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FUNDAMENTALS OF ENGINEERING DESIGN

Methodological instructions for the implementation of practical tasks, independent work and individual tasks in the disciplines «Machine parts» for foreign applicants of the specialty 131 – Applied Mechanics of the first (bachelor's) level of education of full-time education

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Compiled by: V. V. Klitnoi

Reviewer: A.V. Gaydamaka

Department of Machine Components and Hydropneumatic Systems

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Introduction

The basis of any country's industry is mechanical engineering. The quality of machines is laid at the design stage, implemented in production, and manifested in operation. Therefore, future engineering and technical workers must master the knowledge of modern methods of creating machines. The course «Machine Parts» is designed to study and practically master the methods of calculating and designing parts, their connections and assemblies, which are used in the vast majority of modern machines. The course «Machine Parts» is one of the fundamental disciplines in the preparation of a bachelor of mechanical engineering. The knowledge and practical skills obtained in this discipline allow not only to successfully complete a diploma project, but also to start working in production. Mastering the course is based on the knowledge, skills and abilities obtained during the study of higher mathematics, physics, descriptive geometry, engineering and computer graphics, theoretical mechanics, theory of machines and mechanisms, strength of materials, technology of structural materials. According to the results of studying the discipline "Machine Parts", students should know the designs, characteristic types of damage and methods of calculating parts; be able to assess the performance of parts and design assemblies; have skills in working with modern computing equipment. The course «Machine Parts» completes the cycle of general technical disciplines and allows you to begin studying special courses based on the mastered approaches to designing parts of general machine-building machines. Scientific and technical research on the most general issues related to mechanical engineering, regardless of the field of affiliation and the intended purpose of the machines, is aimed at improving design methods. The development of methods for calculating machine parts allows you to reduce material consumption and manufacturing costs, increases reliability, ensures energy saving and competitiveness.

Practical training 1. Fundamentals of engineering design. Mechanical transmissions.

1.1. Basic concepts and definitions.

This section provides an outline of the course "Machine parts".

Machine parts is a discipline that studies the bases of calculation and design of parts and components of general-purpose machinery.

Machine is a device for performing mechanical motion for energy, materials or information conversion to facilitate or replace physical and mental work.

Depending on the nature of the work flow and performed functions, machines are conditionally divided into: energy (motors, generators, and turbines), information (computers), transportation (cars, conveyors), and technology (machines).

Structurally, the machine is a single set of mechanisms, assembly units and machine parts that ensure implementation of inherent functions.
Machine part is part of the machine, made of the same by brand and name material without the use of assembly units. Items which are only used for specific machines are referred to parts for special purposes (turbine blades, ropes, etc.). The parts that are found in almost all of the machines are called general-purpose parts (nuts, pins, gears). The course machine parts deals with calculations and design of parts used for only general purpose.

Parts form sub-assemblies or components.
Unit is a set of parts assembled at the factory-manufacturer by means of assembly operations (bolting, riveting, and welding) and designed to work jointly. A simple assembly is part of a complex one, which in turn is the product node (unit), a complex and so forth. For example, the bearing, support assembly, gearbox. Some units simultaneously perform the function of mechanisms.

1.2. The main technical requirements for the design of machine elements and features of their calculation.

The perfection of the part design is evaluated by its reliability, efficiency and adaptability. In this case, reliability is understood to be the probability of failure-free operation of the part within a specified period of service without unscheduled repairs. Producible parts are the ones that require minimal labor, money and time to produce. Serviceability is a state of the part in which it is able to perform the specified function with the parameters set by the requirements of the technical documentation. The main serviceability criteria are strength, rigidity, wear, heat and vibration resistance. Calculations concerning parts serviceability according to specified criteria are the subject of study of the discipline "machine parts".

Strength - the ability of items not to break down or take permanent residual deformation under the influence of external forces for a specified period of service.

They distinguish static (violation of static strength is usually associated with overloading), and fatigue (loss of fatigue strength is caused by long-acting alternating stresses) strength of parts. Strength calculation is performed by methods of resistance of materials that preclude the possibility of dangerous strains, breakage or surface damage occurrence. The strength is increased by a rational form of parts, removing of stress concentration, and use of surface hardening.

Rigidity - the ability to resist the change in the shape of parts and the size under load.

The rigidity of parts provides the required precision of the machine. Deformations do not only change the size and shape of parts, but also the nature of their pairing. The latter affects the strength and wear resistance. The role of rigidity as a criterion of efficiency is continuously increasing due to the increased rapidity of machines, reducing the weight and dimensions of parts.

Wear resistance is the ability to resist wear and tear of parts, that is, the process of destruction and separation of material from a solid surface under friction, which leads to a gradual change in the size, shape and surface condition of parts.

Wear of parts for a specified period of service shall not cause violations of the nature of its pairing with other items and lead to an unacceptable reduction in its strength. Most of machines (85-90%) break down as a result of wear.

Heat resistance - the ability of components to operate within specified temperature for a specified period of service.

Problems concerning heat resistance are often crucial for machine parts, whose work is connected with high heat generation (output). As the temperature increases the mechanical properties of materials decrease, the viscosity of lubricants reduces, increases wear, the gaps change, the dynamic loads grow.

Vibration resistance is the ability of the structure to work in the desired range of conditions within the allowed fluctuations.

Vibrations reduce the quality of the machine operation, causing additional stress variables in parts, increase the noise. The main task of calculating the vibration is the choice of such structure rigidity, under which there is no danger of resonance occurrence.

In designing of machine parts they perform two types of calculations: projecting - preliminary calculation of the main criteria of efficiency, which takes place with the use of the simplified method for determination of main dimensions of the part. According to the results of design calculations they develop the preliminary version of the parts design; checking calculation is carried out to clarify the conceptual options for necessary (one or two) performance criteria using the refined method of calculation.

1.3. Engineering materials.

When calculating machine parts, one of the most important stages is the choice of materials. The correct choice of material largely determines the quality of parts and machines in general.

When choosing materials for specific parts manufacturing, they need to consider the following factors: correspondence of material properties to the main performance criterion (strength, rigidity, wear resistance, etc.); requirements relating to the function of parts and conditions of its use (anti-corrosive resistance, friction properties, electrical properties and etc.), the cost and scarcity of material.

In engineering they use ferrous (steel, cast iron) and non-ferrous (copper, aluminum) metals, their alloys (bronze, anodized aluminum), non-metallic (plastic, wood, rubber) and combined (metal-ceramic, composite) materials.

When choosing a proper material one has to make a decision on the choice of the method of processing for a particular part (or its segment) that enhances the strength, serviceability and durability of the designed part. To improve the mechanical and other properties of materials there exist the following methods: thermal treatment (annealing, quenching), thermochemical treatment (carburizing, nitriding); mechanical strengthening (cold hardening).

1.4. General information about power transmissions.

Structurally, an unspecified production or transport machine can be divided into three main parts: the engine, transmission gear (power transmission), working organ. The engine and power transmission form the drive - a device for driving the working body (organ) of the car.

The most common are mechanical transmissions of rotational movement, since the rotational movement is easy to make continuous, it’s easier to achieve uniformity, to reduce friction losses. The course "Machine parts" studies only mechanical transmissions of rotary motion, which are simply called transmissions.

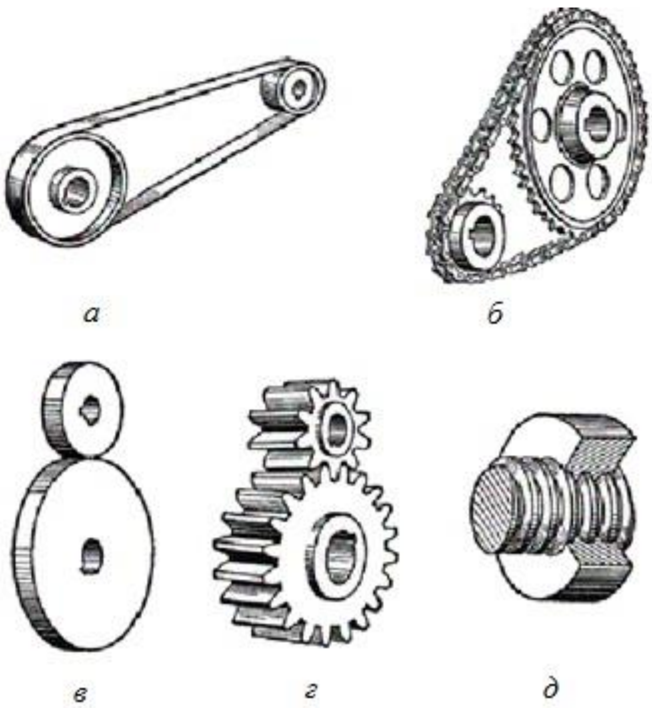


Fig. 1.1.

<u>Transmission</u> is a **mechanism** used for transmitting motion from the motor to the actuator body, as a rule, with rate conversion and the change of torque.

The need to use transmissions is driven by:

• regulation of kinematic and loading parameters;

• transformation of forms of motion (rotary into translator (reciprocating) and vice versa, constant (uniform) into intermittent one);

• provision of a given layout of machines.
Transmitions of rotary motion are divided into the ones with direct contact of rotary bodies and transmissions with flexible coupling, in which the bodies of rotation are connected by a flexible link. Friction transmissions are referred to the first type (Fig. 1.1 in) and gear transmissions (Fig. 1.1 g), and to the second one - belt (Fig. 1.1, a) and chain transmissions (Fig. 1.1, b). Depending on the mode of motion transmission from the driving rotating body to the driven one they distinguish friction transmissions (friction, belt) and gearing transmissions (chain, gear). Screw-and-nut transmissions are also referred to rotary transmissions (Fig. 1.1, d), the purpose of which is to convert rotational motion into linear (translatory/ reciprocating) motion.

In transmissions, links that transmit torque are called driving members (guide links), and taking – driven members (links).

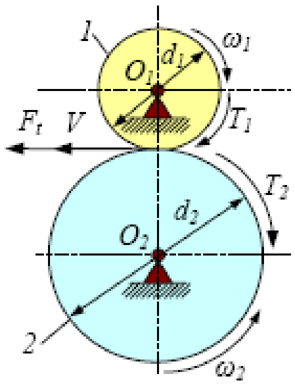


Fig. 1.2

Transmission parameters relating to the driving members are indicated with index *1*, and the driven members – with index *2*. In Figure 1.2: d_1 and d_2 - diameters of the master and slave units (driving members and driven members); $\omega 1$, $T1$; $\omega 2$, $T2$ - angular speed and torque on the driving and driven shafts. The torque on the drive shaft T_1 is the moment of moving forces, its direction coincides with the direction of shaft rotation. The torque on the driven shaft T_2 is the moment of forces of resistance, so its direction is opposite the direction of shaft rotation.

1.5. Basic relations for power and kinematic parameters of transmissions.

The main characteristics required for calculation of any transmission project is output power N_2; rotation speed of the output shaft n_2 and the gear ratio $i = \frac{\omega_1}{\omega_2} = \frac{n_1}{n_2}$, which is the ratio of angular velocities of the master and slave transmission units.

Additional features:

• coefficient of performance (COP) of the transmission $\eta = \frac{N_2}{N_1}$;

• peripheral speed of the master and driven member $v = \omega \cdot \frac{d}{2}$ (m/s);

• circumferential force $F_t = \frac{N}{v}$ (N);

• Torque $T = F_t \cdot \frac{d}{2} = \frac{N}{\omega}$ (Nm).

To determine the power (kW) and rotation speed (1/min) it is convenient to use the following expressions:

$$N = \frac{F_t \cdot v}{1000} = \frac{T \cdot n}{9550}, \tag{1}$$

$$n = \frac{30 \cdot \omega}{\pi} = \frac{60000 \cdot v}{\pi \cdot d}, \tag{2}$$

wherein F_t - is expressed in N; v - in m/s; T - in Nm; ω - in 1/s; d - in mm.

When using a mechanical actuator, consisting of a number of series-connected gears its overall efficiency η is equal to the efficiency factor product of all its gears:

$$\eta = \eta_1 \cdot \eta_1 \cdot \ldots \cdot \eta_n \tag{3}$$

if other devices are part of the drive, where the loss of power is possible (couplings, bearings, ...), this is taken into account when calculating the overall efficiency.The overall gear ratio of the drive i, which consists of several consecutive gears, is equal to the product of all gear ratios of all its gears:

$$i = i_1 \cdot i_1 \cdot \ldots \cdot i_n . \tag{4}$$

Practical training 2. Transmissions with flexible coupling.

2.1. Basic information, design features and kinematics of belt transmissions.

Belt transmissions - are friction transmissions with flexible coupling. The belt transmission usually consists of two pulleys and a drive belt of closed form (Figure 12.1). The load is transmitted by the friction between the pulleys and the belt. To provide the necessary friction, the belt should be tight, which is achieved due to the elastic tension of the belt in its dressing over pulleys or through the use of special tensioners.

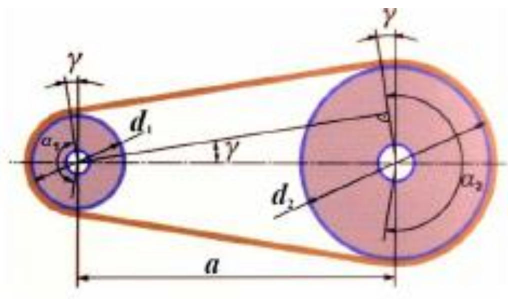


Fig. 2.1.

Belt tension is the main condition of the belt drive (transmission) operation.

Types of belt drives are determined by the shape of the cross section of the belt, the most common of which are the following ones: flat belt (Fig. 2.2, a), V-belt (Fig. 2.2, b), V-ribbed (Fig. 2.2, c), round belt (Fig. 2.2 g) as well as transmissions with timing belt.

a b c d

Fig. 2.2.

In general engineering machinery they use flat and V-belt drives (transmissions).

<u>Belt</u> is a key element of the transmission, which determines its performance. The most common rubber belts with load-bearing elements made of duck (multicord) or cord. The design of such belts is shown in Figure 12.3. Cord belts provide higher transmission efficiency, and they are more flexible and durable.

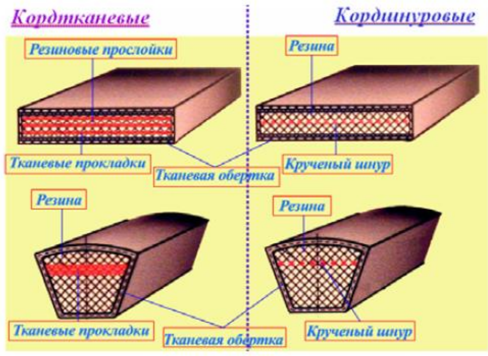


Fig. 2.3.

The main *geometric parameters* of belt drives are the following (Fig. 2.1):

• pulley diameters d_1 and d_2;

• axle spacing a, which is determined by the design of the drive;

• the estimated length of the belt L_p - is the sum of the lengths of the arcs of circumference of pulleys and straight sections of the belt, further according to the determined value out of the standard range they select the nearest large effective length, after selecting takes place, the axle spacing is corrected;

• the angle of wrap on the small pulley is α_1. For flat belt transmissions it is recommended $\alpha_1 \geq 150^\circ$, for V-belt transmissions $\alpha_1 \geq 110^\circ$, with lower values of the wrap angle there may occur reduction of efficiency due to the partial belt slipping under load.

When the belt is in motion in the belt drive they distinguish two types of slip: elastic slip (inevitable during normal transmission operation, occurs due to the difference of forces loading the master and slave loop of the belt) and slippage (complete loss of belt traction with the pulley, which occurs under overloading).

The elastic belt slip characterizes the slip ratio ξ:

$$\xi = \frac{v_1 - v_2}{v_1} \tag{1}$$

wherein v_1, v_2 - the speed of master and slave loops.

Due to the belt slip over the pulleys the *transmission gear ratio* is not constant:

$$i = \frac{\omega_1}{\omega_2} = \frac{v_1 \cdot d_2}{v_2 \cdot d_1} = \frac{d_2}{d_1 \cdot (1-\xi)}\,. \tag{2}$$

Based on the above, we shall note the advantages of belt drives:

• simplicity of design;

• ability to transfer motion over long distances (up to 15 meters);

• ability to work at high speed (high rotation frequency);

• smooth and quiet operation;

• Mitigation of vibration and shock;

• Overload protection of mechanisms at the expense of belt slippage.

The disadvantages of belt drives include:

• large radial dimensions;

• low belt durability;

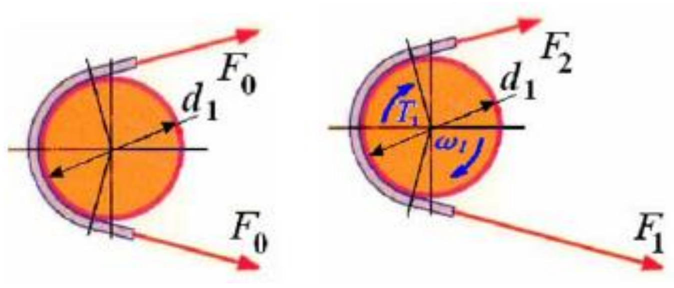
• gear volatility;

• heavy loading of shafts and bearings;

• sensitivity of peripheral environment (oil, moisture, etc.).

2.2. Forces and stresses in the belt.

To create friction between the belt and pulley there is needed the force of belt pre-tension (Fig. 2.4, a). The greater is F_0, the higher is the traction capability of transmission. At rest or idling each loop of the belt is affected only by the pre-tension force F_0.



a b

Fig. 2.4.

When transferring the useful torque T tension in the loops of the belt will change. In the leading loop the tension rises to the power F_1, and in the slave one reduces to F_2 (Fig. 2.4, b). From the equilibrium condition of the driving pulley we can write: $F_1 - F_2 = F_t$, where $F_t = \frac{2 \cdot T}{d_1}$ - circumferential (affective) force. Given the pre-tensioning force, forces in the loops are determined as follows:

$$F_1 = F_0 + \frac{F_t}{2}, \qquad F_2 = F_0 - \frac{F_t}{2}. \tag{3}$$

When the belt follows the pulleys in the belt there arises centrifugal force F_v:

$$F_v = \rho \cdot A \cdot v^2, \tag{4}$$

where ρ - the density of the belt material; A - sectional area of the belt; v - the speed of the belt movement. The influence of centrifugal forces is not usually taken into account at speeds $v \leq 10$ m/s, this mode of operation of belt drives is typical for most general machinery drives.

Force F_1 and F_2 create load on the shafts and bearings, but in practice the load on the shaft is calculated by the approximate formula (error less than 7%):

$$R = 2 \cdot F_0 \cdot \sin\left(\frac{\alpha_1}{2}\right). \tag{5}$$

Stresses $\sigma_0 = \frac{F_0}{A}$, $\sigma_t = \frac{F_t}{A}$, $\sigma_v = \frac{F_v}{A}$, correspond to the forces described above:

$$F_0, F_t, F_v, \tag{6}$$

where A - the sectional area of the belt. In addition to the above-stated stresses in the belt, when it follows the pulley, there additionally arise bending stresses σ_u , which are determined according to the Hooke's law:

$$\sigma_u = E \cdot \frac{\delta}{d}, \tag{7}$$

where $\frac{\delta}{d}$ - the longitudinal strain (deformation) in the belt; δ - the thickness (height) of the belt. Bending stresses do not affect the traction capability of transmission, but varying in zero-to-tension stress cycle, is the main cause of fatigue fracture of the belt.

When the belt drives are in operation, the tension along the length of the belt is distributed unevenly, as it can be seen in the stress distribution diagram (5).

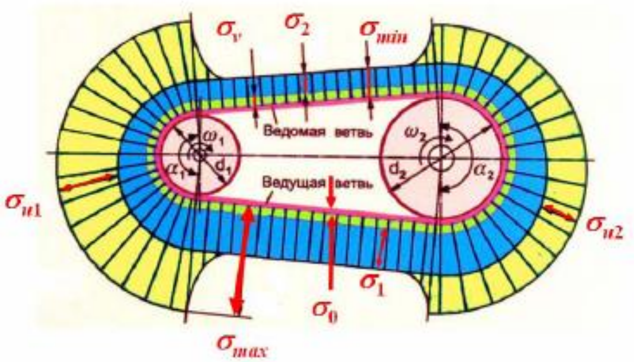


Fig. 2.5.

The maximum stress is observed in the cross-section of the belt in the place where it follows the small pulley (Fig. 2.5).

$$\sigma_{\max} = \sigma_v + \sigma_1 + \sigma_{u1} = \sigma_v + \sigma_0 + \frac{\sigma_t}{2} + \sigma_{u1}. \tag{8}$$

2.3. Performance criteria and calculation of belt transmissions.

The main performance criteria of belting:

<u>towing capacity</u> (the ability of transmission to transfer a given load without slipping);

durability of the belt (belt property to resist fatigue failure).
Traction capability is characterized by experimental curves of relative sliding, combined with the efficiency curves, depending on the loading level of the transmission. The loading level is characterized by the traction factor:

$$\varphi = \frac{F_t}{2 \cdot F_0} = \frac{\sigma_t}{2 \cdot \sigma_0}. \qquad (9)$$

The optimal values of the thrust coefficient for reference: flat-belt transmissions $\varphi_0 = 0,4\ldots0,6$; V-belt transmissions $\varphi_0 = 0,6\ldots0,75$.

Calculation of the belt drive tractive ability is a designing one and comes down to the definition of the calculated cross-sectional area of the belt:

$$A \geq \frac{F_t}{[\sigma_t]}. \qquad (10)$$

In this case, on the basis of expression (9), taking into account the optimal values of the thrust coefficient, the allowed stress value in the belt $[\sigma_t]$ for a particular transmission can be defined as follows: $[\sigma_t] = 2 \cdot \sigma_0 \cdot \varphi_0 \cdot C_\alpha \cdot C_v \cdot C_p \cdot C_\theta$, where C_α - the coefficient of wrap angle by small pulley belt; C_v - the coefficient of the influence of centrifugal forces; C_p - the coefficient of the operating mode ; C_θ - the coefficient which takes into account the transmission mode and the angle of its inclination to the horizon.

When calculating the value of the V-belt (10) the value A in expression (10) is presented in the following form $A = z' \cdot A_0$, where z' - the number of V-belts; A_0 - the sectional area of a V-belt .

Durability of the belt is defined by its resistance to fatigue and depends on the bending stress value σ_u and the number of loading cycles that are proportional to the number of runs of the belt Π.

In calculations of durability (carried out as a testing one) a simplified assessment of belt durability can be carried out based on the limiting condition of the number of its runs within a unit of time:

$$\Pi = \frac{v}{L_p} \leq [\Pi] \qquad (11)$$

where $[\Pi]$- the number of allowed belt runs (for flat belts $[\Pi] \leq 3 \ldots 5 \ c^{-1}$; for V-belts $[\Pi] \leq 10 \ldots 15 \ c^{-1}$).

2.4. Design features, geometry and Kinetostatics of chain drives .

Chain drive is a gear transmission with flexible coupling. It consists of the driving and driven sprockets, enveloped by the chain (Fig. 2.6).

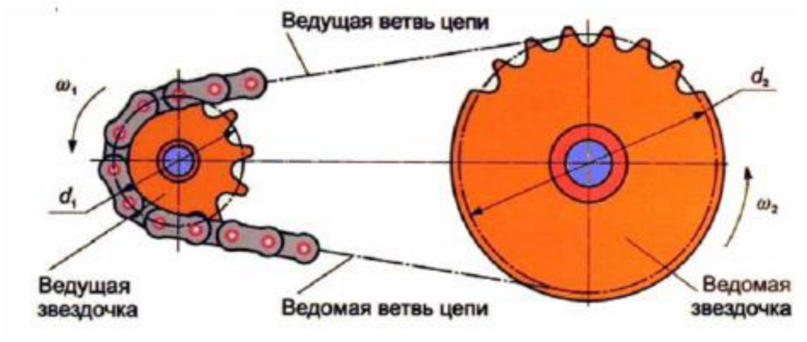
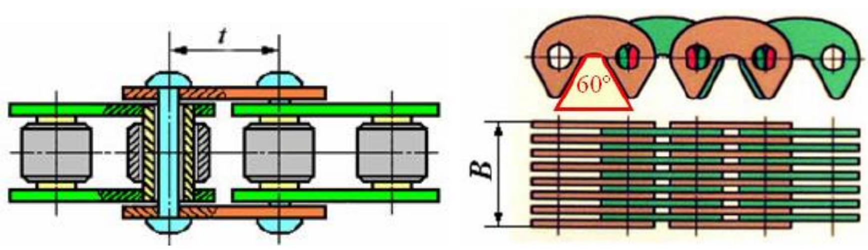


Fig. 2.6.

Chain drives are used in machine tools, transportation, agricultural and other machines to transmit motion between parallel shafts over long distances, when the use of gears is impractical, and the use of belts is impossible.

They distinguish load, traction and driving types of chains. Drive chains are of the following types: sleeve, roller and gear.



a b

Fig. 2.7.

The roller chain is comprised of outer and inner links (Fig. 2.7 a). The external link is composed of two outer plates and rollers, pressed into their holes. The inner link is composed of two inner plates and hubs (mufts) fixedly secured in the openings of inner plates. On the hub pivot there are loose hardened rollers. Rollers, rolling on the sprockets teeth reduce their wear. Roller chains are used at speeds up to 15 m/s.

Sleeve chains have no rollers so they are cheaper and easier than the roller ones, but their wear resistance is lower. Sleeve (bush) chains are used in non-chargeable transmissions at speeds up to 10 m/s.

Roller and sleeve chains can be of single-row or multi-row type. The use of multi-row chains significantly increases the traction capability of the transmission.

For high-speed transmissions of high power they use gears. The links of the gear chain (Fig. 2.7, b) consist of a set of hinge-connected bidentate plates. The working faces of plates are arranged at the angle of 60 °. The number of plates is determined by the width of the chain B, which depends on the transmitted power. Toothed (gear) chains are used with considerable speeds (up to 35 m/s).

At present, in general engineering they more often (more than 90° of the number of chain transmissions) use roller chain transmissions.

The main geometrical parameters of chain drives include the step of the chain t (Fig. 2.7, a) and the width of the chain B (Fig. 2.7, b), pitch circle diameters of sprockets d_∂, which pass through the centers of chain hinges (Fig. 12.6), spacing a (optimal values are recommended to be selected within the range of $a = (30\ldots50)\cdot t$), the chain length L_t (expressed in meters or in the number of units).

The main difference of the chain movement from the belt is that the chain does not cover the circle but the polygon as a geometric model of the sprocket. While turning it, the peripheral speeds of the points are different.

The analysis of this feature suggests that the decrease in the number of sprocket teeth increases the negative kinematic and dynamic features of the transmission chain operation. On the other hand, increase in the number of teeth enhances smooth operation, but as the wear increases, the chain pitch increases as well, and its hinges ascend along the sprocket tooth profile for larger diameter, which can cause popping of the chain. To resolve the above conflicts they focus on operating experience – the number of teeth of the drive sprocket z_1 is selected according to the selected gear ratio, and the number of teeth of the driven sprocket is limited : $z_2 \le 120$.

In practice, the volatility of kinematic characteristics of the transmission chain is not considered, and calculations are carried out according to the average circumferential speed of the chain 30…35 *м/с*.:

$$v = \frac{z_1 \cdot n_1 \cdot t}{60000}. \tag{12}$$

Chain speed is limited by the durability of hinges, strike force in driven engagement and noise. Chain transmissions (drives) with peripheral velocity $v \le 15$ m/s are most widely used, but with careful installation and proper lubricating performance of the drive is provided at the speed of 30…35 m/sec.

<u>The gear ratio</u> of the chain drive taking into account (12):

$$i = \frac{\omega_1}{\omega_2} = \frac{n_1}{n_2} = \frac{z_2}{z_1}. \tag{13}$$

The gear ratio is limited by the size of the drive, the diameter of the driven sprocket, the chain angle of small sprocket wrap and they recommend $i \le 7$.

When the chain drive is in operation they distinguish the following forces acting on the chain:

- <u>circumferential force</u> transmitted by the chain $F_t = \frac{1000 \cdot N}{v} = \frac{2 \cdot T}{d}$;
- <u>pre-tensioning force</u> of the chain from sagging of the return run (slack side) $F_q = K_f \cdot q \cdot g \cdot a$, where K_f - the coefficient of slack chain; q - the weight of one meter of chain (kg); g - acceleration due to gravity (*m/s*^{2}); a - axle spacing (*m*);
- <u>chain tensioning force due to the centrifugal effect</u> $F_v = q \cdot v^2$.

The total tension experienced respectively by the active and the return run of the operating chain transmission:

$$F_1 = K_1 \cdot F_t + F_q + F_v, \qquad F_2 = F_q + F_v, \tag{14}$$

where K_1 - the coefficient, which takes into account the nature of the load.

The load, acting on the gear shaft inclined to the horizon to *40*$^\circ$, is approximately calculated as follows:

$$R = (1{,}15\ldots1{,}20)\cdot F_t\,. \tag{15}$$

The advantages of chain drives include the following:

• setting in motion several shafts by one chain;

• the ability to transfer motion over long distances (up to 8 m);

compared with belt drives:

• compactness;

• ability to transfer high power;

• smaller radial load on the shaft ;

• consistency ratio.

The disadvantages often include:

• Significant noise at operation;

• poor performance at high speeds;

• rapid wear of chain hinges;

• chain elongation in wear and its escape from sprockets.

2.5. Calculation of chain drives.

The main criterion of efficiency for the majority of chains is durability of their hinges. According to this, the calculation of chain drives is made according to the condition in which the pressure in the joints must not exceed the permissible value.

At design calculation of the roller chain the carrying capacity of the chain is determined by the condition: the average calculated pressure in the joint of the chain link p, when the transmission is in operation, must not exceed the permissible $[p]$:

$$p \le [p]. \qquad (16)$$

In this case, the design pressure in the joints is defined as $p = K_E \cdot \dfrac{F_t}{K_m \cdot S_{ОП}}$, where K_E is the service factor; K_m - the factor that takes into account the number of rows of the chain; $S_{ОП}$ - the bearing surface of the hinge, which in turn is expressed in terms of the pitch chain square $S_{ОП} = S_t \cdot t^2$, for driving roller chains $S_t \simeq 0.28$. Considering formula (12.16) and (12.12) the design (calculated) chain pitch value is determined by the formula:

$$t \ge 183 \cdot \sqrt[3]{\frac{N \cdot K_E \cdot 10}{K_m \cdot S_t \cdot z_1 \cdot n_1 \cdot [p]}}. \qquad (17)$$

The determined value of the chain pitch is made consistent with the standard one, and they select a chain.

When carrying out checking calculations they verify the life (durability) of the transmission and its safety factor.

At design calculation of the toothed chain they determine the working width of the chain according to the load:

$$B = \frac{0,25 \cdot F_t \cdot \sqrt[3]{v}}{t} \cdot K_E. \qquad (8)$$

In the course of verification calculation they check the margin of safety of the chain.

Practical training 3. Gears.

3.1. General information and classification of gears.

Gears with round wheels are the most common mechanical transmissions in modern engineering. A simple gear train comprises two toothed wheels, whereby the engagement of the driving and driven wheels takes place. The driven gear of the transmission is called pinion, the driving one - wheel. The term "gear" refers to the wheel and pinion.

Gears are classified:

1. Depending on the nature of motion conversion:
• rotational movement of the drive wheel is converted into rotational motion of the driven one (Fig. 3.1, b);

• the rotational movement of the drive wheel is converted into linear (translatory) motion of the rail (Fig. 3.1, e).

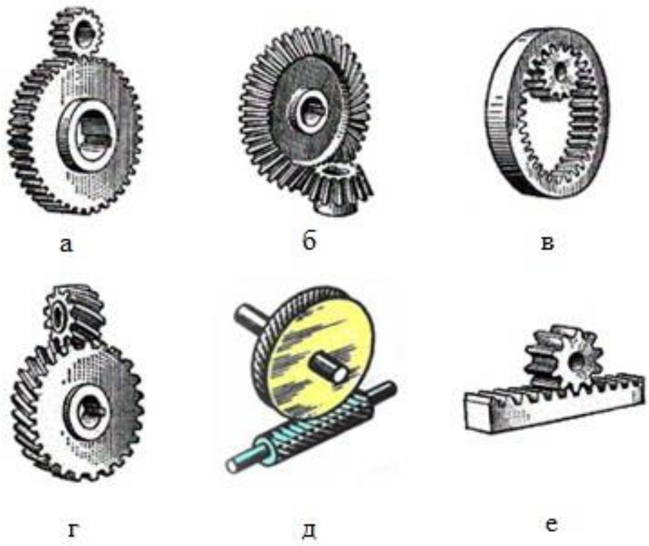


Fig. 3.1.

The rail is considered as an infinitely large diameter gear. Gears are used not only as a pair of gears, but in more complex combinations. In the form of planetary gears consisting of toothed wheels with moving geometric axes, and in the form of transmission waves, in which one of the wheels is a flexible crown.

2. Depending on the relative position of geometrical axes of shafts:

• parallel axes (spur gear (Fig. 3.1 , a));

• intersecting axes (bevel gear (Fig. 3.1, a) - the crossing angle *90*˚);

• intersecting axes (screw gear (Fig. 3.1 g), worm gear (Fig. 3.1 , d)).

The most common are cylindrical and bevel gears, while cylindrical ones are easier to manufacture and assemble. In bevel gears at work there are significant axial forces. Helical (screw) and worm gears, compared with cylindrical and conical ones, have greater smooth running characteristics and the ability to disengage both shafts beyond the transmission in both directions, but their efficiency is lower and the teeth wear out faster due to increased slip. The bearing capacity of helical gears is small, high load-bearing capacity have worm gears, whereupon they are more widely used.

3. Depending on the relative position of the wheels in space:

• external engagement;

• internal engagement (Fig. 3.1 in).

Although the internal engagement is more compact than the external one, but its manufacture and installation is more complex, and therefore transmissions with external gears are more common.

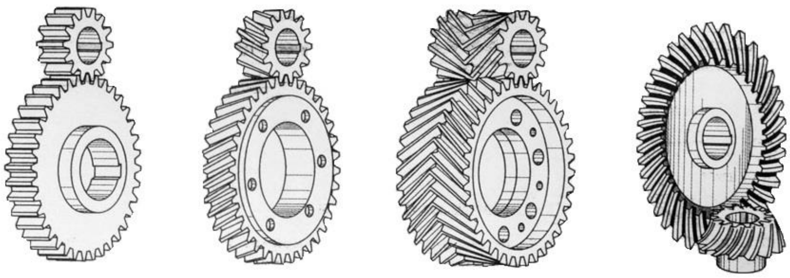
4. Depending on the design (version):
• open transmissions;
• closed transmissions.

In open transmissions gears teeth are not protected from the external environment, such transmissions are used in a manual and low-speed mechanical drive. Setting of transmission in a separate housing guarantees the accuracy of the assembly, better lubrication, higher efficiency, less wear and tear, as well as protection from dust and dirt penetration. Transmission placed in a separate housing and designed to reduce the angular speed and increase the torque on the driven shaft is called gear reducer (speed changer). It is designed to increase the angular velocity of the driven shaft – by means of multiplier.

5. Depending on the number of stages:
• one-stage (one pair of wheels in the engagement);
• multistage (two pairs of gears in the engagement or more).
6. Depending on the location of teeth on the rim of the wheel:

• spur gear (Fig. 3.2 , a);
• helical bevel gear (Fig. 3.2 , b);

• chevron (Fig. 3.2, c);
• with circular teeth (Fig. 3.2, d).



a b c d

Fig. 3.2.

When a spur transmission is in operation a pair of teeth engages along the whole length of the contact that is accompanied by a stroke of the teeth and increased noise, therefore spur transmissions are used at low circumferential speeds. When the helical gears are in operation, the teeth are not immediately engaged over the entire length, but gradually. Advantages: a smooth transmission operation, reduction of dynamic loads, noise reduction. Among disadvantages is the presence of the axial force that additionally puts pressure on shaft bearings. To reduce the axial forces, the helix angle is recommend to be manufactured within the following limits: $\beta = 8 \ldots 20^{\circ}$. Spur and helical bevel gear as well as circular teeth transmissions are used in critical applications at medium and high speeds (15 *м/с*). In a chevron gear, the axial forces on semi-chevrons are opposing, due to this they balance each other and are not transmitted to the supports. The equilibrium of axial forces at chevron gears can increase the helix angle to 40˚, which increases both the load and smooth operation capacity of the transmission.

Chevron transmissions are usually used at high loads and severe operating conditions, at medium and high peripheral speeds.

7. Depending on the shape of tooth profile:
• involute (profile generator – tooth profile (involute)) (Fig. 3.3, a);
• cycloid (profile generator - cycloid) (Fig.3.3, b);
• With Novikov’s gearing (profile generator - arc) (Fig. 3.3, c) .

a b c

Fig. 3.3.

The cycloidal tooth profile is mainly used in clock mechanisms, as it allows getting the wheel with a small number of teeth (up to 5-6). The Novikov's gearing is used in the high-gear, which, for structural reasons, is supposed to have a small size. In practice, they mainly use an involute tooth profile, ensuring their durability, low slip rate in the area of engagement and high efficiency.

The main advantages of gear transmissions are as follows:
• small size;

• constancy of gear ratio;
• higher durability and reliability;
• high efficiency
• ease of use.
The disadvantages include:
• Noise at high speeds;
• high demands on precision of manufacturing and assembly.

3.2. Fundamentals of the theory of gearing.

The kinematic condition of gear teeth profiles fitness is the constancy of the gear ratio.

For this it is necessary to perform the basic law of engagement: to maintain constant transmission ratio it is necessary and sufficient that the normal *NN* to the profiles at the point of contact always cross the center line O_1O_2 at the same point *P*, called the pole of engagement. This point divides the axle spacing into the parts inversely proportional to the angular velocities of gears (Fig. 13.4):

$$i = \frac{\omega_1}{\omega_2} = \frac{O_2P}{O_1P} = const\,. \qquad (1)$$

The basic law of gear (tooth) engagement is satisfied with a lot of curves, but in most cases the teeth are profiled on a curve called the involute. Involute is a curve described by a point lying on the straight line which is rolled circumferentially without slipping. The rolled line is called the rgenerating line and the circle - the main circle or the evolute.

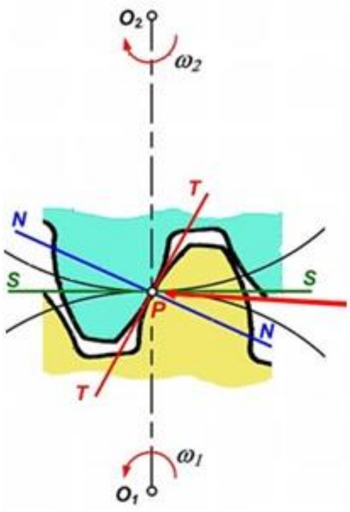


Fig. 3.4.

3.3. The main elements of involute gears.

Let's consider the basic geometric and kinematic parameters of standard involute gears (Fig.3.5).

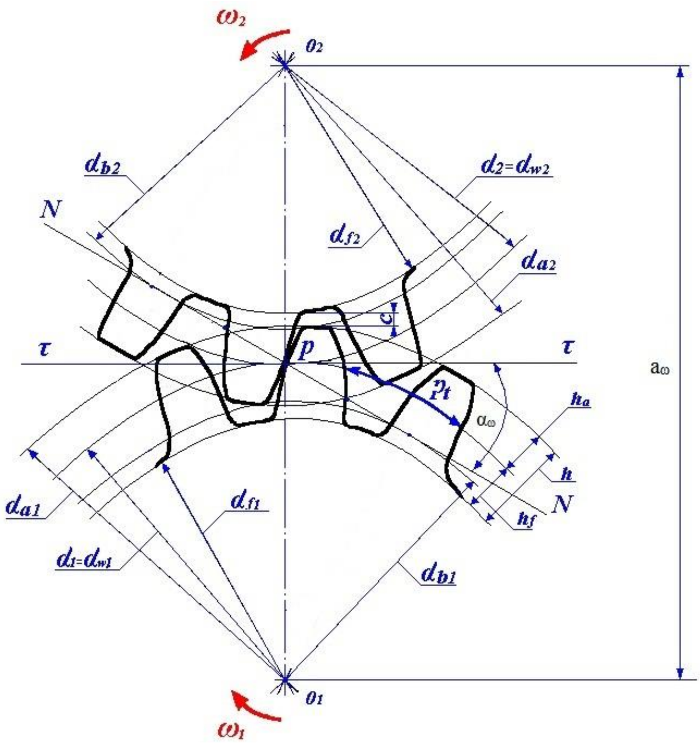


Fig. 3.5.

$d_{\omega1}$, $d_{\omega2}$- pitch diameters of both the gear and the wheel. The initial are called circles, which in the process of engagement are rolled against one another without slipping. To the extent that a_ω - the center distance changes, the pitch diameters will change as well. Thus, the pair of gears can have a set of pitch circles, while an individual wheel does not have a pitch circle, because it is the kinematic concept of a wheel pair.

d_1, d_2- pitch diameters of gears and wheels. Pitch diameters are chosen as the basis for determining the main dimensions of the wheel, they divide a tooth into a tooth crown h_a and a tooth flank h_f. In contrast to initial diameters, pitch circles - are real geometric configurations of individual gears. In most gears, except corrected entanglements, initial diameters and pitch circles coincide (are the same), in which case spacing could be defined as:

$$a_\omega = \frac{d_{\omega 1} + d_{\omega 2}}{2} = \frac{d_1 + d_2}{2}. \qquad (2)$$

d_{b1}, d_{b2} - diameters of the base circle. With the increase in the diameter of the base circle d_b the curvature of the involute is reduced and in case if $d_b \to \infty$ a toothed profile is transformed into a rail with a trapezoidal profile - the main rail. The tooth profile of the main rail matches the original contour of the tooth, which is standardized. This contour is the basis for profiling tools for gear cutting.

d_{a1}, d_{a2} - diameters of top circles of both the gear and the wheel.

d_{f1}, d_{f2} - dedendum circles (root circles) of both the gear and the wheel.

Line *NN*, on which the point of contact of conjugate profiles moves, is called the line of engagement, it makes with the perpendicular to O_1O_2 the pressure angle α_ω. In accordance with the standard $\alpha_\omega = 20^\circ$.

The distance P_t between the corresponding points of adjacent teeth profiles measured along the arc of the pitch circle is called a circular gear pitch (equal to the initial spur gear rail step). From the definition of the step it follows that the length of the pitch circle is equal to $\pi \cdot d = P_t \cdot z$, where the circular gear pitch:

$$P_t = \frac{\pi \cdot d}{z} \qquad (3)$$

where z - the number of teeth on the wheel.

For ease of measurement and calculation they introduce the concept of teeth module:

$$m = \frac{P_t}{\pi}. \qquad (4)$$

Module is the main feature of gearing. To ensure the interchangeability of gears and gear cutting tool standardization, values m are standardized in the range from

0.05 to 100 mm. For a pair of meshed gears the circumferential step P_t, and, hence, the module m should be the same.

All geometric parameters of gears are expressed through the module m. The height of the head and the leg of the tooth of a normal (uncorrected) link is determined as follows:

$$h_a = m; \qquad h_f = h_a + c = m + c, \qquad (5)$$

where c - the radial gap in the link of gears and wheels .

Theoretical thickness of teeth and the width of cavities on the pitch circle are equal. However, to create a lateral clearance required for normal gear pair operation, the tooth is made somewhat thinner so that it enters the cavity freely.

The diameters of the initial, pitch circles of peaks and troughs of teeth of a normal (uncorrected) link is determined by the following formulas:

$$d_\omega = d = m \cdot z, \qquad (6)$$

$$d_a = d + 2 \cdot h_a = m \cdot (z + 2), \qquad (7)$$

$$d_f = d - 2 \cdot h_f = m \cdot (z - 2{,}5) \qquad (8)$$

<u>The arc of engagement</u> is called the path traveled by the spline on the pitch circle during the actual time of its engagement S. A necessary condition for engagement is a requirement that the arc of engagement must be bigger than the step of engagement P_t . The ratio of the length of the arc of engagement to the step is called the <u>overlap ratio</u> $\varepsilon = \frac{S}{P_t}$, it characterizes the average number of pairs of teeth simultaneously meshing (engaged). For cylindrical gears $\varepsilon \geq 1{,}2$ - this means that a pair of teeth is 80 % of time involved in engaging a pair of teeth, the remaining 20 % - two pairs of teeth. The overlap factor characterizes the transmission smoothness and must always be greater than a unit.

Practical training 4. Gears (continued) .

4.1. Geometrical parameters of gears.

The gear has the following features: crown - part of a gear having teeth; hub - part of the gear slipped over the shaft, drive - part of the gear between the hub and the rim.

In a meshed spur the line length of contact is equal to the width of the crown b. The main geometrical parameters of spur gears were discussed in Section 3.3.

In helical wheels the teeth are inclined at the angle β to the axis of the wheel (Fig. 4.1). A variety of helical gears are chevron (herringbone) gears (wheels) which are two helical wheels with matched (but) ends so that the teeth are inclined in opposite directions.

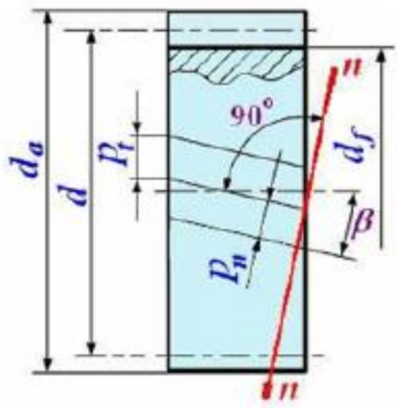


Fig. 4.1.

In helical gears they distinguish two pitches and modules of teeth: in the normal section n-n is the normal tooth pitch p_n and the normal module m_n; in the end section – the circular pitch p_t and circular module m_t.

$$p_t = \frac{p_n}{\cos(\beta)}, \qquad m_t = \frac{m_n}{\cos(\beta)}. \tag{1}$$

Since the helical gears are cut by the same standard tool as the spur ones, the slanting tooth profile in the normal section coincides with the straight tooth profile of module m.

The diameters of initial, pitch circles of peaks and troughs of teeth of normal (uncorrected) cylindrical helical gears is determined by the following formulas:

$$d_\omega = d = \frac{m_n \cdot z}{\cos(\beta)}; \tag{2}$$

$$d_a = d + 2 \cdot h_a = m_n \cdot \left(\frac{z}{\cos(\beta)} + 2 \right); \tag{3}$$

$$d_f = d - 2 \cdot h_f = m_n \cdot \left(\frac{z}{\cos(\beta)} - 2,5 \right). \qquad (4)$$

The analogue of initial and pitch cylinders of cylindrical gears in bevel gears are the initial and pitch (reference) cones. Further we'll consider the bevel gears, for which the axial angle of wheels is $\Sigma = \delta_1 + \delta_2 = 90^\circ$ (see Fig. 4.2), where δ_1, δ_2 are pitch cone angles.

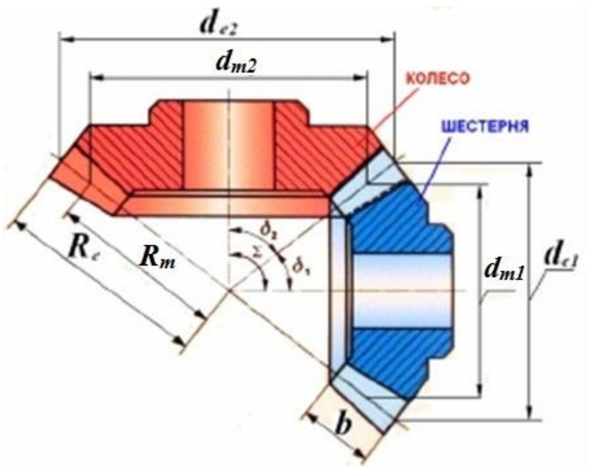


Fig. 4.2.

Since the module of bevel gears teeth in different normal sections are not the same, then there are two pitch modules: m_{te} - external pitch (peripheral) module (in the normal external section); m_m - the average pitch module (in the middle section). For a standard module they take the external peripheral (pitch) module m_{te}.

External and average pitch diameters are expressed as follows:

$$d_e = m_{te} \cdot z; \qquad (5)$$

$$d_m = m_m \cdot z = d_e - b \cdot \sin(\delta), \qquad (6)$$

where b - the width of the bevel gear toothing.

The outer and average cone distance

$$R_e = 0,5 \cdot \sqrt{d_{e1}^2 + d_{e2}^2} = 0,5 \cdot m_{te} \cdot \sqrt{z_1^2 + z_2^2}, \qquad (7)$$

$$R_m = R_e - 0,5 \cdot b. \qquad (8)$$

The angles of pitch cones

$$tg(\delta_1) = \frac{z_1}{z_2}; \qquad \delta_2 = 90^\circ - \delta_1. \qquad (9)$$

The geometrical dimensions of the worm and worm wheel transmission (Fig.14.3) are determined by formulas similar to the ones used for gears.
Worm is a short screw with a trapezoidal thread. Depending on the shape of the outer surface they distinguish cylindrical worms (screw generator - a straight line, such worms are easier to manufacture); globoid worms (have a large load carrying capacity).

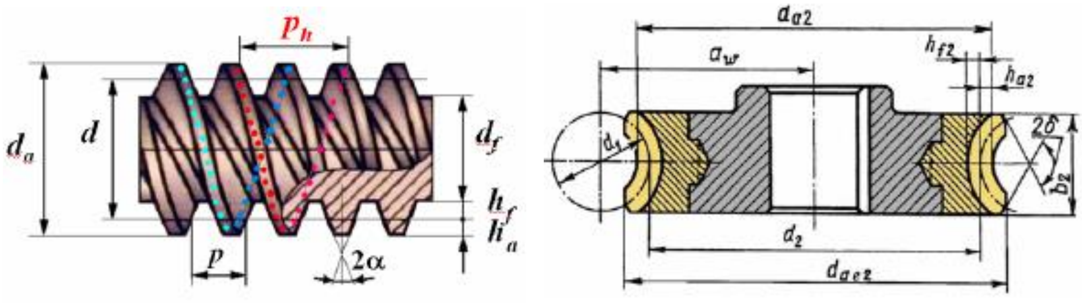


Fig. 4.3.

According to the number of worm threads (entries) they distinguish: single-thread ($z_1 = 1$); double-thread ($z_1 = 2$); quadrifilar ($z_1 = 4$) worms.

The lead of the worm helicoid p_h is the distance between the adjacent sides of the same thread. The axial pitch p is the distance between the lateral sides of two adjacent profiles measured parallel to the axis of the worm, here $p_h = p \cdot z_1$.

α - the angle of thread profile in the axial section ($\alpha = 20^\circ$).
In the worm gear the axial module is m, which is equal to the end module of the worm gear.
Pitch diameters of the worm and wheel:

$$d_1 = m \cdot q; \qquad d_2 = m \cdot z_2, \qquad (10)$$

where q - the worm diameter factor (the number of modules in the pitch diameter of the worm).
The diameters of pits and peaks of both the worm threads and gear teeth:

$$d_f = d - 2,4 \cdot m; \qquad d_a = d + 2 \cdot m. \qquad (11)$$

Center distance:

$$a_\omega = 0,5 \cdot m\left(z_2 + q\right). \qquad (12)$$

4.2. Kinematic and force calculations.

The estimated peripheral (pitch) speed of the cylindrical gear transmission is $v = \omega \cdot \frac{d_\omega}{2} = \omega \cdot \frac{m \cdot z}{2}$. Given that the speed of pitch circles points of meshing gear wheels are identical, we have:

$$v = \omega_1 \cdot \frac{m \cdot z_1}{2} = \omega_2 \cdot \frac{m \cdot z_2}{2}. \qquad (13)$$

Thus, the gear ratio is $i = \frac{\omega_1}{\omega_2} = \frac{z_2}{z_1}$. Further, we'll call the ratio of gear teeth - <u>the drive ratio (gear reduction rate)</u>, and denote as

$$u = \frac{z_2}{z_1}. \qquad (14)$$

When the gear drive is in operation, the load occurring between the enmeshed gear teeth at the pitch point distributed on the contact area (Hertzian flattened band) is replaced by the resultant force F_n directed along the common normal to the profiles of the teeth. For calculation of shafts (rods) and bearings (supports) the force F_n is resolved into components - the <u>circumferential</u> F_t, <u>radial</u> F_r and <u>axial</u> F_a forces.

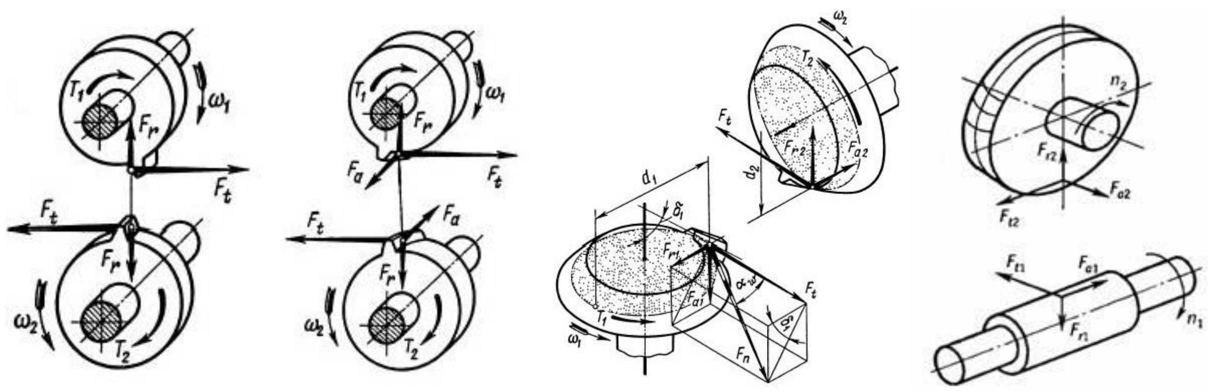
The circumferential force acts tangential to the pitch circle and is determined by the torque and initial diameter of gears:

$$F_t = \frac{2 \cdot T}{d_{\omega 1}}. \qquad (15)$$

For the pinion (tooth gear) the force F_t is a reaction on the part of the driven wheel and is directed opposite the rotation, for the wheel F_t - the force contributing to the movement, it is directed towards the rotation.

The radial force F_r is always directed radially towards the center of the tooth gear, and the axial force F_a - parallel to the wheel axis to the middle of the tooth.

The values of radial F_r and axial F_a forces are shown below for various types of gearing (Fig. 4.4).



a b c d

Figure 4.4.

For spur gear trains (Fig. 4.4, a):

$$F_r = F_t \cdot tg\left(\alpha_\omega\right); \qquad F_a = 0 \tag{16}$$

For helical gears (Fig. 4.4, b):

$$F_r = \frac{F_t}{\cos\left(\beta\right)} \cdot tg\left(\alpha_\omega\right); \qquad F_a = F_t \cdot tg\left(\beta\right). \tag{17}$$

For bevel spur gears (Fig.4.4 c)

$$F_{r1} = F_{a2} = F_t \cdot tg\left(\alpha_\omega\right) \cdot \cos\left(\delta_1\right); \qquad F_{a1} = F_{r2} = F_t \cdot tg\left(\alpha_\omega\right) \cdot \sin\left(\delta_1\right). \tag{18}$$

For worm gears (Fig. 4.4, d):

$$F_{r1} = F_{r2} = F_{t2} \cdot tg\left(\alpha\right); \; F_{a1} = F_{t2}; \; F_{a2} = F_{t1}. \; (19)$$

4.3. Performance criteria and types of tooth gear damage

.

When the two wheels are in contact (enmeshed, engaged), as it was mentioned above, there occurs the normal pressure force F_n acting along the line of contact *NN*, and the friction force $F_f = f \cdot F_n$, wherein f is the coefficient of friction. Under the forces F_n and F_f the tooth is in a precarious state of stress. The main stresses that determine the performance of the gearing are the contact stresses σ_H and bending stresses σ_F. For each tooth they vary in time according to a certain pulsating cycle. The picture of the tooth loading during its operation is shown in Figure 4.5.

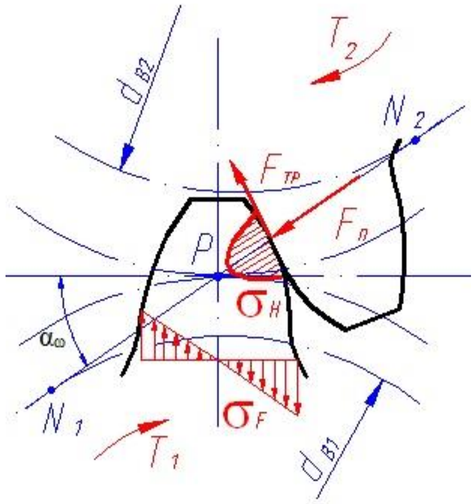
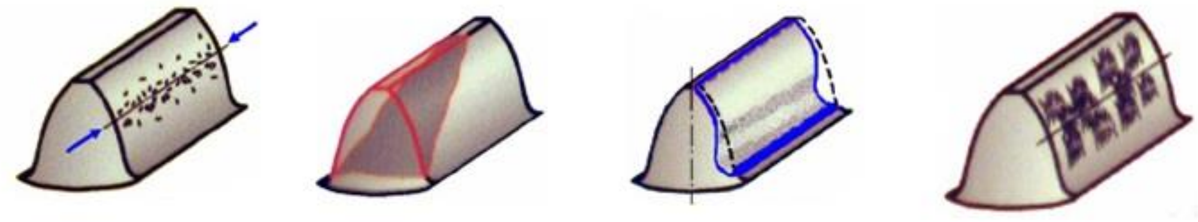


Figure 4.5.

The main elements that determine the efficiency of the transmission are the teeth of the wheels. Re - variable stresses and friction cause the following tooth breakage characteristics:

• fatigue spalling of the surface layers of teeth (characteristic of closed well-lubricated gears).
Spalling begins near the pole line and is a consequence of re-variable contact pressure action (Fig. 4.6, a);
• breakage of teeth (typical for high load fine-grained transmission). Straight teeth are destroyed on the cross section at the base of the tooth, helical - on oblique plane (section). Breakage is a consequence of the re-variable bending stress action σ_F or overload (Figure 4.6, b);

• abrasion of tooth flanks (characteristic of open transmissions). The original tooth profile is distorted, the cross-section of teeth is reduced. Abrasive wear occurs when abrasive particles get into the transmission (Fig. 4.6 in);
• seizing of the teeth surface (characteristic of high load gears operating at high specific loads). As a result of high pressure there occurs oil film rupture. Particles of one tooth material weld to another tooth. Welded particles form build-up, which would damage the surfaces of teeth (Fig. 4.6 g).



a b c d

The breakdown of teeth is prevented by technological and operational measures (increase in precision of machining and assembly, rational choice of materials and heat treatment, selection of lubricants, etc.). Besides, transmission performance depends on correct calculation and design of gears.

As noted earlier, the nature of damage and destruction of teeth affect the working conditions of the transmission. For example, in closed transmissions, working with copious lubrication, the primary damage is fatigue spalling of working surfaces due to contact stresses σ_H, therefore closed transmissions are calculated for contact fatigue and tested for flexural strength. Open gears operating without lubrication and unprotected from the environment, break down as a result of abrasion, resulting in damage of teeth. Thus, the main criterion of the open gears performance is the flexural strength of teeth.

4.4. Principles of tooth gears calculations.

At calculation of cylindrical gear teeth for contact strength the tooth contact is considered at the point of contact P, as the contact of two cylinders with radii ρ_1 and ρ_2, equal to the radii of curvature of the involute at the point of contact.

The highest surface stress in the area of gearing with line contact is given by Hertz formula. For steel wheels with Poisson's ratio $\mu_1 = \mu_2 = 0,3$ the expression has the following form:

$$\sigma_H = 0,418 \cdot \sqrt{\frac{q \cdot E_{\mathrm{Pr}}}{\rho_{\mathrm{Pr}}}}. \qquad (20)$$

where q - normal load per unit of length of contact lines; $E_{\Pr}$ - the reduced modulus of elasticity of wheels material; $\rho_{\Pr}$ - the reduced radius of teeth curvature.

By introducing into the formula (14.20), the coefficients that take into account: the geometrical parameters of the transmission, the properties of the wheels material, uneven and dynamic loads, they obtain the formula for determining $d_{\omega 1}$:

$$d_{\omega 1} = \sqrt[3]{\frac{2 \cdot T_1 \cdot K_{H\alpha} \cdot K_{H\beta} \cdot K_{Hv} \cdot \left(Z_H \cdot Z_M \cdot Z_\varepsilon\right)^2}{\Psi_d \cdot \left[\sigma_H\right]^2} \cdot \frac{(u \pm 1)}{u}}\,. \qquad (21)$$

This formula is used for design calculation of closed cylindrical gears with steel wheels.

After clarification of the initial diameter of the gear they perform verification calculation of the transmission:

$$\sigma_H = Z_H \cdot Z_M \cdot Z_\varepsilon \cdot \sqrt{\frac{2 \cdot T_1 \cdot K_{H\alpha} \cdot K_{H\beta} \cdot K_{Hv}}{b \cdot d_{\omega 1}{}^2} \cdot \frac{(u \pm 1)}{u}} \le \left[\sigma_H\right].$$

(22)

The value $\left[\sigma_H\right]$ is determined according to the contact endurance limit of teeth surfaces $\sigma_{H\lim}$ taking into account the influence on the contact strength: transmission resource, roughness of teeth surface, speed of transmission operation and safety margin. Contact stress $\left[\sigma_H\right]$ for spur transmissions is calculated for gears and wheels and as a calculation one they accept the smallest of them. At calculation of helical gears in which gear teeth have higher hardness than the teeth of the wheel, the calculation contact stress is: $\left[\sigma_H\right] = 0,45 \cdot \left(\left[\sigma_{H1}\right] + \left[\sigma_{H2}\right]\right)$.

When calculating the gearing for flexural strength, the tooth is treated as a cantilever beam loaded with a concentrated force F_n. The force F_n is transferred through the contact line to the axis of the tooth and the resulting point is taken as the vertex of the parabola, which determines the contour of the beam. In determining the normal stress in a dangerous section they use the formulas of strength of materials, taking into account the stress concentration caused by the

particular form of teeth.
Conditions of strength according to bending stress:

$$\sigma_F = Y_F \cdot Y_\beta \cdot Y_\varepsilon \cdot \frac{F_t \cdot K_{F\alpha} \cdot K_{F\beta} \cdot K_{Fv}}{b \cdot m} \leq [\sigma_F]. \qquad (23)$$

The value $[\sigma_F]$ is determined by the endurance limit under bending [σ] Flim taking into account the influence of the transmission resource durability, the surface roughness of intertooth space, and the reversibility of transmission as well as the margin of safety.

Assumptions for stress calculation of bevel and worm gears are similar to those used at calculation of cylindrical gears. For worm gears they additionally perform thermal analysis, because their operation is accompanied by the release of large amounts of heat. With insufficient heat dissipation the properties of lubricating oils deteriorate, there is a risk of jamming and premature transmission failure.
Thermal design is based on the heat balance. The amount of heat released in the gearbox Q_1 is equal to the amount of heat released by the gear-box casing into the environment Q_2. Thus $Q_1 = N_1 \cdot (1 - \eta)$, $Q_2 = K_t \cdot S \cdot (t_M - t_{oc})$, where K_t is the heat-transfer coefficient; S - the cooling area of the gear casing; t_M - oil temperature; t_{oc} - ambient temperature.

If $Q_1 \triangleright Q_2$, it is necessary to provide, for example, excess heat elimination by finning the gear casing, artificial refrigeration (ventilation, additional oil circulation).

Practical training 5. Shafts and axles. Bearings.

5.1. Shafts and axles, general information.

Gears, pulleys, sprockets and other rotating parts of machinery are mounted on shafts or axles.

Axles are only intended to support parts mounted on them. They take the load of the components located on them and work in bending, do not transmit the torque and, therefore, do not experience torsion. The axles are rotating (wagon axle) and fixed ones (the axis of the lifting machine block).

Shafts are used to support mounted parts and torque transmission. Shafts experience bending and torsion, in some cases - tension and compression. Some shafts do not support the rotating parts (cardan shafts, connecting mill rolls, etc.), so that these shafts are only in torsion. Such shafts are called torsion shafts or torsion bars.

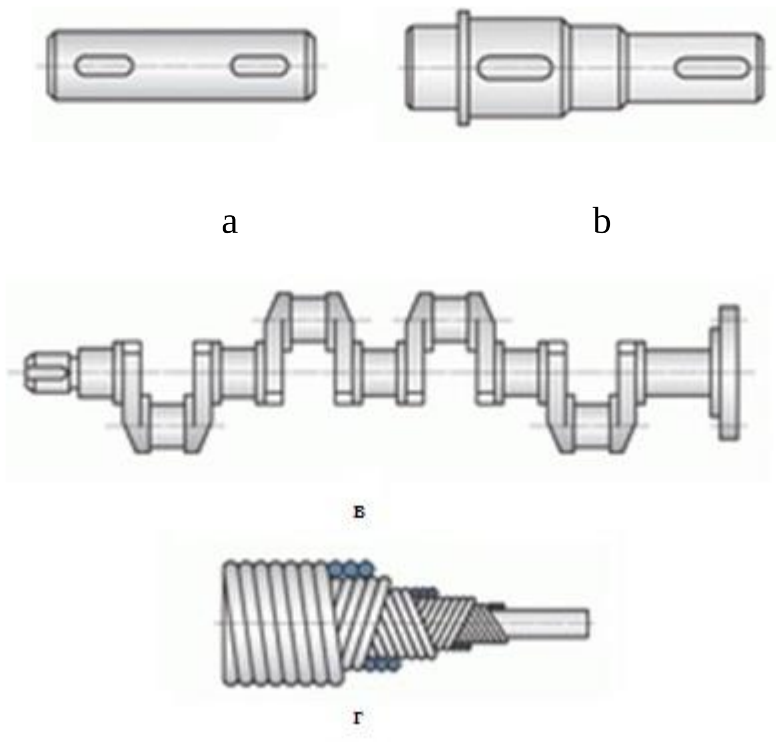


Fig. 5.1.

The shafts are of the following types: direct (Fig. 5.1 , b), crankshafts (Fig. 5.1 in) and flexible (Fig. 5.1 g). The most common are straight shafts. Crankshafts are used for converting reciprocating motion into rotary motion or vice versa. Flexible shafts are multi-start torsion springs of twisted wires used to transmit torque between the units of vehicles changing their relative position when in operation (power tools, dental drills, etc.). Bent shafts and flexible shafts are referred to special parts and are studied in relevant special courses. Depending on the surface of straight shafts they distinguish direct smooth shafts (Fig. 5.1, a) and direct stepped shafts (Fig. 5.1, b). Stepped shape helps separate areas of equal intensity, facilitates installation of parts on the shaft.

The bearing portions of the shaft or axle are called journals at radial loads sensing, or gliders under axial loads sensing. The end pins (journals), which receive the radial load, are referred to as spikes, and pins located at some distance from the end of the shaft - tails. Spikes and neck shaft or axle supported by bearings, bearing part of the heel is the thrust bearing.

Collar is referred to an annular thickening of the shaft, forming with it an integral unit, limiting the freedom of axial movement of shaft parts. The shoulder is referred to as a transition surface from one section to another, serving to lock onto the shaft or axle parts.

The surface of a smooth transition from one stage to another is called a fillet. Fillets have the functions of: a constant radius; a variable radius; with undercut. Transition areas are stress concentrators. To reduce stress concentration in transition sections they cut discharge grooves that allow increasing radii fillets.

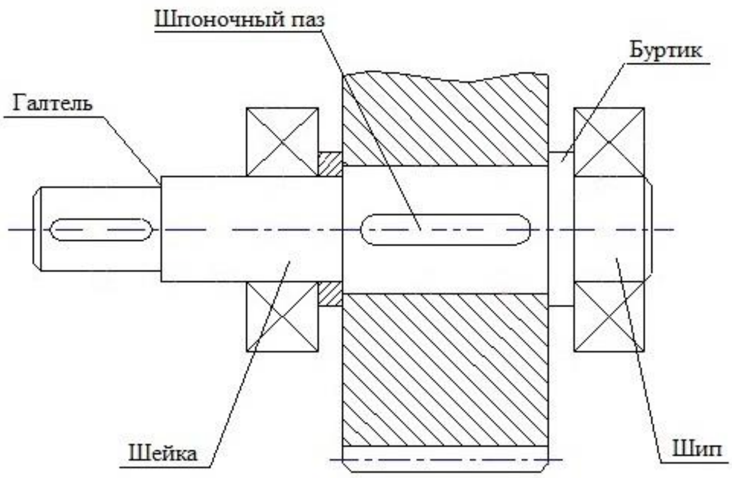


Figure 5.2.

Axles and shafts are in most cases circular continuous and in some cases annular cross sections to reduce the weight of the structure. Separate portions of the shaft have a cross section with a key groove or splines (slots), and sometimes a profile section. In profile connections the joined parts are fastened together by a mutual contact by a non-circular smooth surface and can transfer along with the torque the axial loads. Profiled connections are reliable but not manufacturable, so their use is limited.

The ends of the axles and shafts to facilitate the installation of the moving parts and prevent hand injuries are manufactured with facets.

In Figure 15.2 a general utility gear shaft (gearbox) is shown as an example.

Axles and shafts are made of carbon and alloy structural steels, as they have high strength, the ability to surface and volumetric hardening, possibility to get by means of rolling cylindrical blanks and good machinability by machine tools.

5.2. Calculations of rolls (and axes).

The main criteria of shafts and axles efficiency are the strength and stiffness. The basic design load are twisting T and bending M moments. The effect of compressive or tensile forces is usually small and is not taken into account. The calculation of axes is a special case of the calculation of shafts at torque $T = 0$.

Calculation of shafts consists of two stages: designing and verification. Design calculation is performed only conditionally, as tensional bending moments can be determined only after the study of the shaft design.

After selecting the material for shaft manufacturing taking into account strength conditions under torsion they determine the diameter of its overhang (or cantilever arm in bridges) or supporting portion (if there is no console):

$$d \geq \sqrt[3]{\frac{T}{0,2 \cdot [\tau]}}, \qquad (1)$$

where $[\tau]$- allowable stress under torsion. To compensate for the bending stress and other not considered factors not considered, allowable stresses under torsion are significantly reduced. Thus, for example, gear shafts of general purpose machines take $[\tau]=15\ldots30$ MPa. Expression 1 is written for the shaft of solid circular cross-section. The resulting value of the diameter is rounded to the proximal standard. Thereafter, they develop the entire structure of the shaft; determine the seat of associated parts (gears, sprockets, pulleys, etc.), the seat of bearings, etc. All of these actions are embodied in the thumbnail layout. As a result, they determine the size of all the construction components of the shaft, and then proceed to checking calculations.

Checking calculation for static strength of the shaft is carried out to prevent the occurrence of plastic deformation that may occur during transient overloading, such as starting-up. When calculating they make up a design model, while taking into account that the parts mounted on the shaft transmit it forces and moments in the middle of their width. The own weight of the shaft as well as the weight of the parts mounted thereon, and friction forces in bearings are disregarded. They build diagrams of bending and twisting moments. If the loads act in different planes, they are laid out in two mutually perpendicular components and build diagrams of bending moments separately in each plane. For typical cross sections they calculate the resulting bending moments:

$$M_{\Sigma}=\sqrt{M_X{}^2+M_Y{}^2}\,. \qquad (2)$$

With the help of moments diagram they determine dangerous sections. Then calculate equivalent (reduced) moments according to the energy theory of strength:

$$M_{pr}=\sqrt{M_{\Sigma}{}^2+0,75\cdot T^2}\,. \qquad (3)$$

After that, they specify the design diameter of the shaft at characteristic points:

$$d=\sqrt[3]{\frac{M_{pr}}{0,1\cdot[\sigma]}}\,, \qquad (15.4)$$

where $[\sigma]$- maximum allowable stresses, taken close to the yield strength of the material $[\sigma]\approx 0,8\cdot\sigma_T$.

Checking calculation for fatigue strength for shafts is considered to be the main one, it’s carried out in the form of fatigue strength coefficient n verification in supposedly dangerous sections (the presence of stress concentrators: keyways, contact stresses in the place of parts seat, fillet connections, etc.).Under joint action of bending and torsion the strength margin for the selected cross section is determined by the following formula:

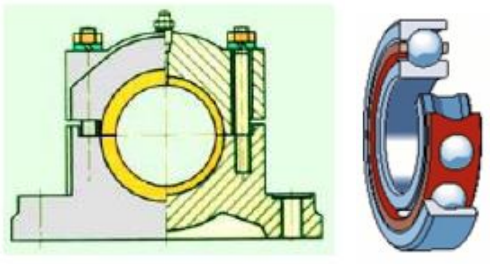
$$n = \frac{n_\sigma \cdot n_\tau}{\sqrt{n_\sigma^2 + n_\tau^2}} \geq [n] \qquad (5)$$

where $n_\sigma = \dfrac{\sigma_{-1}}{K_\sigma \cdot \sigma_a + \psi_\sigma \cdot \sigma_m}$ - the margin of fatigue strength under bending; $n_\tau = \dfrac{\tau_{-1}}{K_\tau \cdot \tau_a + \psi_\tau \cdot \tau_m}$ - margin of fatigue strength under torsion; σ_a, τ_a - peak values of normal and shear stresses; σ_m, τ_m - average values of stresses; ψ_σ, ψ_τ - factors that depend on the mechanical properties of the material; K_σ, K_τ - stress concentration factors under bending and torsion. The permissible value of safety factor is usually taken as $[n] \geq 1{,}5$, but given the increased demands on the stiffness of shafts they accept $[n] \geq 2{,}5 \ldots 3$.

Calculation of shafts for stiffness is carried out when elastic movement occurring under the influence of applied forces may adversely affect the operation of located on the shaft (or axis) parts. The calculation is reduced to determining of deflections, rotation angles, and twist angles and to comparing them with tolerance values.

5.3. Bearings of axles and shafts.

Bearings are pillars of shafts and rotating axles. They accommodate radial and axial loads and transmit them to the machine frame. Bearings provide a predetermined position of rollers and the ability to rotate with minimal friction losses. From the quality of the bearings is largely dependent The performance and durability of machines and mechanisms largely depends on the quality of bearings. According to the type of friction bearings are divided into plain bearings, see Fig. 15.3 a (shaft supporting portion slides on the bearing surface) and rolling bearings, see Fig. 15.3 b (slide is replaced by rolling friction by setting the balls or rollers between the bearing surfaces of the bearing and shaft).



a b

Figure 5.3.

Plain bearings operate only in the presence of lubricant in the clearance between the trunnion of the shaft and bushing. While rotating the shaft engages lubrication into the gap wedge between the pin (journal) and bushing. As a result, there occurs an oil bearing layer with a large hydrodynamic lift under the action of which the shaft emerges.

Lubricant is supplied to the bearing during rotation of the shaft journal (pin) into the area of maximum gap where there’s no hydrodynamic pressure. This is achieved by the presence of oil grooves on the insert that are placed in the unloaded zone.

Bearings are used in those cases when their advantages are convincing:

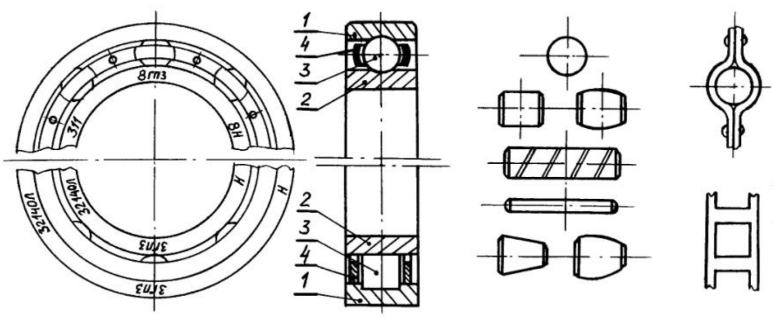
- in high-precision machines;
- in split units;
- for shafts with shock and vibration loads;
- when working in water or other corrosive environments;
- for large-diameter shafts, where there are no roller bearings.

Roller bearings are the most common type of supports of rotating mechanisms and machines parts, and manufactured in large quantities at specialized plants.

Roller bearings have a number of advantages over plain bearings: there’s less resistance to the start-up and rotation at moderate speed rates, high carrying capacity per unit of width of the bearing, easy maintenance, low cost, interchangeability.

However, they have large radial dimensions and weight, smaller radial stiffness, low durability at high rotation speeds (due to overheating), small-scale production unprofitability.

Roller bearings present ready units, and typically consist of an outer -1and inner - 2 rings, rolling elements - 3 and a separator - 4 (Fig. 15.4, a). The rolling elements - balls or rollers (Fig. 15.4, b) roll along the treadmills of rings spaced apart from each other. The separator holds the rolling elements at a predetermined distance from each other and improves lubrication. The separator design (Fig. 15.4, c) depends on the bearing type, and its operating conditions.



a b c

Fig. 5.4.

Standard bearings are classified according to the following criteria:

• according to the direction of the load action relative to the axis of the shaft bearings are divided into: radial (predominantly bearing radial load), thrust (bear only axial loads), combined thrust and radial bearing (bear combined load simultaneously acting on the bearing in the radial and axial directions). Axial bearings (bear axial loads at small radial load simultaneous action);

• according to the shape of rolling elements bearings are divided into ball and roller bearings. Roller bearings according to the form of rollers are divided into bearings with short and long cylindrical rollers, spiral rollers, needle rollers, conical and spherical rollers;

• according to the number of rows of rolling elements bearings are divided into one-, two-, four-and multi-row ones;

• according to the main design features bearings are divided into nonself-aligning bearings and self-aligning ones (allowing rotation of the axis of the inner ring relative to the axis of the outer one).

Bearings of the same bore diameter are subdivided according to boundary dimensions (outer diameter and width) in series. The most common are light and medium series of normal width.

Industries produce roller bearings of five classes of accuracy (0, 6, 5, 4 and 2; notations are given in order of increasing accuracy). Accuracy of bearings is characterized by precision of basic dimensions, accuracy of shape and positional relationship of rings surfaces as well as rotation precision.

Symbolic representation of bearings consists of letters and numbers.

5.4. Performance criteria and practical calculations of bearings.

The main criterion of efficiency of bearings is wear resistance.

The efficiency of roller bearings is evaluated by imputation on the average pressure P on the working surfaces, which guarantees the presence of oil film, and calculations based on the specific action Pv (v - circumferential speed of the tip (journal) surface) of friction forces, which ensures normal thermal conditions and the lack of binding.

$$P = \frac{F_r}{d \cdot l} \geq [P], \qquad Pv \geq [Pv], \qquad (6)$$

where F_r - the radial force acting on the bearing; d and l - the diameter and length of the bearing, respectively.

Roller bearings lose their function because of damage to the surfaces of parts and their destruction. Modern design practice is limited to calculations based on two criteria of efficiency of bearings: fatigue spalling of work surfaces for a given resource (calculation of dynamic load rating) and residual strain (static load rating calculation).

According to static load rating they select bearings with the proviso that the rotating ring has a rotation speed $n \le 1$ rev/min. Selection of bearings is carried out by the condition:

$$P_0 \ge C_0 \qquad (7)$$

where P_0 - the required value of static load rating, if the bearing is loaded with radial and axial forces - $P_0 = X_0 \cdot F_r + Y_0 \cdot F_a$, where X_0, Y_0 - respectively the coefficients of radial and axial loading; C_0 - table-valued static load rating.

At rotation speed of more than 1 r/min selection of bearings is carried out according to the dynamic load capacity. In this case, there's given their longevity in hours or in millions of revolutions. Studies of rolling bearings have established that the durability of the bearing L at 90 % probability of failure-free operation can be determined from the following relationship:

$$L \ge \left(\frac{C_r}{P_r} \right)^p, \qquad (8)$$

where C_r - dynamic load capacity (rating) specified in the catalog for rolling bearings, p – power (degree) indicator ($p = 3$ for ball bearings, $p = 3,33$ for roller bearings) , P_r - reduced (equivalent) load, which takes into account a number of factors affecting the performance of bearings.

For ball radial and **angular contact roller** and angular contact ball bearings the equivalent load is calculated by the following formula:

$$P_r = (X \cdot V \cdot F_r + Y \cdot F_a) \cdot K_б \cdot K_T, \qquad (9)$$

where X , V , Y , $K_б$, K_T - coefficients depending on the design of bearings, loading conditions and exploitation.

According to calculated reduced load P_r and the estimated life of bearings L they carry out the selection of bearings by the following condition:

$$C_r \geq [C_r] \qquad (10)$$

where $[C_r]$ - dynamic load capacity of the bearing according to the catalog.

If the bearing is adopted for structural reasons (for example, according to the diameter of the inner ring), then it is checked by calculating the resource (in hours):

$$L_h = \frac{10^6 \cdot L}{60 \cdot n} \geq [L] \qquad (11)$$

where $[L]$ - required (specified) resource.

Let's note that for the same conditions (nature of load, rotation speed, efficiency factor) there can be used any type of bearings. When choosing the type of bearing they take into account the cost, as well as the experience in operating of units, similar to the one which is being designed.

Practical training 6. Connections. Coupling.

6.1. Connections, threaded coupling.

While manufacturing machines some parts are interconnected in such a way that they form releasable and permanent connections.

Plug-in connectors allow for convenient disassembly of connected parts avoiding their damage. Such compounds include: threaded, keyways, splines, and pin joints (studding). Permanent connections do not make it impossible to make out connected parts without damaging their elements, these include: welded, riveted, soldered joints.

Selection of the connection type of the construction is determined by its structure and purpose as well as economic indicators.

Of all the connections used in mechanical engineering threaded coupling is the most common, since they are the most reliable and convenient for assembly and disassembly, have small dimensions, easy to manufacture. The main disadvantage of these compounds is the lack of reliability under vibration loading.

Threaded connection is made by fasteners by means of the thread. The thread turns to be a forming on a cylindrical or conical rod of grooves with a cross section of certain profile (triangle, trapezoid, etc.), each point of which is located on the helix.

Projections located between the grooves are called threads (turns). Under the coil there is understood that part of the projection, which covers the threaded part within 360°. The projection carving, covering the part more than one time is called the wire.

The threads are distinguished according to:

- the shape of the main surface: cylindrical, conical;
- the form of the thread: triangular, rectangular, trapezoidal, round;
- location of the thread surface: external , internal;

- direction of the helix: right, left ;
- the number of threads: one-, two-, three-and multi-start;

• For the purpose intended: fasteners, fastening and sealing, running.

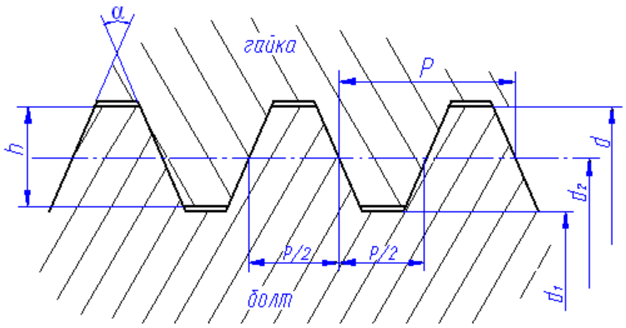


Fig. 6.1.

Widespread threads are standardized. The most common is <u>metric thread</u> - the basic triangular mounting thread. It happens to be with a large and small step. The most common is the thread with a big step, as it has less effect on the wear and production mistakes.

The main geometric parameters of the thread are:

external d, internal d_1 and average d_2 diameters; p - pitch (thread), i.e. the distance between the same parts of two adjacent coils measured in the direction of the center line; α - the angle of the profile, γ - the angle of elevation, i.e. the angle between the helix at the average diameter and the plane perpendicular to its center line; h - the working height of the profile (see Fig. 6.1).

Key fasteners of threaded connections are bolts, screws, studs, nuts and washers. Bolt connection is used for items of small thickness, and also in cases of frequent screwing and unscrewing of the compound, screw and stud connection is carried out at substantially big thickness of one of the parts or in inaccessible locations. Often, they put under a nut a flat circular washer, for example, to reduce the damage to the part surface. Geometric shapes and sizes of screws, nuts, pins are very diverse and described in handbooks and standards.

The thread of fasteners is calculated for shear and crushing. The calculated dependences are shown below:

$\tau = \dfrac{F}{\pi \cdot d_1 \cdot h \cdot K \cdot K_m} \le [\tau]$ - for bolts (screws), $\tau = \dfrac{F}{\pi \cdot d \cdot h \cdot K \cdot K_m} \le [\tau]$ - for nuts, (1)

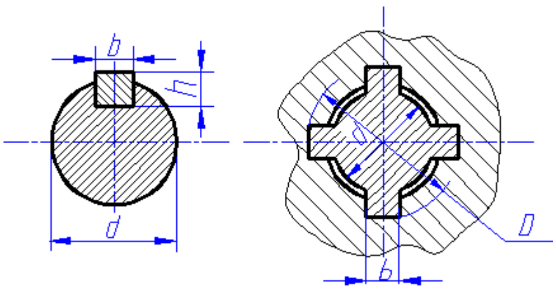
$$\sigma_{см} = \frac{4 \cdot F}{\pi \cdot \left(d^2 - d_1^2\right) \cdot z} \leq [\sigma_{см}] \qquad (2)$$

where K - coefficient of thread completeness, K_m - the coefficient of uneven load distribution on the turns, z - the number of turns, bearing the load.

Since the thread strength of standard fasteners is guaranteed by the National State Standard, strength analysis of these parts thread is not performed.

6.2. Spline and keyway connection.

Spline (Fig. 6.2, a) and keyway (Fig. 6.2, b) compounds are used for fastening parts to shafts that transfer torque (pulleys, sprockets, gears).



a b

Fig. 6.2.

The advantages of the spline joints are: simplicity and reliability of the design and installation, low cost. The disadvantages include: weakening of the shaft and hub group by keyways, limiting of transferred torque.

They distinguish unstressed and strained spline connections. Unstressed compounds are carried out by means of prismatic or segment keys (splines), and strained ones - by means of standard cylindrical pins and wedge keys (splines).

The key located in the groove of the shaft is called the mortise. Prismatic keys are of mortise types, about half of their height is located in the groove of the shaft.

The working faces are their side faces, narrower faces. Keyways in shafts are produced by disc or finger cutters, and in the hub by slotting or pulling.

The most common are prismatic common keys. The disadvantage of parallel keys is difficulties connected with interoperability, which limits their use in mass production.

Circular keys are - mortise and, like a prismatic ones, working by side faces. Circular keys are the most technological ones because of their ease of fabrication and manufacturing of slots for them, as well as the convenience of assembly. Lack of segment keys lies in the fact that they reduce to a greater extent the strength of the shaft due to the necessity of deep grooves. So, they are used for transmissions of relatively small moments.

All V- keys are manufactured with a slope of 1:100. The same inclination is provided for the groove of the hub. Unlike prismatic keys wedge keys working faces are the wide ones and the lateral faces are provided by gaps. Thus, the tension of connections arises due to interference between the shaft and the hub in the radial direction. V-keys use is limited, since they cause the displacement of axis of the hub relative to the shaft of the axis, and in case of short hubs there can occur misalignment of the parts.

When designing key connections the width b and height h of keys is taken according to the State National Standard depending on the diameter of the shaft d. The length of the key l_p is taken depending on the length of the hub and is agreed with the State National Standard. Check The adequacy of taken dimensions is checked by calculation of strength, that is, the calculation of keys is carried out as a screening one. Prismatic and segmental keys are calculated for crushing and shearing, usually checking for shearing is not carried out, as this condition is met using standard cross sections of keys. Accordingly, the strength condition of keyed compounds has the form:

$$\sigma_{см} = \frac{F}{A_{см}} = \frac{4 \cdot T}{d \cdot h \cdot l_p} \leq [\sigma_{см}]. \qquad (3)$$

If in the result of the calculation, it turns out that the key overextended, then they provide two or three keys or use a spline connection.

To connect the hub with the shaft they often use projections on the shaft called splines, which engage in corresponding grooves of the hub. This connection is called a gear or spline. According to the shape of the tooth profile they distinguish connections with rectangular, involute and triangular splines (keys). Compared with key connection spline connections tranfer high torques, provides a more accurate centering of the hub and greater strength of the shaft.

The most common is rectangular connection. It is used with centering of the hub on the outer D and inner d diameters and the lateral sides of the slots. Centering according to b contributes to a more even pressure distribution on the splines, but unlike the centering according to D or d does not provide an accurate alignment of the hub and the shaft, so it is used for transferring high torques when there no stringent requirements for precision centering.

Compared with the rectangular connection, the advantages of the involute connection are as follows: higher strength of splines and improved manufacturability (manufacturing is easier and cheaper). However, for the shafts of small and medium diameter involute splines application is limited due to the high cost of broaching for their manufacture.

Connections with triangular teeth are primarily used in instrument making with limited radial dimensions and small torques.

The number and dimensions of the cross -section of slines (slots) are taken depending on the diameter of the shaft according to the National State Standard. The splines length l is determined by the length of the hub. Calculation of spline connections is carried out as a testing one according to the bearing stress:

$$\sigma_{см} = \frac{2 \cdot T}{d_c \cdot h \cdot l \cdot z \cdot \psi} \leq [\sigma_{см}] \qquad (4)$$

where $d_c = 0,5 \cdot (D + d)$ - the average diameter of slots, z - the number of slots, h - the height of the working surface of the slot, ψ - the coefficient of uneven load distribution between the splines (slots).

6.3. Fundamentals of interchangeability. Limits and fits.

For design, manufacture and repair of machines a big technical and economic importance is the interchangeability of parts and components, through which the design process is accelerated and provided faster assembly and replacement of defective parts.

To ensure the interchangeability of parts, assemblies and systems and streamlining their production on the scale of the enterprise, industry, republic, country, group of countries there exist special standards.

The geometrical parameters of parts are quantified by the size - the numerical values of linear values in selected units of measure. Sizes dimensioned on drawings are referred to as nominal ones. The actual size (the size established by measurement with error margin) may coincide with the nominal size (reference dimension) only by chance, as many factors affecting the accuracy inevitably lead to errors in the size and shape of parts. It is proved that in order to ensure interoperability and normal operation, some items may have a distribution of sizes relative to the nominal size. The maximum and minimum dimensions, among which there may be the actual size of the part, is called – the limit.

The algebraic difference between the real and the nominal size is called a real deviation.

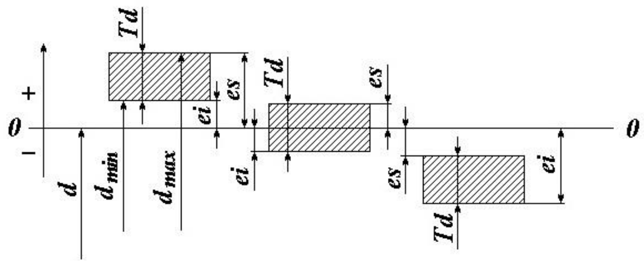


Fig. 6.3.

The difference between the highest and lowest limit dimensions is called - tolerance. The position of the tolerance margin (of the hole - *To*, of the shaft - *Td*) relative to the nominal size is determined by upper and lower limit tolerances. Upper deviation (of holes *ES*, of the shaft *es*) is called the algebraic difference between the maximum limit and the nominal size, the lower deviation (o the hole *EI*, of the shaft *ei*) - the difference between the lowest limit and nominal size. Figure 6.3 shows the layout of tolerance margins of the shaft relative to the zero line, showing a relatively nominal size.

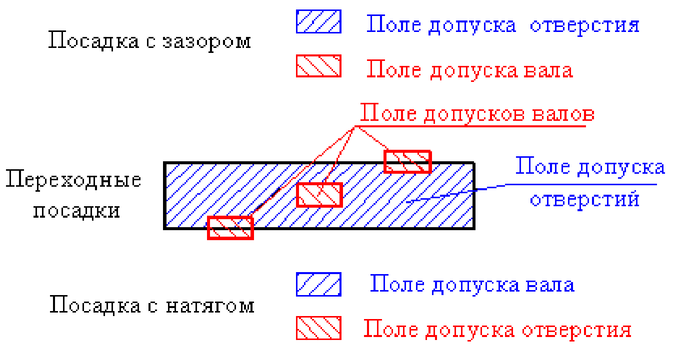
The location of the tolerance margin relative to the zero line is usually denoted a letter (or two letters), of the Latin alphabet: capital letters stand for holes, and small ones – for shafts.

The degree of compliance of the actual size with the nominal one is characteristic for precision of parts manufacturing. A certain quality class (a total of 19) with its value of tolerance margin corresponds to each class (degree) of accuracy. Within each quality class the degree of tolerance margin increases with nominal dimensions enhancement.

The quality class is numbered by serial numbers: 01, 0,1 ... 17. In technical documentation the tolerance margin is referred to as *IT* with quality class number (*IT7*; *IT*14).

Dimensions and extreme deviations for which there is specified the tolerance margin are indicated in the figures/drawings in three ways: by indicating the rated size and margin tolerances: $12^{+0,033}_{+0,006}$, $10\pm0,1$ (deviations equal to zero are not dimensioned); by the designation of tolerance margin consisting of letters and numbers (quality class): $12G8$, $20h10$; by combined method: $12G8^{+0,033}_{+0,006}$, $20h10_{-0.084}$.

The nature of parts connection defined by the difference in their size before assembly is called - fitting. Landing causes more or less freedom of relative to movement of parts and the strength of a fixed connection. Depending on the dimensions of mating parts in the compound there may be occur a gap (the size of the hole prior to assembly is bigger than the size of the shaft) or the preload (the size of the hole prior to assembly is smaller than the size of the shaft). From the relative position of tolerance margins of both the shaft and the hole all fittings are divided into three groups: fittings with a clearance, fittings with preload and transient ones in which it is possible to obtain in a compound both tightness and clearance (Fig. 16.4).



(fitting with a gap, transient fittings, preload fitting; цветные: tolerance margin of the hole, tolerance margin of shafts, tolerance margin of holes, tolerance margin of the shaft, tolerance margin of the hole)

Fig. 6.4.

There are two systems of fitting performance: the system of holes and the system of shafts. The system of holes is a set of fittings in which the tolerance margins of a given size of holes (an appropriate quality class) are the same for all fittings , and a variety of fittings are performed by changing the size of shaft margin tolerances. Such a hole is called the primary one, its tolerance margin is denoted by the letter *H*, which is placed after the nominal size of the parts. Lower deviation of the main hole is equal to zero, the upper one is positive.

The shaft system is called a set of fittings in which shaft deflections are identical (for a given size interval and quality class accuracy), and various fittings are performed by varying the margin tolerances of holes. The tolerance margin of the main shaft are indicated by the letter *h*, it is located "in the body" of the shaft, i.e. the nominal size corresponds to the maximum limit size (the upper deviation is equal to zero, and the lower one is negative).

Both systems are equal, but the hole system is the most preferred one due to reducing the range of cutting tools used.

Fittings on drawings are dimensioned by consecutive writing of the nominal size of mating surfaces and tolerance margin dimensioning of firstly the hole, then the shaft, i.e.: in the hole system $40\frac{H7}{s6}$; in the shaft system $40\frac{P7}{h6}$.

6.4. Coupling. General Information.

Couplings are devices that connect shafts and transfer the torque. An example of the coupling is shown in Figure 16.5.

Some types of couplings have the ability to: absorb vibrations and shocks, protect the machine from overloading, turn on and off the working mechanism of the machine without stopping the engine.

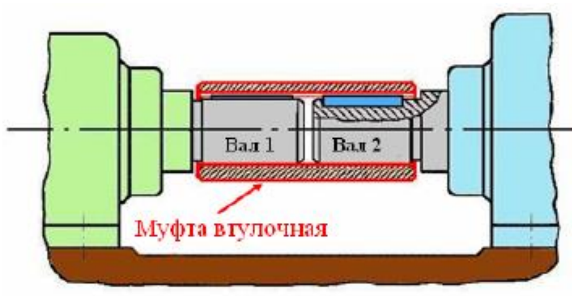


Fig. 6.5.

Classification of mechanical couplings is shown in Fig. 6.6.

Figure 6.6.

(mechanical couplings; incontrollable, controllable, self-controlled, combined; closed coupling, flexible coupling, compensating coupling, cam sleeve, gear clutch, friction clutch, overload clutch, centrifugal clutch, overrunning clutch)

The most common types of couplings are standardized. They are not calculated but selected by the basic parameter of each coupling – the torque T in accordance with the conditional calculation:

$$T = K \cdot T_H \qquad (5)$$

where K - the coefficient of the operation mode of the coupling; T_H - the nominal torque.

Coupling elements that provide the torque transfer are subjected to checking calculation (bolts, pins, springs, resilient members).

Educational edition

Methodological instructions for the implementation of practical tasks, independent work and individual tasks in the disciplines «Machine parts» for foreign applicants of the specialty 131 – Applied Mechanics of the first (bachelor's) level of education of full-time education

Compiler: Volodymyr KLITNOI

Responsible for graduation prof. Gaydamaka A.V.

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