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"National Research Technological University "MISIS" in Almalyk
(Branch of NUST MISIS in Almalyk)
DEPARTMENT OF "MINING"

EDUCATIONAL AND METHODICAL COMPLEX BY SUBJECT

«STATIONARY MACHINES»

(for students of the field of education: specialty
210504 - **«Mining»,** profile: Mining machinery and equipment)

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Educational and methodical

Reliability of mining machines

Developed in accordance with the OS IN:

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Introduction

Among the installations on which the reliability and safety of the operation of a modern mine depends, one of the main places belongs to mine drainage and fan installations. The mine drainage system is designed for pumping water entering the mine workings to the surface.

The importance of pumps in mining can be seen from the following indicators: to extract one ton of coal, it is necessary to pump out, for example, 2.5 tons of water, and in some cases much more.

The productivity of shaft pumps reaches 600 m3/h or more, and the power is up to 1000 kW or more.

The mine fan unit is designed for ventilation of mine workings. The value of fans in mining can be seen from the following indicators: to extract one ton of coal, you need to supply more than 5 tons of air to underground workings. The performance of modern shaft fans reaches 400 m3/sec, and currently projected – 500-600 m3/sec. The power of the fan motors reaches 2-3 thousand kW.

Drainage and fan installations consume 50-70% of the electricity consumed at the mine.

Therefore, they must be not only reliable in operation, but also highly economical installations.

The first part of this course covers the basics of the turbomachine workflow, designs, design methods and operation of mine drainage and fan installations.

The study of these installations is combined in one part of the course, since pumps and fans are organically connected by the commonality of the basic theoretical provisions, devices and their principle of operation. It should also be borne in mind that in drainage and fan installations, the density of the transported medium is almost constant, since the compressibility of air with a small pressure drop in the fan installation can be neglected.

Shaft pumps and fans together with their electric drive represent a single electromechanical complex, therefore, when studying the course, the issues of electrical equipment of drainage and vetilator installations, as well as the principles of automatic and remote control of their operation are considered.

In the development of designs and modern methods of calculating highly economical pumps and fans, an outstanding role was played by domestic scientists – Prof. N.E.Zhukovsky, academicians G.F., Proskura, A.P.German, etc.

A prominent place in the field of the theory of mining turbomachines belongs to acad.V.S.Pak, Professors V.G. Geyer, A.I.Veselov, etc.

BRANCH OF THE FEDERAL STATE AUTONOMOUS EDUCATIONAL INSTITUTION OF HIGHER EDUCATION
"National Research Technological University "MISIS" in Almalyk

COLLECTION OF LECTURES
on the subject

«Stationary machines»

LECTURE No. 1

GENERAL INFORMATION ABOUT HYDRAULIC MACHINES

Plan:

1. Liquid energy
2. Main parameters of liquid transportation machines
3. Energy losses in hydraulic machines
4. Classification of hydraulic machines for the transportation of liquids.

1. The energy of the liquid. It is known from hydraulics that the fluid always moves from areas with a higher specific energy to an area with a lower specific energy, i.e. in the presence of a difference in specific energies — pressures (pressures).

This difference can occur naturally, for example, the natural draft of air in the mine at the difference in its temperatures in the supply and exhaust shafts; the movement of water in the channels with the slope of the bottom, etc. However, in order to force the liquid to move with overcoming height, back pressure and resistance to movement, its flow must be artificially reported an increment of specific energy. The latter can be carried out if a hydraulic machine is installed on the path of fluid movement, the working bodies of which, interacting with the fluid, will transmit energy to the fluid flow from the outside, equal to the sum of work on lifting, overcoming back pressure, resistance to movement and creating kinetic energy. Such an energy source generator is a special hydraulic supercharger machine.

A liquid is a fluid body (medium), which can be in droplet and gaseous states. This unification is due to the unity of the laws that the fluid body obeys.

Superchargers in relation to the mining industry are machines primarily for transporting water and air. Machines for transporting fluid are the main part of fan, drainage and pneumatic installations, as well as air conditioning installations. This group of machines includes pumps, fans, blowers, compressors, vacuum pumps.

The machines receive mechanical energy from the drive motor and convert it into potential, kinetic and thermal energy of the fluid or gas flow being moved.

2. The main parameters of machines for transporting liquids. The main parameters characterizing the operation of pumps, fans, compressors and vacuum pumps are: capacity (supply), pressure (head), power and efficiency.

According to GOST, the terms "performance" and "pressure" are accepted for fans and compressors, and "supply" and "head" — for pumps.

Productivity (supply) is the amount of fluid supplied by a hydraulic machine per unit of time. The capacity (feed) is equivalent to the consumption in the network and can be measured in units of volume, mass and weight.

Accordingly, the capacity (feed) can be volumetric Q (m/s), mass t (kg/s) and weight G (N/s).

The volume, mass and weight quantities of liquids are related by the following ratios:

$$m = pQ; \quad G = pgQ, \tag{1.1}$$

where p – is the density of the liquid, kg/m^3; g – is the acceleration of gravity, m/s^2.

The pressure (head) of the fan (pump) is the difference in the specific mechanical energies of the fluid flow at the outlet of the hydraulic machine and at the inlet to it, i.e. the increment of the specific mechanical energy of the fluid passing through the hydraulic machine. Let's consider this parameter in more detail, using for this purpose the Bernoulli equation known from hydraulics for an elementary trickle of an ideal (inviscid) liquid:

$$E_G=\frac{p}{pg}+z+\frac{c^2}{2g}=const \qquad (1.2)$$

where p – is pressure, Pa; z – is height, m; c – is speed, m/s.

The terms of the equation can be interpreted: z is the specific potential energy of the position; p/rd is the specific potential energy of pressure; $c^2/2g$ is the specific kinetic energy of the fluid flow, and each term of the equation is the energy attributed to one newton of the weight of the flowing fluid and has a unit of measurement J/N or J/N = N.m/N = m.

Multiplying all the terms of equation (1.2) by g, we obtain the equation Bernoulli in the form of

$$E_m=\frac{p}{\rho}+gz+\frac{c^2}{2}=const \qquad (1.3)$$

Each term of this equation represents the specific energy of the mass of the flowing liquid in J/kg (or m2/s2).

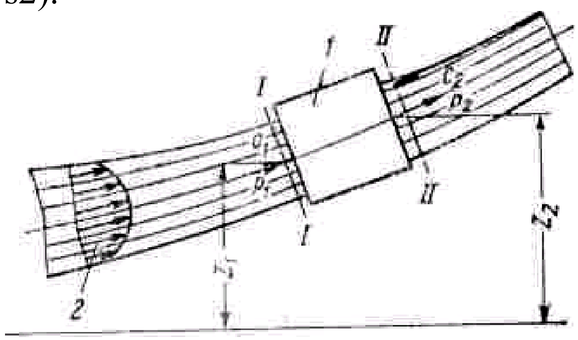


Fig. 1.1. To determine the pressure (head) of a hydraulic machine: 1—hydraulic machine; 2—speed plot

After multiplying equation (1.3) by p, we obtain an equation in which each term will represent the specific energy of 1 m3 of flowing liquid:

$$E_v=p+g\rho z+\frac{\rho c^2}{2}=const$$

The specific energy of the liquid is measured in Dj/m^3 or $\frac{H\cdot м}{м^2}$ = Па.

The specific energy of the fluid flow with a smoothly changing cross section is defined as the average value of the specific energy of the flowing fluid by the expression

$$E_g=\frac{p}{\rho g}+z+\frac{\alpha c_{cp}^2}{2g}=H\ ,$$

which differs from the formula (2.2) for an elementary trickle only by the coefficient a in the expression of kinetic energy, taking into account the uneven distribution of velocities over the flow section.

The coefficient depends on the unevenness of the velocity distribution plot over the cross section

$$\alpha = \frac{1}{c_{cp}^3 F}\int_F c^3 dF \qquad (1.4)$$

where F is the cross—sectional area normal to the fluid flow line. The increment of the specific energy of the fluid flow passing through the hydraulic machine (Fig. 1.1) is defined as the energy difference in the sections //—// and /—/:

weight

$$E_g = z_2 - z_1 + \frac{p_2 - p_1}{\rho g} + \frac{c_2^2 - c_1^2}{2g} \qquad (1.5)$$

volumetric

$$E_v = \rho g(z_2 - z_1) + p_2 - p_1 + \frac{p(c_2^2 - c_1^2)}{2} \qquad (1.6)$$

where EG and EV are the increment of specific energy, respectively, weight (m) and volume (Pa); p1 and p2 are the fluid pressure, respectively, at the inlet and outlet of the hydraulic machine. Pa; c1 and c2 are the fluid velocity, respectively, at the inlet and outlet of the machine, m/ s; z1 and z2 are the geometric height from the reference plane to the cross sections, respectively, at the inlet and outlet of the hydraulic machine, m

. Denote EP by H, and Em by R.

The volumetric increment of the specific energy p has a pressure dimension, Pa. Due to the smallness of the first term of the rd equation (z2—z1) for fans and compressors, they practically accept

$$p = E_v = p_2 - p_1 + \frac{\rho}{2}(c_2^2 - c_1^2) \qquad (1.7)$$

In relation to pumps, the concept of "head" is used, H = EG, i.e. an axial increment of specific mechanical energy having the dimension of meters of a column of liquid (m).

The pressure, unlike the head, depends on the density of the pumped medium. The relationship between pressure and pressure

$$p = \rho g H \qquad (1.8)$$

The total head (pressure) consists of static and dynamic heads (pressures):

$$H = H_{cm} + H_{дин}\,; \qquad p = p_{cm} + P_{дин} \qquad (1.9)$$

Static pressure is the increment of the specific potential energy of a fluid in a hydraulic machine:

a fan ($P_{ст}$. Pa)

$$P_{cm} = P_2 - P_1; \qquad (1.10)$$

pump (H_{st}, m)

$$H_{cm} = z_2 - z_1 + p_2 - p_1 / pg \qquad (1.10')$$

Dynamic pressure (pressure) is the increment of the specific kinetic energy of the fluid in the machine:

pump (H_{din}, m)

$$H_{дин} = \frac{(c_2^2 - c_1^2)}{2g} \qquad (1.11)$$

fan (one. Pa)

$$P_{дин} = \frac{\rho(c_2^2 - c_1^2)}{2} \tag{1.11}$$

Useful power is the power transmitted by a hydraulic machine to the flow of liquid passing through it. Since the head is a specific energy, the net power Np (kW) transmitted by:

by pump

$$N_n = \frac{g\rho QH}{1000} \tag{1.12}$$

fan

$$N_n = \frac{Q\rho}{1000} \tag{1.12'}$$

The efficiency of a hydraulic machine (efficiency) is the ratio of the useful power Np to the power N actually supplied to its shaft, and is determined by the expression

$$\eta = \frac{N_T}{N} \tag{1.13}$$

3. Energy losses in hydraulic machines. Energy losses in fans and pumps are divided into hydraulic, volumetric and mechanical, in compressors — also thermal.

Hydraulic losses are a part of the energy that the fluid flow in a hydraulic machine receives from its working organs and which is spent on overcoming hydraulic resistances when the flow moves in the channels of the machine.

Taking into account the hydraulic pressure losses, the energy actually received by the outgoing fluid flow — the actual pressure of the pumps and the actual pressure of the fans will be:

$$H = H_m — H_г; \quad p = p_m - p_г, \tag{1.14}$$

where Nt and rt are, respectively, theoretical head and pressure; Nt and rg are, respectively, hydraulic head and pressure losses in the pump and fan.

Hydraulic losses are characterized by hydraulic efficiency.:

for pumps

$$\eta_r = 1 - \frac{H_r}{H_T} \tag{1.15}$$

for fans

$$\eta_r = 1 - \frac{p_r}{H_T} \tag{1.16}$$

Volumetric losses are part of the energy received by the fluid flow and lost as a result of fluid leaks through gaps in seals, fluid overflows, removal of part of the fluid (in centrifugal pumps) to balance the axial force, etc. Therefore, the supply (productivity) Q of the hydraulic machine is less than the theoretical supply (productivity) Qt of the impeller by the amount of leaks q. Volumetric losses are characterized by volumetric efficiency.

$$\eta_0 = \frac{Q}{Q_T} = \frac{Q_T - g}{Q_T} = 1 - \frac{q}{Q_T} \tag{1.17}$$

Mechanical losses are a part of the energy received by a hydraulic machine from the engine and spent on overcoming mechanical friction in bearings, in contact seals, in centrifugal pumps on friction of the outer surface of the impeller and other rotor parts on liquid, etc.

Mechanical losses are accounted for by mechanical efficiency.

$$\eta_{м} = 1 - \frac{N_{CT}}{N} \tag{1.18}$$

where Ntr is the power of mechanical friction.

The difference N — Ntr is the hydrodynamic power Ngd transmitted by the fluid impeller,

$$N_{гд} = \frac{g\rho Q_T H_T}{1000}, кВт \tag{1.19}$$

The efficiency of a hydraulic machine is equal to the product of hydraulic, volumetric and mechanical efficiency.;

$$\eta = \eta_{г}\, \eta_{0}\, \eta_{м} \tag{1.20}$$

This efficiency is sometimes called the full or total efficiency of a pump or fan. It characterizes the efficiency of the hydraulic machine and is expressed by the ratio (1.13).

4. Classification of hydraulic machines for the transportation of liquids. Hydraulic machines are classified according to a number of characteristics:

1. According to the type of transported fluidity, they are divided into two groups:

for the transportation of drip liquids (water and hydraulic mixtures) -pumps, dredgers and ground pumps;

for the transportation of gaseous liquids (gases) – fans, blowers and compressors.

Pumps are machines designed to move liquids and communicate energy to them, dredgers and ground pumps are machines for pumping hydraulic mixtures – mixtures of water with soil.

Fans are machines for transporting air (gas) under relatively low pressure, the maximum value of which does not exceed 15,000 Pa.

Blowers and gas blowers are machines for transporting air and gas under pressure up to 0.3 MPa and without cooling it.

Compressors – machines for the production and transportation of air (gases) under pressure over 0.3 MPa and with compressed air (gases) cooling.

2. According to the design of the working body, hydraulic machines used in stationary installations can be divided into two main groups – vane and volumetric. Volumetric machines are divided into piston and rotary.

In blade machines, the working body is the blades, which have only rotational motion and transmit the energy of the liquid. These machines, depending on the fluid flow, are divided into centrifugal, axial and diagonal. The direction of fluid flow in the impeller is radial, axial and diagonal, respectively. Blade machines are characterized by continuous and uniform fluid supply, the absence of friction of the working blades on the body, the ability of the blades to work with high circumferential speeds. In volumetric piston machines, pistons (plungers, rolling pins) with reciprocating motion serve as the working body.

Such machines are characterized by pulsating fluid supply, limited speeds of movement of working bodies with large inertial loads on the drive, working bodies on the housing and the presence of suction and discharge valves. However, they can develop high pressures even at low piston speeds.

The working bodies of volumetric rotary machines are distinguished by their constructive diversity.

The group of rotary machines includes: screw, plate, gear, gear and slide. The common thing that unites them into one subgroup is the presence of one or two working rotors with profile teeth, screws or corrugation. Rotary machines are a combination of piston and blade machines, continuous, but not uniform fluid supply. Rotary machines operate at a high rotor speed and their dimensions are not large. They emit a strong high-pitched noise when working. These machines have a lot of wear of parts.

The movement of water and air through mine workings is carried out due to the difference in the total specific energies of the flow flowing in two sections. However, in practice, it is necessary to artificially create a difference in total specific energies with the help of machines.

Machines designed for this purpose are classified according to the principle of operation and by the type of energy that informs the flow of liquid.

3. According to the type of energy communicated to the fluid flow, dynamic and volumetric machines are distinguished.

4. According to the principle of operation, there are: turbomachines, volumetric machines, jet machines and airlifts.

The largest use has been received by blade turbomachines, which have smaller dimensions and high technical and economic indicators. Centrifugal pumps, fans and compressors are used in mining enterprises. Centrifugal turbomachines make up a significant part of pumping, fan and compressor machines used in mining enterprises. Centrifugal turbomachines can be either single- or multi-stage. Centrifugal turbomachine (Fig. 1.2.) consists of an impeller 2 with blades 3 fixed to the shaft 5, an inlet device 1, a spiral snail-shaped outlet device 4 and a diffuser 6.

Centrifugal turbomachines can have one-way suction impellers (Fig. 1.2.), i.e. with fluid supply to the wheel on one side, and two-way suction, i.e. with fluid supply from both sides to increase the flow and compensate for the axial force on the impeller.

The fluid flow is brought to the impeller in the axial direction, and in the wheel area acquires a radial direction. When the impeller rotates in the fluid flow, pressure differences occur on both sides of each blade.

The forces from the pressure of the blades on the fluid flow create a forced rotational and translational motion of the fluid, increasing its pressure and velocity. Thus, the flow receives an increment of energy only in the impeller. The remaining elements of the centrifugal turbomachine are stationary, and in them one type of energy can be converted into another. In the diffuser, the kinetic energy of the flow is converted into potential energy.

In an axial turbomachine (Fig. 1.3), the fluid flow is parallel to the axis of rotation of the impeller mounted on the shaft and rotated in a cylindrical casing. The impeller of the axial turbomachine consists of a sleeve 1 with blades 2 fixed on it at some angle.

Smooth flow of liquid to the impeller is provided by the collector and the front fairing, unwinding the flow behind the impeller – straightening apparatus. The liquid exit from the axial machine is carried out through an annular diffuser.

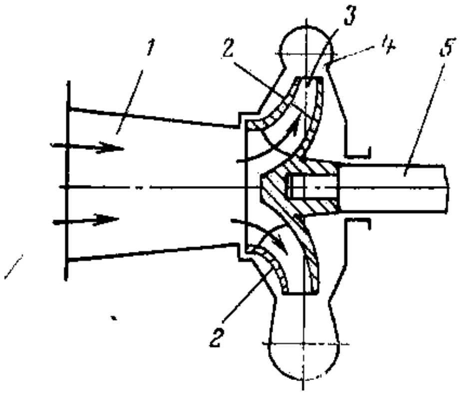


Fig. 1.2. The scheme of the centrifugal turbomachines

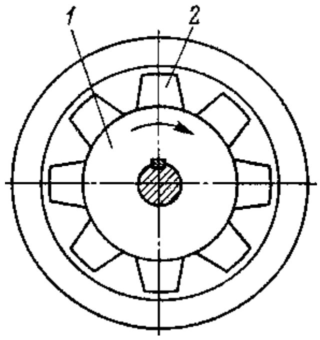


Fig. 1.3. The scheme of the axial turbomachine

In diagonal pumps (fig.1.4) the fluid moves from the center to the periphery of the impeller at an angle Y, and at the axial - parallel to the axis of rotation.

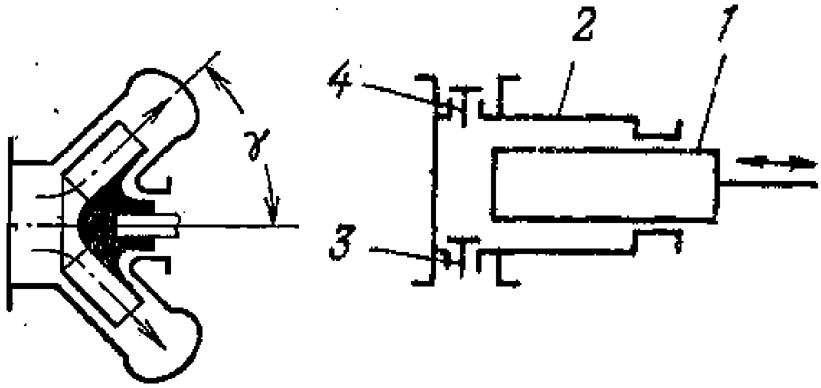


Fig. 1. 4. Diagram of the diagonal pump

In addition to the purely formal feature associated with the direction of fluid movement in the impeller, the separation of vane pumps into three types has a fundamental basis. In the working process of centrifugal pumps, centrifugal forces are of primary importance. In diagonal pumps , the ratio between the forces of viscous interaction with the liquid and centrifugal depends on the angle Y .

In axial pumps, energy conversion is determined mainly by the viscous interaction of the fluid with the blades, therefore diagonal pumps are considered as an intermediate type between centrifugal and axial.

Volumetric pumps are pumps in which the liquid medium moves by periodically changing the volume of the chamber occupied by it, which alternately communicates with the inlet and outlet openings of the pump. Depending on the form of movement of the non-working organ, this class of pumps is divided into two groups: rotary (rotary) and reciprocating.

Reciprocating pumps. This group of pumps has a rectilinear reciprocating motion of the working bodies, regardless of the nature of the movement of the leading link. The following types of reciprocating pumps are used for mass fluid supply: plunger, piston and diaphragm.

The working body of the plunger pump (Fig. 1.5) has the form of an elongated cylinder. The rotational motion of the pump drive shaft is converted into reciprocating motion of the plunger 1 by means of a crank mechanism. When the plunger

1 is pulled out of the cylinder 2, the volume of the pump's working chamber increases, the pressure in it drops, the suction valve 3 opens under the influence of atmospheric pressure and liquid enters (is sucked) into the chamber. When the plunger is reversed, the volume of the working chamber decreases. Due to the higher pressure in the chamber, the suction valve closes, and the discharge valve 4 opens, and through it the liquid is forced out of the pump.
A piston pump is not fundamentally different from a plunger pump. The length of the piston in proportion to its diameter is much smaller than that of the plunger. When moving, the piston touches the walls of the housing with its entire cylindrical surface, while the plunger comes into contact with the housing only in the seal assembly located at the entrance to the working chamber. In the diaphragm pump, the role of the displacer is performed by one of the walls of the working chamber, made of rubber. The diaphragm is pressed into the chamber and its pulling is carried out periodically by means of a pusher with an eccentric drive.
Rotary pumps. In this group of pumps, the volume of the working chambers is changed during the rotational movement of the working body- the rotor. Three types of rotary pumps are used to supply liquid at quarries: screw, hose and plate pumps. The working bodies of screw pumps are one, two or three screw rotors. If two screw rotors touch so that the protrusions of one enter the depressions of the other, then during rotation the overlap point of the depression with the protrusion moves along the axis of rotation, causing the displacement of liquid from the cavity along screw lines in one direction and suction in the other.
The hose pump, the schematic diagram of which is shown in Fig.1.6, consists of a flexible rubber hose 1 fixed in the housing 2 and a gear rotor 3 bearing the forces. When the rotor rotates, the rollers squeeze the hose into two parts. As the pinch point moves, one of the parts will decrease in volume, and the other will increase. The liquid is absorbed into the first, and is displaced from the second.
The plate pump (Fig. 1.7.) consists of a housing 1, in which the rotor 2 rotates eccentrically, bearing freely seated plates 3 in radial grooves. When the rotor rotates, centrifugal forces push the plates out of the grooves, pressing them against the inner surface of the cylindrical housing. Isolated chambers 4 are formed between each pair of plates and the surfaces of the cylinder, rotor and side covers, the volume of which varies from 0 to maximum and again to 0. That part of the crescent-shaped space between the rotor and the housing, where the volume of the chambers increases, communicates with the suction nozzle, and the other with the discharge.

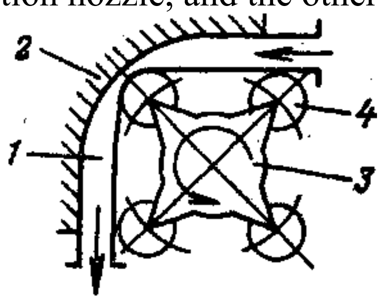


Fig.1.6. Hose pump diagram

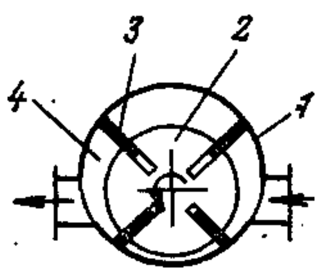


Fig. 1.7. Plate pump diagram

Friction pumps. In pumps of this group, the liquid medium moves under the influence of friction forces arising on contact with a moving working body. According to the type of working bodies, these pumps are divided into the following types: disk, scoop, vibration, screw, vortex, free-vortex, jet and airlifts. The working body of the screw pump (Fig.1.8, a) is made in the form of a screw rotor 1 rotating in a cylindrical housing 2. Due to the friction forces arising on contact with the screw surface of the screw, the liquid moves in the direction of its axis of rotation.

In the free-vortex pump (Fig.1.8, b) rotating in the housing 2, the rotor 1 is made in the form of a concave disk with blades. The fluid involved in the rotational motion by friction forces moves under the action of centrifugal forces mainly outside the rotor from its center to the periphery.

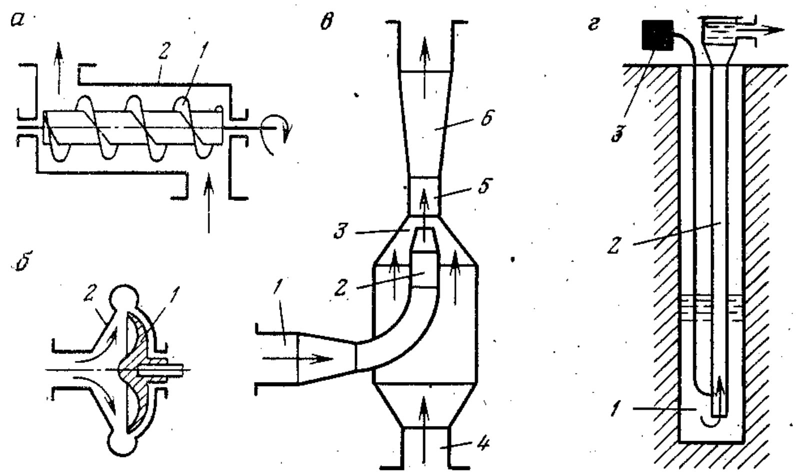


Figure 1.8. Schemes of friction pumps

Jet pumps and airlifts differ from all other pumps in that in them the role of a moving working body is performed by a liquid.

In the jet pump (Fig. 1.8, c), the working fluid flows through the inlet 1 to the nozzle 2. Leaving the nozzle at high speed and entering the mixing chamber 3, the working fluid engages in motion the liquid being sucked in it and in the suction pipeline 4. In the neck five, the energy exchange between the working and the sucked liquids is completed, and in the expanding diffuser 6, a gradual decrease in speed occurs with a partial transformation of the dynamic pressure of the liquid into its static pressure.

The airlift (Fig. 1.8, d) is a water-lifting pipe 2 lowered into the water collector 1.At the lower cutoff, compressed air is supplied to the pipe 2 from the compressor 3. Due to the lower density, air bubbles float in the liquid, involving it in motion by friction forces. The rise of the water-air mixture through the pipe 2 is also facilitated by its lower density compared to the density of the liquid medium in the water collector 1.

At drainage installations, the most widespread are vane pumps with soft pressure characteristics. Friction pumps have a rigid head characteristic. Volumetric pumps have a rigid flow characteristic. The pressure developed by them is theoretically not limited by anything and depends only on the condition created by the drive on the working organ and the strength of the pump design. The overlap of the discharge when the volumetric pump is running is unacceptable.

LECTURE No. 2

THE MAIN TYPES OF TURBOMACHINES AND THEIR ELEMENTS

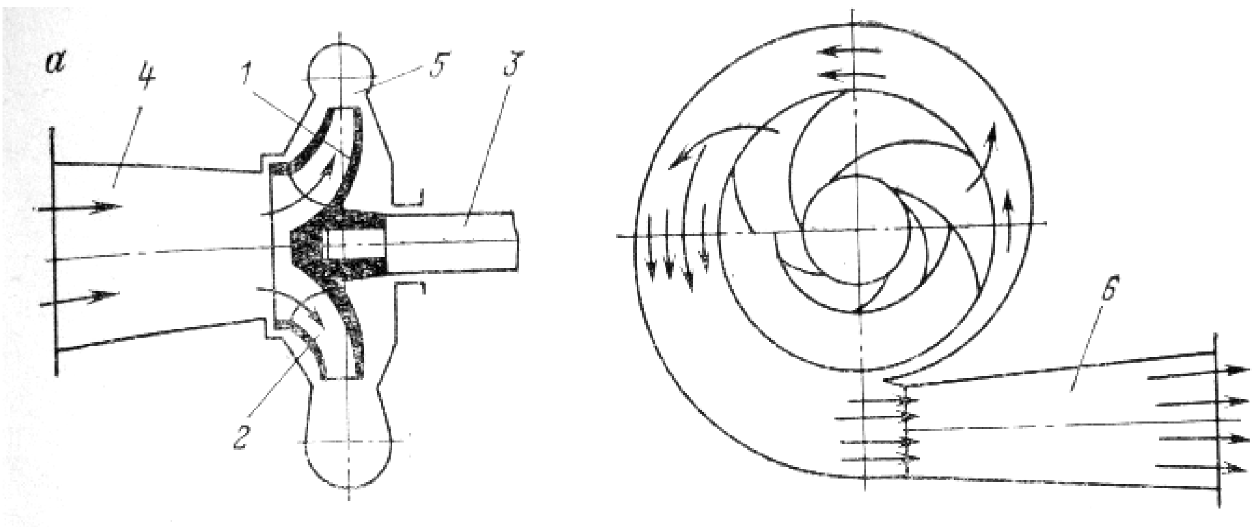
Plan:

1. The principle of operation and the main elements of turbomachines
2. Kinematics of fluid flow in the impeller
3. The basic energy equation of turbomachines
4. Fundamentals of vortex theory of turbomachines

1. The principle of operation and the main elements of turbomachines. Turbomachines used in the mining industry: pumps designed for pumping and supplying water; fans that ventilate mine workings; turbochargers that produce compressed air are characterized by a single operating principle. Depending on the direction of fluid flows relative to the axis of rotation of the impeller, they can be centrifugal, axial and meridional (diagonal). In the mining industry, the last group of turbomachines has limited use. Axial turbomachines are used in mining enterprises mainly as fans.

The centrifugal turbomachine (Fig. 2.1) consists of an impeller 1 with blades 2 mounted on a shaft 3, an inlet device 4, a spiral snail-shaped outlet device 5 and a diffuser 6.

The fluid flow is brought to the impeller in the axial direction and at the entrance to the latter changes its direction and in the inter-blade channels of the wheel moves already in the radial direction, moving along the blades from the entrance to the wheel to the exit from it.



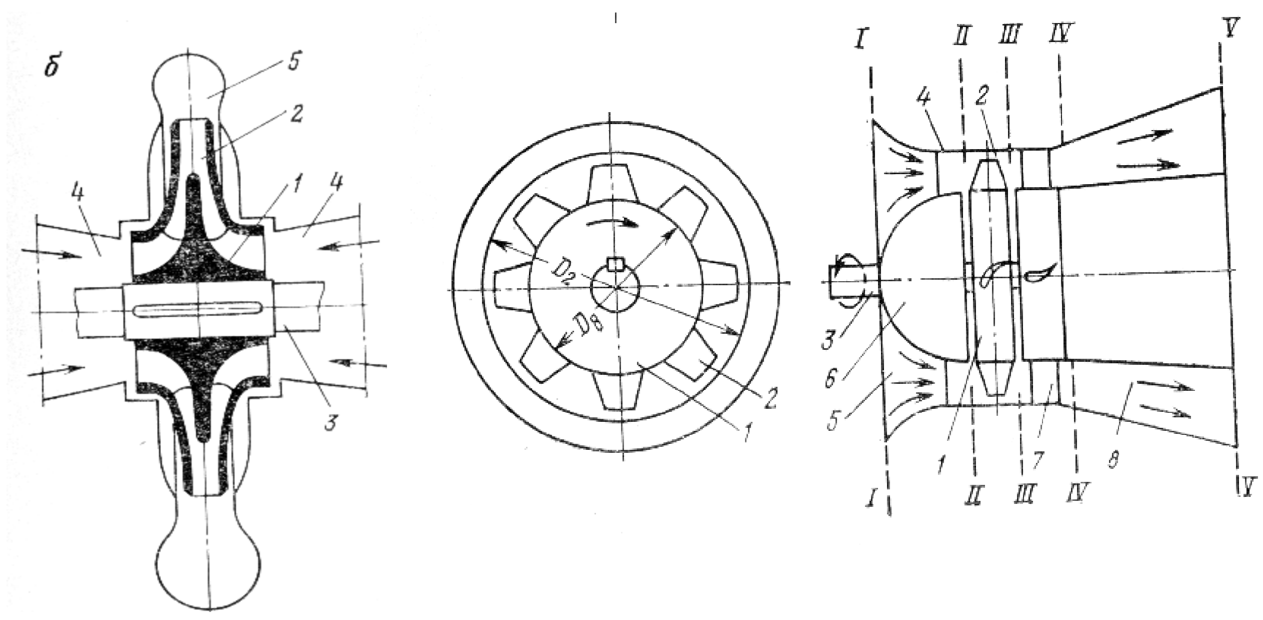


Figure 2.1 Centrifugal turbomachine

A centrifugal turbomachine can have a one-way suction impeller, i.e. with liquid supply to the wheel on one side, and with two-way suction, i.e. with two-way liquid supply, to increase productivity (supply).

The axial turbomachine (Fig. 2.2) consists of an impeller 1 with blades 2, a shaft 3, a casing 4 with an inlet device (collector) 5, a front fairing 6, an output device — a straightening device 7 and a diffuser 8. The straightening device installed behind the impeller serves to unwind the fluid flow coming out of the wheel twisted. From the axial turbomachine, the fluid flow is supplied to the impeller and diverted from it in the axial direction. The impeller of a turbomachine, being its main element, receives energy from the engine and transmits it through the blades to the fluid flow, while increasing its pressure (head).

The impeller blade is a wing — a slightly curved, conveniently streamlined body with a rounded part running into the flow and a pointed end, and the impeller is a lattice of such jointly working wings. The designs of the blades of centrifugal and axial turbomachines have significant differences.

To reduce the turbulence of the fluid flow at the inlet and the shockless entry into the impeller, a special fairing 6 is installed in front of axial turbomachines, while the centrifugal fairing is performed at the same time with the impeller (see Fig. 2.1, a).

The supply device (supply) provides fluid supply to the impeller with a uniform, if possible, flow velocity field along its cross section.

The purpose of the discharge device is to collect the flow coming out of the impeller at high speed, convert its kinetic energy into potential pressure energy and divert the liquid to the discharge pipe or the next impeller. In the removal of axial machines, partial or complete unwinding of the flow twisted by the impeller can also occur. The flow in the outlet due to the smooth expansion has a diffusor flow character, i.e., the fluid velocity decreases and the pressure increases. If there is a diffuser behind the discharge device in the latter, the flow velocity is further reduced and the kinetic energy of its movement is converted into potential energy (static pressure).

It should be noted that the increment of the specific energy of the flow occurs only in the impeller, in the other elements — energy conversion and reduction of the total pressure due to energy losses to overcome resistances.

Axial shaft turbomachines are performed only with sequential connection of impellers. Centrifugal and axial turbomachines are usually combined into one group of vane (blade) machines. This is due to the fact that they can be considered as limiting cases of diagonal machines (Fig. 2.4, b). In this view, a centrifugal turbomachine is a diagonal machine with an angle of □ = 90 ° (Fig. 2.4, a), and an axial one with an angle of □ = 0° (Fig. 2.4, b). This unity does not exclude significant design differences between axial and centrifugal machines.

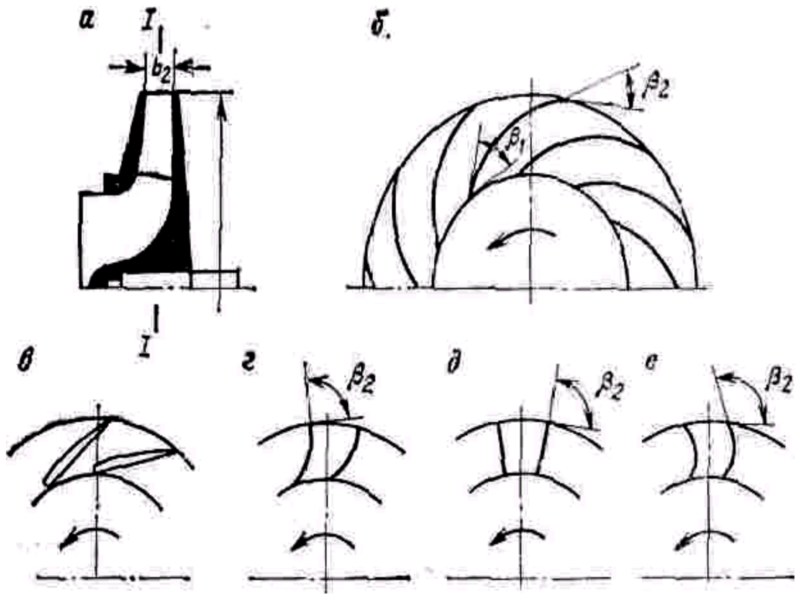
2. Kinematics of fluid flow in the impeller. The fluid movement in the flow channels of turbomachines has a very complex spatial character. The flow parameters vary both along the width of the wheel and along the circumference of a fixed radius.

Fig. 2.4. Turbomachine impellers diagrams

To simplify, the three-dimensional model of the fluid flow in the impeller is replaced by a two-dimensional one that preserves the basic properties of the flow. Such a model is used, in particular, when considering the kinematics of the flow, choosing as its kinematic parameters the velocity of liquid particles near the inlet and outlet edges of the blades. The values of velocities are understood as their values averaged over the pitch and width of the interscapular channel.

The centrifugal impeller of a turbomachine has an inlet section for fluid flow in a plane perpendicular to the axis of rotation, and an outlet in a cylindrical surface with an axis coinciding with the axis of rotation.

To obtain a two—dimensional model of the flow in a centrifugal wheel, it is conditionally dissected by a plane /-I perpendicular to the axis of rotation (Fig. 2.5, a). In this case, the sections of the blades forming a radial (circular) lattice are obtained (Fig. 2.5, b).



2.5. Centrifugal impeller (a) and blade profiles: backward curved leaf (b); wing-shaped (c); radial curved leaf (d); flat (e) and curved forward (e)

The fluid flow in the radial grating is assumed to be plane-parallel, i.e. the same in width of the wheel.

The analysis of the flow kinematics within the impeller is based on the construction of parallelograms of fluid flow velocities at the inlet and outlet of the impeller. To build them, it is necessary to know the magnitude and direction of the speeds, which are determined by the dimensions of the impeller, the geometry of its flow channels and the operating mode. At the same time, the shape and profile of the working blades have a decisive influence. They are performed bent backwards, β < 90° (Fig. 2.5, b, c), radial, β2 = 90° (Fig. 2.5, d, e) and bent forward, β2 > 90° (Fig. 2.5, e), profiled in cross section (Fig. 2.5, e) and thin, practically

unprofiled (leaf) (Fig. 2.5, d, d, e). In Fig. 2.5, β1 and β2 denote the input and output angles of the blades between the tangents to the circles of the gratings and the blades at their input and output edges. Passing through the impeller during its rotation, the fluid participates in portable (together with the impeller) and relative (relative to the wheel) movements with velocities and and w. The absolute velocity of the fluid particles is equal to the geometric sum:

$$\vec{c} = \vec{u} + \vec{w} \qquad (2.1)$$

Absolute velocity is the velocity of a liquid particle relative to a stationary body.

The portable speed and in absolute value is equal to

$$u = \frac{2\pi rn}{60} \qquad (2.2)$$

and is directed tangentially to the circle of radius g; p is the speed of rotation of the impeller.

The relative velocity w, with which the flow moves in the inter-blade channels, also varies in magnitude and direction.

Axial impeller of the turbomachine. Unlike the wheel of a centrifugal machine, the sections of the inlet and outlet of the fluid flow of the axial impeller are in planes perpendicular to the axis of its rotation. The liquid moves through the wheel translationally and simultaneously twists in the direction of rotation.

We dissect the impeller (Fig. 2.7, a) with a cylindrical surface with radius r and select an annular trickle of liquid with a thickness of Δr, within which the flow parameters (velocity and pressure) can be considered constant (due to the smallness of Δr).

By turning the cylindrical surface of the cut into a plane, we obtain the so-called flat lattice of profiles (Fig. 2.7, b) of the axial impeller. The main parameters of this lattice are: blade width (chord length) b; lattice width B; number of blades z; blade angle θ formed by its chord and the century-

the velocity torus and; the angles of entry and exit of the blades β1 and β2. An important parameter is the lattice pitch t = equal to the distance between the similar points of the blade sections measured in the direction of rotational motion of the lattice. The ratio b/t is called the lattice density, and t/b is the relative pitch.

When the impeller rotates, the particles of the liquid flowing through the lattice participate in relative motion along the lattice (with relative velocity w1 at the entrance to the lattice and w2 at the exit from it) and in portable motion — with circumferential velocity and = wr. At a constant angular velocity y for a cylindrical surface of a given radius r, the velocity i = const.

In the absence of a swirling flow in front of the impeller , the liquid flows to the grate at an absolute speed and has an absolute speed at the outlet of the grate . Figure 2.7 shows the triangles of velocities at the entrance and exit from the grid.

Based on the flow continuity equation for incompressible ca1 and ca2 fluid, it can be proved that the axial velocities of ca1 and ca2 at the inlet and outlet of the impeller of the turbomachine are the same: ca1 = ca2 = ca is the speed at which particles move along the axis of the impeller.

The relative velocity w1, at the entrance to the lattice is directed at the angle of attack δ — the angle between the tangent to the median line of the blade and the relative velocity at the entrance.

Passing through the lattice, the fluid flow from the interaction with the blades bends and the relative velocity w changes its direction, deviating towards the rotation of the lattice. N. E. Zhukovsky and S. A. Chaplygin showed that the curved flow by the interaction effect can be replaced by an equivalent rectilinear flow with an average relative velocity

$$\vec{w}_{cp} = \frac{\vec{w}_1 + \vec{w}_2}{2} \qquad (2.6)$$

This conclusion is important for analyzing the working process of an axial turbomachine.

By combining the velocity triangles of the liquid particles at the inlet and outlet of the lattice, we obtain a velocity plan from which we determine the angle of inclination of the velocity vector βcp and its absolute value:

$$tg\beta_{cp} = \frac{c_a}{u - \frac{c_u}{2}} \qquad (2.7)$$

$$w_{cp} = \sqrt{c_a^2 + \left(u - \frac{c_u}{2}\right)^2} \qquad (2.8)$$

where c_a – is the projection of the absolute velocity vector c2 on the direction of the vector and .

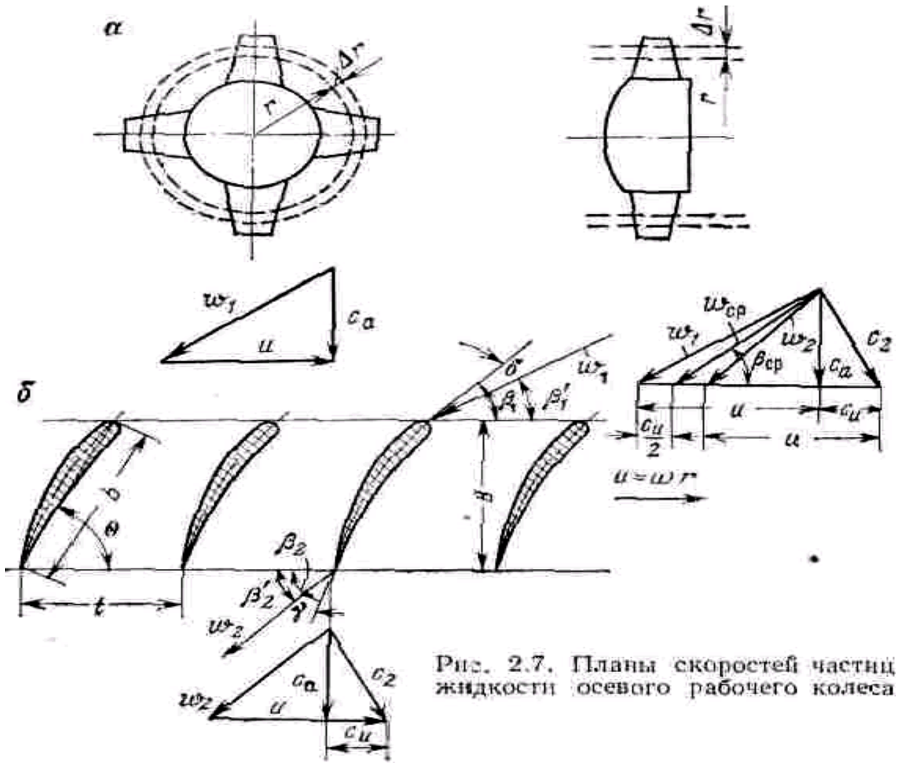


Рис. 2.7. Планы скоростей частиц жидкости осевого рабочего колеса

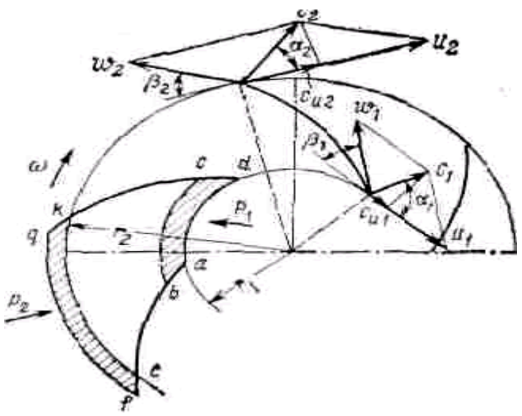


Figure 2.8 Diagram for the derivation of the Euler equation

The cu velocity is called the flow twist velocity.

If the absolute value of w1 > w2, then the lattice has a retarding effect on the flow and is called diffusor. If the relative flow velocity in the impeller increases (w1 <w2), then the lattice is called confusory, at a constant velocity w (w1 = w2) — active. In shaft fans, the diffusor grille has received the greatest use, the active one is practically not used.

The theoretical productivity (feed) of the QT (m3/s) of the impeller is determined by the expression

$$Q_T = c_a \pi / 4\left(D_2^2 - D_B^2\right) \qquad (2.9)$$

where d_B is the diameter of the impeller sleeve, m.

3. The basic energy equation of turbomachines. Due to the complexity of fluid movement in the flow channels of real turbomachines, they resort to idealizing the processes occurring in the impeller. In particular, when theoretically considered, the concept of a theoretical turbomachine or, more correctly, a theoretical impeller is widespread, since theoretically only the latter is considered without diverting and underwater elements.

The theoretical impeller of a turbomachine will be called an imaginary wheel in which all the energy transmitted by the drive to the impeller is completely transferred by the fluid blades, i.e. there is no energy dissipation.

Consider a workflow based on the jet theory. According to this theory, the fluid flow is considered to consist of countless elementary jets separated by infinitely thin blades. In this case, the relative velocity of each trickle at any point will be tangent to the blade, i.e. the angles 6 and y (see Figure 2.6) will be zero, β1 = β'1 and β2 = β'2.

It should be noted that such a scheme is very conditional, since with an infinite number of blades and the separation of the flow into elementary streams, the pressure on both sides of the blades will be equal at the same radii and the impeller will not be able to create pressure. Other assumptions: the fluid flow is assumed to be axisymmetric, and the fluid is inviscid.

To obtain the basic equation of the impeller, we will use the theorem on the change in the moments of the amount of motion of the material system. This theorem makes it possible, without knowledge of complex phenomena occurring during the flow of fluid in the wheel, to present the general nature of fluid movement with the necessary quantitative indicators.

Let us determine for the centrifugal wheel (see Fig. 2.8) the moment M of external forces, due to which there is a change in the moment of the amount of movement of the fluid enclosed in the wheel.

During the time dt, a mass dm equal to the mass of the liquid abcd will enter the impeller channel, and at the same time a mass dm equal to the mass ekgf will leave the wheel channel. The amount of mass movement dm at the entrance to the wheel dts1, and at the exit from it dts2 .

In 1 s, with the mass productivity (feed) of the impeller t, the amount of flow movement will be mc1 and tc2, respectively.

4. Fundamentals of the vortex theory of turbomachines. Fluid movement in the impeller. The relative motion in the inter-blade channels can be schematically represented as the sum of three fluid motions: in a stationary lattice; vortex motion inside the inter-blade channels caused by the rotation of the lattice; circulating fluid motion around the blades. In this case, the relative velocity in the channels of the impeller can be considered as the sum of three speeds:

$$w = w_{\text{ср}} + w_{\text{ц}} + w_{\text{о}} \qquad (2.10)$$

where wpc is the velocity of fluid flow in a stationary wheel;

wc — the velocity of vortex motion in the channel;

wo is the velocity of the flow circulation around the blade.

The diagrams of the velocities wsr of the fluid flow relative to the fixed lattice are shown in Fig. 2.9, a (channel /). In this case, if we neglect the influence of the friction forces of the liquid on the surface of the blades, the velocity of the liquid streams along the width of the channels are the same and decrease to the periphery of the wheel due to an increase in the cross-section of the channels.

The third circulation movement (see Figure 2.9, a, channel III) is caused by the action of the blade on the fluid flow like a wing; its speed depends on the profile of the curvature of the blades and the angle of attack 6 (see Figure 2.6).

For blades bent backwards (□2 < 90°), the velocities of the circulation motion are subtracted from the velocities of the vortex motion, which reduces the difference in relative velocities on the working and back sides of the blades. When the blades are bent forward (Ra > 90 °), the speeds of both movements add up, which increases the difference in relative speeds and contributes to the transfer of energy by the impeller to the fluid flow. In the case of radial blades and an unstressed flow inlet into the profile grid (which can be achieved theoretically by installing a device for pre-twisting the flow cu1 □ 0 in front of the wheel) wc = 0, and there is no circulation movement.

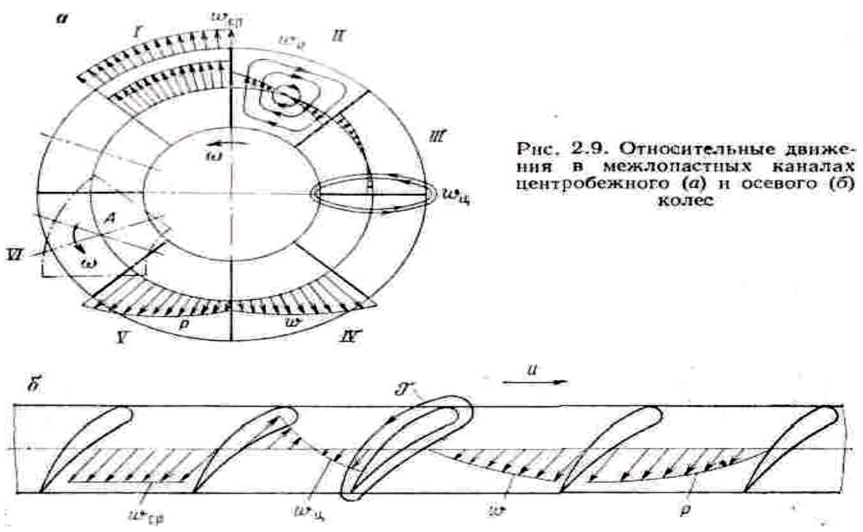


Рис. 2.9. Относительные движения в межлопастных каналах центробежного (а) и осевого (б) колес

Summing the plots of the three speeds gives a plot of the relative velocity w, from which it can be seen that the speed decreases from the back of the blade to the working side (channel IV), and the pressure p, according to Bernoulli's law, on the contrary, from the working to the back of the blade (channel Y). The pressure difference on the working and back sides of the blade is a necessary condition for the transmission of mechanical energy by the impeller to the fluid flow.

In an axial turbomachine, there is no vortex motion inside the inter-blade channels and the flow movement in the profile grid can be considered as the sum of two fluid movements: in a fixed grid and circulating around the blades (Fig. 2.9, b}:

$$w = w_{ср} + w_{ц} \qquad (2.11)$$

The circulating motion of the fluid around the blades, as will be shown below, determines the interaction of the blade arrays with the flow, and without it, the operation of turbomachines is impossible. The quantitative assessment of vortex motion is carried out by the so-called velocity circulation.

Circulation of the flow around the blades. Recall that velocity circulation is a kinematic characteristic of the vortex motion of liquid particles (vortex) around some instantaneous (like a solid) axis, stationary or moving, and represents the work of the linear velocity vector along a closed contour (Fig. 2.10):

$$\Gamma = \oint c \cos \varphi ds \qquad (2.12)$$

where c is the flow velocity at a given point of the contour, m/s; ds is the element of the contour length, m; (p is the angle between the velocity vector c and the element az, degree.

In the simplest case, when the contour is bounded by a circle,

$$\Gamma = 2\pi rc. \qquad (2.13)$$

Velocity circulation has the important property that it is not determined by the shape of a closed loop and depends only on the magnitude of the vortex that it is caused by.

The circulation along the contour is equal to the sum of the circulation and inside the contour:

$$\Gamma = \sum_{0}^{n} \Gamma_i \qquad (2.14)$$

The basic equation of the vortex theory. Multiplying both parts of the above equations (2.15) and (2.15') by the angular velocity w of the wheel, we obtain the expressions:

$$\omega \Gamma_{ц} = 2\pi(u_2 c_{u2} - u_1 c_{u1}) \qquad (2.15)$$

$$\omega \Gamma_{o} = 2\pi(u\ c_{u2} - u_1 c_{u1}) \qquad (2.15^{/})$$

in the right part of which the binomial enclosed in brackets represents the theoretical pressure of HT, determined by the equation of L. Euler. From here we get an expression for the theoretical pressure of the impeller through circulation in the form

$$H_T = \frac{\Gamma \omega}{2\pi g}. \qquad (2.16)$$

where Γ is the total circulation generated by the impeller.

Consequently, the theoretical head of a turbomachine does not depend on the type of fluid, but is entirely determined by the circulation around the blades, their number and the speed of rotation of the impeller.

LECTURE No. 3

MAIN TECHNOLOGICAL CHARACTERISTICS OF TURBOMACHINES

Plan:

1. Individual and valid characteristics of turbomachines
2. The effect of a finite number of blades and performance (feed) on the operation of turbomachines.
3. Supply and branch lines and their influence on the characteristics.

1. **Individual and valid characteristics of turbomachines.** The characteristics of turbomachines are called graphical representation of dependencies $H = f(Q)$, $N = f(Q)$, $\eta = f(Q)$, $H_m = f(Q)$. These dependencies are given when $n = const$. The most important is the dependence of the pressure on the feed.
Consider the Euler equation and the parallelogram of velocities at the exit of the wheel

$$H_m = u_2\, v_{2u} / g\,, \qquad v_{2u} = u_2 - v_{2r} \cdot ctg\, \beta_2\,,$$

besides

$$Q_m = \pi\, D_2\, b_2\, v_{2r} \qquad \text{or} \qquad v_{2r} = Q_m \,/\, \pi\, D_2\, b_2\,,$$

then

$$v_{2u} = u_2 - ctg\, \beta_2\, Q_m / \pi\, D_2\, b_2\,.$$

After substituting into the Euler equation we get

$$H_m = (u_2 - ctg\, \beta_2\, Q_m / \pi\, D_2\, b_2)\, u_2 / g\,.$$

Circumferential speed u_2 defined by $u_2 = \pi\, D_2\, n \,/\, 60$. After performing the transformations , we get:

$$H_m = (\pi\, D_2\, n)^2 / 3600\, g \;-\; Q_m\, n\, ctg\, \beta_2 / 60\, g\, b_2\,.$$

For simplification, we will take (for centrifugal machines):

$$(\pi\, D_2\, n)^2 / 3600\, g \;=\; const = C$$

$$n\, ctg\, \beta_2 / 60\, g\, b_2 \;=\; const = E$$

We get the dependency $H_m = C - E\, Q_m$. This is the equation of a straight line, the position of which depends on the angle β_2. Values n, D_2 u b_2 set. The theoretical pressure is higher the more the blades are bent forward (Fig. 3.1).

The pressure of the machine is achieved by increasing the absolute speed v_2. In the expanding channels of the diffuser, the dynamic pressure is converted into static pressure, while energy losses are observed, which are greater the greater v_2.

Currently, clean water pumps are used with impellers whose blades are bent backwards, i.e. $\beta_2 = 20 \div 35°$, for ground pumps $\beta_2 \leq 50°$. The blades bent forward are used on some types of local ventilation fans.

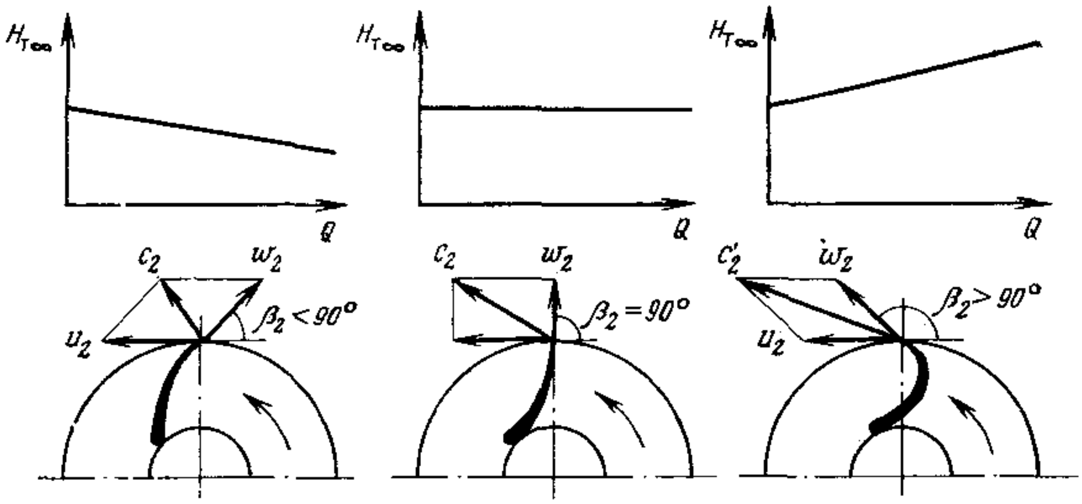


Fig. 3.1. Theoretical pressure characteristics of pumps with different profiles of impeller blades

The actual pressure differs from the theoretical one by the amount of pressure loss in the flow part of the machine.

The amount of pressure loss consists of the following components:

1. Mechanical pressure losses hm, friction losses in bearing assemblies and seals, friction of external surfaces on the liquid. For this machine, the losses are hmn = const , and depend on the design, size and operational condition of the machine. They are evaluated by the mechanical efficiency of the pm .

2. Braking losses ht, losses due to the occurrence of reverse flows at the boundary of the inter-blade channel, these flows slow down the main flow. Losses occur with small feeds and decrease with increasing feed.

3. Pressure losses due to the finite number of blades hk occur, as there is an overlap of potential and inertial vortices on the relative flow. hc losses decrease somewhat with increasing feed.

4. The friction losses of the flow htr, in the turbulent mode, are proportional to the square of the flow rate (supply) of the flow.

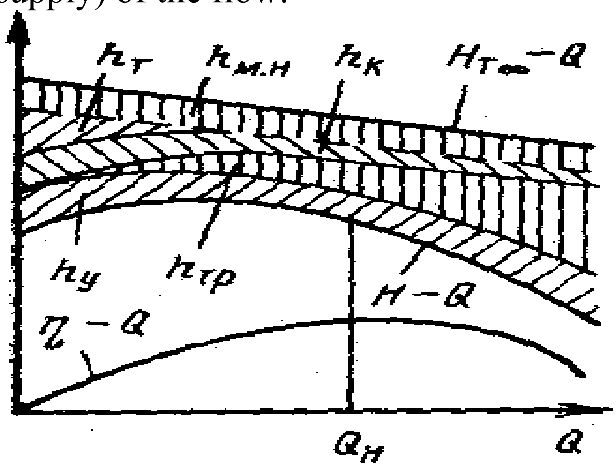


Fig. 3.2. Diagram of the turbine head loss

5. Shock losses of pressure hy, losses at the moment of liquid entry into the interblade channel and at the exit from it.

Losses ht , hc , ht ,hy are called hydraulic losses hg and are characterized by hydraulic efficiency pg .

6. Volumetric losses arise due to a decrease in the actual supply compared to the theoretical one and are characterized by a volumetric efficiency of the software.
The total efficiency of the turbomachine is determined by

$$\eta = \eta_{м} \eta_{г} \eta_{o}$$

The operating characteristics of a turbomachine are called dependency graphs
H = f (Q), N = f (Q), η = f (Q) taking into account all losses at *n = const*.

Characteristic points of the curve H = f (Q) are the points M, L и N . Point M determines the pressure of the machine when the valve is closed, point L is the maximum pressure. The LM section is the zone of unstable operation of the machine. The point N characterizes the normal flow at maximum efficiency. and is always located below the point M, which allows the machine to overcome the geodetic lifting height, hydraulic losses and create the necessary high-speed pressure.

2. **The effect of a finite number of blades and performance (feed) on the operation of turbomachines.** The influence of a finite number of blades. The fluid flow in the channels of the impeller with an infinite number of blades differs significantly from the flow in a real impeller with a finite number of them. Due to the presence of the thickness of the blades in the channels of the impeller, the flow narrows, and at the outlet it expands.

At the entrance to the impeller, at the moment when the liquid enters the inlet edges of the blades of the rotating wheel, the velocity vector changes from the liquid. At the same time, energy is lost. Energy losses depend on the performance (feed), the number of blades and their thickness. At a certain performance, they will be minimal.

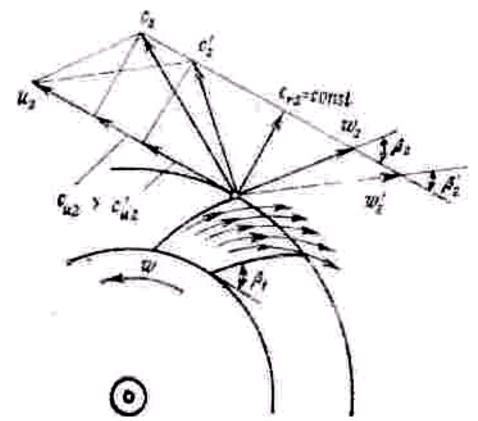


Fig.3.3 Deviation of the flow at the outlet of the centrifugal impeller with a finite number of blades

After the flow exits the impeller, the field of relative velocities is leveled and the flow, under the action of an axial vortex and circulation, deviates from the direction set by the tangents to the blades at their exit, in the direction opposite to the rotation of the wheel. The angle β2 between the average relative velocity β_2 of the aligned flow and the reverse direction of the velocity u_2 turns out to be actually less than the output angle of the blade β2 (Fig. 3.3.).

This leads to an increase in the relative velocity ($w_2 > w_2$) and to a decrease in the twisting speed ($c_{u2} < c_{u2}$), which causes a decrease in the theoretical pressure.

Thus, the theoretical head of HT with a finite number of blades will be less than the head of HT, determined by the equation of L. Euler under the assumption of an infinite number of impeller blades (H_T < H_T). This difference depends on the performance and, according to calculations, can be up to 30%, Various correction factors are used to determine the change in the theoretical head, taking into account the finite number of blades z of the impeller. The k_z coefficient is determined by the results of machine tests and calculations based on approximate dependencies under various assumptions obtained by a number of authors (for pumps by Proskura, Stodola, Maisel, etc.).

The thickness of the blades of a real turbomachine also affects the performance (feed) of turbomachines.

The impact of productivity (feed). Figure 3.4 shows the plans of speeds at the inlet and outlet to the impellers of a centrifugal machine at different capacities (feeds) Q. With a change in Q, the radial velocity of sg proportional to it changes in magnitude and the direction and magnitude of the absolute velocity of c change. Speeds without a stroke correspond to small Q values and with strokes to large Q values.

At the input to the impeller , a change in performance Q causes a change in the vector and its direction . With increasing productivity, the Q angle decreases, and with its decrease it increases. At small and large Q, local flow separation (creation of a vortex zone) may occur at the inlet edge of the blade, which leads to an increase in hydraulic losses and a decrease in pressure. At low costs, the separation zone is located on the back side of the blades, at large - on the working side. Energy losses at the input will be minimal if the angle is close to the input angle of the blade.

3. Supply and branch lines and their influence on the characteristics. The shape, size, location and design of devices for supplying and removing fluid from the impeller of a turbomachine affect not only the amount of energy transmitted to the flow in the turbomachine, but also its entire characteristics.

The supply devices of turbomachines must meet the following requirements:

to ensure uniform, axisymmetric flow distribution over the input section of the impeller, which improves the efficiency of the machine;

the speeds in the supply sections should not be high and increase gradually to its value in the input section to obtain minimal losses in the supply;

the design of the supply should create a convenient interface of the machine with the pipeline.

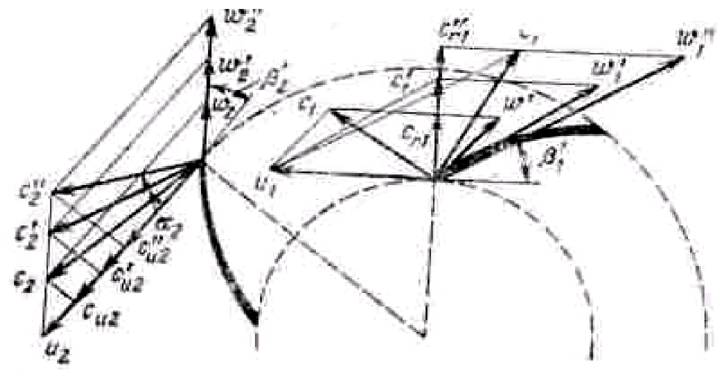


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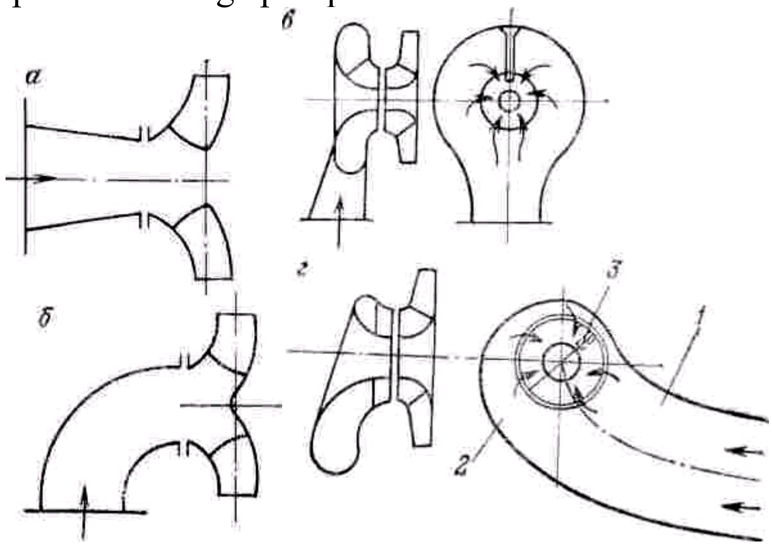
Fig. 3.4. Plans of the impeller speeds at different turbomachine capacities

The supply devices (Fig. 3.5) are made in the form of confusers with straight (a) and curved (b) axes, annular chambers (c), spiral supply chambers (d) and return channels of blade taps of multistage pumps.



3.5. Turbomachine feeders: confusers with straight (a) and curved (b) axes, annular (c) and spiral (d) feeders

In a straight-axis confuser, the fluid velocity increases by 15-20%. Accelerated motion ensures the alignment of the flow velocity field before its entry into the impeller.

The spiral supply consists of an inlet pipe 1 and a spiral channel 2 ending with an edge 3, which separates the flows entering the wheel directly from the branch pipe and from the spiral channel. The spiral supply, unlike the annular one, avoids the formation of a vortex zone behind the shaft and helps to equalize the flow velocity field.

The diverting devices must ensure that the fluid flow is diverted from the impeller with the least losses, if possible without violating the axisymmetry of the flow in the wheel and with a decrease in the flow rate to the speed in the initial section of the external network.

Four types of taps are used in turbomachines: annular, spiral, shovel and channel.

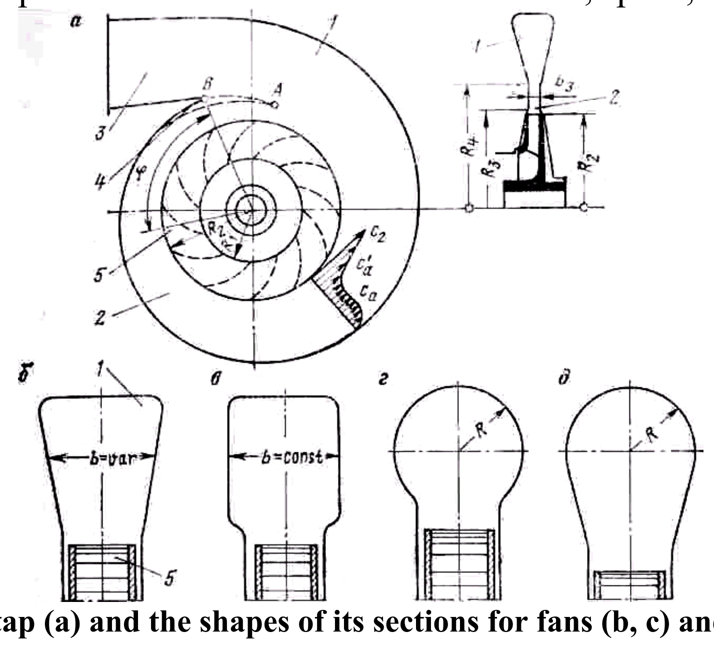


Fig. 3.6. Spiral tap (a) and the shapes of its sections for fans (b, c) and pumps (d, e): 1 — spiral channel; 2 — annular chamber; 3 — diffuser; 4 — yavyk; 5 — impeller

The annular tap is a cylindrical annular chamber of constant width covering the impeller of the machine. The spiral tap has a spiral channel 1 surrounding the impeller and is usually combined, as shown in Figure 3.6, with an annular chamber 2, bounded by radii R3 and R4 - The section of the spiral channel in fans is usually trapezoidal, with a width b and rectangular increasing to the periphery, in pumps — cylindrical and pear-shaped with radii R.

The turbomachine operation is significantly influenced by the shape and location of the tongue, which determines the amount of liquid circulating in the spiral chamber. Depending on the length of the tongue, liquid enters the chamber, starting from point A or B. The tongue of the correct shape helps to equalize the flow in the diffuser; its close location to the impeller causes a sharp increase in noise.

Blade and channel taps are used in multistage turbomachines and will be discussed below.

Hydraulic losses in turbomachines are caused by: friction of the real fluid against the surface of the channels of the flow part of the turbomachine (in the inlet, impeller and outlet); changes in the magnitude and direction of flow velocities in the channels, in particular at the entrance to the impeller; internal friction in the fluid due to vortex formation during fluid movement in the impeller.

LECTURE №. 4

ENERGY LOSS IN THE ACTUAL CHARACTERISTICS OF TURBOMACHINES

Plan:

1. Characteristics of external networks.
2. Operating modes of turbomachines.
3. Joint operation of several pumps (fans) on a common network.

1 Characteristics of external networks. The hydraulic machine is connected to an external network: pumps and compressors — with a pipeline system, a fan — with a mining system. To move water or air in an external network in a hydraulic machine, a certain increment of specific energy is created, which is necessary to lift the liquid from the mine (mine), overcome static back pressure in the network and hydraulic resistances, as well as to communicate a certain velocity to the fluid flow from the outlet section of the pipeline.

The characteristic of the external network is understood as the relationship between the pressure (pressure) of the Hc, and the amount of flowing liquid Qc—the flow rate in the network. In general, the pressure Ns (m) required to move a certain amount of fluid in the external network, in accordance with the Bernoulli equation, can be represented in the case of zero fluid velocity on the suction side in the following form:

$$H_c = z_r + \frac{p_{a2} - p_{a1}}{gp} + c^2/2g + \Delta H, \tag{4.1}$$

where zr is the lifting height of the liquid from the shaft, m;

p_{a2}-p_{a1} back pressure p_{a2} – p_{a1} pressure, respectively, on the surface and horizon of the shaft, Pa;

p is the density of the liquid, kg/m3;

c^2 – high-speed pressure at the outlet of the network, m;

ΔH is the pressure expended to overcome resistances during fluid movement in the external network (pressure loss).

Mine drainage network. The pump must create a pressure H sufficient to lift the water to the required height, overcome atmospheric back pressure and harmful resistances in the pipeline, and communicate the kinetic energy of the liquid.

The height to which the liquid rises is called the geodesic height z_r (Fig. 4.1, a):

$$z_r = z_в + z_н, \tag{4.2}$$

where zb and zn are the suction and discharge heights, respectively, m.

Static backpressure has a negative sign, since p2 is slightly less than p1. Since the absolute value of this backpressure is insignificant, it is neglected.

The pressure loss in the pipeline is known to be proportional to the square of the velocity or flow rate:

$$\Delta H = \xi \frac{c^2}{2g}, \tag{4.3}$$

where ξ is the proportionality coefficient, taking into account the configuration, dimensions, roughness and material of pipelines

The high-speed pressure c2/2g at the outlet of the pipeline can be considered as a loss of pressure to overcome the resistance at the outlet with a coefficient of $\xi = 1$.

Taking into account expressions (4.2) and (4.3), we obtain the following equation for determining the pressure:

$$H = z_r + (\xi + 1)\frac{c^2}{2g} \qquad (4.4)$$

The velocity c of the liquid in the pipeline section F is related to its flow rate by the expression Qc $c = \frac{Q_c}{F}$ replacing the velocity with the flow rate in expression (4.4), we get

$$H_c = z_r + \frac{(\xi + 1)}{2F^2 g} Q_c^2$$

Denoting z_r = H и $\xi_T = \frac{\xi + 1}{2F^2 g}$, when ξ_T – the resistance coefficient of the pipeline, c2/m, we obtain the following expression of the external characteristics of the drainage pipeline:

$$H_c = H_r + \xi_T \qquad (4.5)$$

In the practice of calculating the characteristics of external networks, local pipeline resistances are usually taken into account, taking instead of the actual equivalent length of the pipeline, i.e. such a length of the pipeline is added to the actual length, the pressure losses in which are equal to local losses.In particular, it can be assumed that pressure losses in the suction valve are equivalent to losses in a straight pipeline with a length of 10 m; pressure losses in knees, tees, valves and check valves (in each element) are equivalent to losses in a pipeline with a length of 5 m.

Shaft ventilation network. The mine ventilation network is a network of underground mine workings. Natural traction can create some positive or negative back pressure in the ventilation network.

The value of the natural thrust of the rt is determined by the expression

$$p_T = gz_r\,(\rho_a - \rho), \qquad (4.7)$$

the density of air, respectively, atmospheric and outgoing from the mine, kg / m3.

Fan installations operating on injection create pressure in the workings exceeding atmospheric pressure. Injection ventilation is allowed only for non—gas mines and only as an exception - for shallow mines of the first gas category.

With suction ventilation, air enters through the main trunk and then enters the mine wing workings through the trunk yard, washes the treatment faces and from there through the ventilation drifts goes to the ventilation shaft and then goes into the atmosphere through the fan installation. The lowest pressure of the rc takes place at the entrance to the fan.

The pressure difference created by the fan must overcome the harmful resistance to the movement of air in the workings and the inertial properties of the air, which needs to be informed of a certain kinetic energy. Thus, the equation for the mine pipeline is also applicable for the mine ventilation network, with the only difference that the value of the geodetic height is assumed to be zero.

$$p_c = \xi_c Q_c^2 \tag{4.8}$$

where ξc is a coefficient depending on the roughness of the walls of the workings, their length, cross—section and shape (N ·s2 · m-8).

The characteristic is a parabola with a vertex at the origin.

The pressure generated by the shaft fan installation is spent mainly on overcoming the resistance to air movement in underground workings.

In the practice of calculations and research, the concept of an "equivalent opening" is used to assess the resistance of ventilation networks.

An equivalent hole is such a conditional round hole in a thin wall, the air flow through which, at a pressure difference equal to the mine depression, is the same as in the mine ventilation network. In other words, an equivalent hole of a mine (mine) means a conditional hole whose resistance is equal to the resistance of the entire mine (mine).

It is known from hydraulics that the flow rate of liquid Q (m3 /s) through a thin hole

$$Q = \varphi A\sqrt{2gH}, \tag{4.9}$$

where ψ = 0.62÷0.65 is the narrowing coefficient of the jet; A is the area of the equivalent hole, m2; H is the pressure difference, m.

Taking the depression pc = drN, as well as the coefficient ψ = 0.65 and the air density p = 1.2 kg /m2, we obtain the following expressions for the ventilation network:

$$Q_c = 0{,}839A\sqrt{p_c}$$

$$A = 1{,}19\frac{Q_c}{\sqrt{p}}; \tag{4.10}$$

$$p_c = 1{,}42\left(\frac{Q_c}{A}\right)^2; \tag{4.11}$$

The last equation is an analytical expression of the characteristics of the ventilation shaft network through an equivalent hole.

2. Operating modes of turbomachines. Operating mode. To obtain data on the operating mode of a turbomachine connected to the network, it is necessary to compare the characteristics of the external network and the individual turbomachine.

It is obvious that in the steady-state operating mode, the flow of liquid through the network should be equal to the performance (supply) of the machine and it should develop the pressure necessary to overcome the network resistances, i.e. in the steady-state operating mode, the flow and pressure values should simultaneously satisfy both the characteristics of the network and the characteristics of the turbomachine.

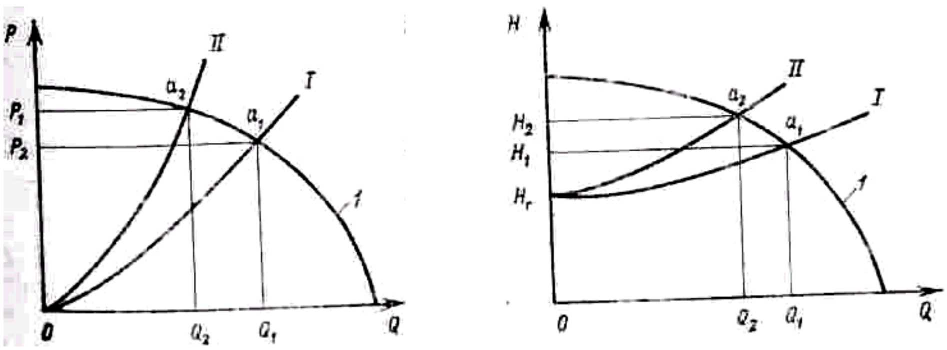


Fig. 4.1. Determination of the operating mode of the fan (a) and pump (b)

Stability of work and surge. The stability of the turbomachine is an important condition for its normal operation. The operation of a turbomachine is stable, in which, after eliminating the causes that caused the change in the operating mode, it is automatically restored. A change in the operating modes of pumps and fans occurs when the resistance of the network or the rotational speed of the turbomachine changes, which throws the machine—network system out of balance.

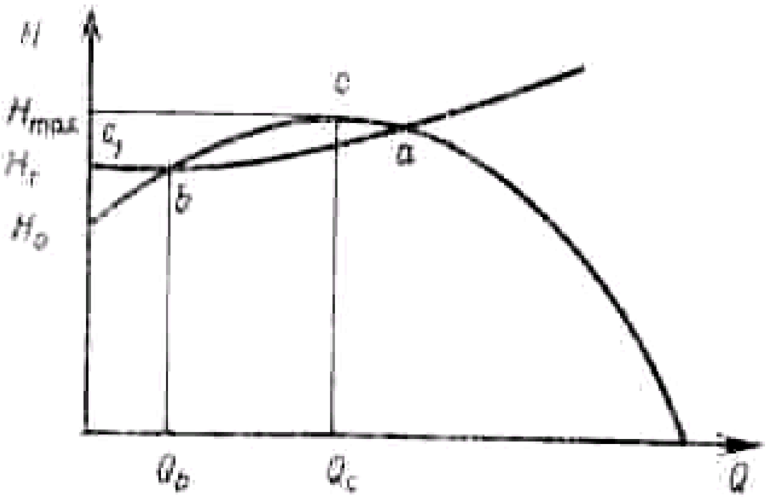


Figure 4.2. Characteristics of the turbomachine and the external network intersecting at two points

The phenomenon of surging can occur not only in pumps, but also in other turbomachines (fans, compressors) operating on an external network. Surging occurs in the presence of an unstable section of the turbomachine characteristic. The unstable part of the characteristic is that part of it where the ascending part of the characteristic of the machine passes steeper than the characteristics of the network, i.e. the condition (4.13) is not fulfilled.

Shaft centrifugal fans have smoothly changing characteristics, work on the mine network without geodetic height and always have stable modes.

Shaft axial fans, although they also work for a network without geodetic height, but may have unstable operating modes in the presence of depressions and gaps in their pressure characteristics.

Working areas of characteristics. During the operation of the mines, there is a continuous movement of the treatment faces and a change in the length of the mine workings. As a result, the characteristics of mine networks and the operating modes of turbomachines change.

The efficiency and stability of their operation are important operating conditions for stationary installations. The criterion for quantifying efficiency is the specific consumption of electricity, i.e. the amount of energy spent on moving a unit of liquid. To a large extent, the efficiency is determined by the efficiency of the turbomachine. Obviously, the rational area of use of turbomachines will be the area in which the efficiency has values equal to or greater than the minimum allowable. Minimum allowable efficiency it depends on the level of development of turbomachines and is currently 0.6 for main ventilation shaft fans.

Sections of individual operational characteristics of turbomachines that meet the requirements of efficiency and stability of modes are called working sections x a r a k t e r i s t and K.

3. Joint operation of several pumps (fans) on a common network

General information. The joint operation of several turbomachines on one common network is resorted to at mining enterprises in cases when the pressure or flow rate generated by one turbomachine at the limiting parameters (rotational speed, blade angle, etc.) is insufficient. All cases of collaboration can be reduced to two options — parallel and sequential operation of machines. For example, if one pump does not provide pumping of the daily inflow of water, then two pumps are included in parallel operation. If the pressure generated by one pump is insufficient, then sequential operation of the pumps is provided. In parallel operation, the fluid from both turbomachines enters the common network. In sequential operation, the liquid supplied by one machine passes through another and receives additional energy in it. Multi-stage turbomachines are also structurally combined single-stage machines operating in series or in parallel.

Consistent work. Sequential operation of turbomachines is used to increase the head (pressure) in the external network.

At the same time, the feed (performance) of the machines is the same ($Q_1=Q_2=Q$), a the head (pressure) is equal to the sum of the heads (pressures) of both machines ($H=H_1+H_2$).

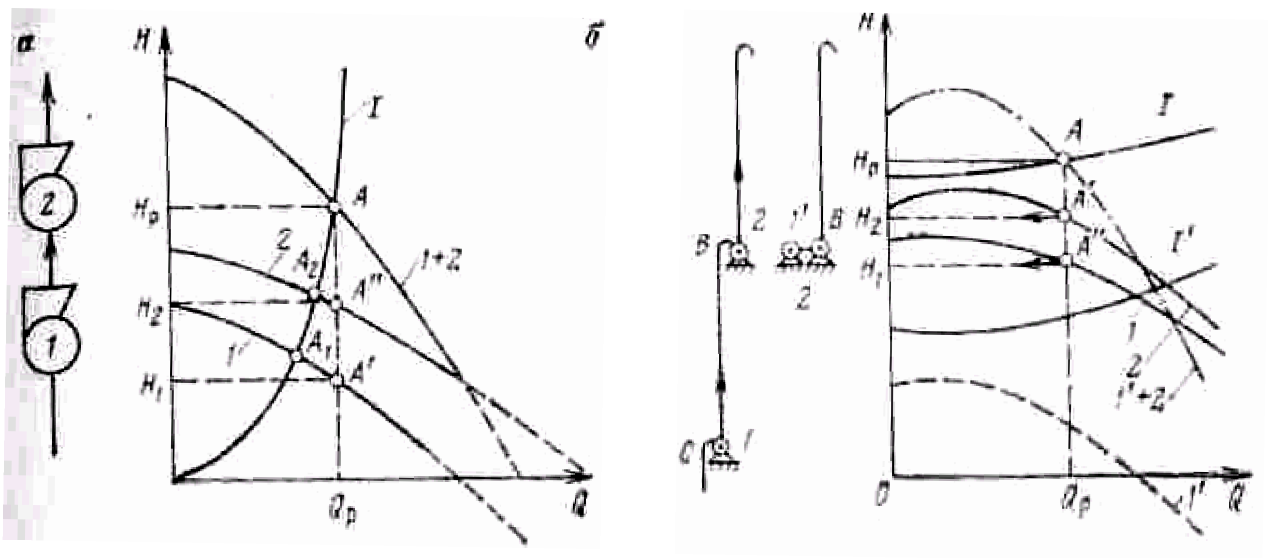


Figure 4.3. Sequential operation of turbomachines when they are located in one place (a) and at a distance (b)

The machines are installed side by side. To obtain the total pressure characteristics of 1+2 turbomachines, it is necessary to add up the ordinates of characteristics 1 and 2 (Fig. 4.3). The intersection point A of the total characteristic 1 + 2 with the characteristic of the network will indicate the operating mode of jointly operating machines, and points A' and A" — the operating modes of individual turbomachines. It should be noted that the parameters of the operating modes (points A' and A") when machines work together differ significantly from the parameters of the modes when they work separately on the same external network (points A1 and A2).

The cars are located at a distance from each other. In order to obtain the total pressure characteristic of the turbomachines, it is first necessary theoretically to replace the turbomachine 1 located at point C with the adjacent pipeline CB with an equivalent machine 1’ located at point B next to the turbomachine 2, i.e. to bring the turbomachines to a common point B. To do this, it is necessary to build a pressure characteristic 1’ by subtracting from the characteristics of the turbomachine 1 from the same 0 ordinates of the characteristics /' of the CB section, i.e. the pressure consumed for lifting the liquid and overcoming the resistance of the pipeline. The total characteristic is obtained by adding the ordinates of the characteristics G and 2. Point A of the intersection of the total characteristic of turbomachines with the characteristic of the network / determines the operating mode of sequentially operating turbomachines, and points A' and A" are the operating modes of individual turbomachines 1 and 2.

To ensure the efficient operation of sequentially connected turbomachines, it is necessary that their optimal performance is approximately the same.

When installing machines at a distance, this requirement applies to the above and real 2 machines.

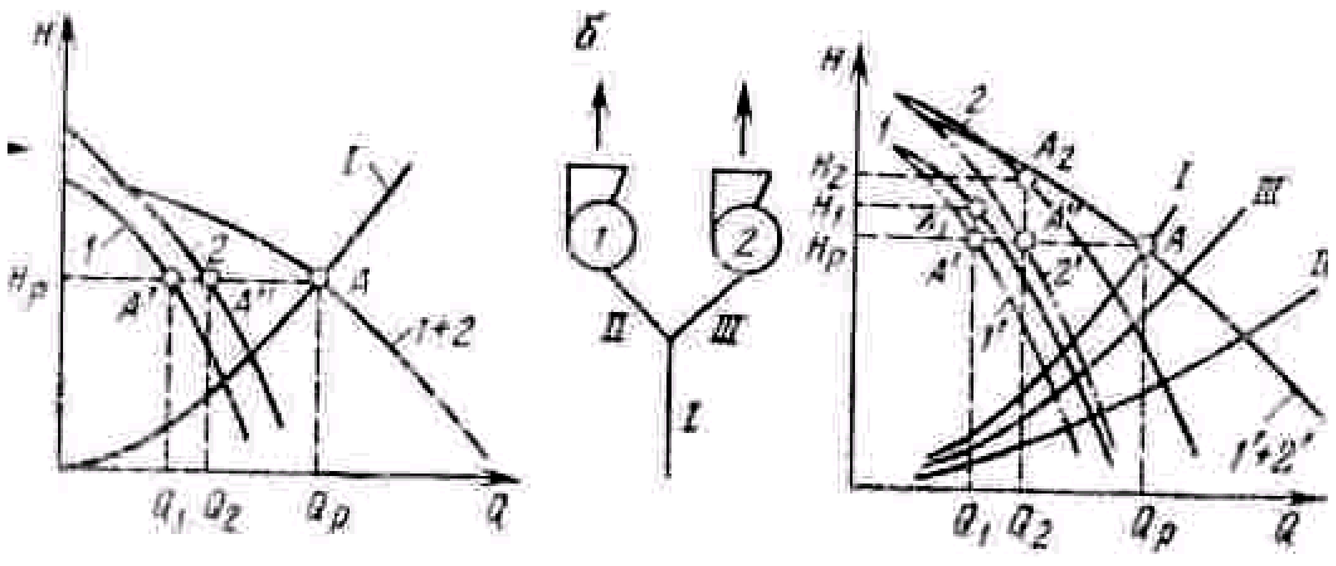


Figure 4.4. Parallel operation of turbomachines when they are located in one place (a) and at a distance (b)

Efficiency. is consistently determined from the expression of the pumps included

$$\eta = \frac{g\rho Q(H_1 + H_2)}{1000(N_1 + N_2)} \text{или} \, \eta = \frac{H_1 + H_2}{H_1/\eta_1 + H_2/\eta_2}, \qquad (4.14)$$

where n1 and n2 are the efficiency of machines 1 and 2; N1 and N2 are the power consumed by machines / and 2, kW.

Similarly, the efficiency of fans is determined:

$$\eta = \frac{p_1 + p_2}{p_1/\eta_1 + p_2/\eta_2}, \qquad (4.15)$$

where p1 and p2 are the pressure developed by fans 1 and 2.

The most economical operation of sequentially connected machines will be if each machine, at the required total head, operates in the maximum efficiency mode.

When several turbomachines work together, their total characteristics are constructed in the same way as for two machines.

LECTURE No. 5
GENERAL INFORMATION ABOUT DRAINAGE INSTALLATIONS AND DRAINAGE

Plan:

1. Underground water flows.

2. Purpose and classification of drainage installations.

3. Technological schemes of stationary drainage.

4. Pumping chambers and water collectors.

1. Underground water flows. The inflow of groundwater varies widely at various mining enterprises and reaches 20,000 m3/h, and at very watered mines exceeds this value. At the same enterprise, the water supply does not remain constant throughout the year. In this regard, there are normal and maximum water flows. The latter are noted in the spring or autumn period. The inflow of water during these periods in mines up to 100 m deep increases 2-4 times or more.

The water content of a mine or mine can be expressed in absolute or relative values. Absolute water availability (water inflow) expresses the total amount of water (m3) entering the mine per unit of time. Relative water availability is estimated by the coefficient of water availability, i.e. the ratio of the mass of the annual inflow of water to the mass of the mineral extracted during the same period. The coefficient of water availability for coal mines ranges from 0.38 (Karaganda basin) to 25 (Moscow region basin) and more.

Mine waters contain chemicals and mechanical impurities that adversely affect the equipment of drainage installations.

2. Purpose and classification of drainage installations. A drainage system is a complex of technical means for removing water from mine workings and releasing it to the surface. Drainage installations are used for pumping groundwater, which, depending on the purpose, are divided into central, main, precinct, auxiliary, pumping, tunneling and borehole.

The central drainage system is designed to pump water from several mines; the main drainage system is designed to deliver directly to the surface of the inflow of water of the entire mine.

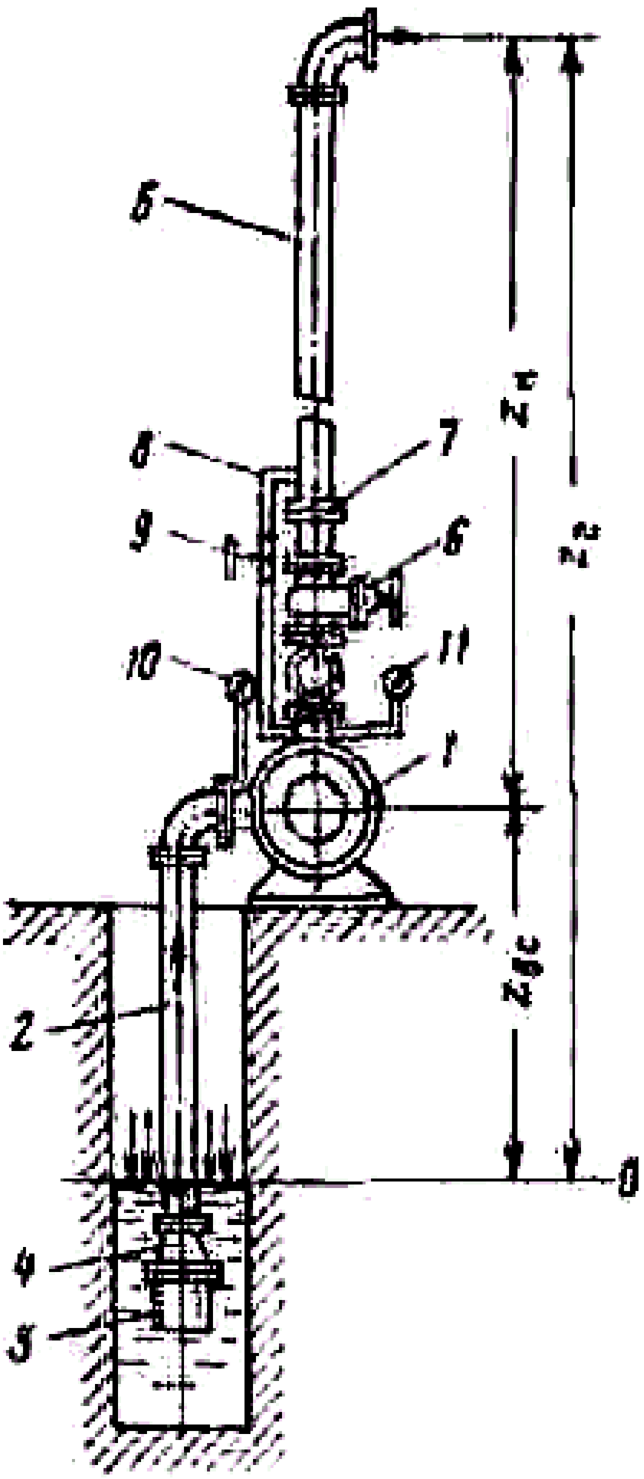
Precinct drainage installations pump water from the sites to the main catchment area or to the surface from the workings of any section of the mine or mine.

Auxiliary drainage installations are located on sections, slopes, sumpfs and serve for pumping water into the catchment of the main or central drainage installation. When the soil of the formation is undulating, pumping units are used to pump water from the plots into the catchment area of the main drainage system.

Drainage installations used in the sinking of slopes, inclined and vertical shafts of mines are called sinking. They move as the face moves or the water level drops. Installations for lowering groundwater are called borehole.

Central, main, auxiliary and district drainage installations, as a rule, are placed in special chambers and are stationary.

Mobile drainage installations are used during mining operations.



Drainage installations (Fig. 5.1) are equipped mainly with centrifugal pumps. The installation consists of a pump 1 with a motor, a suction pipeline 2 with a receiving grid 3 and a valve 4, a discharge pipeline 5 with a gate valve 6 and a check valve 7, a tube 8 with a valve 9 for filling the pump with water before starting it. The pressure in the suction 2 and discharge 5 pipelines is measured by a vacuum gauge 10 and a pressure gauge.

Figure 5.1. The scheme of the drainage system

The vertical distance from the water level in the intake tank (well) to the pump axis is called the geodesic (geometric) suction height zc, and the vertical distance from the pump axis to the drain opening of the pipeline is the geodesic (geometric) discharge height zn. The sum of the geodesic heights of suction and discharge is the geodesic (geometric) height of the supply zg, which, in essence, is the full geodesic height of the water lift.

3. Technological schemes of stationary drainage. Technological schemes of drainage are determined by the depth and number of horizons being developed, the method and order of mining deposits.

When developing a single horizon, the most acceptable is a stepless drainage scheme, when water is collected in the main drainage basin and pumps directly pump it to the surface. If the pressure of one pump is not enough, then use sequential operation of pumps (Fig. 5.2, b) installed in one chamber, or use step schemes (Fig. 5.2, c, d) with the placement of pumps on different horizons. In this case, the pumps can be switched on without and with an intermediate water collector.

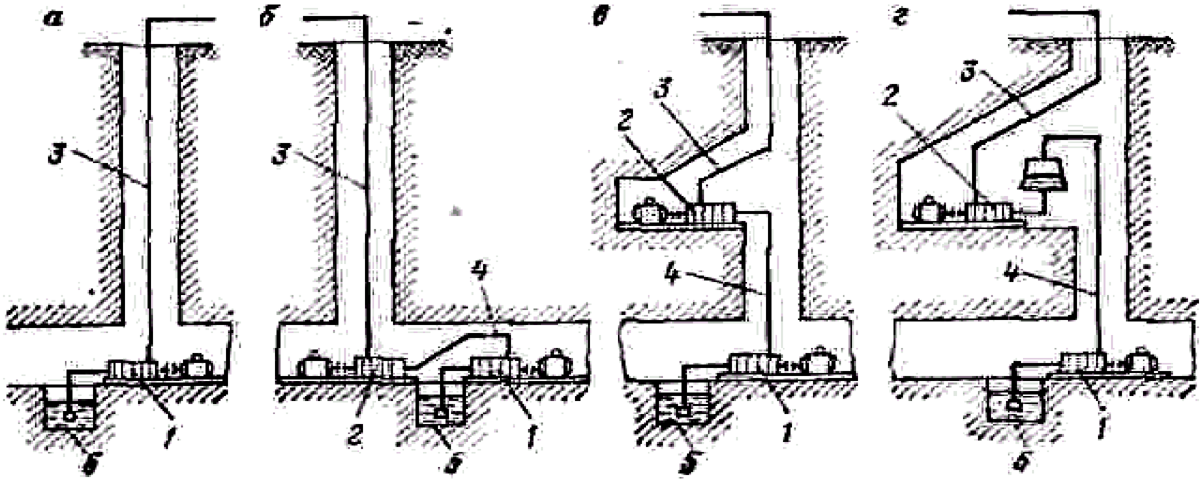


Fig. 5.2. Drainage scheme for the development of one horizon:
1 and 2 — pumps; 3 and 4 — pipelines; 5 — water collector

The simplest is the stepless drainage scheme, in which one drainage system is sufficient. When using this scheme, the volume of mining operations required for drainage is reduced, electromechanical equipment is simplified and labor safety is reduced, but pressure increases, capital costs for the manufacture of pumps and high-pressure fittings increase, engine power increases. Nevertheless, this scheme is preferable, and they strive to increase the possible depth of step-by-step drainage by creating high-pressure pumps.

A step-by-step scheme with an intermediate catchment allows for lower pressures, which increases the safety and reliability of work, but this scheme requires the construction and maintenance of an intermediate catchment, the cost of which is significant.

When developing two or more horizons with independent tributaries, stepless drainage is possible separately from each horizon (Fig. 10.3, a) or with pumping (Fig. 5.3, b, c, d).

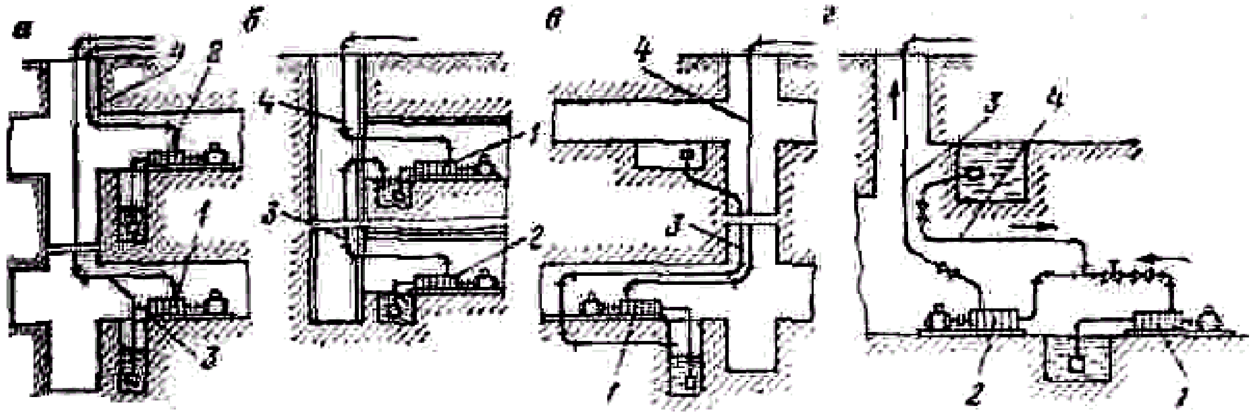


Figure 5.3. Drainage scheme for the development of two horizons:
1 and 2 — pumps; 3 and 4 — pipelines

The choice of the latter schemes is determined by water flows on individual horizons. For example, if the water flows of the upper horizons are large, then the scheme shown in Figure 5.3 is rational, b. If the water flows of the lower horizons are large, then it is better to use a scheme with a bypass of water from the upper horizons to the lower, in which it is not required to install). Powerful pumping stations on different horizons. The water pressure from the upper horizon can be partially used (Fig. 5.3, d) if water from this horizon is piped directly to the suction path of the pumps. When choosing a drainage scheme, it is also necessary to take into account the acidity of waters at different horizons and other factors. Acidic water is recommended to be pumped out separately.

In the case of a significant removal of a mine or mine site from the main drainage system and a small depth of the developed horizon, water is pumped to the surface through special wells or pits from the site installations.

The choice of the technological scheme of drainage is carried out on the basis of a technical and economic comparison of options, depending on the local mining and geological conditions of the mining enterprise, the sequence of mining horizons and other factors.

4. Pumping chambers and water collectors. Pumps and equipment for automation of drainage installations are usually placed in special mine workings — pumping chambers. The pumping chamber 1 is connected by an inclined passage 8 to the trunk yard and a pipe-cable passage 4 — to the trunk 5, and with the help of pipes with valves — to the water collector (Fig.. 5.4).

Usually the pumping chamber is located on a fresh jet in the yard and adjacent to the underground substation. The dimensions of the pumping chamber are determined by the number and dimensions of the pumping units used, which, according to safety rules, must be at least three (one in operation, one in reserve and one in repair). The device of the pumping chamber must ensure the safe operation of the equipment, its convenient delivery and installation, as well as free access during the maintenance of pumping units. To prevent flooding of the pumping chamber, its floor should be positioned 0.5 m above the level of the trunk yard.

To accommodate the suction pipelines, the pumping chamber is equipped with group or individual wells 8 connected to the water collector 2 through a receiving collector or directly.

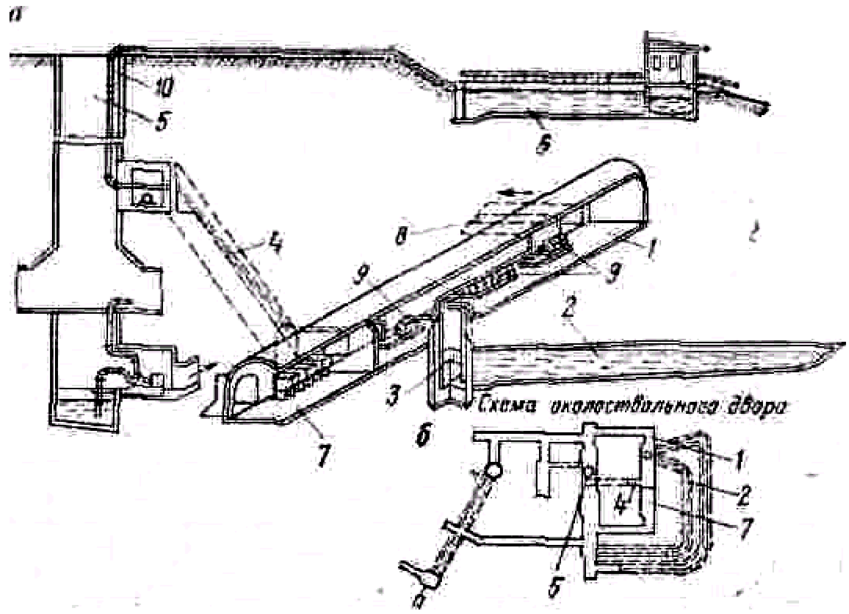


Fig. 5.4. Technological scheme of the main drainage (a) and mining plan (b):

1 — pumping chamber; 2 — water collector; 3 — suction well; 4 — pipe walker; 5 — trunk; 6 — mine water sump; 7 — underground electrical substation; 8 — walker in the yard; 9 — pumps; 10 — pipeline

Pumping chambers can be located above and below the water level in the catchment area. In the first case, the pumps have a positive, and in the second a negative suction height, i.e. they work with a backup, which is a favorable circumstance, since it eliminates the appearance of cavitation in the pumps and the need to fill them. However, the disadvantages of chambers located below the water level in the catchment (submersible) are the difficulty of ensuring their tightness (water from the catchment enters the chamber) and large capital costs.

Submersible chambers are not widely used in coal mines due to the fracturing of rocks and the difficulty of sealing the chambers, but are used in many mines.

Water collectors are workings for collecting water and clarifying it. The catchment area of the main or precinct drainage should consist of two or more workings. In mines that are dangerous for water breakthrough, according to the rules of technical operation, the capacity of the catchment basins of the main and precinct installations should be calculated for an eight- and four-hour inflow of water, respectively, for the remaining mines, the catchment basins are calculated for a four- and two-hour normal water flow, respectively.

The water, passing at a low speed through the catchment area, is clarified. Water collectors are sometimes divided into two parts (in one part the water is clarified) or a special preliminary sump is made to clarify the water in the horizontal development.

From the water collector, water enters the water intake wells, from where it is pumped to the surface through suction pipelines.

Due to the settling of solid particles and the clarification of water, the catchments are silted up. They should be cleaned at least 2 times a year. Contamination of the catchment area by more than 30% is not allowed.

A progressive direction is the use of tubeless drainage, which does not require a special pumping chamber and greatly simplifies the drainage of mines. For tubeless drainage, submersible pumps are used, for example, of the ECV type, installed in the pipe compartments of the trunks and below the water level in the catchments.

A promising direction is also the installation of submersible pumps directly in the water collectors, which significantly reduces the size of the pumping chambers.

LECTURE No. 6
FORCES ACTING ON THE PUMP IMPELLER AND THEIR BALANCING
Plan:

1. Forces acting on the pump impeller.

2. Pump characteristics and operating modes

3. Methods of regulation of operating modes.

1. Forces acting on the pump impeller. Axial and radial forces act on the pump rotor. These forces arise during the operation of all turbomachines, but they are significantly less in fans than in pumps.

The axial force appears during the operation of the pump due to the inequality of pressures on the front and rear outer walls of the impeller (Fig. 6.1) In the spaces A and B between the outer walls of the impeller and the inner walls of the housing, the liquid coming out of the impeller gets under pressure p2. The pressure p2, depending on the gap, is equal to or less than the fluid pressure at the outlet of the impeller. Regardless of the nature of the pressure diagram, approximately the same and oppositely directed forces (mutually balanced) act on the areas enclosed between the circles of the radii R2 and Ru (the radius of the front wheel seal) and the areas of the outer walls of the impeller.

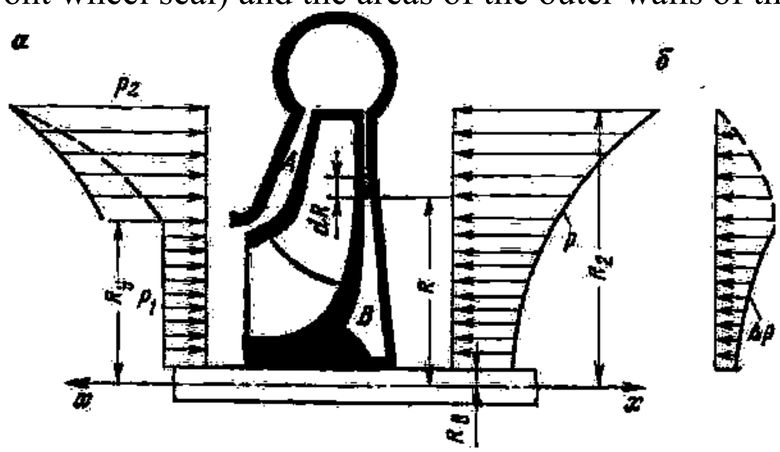


Figure 6.1 Pressure diagrams on the impeller (a) and overpressure diagram (b)

The area of the rear (root) disk of the impeller is not hydraulically balanced, bounded by the circles of radii Ru and Rv (radius of the wheel shaft) and equal to ( ). Pressure acts on this area of the disk from its outer side and on the inner pressure at the inlet of the impeller is determined by the suction conditions: if suction is carried out in the presence of a positive suction height (z > 0) and when working with a backup, i.e. with a negative suction height. However, in both cases there is less fluid pressure at the outlet of the impeller. Therefore, the wheel will be under a pressure difference:

$$\Delta p = p - p_1 \qquad (6.1)$$

with the plot shown in Fig. 11.1, b. As a result, the axial force of the To acts on the impeller, directed towards its suction hole:

$$T_0 = \int_{R_B}^{R_y} (p - p_1) 2\pi R dR \qquad (6.2)$$

where p is the current pressure, the maximum value of which is Pa.

To determine the strength of the Tu, it is necessary to establish the dependence p = = f(R), for this we consider an elementary ring of liquid with a width of dR. The liquid located in space B, due to friction against the walls of the wheel, is driven into rotational motion with a certain angular velocity, depending on the gaps between the walls, the roughness of the latter and the viscosity of the liquid. It is assumed that the fluid in the spaces A and B comes into rotation with an angular velocity = 0.5 angular velocity of the impeller . When the liquid rotates due to centrifugal forces, back pressure occurs.

For an elementary volume of liquid (ring dR), the pressure increment is due to centrifugal forces:

$$dp = \rho (\frac{\omega}{2})^2 R dR \qquad (6.3)$$

Let 's integrate this expression:

$$\int_{p}^{p_2} dp = \int_{R}^{R_2} \rho (\frac{\omega}{4})^2 R dR$$

We will get

$$p_2 - p = \frac{\rho \omega^2}{8} (R_2^2 - R^2)$$

Where from

$$p = p_2 - \frac{\rho \omega^2}{8} (R_2^2 - R^2) \qquad (10.15)$$

$$p = p_2 - \frac{\rho \omega^2 R_2^2}{8} (1 - \frac{R^2}{R_2^2}) \qquad (6.4)$$

It can be seen from the resulting equation that the dependence p = f (R) has a parabolic character (see Fig. 6.1, a).

Substituting the expression for p from (11.4) into equation'(6.3), after integration we obtain the value of the axial force

$$T_0 = [p_2 - p_1 - \frac{\rho \omega^2}{8} (R_2^2 - \frac{R_y^2 + R_2^B}{2})] \pi (R_y^2 - R_B^2) \qquad (6.5)$$

An oppositely directed force acts on the inner surface of the wheel, caused by a change in the amount of flow movement,

$$T' = -m(c_1 - c_2) = \rho Q (c_1 - c_2)_0 \qquad (6.6)$$

Since c 2 = 0 for radial centrifugal pumps, it is practically accepted for centrifugal pumps

$$T_0' = -\varphi \rho Q c_1$$

where is the experimental coefficient depending on ns.
For small ns values, the coefficient =1.0.

The resulting axial force T acting on the impeller,

$$T = T_0 + T_0' \qquad (6.7)$$

In magnitude, the axial force of the is significantly less than That. In practice , an approximate equation is often used to determine the axial force

$$T = r_n \pi (R_y^2 - R_B^2) \rho H \quad (6.10)$$

Where is the experimental coefficient, depending on ns, for a wheel with a one—way fluid supply and a through shaft rn = 0.6 at ns = 60 and rn = 0.8 at ns = 200; H is the wheel head, m.

For multistage centrifugal pumps, the total axial force

$$T_{\sum} = zT \qquad (6.11)$$

where z is the number of steps.

The value reaches 0.15—0.20 MN.

The axial force increases significantly with increasing leaks through the seal between the housing and the front wheel disc. In case of failure of the seal, the pressure plot on the front disc is limited by the dotted curve (see Fig. 6.1, a). The axial force in this case can increase up to three times the value.

Methods of balancing the axial force. Due to the high value of the axial force, its balancing by the usual mechanical method — the use of thrust bearings is not rational, since the design of the pump becomes heavier and more complicated and the mechanical efficiency decreases. In pumps, hydraulic methods of balancing the axial force are resorted to. The following five main methods of full or partial axial force equalization are used.

1. The use of impellers with two-way suction. With this method, in the ideal case (symmetrical fluid supply to the impeller and the same sealing quality), the axial force is zero.

2.Arrangement of impellers in pairs or groups on the shaft of a multistage pump, in which the axial forces are mutually balanced. In this case, the fluid bypass sequence is selected in such a way that the resulting axial force is close to zero (Fig. 6.2). With this method, large bypass channels between the wheels are required, and complete balancing is not achieved and thrust bearings have to be used. The design of the pump becomes more complicated.

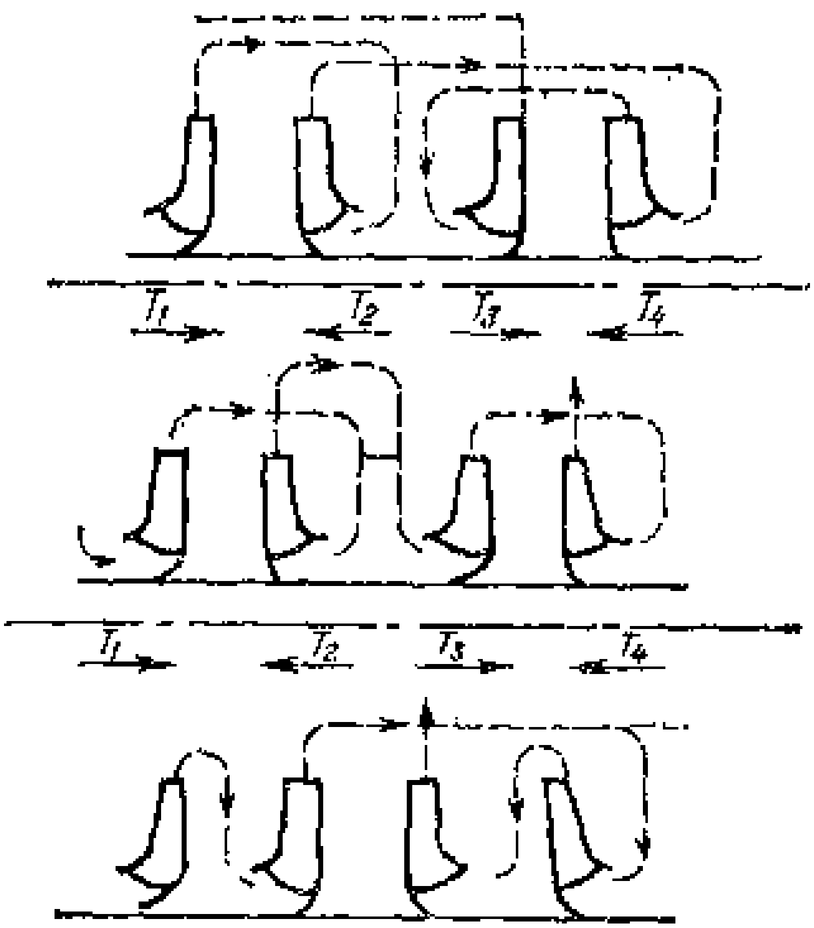


Рис. 11.2. Примеры схем движения

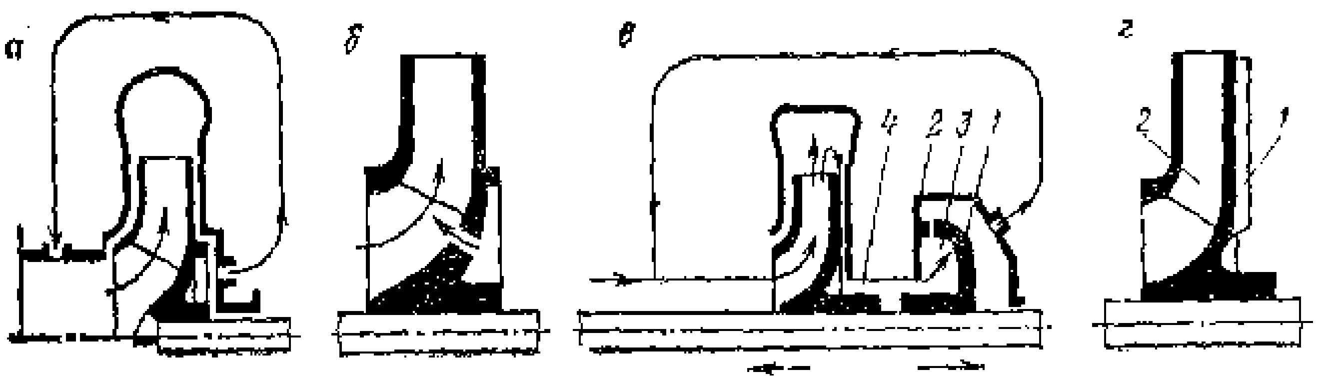


Fig. 11.3 Schemes of balancing the axial force of the impellers by equalizing the pressures on the outer surfaces of the discs (a and b), using the unloading disc (b) and unloading blades on the rear wheel disc (d) 4.

3. The use of an additional ring seal on the rear disc of the wheel, which divides the surface of the disc into two parts. A pressure approximately equal to the pressure in the suction channel acts on the central part. This is achieved by combining the space of the central part with the suction channel by a bypass tube (Figure 6.3, a) or a hole in the rear disc (Figure 6.3, b). By appropriate selection of the diameter of the sealed part of the rear disc and the size of the bypass holes, it is possible to achieve a complete absence of axial force. However, at the same time, internal fluid flows in the machine increase and its volumetric efficiency decreases.

The use of an unloading disk (Fig. 6.3, c). A disk rigidly mounted on the drive shaft with the end surface of the housing 2 forms a sealed chamber 3, into which water from the last stage of the pump enters through a slot 4. Water exerts pressure on the disk in the opposite direction of the axial force. The pressure in front of the disc is greater than the pressure behind the disc by the amount of resistance of the radial slit . Then an axial force acts on the unloading disc.

$$T_D = \psi F_D (p_3 - p_4) = \varphi\pi(R_D^2 - R_{вт}^2)\Delta p_D \qquad (6.12)$$

where is the coefficient that takes into account the uneven distribution of pressure over the surface of the disk;

Fd — disk area,

Rd and Rvt are the radii of the disk and its sleeve, respectively.

The Td force of the fluid pressure on the disc in all pump operating modes is equal to the resulting axial force $T_{\sum} = (T_Д = T_{\sum})$. This equality is achieved automatically due to some movement of the pump rotor along the axis and a change in the size of the gap between the disk and the housing 2. If the fluid pressure forces on the discharge disc exceed the axial force, the shaft is displaced, the gap in the O-ring seal between the disc and the housing increases, as a result, due to increased fluid leaks, the pressure in chamber 3 decreases and the force acting on the discharge disc decreases. The water flow through the discharge opening slot should not exceed 3%, since the flow rate reduces the volumetric efficiency of the pump. However, this method of unloading bearings is widely used in multistage pumps, as it provides minimal dimensions in the axial direction.

5. The use of unloading blades on the rear disk of the impeller (Fig. 6.6, d). The unloading blades on the outside of the rear disk have a small thickness, and their geometry is similar to the profile of the working blades 2 of the impeller. By selecting the dimensions of the unloading blades, it is possible to achieve the necessary vacuum on the rear disk and complete compensation of the axial force. However, some of the energy supplied to the pump shaft is lost.

The radial force occurs when the operating mode of the pump with a spiral casing deviates from the calculated one or in the case of an incorrectly configured outlet channel. In magnitude, it can be 5-10 times greater than the gravity of the rotor. When the pump is operating in non-calculated modes, the cross-sectional area of the spiral outlet does not correspond to the amount of liquid passing through it. At low flows Q, the velocity of the liquid in the spiral outlet in the direction of the outlet decreases, and the pressure increases (Fig. 6.3 a). With Q feeds exceeding the calculated Qp, the speed increases and the pressure drops. Since the pressure distribution in both cases is uneven around the circumference of the impeller, there is a radial force FR directed at an angle a to the axis of the outlet pipe.

The force FR is directed at Q < Q p at approximately an angle of a 20° and at Q > Qr at an angle of 150 °. The FR force causes the shaft to deflect, which leads to increased wear of the seals. The radial resultant force of the hydrodynamic fluid pressure on the wheel vanishes or decreases significantly if the flow is diverted from the impeller by two or more channels of the same design. In pumps with spiral channels, a decrease in FR is possible when using two spiral bends offset by 180 ° relative to each other, or when dividing the spiral channel by a partition into two channels. The radial forces FR and F1R are partially balanced. In sectional multistage pumps, the annular bends are equipped with guide vanes, through which a smooth, with small shocks and vortices, the transition of liquid from the impeller channel to the outlet is ensured. Each pair of guide vanes forms a separate spiral branch. The radial forces arising in each of these bends are mutually compensated due to their axisymmetricity.

2. Characteristics of pumps and their operating modes The main technical data characterizing the operation of the pump are: supply Q (m3/c) — the volume of liquid supplied by the pump per unit of time; pressure H (m) — the specific energy supplied by the liquid pump; pressure at zero supply H0 (m) pump pressure when closed the valve installed at the pressure nozzle of the pump; vacuum suction height H vac (m)—the suction height determined by the vacuum gauge; permissible vacuum suction height H vac, additional (m) — the height at which the pump is operated without changing its main technical parameters; power (kW) on the pump shaft — the power consumed by the pump. Individual characteristics of pumps used in mining enterprises are usually expressed by dependencies H =F (Q), H $_{\text{рак}}$ =F $_1$, (Q), N=$f_2(Q)$ и $\eta = f_3(Q)$ Figure 5.8 shows as examples the individual characteristics of centrifugal pumps CNK 300-120-600, presented graphically. For multistage sectional pumps, it is customary to give characteristics for one impeller. Universal characteristics are also sometimes used to characterize pumps used in other industries.

The characteristics of pumps and pipelines change during operation. Due to abrasive wear of the impellers and changes in the cross-section of the channels, the pump head decreases.

For normal operation of centrifugal pumps, the following conditions must be met.

To prevent unstable pump operation modes, the static back pressure Hr in the pipeline network, equal to the geodetic height zr, should be less than the pump head But at zero supply (with the valve closed), for shaft pumps Hr < 0.9 No.This is due to the fact that in practice, small fluctuations in the rotation frequency of the pump motor are possible (for example, when the frequency of the current in the power grid changes), which lead to a change in the pressure characteristic and, in particular, to a decrease in the No value. At Hr > H0, the pressure characteristic of the pump intersects with the characteristic of the pipeline at two points and unstable operation of the pump (surging) occurs. In order for the operation of the pump to be economical, its mode must be within the working area of the characteristic, limited by the values of efficiency = 0.95 max. Cavitation should not occur during operation of the pump.

3. Methods of regulation of operating modes. The operating modes of drainage systems are regulated by changing the characteristics .pipelines and pumps. The most widespread in practice is a simple, although less economical, first method of regulation, using a valve in the discharge pipeline.

With characteristics I and, respectively, the pipeline and pump, the operating mode is represented by point a, located behind the working area. To establish the nominal supply Qn and pressure Nnom regulation is carried out by a gate valve, as a result of which a characteristic of the pipeline is obtained, depicted by curve III (the operating mode is represented by a dot and). Since this increases the resistance of the pipeline, the pressure increases by the amount of Nc and additional power is consumed. For high-speed pumps, instead of throttling, it is more economically advantageous to regulate the discharge of part of the liquid through the bypass pipeline into the suction tract of the pump or discharge into the water collector.

In the second method, the characteristics of pumps can be regulated by changes in the number of impellers and the speed of rotation of the pump shaft, trimming of impellers, etc. When the number of wheels decreases, the pump characteristic is represented by curve 2, and the operating mode is represented by point E. Up to the required mode c, the adjustment is carried out by a gate valve, with the help of which the characteristic of the pipeline is set, shown by CURVE II.

Regulation of the pump operation by changing the speed of rotation minimizes losses, eliminates changes in the characteristics of the network, but is associated with the use of a more complex adjustable drive. When the rotation speed changes, congruent characteristics H =f(Q) are obtained, which can be constructed using similarity equations. However, the regulation by changing the rotation speed of the pump shaft is not widespread, since the pressure H of the pump changes, and the height of the water rise in mining enterprises is in most cases constant.

Changing the individual characteristics of multi-section pumps by trimming the impellers, i.e. reducing their outer diameter D2 to D'2, has been widely used in mining enterprises, but is currently rarely used.

Reducing the diameter of the wheel gives a new triangle of speeds at the output, similar to the original, but with a portable speed u2 <u2. In the new triangle, the directions of absolute and relative velocities are preserved (approximately), and the values of these velocities and the circumferential velocity decrease

according to the ratio of diameters D'2/D2. The parameters of pumps Q' ,H' and N' with clipped impellers can be determined from the expressions:

$$Q' = Q(\frac{D'_2}{D_2})^3; \mathrm{H}' = \mathrm{H}(\frac{D'_2}{D_2})^2; \mathrm{N} = N(\frac{D'_2}{D_2})^5; \quad (6.14)$$

According to these formulas, it is possible to approximate the new characteristics of the pump. To change the operating mode of the pump, you should have several sets of wheels and replace them. When the outer diameter of the wheel is reduced, for example, by 10 and 20%, the efficiency decreases by 1 and 4%, respectively.

LECTURE No. 7
CENTRIFUGAL PUMPS

Plan:
1. Elements of centrifugal pumps.
2. Single-stage pumps.

1. Elements of centrifugal pumps. The main elements of centrifugal pumps: impellers, shaft, body parts, sealing devices, guiding devices, unloading devices, bearings. Figure 7.1 shows the simplest centrifugal single-stage cantilever pump 2K-6. It consists of a wheel / cantilevered to the shaft 2, a spiral housing 3 with suction 4 and discharge 5 nozzles, an oil seal 7 with a lid 8, a seal 11, two ball bearings 10 mounted on a bracket 9, and a coupling 12, In the housing 3 there are holes with plugs for pouring and draining water. The shaft and the bronze sleeve 6 of the sliding bearing are lubricated with water flowing through the channel from the impeller to the oil seal 7.

The impellers of shaft pumps are of two types: closed and semi-open without a front disc. Wheels are manufactured: for non—aggressive water — cast iron; for acidic water - from chromium-nickel and chromium-plated steel, chromium or siliceous cast iron, acid-resistant bronze and plastics

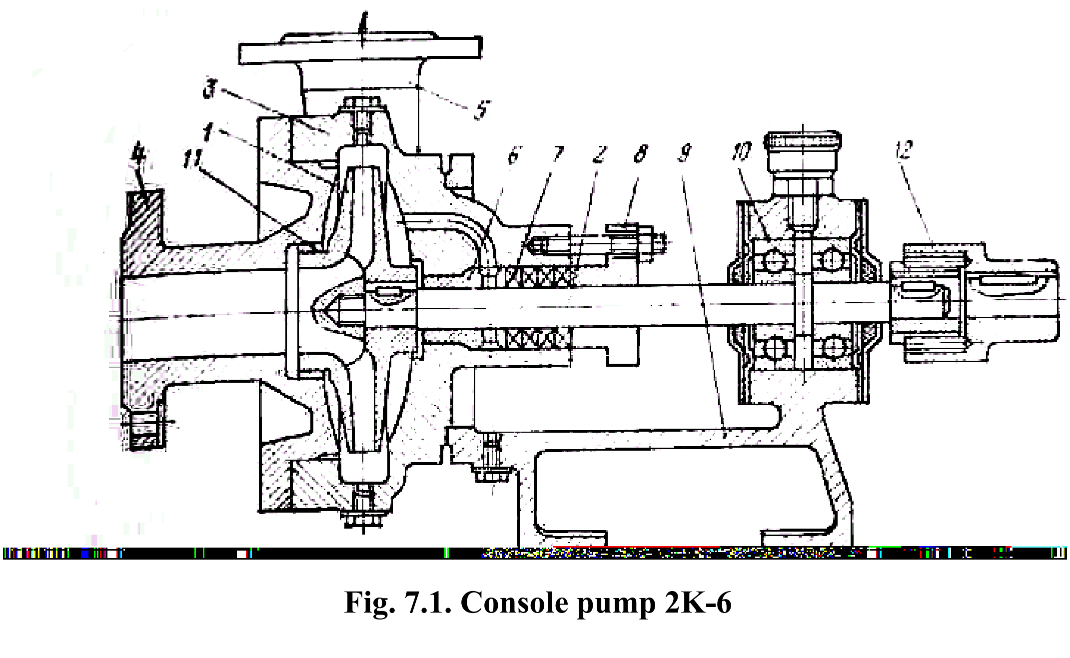


Fig. 7.1. Console pump 2K-6

Semi-open type wheels are used for transporting contaminated and thick liquids. Closed-type wheels have found the greatest use. Impellers are made with one-way or two-way fluid supply.

Wheel blades (Fig. 7.2) with rational profile shapes have a thickened rounded front part and a thin trailing edge (a). This shape of the blade profile is favorable from the point of view of strength, but with small working dimensions Fig. 7.2. The shape of the profiles of the working blades of the wheel thickened leading edges of the working blades significantly constrain its input section. Therefore, other blade profiles are also used: of the same thickness (b) and with a thickened middle part (b).

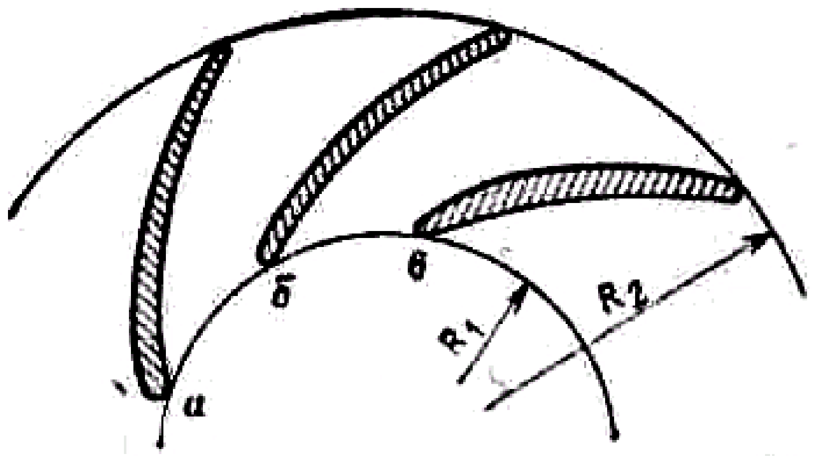


Fig. 7.2. Pump wheel blades

Since the relative velocity of fluid flow around the surfaces of the impeller is high, in order to reduce hydraulic losses and increase efficiency, the surfaces must be carefully treated.

The supply devices of modern pumps are made in the form of a rectilinear conical nozzle, annular and spiral (often also called semi-spiral) supplies. The conical nozzle (confuser) meets the requirements for the supply devices, but can only be used in single-stage cantilever pumps. The annular supply is a channel of constant cross-section, which is located around the circumference of the entrance to the impeller. With this design, the uniformity of the velocity field is disrupted due to the formation of a vortex zone around the shaft. The annular supply is arranged only in some pump designs that have a connector perpendicular to the shaft. The spiral supply is a channel of variable cross-section along the circumference of the entrance to the wheel / Due to this, one part of the flow enters the wheel without flowing around the shaft, and the other part smoothly flows around the shaft, evenly distributed around the circumference. This supply is currently used in most designs of multistage pumps.

Discharge devices of pumps. As such devices, a spiral tap, a guide device and an annular tap are used.

In multistage pumps with sequential connection of wheels, it is necessary to transfer fluid from the previous wheel to the next one. For this purpose, in sectional pumps (of the CNS type), guide devices are installed between the impellers.

Guiding devices are of two types: shovel and channel. In the case of shovel devices, the channels of the blades located on both sides form diverting 1 and reverse 2 (leading to the next wheel) channels. The fluid flowing from the impeller into the radial grating of the guide vanes located behind it reduces the speed by 2-2.5 times. Further, the fluid flow, rotating in the annular channel 3 by 180 °, enters the reverse radial grating, where the final unwinding of the flow and an additional slight decrease in velocity due to the diffusor effect of the grating usually occur.

Annular outlet is a channel of constant cross—section located along the circumference of the exit from the impeller. Such taps are used in pump designs designed for pumping contaminated liquids, and in coal pumps.

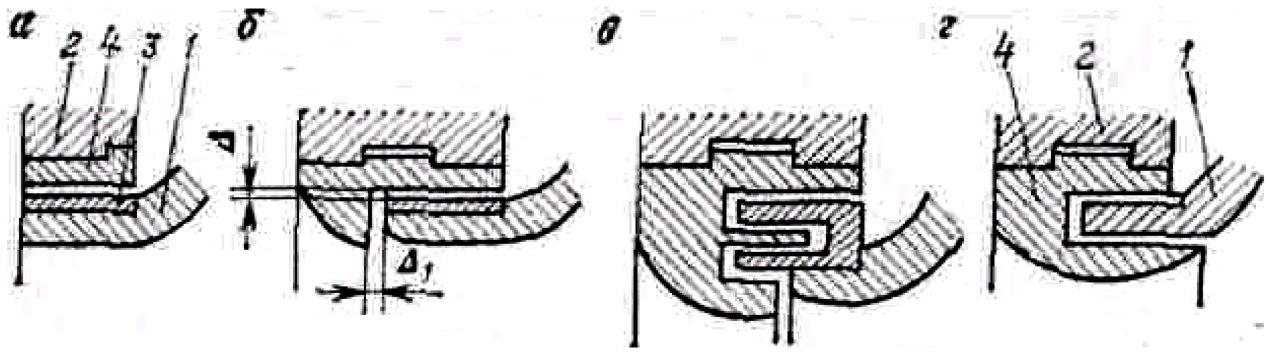


Figure 7.3. Some types of impeller sealing

Bearings. In centrifugal pumps, sliding bearings with liquid lubrication or rolling bearings with liquid or thick lubrication are used. In some designs of borehole pumps, rubber or plastic sliding bearings with water lubrication are used.

Impeller seals are used to reduce the flow of water from the pressure cavity into the suction pipe, that is, reduces the volumetric efficiency. Seals are always made with slits, without friction between solid sealing elements (Fig. 7.3). The simplest is an annular seal (Fig. 7.3, a, b). Smaller leaks give labyrinth seals (Fig. 7.3, c, d), but they are more complicated. The walls of the seals are subject to quite intense wear. The gaps increase due to the possible so-called slit cavitation and rapid movement of the liquid, and especially if the pumped liquid contains even a small number of abrasive particles. With this in mind, seals are mostly performed with removable rings.

It should be noted that the radial clearance L is always made much smaller than the axial Ad, since the accuracy of radial landings is higher than the axial dimensions for installing the wheel.

The oil seals are designed to prevent fluid leaks from the pump along the shaft. The oil seal should not be completely sealed, since it is necessary to pass liquid through it to remove heat that appears due to friction.

The pressure in the sealing cavity of the shaft pumps is higher than atmospheric. For this purpose, seals with a hydraulic lock are used (see Fig. 7.1). The lock is a hollow ring into which liquid flows from the impeller under pressure and prevents air from entering the pump through the oil seal,. At the pumps for the supply of hydraulic mixture, valves are used on both sides and clean water is supplied to them from an outside source. Stuffing box — cotton, hemp or linen cords soaked in thick fat, etc..

To reduce the intensity of shaft wear, a protective jacket is placed on the shaft at the point of passage through the oil seal.

2. Single-stage pumps. Single-stage horizontal and vertical pumps with spiral bends are used at mining enterprises. They are used on the site drainage, as well as as auxiliary and special. For water supply, the most common types of pumps are cantilever

pumps of type K and pumps with double suction wheels (according to GOST 10272-77) of type D (old code NDv, VAt, NDn, D).

D—type pumps are horizontal, with a two-way inlet impeller and a horizontal housing connector.

In the lower part of the housing, the suction and discharge pipes are horizontally located, directed in opposite directions at an angle of 90 ° to the axis of the pump. This arrangement of the nozzles and the horizontal connector of the housing allow you to disassemble, inspect and replace the working bodies without removing the pump from the foundation and without dismantling the electric motor and pipelines.

Pumps of type D provide a supply within 90-1700 m/h at a pressure rarely exceeding 100 m, and are used for pumping clean water.

LECTURE No. 8
MULTISTAGE PUMPS

Plan:

1. Nomenclature of centrifugal sectional pumps.
2. Design of sectional pumps.
3. Nomenclature and design of multistage spiral pumps.
4. Vertical multistage pumps.

Multistage horizontal pumps

Due to the need to provide significant pressure when pumping water in mines and mines, multistage centrifugal pumps, which are used in main drainage installations and in auxiliary (precinct) installations, have become the most widespread. Multistage pumps are divided into two groups — sectional and spiral. In the first case, the pump consists of the same type of sections, each of which is a separate pump stage. Sectional pumps are characterized by relatively small dimensions and weight.

If necessary (to regulate the pressure), the number of sections in the pump can be changed relatively simply. Disadvantages of pumps: during repair, it is necessary to disconnect the suction and discharge pipelines, difficulties in controlling gaps during assembly, etc.

Spiral pumps have a more advanced flow part, which causes a higher efficiency of machines of this type. However, they are cumbersome due to the large size of the bypass channels between the steps.

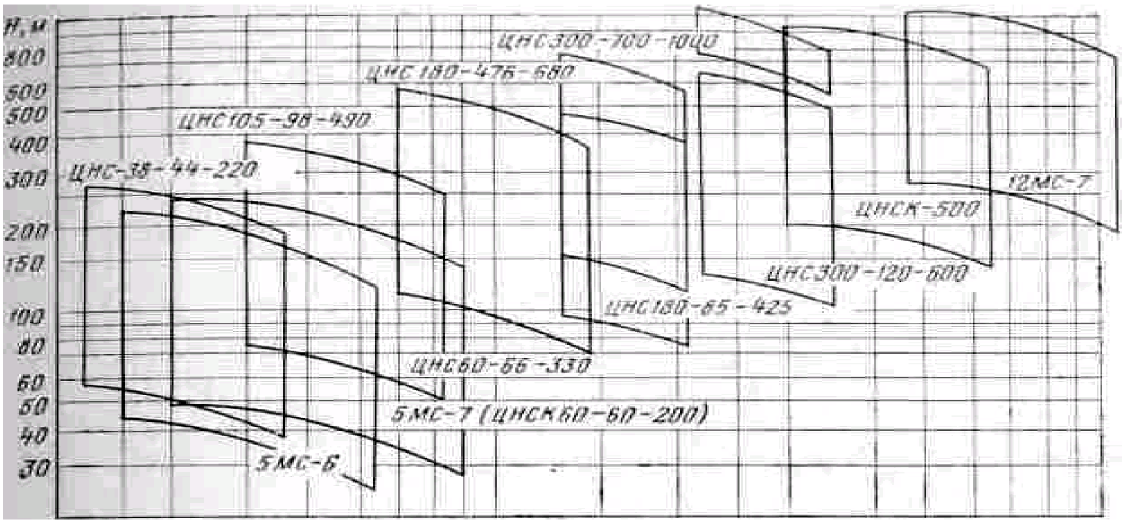
At mining enterprises, it is customary to use horizontal sectional pumps when feeding up to 300-450 m/h, and spiral pumps when feeding up to 450—1500 m/h.

1. Nomenclature of centrifugal sectional pumps. The main ones for the main and precinct mine drainage are multistage centrifugal sectional pumps manufactured by Yasnogorsk Machine-Building Plant, types of CNS and MS.

In accordance with GOST 10407-70, the designations of pump sizes include: initial letters — the name of the pump — CNS (sectional centrifugal pump), numbers after the letters — the nominal pump flow (m / h), numbers after the hyphen — the head in the design mode in meters (with a minimum and maximum number of steps). The letter K after the Central nervous system means that the pump is designed for pumping an aggressive medium (i.e. acid-resistant). For example, the CNS 300-120-600 is a centrifugal pump with a nominal flow of 300 m3/h and a head from 120 to 600 m.

In the designation of pumps of the earlier MS series, the first digit is the diameter of the inlet pipe (mm), reduced by 25 times and rounded, MS is multistage, sectional, K is acid-resistant, the digit after the hyphen is the speed coefficient, reduced by 10 times and rounded.

Centrifugal pumps are available with a rotation speed of 1450 and 2950 rpm and a specific rotation speed of 60-70 and 90-100 rpm, respectively.



• 24 25 30 40 50 КО 80 100 150 200 250 300 Ш 500 600 $м^3/ч$

Figure 8.1. Areas of industrial use of the main shaft pumps of the CNS type

Pump housings consist of separate sections, therefore, with constant supply, the pressure changes by installing the required number of impellers and guide devices; at the same time, only the lengths of the shaft, tie pins and bypass pipe change.

Figure 8.1 shows the areas of industrial use (working areas) of the main shaft pumps of the CNS type.

2. Design of sectional pumps. The sectional centrifugal pump (Fig. 8.2) consists of a rotor with impellers 4 and a sectional housing / with suction 2 and discharge 7 nozzles.

The bulk 1 of the rotor is mounted, in addition to the impellers 2, a thrust ring 3, a protective jacket 4, a remote sleeve 5 and an unloading disc 6, tightened on the shaft by a nut 7.

The housing consists of housing sections 8 with guiding devices 9 inside them, suction 10 and discharge 11 covers, tightened with studs 12. The joints of the sections are sealed with a rubber cord.

The rotor supports are two deep groove roller bearings 13 mounted in brackets 14 on a sliding fit that allows the rotor to move axially and rotate the outer ring of the bearing.

The liquid entering through the gap between the remote sleeve 5 and the sleeve 15 presses on the disc 6 with a force equal to the sum of the FORCES acting on the pump rotor, but directed in the opposite direction. The equality of these efforts is established automatically. The liquid from the chamber is drained through the discharge tube into the water collector.

The stuffing box is stuffed with hemp stuffing, braided and impregnated with an antifriction compound. The suction of air through the oil seal is prevented by a hydraulic seal. To do this, a liquid is supplied from the first stage of the pump through the hole in the suction cover 10 into the cavity of the hydraulic seal, forming a liquid ring during the operation of the pump, which prevents air suction. In this case, part of the liquid from the cavity of the hydraulic seal seeps through the shaft jacket 4 and cools the oil seal, the rest of the liquid enters the cavity of the suction cover through the gap between the sleeve and the jacket 4 and then the first wheel. For the normal operation of the unit, such a tightening of the oil seal is necessary so that water seeps through it in an amount of 0.5 l / min. Excessive tightening of the oil seal prevents water from seeping through the oil seal and increases friction losses.

In some designs of sectional pumps, liquid is fed into the hydraulic seal of the oil seal from the discharge device of the pump through the bypass tube.

The design of pumps of the CNS and CSC types differs from the one considered mainly by the design of guide devices, which can be made at the same time with the sections of the housings and the designs of the pump shaft seal assembly from the suction side and the unloading unit from the axial force, the designs of shaft supports and body parts.

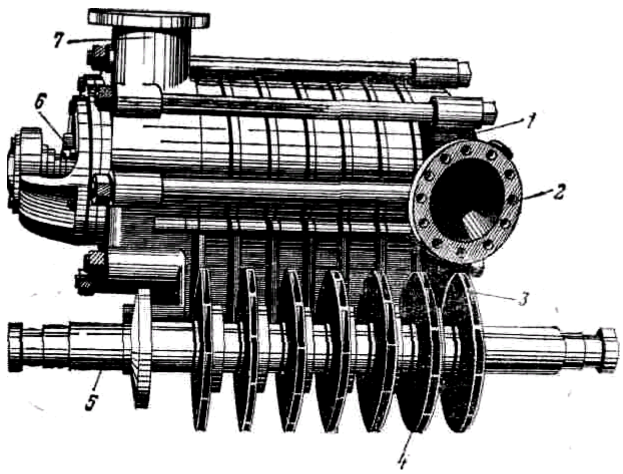


Figure 8.2. General view of the sectional pump
1 - pump body; 2 - suction nozzle; 3 – blades; 4 – impeller; 5- shaft; 6 - oil seal; 7 - discharge nozzle

The main difference between pumps of the TSNSK type and the CNS type is in the materials of parts made of acid-resistant steel or cast iron.

3. Nomenclature and design of multistage spiral pumps. Spiral pumps differ from sectional center-line pumps by the absence of guide devices replaced by spiral chambers in the pump housing, and by the symmetrical arrangement of the inlet openings of the impellers, which makes it possible to abandon hydraulic unloading devices. In some designs of such pumps, the first impeller has a two-way suction. The distinctive features of spiral pumps are the presence of a horizontal housing connector and an even number of wheels (a wheel with a two-way inlet is considered double) to balance the axial force. Spiral multistage pumps have become more widespread in mines, especially in difficult mining and geological conditions and with a negative suction height. These pumps have flatter characteristics, longer service life and are subject to less vibrations than sectional pumps. However, spiral pumps have large dimensions and weight.

In the mining industry, M-type pumps are used for drainage, in particular spiral pumps 5MD and 8MD of the Sumy plant, etc. These pumps are horizontal, three- and five-wheeled.

The MD type pump consists of a housing comprising two parts and 2 with a connector in a horizontal plane; impellers 3, 4 and 5 mounted on a shaft 6', a bypass channel 7 and a coupling 8.

The liquid enters the impeller 3 through the suction pipe 9 through a spiral supply, from where it is fed into the chamber 11 through the spiral channel 10, channel 7 and then into the impeller 4. From the impeller 4, the liquid enters the impeller 5 through the spiral channel 12, after which it enters the spiral channel 13 and the discharge pipe 14. The shaft 6 is sealed with a stuffing box 15 held by bushings 16. The axial force is balanced by the counter arrangement of the suction holes of the impellers, the residual unbalanced force is perceived by the ball fifth. The body and impellers are made of cast iron, the shaft is made of steel.

4. Vertical multistage pumps. Vertical pumps are used to supply water from wells and wells. At mining enterprises, these pumps are used in stationary drainage installations (main and precinct), in installations for preliminary drainage of deposits through wells, as well as in mobile installations when drilling vertical shafts and pumping flooded workings.

Vertical multistage pumps are manufactured in two types — non-submersible and submersible.

Non-submersible vertical pumps are characterized by the fact that their electric motor is not immersed in water and is installed above the water level, only the pump is immersed in water. Submersible vertical pumps are immersed in water together with an electric motor.

Non-submersible vertical pumps include suspended tunneling pumps designed for drainage during the sinking of mine shafts (types PPN and VP), and artesian pumps for drainage, water reduction (at mining enterprises) and water supply by water intake from wells.

Non-submersible artesian pumps are manufactured of ATN, A and NA types with a shaft length of up to 100 m and a flow rate of 30 to 1200 m3/h with minimum well diameters from 200 to 600 mm. The pump is located in a well filled with water, and the electric motor is located on the surface. The impellers 2 of the pump are suspended from a long drive shaft 1 placed inside a column of discharge pipes. The shaft is assembled from segments of round steel 2.5—3.5 m long, connected by means of couplings, and rotates in rubber bearings 7, placed in cross-shaped brackets, which are mounted inside the pipeline between the joints. Centrifugal impellers of closed type. Since the diameter of the wheels is limited by the diameter of the well, the pressure developed by each wheel is small and it is necessary to increase the number of wheels, especially with small well diameters. So, at the pump ATN-8-1-22 , having a maximum diameter of 188 mm, the number of wheels is 22.

The main disadvantage of such pumps is the presence of a drive shaft in the discharge pipe column, which limits the pumping depth and increases the installation weight and resistance in the pipes, as well as complicates the installation and operation of the pump. Therefore, at present, borehole pumps with a submersible electric motor lowered into the well together with the pump and the discharge pipeline are becoming increasingly common. Modern submersible multistage vertical pumps are equipped with submersible electric motors in which water fills all internal cavities and is used to cool the winding and lubricate its bearings. The electric motor has a moisture-proof insulation of the windings and is located below the pump. Non-metallic materials replacing steel and cast iron are widely used in these pumps: polypropylene, polyamide, impact-resistant polystyrene, wear-resistant rubber, etc.

Pumps are suspended in wells on air-gas-wire pipes connected to each other by a conical thread.

The main type of submersible pumps are centrifugal pumps of the ECV type (electric centrifugal for water). They provide a supply from 4 to 375 m3/h and a maximum head of up to 540 m.

5. Special pumps. Special pumps include pumps for transporting a mixture of solid material and water — coal pumps and slurry pumps designed for hydraulic lifting of coal from mines with a hydraulic method of its extraction, transportation of hydraulic laying of the developed space, pumping of unlit mine waters, cleaning of sumpfs and water collectors from sludge, transportation of hydraulic mixtures at processing plants, etc

Coal pumps allow transporting coal a hydraulic mixture with a ratio of solid materials with a grain size of up to 100 mm to a liquid by weight of no more than 1 : 5.

The main types of angle pumps are 10U4, 12U10, 14U7 and 12UV6 (the first digit is the diameter of the suction pipe, mm, reduced by 25 times; the letter U is the angle pump; the letter B is high—pressure; the last digit is the coefficient of specific speed, reduced by 10 times and rounded). The 12UV6 coal pump is a two—stage spiral pump with a horizontal connector: the other types of coal pumps are single-stage pumps characterized by a number of design features related to the transportation of particles and pieces of coal. In particular, their impellers are made with wide passageways and a minimum number of working blades.

Centrifugal slurry pumps are single-stage cantilever pumps with closed (6SH8 and 8SH8) and open impellers (VSHN-150, SHN-200-1, SHN-150-1).

Pumps of the SHN type pump slurry water with a solid particle content of up to 50% with a maximum diameter of these particles up to 20 mm.

Coal pumps and slurry pumps have a flow part, the elements of which are made of materials that are particularly resistant to waterjet wear; replaceable parts are provided in their designs; the speed of rotation of the impeller is usually taken lower than that of water pumps, etc. The efficiency of special pumps is lower and is 0.53—0.68 for coal pumps.

LECTURE No. 9
EQUIPMENT AND EQUIPMENT OF DRAINAGE INSTALLATIONS

Plan:

1. Requirements for drainage stationary installations
2. Pipelines of drainage installations.
3. Technological equipment for monitoring and control of single-drain installations
4. Electric drive and automation equipment of drainage installations.

1. Requirements for drainage stationary installations. A stationary drainage system consists of pipeline pumps, an electric drive and automation equipment. Stationary drainage installations should provide a minimum of capital and operational costs for pumping water. To do this, pumping units must have a high efficiency, wide areas of operating modes and during the entire service life their modes must not leave these areas, which is achieved by the correct selection of equipment, control of the supply and pressure of pumps, as well as their timely inspection and repair.

Pumps should have minimal dimensions, and their installation, disassembly and maintenance should be simple.

The supply of the working pumps of the drainage system should ensure the pumping out of the daily inflow in no more than 16 hours.

The number of pumps of stationary installations with a water inflow of more than 50 m3/h should be at least three, of which one is working, the second is backup and the third is replaceable.

The central and main drainage installations must have at least two injection pipeline staves — working and backup. For installations with four pumps, of which two are permanently operating, as well as for pumping acidic waters, three discharge pipelines should be provided. The number of pressure stations at the mines is also determined from the CONDITION of the water flow rate in the pipes, which should not exceed 2.5 m /s with one disconnected pressure pipeline.

To control the operating modes, the main and district drainage installations must be equipped with a pressure gauge and a control flow meter.

Modern drainage installations should be equipped with automation tools and work without the constant presence of people.

2. Pipelines of drainage installations. Pipelines must have sufficient strength of parts and high quality seals to ensure reliable operation of drainage systems and safe maintenance; corrosion resistance; minimum weight and metal consumption; availability for inspection, repair and replacement of elements. Capital and operating costs for pipelines should be minimal.

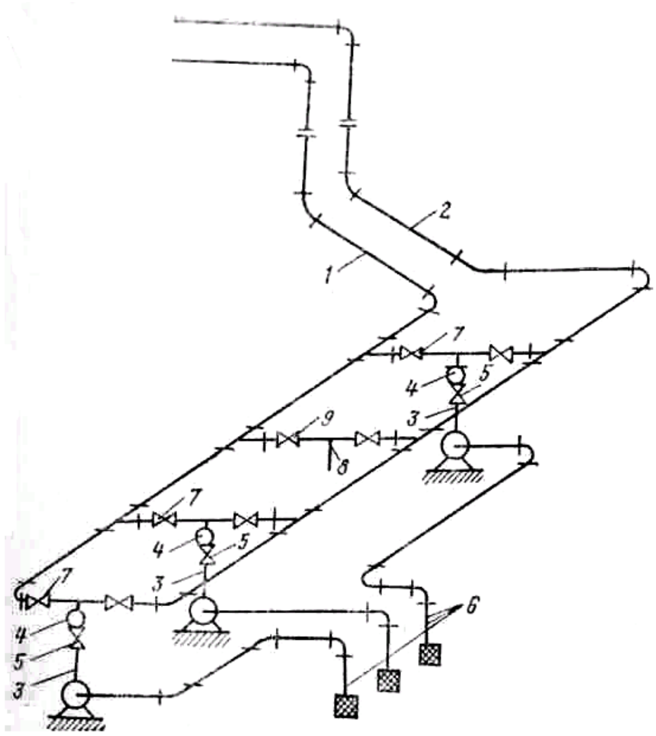


Fig. 9.1. Typical scheme of main drainage pipelines:
1 and 2 — working and backup main pipelines; 3 — connecting pipe; 4 — check valve; 5 — shut—off valve; 6 — suction pipelines; 7 — distribution valve; 8 and 9 - pipe and valve for releasing water from the staves into the water collector

The external network of pipelines (Fig. 9.1) of drainage installations consists of suction pipelines, pumps and main pipelines through which water is pumped to the surface. The pipeline scheme should provide for the possibility of connecting each of the pumps to any of the two or three main discharge pipelines.

Drainage installations, depending on the supply, are equipped with pipelines with a diameter of 100 to 600 mm when pumping water at a pressure of 1-10 MPa. For the pipeline, standard steel pipes are used, and less often cast iron (at a water pressure of up to 1 MPa), connected to each other by movable or fixed flanges, sometimes by electric welding. Rubber or rubberized material is used as gaskets when connecting pipes; at high pressures, lead or red copper gaskets are used.

The pipeline is equipped with fittings (Fig. 9.2), which includes a receiving valve with a grid (on the suction pipeline), a check valve, a shut-off valve with manual or hydraulic drive, a temperature stuffing box compensator, support pipes, support and conventional elbows, transition tees, etc.

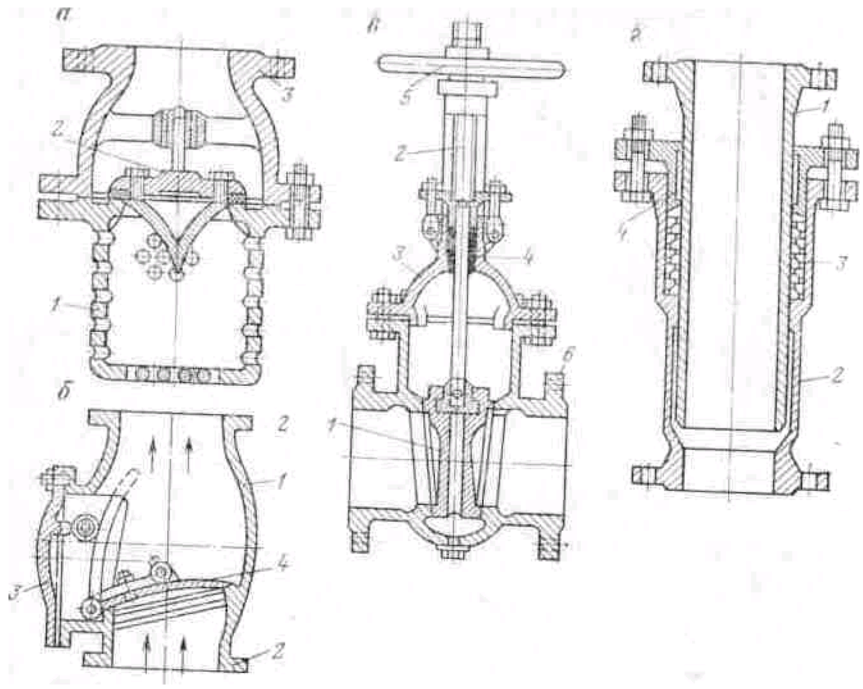


Fig. 9.2. Fittings of the drainage system pipeline:
a — intake valve with suction mesh: 1 — mesh; 2 — valve; 3 — flange; 6 —check valve: 1 — housing; 2 — flange; 3 — cover; 4 — valve; b — manually operated valve: 1 — shut-off valve part; 2 — spindle; 3 — cover; 4 — oil seal; 5 — flywheel; 6 — flange; g — oil seal compensator: 1,2— pipes; 3 — oil seal; 4 — axle boxes

Pipelines and fittings made of chromium—nickel steel or of cheaper material - gray cast iron or carbon steel, lined from the inside with rubber, wood (spruce, fir) and other materials can be used for pumping acidic waters.

The vertical discharge pipeline with a length of less than 200 m is installed on a metal support elbow, which perceives the weight of the stack. With a longer length, the weight of the discharge pipeline is distributed over several intermediate supports. In this case , the pipeline is divided into sections with a length of 150-250 m .

An intermediate support is installed at the lower end of each section, and a compensator is installed at the upper end, which, in addition to distributing the weight of the stake on the supports, compensates for temperature elongations, prevents additional stresses in the pipeline during deformations of the support, as well as facilitates installation and repair of the pipeline.

From horizontal displacements, the pipeline is held by clamps, which are installed after 25-35 m and attached to the barrel shootings or special beams.

According to the rules of technical operation of coal mines, pressure pipelines are subjected to a hydraulic pressure test of 1.25 working pressure before commissioning.

During operation of drainage systems in transient modes, for example, when starting a pump or its sudden stop (for example, during a power outage), a hydraulic shock may occur.

N.E. Zhukovsky showed that with the instant closure of the valve, the maximum pressure increase

$$p = \rho c v_0, \quad (9.1)$$

where c is the velocity of propagation of the shock wave, m / s; —the velocity in the pipeline until the valve closes, m / s.

Drainage installations have a long length of pipelines and have significant inertia, therefore, the values of pressure increase during hydraulic shock are lower than those determined by the formula (9.1). Theoretically, the pressure during hydraulic shocks in drainage installations does not exceed 1.5 pct. Measures of protection against hydraulic shocks are the installation of: a relief valve, a second check valve in the middle part of the discharge pipeline, special shock dampers, etc.

The rules of technical operation of coal and shale mines regulate the mandatory use of safety devices to reduce hydraulic shock on drainage installations pumping water to a height of more than 400 m.

3. Technological equipment for monitoring and control of single-discharge installations. When monitoring and controlling drainage installations, it is necessary to measure the water level in the catchment, the supply, the pressure (pressure), the temperature of the pump bearings and the electric motor (Fig. 9.3). Sensors are used to measure these parameters, which are combined with relays that operate at certain preset parameter values.

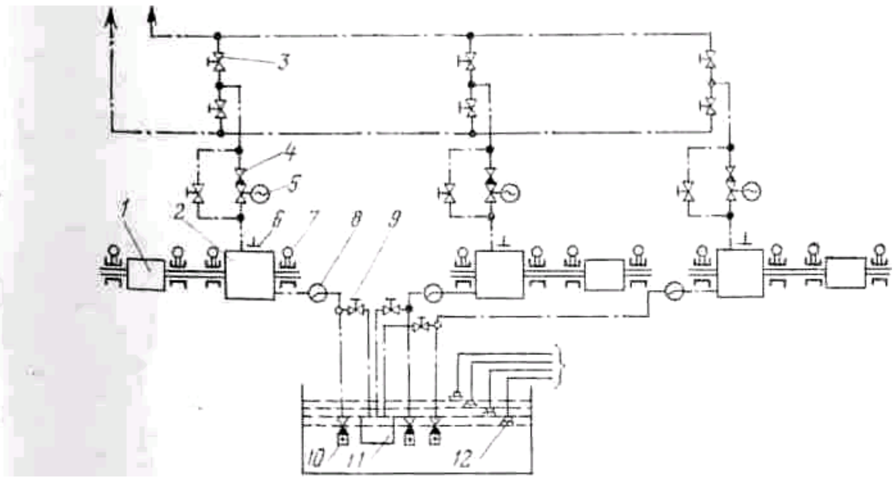


Figure 9.3. Diagram of an automated drainage system:
1— main pump electric motor; 2 — main pump; 3 — manually operated valve on the discharge pipeline; 4 — check valve; 5 — electric drive of the engine; 6 — pressure switch; 7 — temperature sensor for temperature control; pump and electric motor bearings; 8 — relay 9 — manual valve in the pump filling system; 10 — suction valve; 11— filling pump; 12 — electrode level sensor

Level sensors are used in automation systems of drainage installations to automate their operation in the function of the water level in the catchment. There are two types of sensors — float and electrode. The most widespread are electrode level sensors of the ED type, built on the principle of electric current conductivity by mine waters and their dielectric properties.

During operation, the electrode sensors are suspended from the cable in the water collector at the marks corresponding to the controlled water levels.

The pump supply relay performs, along with the control of the pump supply, its hydraulic protection. Hydraulic protection is commonly referred to as blocking from turning on an unloaded pump and turning off a working pump when it loses its supply. The relay is installed on a horizontal section of the suction pipeline.

Pressure switches are used to control the filling of pumps, the water level in the water collectors of buried pumping chambers and other purposes. Various sensitive elements are used in the pressure switch: compacted and non-compacted pistons, bellows and an ordinary flat membrane.

4. Electric drive and automation equipment of drainage installations. Electric drive of pumps. Asynchronous electric motors with a short-circuited rotor and less often asynchronous electric motors with a phase rotor are mainly used as an electric drive for pumps of drainage installations. For pumps of the CNS type, electric motors with a short-circuited rotor are used in the normal version of a single series A and AO.

For drainage installations operating in explosive conditions, electric motors of the MA, KO and VAO series with a voltage of 380/660 V and electric motors of increased reliability and blown under excessive pressure of the A, AO, AP, AZP series, explosion-proof engines of the VAO series with a capacity of up to 1600 kW (rotation speed 1500 rpm) and the Ukraine series are used (rotation speed 3000 rpm) with a power of up to 630 kW at a voltage of 6000 V.

Complete equipment for automation of installations. For automation of stationary drainage installations, Konotop plant "Krasny Metallist" produces automated control equipment AV-5 and AV-7 — for precinct drainage, AVO-3 — for single drainage, AVN-1M — for drainage installations with low-voltage engines, UAV— for main drainage systems with low-voltage and high-voltage engines, VAV-for the main drains of mines that are dangerous for gas or dust. The most advanced is the UAV and VAV equipment.

The unified automation equipment of the UAV type is designed for automatic control of mine drainage installations with a number of pumping units up to 16 with asynchronous low-voltage and high-voltage electric motors with a short-circuited rotor. The circuit of the equipment is assembled from multi-contact relays in combination with semiconductor elements.

Explosion-proof equipment of the VAV type is designed for automatic control of drainage installations with up to nine pumping units with high-voltage and low-voltage electric motors in mines developing formations,

Dangerous or threatened by sudden emissions of coal and gas. Automatic control of pumping units is carried out according to the water level in the catchment area. Depending on the set program, the pumping units are switched on at the upper, elevated and emergency levels. When a faulty pumping unit operating in automatic mode is turned off, a backup one is turned on. Pumps can work with and without controlled valves.

The unified automation kit of the KAV type is designed for automatic control of drainage installations in the RV and PH versions, respectively, with a number of pumps up to 10 and 16.

The versatility of the kit allows them to replace all currently manufactured sets of automation equipment for mine drainage installations.

The expansion of functionality and greater flexibility of the kit structure, which allows automating almost any technological scheme of mine drainage, are achieved by software restructuring of the information processing algorithm and the aggregate construction of the technical means of the kit.

LECTURE No. 10
DESIGN AND INSTALLATION OF DRAINAGE SYSTEMS

Plan:
1. Tests of pumps of drainage installations
2. Maintenance of drainage installations.
3. Design methodology of the drainage system.

1. Tests of pumps of drainage installations. At mining enterprises, characteristics and comparison of them with factory ones to establish the technical condition of pumps. The pumps are tested after the installation of drainage systems, as well as after each major repair.
In this case, the pressure, supply and power consumption are measured in the range of operating modes that overlap the working area.
A Venturi tube and a diaphragm with a differential pressure gauge are used to measure the flow.
The pump pressure is measured by a pressure gauge and a vacuum gauge. The wiring diagram of the devices during normal tests is shown in Fig. 10.1.

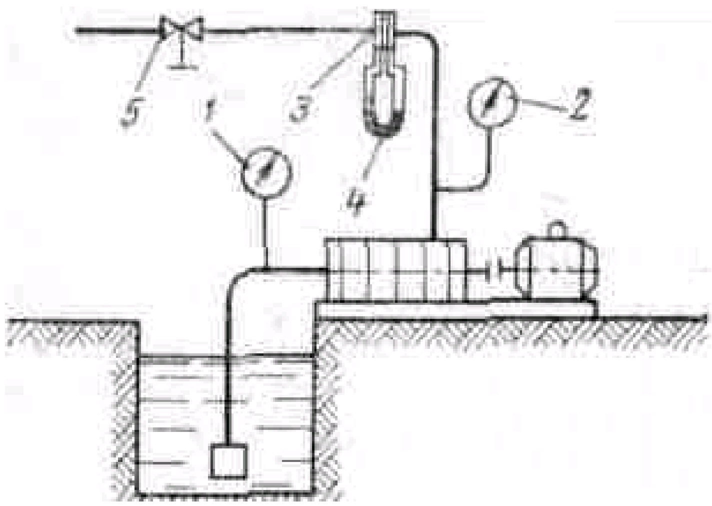


Fig. 10.1. Installation diagram of instruments and equipment for industrial testing of pumps: 1 — vacuum gauge; 2 — pressure gauge; 3 —measuring diaphragm; 4 — differential pressure gauge; 5 — valve

2. Maintenance of drainage installations. For normal operation of drainage installations, a number of requirements must be met: air must not penetrate the pump; the intake valve of the suction pipeline must be positioned at least 2.5 m below the liquid level, the distance between the intake valve grid and the bottom of the well must be at least 0.5 m, and from the walls of the well to the intake valve with a grid of at least 0.3 m; the live section of the intake valve mesh should be 4-5 times larger than the section of the pipeline; the diameter of the suction pipeline should not be less than the diameter of the suction pipe; with a suction line length of more than 10 m, the inner diameter of the pipeline should be larger than the diameter of the pump inlet pipe and connected to it through a confuser: pressure pipelines should have independent supports and not transfer forces to the pump; the work of pumps should be organized so that all pumps are in operation, alternating in a certain order; before starting the pump and the suction the pipeline must be filled with water up to the discharge pipe. In this regard, in modern automated drainage systems, pumps and suction pipelines are filled with water immediately before start-up, or they are constantly filled with water, or they work with a backup created by booster (auxiliary) pumps.

The pumps can be filled (Fig. 10.2) from the battery tank (a), directly from the discharge pipeline (b), by an auxiliary pump (b), using a vacuum pump (d) and a backup (booster) pump (e).

When filling from the accumulator tank (Fig. 10.2, and) at the beginning of pump 3 operation, water enters it from the accumulator tank.

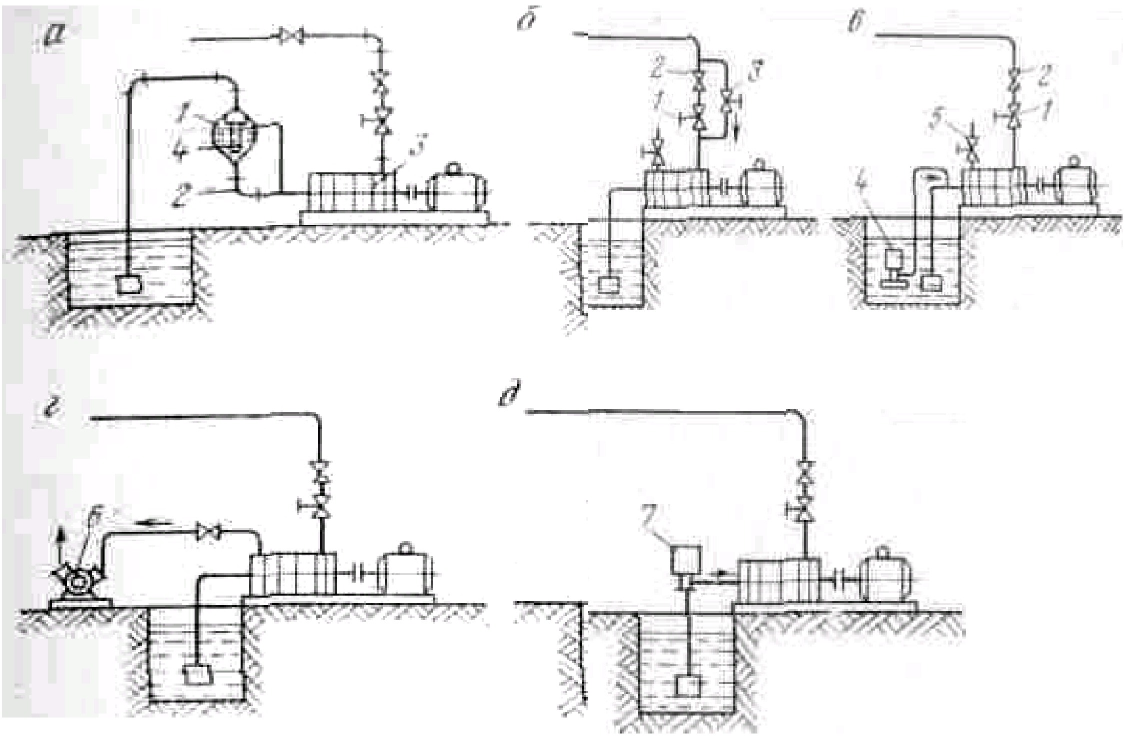


Fig. 10.2. Pump filling schemes:

a - from the accumulator tank: 1 - accumulator tank; 2 — suction pipeline; 3 - pump; 4 — ejector; b — from the discharge pipeline; c — auxiliary pump; d — vacuum pump; d — back—up pump; 1 - main valve; 2 — check valve; 3 — auxiliary valve; 4 — auxiliary pump; 5 — valve; 6 - vacuum pump; 7 — backup (booster) pump together with air from the ejector 4.

With an increase in the vacuum in the suction pipeline 2, the latter and the accumulator tank 1 are filled with water and the normal operation of the pump begins.

Filling of the pump from the discharge pipeline (Fig. 10.2, b) is carried out through a pipe installed bypassing the main valve 1 and the check valve 2. When filling, the valve 3 of the pipe opens.

When filling with an auxiliary pump 4 (Fig. 10.2, e), the valve 5 is opened to release air. When using a vacuum pump for filling (Fig. 15.2, d), the discharge pipeline is closed with a valve and the vacuum pump 6 sucks air from the pump and the suction pipeline.

Filling with booster pump 7 (Fig. 10.2, e) is carried out with the valve open on the pressure line. The water comes from the catchment area.

With manual control of the pumps, their filling can be carried out manually.

From the point of view of simplicity of automation of filling and reliability of operation of pumps the hydraulic scheme of drainage installations with constant filling of pumps with water is the best. Such a scheme can take place for centrifugal pumps with a horizontal axis when placing the pump chamber below

The water level in the water collector and when using vertical centrifugal pumps with immersion of the first impeller

It is recommended to start the pumps when the valve of the discharge pipeline is closed. At the same time, a smooth increase in the speed of water movement in the discharge pipeline is ensured, eliminating the occurrence of hydraulic shock in high-pressure pipelines and large overloads of the electric motor. When starting with an open valve, the electric motor may overheat and fail. Therefore, such a start-up is used only for low-flow installations. Its advantages are the comparative simplicity of the automation scheme and the quick start of the drainage system in operation.

Centrifugal pumps are stopped by first closing the valve on the pressure line, then the valve at the vacuum meter, and then turning off the electric motor. Indicators of normal operation of the pump are the amount of water flowing out of the drain tube, which should be 3-6% of the nominal pump supply, and its heating, which should not exceed 2 ° C. Evidence of normal tightening of the oil seals is the outflow of a small (up to 0.5 l/min) amount of water from the oil seal device. With a strong tightening of the oil seals, their heating occurs, additional power consumption, as well as increased wear of the oil seal elements.

Maintenance of pumps is reduced to monitoring their operation, monitoring the operating modes of the devices, timely lubrication, checking and adjusting the unloading device, tightening and replacing the seals, cleaning the suction device mesh from foreign objects.

The pipeline and water collectors must be periodically cleaned of precipitation. To clean the pipes, water is drained from the stave and washed from above with a jet of neutral water under pressure. Pipes can also be cleaned using hollow hedgehog balls or other special projectiles that move under the action of moving water. The balls moving in the pipeline by means of a winch and a rope are also used.

A pump operation log should be kept at the main drainage installations. The frequency and volume of preventive repairs are set in accordance with the factory instructions.

3. Design methodology of the drainage system. To calculate the drainage system, the following initial data are required: the value of the normal Qn and the maximum Q of the water inflows; the full geometric height of the water supply; the physico-chemical characteristics of the water necessary for the correct choice of equipment and the category of the mine by gas or dust (for coal mines), When calculating the installation, the pipeline is calculated, a pump and an electric motor are selected, a pipeline diagram is drawn up and the pump operation mode is graphically determined.

Calculation of the pipeline. When calculating the pipeline, the inner diameter and thickness of the pipes are set, as well as the amount of pressure loss in the pipeline.

The inner diameter of the discharge pipeline is determined by the formula

$$d_в = \sqrt{\frac{4Q}{\pi v_1 3600}}$$

where Q is the flow rate of water through the pipeline, is the velocity of water in the pipeline, m/s.

Based on experience, it is usually taken in the discharge pipeline and, ÷ 2.5 m / s, in the suction vT = 0.9 ÷1.2 m / s.

In accordance with the obtained value of d b, the pipes of the discharge pipeline are selected.

The diameter of the supply pipeline is usually taken to be 25-50 mm larger than the pressure one.

The wall thickness is selected depending on the water pressure in the pipeline, determined by the formula

$$p_{\max} = (0{,}011 \div 0{,}0115)_{zr}$$

Where Pmax is the water pressure in the pipeline, MPa;

zг is the geometric height of the discharge, m. Pressure losses in the pipeline from resistances in the straight section are

$$H_n = \xi \frac{l}{d_n} \frac{v_T^2}{2g} \qquad (10.2)$$

Where ξ is the coefficient of resistance depending on the roughness of the inner surface of the pipes; l is the length of straight sections of the pipeline of the same diameter, m.

The value of the coefficient for pipes that have been in operation for some time can be determined by the formula of F. A. Shevelev

$$\xi = \frac{0{,}021}{d_в^{0,3}} \qquad (10.3)$$

For mine conditions, ξ = 0.03 can be taken.

Pressure losses in the intake valve and the grid of the Hp gate valve, the check valve Hо.к. the tee Pe and in the cone transitions Hк.п. are determined by the formula

$$H = \xi \frac{v_T^2}{2g} \qquad (10.4)$$

Pressure losses in a round knee are determined from the expression

$$H_k = \xi \frac{\alpha_k v_T^2}{90 \cdot 2g} \qquad (10.5)$$

Where α_k is the bend angle of the knee.

The coefficient of resistance in the intake valve and the grid is assumed to be equal when the diameter of the suction pipeline is 0.3 m in the tee $\xi_{к\cdot с} = 3{,}7$ with a diameter of 0.5 м $\xi_{к\cdot с}$=2,51; in the tee $\xi_T = 2$; in the latch when opening it by a quarter $\xi_3 =$ 17, when opening half ξ_3 2,06, when opening by three quarters $\xi_3 =$ 0,26; in the check valve at the opening angle 30—40° $\xi_3 = 30 \div 14$ and at the opening angle 50—60° $\xi_3 =$ 6,6-3,2; in the knee with the ratio of the pipe radius to the bend radius 0,5 $\xi_к$=0,294, by 0,7 $\xi_к$=0,661,' by 1,0 $\xi_к$=1,978; in cone transitions when switching to larger diameter pipes ξ_n= 1,0; with reverse transition and taper angle 10—15° ξ_n= 0,5, at the taper angle 20—30° $\xi_к$= 0,6-0,7.

The manometric pressure H_m (pressure for injection) is determined by the formula

$$H_М = z_r = +\sum H \qquad (10.6)$$

Where z_r — geodetic feed height, м; $\sum H$ — total losses in the discharge pipeline, m.

With approximate calculations $\sum H$ you can take 10—12% from 2г. Due to the reduction in the cross-section of pipes during operation due to the deposition of precipitation on their inner surface, the value of the pressure gauge, determined by the formula (10.6), should be increased by 5-8%.

Selection of pumps. When choosing pumps, it is assumed that the normal daily inflow should be pumped out by one pump in no more than 16 hours. Hence the minimum pump supply

$$Q_{\min} = \frac{24Q_{n\cdot n}}{16} \qquad (10.7)$$

The pump head H must be equal to or greater than the pressure gauge . The values of Qmin and H are plotted on the graph of industrial use areas and determine the type of pump. If the required mode (Qmin and H) can be provided by several pumps, then the final choice of the pump is made on the basis of a technical and economic comparison of the options.

If the pressure is sufficient and the supply is small, consider the possibility of parallel operation of pumps.

According to the characteristics of the selected pump, the supply pressure is determined in the optimal mode (i.e. at $\eta_{\max}$) and the pressure with the valve closed Ho.

In the case of centrifugal sectional pumps, these parameters are determined per impeller. The number of impellers is set by the expression

$$z = \frac{H}{H_k} \qquad (10.8)$$

rounded to the nearest larger number. The selected pump is checked for stability by the expression

$$z_r = H_r \leq 0{,}95\,\mathrm{H}_0 \qquad (10.9)$$

For sectional pumps H0=zH0.k where Ho.k is the pressure with the valve closed on one impeller.

If the condition (10.9) is not met, then it is necessary to increase the number of impellers or choose another pump.

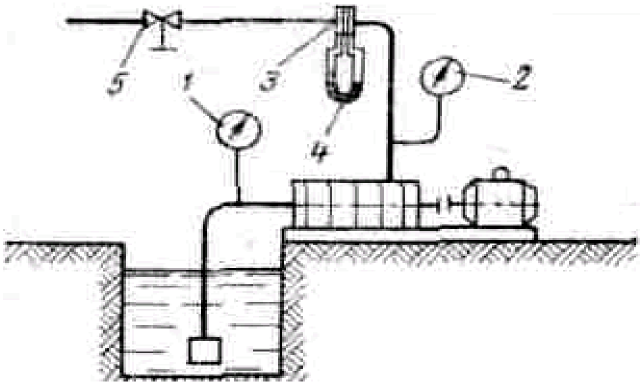


Fig. 10.3. Installation diagram of devices and equipment for industrial testing of pumps:
1 — vacuum gauge; 2 — pressure gauge; 3 —measuring diaphragm; 4 — differential pressure gauge; 5 — valve

LECTURE No. 11
GENERAL INFORMATION ABOUT FAN INSTALLATIONS
Plan:

1. Purpose and classification.
2. Features of fan installations.
3. Characteristics and areas of industrial use of fans

1. Purpose and classification. Fan installations are designed for continuous ventilation of mine workings and mines and the creation of normal atmospheric conditions in them. According to their purpose, they are divided into main fan installations, auxiliary and local ventilation.

The main fan installations are used to ventilate all operating workings of a mine or mine, with the exception of blind (dead-end) faces. They are placed on the surface at the mouths of hermetically sealed shafts or tunnels in the center of the mine field with a central ventilation scheme and on its flanks with a diagonal scheme and pass all the air passing through the mine (mine) or its wing.

The fan installation includes a fan (fans) and electric motors connected to it, input and output devices (supply channels, diffuser, output part and auxiliary devices for switching and reversing the air jet), as well as start-up and control equipment and sound-absorbing devices.

A fan is an assembly consisting of a housing, a rotor, guides and straightening devices, with a collector and an input box attached to it.

Auxiliary fan installations are designed for ventilation of shafts and capital workings during their penetration, chambers and workings of the near-trunk yard during the operation of mines and mines, as well as their individual sections. They are located on the surface near the trunk or pit.

Local ventilation fan installations consist of fans with drives, ventilation ducts, starting equipment and control and automation equipment. They are supplied with one and less often two fans.

In accordance with the purpose, fans are also divided into main, auxiliary and local ventilation fans. These groups of fans differ significantly in their parameters, and local ventilation fans differ in their design.

Fans used in the mining industry are turbomachines, which, according to the principle of operation, are divided into two large groups — axial and centrifugal fans.

According to the number of stages, the fans are single-stage (centrifugal) and multi-stage (axial).

Depending on the location of the shaft, the fans can be horizontal and vertical.

Comparative evaluation of axial and centrifugal fans. Both types of fans have their advantages and disadvantages.

In comparison with centrifugal axial fans, it is easier to reverse the air stream (without bypass channels) and have greater possibilities of regulating operating modes (by turning the guide vanes and impellers). They have smaller dimensions and weight with significant productivity, as well as a higher internal efficiency. It is more convenient to include them for sequential work.

At the same time, axial fans have the following disadvantages:

the shape of their characteristics is saddle—shaped, which creates prerequisites for unstable operation of fans, especially during parallel operation; strong noise when working at speeds of 90-95 m / s or more; rotor bearings are inaccessible for inspection, which leads to a decrease in the reliability of fans; greater sensitivity to the accuracy of rotor balancing.

Centrifugal fans do not have these disadvantages. The circumferential speeds of their impellers reach 120-125 m/s.

They have large heads compared to axial ones. However, the depth of economical regulation of centrifugal fans is less than that of axial fans. With large feeds and small pressures, a low rotation speed of their impellers is obtained and the installation of a gearbox between the fan and the engine is required.

Mass-produced axial and centrifugal fans are approximately equivalent in terms of efficiency.

2. Features of fan installations. The ventilation units of the main ventilation work according to the suction, discharge and combined ventilation schemes of the mining enterprise. Local ventilation fan installations provide injection ventilation.

The operation of the suction fan unit (Fig. 11.1) differs from the pumping operation in that the air spent in the mine is usually ejected by the fan through the diffuser into the atmosphere. In this case, the kinetic energy of the air flow is lost.

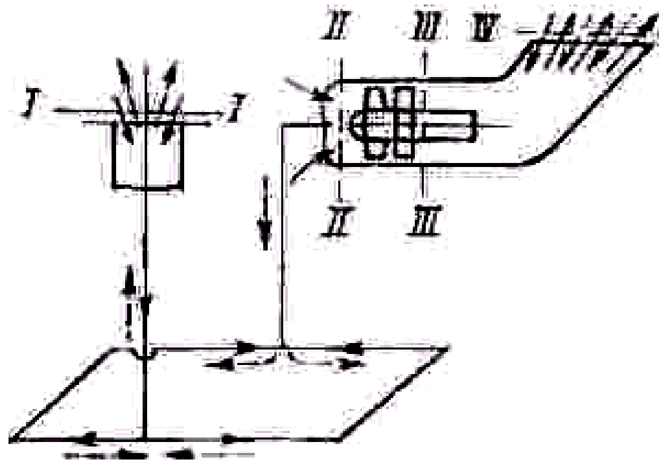


Fig. 11.1. Operation diagram of the fan unit

When working on suction, the operating mode of the fan is determined by the characteristic of static pressure, and the dynamic pressure lost at the outlet is attributed to losses not of the network, but of the fan installation. Since the vast majority of the main ventilation fan installations operate on suction, static pressure is given in the technical specifications for fans, except for local ventilation fans. When the flow is twisted in the opposite direction, the speed has a plus sign and the pressure increases.

The depth of regulation by means of a guide device for centrifugal fans is less than for axial fans.

The rotation of the blades is regulated in axial fans and by turning some parts of the impeller blades in centrifugal fans. This method is especially appropriate for cases when the main adjustable parameter is the pressure behind the fan. When the angle of installation of the blades of the axial impeller changes, the angle of attack 6 changes, which leads to a change in the circulation of G around the blade. As the angle of attack increases, the circulation of the impeller and the fan pressure increase.

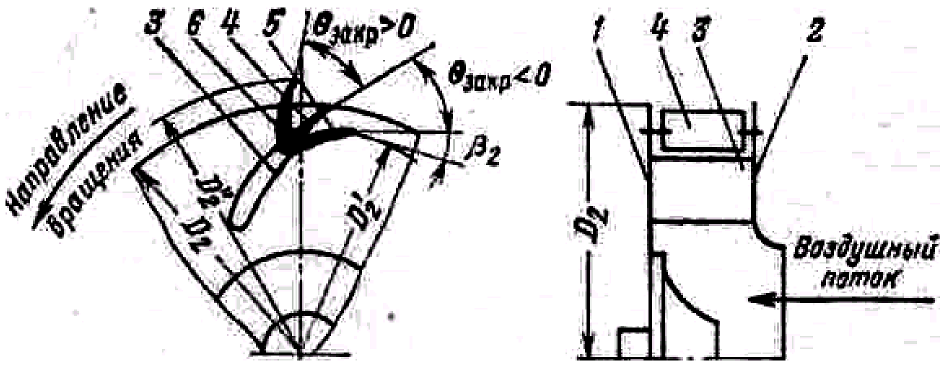


Fig. 11.2. Adjustment by turning the flaps of the impeller blades:
1 and 2 — root and cover discs; 3 — fixed part of the blade; 4
blade flaps; 5 and 6 — flap positions, respectively, at negative and positive angles of its installation

Shaft fans can be regulated using two or three methods simultaneously, for example, by changing the position of the guide vanes and the speed of rotation of the impeller.

The choice of one or another way of regulating fans should be decided taking into account its greatest efficiency, simplicity and ease of maintenance.

3. Characteristics and areas of industrial use of fans

Individual characteristics of the fans. The aerodynamic quality of fan installations and fans is characterized by the following main parameters: performance Q (m3 /s), static pressure pct when the fan is working for suction or full pressure p when working for injection (Pa), power on the fan shaft N (kW) and its static pst or full efficiency n.

Modern fans are equipped with means of regulating their operating modes. In this regard, the factory characteristic of the fans is a family of individual characteristics built at different angles of installation of the blades of guide devices, blades of impellers and straightening devices (axial fans) and at different rotational speeds of the drive motor (centrifugal fans).

Areas of industrial use of fans. Individual characteristics form the field of operating modes of the fan. The field of industrial use of the field of operating modes is limited by the conditions of economical and stable operation of the fan. It is generally assumed that the main ventilation shaft fan unit works economically if its efficiency is not lower than 0.6, and the local ventilation fan unit — at a minimum efficiency of 0.5. For installations with centrifugal fans having stable characteristics, this condition is the only one, and the scope of industrial use of installations is limited by a curve drawn through the points of individual characteristics p = f (Q) at pu.ct = 0.6. From above, the area of economical operation of modern centrifugal fans regulated by guiding devices is limited by individual aerodynamic characteristics when the blades of the guiding device are "Open" (0 °) or when they are installed at a negative angle of 9".a (no more than -25 °).

For installations with centrifugal fans controlled by changing the speed of rotation, these areas from above and below are limited by the characteristics at the position of the guide vanes "Open" (0 °) and, respectively, at the maximum and minimum rotational speed of the drive motor, and on the right and left — by the curves of modes with a single value of efficiency equal to 0.6.

For installations with axial fans, the area of economical operation on the right and left is limited by aerodynamic characteristics, respectively, at the maximum (45°) and minimum (15°) angles of the installation of the impeller blades.

Сверху эта область ограничена кривой, полученной из условия обеспечения при реверсировании вентилятора не менее чем 60% производительности при его нормальной работе и 20%-ном запасе сопротивления шахтной сети по отношению к границе однозначной работы. Снизу зона экономичной работы ограничена кривой вентиляционных режимов с величиной к. п. д., равной 0,6. Для установок с осевыми вентиляторами, имеющими индивидуальные характеристики с впадинами и разрывами, граница слева области промышленного использования определяется исходя из условия обеспечения устойчивой и однозначной работы вентиляторной установки при нормальном направлении воздушного потока и его реверсировании (левая граница). Принимается, что давление, развиваемое вентилятором, не должно превышать 90% максимально возможного давления вентилятора.

Наиболее, широкую область экономичной работы, а следовательно, наибольшую глубину регулирования имеют центробежные вентиляторы, регулируемые изменениями частоты вращения *и* ротора вентилятора и угла установки $\Theta_{закр}$ закрылков рабочего колеса, и осевые вентиляторы, регулируемые изменением угла установки лопаток $\Theta_{р.к}$ рабочего колеса. Область экономичной работы при регулировании изменением частоты вращения увеличивается почти в 2 раза по сравнению с регулированием остальными способами. Особенно важным в этом случае является то, что увеличение происходит за счет зоны с малыми производительностями и депрессиями. Расширение зоны экономичной работы позволяет с большей вероятностью обеспечить экономичность вентиляторной установки.

Определение экономичных рабочих режимов вентиляторной установки производится по области промышленного использования и характеристике сети (кривая 1), которые представляются графически и совмещаются на одном рисунке (рис. 11.3). Отрезок *aЬ,* лежащий в пределах области. промышленного использования, определяет не один, а множество режимов в зависимости от выбранного значения параметра регулирования. В приведенном примере области промышленного использования вентилятора ВЦД-47 регулируемым параметром является частота вращения рабочего колеса. Следовательно, экономичный режим работы вентилятора будет определяться выбранной частотой вращения ротора вентилятора в пределах отрезка характеристики сети *aЬ*.

Представляет интерес определение области промышленного использования при совместной параллельной и последовательной работе вентиляторов. Эта задача проще всего решается графическим методом.

Методика построения базируется на известных методах построения суммарных характеристик вентиляторов при последовательной и параллельной их работе. Ее нетрудно понять из приведенного рис. 11.4, на котором обозначены S_1 и S_2 – области промышленного использования соответственно вентиляторов ВОКР-1,8 при n = 1000 об/мин и ВЦ-25 при n =750 об/мин, взятых в качестве примера; S_3 и S_4 — области промышленного использования вентиляторов соответственно при последовательной и параллельной работе двух вентиляторов.

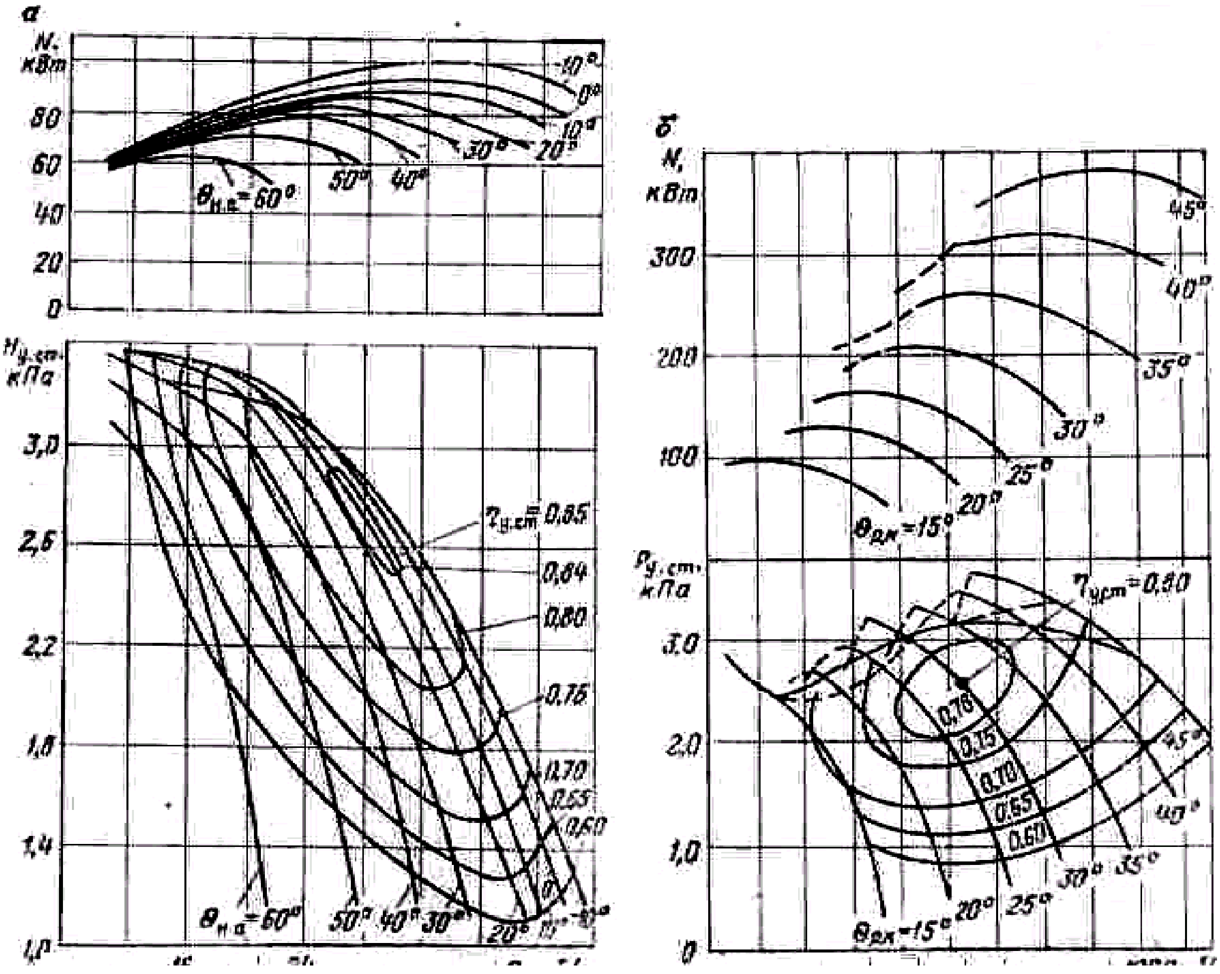


Fig. 11.3. Individual characteristics of the centrifugal fan VSHTS-16 (a) and axial fan VOD-21 (b)

To determine the contour of the area of industrial use of two fans connected in series, it is necessary to add up the ordinates of the points of the contours of the areas of each fan. For example, the ordinate of the point z of the S3 region is determined by the sum of the ordinates of points x and y, and points z1 — by the sum of the ordinates of points x1 and Y1, etc. At the same time, it is obvious that the addition of ordinates makes sense only for points with abscissa values from Q1 and Q2.

To determine the contour of the area of industrial use of two fans connected in parallel, it is necessary to add up the abscissae of the points of the contours of the areas of each fan.

Addition makes sense within the segment of the ordinate axis from p1 to p2 to the right of which the contour points of both regions S1 and S2 are located simultaneously.

If two fans with the same aerodynamic characteristics work together, then the area of their industrial use is a decrease of 2 times the scale, but the ordinate axis with sequential operation and a decrease of 2 times the scale along the abscissa axis with parallel operation of the fans. Operating modes are determined by plotting network characteristics on graphs.

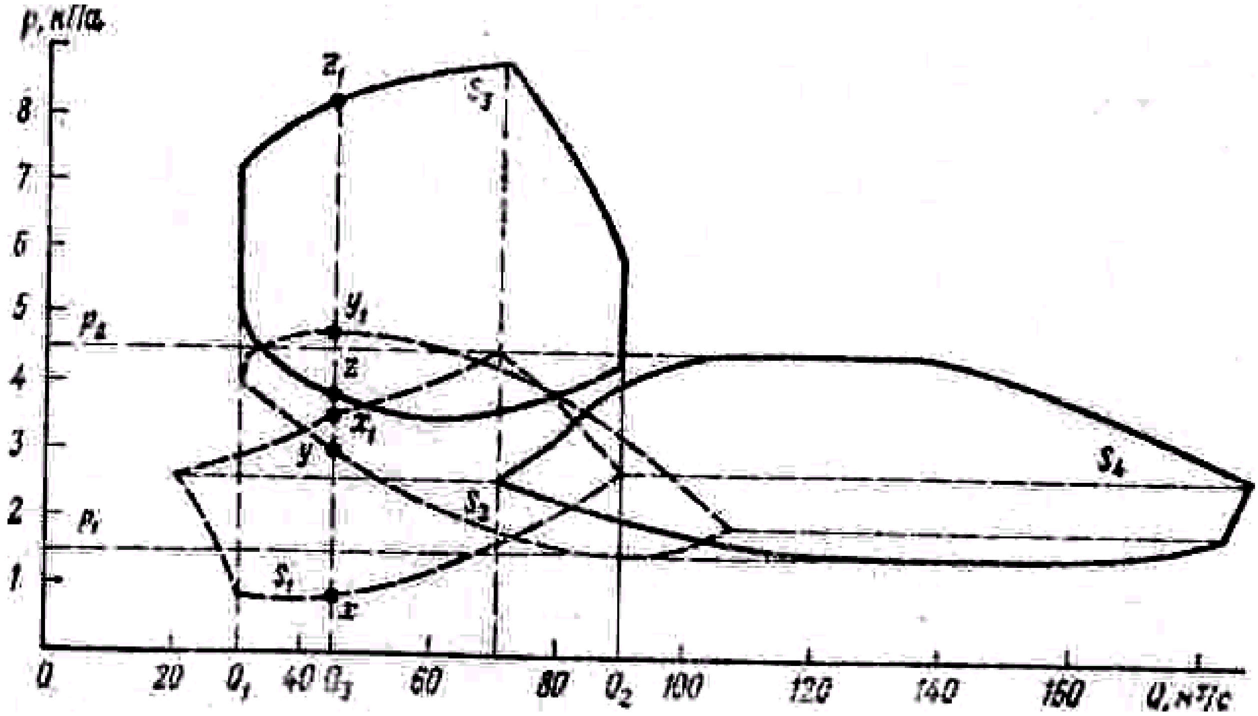


Figure 11.5. Definition of the area of industrial use of two fans in their sequential and parallel operation.

All network performance modes that fall within the scope of S3 or S4 industrial use are economical.

Determination of the weighted average efficiency In accordance with GOST 11004-75, one of the important parameters characterizing the efficiency of the operation of the main ventilation fan unit is the weighted average efficiency, determined in the so-called normal area, which is allocated in the field of industrial use of the fan unit. The right and left boundaries of the normal region are formed by two vertical lines drawn so that the minimum Qmin performance is 2 times less than the maximum Qmax between them there would be modes with the highest efficiency. From above, the normal area is limited by the pressure curve at the limiting angle of installation of the impeller blades or the guide device of this fan, and from below — by a line drawn through the points corresponding to 0.5 pmax. To simplify further calculations to determine the weighted average efficiency, the upper and lower boundary curves are replaced by broken lines with no more than two breaks.

Weighted average efficiency of the installation

$$\eta_{y.cm.cp} = \sum_{1}^{25}(Qp_{y.cm}) : \sum_{1}^{25}\left(\frac{Qp_{y.cm}}{\eta_{y.cm}}\right). \qquad (11.7)$$

LECTURE No. 12
SHAFT FANS
Plan:

1. Centrifugal fans
2. Classification of shaft fans
3. Purpose and nomenclature of shaft fans

1. Centrifugal fans. Classification. Shaft centrifugal fans are classified mainly according to the method of air supply to the impeller: with one-way (Fig. 12.1) and two-way suction.

Fans with two-way suction have higher performance compared to fans with one-way suction.

Aerodynamic schemes. Fans are developed on the basis of typical aerodynamic schemes (models of fans), worked out in laboratory conditions. The aerodynamic scheme of the fan is usually called the flow diagram of the fan with a set of basic dimensionless design parameters that determine the appropriate aerodynamic characteristics.

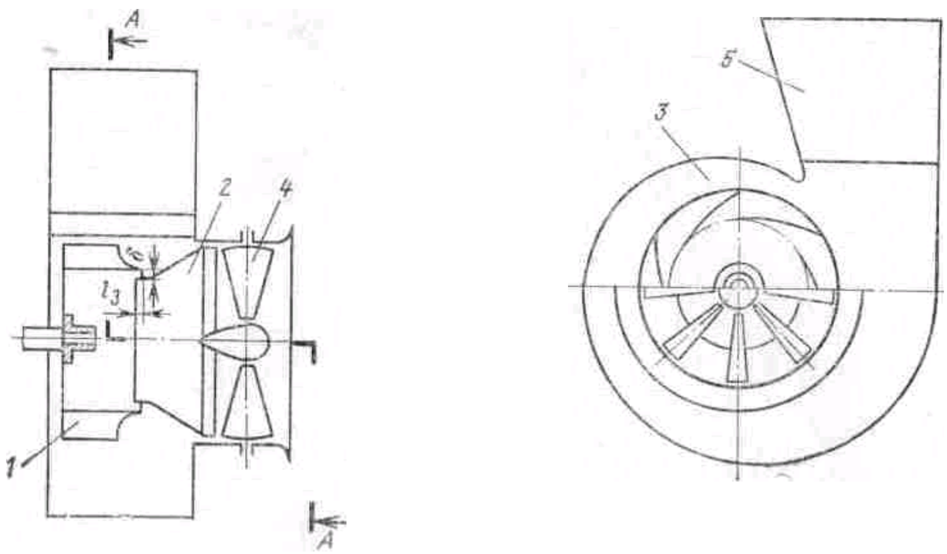


Fig. 12.1. Scheme of a centrifugal fan of one—way suction:
1— impeller; 2 — inlet pipe; 3 — spiral housing; 4 - guide device; 5 — diffuser

The fan is considered to be right or left rotation depending on the rotation of the impeller clockwise or counterclockwise when looking at the fan from the drive side.

Elements of fans. The process in the impeller is significantly influenced by the input elements: their profile, the diameter of the inlet, the size and configuration of the gap in the seal between the impeller 1 and the nozzle 2 (see Fig. 12.1), etc. The wrong choice of these elements causes distortion of the velocity field at the entrance to the wheel, separation of the flow from the surfaces of the flowing part of the wheel and a decrease in the fan efficiency as a result.

In shaft fans working for injection, cylindrical nozzles with a free inlet are used, in fans working for suction, smooth, conical and composite nozzles with a significant degree of confusability are used to reduce turbulence and equalize the air flow velocity. Loss of flow energy at the fan inlet:

$$\Delta p_{вх} = \xi \frac{c_1^2}{2}, \qquad (12.1)$$

where c1 is the speed in front of the impeller, m/ s; is the loss coefficient.

There is a gap b between the inlet pipe 2 and the front disc of the wheel 1 (see Figure 12.1), through which additional air flows into the impeller. With a rational choice of the seal shape, the axial overlap of the radial width of the gap, the main flow is pressed against the inner side of the disk and flows around it without separation; the optimal values of lz = 0.007 and = 0.002. Air leaks through the gap in the optimal operating mode of modern fans are 2-4% of the nominal capacity.

The impellers of the fans are made of single- and double-suction with profiled wing-shaped blades bent backward, ra < 90 ° (Fig. 12.2). The impellers of the old fans were equipped with structurally simpler flat blades 1 and 5-shaped blades 2, providing greater pressure compared to flat ones; efficiency of fans with both types the blades are lower than the efficiency of fans with wing-shaped blades.

To regulate the operating modes, the fans are equipped with axial guide devices 4 (see Fig. 12.1). These devices are equipped with 10 or 12 flat blades made in the form of a sector and rotating at fans of the VC and VCD types at an angle from -20 to +110 °. At angles from 0 to 90 °, the air in the guide device twists in the direction of rotation of the wheel, reducing the fan performance and power consumed by it. When the angle of installation of the blades is 90 °, the blades of the latter completely block the suction opening of the fan; such an installation of the blades is used when starting the fan. The angle of 0° corresponds to the installation of the blades in the plane of the axis of the fan shaft, and the twist of the flow is excluded. When the guide vanes are rotated at negative angles, the air in it is twisted against the direction of rotation of the impeller, which leads to an increase in fan performance and power consumption. At large negative angles of installation of the blades (-20 ° or more) the fan operation mode is characterized by flow separation from the blades at the inlet to the impeller and increased rotor vibrations.

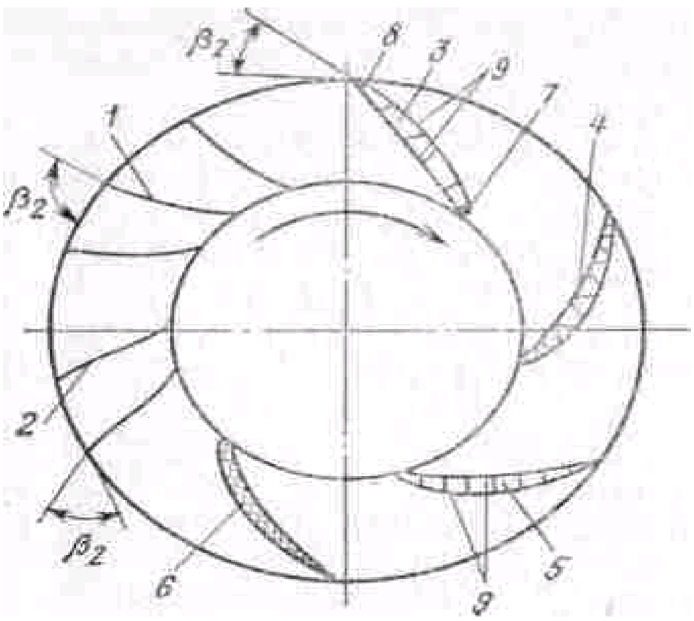


Fig. 12.2. Profiles of centrifugal fan blades:
1 — leaf; 2 — 3-shaped: 3 — wing-shaped with C- and U-shaped spars; 4 and 5 — with truss and honeycomb fins; 6 — three-layer; 7 and 8 — spout and tail blades; 9 — upper and lower shells

The output devices of centrifugal fans are, as a rule, spiral housings that form the flow and guide it from the output section of the wheel to the fan outlet, as well as partially converting the dynamic flow pressure into static.

Nomenclature and purpose. At mines and mines, centrifugal fans of the main ventilation of the following types are currently being operated: VC, VCD, VSHC, VRCD, VCP, VCZ and a local ventilation fan VC-7 (V—fan, C—centrifugal, O—one-way suction, D - two-way suction, R- mine, Sh-pit, P—tunneling, 3-with adjustable flaps). According to GOST 11004-75, centrifugal fans are marked: one-way suction VC and two-way suction VC. The number in the marking means the diameter of the impeller in decimeters.

Currently, the plants produce the following centrifugal fans designed for the main ventilation of mines and mines: VCD-16, VC-25, VC-31.5 (old code VC-32), VCZ-32, VCD-31.5 (old code VCD-32M) and VCD-47 "North" and for ventilation shafts and near-barrel workings: VC-11 and VSHC-16 and only for ventilation of the faces of mine shafts VCPD-8UM and VCP-16, as well as fans of local ventilation VC-7.

It is planned to produce VC-31.5P fans (with rotary blade flaps) instead of VCZ-32.

The VCD-47 "North" fan was created for the intended purpose for ventilation of deep polymetallic mines of the Far North, but can also be used for ventilation of the deepest coal and other mines.

Centrifugal fans are applicable for suction, discharge and combined ventilation schemes of mining enterprises. Large centrifugal fans are manufactured by Donetsk Machine—Building Plant named after LKU, fans with a diameter of the impeller up to 2.5 m inclusive are manufactured by Artemovsky Machine-Building Plant.

All centrifugal fans currently manufactured are developed on the basis of aerodynamic schemes of the M. M. Fedorov IGTMK, the Dongiprouglemash Institute and the SLE of the Artemovsky and Kamensky machine plants. Figure 12.3 shows a summary graph of the areas of industrial use of main ventilation fan units with centrifugal fans produced by factories. Fans are made of one-way and two-way suction. The impellers of the one-way suction fans have 8 blades, and the two-way suction has 16 (8 on each side), except for the VCD-47 fan, which has 12 blades.

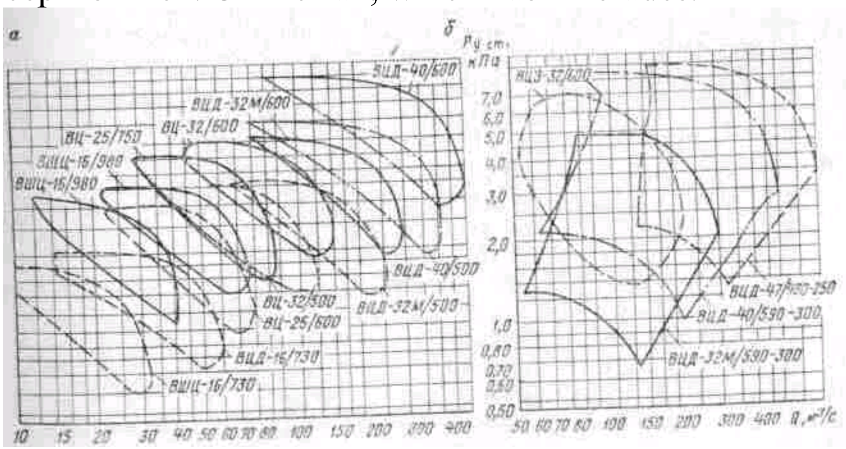


Fig. 12.3. Summary graph of the areas of industrial use of centrifugal fans of the main ventilation, regulated by axial guide devices (a) and with the help of rotary flaps and changes in the speed of rotation of the impeller (b)

The design of the fans. One-way suction fans have two layout schemes: with a cantilever arrangement of impellers on the rotor shaft (VTS-11, VSHTS-16, VTSP-16, VTS-25) and with the arrangement of impellers on the shaft between the bearing supports. The latter scheme is implemented in larger VC-31.5 fans and

VCZ-32 this arrangement reduces the load on the bearings and eliminates resonant transverse vibrations of the main shaft.

The centrifugal fan of one-way suction (VC-11, VC-16 (Fig. 12.4), VC-16, VC-25, VC-31.5 and VC-31.5P) consists of a rotor / with an impeller 2, an axial guide device 3, an inlet manifold 4 and a branch pipe 7, a spiral casing 5 and frames 6. The first three fans (with a diameter of the impeller up to 1600 mm) have a common welded frame on which the entire mechanical part of the fan and its drive motor are mounted. Larger fans do not have a single rigid frame, and the rotor shaft rotates in two separate bearing supports mounted on separate frames mounted on a concrete base.

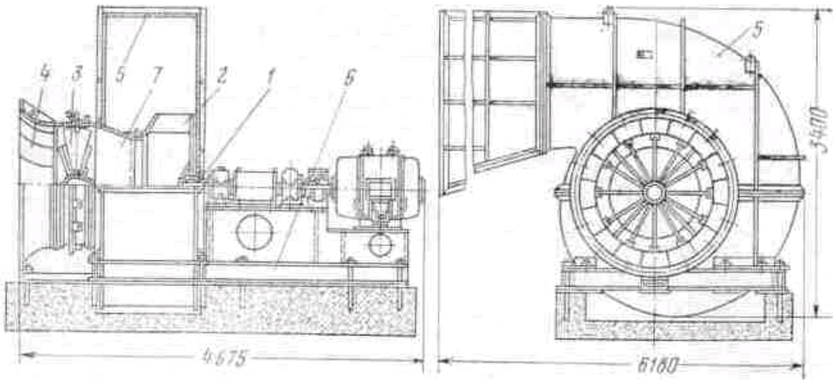


Fig. 12.4. One-way suction fan VSHTS-16

The impeller consists of wing-shaped backward-curved blades welded to the front conical cover and rear flat root discs. The rear wheel disc is welded or bolted to the hub. The cover disk on the diameter of the entrance can be reinforced with a toroidal cast labyrinth ring. The root disk of the impeller is reinforced by connecting it to the fairing, which simultaneously improves the conditions for turning the airflow in the impeller.

The blades are hollow, welded, with stiffening ribs inside.

The fans VC-31.5P and VCD-31.5P have blades equipped with rotary flaps. The internal space of these blades and the blades of some other fans, for example, VCD-47 "North", is filled with foam foam, foamed directly in the shells. This made it possible to abandon the connection of the shells with the spars and reduce the weight of the blades.

The impeller is fastened to the drive shaft by a keyway connection. The shafts rotate in radial double-row spherical and radial-thrust twin bearings cooled in large fans due to forced oil circulation. The rotor shaft is connected to the motor shaft by an elastic finger or, more often, a toothed coupling.

The stator part of the fan (see Fig. 12.4) consists of a spiral casing 5, an inlet manifold 4 and an inlet pipe 7, which with its narrow end enters the labyrinth seal of the impeller, forming an annular labyrinth gap with the latter. To increase rigidity, the casing and the box are equipped with fins closed in frames. The lower part of the spiral outlet of the VC-25 and larger fans is formed by a channel in concrete. The housings are made detachable, excluding the VC-11 fan housing.

The axial guiding device (Fig. 12.5) has flat leaf blades with a mechanism for their simultaneous rotation and a fairing located along the axis and suspended in the housing on extensions or mounted on the rotor shaft. The blades consist of a web and two trunnions welded to it and are supported by bronze bushings in the fairing and nylon or bronze bushings in the housing. On protruding from the housing of the guide device.

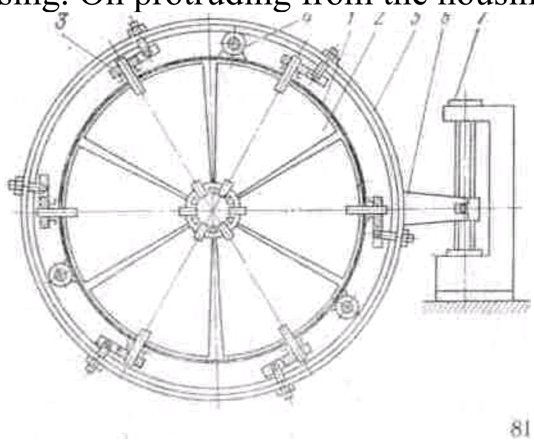


Figure 12.5. Guide device with external drive ring:
1-the body of the device; 2 - the blades; 3 - the lever of the blade; 4 - the roller; 5 — the drive ring;
6 — lever; 7 — drive column

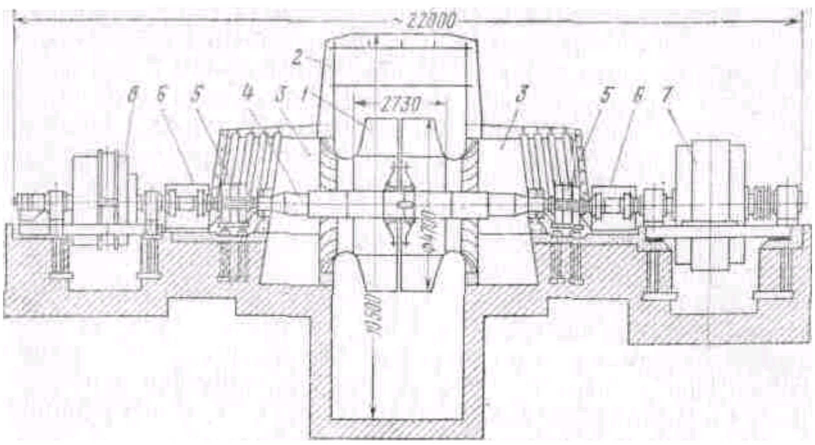


Fig. 6.6. VCD-47 "North" fan:
1 - impeller; 2 — casing; 3 — suction boxes; 4 — shaft; 5 — bearing supports; 6 — gear couplings; 7 and 8 — electric motors

Centrifugal fans of double suction (VCPD-8UM, VCD-16, VCD-32M, VCD-40 and VCD-47 "North") have a layout scheme with the placement of impellers on the drive shafts between their bearing supports. The design of these fans is in many ways similar to the design of the considered one-way suction fans. The main design features of double-suction fans are the presence, excluding the fans of the VCD-47 "North", two axial guide devices, a double-suction impeller for large fans (VCD-32M, VCD-40 and VCD-47) and the output end of the shaft for connection by means of a gear coupling of the second drive motor. Figure 12.6 shows the most powerful centrifugal fan VCD-47 "North" as an example. Its impeller consists of two half-wheels. This design of the wheel facilitates its transportation to the installation site of the fan. The half-wheels are connected to each other and to the hub rim with bolts.

The VCD-47 "Sever" fan provides high efficiency of operation in a wide range of ventilation modes that differ from the optimal pressure by 4 times, which is achieved by using a fan electric drive speed control system made according to the scheme of a combined asynchronous valve-machine cascade.

The two-way suction fans VCD-31.5 M, VCD-47 "North" have aerodynamic qualities that ensure the supply of large amounts of air at high pressures.

Centrifugal fans have become widespread in mining enterprises, which is explained by the significant improvement in their technical and economic indicators over the past 15-20 years. Thus, the maximum static efficiency of the main ventilation unit increased from 0.72 to 0.86, the weighted average static efficiency in the field of operation from 0.52 to 0.75; the dimensions and cost of fans decreased by 1.5—2.0 times.

LECTURE No. 13
AXIAL FANS
Plan:

1. Classification
2. Nomenclature and design of axial fans.
3. Nomenclature and design of local ventilation fans

1. Classification. Axial fans are classified into single-stage and multi-stage according to the number of working stages (the number of impellers). The first are created and used for local ventilation, the second — for the main ventilation. Multi-stage fans provide higher pressure at the same performance as single-stage fans. The number of steps in modern axial fans of the main ventilation is assumed to be equal to two.

Fan diagrams. Modern axial shaft fans are assembled according to the schemes of local ventilation ON + RK and ON + RK + SA and the main ventilation of RK + ON +RK-SA and RK +RK. Designated: RC - impeller, ON — guiding device, CA - straightening device. They are designed to give the air flow the necessary direction when entering the impeller, and between the two impellers of a multi—stage fan, they are also designed to untwist the flow; CA -to untwist the flow when exiting the wheel in the direction opposite to its rotation. The output angles of the blades ON between the impellers

and the blades of the CA can be set such that the flow will change its direction even to the opposite. The direction of the flows can be traced on the velocity plans, which indicate the absolute velocities at the inlet and outlet: and — the guiding apparatus, and — the straightening apparatus, and —the impeller. The two-stage RC+NA+RC+CA fan circuit will provide approximately 2 times more pressure than the RC + CA circuit, with the same performance. Total pressure generated by the axial fan,

$$p = p_{вых} - \rho \frac{c^2_{вых}}{2}, \qquad (13.1)$$

It is obvious that the static and dynamic components of the pressures will be:

$$p_{ст} = p_{вых} - p_B \text{ и } p_{дин} = \rho \frac{c^2_{вых}}{2} \quad (13.2)$$

Counter-rotating fans have two impellers I and II (RC + RC scheme), rotating towards each other without a guide and straightening apparatus. In the absence of a guide device, the fan inlet speed = 0, the absolute inlet speed c11 = ca. At the exit of the wheel I, the flow has a twisting speed and an absolute speed with 21. Rotating counter, the second impeller will spin the flow. With its full unwinding, si21 = 0 and c2 11 = ca11

Wheels in general can have different rotational speeds.

The pressure they develop is determined by the expressions for the first wheel

$$p_1 = \rho_1 c_{u2I} \qquad (13.3)$$

for the second wheel

$$p_{II} = \rho u_2 (c_{u2II} - c_{u2I}) \qquad (13.4)$$

By $u_I = -u_{II} = u$ and full unwinding of the flow, the total pressure developed by both wheels,

$$p = p_I + p_{II} = 2\rho u c_{u2I} \qquad (13.5)$$

Full unwinding of the flow is possible with an equal distribution of theoretical pressure between the impellers. This distribution of pressure between the wheels is the most advantageous, while the speed $\omega_{cpI} < \omega_{cpII}$, and the angle $\beta_{cp\,1} > \beta_{cp\,II}$ in this connection, the angle of installation of the blades of the first wheel is greater than the second.

With equal parameters with conventional two-stage fans, the counter-rotating fan has a smaller axial size and weight and greater reverse performance.

Fans with meridional flow acceleration (Fig. 13.1) allow to have a higher pressure value than conventional axial fans. The pressure of axial fans is limited by the diffusivity of the blades, which can lead to the separation of the flow from the blades. The diffusivity can be reduced by giving the flow a meridional acceleration by installing a conical or spherical impeller sleeve. At the same time, the axial flow velocity increases. In fans with significant meridional acceleration, axial velocities can be 1.5—2.0 times greater than at the inlet, which leads to large dynamic pressures in front of the diffuser.

Figure 13.1 shows the velocity triangles for an ordinary axial wheel (thin lines) and for a wheel with meridional acceleration (thick lines) for two cases; the relative velocity of the meridional wheel is less than the relative velocity and greater than the relative velocity of the ordinary axial wheel ( ) and the relative velocity of the meridional wheel is greater than the velocity . The flow velocities c1 at the entrance to the meridional impeller in both cases and in a conventional axial wheel are assumed to be the same.

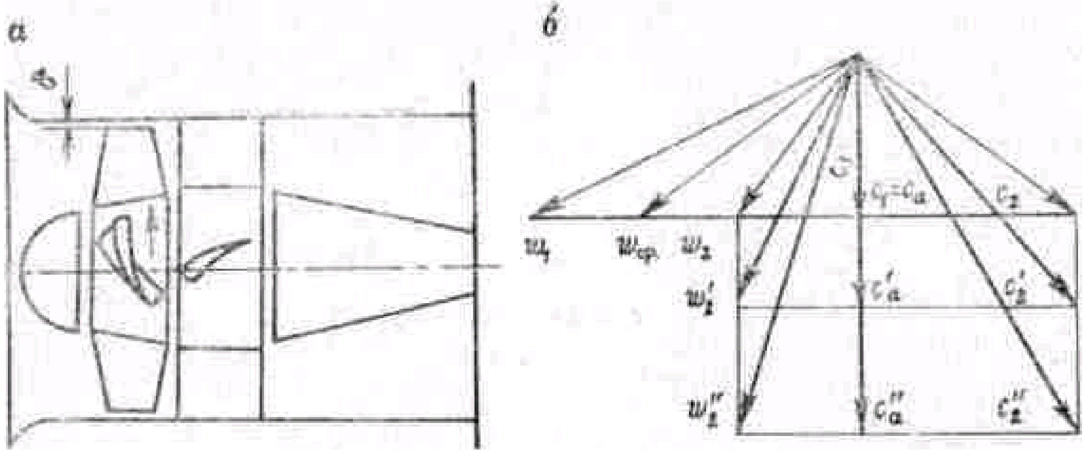


Fig. 13.1. Diagram (a) and speed plans (b) of a fan with a meridional by accelerating the flow

It can be seen from the velocity triangles that the absolute flow velocities c2 and c'1 at the outlet of the meridional wheels are much larger in absolute magnitude and, consequently, the dynamic pressure in the meridional wheel increases significantly. As the relative speeds increase, the static head decreases, but the total theoretical head will exceed the static head of a conventional axial wheel. At the same time, the increment of static pressure practically does not occur, only its dynamic component increases. At the same time, the losses caused by the gap b between the wheel blades and the fan casing practically disappear, as a result of which the efficiency of the fan is noticeably improved. However, the air flow receives energy in the wheel in the form of dynamic pressure, which turns into static in the CA and the diffuser, which somewhat worsens the efficiency. Meridional single—stage fans develop greater pressure than conventional axial fans with a sufficiently high efficiency (h,; t.t. = 0.87-0.89), and can be used instead of two-stage axial fans. Their characteristics are flatter, the area of unstable operation is smaller and the level of noise generated by them is lower. These advantages of fans with meridional acceleration determine the prospect of their application. In recent years, such fans have been widely used type A

Elements of fans. The collector 4 and the fairing 5 are installed at the entrance to the first blade unit and are designed to smoothly increase the axial velocity of the c1 flow without large losses and ensure a uniform velocity field. The collector can be outlined along the arc of a circle with radius g > 0,2D2, where D2 is the diameter of the impeller. The fairing has the shape of a hemisphere or a semi-ellipsoid. The impellers of axial fans are equipped with profiled blades — conventional and twisted. The shape of the blades is chosen such that the necessary circulation is provided around them to create sufficient pressure and at the same time the flow of air particles in the radial direction to the periphery would be excluded due to the action of centrifugal forces from the twisting of the flow. In fact, it can be seen from L. Euler's equation that the transfer of energy to the flow by the blades is inevitably associated with the twisting of the flow. Centrifugal forces arise in the swirling flow, which can cause air to flow from one annular layer to another to the periphery of the fan. Such overflowing is undesirable, as it causes significant energy losses. It is obvious that there will be no flow of particles if the pressures along the radii are constant (p=const) or according to the L. Euler equation.

$$T = const$$

$$\text{или} \qquad (13.6)$$

$$rc_{u2} = const$$

It follows from this condition that in order to ensure the constancy of the pressure, the twisting speed must change according to the hyperbolic law:

$$c_{u2} = \frac{const}{r} \qquad (13.7)$$

Such a change in speed along the radius of the impeller is achieved by the design of blades with variable widths b and angles . In this regard, modern axial fans use twisted blades with variable cross-section along the length. The velocity field behind the impeller with uncoiled blades has a significant unevenness. The gap 6 between the ends of the blades and the fan casing has a great influence on the operation of the fan.

Relative clearance (the gap related to the length of the blade 1л) $\bar{\delta} = \frac{\delta}{l_л} 100\%$ for high—quality domestic fans, it is 0.8-1.0%. With a relative clearance of only 1.5%, the pressure decreases by 15-20% compared to the calculated value at zero clearance. The decrease in pressure is explained by the reverse flow of air through the annular gap from the area of increased to the area of reduced pressure.

The guiding and straightening apparatus are crowns of fixed or rotary blades. The main parameters are the density of the lattice b / t, the angle of installation and the aerodynamic characteristics of the profile of the blades. For radial equilibrium of the flow, the distribution of the twisting velocities of the flow in the apparatuses should be the same as in the impellers. To work in a curved flow, the profiles of the blades CA and NA must be deformed according to the curvature of the flow. In order to avoid energy losses, the blades should be installed so that the flow enters their input edges tangentially or at a slight angle. In order for this to be observed under different operating modes of the fans, the blades ON and CA should be rotated.

The diffuser and the outlet part are designed to convert the dynamic rdin into static pct pressure. The shape of the diffuser, common in shaft axial fans, is shown in Fig. 12.7, a. It is advisable to provide a small cylindrical section between the straightening apparatus and the diffuser to equalize the flow velocity field. The transition from a cylindrical to a conical section should be smooth.

The most important parameters of diffusers are: the degree of expansion of the diffuser

$$\chi = \frac{F_2}{F_1} = \frac{c_{a1}}{c_{a2}} \quad (13.8)$$

F_1 и F_2 — cross-sectional areas, м^2

c_{a1} и c_{a2} — flow rates at the inlet and outlet of the diffuser;

opening angles of cones φ_1 и φ_2 and where D and the relative length of the diffuser $\overline{L} = L / D$ initial diameter of the diffuser.

Shaft fans have a conical part with a length of $\overline{L} = 2$ and the opening angles of the cones $\varphi_1 = 3^0$, $\varphi_1 = 6°$.

2. Nomenclature and design of axial fans. Nomenclature and purpose. In the mining industry of the USSR, axial fans of the following types are operated: VOD, VOK, VOKD, VOKR (V—fan, O—axial, K—with twisted blades, D—multistage, P—reversible. According to GOST 11004-75, axial fans are marked: single—stage — IN and multi-stage - IN. The first domestic axial fans were produced on the basis of the B-series (high-pressure) fans developed at TsAGI in 1938-1939. Due to various shortcomings, these fans were discontinued in 1957. They were replaced by fans of the VOC and VOC types with twisted blades, which were distinguished by a more perfect aerodynamic scheme and smoother performance control, and fans of the VOC type and the possibility of reversing the air jet by the fan itself.

Currently, recently developed fans of the VOD type (VOD-11, VOD-16, VOD-21, VOD-30, VOD-40 and VOD-50) are being produced (the figure in the fan marking is the diameter of the impeller at the ends of the blades in decimeters). Fans are manufactured by Artemovsky Machine-building Plant. Axial fans of the WATER type are intended for ventilation of shallow mines and mines, the total mine depression of which does not exceed 4 kPa. The VOD-11 fan is also used as an auxiliary fan for ventilation of trunks and near-barrel workings during their construction, in air-conditioning installations, etc.

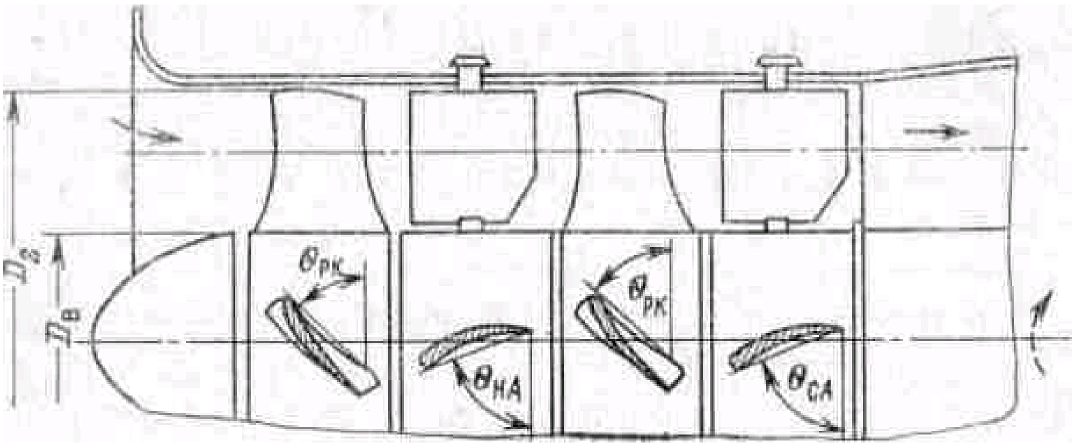


Fig. 13.2. Diagram of the axial fan of the N. E. Zhukovsky type of water

Fans of the VOD type, with the exception of VOD-11, are reversible and provide the performance required by safety regulations with a reverse of more than 60% of the normal performance; on special order, they can also be performed non-reversible. These fans are applicable for both suction and discharge ventilation. The VOD-21, VOD-30, VOD-40 and VOD-50 fans were developed by Dongiprouglemash according to the K-84 aerodynamic scheme (K - twisted blades, 84 — speed coefficient) of TsAGI named after N. E. Zhukovsky and Dongiprouglemash. The VOD-16 fan was created by the Artemovsky Machine-Building Plant according to the aerodynamic scheme of the M-1 IGMTC named after M. M. Fedorov and TsAGI VOD, has counter rotation of the impellers. WATER type fans (Fig. 13.2) are two—stage. To reduce noise, they do not have input guide devices and are assembled according to two conditional schemes RK + RK (VOD-11 and VOD-16 fans) and RK + NA + RK + SA (other types of WATER fans).

The fan impellers are equipped with 12 blades, with the exception of the impeller of the second stage of the VOD-16 fan, which has 10 blades. Installation of RC fan blades, with the exception of VOD-16, is carried out within 15-45 °. When the air jet is reversed, the blades ON and CA rotate at angles of 153 and 158 ° and the direction of their convexity changes to the opposite.

Summary graphs of the areas of industrial use of axial fans of the WATER type and the field of ventilation modes of mines and mines covered by them are shown in Fig. 13.3.

The design of axial fans. The VOD-21, VOD-30, VOD-40 and VOD-50 fans have a similar design.

The fan (Fig. 13.4) consists of a rotor with impellers and 2 (I and II stages), a housing (casing) with an intermediate guide 3 and a straightening 5 apparatus and rotation mechanisms 4 and 6 blades, a front fairing 7, a main shaft 8, a transmission shaft 12, a collector 9, a diffuser 10 and a brake 11. The fan rotor is connected to the electric motor 13 by means of a transmission shaft 12 and couplings 14.

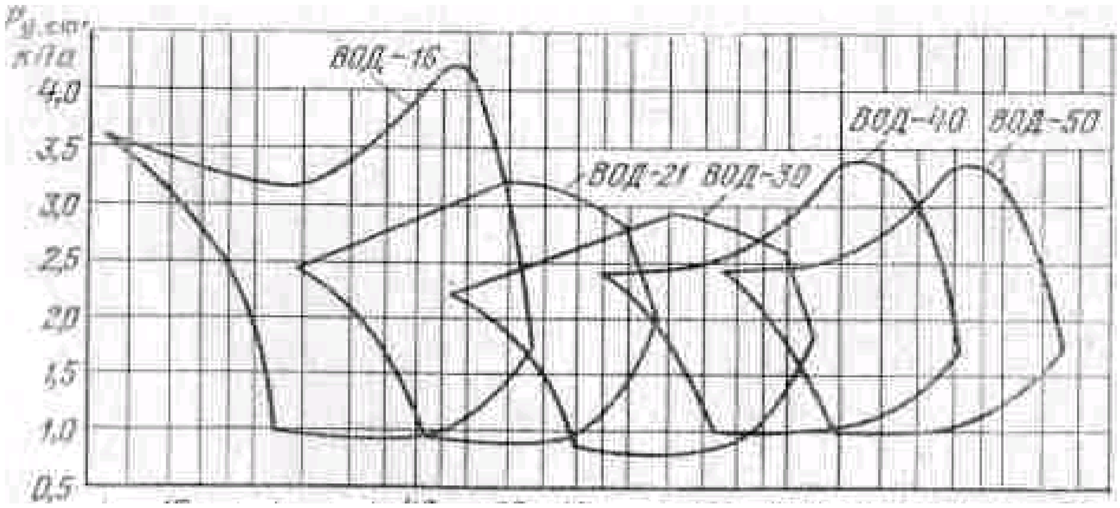


Fig. 13.3 Summary graph of areas of industrial use of axial fans of the WATER type

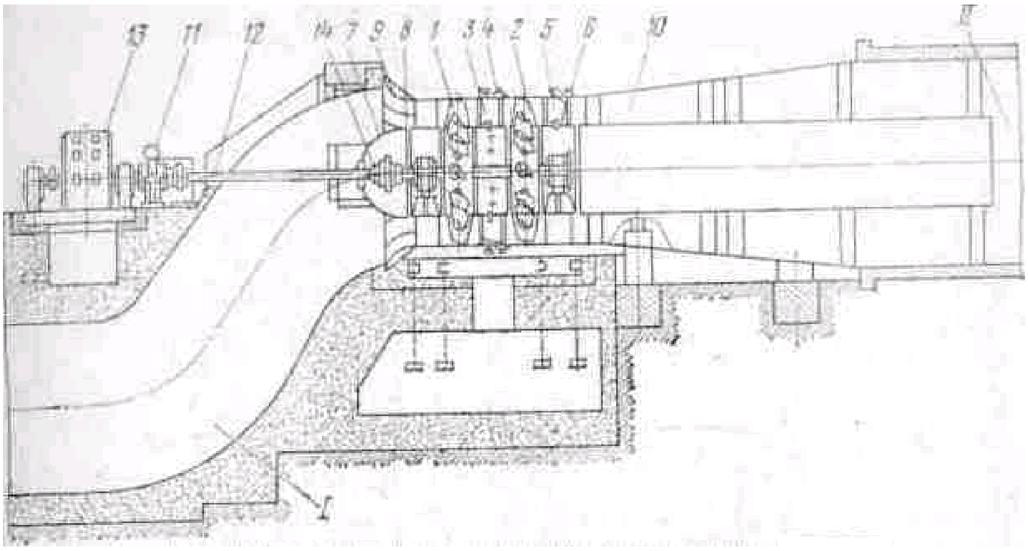


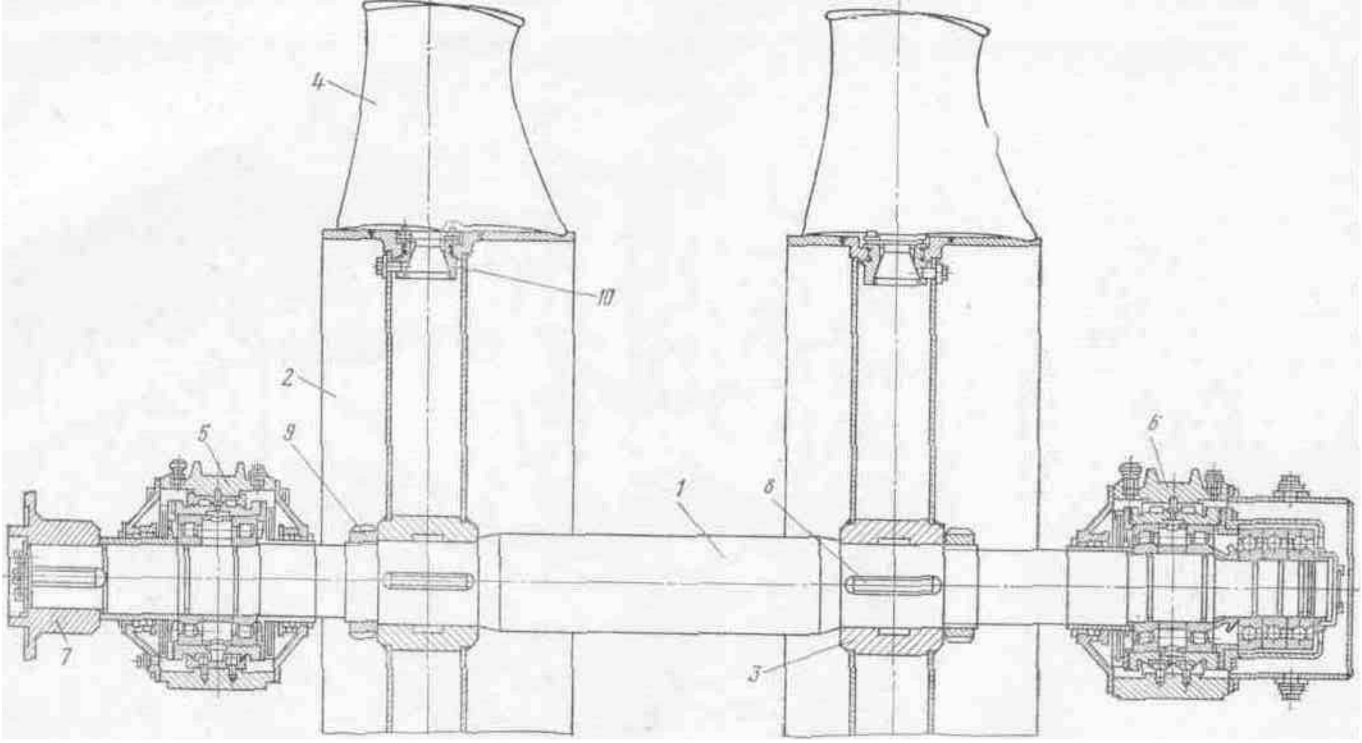
Figure 13.4 . General view of the axial fan VOD-30

With suction ventilation of a mine or mine, air enters the fan from the ventilation network through the supply channel I and exits through the diffuser into the outlet channel II. In the diffuser, the dynamic pressure is partially converted into static pressure.

The blades 4 (Fig. 13.4) of the impellers of WATER—type fans are profiled, screw-shaped, welded-riveted, hollow, consist of a bearing shank, two sheets of sheathing, reinforcing ribs and bottoms, are made of steel or of a lighter material — aluminum alloy or fiberglass.

The blades are attached to the bushings of 2 impellers by special gates 10, which allow them to be turned manually within the installation angles of 15-45 ° when the fan is stopped to regulate performance and pressure. The fan rotor rotates in two bearing units — front and rear. Radial load is perceived by paired bearings with cylindrical rollers mounted in spherical clips (fans VOD-30, VOD-40, VOD-50), or spherical roller bearings (fans VOD-21, VOD-11) mounted in both supports. Axial load is perceived by angular contact ball bearings placed in the rear support.

The lubrication of the bearings of the VOD-21, VOD-16 and VOD-11 fans is consistent, refractory, periodically replenished with the help of oil presses placed on the upper part of the housing. Lubrication of the bearings of the fans VOD-21, VOD-30, VOD-40 and VOD-50 is forced, circulating. The oil is supplied from the oil station.



The shafts of the rotor and the electric motor are connected by an outboard transmission shaft with the help of single-sided gear fans VOD-30, VOD-40 and VOD-50 and with the help of. finger couplings for fans VOD-11, VOD-16 and VOD-21.

The WATER type fan is equipped with a pad brake with an electromagnetic drive that ensures the rotor stops for 2-2.5 minutes.

The housing of the VOD-30 fans. VOD-40 and VOD-50 consists of front and rear support blocks and a casing.

The WATER-type fan housing and the supporting structures placed in it, the collector, the outer and inner cones of the diffuser are welded from sheet steel and long products. INSIDE the rear support block of the housing of the VOD-1, VOD-30, VOD-40 and VOD-50 fans, there is a straightening apparatus with 11 rotary and 3 fixed bearing blades. In the casing of these fans, located between the front and rear support blocks, an intermediate guide device with rotary blades is built in, which are rotated by a special mechanism from the servo motor at an angle of up to 180 when reversing the air jet and at an angle of up to 36 ° from the starting position for fine performance control. The angles of rotation of the blades are limited by limit switches.

The mechanism of simultaneous rotation of the guide vanes is similar in principle to the mechanism shown in Fig. 6.5. It is equipped with a rotating ring covering the fan casing, which turns the rollers with the guide vanes fixed on them with the help of ropes.

The performance and pressure of WATER-type fans are regulated by turning the blades of the impellers manually when the drive is stopped; fine adjustment within 5-10 is carried out by turning the blades of the guide device.

To work with less pressure, you can reduce the number of blades of the impeller of the II stage to six.

Fans VOD-21, VOD-30, VOD-40 and VOD-50 are reversed by changing the direction of rotation of the drive motor with simultaneous rotation of the blades of the guide and straightening apparatus.

3. Nomenclature and design of local ventilation fans Purpose and nomenclature. Axial and less often centrifugal fans of local ventilation are used to ventilate dead-end workings in coal and ore mines. A wide range of modes and conditions for the ventilation of these workings required the creation of a number of axial fans with electric and pneumatic drives. Currently, a number of axial fans of local ventilation with meridional acceleration of the flow of types VM-M and VMP-M, as well as fans VKM-200A and VMP-4 are being mass-produced. In the designation of the brand of fans: B—fan, M—local, P — pneumatic, the numbers mean the diameter of the outlet pipe in decimeters and the model number, M — after the numbers means modernization.

Electric axial fans have explosion-proof RV design and can be used in mines dangerous for gas or dust. Pneumatic fans are used to ventilate dead-end workings where the use of electric fans is not allowed by the PB. Since the cost of pneumatic energy for fans of the same hydraulic power exceeds 6-7 times the cost of electricity, it is advisable to use pneumatic fans only in conditions in which electric fans cannot be used.

Electric fans of the VM-M type are performed according to the NA + RK + SA scheme; pneumatic fans of the VMP-M type are performed according to the NA + RK scheme. The guiding devices (ON) of the VM-ZM and VM-4M and VMP-M fans are made unregulated, the VM-5M, VM-6M, VM-8M and VM-12 fans are adjustable; the output straightening devices (CA) of electric fans are unregulated. The currently manufactured local ventilation fans are single-stage and, with the exception of VMP-4 fans, have impellers with a conical sleeve that provides meridional acceleration of the air flow. In this case, the vast majority of the energy supplied to the impeller is converted into the high-speed flow pressure.

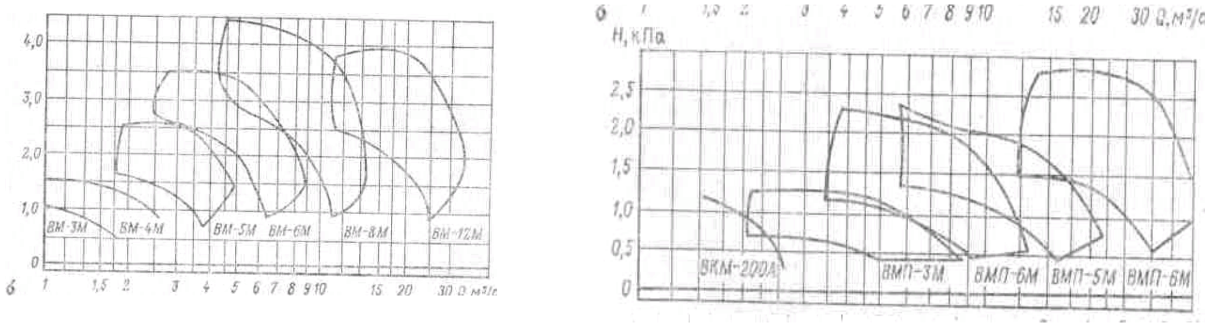


Fig. 13.6. Summary graphs of areas of industrial use of axial fans of local ventilation with electric (a) and pneumatic (b) drives

Figure 13.6 shows summary graphs of the areas of industrial use of local ventilation fans.

The design of the fans. Electric fans of the VM type (Fig. 13.7) have the same design scheme and differ in the size and design of individual units. The main nodes of the fans are the input guide device 1, the impeller 2, the straightening device 3, the built-in electric motor 4, the cable entry 5 and the slide 6.

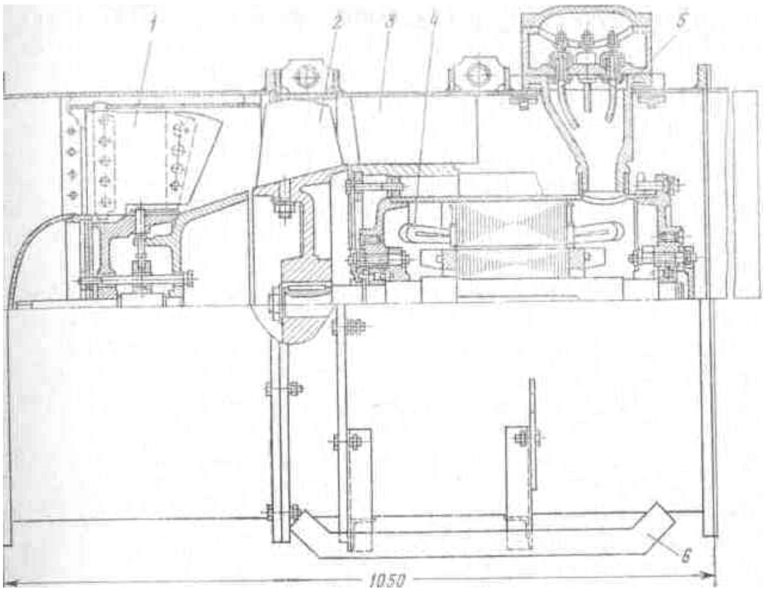


Fig. 13.7. VM-6M electric fan

The guiding device of the VM-ZL1 and VL1-4M fans is equipped with twisted steel sheet blades rigidly fixed between the outer and inner shells. The adjustable guide devices of the remaining fans of this type are equipped with profile elastic blades made of rubber with reinforced steel plates with inlet and outlet edges.

The pressure and performance of the fan are regulated on the move by means of a simultaneous rotation mechanism that provides a smooth stepless deflection of the blade flaps at an angle from 45° to 50°. An additional shell is installed in the guide unit, forming an annular channel of the air separator with the body, eliminating depressions on the pressure characteristics and surging modes of operation. This significantly expands the scope of industrial use of the fan and increases the reliability of its operation.

The fan impeller consists of a conical steel sleeve, cast or welded (VM-12L1), and profile twisted blades fixed to it. The blades of all VM-M fans are made of nylon resin VSTU 73-1008-63, with the exception of the blades of VM-12L1 fans, in which they are made of thick-sheet steel. Nylon blades do not accumulate dangerous static charges and if they are touched by the body, no sparks are formed that can ignite an explosive environment.

The fan housing, the bushing installed in it and the blades of the straightening apparatus welded between them form a rigid supporting structure into which the electric motor is built.

The VM-12M fan is additionally equipped with a diffuser.

The pneumatic fans VMA-ZM, VMP-5M, VMP-6M and VKM-200A have a similar design and differ in the size of the nodes and aerodynamic parameters. Fans of this type are equipped with a pneumatic turbine drive. The impeller on the periphery has a turbine — rim with profile nylon blades installed in the groove of the rim. The air is supplied to the blades by an adjustable nozzle device, which provides compressed air to the turbine through one, two or three nozzles, which corresponds to the operation of the fan at reduced, normal and enhanced modes.

The pneumatic fan drive has a small mass, is economical, simple and easy to maintain.

Axial fans of local ventilation for partial noise reduction can be equipped with noise silencers of the GSH type developed by Dongiprouglemash.

Local ventilation fans and noise mufflers of the GSH type are manufactured by the Tomsk Electromechanical Plant named after V. V. Vakhrushev, Artemovsky Machine Plant, etc.

LECTURE No. 14
MAIN VENTILATION FAN INSTALLATIONS

Plan:

1. Requirements for installations
2. Electric drive and automation equipment of fan installations
3. Installation diagrams

1. Requirements for installations. Fan installations in accordance with the PB should have a performance reserve of 20 to 45% and ensure the reversal of the air flow in no more than 10 minutes, while the performance should be at least 60% of normal.

Safety regulations also require that modern installations be equipped with two identical fans—working and backup.

Installations must be reliable and low-noise, have high efficiency and the ability to adjust parameters within wide limits. In order to avoid excessive energy losses by the air flow, they must be equipped with input and output elements (supply channels, tees, input boxes, diffusers) with a small hydraulic resistance.

Fan installations should have as small dimensions as possible in plan.

To work in harsh climatic conditions, the fan installation scheme must exclude freezing of switching devices, backup fan and other elements.

The efficiency and reliability of the main ventilation fan unit largely depend on its layout scheme and auxiliary equipment. The fewer ventilation ducts and auxiliary equipment (windows or doors), the fewer air leaks and leaks and the more reliable the operation of the fan unit. The layout scheme should be chosen so that air leaks and its suction are minimal. According to the standards, they should not exceed 10%.

The method of reversing the jet is determined by the properties of the fans. In fan installations equipped with centrifugal fans, the reversal of which is impossible, the direction of the air jet is changed by a system of LED or vertical doors and bypass channels. This leads to a complication of installations with centrifugal fans compared to installations equipped with reversible axial fans.

2. Electric drive and automation equipment of fan installations. Electric fan drive. In most cases, the main ventilation fans are equipped with an unregulated electric drive with asynchronous or synchronous electric motors.

Depending on the power, various types of electric motors are used in electric fan drives: at power consumption up to 100-150 kW — low—voltage asynchronous motors with a short—circuited rotor; at power from 150 to 350 kW - low-voltage synchronous motors with a voltage of 380 V; at power over 350 kW - high-voltage synchronous motors (voltage 6 kV) SD, SDS and SDV and high-voltage asynchronous with a phase rotor of the AKN and AKS series with a power from 400 to 1000 kW at a voltage of 3000/6000 V and a rotation speed from 290 to 490 rpm.

Fan installations with an adjustable drive are equipped with: with stepwise regulation—multi—speed asynchronous electric motors, units of two asynchronous motors or synchronous and asynchronous motors at different speeds; with stepless regulation—DC motors controlled by the G—D (generator—motor) and UPV-D (controlled rectifier—motor) systems; cascade units of three-phase current and other systems.

With a small control range and short-term use of a reduced rotational speed, asynchronous electric motors with resistances in the rotor circuit are used in fan installations.

Technological control equipment. The condition of the fan unit is characterized by a large number of parameters:

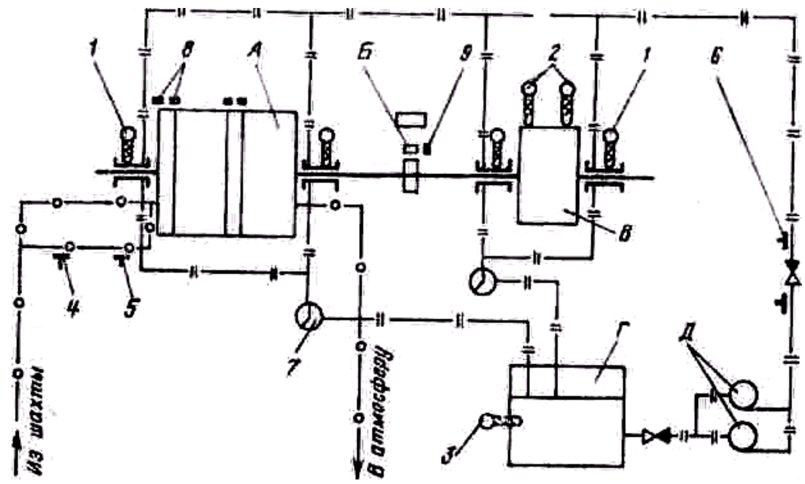


Fig. 14.1. Layout of the sensors for monitoring the fan installation:
1 — bearing temperature; 2 — temperature of the electric motor windings; 3 — oil temperature; 4; 5 — pressure and air flow; 6 — oil pressure; 7 — oil flow; 8 — positions of the guiding and straightening apparatus of the fan; 9 —brake position; A — fan; B — brake; C — electric motor; D — oil tank; D — oil pumps.

Air pressure and performance, the temperature of the fan bearings and its drive motors, the temperature of the windings and oil in the bearing bearing lubrication system, oil pressure and flow, etc. An approximate layout of the control sensors on the fan unit is shown in Fig. 14.1.

In accordance with the Safety Rules in coal and shale mines, during the operation of the main fan installations, their performance and pressure must be measured during direct and reverse operation. To do this, the main fan installations of non-gas mines are equipped with depressionometers, category III mines, ultra-large and hazardous in emissions with a self-recording depressionometer and a flow meter.

The operation of the fan unit without a driver, regardless of the gas mine category, is allowed only when the installation is equipped with two self-recording devices (in the engine room and in the control room) that constantly record the fan performance and the pressure created by it, and devices that signal to the remote control about deviations of the fan unit from the specified parameters for performance and pressure.

To monitor pressure and performance, equipment is used, including sensors, primary and secondary devices.

Performance and pressure sensors are installed in the control section of the fan unit and are intended to receive pneumometric pulses, the value of which varies in proportion to the measured pressure (pressure sensors) or in proportion to the square of the air flow velocity (performance sensors). The pneumometric pulse, which is a pressure drop, is supplied to the primary device by pneumatic pulse tubes.

The primary device is intended for direct measurement of air flow or pressure and conversion of their values into a proportional electrical signal with its subsequent transmission to the secondary device. When installing two secondary devices (one in the building of the fan installation and the second on the dispatcher's console), an electrical signal to the second secondary device is supplied from the output converter of the first secondary device.

Differential pressure gauges are used as primary instruments for measuring pressure and air flow. Currently, membrane and bellows diffmanometers are mainly used. Membrane diffmanometers with an induction converter DMI are available in two modifications: flowmeters DMI-R and flowmeters (pressure gauges) DMI-T.

As secondary devices, devices showing both the DS-1, DSRF, DSMR2 and DSM2 recorders and the SD-2 and KSD-R recorders are used. They are designed to measure and record air performance and pressure.

Complete automation equipment. Complete automation equipment is available for automation of the main ventilation fan installations. Modern automation equipment ensures the operation of the fan unit without the constant presence of maintenance personnel and the possibility of operational regulation of the fan operation mode to maintain optimal ventilation parameters.

The automation equipment of the main ventilation fans of the CC is operated at the existing mines.VG and ERVGP-2, as well as equipment for automation of ADSHV pit fans. The sets of equipment provide automatic control and monitoring of the state of the fan unit from the engine room from the control panel.

This equipment has been discontinued and is being replaced by a more advanced standardized equipment for the automation of shaft fans UKAV-2, which is mass-produced by the KHEMZ plant.

The UKAV-2 equipment is intended for automation of installations equipped with one or two axial (non-reversible or reversible) or centrifugal fans (one-way or two-way suction) electrically driven by synchronous (asynchronous) high and low voltage motors, as well as a two-motor reversible electric drive of counter-rotating fans.

The expediency of using a particular electric drive system and automation equipment is determined by their technical and economic indicators, of which the main ones are economic efficiency and payback of additional capital costs.

3. Installation diagrams. Installations with centrifugal fans. The technological scheme of the installation with two fans of the VC-25 or VC-31.5 type is shown in Fig. 14.2 a. When the fan is running / in suction mode, the air from the shaft is ejected through the diffuser into the atmosphere. To reverse the air flow with this fan, the diffuser and the common supply channel are blocked by the 5 and 3, respectively, and the bypass channel is connected to the shaft. At the same time, fresh air is taken from the atmospheric booth, in which the lyada 4 opens, and is pumped into the shaft through the bypass channel by the fan 1. To operate the fan 2, the LED 6 rises and opens the supply channel of this fan, and the led 7 descends, blocking the fan channel 1. The positions of the remaining led in the suction and discharge mode correspond to their positions when operating in these fan modes 1. Electric winches 8 serve to move the led, the position of the LED is controlled by limit switches 9. The technological schemes of fan installations with fans VCD-2M, VCD-40 and VRCD-4,5 are similar and differ only in the type and number of drives.

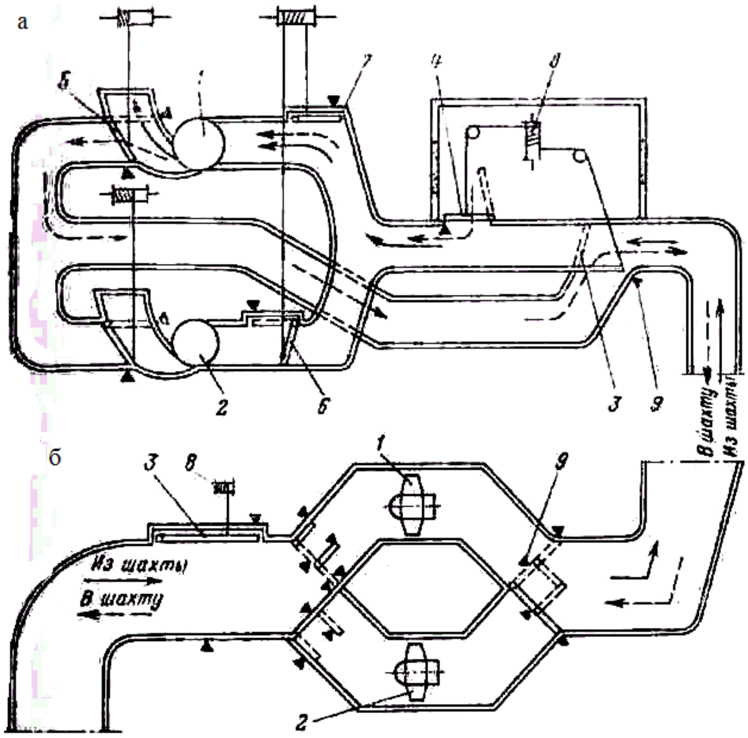


Fig. 14.2. Flow diagram of VC-25 and VC-32 fans of fan installations with centrifugal (a) and axial WATER type fan (b):
1 and 2 — fans; 3, 4, 5, 6, 7 — lyady; 8 — winch; 9 — limit switches

When working on suction, the lade 5 of the working fan 1 is in the upper position, and the similar lade of the backup fan 2 is lowered and blocks the supply channel. The lade 5 of the atmospheric booth and the cutting-off lade 4 overlap the openings connecting the suction side of the fan with the atmosphere, and the discharge side with the shaft of the mine. The LED 6 of the diffuser is lowered and covers the opening of the bypass channel. When reversing the air flow, the rows 4 and 5 switch and open the openings, and the row 6 block the output channel of the diffuser

At the same time, the air sucked in by the fan passes through the atmospheric booth, the opening of the lade 6, the bypass channel, the opening of the lade 4 and is pumped into the shaft of the mine.

Installations with axial fans. The main ventilation shaft installation with WATER type fans consists of two identical fans connected to the main ventilation duct through branching supply channels, in which devices are installed to disconnect the idle (backup) fan from the ventilation network (Fig. 14.2, b). When the fan 1 is operating in suction mode, air is sucked out of the shaft and ejected through the diffuser into the atmosphere. In the case of reversing the airflow by this fan by changing the direction of rotation of the drive motor, all the blades occupy the same position as in normal mode. The air is sucked through the diffuser from the atmosphere and pumped into the shaft. In case of switching to ventilation by fan 2 in any of the suction or discharge modes, the pads are transferred to the positions shown by dotted lines. Lyada 3 always occupies the upper position during operation of any of the fans and descends only when both fans are stopped or during fire-fighting measures, blocking the common channel and thereby excluding the flow of air from the mine or into the mine due to natural traction.

The technological schemes of fan installations for all fans of the WATER type are practically similar and differ only in the number and types of lads.

When using reversible fans of the WATER type, the bypass channel, the atmospheric booth and the corresponding auxiliary equipment are excluded, which significantly simplifies the technological scheme, reduces air leaks and increases the reliability and efficiency of such fan installations.

A general view of the typical layout of a shaft fan installation equipped with VOD-21, VOD-30, VOD-40 and VOD-50 fans is shown in Fig. 14.3. The installation consists of two fans 1 (working and backup) with drive electric motors 2, auxiliary equipment 3 for switching the air jet from the working fan to the backup, noise muffler 5, unified sets of automation equipment 4 and lubrication system 6.

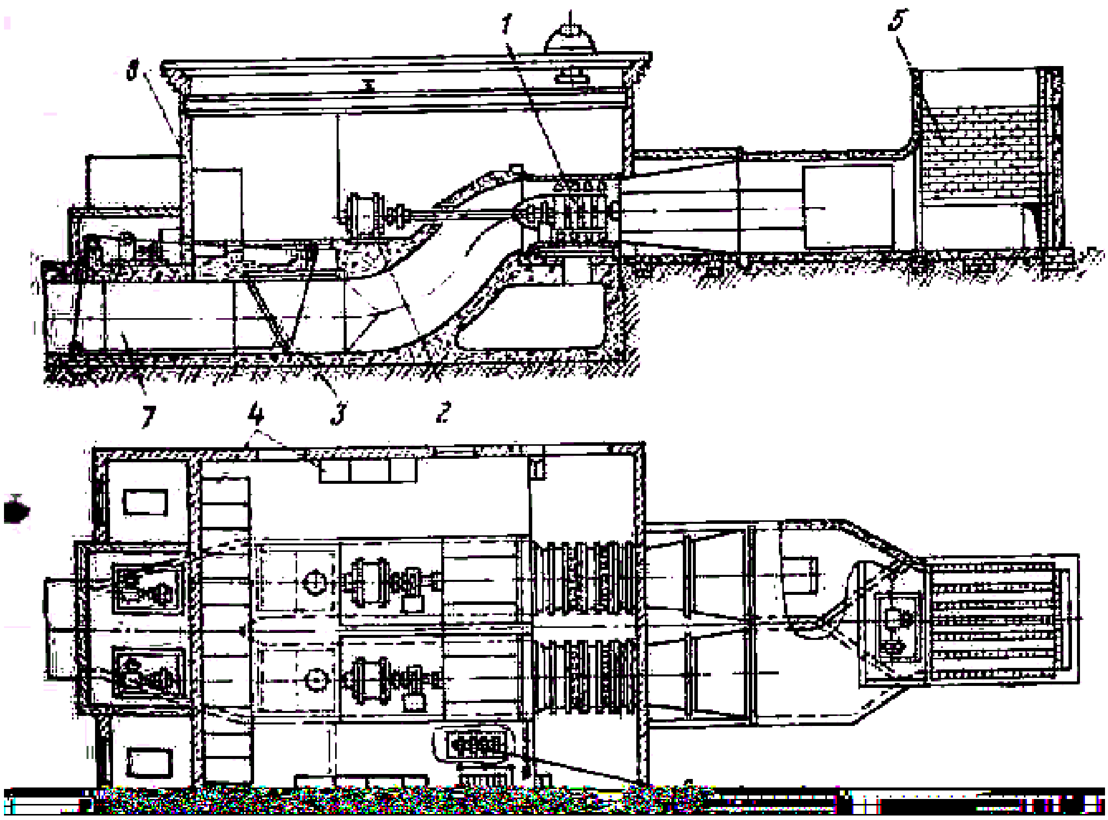


Fig.14.3 Fan installation with axial fans of the WATER type

The fan unit is located in building 8 on the foundation. The actual fans can be located outside the building, depending on local conditions. In this case, they should have a light shelter or anti-noise insulation. The fans are connected to the shaft shaft by a reinforced concrete channel 7 of rectangular cross-section supplying air, which is bifurcated into two channels of circular cross-section, suitable for the inlet openings of the working and backup fans. The output reinforced concrete channels are of rectangular cross—section. They are connected to a noise muffler 5 common to both fans, installed at the air outlet from the fan unit.

LECTURE No. 15
OPERATION AND DESIGN
FAN INSTALLATIONS

Plan:

1. . Testing of fan installations
2. Maintenance of fan installations
3. Design methodology of the main ventilation unit

1. Testing of fan installations. Tests of fan installations at mining enterprises are carried out in full and partial. The purpose of complete tests is to obtain data for the construction of individual characteristics. Partial industrial tests of fan installations are carried out periodically to identify the actual operating modes and their compliance with the field of industrial use. During the tests, devices installed on the fan unit are used to control its parameters during operation, or temporarily installed devices are used.

To measure pressure, various types of diffmanometers are used, which in mining practice are usually called a depressionometer. For clarity, consider the measurement of pressures in the ventilation duct with a liquid U-shaped depressionometer.

The simplest U-shaped depressionometer consists of a curved glass tube with a diameter of 5-8 mm filled with mercury or tinted with water. A millimeter scale is placed between the tubes.

To measure static pressure, one end of the U-shaped depressiometer 3 is connected with a rubber tube to a hole in the wall of the ventilation duct I (Fig. 15.1). The difference in the levels of hst liquid in the depressiometer is proportional to the static pressure of the pct.

Since the pressure in the ventilation duct with suction ventilation is less, and with discharge ventilation it is more atmospheric, the liquid level in the channel of the depressionometer tube communicating with the atmosphere will be lower in the first case, and higher in the second than in the channel connected to the ventilation duct.

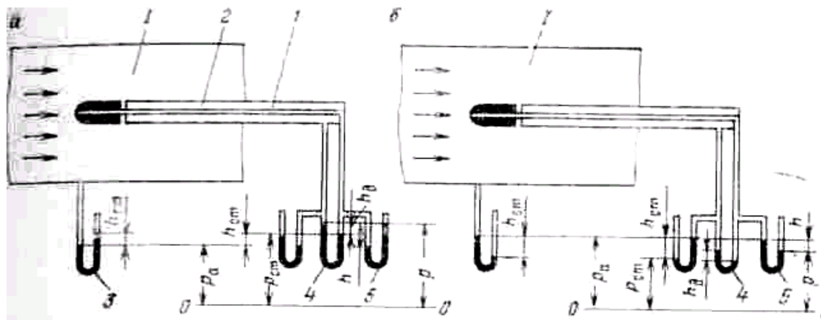


Fig. 15.1. Pressure measurement schemes in the discharge (a) and suction (b) Pipelines

To measure the total pressure and its dynamic component, depressometers are switched on, as shown in Fig. 14.1.

In this case, a Prandtl tube with two channels is often used. The central opening of channel 1 is directed towards the flow and serves to measure the total pressure p, and the side openings of channel 2 are for measuring static pressure pct. By connecting both channels with rubber (pneumatic pulse) tubes with a depressionometer 4, it is possible to measure the dynamic pressure. The total pressure p is measured by connecting the depressionometer 5 to channel 1 of the Prandtl tube. Its value is proportional to the difference in the levels of 1 depressiometer fluid.

With a significant unevenness of static pressure across the cross section, the pressure is measured at six to eight points along the perimeter of the ventilation duct and its average value is calculated by calculation or with a small unevenness (less than 10%) using integral tubes. To do this, several evenly distributed holes are drilled along the perimeter of the ventilation duct 1 and connected with fittings to a tube 2 covering the channel along the perimeter and having a terminal 3 to the pressure gauge (Fig. 15.2, and).

To measure the performance (flow rate) of fan stations at mining enterprises, anemometers and diffmanometers are used — flow meters connected to Prandtl tubes and integral full-pressure tubes. When testing fans in bench conditions, various flowmeters are used: Venturi pipe, TsAGI measuring collector, measuring diaphragms, etc.

Some of these devices are also used for testing fan installations in industrial conditions.

Type A

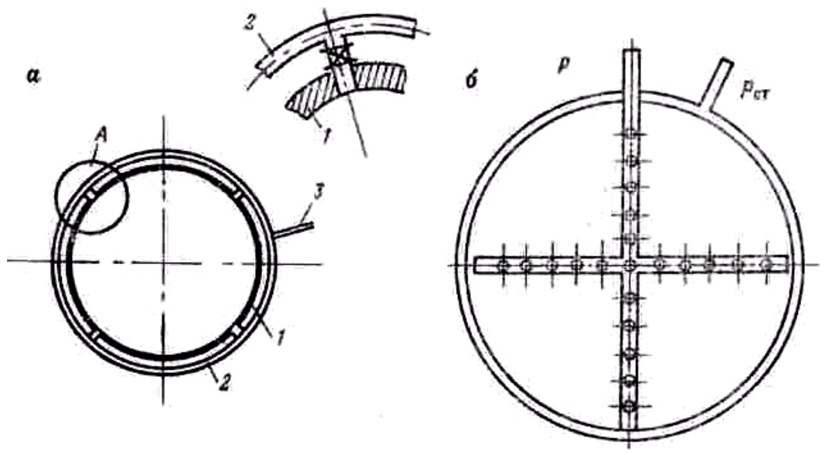


Fig.15.2. Integral tubes for measuring the average static pressure (a) and air flow through the ventilation duct (b)

Productivity Q (m3/s) (flow rate) is often determined by the magnitude of the dynamic pressure at a known section P of the ventilation duct:

$$Q=c_{cp}F, \qquad (15.1)$$

Where C_{cp} — average flow rate,

$$c_{cp} = \sqrt{2p_{д.cp} / p} \qquad (15.2)$$

The average dynamic pressure of the rd.sr is taken in connection with the uneven distribution of flow velocities across the sections of the ventilation ducts and is determined by pressure measurements not in one, but in a number of cross-section points. If the full p and static pct pressures are measured, then the rd.cp is set according to their average values:

$$p_{д.ср} = p_{п.ср} - p_{ст.ср} \qquad (15.3)$$

Averaging can be carried out by calculation or using averaging (integral) tubes (Fig. 15.2, b) with holes directed towards the flow. The holes are located at the points where the Prandtl tubes were to be measured. The air entering their holes is supplied to the diffmanometer through the outlet fitting of one of the ends of the tubes; the free ends of the tubes are sealed. To measure static pressure, the diffmanometer is also connected to the outlet fitting of tubes located along the perimeter of the channel.

Industrial tests of fan installations include measuring the plant's performance, pressure, power consumed by the fan, determining the speed of rotation of the main fan shaft, static efficiency, and external leaks in one or different fan operating modes. Various methods of aerodynamic testing of fan installations are used in the mines.

In installations not equipped with devices for monitoring air flow and pressure, the amount of air passing through the ventilation duct is determined by the air flow velocity in the ventilation duct, measured at a speed of up to 5 m/s by wing-type anemometers and at a speed of more than 5 m/s by cup-type anemometers.

The air flow rate Q (m3/s) is determined by the expression

$$Q = F\frac{1}{n}\sum_{i=1}^{n} c_i, \qquad (15.4)$$

where F is the cross—sectional area of the ventilation duct, m2;

n is the number of measurements for the sections into which this section is divided;

ci is the air flow velocity on the site, m/s.

The size of the plot is assumed to be 0.5 m2, three measurements of 30 s are made on each plot.

The static pressure of the fan is determined by a pressure gauge in the supply ventilation duct (in front of the impeller).

The simplest and most reliable determination of performance and pressure is carried out by measuring the static pressure drop in two sections of the ventilation duct of different areas. As measuring stations, crosses made of two pipes can be used, hermetically sealed at the ends and having evenly spaced holes with a diameter of 3-4 mm along the entire length (see Fig. 15.2, b). The holes in the tubes are oriented to the flow.

Productivity and pressure are determined by formulas derived from the Bernoulli equation:

$$Q = \sqrt{\frac{2F_1^2(p_{1-1} - p_{2-2})}{\rho\left[(1+\xi)\frac{F_1^2}{F_2^2} - 1\right]}}; \qquad (15.5)$$

$$p_{y.cm} = p_{1-1} - \frac{\rho Q_0^2}{2F_1^2} \qquad (15.6)$$

Where $p_{1\text{-}1}$, $p_{2\text{-}2}$, F_1, F_2 – accordingly, the flow pressure in the sections and the cross-section area *1— 1* и *2—2* ventilation duct;

ρ – air density, кг/м^3;

ξ = 0,1-0,2 — the coefficient of resistance in the area between the sections; Qo— performance when installing the guide device at an angle equal to zero, м^3/с.

The power N on the fan shaft is determined according to measurements of the active power of the electric drive by known methods using two wattmeters; the rotation frequency n of the shaft is according to passport data or is measured by a tachometer.

To change the characteristics of the ventilation network during testing, it is recommended to disconnect the installation from the shaft network by a channel and suck in atmospheric air through a backup fan, adjusting the resistance of the network by turning the blades of the axial guide device.

In the event that it is not possible to disconnect the shaft ventilation network from the fan unit, different fan operating modes can be obtained with simultaneous air supply from the shaft and through a backup fan.

According to the values of performance, pressure and k, p. d. obtained during testing of the fan installation, the actual mode of its operation is determined. If the mode is outside the scope of industrial use of the fan unit, its regulation is performed.

2. Maintenance of fan installations. Long-term and uninterrupted operation of fans is possible with proper care of them.

During the operation of the main ventilation fan installations, their inspections, revisions, repairs and adjustments are carried out. Inspections of fans and auxiliary equipment of installations are carried out every shift when accepting a shift by a machinist or on automated installations — by an electrician on duty, daily by an electrician in the day shift, weekly by the chief mechanic or his deputy and quarterly by an electrician or a team under the guidance of a mechanic. According to the results of inspections, routine repairs are carried out within the time allocated for the inspection. Defects and malfunctions noticed during inspections are recorded in the "Book of inspection of fan installations and reversal checks". The results of control reversals of the installation, which are performed at least 2 times a year, are recorded in the same book. The commissioning team under the guidance of a mechanic carries out 1 audit and adjustment of the fan unit once a year and 1 every two years—technical tests and adjustment.

Based on the materials of inspections and ongoing repairs, and depending on the wear of the main parts and components of the unit, schedules of medium and major repairs carried out with the involvement of the forces of the central workshops or ore repair plant are compiled.

Once every two years, an audit, technical tests and commissioning of all elements of the fan installation are carried out by the commissioning team of a specialized organization.

Preventive inspection and revision of local ventilation fans of the VM-M type are carried out at least once every six months.

When servicing fans, it is necessary to strictly observe the frequency of replacement of bearing lubrication.

3. The methodology of designing the installation of the main ventilation. The choice of the fan and the method of its regulation. To select fans, it is necessary to know or determine the required performance of the fan unit, the depression of the mine (mine) during its operation (obtained as a result of the calculation of the ventilation network), i.e. p and rpmx and their corresponding performance values 0, as well as to have the characteristics of fans with the designation of areas of their industrial use.

In the first approximation (or at the first stage of design), the required performance of the fan unit is determined by the formula

$$Q_в = Q_ш k_{у.в.}, \tag{15.7}$$

Where $Q_{Ш}$ — the air consumption required for ventilation of the mine (mine), m3 / s;

$k_{у.в}$ is a coefficient that takes into account air leaks through the overhead structures and fan channels. It is assumed to be equal to: 1.25 —when installing fans on a skip trunk; 1.20 — on a cage; 1.10 — on trunks and pits and 1.30 — on pits (which are used for lowering and lifting materials.

A more accurate accounting of air leaks through overhead structures and fan ducts can be carried out by summing air leaks through individual elements of structures and their interfaces when the depression and parameters of these structures are known. Usually such accounting is carried out at the second stage of design. In addition, when choosing fans, an additional margin of at least 20% is provided for their performance.

The selection of fans is made according to the schedules of their industrial use areas. If the values of productivity and depression obtained during the calculation of mine ventilation for various periods of operation do not fit into the field of industrial use of any manufactured fan, then the possibility of using a fan with a single change of its drive motor is analyzed. If at the same time it does not fit into the field of industrial use, then the use of two or more types of fans is envisaged during the service life of the enterprise.

If it is possible to provide the necessary design modes with several types of fans, their final choice is made from the conditions of minimum annual costs based on a technical and economic comparison of fans and taking into account all other factors.

Determination of the fan performance reserve. To determine the performance reserve of the selected fan during the period of maximum air flow, it is necessary to impose a characteristic of the ventilation network of the mine on its aerodynamic characteristics, determining it by the formula for the minimum equivalent opening — a quadratic parabola with a vertex at the origin passing through a point with coordinates (Q, pmax). The abscissa of the intersection point of this curve with the upper or right boundary of the industrial use zone of the fan shows the value of the maximum possible fan performance. The fan performance reserve is calculated by the formula

$$\Delta Q = \left(\frac{Q_{\max}}{Q} - 1 \right) 100\%. \tag{15.8}$$

Determination of fan power. The power of the fan drive motor is determined for each stage of its rotation frequency at the maximum and minimum equivalent openings of the shaft during its operation

$$N = \frac{Q_p}{1000\eta_{y.cm}}.$$

The values of the parameters Q, p, pu.st are taken as the coordinates of the corresponding operating point on the fan characteristic.

The engine is accepted by the power

$$N_{дв} = kN_{\max} \quad (15.9)$$

Where $N_{\max}$ — the maximum power on the fan shaft at this speed stage; k = 1,1-1,15 – power reserve factor in case of overloads.

Calculation of the annual cost of operation of the fan unit. The given annual costs for the operation of the fan installation of the main ventilation of the mine are determined by the formula

$$C = C_э + C_a + C_p + C_{об} + C_м + E_н C_в, \quad (15.10)$$

Where $C_э$ — the cost of electricity consumed by the fan unit on average per year, руб.;

$C_а$ — annual depreciation charges for the fan unit, руб.;

$C_р$ — annual depreciation charges for the fan unit y.e.;

$C_{об}$ — annual maintenance costs, y.e.;

$C_м$ — the cost of auxiliary materials spent on a fan installation per year, y.e.;

E = 0,14 — regulatory efficiency coefficient of capital investments;

$C_в$ — the cost of the fan installation, y.e.

LECTURE No. 16
GENERAL INFORMATION ABOUT PNEUMATIC INSTALLATIONS

Plan:

1. Purpose of pneumatic installations
2. Classification of compressors
3. Main parameters of compressors

1. Purpose of pneumatic installations. Pneumatic installations at mining enterprises are designed for the production and transportation of compressed air used to power pneumatic drives of mining equipment, drill and jackhammers, loading and stowing machines, combines and winches, fans, etc. Pneumatic installations are widely used in coal mines that develop steep formations, where the use of electricity is prohibited under safety conditions and compressed air is the only type of energy, and in mines where ore mining is carried out by drilling and blasting. At other coal mines and mines, compressed air is used only to a limited extent, mainly for auxiliary machines and mechanisms.

The pneumatic installation includes a compressor station and an air supply network.

With high consumption, compressed air is produced by stationary compressor stations located on the surface. In cases where a relatively small amount of air is needed, especially from places remote from the trunk, mobile compressor units located underground on the plots are used.

In some mining enterprises, it is economically feasible to use an air supply system with booster compressors. When using such a system, air compressed to a pressure of only 0.3—0.4 MPa in a stationary compressor station on the surface of the mine (mine) is supplied through air ducts to mobile compressors located near the places of its consumption. In booster compressors, the air pressure increases to the required value. Cost savings are obtained by reducing energy losses during the transportation of compressed air due to its lower pressure and energy losses on hydraulic resistances.

From stationary compressor units, compressed air is transported to consumers through air ducts of pneumatic networks.

Pneumatic networks of modern mining enterprises consist of branched air ducts with a large number of different consumers. The place of branching is called a node.

Pipelines between nodes, nodes and switchgear are called a backbone network or simply a highway (main highway, district highway, etc.) by consumers. The branches of the pipes to which consumers are connected are called end elements. The total length of air ducts at some mines and mines reaches tens of kilometers.

2. Classification of compressors. A compressor is a machine designed to convert the mechanical energy of the drive into useful potential and kinetic energy of a gas. The compressor increases the gas pressure and moves it from the low to the high pressure area. At mining enterprises, air is compressed by compressors of pneumatic installations.

According to the method of gas compression, compressors are divided into two groups — volumetric compression (displacement compressors), in which the pressure of gas (air) increases by reducing the working space; these include reciprocating, screw, rotary compressors, etc.;

kinetic compression, in which gas (air) is compressed during the forced movement of gas (air) during force interaction with the blades of rotating wheels, these include turbochargers — centrifugal and axial.

The first group of compressors is sometimes also called displacement compressors, and the second — blade compressors due to the presence of blades (blades) in them, with which the compression process is carried out,

According to the design of the working bodies, compressors are distinguished: reciprocating, vane (turbochargers), screw, rotary, etc.

According to the type of compressed gas, compressors are divided into air, ammonia, freon, etc.

According to the magnitude of the pressure created , there are:

compressors, called vacuum pumps, sucking gas (air) out of a vacuum space and compressing it to atmospheric or slightly higher pressure;

blowers (gas blowers) — machines compressing air (gas)

up to 0.3 MPa;

low pressure compressors (0.3—1.0 MPa);

medium pressure compressors (1.0—10.0 MPa);

high-pressure compressors (10-250 MPa).

In the mining industry, low-pressure compressors are most widely used. Vacuum pumps are used to suck methane out of coal seams.

3. The main parameters of compressors. The main parameters characterizing the operation of the compressor are: volumetric capacity Q (m3/s or m3/min), reduced to suction conditions; initial (before compression) rp and final (after compression) rc (Pa, kPa, MPa) pressure; degree of pressure increase; initial TN and final Tc, temperature compressed gas; power N (kW) on the compressor shaft. There are excessive (relative to atmospheric ri) and absolute rad (taking into account atmospheric pressure) pressure; the first is indicated in the compressor passport, the second is used in thermodynamic calculations. The temperature of the compressed gas (air) in thermodynamic calculations is expressed in units of Kelvin (K), $T = 273 + t$, where t is the temperature in degrees Celsius (° C).

The most widely used compressors in the mining industry are: two-stage reciprocating compressors with a capacity of 10; 20; 30; 50; 100 m3/m centrifugal compressors with a capacity of 115; 250; 500 m3/min and screw compressors with a capacity of 5; 12.5; 25 m3/min

Axial turbochargers, widely used in a number of industries, are not used in the mining industry due to the low final pressure of compressed air.

LECTURE No. 17
RECIPROCATING COMPRESSORS
Plan:

1. Principle of operation and classification
2. The working process of the reciprocating compressor
3. The actual process in a single-stage compressor

1. The principle of operation and classification. Reciprocating compressors are volumetric compressors: gas compression processes occur in a closed space, which is the internal volume of the working cylinder. The organ acting on the gas in order to change its volume is a piston moving inside the cylinder. The cylinders are equipped with regulating bodies — suction and discharge valves for suction, compression and injection of air (gas) into the network. The time—consistent implementation of three stages of the working process — suction, compression and injection - is a characteristic feature of the operation of a reciprocating compressor.

Reciprocating compressors are distinguished by a variety of design schemes and designs. In addition to the general classification of compressors, reciprocating compressors are divided according to the type of cylinders, their location in space and combination.

By type of cylinders — compressors with cylinders of simple (Fig. 17.1, a) and double (Fig. 17.1, b) action, as well as with a differential cylinder (Fig. 17.1, c). The latter is used only in multistage compressors.

According to the number of steps — single-stage (Fig. 17.1, a), two-stage (fig. 17.1, d, e, w, z), three-stage and more. In modern compressors, the number of stages does not usually exceed seven.

According to the number of cylinders — single-cylinder (Fig. 17.1, a, b), two-cylinder (Fig.17, e, w, z, d), three-cylinder and more. Figure 17.1, g. shows a diagram of a single-stage two-cylinder compressor.

According to the number of rows in which the cylinders are located — (one Fig. 17.1, g), two - (Fig. 17.1, d) and multi-row.

According to the orientation of the cylinders — horizontal, vertical and angular (with an angle between the cylinders): rectangular (fig. 17.1, e), V-shaped (fig.17.1, e), etc.

In the mining industry, horizontal compressors made according to oppositional schemes, with counter (oppositional) movement of pistons, providing balance of piston forces, are increasingly used.

Compressors made according to oppositional schemes, due to the balance of piston forces, are very promising and are increasingly being used in the mining industry.

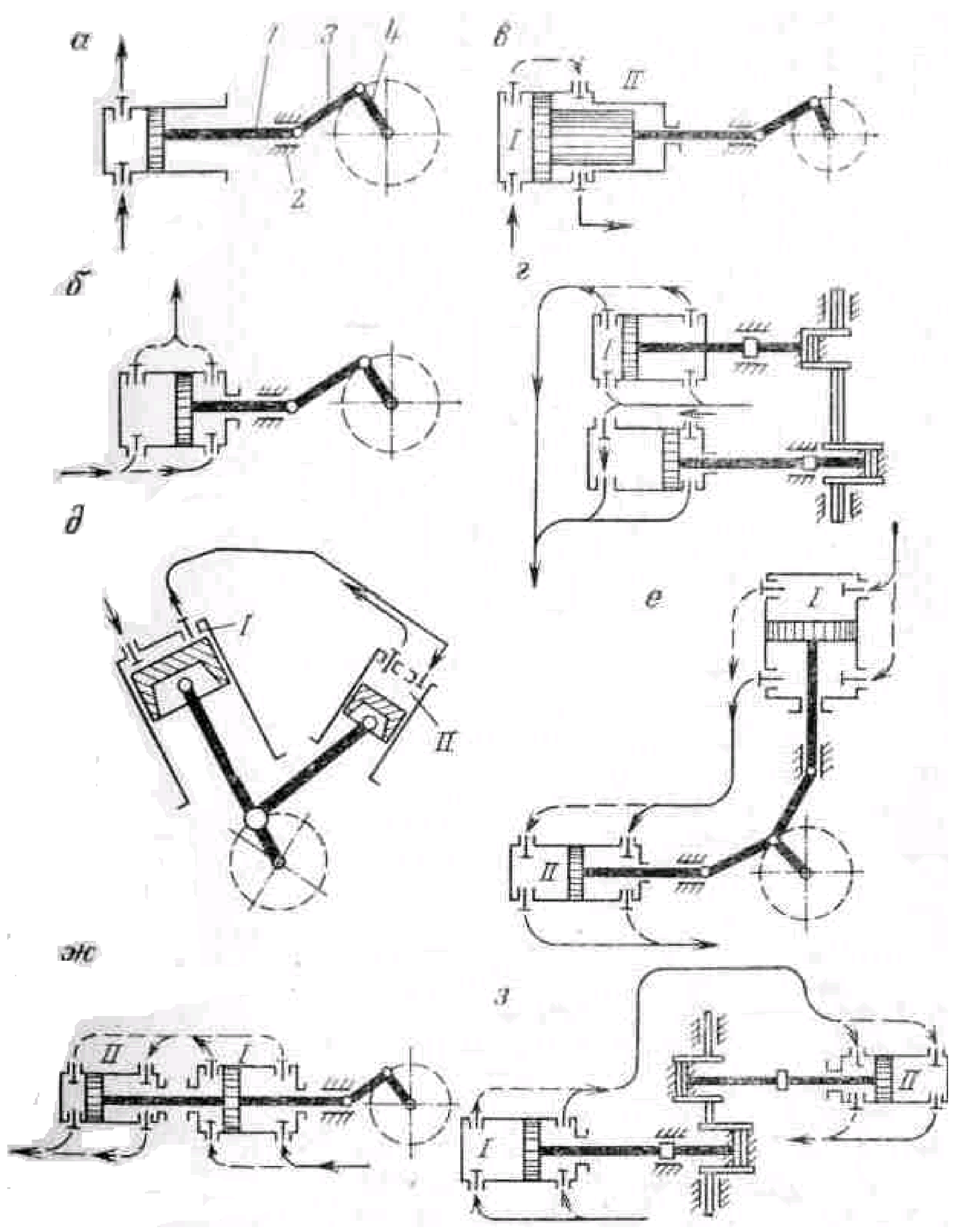


Fig. 17.1. Schemes of reciprocating compressors

As drives of compressor pistons, crank-connecting rod mechanisms are usually used, consisting of a rod (see Fig. 17.1, a), a crosshead 2, a connecting rod 3, a crank 4 and designed to convert rotational motion in water into reciprocating motion of the piston. The drives of small compressors may not have a crosshead, while the connecting rod is directly connected to the piston by means of hinges

2. The working process of the reciprocating compressor. It is convenient to consider the working process in an ideal reciprocating compressor using the process diagram in the coordinate system p — i (Fig. 17.2, a).

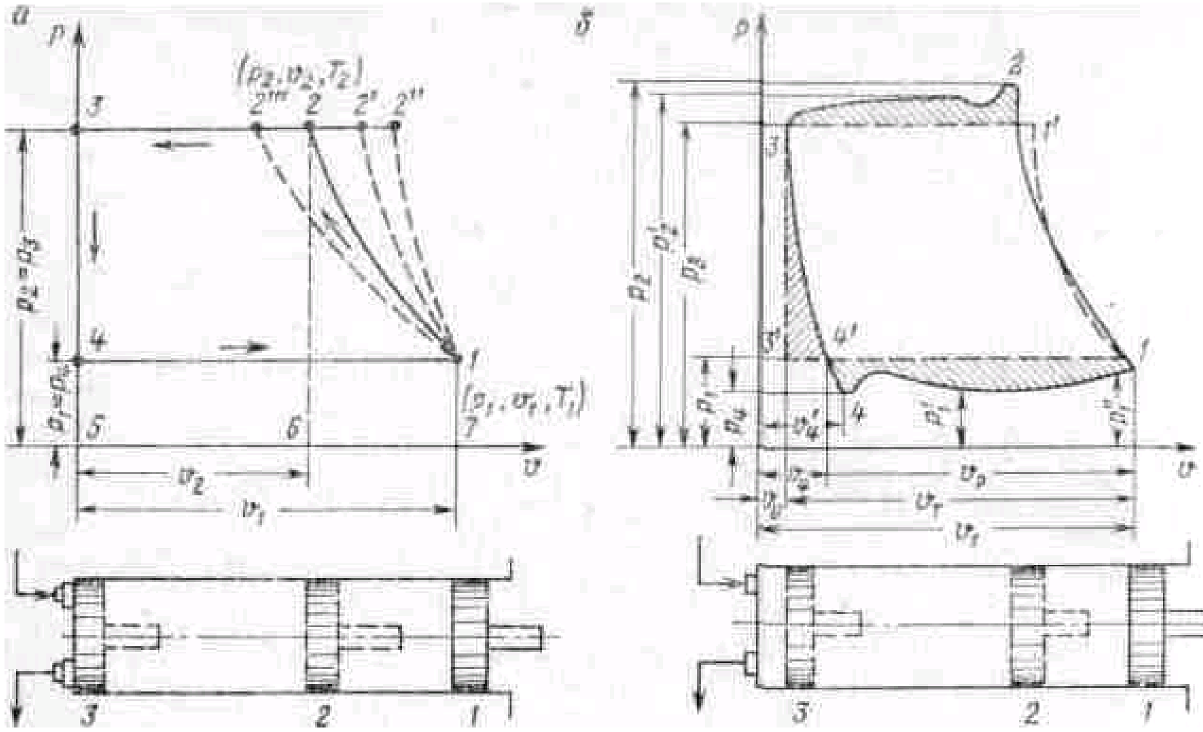


Fig. 17 2. Diagrams of a single-stage reciprocating compressor; ideal (a) and real (b)

The rightmost position of the piston in the cylinder corresponds to point 1 — the cylinder is filled with air (gas) with parameters p1, v1 and T1, the suction valve is closed. When moving to the left, the piston compresses the air (gas) enclosed in the cylinder. The compression process is generally characterized by polytrope 1-2 and ends at point 2, where the gas is characterized by parameters p2, v2 and T2. Depending on the conditions, compression can be carried out by isotherm 1-2", polytrope 1-2, adiabate 1-2' and polytrope 1-2" with a large value of the index n.

In an ideal compressor, in which there is no resistance of the discharge valves, the moment of the end of compression coincides with the moment of opening of the discharge valves and the beginning of gas injection into the pressure pipeline. When the piston moves from point 2 to point 3, the air from the cylinder is pushed into the discharge pipeline, while the pressure p2 and temperature T2 change. The process proceeds along the line 2-3 (2'-3, 2"-3, 2"-3), which is called the discharge line. In the leftmost position of the piston (point 3), the discharge valve(s) closes. When the piston starts moving to the right, the pressure in the cylinder drops to the pressure p1 in the suction line and the suction valve opens. When the piston moves to the right, the cylinder is filled with gas (suction); line 4-1 is called the suction line. The suction process ends at point 1, and then the cycle repeats.

The work expended in the compressor cycle is proportional to the area f of the diagrams:

compression work

$$L_{сж} = Rf_{1-2-6-7}; \qquad (17.1)$$

pumping operation

$$L_{H} = Rf_{2-3-5-6}; \qquad (17.2)$$

suction operation

$$L_{вс} = Rf_{1-4-5-7}; \qquad (17.3)$$

Here R is the scale factor.

The minus sign in the expression (17.3) indicates that at the moment of suction, the system, together with the incoming gas, receives from the outside part of the energy, by the amount of which the total amount of work per cycle decreases.

Thus, the total operation of the compressor cycle

Bringing the work expended to specific, i.e. to the work spent on compressing 1 kg of gas, we get

3. A valid process in a single-stage compressor. The operation of a real compressor and the thermodynamic processes that occur at the same time, in fact, differ significantly from the work and processes occurring in an ideal compressor. This difference primarily lies in the fact that in the cylinder of a real compressor after the end of the injection process (the leftmost position of the piston) there remains a certain amount of gas volume v0 compressed to pressure, injection p3 During the suction process, this gas expands and fills part of the cylinder volume, reducing the compressor performance. Therefore, the cylinder space filled with this residual gas is called "harmful".

The harmful volume of gas consists of the volume formed in the gap between the piston surface (up to the first O-ring) and the cylinder surface, as well as the volume of the valve box and gas channels in the valve to the working plate.

The second feature of the operation of a real compressor is the fact that during its operation there is a continuous change in the parameters of the state p, v and T, due to the presence of energy costs to overcome hydraulic resistances when moving gas inside the cylinder from the intake pipeline to the pressure, as well as the presence of heat exchange having a different intensity for each moment of the compressor cycle.

The diagram of the cycle of a real compressor is shown in Fig. 17.2, b. In the same diagram, for comparative evaluation, dotted lines are drawn diagram 1-1'—3—3' for the case if the compressor worked on an ideal cycle, and the expansion of the gas remaining in the harmful space would not affect the performance of the compressor. The actual compressor cycle differs from the theoretical one.

The compression process (line 1-2) generally occurs along a polytrope with a variable index n during compression. Compression ends at point 2 at a pressure that is greater than the pressure in the pressure line p3 by the amount of the pressure drop required to overcome the resistance of the pressure valve springs and the inertia forces of the movable elements of the latter $\Delta p_H = p_2 - p_3$.

The injection process (line 2-3) after opening the valves is characterized by a decrease in pressure (the necessary pressure drop decreases) and then some of its increase to R., due to an increase in the speed of the piston and, consequently, the gas velocity. The maximum pressure value corresponds to the maximum speed of the piston in its middle position.

LECTURE No. 18
PISTON COMPRESSOR PERFORMANCE
Plan:

1. Piston compressor performance
2. Regulation of reciprocating compressors
3. Nomenclature and design of shaft reciprocating compressors.

1. The performance of a reciprocating compressor. The capacity of the reciprocating compressor (m3/min), respectively, for single and double-acting cylinders:

$$Q = \lambda F_1 sn \text{ и } Q = \lambda(F_1 + F_2)sn \qquad (18.1)$$

Where λ - a coefficient that takes into account the influence of various factors that occur during the operation of a real compressor and lead to its reduction;

F_1 and F_2 — piston areas on one side and the other, м^2;

s- piston stroke, м.

coefficient λ they are called the performance coefficient. It consists of a complex of coefficients (volume, pressure, tightness, heating, humidity):

$$\lambda = \lambda_0 \lambda_p \lambda_\Gamma \lambda_T \lambda_B \qquad (18.2)$$

For modern reciprocating compressors λ= 0,7÷0,90.

The volume coefficient 0 takes into account the decrease in productivity due to the presence of harmful space and represents the ratio of the volumes of the working vP to the geometric v_r

$$\lambda_0 = \frac{v_P}{v_T} \qquad (18.3)$$

Where $v_p = v_T + v_0 - v_4$. Given that the mass of the gas at points 3 and 4' is the same, and applying the polytrope equation, we can write:

$$v_0 p^{1/n}{}_3 = v_4 p^{1/n}{}_1$$

Where from $$v_4 = v_0 (\frac{p_3}{p_1})^{1/n}$$

$$\lambda_0 = \frac{v_T + v_0 - v_0 (p_3/p_1)^{1/n}}{v_T} = 1 - a(\varepsilon^{1/n} - 1) \qquad (18.4)$$

Substituting in .(18.3), we get

where *a* — relative size of the harmful space, $a = v_0/v_T$; ε — degree of pressure increase, $\varepsilon = p_3/p_1$

In modern compressors, a = 0.02-0.1 and is more important for high-pressure cylinders. When designing compressors, they strive for a maximum increase of 0 by reducing the harmful space, i.e. the value of a.

Harmful space is one of the factors limiting the degree of pressure increase in one stage. This follows from equation (18.4). With a sufficiently large value, the value 0 can be equal to zero.

$$\varepsilon^{1/n} - 1 = \frac{1}{a}$$

where is the maximum degree of pressure increase

$$\varepsilon = (\frac{1}{a} + 1)^n \qquad (18.5)$$

So, for a = 0.1 and the polytrope of expansion n= 1.2, the limit

The value e = 17.8. physically, this means that with the specified parameters, the compressor will run idle. The gas enclosed in the harmful space will fill the entire volume of the cylinder during suction.

The pressure coefficient p takes into account the decrease in compressor performance due to the fact that the air (gas) pressure in the cylinder during suction is lower than the pressure p1 in the suction pipeline (p 0.90÷0.95).

The tightness coefficient g takes into account a decrease in productivity due to gas leaks from the injection area to the suction area or into the atmosphere through the gaps between the cylinders and piston rings, in the oil seals and working valves (g 0.95-0.98).

The heating coefficient t takes into account a decrease in actual productivity compared to theoretical due to a decrease in the specific volume of gas due to with its heating in the cylinder during suction.

The humidity coefficient b is used when taking into account the decrease in productivity due to condensation of moisture during compression. This change is small for air, and in relation to shaft compressors in can not be taken into account when calculating performance.

2. Regulation of reciprocating compressors. The performance of compressors has to be regulated due to the variable flow of compressed air in the network.

The short-term discrepancy between the flow rate and the performance of the compressors (compressor) is compensated by the air collector installed at the compressor station when using reciprocating compressors and the capacity of the pneumatic network. In other cases, regulation is carried out by special control systems, which should ensure smooth performance changes and cost-effectiveness of regulation.

Performance regulation can be carried out periodically by stopping the air supply to the network, stepwise and smooth performance changes. The periodic termination of the supply can be performed by stopping the compressors. Step-by-step control of the compressor station performance is effectively performed by changing the number of simultaneously operating compressors. Smooth regulation can be carried out in the following ways.

The regulation by changing the drive speed is most economical in terms of energy costs for compression. However, due to the great complexity of the regulated electric drive, this method has not yet found practical application on reciprocating compressors used

in the mining industry.

Throttling control on the suction line, i.e. an increase in the resistance (partial overlap) of the suction pipeline, provides a decrease in pressure in the suction cavity of the compressor cylinder from p1 to p1reg. This leads to a decrease in the amount of incoming air (gas) into the cylinder in one stroke of the piston from v1 to v1 reg (Fig. 18.1, a). At the same time, the degree of increase in air pressure and its final temperature increases, which is unacceptable for shaft compressors. Regulation by throttling the suction has not become widespread**.**

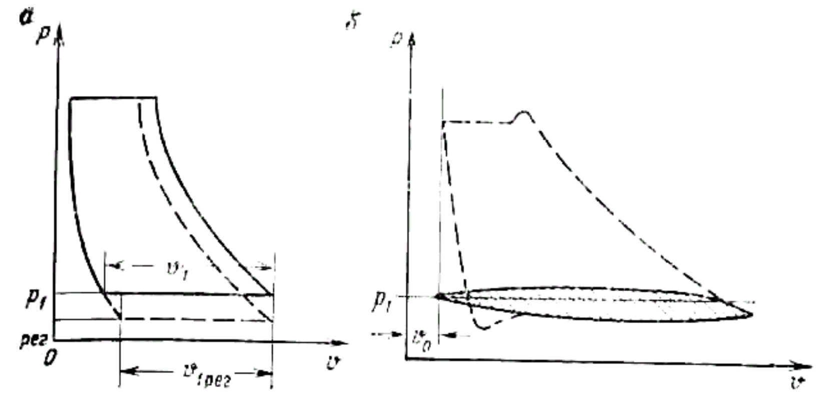


Fig. 18.1. Indicator diagrams for regulation by throttling on suction (a) and pressing of suction valves (b)

Air (gas) bypass control from injection to suction using, for example, a bypass pipeline ensures a change in compressor performance and, at zero performance, idling. The control method is uneconomical and can lead to an increase in temperature.

The control of the suction valves by pressing is based on the fact that during compression, part of the compressed air is pushed back into the suction pipeline when the suction valves are partially (leaky) closed.

To regulate this method, the suction valves are equipped with a special device that squeezes the valve plates during compression.

The indicator diagram of a cylinder with a control device for pressing valve plates is shown in Fig. 18.1, b, its area is shaded. In the same figure, the dotted line shows an indicator diagram for comparison when the device is turned off or not.

Regulating the impact on the suction valves is a simple and common way to reduce the performance of general purpose reciprocating compressors. The disadvantage of this method is the energy consumption at idle of the compressor, which is about 15% of the power consumed by the compressor at full load.

Regulation by the inclusion of additional harmful spaces allows to reduce productivity by reducing the amount of intake air (gas). Compressed air (gas) during injection, enclosed in an additional harmful space, during suction

partially fills the working space of the cylinder, reducing the compressor performance accordingly.

In terms of economy, this method of regulation is equivalent to regulation by pressing the suction valve plates.

The last three methods are used to regulate the performance of reciprocating compressors at mining enterprises.

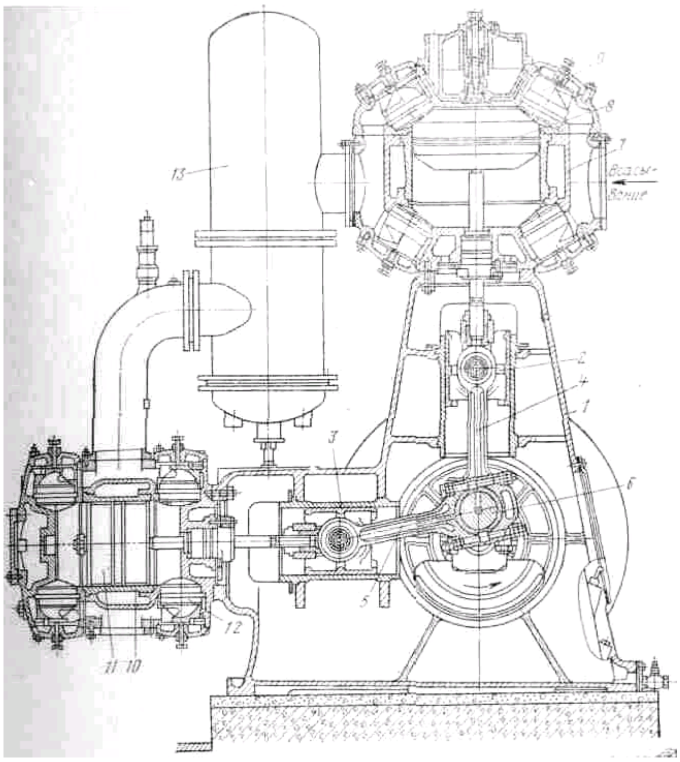


Fig. 18.2. General view of the compressor 305VP-30/8:
1 — frame; 2 and 3 — crossheads; 4 and 5 — connecting rods; 6 — crankshaft;
7 — vertical cylinder; 8 and P — pistons; 9 and 12 — valves;
10 — horizontal cylinder; 13 — intermediate refrigerator

3. Nomenclature and design of shaft reciprocating compressors. A large number of types of reciprocating compressors are used in the mining industry. Currently, upgraded air angular compressors of type P (302VP-10/8, 202VP-20/8, 305VP-30/8, Fig. 18.2) and multi-row air compressors with horizontal arrangement of cylinders of type M (4M10-100/8, Fig. 18.3, 2M10-50/8) are produced and recommended for use as stationary. Conventional designation of type P compressors: letters VP — rectangular air, after the letters in the numerator of the fraction — productivity (m3/min), in the denominator — excess discharge pressure (kg/cm2), the numbers before the letters show the nominal load on the rod (ts), the number before zero is the modification number of the upgraded compressor

In the conventional designation of compressors of type M: M is a multi—row base, 10 is the value of the piston force (tc) of one row (the piston force is the sum of the forces acting on the piston: air pressure in the cylinder, inertia of moving reciprocating masses and friction); the numerator of the fraction is the compressor capacity (m3/min), the denominator of the fraction — overpressure (kgf/ cm2), the number before the letter M is the number of cylinders. Type M compressors have two or four cylinders horizontally arranged, an intermediate refrigerator between the first and second stages. The compressors are made according to the opposite scheme. Their crank-connecting rod mechanisms are located on both sides of the crankshaft: moreover, the cranks of adjacent rows are shifted 180 ° and have a mutually opposite movement, which ensures the balance of the moving masses (the compressor is made without a flywheel) and piston forces.

The following compressors of earlier releases are also widely used in the mining industry: VP-50/8, VP-20/8M, 2VG-100/8, 5G-100/8, 55V, etc. The most important components of reciprocating compressors are air distribution devices — valves.

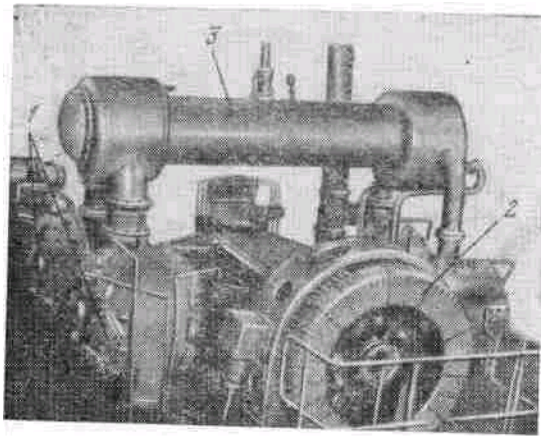


Fig. 18.3. General view of the piston compressor 4M 10-100/8:
1- cylinders; 2 — electric motor; 3 - intermediate refrigerator

The specific energy consumption and compressor performance largely depend on their operation. In modern reciprocating compressors, self-acting plate valves are used, the opening and closing of which occurs automatically under the influence of the difference in air (gas) pressure on the valve plates from the cylinder side and the suction or discharge nozzle of the compressor.

The main elements of the valves (Fig. 18.4) are valve plates (one or more), a seat, limiters of the plates and springs (the role of the latter can also be performed by self-spring plates).

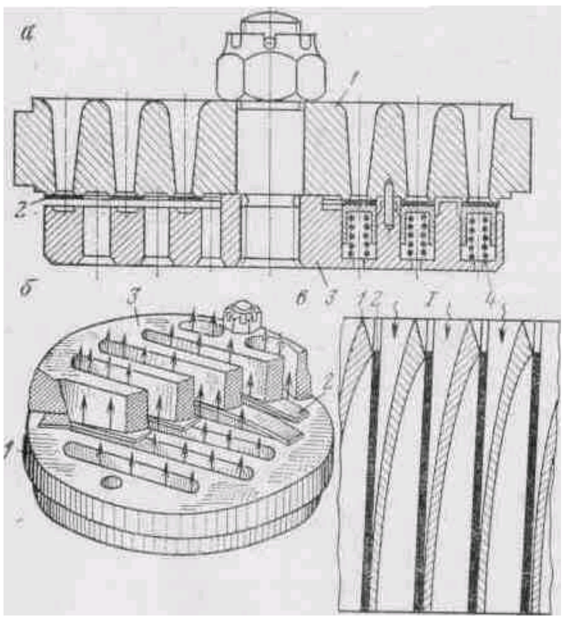


Fig. 18.4. Valves:
a — ring; b — strip; c — direct—flow closed (1) and open (II), 1 — seat; 2 — plate; 3 - limiter (stop); 4 — spring

The main parameters that determine the operation of the valve include: the area of the flow section, the height of the lifting plates, the air (gas) velocity in the channels of the valves and the tension of the springs. For quiet and silent operation, the lifting height of the plates is 2-4 mm.

Plate valves are supplied with plates in the form of disks of rings, strips and are called disk, ring, strip. In the direction of the air flow relative to the plates, the valves are divided into valves with perpendicular (non-direct-flow) and parallel (direct-flow) arrangement of the plates.

The above-mentioned valves (disc, ring and strip) belong to the first group of non-direct valves, in which the air (gas) repeatedly changes direction. The second group of direct—flow valves are equipped with spring plates, which bend under pressure along the profile of the seat, and air passes through the valve without changing its direction, which provides them with a number of advantages: a larger flow section with the same valve sizes, reduced pressure losses, smaller volumes of harmful space and the mass of plates, tightness of the plates to the support surfaces and others .

The valves must meet the following general requirements: ensure the density of closing channels, have minimal inertia and sufficient speed, have a small volume of harmful space and little resistance to gas flow, have wear resistance and strength. Valve plates work in difficult conditions —alternating loads and temperatures of 430-450 K, therefore they are made of high-quality alloy steels. During operation, the valve boxes must be intensively cooled.

The cylinders of low and medium pressure compressors are cast from high-quality cast iron, and high—pressure multi-stage compressors are made of steel. Pistons are cast iron.

Double-acting compressors are equipped with crank-pin mechanisms with crosshead heads. The slider of the head is connected tightly to the piston rod and pivotally by means of a finger with a connecting rod. In single-acting compressors, the connecting rod is directly connected to the piston. The other end of the connecting rod is connected to the crankshaft. The crankshaft, depending on the number of connecting rods, has one or more elbows located between the supporting shaft necks.

In reciprocating compressors, sliding bearings are widely used, which are used for the main shafts of compressors with a capacity of up to 100 m3/min (compressors 1RG-100/8, 4M10-100/8, etc.). Rolling bearings are mainly used for the main shafts of compressors with a capacity of up to 50 m3/min (type P compressors). The crankshaft is equipped with a flywheel necessary to equalize the stroke of the piston. Usually it is also a pulley. If the engine rotor is mounted on the compressor shaft, then it also performs the role of a flywheel.

LECTURE No. 19
CENTRIFUGAL AND ROTARY COMPRESSORS
Plan:

1. The principle of operation and the device of the compressor
2. The working process of the centrifugal compressor
3. Nomenclature and design of centrifugal compressors and installations
4. Classification of compressors
5. The working process of the screw compressor and its performance.

1. The principle of operation and the device of the compressor. According to the principle of operation, centrifugal compressors are similar to centrifugal fans and pumps. Compression of air (gas) is carried out by aerodynamic forces arising from the interaction of the blades of the rotating impeller with the flow of air (gas).

Since the degree of pressure increase in one impeller (in one stage) is small, multistage compressors consisting of several stages in which air is compressed sequentially are used in mining enterprises. In this case, the air flow from the previous to the next stage can flow directly or passing through an intermediate refrigerator and cooling in it. The stage from which the flow

enters the intermediate refrigerator or the pneumatic network is called the end stage.

Air (gas) is supplied to the impeller through an inlet device, which should ensure a uniform distribution of the flow velocity along its cross section. To do this, the input device is performed carefully designed and with the confusory nature of the flow flow. The rate of flow entry into the impellers of modern centrifugal compressors reaches 100-150 m/s or more. Increasing the speed has a positive effect on the uniformity of the flow and allows you to reduce the dimensions of the compressor.

To pre-twist the flow in front of the impeller, the input devices of some compressors, for example, fresnel compressors, are equipped with a blade guide device similar to the fan guide devices (see Fig. 19.1.).

From the impeller 1, a stream of compressed air (gas), the absolute velocity of which reaches 500-600 m / s, usually enters the annular bladeless diffuser 2 and then into the diffuser 3 (Fig. 19.1). The diffuser 2 is an annular space of small radial extent and serves mainly to equalize the velocity field of the flow coming out of the working wheels. The diffuser 3 can be made with a blade or, less often, a channel. It is designed to convert kinetic energy into potential energy. The conversion process in it occurs with greater efficiency than in a bladeless diffuser. From the blade diffuser 3 of the intermediate section of the compressor, the flow flows along the knee 4 into the reverse guide blade 5, which simultaneously plays the role of a diffuser, additionally converting potential kinetic energy.

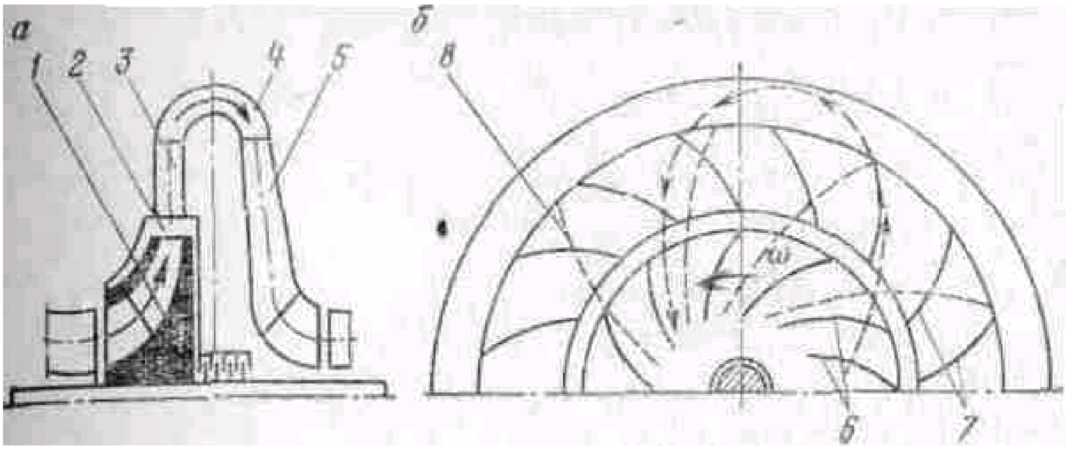


Fig.19.1. Diagrams of the intermediate section of the centrifugal compressor (a) and its section along the gas flow line: 1 — impeller; 2 and 3 — respectively blade—less annular and blade diffusers — elbow; 5 — reverse guide blade; 6 — impeller blades; 7 and 8 - respectively diffuser blades 3 and the reverse guide device 5 (the latter are shown in dotted lines)

From the blade diffuser 3 of the end section of the compressor, the flow enters the output device — the spiral chamber (snail) and then into the end diffuser.

In comparison with reciprocating centrifugal compressors, they have the following advantages: greater compactness (smaller dimensions and weight of the compressor and the foundation); absence of reciprocating masses, as well as suction and discharge valves (there is a check valve); high speed of rotation of the drive shaft and high compressor performance (usually more than 100 m3/min) uniformity compressed air supply; operation safety due to the absence of lubricating oil impurities in the compressed air; low consumption of lubricating oil; possibility of direct connection to a high-speed turbine or electric motor; cheaper maintenance.

Disadvantages of centrifugal compressors: the difficulty and even impossibility of manufacturing low-performance compressors; a slightly smaller k, P. D. of the compressor itself; the limited degree of compression and the difficulty of obtaining high pressures (For mining enterprises, where small pressures are mainly required, this disadvantage is not significant); the possibility of stable operation within certain limits of productivity as with all turbomachines, more complex parallel operation; impossibility of frequent switching on and off of the compressor.

It is preferable to use centrifugal compressors at large compressor stations (200 m3/min).

2. The working process of a centrifugal compressor. The working process begins in the compressor input device, where, due to the confusory nature of the flow, its velocity and kinetic energy increase, and due to energy losses, the temperature also increases somewhat.

A continuous compression process occurs in the impeller, characterized by a change in the thermodynamic parameters of the state p, V, etc. With an increase in pressure, the temperature simultaneously increases. In turbochargers, there is no pronounced separation of the processes that make up the working cycle, as is the case with reciprocating compressors. The work spent on compression is spent on increasing the potential and kinetic energy of the flow, as well as on overcoming hydraulic resistances in the channels.

The compression process follows general patterns. In the absence of air cooling inside the flow part of the stage (section) of the centrifugal compressor, the process of air compression in the impeller can be considered adiabatic, i.e. without heat exchange with the environment. Then the compression process in an ideal compressor in the T — 8 coordinate system will be characterized by an adiabatic 1-2, and the total work spent on compression will be an area of 5-1-2-3— 4.

The actual compression process is polytropic, since the compressed air (gas) is heated by converting the energy spent on overcoming friction into heat. The compression process proceeds along the polytrope 1-2, and the compression work is equivalent to the area 6-2-3-4-6 in the diagram T — .5. Taking into account internal losses, the specific compression work in the impeller can be determined by the formula

The working process of compressing air (gas) proceeds after the impeller — in diffusers: blade-less, blade-like, in the reverse guide device of the intermediate section, as well as in the outlet behind the end section. In diffusers, kinetic energy is converted into potential pressure energy, which is accompanied by an increase in the temperature of the compressed air (gas) at the outlet of the compressor.

For a multi-stage compressor, the theoretical foundations of multi-stage compression are valid.

In turbochargers, due to the significant degree of compression, cooling of the compressed gas is used. In this case, cooling can be carried out in the following ways: cooling by supplying water to special cavities inside the compressor housing, an expensive and inefficient method; cooling of air (gas) in the removed intermediate refrigerators after each section, in which there are several impellers;

combined air (gas) cooling (a combination of the first and second options), structurally complex and expensive, is the most effective method.

In compressors operated at mining enterprises, cooling is used in out-of-the-way refrigerators.

Characteristics of centrifugal compressors. The characteristics of centrifugal compressors are called dependencies p=f(Q), N=f1(Q) and . Figure 19.2 shows the approximate nature of these dependencies at a constant speed of rotation of the compressor rotor, n = const. The operating mode of the compressor, like other centrifugal machines (pumps, fans), is at the intersection of its characteristic p=f(Q), with the characteristic I of the external network, which in this case is the pneumatic network. The performance of a centrifugal compressor, unlike a reciprocating one, significantly depends on the network pressure. The characteristics of centrifugal compressors have a maximum. The nominal operating mode of the compressor corresponds to and is characterized by the parameters Qn ph and Nn, the operating mode of the compressor at point R will be critical, the parameters corresponding to this mode are critical.

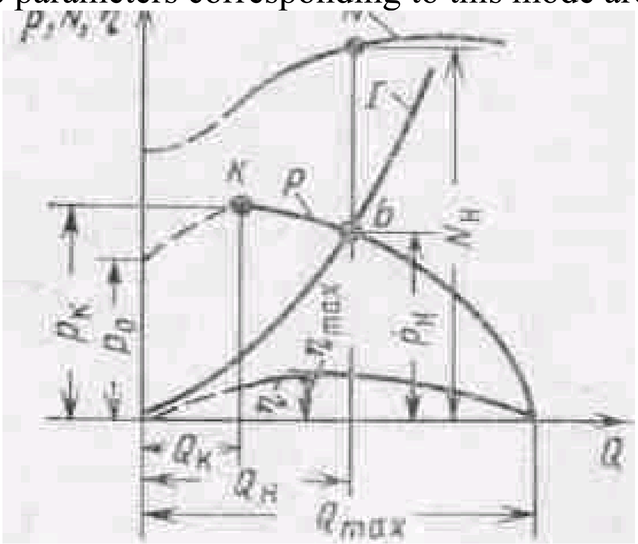


Fig. 19.2. Full characteristics of the centrifugal compressor

The operating modes at the points located on the right branch of the characteristic are stable. If, due to a decrease in air flow from the network, the operating mode turns out to be at a point located on the left branch of the characteristic, then the operation of the compressor will be unstable and a phenomenon may occur, in which there is a drop in productivity and pressure, shaking of the machine and air ducts. Surging has a harmful effect on compressors, and during their operation measures are taken to exclude their operation on the left branch of the characteristic. In the designs of compressor units, special anti-surge devices are provided to prevent surging.

The anti-surge device has one or more high-speed valves connecting the discharge pipe or part of the compressor stages to the atmosphere and triggered by the sensor if the compressor operation mode is approaching critical. When the anti-surge device is triggered, the compressed air partially escapes into the atmosphere, the air flow through the compressor increases and the point of joint operation of the compressor with the network shifts to the right of the critical one.

As for fans, universal characteristics are built for compressors. In the practice of using compressors at mining enterprises, they are rarely used.

Regulation of centrifugal compressors. The regulation of the parameters of the operation of centrifugal compressors, as well as centrifugal pumps, is possible in the following ways: throttling on the injection or suction line, changing the speed of rotation, turning the blades of the guide inlet apparatus.

Throttling control on the discharge line consists in artificially changing the characteristics of the network by increasing its resistance when closing the valve or throttle installed at the beginning of the network. This method of regulation is the most uneconomical, since part of the energy transmitted to the air (gas) in the compressor is lost in the throttle, and the specific energy consumption per 1 m3 of compressed air increases significantly (depending on the depth of regulation).

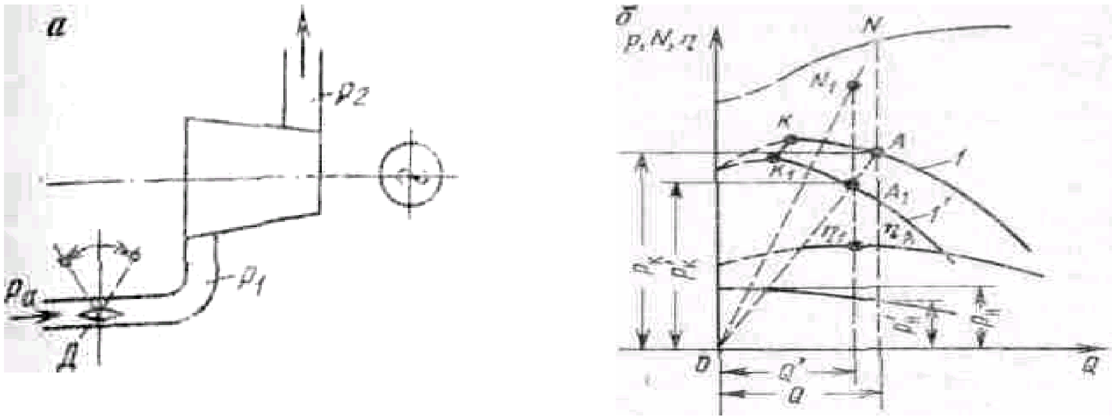


Fig. 19.3. Diagram (a) and characteristics (b) of the compressor when throttling on the suction line

Throttling control on the suction line. With this method of regulation, the characteristics of the compressor p=f(Q) and N= f1 (Q) change, shifting down from the nominal (Fig. 19.3). The characteristics obtained during throttling can be constructed by taking unchanged the air temperature at the inlet to the impeller (T = T), the degree of pressure increase and the gas constant (R=R) based on proportionality formulas:

$$p'_k = \frac{p'_н}{p_н} p_к; Q'_k = \frac{p'_н}{p_н} Q; N'_k = \frac{p'_н}{p_н} N; \qquad (19.1)$$

When the suction pressure decreases from pH to pH, the parameters of the air (gas) at the outlet Q, pk, N from the compressor change to Q, pk, N. Point

A on the characteristic corresponds to point A1 on the characteristic during throttling: r and r1 are critical points on these characteristics.

Throttling control on the suction line is about 6-8% more economical than on the discharge line, especially at high compression ratios. With this method, the area of unstable operation of the compressor is also reduced by the characteristic, which is important to ensure reliable operation of compressors. Taking into account these advantages, the method of regulating the operating mode by throttling the air in the suction pipe is widely used to maintain the required performance of compressors operated at mining enterprises.

The control by twisting the flow in front of the impeller with the help of a blade guide device in compressors is similar to the regulation of fans. In air centrifugal compressors, guiding devices are usually not provided, in connection with which this method is not used to regulate the operating modes of such compressors. However, the method finds application, for example, in freon compressors used in refrigerating machines. When regulated by this method, a gain in specific energy costs of up to 20% is achieved.

Regulation by changing the rotation speed of the compressor rotor is theoretically the most economical, but it is associated with the need to have an adjustable drive and is not used in the mining industry.

Thus, at present, air centrifugal compressors are regulated only by throttling the air in the suction nozzle.

3. Nomenclature and design of centrifugal compressors and installations

In the mining industry, centrifugal turbochargers of the Khabarovsk plant "Energomash" —K are used to compress air-500-61-1 , To-250-61-1 (discontinued) and to-250-61-2 and the Kazan Compressor Plant — CC-135/8 and CC-115/9. In the designation: K — compressor, C — centrifugal, the number after the letters indicates the compressor capacity in m3/min. The compressor unit (Fig. 19.4) consists of a turbocharger proper, intermediate refrigerators, a drive with a gearbox and a lubrication system.

Centrifugal turbochargers (Fig. 19.4) are multi-stage and are divided into sections of two or three stages (impellers) in each. Moreover, the wheels of one section are made of the same diameter, and the wheels of different sections are made of different diameters. Between the steps, compressed air is cooled by passing through intermediate refrigerators.

The impellers of modern centrifugal compressors are closed, i.e. they have front and rear discs; they are equipped with blades bent backwards with an output angle of 40-50 °; the number of blades is 14-28. Such wheels operate at relatively high circumferential speeds (200-300 m/s) and provide high pressures with relatively small dimensions. The rotor shaft is flexible, i.e. it is designed in such a way that its nominal rotational speed exceeds the critical speed. Due to the high rotor speed, turbochargers use plain bearings with babbit-filled steel inserts. Lubrication of bearings is forced.

The compressors are equipped with internal (along the cover discs of the impellers, shaft and dummies) and end (along the shaft) labyrinth seals, which reduce the flow of air, gas between the steps inside the machine, air suction and leakage from the compressor.

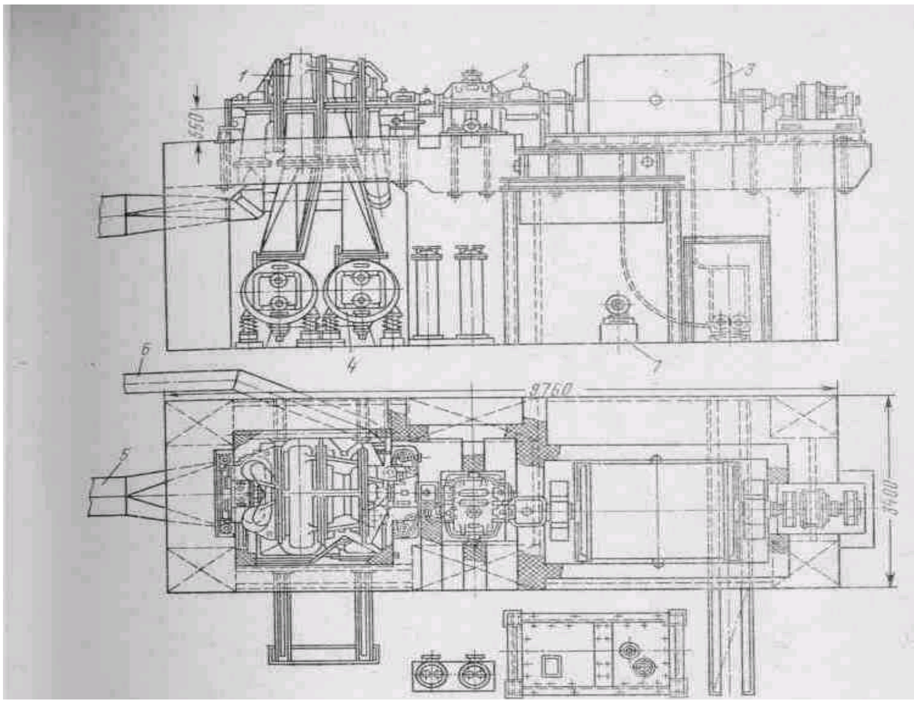


Fig. 19.4 Compressor unit To-500-61-1:
1- turbocharger; 2 — gearbox; 3 — electric motor; 4 — intermediate refrigerators; 5 — suction pipe; 6 — discharge pipe; 7 —oil pumps

Compressors To-500-61-1 (see Figure 18.6), to-500-61-2 and to-250-61-1 they have a similar design of nodes and similar characteristics. The compressors have six compression stages, their impellers are combined into three sections, two wheels per section. With the help of diaphragms 6, the inner chamber of the housing is divided into pressure stages. Each diaphragm is equipped with a channel diffuser and a reverse guide device.

The axial force acting on the rotor impellers is balanced by the pressure of compressed air on the discharge piston (dummis) 7 mounted on the shaft 2 behind the last impeller. The part of the axial FORCE that is not balanced with the help of dummis is perceived by a special thrust bearing 3. The compressor housing is cast iron, cast, with horizontal vertical connectors.

The compressors are equipped with forced lubrication of the rubbing parts and an automatic regulating device — a throttle valve in the suction pipeline, through which a constant discharge pressure is maintained, as well as a surge protection regulator that automatically releases excess air into the atmosphere when the compressor reaches critical performance.

The CC-135/8 and CC-119/9 compressors are structurally similar and differ in the gear ratio of the gearbox and the values of some parameters. A compressor of the CC type is an externally cooled machine without guiding devices with a spiral air outlet after each stage and an external air bypass from stage to stage. The stages are divided into two sections — low and high pressure, enclosed in two separate enclosures.

Compressor drives use single-stage gearboxes with two-stage chevron gear pairs and sliding bearings that increase the speed of rotation.

Intermediate air coolers of centrifugal compressors are shell—and-tube type. The cooled compressed air enters the inter-tube space and washes the bundles of tubes through which the cooling water circulates.

ROTARY COMPRESSORS

4. Classification of compressors Rotary are called compressor machines with rotational movement of the piston. These include a large group of compressors, in which the working body is one or two rotors. The process of gas compression in such compressors proceeds in a decreasing closed volume enclosed between a rotating rotor and a housing or between two rotors. They belong to volumetric type machines and

do not differ from reciprocating ones by the principle of gas compression.

There are screw compressors, rotary-plate, water-ring and two-rotor.

Advantages of rotary compressors: compactness at relatively high capacities, absence of working valves, uniformity of supply and dynamic balance,

Their disadvantage is the limited final pressure (no more than 1.2 MPa), which narrows the range of their application.

5. The principle of operation and the device of screw compressors Distinguish two types of screw compressors: dry compression that compress air (gas) not contaminated with oil, and oil filling, into the compression cavity of which oil is injected in large quantities (oil weight is 6-8 times the mass of compressed air), cooling compressed air (gas) and sealing gaps between working bodies.

Screw compressors (Fig. 19.5) belong to the group of volumetric machines in which air is compressed by reducing its volume in the working cavity formed by the teeth and depressions of two parallel rotor screws that do not come into contact with each other and with the compressor housing when rotating.

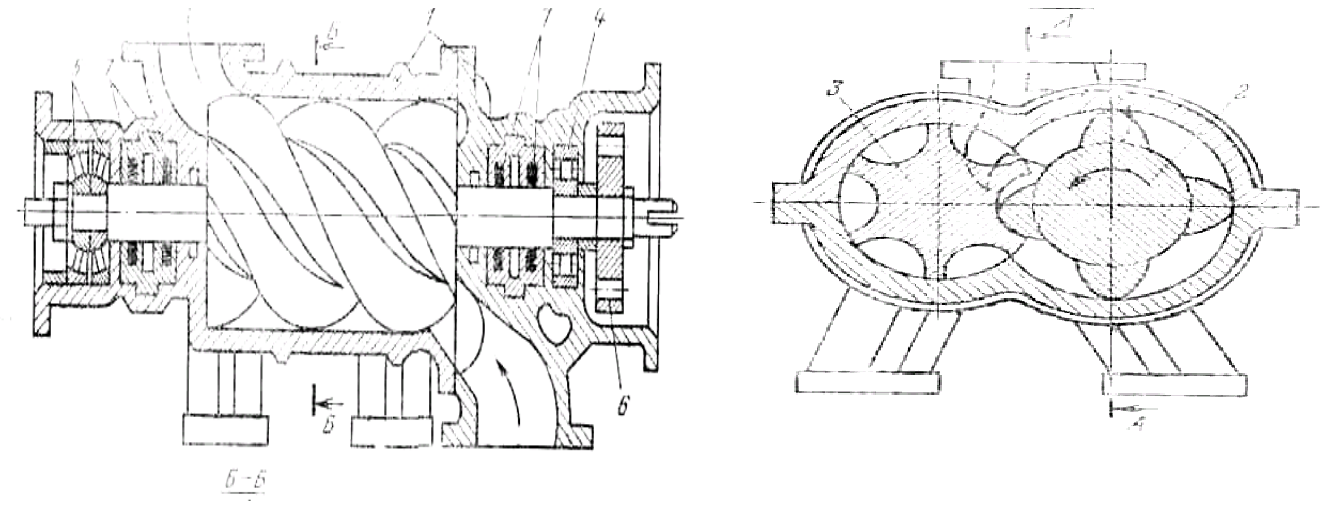


Fig. 19.5 Diagram of the screw compressor.
1- housing, 2-drive rotor, 3-driven rotor, 4, 5-rotor bearings, 6- gear, 7- seals

The air from the suction pipe enters the screw channels between the rotors by the body; after turning the rotors at a certain angle, the two cavities are connected to each other, forming a paired cavity. the air that has fallen into this space is isolated from the suction cabin, and then compressed when the teeth of one rotor gradually fill the cavities of the other. Compression continues until the continuously decreasing volume of the steam cavity with compressed AIR approaches the edge of the discharge window, i.e. until the cavity filled with compressed air is connected to the exhaust window. Compression ends, then the compressed air is ejected. The rotors are shaped like screws with a large lifting angle and, as a rule, a different number of teeth (for example, four on the master and six on the slave). The rotational speed of the leading 2 rotor is higher than the driven 3 (inversely proportional to the number of teeth). The leading rotor has convex teeth, and the driven rotor has concave teeth. The rotors rotate in sliding or rolling bearings and have gears, thanks to which their rigid kinematic connection is carried out. The necessary gap is provided between the rotors, eliminating direct contact between their screw surfaces and reducing their wear. At the ends of the screw part of the rotors, as well as on the outer diameter, sealing tendrils are provided, made as a whole by rotors or minted into milled narrow grooves.

5. The working process of the screw compressor and its performance. The working process in a screw compressor is similar to the working process of a piston compressor (suction, compression and injection). Its peculiarity is the constant value of the final pressure rc, regardless of the pressure pc in the discharge pipeline. The indicator diagram is slightly different from the diagram of a reciprocating compressor. When a screw compressor is operating in the design mode, the rc and diagram are similar to the theoretical diagram of a reciprocating compressor that has practically no harmful space (Fig. 19.6, a). If rc > pc, the gas pressure decreases when it exits the compressor (Fig. 19.6, b). At rc < rs, the pressure of compressed gas at the compressor outlet increases sharply (Fig. 19.6, b). In non-calculation modes, additional power costs increase.

Additional energy costs for compression are determined by the shaded areas of the diagrams.

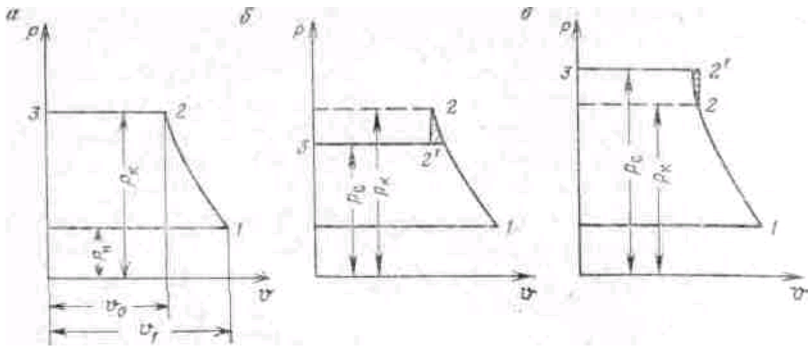


Fig. 19.6. Diagrams of the theoretical working process of a screw compressor at rc= rs (a), rc > rs (b) and rc< rs (c)

The compression process in screw compressors, due to the high rotor speed, occurs in a short period of time <0.01 sb and heat exchange between the walls of the compressor with compressed air practically does not have time to occur, Therefore, theoretically, the compression process in uncooled dry compressors (without injecting water or oil into the compression cavity) can be considered adiabatic. The actual process, due to the release of additional heat of friction, is polytropic with a polytropic index n = 1.45÷1.6. The compressed air is cooled in intermediate and terminal refrigerators. When oil is injected into the suction pipe, the polytrope index can be reduced for oil-filled compressors n=1,1-1,4. In the practice of screw compressor engineering, the degree of pressure increase 4÷5 is considered the most economical.

The performance Q (m3/min) of a screw compressor is determined by the expression

$$Q = (F_1 + F_2) z_1 \ln_1 \lambda \qquad (19.2)$$

where F1 and F2 are the cross—sectional areas of the depressions, respectively, of the leading and driven rotor, m2; z1 is the rotor; l is the length of the depressions, m; n1 is the rotational speed of the leading rotor, rpm; 0.8-0.9 is the coefficient characterizing the tightness of screw compressors.

LECTURE No. 20
SCREW AND ROTARY PLATE COMPRESSORS
Plan:

1.Screw compressors

2. Rotary plate compressors

3. Liquid-ring compressors

1.Screw compressors. Screw compressors at mining enterprises are used in stationary and mobile installations.

Installations with dry compression screw machines include a compressor, a multiplier reducer and a drive electric motor, air (gas) coolers, shut-off and control valves, noise absorbers, as well as systems for: water cooling, lubrication, automation and protection.

Installations with oil-filled compressors are additionally equipped with compressed air purification systems from oil. The compressed oil-air mixture enters the oil collector, where about 90% of the oil is separated. Next, the air enters the oil separator for repeated, deeper separation from the oil and is then pumped into the air duct to the consumer.

Mobile installations 6VKM-25/8, 6V KM-13/8 have become widespread at mining enterprises. The numbers after the letters indicate: in the numerator — productivity (m3 /min) in the denominator — final pressure (kg / cm2) and ZIF-SHV-5, developing a pressure of 5 kgf / cm2.

6B1 type stations<M are equipped with oil-filled screw compressors In K-11 with air and water cooling of oil.

The compressor station ZIF-SHV-5 is mounted on wheels and can move along rails.

Screw compressors have the following advantages over other types of compressor machines.

Compared with piston: high reliability and durability due to the absence of valves and parts that perform reciprocating movements, and parts with rubbing surfaces; uniformity of air (gas) supply, which eliminates the need for large-capacity air collectors; a smaller volume of harmful space (about 1%); significantly lower specific metal consumption and the dimensions of the installation; complete balance of the rotors and, as a result, no need for heavy foundations.

Compared with centrifugal ones: absence of surge zones on the characteristics; slight change in performance and efficiency when the degree of pressure increase varies within wide limits; reliability of operation on dusty gases; the possibility of compression of gases with a high content of liquid phase; insensitivity to shocks and shocks.

Screw compressors are 10-15 times lighter than reciprocating compressors and 3-4 times lighter than centrifugal compressors, with the same performance. The capacity of screw compressors can reach 1000 m3/min.

Along with this, screw compressors have a lower efficiency than piston compressors; their efficiency decreases when the pressure of compressed air behind the compressor deviates from the nominal. Screw compressors have a high cost due to the high accuracy class of processing screw profiles and create a lot of noise during operation, especially at a rotor speed of 15,000-18,000 rpm.

However, the advantages of screw compressors noted above determine the broad prospects for their use, including in mining enterprises.

2. Rotary plate compressors. The rotary plate compressor (Fig. 20) consists of a cylindrical body 3 and an eccentrically located rotor 2. The surfaces of the cylinder and rotor form a crescent-shaped space. Slots are provided on the rotor along its length, slightly inclined to the direction of movement. The thin plates inserted into the grooves, when the rotor rotates, are pushed out of the grooves by centrifugal forces, dividing the crescent-shaped space into separate closed chambers of different volumes. The gap between the plates and the cylinder surface is sealed with grease.

The volume of chambers between the plates is minimum at the bottom and maximum at the top. When the rotor rotates, gas enters from the suction nozzle, filling chambers A, compresses in chambers B due to a decrease in their volume and is pushed into the discharge pipeline. Figure 20.1, b shows an indicator diagram of the working process in a chamber enclosed between two plates. Gas compression occurs along the polytrope a —b during the rotation of the chamber from point a (maximum chamber volume) to point b, i.e. until the chamber is connected to the discharge pipe. Along the line b — c, the compressed gas is pushed into the pressure pipe.

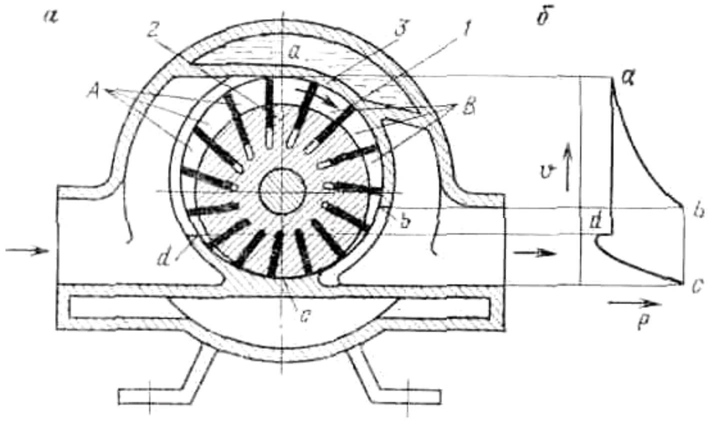


Fig. 20.1. Diagram of a rotary plate compressor (a) and an indicator diagram of the working process in the chamber (b)

The gap between the rotor and the cylindrical surface of the housing in its lower part forms a harmful space. When the rotor rotates from point c to point d, the gas residues enclosed in the harmful space expand in the chamber (polytrope c — d — in the indicator diagram). Starting from point d, the chamber is connected to the suction pipe and suction occurs (line d-a on the indicator diagram).

The ratio of the volume of the chamber in the upper position to the volume of the chamber before its connection to the pressure nozzle determines the degree of pressure increase e, which depends on the magnitude of the eccentricity of the rotor. At high e, the number of chambers is usually at least 20 in order to reduce the pressure drop between adjacent chambers, reduce gas leaks and increase the productivity coefficient.

The performance coefficient K shows the ratio of the actual Q and theoretical $Q_т$ productivity: $\lambda = \dfrac{Q}{Q_T}$ it depends on the size of the harmful space, the degree of gas heating in the suction zone and other factors (λ = 0,5÷0,8).

Rotary plate compressors are widely used as vacuum pumps. The main feature of their operation in the vacuum pump mode is a high degree of pressure reduction. For example, if a single-stage vacuum pump is able to create a vacuum 0,005 МПа, то e = 22.

The specific work of compression significantly depends on the degree of pressure increase.

The maximum operation corresponds to the degree of pressure increase: 3.24.

Rotary plate compressors have efficiency.: =0.62÷0.67 and = 0.75÷0.78. Advantages of rotary compressors: compactness and simplicity of design with relatively high productivity, lack of working valves and dynamic balance, uniformity of air (gas) supply, etc.; disadvantages: large friction losses and, therefore, increased wear of rubbing elements, difficulty of sealing at high pressures, the need for high precision processing and assembly. These disadvantages limit the relative speeds of the rubbing surfaces (15-20 m/s) and the final pressure, the second of modern plate compressors does not exceed 1.2 MPa.

3. Liquid-ring compressors. Liquid-ring compressors (Fig. 20.2), according to the principle of operation, are a kind of rotary plate compressors. When rotating the impeller (rotor) with blades eccentrically located in the housing 2, the liquid supplied to the housing is captured and driven into rotational motion. In this case, the liquid is thrown by centrifugal forces to the body, and a rotating liquid ring 5 is formed.

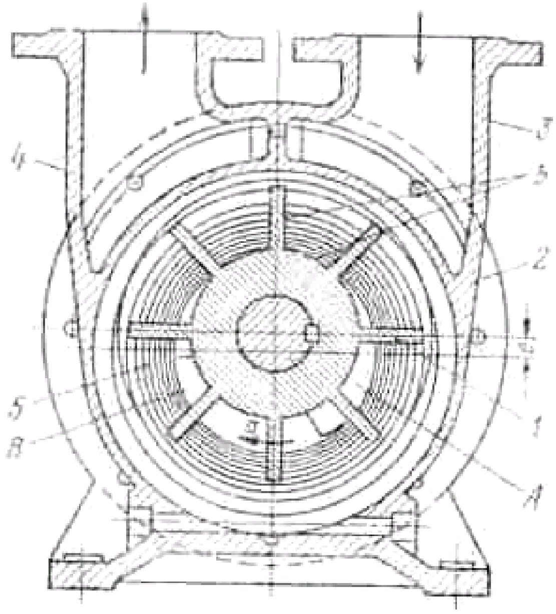


Fig. 20.2. Diagram of a liquid-ring compressor

Water is usually used as the working fluid. A crescent-shaped working space is formed between the rotor bushing and the water ring. The working process in the compressor is similar to the process of a rotary plate compressor — the suction of gas coming from the suction pipe 3 is carried out in chambers A, and its compression is carried out in chambers B, from which the gas is then pushed into the discharge pipe 4.

The amount of liquid (water) in the compressor should be sufficient to eliminate the gap between the cylindrical surface of the rotor and the liquid ring in the area between the injection and compression chambers to avoid the formation of harmful space. The rotor blades do not touch the cylindrical surface, as a result of which the wear of the blades, their friction and the associated energy losses are eliminated. This is the meaning of using a liquid ring, and in addition, the liquid in the compressor serves to cool the gas and seal the gaps.

Single-stage liquid-ring compressors develop a pressure of 0.25—0.5 MPa with a capacity of 0.003—2.4 m3/s. The rotor speed is up to 60 rpm. The disadvantage of compressors is their low overall efficiency. Due to the sealing of gaps with liquid, a high volumetric efficiency is provided. This circumstance has led to the expediency of using liquid-ring compressors as vacuum pumps to create a vacuum and suck gas out of formations. They allow you to achieve a vacuum of 98%.

Water-ring machines for compressing air or inert gases that do not dissolve in water are manufactured in two types: simple and double—acting and, depending on the purpose, in two versions: for operation as a vacuum pump - VN and compressor — K. To increase efficiency, the blades are bent backwards, and the inner surface of the body is elliptical.

LECTURE No. 21
OPERATION AND DESIGN
PNEUMATIC INSTALLATIONS

Plan:

1. Compressor tests
2. Maintenance of pneumatic installations
3. Methods of designing a pneumatic installation

1. Testing of compressors. The compressors are tested to determine their characteristics under operating conditions and to find out whether the factory data correspond to the actual ones, as well as to identify the causes of their deterioration. The tests are carried out under steady-state operating conditions of the compressors. At the same time, the performance and power consumption of the compressor engine are measured, the temperature regime and the pressure distribution by stages are set.

To determine the performance, methods are used to measure the flow of compressed air on the injection line (after the air collector) using throttle diaphragms, air collectors and indicator diagrams.

Pressure and pressure difference are measured by pressure gauges and diffmanometers.

When testing reciprocating compressors, due to simplicity, the method of determining pressure, productivity and power from actual compression diagrams in p-v coordinates recorded using mechanical pressure indicator devices, called indicator diagrams, has become widespread.

The cylinder 1 of the indicator (fig. 21.1, and) communicates with the cylinder 4 of the compressor. A spring-loaded piston 2 with a rod 5 and a pencil 8 drawing a diagram moves proportionally to the pressure in the compressor cylinder. At the same time, the bar 6 with the paper moves in a direction perpendicular to the movement of the pencil using the lever 7. The mechanical indicator has inertia. More precisely, the indicator diagram can be obtained using pressure and displacement sensors on the screen of the cathode ray tube of the oscilloscope.

The value of the average indicator pressure is determined based on the planimetry of the area of the indicator diagram according to the formula

$$p_И = r_P \frac{f_И}{s_u} \qquad (21.1)$$

where r_p— pressure scale; f_u and sи— accordingly, the area and the total length of the indicator chart.

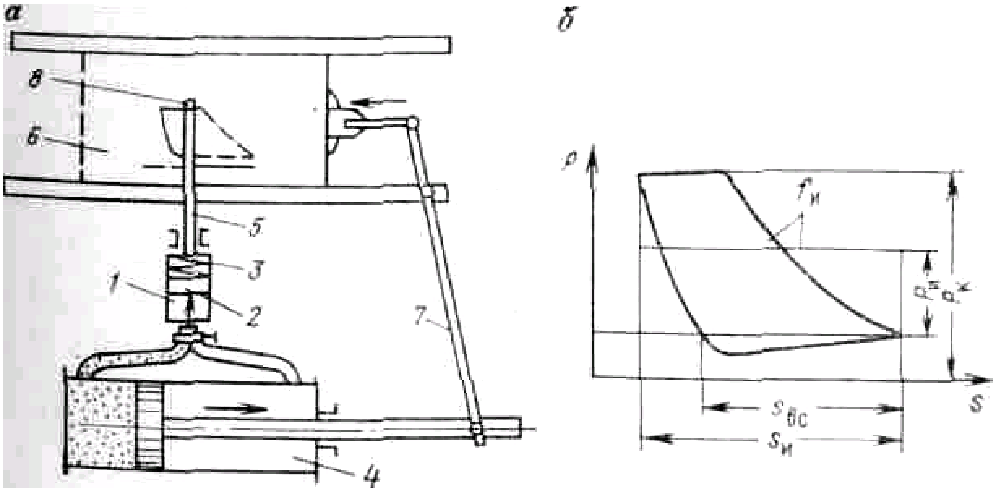


Fig.21.1. Indicator diagram (a) and indicator diagram (b): indicator cylinder; 2 — piston; 3 — spring; 4 — compressor cylinder; 5- rod; 6 — strip with paper; 7 — lever; 8 — pencil

Thus, when determining p, the area of the indicator diagram is replaced by an equally large rectangle with the dimensions of the sides equal to ri and si (Fig. 21.1, b). The pressure at the end of injection is determined directly from the diagram.

The performance is set approximately by the volumetric suction coefficient λ_0, equal treatment $\lambda_0 = \frac{s_{вс}}{s_u}$ where $s_{вс}$—the length of the suction segment along the atmospheric line, mm. Usually the value should be 0.85—0.95. Volumetric feed ratio of each cylinder cavity (0.94÷0.90). The performance of the compressor at the known is determined by the formulas.

Indicator diagrams allow you to determine the actual (indicator) power of the compressor. For a single-stage compressor with a double-acting cylinder, the indicator power (kW) is determined by the formula

$$(N_И = \frac{p_u(F_1 + F_2)sn}{1000 \cdot 60}) \qquad (21.2)$$

where F_1 and F_2 — working surfaces of the compressor piston, м^2;

s — piston stroke, м;

n- number of piston strokes per minute.

Indicator power of multi-cylinder single- and multi-stage compressors

$$N_{u\sum} = \sum_{i=1}^{n} N_u \qquad (21.3)$$

where n- number of cylinders (or number of stages).

According to the test results of reciprocating compressors, the indicator efficiency is also determined, which characterizes the deviation of the indicator power from the theoretical one. Depending on the chosen theoretical thermodynamic process, there are indicator KD, isothermal and adiabatic:

$$\eta_{и.из} = \frac{N_{из}}{N_{и}}; \qquad \eta_{и.ад} = \frac{N_{иэ}}{N_{и}}; \quad (21.4)$$

The value of the indicator efficiency of the compressor when calculating isothermal compression $\eta_{и.из}$= 0,75 ÷ 0,85, when calculating adiabatic compression $\eta_{и.ад}$= 0,9÷0,94.

Indicator diagrams allow you to identify malfunctions in the operation of reciprocating compressors. So, if the suction valve is loose, the compression line becomes more positive, and the expansion line is steeper; in the case of a leak in the discharge valve due to leaks of compressed air from the discharge air line into the compression cylinder, the compression line becomes steeper, and the expansion line is more positive. When the suction and discharge valves are pinched, the processes of compression and expansion, respectively, begin with a delay. With rigid valve plates, pressure losses in the valve increase. With weak plates, there is some delay in their landing, which causes the valves to knock.

For a simple-acting compressor, the wear of the piston seals (piston rings) causes a decrease in its performance and an increase in air temperature; the indicator diagram in this case has the same appearance as with a loose suction valve. In a double-acting compressor, the diagram looks different due to air flows in adjacent cavities.

When testing the turbocharger, the following are measured: temperature, air pressure at the inlet to the wheel section and at the outlet; power on the turbocharger shaft; cooling water flow through each intermediate refrigerator; the initial temperature of cooling water and its temperature at the outlet of the refrigerators, the volume of air sucked in by the turbocharger; air flow through the dummies; air pressure losses in intermediate refrigerators.

2. Maintenance of pneumatic installations. Operation of compressor stations consists in starting, stopping and monitoring their operation mode. The compressor is started in the following sequence: starting the cooling system pumps and the oil pump (if it has an independent drive), installing valves on the exhaust pipe, starting the compressor drive motor and then connecting the pipeline for normal operation.

During the operation of the compressors, the following are monitored: air temperature and pressure of compressed air, oil in the lubrication system, cooling water, bearing temperature and the load of the drive motor.

During the operation of reciprocating compressors, periodic purging of air collectors and air coolers, oil separators, as well as testing of safety valves are performed.

During the operation of turbochargers, the daily refrigerators are purged - and the serviceability of the anti-surge device is checked. Before stopping the turbocharger, open the exhaust valve and cover the throttle valve on the vasa.

To maintain the efficiency of compressors at the required level, regular cleaning of their cooling systems from silty deposits and scale is necessary. The cooling jackets of the cylinders of reciprocating compressors and the tubes of intermediate refrigerators are cleaned of silty deposits by intensive washing with water when small portions of compressed air are supplied. Cooling systems are usually descaled with a solution of inhibited hydrochloric acid HC1, which, interacting with the scale, forms water-soluble calcium chloride CaC12, water and carbon dioxide, which contributes to the loosening of the remaining scale, a significant part of which is removed during subsequent washing.

An important condition for preventing the formation of scale is the treatment of cooling water during the operation of compressors in order to soften it.

When operating reciprocating compressor units, there is a risk of explosions and fires in compressors and air ducts. The cause of explosions and fires, which often lead to serious consequences, is the spontaneous ignition of carbon deposits in compressor units and in air ducts.

The essence of the self-ignition process is as follows. The oil entering the cylinders of the piston lubrication compressor is carried away in a sprayed form by compressed air and settles as an oil layer on the surface of the air duct of the compressor unit and pipelines. Under the action of compressed air oxygen, oil deposits are oxidized with the release of heat, which is carried away by air at a steady temperature. With an increase in the thickness of the deposits and the air velocity, the heat transfer deteriorates and the oxidation reactions of oil deposits accelerate. If the amount of heat released is greater than the amount of heat given off by thermal conduction and convection, the temperature of the deposits increases and their spontaneous ignition may occur. During gorenje carbon monoxide is released, which in the presence of an open flame can give an initial explosion. Subsequent explosions occur under the influence of a shock wave, which, spreading at high speed, tears off the oil film from the walls of the pipes, which then evaporates and sprays in the form of fog. In some cases, gorenje is not accompanied by an explosion, but the carbon monoxide released at the same time, coming together with compressed air to the workplace, creates a danger of poisoning workers. The amount of carbon-oil deposits depends on the amount of lubricant supplied to the cylinders, the brand of oil, the duration of operation of the compressor without cleaning, dustiness of the intake air and other reasons. The condition for safe operation during the operation of reciprocating compressors is periodic cleaning of the discharge lines from the accumulated carbon-oil deposits in them.

1. It is necessary to clean the air cavities of compressors, terminal and intermediate coolers, air collectors and air ducts from carbon-oil deposits at least once every six months with 5-10% caustic soda solution or 3% sulfone solution. The solution is fed into the injection communications by pumps and circulates for several hours until the communications are completely cleaned. After that, the communications are washed with water from the water main and purged with air from the compressor.

3. The method of designing a pneumatic installation. The design of a pneumatic installation includes: determination of the main parameters of the compressor station (performance and pressure), selection of the type and number of compressors, their drive motors, starting equipment and automation equipment; drawing up a diagram of the pneumatic network and its calculation; technical and economic calculations of the main indicators of the pneumatic installation and mining enterprise.

The basis for the design of service d: mining plans quantity, location, characteristics and operating modes of compressed air.

The performance of the compressor station Qk.c s (m3/min) is calculated based on meeting the compressed air needs of all its consumers working in shifts with the highest air consumption, and compensating for compressed air leaks in the network:

$$Q_{k.c} = Q_n + Q_{ym} = \mu \sum_{i=1}^{я} n_i q_i \varphi_i r_в r_3 + q_{ym} \sum l + q'_{ym} m \qquad (21.5)$$

where $Q_п$ — compressed air consumption by consumers;

$Q_{ут}$ leaks of compressed air in the pneumatic network and in places where consumers are connected;

$\mu =$ 1,05÷1,1 — stock ratio for unaccounted mechanisms;

I — group number of the same type of compressed air consumers;

z — number of consumer groups; n_i — number of consumers in the group;

q_i — nominal air consumption by one consumer of this group during its continuous operation;

φ_i — the coefficient of increase in the consumption of compressed air by the consumer as a result of its wear, is taken for jackhammers and drill hammers and piston engines to be equal to 1.15, for gear engines 1.2, for turbine engines 1.0; gw— the coefficient of simultaneity of consumers (the ratio of working consumers to their total number), is approximately taken for jackhammers and drill hammers, depending on their numbers:

Number of hammers ……….. 10 11—30 31-60 60

$r_в$ ………………………1—0,85 0,85—0,75 0,75-0,65 0,65

for pneumatic engines, it is selected depending on their operating conditions, for drives with high air consumption (combines and others) it is accepted in the same way as for hammers;

r_3 — the load factor, which takes into account the change in the consumption of compressed air by the consumer due to the difference in the actual load from the nominal and during regulation, is accepted for the drives of combines, winches, drilling machines, jackhammers and drill hammers equal to 1.0, rock loading machines -0.25 and fans of local ventilation — 0.7;

$q_{ут}$ и $q'_{ут}$ — accordingly, the permissible leaks of compressed air through the leaks of the main air pipeline for 1 km of its length l and at each point of connection of the consumer, (m3/min) m is the total number of consumers connected to the pneumatic network, including non—working,

Leakage rates qut and q'day at an overpressure of 0.5 MPa are taken respectively 3 and 0.4 per (m3/min) 1 km of air pipeline and per consumer unit.

The expression (21.5) is inaccurate, since the coefficients gb and r3 can be taken only approximately.

The pressure of compressed air at the outlet of the compressor station is determined based on the condition of ensuring the working pressure pp at the most remote consumers and the permissible pressure loss of the PC network. Approximately, it is possible to take the average value of specific (per 1 km of pipeline length) pressure losses in a metal pipeline equal to 0.03 MPa, regardless of the flow rate and pressure loss in the hoses of the lava air pipeline. In this case, the required pressure p (MPa) of compressed air at the compressor station is determined by the formula

$$p = p_p + 0,03L + 0,03 \qquad (21.6)$$

where L is the distance from the compressor station to the most remote consumer, km. Usually, the required air pressure of a compressor station installed on the surface is 0.8—0.9 MPa, and mobile ones — at least 0.6 MPa.

The choice of the type and number of compressors is made according to the calculated values of performance and operating pressure, as well as the advantages and disadvantages of individual types of compressors. Currently, based on efficiency, they are guided by: at the working capacity of the compressor station 200-500 (m3/min) — 4M10-100/8 reciprocating compressors, at the station capacity over 500 — K-500 centrifugal compressors. Compressor stations with a capacity of less than 200 (m3/min) are equipped with the same type of reciprocating compressors 2M10-50/8, 305VP-30/8, 202VP-10/8, etc. Centrifugal compressors are recommended to be used together with a certain number (up to 25% in performance) of reciprocating compressors, which together with air supply a certain amount of oil to the pipeline, which reduces corrosion of the internal surfaces of pipes and equipment.

The calculation of pneumatic networks due to their branching of a large extent and changes in the parameters of compressed air during its movement in air ducts is a complex task associated with technical and economic analysis. The most complete solution to this problem can be carried out by modeling methods on digital computers or analog machines.

Simplified methods for calculating pneumatic networks using nomograms have been developed. The calculation of pneumatic networks according to these methods is reduced to determining the diameters of the air ducts according to the specified pressure losses on individual sections of the network and finding the actual pressure losses with accepted standard pipes,

The pressure losses are set based on the need to ensure sufficient pressure from consumers and the total regulatory pressure drop of ah in the network is no more than 0.20 MPa. Based on the operation of pneumatic networks and calculations, it has also been established that the most economically and technically advantageous is the average air velocity in pipelines of 6-8 m/s.

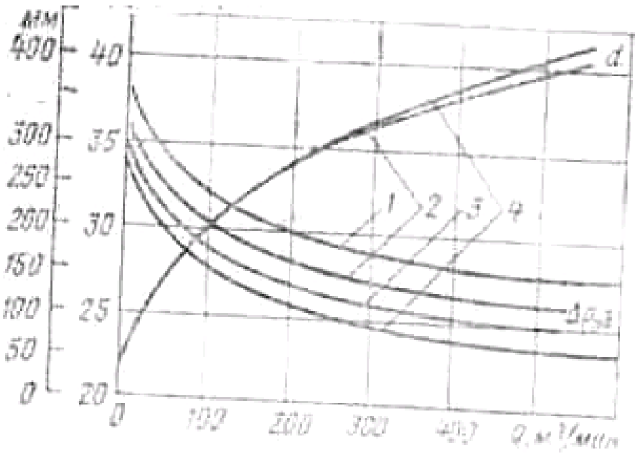


Fig. 21.2. Optimal pressure losses and the diameter of the pipeline d of the air supply network: 1, 2, 3, 4 — at an initial pressure of 0.4; 0.5; 0.6; 0.7 MPa, respectively

Approximately the diameter of the pipeline and the optimal value of specific (per 1 km) losses can be determined according to the graphs shown in Fig. 21.2.

Technical and economic indicators of a pneumatic installation give an assessment of the efficiency of using compressed air energy at a mining enterprise. The most important indicators are: the design cost of 1 m3 of air, its specific consumption per 1 ton of extracted minerals, operational and capital costs for a pneumatic installation.

The average cost of 1 m3 of compressed air (in a free state) is approximately 0.1 kopecks.

About 0.1 kWh of electricity is consumed to generate 1 m3 of compressed air. The actual consumption of compressed air is 30-90 m3 per 1 ton of ore in the Krivoy Rog basin and about 80 m3 per 1 ton of coal in the Donetsk basin.

An important technical indicator of a pneumatic installation is its efficiency., equal to

$$\eta_{П.у} = \eta_{ку_у}\eta_с\eta_д \qquad (21.7)$$

$\eta_{к.у}$ — the full efficiency of the compressor unit, which takes into account the efficiency of the compressor and the drive motor, as well as the energy costs of the compressor unit lubrication system,

$$\eta_{к.у} = \frac{N_В}{N_{k.c}}$$

$N_В$ – the power of compressed air at the outlet of the compressor (compressor station), kW;

$N_{k.c}$ — the power consumed by the compressor station from the network, kW. The actual total efficiency of the compressor station is in the range of 0.45÷0.60;

η_c = 0.4—0.5 is the total efficiency of the pneumatic network, taking into account energy losses caused by air leaks, heat exchange by hydraulic resistances;

$\eta_д$ = 0,2-0,4 — full P.D. of pneumatic motors of consumers.

The total efficiency of the pneumatic installation is low and is 0.10÷0.25 (calculated). The actual efficiency of pneumatic installations at mining enterprises is lower and is 0-by—0.08, which is due to large leaks of compressed air in the pneumatic network, a decrease in the efficiency of compressor installations I; pneumatic engines.

Technical and economic indicators of pneumatic installations of mining enterprises can be improved by increasing the efficiency of installations and, first of all, reducing leaks of compressed air, utilization of heat removed during cooling of compressors and other measures.

BRANCH OF THE FEDERAL STATE AUTONOMOUS EDUCATIONAL INSTITUTION OF HIGHER EDUCATION "National Research Technological University "MISIS" in Almalyk

DEPARTMENT OF "MINING"

METHODICAL INSTRUCTIONS
for performing laboratory work

in the discipline "STATIONARY MACHINES"

Almalyk 2022

LABORATORY WORK No. 1

STUDY OF DESIGNS OF CENTRIFUGAL TURBOMACHINES

The purpose of the work is to study the principle of operation and the main elements of turbomachines, the kinematics of fluid flow in the impeller.

1. The principle of operation and the main elements of turbomachines. Turbomachines used in the mining industry: pumps designed for pumping and supplying water; fans that ventilate mine workings; turbochargers that produce compressed air are characterized by a single operating principle. Depending on the direction of fluid flows relative to the axis of rotation of the impeller, they can be centrifugal, axial and meridional (diagonal). In the mining industry, the last group of turbomachines has limited use. Axial turbomachines are used in mining enterprises mainly as fans.

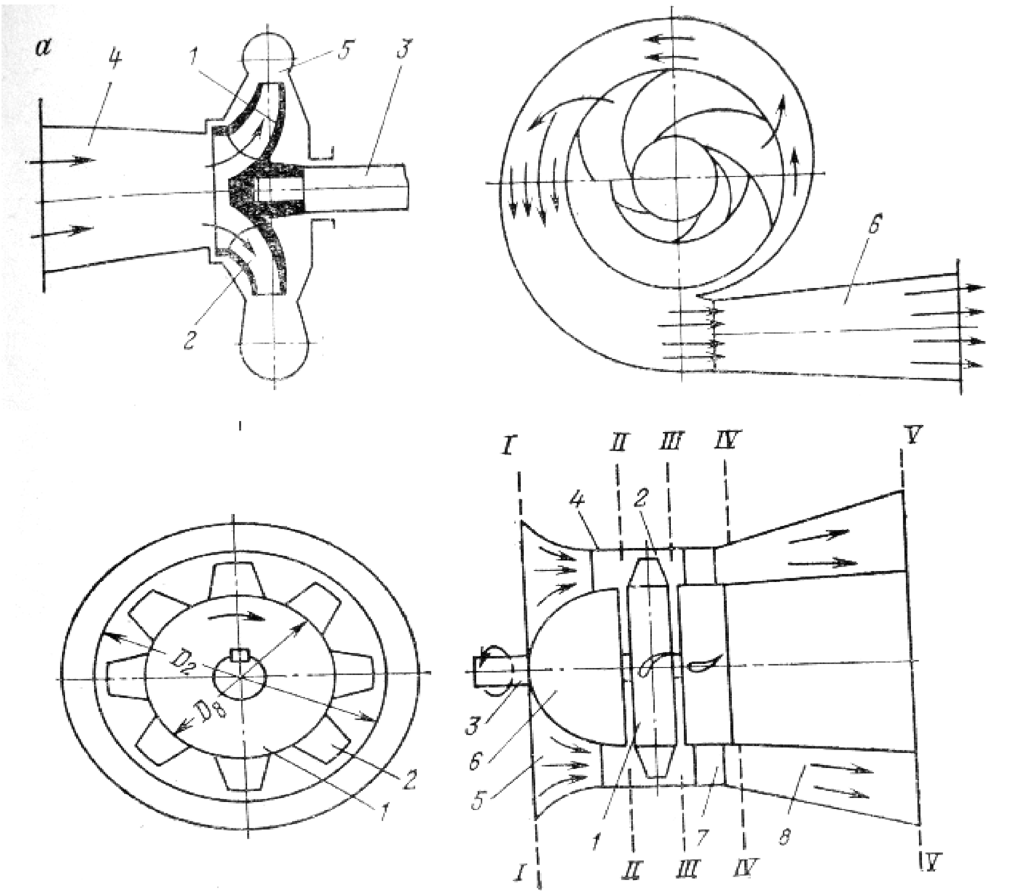


Figure 1.1 Centrifugal turbomachine

The centrifugal turbomachine (Fig. 1.1) consists of an impeller / with blades 2, mounted on a shaft 3, an inlet device 4, a spiral snail-shaped outlet device 5 and a diffuser 6.

The fluid flow is brought to the impeller in the axial direction and at the entrance to the latter changes its direction and in the inter-blade channels of the wheel moves already in the radial direction, moving along the blades from the entrance to the wheel to the exit from it.

A centrifugal turbomachine can have a one-way suction impeller, i.e. with liquid supply to the wheel on one side, and with two-way suction, i.e. with two-way liquid supply, to increase productivity (supply).

The impeller blade is a wing — a slightly curved, conveniently streamlined body with a rounded part running into the flow and a pointed end, and the impeller is a lattice of such jointly working wings. The designs of the blades of centrifugal and axial turbomachines have significant differences.

To reduce the turbulence of the fluid flow at the inlet and the shockless entry into the impeller, a special fairing 6 is installed in front of axial turbomachines, while the centrifugal fairing is performed at the same time with the impeller (see Fig. 1.1, a).

The supply device (supply) provides liquid supply to the impeller with a uniform, if possible, flow velocity field along its cross section.

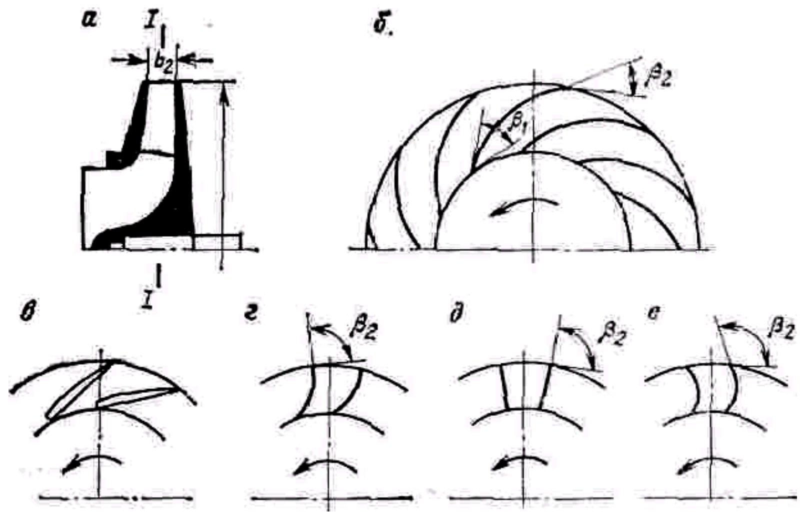
The purpose of the discharge device is to collect the flow coming out of the impeller at high speed, convert its kinetic energy into potential pressure energy and divert the liquid to the discharge pipe or the next impeller. In the removal of axial machines, partial or complete unwinding of the flow twisted by the impeller can also occur. The flow in the outlet due to the smooth expansion has a diffusor flow character, i.e., the fluid velocity decreases and the pressure increases. If there is a diffuser behind the discharge device in the latter, the flow velocity is further reduced and the kinetic energy of its movement is converted into potential energy (static pressure).

It should be noted that the increment of the specific energy of the flow occurs only in the impeller, in the other elements — energy conversion and reduction of the total pressure due to energy losses to overcome resistances.

Axial shaft turbomachines are performed only with sequential connection of impellers. Centrifugal and axial turbomachines are usually combined into one group of vane (blade) machines. This is due to the fact that they can be considered as limiting cases of diagonal machines (Fig. 1.2, b). In this view, a centrifugal turbomachine is a diagonal machine with an angle of □ = 90° (Fig. 1.2, a), and an axial one with an angle of □ = 0° (Fig. 1.2, b). Such unity does not exclude significant design differences between axial and centrifugal machines.

2. Kinematics of fluid flow in the impeller. The fluid movement in the flow channels of turbomachines has a very complex spatial character. The flow parameters vary both along the width of the wheel and along the circumference of a fixed radius.

Figure 1.4. Turbomachine impellers diagrams



1.3. Centrifugal impeller (a) and blade profiles: backward curved leaf (b); wing-shaped (c); radial curved leaf (d); flat (e) and curved forward (e)

To simplify the three-dimensional model of fluid flow in the impeller is replaced by a two-dimensional one that preserves the basic properties of the flow. Such a model is used, in particular, when considering the kinematics of the flow, choosing as its kinematic parameters the velocity of liquid particles near the inlet and outlet edges of the blades. The values of velocities are understood as their values averaged over the pitch and width of the interscapular channel.

The centrifugal impeller of a turbomachine has an inlet section for fluid flow in a plane perpendicular to the axis of rotation, and an outlet in a cylindrical surface with an axis coinciding with the axis of rotation.

To obtain a two—dimensional model of the flow in a centrifugal wheel, it is conditionally dissected by a plane /-I perpendicular to the axis of rotation (Fig. 1.3, a). In this case, the sections of the blades forming a radial (circular) lattice are obtained (Fig. 1.3, b).

The fluid flow in the radial grating is assumed to be plane-parallel, i.e. the same in width of the wheel.

The analysis of the flow kinematics within the impeller is based on the construction of parallelograms of fluid flow velocities at the inlet and outlet of the impeller. To build them, it is necessary to know the magnitude and direction of the speeds, which are determined by the dimensions of the impeller, the geometry of its flow channels and the operating mode. At the same time, the shape and profile of the working blades have a decisive influence. They are performed bent backwards, $\beta < 90°$ (Fig. 1.3, b, c), radial, $\beta 2 = 90°$ (Fig. 1.3, d, e) and bent forward, $\beta 2 > 90°$ (Fig. 1.3, e), in cross section—profiled (Fig.1.3, e) and thin, practically unprofiled (leaf) (Fig.1.3, d, d, e). In Fig. 1.3, β1 and β2 denote the input and output angles of the blades between the tangents to the circles of the gratings and the blades at their input and output edges. Passing through the impeller during its rotation, the fluid participates in portable (together with the impeller) and relative (relative to the wheel) movements with velocities and and w. The absolute velocity of the fluid particles is equal to the geometric sum:

$$\vec{c} = \vec{u} + \vec{w} \qquad (1.1)$$

Absolute velocity is the velocity of a liquid particle relative to a stationary body.

The portable speed and the absolute value is equal to

$$u = \frac{2\pi rn}{60} \tag{1.2}$$

and is directed tangentially to the circle of radius g; p is the speed of rotation of the impeller.

The relative velocity w, with which the flow moves in the interscapular channels, also varies in magnitude and direction.

Axial impeller of the turbomachine. Unlike the wheel of a centrifugal machine, the sections of the inlet and outlet of the fluid flow of the axial impeller are in planes perpendicular to the axis of its rotation. The liquid moves through the wheel translationally and simultaneously twists in the direction of rotation.

We dissect the impeller (Fig. 1.4, a) with a cylindrical surface with radius r and select an annular trickle of liquid with a thickness of Δr, within which the flow parameters (velocity and pressure) can be considered constant (due to the smallness of Δr).

Turning the cylindrical surface of the cut into a plane, we get the so-called flat grid of profiles (Fig. 1.4, b) of the axial impeller. The main parameters of this lattice are: the width of the blade (chord length) b; the width of the lattice B; the number of blades z; the angle of installation of the blade θ, formed by its chord and the velocity vector and; the angles of entry and exit of the blades $\beta 1$ and $\beta 2$. An important parameter is the lattice pitch t = equal to the distance between the similar points of the blade sections measured in the direction of rotational motion of the lattice. The ratio b/t is called the lattice density, and t/b is the relative pitch.

When the impeller rotates, the particles of the liquid flowing through the lattice participate in relative motion along the lattice (with a relative velocity w1 at the entrance to the lattice and w2 at the exit from it) and in portable motion — with a circumferential velocity and = wr. At a constant angular velocity y for a cylindrical surface of a given radius r, the velocity i = const.

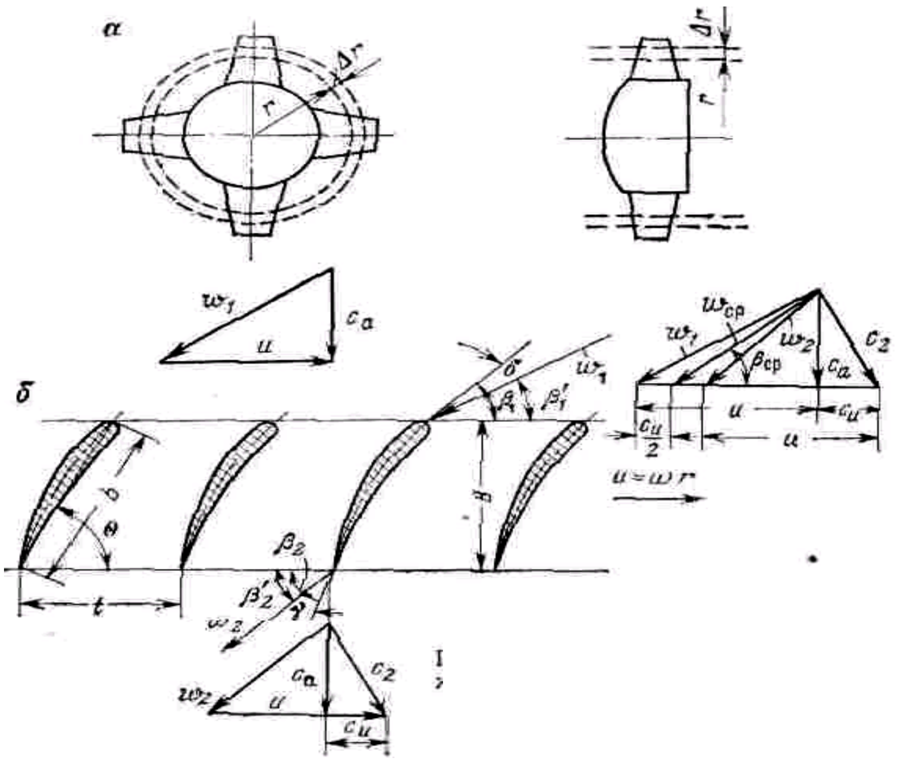


Figure 1.4 Velocity plans of the fluid particles of the axial impeller

In the absence of twisting of the flow in front of the impeller , the liquid flows to the grate at an absolute speed and has an absolute speed at the outlet of the grate . Figure 1.4 shows the triangles of velocities at the entrance and exit from the grid.

Based on the flow continuity equation for incompressible ca1 and ca2 fluid, it can be proved that the axial velocities of ca1 and ca2 at the inlet and outlet of the impeller of a turbomachine are the same: ca1 = ca2 = ca is the speed at which particles move along the axis of the impeller.

The relative velocity w1, at the entrance to the lattice is directed at the angle of attack δ — the angle between the tangent to the median line of the blade and the relative velocity at the entrance.

Passing through the lattice, the fluid flow from the interaction with the blades bends and the relative velocity w changes its direction, deviating towards the rotation of the lattice. N. E. Zhukovsky and S. A. Chaplygin showed that the curved flow by the interaction effect can be replaced by an equivalent rectilinear flow with an average relative velocity

$$\vec{w}_{cp} = \frac{\vec{w}_1 + \vec{w}_2}{2} \qquad (1.6)$$

This conclusion is important for analyzing the working process of an axial turbomachine.

By combining the velocity triangles of the liquid particles at the inlet and outlet of the lattice, we obtain a velocity plan from which we determine the angle of inclination of the velocity vector βcp and its absolute value:

$$tg\beta_{cp} = \frac{c_a}{u - \frac{c_u}{2}} \qquad (1.7)$$

$$w_{cp} = \sqrt{c_a^2 + \left(u - \frac{c_u}{2}\right)^2} \qquad (1.8)$$

where cu is the projection of the absolute velocity vector c2 on the direction of the vector and .

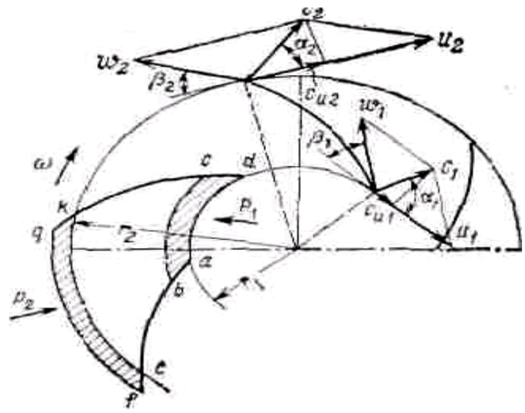


Figure 1.5 Diagram for the derivation of the Euler equation

The cu velocity is called the flow twist velocity.

If the absolute value of w1 > w2, then the lattice has a retarding effect on the flow and is called diffusor. If the relative flow velocity in the impeller increases (w1 <w2), then the lattice is called confusory, at a constant velocity w (w1 = w2) — active. In shaft fans, the diffusor grille has received the greatest use, the active one is practically not used.

The theoretical productivity (feed) of the QT (m3/s) of the impeller is determined by the expression

$$Q_T = c_a \pi / 4\left(D_2^2 - D_B^2\right) \qquad (1.9)$$

where dB is the diameter of the impeller sleeve, m.

LABORATORY WORK No. 2
STUDY OF CENTRIFUGAL PUMP DESIGNS

The purpose of the work is to study the design of centrifugal pumps.

GENERAL INFORMATION

With the transition of developments to deeper horizons, the existing type of pumps could not satisfy the field of possible modes. Therefore, multistage pumps of the CNS type have become widespread (Figure 2.1).

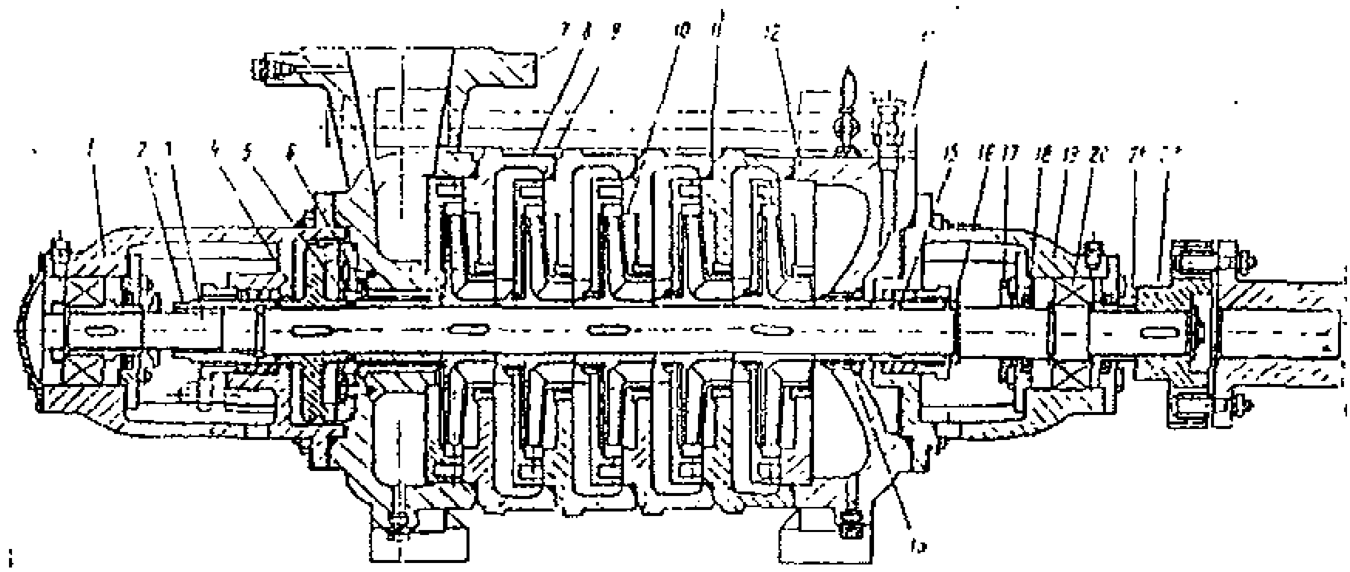


Figure 2.1 Sectional centrifugal pump

According to GOST 10407-70, the designation of pumps includes the initial letters - the name of the Central nervous system pump. After the letters - the nominal supply of the pump is set in the design mode (m3), the pressure at the minimum and maximum number of steps (m). The letters after the Central nervous system mean that the acid-resistant pump is K (for operation in an aggressive environment), for hot water - G and M - for operation in oil. For example: TSNSK-500-100...800 - sectional acid-resistant centrifugal pump with a supply of 500 m3/h with a head with a minimum number of steps of 160m, with a maximum of 800m.

The structural elements of the pump body and impeller are made of modified cast iron when working with a neutral liquid or from chromium-nickel steel in an aggressive environment. The pump consists of the following main components: housing, rotor, support brackets, tie pins (Figure 2.1).

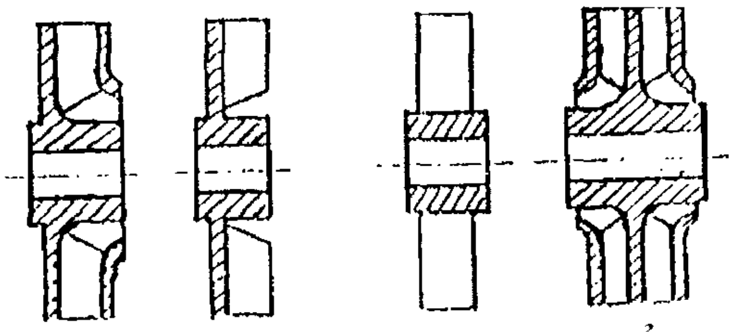


Figure 2.2. Pump impellers designs

The rotor includes a shaft 3, on which impellers 10, an unloading disc (hydraulic heel) 4, a remote sleeve 6 and a rotor nut 2 are planted by means of dowels.

The pump housing is sectional, the housing section consists of guide devices (disks) 8, suction covers 12 and discharge 7, front 19 and rear brackets 1. The housing of the guide devices are tightened with tie pins 14, their joints are sealed with annular rubber seals P.

The impeller is the main element of the design of a centrifugal pump, since it converts the mechanical energy of the pump shaft into the energy of the fluid flow. The impeller is made by casting. It consists of a dullard, anterior and posterior discs, between which there are shoulder blades. Depending on the physical and mechanical properties of the pumped medium, the wheels are made closed, semi-open or open, with one-sided and two-sided liquid supply (Fig 2.2).

The shaft is a node that transmits torque from the engine to the impeller. The shaft is steel, forged, chiffon, heat-treated, has poses for installing dowels. Support necks are made to fit the bearings on the shaft. The shaft is connected to the electric motor by means of.

Bracket - serves to perceive the weight of the rotor. The bracket is made in the form of a solid cast iron structure or chrome-nickel steel. The flange of the bracket is attached to the lid by means of bolts. At the other end of the bracket there is a support shoe on which a shaft with a bearing sits.

The covers are cast from cast iron or chrome-nickel steel. They are the binding or supporting element of the pump. Between the covers, sections are assembled into a single block under the means of tie pins. The front cover is made taking into account the placement of the shaft outlet, the hydraulic seal sleeve and sealing seals designed to isolate the suction chamber in it together.

The back cover is cast together with the discharge pipe and has a snail-shaped chamber inside. The chamber is connected to the discharge side of the last impeller.

Guiding devices - serve to direct the flow of fluid coming from the previous wheel to the next one and to partially transform the dynamic component of the pressure to the static one. The guiding device is carried out together with the housing or separately. If performed separately, then it is planted in the section. The guiding device is cast from the same material as the impeller and is equipped with internal blades.

Oil seals and hydraulic seal. Together with the shaft outlet from the housing, there is a mechanical and hydraulic seal.

The mechanical seal is the oil seals. They are installed in the front and rear pump covers. Oil seals are made of cotton or asbestos bundles impregnated with thick fat. The hydraulic control or hydraulic lock is installed on the suction side of the pump (Fig. 2.3).

The water coming out of the discharge device is fed through the bypass pipe 9 through the hole 8 in the suction cover 1 into the cavity of the hydraulic seal 7, forming a liquid ring during operation of the pump, preventing air from being sucked out of the atmosphere through the stuffing box 5. From the cavity of the hydraulic seal, part of the water seeps through the jacket of the shaft 6, cooling the oil seal, the other part enters the cavity of the suction cover at the entrance to the first wheel through the gap between the sleeve of the hydraulic seal 4; excess water to lower the pressure on the oil seal is discharged through the fitting 2 or sent to the supply pipeline.

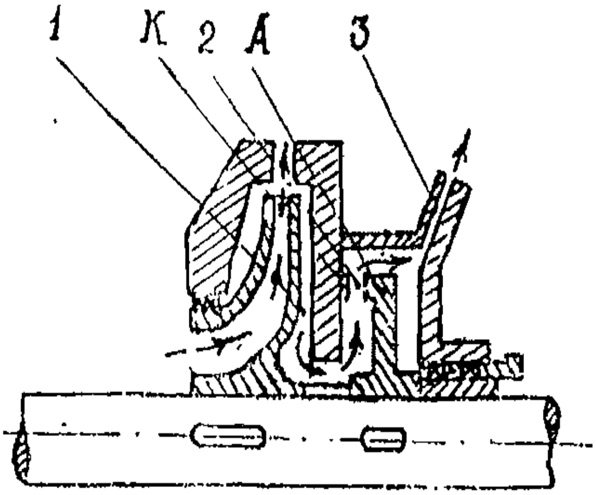


Figure 2.3. Diagram of the unloading device

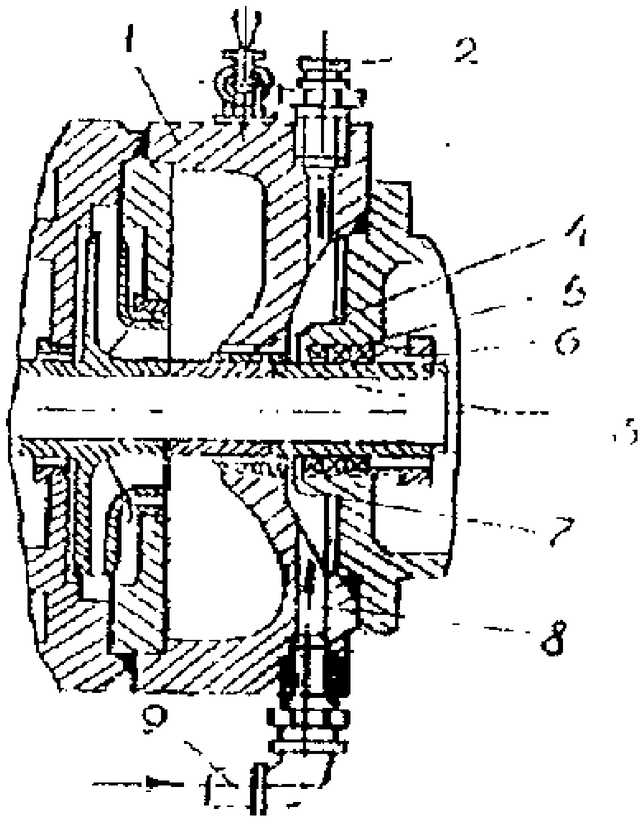
Unloading disk. The unloading disc A (Fig. 2.3) is mounted on the shaft by means of a key behind the last impeller in the unloading chamber and can move axially with the rotor.

The unloading device is designed to prevent the axial force generated when the fluid enters the impeller. The disk divides the unloading chamber into two cavities: a high-pressure cavity (Fig. 2.4) and a low-pressure cavity (right).

When the pump is running, part of the liquid, leaving the last impeller, enters the left chamber cavity through the radial gap 1. Then it follows through a narrow shawl gap 2 into the right cavity and finally, through the tube 3, it is discharged into the atmosphere.

During the operation of the pump, there is a pressure in the left cavity equal to the pressure of the last stage. The liquid passing through a narrow slit gap 2 loses most of its energy, and a pressure close to atmospheric pressure acts on the right surface of the hydraulic disk. When the tube 3 is open, the pressure difference on both sides of the disc causes a force opposite to the axial one. The magnitude of this force will depend on the area of the discharge disk and the gap 2 between the ring K and the disk A.

The unloading device automatically adapts to changes in axial force. When the axial force increases, the shaft with impellers and discs moves to the suction sides. At the same time, the gap 2 decreases, as well as the amount of water flowing out of the tube 3. The pressure on the disc increases and it moves in the direction opposite to the axial force.



If the axial force decreases for some reason, a nasty displacement of the disk occurs (Fig. 2.4). The slot gap 2 increases, which entails a decrease in the magnitude of the balancing force. The longitudinal displacement of the shaft and the hydraulic disc automatically ensures the equilibrium of the axial and balancing forces.

LABORATORY WORK No. 3
STUDY OF GROUND PUMP DESIGNS
The purpose of the work is to study the design of ground pumps

In accordance with GOST 1965g. the symbol of the pump brand is as follows: 20GRT-8. The numbers before the brand indicate the diameter of the suction pipe in inches, and after the brand - the speed coefficient of 80.

20GRT-8 is a heavy-duty double-hull ground pump with a suction pipe of 20 inches (500mm) and a coefficient of 80.

According to design features, they are divided: with an increased cross-section size by 25% compared to the nominal value (Y) with a reduced cross-section (O) with wear-resistant metal and rubber parts (P) corundated against abrasive wear (K) single (L) and double-hull (T). In double-hull pumps, the inner housing they are made replaceable from wear-resistant metal.

Currently, the marking of ground pumps has been changed. The letter designations remained the same (GRU, GR, GRT), but instead of the numbers that characterized the diameter of the suction pipe and the speed coefficient, the pump water supply (m3/h) and pressure (MPa * 10) are now indicated behind the letter designation. For example, a soil pump 16GRU1600/50. This is a pump with a supply of 1600mZ /h and a pressure of 0.50 MPa.

The index "L" is not indicated in the new designation. In the absence of the "T" index, it is assumed that the ground pump has a single-body design.

Using the example of GRK1600/50 ground pumps, we will consider the designs of pumps of a single series.

At the GRK1600/50 ground pump, the housing has a vertical connector, the front 3 and rear 6 halves of the housing are connected by bolts 1. There are no front and rear covers. The GRK1600/50 pump has no protective discs (armored discs) and an internal housing. They are replaced by a corundum lining 4 which covers both parts of the body and the suction pipe. Corundation is carried out by applying corundum grain on a bakelite bundle. The impeller 5 is made of a special alloy, it is attached to the shaft 14 with a nut 2 and a lock nut. The gap between the impeller and the front of the housing is adjusted by moving the shaft using adjusting bolts. The pressure pipe is directed vertically upwards. The stuffing box seals are located in the cavity 9 of the housing 6 and consist of a water distribution ring 10, a stuffing box 11 and a pressure ring 12 formed from two halves. Clean water at a pressure of 0.7 MPa with a flow rate of 25 m/h is fed through the hole 8 into the oil seal housing. The supporting part of the pump is the control of the frame 7, to which the housing is attached on studs. The shaft. rotates in supports located in the frame. One support consists of a spherical deep groove roller bearing 13, the other consists of a roller bearing 16 and a thrust ball bearing 15. The bearings are lubricated with grease, carried out through a "press-oiler 17. The pump is connected to the electric motor by an elastic coupling 18.

The GRK1600/50 pump, like all corundated pumps, can be used for hydraulic transportation of abrasive soils (sand, slag, ash), but in the absence of large-scale inclusions such as gravel, pebbles, because corundum, if present, quickly chips off.

The principle of operation of ground pumps is the same as water pumps. They are cantilever type centrifugal pumps with one-way liquid supply.

However, the operating conditions of ground pumps differ significantly from the operating conditions of water pumps, therefore they are structurally solved differently. Due to the presence of abrasive materials and large inclusions in the hydraulic mixture, its increased density in relation to water, ground pumps are more massive, their impeller has fewer blades, which allows large inclusions to pass through the flow part. For the same purpose, the wheel width has been increased, the front and rear discs are located in parallel planes, and the cross-sectional area of the ground pump housing has been increased. The rotation speed of the impeller of the soil pump is usually less than that of the water pump. Reducing the speed of rotation and increasing the flow section reduces the speed of movement of the hydraulic mixture in the flow part of the ground pump, which reduces abrasive wear.

A smaller number of revolutions, a reduction in the number of blades, the worst configuration of the flow part of the ground pump from a hydraulic point of view also cause a lower efficiency compared to water pumps, a lower head. To protect the front and back covers from abrasive wear, replaceable elements - armored discs are installed. In order to prevent the ingress of soil particles from the pulp into the space between the armored disk and the impeller, clean water is supplied to this cavity. In modern pumps designed for hydrotransportation of soil of high abrasiveness, it is provided for booking the body with replaceable inner jackets made of wear-resistant materials. The impeller is usually made of cast steel or special wear-resistant alloys. During operation, its elements, primarily the blades, wear out and are fused with electrodes.

Due to the uneven wear of the impeller, its decentering (unbalance) may occur, which increases the load on the shaft, therefore it is made of a larger diameter than that of pumps designed for water supply.

The main elements of the ground pumps are a snail-shaped housing, an impeller, a shaft, bearings supporting and thrust, front and rear covers, a bed, a suction pipe, a pressure pipe, armored discs, a shaft coupling for connection to an electric motor and oil seals.

Soil pumps are usually designed with a horizontal shaft arrangement, since at the same time the technological schemes of dredging plants are simpler and more convenient to operate.

The pressure pipe in modern ground pumps is directed vertically upwards or has a lower horizontal direction. Vertical direction is convenient for arrangement on dredgers, horizontal - on stationary and mobile installations.

The snail is made in the form of a concentric channel or slightly narrowed to the pressure pipe. The body of the ground pump is cast. In heavy-type ground pumps designed for hydraulic transportation of abrasive rocks, a double casing is manufactured - internal (replaceable jackets) and external. The connector of the ground pump housing can be carried out in both horizontal and vertical planes. The most common type of connector is in the horizontal diametric plane. This type of connector is adopted in GR type ground pumps.

The impeller of the ground pump can be open, semi-open and closed. Open and semi-open wheels are practically not used due to low efficiency. Impellers with four and three blades are used in pumps of the GR brand.

The mounting of the wheel on the shaft can be cylindrical or conical. Cylindrical landing is rarely used, although the impeller is more accurately fixed on the shaft. With a cylindrical landing, very precise processing of the landing surfaces is required and it is difficult to remove the wheel hub from the shaft.

The most commonly used conical landing of the impeller. The taper of the hole in the hub and shaft ends is 1:10 -, 1:20. With a cone landing, it is easier to remove the wheel and fit it. The disadvantages of a conical fit should include less accurate fixation of the impeller than with a cylindrical one, relative to the housing in the longitudinal direction, which can lead to a "beating" of the impeller.

The shaft of the ground pump is made of carbon and alloy steels. It is mounted in two support bearings. For the perception of axial forces, a thrust bearing is used, which in most cases is installed in the same housing with a support bearing far from the wheel. In the largest ground pumps, the shaft is rigidly connected to the motor shaft, therefore it lies in one bearing, and the motor shaft lies in its two bearings. Rolling and sliding bearings are used in ground pumps. Babbit inserts are used in sliding bearings, rolling bearings use radial, radial-thrust and thrust ball and roller types.

The back cover of the housing is manufactured as a separate part of the ground pump. In the center of the back cover there is a hole for inserting the shaft into the pump housing. There is an oil seal between the cover and the shaft. The front cover is located on the side of the suction pipe. In a number of ground pumps, the front and rear covers are protected from wear by special replaceable bottoms (armor plates). The bed of the ground pump is designed for mounting support and thrust bearings on it. It is usually a cast steel structure. The connection of the shaft of the ground pump with the shaft of the electric motor is carried out by means of an elastic coupling with fingers. Great importance in ground pumps is attached to the system of protection of its main parts from abrasive wear. For this purpose, in addition to the device of protective jackets, liners and armor plates, a pressure water supply system is used in the gaps between the front and rear covers and the impeller.

To the stuffing box seal of the hub or shaft of the impeller, the extraction water must be supplied with a pressure 50-100 kPa greater than the pressure developed by the ground pump.

There are radial protrusions on the outer surface of the impeller discs, which, when rotating, create back pressure in the gap between the impeller and the bearings.

Good results in reducing abrasive wear are achieved with the use of alloyed and manganese steels, high-chromium cast iron with nickel and alloyed cast iron. The use of alloy steels with subsequent heat treatment makes it possible to increase the service life of the impellers before surfacing by 5 times or more than with conventional medium-carbon steels.

Rubber lining of wear parts of ground pumps has been widely used abroad. However, for this purpose, only particularly elastic rubbers can be used, which do not lose their properties when cut by sharp edges of clastic materials.

LABORATORY WORK No. 4

STUDY OF AXIAL FAN DESIGNS

The purpose of the work is to study the designs of axial fans

GENERAL PROVISIONS

Laboratory work is provided after listening to the theoretical course.

The student must be prepared to perform laboratory work. He must know the purpose and classification of fans for open and underground mining.

According to the intended purpose, fan installations are divided into main, auxiliary and local ventilation. By location - on the surface and underground. According to the method of ventilation - installations with fans working for suction and injection.

Fans are divided into: according to the developed pressure - low, medium and high pressure; according to the conditions of suction - single and double-sided suction; in the direction of flow - centrifugal, axial.

The student should know the basic parameters of fans: pressure, supply, power and efficiency

AXIAL FANS

The main elements of a single-stage axial fan are: an impeller, a housing, a collector, a front fairing as a spherical straightening device and a diffuser. The diffuser consists of two shells placed one into the other: the outer cone - the actual diffuser and the inner (on most cylindrical fans) - the rear fairing.

To increase the pressure, the axial fan is usually made two-stage with two sequentially connected impellers, an intermediate guide device between them and a straightening device behind the last impeller. Sometimes an input guide device is installed in front of the first impeller.

The impellers, together with the shaft on which they are fixed, form

a fan rotor, which is driven directly from

the electric motor. The shaft supports are ball or roller

bearings.

The collector and fairing are designed to ensure the correct air supply to the wheel blades so that the air flow is directed along the fan axis with as uniform a velocity field as possible. The action of the collector is most effective when there is a cylindrical section of the housing with a length of at least 0.5 wheel diameter between it and the wheel. In the absence of a collector, the fan pressure decreases by 10... 20%, and the efficiency is 10 ... 15%. The fairing is stationary and in its absence, the pressure decreases by about 20%.

The guiding and straightening devices, which are fixed wheels with radial blades, are necessary for unwinding the flow and, consequently, increasing the efficiency of the fan. The rotary blades of the intermediate guide and straightening devices provide the possibility of regulating the operating mode of the fan, reversing the ventilation jet. For regulation, an input guide device with rotary blades is sometimes used.

One of the main components of the fan is the diffuser, thanks to which a significant part of the dynamic pressure (at least 70%) must be converted into static pressure.

The blades are mounted on the impeller hub at regular intervals at an angle to the plane of its rotation. The most rational blade is wider at the hub than at the periphery. The best design is a twisted blade with a shape similar to that of an aircraft propeller blade.

The blades are made hollow (Fig. 4-a) with a rod for fixing it on the bushing and cast (Fig. 4.1.) from aluminum or magnesium alloys. The hollow blade consists of: a rod 1 riveted to it with a twisted steel skin 2 2-3 mm thick, welded to the skin of the rib 3, acting as reinforcement against abrasion by coal dust, welded to the edge of the upper and lower bottoms 4.

The production of blades is possible from plastics. Such blades are manufactured with a higher degree of accuracy, eliminating the danger of sparking when the blade may touch the fan housing, racks in a chemically aggressive environment.

From the condition of reliability of operation and noise reduction of the running fan, the maximum circumferential speed at the ends of the blades should be no more than 95m/sec.

Up to 14 blades are mounted on one bushing, their attachment points must provide the possibility of installation at different angles relative to the plane of the wheel, which is necessary to regulate the performance and pressure of the fan.

In a running fan, under the influence of a pressure difference, part of the air flows through the gap between the end of the blade and the housing from the side of the air outlet from the impeller towards the entrance to it, while the pressure decreases and the efficiency of the fan decreases. However, excessive reduction of the gap may cause the blade to touch the fan housing. The size of the gap depends on the type of fan and usually should not exceed 15% of the blade length. When the fan is running, due to the difference in flow pressure before and after the impeller, an axial force acting on the rotor and directed towards the flow inlet into the fan arises. The axial force is perceived by the thrust bearing.

Two-stage reversible fans VOD 221, VOD 40, VOD 50 (V - fan, O-axial, D- two-stage, figures-diameter of the impeller in cm) are designed for the main ventilation of mines at the required static fan pressure of no more than 300 Pa and air flow from 50 to 600 mZ / sec. These fans are designed according to the aerodynamic scheme K-84 TsAGI named after N.E. Zhukovsky (K- twisted blades , 84 - speed coefficient).

The main fan unit with WATER type fans (Fig. 4.2) consists of: backup and working fans 1 and 2 with synchronous electric motors 3 and 4; lubrication systems 5; electrical and automation equipment 6 and devices 7 and 8 (bloopers or doors controlled by winches with motor reducers) to switch any of the fans to work and turn off the other; supply 9 and output 10 channels; noise muffler.

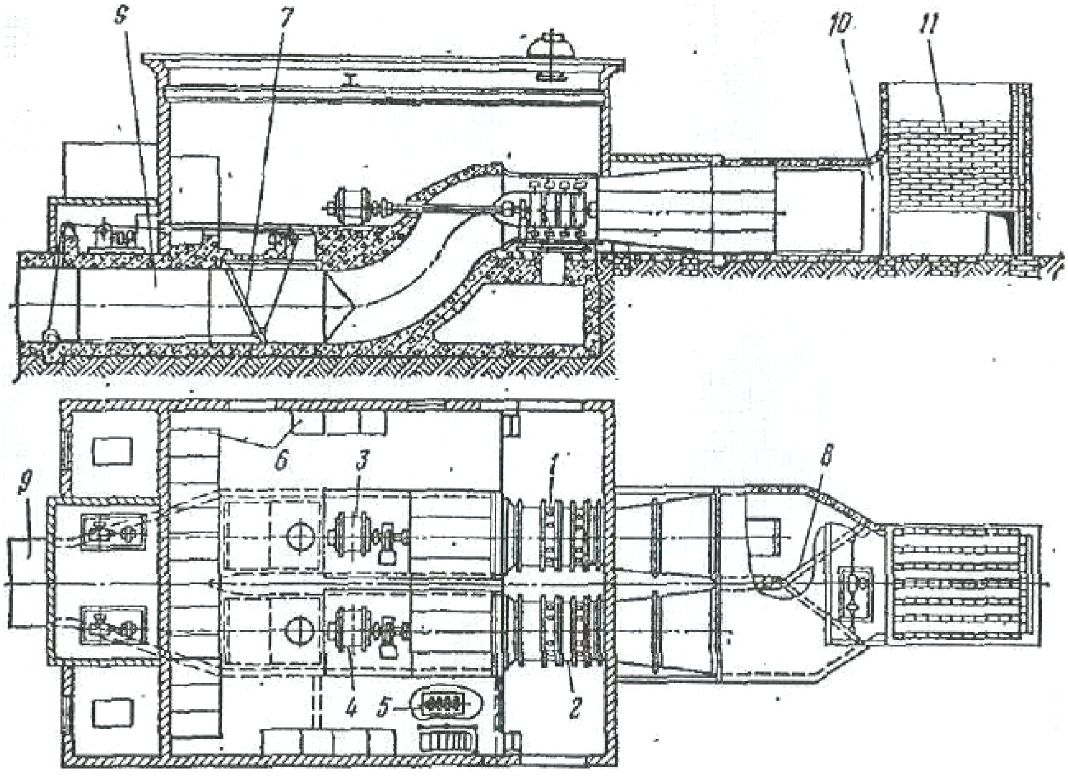


Fig. 4.2 Fan installation with WATER type fans

The noise muffler is made of sound-absorbing cinder blocks and has 5...7 parallel walls, common for a working and backup fan, which eliminates the penetration of outside cold air to the backup fan in winter, and consequently, its freezing.

The VOD-21, VOD-30, VOD-40 AND VOD-50 fans (Fig. 4.3.) have a housing 1, frame 2, rotor 3, front 4 and rear 5 support blocks, guide 6 and straightening 7 apparatuses, collector 8, as 9, diffuser 10, transmission shaft 11 with coupling 12 for connection to a synchronous electric motor 13.

The rotor of the WATER fan is shown in Figure 4.3. Impellers 3, 4 and half coupling 5 are fixed on the shaft 1 with bearings 2, which perceive both radial and axial loads. The wheel is fixed on the shaft by a key 6 and from axial displacement by a nut 7. Bushings 8 impellers (the diameter of the sleeve is 0.6 of the diameter of the impeller) they are welded to avoid the appearance of imbalance, sealed from moisture and dust ingress into them. There are 12 blades on the sleeve 9. The blades are hollow welded-riveted, consist of two sheets of sheathing, reinforcing ribs, bottoms and shank. Thanks to the fastening gate 10, the blades, when the fan stops, can be turned over manually in the range of 15....45 °. In order to regulate the operating mode of the fan, the number of blades of the second stage wheel can be reduced by 2 times.

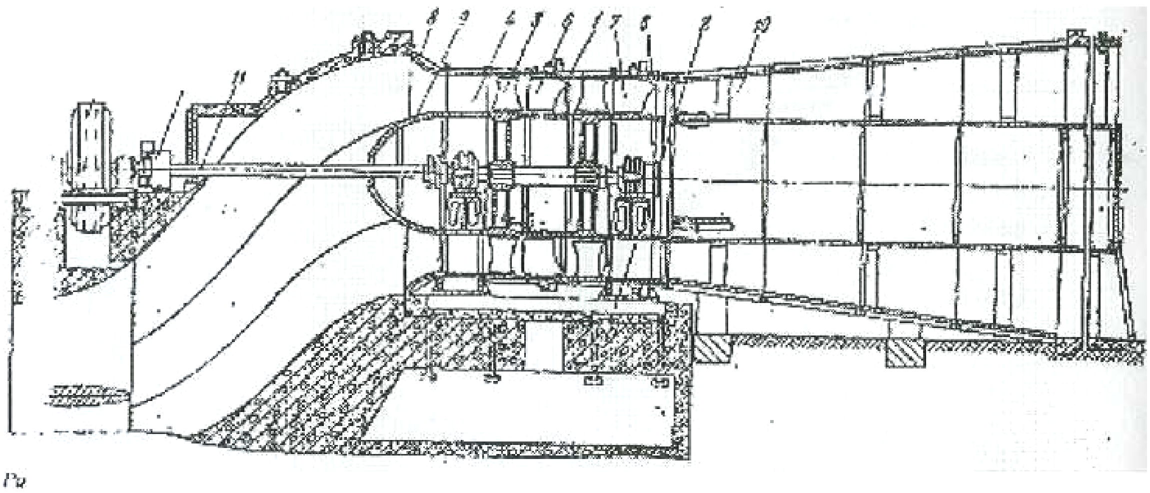


Figure 4.3 Water type fan

The transmission shaft is made suspended with a gear (in the VOD-21 fan - with finger couplings).

The intermediate guide device has 14 rotary blades that can rotate at an angle of up to 100°. the straightening device has 14 blades, of which 11 are rotary, and 3 are non-rotating.

The fans are equipped with a pad brake with an electromagnetic drive, which stops the rotor in 2.. 2.5 minutes.

To reverse the air jet, it is necessary to turn off the fan motor and brake the rotor, turn the blades of the guide and straightening apparatus 180 °, brake the rotor and start the engine in the opposite direction.

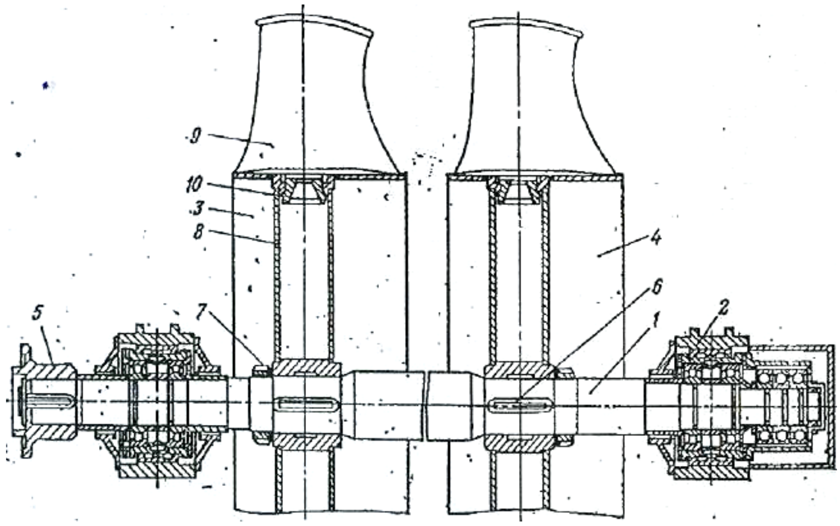


Fig. 4.4 Water-type fan rotor

LABORATORY WORK No. 5

STUDY OF CENTRIFUGAL FAN DESIGNS

The purpose of the work is to study the designs of centrifugal fans

1. Centrifugal fans. Classification. Shaft centrifugal fans are classified mainly according to the method of air supply to the impeller: with one-way (Fig. 5.1) and two-way suction.

Fans with two-way suction have higher performance compared to fans with one-way suction.

Aerodynamic schemes. Fans are developed on the basis of typical aerodynamic schemes (models of fans), worked out in laboratory conditions. The aerodynamic scheme of the fan is usually called the flow diagram of the fan with a set of basic dimensionless design parameters that determine the appropriate aerodynamic characteristics.

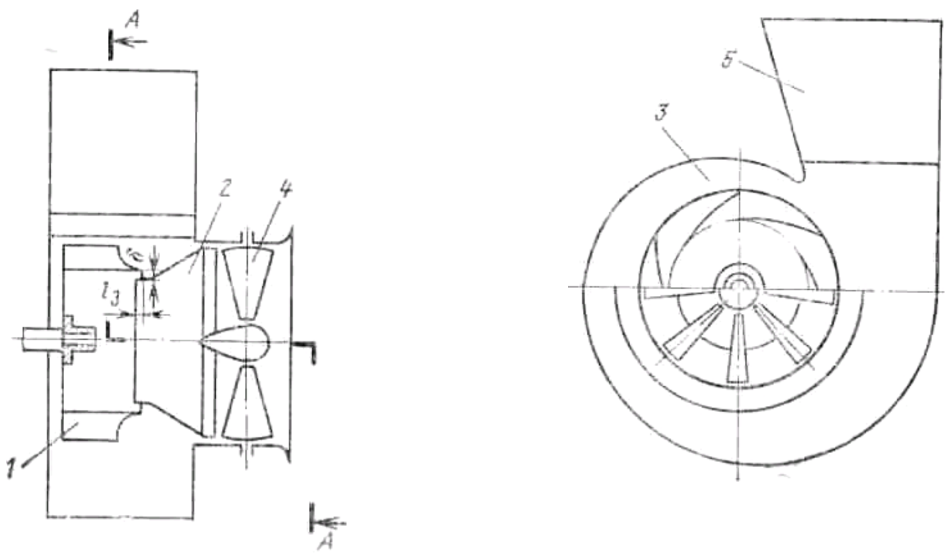


Figure 5.1. Diagram of the centrifugal fan of one-way suction: 1 — impeller; 2 — inlet pipe; 3 — spiral housing; 4 — guide device; 5 — diffuser

The fan is considered to be right or left rotation depending on the rotation of the impeller clockwise or counterclockwise when looking at the fan from the drive side.

Elements of fans. The process in the impeller is significantly influenced by the input elements: their profile, the diameter of the inlet, the size and configuration of the gap in the seal between the impeller 1 and the nozzle 2 (see Fig. 5.1), etc. The wrong choice of these elements causes distortion of the velocity field at the entrance to the wheel, separation of the flow from the surfaces of the flowing part of the wheel and a decrease in the fan efficiency as a result.

In shaft fans working for injection, cylindrical nozzles with a free inlet are used, in fans working for suction, smooth, conical and composite nozzles with a significant degree of confusability are used to reduce turbulence and equalize the air flow velocity. Loss of flow energy at the fan inlet:

$$\Delta p_{вх} = \xi \frac{c_1^2}{2}, \qquad (5.1)$$

where c1 is the speed in front of the impeller, m/ s; is the loss coefficient.

There is a gap b between the inlet pipe 2 and the front disc of the wheel 1 (see Figure 5.1), through which additional air flows into the impeller. With a rational choice of the seal shape, the axial overlap of the radial width of the gap, the main flow is pressed against the inner side of the disk and flows around it without separation; the optimal values of lz = 0.007 and = 0.002. Air leaks through the gap in the optimal operating mode of modern fans are 2-4% of the nominal capacity.

Fan impellers are made of single- and double-suction with profiled wing-shaped blades bent backward, ra < 90 ° (Fig. 5.2). The impellers of old fans were equipped with structurally simpler flat blades 1 and 5-shaped blades 2, providing greater pressure compared to flat ones; efficiency of fans with both types the blades are lower than the efficiency of fans with wing-shaped blades.

To regulate the operating modes, the fans are equipped with axial guide devices 4 (see Fig. 5.1). These devices are equipped with 10 or 12 flat blades made in the form of a sector and rotating at fans of the VC and VCD types at an angle from -20 to +110 °. At angles from 0 to 90 °, the air in the guide device twists in the direction of rotation of the wheel, reducing the fan performance and power consumed by it. When the angle of installation of the blades is 90 °, the blades of the latter completely block the suction opening of the fan; such an installation of the blades is used when starting the fan. The angle of 0° corresponds to the installation of the blades in the plane of the axis of the fan shaft, and the flow is excluded from twisting. When the blades of the guide device are rotated to negative angles, the air in it is twisted against the direction of rotation of the impeller, which leads to an increase in the fan performance and power consumption. At large negative angles of installation of blades (-20 ° or more) the fan operation mode is characterized by flow separation from the blades at the inlet to the impeller and increased rotor vibrations.

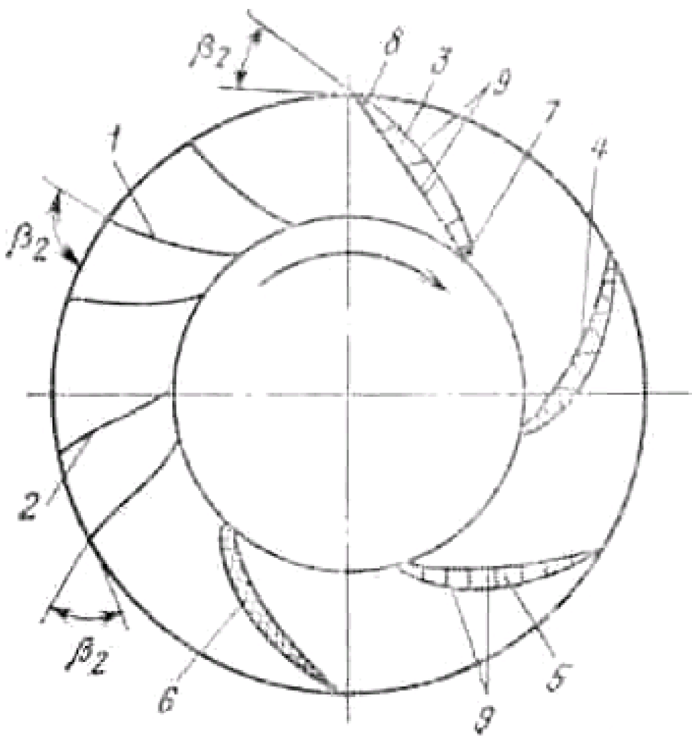


Fig. 5.2. Profiles of the working blades of centrifugal fans:
1 — leaf; 2 — 3-shaped: 3 — wing-shaped with C- and U-shaped spars; 4 and 5 — with truss and honeycomb fins; 6 — three-layer; 7 and 8 — spout and tail blades; 9 — upper and lower shells

The output devices of centrifugal fans are, as a rule, spiral housings that form the flow and guide it from the output section of the wheel to the fan outlet, as well as partially converting the dynamic flow pressure into static.

Nomenclature and purpose. At mines and mines, centrifugal fans of the main ventilation of the following types are currently being operated: VC, VCD, VSC, VRCD, VCP, VCC and a local ventilation fan VC-7 (V—fan, C—centrifugal, O—one-way suction, D - two-way suction, R- mine, Sh-pit, P—tunneling, 3-with adjustable flaps). According to GOST 11004-75, centrifugal fans are marked: single-sided suction VC and double-suction VC. The number in the marking means the diameter of the impeller in decimeters.

Currently, the plants produce the following centrifugal fans designed for the main ventilation of mines and mines: VCD-16, VC-25, VC-31.5 (old code VC-32), VCZ-32, VCD-31.5 (old code VCD-32M) and VCD-47 "North" and for ventilation trunks and near-barrel workings: VC-11 and VSHC-16 and only for ventilation of the faces of mine shafts VCPD-8UM and VCP-16, as well as fans of local ventilation VC-7.

It is planned to produce VC-31.5P fans (with rotary blade flaps) instead of VCZ-32.

Centrifugal fans are applicable for suction, discharge and combined ventilation schemes of mining enterprises. Large centrifugal fans are manufactured by Donetsk

Machine—Building Plant named after LKU, fans with a diameter of the impeller up to 2.5 m inclusive are manufactured by Artemovsky Machine-Building Plant. All currently manufactured centrifugal fans are developed on the basis of aerodynamic schemes of TMK named after M. M. Fedorov, the Institute of Dongiprouglemash and SLE of Artemovsky and Kamensky machine plants.

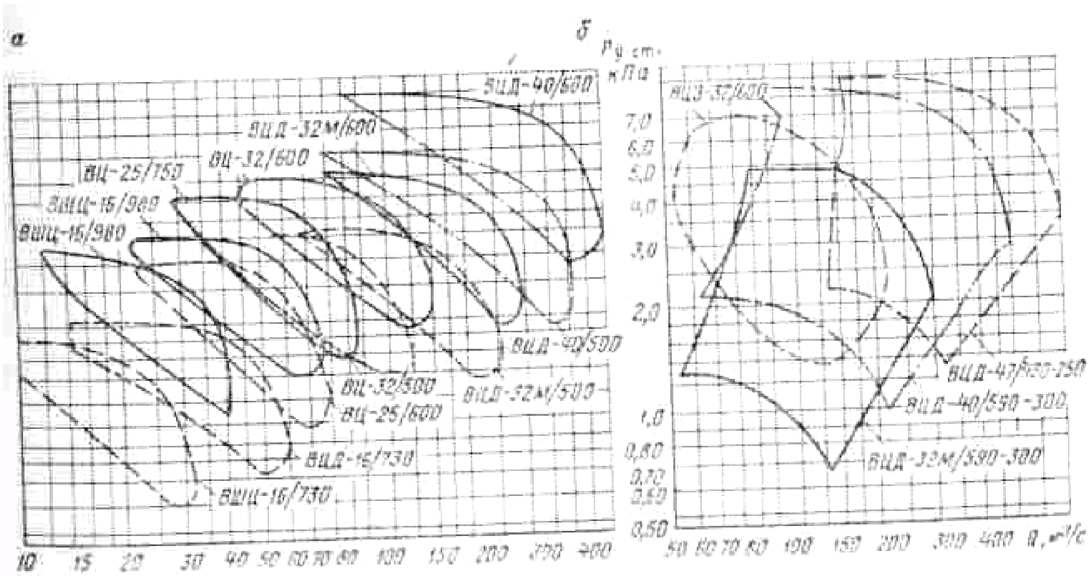


Figure 5.3. Summary graph of the areas of industrial use of centrifugal fans of the main ventilation, regulated by axial guide devices (a) and with the help of rotary flaps and changes in the speed of rotation of the impeller (b)

Figure 5.3 shows a summary graph of the areas of industrial use of main ventilation fan units with centrifugal fans produced by factories. Fans are made of one-way and two-way suction. The impellers of the one-way suction fans have 8 blades, and the two-way suction has 16 (8 on each side), except for the VCD-47 fan, which has 12 blades.

The design of the fans. One-way suction fans have two layout schemes: with a cantilever arrangement of impellers on the rotor shaft (VTS-11, VSHTS-16, VTSP-16, VTS-25) and with the arrangement of impellers on the shaft between the bearing supports. The latter scheme is implemented in larger VC-31.5 fans and

VCZ-32 This arrangement reduces the load on the bearings and eliminates resonant transverse vibrations of the main shaft.

The centrifugal fan of one-way suction (VC-11, VC-16 (Fig. 5.4), VC-16, VC-25, VC-31.5 and VC-31.5P) consists of a rotor / with an impeller 2, an axial guide device 3, an inlet manifold 4 and a branch pipe 7, a spiral casing 5 and frames 6. The first three fans (with a diameter of the impeller up to 1600 mm) have a common welded frame on which the entire mechanical part of the fan and its drive motor are mounted. Larger fans do not have a single rigid frame, and the rotor shaft rotates in two separate bearing supports mounted on separate frames mounted on a concrete base.

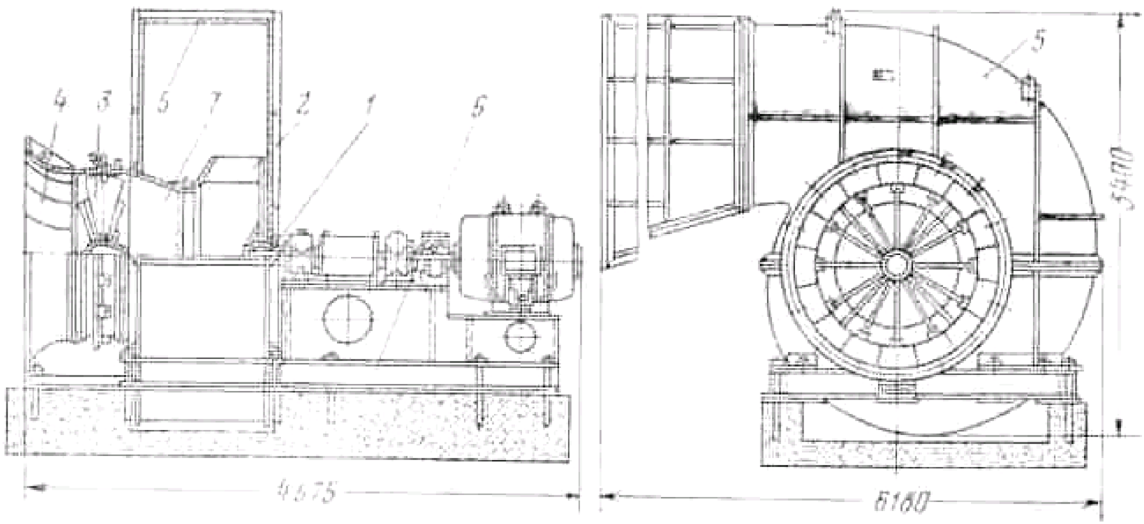


Fig. 5.4. One-way suction fan VSHTS-16

The impeller consists of wing-shaped backward-curved blades welded to the front conical cover and rear flat root discs. The rear wheel disc is welded or bolted to the hub. The cover disk on the diameter of the entrance can be reinforced with a toroidal cast labyrinth ring. The root disk of the impeller is reinforced by connecting it to the fairing, which simultaneously improves the conditions for turning the airflow in the impeller.

The blades are hollow, welded, with stiffening ribs inside.

The fans VC-31.5P and VCD-31.5P have blades equipped with rotary flaps. The internal space of these blades and the blades of some other fans, for example, VCD-47 "North", is filled with foam foam, foamed directly in the shells. This made it possible to abandon the connection of the shells with the spars and reduce the weight of the blades.

The impeller is fastened to the drive shaft by a keyway connection. The shafts rotate in radial double-row spherical and radial-thrust twin bearings cooled in large fans due to forced oil circulation. The rotor shaft is connected to the motor shaft by an elastic finger or, more often, a toothed coupling.

The stator part of the fan (see Fig. 5.4) consists of a spiral casing 5, an inlet manifold 4 and an inlet pipe 7, which with its narrow end enters the labyrinth seal of the impeller, forming an annular labyrinth gap with the latter. To increase rigidity, the casing and the box are equipped with fins closed in frames. The lower part of the spiral outlet of the VC-25 and larger fans is formed by a channel in concrete. The housings are made detachable, excluding the VC-11 fan housing.

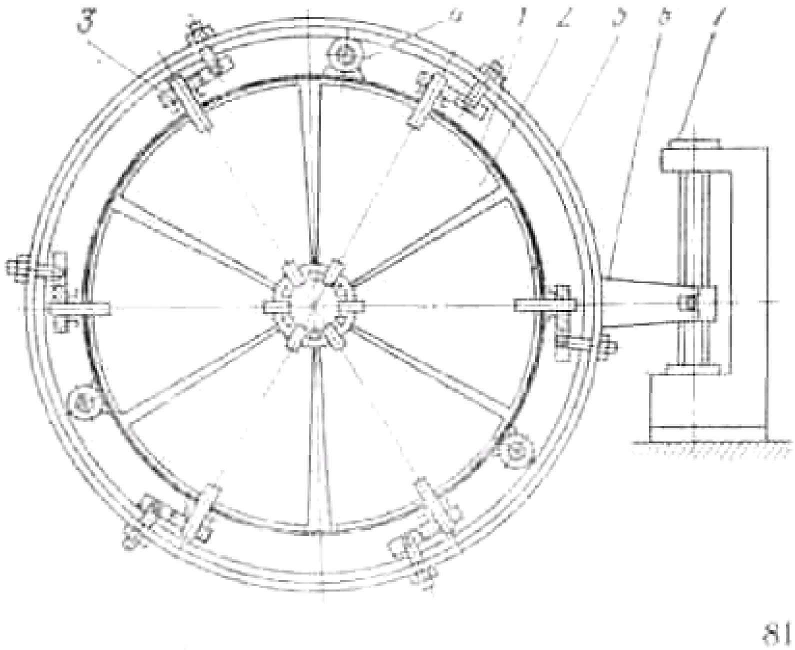


Figure 5.5. Guide device with external drive ring:
1-the body of the device; 2 - the blades; 3 - the lever of the blade; 4 - the roller; 5 — the drive ring; 6 — lever; 7 — drive column

The axial guiding device (Fig. 5.5) has flat leaf blades with a mechanism for their simultaneous rotation and a fairing located along the axis and suspended in the housing on extensions or mounted on the rotor shaft. The blades consist of a web and two trunnions welded to it and are supported by bronze bushings in the fairing and nylon or bronze bushings in the housing. On protruding from the housing of the guide device.

Centrifugal fans of two-way suction (VTSPD-8UM, VTSD-16, VTSD-32M, VTSD-40 and VTSD-47 "North") have a layout scheme with the placement of impellers on the drive shafts between their bearing supports. The design of these fans is in many ways similar to the design of the considered one-way suction fans. The main design features of double-suction fans are the presence, excluding the fans of the VCD-47 "North", two axial guide devices, a double-suction impeller for large fans (VCD-32M, VCD-40 and VCD-47) and the output end of the shaft for connection by means of a gear coupling of the second drive motor. Figure 6.6 shows the most powerful centrifugal fan VCD-47 "North" as an example. Its impeller consists of two half-wheels. This design of the wheel facilitates its transportation to the installation site of the fan. The half-wheels are connected to each other and to the hub rim with bolts.

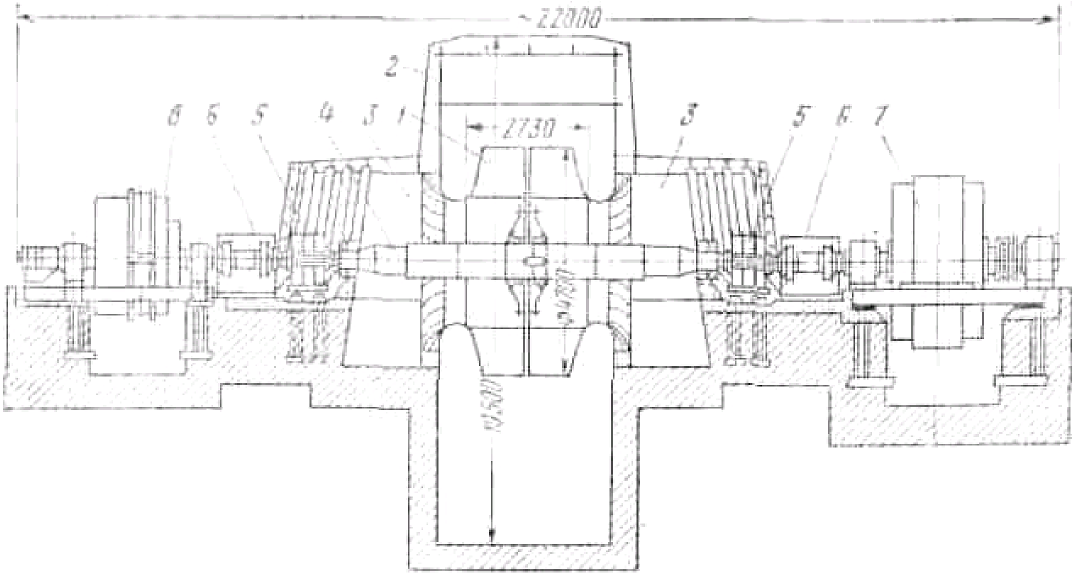


Fig. 5.6. VCD-47 "North" fan:
1 - impeller; 2 — casing; 3 — suction boxes; 4 — shaft; 5 — bearing supports; 6 — gear couplings; 7 and 8 — electric motors

The VCD-47 "Sever" fan provides high efficiency of operation in a wide range of ventilation modes that differ from the optimal pressure by 4 times, which is achieved by using a fan electric drive speed control system made according to the scheme of a combined asynchronous valve-machine cascade.

The two-way suction fans VCD-31.5 M, VCD-47 "North" have aerodynamic qualities that ensure the supply of large amounts of air at high pressures.

Centrifugal fans have become widespread in mining enterprises, which is explained by a significant improvement in their technical and economic indicators over the past 15-20 years. Thus, the maximum static efficiency of the main ventilation unit increased from 0.72 to 0.86, the weighted average static efficiency in the field of operation from 0.52 to 0.75; the dimensions and cost of fans decreased by 1.5—2.0 times.

LABORATORY WORK No. 6

STUDY OF AUXILIARY EQUIPMENT DESIGNS

The purpose of the work is to study the designs of auxiliary equipment

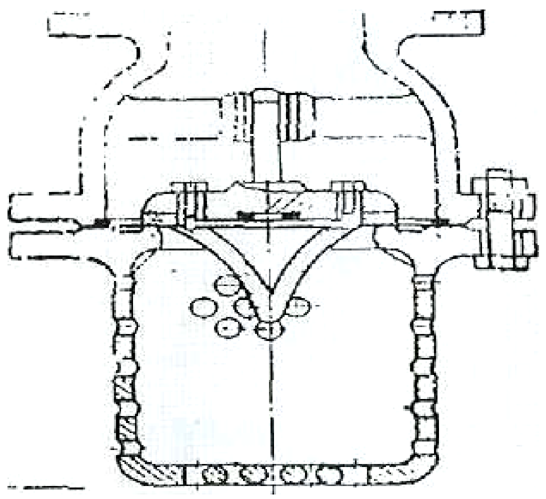
AUXILIARY EQUIPMENT

Centrifugal pumps and pipelines are equipped with check valves, automated valves, various kinds of taps and other pipe fittings for safe operation.

Intake valve with mesh (fig.6.1) is used in drainage installations with a positive suction height to retain water in the suction pipeline of the pump and to protect the pump from ingress of foreign bodies.

For high-performance pumps, intake valves with two guides are used, which have an increased area of the live section of the cylindrical lattice and a large area of the valve itself. This ensures a tight fit in the saddle when the pump stops and reduces water leakage from the suction pipeline.

In Fig.6.2 a receiving device with a ball check valve developed at the M.M. Fedorov IGM and TC is presented. It consists of a welded steel body 1, a seat 3 directed at the landing point of the ball by stainless steel, and a hollow steel ball 2 covered with a thick layer of hard rubber.



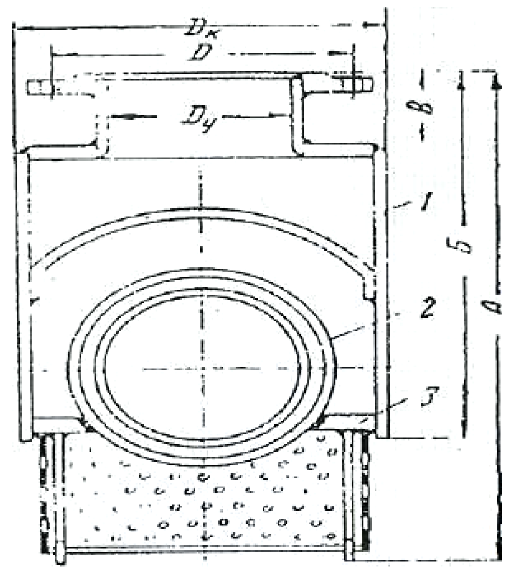
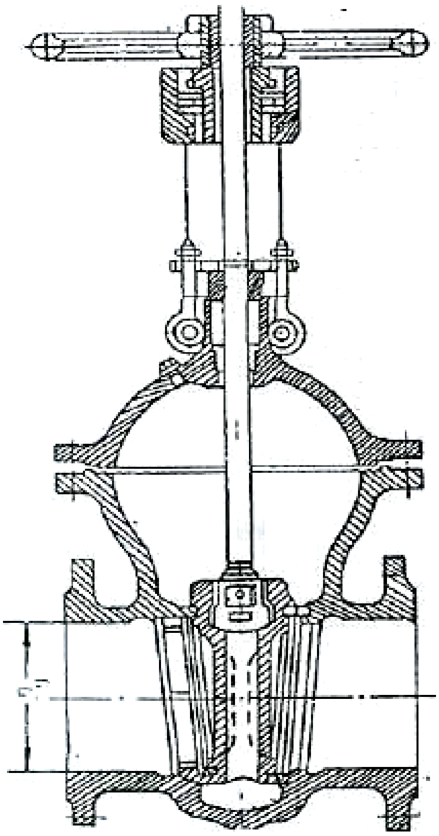


Fig. 6.1 Receiving valve Fig. 6.2 Flanged receiving device with mesh, type with ball check valve 16h 40r (P=2.5 kgf/cm2)

The valves are used as a shut-off device for discharge and suction pipelines with a nominal diameter of 100-600 mm (Fig.6.3). In automated drainage installations of high performance, electric valves are used to start, stop the pumping unit and switch the large-diameter pipeline in complex hydraulic schemes of pipe collectors. Manual control in these conditions presents significant difficulties. Currently, hydraulic-driven valves are beginning to be used, providing high reliability and safety in operation.

The rotary check valve is used to hold the column in the stakes (Fig.6.4). It has a barrel-shaped body and a lid on bolts or studs. The valve disc is suspended on an axis lying in the tides of the body, from which the position remains unchanged when the bolts are tightened. By special order, check valves with a diameter of 200-500 mm are manufactured with a bypass valve that allows pre-filling the pump or draining water from the pump. The improvement of the design of check valves is carried out in the direction of the development of contact and contactless devices for monitoring their operation, as well as check valves with slow landing.



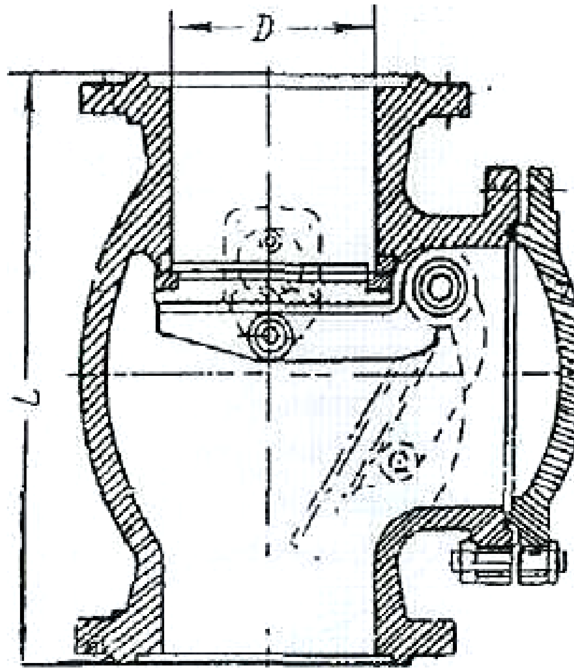


Fig. 6.3 Valve Fig. 6.4 Rotary check valve flanged steel type 19c17br

Compensators are used to eliminate dangerous stresses in pipelines from temperature expansions, facilitate installation and more evenly distribute the load from the weight of the pipeline to the support beams in the shaft (Fig.6.5). It is known that the temperature in the shaft of the mine varies widely throughout the year and the temperature elongations of pipelines reach 10-12 mm per 100 m length with a difference of 10 S and an expansion coefficient of steel pipes 11.5x10 1 / deg. Compensators choose a variable length of the pipeline and thereby protect it from an accident. Stuffing box compensators are used with a normal or large stroke of the stuffing box, and the reliability of their operation depends on the design and quality of the packing. The best results compared to leather cuffs were shown by stuffing box compensators with tarred hemp stuffing. When pumping acid mine waters, a stuffing in the form of cuffs made of acid-resistant rubber is used.

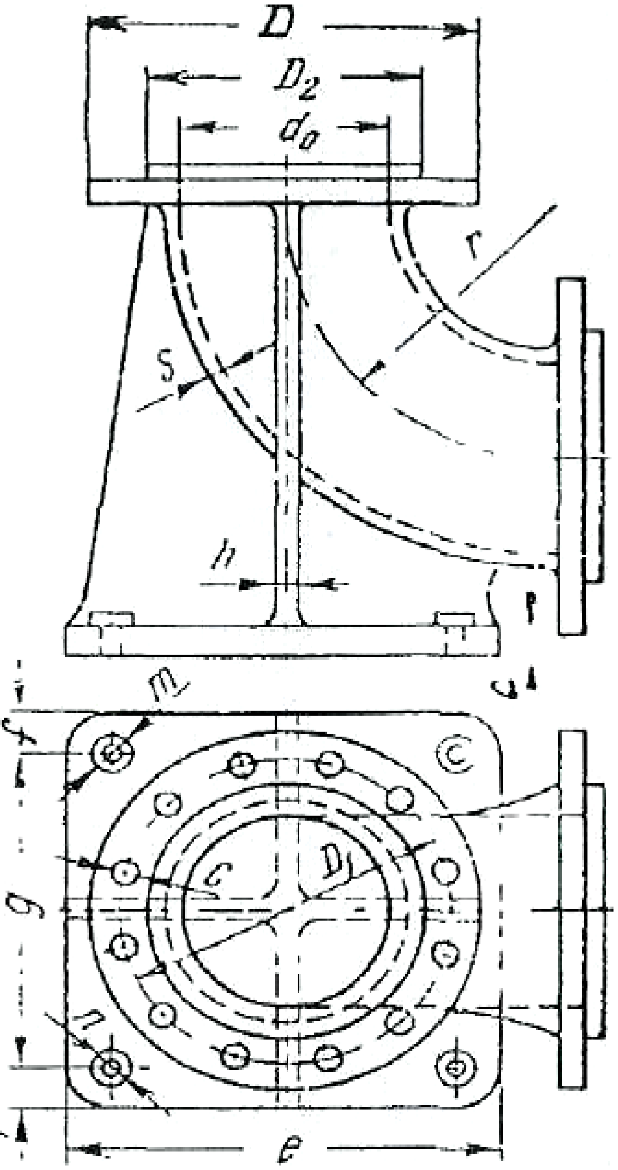


Figure 6.5 Support knee and its dimensions. (P = kgf/cm2)

The support knee is used in drainage installations to perceive the weight of the pipeline and the column of water in it (Fig.6.6). The support elbow is installed at the entrance of the discharge pipeline from the pipe passage into the shaft of the mine.

Pipeline fittings (valves, check valves) for the equipment of drainage installations with high-pressure pumps are currently produced by machine-building plants at a pressure of 10-380 kgf/cm2 and with a nominal passage diameter of 100-350 mm. Shaped parts of pipelines (elbows, tees) are made of steel casting and are widely used in the equipment of drainage installations.

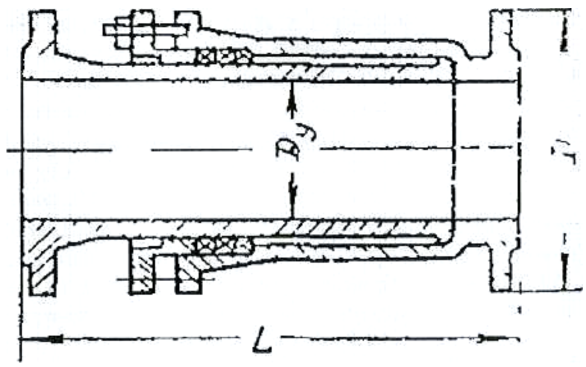


Figure 6.6 Temperature compensator and its dimensions.

LABORATORY WORK No. 7
STUDY OF PISTON COMPRESSOR DESIGNS.

The purpose of the work is to study the designs of reciprocating compressors general information

The classification of compressors is made according to their characteristic features.

-according to the method of exposure to air (gas) - volumetric or displacement compressors (reciprocating, rotary, screw, plate, etc.), in which the air (gas) pressure increases due to a decrease in the working volume, - vane (turbochargers), where an increase in air (gas) pressure occurs due to the force interaction of air (gas) with blades of rotating impellers of the machine;

- by the type of compressible gas - air, ammonia, freon, etc.
- by the value of the final pressure - low-pressure compressors (0.3-1.0 MPa or 10 atm.), medium-pressure compressors (1.0-10 MPa or 10-YUOATM), high-pressure compressors (over 10 MPa or 100 atm.)

The variety of design schemes and designs allows piston compressors to be classified according to the following criteria. By the method of action:

- simple-acting compressors (Fig. 7.2, 7.6)
- double-acting compressors (Fig.. 7.3,7.5. 7.7, 7.8, 7.9)
- differential compressors (fig. 7.4.) By the number of steps:
- single-stage (Fig. 7.2, 7.3, 7.4) -two-stage (Fig. 7.5, 7.7, 7.8)
- three-stage and more. By the number of cylinders:
- single-cylinder (fig.7.2, 7.3.) -two-cylinder (Fig.. 7.5, 7.6, 7.7, 7.8, 7.9)
- multi-cylinder.

By the arrangement of the cylinders:

- horizontal,
- vertical,
- angular (rectangular fig. 7.7, Y-shaped fig.7.6.)

Reciprocating compressors are widely used in the mining industry. They are equipped with stationary compressor stations when there is a need for compressed air, in an amount of up to 500 mZ/min.

Currently, angular compressors of type P and horizontal type M are produced.

Angular compressors — 302VP-10/8, 202VP — 20/8, 305VP — 30/8, etc. have the following symbols: VP — rectangular air, after the letters of the digits in the numerator of the fraction — productivity (m3 / min.), in the denominator — excess discharge pressure (kgf / cm).

The numbers before the letters show the nominal load on the rod (tc), the numbers before zero are the number of the compressor modification.

Horizontal compressors of types- 2M10-50/8 and 4M10-100/8 have designations: M-multi-row, 10-piston, single row force (tc), fraction numerator-compressor capacity (m/min), fraction denominator-overpressure (kgf/ cm2), digits before the letter M- the number of cylinders.

Previously released angular compressors 2VG-100/8, 5G-100/8, 55V, etc. are also in operation.

The device and the principle of operation of reciprocating compressors. Reciprocating compressors belong to volumetric machines. The scheme of the reciprocating compressor is shown in Fig. 7.1. A piston 2 is placed in the cylinder I, which is pivotally connected with the upper head of the connecting rod 4 by a finger 3. The connecting rod is pivotally connected with the neck of the crankshaft 5 by the lower head.

When the crankshaft rotates by means of a crank mechanism, reciprocating motion is communicated to the piston. The positions in which the piston changes movements are called extreme positions (dead spots). The distance between the extreme positions of the piston is called the stroke of the piston. The stroke of the piston is made in half a revolution of the crankshaft. The suction 6 and discharge 7 valves are located in the cylinder cover. When the piston moves down from the extreme position (from the upper dead center), a discharge is created in the cylinder. Under the influence of atmospheric air, the suction valve opens and the cylinder is filled with air.

When the piston reaches its lowest position, the entry of atmospheric air into the cylinder ends. When the piston moves in the opposite direction, the volume of air will decrease and the pressure will increase. Under the influence of excessive pressure (pressure greater than atmospheric) air, the suction valve automatically closes. Air compression will continue until the pressure in the cylinder reaches the air pressure in the network. In this case, the discharge valve opens and the compressed air will be pushed out of the cylinder by the piston into the network. The ejection of compressed air ends when the piston reaches the highest position.

For one complete rotation of the crank, one complete movement of the piston or a full compressor operating cycle is performed, consisting of cycles: suction, compression and heating of air.

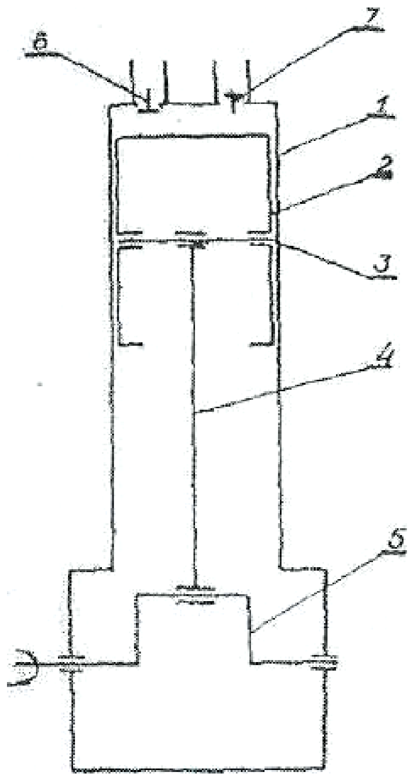


Figure 7.1 Diagram of a single-stage simple-acting reciprocating compressor:

1- cylinder; 2- piston, 3- pin, 4-connecting rod, 5- crankshaft, 6-suction valve, 7- discharge valve.

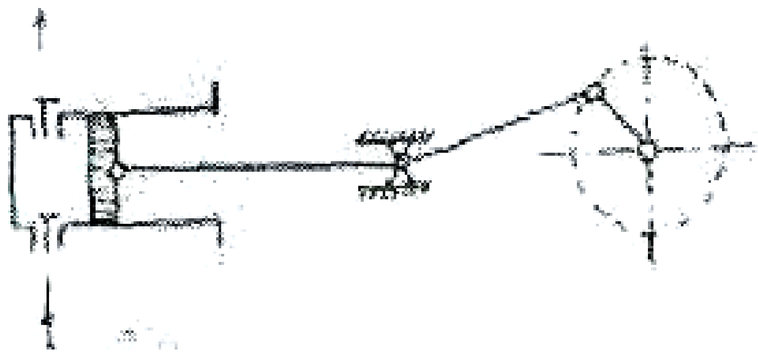


Fig. 7.2 single-cylinder, horizontal, single-stage piston compressor of simple operation.

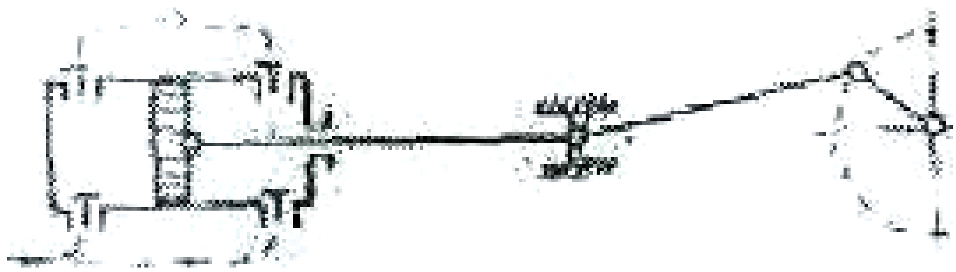


Figure 7.3 single-cylinder, horizontal, single-stage double-acting piston compressor.

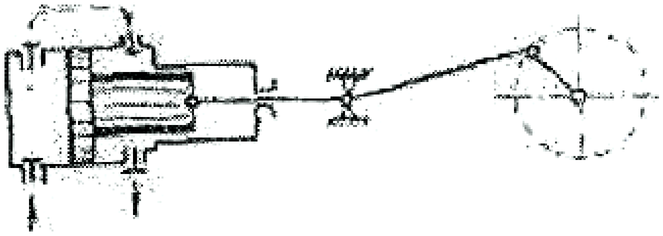


Figure 7.4 piston compressor with differential cylinder.

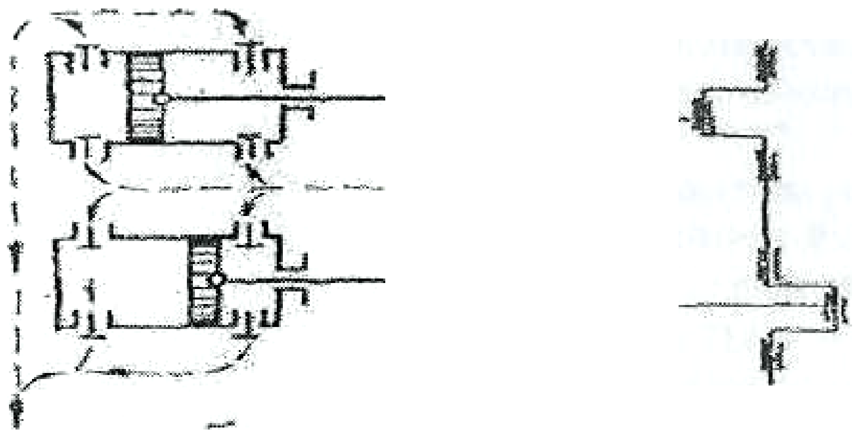


Figure 7.5 Two-cylinder, horizontal, single-stage double-acting piston compressor.

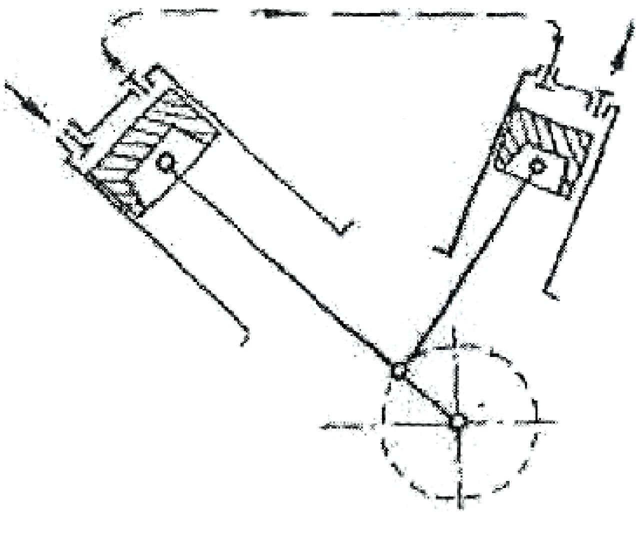


Fig. 7.6 Two-cylinder, angular V-shaped, two-stage piston compressor of simple action.

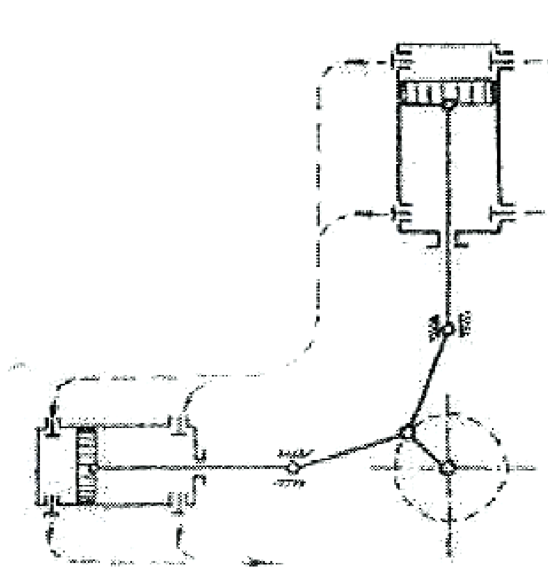


Fig. 7.7 Two-cylinder, rectangular, two-stage double-acting piston compressor

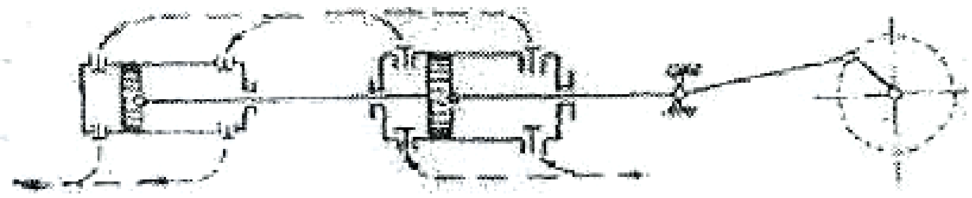


Fig. 7.8 Homogeneous, two-cylinder, two-stage double-acting piston compressor

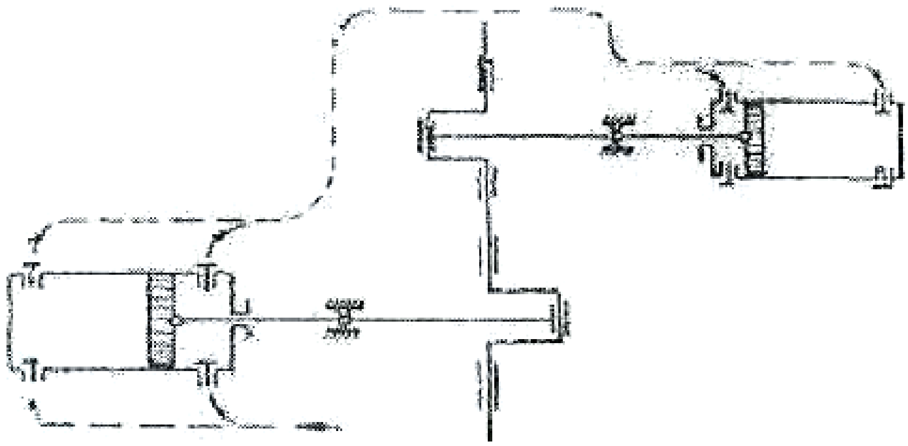


Fig. 7.9. Two-row, two-cylinder, two-stage double-acting piston compressor.

LABORATORY WORK No. 8
STUDY OF ROTARY COMPRESSOR DESIGNS.
The purpose of the work is to study the designs of rotary compressors
general information.

The classification of compressors is made according to their characteristic features.

According to the method of exposure to air (gas):

-volumetric or displacement compressors (reciprocating, rotary, screw, plate, etc.), in which the air (gas) pressure increases by reducing the working volume:

-vane (turbochargers), where an increase in air (gas) pressure occurs as a result of the force interaction of air (gas) with the blades of the rotating impellers of the machine. By the kind of compressible gas:

- air,
- ammonia,
- frescoes, etc.

By the value of the final pressure;

- low pressure compressors (0.3-1.0 MPa or 10 atm.).
- medium pressure compressors (1.0-10 MPa or 10-100 atm.).
- high pressure compressors (over 10 MPa or 100 atm.).

ROTARY PLATE COMPRESSOR

The rotary plastic compressor (Fig.8.1.) is a cylindrical cone 1, inside which the rotor 2 is eccentrically placed.

The diameter of the cylinder is much larger than the diameter of the rotor and therefore the surface of the cylinder and the rotor form a crescent-shaped space. The slots where the plates 3 are placed are cut on the rotor. When the rotor rotates, the plates slide out of the slots, dividing the crescent-shaped space into chambers, the volumes of which are minimal, at the bottom and maximum at the top, when the rotor rotates, the direction of rotation is shown in (Fig. 8.1) with the arrow, the volumes of chambers "A" increase as they move, and chambers "B" they are decreasing. The air from the suction pipe enters the chambers "A" is compressed in the chambers "B" and from the point "B" is pushed into the heating pipeline.

From point "C" to point "i" there is an expansion of air from the harmful space (the harmful space is formed by the presence of a gap between the rotor and the cylinder of its lower part).

Thus, the principle of operation of a rotary plastic compressor is based on the compression of air in gradually decreasing chambers bounded by the inner surface of the cylinder, the rotor and its plates, during the rotation of the rotor

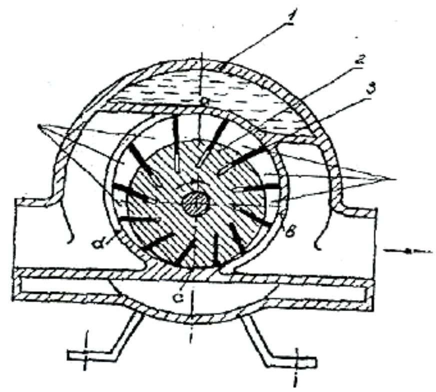


Figure 8.1 Rotary plate compressor.

SCREW COMPRESSORS

Screw compressors belong to volumetric machines 3 two cylinders, the side surfaces of which are interconnected, the leading 1 and the driven 2 rotors are placed (Fig.8.3.). The rotors are shaped like screws with a large lifting angle and a different number of teeth (as shown in Fig.8.3. there are four teeth on the driving rotor and six teeth on the driven rotor). The teeth are convex at the leading rotor and concave at the driven one. The rotors rotate in bearing supports 3 and 4 and are clinimatically connected by means of gears 5. The speed of rotation of the driving rotor is greater than the driven one. The screw surfaces do not touch each other, the presence of the necessary minimum gap between the rotors.

When the rotors rotate, atmospheric air from the suction pipe 6 enters the screw valves between the rotors and the housing 8, which at that moment turned out to be open from the ends. With the rotation of the rotors, the cavities will move, and the intake of atmospheric air into them will stop. Further rotation of the rotors leads to a gradual compression of air as the teeth of one rotor fill the cavities of the other. The air compression ends after it ends after" when the compressed air filling cavities are connected to the discharge pipe 7. The compressed sequence is shown in Fig.8.2. The 6VV-25/9 screw compressor is designed to supply compressed air to pneumatic machines and mechanisms used in the sinking of vertical shafts of mines, horizontal mining in the coal mining industry and installed on the surface of mines in an environment not dangerous for dust and gas.

The compressor unit 6VV-39 is designed to work as part of drilling rigs of the SBS type.

The compressor unit, the oil cooling unit and the control panel are complete, ready-to-operate unit tests.

The compressor operates according to the following scheme:

The air through the filter air valve, inlet enters the compressor, where it is compressed to the specified parameters and simultaneously mixed with oil injected through drilling. The oil is fed into the working cavity for lubrication temperature reduction reduction of internal constriction of the compressing air. From the compressor, the oil-air mixture of the spring, through the return valve enters the frame of the oil tank, the check valve is installed to precede the return flow of compressed air after the compressor is installed.

In the oil tank frame, drip oil is separated from the oil-air mixture. The air separated from the drip oil mixture passes through an oil separator mounted in the frame of the oil tank. Here, a more thorough separation of the oil from the oil-air mixture takes place. The oil separated in the oil separator is fed to the compressor. The purified air, then through the maintenance pressure valve, enters the consumer.

The compressor is a screw volumetric machine based on working bodies, which are two rotors rotating in rolling bearings.

The air sucked in by the compressor enters the suction cavity of the compressor. When the rotors rotate, the incoming air fills the entire length of the screw depressions. Then the volumes of air filling the depressions of the screws are cut off from the suction cavity and gradually compressed by the teeth entering these depressions. Oil is injected into the compression inlet of the screw cavity in order to take away the heat released by compressing the air sealing the gaps on the screw surfaces of their lubrication.

Compression of the oil-air mixture ends at the moment of connection of the depressions with the compressor discharge window.

The compressor consists of the following main components and parts:

Compressor housing, suction cavity, boring block, rolling bearings, rotor screws, drive rotor, pistons, bleed valve, bypass valve, pressure maintenance valve, shut-off valve, safety valve, tank frame, air filter.

Advantages of rotary compressors:

- simplicity of construction,
- compactness,
- no valves,
- uniform air supply,
- better balance. Disadvantages of rotary compressors:
- increased wear of the rubbing parts, as a result of the limited speed and rotation of the rotor;
- large friction losses;
- worst air cooling conditions in the compressor.

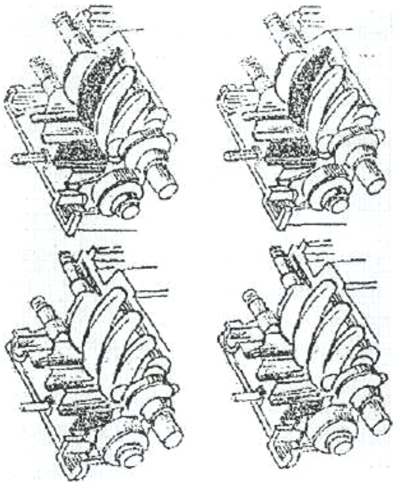


Figure 8.2 Diagram of the sequence of air compression in a screw compressor

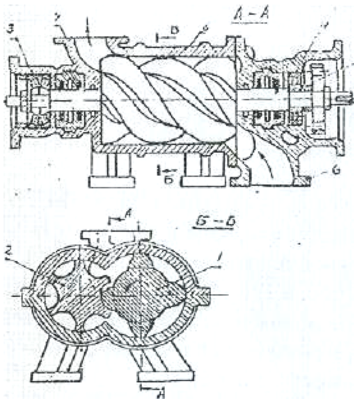


Figure 8.3 Diagram of the screw compressor.

BRANCH OF THE FEDERAL STATE AUTONOMOUS EDUCATIONAL INSTITUTION OF HIGHER EDUCATION
"National Research Technological University "MISIS" in Almalyk

DEPARTMENT OF "MINING"

METHODICAL INSTRUCTION

to perform practical work on the discipline "Stationary machines"

Almalyk 2022

PRACTICAL WORK No. 1

CALCULATION OF THE MAIN DRAINAGE SYSTEM OF THE MINE

Initial data for the calculation

1. The normal daily inflow of water into the mine is QH = 5600 m3/day.

2. The maximum daily water flow is Qmax = 14800m3/day.

3. The depth of the mine shaft — Nsh = 480 m.

4. The length of the pipeline on the surface is L1=210 M.1.

CALCULATION OF INITIAL PARAMETERS FOR THE SELECTION OF DRAINAGE PLANT EQUIPMENT

1.1 Estimated capacity of the pumping station of the main drainage plant of the mine:

a) by normal water flow

$$Q_p = Q_H/T_H = 5600/20 = 280 \text{ м}^3/\text{ч};$$

б) by maximum water flow

$$Q_{pm} = Q_{max}/T_H = 14\,800/20 = 740 \text{ м}^3/\text{ч},$$

where T_N = 20 h is the standard number of hours for pumping daily water flows according to Safety Rules (PB).

Economically feasible speed of water movement through the pipes of the injection pump.

$$V_{эк} = 0{,}54\sqrt[4]{Q_P} = 0{,}54\sqrt[4]{280} = 2{,}21 \text{ м/с},$$

where Q_P is the estimated capacity of the drainage system according to the normal daily water flow, m3 / h

1.3. The estimated diameter of the discharge pipeline

$$D_p = \sqrt{\frac{4Q_P}{3600\pi v_{эк}}} = \sqrt{\frac{4\cdot 280}{3600\cdot 3{,}14\cdot 2{,}21}} = 0{,}212\,м .$$

1.4. Calculated coefficient of linear hydraulic resistances of pipelines

$$\lambda_P = \frac{0{,}0195}{\sqrt[3]{D_P}} = \frac{0.0195}{\sqrt[3]{0.212}} = 0.0327$$

where Dp is the estimated diameter of pipelines, m

1.5. Geodetic height of water rise to the surface

$$H_г=H_ш+H_{вс}+h_п=480+4,5+15=486 \text{ м}.$$

where $H_{вс}$ = 4-5 m is the approximate suction height of the pumps; $h_п$ = 0.5-2 m is the height of the water rise above the surface of the mine.

1.6. Estimated length of pipelines

$$L_р = H_ш+L_{вс}+l_{тх}+l_{нк}+l_1= 480 + 10 + 18 + 25 + 210 = 743 \text{ м},$$

where $L_{вс}$ = 8-12 м - suction pipe length; $l_{тх}$ =15-20 м - the length of the pipeline in the pipe running; $1_{нк}$ = 20-30 м — дthe length of the pipeline in the pumping chamber of the drainage system.

1.7. Design head of the drainage pumping station

$$H_P=H_г+\left(1+\lambda_p\frac{L_P}{D_P}+\Sigma\xi_P\right)\frac{v_{эк}^2}{2g}=486+\left(1+0.0327\frac{743}{0.212}+28\right)\frac{2.21^2}{2\cdot 9.81}=521.7м$$

Where $\sum\xi_P$ = 25-30 — the estimated sum of the coefficients of local hydraulic resistances of the pipeline system.

2. CALCULATION AND SELECTION OF PIPELINES

2.1. Design water pressure in the discharge pipe

$$P_P=10^{-6}\rho gH_P=10^{-6}\cdot 1025\cdot 9.81\cdot 521.7=5,25МПа,$$

where p = 1020-1030 кг/м^3 — the density of pumped mine water.

2.2. The minimum wall thickness of the pipes of the injection pump according to the strength conditions

$$\delta_0=\frac{1875\,p_\rho D_\rho}{\sigma_\rho}=\frac{1875*5,25*0,212}{412}=5.07мм,$$

where Dp — estimated pipe diameter, м; σ_B — temporary tear resistance of the pipe material, MPa. We accept St4sp grade steel with temporary tear resistance for pipelines σ_B = 412 МПа.

2.3. Estimated wall thickness of pipes

$$\delta_0=1,18[\delta_0+(0,25+\delta_{кн})12]=$$

$$=1,18[5,07+ (0,25 + 0,2)12] = 12,4 \text{ мм},$$

where 1.18 is a coefficient that takes into account the minus tolerance of the pipe wall thickness; — the rate of corrosion wear of the inner surface of pipes, mm / year; t = 10-15 years — the estimated service life of pipes.

For acid mine waters with a hydrogen index pH 6-7 the rate of corrosion wear is $\delta_{КН}$ = 0,20 мм/год.

2.4. The choice of pipes for the injection pump is made according to the calculated internal diameter $D_р$= 212мм and wall thickness δ_p =12,4 мм. We accept pipes with an internal diameter for the discharge stave $D_н$= 217мм and wall thickness δ = 14мм

$$\text{Труба } \frac{245*14*8000 \;\; ГОСТ\ 8732-78}{Б\ Ст4сп\ \ ГОСТ\ 8731-74}$$

2.2. For the suction pipeline (Dbs = Dn + 25 mm), we accept pipes with an internal diameter of Dbs = 231mm and a minimum wall thickness δ = 7мм:

$$\text{Труба } \frac{245*7*8000 \;\; ГОСТ\ 8732-78}{Б\ Ст4сп\ \ ГОСТ\ 8731-74}$$

2.6. We accept the number of pipelines of the injection pump:

$Z_{тр}$ = 2 (working and standby).

3. SELECTION OF PUMPS AND THEIR CONNECTION SCHEMES

3.1. The choice of pumps is made according to the calculated flow rate $Q_р$ = 280 м^3/ч and pressure $H_р$ = 521,7м with a focus on multistage sectional pumps of the CNS brand. In accordance with the fields of operating modes shown in Fig. 1, we accept a CNS brand pump for drainage 300-120...600 with the following technical characteristic: nominal feed — $Q_н$ = 300 м^3/ч; nominal pressure — $H_н$ = 120x600 м; the maximum efficiency is 0.71; the rotation speed is n = 1475 rpm; the number of stages is i_{st} = 2-10.

3.2. Pressure characteristic of the pump stage of the CNS 300-120 brand...600 is shown in Table 1.

3.3 Estimated number of pump stages

$$i_{СТ} = \frac{H_p}{H_1} = \frac{521,7}{59} = 8,84$$

Table 1.1

$Q, м^3/ч$	0	75	150	225	300	375
H_1,м	67	68	67.5	66	60	48,5
η, %	0	36	59	69	71	66
$\Delta h_Д$,м	—	—	—	3,2	4,0	5,8

where H1 = 59 m is the head of the pump stage at a flow rate close to the design capacity of the drainage system. We accept iCT = 9.

3.4 The number of working pumps and their connection scheme.

We accept ZP = 1, since the calculated head and flow are provided by one pump.

3.5 The number of hot reserve pumps is assigned from the following conditions: pumps must be of the same type; the volume of the reserve is not less than 100%; the total supply of working and hot reserve pumps must ensure the calculated capacity of the drainage system according to the maximum daily water flow Qpm = 740 m3/h. We accept Zgr = 2.

3.6. The number of cold reserve pumps is selected from the condition that their total capacity should be at least 50% of the total capacity of the working pumps. In addition, the pumps must be of the same type.

We accept Z_{xp} = 1.

3.7. Total number of pumps at the drainage pumping station

$$Z_Н = Zp+Zгр + Zxp= 1 +2+ 1 =4.$$

4. SWITCHING HYDRAULIC SCHEME OF THE DRAINAGE PUMPING STATION

Pumps on the drainage system must be connected to pipelines in such a way that any of them can be connected to any pipeline of the discharge stave. With the diameter of the pipes of the injection pump D 300mm, a typical switching circuit with an annular pipeline at the ceiling of the pumping chamber and a common receiving sump is usually used.

The switching scheme of pumping equipment is shown in Fig. 1.

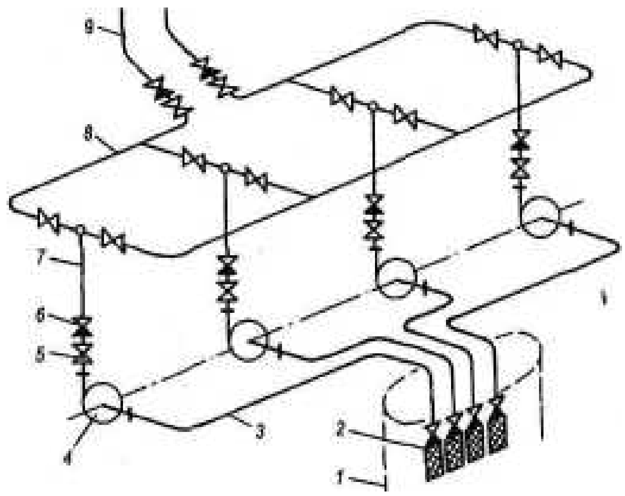


Fig. 1. Switching diagram of the main drainage system:

1 — receiving well; 2 — intake device with a receiving grid; 3 — suction pipeline; 4 — pumping units; 5 — valves; 6 — check valves; 7 — riser; 8 - ring collector; 9 — discharge pipeline

5. CALCULATION OF THE PRESSURE CHARACTERISTICS OF THE EXTERNAL NETWORK

5.1. Coefficients of linear hydraulic resistances:

а) of the discharge pipeline

$$\lambda_H = \frac{0{,}0195}{\sqrt[3]{D_H}} = \frac{0{,}0195}{\sqrt[3]{0{,}217}} = 0{,}0324$$

б) suction pipe

$$\lambda_H = \frac{0{,}0195}{\sqrt[3]{D_{BC}}} = \frac{0{,}0195}{\sqrt[3]{0{,}231}} = 0{,}0319$$

where D_H= 0,217 м and D_{BC} = 0,231 м — the diameters of the discharge and suction pipelines, respectively.

5.2. Length of pipelines:

а) suction $L_{вс}$ = 10 м;

б) injection $L_н = L_р - L_{вс:}$ = 743 - 10 = 733 м.

5.3. The results of calculating the sum of the coefficients of local hydraulic resistances of the suction pipeline are shown in Table 1.2.

Table 1.2.

Устройство	Коэффициент сопротивления ξ_i	Количество n_i	$\xi_i n_i$
Приемный клапан с сеткой	4,4	1	4,4
Нормальное колено с углом поворота 90^0	0,182	3	0,546
Сальниковый компенсатор	0,2	1	0,2
ИТОГО	$\sum\xi_{BC} = 5{,}15$		

5.4. Number of flange joints in the discharge pipe

$$Z_{CT} = \frac{L_H}{l_T} = \frac{733}{8} = 92$$

where l_T = 8м — length of one standard length pipe.

5.5 The results of calculating the sum of the coefficients of local resistances of the discharge pipeline are shown in Table. 1.3.

Table 1. 3

Устройство	Коэффициент сопротивления ξ_i	Количество n_i	$\xi_i n_i$
Задвижки открытые	0,08	3	0,24
Обратные клапаны	5,5	2	11,0
Нормальное колено с углом поворота 90°	0,168	3	0,504
Нормальное колено с углом поворота 135°	0,084	2	0,168
Тройники	0,75	2	1,5
Сальниковый компенсатор	0,2	6	1,2
Воздушная колонна	0.5	1	0,5
Фланцевые стыки	0,15	92	13,8
ИТОГО	$\sum\xi_{BC} = 28{,}9$		

5.6. Generalized external network resistance coefficient

$$R_c = \left(1+\lambda_{вс}\frac{l_{вс} D_H^4}{D_{вс} D_{вс}^4}+\lambda_H \frac{L_H}{D_H}+\sum\xi_{вс}\frac{D_H^4}{D_{вс}^4}+\sum\xi_H\right)\frac{6{,}25*10^{-8}}{g D_H^4}=$$

$$\left(1+0{,}{,}319\frac{10*0{,}217^4}{0{,}231*0{,}231^4}+0{,}0324\frac{733}{0{,}217}+5{,}15\frac{0{,}217^4}{0{,}231^4}+28{,}9\right)*\frac{6{,}25*10^{-8}}{9{,}81*0{,}217^4}=4{,}15*10^{-4}\ ч^2/м^5$$

5.7 The calculation of the pressure characteristics of the external network is carried out according to the formula

$$H_c = H_г + R_c Q^2$$

where Q — pump flow rate, м^3 /ч.

The calculation results are shown in Table. 1.4.

Table 1. 4.

Q, м^3/ч	0	75	150	225	300	375
H_c,m	486	490,7	495,3	507,0	523,4	544,4

6. DETERMINATION AND ANALYSIS OF THE OPERATING MODE OF THE DRAINAGE SYSTEM

6.1. A summary table for the graphical definition of the operating mode is presented in table. 1.5

Table 1.5

Q, м^3/ч	0	75	150	225	300	375
$Q_{нс}$м^3/ч	0	75	150	225	300	375
H_1, м	67	68	67.5	66	60	48,5
$H_{нс}$,м	603	612	607,5	594	540	436,5
H_c, м	486	490,7	495,3	507,0	523.4	544,4
η,%	0	36	59	69	71	66
$\Delta h_Д$,м	—	—	—	3,2	4,0	5,8
$H_{вп}$,м	—	—	—	6,51	5,71	3,91

Notes to the table. 1.5

1. The capacity of the pumping station is determined as follows:

$$Q_{нс} = Q z_{пр} = Q*1 = Q$$

where Znp — number of working pumps in parallel connection.

2. Pumping station pressure

$H_{нс}=H_1 i_{ст}$

where $i_{ст}$ — total number of working pump stages.

3. The permissible vacuum suction height of the pumps is calculated by the formula

$$H_{вд}=\frac{p_0-p_П}{\rho g}-\Delta h_Д$$

where $p_0 = 10^5$ Па — atmospheric pressure; $p_П = 2337$ Па — saturated water vapor pressure at temperature t = 20 °C.

6.2. Graphical definition of the operating mode of the drainage system is shown in Fig. 1.2. The operating mode of the drainage system is characterized by the following parameters:

1. Valid pumping station feed — $Q_Д = 314$ м³/ч.
2. Actual pressure — $H_Д=527$ м.
3. Efficiency in actual operating mode — $\eta_Д = 0{,}70$.
4. Permissible vacuum suction height at the actual operating mode $H_{вдд}=5{,}45$ м.

6.3 Checking the operating mode:

6.3.1. Ensuring the estimated flow rate

$Q_Д \geq Q_P$ => 314>280. The condition is met.

6.3.2. Ensuring the stability of the operating mode

$H_г \leq 0{,}9H_0$, где H_0 — pump head at zero supply.

486 < 0,9*603 = 542,7. The condition is met

6.3.3. Efficiency of the operating mode

$\eta_Д \geq 0{,}90 \cdot \eta_{max}$ =>0.70> 0,9 · 0,71 = 0,639. The condition is met.

6.3.4. No cavitation during pump operation

$$H_{max}=H_{BC}+\left(\lambda_{вс}\frac{L_{вс}}{D_{вс}}+\sum\xi_{вс}\right)\frac{v_{вс}^2}{2g}\leq H_{вдд}$$

where v_{BC} — the speed of water in the suction line at the actual supply,

$$v_{вс}=\frac{4Q_д}{3600\cdot\pi D_{вс}^2}=\frac{4\cdot 314}{3600\cdot 3{,}14\cdot 0{,}231^2}=2{,}08 \text{ м/с}.$$

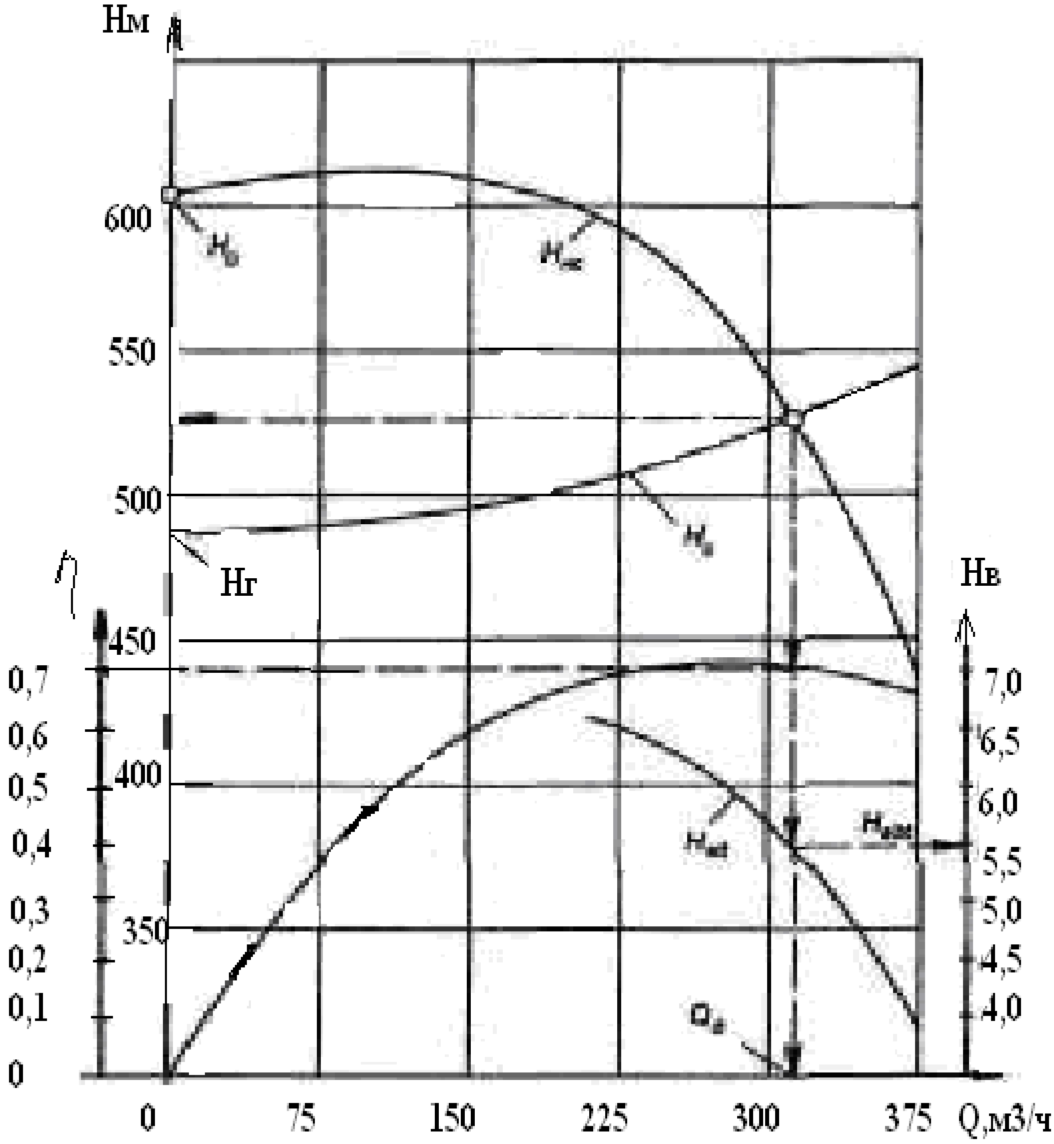


Fig. 1. 2. Graphical definition of the operating mode of the drainage system.

$$H_{вак} = 4{,}5 + \left(\left(0{,}0318\frac{10}{0.231} + 5{,}15\right)\frac{2{,}08^2}{2\cdot 9{,}81}\right) = 5{,}94 \text{ м}$$

$H_{вак}$ =5,94 м > $H_{вдд}$ = 5,45 м. Условие не выполняется.

7. PERMISSIBLE SUCTION HEIGHT AND ENSURING THE SUCTION CAPACITY OF PUMPS

7.1. Permissible suction height of pumps

$$H_{вак} = H_{вдд} - \left(\lambda_{вс}\frac{L_{вс}}{D_{вс}} + \Sigma\,\xi_{вс} \right)\frac{v_{вс}^2}{2g} = 5{,}45 - \left(0{,}0318\frac{10}{0.231} + 5{,}15\right)\frac{2{,}08^2}{2\cdot 9{,}81} = 4{,}01 \text{ м.}$$

7.2. Ensuring the necessary suction capacity of pumps and operation without cavitation. Since the sl is > 3.5 m, no additional technical means are required to ensure the cavitation operation of the drainage system. It is enough to position the pumping units in such a way that the axis of rotation is at a height of no more than 4 m above the water level in the catchment.

8. PUMP DRIVE AND ENERGY CONSUMPTION OF THE DRAINAGE SYSTEM

8.1. Estimated power of the electric pump drive

$$N_р=(1{,}1\text{-}1{,}15)10^{-3}\rho g\frac{H_Д \bullet Q_Д}{3600 \bullet \eta_Д \bullet Z_{ПР}}=1{,}15\cdot10^{-3}1025\cdot9{,}81\frac{527\bullet314}{3600\bullet0{,}7\bullet1}=759{,}3\ кВт.$$

8.2 As a pump drive, we accept electric motors of the VAO 630 M4 brand with the following technical characteristics:

1. Power N = 800 кВт.
2. Synchronous rotation speed п = 1500 об/мин.
3. Supply current voltage V= 6000 В.
4. Engine efficiency $ц_я$ = 0,954.
5. $\cos\phi$= 0,9.

8.3. Estimated number of machine hours of pump operation per day:

a) when pumping out the normal inflow

$$T_H=\frac{Q_H}{Q_Д}=\frac{5600}{314}=17{,}83\ ч.$$

б) when pumping out the maximum water flow

$$T_M=\frac{Q_{MAX}}{Q_Д}=\frac{14800}{314}=47{,}13\ ч.$$

8.4. Electricity consumption by pumping equipment of the drainage system

$$E_г=1{,}05\cdot10^{-3}\rho g\frac{H_Д Q_g}{3600\cdot\eta_Д\cdot\eta_{ЭС}}[(365-N_M)T_H+N_M T_M]=1{,}05\cdot10^{-3}$$

$$1025\cdot9{,}81\ \frac{527\cdot314}{3600\cdot0{,}7\cdot0{,}954\cdot0{,}96}X\ [(365-60)17{,}83+60\cdot47{,}13]=$$

$6257\cdot10^6$кВт ·ч/год,

where N_n = 60 сут — the number of days per year with the maximum water flow;

$\eta_{ЭС}$ =0,92:0,96 — Efficiency of the power supply network.

8.5. The specific consumption of electricity attributed to the unit volume of pumped water,

$$e = \frac{E_r}{(365 - N_M)Q_H + N \bullet Q_{\max}} = \frac{6{,}257 \bullet 10^6}{(365 - 60)5600 + 60 \bullet 14800} = 2{,}41 \text{ кВт·ч/м}^3$$

PRACTICAL WORK No. 2

CALCULATION OF THE INSTALLATION OF THE MAIN VENTILATION OF THE MINE

Initial data for the calculation

1. Required air consumption for mine ventilation —$Q_в$=90м^3/с.

2. Estimated depression of the mine ventilation network:

maximum — h_{max} = 3100 Па;

minimum — h_{min} = 2200 Па.

3. The location of the fan unit is at the cage trunk.

OPERATIONAL CALCULATION OF THE FAN

1. Estimated fan supply

$$Q_p=k_y Q_в = 1{,}2*90=108\ м^3/с,$$

where ku = 1,2 is a coefficient that takes into account leaks through the shaft and the supply channels of the fan installation. When the fan unit is located at the cage trunk, ku = 1,2.

2. Select the type of fan. The selection is made using the diagrams of the fields of the operating modes of the fans of the VOD and VC series (VCD). Two points are plotted on the diagrams, one of which has the coordinates Qp and hmax, and the other — Qp and hmin,. If both points fall simultaneously into the fields of operating modes of two types of fans, then preference is given to the one that provides the necessary operating modes with a larger value of the weighted average static efficiency. In our case, the design parameters correspond to one fan — VC32/600 with an impeller diameter of 3.2 m and a rotation speed of 600 rpm.

The aerodynamic characteristics of the selected fan are shown in Fig. 3, from which it can be seen that the calculated fan supply is realized with the following values of static efficiency: at maximum depression (point 2)— = 0.74; with minimal depression (point 2) — = 0.62.

Weighted average static efficiency of the fan

$$\eta_{cp} \frac{h_{\max}+h_{\min}}{h_{\max}/\eta_{\max}+h_{\min}/\eta_{\min}} = \frac{3100+2200}{3100/0{,}74+2200/0{,}62} = 0{,}685$$

3. Generalized coefficients of ventilation network resistance:

• with maximum depression

$$R_{max}=h_{max}/Q_p^2 = 3100/108^2 = 0{,}2658;$$

• with minimal depression

$$R_{min} = h_{min} / Q_p^2 = 2200/108^2 = 0,1885.$$

4. The calculation of the aerodynamic characteristics of the ventilation network is carried out according to the formula

$$h_{c.i}=R_iQ^2.$$

The calculation results are presented in Table. 6.

Table 6

$Q, м^3/с$	60	90	108	120	135	150
$h_{c.max}=R_{max}Q^2$, Па	957	2153	3100	3827	4844	5980
$H_{c.min}=R_{min}Q^2$, Па	679	1528	2200	2716	3438	4244

5. Graphical determination of the parameters of the operating modes of the fan (Fig. 2.1).

5.1. At the calculated operating modes (points 1 and 2 in Fig. 2.1):

• at maximum depression

$Q_1 = 108$ $м^3/с$; $h_1 = 3100$ Па; $\Theta_{на1} = 35°$; $\eta_{min} = 0,74$;

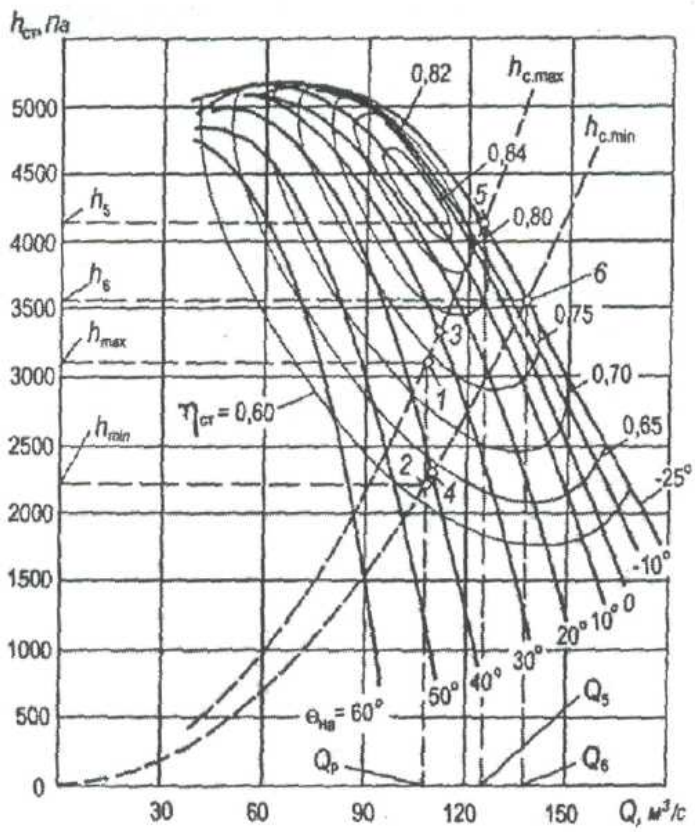


Fig. 3. Aerodynamic characteristics of the fan and ventilation network.

• with minimal depression

$Q_2 = 108$ $м^3/с$; $h_2 = 2200$ Па; $\Theta_{на2} = 42°$; $\eta_2 = 0,62$.

5.2. With forced operating modes (points 5 and 6 in Fig. 3):

• with maximum depression

Q_5 = 125 м^3/с; h_5 = 4140 Па; $\Theta_{на5}$ = -25°; η_5 = 0,79;

• with minimal depression

Q_6 = 137 м^3/с; h_6 = 3560 Па; $\Theta_{на6}$ = -25°; η_6 = 0,76.

6. Reserve performance of the fan unit under the design modes of ventilation:

• with maximum depression

$$\Delta\overline{Q}_1 = 100(\frac{Q_5}{Q_p} - 1) = 100(\frac{125}{108} - 1) = 15{,}7\,\%$$

• with minimal depression

$$\Delta\overline{Q}_2 = 100(\frac{Q_6}{Q_p} - 1) = 100(\frac{137}{108} - 1) = 26{,}8\,\%$$

7. Rated power on the fan shaft:

• with maximum depression

$$N_1 = 10^{-3}\, h_1 Q_1/\eta_1 = 10^{-3} \cdot 3100 \cdot 108/0{,}74 = 452{,}4 \text{ кВт};$$

$$N_5 = 10^{-3}\, h5 Q_5/\eta_5 = 10^{-3} \cdot 4140 \cdot 125/0{,}79 = 655{,}1 \text{ кВт};$$

• with minimal depression

$$N_2 = 10^{-3} h_2 Q_2/\eta_2 = 10^{-3} \cdot 2200 \cdot 108/0{,}62 = 383{,}2 \text{ кВт};$$

$$N_6 = 10^{-3} h_6 Q_6/\ \eta_6 = 10^{-3} \cdot 3560 \cdot 137/0{,}76 = 641{,}7 \text{ кВт}.$$

8. The choice of an electric motor for the fan is made according to the largest of the calculated capacities obtained above, taking into account a reserve of at least 10-15 %:

$$N_p = (1{,}1\text{-}1{,}15)\ 655{,}1 = 720{,}6\text{-}753{,}4 \text{ кВт}.$$

As a fan drive, we accept a synchronous electric motor SDV-15-39-10 with the following characteristics: rated power N = 800 kW; rotation speed n = 600 rpm; efficiency = 0.943.

9. Average annual electricity consumption

$$E_г = 1{,}05 \frac{N_1 + N2}{2\eta_{cp}\eta_{эс}\eta_{эд}} = 1{,}05 \frac{452.24 + 383.2}{2 \cdot 0{,}685 \cdot 0{,}943 \cdot 0{,}95} = 6{,}26 \cdot 10^6 \text{ кВт} \cdot \text{ч}$$

where $\eta_{ЭС}$ = 0,94-0,96 — Efficiency of the power supply network.

PRACTICAL WORK No. 3

CALCULATION OF A STATIONARY PNEUMATIC SHAFT INSTALLATION

Initial data for the calculation

1. The design scheme of the pneumatic installation is shown in Fig. 3.1.

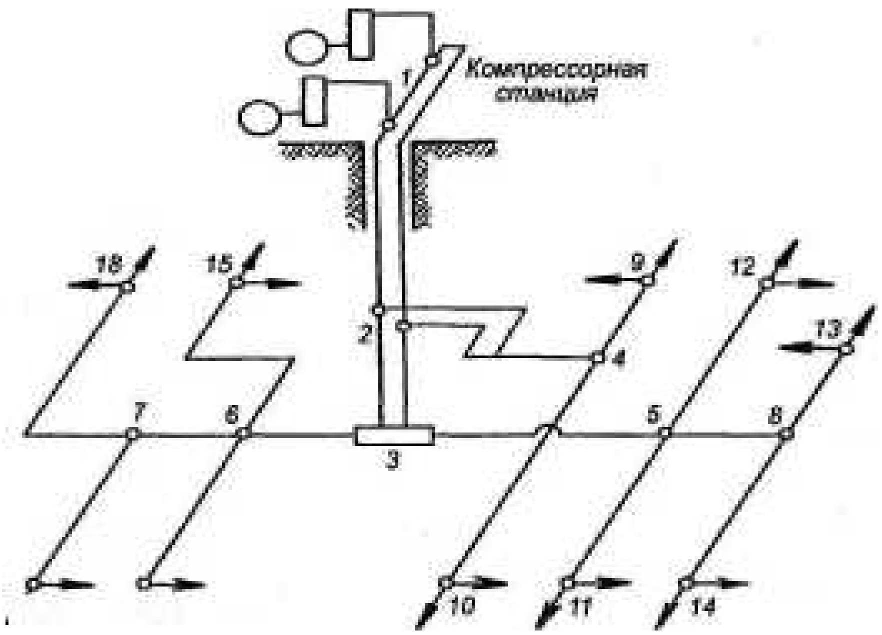


Fig. 3.1. Design scheme of the pneumatic installation

2. The distance between the nodal points of the pneumatic networks:

$L_{1\text{-}2}$ = 520 м; $L_{2\text{-}3}$ = 160 м; $L_{2\text{-}4}$= 420 м; $L_{3\text{-}5}$ = 180 м; $L_{3\text{-}6}$ = 240 м;

$L_{4\text{-}9}$ = 640 м; $L_{4\text{-}10}$ = 580 м; $L_{5\text{-}8}$ = 340 м; $L_{5\text{-}11}$ = 460 м; $L_{5\text{-}12}$ = 420 м;

$L_{6\text{-}7}$ = 440 м; $L_{6\text{-}15}$ = 320 м; $L_{6\text{-}16}$ = 400 м; $L_{7\text{-}17}$ = 420 м; $L_{7\text{-}18}$=510 м;

$L_{8\text{-}13}$ = 440 м; $L_{8\text{-}14}$ = 460 м.

3. Technical characteristics of compressed air consumers are presented in Table. 1.

4. Distribution of compressed air consumers by consumption points:

point 9— (1гп + 1ск + 6мо + 1бм + 1вм);

point 10 — (1гв + 1бс + 1вм + 1мл + 1нп);

point 11 — (1пм + 12мо + 2бм + 1вм + 1нп);

point 12 — (1ко + 2ск + 1вм + 2мл + 1нп);

point 13 — (1бс + 6мо + 1бм + 1вм + 2сл);

point 14 — (1ко + 2ск+ 1вм + 1мл + 1нп);

point 15 — (1гп + 1ск + 5мо + 1бм);

point 16 — (2гв + 6мо + 1бм + 2мл + 2нп);

point 17— (8мо + 2бм + 2вм + 2сл);

point 18 — (2пм + 10мо + 1бм + 1вм).

Note. The numbers before the letter designations of compressed air consumers indicate their number in the corresponding points.

Table 1.

Technical characteristics of compressed air consumers

Потребители сжатого воздуха (обозначение)	Количество n_i	q_i м³/мин	p_{ni} бар	K_{3i}	ψ_i	k_{Bi}
Очистные комбайны (ко)	2	48	4,5	1.0	1,10	1,0
Погрузочные машины (пм)	3	22	5,0	0,25	1,10	0,4
Грейферы проходческие (гп)	2	20	5,6	0,38	1,15	0,45
Маневровые лебедки (мл)	6	10	3,5	0,80	1,20	0,15
Скребковые конвейеры (ск)	6	16	3,5	0,80	1,10	1,0
Буросбоечные станки (бс)	2	16	4,5	1,0	1,20	0,5
Отбойные молотки (мо)	53	1.4	4,0	1,0	1,15	0,4
Бурильные молотки (бм)	9	3,5	5,0	1,0	1,15	0,65
Вентиляторы ВМП (вм)	9	5,0	4,0	0,7	1,0	1,0
Пневматические насосы (нп)	6	2,5	4,0	0,8	1,05	1,0
Скреперные лебедки (сл)	4	2,5	5,0	0,8	1,20	0,5
Гировозы (гв)	3	35	5,5	1,0	1,15	0,3

1. CALCULATION OF COMPRESSOR STATION PERFORMANCE

1.1 Total number of compressed air consumers

$$n_c=\sum n_i=n_1+n_2\ldots n_m= 2 + 3 + 2 + 6 + 6 + 2+53 + 9 + 9 + 6 + 4 + 3=105.$$

1.2. Weighted average coefficient of inclusion of consumers

$$K_B=\frac{\sum_1^m n_i q_i k_{3i}\psi_i k_{Bi}}{\sum_1^m n_i q_i k_{3i}\psi_i}=\frac{375,8}{617,7}=0.608$$

where ni is the number of i—type consumers of the same name; qi is the nominal consumption of compressed air by one consumer of the specified type; k3i is the load factor; i is the coefficient that takes into account the increase in air consumption due to wear of machines and mechanisms; kBI are the coefficients of inclusion of machines and mechanisms; m is the number of groups of the same type of consumers. Numerical values of parameters and coefficients are accepted in accordance with Table.1.

$\sum_{1}^{m} n_i q_i k_{3i} \psi_i k_{Bi}$ = 2 * 48 * 1,0 * 1,1 * 1,0 + 3 * 22 * 0,25 * 1,1 * 0,4 + 2 * 20 * 0,38 * 1,15 * 0,45 + 6 * 10 * 0,8 * 1,2 * 0,15 + 6 * 16 * 0,8 * 1,1 * 1,0 + 2 * 16 * 1,0 * 1,2 * 0,5 + 53 * 1,4 * 1,0 * 1,15 * 0,4 + 9 * 3,5 * 1,0 * 1,15 * 0,65 + 9 * 5,0 * 0,7 * 1,0 * 1,0 + 6 * 2,5 * 0,8 * 1,05 * 1,0 + 4 * 2,5 * 0,8 * 1,2 * 0,5 + 3 * 35 * 1,0 * 1,15 * 0,3 = 375,8 м^3/мин;

$\sum_{1}^{m} n_i q_i k_{3i} \Psi_i$ =2 * 48 * 1,0 * 1,1 + 3 * 22 * 0,25 * 1,1 + 2 * 20 * 0,38 * l,15 + 6 * 10 * 0,8 * 1,2 + 6 * 16 * 0,8 * 1,1 + 2 * 16 * 1,0 * 1,2 + 53 * 1,4 * 1,0 * 1,15 + 9 *3,5 * 1,0 * 1,15 + 9 * 5,0 * 0,7 * 1,0+6 * 2,5 * 0,8 * 1,05 + 4 * 2,5 * 0,8 * 1,2 + 3 * 35 * 1,0 * 1,15 = 617,7 м^3/мин.

The coefficient of simultaneity of the operation of consumers of k_o is determined graphically by the specified ps and K_t using a diagram, k_o = 0.72.

Estimated total air consumption by consumers

$Q_n = \mu K_0 \sum_{1}^{m} n_i q_i k_{3i} \Psi_i$=1,10,72-617,7 = 489,2 м^3/мин,

where μ = 1,05+1,15 — reserve ratio for unaccounted compressed air consumers in the mine and on the surface.

1.5. Total length of pneumatic network pipelines

$\sum_{1}^{m} L_{i-j}$ = $L_{1\text{-}2}$+$L_{3\text{-}3}$…= 520 + 160 + 420+180 + 240 + 640 + 580 + 340 + 460 + 420 + 440 + 320 + 400+420 + 510 + 440 + 460 = 6950м.

1.6. The amount of leaks due to leakage of the pneumatic network piping system

$Q_{yc} = \frac{p+1}{6} a \sum_{1}^{m} L_{i-j} = \frac{5+1}{6}$ 0,003·6950=20,85 м^3/мин,

where p = 5 бар — the calculated average pressure of compressed air at the points of consumption; a = 0.003 m3 / (min-m) is the permissible value of specific losses of compressed air at the calculated pressure of consumers of 5 bar, attributed to the unit length of the pipeline

1.7. The amount of leaks in air consumption points

$Q_{ус}= \frac{p+1}{6} b\sum_{1}^{m} n_i = \frac{5+1}{6}$ 0,4-105 = 42,0 м³/мин,

where b= 0,4 $м^3$/мин — the standard value of leaks in the connecting elements at a compressed air pressure of 5 bar.

1.8. Estimated capacity of the compressor station

$Q_{кс}+ Q_{п} +Q_{ус}+Q_{уп}$=489,2+20,8+42,0 = 552,0 $м^3$/мин.

2. CALCULATION OF COMPRESSED AIR TRAVEL EXPENSES

2.1. The calculation of the travel flow of compressed air in the peripheral sections of the pneumatic network is carried out according to the formula

$$Q_{i-j}=\sum_{1}^{m} n_i q_i k_{3i} \psi_i k_{Bi} + a L_{i-j} + b\sum_{1}^{m} n_i$$

2.2. The travel flow rate of compressed air in the sections of the main pipeline is determined as follows:

$$Q_{i-j}=\sum (Q_{i-j}) + a L_{i-j}$$

where Q_{i-j} — travel expenses of the connected peripheral sections of the pneumatic network.

2.3. The results of the calculation of travel expenses are presented in Table.2.

Table. 2

Обозначение участка сети, i-j	Потребление сжатого воздуха в конечном пункте участка или перечень потребителей, подключенных к конечному пункту	Длина участка L_{i-j}, м	Количество потребителей n_{i-j}	Путевой расход воздуха Q_{i-j} м^3/мин
4-9	1гп+1ск+6мо+1бм+1вм	640	10	33,9
4-10	1гв+1бс+1вм+1мл+1нп	580	5	32,4
5-11	1пм+12мо+2бм+1вм+1нп	460	17	29,2
5-12	1ко+2ск+1вм+2мл+1нп	420	7	23,5
6-15	2гп+1ск+5мо+1бм	320	8	28,0
6-16	2гв+6мо+1бм+2мл+2нп	400	13	44,1
7-17	8мо+2бм+2вм+2сл	420	14	26,7
7-18	2пм+10мо+1бм+1вм	510	14	24,5
8-13	1бс6мо+1бм+1вм+2сл	440	11	27,7
8-14	1ко+2ск+1вм+1мл+1нп	460	6	91,8
5-8	$Q_{8-13}+Q_{8-14}$	340	-	120,5
6-7	$Q_{7-17}+Q_{7-18}$	440	-	52,5
3-6	$Q_{6-7}+Q_{6-15}+Q_{6-16}$	240	-	125,3
3-5	$Q_{5-8}+Q_{5-11}+Q_{5-12}$	180	-	173,7
2-3	$Q_{3-5}+Q_{3-6}$	160	-	299,5
2-4	$Q_{4-9}+Q_{4-10}$	420	-	67,6
1-2	$Q_{2-3}+Q_{2-4}$	520	-	368,7

3. CALCULATION OF PRESSURE LOSSES IN THE PNEUMATIC NETWORK

3.1. The average pressure in the air ducts of the pneumatic

$$P_c=P_{п,мах}+\frac{\Delta P_{мах}}{2}=5{,}6+\frac{1{,}8}{2}=6{,}5 \text{ бар},$$

where rp.Pp, max = 5.6 bar is the maximum operating pressure of consumers in accordance with Table. П-1;

$\Delta P_{мах}$ = 1,5-5-2,0 бар — preset maximum pressure loss in the air ducts of the pneumatic network.

Standard specific pressure loss

$\Delta P_{ун}$ = 0,0002/0,0004 бар/м = 20/40 Па/м.

3.2. According to the specified travel expenses Qi-j (Table.3), as

well as the average pressure pc = 6.5 bar and the standard value = 30 Pa / m graphically using a nomogram, the diameters of the Qi-j pipes and the actual specific losses in the sections of the pneumatic network are determined.

3.3. The total pressure loss in the sections of the pneumatic network is determined by the formula

$$\Delta P_{i-j} = 1{,}5 \bullet \Delta P_{УД} L i - j$$

where 1,15 — a coefficient that takes into account local pressure losses at the pipe joints and in the start-up and safety valves

3.5. The results of calculations of pressure losses in sections of the pneumatic network are presented in Table 3.3.

Table. 3

i-j	Q_{i-j} $м^3$/мин	Q_{i-j} , мм	$\Delta P_{óí}$ Па/м	L_{i-j} ,м	ΔP_{i-j}, бар
4—9	33,9	125	45	640	0,331
4—Ю	32,4	125	43	580	0,287
5—11	29,2	125	33	460	0,175
5—12	23,5	125	22	420	0,106
6—15	28,0	125	32	320	0,118
6—16	44,1	150	31	400	0,143
7—17	26,7	125	24	420	0,116
7—18	24,5	125	25	510	0,147
8—13	27,7	125	32	440	0,162
8—14	91,8	200	31	460	0,164
5—8	120,5	200	41	340	0,160
6—7	52,5	150	47	440	0,238
3—6	125,3	200	42	240	0,116
3—5	173,7	250	43	ISO	0,089
2—3	299,5	300	46	160	0,085
2—4	67,6	150	72	420	0,348
1—2	368,7	350	60	520	0,359

4. DESIGN PRESSURE OF THE COMPRESSOR STATION

4.1. The design pressure of the compressor station according to the maximum operating pressure of consumers (consumption point 9)

P= $p_{П.MAX} + \sum\Delta P_{i-j} = \Delta P_{п.мах} + \Delta P_{1-2} + \Delta P_{2-4} + \Delta P_{4-9}$= 5,6 + 0,359 + 0,348 + 0,331=6,64 бар,

where $p_{п.max}$ = 5,6 бар — максимальное рабочее давление потребителей в соответствии с табл. 7; $\sum\Delta P_{i-j}$— total pressure losses in the air ducts from the compressor station to the consumer with the maximum operating pressure.

4.2. Design pressure of the compressor station along the longest section of the pneumatic network (consumption point 18)

P= $p_{П.MAX} + \sum\Delta P_{i-j}$=$P_п$+$\Delta P_{1-2} + \Delta P_{2-3} + \Delta P_{3-6} + \Delta P_{6-7} + \Delta P_{7-18}$= 5,0 + 0,359 + 0,085 + 0,116 + 0,238 + 0,147 = 5,945 бар,

where $P_п$ = 5,0 бар — maximum operating pressure of the consumer at the most remote point of air consumption; $\sum\Delta P_{i-j}$— total pressure loss in the air ducts to the most remote compressed air consumer.

4.3. As a calculation, we take the largest of the two calculated pressures, т. е. p = 6,64 бар.

5. SELECTION OF COMPRESSOR STATION EQUIPMENT

5.1. The choice of compressors is made according to the design capacity of the compressor station QK = 552 m3/min and the design pressure p = 6.64 bar. In the example under consideration, two variants of the compressor equipment of a pneumatic installation are possible:

a) 5 identical piston working compressors of the brand
4VM10-P0/9 and 2 backup compressors of the same brand (total:
5 + 2 = 7 compressor units);

б) 2 working and 1 backup turbochargers of the brand
K-250-61-2 , as well as 1 working and 1 backup piston
compressor 4BMI0-63/9 (total: 3 + 2 = 5 compressor units).

The first variant of the equipment of a pneumatic installation with the same type of compressors 4VM10-110/9 having the following technical characteristics is selected: performance — QK = 110m /min; overpressure —p = 8 bar; electric drive power — 630 kW; compressor weight — 18.7 t.

5.2. Total number of compressors

$$Z_К = Z_{раб} + Z_{рез} = 5 + 2 = 7.$$

5.3. If the number of compressors is > 3 one common air collector is provided for each pair of compressor units. Therefore, the required number of air collectors at the compressor station

$$Z_B = Z_K/2 = 7/2 = 3,5$$

We accept $Z_в = 3$, taking into account, what$Z_{раб} = 5$.

5.4. Estimated capacity of the air collector

$$Q_B = 1,6\sqrt{2Q_B} + 1,6\sqrt{2*110} = 23,7м^3$$

5.5. We accept the air collector of the brand B-25 with the following technical characteristics: capacity — 25m3; inner diameter — 2.0 m; shell thickness — 9mm; bottom thickness — 12mm; air collector weight — 4,615 t.

5.6. Oil separators of the V-3.20 brand with the following technical characteristics are provided for air collectors: capacity — 3.2 m; overpressure — 8 bar; weight — 1.09 t.

5.7. A vertical shell-and-tube end cooler of the brand is provided for each compressor unitHC-100 with the following technical characteristics: cooling surface — 180m2; excess air pressure — 8 bar; excess cooling water pressure — 2 bar; inlet air temperature — 140 °C; outlet air temperature — 30 ° C; cooling water temperature — 20 ° C; cooler weight — 2.74 t.

5.8. 5.8. The selection of filters for cleaning the intake atmospheric air is made according to the calculated performanceof the compressor station. We accept a self-cleaning mesh filter of the KT—40 brand with the following technical characteristics: working cross—section — 3.94 m2; estimated air consumption — 655 m3/min; amount of oil - 290kg; weight - 0.65 t.

6. CALCULATION OF THE COMPRESSOR STATION COOLING SYSTEM

6.1. The degree of air compression in the compressor stage

$$\varepsilon_1 = \sqrt{p_2/p_1} = \sqrt{9/1} = 3$$

where p_2 и p_1 — pressure at the outlet and inlet of the compressor, respectively.

6.2. Air temperature at the compressor outlet

$$T_2 = T_1\,\varepsilon_1^{\frac{n-1}{n}} = 293 * 3^{\frac{1,25-1}{1,25}}\ 365\ К,$$

where T_1= 293 К (20°С) — atmospheric air temperature;

n = 1,20+1,25 — an indicator of the polytropy of air compression in the compressor.

6.3. Specific amount of heat removed by the water jacket of the compressor cylinder:

$$q_\tau = \frac{k-n}{n-1} C_V (T_2 - T_1) = \frac{1,4-1,25}{1,25-1} 0.721(365\text{-}293) = 31,1\ кДж/кг,$$

where k =1,4 — adiabatic air compression index; Cv - 0.721 kJ/(kgK) — isochoric heat capacity of air.

6.4. Specific heat removed in the intermediate and end coolers of the pneumatic installation:

$$q_{vx} = C_n(T_H - T_K) = 1{,}005(365\text{-}293) = 72{,}4 \text{ кДж/кг},$$

Where C_p = 1,005 кДж/(кг-К) — isobaric heat capacity of air; $T_н$ и $T_к$, — the air temperature, respectively, at the beginning and end of the movement path in the coolers, К ($T_н = T_2$ и $T_к = T_1$).

6.5. The total specific amount of heat discharged in the compressor unit:

$$q_\tau = Z_{c\tau} q_\tau + Z_x q_{c\tau} = 2\text{-}31{,}1+2\text{-}97{,}5 = 207 \text{ кДж/кг},$$

where г<гг и гх — the number of stages and the number of coolers in one compressor unit ($z_{ст}$ = 2 и $Z_х$. = 2).

6.6. The total amount of heat removed by the cooling system of the compressor unit per unit of time:

$$E_\tau = 60\rho_{вс} q_{тк} Q_к = 60*1{,}2*207*110 = 1{,}64*10^6 \text{ кДж/ч},$$

where $p_{вс}$ =1,2 кг/м^3 — air density under normal atmospheric conditions; Qk = 110 m3/min — compressor capacity, м^3/мин.

6.7. Estimated cooling water consumption per compressor unit

$$Q_{ов} = \frac{ET}{\rho_в C_в (t_в 2 - t_в 1)} = \frac{1{,}64 - 10^6}{1000 * 4{,}2(40-20)} = 19{,}5 \text{ м/ч},$$

where rv = 1000 kg/m3 is the density of cooling water; Cв = 4.2 kJ / (kgK) is the heat capacity of cooling water; tv1 and tv2 are the temperature of cooling water, respectively, at the inlet and outlet of the cooling system.

6.8. Total water consumption by the cooling system

$$Q_p = Z_{раб} Q_{ов} = 1000\text{-}4{,}2(40\text{-}20)$$

6.9. Required area of splash pools

$$П_{бб} = (0{,}8 / 1{,}3)\ Q_р = (0{,}8 / 1{,}3)\ 97{,}5 = 78 /27 \text{ м}^2.$$

BRANCH OF THE FEDERAL STATE AUTONOMOUS EDUCATIONAL INSTITUTION OF HIGHER EDUCATION "National Research Technological University "MISIS" in Almalyk

GLOSSARY

by discipline: "STATIONARY MACHINES"

Almalyk 2022

Glossary

The physical atmosphere is the average atmospheric air pressure at sea level at , equal to 10333 kg/m2 and corresponding to the height of the mercury column of 760 mm or the height of the water column of 10.33 m .

The technical atmosphere is a pressure equal to 10,000 (or 1) and corresponding to the height of the mercury column of 736 mm or the height of the water column of 10 m.

Productivity is the supply or flow rate, i.e. the amount of fluid transported by the turbomachine per unit of time, measured in volume (m3/sec, m3/min, m3/hour) or weight (kg/sec, kg/min, kg/hour) units;

Head – pressure or pressure generated by a turbomachine. The pressure measurement units are the heights of columns of liquid (water, mercury, alcohol, etc.) or atmosphere: physical and technical.

The slider is used in crank mechanisms of double-acting compressors. It is connected tightly to the piston rod, and movably with the connecting rod using a finger.

The pumping unit in the mine is necessary for pumping water to the surface. There is a local drainage system for pumping water from a site (or a group of sites) to the level of the mine's trunk yard and a main drainage system for issuing water from the level of the trunk yard to the surface.

A pneumatic installation is necessary to obtain compressed air used in the mine for the operation of jackhammers and drill hammers, pneumatic motors, winches, district pumps, etc.

Pistons of single-stage compressors of simple action are made hollow, in the form of a cup, in the inner part of which a connecting rod pin is mounted, fixed in the holes of the piston wall. In double-acting compressors, disc pistons are used. For better sealing of the piston in the cylinder, spring rings made of gray cast iron are used, placed in recesses along the circumference of the piston.

The compressor station operates in three shifts, coal works are carried out in two shifts, the mine operates on a continuous working week.

BRANCH OF THE FEDERAL STATE AUTONOMOUS EDUCATIONAL INSTITUTION OF HIGHER EDUCATION
"National Research Technological University "MISIS" in Almalyk

Methodological instructions for the implementation of the course project

by discipline:
"STATIONARY MACHINES"

GENERAL REQUIREMENTS FOR THE COURSE PROJECT

The course project includes two parts: graphic and computational and explanatory. The graphic part of the project consists of two demonstration sheets - drawings in A1 or A2 format, made in pencil or using computer graphics, in compliance with the established GOST standards for rational filling of sheets and each sheet must have an angular stamp.

The first sheet contains: a technological layout of the equipment accepted by the student in the project, indicating technical data, as well as general types of equipment.

On the second sheet, referred to as the special part of the project, there are graphs of parameter dependencies explaining the devices, the principle of their operation, functional and correlation dependencies.

The settlement and explanatory note, in the volume of 8,000-12,000 words, should be written in ink or paste or typed on a computer on standard writing paper and reset; should differ in brevity and clarity of presentation and divided into sections and subsections, according to the requirements for the design of text documents.

All decisions and calculations made should be illustrated with diagrams and diagrams, the designations on which should be explained in the text of the explanatory note.

Calculation formulas are written in a letter image, and then in a digital one. The result is specified with dimension. All explanations and notations to the formulas are given before determining the numerical value of the result.

The course project is drawn up in accordance with the requirements of the ESCD.

The explanatory note should have the following content and sequence of presentation:

title page (Appendix A);

assignment (on the form issued by the teacher);

CONTENTS (on a separate sheet with the pages indicated);

introduction;

1 MAIN PART;

2 SPECIAL PART;

conclusion;

list of literature;

Applications (if any).

During the design process, the student receives advice from the teacher-project manager on calculations, design of an explanatory note and drawing up drawings. Completed projects are submitted to the department for review a week before the defense. Before the defense, the projects are returned to the students to familiarize themselves with the comments of the supervisors and make corrections on them.

For the defense, the student prepares a (5-10 minute) report in which it is necessary to say about the specified mining conditions, the calculated complex of equipment, the content of drawings, the capacities developed by the engines of machines, their expected productivity. A special place in the report should be given to the special part of the project (50% of the report).

On the day of defense (after the report), the members of the admissions committee ask the student questions about the principle of operation and the device of the designed machines and the modes of their operation provided. The student answers questions on structural, kinematic, hydraulic schemes of machines and work schedules given in the drawings of the project.

When evaluating the project, the quality of the drawings and explanatory notes, the content of the report and the completeness of the answers to the questions of the commission members are taken into account.

In case of unsatisfactory answers to questions, the project can be defended again after a week of preparation.

1. GENERAL PROVISIONS FOR THE DESIGN AND DETERMINATION OF COMPRESSED AIR FLOW.

When designing pneumatic installations, the general principle of supplying a mining enterprise with compressed air is determined, followed by the operational calculation of the main and auxiliary equipment. The initial material for the design of the operational calculation of the equipment is: a mining plan or a plan of an industrial site of a mining enterprise to be provided with compressed air. The number and technical characteristics of compressed air consumers. Location of compressed air consumers for the estimated period of operation of the mining enterprise. Schedules of work of consumers by shifts.

The listed documents are the basis for drawing up a design scheme for the supply of compressed air to the enterprise, on which, in compliance with certain scale ratios, the location of compressor stations, air distribution units and compressed air consumers is graphically displayed, as well as the general scheme of branching pipelines of the pneumatic network, an example of which is shown in Fig. 1. In the diagram, the numbers 1,2,3 ...,11 indicate the branching points of pipelines and the distance lij between them is indicated.

The compilation of the calculation scheme is preceded by the choice of the general principle of supplying the mining enterprise with compressed air. The supply can be based on the use of several precinct (mobile or semi-stationary) compressor stations or one central (stationary) pneumatic installation. The choice of the principle of compressed air supply is made on the basis of a technical and economic comparison of options when forming an ingenious plan of a mining enterprise.

The focus on the use of precinct compressor stations contributes to a significant reduction in the length of the pneumatic network and, as a result, a reduction in leaks and pressure losses. However, it should be borne in mind that such a principle of compressed air supply is unacceptable for mines that are dangerous for gas and dust. In addition, with the increase in total air consumption, the economic efficiency of centralized compressed air supply, as a rule, increases.

The use of central pneumatic installations is considered economically justified when the length of air ducts to consumers does not exceed 5-6 km. The cost-effective distance of compressed air supply to consumers increases when large diameter pipes (600-1000 mm) are used on main air pipelines, as well as when switching to new materials for pipelines (fiberglass, polyethylene, enameling, etc.). In domestic mining practice, there is a positive experience of centralized supply of compressed air to several mines from one district compressor station.

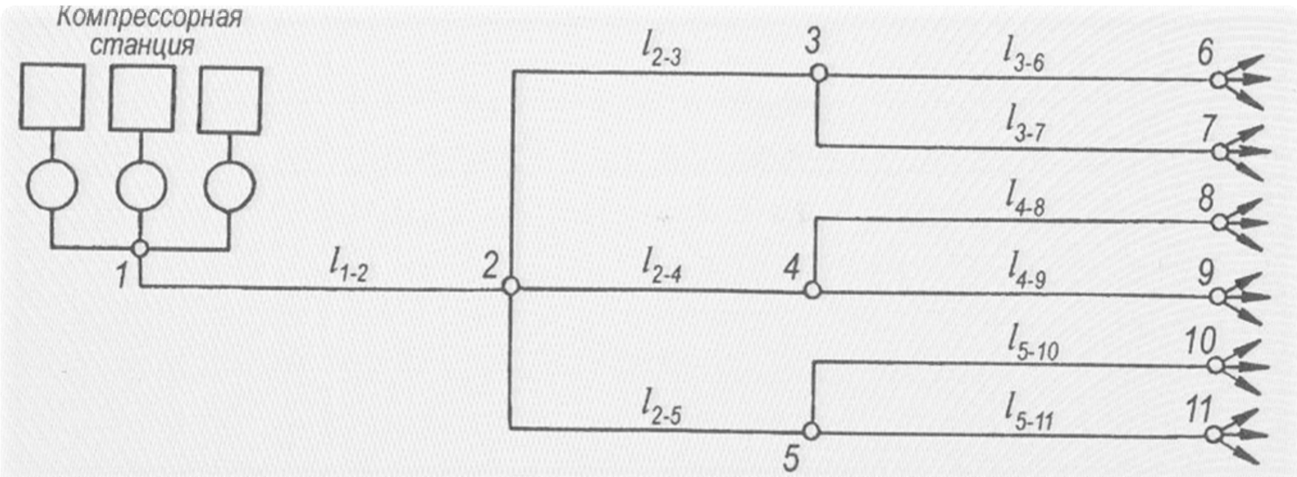


Fig. 1. Approximate scheme of branching of pneumatic network pipelines

In some cases, the principle of combined compressed air supply is used, when the overwhelming majority of consumers with a working pressure of 5-7 bar are serviced by a central pneumatic installation, and individual consumers whose working pressure in mining conditions can reach 25-80 bar are provided with compressed air from precinct or individual mobile compressor stations.

When designing pneumatic installations, it becomes necessary to determine not only the total air consumption of the enterprise, but also to calculate the so-called travel costs of compressed air in sections of the pneumatic network. According to the calculated total air consumption, taking into account the leaks of compressed air, the required performance of the compressor station is determined and the selection of compressors that provide the specified performance is made. Knowledge of the travel costs of compressed air in the sections between the branching points is necessary to determine the rational diameter of the pipes and the subsequent calculation of pressure losses in the pneumatic network.

The estimated total air consumption by consumers (air consumption of the enterprise) is determined as follows:

$$Q_п = \mu K_0 \sum_{1}^{m} n_i q_i k_{3i} \Psi_i \qquad (1)$$

where μ is the reserve coefficient for unaccounted compressed air consumers; K0 is the weighted average coefficient of simultaneous operation of consumers; qi is the nominal consumption of compressed air by one type i consumer; ni is the number of consumers of the same name of the specified type; ψi is the coefficient that takes into account the increase in air consumption during operation of machines and mechanisms due to their wear; k3i is the load factor that takes into account the change in the average air flow compared to the nominal one due to the difference in the actual load from the nominal one, as well as as a result of regulating the operating mode of machines and mechanisms; m is the number of groups of the same type of consumers.

The numerical values of the parameters qi and ni, as well as the coefficients ψi and k3i are determined on the basis of a general list of consumers with their technical characteristics, which is one of the source documents for the design of a pneumatic installation. In other cases, they are determined by the corresponding reference books.

The weighted average coefficient of simultaneity K0 of the operation of air consumers is usually determined graphically (Fig.2) depending on the total number of PS consumers, as well as the numerical value of the weighted average coefficient of Kv inclusion of working machines and mechanisms.

Total number of compressed air consumers

$$n_c=\sum n_i=n_1+n_2+\ldots+n_m \quad (2)$$

The weighted average coefficient of inclusion of compressed air consumers is the ratio of the average total air consumption to its maximum value and is determined by the following formula:

$$K_B=\frac{\sum_1^m n_i q_i \psi_i k_{3i} k_{Bi}}{\sum_1^m n_i q_i \psi_i k_{3i}} \quad (3)$$

where kvi are the coefficients of inclusion of machines and mechanisms, numerically equal to the ratio of their net working time to the total duration of the working shift; determined on the basis of the above-mentioned list of consumers or reference books.

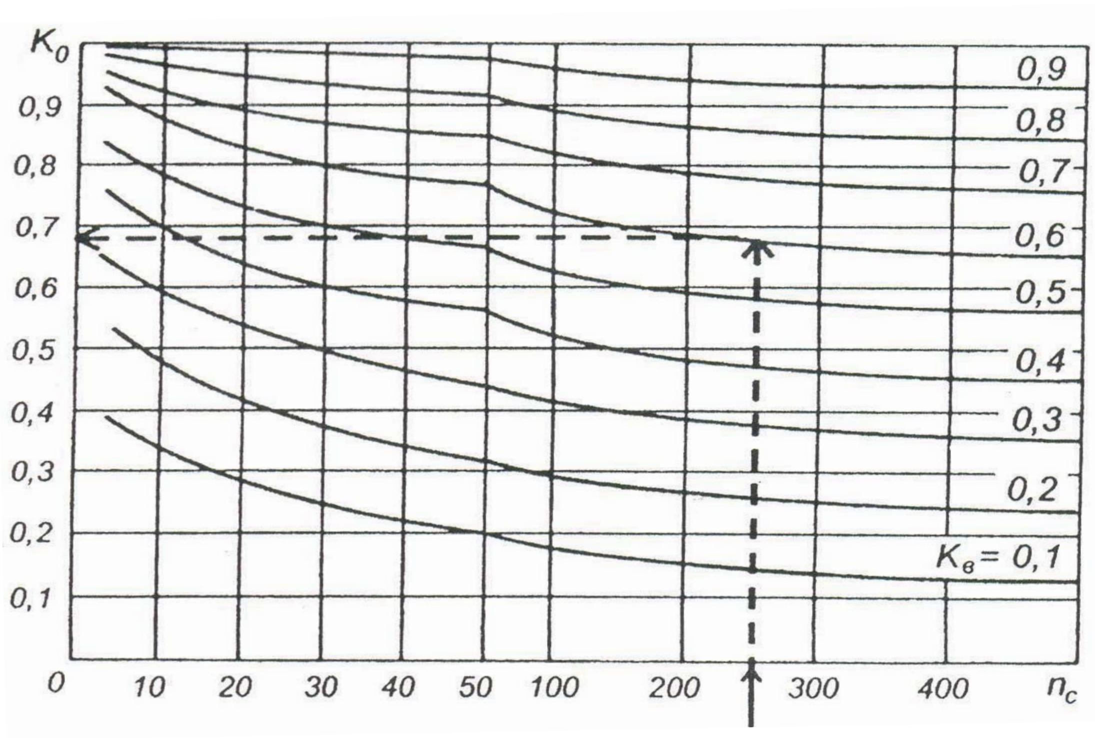


Fig.2 Graphical determination of the weighted average coefficient of simultaneous operation of compressed air consumers.

The method of graphical determination of K0 by the given Σni and Kv is shown in Figure 2 with dotted lines and arrows. If the calculated Kv value does not coincide with the numerical values indicated on the corresponding curves, then the interpolation method can be used in this case.

The design capacity of the compressor station must take into account possible leaks of compressed air in the pneumatic network and is therefore determined as follows:

$$Q_{кс}= Q_{п} + Q_{ус} + Q_{уп} \quad (4)$$

where Qc and Qyn are the estimated costs of compressed air to compensate for leaks, respectively, in the pneumatic network and in air consumption points.

The amount of leaks due to leakage of the pneumatic network piping system is determined by the formula:

$$Q_{ус}= \frac{p+1}{6} a \sum_{1}^{m} L_{i-j} \qquad (5)$$

where p is the calculated pressure of compressed air at the points of consumption, bar; a is the permissible value of specific losses of compressed air, attributed to the unit length of the pipeline, at a calculated pressure of consumers of 5 bar; Σli–j is the total length of pipelines of the pneumatic network, determined by the design scheme of the pneumatic installation; li-j is the length of individual sections the pneumatic network between the points of branches i and j (see Fig.1).

The amount of leaks in the points of air consumption due to the leakiness of the connecting joints and connecting hoses

$$Q_{уп} = \frac{p+1}{6} b \sum_{1}^{m} n_i \qquad (6)$$

where b is the standard value of leaks in the connecting elements of compressed air consumption points at a pressure of 5 bar.

Considering that in the conditions of mining, the nominal pressure of the main consumers of compressed air varies insignificantly relative to the average value of 5 bar, at the stage of operational calculation it is possible to take p = 5 bar and then formulas (5) and (6) are converted:

$$Q_{ус}= \frac{p+1}{6} a \sum_{1}^{m} L_{i-j} \quad \text{и} \quad Q_{уп} = \frac{p+1}{6} b \sum_{1}^{m} n_i \qquad (7)$$

The value of normal leaks of compressed air at p = 5 bar is usually taken as follows: a = 0.003 m3 / (min * m) – for 1 m of the total length of the air ducts; b = 0.4 m3/min per consumer (machine or mechanism).

The travel expenses of compressed air are calculated starting from the peripheral (end) sections of the pneumatic network adjacent directly to the points of air consumption (branch points 6-11 in Fig.1). For these sections, the travel expense of compressed air is determined by the formula

$$Q_{i-j}=\sum_{1}^{m} n_i q_i k_{3i} \psi_i k_{Bi} + a L_{i-j} + b \sum_{1}^{m} n_i \qquad (8)$$

where qi, ni and m have the same value as in formulas (1) and (3), but in relation to the considered end section of the pneumatic network and those compressed air consumers that this section of the network serves.

For sections of main air pipelines bounded by two branching points i and j, compressed air consumption at the final (jth) point is defined as the sum of expenses on the end sections connected to it by consumers, and the estimated travel expense is determined as follows:

$$Q_{i\text{-}j}=\sum(Q_{i-j})+a\,L_{i-j} \qquad (9)$$

where li-j is the length of the considered section of the main pipelines of the pneumatic network, and the first term displays the total air flow in the peripheral pipelines connected to it.

Last of all, the travel flow is calculated on the section of the main pipeline directly adjacent to the compressor station. The calculated travel air flow on this section of the pneumatic network will differ slightly from the calculated performance of the compressor station, since the formulas used to determine them differ.

2. DETERMINATION OF PRESSURE LOSSES IN THE PNEUMATIC NETWORK

The general principle of the operational calculation of the pneumatic network is the same as that for external networks of pumping and fan installations. The calculation and selection of pipelines is based on the preliminary setting of the optimal air velocity through the pipes, and the subsequent determination of pressure losses is completed by constructing the pressure characteristics of the pneumatic network and graphical analysis of the operating mode of the compressor station.

The optimal ratio between energy losses in the pneumatic network and capital costs for the construction of air pipelines corresponds to the average speeds of compressed air in the range of 4-8 m/s. Moreover, it is recommended to take lower speeds in the initial sections of the air pipelines immediately after the compressor station, gradually increasing them in the sections of branches and remote sections of the main pipeline.

The diameter of pipes in sections of the pneumatic network at a pre-selected average speed vc = 4 ÷8 m /s is determined by the following formula:

$D_{i\text{-}j}$=**Ошибка! Источник ссылки не найден.** (10)

where $Q_{i\text{-}j}$ – air consumption in the section of the pneumatic network between the points of branching i and j, m3/min.

The exact calculation of the pressure characteristics of the pneumatic network is complicated, is performed only in exceptional cases and requires a number of data that can be obtained only during the operation of the pneumatic installation. The complexity of the calculation is explained by the branching of the pneumatic network, a large number of simultaneously operating consumers, taking into account the pressure characteristics of which is necessary in the calculation, as well as changes in the speed of air movement due to the way its pressure changes and fluctuations in air consumption by consumers. Pressure changes in pneumatic networks, in addition to energy losses due to friction, are significantly affected by leaks of compressed air and heat exchange with the environment.

The calculation of pressure losses in air ducts during the operational calculation of a pneumatic installation is usually simplified by performing it only for the mode of calculated air supply along the longest section of the air pipeline to the most remote point of air consumption or to the point of air consumption with the highest operating pressure.

The pressure losses Δp due to the aerodynamic resistance of the pneumatic network are calculated using the following formula:

Δp =**Ошибка! Источник ссылки не найден.** (11)

where Δp i-j are pressure losses due to the aerodynamic resistance of individual sections of the pneumatic network, bar; λc is the coefficient of linear aerodynamic resistances of rectilinear sections of air ducts; rs is the average density of compressed air, kg/m3; vc is the average air velocity in sections of the pneumatic network, m/s.

To determine the coefficient of aerodynamic drag, the formula of A.F.Shevelev is usually used:

Ошибка! Источник ссылки не найден. (12)

where D i-j is the diameter of the pipeline on the network section, m.

The average density of compressed air in individual sections of pipelines is determined in accordance with the Clapeyron equation as follows:

Ошибка! Источник ссылки не найден.

(13)

where p_0 is the density of air under normal atmospheric conditions, and rs and Ts are the average values, respectively, of the pressure and absolute temperature of compressed air in the sections of the pneumatic network. The multiplier 1.15 in formula (11) takes into account additional local pressure losses in pipe fittings and other devices installed in the pneumatic network.

The average pressure on the pipeline sections is determined by the following formula:

Ошибка! Источник ссылки не найден.Δ**Ошибка! Источник ссылки не найден.**
(14)

where p_i и p_j – the pressure is respectively at the beginning and end of the network section.

If, in formula (11), the average velocity of compressed air is expressed in terms of the flow rate at the site under normal atmospheric conditions in accordance with formula (10), take into account expressions (12), (13) and (14), and also assume that the air at the site of the pneumatic network is cooled to normal temperature, then we get the following calculation formulas for determining pressure losses:

Δ**Ошибка! Источник ссылки не найден.**

or

Δ**Ошибка! Источник ссылки не найден.** (15)

where the constant B is a generalized aerodynamic parameter of the section of the air pipeline and is determined by the formula

B=0,42·**Ошибка! Источник ссылки не найден.** (16)

The calculation of pressure losses according to formulas (15) and (16) can be performed at a given pressure at the most remote consumer, gradually moving from the end sections to the main pipeline at the compressor station. The final pressure in each subsequent section will increase by the amount of pressure loss in the previous sections.

If the pressure developed by the compressor station is set, then the calculation is carried out in reverse order – from the compressor to the consumer. In this case, the pressure at the beginning of each subsequent section of the pipeline is reduced by the amount of pressure loss in the previous sections.

The operational calculation of the pneumatic network is greatly simplified when using a nomogram for this purpose (Fig.3), based on the formulas discussed above. The calculation is made according to the preset travel costs of compressed air in the sections, the permissible maximum pressure loss in the air ducts and the average pressure, which is usually assumed to be the same for all sections of the pneumatic network and is determined with sufficient accuracy for practical purposes as follows:

Ошибка! Источник ссылки не найден. or **Ошибка! Источник ссылки не найден.** (17)

where p and rp.max are, respectively, the pressure at the outlet of the compressor station and the maximum operating pressure of consumers; Δp max = 1.5–2 bar is the preset maximum pressure loss in the air ducts of the pneumatic network.

The calculated specific pressure loss attributed to the unit length of the air ducts is determined by the formula

Δ**Ошибка! Источник ссылки не найден.** (18)

where $\Sigma l_{i\text{-}j}$ – the length of the air ducts from the compressor station to the most remote point of air consumption.

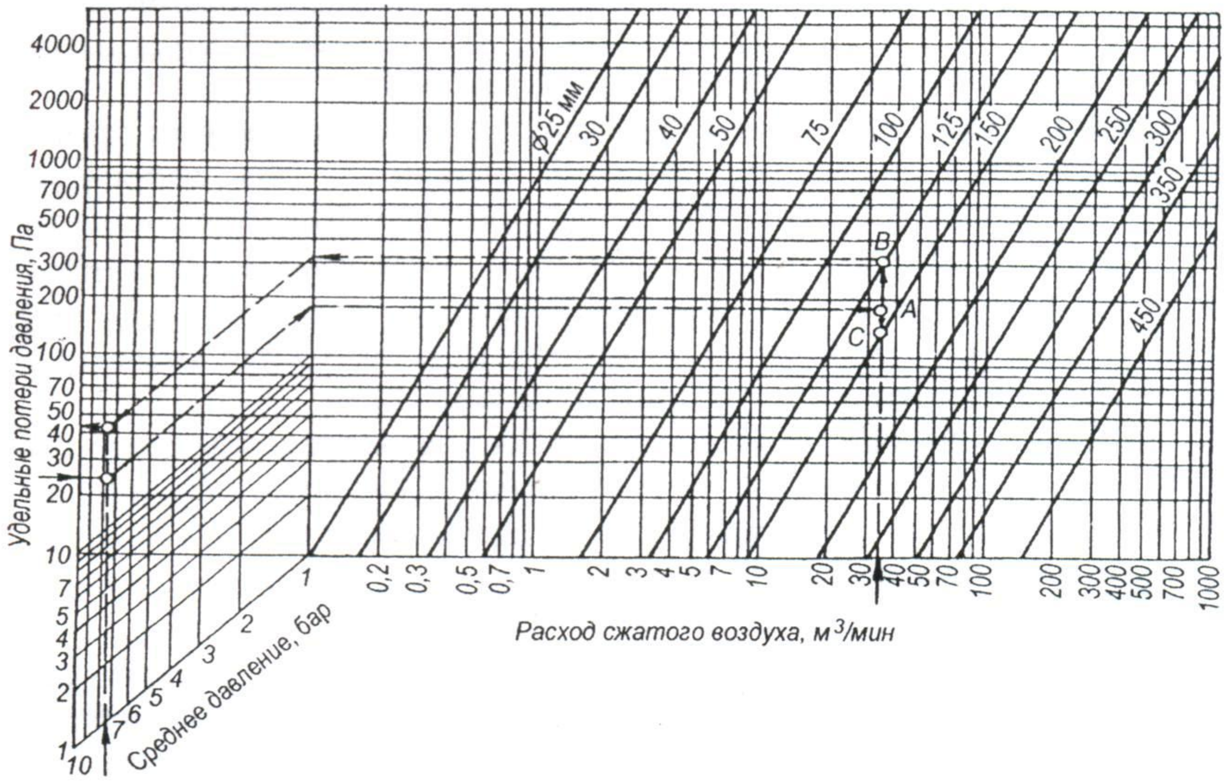
During operational calculations, the standard specific pressure loss Δrun = 0.0002÷0.0004 bar / m (20÷40 Pa / m) is usually set, which corresponds to the optimal value of the air velocity calculated from the air flow rate under normal atmospheric conditions. Calculation of air ducts based on the nomogram shown in Fig.3, in this case, is performed in the following order. The formulas (17) determine the average pressure in the pneumatic network. According to the given Δrunes = 20÷40 Pa / m, as well as pc and Qi-j, the base point A is found on the nomogram, as shown by the dotted lines in Fig.3. On the flow vertical Qi-j, the intersections with the pipe diameter lines (points B and C) closest to point A are found. For the example shown in Fig.3, this means that it is advisable to pump the specified flow rate Qi-j of compressed air through pipes with a diameter of 125 and 150 mm. By choosing one of two appropriate diameters (for example, at point B), in the opposite direction, as shown by the dotted lines, the actual specific pressure loss of ore is determined, and then the total pressure loss on the pipeline section, bearing in mind that

Δ**Ошибка! Источник ссылки не найден.** (19)

The pressure of the pipeline at its initial and final points, and in the following columns the accepted and calculated values are given, respectively, Qi-j , pc , Δrun , D i-j , Δore , li-j and Δp i-j. After filling in the table, the most remote point of compressed air consumption is determined, as well as the point with the maximum working pressure at the consumer. For these two air supply options , the required pressure of the compressor station is calculated according to the formula

p=**Ошибка! Источник ссылки не найден.** (20)

where $\Delta\Sigma p_{i\text{-}j}$ - total pressure loss in the pipeline from the compressor station to the selected consumer.



The largest of the two calculated pressures obtained is taken as the calculated pressure of the compressor station, according to which the main equipment of the pneumatic installation is selected - compressors.

The calculation of the air ducts of the pneumatic network can be carried out according to an even more simplified scheme – without correcting the specific pressure losses after selecting the pipe diameter. With this calculation, the graph of ores in the resulting table is excluded, and the value of the normative specific pressure losses of runes for each section of the pipeline of the pneumatic network is determined by the corresponding graphs in Fig.4.

The graphs shown in Fig.4 are the dependences of the optimal values of the normative specific pressure losses Δrunes and pipe diameters D i-j of air ducts on the travel air flow Qi-j under normal atmospheric conditions. Moreover, the graphs for determining the runes are represented by several curves corresponding to different numerical values of the average pressure of the pc in the pneumatic network. The optimal values of the parameters Δrune and D i-j are determined on the basis of a technical and economic comparison of options and correspond to the minimum costs for the construction and operation of air pipelines.

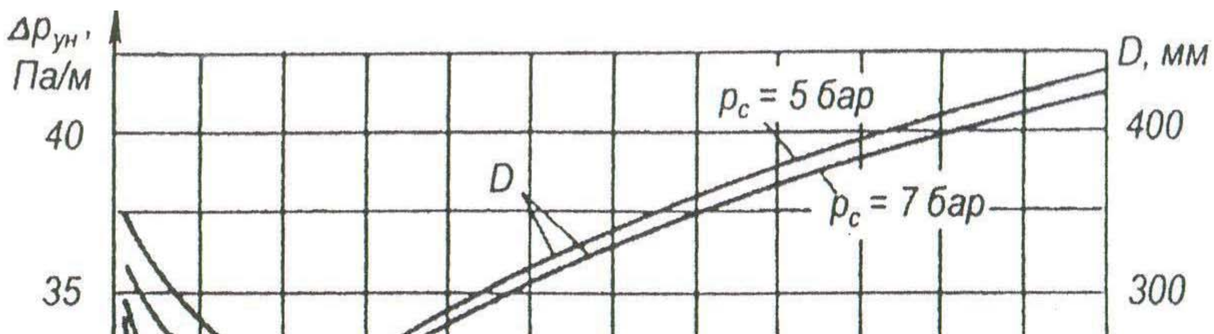


Fig.4. Graphs for determining the standard specific pressure losses in air ducts and their diameter

After determining the Δrune and D i-j according to the specified travel air flow rate Qi-j and the average pressure pc, the final choice of pipe diameter is made, rounding the calculated values to the nearest standard ones. The calculation of pressure losses in the section of the air pipeline is performed according to the formula (19), assuming Δore = Δ run, and the design pressure of the compressor station is determined as follows:

p=**Ошибка! Источник ссылки не найден.** (21)

where Δp_3 = 0,3 бар – a pressure reserve that takes into account the above-mentioned refusal to correct specific pressure losses on sections of air ducts after rounding their diameters. In other words, Δp3 takes into account the difference between the actual specific pressure losses from the calculated optimal ones with the finally selected diameters of the air ducts of the pneumatic network.

3. CALCULATION AND SELECTION OF THE MAIN PNEUMATIC INSTALLATION EQUIPMENT

The operational calculation and selection of the main equipment of a stationary pneumatic installation is preceded by the preparation of the following source documents: schemes for branching pipelines of the pneumatic network from the compressor station to consumers, indicating the distances between the branching points; a list of compressed air consumers presented in the form of a table, indicating the nominal pressure and air flow by each of them, the number of the same type, as well as numerical values of loading coefficients, switching on and increasing air flow due to wear of machines and mechanisms; tables of distribution of compressed air consumers by air consumption points. The operational calculation of the equipment contains six calculation blocks performed in the sequence below.

3.1. Determination of the required performance of the compressor station. The algorithm for performing this calculation block is as follows:

• the total number of compressed air consumers is determined by the formula (2);

• according to the formula (3), the weighted average coefficient of inclusion of consumers is calculated;

• graphically, as shown in Fig.2, the coefficient of simultaneous operation of consumers is determined;

• taking the reserve coefficient for unaccounted compressed air consumers $\mu = 1.05 \div 1.15$, the calculated total air consumption by consumers is determined by the formula (1);

• in accordance with the data of the branching scheme, the total length of the pipelines of the pneumatic network is determined as the sum of the lengths of individual sections;

• according to the formula (5), the amount of leaks due to leakage of the pneumatic network piping system is determined;

• according to the formula (6), the amount of leaks in air consumption points is calculated;

• the required capacity of the compressor station is determined by the formula (4) as the sum of the estimated air consumption by consumers and possible leaks due to leakage of the pneumatic network and connecting elements of the points of consumption.

3.2. Calculation of compressed air travel expenses. The calculation begins with the peripheral sections of the pneumatic network, for which the formula (8) and the data of the distribution table of air consumers by consumption points are used. Then, according to the formula (9), travel expenses on the main sections of pipelines are calculated, sequentially moving from remote sections to the section adjacent to the compressor station.

The results of the calculation of travel expenses are drawn up in the form of a table, in the first column of which the designation of the section is given in the form of a numerical index i – j, where i and j are the numbers of branch points, respectively, at the beginning and end of the pipeline section. The following columns of the table display: compressed air consumption at the end point of the network section or a list of consumers connected to it (for peripheral sections); the length of the pipeline section; the number of consumers connected at the end point (for peripheral sections); the calculated value of the travel flow on the section of the pneumatic network.

3.3. Calculation of pressure losses in the pneumatic network. The sequence of execution of this calculation block is as follows:

• according to the formula (17), the average pressure in the air ducts of the pneumatic network is determined;

• the standard specific pressure loss in the air ducts is set Δrun = 0.0002÷0.0004 bar/m;

• according to the specified travel costs Qi-j, as well as the average pressure of the rs and the standard value of the runes, the diameters of the pipes D i-j and the actual specific losses of ores in the sections of the pneumatic network are determined graphically, as shown in Fig.3.;

• according to the formula (19), the total pressure loss in the sections of the pneumatic network is determined.

The results of calculations of pressure losses in sections of the pneumatic network are drawn up in the form of a table, the first column of which is intended for the numerical index of the section. The remaining columns of the table show the travel flow rate at the site, the average pressure of compressed air, the accepted value of the normative specific pressure loss, the result of determining the diameter of the pipes, the actual value of the specific pressure loss and the total pressure loss at the site of the pneumatic network.

3.4. Design pressure of the compressor station. According to the scheme of branching pipelines of the pneumatic network, the point of air consumption most remote from the compressor station and the point with the maximum operating pressure of the consumer are determined. The pressure calculation is performed for two options:

a) according to the maximum operating pressure of consumers

p=**Ошибка! Источник ссылки не найден.**$\Delta\Sigma p_{i-j}$ (22)

where $\Delta\Sigma p_{i-j}$ - total pressure losses in the air ducts from the compressor station to the consumer with the maximum operating pressure rp.max ;

b) along the longest section of the pneumatic network (the most remote point of air consumption)

p=**Ошибка! Источник ссылки не найден.**$\Delta\Sigma p_{i-j(max)}$ (23)

where $\Sigma\Delta p_{i-j(max)}$ - the total pressure loss in the air ducts to the most remote compressed air consumer, and the rp is its operating pressure.

The largest of the two calculated pressures is taken as the calculated one for the selection of compressors.

3.5. Selection of compressor station equipment. The main parameters for the selection of equipment are the design capacity Qcs and the design pressure p of the compressor station.

3.5.1. The choice of compressor type for stationary compressor stations is usually made on the basis of a technical and economic comparison of options. Special studies have established that with the design capacity of the compressor station less than 200 m3/min, the most appropriate reciprocating compressors of the same type are horizontal or rectangular type with a nominal flow from 10 to 50 m3/min.

At the design capacity of the compressor station of 200-500 m3/min, horizontal reciprocating compressors with a nominal flow of 100-125 m3/min should be used. With a station capacity of 500-1000 m3/min, centrifugal compressors with a supply of 115 and 250 m3/min are advisable. When the calculated supply of the station is more than 1000 m3/min, it is necessary to focus on centrifugal compressors with a nominal air supply of 500 m3/min.

It is recommended to use centrifugal compressors in a set with a certain number (up to 25% of the design capacity of the compressor station) of reciprocating compressors, which are put into operation during periods of cycles of reduced air consumption. In addition, reciprocating compressors are also necessary to supply a certain amount of oil to the external pneumatic network in order to eliminate corrosion of the internal surfaces of pipes and auxiliary equipment.

3.5.2. The required number of working compressors for equipping the station, if the units are of the same type, is determined as follows:

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where Q – nominal capacity of the selected compressor.

3.5.3. The number of backup compressors depends on their type. If reciprocating compressors are selected as workers, but with their number from 1 to 3 in reserve, it is enough to have one compressor unit of the same type as the workers. With the number of working piston units. Turbochargers with 1-2 workers are reserved by one unit, and with 3 or more working compressors – by two units.

3.5.4. Total number of compressors at the compressor station

Ошибка! Источник ссылки не найден. (25)

where **Ошибка! Источник ссылки не найден.** - number of backup compressors.

3.5.5. Type and number of air collectors. Each compressor is equipped with an individual air collector (. With a larger number of them, one common air collector is provided for each pair of compressors (**Ошибка! Источник ссылки не найден.**.

3.5.6.. The selection of filters for cleaning the intake atmospheric air is made according to the calculated performance of the compressor station. At the same time, preference is given to self-cleaning mesh filters of the brand Кт.

3.6. Calculation of the water supply system.

The sequence of calculation of the water supply system of the compressor station with cooling water can be taken as follows:

- the degree of air compression in the compressor stage is determined

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where p_2 и p_1 – pressure, respectively, in the discharge and suction pipes of the compressor.

- the air temperature at the compressor outlet is calculated by the formula (27)

Ошибка! Источник ссылки не найден. ; (27)

where T_1**=273 К** – the temperature of the intake atmospheric air; n=1,20Ошибка! Источник ссылки не найден. 1,25 – the indicator of the polytrope of air compression in the compressor;

- total water consumption by the cooling system

$$Q_p = z_{раб}\, Q_{ов}\ ; \quad (28)$$

- required area of splash pools

$П_{бб}$ =(0,8**Ошибка! Источник ссылки не найден.**1,3)Q_p ; (29)

The water consumption for cooling compressor units can be set according to the manufacturers' data or approximately according to the average specific norms assigned to 1 m3 of air: for two-stage reciprocating compressors with an excess pressure of 8-9 bar, the specific water consumption is 4.5–5.5 l/m3; for single-stage reciprocating compressors – 1.5–2 l/m3, for turbochargers with an excess pressure of 7 - 8 bar - 5-6 l/ m3.

The operational calculation is completed by compiling a list of the main equipment of the pneumatic installation, indicating its technical characteristics.

4. EXAMPLE OF CALCULATION OF A STATIONARY PNEUMATIC SHAFT INSTALLATION.

Initial data for the calculation

1. The design scheme of the pneumatic installation is shown in Fig. 4.

2. The distance between the nodal points of the pneumatic network:

L_{1-2} = 520 м; L_{2-3} = 160 м; L_{2-4}= 420 м; L_{3-5} = 180 м; L_{3-6} = 240 м;

L_{4-9} = 640 м; L_{4-10} = 580 м; L_{5-8} = 340 м; L_{5-11} = 460 м; L_{5-12} = 420 м;

L_{6-7} = 440 м; L_{6-15} = 320 м; L_{6-16} = 400 м; L_{7-17} = 420 м; L_{7-18}=510 м;

L_{8-13} = 440 м; L_{8-14} = 460 м.

3. Technical characteristics of compressed air consumers are presented in Table 1.

4. Distribution of compressed air consumers by consumption points:

point 9— (1гп + 1ск + 6мо + 1бм + 1вм);

point 10 — (1гв + 1бс + 1вм + 1мл + 1нп);

point 11 — (1пм + 12мо + 2бм + 1вм + 1нп);

point 12 — (1ко + 2ск + 1вм + 2мл + 1нп);

point 13 — (1бс + 6мо + 1бм + 1вм + 2сл);

point 14 — (1ко + 2ск+ 1вм + 1мл + 1нп);

point 15 — (1гп + 1ск + 5мо + 1бм);

point 16 — (2гв + 6мо + 1бм + 2мл + 2нп);

point 17— (8мо + 2бм + 2вм + 2сл);

point 18 — (2пм + 10мо + 1бм + 1вм).

Note. The numbers before the letter designations of compressed air consumers indicate their number in the corresponding points.

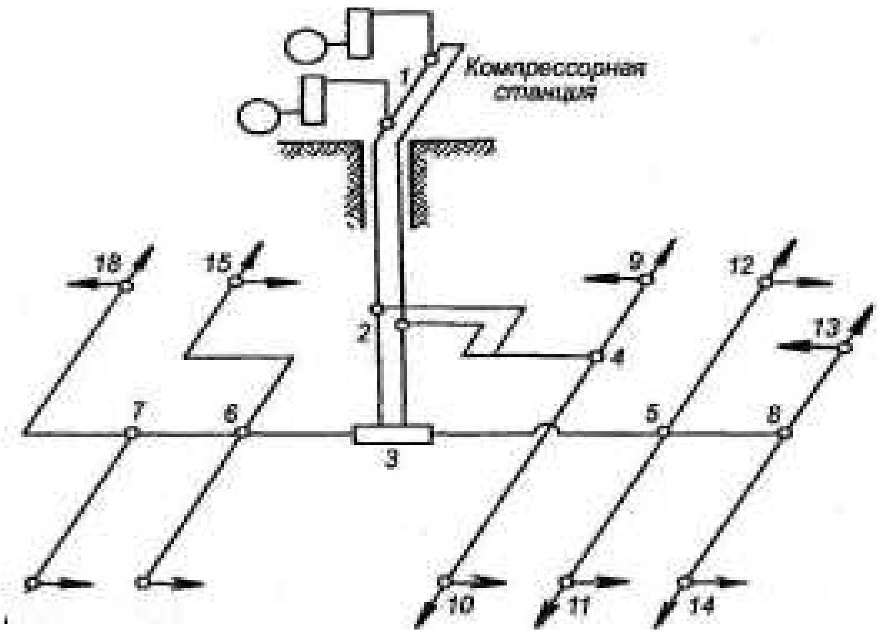

Fig. 4. Design scheme of the pneumatic installation.

Table 1.

Technical characteristics of compressed air consumers

Потребители сжатого воздуха (обозначение)	**Количество n_i**	**q_i м³/мин**	**p_{ni} бар**	**K_{3i}**	**Ψ_i**	**$k_{вi}$**
Очистные комбайны (ко)	2	48	4,5	1.0	1,10	1,0
Погрузочные машины (пм)	3	22	5,0	0,25	1,10	0,4
Грейферы проходческие (гп)	2	20	5,6	0,38	1,15	0,45
Маневровые лебедки (мл)	6	10	3,5	0,80	1,20	0,15
Скребковые конвейеры (ск)	6	16	3,5	0,80	1,10	1,0
Буросбоечные станки (бс)	2	16	4,5	1,0	1,20	0,5
Отбойные молотки (мо)	53	1.4	4,0	1,0	1,15	0,4
Бурильные молотки (бм)	9	3,5	5,0	1,0	1,15	0,65
Вентиляторы ВМП (вм)	9	5,0	4,0	0,7	1,0	1,0
Пневматические насосы (нп)	6	2,5	4,0	0,8	1,05	1,0
Скреперные лебедки (сл)	4	2,5	5,0	0,8	1,20	0,5
Гировозы (гв)	3	35	5,5	1,0	1,15	0,3

1. CALCULATION OF COMPRESSOR STATION PERFORMANCE

1.1. Total number of compressed air consumers

$$n_c=\sum n_i=n_1+n_2\ldots n_m= 2 + 3 + 2 + 6 + 6 + 2+53 + 9 + 9 + 6 + 4 + 3=105.$$

1.2. Weighted average coefficient of inclusion of consumers

$$K_B=\frac{\sum_1^m n_i q_i k_{3i}\psi_i k_{Bi}}{\sum_1^m n_i q_i k_{3i}\psi_i}=\frac{375,8}{617,7}=0.608$$

where n_i — the number of consumers of the same name of the i-th type; qi — the nominal consumption of compressed air by one consumer of the specified type; k3i — the load factor; i— the coefficient that takes into account the increase in air consumption due to wear of machines and mechanisms; kBI- the coefficients of inclusion of machines and mechanisms; m — the number of groups of the same type of consumers. Numerical values of parameters and coefficients are accepted in accordance with Table.1.

$\sum_1^m n_i q_i k_{3i}\psi_i k_{Bi}$ = 2 * 48 * 1,0 * 1,1 * 1,0 + 3 * 22 * 0,25 * 1,1 * 0,4 + 2 * 20 * 0,38 * 1,15 * 0,45 + 6 * 10 * 0,8 * 1,2 * 0,15 + 6 * 16 * 0,8 * 1,1 * 1,0 + 2 * 16 * 1,0 * 1,2 * 0,5 + 53 * 1,4 * 1,0 * 1,15 * 0,4 + 9 * 3,5 * 1,0 * 1,15 * 0,65 + 9 * 5,0 * 0,7 * 1,0 * 1,0 + 6 * 2,5 * 0,8 * 1,05 * 1,0 + 4 * 2,5 * 0,8 * 1,2 * 0,5 + 3 * 35 * 1,0 * 1,15 * 0,3 = 375,8 м^3/мин;

$\sum_1^m n_i q_i k_{3i}\psi_i$ =2 * 48 * 1,0 * 1,1 + 3 * 22 * 0,25 * 1,1 + 2 * 20 * 0,38 * 1,15 + 6 * 10 * 0,8 * 1,2 + 6 * 16 * 0,8 * 1,1 + 2 * 16 * 1,0 * 1,2 + 53 * 1,4 * 1,0 * 1,15 + 9 *3,5 * 1,0 * 1,15 + 9 * 5,0 * 0,7 * 1,0+6 * 2,5 * 0,8 * 1,05 + 4 * 2,5 * 0,8 * 1,2 + 3 * 35 * 1,0 * 1,15 = 617,7 м^3/мин.

1.3. The coefficient of simultaneity of the operation of consumers is determined graphically by the specified ts and Kv using the diagram in Fig 2, K_o = 0,72.

1.4. Estimated total air consumption by consumers

$Q_n=\mu K_0 \sum_1^m n_i q_i k_{3i}\psi_i$ =1,1·0,72·617,7 = 489,2 м^3/мин,

where μ = 1,05 - 1,15 — reserve ratio for unaccounted compressed air consumers in the mine and on the surface.

1.5. Total length of pneumatic network pipelines

$\sum L_{i-j}$ = L_{1-2}+L_{3-3}…= 520 + 160 + 420+180 + 240 + 640 + 580 + 340 + 460 + 420 + 440 + 320 + 400+420 + 510 + 440 + 460 = 6950м.

1.6. The amount of leaks due to leakage of the pneumatic network piping system

$Q_{ус}= \frac{p+1}{6} a \sum L_{i-j} = \frac{5+1}{6} 0{,}003 \cdot 6950 = 20{,}85$ м^3/мин,

where p = 5 бар — the calculated average pressure of compressed air at the points of consumption; a = 0.003 m3 / (min * m) is the permissible value of specific losses of compressed air at the calculated pressure of consumers of 5 bar, attributed to the unit length of the pipeline.

1.7. The amount of leaks in air consumption points

$Q_{уп}= \frac{p+1}{6} b \sum n_i = \frac{5+1}{6} 0{,}4 \cdot 105 = 42{,}0$ м^3/мин,

where b= 0,4 м^3/мин — the standard value of leaks in the connecting elements at compressed air pressure 5 бар.

1.8. Estimated capacity of the compressor station

$$Q_{кс} + Q_{п} + Q_{ус} + Q_{уп} = 489{,}2+20{,}8+42{,}0 = 552{,}0 \text{ м}^3\text{/мин.}$$

2. CALCULATION OF COMPRESSED AIR TRAVEL EXPENSES

2.1. The calculation of the travel flow of compressed air in the peripheral sections of the pneumatic network is carried out according to the formula

$$Q_{i-j} = \sum_{1}^{m} n_i q_i k_{3i} \psi_i k_{Bi} + a L_{i-j} + b \sum n_i$$

2.2. The travel flow rate of compressed air in the sections of the main pipeline is determined as follows:

$$Q_{i-j} = \sum (Q_{i-j}) + a L_{i-j}$$

where Q_{i-j} — travel expenses of the connected peripheral sections of the pneumatic network.

2.3. The results of the calculation of travel expenses are presented in Table.2.

table. 2

Обозначение участка сети, i-j	Потребление сжатого воздуха в конечном пункте участка или перечень потребителей, подключенных к конечному пункту	Длина участка L_{i-j}, м	Количество потребителей n_{i-j}	Путевой расход воздуха Q_{i-j} м^3/мин
4-9	1гп+1ск+6мо+1бм+1вм	640	10	33,9
4-10	1гв+1бс+1вм+1мл+1нп	580	5	32,4
5-11	1пм+12мо+2бм+1вм+1нп	460	17	29,2
5-12	1ко+2ск+1вм+2мл+1нп	420	7	23,5
6-15	2гп+1ск+5мо+1бм	320	8	28,0
6-16	2гв+6мо+1бм+2мл+2нп	400	13	44,1
7-17	8мо+2бм+2вм+2сл	420	14	26,7
7-18	2пм+10мо+1бм+1вм	510	14	24,5
8-13	1бс6мо+1бм+1вм+2сл	440	11	27,7
8-14	1ко+2ск+1вм+1мл+1нп	460	6	91,8
5-8	$Q_{8-13}+Q_{8-14}$	340	-	120,5
6-7	$Q_{7-17}+Q_{7-18}$	440	-	52,5
3-6	$Q_{6-7}+Q_{6-15}+Q_{6-16}$	240	-	125,3
3-5	$Q_{5-8}+Q_{5-11}+Q_{5-12}$	180	-	173,7
2-3	$Q_{3-5}+Q_{3-6}$	160	-	299,5
2-4	$Q_{4-9}+Q_{4-10}$	420	-	67,6
1-2	$Q_{2-3}+Q_{2-4}$	520	-	368,7

3. CALCULATION OF PRESSURE LOSSES IN THE PNEUMATIC NETWORK

3.1. The average pressure in the air ducts of the pneumatic network.

$$P_c=P_{п,мах}+\frac{\Delta P_{мах}}{2}=5,6+\frac{1,8}{2}=6,5 \text{ бар},$$

where рп.$P_{п,мах}$ = 5,6 бар — the maximum operating pressure of consumers in accordance with Table. 7;

$\Delta P_{мах}$ = 1,5 - 2,0 бар — preset maximum pressure loss in the air ducts of the pneumatic network.

3.2. Standard specific pressure loss

$\Delta P_{ун}$ = 0,0002/0,0004 бар/м = 20/40 Па/м.

According to the specified travel expenses Qi-j (Table. 8), aswell as the average pressure pc = 6.5 bar and the standard value = 30 Pa / m graphically using the nomogram shown in Fig.17.3, the diameters of the Di - j pipes and the actual specific losses in the sections of the pneumatic network are determined.

3.4. The total pressure loss in the sections of the pneumatic network is determined by the formula

$$\Delta P_{i-j} = 1{,}15 \cdot \Delta P_{УД} L i - j$$

where 1.15 is a coefficient that takes into account local pressure losses at the pipe joints and in the start—up and safety valves.

3.5. The results of calculations of pressure losses in sections of the pneumatic network are presented in Table.3.

Table 3

i-j	Q_{i-j} м^3/мин	Q_{i-j}, мм	$\Delta P_{уд}$ Па/м	L_{i-j}, м	ΔP_{i-j}, бар
4—9	33,9	125	45	640	0,331
4—10	32,4	125	43	580	0,287
5—11	29,2	125	33	460	0,175
5—12	23,5	125	22	420	0,106
6—15	28,0	125	32	320	0,118
6—16	44,1	150	31	400	0,143
7—17	26,7	125	24	420	0,116
7—18	24,5	125	25	510	0,147
8—13	27,7	125	32	440	0,162
8—14	91,8	200	31	460	0,164
5—8	120,5	200	41	340	0,160
6—7	52,5	150	47	440	0,238
3—6	125,3	200	42	240	0,116
3—5	173,7	250	43	ISO	0,089
2—3	299,5	300	46	160	0,085
2—4	67,6	150	72	420	0,348
1—2	368,7	350	60	520	0,359

4. DESIGN PRESSURE OF THE COMPRESSOR STATION

4.1. The design pressure of the compressor station according to the maximum operating pressure of consumers (consumption point 9)

P= $p_{П.MAX} + \sum\Delta P_{i-j} = P_{п.мах} + \Delta P_{1-2} + \Delta P_{2-4} + \Delta P_{4-9}$= 5,6 + 0,359 + 0,348 + 0,331=6,64 бар,

Where $p_{п.max}$ = 5,6 бар — the maximum operating pressure of consumers in accordance with Table.7; $\sum\Delta P_{i-j}$— total pressure losses in the air ducts from the compressor station to the consumer with the maximum operating pressure.

4.2. Design pressure of the compressor station along the longest section of the pneumatic network (consumption point 18)

P= $p_{П.MAX} + \sum\Delta P_{i-j}$=Pп+$\Delta P_{1-2} + \Delta P_{2-3} + \Delta P_{3-6} + \Delta P_{6-7} + \Delta P_{7-18}$= 5,0 + 0,359 + 0,085 + 0,116 + 0,238 + 0,147 = 5,945 бар,

where $P_п$ = 5,0 бар — maximum operating pressure of the consumer at the most remote point of air consumption; $\sum\Delta P_{i-j}$— total pressure loss in the air ducts to the most remote compressed air consumer.

4.3. We take the largest of the two calculated pressures as the calculated one, т. е. р = 6,64 бар.

5. SELECTION OF COMPRESSOR STATION EQUIPMENT

5.1.The selection of compressors is made according to the design capacity of the compressor station QK = 552 m3/min and the design pressure p = 6.64 bar. In the example under consideration, two variants of the compressor equipment of a pneumatic installation are possible:

a) 5 identical piston working compressors of the brand

4VM10-110/9 and 2 backup compressors of the same brand (total: 5 + 2 = 7 compressor units);

b) 2 working and 1 backup turbochargers of the brand

K-250-61-2 , as well as 1 working and 1 backup reciprocating compressors 4BM10-63/9 (total: 3 + 2 = 5 compressor units).

The first variant of the equipment of a pneumatic installation with the same type of compressors 4VM10-110/9 having the following technical characteristics is selected: performance — QK = 110m3 /min; overpressure — p = 8 bar; electric drive power — 630 kW; compressor weight — 18.7 t.

5.2. Total number of compressors

$Z_К = Z_{раб} + Z_{рез} = 5 + 2 = 7.$

5.4. If the number of VC compressors is > 3 one common air collector is provided for each pair of compressor units. Therefore, the required number of air collectors at the compressor station$Z_B=Z_K/2=7/2=3,5$

We accept $Z_в = 3$, considering that $Z_{раб} = 5$.

5.5. Estimated capacity of the air collector

$$Q_B = 1,6\sqrt{2Q_к} + 1,6\sqrt{2*110} = 23,7м^3$$

5.6 we accept an air collector of the B-25 brand with the following technical characteristics: capacity — 25m3; inner diameter — 2.0 m; shell thickness — 9mm; bottom thickness — 12mm; air collector weight — 4,615 t

5.7 . Oil separators of the V-3.20 brand with the following technical characteristics are provided for air collectors: capacity — 3.2 m; overpressure — 8 bar; weight — 1.09 t

5.8 .For each compressor unit, we provide

5.9 a vertical shell-and-tube end cooler of the brand HC-100 with the following technical characteristics: cooling surface — 180m2; excess air pressure — 8 bar; excess cooling water pressure — 2 bar; inlet air temperature — 140 °C; outlet air temperature — 30 ° C; cooling water temperature — 20 ° C; cooler weight — 2.74 t

5.10 The selection of filters for cleaning the intake atmospheric air is made according to the calculated performance

5.11 compressor station. We accept a self-cleaning mesh filter of the KT—40 brand with the following technical characteristics: working cross—section — 3.94 m2; estimated air consumption — 655 m3/min; amount of oil - 290kg; weight - 0.65 t.

6. CALCULATION OF THE COMPRESSOR STATION COOLING SYSTEM

6.1. The degree of air compression in the compressor stage

$$\varepsilon_1 = \sqrt{p_2/p_1} = \sqrt{9/1} = 3$$

Where p_2 и p_1 — pressure at the outlet and inlet of the compressor, respectively.

6.2. Air temperature at the compressor outlet

$$T_2=T_1\,\varepsilon_1^{\frac{n-1}{n}}=293\cdot 3^{\frac{1,25-1}{1,25}}\ 365\ К,$$

where T_1= 293 К (20°С) — atmospheric air temperature;

n = 1,20**Ошибка! Источник ссылки не найден.**1,25 — the polytrope index of air compression in the compressor.

6.3. Specific amount of heat removed by the water jacket of the compressor cylinder:

$$q_{\text{T}}=\frac{k-n}{n-1}C_V(T_2-T_1)=\frac{1{,}4-1{,}25}{1{,}25-1}0.721(365\text{-}293)=31{,}1\ \text{кДж/кг},$$

where k =1,4 — adiabatic air compression index; Cv - 0.721 kJ/(kgK) — isochoric heat capacity of air.

6.4. Specific heat removed in the intermediate and end coolers of the pneumatic installation:

$$q_{\text{TX}}=C_n(T_H-T_K) = 1{,}005(365\text{-}293) = 72{,}4\ \text{кДж/кг},$$

where C_p = 1,005 кДж/(кг-К) — isobaric heat capacity of air; Tn and Tc, — air temperature, respectively, at the beginning and end of the path of movement in coolers,К ($T_н = T_2$ и $T_к = T_1$).

6.5. The total specific amount of heat discharged in the compressor unit:

Ошибка! Источник ссылки не найден.=2-31,1+2-97,5 = 207 кДж/кг,

where **Ошибка! Источник ссылки не найден.** и **Ошибка! Источник ссылки не найден.** — the number of stages and the number of coolers in one compressor unit ($z_{ст}$ = 2 и $Z_{х.}$ = 2).

6.6. The total amount of heat removed by the cooling system of the compressor unit per unit of time:

Ошибка! Источник ссылки не найден.=60*l,2*207*110 = l,64*10^6 кДж/ч,

where $\rho_{вс}$ =1,2 кг/м^3 — air density under normal atmospheric conditions; $Q_к$ = 110 м^3/мин — compressor performance, м^3/мин.

6.7. Estimated cooling water consumption per compressor unit

$$Q_{ов}=\frac{E_T}{\rho_в C_в(t_{в2}-t_{в1})}$$ =**Ошибка! Источник ссылки не найден.**= 19,5 м^3/ч,

where $p_в$ = 1000 кг/м^3 — cooling water density; $C_в$ = 4,2 кДж/(кг·К) — heat capacity of cooling water; $t_{в1}$ и $t_{в2}$ — the temperature of the cooling water at the inlet and outlet of the cooling system, respectively.

6.8. Total water consumption by the cooling system

$$Q_p=Z_{раб}Q_{ов}=5\cdot 19.5=97.5\ \text{м}^3/\text{ч}.$$

6.9. Required area of splash pools

$П_{бб}$ = (0,8 **Ошибка! Источник ссылки не найден.** 1,3) Q_p =(0,8 **Ошибка! Источник ссылки не найден.**1,3) 97,5 = 78 **Ошибка! Источник ссылки не найден.**127 м^2.

5. TASK AND INITIAL DATA FOR THE DESIGN OF PNEUMATIC INSTALLATIONS.

According to the initial data presented below (Table.5) design a pneumatic installation for mines developing steep formations. The scheme of the air supply network and the technical characteristics of compressed air consumers are shown in Figure 1 and in Table.4. The order of mining the mine field is direct.

Note. The numbers before the letter designations of compressed air consumers indicate their number in the corresponding points. Ag-mine productivity, Lsh-mine depth.

TAble.5

№	**$A_г$** млн. т	**$L_ш$** м.	**Расстояние между узловыми точками пневматической сети**	**Распределение потребителей сжатого воздуха по пунктам потребления**
1	**4**	**400**	L_{1-2} = 280 м; L_{2-3} = 120 м; L_{2-4}= 180 м; L_{3-5} = 140 м; L_{3-6} = 200 м; L_{4-9} = 360 м; L_{4-10} = 300 м; L_{5-8} = 240 м; L_{5-11} = 220 м; L_{5-12} = 230 м; L_{6-7} = 310 м; L_{6-15} = 250 м; L_{6-16} = 270 м; L_{7-17} = 260 м; L_{7-18}=320 м; L_{8-13} = 170 м; L_{8-14} = 230 м.	пункт 9— (1гп + 1ск + 4 мо + 1бм + 1вм); пункт 10 — (1гв + 1бс + 1вм + 1мл); пункт 11 — (1пм + 10 мо + 1бм + 1нп); пункт 12 — (1ко + 1ск + 1вм + 1мл + 1нп); пункт 13 — (1бс + 4 мо + 1бм + 2сл); пункт 14 — (1ко + 1ск+ 1вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 1бм); пункт 16 — (2гв + 6мо + 1мл + 1нп); пункт 17— (8мо + 2бм + 2вм + 1сл); пункт 18 — (1пм + 9мо + 1бм + 1вм).
2	**4.2**	**420**	L_{1-2} = 280 м; L_{2-3} = 140 м; L_{2-4}= 190 м; L_{3-5} = 140 м; L_{3-6} = 200 м; L_{4-9} = 360 м; L_{4-10} = 300 м; L_{5-8} = 250 м; L_{5-11} = 240 м; L_{5-12} = 230 м; L_{6-7} = 340 м; L_{6-15} = 260 м; L_{6-16} = 290 м; L_{7-17} = 260 м; L_{7-18}=320 м; L_{8-13} = 170 м; L_{8-14} = 230 м.	пункт 9— (1гп + 1ск + 4 мо + 1бм + 1вм+1пм); пункт 10 — (1гв + 1бс + 1вм + 1мл+1пм); пункт 11 — (1пм + 10 мо + 1бм + 1нп); пункт 12 — (1ко + 1ск + 1вм + 1мл + 1нп); пункт 13 — (1бс + 6 мо + 1бм + 2сл+1вм); пункт 14 — (1ко + 1ск+ 1вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 1бм+1вм+1сл); пункт 16 — (2гв + 6мо + 2мл + 1нп+2бм+2ск); пункт 17— (8мо + 2бм + 2вм + 1сл+1нп+1сл); пункт 18 — (1пм + 9мо + 1бм + 1вм+1мл+1нп).
3	**4.4**	**440**	L_{1-2} = 300 м; L_{2-3} = 140 м; L_{2-4}= 200 м; L_{3-5} = 180 м; L_{3-6} = 220 м; L_{4-9} = 380 м; L_{4-10} = 320 м; L_{5-8} = 260 м; L_{5-11} = 230 м; L_{5-12} = 240 м; L_{6-7} = 330 м; L_{6-15} = 250 м; L_{6-16} = 270 м; L_{7-}	пункт 9— (1гп + 1ск + 4 мо + 2бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 2вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 1бм +

			$_{17}$ = 260 м; $L_{7\text{-}18}$=320 м; $L_{8\text{-}13}$ = 210 м; $L_{8\text{-}14}$ = 250 м.	2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 8 мо + 2бм + 2сл+1пм+2мл); пункт 14 — (1ко + 1ск+ 2вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 2бм+1мл+1вм); пункт 16 — (2гв + 6мо + 1мл + 1нп+1ск+1ск); пункт 17— (8мо + 2бм + 2вм + 1сл+2ск+1нп); пункт 18 — (1пм + 9мо + 2бм + 1вм+2сл).
4	**4.6**	**460**	$L_{1\text{-}2}$ = 300 м; $L_{2\text{-}3}$ = 160 м; $L_{2\text{-}4}$= 210 м; $L_{3\text{-}5}$ = 180 м; $L_{3\text{-}6}$ = 260 м; $L_{4\text{-}9}$ = 400 м; $L_{4\text{-}10}$ = 300 м; $L_{5\text{-}8}$ = 280 м; $L_{5\text{-}11}$ = 250 м; $L_{5\text{-}12}$ = 240 м; $L_{6\text{-}7}$ = 380 м; $L_{6\text{-}15}$ = 270 м; $L_{6\text{-}16}$ = 280 м; $L_{7\text{-}17}$ = 260 м; $L_{7\text{-}18}$=360 м; $L_{8\text{-}13}$ = 320 м; $L_{8\text{-}14}$ = 350 м.	пункт 9— (1гп + 1ск + 4 мо + 1бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 2вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 2бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 8 мо + 2бм +2мл); пункт 14 — (1ко + 1ск+ 1вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 2бм+1мл+1вм); пункт 16 — (2гв + 6мо + 1мл +1ск+1ск); пункт 17— (10мо + 2бм + 1вм + 1сл+2ск); пункт 18 — (1пм + 9мо + 2бм +2сл).
5	**4.8**	**480**	$L_{1\text{-}2}$ = 310 м; $L_{2\text{-}3}$ = 170 м; $L_{2\text{-}4}$= 185 м; $L_{3\text{-}5}$ = 140 м; $L_{3\text{-}6}$ = 200 м; $L_{4\text{-}9}$ = 370 м; $L_{4\text{-}10}$ = 300 м; $L_{5\text{-}8}$ = 230 м; $L_{5\text{-}11}$ = 225 м; $L_{5\text{-}12}$ = 230 м; $L_{6\text{-}7}$ = 310 м; $L_{6\text{-}15}$ = 250 м; $L_{6\text{-}16}$ = 255 м; $L_{7\text{-}17}$ = 270 м; $L_{7\text{-}18}$=320 м; $L_{8\text{-}13}$ = 290 м; $L_{8\text{-}14}$ = 260 м.	пункт 9— (1гп + 1ск + 8 мо + 1бм + 1вм); пункт 10 — (1гв + 1бс + 1вм + 1мл+1пм); пункт 11 — (1пм + 11 мо + 1бм + 1нп+1мл); пункт 12 — (1ко + 1ск + 1вм + 1мл + 1нп); пункт 13 — (1бс + 7 мо + 2бм + 2сл+1ск+1нп); пункт 14 — (1ко + 1ск+ 1вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 1бм+1вм+1сл); пункт 16 — (2гв + 6мо + 1мл + 1нп+1бм); пункт 17— (8мо + 2бм + 2вм + 1сл+1мл+1нп); пункт 18 — (1пм + 9мо + 1бм + 1вм+1нп).
6	**5**	**500**	$L_{1\text{-}2}$ = 320 м; $L_{2\text{-}3}$ = 180 м; $L_{2\text{-}4}$= 180 м; $L_{3\text{-}5}$ = 160 м; $L_{3\text{-}6}$ = 210 м; $L_{4\text{-}9}$ = 370 м; $L_{4\text{-}10}$ = 300 м; $L_{5\text{-}8}$ = 250 м; $L_{5\text{-}11}$ = 220 м; $L_{5\text{-}12}$ = 220 м; $L_{6\text{-}7}$ = 310 м; $L_{6\text{-}15}$ = 250 м; $L_{6\text{-}16}$ = 280 м; $L_{7\text{-}}$	пункт 9— (1гп + 1ск + 10 мо + 1бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 1вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 1бм +

			$_{17}$ = 270 м; L_{7-18}=320 м; L_{8-13} = 290 м; L_{8-14} = 350 м.	2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 7 мо + 2бм + 2сл+1пм); пункт 14 — (1ко + 1ск+ 2вм + 1мл +1бм); пункт 15 — (1гп + 1ск + 5мо + 2бм+1вм); пункт 16 — (2гв + 6мо + 1мл + 1нп+1ск+1ск); пункт 17— (8мо + 2бм + 2вм + 1сл+1ск); пункт 18 — (1пм + 9мо + 2бм + 1вм+2сл).
7	**5.2**	**520**	L_{1-2} = 310 м; L_{2-3} = 210 м; L_{2-4}= 280 м; L_{3-5} = 260 м; L_{3-6} = 240 м; L_{4-9} = 400 м; L_{4-10} = 300 м; L_{5-8} = 240 м; L_{5-11} = 220 м; L_{5-12} = 230 м; L_{6-7} = 360 м; L_{6-15} = 250 м; L_{6-16} = 270 м; L_{7-17} = 260 м; L_{7-18}=320 м; L_{8-13} = 270 м; L_{8-14} = 330 м.	пункт 9— (1гп + 1ск + 11 мо + 2бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 2вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 2бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 8 мо + 2бм + 2сл+1пм+2мл); пункт 14 — (1ко + 1ск+ 2вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 2бм+1мл+1вм); пункт 16 — (2гв + 6мо + 1мл +1ск); пункт 17— (9мо + 2бм + 2вм + 1сл+2ск); пункт 18 — (1пм + 9мо + 2бм + 1вм+1сл).
8	**5.4**	**540**	L_{1-2} =320 м; L_{2-3} = 220 м; L_{2-4}= 280 м; L_{3-5} = 240 м; L_{3-6} = 300 м; L_{4-9} = 460 м; L_{4-10} = 400 м; L_{5-8} = 440 м; L_{5-11} = 320 м; L_{5-12} = 330 м; L_{6-7} = 410 м; L_{6-15} = 350 м; L_{6-16} = 370 м; L_{7-17} = 360 м; L_{7-18}=420 м; L_{8-13} = 370 м; L_{8-14} = 330 м.	пункт 9— (1гп + 1ск + 10 мо + 1бм + 1вм); пункт 10 — (1гв + 1бс + 1вм + 1мл); пункт 11 — (1пм + 10 мо + 1бм + 1нп); пункт 12 — (1ко + 1ск + 1вм + 1мл + 1нп); пункт 13 — (1бс + 10 мо + 1бм + 2сл); пункт 14 — (1ко + 1ск+ 1вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 1бм); пункт 16 — (2гв + 8мо + 1мл + 1нп); пункт 17— (8мо + 2бм + 2вм + 1сл); пункт 18 — (1пм + 9мо + 1бм + 1вм).
9	**5.6**	**560**	L_{1-2} = 300 м; L_{2-3} = 260 м; L_{2-4}= 190 м; L_{3-5} = 140 м; L_{3-6} = 200 м; L_{4-9} = 360 м; L_{4-10} = 300 м; L_{5-8} = 250 м; L_{5-11} = 240 м; L_{5-12} = 230 м; L_{6-7} = 340 м; L_{6-15} = 260 м; L_{6-16} = 290 м; L_{7-17} = 260 м; L_{7-18}=320 м; L_{8-13} = 470 м; L_{8-14} = 330 м.	пункт 9— (1гп + 1ск + 10 мо + 1бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 1вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 1бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 7 мо + 2бм +

				2сл+1пм); пункт 14 — (1ко + 1ск+ 2вм + 1мл +1бм); пункт 15 — (1гп + 1ск + 5мо + 2бм+1вм); пункт 16 — (2гв + 6мо + 1мл + 1нп+1ск+1ск); пункт 17— (8мо + 2бм + 2вм + 1сл+1ск); пункт 18 — (1пм + 9мо + 2бм + 1вм+2сл).
10	**5.8**	**580**	L_{1-2} = 300 м; L_{2-3} = 280 м; L_{2-4}= 400 м; L_{3-5} = 380 м; L_{3-6} = 420 м; L_{4-9} = 480 м; L_{4-10} = 420 м; L_{5-8} = 360 м; L_{5-11} = 430 м; L_{5-12} = 340 м; L_{6-7} = 330 м; L_{6-15} = 350 м; L_{6-16} = 370 м; L_{7-17} = 360 м; L_{7-18}=420 м; L_{8-13} = 410 м; L_{8-14} = 450 м.	пункт 9— (1гп + 1ск + 4 мо + 2бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 2вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 2бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 8 мо + 2бм + 2сл+1пм+2мл); пункт 14 — (1ко + 1ск+ 2вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 2бм+1мл+1вм); пункт 16 — (2гв + 6мо + 1мл + 1нп+1ск+1ск); пункт 17— (9мо + 2бм + 2вм + 1сл+2ск+1нп); пункт 18 — (1пм + 9мо + 2бм + 1вм+2сл).
11	**6**	**600**	L_{1-2} = 400 м; L_{2-3} = 200 м; L_{2-4}= 310 м; L_{3-5} = 380 м; L_{3-6} = 360 м; L_{4-9} = 500 м; L_{4-10} = 400 м; L_{5-8} = 380 м; L_{5-11} = 350 м; L_{5-12} = 340 м; L_{6-7} = 480 м; L_{6-15} = 370 м; L_{6-16} = 380 м; L_{7-17} = 360 м; L_{7-18}=460 м; L_{8-13} = 420 м; L_{8-14} = 450 м.	пункт 9— (1гп + 1ск + 6 мо + 1бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 2вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 2бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 8 мо + 2бм +2мл); пункт 14 — (1ко + 1ск+ 1вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 6мо + 2бм+1мл+2вм); пункт 16 — (2гв + 9мо + 1мл +1ск+1ск); пункт 17— (10мо + 2бм + 2вм + 1сл+2ск); пункт 18 — (1пм + 9мо + 2бм +2сл).
12	**6.2**	**620**	L_{1-2} = 280 м; L_{2-3} = 120 м; L_{2-4}= 180 м; L_{3-5} = 140 м; L_{3-6} = 200 м; L_{4-9} = 360 м; L_{4-10} = 300 м; L_{5-8} = 240 м; L_{5-11} = 220 м; L_{5-12} = 230 м; L_{6-7} = 310 м; L_{6-15} = 250 м; L_{6-16} = 270 м; L_{7-17} = 260 м; L_{7-18}=320 м; L_{8-13} = 170 м; L_{8-14} = 230 м.	пункт 9— (1гп + 1ск + 4 мо + 1бм + 1вм); пункт 10 — (1гв + 1бс + 1вм + 1мл); пункт 11 — (1пм + 10 мо + 1бм + 1нп); пункт 12 — (1ко + 1ск + 1вм + 1мл + 1нп); пункт 13 — (1бс + 4 мо + 1бм + 2сл); пункт 14 — (1ко + 1ск+ 1вм + 1мл +

				1нп); пункт 15 — (1гп + 1ск + 4мо + 1бм); пункт 16 — (2гв + 6мо + 1мл + 1нп); пункт 17— (8мо + 2бм + 2вм + 1сл); пункт 18 — (1пм + 9мо + 1бм + 1вм).
13	**5**	**420**	L_{1-2} = 280 м; L_{2-3} = 140 м; L_{2-4}= 190 м; L_{3-5} = 140 м; L_{3-6} = 200 м; L_{4-9} = 360 м; L_{4-10} = 300 м; L_{5-8} = 250 м; L_{5-11} = 240 м; L_{5-12} = 230 м; L_{6-7} = 340 м; L_{6-15} = 260 м; L_{6-16} = 290 м; L_{7-17} = 260 м; L_{7-18}=320 м; L_{8-13} = 170 м; L_{8-14} = 230 м.	пункт 9— (1гп + 1ск + 4 мо + 1бм + 1вм+1пм); пункт 10 — (1гв + 1бс + 1вм + 1мл+1пм); пункт 11 — (1пм + 10 мо + 1бм + 1нп); пункт 12 — (1ко + 1ск + 1вм + 1мл + 1нп); пункт 13 — (1бс + 6 мо + 1бм + 2сл+1вм); пункт 14 — (1ко + 1ск+ 1вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 1бм+1вм+1сл); пункт 16 — (2гв + 6мо + 2мл + 1нп+2бм+2ск); пункт 17— (8мо + 2бм + 2вм + 1сл+1нп+1сл); пункт 18 — (1пм + 9мо + 1бм + 1вм+1мл+1нп).
14	**4.5**	**440**	L_{1-2} = 300 м; L_{2-3} = 140 м; L_{2-4}= 200 м; L_{3-5} = 180 м; L_{3-6} = 220 м; L_{4-9} = 380 м; L_{4-10} = 320 м; L_{5-8} = 260 м; L_{5-11} = 230 м; L_{5-12} = 240 м; L_{6-7} = 330 м; L_{6-15} = 250 м; L_{6-16} = 270 м; L_{7-17} = 260 м; L_{7-18}=320 м; L_{8-13} = 210 м; L_{8-14} = 250 м.	пункт 9— (1гп + 1ск + 4 мо + 2бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 2вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 1бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 8 мо + 2бм + 2сл+1пм+2мл); пункт 14 — (1ко + 1ск+ 2вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 2бм+1мл+1вм); пункт 16 — (2гв + 6мо + 1мл + 1нп+1ск+1ск); пункт 17— (8мо + 2бм + 2вм + 1сл+2ск+1нп); пункт 18 — (1пм + 9мо + 2бм + 1вм+2сл).
15	**5.2**	**520**	L_{1-2} = 310 м; L_{2-3} = 210 м; L_{2-4}= 280 м; L_{3-5} = 260 м; L_{3-6} = 240 м; L_{4-9} = 400 м; L_{4-10} = 300 м; L_{5-8} = 240 м; L_{5-11} = 220 м; L_{5-12} = 230 м; L_{6-7} = 360 м; L_{6-15} = 250 м; L_{6-16} = 270 м; L_{7-17} = 260 м; L_{7-18}=320 м; L_{8-13} = 270 м; L_{8-14} = 330 м.	пункт 9— (1гп + 1ск + 11 мо + 2бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 2вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 2бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 8 мо + 2бм + 2сл+1пм+2мл); пункт 14 — (1ко + 1ск+ 2вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо +

				2бм+1мл+1вм); пункт 16 — (2гв + 6мо + 1мл +1ск); пункт 17— (9мо + 2бм + 2вм + 1сл+2ск); пункт 18 — (1пм + 9мо + 2бм + 1вм+1сл).
16	**5.4**	**540**	L_{1-2} =320 м; L_{2-3} = 220 м; L_{2-4}= 280 м; L_{3-5} = 240 м; L_{3-6} = 300 м; L_{4-9} = 460 м; L_{4-10} = 400 м; L_{5-8} = 440 м; L_{5-11} = 320 м; L_{5-12} = 330 м; L_{6-7} = 410 м; L_{6-15} = 350 м; L_{6-16} = 370 м; L_{7-17} = 360 м; L_{7-18}=420 м; L_{8-13} = 370 м; L_{8-14} = 330 м.	пункт 9— (1гп + 1ск + 10 мо + 1бм + 1вм); пункт 10 — (1гв + 1бс + 1вм + 1мл); пункт 11 — (1пм + 10 мо + 1бм + 1нп); пункт 12 — (1ко + 1ск + 1вм + 1мл + 1нп); пункт 13 — (1бс + 10 мо + 1бм + 2сл); пункт 14 — (1ко + 1ск+ 1вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 1бм); пункт 16 — (2гв + 8мо + 1мл + 1нп); пункт 17— (8мо + 2бм + 2вм + 1сл); пункт 18 — (1пм + 9мо + 1бм + 1вм).
17	**5.6**	**560**	L_{1-2} = 300 м; L_{2-3} = 260 м; L_{2-4}= 190 м; L_{3-5} = 140 м; L_{3-6} = 200 м; L_{4-9} = 360 м; L_{4-10} = 300 м; L_{5-8} = 250 м; L_{5-11} = 240 м; L_{5-12} = 230 м; L_{6-7} = 340 м; L_{6-15} = 260 м; L_{6-16} = 290 м; L_{7-17} = 260 м; L_{7-18}=320 м; L_{8-13} = 470 м; L_{8-14} = 330 м.	пункт 9— (1гп + 1ск + 10 мо + 1бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 1вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 1бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 7 мо + 2бм + 2сл+1пм); пункт 14 — (1ко + 1ск+ 2вм + 1мл +1бм); пункт 15 — (1гп + 1ск + 5мо + 2бм+1вм); пункт 16 — (2гв + 6мо + 1мл + 1нп+1ск+1ск); пункт 17— (8мо + 2бм + 2вм + 1сл+1ск); пункт 18 — (1пм + 9мо + 2бм + 1вм+2сл).
18	**5.8**	**580**	L_{1-2} = 300 м; L_{2-3} = 280 м; L_{2-4}= 400 м; L_{3-5} = 380 м; L_{3-6} = 420 м; L_{4-9} = 480 м; L_{4-10} = 420 м; L_{5-8} = 360 м; L_{5-11} = 430 м; L_{5-12} = 340 м; L_{6-7} = 330 м; L_{6-15} = 350 м; L_{6-16} = 370 м; L_{7-17} = 360 м; L_{7-18}=420 м; L_{8-13} = 410 м; L_{8-14} = 450 м.	пункт 9— (1гп + 1ск + 4 мо + 2бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 2вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 2бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 8 мо + 2бм + 2сл+1пм+2мл); пункт 14 — (1ко + 1ск+ 2вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 2бм+1мл+1вм); пункт 16 — (2гв + 6мо + 1мл + 1нп+1ск+1ск); пункт 17— (9мо + 2бм + 2вм +

				1сл+2ск+1нп); пункт 18 — (1пм + 9мо + 2бм + 1вм+2сл).
19	**6**	**600**	L_{1-2} = 400 м; L_{2-3} = 200 м; L_{2-4}= 310 м; L_{3-5} = 380 м; L_{3-6} = 360 м; L_{4-9} = 500 м; L_{4-10} = 400 м; L_{5-8} = 380 м; L_{5-11} = 350 м; L_{5-12} = 340 м; L_{6-7} = 480 м; L_{6-15} = 370 м; L_{6-16} = 380 м; L_{7-17} = 360 м; L_{7-18}=460 м; L_{8-13} = 420 м; L_{8-14} = 450 м.	пункт 9— (1гп + 1ск + 6 мо + 1бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 2вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 2бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 8 мо + 2бм +2мл); пункт 14 — (1ко + 1ск+ 1вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 6мо + 2бм+1мл+2вм); пункт 16 — (2гв + 9мо + 1мл +1ск+1ск); пункт 17— (10мо + 2бм + 2вм + 1сл+2ск); пункт 18 — (1пм + 9мо + 2бм +2сл).
20	**4.6**	**460**	L_{1-2} = 300 м; L_{2-3} = 160 м; L_{2-4}= 210 м; L_{3-5} = 180 м; L_{3-6} = 260 м; L_{4-9} = 400 м; L_{4-10} = 300 м; L_{5-8} = 280 м; L_{5-11} = 250 м; L_{5-12} = 240 м; L_{6-7} = 380 м; L_{6-15} = 270 м; L_{6-16} = 280 м; L_{7-17} = 260 м; L_{7-18}=360 м; L_{8-13} = 320 м; L_{8-14} = 350 м.	пункт 9— (1гп + 1ск + 4 мо + 1бм + 1вм+1мл); пункт 10 — (1гв + 1бс + 2вм + 1мл+1ск+1нп); пункт 11 — (1пм + 10 мо + 2бм + 2нп); пункт 12 — (1ко + 1ск + 2вм + 1мл + 2нп+1пм); пункт 13 — (1бс + 8 мо + 2бм +2мл); пункт 14 — (1ко + 1ск+ 1вм + 1мл + 1нп); пункт 15 — (1гп + 1ск + 4мо + 2бм+1мл+1вм); пункт 16 — (2гв + 6мо + 1мл +1ск+1ск); пункт 17— (10мо + 2бм + 1вм + 1сл+2ск); пункт 18 — (1пм + 9мо + 2бм +2сл).

BRANCH OF THE FEDERAL STATE AUTONOMOUS EDUCATIONAL INSTITUTION OF HIGHER EDUCATION "National Research Technological University "MISIS" in Almalyk

application

in the discipline "Stationary machines"

BRANCH OF THE FEDERAL STATE AUTONOMOUS EDUCATIONAL INSTITUTION OF HIGHER EDUCATION
"National Research Technological University "MISIS" in Almalyk

DEPARTMENT OF "MINING"

Registered	"I APPROVE"
	Vice-Rector for Academic Affairs
№_________________________	_______________С.Худояров
«____»_______________2021 г.	«_____»____________2021 г.

WORKING CURRICULUM
According to the course: STATIONARY MACHINES
For specialists

Область знаний	**300 000**	Производственно-техническая сфера
Область образования	**2140070**	Инженерное дело
Направление образования	**21.05.04.**	Горное дело
Специальность	**СГД-16-9**	Горные машины и оборудование

Term	**5**	**6**
Total number of hours		190
Of these:		
Lecture	36	28
Practical exercises	36	32
Laboratory classes	18	16
Independent work/ / Course project	32	24/(50)

Almalyk-2021.

The program was compiled (and):
к.т.н., доц. Кахаров Сергей Каримович

Work program
Stationary machines

Developed in accordance with the OS in:
Independently established educational standard of higher education Federal State Autonomous Educational Institution of Higher Education "National Research Technological University "MISIS" in the specialty 21.05.04 MINING (Order No. 602 O.V. dated 02.12.2015)

Compiled on the basis of the curriculum:
21.05.04 MINING, 21.05.04-SRS-16-9.PLS Mining machinery and equipment approved by the Scientific Council of the Federal State Educational Institution of Higher Education of NUST MISIS on 21.05.2020, Protocol No. 10/zg

The working program was approved at the meeting of the department
Department of Mining Equipment, Transport and Mechanical Engineering

Protocol of 09.06.2021, No. 10

Head of the Department Maxim Grigoryevich Rakhutin

INTRODUCTION

Mine water-filling, fan and pneumatic installations are an important and integral attribute of the energy-mechanical support of mining operations. The development of a mountain massif in the exceptional majority of cases is associated with the flow of water into the mine workings, which is pumped to the surface using water-draining plants. All technological processes of mining enterprises are more or less accompanied by the inevitable pollution of atmospheric air (dust formation, release of harmful gases, etc.) The task of constantly updating atmospheric air in mine workings is performed by fan installations. By means of pneumatic installations, the needs of mining enterprises for compressed air, which is used as an energy carrier in many mining machines and mechanisms, are met.

The study of the general structure, operational features and principles of calculating the equipment of drainage, fan and pneumatic installations in one volume or another is provided for in the curricula for the preparation of bachelors of mining specialties.

The effective operation of complex mining equipment largely depends on the level of theoretical and practical training of specialists. Bachelor in the direction 5312200 – "Mining electromechanics" and 5320300 – "Technological machines and equipment" must know the classification, design, device, scope, operating mode and technological indicators of drainage, fan and pneumatic installations.

The discipline is related to the courses: "Fundamentals of mining engineering", "Electrical and structural materials", "Hydraulics and hydraulic machines", "Theory of machines and mechanisms", "Machine parts", "Fundamentals of Mining", "Drilling and blasting", "Theoretical foundations of electrical engineering" and is the basis for studying other disciplines of the direction.

COURSE CONTENT
5-semester (36 hours).
SECTION 1. THE ROLE OF STATIONARY MACHINES IN MINING. (10 hours)

Introduction. Basic information about turbomachines (2 hours).

Sections of stationary machines. The main parameters of machines for transporting liquids. Energy losses in turbomachines. Classification of turbomachines of machines for the transportation of liquids.

The main types of turbomachines, their elements and the principle of operation (4 hours).

The principle of operation and the main elements of turbomachines. Kinematics of fluid flow in the impeller. The basic energy equation of turbomachines. Fundamentals of the vortex theory of turbomachines.

The main technological characteristics of turbomachines (2 hours).

Individual and actual characteristics of turbomachines. The effect of a finite number of blades and performance on the operation of turbomachines. Inputs and outputs, and their impact on the characteristics.

Theoretical foundations of turbomachines (2 hours).

Characteristics of external networks. Operational modes of turbomachines. Joint operation of several pumps on a common network.

SECTION 2. GENERAL INFORMATION ABOUT DRAINAGE INSTALLATIONS AND DRAINAGE (26 HOURS)

General information about drainage installations and drainage (4 hours). Underground water flows. Purpose and classification of drainage installations Technological schemes of stationary drainage. Pumping chambers and water collectors

of the force acting on the pump impeller and their balancing (4 hours).

Forces acting on the pump impeller. Characteristics of pumps and their operating modes. Methods of regulation of operating modes.

Centrifugal pumps, types and designs (4 hours).

Elements of centrifugal pumps. Single-stage pumps.

Multistage pumps (6 hours).

The nomenclature of centrifugal sectional pumps. Nomenclature and design of multistage spiral pumps. Nomenclature and design of multistage spiral pumps. Vertical multistage pumps

Equipment and equipment of drainage installations (4 hours).

Requirements for drainage stationary installations. Pipelines of drainage installations Technological equipment for monitoring and control of drainage installations. Electric drive and equipment for automation of drainage installations

Operation and design of drainage systems (4 hours).

Tests of pumps of drainage installations. Maintenance of drainage installations. Method of designing a drainage system. Compressor lubrication systems. Electric drive and automation

6-semester (28 hours).

SECTION 3. GENERAL INFORMATION ABOUT SHAFT FAN INSTALLATIONS (12 HOURS).

General information of fan installations (2 hours).

Purpose and classification of fans and fan installations. Features of operation of fan installations. Characteristics and areas of industrial use of fans

Shaft fans (2 hours)..

Centrifugal fans. Features of operation of fan installations. Characteristics and areas of industrial use of fans

Axial fans (2 hours).

Classification. Nomenclature and design of axial fans. Nomenclature and design of local ventilation fans.

Main ventilation fan installations (2 hours).

The requirements for installations. Electric drive and automation equipment of fan installations. Installation diagrams

Operation and design of fan installations (4 hours).

Testing of fan installations. Maintenance of fan installations. The methodology of designing the installation of the main ventilation.

SECTION 4. GENERAL INFORMATION ABOUT MINE PNEUMATIC INSTALLATIONS (16 HOURS).

General information about pneumatic installations (2 hours).

Purpose of pneumatic installations. Classification of compressors. Main parameters of compressors

Reciprocating compressors (4 hours).

The principle of operation and classification. The working process of an ideal reciprocating compressor. The actual process in a single-stage compressor

Calculation of the piston compressor performance (2 hours).

The performance of a reciprocating compressor. Regulation of reciprocating compressors. Nomenclature and design of shaft reciprocating compressors

Centrifugal and rotary compressors (2 hours).

The principle of operation and device of the compressor. The working process of a centrifugal compressor. Nomenclature and design of centrifugal compressors and installations. Classification of compressors.

Screw and rotary plate compressors (2 hours).

Screw compressors. Rotary plate compressors.Liquid-ring compressors of loaders.

Operation and design of pneumatic installations (4 hours).

Testing of compressors. Maintenance of pneumatic installations. Method of designing a pneumatic installation.

1. TOPICS OF LABORATORY WORK.

5-semester (8 hours).

2. Study of designs and basic elements of centrifugal turbomachines. (2 hours)
3. Study of the designs and principle of operation of single-stage and multi-stage pumps. (2 hours)
4. Study of pipes, their auxiliary equipment and connection schemes with pumps. (2 hours)
5. Experimental testing of pumps and determination of their operational characteristics. (2 hours)
6. -semester (4 hours).
7. Study of the aerodynamic scheme, structures and main parts of axial and centrifugal fans. (2 hours)
8. Study of circuits, main components, design and principle of operation of single-stage and multi-stage reciprocating compressors. (2 hours)

TOPICS OF PRACTICAL CLASSES.

5-semester (36 hours).

1. Choosing the type of pump. (2 hours).
2. Selection of the type of pumping unit (type of pumping station). (4 hours).
3. Arrangement of equipment within the station (4 hours).
4. Determination of the parameters of the operating mode of the installation (4 hours).
5. Collector selection (2 hours).
6. Calculation and selection of pipelines, determination of their characteristics (4 hours).
7. Determination of the operating mode of technological indicators of the drainage system (4 hours)
8. Determination of the energy consumption of the drainage system (2 hours).
9. Calculation of geometric parameters of water collectors and placement of pumps in the pumping chamber (4 hours).
10. Selection of installation automation equipment (4 hours).
11. Calculation of technical and economic indicators of the installation (2 hours).

6-semester (32 hours).

1. Select the type of fan. (2 hours)
2. Calculation and determination of the characteristics of the external network of the fan installation. (4 hours)
3. Determination of the operating mode and technological parameters of fan installations. (4 hours)
4. Selection of the electric drive of fan installations. (2 hours)
5. Determination of economic indicators of fans. (2 hours)
6. Determination of compressed air flow in the pneumatic network. (4 hours)
7. Selection of the type and number of compressor and determination of spare parameters of the device. (4 hours)

8. Calculation and selection of pipelines. (4 hours)

9. Calculation and selection of auxiliary equipment of the compressor unit (4 hours)

10. Determination of technical and economic indicators of pneumatic installations. (2 hours)

COURSE PROJECT.

The course project is designed to consolidate and deepen the knowledge gained by students in the course of studying the course. To teach students to apply the acquired knowledge in the independent solution of technical issues related to complex mechanization and automation of the main production processes in the development of mineral deposits, operation and modernization of some units of installations. To instill in students a sense of responsibility for the assigned work and personal initiative in solving the task. To prepare students for independent work with reference literature, current GOST standards, departmental standards, engineering calculation methods adopted in design organizations and industry and to develop skills in drawing up calculation and explanatory notes. The ultimate goal of course design is to prepare students for the completion of qualifying final work.

Topics of the course project:

a) Calculation and design of mine drainage installations.

b) Calculation and design of shaft fan installations.

c) Calculation and design of mine pneumatic installations.

CRITERIA for ASSESSING

students' knowledge based on the rating system for the subject "Stationary machines".

The main focus of the national training program is to improve the quality of education. In higher educational institutions, students' knowledge is evaluated according to the rating system. Assessment of students' knowledge according to the rating system allows students to constantly work to improve the acquired knowledge, and is also an incentive for the development of creative activity in the learning process.

These evaluation criteria for the discipline “Stationary machines” are recommended for wide use in assessing students' knowledge and provide complete information to students on the number of points they can receive during the current, intermediate and final controls.

At the first lesson, students get acquainted with the evaluation criteria, types of control, with the maximum score for each control, as well as with passing scores.

1. Types of control and evaluation procedure

In accordance with the curriculum of the specialty 21.05.04 MINING, 21.05.04-SRS-16-9.PLX Mining machines and equipment, the discipline “Stationary machines” is held in the third year for 5, 6 semesters. In order to correspond the degree of knowledge and assimilation to the state educational standard, the following types of control are provided:

current control is a method of determining the acquired knowledge in the discipline “Stationary machines” during practical classes. The assessment is carried out by checking the solutions of the tasks according to the variants of each practical work, as well as a survey on the topic;

intermediate control is a method of determining the degree of knowledge on the topics covered. Intermediate control is carried out twice during the semester, the form and score are determined in accordance with the allotted amount of hours according to the curriculum;

final control is a method of determining the theoretical knowledge and practical skills obtained. The final control is based on the basic concepts and is carried out in writing.

Students are evaluated on a 100-point scale during each semester in the subjects. 70 points are allocated for intermediate and current control, 30 points for final control.

1. Rating table.

п/н	Курс	Семестр	Количество недель	Общее количество часов (рейтенговый балл)	Лекция	Практические занятия	Лабораторные занятия	Часы для самостоятельной работы	Аб-аудиторный балл Сб-балл за самостоятельную	Виды контроля												Для дисциплины с курсовым проектом
										Итого часов в %	ТК	ТК – 1	ТК – 2	ПК	ПК – 1	ПК– 2	∑ТК+ПК	Прρходной балл	ИК	Форма проведения ИК	Показатель усвоения	
1	3	5	18	140	36	36	18	32	Аб/Сб	100	5	5	5	5	5	5	5	3	5	уст	5	
2	3	6	16	112	28	32	16	24	Аб/Сб	100	5	5	5	5	5	5	5	3	5	уст	5	1

3. “Stationary machines”

3.1. Rating development
5th semester

П.н	Вид контроля	Кол во	Балл и	Итого баллов
1.	**ТК общий балл 35**			
1.1.	Практические занятия	18	5	5
1.2.	Лабораторные работы	9	5	5
2.	**ПК общий балл 35**			
2.1.	Первая промежуточная работа (3 вопроса)	1	5	5
2.2.	Вторая промежуточная работа (3 вопроса)	1	5	5
2.3.	Самостоятельная работа	1	5	5
∑ТК+ПК				**5**
3.	**ИК**			
3.1.	Итоговая контрольная работа	1	5	5
Итого				**5**

6th semester

П.н	Вид контроля	Кол во	Балл	Итого баллов
1.	**ТК общий балл 35**			
1.1.	Практические занятия	16	5	5
1.2.	Лабораторные работы	8	5	5
2.	**ПК общий балл 35**			

2.1.	Первая промежуточная работа (3 вопроса)	1	5	5
2.2.	Вторая промежуточная работа (3 вопроса)	1	5	5
∑ТК+ПК				**5**
3.	**ИК**			
3.1.	Итоговая контрольная работа	1	5	5
Итого				**5**

3.2 Evaluation criteria for current control

The student who performs the practical work is given grades 5-excellent if the work is done qualitatively, but according to the level of response to the assessment questions 4-good, if not completely done, then according to the degree of completion of the assessment 3-satisfactory, if the answers to the questions do not correspond as is, also the answers are not fully then the answer to this question is -0.

The intermediate control (first) is made in the form of a written work, in the form of answers to 4 questions. Each answer is evaluated according to the 2,3,4,5 point system:

if the essence of the question is fully disclosed, the answers are accurate and complete, then – score 5;

if the answer to the question is generalized, the essence is not fully disclosed and some facts are missing – then – otsunka 4;

if there is an attempt to answer the question, but there are confusions, then – score 3

List of questions for the first interim control

1. The main parameters of machines for transporting liquids.
2. Energy losses in turbomachines.
3. Classification of turbomachines of liquid transportation machines
4. The principle of operation and the main elements of turbomachines.
5. Kinematics of fluid flow in the impeller.
6. The basic energy equation of turbomachines.
7. Fundamentals of the vortex theory of turbomachines.
8. Individual and actual characteristics of turbomachines.
9. The effect of a finite number of blades and performance on the operation of turbomachines.
10. Feedings and taps, and their effect on the characteristics.
11. Characteristics of external networks.
12. Operating modes of turbomachines.
13. Joint operation of several pumps on a common network.
14. Purpose and classification of drainage installations
15. Technological schemes of stationary drainage.
16. Pumping chambers and water collectors

17. Forces acting on the pump impeller.
18. Characteristics of pumps and their operating modes.
19. Methods of regulation of operating modes.
20. Elements of centrifugal pumps.
21. Single-stage pumps.
22. Vertical multistage pumps.
23. Requirements for drainage stationary installations.
24. Pipelines of drainage installations
25. Technological equipment for monitoring and controlling drainage installations.
26. Electric drive and equipment for automation of drainage installations
27. Tests of pumps of drainage installations.
28. Maintenance of drainage installations.
29. Design methodology of the drainage system.

List of questions for the second interim control

- Purpose and classification of fans and fan installations.
- Features of the fan installations.
- Characteristics and areas of industrial use of fans.
- Centrifugal fans.
- Features of the fan installations.
- Characteristics and areas of industrial use of fans
- Nomenclature and design of axial fans.
- Nomenclature and design of local ventilation fans.
- Electric drive and automation equipment for fan installations.
- Testing of fan installations.
- Maintenance of fan installations.
- The methodology of designing the main ventilation unit.
- Purpose of pneumatic installations.
- Classification of compressors.
- Basic parameters of compressors.
- Principle of operation and classification.
- The working process of an ideal reciprocating compressor.
- A valid process in a single-stage compressor.
- The performance of the reciprocating compressor.
- Regulation of reciprocating compressors.
- Nomenclature and design of shaft reciprocating compressors
- The principle of operation and device of the compressor.
- The working process of a centrifugal compressor.
- Nomenclature and design of centrifugal compressors and installations.
- Classification of compressors.
- Screw compressors.
- Rotary plate compressors.
- Liquid-ring compressors of loaders.

- Testing of compressors.
-Maintenance of pneumatic installations.
- Methods of designing a pneumatic installation.

List of questions for the final control of the discipline "Stationary machines"

- The main parameters of machines for transporting liquids.
- Energy losses in turbomachines.
- Classification of turbomachines of machines for the transportation of liquids
- The principle of operation and the main elements of turbomachines.
- Kinematics of fluid flow in the impeller.
- The basic energy equation of turbomachines.
- Fundamentals of the vortex theory of turbomachines.
- Individual and valid characteristics of turbomachines.
- The effect of a finite number of blades and performance on the operation of turbomachines.
- Drains and bends, and their effect on the characteristics.
- Characteristics of external networks.
- Operating modes of turbomachines.
- Joint operation of several pumps on a common network.
- Purpose and classification of drainage installations
- Technological schemes of stationary drainage.
- Pumping chambers and water collectors.
- Forces acting on the pump impeller.
- Characteristics of pumps and their operating modes.
- Ways to regulate operating modes.
- Elements of centrifugal pumps.
- Single-stage pumps.
- Vertical multistage pumps.
- Requirements for drainage stationary installations.
- Pipelines of drainage installations
- Technological equipment for monitoring and controlling drainage installations.
- Electric drive and equipment for automation of drainage installations .
- Testing of pumps of drainage installations.
- Maintenance of drainage installations.

Method of designing a drainage system.
- Compressor lubrication systems.
- Purpose and classification of fans and fan installations.
- Features of the fan installations.
- Centrifugal fans.
- Features of the fan installations.
- Nomenclature and design of axial fans.

-Nomenclature and design of local ventilation fans.
- Testing of fan installations.

- Maintenance of fan installations.
- The methodology of designing the main ventilation unit.
- Purpose of pneumatic installations.
- Classification of compressors.
- Basic parameters of compressors.
- Principle of operation and classification.
- The working process of an ideal reciprocating compressor.
- A valid process in a single-stage compressor.
- The performance of the reciprocating compressor.
- Regulation of reciprocating compressors.
- Nomenclature and design of shaft reciprocating compressors.
- The principle of operation and device of the compressor.
- The working process of a centrifugal compressor.
- Classification of compressors.
- Screw compressors.
- Rotary plate compressors.
- Liquid-ring compressors of loaders.
- Testing of compressors.
- Maintenance of pneumatic installations.
- Methods of designing a pneumatic installation.

3.5. The procedure for the final control

The final control in this discipline is carried out at the end of the 6th semester in accordance with the approved schedule.

The final control work is accepted in writing. Each option consists of three questions and reference words. The questions should meet the requirements of the material passed.

At the beginning of the academic year, the list of questions and tickets are updated by the teacher and approved at a meeting of the department.

After the final control, the teacher is obliged to check and evaluate the student's work within two days, as well as to inform them about the score received.

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1. Гейер В.Г., Тимошенко Г.М., «Шахтные вентиляторные и водоотливные установки».Недра. М.1981 г.
2. Картавий Н.Г. «Стационарные машины». Недра. М.1981 г.

Дополнительная литература

3. Братченко Б.Ф. «Стационарные установки шахт». Недра. М. 1977 г.
4. Шерматов Ш.М. «Сув чикариш, вентилятор ва пневматик қурилмалар» фанидан маърузалар. Биринчи қисм. 1993й.
5. Гришко А.П. «Стационарные машины». Том 2. Рудничные водоотливные, вентиляторные и пневматические установки: Учебник для вузов. —М.: Издательство ≪Горная книга≫, 2007. — 586 с

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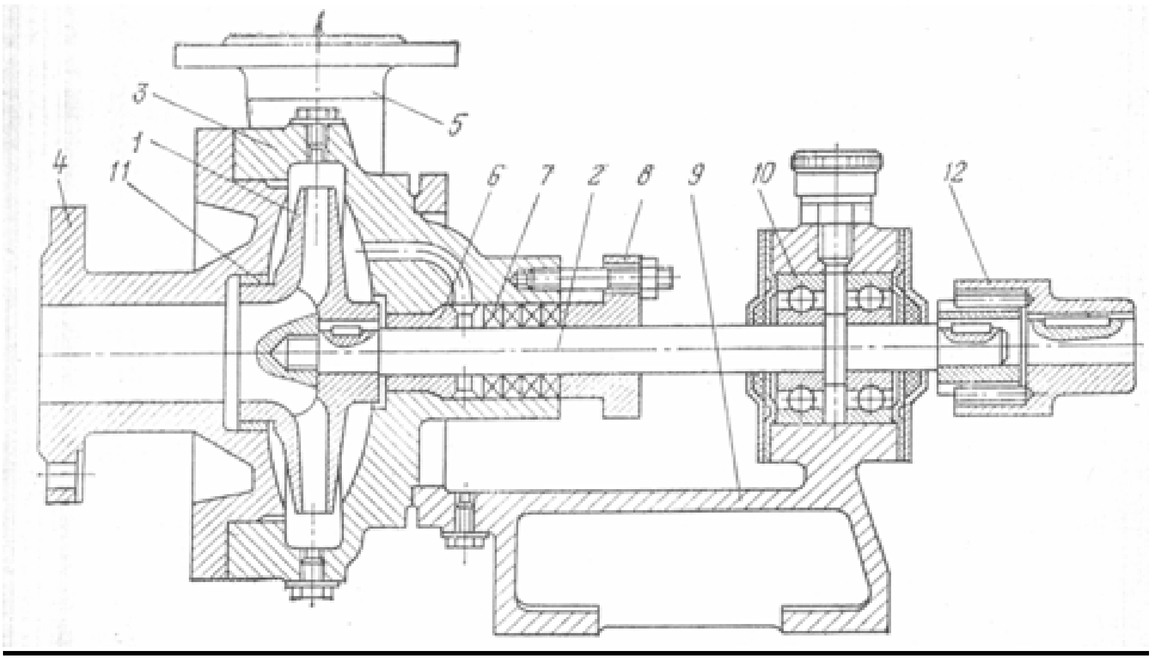

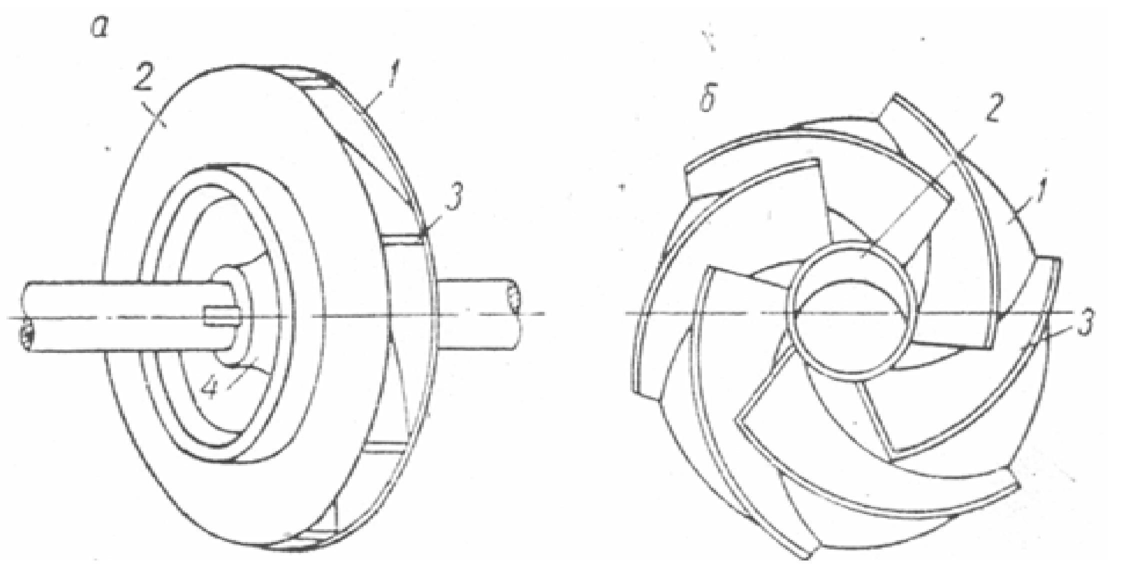

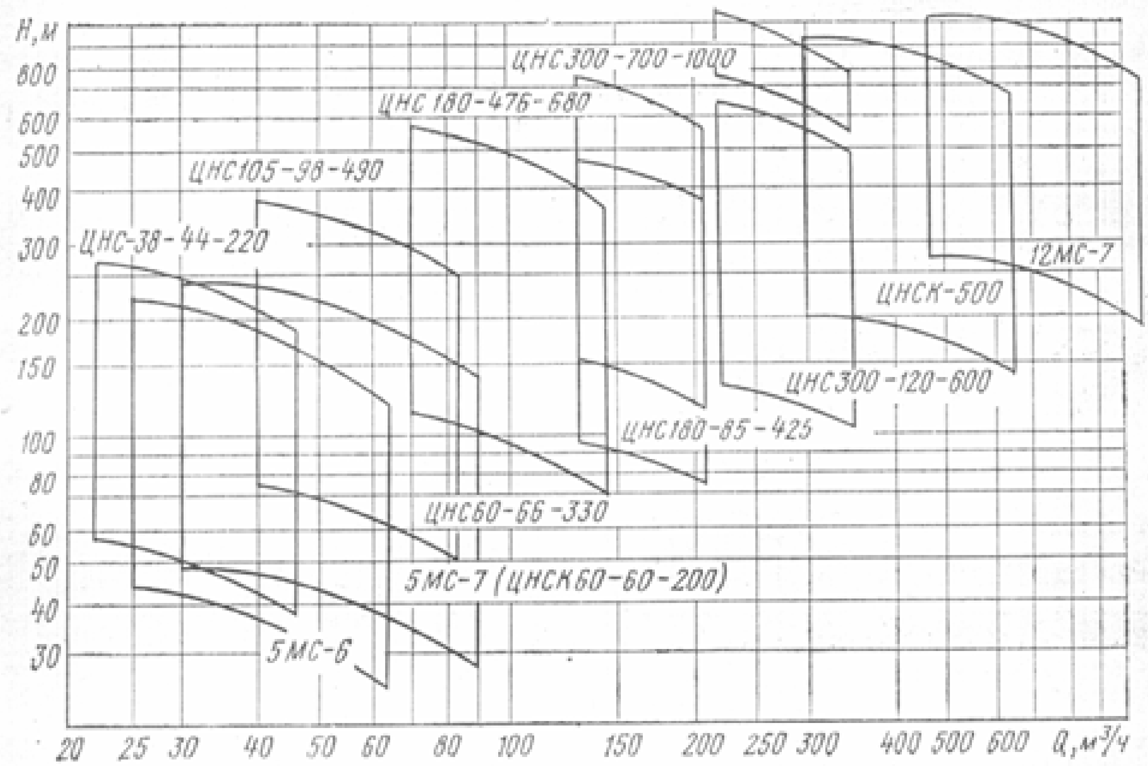
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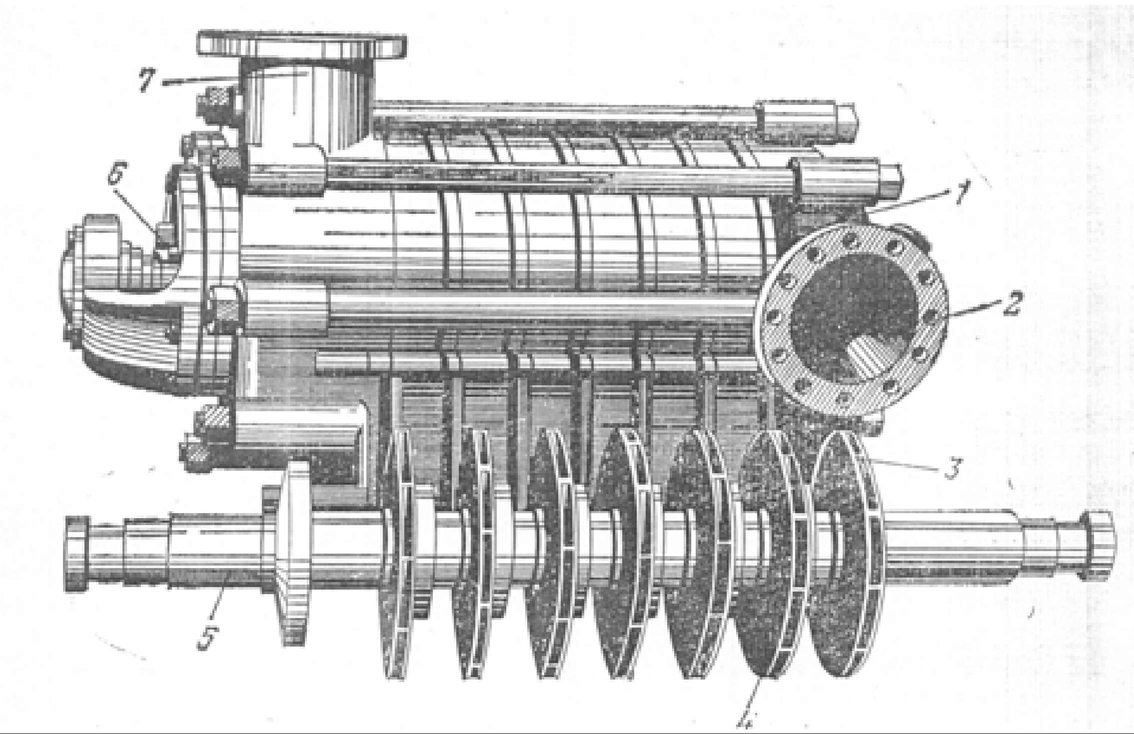
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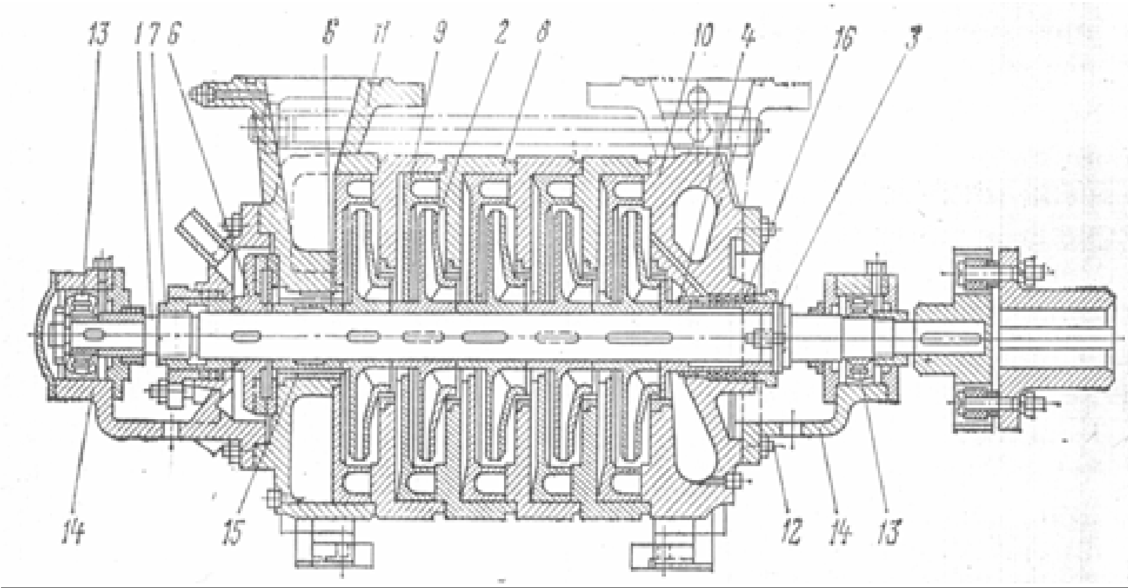
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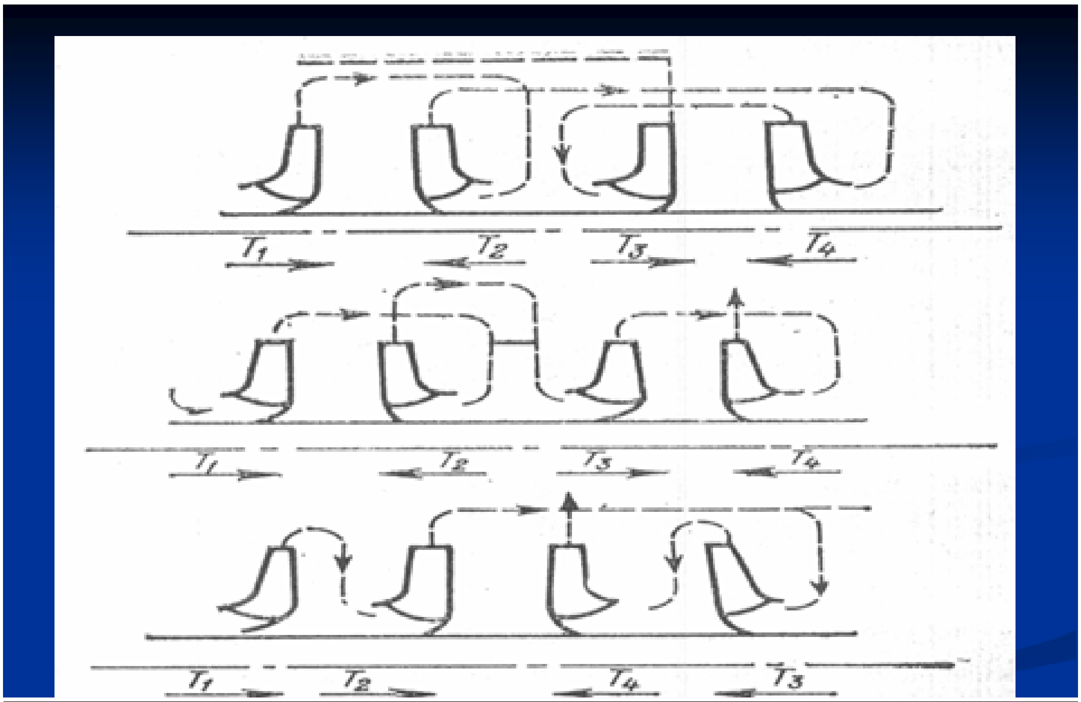
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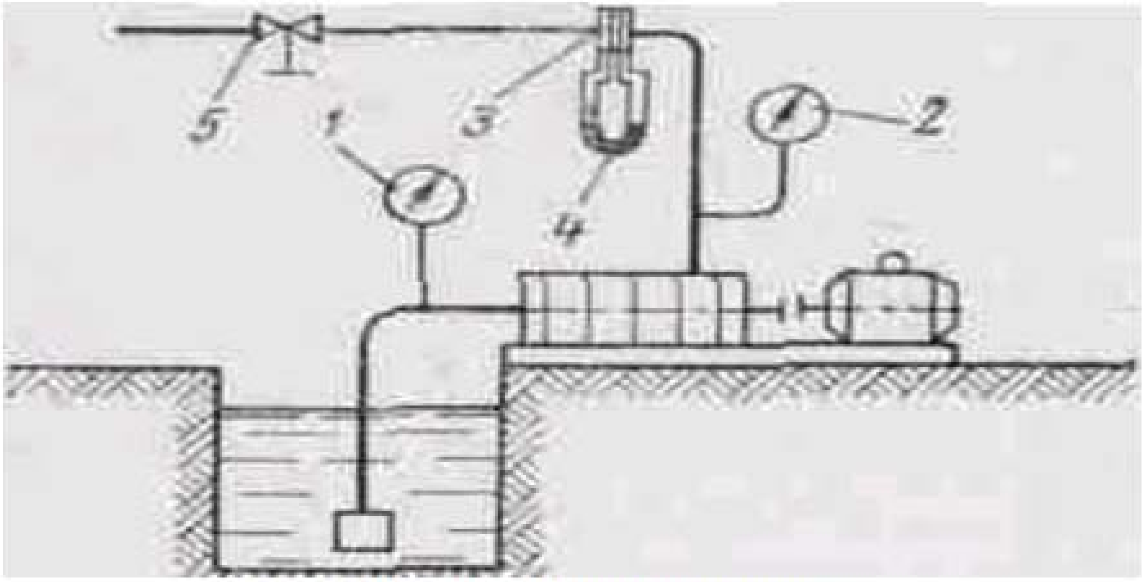

- Схема установки приборов
и аппаратуры при промышленных испытаниях насосов:
1 — вакуумметр; *2* — манометр; *3* —измерительная диафрагма; *4* — дифференциальный манометр;
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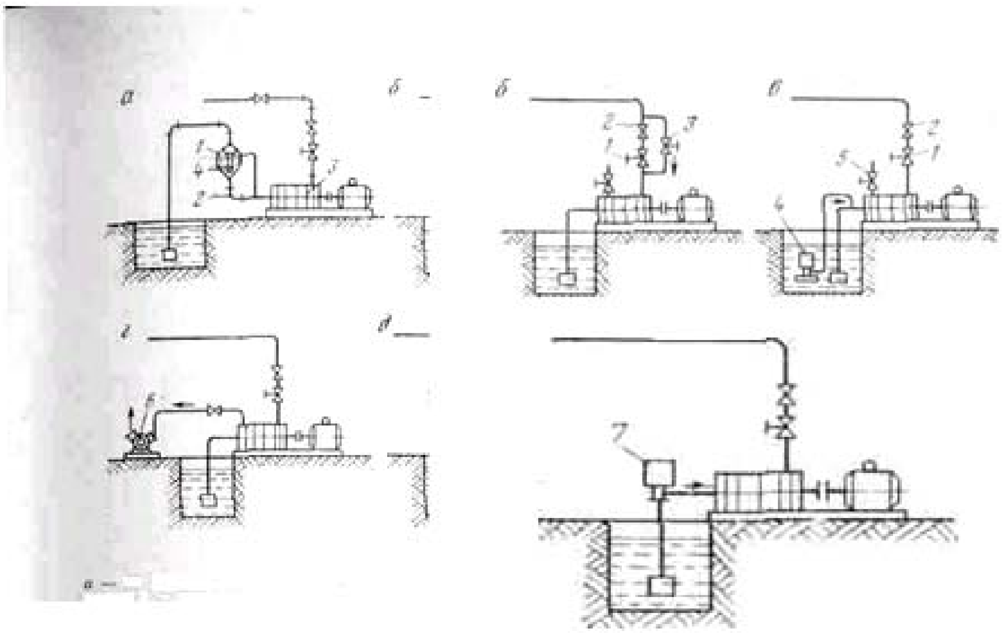
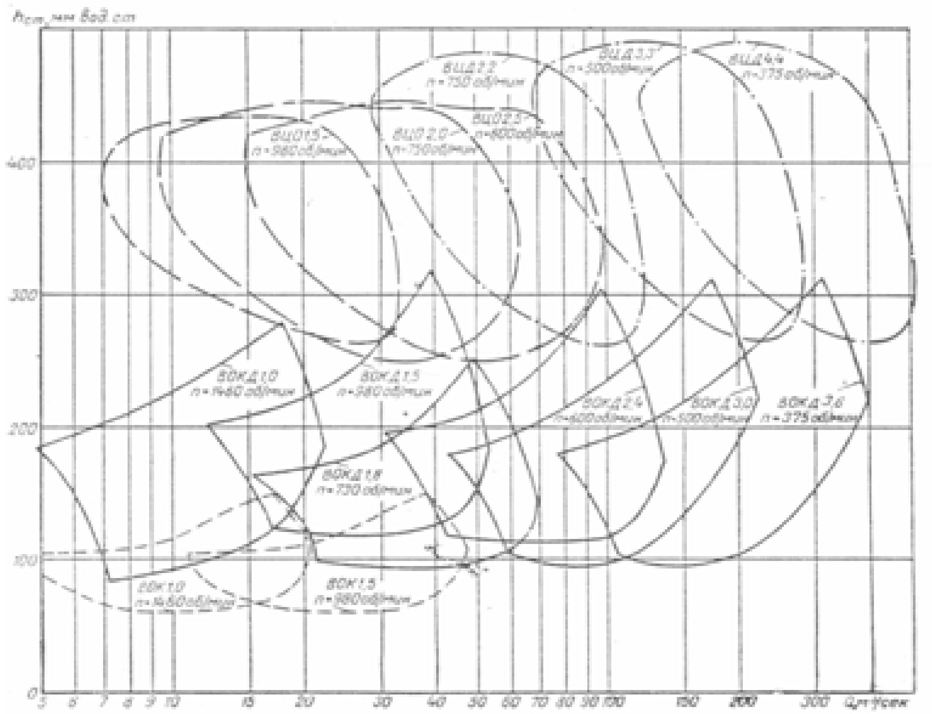


- а - из бака-аккумулятора: 1 — бак-аккумулятор; *2* — всасывающий трубопровод; *3* — насос; *4* — эжектор; б — из нагнетательного трубопровода; в — вспомогательным насосом; *г* — вакуум-насосом; *д* — подпорным насосом; 1 — главная задвижка; 2 — обратный клапан; 3 — вспомогательная задвижка; 4 — вспомогательный насос; 5 — задвижка; 6 - вакуум-насос; 7 — подпорный (бустерный) насос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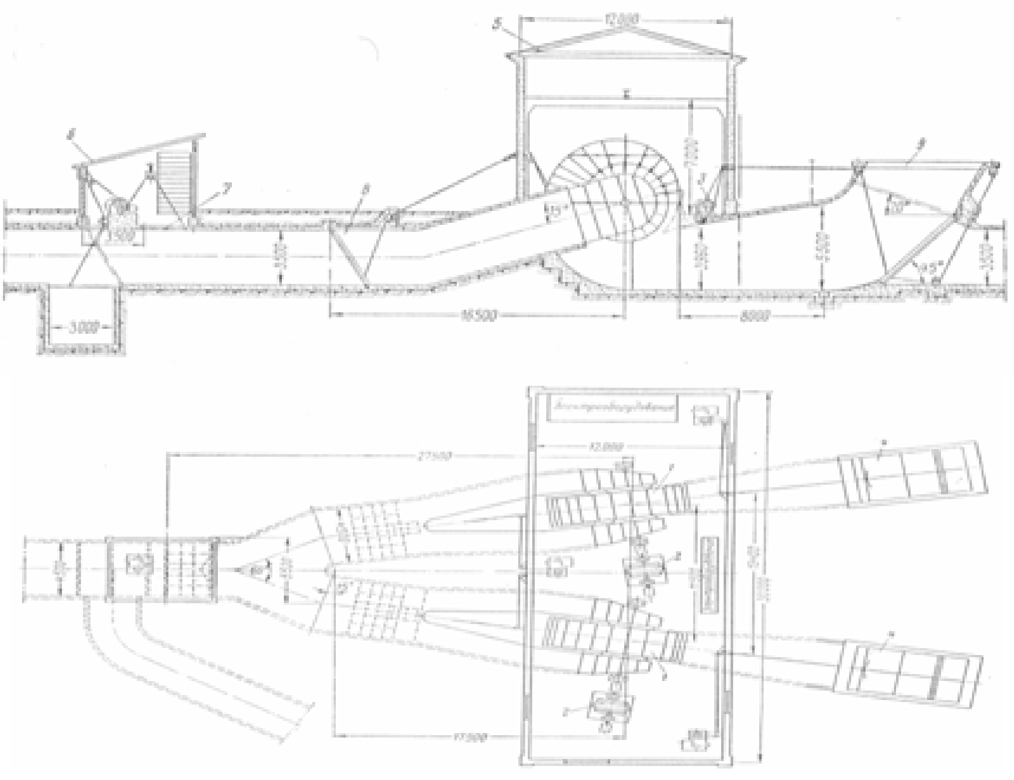

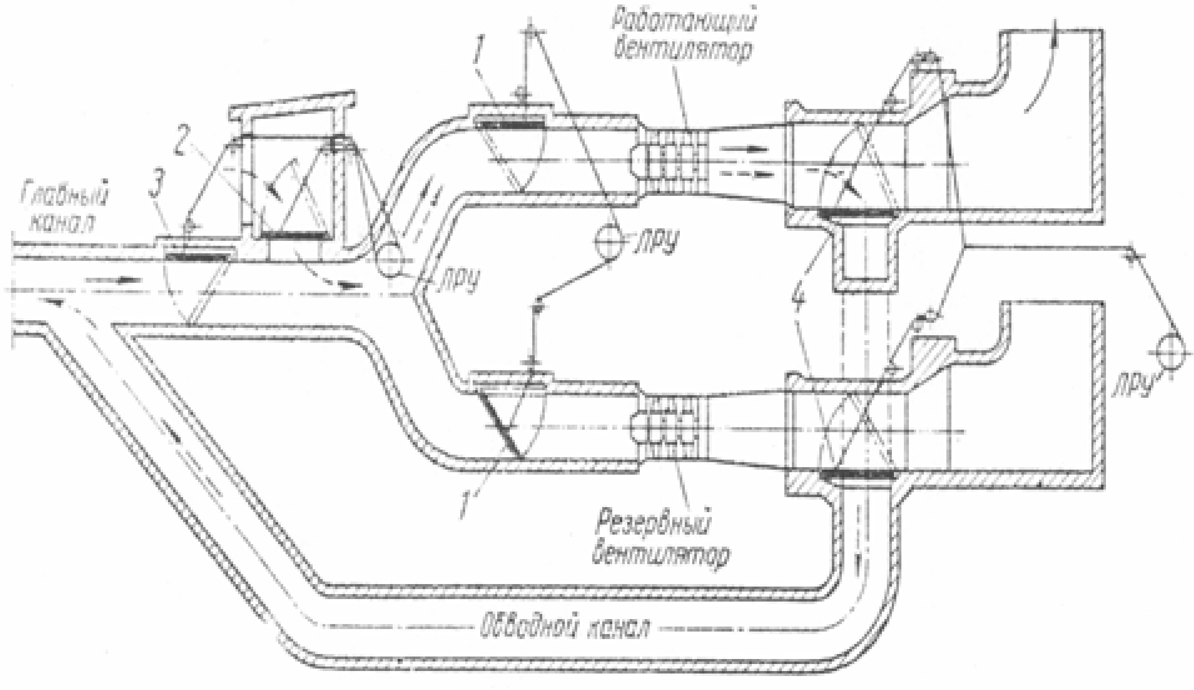
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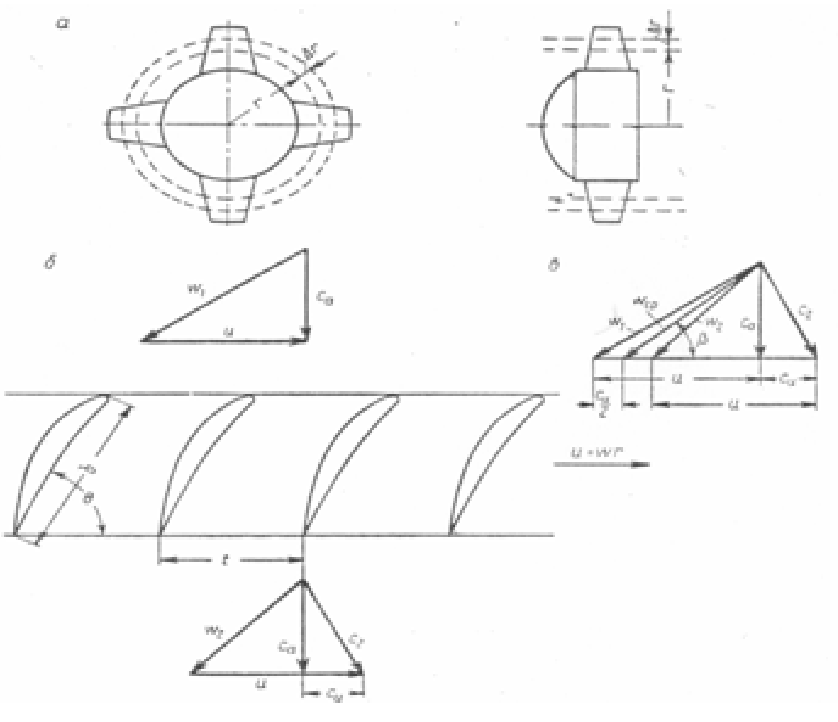
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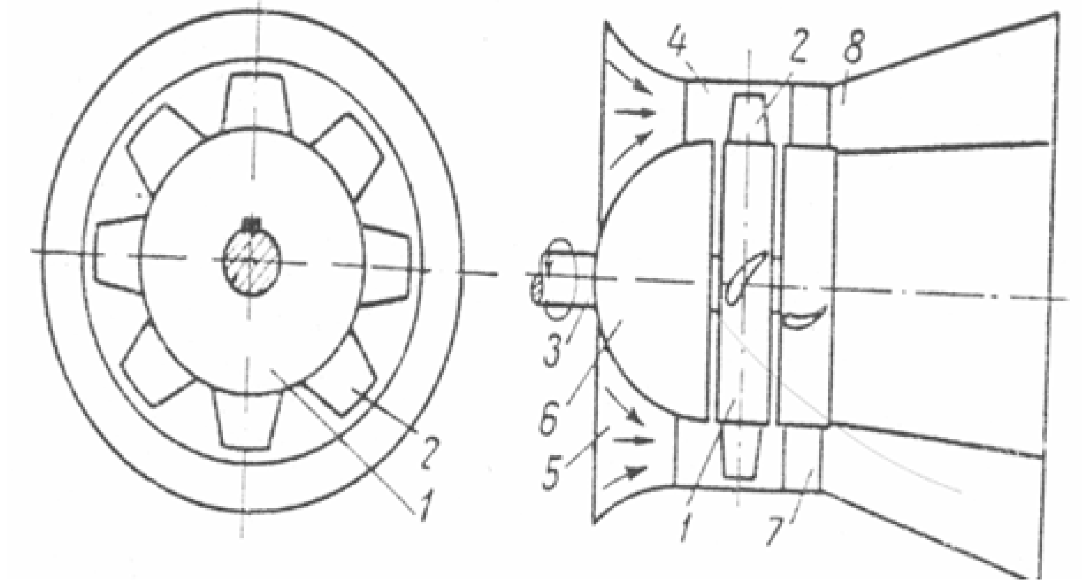
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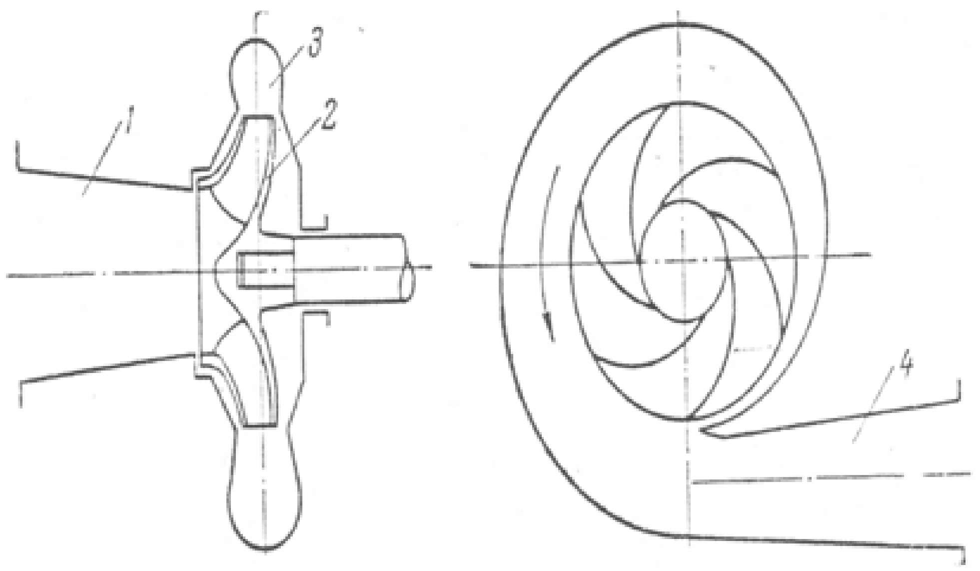
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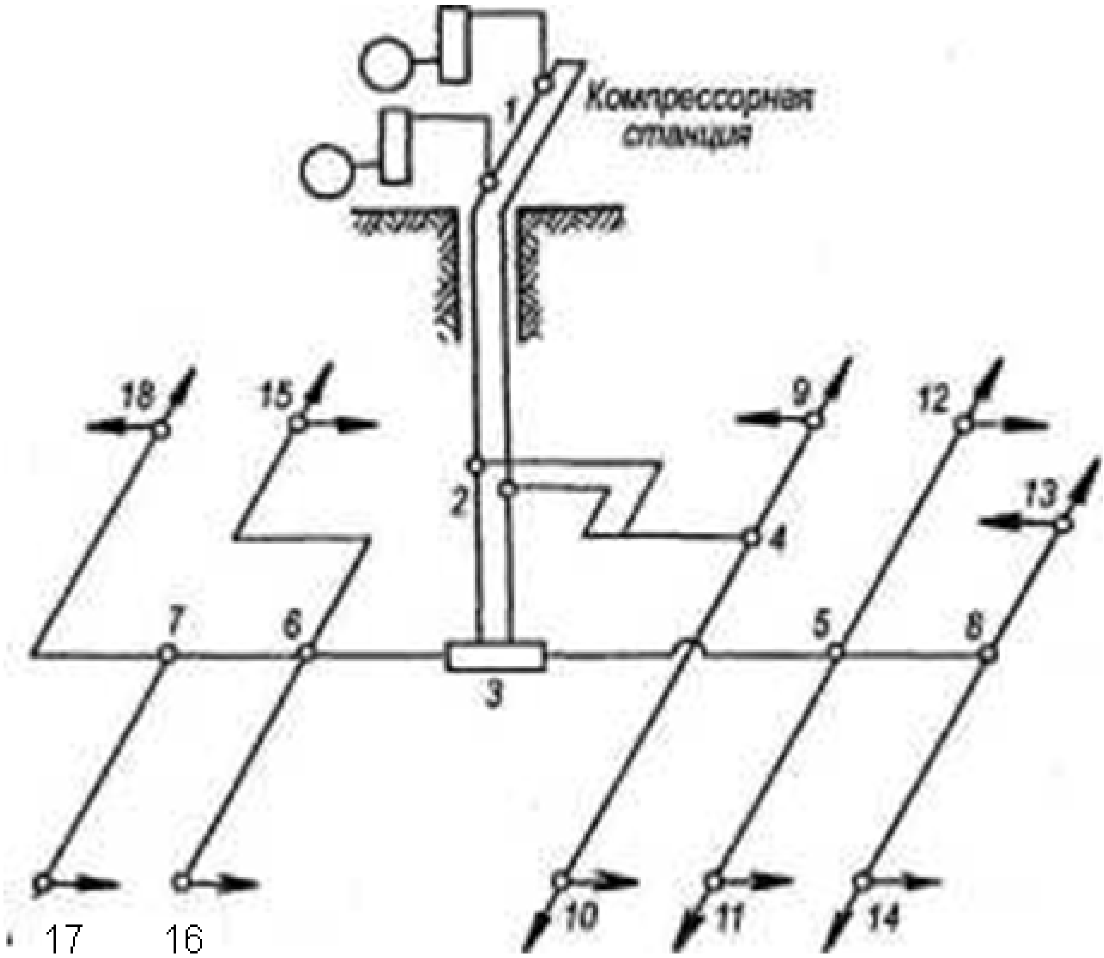
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The list of control questions for self-control in the discipline: "Stationary machines"

1. The main parameters of machines for transporting liquids.

2. Energy losses in turbomachines.

3. Classification of turbomachines of liquid transportation machines

4. The principle of operation and the main elements of turbomachines.

5. Kinematics of fluid flow in the impeller.

6. The basic energy equation of turbomachines.

7. Fundamentals of the vortex theory of turbomachines.

8. Individual and actual characteristics of turbomachines.

9. The effect of a finite number of blades and performance on the operation of turbomachines.

10. Feedings and taps, and their effect on the characteristics.

11. Characteristics of external networks.

12. Operating modes of turbomachines.

13. Joint operation of several pumps on a common network.

14. Purpose and classification of drainage installations

15. Technological schemes of stationary drainage.

16. Pumping chambers and water collectors

17. Forces acting on the pump impeller.

18. Characteristics of pumps and their operating modes.

19. Methods of regulation of operating modes.

20. Elements of centrifugal pumps.

21. Single-stage pumps.

22. Vertical multistage pumps.

23. Requirements for drainage stationary installations.

24. Pipelines of drainage installations

25. Technological equipment for monitoring and controlling drainage installations.

26. Electric drive and equipment for automation of drainage installations
27. Tests of pumps of drainage installations.
28. Maintenance of drainage installations.
29. Design methodology of the drainage system.
30. Purpose and classification of fans and fan installations.
31. Centrifugal fans
32. Features of fan installations.
33. Nomenclature and design of axial fans.
34. Nomenclature and design of local ventilation fans.
35. Testing of fan installations.
36. Maintenance of fan installations.
37. Methods of designing the main ventilation installation.
38. Purpose of pneumatic installations.
39. Classification of compressors.
40. Main parameters of compressors
41. The principle of operation and classification.
42. The working process of an ideal reciprocating compressor.
43. The actual process in a single-stage compressor
44. The performance of a reciprocating compressor.
45. Regulation of reciprocating compressors.
46. Nomenclature and design of shaft reciprocating compressors
47. The principle of operation and device of the compressor.
48. The working process of a centrifugal compressor.
49. Classification of compressors.
50. Screw compressors.
51. Rotary plate compressors.
52. Liquid-ring compressors of loaders.
53. Compressor tests.
54. Maintenance of pneumatic installations.

55. Methods of designing a pneumatic installation.

TOPICS OF PRACTICAL CLASSES (34 hours).

1. Performance of stationary machines. (2 hours)
2. Calculation of the main drainage system of the mine (2 hours)
3. Calculation and selection of pipelines (2 hours).
4. Determination and analysis of the operating mode of the drainage system (4 hours)
5.. Pump drive and energy consumption of the drainage system (2 hours).
6. Calculation of the main drainage installation of the quarry (4 hours).
7. Calculation and selection of pipelines (2 hours)
8. Pump drive and energy consumption of the drainage system (2 hours)
9. Calculation of the installation of the main ventilation of the mine (2 hours)
10. Operational calculation of the fan (4 hours)
11. Calculation of mine stationary pneumatic installation (4 hours)
12. Selection of compressor station equipment (4 hours)

TOPICS OF LABORATORY WORK (20 hours).

1. Study of designs of centrifugal turbomachines. (4 hours)
2. Study of designs of centrifugal pumps. (2 hours)
3. Study of auxiliary equipment designs. (4 hours)
4. Study of axial fan designs (2 hours)
5. Study of centrifugal fan designs (2 hours)
6. Study of piston compressor designs (4 hours)
7. Study of rotary compressor designs (2 hours)

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11. Картавий Н.Г. «Стационарные машины». Недра. М.1981 г.

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