheel of normal high speed (n_s= 80 - 150) should be adopted.
Answer: wheel of normal speed (n_s=80-150).
Example 5.
Centrifugal pump type 20NDn with capacity Q = 555l/sec at head H = 13,5m has speed n = 730 rpm. Determine the cavitation coefficient and the maximum possible suction height for this pump. It is known that the temperature of pumped water is 20^{0}C, losses in the suction line $h_{\omega вс} = 0{,}50м$, coefficient C in the formula
$$\sigma = \left(\frac{n_s}{C}\right)^{4/3}$$
is 800.
Solution.
Let's first set the speed factor by the formula
$$n_s = \frac{3{,}65n\sqrt{Q}}{H^{3/4}}$$
$$\sigma = \left(\frac{n_s}{C}\right)^{4/3}$$
Then by the formula
determine the cavitation coefficient:
$$\sigma = \left(\frac{282}{800}\right)^{4/3} = 0{,}247,$$
and
$$h_{BC} = \frac{p_{am} - p_n}{\gamma} - h_{\omega вс} - \varphi\Delta H$$
maximum suction height, assuming for water at 20^{0}C the pressure p_n=0.236m, coefficient $\varphi = 1{,}2$:
$$h_{BC} = \frac{p_{am} - p_n}{\gamma} - h_{\omega вс} - \varphi\sigma H = 10{,}0 - 0{,}236 - 0{,}50 - 1{,}2 \cdot 247 \cdot 13{,}5 = 5{,}26м.$$
The value of vacuum generated in the pump is found by the formula
$$h_{BC} = h_{вак} - h_{\omega вс} - \frac{c_1^2}{2g}$$
at pump inlet velocity c_1 = 1.5m/sec. Then
$$h_{вак} = h_{вс} + h_{\omega вс} + \frac{c_1^2}{2g} = 5{,}26 + 0{,}50 + \frac{1{,}5^2}{19{,}62} = 5{,}85м.$$

Example 6.

The centrifugal pump making $1200об/мин$, showed the following data in the test:

$Q, л/с$	0	10,8	21,2	29,8	40,4	51,1
$H, м$	23,5	25,8	25,4	22,1	17,3	11,9
$N, кВт$	5,16	7,87	10,1	11,3	12,0	18,5

A solution was pumped with respect to density $1{,}12$. Determine the efficiency of the pump for each capacity and plot the pump.

Solution.

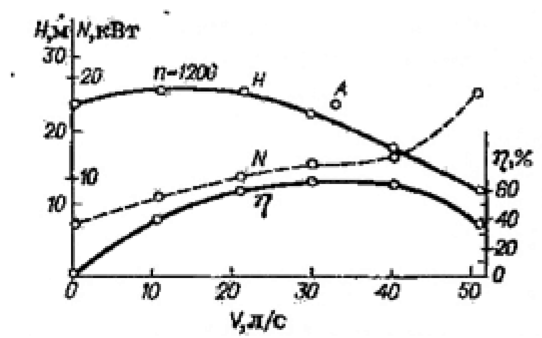
The efficiency of the pump is determined from the equation:

from where $N = Q\rho gH/(1000\eta)$

$$\eta = Q\rho gH/(1000N)$$

The following pump efficiency values are calculated with this formula:

$Q, л/с$	0	10,8	21,2	29,8	40,4	51,1
η	0	0,39	0,587	0,643	0,637	0,36



The pump characteristic is shown in Fig.

Answer:

$Q, л/с$	0	10,8	21,2	29,8	40,4	51,1
η	0	0,39	0,587	0,643	0,637	0,36

Example 7.

It is required to supply $115м^3/ч$ of solution relative to density $1{,}12$ from the tank to the apparatus with height $10{,}8м$, counting from the liquid level in the tank. The pressure in the apparatus is $p_{изб} = 0{,}4кгс/см^2 (40кПа)$, the pressure in the tank is atmospheric. The pipeline has a diameter of $140 \times 4{,}5мм$, its design length (its own length plus the equivalent length of local resistances) $140м$. Can the centrifugal pump of the previous example be used if the friction coefficient in the pipeline λ is assumed to be $0{,}03$?

Solution.

Determine the required head that the pump must deliver.

Fluid velocity:

$$\omega = 115/(3600 \cdot 0{,}785 \cdot 0{,}131^2) = 2{,}37м/с$$

Speed Pressure:

$$h_{ск} = 2{,}37^2/(2 \cdot 9{,}81) = 0{,}286м$$

Head loss due to friction and local resistances:

$$h_{тр+м.с} = \frac{\lambda(L + L_э)}{d} h_{ск} = \frac{0{,}03 \cdot 140}{0{,}131} 0{,}286 = 9{,}16 м$$

The required total head of the pump is calculated according to the formula:

$$H = \frac{p_2 - p_1}{\rho g} + H_Г + h_П$$

$$H = \frac{0{,}4 \cdot 10000 \cdot 9{,}81}{1120 \cdot 9{,}81} + 10{,}8 + 9{,}16 + 0{,}286 = 23{,}8 м$$

Required pump capacity:

$$Q = 115 \cdot 1000 / 3600 = 32 л/с = 0{,}032 м^2/с$$

Referring to the figure, we see that the point A with coordinates $Q = 32 л/с, H = 23{,}8 м$ lies above the pump characteristic curve and therefore this pump will not be able to deliver the required capacity at $n_1 = 1200 об/мин$ (at $H = 23{,}8 м$ the pump can only deliver $26 л/с$). However, if the speed is increased slightly, the pump will be suitable. Using the relationship

$$Q_1 / Q_2 = n_1 / n_2 \text{ и } H_1 / H_2 = (n_1 / n_2)^2$$

$$N_1 / N_2 = (n_1 / n_2)^2$$

the required new speed can be selected . n_2

If, for example, we take $n_2 = 1260 об/мин$ and recalculate the data of the example $\eta = \eta_н \eta_п \eta_д$ using the formulas

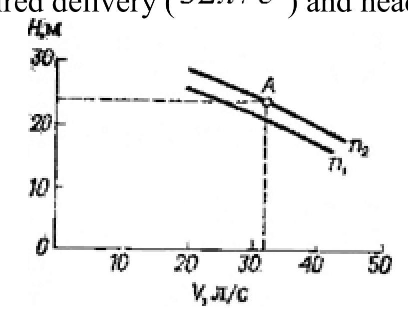
$Q_1 / Q_2 = n_1 / n_2$ и $H_1 / H_2 = (n_1 / n_2)^2$

$N_1 / N_2 = (n_1 / n_2)^2$

by this new speed, we obtain the following results:

	$Q_1, л/с$	21,2	29,8	40,4
$n_1 = 1200 об/мин$	$H_1, м$	25,4	22,1	17,3
	$Q_2, л/с$	22,3	31,3	42,5
$n_2 = 1260 об/мин$	$H_2, м$	28,0	24,4	19,1

By plotting the pump characteristic curve at $n_2 = 1260 об/мин$, we find that at this speed the pump can deliver the required delivery ($32 л/с$) and head ( ). $23{,}8 м$



The power consumed by the pump at the new speed is determined by the following formula

$N = Q\rho gH/(1000\eta)$,

assuming roughly that the efficiency of the pump η has not changed. We take its value from the example $\eta = \eta_н \eta_п \eta_д$, in which it was found that for $Q = 30 \div 40 л/с$ the efficiency of the pump $\eta \approx 0{,}64$.

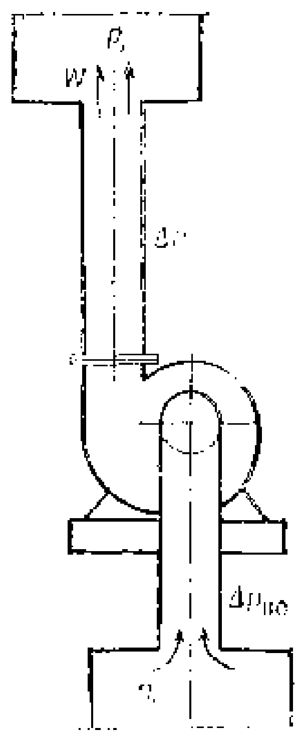
Power consumed by the pump at : $n_2 = 1260 об/мин$

$N = 32 \cdot 1120 \cdot 9{,}81 \cdot 23{,}8/(1000 \cdot 1000 \cdot 0{,}64) = 13{,}1 кВт$

Response: $N = 13{,}1 кВт$

Example 8.

Determine the pressure developed by the fan that supplies nitrogen ($p = 1{,}2 кг/м^3$) from the gas storage tank to the plant. Overpressure in the gas storage $60 мм.вод.ст.$, in the plant $74 мм.вод.ст.$ Losses in the suction line $19 мм.вод.ст.$, in the discharge line $35 мм.вод.ст.$ Nitrogen velocity in the discharge line $11{,}2 м/с$.



Solution.

Pressure difference between the discharge and suction points:

$p_2 - p_1 = (74 - 60)9{,}81 = 137 Па$ or $14 мм.вод.ст.$

Total losses in suction and discharge pipework:

$\Delta p_{ВС} + \Delta p_{Н} = (19 + 35)9{,}81 = 530 Па$ or $54 мм.вод.ст.$

Velocity pressure at the pipeline outlet:

$\omega^2 p/2 = 11{,}2^2 \cdot 1{,}2/2 = 76$ or $7{,}7 мм.вод.ст.$

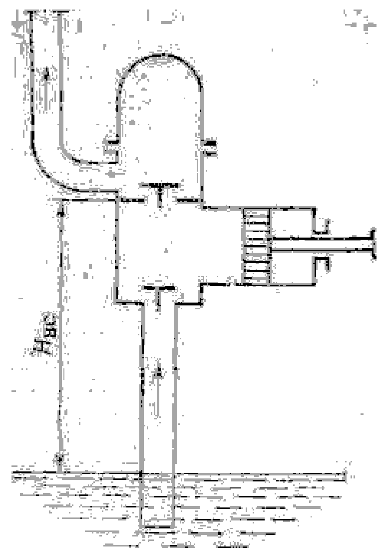
Pressure generated by the fan:

$\Delta p = 137 + 530 + 76 = 743 Па$ or $76 мм.вод.ст.$

Answer: $\Delta p = 743 Па$ or $76 мм.вод.ст.$

Tasks for self-solving.

Problem 1.A simple-acting piston pump with piston diameter $160 мм$ and stroke $200 мм$ is required to supply $430 л/мин$ a liquid of relative density $0,93$ from a collection tank to an apparatus with pressure $p_{изб} = 3,2 кгс/см^2$. The pressure in the collector is atmospheric. Geometric lift height $19,5 м$. Total head loss in the suction line $1,7 м$, in the discharge line $8,6 м$. What speed should be given to the pump and what power of the electric motor should be installed, if we assume the pump delivery ratio $0,85$ and the efficiency of: pump $0,8$, transmission and electric motor by ? $0,95$



Answer: $n = 126 об/мин$; an electric motor with a power of $6,8 кВт$

Literature: Romankov P.G. and... "Examples and tasks of the course "Processes and apparatuses of chemical industry" p.51.

Problem 2.Determine the power consumed by a single-stage reciprocating compressor that compresses $460 м^3/ч$ (count at $0°C$ and $760 мм.рт.ст.$) ammonia from $p_{абс} = 2,5 кгс/см^2$ to $p_{абс} = 12 кгс/см^2$. The initial temperature of ammonia is $10°C$, the compressor efficiency is . $0,7$

Response: $N = 33,4 кВт$

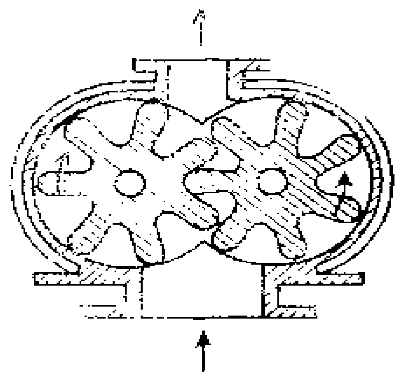
Used literature: Romankov P.G. and... "Examples and tasks of the course "Processes and apparatuses of chemical industry" p.57.

Task 3 It is required to supply compressed air under pressure $p_{изб} = 0,45 МПа$ in quantity $80 кг/ч$. Will a simple-acting single-stage reciprocating compressor having cylinder diameter $180 мм$, piston stroke length $200 мм$ and making $240 об/мин$ be suitable for this circuit ? The harmonic space is 5% of the volume described by the piston. Assume the polytropy of expansion to be . $1,25$

Answer: The compressor will not provide the specified capacity
Literature: Romankov P.G. and... "Examples and tasks of the course "Processes and apparatuses of chemical industry" p.58.
Problem 4: Determine the power consumed by a carbon dioxide reciprocating compressor with a capacity of 5.6 m^3/h (under suction conditions). The compressor compresses CO_2 from 20 to 70 kgf/cm^2 (absolute pressure).Initial temperature - 15°C. The efficiency of the compressor is 0.65. Solve the problem both analytically and with the help of T-S diagram for carbon dioxide.
Answer: 4.6 kW
Literature: Romankov P.G. and... "Examples and tasks of the course "Processes and apparatuses of chemical industry" p.60.

Problem 5.Determine the volumetric efficiency of the compressor of the previous problem, if the harmful space is 6% of the volume described by the piston, and the polytropy of expansion $m =$ 1.2.
Answer: 0.89
Reference: Romankov P.G. Examples and tasks of the course "Processes and apparatuses of chemical industry" p.60
Problem 6.At what discharge pressure will the volumetric efficiency of a single-stage reciprocating compressor compressing ethylene fall to 0.02? The suction pressure is 1 kgf/cm^2. Consider the expansion of the gas from the harmful space to be adiabatic. The volume of the harmful space is 7% of the volume described by the piston.
Response: $20{,}3 \cdot 10^5\ Па$

Task 7 In the suction pipeline before the centrifugal fan there is a vacuum $15{,}8 мм.вод.ст.$, the pressure gauge on the discharge pipeline after the fan shows overpressure $20{,}7 мм.вод.ст.$ Flow meter, shows the air supply $3700 м^3/ч$. The suction and discharge pipes have the same diameter. The speed in $1 мин$ is equal to 960. The fan consumes $0{,}77 кВт$. Determine the pressure developed by the fan and the efficiency of the fan. How will the capacity of the fan change if the fan speed is increased to $1150 об/мин$, and what power will be consumed at the new speed?
Answer: ; ; $\eta = 0{,}48$ $\Delta p = 354 Па$ $Q_2 = 4430 м^3/ч$; $N_2 = 1{,}33 кВт$
Reference: Romankov P.G. Examples and tasks of the course "Processes and apparatuses of chemical industry" p.55
Problem 8 Determine the feed coefficient of a gear pump making $440 об/мин$. The number of teeth on the gear 12, the tooth width $42 мм$, the area of the tooth cross-section, bounded by the outer circle of the neighbouring gear, $960 мм^2$. The pump feeds . $0{,}312 м^3/мин$
Response: $\eta_v = 0{,}735$



Task 9: The pump pumps 30% sulphuric acid. Manometer reading on the discharge pipe 1.8 kgf/cm^2 = 0.18 MPa, vacuum gauge reading (vacuum) on the suction pipe in front of the pump 29mmHg. The pressure gauge is connected at0,5 м above the vacuum gauge. The suction and discharge pipes are the same diameter. What is the head of the pump?
The answer is: .15,6 м
Problem 10. A pump pumps a liquid with density 960 kg/m^3 from a tank with atmospheric pressure into an apparatus with pressure of
p_{eq} = 37kgf/cm^2 = 3.7 MPa. Lift height16 м . Total resistance of suction and discharge line65,6 м . Determine the total head developed by the pump. Determine the efficiency of the pumping unit. A pump delivers 380 l/min of fuel oil of relative density 0.9. Total head30,8 м . The power consumed by the motor is 2.5 kW.
The answer is: .467 м
Problem 11.A centrifugal pump making 1800 rpm has to pump 140 m^3/h of water having temperature 30ºC. The mean atmospheric pressure at the pump location is745 мм Hg. The total head loss in the suction line is4,2 м . Determine the theoretically permissible suction height.
The answer is: no more than .2,2 м
Problem 12.A centrifugal pump pumping 280 l/min of water generates a head of
H = 18m. Is this pump suitable for pumping liquid of relative density 1.06 in quantity of 15 m^3/h through a pipeline of diameter 70 x2,5 мм from a collection tank with atmospheric pressure p_{eq}= 0.3 kgf/cm^2? Geometric lift height8,5 м . Design length of the pipeline (natural length plus equivalent length of local resistances)124 м .Coefficient of friction in the pipeline λ = 0.03.Determine also how much power the motor will need to be installed if the efficiency of the pumping unit is 0.55.
The answer is 1.86 kW.
Task 13. Centrifugal pump for pumping water has the following passport data: Q = 56 m^3/h, H= 42 м , N = 10.9 kW at n = 1140 rpm. Determine: 1) efficiency of the pump; 2) its capacity, developed head and power consumption at n = 1140 rpm, considering that the efficiency remains unchanged.
Response: $\eta = 0{,}59; Q = 71{,}2 м^3/ч;\ H = 68 м$
Problem 14.When testing a centrifugal pump, the following data are obtained:
Q, l/min 0 100 200 300 400 500
H, m 37.2 38.0 37.0 34.5 31.8 28.5
How much liquid will this pump deliver through a pipeline of diameter 76 x4 мм , length355 мм (natural length plus equivalent length of local resistances) at geometric delivery height4,8 м ? Friction coefficient λ = 0.03; $\Delta p_{доп}$ = 0. (Plot the pump and pipework characteristics and find the operating point.)
Answer: Q_1 = 0.4 m^3/min

Task 15. A centrifugal fan, making 960 rpm, supplies 3200 m^3/h of air, consuming 0.8kW. The pressure (overpressure) produced by the fan is44 мм vg. Hg. What will be the supply, pressure and power input of this fan at n = 1250 rpm? Also determine the efficiency of the fan.

The answer is: . $\eta = 0{,}48; Q = 4170 м^3/ч; \Delta p = 734 Па; N = 1{,}77 кВт$

Test questions on the discipline
" HYDRAULIC ACTUATORS "

Hydraulic shock is caused by...?	*Really Fa increase in pressure	Time changes	Speed changes	Changes in the direction of fluid flow
Where does the liquid in a centrifugal pump come from the suction pipe?	*Impeller blades	Pump casing	To the external environment	All answers are correct
Gear pumps refers to...	*To positive displacement pumps	Centrifugal pumps	To vane pumps	To the propeller pumps
In which case will there be an interaction between the jet and a solid obstacle?	*When the jet encounters a solid moving or stationary obstacle	When the volume of the jet is proportional to the pressure	When the jet pressure increases	When the jet pressure decreases
What is the purpose of installing a strainer in the suction pipe of the pump?	* To retain all kinds of impurities *	To create the head	To accelerate pump operation	To prevent the pump from running
State the characteristic feature of most rotary piston machines.	*Absence of suction and pressure valves	Absence of suction valves	No pressure valves	All answers are correct
What is a distinguishing feature of positive displacement	*The reciprocating or rotating movement of a	Progressive movement of the displacer	Rotary motion of the displacer	All answers are correct

hydraulic machines?	displacer in the form of a sliding or rotating piston.			
What are the different types of positive displacement hydraulic machines, depending on their construction?	*Piston, rotor-tooth, rotor-piston, rotor-plastic, rotor-plastic	Piston, plate	Rotary piston	Rotor-plastic
What does the pressure in a positive displacement hydraulic machine depend on?	*From external load	From internal load	From external and internal load	From gravity
Where does the reactive force generated by the fluid flowing out of the nozzle go?	*In the opposite direction to the jet's movement.	Directly relative to the jet	Parallel to the jet movement	Perpendicular to the jet movement
Piston pumps are a...	*To positive displacement pumps	To vane pumps	Centrifugal pumps	To slip-on pumps
In which case the flow is stabilised by local resistance	*If local resistances are spaced at least 20 pipe diameters apart	If the temperature does not exceed 100^0 C	If the speed is less than 5 $м/сек$	If the pressure is constant
What is called the propagation velocity of a shock wave?	*Velocity of elastic strain propagation	Fluid velocity	Average speed	Average speed
. Which scientist first experimentally and theoretically studied the phenomenon of hydraulic shock?	*N.E. Zhukovsky	Archimedes	Euler	Pascal
What are the different types of hydraulic machines?	* Vane and volumetric	Axial and piston	Horizontal and vertical	Small and large
What are ehrlifts?	*Device for lifting water from wells	Crane	latch	valve

	with compressed air			
Which type of hydraulic machine do piston and rotary pumps belong to?	*Volume hydraulic machines	Hydraulic vane machines	Simple hydraulic machines	Complex hydraulic machines
What are ejectors?	*Water jet pumps, in which the liquid is lifted by utilising the kinetic energy of the jet.	Piston pumps	Vane pumps	Centrifugal pumps
Which hydraulic machine is based on the principle of utilising the pressure produced by hydraulic shock?	*Hydraulic ram	Hydraulic pump	Hydraulic motor	Hydraulic accumulator
Which hydraulic machines are used to convert fluid pressure energy into mechanical energy?	*hydraulic motors	presses	accumulators	jacks
What is called the head of a pump?	*Energy collected from each kilogram of pumped liquid	Head strength	Head velocity	Head volume
What is called the pump capacity of a pump?	*The volume of liquid delivered by the pump into the pipeline per unit of time	Fluid velocity	Liquid temperature	Fluid energy
Which type of pumps are centrifugal, diagonal and axial pumps?	*Vane pumps	Piston pumps	Propeller pumps	Water jet pumps
Which hydraulic machines are used to deliver fluid under pressure?	*pumps	Presses	battery	jacks
What is called the geodetic suction height?	*Distance from the water level in	Piezometric height	Geometric height	Liquid column height

	the well to the centre of the pump			
What is called the usable power of a pump?	*Power produced by the pump to lift and move the liquid	Force produced by the pump	Temperature produced by the pump	Force produced by the pump to lift the liquid
What is the purpose of installing a check valve fitted with a strainer at the end of the suction pipe?	*To prevent fluid from leaking out of the pump	To keep the liquid from evaporating	To prevent fluid from flooding the pump casing	To increase the velocity of the fluid
What is the head of a multistage pump?	*Pressure of one wheel multiplied by the number of wheels	The pressure of two wheels	The pressure of three wheels	All answers are correct
How is the pump capacity determined?	*The volume of liquid delivered by the pump to the discharge line	Fluid head in the discharge pipework	Fluid velocity in the discharge line	Head capacity in the discharge pipework
What does the formula represent: $\eta = \eta_{н} \eta_{пер} \eta_{дв}$?	*Total efficiency of the pump unit	transmission efficiency	motor efficiency	Volumetric efficiency
When is a pump considered to be correctly selected?	*If the operating point meets the optimum operating condition of the pump	If the point corresponds to the minimum operating mode of the pump	If the point does not meet the minimum operating mode of the pump	If the point does not correspond to the maximum operating mode of the pump
State the common feature for all reciprocating pumps.	*These pumps are characterised by a certain irregularity in delivery and are practically independent of the discharge pressure	There are blades	The delivery of these pumps is entirely dependent on the discharge pressure	These types of pumps are not used anywhere
Which hydraulic machines are called rotary piston hydraulic machines?	*Multi-cylinder machines in which the fluid is displaced by several pistons in succession	Pumps in which the cylinder blocks are arranged radially to the block axis	Single-cylinder machines in which the fluid is displaced sequentially by a single piston	Multicylinder machines in which the liquid is displaced without pistons
What are radial piston machines?	*Machines in which the cylinder blocks are arranged radially to the block axis	Multi-cylinder pumps, in which the liquid is displaced by	Pumps in which the cylinder blocks are arranged perpendicu-larly	All answers are correct

		several pistons in sequence		
What is the phenomenon of cavitation in vane machines?	*Vapour bubbles escaping from the liquid disappear and pressure builds up, causing the blade to collapse	Pump stop	Pump heating	A sudden loss of energy
What is called a hydraulic turbine?	*A hydraulic motor used to convert the energy of fluid flow into mechanical energy at the turbine shaft.	A hydraulic motor used to convert fluid flow energy into kinematic energy at the turbine shaft.	A hydraulic motor used to convert the energy of a fluid flow into potential energy at the turbine shaft.	All answers are correct
Which centrifugal pumps are used to create high heads?	*Multistage pumps	Piston pumps	Two-stage pumps	Gear pumps
What is called the head of a pump?	*Energy collected from each kilogram of pumped liquid	Head strength	Head velocity	Head volume
Which type of pumps are centrifugal, diagonal and axial pumps?	*Vane pumps	Piston pumps	Propeller pumps	Water jet pumps
What is called the pump capacity of a pump?	*The volume of liquid delivered by the pump into the pipeline per unit of time	Fluid velocity	Liquid temperature	Fluid energy

SELF-STUDY ASSIGNMENT.

1-option

1. Energy losses in the area of local resistances.
2. System resistance coefficient.

Task

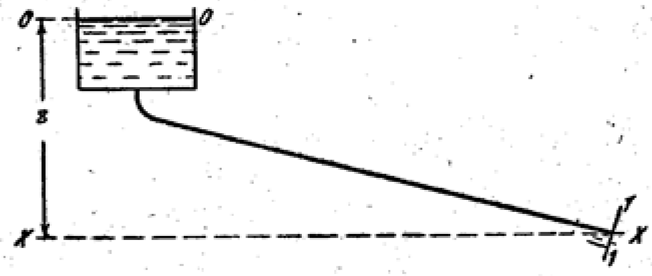
A flow meter "Venturi tube" is installed on a pipeline with a diameter of 160 x5 мм , the inner diameter of the narrow part of which is equal to60 мм . Ethane flows through the pipeline under atmospheric pressure at 25 °°C. The water diffmanometer reading of the Venturi tube is H =32 мм . Determine the mass flow rate of ethane flowing through the pipeline (in kg/h), assuming a flow coefficient of 0.97.
Answer: 280 kg/h

2-option

1. Hydraulic shock in pipes. Methods of reducing hydraulic shock
2. Principle of operation and design of the hydraulic ram.

Task

Determine the flow rate Q in the pipeline length L = 2000m, diameter d = 15cm, supplying water from an open pressure tank in the atmosphere, if the difference in marks between the water surface in the tank and the outlet cross-section of the pipeline z = 10m local losses in the decision to neglect.



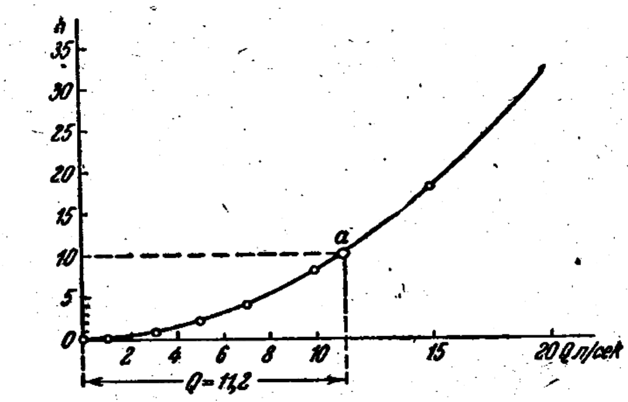
Response: $Q = 11,2\ л/сек.$

3-option

1. Nozzles. Classification and field of application.
2. Fluid flow at variable head.

Task

Determine the flow rate Q in the pipeline length L = 2000m, diameter d = 15cm, supplying water from an open pressure tank in the atmosphere, if the difference in marks between the water surface in the tank and the outlet cross-section of the pipeline z = 10m local losses in the solution neglect. Solve graphically by plotting the hydraulic characteristic of the pipeline.



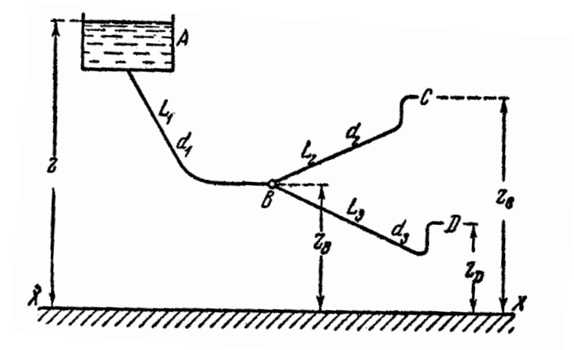
Answer: Q = 11.2 l/sec.

4-option

1. Classification of orifices and basic flow characteristics.
2. The flow of liquid through the opening of a thin wall.

Task

Determine the flow rate of water Q_1 and Q_3 supplied to points C and D through a system of pipes from the pressure tank A. The lengths and diameters of the individual sections of the pipeline are respectively equal to: $L_1 = 1500$m, $L_2 = 1000$m, $L_3 = 800$m, $d_1 = 20$cm, $d_2 = 15$cm, $d_3 = 15$cm; the mark of the water horizon in the reservoir relative to some horizontal plane of comparison (x-x) z = 50m, the mark of the branching point $z_B = 20$v, the mark of the flow points C and D: $z_C = 25$m, z_D 10m.



Answer: $Q_1 = 38.3$ and $Q_3 = 25.8$l/sec.

5-option

1. Active and reactive interactions between a jet and a solid obstacle.
2. Bernoulli's equation for the relative motion of a fluid.

Task

Determine the flow rate of air at a temperature of 20 ºC, passing through the steel wire diameter d = 15cm, length L = 1000m, the pressure created by the compressor at the beginning of the pipeline p_1 = 15at, and the pressure at the end of the pipeline required for the operation of pneumatic machines p_2 = 8at.

Response: $G = 7.01 кГ / сек$

6-option

1. Determination of the force of total fluid pressure on plane figures.
2. Forces of total fluid pressure on a curved surface.

Task

Determine the flow rate of liquid passing through a pipe with diameter d = 40mm, if the average flow velocity v = 1.2 m/sec. The pipe is filled completely.

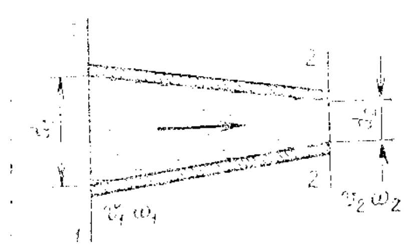
Response: $Q = 0.0015 м^3 / сек = 1.5 м^3 / сек = 5400 м^3 / ч = 5.4 м^3 / ч.$

7-option

1. Live section and fluid flow rate.
2. Methods and instruments for measuring fluid flow and velocity.

Task

We have a conical pipe for which it is known that d_1 = 400mm, average velocity in section 2-2 v_2 = 1.0 m/sec. Determine the average velocity v in section 1-1 and the flow rate of liquid in the pipe. Movement in the pipe is pressure.



Response: $v_1 = 0.25 м / сек.$

1. Hydrvalic, geometric and piezometric gradients.
2. The continuity equation of molasses.

Task

D = 200mm, d = 100mm, h = 0.5m, μ = 0.98, liquid-water. Determine the liquid flow rate.
Answer: Q=25l/sec

7-variant

1. The concept of hydrodynamic similarity and criteria about similarity.
2. Hydrodynamic theory of lubrication.

Task

Determine the absolute viscosity of water in the technical system of units at temperature t = 100 C.
Answer: μ = 26,30 кГ - sec/m^2.

9-option

1. Bernoulli's equation for an elementary jet of a viscous fluid.
2. Uniform fluid motion in pipes and open channels

Task

Oil of specific gravity γ = 900 kG/m^2 has viscosity in Engler degrees ^{0}E = 5. Determine the kinematic and absolute viscosities in the physical system of units.

Answer: V = 0.353 stokes, $\mu = 0{,}318$ poise $\frac{г}{см \cdot сек}$

10-option

1. Bernoulli's equation for an elementary jet of an ideal fluid.
2. Geometric and physical (energy) meaning of Bernoulli's equation

Task

Determine the character of the regime at movement of water on a pipeline with diameter d = 10cm, if the flow rate Q = 4 л .sec and water temperature t = 14 .^{0}C
The answer is: turbulent regime.

11-option

1. The concept of smooth and rough pipes.
2. Determination of head losses in turbulent flow regime.

Task

In the laboratory, the question of the hydraulic resistances that will occur in the designed water pipe d_1=1m is investigated. The research is conducted on water. The diameter of the laboratory pipeline is assumed to be d_2=0,1m. Determine what flow rate Q_2 should be passed through this pipeline to fulfil the conditions of dynamic similarity.
Response: $Q_2 = 0{,}00785$

12-option

1. Energy losses in the area of local resistances.
2. System resistance coefficient.

Task

Determine the mode of water movement in a pipe with diameter d=100mm at flow velocity v = 1.5 m/sec; kinematic viscosity coefficient $\upsilon = 0.01 см^2 / сек.$
Answer: the mode of motion is turbulent

13-option

1. Hydraulic shock in pipes. Methods of reducing hydraulic shock
2. Principle of operation and design of the hydraulic ram.

Task

Determine the mode of oil movement along the flume of rectangular section with the base150 мм and layer height 100mm, flow velocity v=0,2m/sec; kinematic viscosity coefficient .$\nu = 0.5 см/сек$
Answer: The mode of motion is turbulent

14- option

1. Nozzles. Classification and field of application.
2. Fluid flow at variable head.

Task

Determine the hydraulic resistance coefficient λ for the case of water flow through the cast iron pipeline with diameter d = 0.2m with average velocity v = 0.01 cm^2/sec.
Response: $\lambda = 0.0209$.

WORKING CURRICULUM
in the discipline of HYDRAULIC DRIVE OF MINING MASHINES

EVALUATION CRITERIA
on discipline
" HYDRAULIC ACTUATORS "

The following controls are provided to assess the degree of learning:
current control - a method of assessment and evaluation of knowledge and skills on lecture and practical materials of the discipline "Hydraulics and Hydraulic Drives". **Current control is carried out by means of discussions, oral questioning on laboratory classes, checking homework on practical classes based on the characteristics of the discipline** "Hydraulics and Hydraulic Drives"
interim control - a method of assessment and evaluation of students' knowledge and practical skills during the academic term at the end of the chapter of the study programme. **Intermediate control is conducted twice a semester in the form of written work based on the volume of total hours allocated to the academic discipline;**
final control is a method of assessment and evaluation of students' progress in theoretical knowledge and practical skills in the discipline "Hydraulics and Hydraulic Drives". The final control is conducted in the form of written work based on reference words and definitions.

1. List of final control questions

1. Purpose of hydraulic machines and brief information about them.
2. Principle of operation of vane machines.
3. The basic equation of vane machines.
4. Classification of vane pumps.
5. Basic definitions used in pump theory.
6. Classification of centrifugal pumps.
7. Schematic diagram and principle of operation of a centrifugal pump.
8. The basic equation of a centrifugal pump.
9. Impeller blades, diffuser and pump guide apparatus.
10. Characteristics of centrifugal pumps.
11. Operation of the centrifugal pump on the pipeline and determination of the pump operating point.

12. Joint operation of several centrifugal pumps.
13. Similar to vane pumps.
14. Dependence of head, capacity and power of a centrifugal pump on speed.
15. Speed factor.
16. The suction process and the phenomenon of cavitation.
17. Examples of vane pump designs manufactured by our industry.
18. Basic operation of centrifugal pumps and basic concepts of axial flow pumps.
19. Purpose of hydraulic turbines. Methods of creating head.
20. Basic equation of hydraulic turbines. Rapidity coefficient
21. Classification and examples of hydraulic turbine constrictions.
22. Purpose and principle of operation of hydrodynamic gears.
23. Basic equations of hydrodynamic transmissions.
24. Hydraulic coupling workflow.
25. Characteristics and design diagrams of hydraulic couplings.
26. Workflow of the torque converter.
27. Characteristics of torque converters.
28. Torque converter designs.
29. Application of hydrodynamic gears.

2. **Questions for mid-term follow-up**.

1. Principle of operation and purpose of volumetric hydraulic machines.
2. Basic parameters of volumetric hydraulic machines.
3. Working fluids of volumetric hydraulic machines.
4. Device diagrams and working process of piston pumps.
5. Piston pump delivery schedules.
6. Piston pump designs for water and other liquids.
7. Design and principle of operation of rotary piston hydraulic machines.
8. Radial piston pumps and hydraulic motors.
9. Axial piston pumps and hydraulic motors.
10. Rotary vane pumps and hydraulic motors.
11. Gear pumps and hydraulic motors.
12. Screw pumps and hydraulic motors.
13. Switchgear.
14. Throttling devices.
15. Valves .
16. Auxiliary devices.
17. Purpose and classification of hydraulic drives.
18. Hydraulic calculations of hydraulic drive elements.
19. Hydraulic actuators for reciprocating motion.
20. Rotary motion hydraulic drives.
21 Hydraulic tracking actuator

3. INDEPENDENT WORK TOPICS

1. Pressure distribution in a viscous fluid flow under smoothly varying motion.

2. Determination of head losses in turbulent flow regime.
3. Energy losses in the area of local resistances.
4. System resistance coefficient.
5. Hydraulic shock in pipes. Methods of reducing hydraulic shock
6. Principle of operation and design of the hydraulic ram.
7. Nozzles. Classification and field of application.
8. Fluid flow at variable head.
9. Classification of orifices and basic flow characteristics.
10. The flow of liquid through the opening of a thin wall.
11. Determination of the force of total fluid pressure on plane figures.
12. Forces of total fluid pressure on a curved surface.
13. Live section and fluid flow rate.
14. Hydrvalic, geometric and piezometric gradients.
15. The continuity equation of molasses.
16. The concept of hydrodynamic similarity and criteria about similarity.
17. Hydrodynamic theory of lubrication.
18. Bernoulli's equation for an elementary jet of a viscous fluid.
19. Uniform fluid motion in pipes and open channels
20. Bernoulli's equation for an elementary jet of an ideal fluid.
21. Geometric and physical (energy) meaning of Bernoulli's equation
22. Differential equation of continuity for an ideal fluid.
23. Pressure distributions in a molasses viscous fluid under smoothly varying motion.
24. Determination of head losses in turbulent flow regime.
25. Energy losses in the area of local resistances.
26. System resistance coefficient.
27. Hydraulic shock in pipes. Methods of reducing hydraulic shock
28. Principle of operation and design of the hydraulic ram.
29. Nozzles. Classification and field of application.

GLOSSARY OF DISCIPLINE:
" HYDRAULIC ACTUATORS "

A hydraulic accumulator is a device used to , i.e. accumulate, collect energy.
Hydraulic machines - machines used to convert mechanical energy of the motor into energy of the moving fluid (pumps) or hydraulic energy of the fluid flow into mechanical energy (hydraulic motors).
A hydraulic press is a hydraulic machine used to produce large compressive forces.
Hydraulic cross-sectional radius is the ratio of *the* live **section** area to the wetted perimeter $R= F/A$.
Hydraulic shock is a sudden change in pressure in a pressure pipe due to a sudden change in the velocity of the fluid in time.
Hydraulic drive (transmission) - consists of vane hydraulic machines - pump and turbine wheels, extremely close to each other and coaxially arranged.
Hydraulic resistance - resistance appearing in a moving fluid due to the action of external or internal friction forces and manifested in head losses.
Hydraulic motors *are* hydraulic machines for converting fluid pressure energy into mechanical energy.

Hydrodynamics *is the* section of hydraulics that studies the laws of fluid motion.
Hydromechanics is a science studying the laws of equilibrium and motion of liquids, in which only strictly mathematical methods are used to obtain general theoretical solutions to various problems related to the equilibrium and motion of liquids.
Hydraulic drive is a set of devices - hydraulic machines and hydraulic apparatuses, designed to transfer mechanical energy and fluid conversion.
Hydrostatic pressure is the compressive stress that occurs within a resting fluid.
Depth in compressed section is the minimum depth of flow in a stream behind a spillway, at the crest of a spillway with a wide sill, or when flowing from an orifice where the fluid motion can be considered to be smoothly varying.
Stream depth is the distance from the bottom of a stream to its upper boundary (usually the free surface) measured in the vertical longitudinal plane normal to the bottom line.
An ideal liquid is a liquid characterised by absolute mobility, i.e. absence of tangential stresses in the liquid, and absolute invariability in volume at temperature change or under the action of any forces, i.e. absence of compression and stretching deformations.
OverpressurePизб is the difference between absolute **pressure** P and atmospheric pressure . PaPизб= P*Pa
Excess piezometric height - height to which a liquid can rise under the action of pressure at a given point, on the free surface of which the pressure of the external gaseous medium (atmospheric pressure) acts.
Cavitation (from Latin word "cavitas" - cavity) - formation of cavities filled with vapour or air (gas) in a moving liquid.
Cavitation mode of a pump is the mode of pump operation under cavitation conditions causing a change in the main technical parameters.
Short pipelines *are* pipelines in which head losses are mainly composed of local losses.
Feed irregularity coefficient is the ratio of the maximum ordinate of the graph to the average.
The compression ratio is the ratio of the compressed section area ωcj to the hole area ω.
The free surface curve is the line of intersection of the free surface of the flow with a vertical surface drawn through the flow axis.
Laminar mode of fluid motion - the fluid moves in layers without transverse mixing, and there are no velocity and pressure pulsations. The criterion for determining the mode of motion is the dimensionless Reynolds number.
A current line is the direction of motion of the various particles belonging to that line.
Local friction head loss - reduction of total head due to the work of internal friction forces of the fluid at local deformation of the flow.
Local resistance - resistance causing sharp deformation of the flow.
Head is the height of the liquid column above the level under consideration.
Pump head is the energy imparted by the pump to each kilogram of liquid pumped.
Pumps *are* hydraulic machines used to convert the mechanical energy of a motor into the energy of the fluid being moved.
Nominal pump mode is the operating mode of the pump that provides the specified technical data.
A reverse shock wave is a pressure drop transmitted from layer to layer and propagating towards the gate.
The volumetric flow rate of a pump is the volume of liquid delivered by the pump per unit of time.
Positive displacement pumps *are* hydraulic machines used for pressurised fluid delivery.
Optimum pump operation is the operation of the pump at the highest efficiency value.
Smoothly varying motion - motion in which the curvature of the current line and the angle of divergence between them are very small and tend to zero in the limit.

Surface tension (capillarity) is a property of a liquid that is due to mutual attraction forces that arise between the particles of the surface layer and cause its stressed state.
Surface forces are forces acting on the surface of the fluid volumes being investigated, such as the forces of piston pressure on the surface of the fluid.
Total head is the sum of piezometric and velocity head.
Pump capacity *is* the volume of liquid delivered by the pump into the pipeline per unit of time.
Flow rate is the amount of fluid flowing through the live section of a stream per unit time.
The free surface of a liquid is the interface surface between the liquid and an external gaseous medium.
Free flow - the flow is not restricted by solid walls at all.
Compressibility is the property of a liquid to change its volume under the action of pressure.
A siphon is a short pipe that moves fluid from the supply tank A to the receiving tank B.
Velocity head is the height to which a fluid can rise above a given point in space under the action of the flow velocity of that point.
Average velocity of fluid flow is a conditional velocity equal to the ratio of flow rate to live section area.
Turbulent motion - motion in which all particles of a fluid move chaotically, i.e. the layering is broken.
The specific gravity of a liquid is the weight of a unit of its volume.
Specific volume is the volume occupied by a unit mass of liquid.
Specific fluid flow rate is the fluid flow rate per unit width of the live section.
The specific energy of a liquid is a measure of the mechanical energy of a liquid, equal to the energy belonging to a unit mass of that liquid related to the acceleration of free fall.
Steady motion *is a* type of motion in which velocities, accelerations, pressures, depths do not change with time, and depend only on the position in the fluid flow of the point under consideration, being a function of coordinates.
Pump characteristic - graphically expressed dependence of pump head, power and efficiency on pump capacity (delivery) at constant speed.
The characteristic of a piston pump *is the* dependence of fluid delivery on discharge pressure when the speed and viscosity are constant.
An elementary jet is a bundle of current lines drawn for the same instant of time.

Literature

1. Makhovikov B. S., Krivenko E. M. M., Gudilin N. S., Pastoev I. L. Hydraulics and Hydroprivod, textbook..Moscow: Gornaya kniga, 2017.
2. V.A.Kudinov. Hydraulics M: Higher School 2006.
3. A.D.Girgidov. Mechanics of Fluid and Gas (Hydraulics). Saint-Petersburg. Publishing house of SPbSPU. 2004.
4. Pastoev I. L., Berlizev N. I., Elenkin V. F. Hydropneumatic drive: textbook for students of higher educational institutions. F. Hydropneumatic drive: textbook for students of higher educational institutions, trained in the speciality "Mining Engineering", speciality "Mining Machines". "Mining Engineering", speciality "Mining Machines". Moscow: MGGU Publishing House, 1997.
5. V.S.Dulin, A.N.Zarya. Hydraulics and hydraulic drive. M. "Nedra" 1991.
6. E.Z.Rabinovich. Hydraulics M. "Nedra". 1980.V.A.Bolshakov, Hydraulics M. "Vysshaya Shkola", 1985.
7. A.V.Andreevskaya. Hydraulics. M. "Energia" 1980.

Internet sites

1.www.techno.edu.ru/db/sect/110
2.www.emomi.com/
3.www.toehelp.ru/theory/ter_meh/contents.html
4.www.mai.exler.ru/education/classified/theormech/
5.www.miit.ru/institut/ipss/faculties/trm/main.htm
6.www.protgu.ru/kurs2/teormex.html
7.www.physmech.ru/modules.php?name=Pages&pa=showpage&pid=10