

Ministry of Education and Science of Ukraine

National Aviation University

Lecture Notes

subject “**Mechanics**”

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Introduction in the applied (technical) mechanics. The primary goals of a course.

Mechanics - the science area which purpose is studying of movement and an intense condition of elements of machines, building designs, continuous environments, etc.

The mechanic can be divided into theoretical and applied (technical). Both of them are interconnected.

As the term the applied mechanics understand the area of mechanics devoted to studying of movement and an intense condition of real technical objects - designs, machines, работотехнических systems, taking into account the basic laws established in the theoretical mechanics.

The course is based on such общейинженерных disciplines as: mathematics, physics, materials technology, the theoretical mechanics.

Course "the Applied mechanics" finishes a cycle общейинженерных disciplines in which bases of the theory and calculation of designs of machines are examined.

The basic definitions of section "Resistance of materials".

"Resistance of materials" - science area about engineering methods of calculation on durability of engineering designs.

The bar is a firm body, at which one of the sizes much more than two others.

The core is a bar which works on a stretching or compression.

The basic hypotheses of resistance of materials:

1. The material is considered continuous, homogeneous and isotropic.
2. Deformations insignificant (small) in comparison with the sizes of bodies.
3. Between deformations and loadings there is a linear dependence.
4. A superposition principle: the result of interaction on a body of system of loadings is equal to the sum of results of interaction of the same loadings enclosed consistently in any order.
5. Principle Sen-Venana. In points of a body enough removed from a place of the appendix of loadings the size of internal forces depends a method of the appendix of loadings a little.

Method of sections.

Durability of real bodies is defined by internal forces which arise in them under the influence of external loadings.

For definition of internal forces use a method of sections. In a place interesting us the section is conditionally spent, and balance (cut) cut off part is then examined.

Internal forces are defined as the sum of external forces (loadings) enclosed to the cut (cut off) part.

Components of internal forces:

N - normal force

R - equally effective force

$Q_y$  and  $Q_z$  - cross-section (перерезывающие)  
forces  $M_z$  and  $M_y$  - the bending moments  $M_k$  -  
twisting the moment

Types of simple deformations

1. A stretching (compression).

In cross-section sections of a bar there are only normal forces.

Cores, thread connections

2. Shift (cut).

In cross-section sections arise only cross-section (перерезывающие) forces.

Rivets, welded seams

3. Torsion.

In cross-section sections there are only twisting moments.

Shaft

4. A bend.

In cross-section sections arise the bending moment and перерезывающая force.

Axes and shaft

Deformations and pressure.

Deformation is a change of the form and the sizes of a body.

Elastic deformations disappear at removal of loadings. All designs work in a zone of elastic deformations.

Residual deformations (plastic) - remain at removal of loadings.

As pressure is called the size (quantity) of the internal force falling to unit of the area of section.

$\sigma$  - normal pressure which is directed perpendicularly to section.

$\tau$  - касательное the pressure laying in a plane of section.

#### Tension (compression).

Such kind of deformation arises at action on a bar of system of the forces directed along its axis.

Along an axis it is expedient to represent distribution of internal forces graphically.

Normal forces we will define by means of a method of sections.

Normal force is numerically equal to the sum of projections of external forces on a longitudinal axis of a bar (forces are enclosed to the cut part).

The rule of signs is thus considered:

Compression

Stretching



**Lecture № 1**  
**Details of machines – the General data**

"Details of machines" is a subject devoted to consideration of calculation and designing bases of details and the knots of a general purpose meeting in various mechanisms and machines.

Mechanism – is called the system of bodies intended for transformation of movement of one or several bodies in demanded movements of other bodies.

Machine – is called mechanism or a combination of mechanisms which serve for simplification or replacement physical or brainwork of the person.

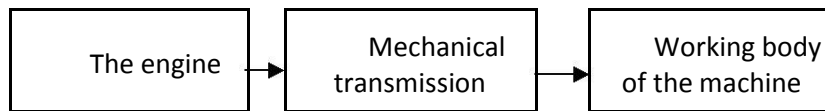
Detail is the part of the machine produced without application of assembly operations (a bolt, a gear, key).

Knot – is a large assembly unit (a transmission, a reducer), which is a product component.

The mechanism – is a machine part in which working process is realized in performance of certain mechanical movements.

**Mechanical transmissions**

Mechanical transmission is the mechanism that transfers (transmits) energy from the engine to working body of the machine with transformation of parameters of movement.



In most cases mechanical transmissions transform parameters of rotary movement, changing the module or a direction of speed, and sometimes and character of movement.

Requirement of introduction of mechanical transmission between the engine and working bodies of the machine as a drive component it is caused by such reasons:

- required speeds of movement of the machine working bodies, as a rule, are not coordinated with the most favourable speeds of the engine;
- in some machines it is necessary to adjust speed of movement, however regulation of the engine speed is not always expedient and possible;
- the engine shaft carries out rotary movement with constant speed, and in machines non-uniform movement, or translational movement, screw and so forth sometimes is necessary.

Therefore, the basic purpose of mechanical transmissions is a coordination of parameters of the machine working body movement with parameters of the engine shaft movement.

Transferring mechanical energy, transmissions can simultaneously carry out following functions:

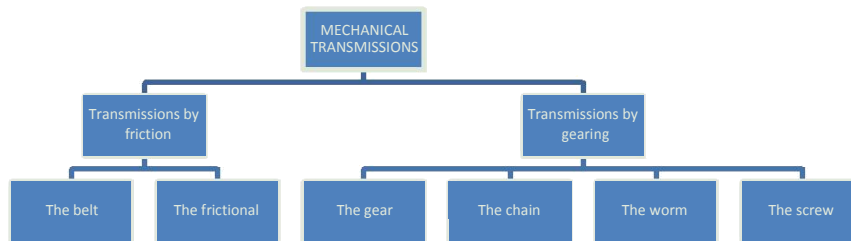
- to choose optimum speeds of movement;
- to increase and decrease angular speeds, accordingly increasing or decreasing rotating moments;
- to reverse movement (a straight line and reverse motion);
- to transform one kind of movement to another (rotary in reciprocating, uniform in discontinuous motion);
- to distribute engine work between several working bodies.

In modern mechanical engineering mechanical, pneumatic, hydraulic and electric transmissions are applied.

In a course "Details of machines" consider only mechanical transmissions - as the most widespread.

All mechanical transmissions divide on two basic groups:

- Transmissions which are based on use of a friction forces (to them relate belt and friction transmissions);
- Transmissions which are based on gearing (to them relate gear, worm gear, chain and screw gears).



### 1. Belt gear

- 1,2 - pulleys;
- 3 - a continuous belt

### 2. Friction gear

- 1,2 - frictional roll  
(there may be with constant, or variables a reduction ratio)

### 3. Tooth gearing

- 1 - a gear wheel (pinion);
- 2 - a wheel (cogwheel)  
(there may be: straight, bevel, and herringbone)

### 4. Worm gear

- 1 - a worm;
- 2 - a worm wheel

### 5. Chain transmission

- 1,2 - star gear;
- 3 - a chain

### 6. A screw gear

- 1 - a nut;
- 2 - the screw

Transmissions of direct contact - frictional, gear, worm, screw.

Transmissions by flexible connection - belt, chain.

In modern mechanical engineering the most widespread are tooth gearings.

The most simple tooth gearing consists of a driving gear (a pinion which transmits the rotary moment), a cogwheel and a strut and is called - a mechanical step.

Main parameters of mechanical transmission are:

- power on entrance  $P_1$  and exit  $P_2$ , angular speeds of drive shaft  $\omega_1$  and driven shaft  $\omega_2$ , or their frequency of rotation ( $n_1, n_2$ ).

These parameters are necessary and sufficient for performance of design calculation of any transmission.

Derivative parameters are:

- mechanical transmission efficiency; efficiency characterizes degree of perfection of mechanical transmission.

For the multistage transmission consisting of several consistently connected transmissions, the general efficiency is:

$\eta_{gen} = \eta_1 \cdot \eta_2 \cdot \eta_3 \cdot \dots \cdot \eta_n$ ; where  $\eta_1 \cdot \eta_2 \cdot \eta_3 \cdot \dots \cdot \eta_n$  - efficiency of each kinematic pair, which enters into a drive;  
Peripheral velocity speed of a drive or driven segment, m.p.s.

$$V = \frac{\omega \cdot d}{2} \quad \text{where } d - \text{diameter of a wheel, a pulley;}$$

Peripheral force,  $N$

$$F = \frac{P}{v} = \frac{2T}{d} \quad \text{where } P - \text{power, Watts;}$$

Rotating moment,  $N \cdot m$

$$T = \frac{P}{\omega} = F \cdot \frac{d}{2}$$

Rotating moment  $T_1$  - is the moment of moving forces, its direction coincides with a direction of a shaft rotation.

Moment  $T_2$  - the moment of resistance forces (a driven shaft),

Therefore a direction of its rotation an opposite to direction of rotation of a shaft.

The reduction ratio:

$$U = \frac{\omega_1}{\omega_2} = \frac{d_2}{d_1} = \frac{z_2}{z_1}$$

The reduction ratio is the relation of angular speeds of drive  $\omega_1$  and driven  $\omega_2$  segments.

For multistage transmission:

$$U_{gen} = U_1 \cdot U_2 \cdot U_3 \cdot \dots \cdot U_n \quad \text{where } - U_1 \cdot U_2 \cdot U_3 \cdot \dots \cdot U_n \text{ is reduction ratio of every stage.}$$



## Lecture № 2

### Tooth gearings. The general data.

In tooth gearings movement is transferred with the help of cogwheels pair. The smaller wheel is called a gear wheel (pinion), and the larger – a wheel. Pinion parameters have an index – 1, and the wheels – 2.

#### Advantages of tooth gearings:

- High loading ability and compactness (in comparison with belt and frictional);
- Small dimensions, high reliability and the big durability (resource of a reducer of 30000 hours);
- High efficiency – 0,96 – 0,98;
- Constancy of the reduction ratio (a slippage is not present);
- The big range of transferred capacities (from several Watt up to 300 MWatt);
- Simplicity of service.

#### Lacks of tooth gearings:

- Rather high requirements to manufacturing and installation;
- Noise at the big speeds.

#### Classification of tooth gearings.

##### 1) On an arrangement of the shaft axes:

- With the parallel axes executed with cylindrical wheels (with internal and external gearing);
- With crossed axes (conic gearing);
- With crossing axes (worm, screw gearing);

##### 2) On character of the movement transformation:

- Rotary into the rotary;
- Rotary into the reciprocating;

##### 3) On an arrangement of teeth concerning the forming line:

- spur;
- helical;
- herringbone

##### 4) In the form of a tooth profile:

- involutes;
- cycloid;
- circular (by Novikov);

##### 5) In the form of wheels:

- With round wheels;
- With non-round wheels;

##### 6) On constructional execution:

- The opened;
- The closed;

##### 7) On size of transferred effort:

- The power;
- Not power;

##### 8) On district speeds:

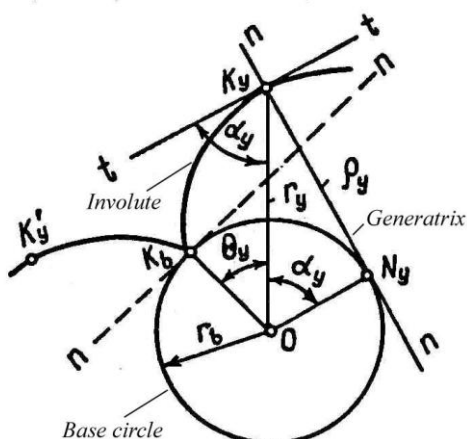
- Low-speed - 3 km/s;

- Medium-rate - = 3 – 15 km/s;

- High-speed - 15 km/s;

##### 9) To destination:

- Reducers;
- Step-up gearing (multiplier).



#### *Formation of the involute gearings.*

Now in the technique there are used tooth gearings almost exclusively with involute gearing of wheels. In the involute

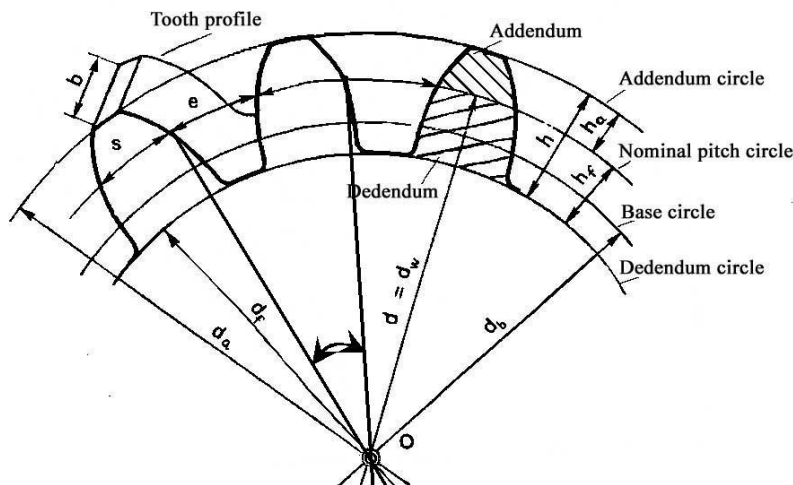
cogwheel a lateral profile of a tooth is involute of circle with the centre in the axis of wheel.

Involute of a circle named a curve which is described by point S of straight line NN rolled without sliding on a motionless circle of radius  $r$ . This circle is called – the basic, and a rolled straight line – as a making straight line (generatrix).

*Involute properties:*

1. Involute is a symmetrical curve with two branches that are begun on the base circle;
2. The tangent to the base circle at any point is always the normal to the involute;
3. Segment  $KyNy$  is the radius of the curvature of the involute at point  $Ky$ . The centre of the curvature is always located on the base circle;
4. The pressure angle and the radius of curvature at the initial point of the involute are equal to zero. While removing points of the involute from the base circle these parameters are increased;
5. When the radius of the base circle is increased the radius of curvature of the involute is reduced. If  $r_b$  tends to infinity the involute is transformed into a straight line.

*Basic parameters of standard involute straight spur gear*



Cogwheel **pitch** is called the distance between the same points of two next teeth, measured on an arch of a corresponding circle. Arches  $p$  i  $p_b$  correspond to the same angular step  $\tau = p/r = p_b/r_b$ . The step is proportional to radius of a corresponding circle. The angular step also can be defined from the formula  $\tau = 360^\circ/z$ , where  $z$  – quantity of teeth of a cogwheel.

The important parameter of a cogwheel is the step  $p_b$  on the base circle. It is obvious, that a step  $p$ , multiplied by number of teeth  $z$ , will represent length of the base circle:

$$\pi d = p z$$

Whence it follows  $d = m z$ .

The value  $m$  which is the relation of pitch  $p$  to number  $\pi$ , is called the wheel **module**. The module is measured in **millimeters**. Its size depends on what circle it is measured. Values of the module on the nominal pitch circle **are standardized**.

Gears, in which the thickness of tooth  $s$  on the nominal pitch circle is equal to width of space width  $e$ , are called the wheels **without displacement**. Such wheels named also **normal** (or **zero**). In the process of movement transferring to cogwheels there are so-called **initial circles** which are rolled by each other without sliding. Wheel parameters on an initial circle attribute index  $W$ . So,  $d_W, p_W, m_W$  – accordingly, diameter, a step, the module of initial circle. At wheels of a gearing without displacement initial circle coincide with the nominal pitch circle, therefore  $d_W = d; p_W = p; m_W = m$ .

Initial circles divide a teeth of wheels on height on two parts: **a head** of a tooth (**tooth addendum**) with height  $h_a$  and **a leg** of a tooth (dedendum) with height  $h_f$ . Formulas for definition of the gear sizes without displacement are placed in the tab.

*The table*

Formulas for definition of parameters of a gear without displacement

Parameter	The formula
Height of a head of a tooth (Tooth addendum)	$h_a = m$
Height of a leg of a tooth (Dedendum)	$h_f = 1,25m$
Whole height of a tooth	$H = 2,25m$
Step on the nominal pitch circle (Pitch)	$P = \pi m$
Step on the base circle	$p_b = \pi m$
Tooth thickness measured along the nominal pitch circle	$S = p/2 = \pi m/2$
Space width measured along the nominal pitch circle	$e = p/2 = \pi m/2$
Diameter of the nominal pitch circle	$d = mz$
Diameter of the base circle	$d_b = mz \cos \alpha$
Diameter of the addendum circle	$d_a = m (z \pm 2)$
Diameter of the dedendum circle	$d_f = m \cdot (z - 2,5)$

Calculation of a straight spur gearing a)

*Forces which operate in gearing*

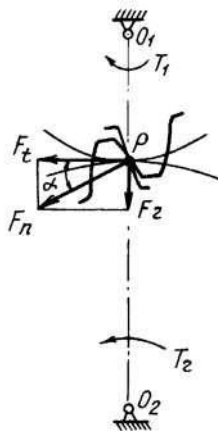


Fig. Forces operating in the straight teeth gearing

Forces of interaction between teeth can be defined in a gearing pole  $p$  (fig.).

Distributions on a contact line of loading replace with equally effective force which is directed along the line of gearing. Normal force display on two components:

- District force  $F_t = F_n \cos \alpha = d^1 \frac{2T}{1}$

- Radial force

$F_r = F_n \operatorname{tg} \alpha.$

*b) Calculation of teeth on contact strength*

At calculation of teeth on contact strength accept following assumptions:

- Calculation carry out in a gearing pole;

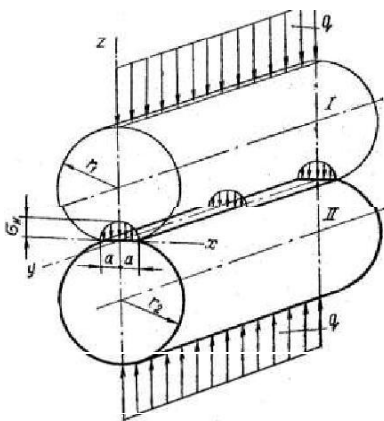
- Contact of two teeth replace with contact of two cylinders in which radiuses are equalled to radiuses of curvature of teeth in a pole of gearing (fig).

The theory of contact stresses is developed by Hertz and for two cylinders looks like

$$\sigma_H = 0,418 \sqrt{\frac{q E_{red}}{\rho_{red}}}$$

Where  $q = \frac{F K}{b}$  - rated contact loading on a tooth;

$K_H = k_H = k_{H\alpha} \cdot k_{H\beta} \cdot k_{HV}$  - factor of contact stresses  $k_{H\alpha}$ ;  $k_{H\beta}$ ;  $k_{HV}$  - Accordingly: factor which considers non-uniformity of loading between a teeth; factor which considers non-uniformity of division of loading on width of a rim, factor which considers dynamic loading on a tooth. All factors are tabular.



$E_{red} = \frac{2E \cdot E}{E + E}$  The reduced module of elasticity;

$\frac{1}{\rho_{red}} = \frac{1}{\rho_1} + \frac{1}{\rho_2} = \frac{2}{d \sin \alpha} \left( \frac{u+1}{u} \right)$  - Reduced radius of curvature.

Considering, that factor of width of rim  $\psi_{bd} = \frac{b}{d}$ ; and factor of width of a tooth of wheel  $\psi_{ba} = \frac{b}{a_w}$ , we will substitute these data in the formula of Hertz and after mathematical transformations we will receive formulas:

- For interaxial distance  $a_w = K_a (u+1) \sqrt{\frac{T \cdot k}{2 H \beta} \frac{1}{[\sigma]^2 u^2 \psi_{ba}}}$ ,

Where  $K_a = 49,5$  - for straight teeth gearing;

$K_a = 43,0$  - for helical teeth gearing.

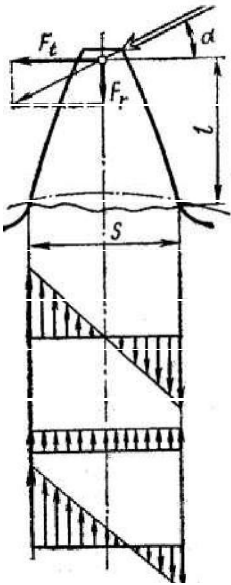
- For contact stresses  $\sigma_H = \frac{330 a}{w} \sqrt{\frac{T \cdot k \cdot (u+1)^3}{b \cdot u^2}} \leq [\sigma]_H.$

*c) Calculation on bending strength.*

At calculation on bending strength a tooth is considered, how a console beam (fig.).

Rated loading is put to tooth top, that is choose the most dangerous case from the point of view of occurrence of the maximum stress of bending in its base.

Let's transfer a point of application of force  $F_n$  on an axis of symmetry of a tooth also we will spread out this force on two components:  $F_r$  – radial and  $F_t$  – district forces.



$$\sigma_F = \sigma_{\text{ax}} - \sigma_{\text{cm}} = \left( \frac{F \cdot l}{W} - \frac{F}{A} \right) \cdot k_T$$

where  $W$  is the polar moment of resistance;  $A = S \cdot b$  - the compression area;  $k_T$  - theoretical factor of stress concentration.

After mathematical transformations we will receive thhe formula

$$\sigma_F = \frac{F \cdot Y \cdot k}{b \cdot m} \leq [\sigma]$$

Where  $Y_F$  - factor of the form of a tooth;  $k_F = k_{F\beta} \cdot k_{FV}$  - factor of bending stresses, accordingly the factor which considers non-uniformity of division of loading on tooth height; factor which considers dynamic loading on a tooth at bending. All factors are tabular.

### Helical spur gearings

Helical teeth gearings are applied in responsible mechanisms at average and high speeds. Helical teeth wheels cuts with the same tools, as straight teeth, a copying method, (on milling machine tools), a run-in method (on tooth mortising machine tools). A tooth inclination receive tool turn on an angle

#### a) The general data.

The spur gears, which line of teeth it is inclined under some angle to forming the separator cylinder are called helical teeth. The angle make s 8 ... 15° (sometimes to 20)

*Advantages of helical teeth gearings:*

- High smoothness of gearing (that higher, than angle there is more);
- Considerably smaller noise and dynamic loadings;
- Higher loading ability.

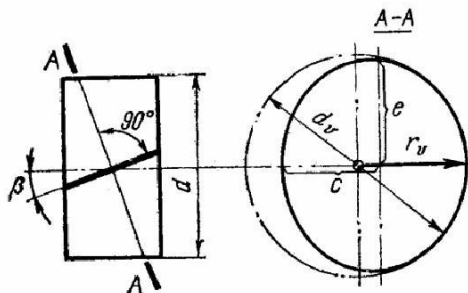
*Lacks:*

- Presence of axial force ( $F_a$ ), tending to shift a wheel with shaft along an axis;
- Necessity of axial fixing of a shaft.

#### 2.1.1 b) Features of calculation of a helical teeth gearing

##### 1) Geometrical parametres

It is possible to define the sizes of helical teeth gears in normal and face section (fig.).



Accordingly we receive a normmal and face step.

$$P_n = P_S \cos \beta,$$

That is  $m_n = m_S \cos \beta$ .

The normal module is standard.

$$d = \frac{m \cdot z}{\cos \beta} - \text{Diameter of a nominal pitch circle};$$

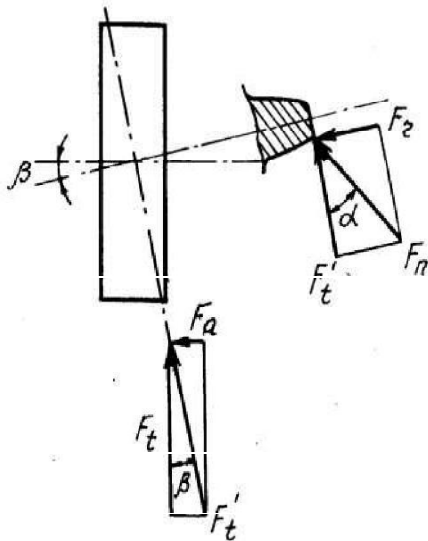
$$d_a = \frac{m \cdot (z + 2)}{\cos \beta} - \text{Diameter of an addendum};$$

$$d_f = \frac{m \cdot (z - 2,5)}{\cos \beta} - \text{Diameter of a dedendum};$$

$$a_w = \frac{d_1 + d_2}{2} = \frac{m_n \cdot z_{\Sigma}}{2 \cos \beta} - \text{Interaxial distance}.$$

Parameters of a helical teeth gear in normal section define from parameters of an equivalent gear. The normal section to a tooth creates an ellipse with semiaxes  $c = r$ ;  $e = \frac{r}{\cos \beta}$ . Teeth which are located on a small axis of an ellipse take part in gearing. The radius of curvature of an ellipse on a small axis equals

$$r_e = e^2/c = r/\cos^2 \beta, \text{ or } d_e = d/\cos^2 \beta. \text{ Accordingly } z_e = z/\cos^3 \beta.$$



#### b) Forces operating in the gearing

Normal force of interaction  $F_n$  distribute on three components:

- $F_t = \frac{2T}{d}$  - district force;
- $F_a = F_t \operatorname{tg} \alpha$  - axial force;
- $F_r = \frac{F_t \operatorname{tg} \alpha}{\cos^2 \beta}$  - radial force

#### c) Features of calculation on contact and bending strength

Calculation on contact strength carries out on parameters of an equivalent gear under the Hertz formula

$$\sigma_H = 0,418 \sqrt{\frac{qL_{red}}{\rho_{red}}},$$

where  $\sigma_H$  - rated contact loading on a helical gearing tooth;  
 $k_{H\alpha}$  - factor which considers non-uniformity of loading between a

Fig. Force operating in the helical teeth gearing

teeth;

$$l_{\Sigma} = \frac{b \cdot \varepsilon}{\cos \beta} \alpha - \text{length of a contact line; } \varepsilon - \text{overlapping factor}.$$

$$\frac{1}{\rho_{red}} = \frac{2 \cos^2 \beta}{d \sin \alpha} \left( \frac{u+1}{u} \right).$$

After substitution of data in the formula of Hertz and mathematical transformation we will receive formulas:

$$- \text{Interaxial distance } a_w = K_a (u+1) \sqrt{\frac{T \cdot k}{2 H \beta} \frac{1}{[\sigma]^2 \cdot u^2 \psi_{ba}}},$$

where  $K_a = 43,0$  - for helical teeth gearing;

- Calculation formulas on contact strength

$$\sigma_H = \frac{270 a_w}{b \cdot u^2} \sqrt{\frac{T \cdot k (u+1)^3}{2 H}} \leq [\sigma]_H.$$

At calculation of a helical gearing on a bending use the formula similar as for calculation of a straight gearing

$$\sigma_F = \frac{F_t \cdot Y_F \cdot k_F \cdot Y_{\beta} \cdot k_{F\alpha}}{b \cdot m} \leq [\sigma]_F,$$

where:  $Y_{\theta}$  is the factor which consider increasing of strength at a bending;  $K_{F\alpha} = \frac{4 + \left( \frac{\varepsilon - 1}{\alpha} \right)^{n-6}}{4\varepsilon}$  – factor, considering non-uniformity of division of loading between teeth;  $n$  - degree of accuracy of manufacturing of a detail.

## Lecture № 4

### Bevel gearings

#### 1. The general data.

Bevel gearings are applied to transferring the rotary movement between the shafts which axes are crossed (fig). The interaxial angle is defined by constructive necessity and can change within  $10-170^{\circ}$ . The bevel gearings with crossing of axes under an angle  $=90^{\circ}$  are used more often. Such gearings are called *orthogonal*.

Lacks of bevel gearings:

- For toothings of bevel wheels special machine tools and tools are required;
- Crossing of axes of wheels forces one of them to instal cantilever (more often a pinion), that leads to increasing of non-uniformity of the loading distribution on length of a tooth;
- In gearing there is an additional axial force for which perception it is necessary to instal radial-thrust bearings;
- Increased requirements to accuracy of manufacturing and installation;
- Pinion bearings are installed in a special cage where by installation of special gaskets the wheels top cones coincidence are adjusted .

All these lacks lead to decrease in loading ability of bevel gearing. In comparison with spur gearing it makes 85 % (enter factor = 0,85)

#### 2. A bevel wheel.

By analogy to spur gearing, bevel gearing also has initial cones which are rolled on one another without sliding. Because in bevel gearing angular correction initial and separatory cones coincide is not made.

The gearing module is the relation of separatory diameter to quantity of teeth  $m = d/z$ , therefore in a bevel wheel the module is variable magnitude as separatory diameters it is possible to carry out much. At definition of the geometrical sizes of bevel wheels accept the greatest module (external).

At calculations on strength the sizes on mean section of a wheel are used. Then the mean module.

The geometrical sizes of a bevel gear:

- External pitch diameter;
- Addendum at the outer section;
- Dedendum at the outer section;
- Width of a tooth;
- Outer cone distance;
- Mean cone distance;
- Mean pitch diameter

#### 2. Bevel gearing

The key parametres of bevel gearing are (fig.):

- $h = m$
- $h_f^a = 1,2m_e$  - height of a tooth head;
- $h = 2,2m_e$  - height of a tooth leg;
- $h = 2,2m_e$  - tooth height;
- $d_{ea1} = d_{e1} + 2m_e \cos \delta_1$ ;  $d_{ea2} = d_{e2} + 2m_e \cos \delta_2$  - diameters of addendum;
- $d_{ef1} = d_{e1} - 2,4m_e \cos \delta_1$ ;  $d_{ef2} = d_{e2} - 2,4m_e \cos \delta_2$  - diameters of dedendum;
- $R_e = 0,5m_e z_{\Sigma}$  - external cone distance,  $z_{\Sigma} = \sqrt{z_{12}^2 + z_{22}^2}$ ;
- $R_m = R_e - 0,5b_e$  - среднее cone distance;
- $\psi_{bRe} = \frac{b_e}{R_e} = 0,285$  - factor of a rim width of a tooth.

Reduction ratio of bevel gearing

$$U = \frac{n_1}{n_2} = \frac{\omega_1}{\omega_2} = \frac{z_2}{z_1} = \frac{\cos \delta_1}{\sin \delta_1}$$



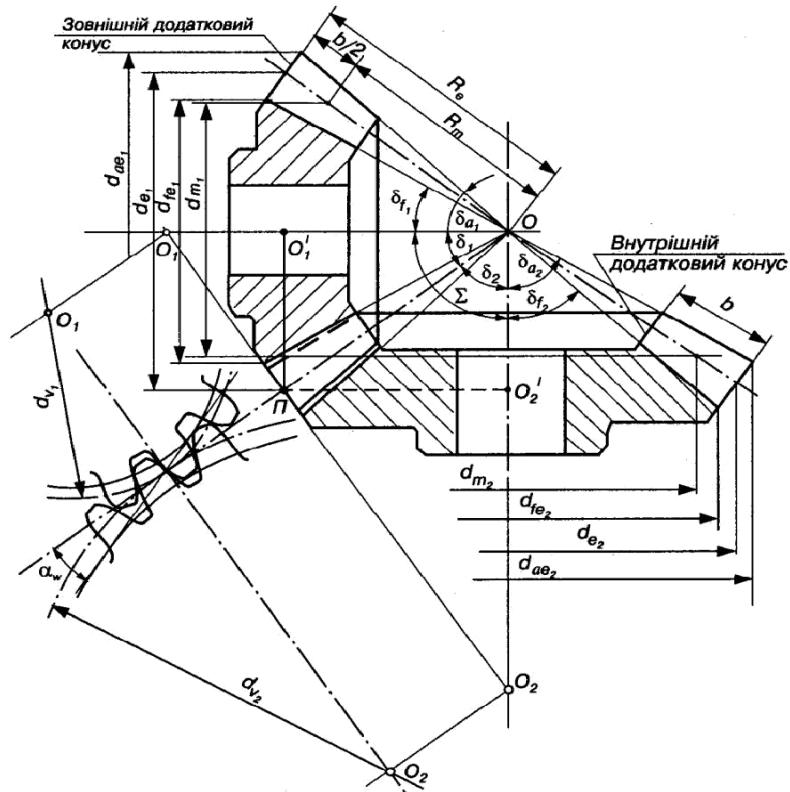


Fig.

*Forces operating in gearing*

Normal force  $F_n$  distribute on:

- $F_t = \frac{2T}{d_{m2}}$  - district force;
- $F_{r1} = F_{a2} = F_t \operatorname{tg} \alpha \cos \delta_1$  - radial force;
- $F_{a1} = F_{r2} = F_t \operatorname{tg} \alpha \sin \delta_1$  - axial force.

Normal force express through district force

$$F_n = \frac{F_t}{\cos^2 \alpha}$$

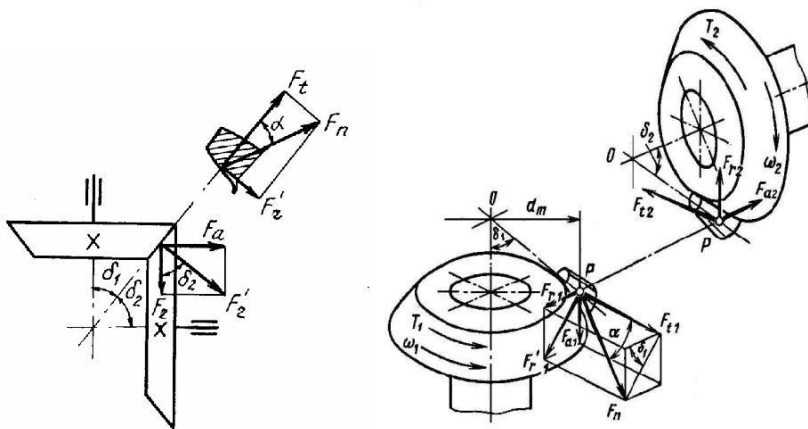


Fig. Forces which operate in conic gearing

*Features of calculation on contact and bending strength*

Calculation on contact strength carry out on parametres of an equivalent gear. The form of a tooth of a bevel wheel in normal section an additional cone same, as well as at a spur gear (fig.). An equivalent gear obtain as evolvent of an additional cone.

Diameters of equivalent gears:

$$d_{E1} = d_{e1} / \cos \delta_1 ; d_{E2} = d_{e2} / \cos \delta_2 .$$

If to express diameters through  $m$  and  $z$  we will receive quantity of teeth of an equivalent wheel:

$$z_{E1} = z_1 / \cos \delta_1 ; z_{E2} = z_2 / \cos \delta_2 .$$

Calculation on contact strength carry out under the Hertz formula

$$\sigma_H = 0,418 \sqrt{\frac{qE}{\rho_{36}}}$$

Where - designed contact loading on a tooth.

$$\frac{1}{\rho_{36}} = \frac{1}{\rho_1} + \frac{1}{\rho_2} \quad \frac{1}{\rho_1} = \frac{2 \cos \delta_1}{d_{e1} \sin \alpha} ;$$

Reduced radius of curvature, where

$$\frac{1}{\rho_2} = \frac{2}{d_{e2} \sin \alpha}$$

After substitution in the formula of Hertz and mathematical transformation we will receive

$$\sigma_H = \frac{335}{m} \sqrt{\frac{T_K \left( \sqrt{u^2 + 1} \right)^3}{2 H b u^2}} \leq [\sigma]_H$$

The designing formula for calculation will become

$$d_{e2} = K \sqrt{\frac{T_2 \kappa_H \beta u}{[\sigma]^2 (1 - 0,5 \psi) \psi}} \quad \left( \begin{matrix} b_{Re} \\ b_{Re} \end{matrix} \right)$$

Where  $K_d = 99$  - skilled factor.

For calculation on bending strength use the formula

$$\sigma_F = \frac{F Y_K}{b m \vartheta} \leq [\sigma]_F$$

Where  $\vartheta = 0,85$  - skilled factor which specifies, that bearing ability of conic gearing is equaled 85 % from bearing ability of spur tooth gearings.

**Lecture № 5**  
**Features of calculation of a worm gearing**

*The general data*

Worm gears concern transmissions of gearing and are used in a case when it is necessary to transfer rotary movement between the shafts whose axes are crossed straight lines. The angle between axes, as a rule, is equal  $90^\circ$ . Movement in worm gears is transformed by a principle of screw pair, or by a principle of an inclined plane.

Advantages of worm gears:

1. Possibility of reception of the big transfer numbers.
2. Smoothness and noiselessness of work.
3. The raised kinematic accuracy.
4. Self-braking possibility.

Lacks of worm gears:

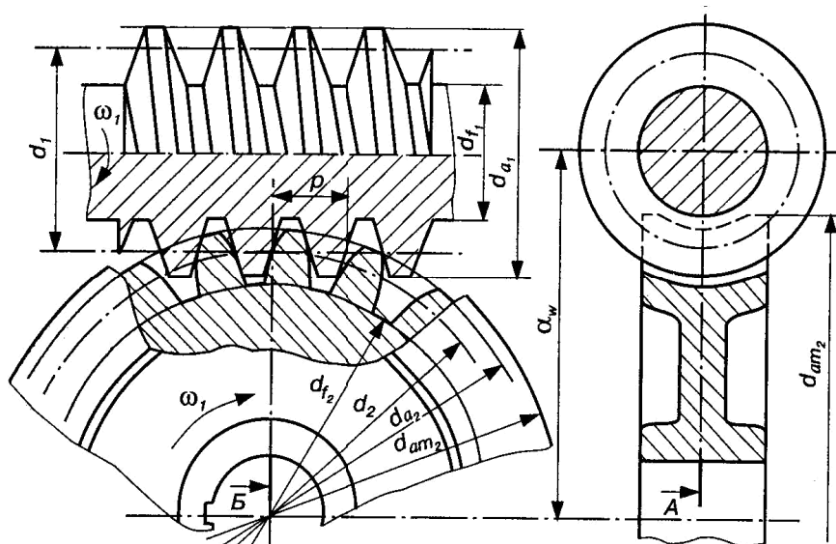
1. Rather low efficiency.
2. The raised wear and propensity to jamming.
3. Necessity of use for worm gears an expensive antifrictional materials (bronzes).
4. Increased requirements to accuracy of assemblage.

Worm gearings are more difficult and more expensive than other gearings. They have rather low efficiency and propensity to jamming which limits their usage by area of low and average powers at periodic and short-term work. Power of worm gearings, as a rule, does not exceed 50 - 60 kW. At the big powers and long work, losses in worm gearings are so considerable, that their use becomes unprofitable (inexpedient).

Worm gearings are used for reduction of a conducted shaft rotation frequency. By means of a worm gearing it is possible to realise the big reduction ratio  $u = 7 - 100$  and more. Such gearings are used in hoists and transport machines, various metalcutting machine tools, vehicles etc.

*Geometrical parameters*

Basically the driver is the worm than geometrical parameters of a worm there will be with an index 1, and geometrical parameters of worm gear with an index 2 (fig.).



-  $h_a = m$  - a tooth Head;

-  $h_f = 1,2m$  - a tooth Leg;

-  $h = 2,2m$  - tooth height;

$d_1 = mq$ ;  $d_2 = mz_2$  - Pitch diameters, accordingly a worm and a worm gear; where  $q$  - factor of diameter of a worm;

-  $d_{a1} = d_1 + 2m$ ;  $d_{a2} = d_2 + 2m$  - Addendum circles diameters;

- $d_{f1} = d_1 - 2,4m; d_{f2} = d_2 - 2,4m$  – Dedendum circles diameters
- $d_{am2} = d_{a2} + \frac{6m}{z_2+2}$  - the Maximum diameter of a worm gear.

The main parameters of the worm are:  $\alpha$  - profile angle ( $\alpha=20^\circ$ );  $m = \frac{P}{\pi}$  - axial module;  $P$  - axial pitch;

$z_1$ - number of starts of the worm (з стандартом  $z_1 = 1; 2; 4$ ). Is recommended that  $z_1 = 4$  when  $u = 8... 15$ ;  $z_1 = 2$  when  $u = 15...30$ ;  $z_1 = 1$  when  $u \geq 30$ ;  $q = \frac{d_1}{m}$  - число осьових модулів в дільному діаметрі черв'яка  $d_1$ .  $m$

Значення  $m$  і  $q$  стандартизовані.

Worm gear efficiency count by the formula

$$\eta = \frac{P}{P_1} = \eta_1 \eta_2 \eta_3$$

Where  $P_1, P_2$  - power on the driving and a driven shafts;

$\eta_1; \eta_2; \eta_3$  – efficiency reflecting losses of power in gearing, bearings and oil moving in the gearing case accordingly.

If the driver is the worm, gearing efficiency count by the formula

$$\eta_1 = \frac{\text{tg} \gamma}{\text{tg}(\gamma + \phi)}$$

Where  $\phi$  is a summary angle of a friction which depends on speed of sliding (tabular).

$\gamma$  - lifting angle of the helical line

$$\text{tg} \gamma = \frac{P \cdot z_1}{\pi \cdot d_1} = \frac{\pi \cdot m \cdot z_1}{\pi \cdot m \cdot q} = \frac{z_1}{q}, \quad (2.1)$$

If the driver is the worm gear, gearing efficiency count by the formula

$$\eta_1 = \frac{\text{tg}(\gamma - \phi)}{\text{tg} \gamma}$$

At  $\gamma \leq \phi$  movement from a gear it is impossible, as, there is a self-braking phenomenon.

Power losses in bearings and for oil moving  $\eta_2, \eta_3$  are estimated  $\approx 0,95-0,96$ ; therefore the general efficiency of a worm gear count by the formula

$$\eta = (0,95 - 0,96) \frac{\text{tg} \gamma}{\text{tg}(\gamma + \phi)}$$

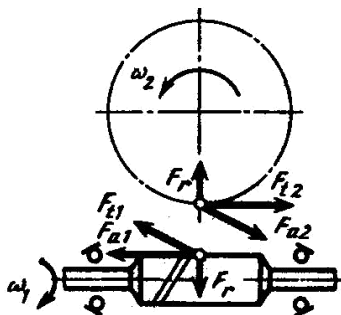
### The forces operating in gearing

The general normal force  $F_n$  distribute on three components for a worm and a worm gear:

-  $F_{t1} = F_{a2} = \frac{2T_1}{d_1}$  District force on a worm and axial force on a worm gear;

-  $F_{t2} = F_{a1} = \frac{2T_2}{d_2}$  District force on a worm gear and axial force on a worm;

-  $F_{r1} = F_{r2} = F_{t2} \text{tg} \gamma$  Radial forces on a worm and a worm gear.



Features of calculation on contact and bending strength

Calculations on contact strength carry out under the Hertz formula

$$\sigma_H = 0,418 \sqrt{\frac{qE}{\rho^{3/2}}}$$

Where  $q = \frac{F \kappa}{n H \Sigma}$  - rated contact loading on a tooth;

$$\frac{d \delta \varepsilon \rho}{\Sigma \cos \gamma}$$

=<sup>a</sup> Length of a contact line  $\delta$  - a coverage arch;

$\varepsilon \alpha$  - Overlapping factor;  $\rho = 0,75$  - the factor considering reduction of length of a contact line.<sup>1</sup>

$$\frac{1}{\rho_1} = \infty \text{ In axial section; } \frac{1}{\rho_2} = \frac{2 \cos^2 \gamma}{d \sin \alpha}$$

After substitution in the formula of Hertz and mathematical transformation obtain the design formula:

$$a_w = \left( \frac{z}{2} + 1 \right)^3 \sqrt{\left( \frac{170}{z} \right)^2 \left[ \frac{2 \cos^2 \gamma}{d \sin \alpha} \right]^2} ;$$

And the formula for calculation of contact stresses:

$$\sigma_H = \frac{170}{\left( \frac{z}{2} + 1 \right)^3} \sqrt{\frac{T \kappa}{a_w} \left( \frac{z}{2} + 1 \right)^3} \leq [\sigma]_H$$

For calculation of teeth of a worm gear on bending strength use the formula:

$$\sigma_F = \frac{0,6 F \kappa Y \xi}{b_2 m} \leq [\sigma]_F$$

Where  $\xi$  - the factor considering easing of teeth from deterioration.

**Lecture № 11**  
**Calculation of shafts**

Separate elements of machines, mechanisms, and also mechanical parts of the electronic devices which are carrying out rotary movement are installed on shafts and axes.

The axis is a detail of the extended cylindrical form, which supports elements of machines in their rotating movement, not transferring the useful twisting moment.

The shaft is a detail which is intended for transfer of twisting movement and maintenance of elements of machines in their rotary movement.

Constructively shafts can be: cylindrical with constant diameter, step and with the gear rim which cut on them, and flexible.

The basic materials of shafts is the carbonaceous and alloyed steels. For the shafts which sizes are choosing on rigidity conditions, there were used mainly steel 5, steel 6. To the majority of other cases are applied steels 45, 50, 40X, 40XH and others.

Basically there are used three kinds of shaft calculation: design, on static strength and on fatigue strength.

*Design calculation of the shafts*

During performance of design calculation at the initial stage there are only known details which will be installed on the shaft, and operating external loadings. Therefore conditionally define diameter of a shaft in its characteristic sections on strength conditions on torsion:

If to accept the polar moment of resistance of section in which twisting moment  $T$  operates, it is possible to define the necessary diameter of a shaft

$$\tau = \frac{T}{W_p} \leq [\tau].$$

If to accept the polar moment of resistance of section  $W_p = \frac{\pi d^3}{16} \approx 0,2d^3$  in which operates the twisting moment  $T$  it is possible to define the necessary diameter of the shaft  $d \geq \sqrt[3]{\frac{T}{0,2[\tau]}}$ .

For steel shafts admissible stresses are in limits

$$[\tau] = (15 - 30) \text{ MPa.}$$

Having the sizes of details which are installed on a shaft, and considering ways of their installation and fixing developing all design of a shaft (fig.).

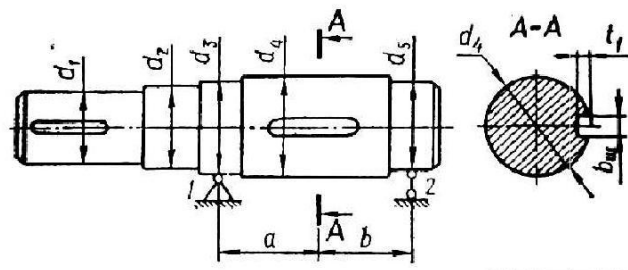


Fig.

The outlet end of a shaft  $d_1$  increasing by 5-10 % in connection with presence of a spline groove. If the shaft by means of a coupling incorporates to electric motor shaft their diameters are co-ordinated that the sizes of diameters of connected shafts differed no more than 20 %.

Diameter of a shaft  $d_1$  selects in accordance with the standard (GOST). Diameter of a shaft  $d_2$  choose under the formula  $d_2 = d_1 + (2...5) \text{ mm.}$

On a shaft in diameter  $d_3$  there are bearings. It calculates under the formula  $d_3 = d_2 + (3...8) \text{ mm}$  so that the total result was multiple 5.

On a shaft in diameter  $d_4$  the gear settles down. This diameter is calculated under the formula  $d_4 = d_3 + 5 \text{ mm.}$

*Shaft Calculation on fatigue strength*

Calculation of shaft on fatigue strength is the basic and obligatory. Calculation on fatigue strength in most cases are carried out in the form of check of the safety factors in certain sections.

## Couplings

### 1. The general data.

Couplings name the devices connecting shafts and transfers twisting moments. Application of couplings is connected with that the majority of machines consists of separate components which have input and output shafts. Connection of shafts and transfer of the twisting moment is the general, but not unique purpose of couplings.

Also they carry out other functions:

- Compensate an error of a relative positioning of shafts;
- Reduce dynamic loadings;
- Protect elements of the machine from an overload;
- Switch on and off working bodies of the machine without engine shutdown.

All couplings of the machine drives by a work principle divide into three basic types: mechanical, electric and hydraulic.

Couplings are classified on four big classes:

uncontrollable, controllable, self-directed and combined. The most applied are uncontrollable couplings which structure includes blind, elastic and compensatory couplings.

By an operation principle there are:

- Constants - carry out constant connection of shafts among themselves;
- Coupling - allow coupling and unhooking of shafts during operation;
- Self-directed (automatic) - automatically separate shafts at change of the set operating mode of the machine.

On a kind of work there are:

- Rigid - which transmits pushes, shocks and vibration together with the twisting moment;
- Elastic - which amortise, pushes, shocks and vibrations by transfer of the twisting moment thanks to presence of elastic elements - springs, rubber plugs, etc.

The basic characteristic of each coupling is a twisting moment  $T_p$  on which transmitting it is calculated. Other important indicators of designs of couplings is limiting frequency of rotation, overall dimensions, weight and the moment of inertia of couplings.

Standard couplings do not calculates, but selects on the twisting moment from the corresponding guides

$$T_p = k_H \cdot T,$$

Where  $T$  - the twisting moment;  $k_H$  is the factor considering overloads or an operating mode of a coupling.

$k_H = 1,0-1,5$  - for quiet loadings;  $k_H = 1,5-2,0$  - for machines with average weights and variable loading;  $k_H = 2,0-3,0$  - for machines with big weights and shock loadings.

Sometimes at some couplings calculate separate elements (bolts, screws, pins, elastic elements, etc.). The blind flanged (fig. 1) and elastic coupling МУВП (fig. 2) may be an example of such calculations.

In a flanged coupling the bolt, connecting two semicouplings can be installed in an aperture without a clearance and with a clearance.

If the bolt is installed in an aperture with a clearance (fig., a variant I) the twisting moment is transmitted by means of the forces of a friction, arising in a plane of a joint of semicouplings from a tightening of bolts.

Calculations are carried out on conditions

$$F_{frict} = F_{tight} \cdot f \cdot i > F, \text{ i.e. } F_{tight} > \frac{F}{fi};$$

$$\text{where } F = \frac{2T}{D_0}.$$

In general

$$\sigma_p = \frac{F_{tight}}{A} \text{ where } A = \frac{\pi d^2}{4} \text{ is the area of a bolt cross section.}$$

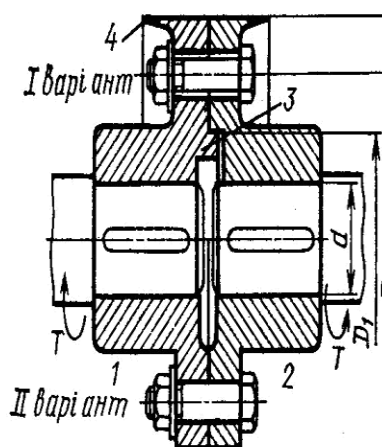


Fig.

$$\sigma_p = \frac{8T_p}{\nu_0 d_1^2 f i \pi z} \leq [\sigma] \quad \text{or, } d = \sqrt{\frac{8T_p}{D [\sigma]_p f i \pi z}} \quad \text{where } z - \text{quantity of bolts.}$$

If the bolt is installed in an aperture without a clearance (fig. 1, a variant II) it counts on strength of a cut

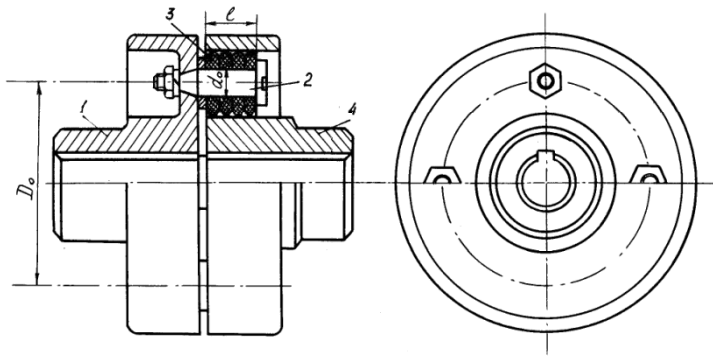


Fig.

$$\tau_{cut} = \frac{F}{A_{cut}} \leq [\tau]_{cut}; \quad \tau_{sp} = \frac{8T_p}{2 D_0 \pi d_0 z} \leq [\tau]_{sp}; \quad d = \sqrt{\frac{8T_p}{D [\tau]_{sp} \pi z}}$$

Verifying calculation of a coupling with elastic element carried out under condition of pressure restriction on surfaces of an elastic element

$$P = \frac{2T_p}{D_0 d z} \leq [P], \quad [P] = 1,8-4,0 - \text{for rubber.}$$

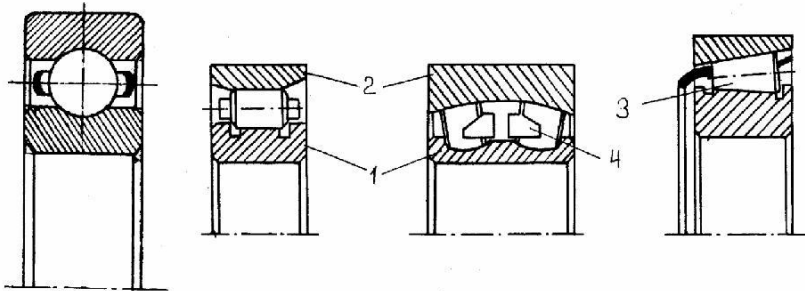


Bearings are machine elements intended to support axles and shafts.

Depending on a kind of a friction bearings divided into: Rolling contact bearings and Sliding contact bearings.

**Rolling contact bearings**

The supporting elements of axles, shafts and other details working on use of a principle of a rolling friction are Rolling contact bearings . They are the ready units which basic elements are rolling bodies - balls or rollers. In most cases the bearing coonsists of inner 1 and outer 2 rings, rolling elements 3 and a separator 4.



The industry delivers bearings in a wide range, both in the sizes, and on weight (external diameter from 1,5 to 2600 mm, weight from 0,5 g to 3,5 tons). The basic sizes of bearings are standardised.

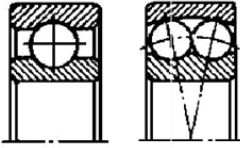
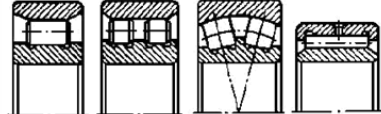
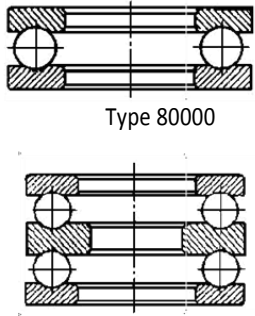
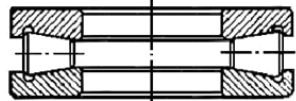
*The advantages of rolling contact bearings are:*

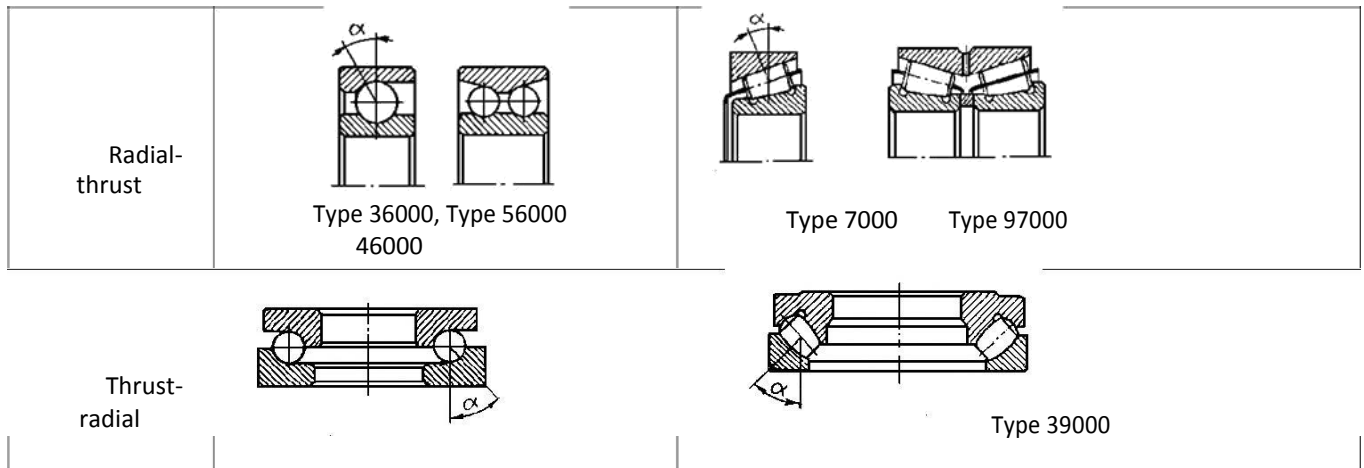
- Small losses on a friction that provides high efficiency;
- High bearing ability;
- Insignificant losses of lubricants;
- Small overall dimensions in an axial direction.
  
- Have a grater size in the radiial direction;
- The limited service life;
- Low ability to absorb shock loadings;
- The raised noise on high speeds of rotation.

**Classification of bearings**

Table 7.1

**The basic types of rolling contact bearings**

Type of bearings	The ball bearing	The rooller bearing
The radial	 <p>Type 0000    Type 1000</p>	 <p>Type    Type    Type    Type 2000    82000    3000    74000</p>
Thrust	 <p>Type 80000</p> <p>Type 38000</p>	 <p>Type 9000</p>



2.3.0.

#### Type 68000

#### **Bearings divide:**

1. According to the form of rolling elements - ball and roller.  
Roller, depending on the roller form, can be:  
With cylindrical, tapered, barrel-shaped, needle and helical rollers.
2. According to the direction of the restrained loading
  - Radial (for radial loadings);
  - Radial-thrust (for combined radial and axial loadings);
  - Thrust (for axial loadings);
  - Thrust-radial (for axial and radial loadings).
3. According to the number of rows of the rolling elements:  
Single-row, double-row, quadruple-row
4. According to the ability to compensate for shaft misalignment:  
Non-self-aligning and self-aligning (spherical), allowing a distortion up to
5. According to the load-carrying capacity (or to dimensions).  
At the same diameter of internal ring they divided on a series on the radial sizes:  
Extralight, very light, light, medium, heavy.
6. Depending on the width of the bearing:  
Especially narrow, narrow, normal, wide, especially wide.

Rolling contact bearings of light and medium series of normal width are most extended.

For rolling contact bearings there are 5 classes of accuracy:

Class of accuracy 0 6 5 4 2

Relative cost 1 1,3 2 4 10

In the general mechanical engineering the bearings of 0 class of accuracy are applied.

#### 2.1.2 Designation of bearings.

Designation marks on the end faces of rings of some figures and letters conditionally designating internal diameter, a series, type, a design version, a class of accuracy, etc.

Last two figures specify diameter of an inner ring of the bearing:

00 - d = 10 mm; 01 - d = 12 mm; 02 - d = 15 mm; 03 - d = 17 mm. For bearings since d = 20.495mm. - the size of internal diameter is defined by multiplication of last figures on 5.

The third figure on the right specifies a bearing series:

- Very light - 1, light - 2, medium - 3, heavy - 4 etc.

The fourth figure on the right designates bearing type:

- Radial ball single-row - 0 (if more to the left there are no figures 0 do not specified);

- Radial ball double-row - 1;

- Radial-thrust ball - 6;

- Roller helical (radial-thrust) - 7;

- Thrust ball - 8;

- Thrust roller - 9.

For example 7312 - diameter of an inner ring  $12 \times 5 = 60\text{mm.}$ , the helical roller bearing of a medium series.

The accuracy class is marked to the left of a bearing symbol

5 - 206 (0th class of accuracy is not designated).

### **Materials.**

Balls, rollers and rings make of steel ШХ - 15, ШХ - 15ГЦ, ШХ - 20ГЦ - high-carbon chromic steels; Separators - from a soft sheet steel by punching.

#### *2.1.3 The basic types of bearings*

1. Radial single-row ball-bearings - receive radial and axial loading  $F_a/F_r < 0,2, 0,25$

The most widespread in mechanical engineering, suppose a distortion -  $10^{-4}$ .

2. Radial ball double-row spherical -

are intended for receive of radial loading. Accept a distortion ( $2 \dots 3 \cdot 10^{-4}$ ).

3. Ball radial-thrust -

Receives radial and axial loadings. Radial load-carrying capacity of these bearings on 30-40 % more than in radial single-row ball-bearings.

4. The radial roller bearing with short rollers - perceives only radial loading. Loading ability on 70 % more than in the ball.

5. Helical roller bearings -

Are intended for receive of radial and axial loadings. Have load-carrying capacity in comparison with ball radial-thrust bearings.

6. Thrust ball - and roller bearings -

Receive only axial loading. Are applied at small and average speeds of rotation.

7. Needle - have rollers of the big length and small diameter. Do not accept axial loading. Can work at the big radial loadings.

#### *2.1.4 Selection of rolling contact bearings.*

Distinguish selection of bearings for static and dynamic basic load rating (at  $n > 10$  rpm).

Condition of selection  $C_{req} \leq C_{nominal}$  (catalogue)

The nominal basic load rating (C) is such constant loading which the bearing can sustain during 1million revolutions

without display of signs of fatigue

not less than in 90 % from certain number of the examined bearings. C - are shown in catalogues.

At designing it is necessary to be guided first of all by application of ball radial single-row bearings.

#### *2.1.5 Calculations of rolling contact bearings*

The rolling contact bearings, which frequency of rotation no more ( $\leq$ ) then 10 rpm calculates on the static basic load rating

$$R_0 \leq C_0,$$

Where -  $R_0 = X_0 R_r + Y_0 R_a$  designed static basic load rating;

$R_r$ ;  $R_a$  - Radial and axial loading on the bearing accordingly;  $X_0$ ;  $Y_0$  - factors of radial and axial loading accordingly.

The rolling contact bearings, which frequency of rotation is more then 10 rpm count on the base dynamic load rating C

$$C = P \sqrt[m]{L} \leq [C],$$

Where  $C$ ;  $[C]$  - dynamic load rating, rated and tabular accordingly; R - loading on the bearing; L - a service life in million revolutions;  $m = 3$  for ball,  $m = 10/3$  for roller bearings.

More often rolling contact bearings are calculated on durability

$$L = \left(\frac{C}{P}\right)^m \text{ (million revolutions).}$$

$$L = \frac{L \cdot 10^6}{60 \cdot n} \text{ (hours) - rated durability in hours, it should be not less than 10 thousand hours. }^h$$

Loading practically to calculate on the bearing not probably therefore it replace with the equivalent:

$$P_{екв} = (XV R_r + Y R_a) k_r k_\sigma,$$

Where - radial and axial loading on the bearing accordingly; X; Y - factors of radial and axial loading

accordingly; V - factor of a ring of the bearing

$(V = 1 \text{ if the internal ring } V = 1,2 \text{ works - if the external ring) works; } k_{\tau} - \text{temperature factor } k_{\tau} = 1 \text{ to } t \leq 80^{\circ};$   
 $k_{\sigma} - \text{Factor of safety } k_{\sigma} = 1 - \text{at uniform loading } k_{\sigma} = 1,1 - 1,2 - \text{loading with easy pushes } k_{\sigma} = 1,3 \text{ } 1,8 - \text{shock loadings.}$

For radial bearings as axial loading is not present.

Radial loading is equaled to total reactions of support. As Рєв pays off for each bearing separately,

$$R_{r1} = R_{\Sigma 1} \quad R_{r2} = R_{\Sigma 2} \quad (\text{calculation of shafts on static strength see}).$$

Factors X and Y find behind a parity which compare to value of a table value e - factor of axial loading. For ball radial-thrust bearings with a angle

$\alpha > 180$ , and roller radial-thrust bearings at calculation axial components are considered:

$$S_1 = eR_1; S_2 = eR_2 \quad \text{- For radial-thrust ball bearings;}$$

$$S_1 = 0,83eR_1; S_2 = 0,83eR_2 \quad \text{- For radial-thrust roller bearings.}$$

### 2.1.6 Lubrication of bearings.

Lubrication of bearings essentially influences on it durability. It reduces a friction, reduces contact stresses, protects from corrosion, promotes of bearings cooling.

For lubrication are applied plastic (dense) greases and liquid oils. Liquid lubrication is more effective for cooling and reduction of losses. The excessive quantity of lubrication worsens work of bearings. It is necessary to protect carefully the bearing units from the dust and dirt ingress. Otherwise the durability sharply decreases.

#### **Principal causes of availability loss.**

Fatigue spalling - after their long operation in normal conditions.

Wear - at insufficient protection against abrasive particles (a dust, a dirt), is a destruction principal type in cars and other mechanisms.

Destruction of separators - especially at high-speed machines.

Splitting of rings and rolling elements - shock and vibrating loadings (wrong installation, a distortion, jamming).

Residual deformations - in hard loaded slow-speed bearings.

**Lecture № 8**  
**Demountable joints.**

**Key joints.**

**1. The general data.**

Keyed joints, serve for fastening of details on axes and shafts. Key and gear hubs (a pulley or other detail) consist of a shaft.

Advantages - simplicity of a design and comparative ease of installation and disassembling.

Lacks - key grooves, weaken a shaft and a hub.

All keyed joints are subdivided on strained and not strained:

- Strained - wedge and tangential;
- Not Strained - prismatic and segment.

Materials of key - alloyed or carbon steels with  $\sigma_B$  not more than 500 MPa.

For manufacturing all kinds of key use rolled sheets from the carbonaceous or alloyed steel with strength  $\sigma_B$  not less than 500 MPa and  $[\sigma_{3M}] = 100$  MPa. However strength of key should be less than strength of a material of a shaft and details which are located on it.

**2. Connection by wedge key (fig. 3 c).**

Working sides are wide sides of a key. Mounting displaces the centres of a shaft and a hub on size.

Wedge keys are made with heads and without heads. They are applied in slow-speed transmissions (in the conditions of modern manufacture their application is reduced).

**3. Connection by prismatic key.**

The basic keyed joint in mechanical engineering demands manufacturing of a shaft and a key seating with the big accuracy.

The moment is transmitted from a shaft to a hub by lateral narrow sides of a key. Thus there are stresses of a bearing  $\sigma_{3M}$ .

For purpose of calculation simplification, accept that the key embedded in a shaft on half of the height, stress  $\sigma_{3M}$  is distributed in regular intervals on height and length of a key, and in longitudinal section of a key - stress on a shearing  $\tau$ .

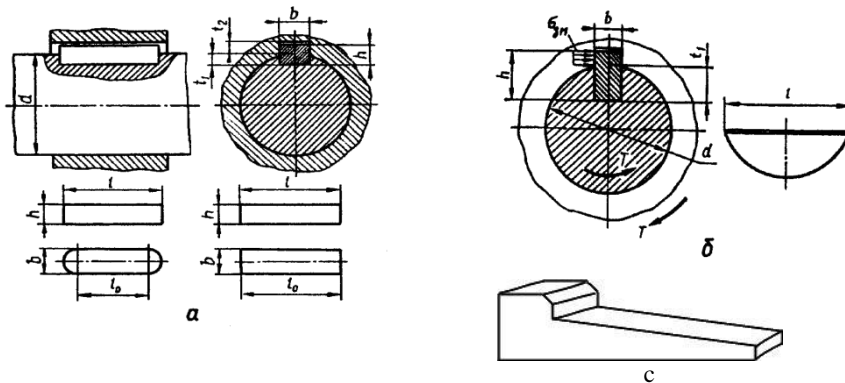


Fig. 3

Prismatic key (fig. 3 a,) can be with round end faces, therefore length of the key, which lateral surface  $= -b$

transmits twisting moment count under the formula  $p$ . According to the standard width  $b$  and height  $h$  of the key choose depending on diameter of a shaft. The standard also regulates depth of a groove on a shaft  $t_1$  and in the plug  $t_2$ . Length of a key  $l$  choose on width of the detail located on a shaft.

Segment key (fig. 3 b,) are used mainly for connection of details on poorly loaded sites of shafts as the deep groove considerably weakens a shaft.

At designing of a keyed joint width and height of key accept on corresponding state that depending on diameter of a shaft. Length of a key assumed depending on length of a hub and coordinate which state volume of the key. Correctness of the chosen sizes of a key are checked by corresponding calculations on strength. Hence, calculation of keyed joints on strength carry out as verifying.

Calculation key on strength of a shearing and a bearing.

- On a shearing:

$$\tau = \frac{F}{A} \leq [\tau] \quad F = \frac{2T}{d}; A = b \cdot p$$

$$\tau = \frac{2T}{db} \leq [\tau]$$

But in practice there are used calculation on strength of a bearing:

$$\sigma = \frac{F}{A} \leq [\sigma] \quad ; F = \frac{2T}{d}; A = (h - t) \cdot p$$

$$\sigma = \frac{2T}{d(h - t)} \leq [\sigma]$$

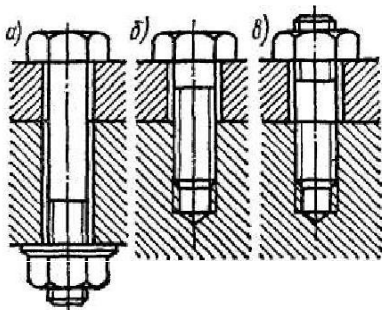
If as a result of key calculation it will be it's overloading, two or three key are provided. Two prismatic key are installed under an angle 180°, and three prismatic key or two wedge - under an angle 120°.

### Threaded joints. The general data.

Threads named such joints which are carried out by means of the details having a thread. Threaded joints refer to demountable joints, i.e. probably disassembly of the joint details without their damaging. Wide usage of threaded joints in mechanical engineering is caused by their simplicity, high bearing ability, reliability, and also convenience of connection and separation of details.

#### 1. The basic types of fixing elements.

For connection of details there are used bolts (fig. a), screws (fig. b) or pins (fig. c).



**Bolt** is a continuous cylindrical core with a hexahedral head on the one hand, and a thread on another. Bolts may be with hexahedral, round, square, rectangular heads; special and a general purpose; the normal and raised accuracy. Special, base and eyes-bolts.

**Nut** is a detail with an inner thread. They have very various configuration (with one or with two facets, high, low, castle nut

etc.) Between a nut and connected details enclose washers.

The **washer** is a round plate, a small thickness with a hole under a bolt core, they also are very various. They are used for increasing a basic surface and prevention of a detail from scores.

The **screw** is a bolt, with one of connecting details is a nut. Depending on the sizes and destination screws have various heads (hexahedral, round, secret under a screw-driver, etc.).

**Pin** - the cylindrical core having a thread from both ends. They are applied when length of a bolt is less than thickness of connected details. Pins are usual happened with groove.

All details of threaded joints are standardised.

#### 2. Thread profiles.

Depending on the surface for m on which the thread is cut, distinguish cylindrical and helical threads. The contour of section of a thread in a plane which is passing through its axis is called a **thread profile**. The thread profiles may be: triangular (fig. a), buttress (fig. b), trapezoid (fig. c), rectangular (fig. d), round (fig. e).

#### 3. Methods of a thread manufacturing:

- Manually in individual manufacture;



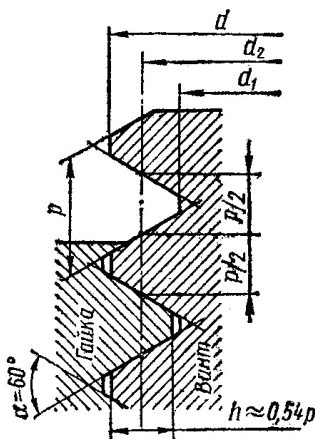
- Threading on screw-cutting lathe;  
- Milling on special machine tools; -  
Rolling on special machine tools;

- Extrusion in thin-walled and stamped products from a tin and plastic.

4. Geometrical parameters

The thread parameters are:

- A thread step ( $p$ );
- A angle of a screw line
- External diameter of a
- Internal diameter of a
- Mean diameter of a For
- the rated area of cross-



of a thread.

lifting on its mean diameter (fig.);

thread  $d$ ;  
 thread  $d_1$ ;  
 thread  $d_2$ .  
 section section is

$$A = \frac{\pi \cdot d^2}{4}$$

threads.

5. The basic types of

5.1. Fixing threads:

- Metric with a triangular profile ( $\alpha = 60^\circ$ ) is the basic fixing thread (for fixing metric threads  $n = 1$  - number of a thread starts)

M 16 x 1,5 metric thread with small step  $p = 1,5$  mm and external diameter  $d = 16$  mm.

The metric thread can be with the normal step  $p = 2,5$  mm, the small steps: 2, 1,5, 1,0, 0,75 and 0,5 mm.

- Pipe with a triangular profile ( $\alpha = 55^\circ$ ), tops and hollows are rounded off;
- A round thread. The profile of heads is formed the circles interfaced by arches, a profile angle  $\alpha = 30^\circ$ .
- A thread for screws from wood.

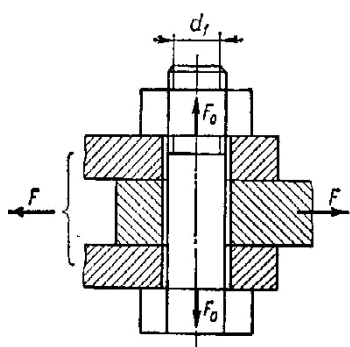
5.2. Running threads (screw mechanisms):

- Rectangular - it is threaded on screw-cutting lathe;
- Trapezoid symmetric - the baasic for screw-nut transmissions, and worm gearings;
- Buttress - in press, cargo hooooks with unilateral loadings.

The choice of a thread profile, depending on its application, is defined by many factors. The main of them are strength, adaptability to manufacture and force of a friction in a thread. For example, the fixing thread should have high strength and high friction force to warn unscrewing. Threads of screw mechanisms should be with a small friction to raise efficiency, and to reduce damage.

The basic material of threaded joints is the carbon steel 2; 3; - Steel 40. For special connections sometimes there are used alloyed steel 40X; 30XH; 40XH and others.

a) Calculation of the tighten bolt with a gap and loaded by shear



Condition of the shear absence in jointed details:

$$F \leq F_T = F_{zam} f l$$

were  $F$  is external shear force;  $F_T$  - friction force in one plane of details junction;  $f$  - slipping friction factor;  $l$  - number of pairs of joint plane.

Calculate the bolt rod from tightening force

$$F_{zam} \geq \frac{F}{f l}$$

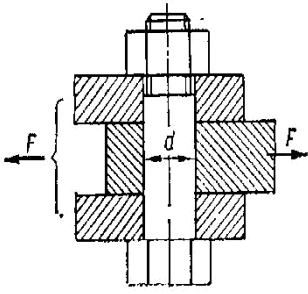
$$\sigma_p = \frac{F_{zam}}{\frac{\pi d_1^2}{4}} = \frac{1,3 \cdot 4F}{\pi d_1^2 f l} \leq [\sigma]$$

$$d_1 \geq \sqrt{\frac{4 \cdot 1,3 F}{\pi [\sigma] f l}}$$

The main disadvantage of such connection is that the connection is immovable, if the tightening force is 5-10 times greater than the external shear force.

b) Calculation bolts installed inn the hole without gap and loaded by external forces





External force  $F$  is transmitted directly to the bolt, because friction between the parts are not included, and tightening the bolt is not required. Bolt in this joint is calculated for strength condition on the shear:

$$\tau_{3p} = \frac{F}{A_{3p}} = \frac{4F}{\pi d_0^2 z} \leq [\tau]_{3p}$$

$$d_0 \geq \sqrt{\frac{4F}{\pi [\tau]_{3p} z}}$$

Also, the can be calculated for strength condition on bearing if one of the joined parts has a much smaller thickness ( $\delta_{\min}$ ) in comparison with other details.

$$\sigma_{3M} = \frac{F}{d \delta_{\min} z} \leq [\sigma]_{3M}$$

