

V. Dotsenko, Yu. Koveza

DESIGN OF MACHINE ELEMENTS

2021

MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE
National Aerospace University
«Kharkiv Aviation Institute»

V. Dotsenko, Yu. Koveza

DESIGN OF MACHINE ELEMENTS

Tutorial

Kharkiv “KhAI” 2021

UDC 621.81.001.66(075.8)
D76

Стисло англійською мовою викладено класичний курс «Деталі машин і основи конструювання» відповідно до галузі знань "Авіація і космонавтика". Особливу увагу приділено спеціальним вимогам до деталей літальних апаратів – вимогам мінімальної маси і габаритних розмірів, високої надійності, довговічності, питомої міцності, працездатності – у різних, у тому числі в екстремальних умовах.

Для студентів машинобудівних вузів.

Reviewers: Candidate of science A.V. Ustynenko,
Candidate of science E. V. Pavlyuk

Dotsenko, V.

D76 Design of machine elements [Digital resource] : tutorial / V. Dotsenko, Yu. Koveza. – Kharkiv : National Aerospace University «Kharkiv Aviation Institute», 2021. – 224 p.

The classic course "Design of machine elements" according to the field of knowledge "Aviation and cosmonautics" is stated in a concise form in English. Particular attention is paid to special requirements for aircraft parts - the requirements of minimum weight and overall dimensions, high reliability, durability, specific strength, workability - in different, including - in extreme conditions.

For students of engineering universities.

Figs 244. Tables 25. References: 10 titles

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INTRODUCTION

The purpose of the «Design of Machine Elements» course is the first acquisition of engineering skills in calculation and designing of standard parts and machine parts based on theoretical knowledge.

The subject of DME is the creation of new and better machines and improving the existing ones. A new or better machine is one, which is more economical in the overall cost of production and operation. The process design is a long and time-consuming one. From the study of existing ideas, a new idea has to be conceived. The idea of is then studied keeping in mind its commercial success and given shape and form in the form of drawings. In the preparation of these drawings, care must be taken of the availability of resources in money, in men and in materials required for the successful completion of the new idea into an actual reality. In designing a machine component, it is necessary to have a good knowledge of many subjects such as Mathematics, Engineering Mechanics, Strength of Materials, Theory of Machines, Workshop Processes and Engineering Drawing.

Course Terminology

Designing a product means defining its shape, size and arrangement of product individual components, as well their functional interaction.

Designing is creating of a product IMAGE (not the product itself).

This image of the material object is commonly presented in the form of clear and readable drawings, descriptions, notes, symbols and technical specifications. Designing may be classified as

qualitative – it is defining components and assemblies shape, their arrangement and interaction;

quantitative – it is determining of size and number of elements. Means of quantitative design:

- calculation;
- reference tables;
- experiment;
- professional experience and intuition.

According to the Oxford English Dictionary, machine is an independently functioning structure.

We will consider **Machine** as a technical system, which produces useful work and is mainly characterized by flow of energy and conversion of energy.

Machines are usually a source of mechanical energy, which is obtained from any other: electrical, chemical, potential, etc.

Assembly (assembly unit) is a set of interacting components, connected together for a common functional purpose (gearbox, coupling, bearing, Fig. I.1).

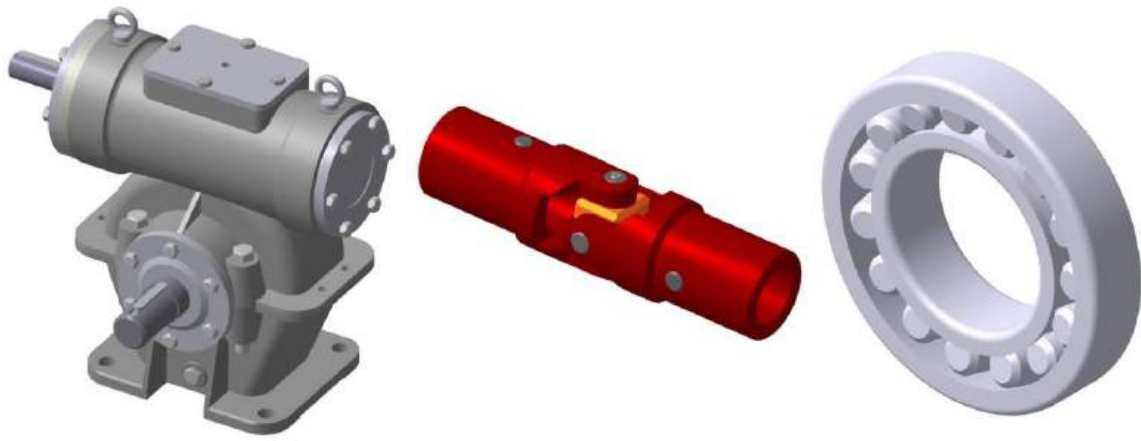


Fig. I.1

Mechanism is a structure or arrangement constructed of individual machine components and sub-assemblies.

Element (part, component) is a product that is manufactured without using assembly operations (for example, a shaft, a bolt, a nut, Fig. I.2).



Fig. I.2

General (common) parts perform almost the same functions in any machine (gears, bolts, bearings, etc.)

Specific ones are used in specialized types of machines.

Usually there are fewer specific parts in a machine than the common parts.

Course Goals

During the course studying, many difficulties occur. They are:

1. A lot of empirical equations and correcting factors.
2. Multi-variant solutions.
3. The need to use standard parts and solutions.
4. The need to find a solution that satisfies conflicting requirements.

That is why after finishing of our course studying a student must be able to:

I. Transit mentally from a diagram/drawing to the structure and vice versa (Fig. I.3).

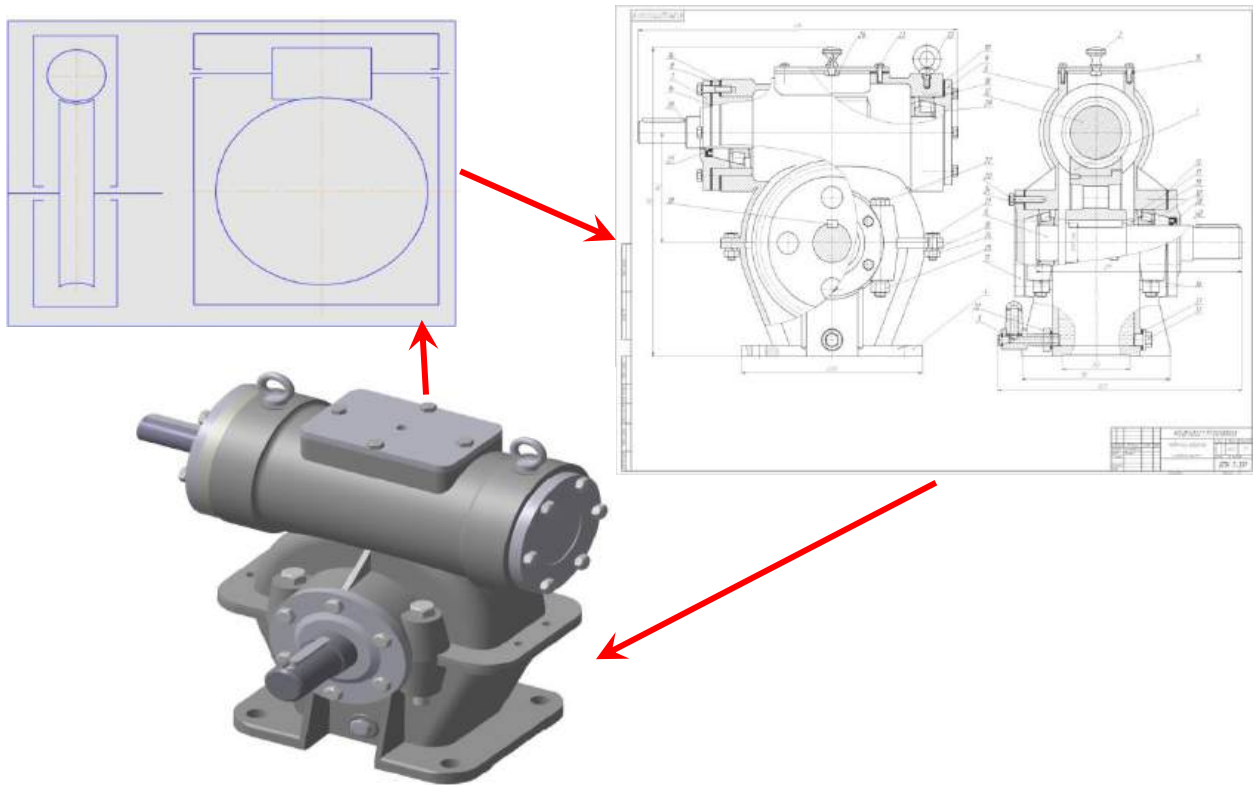


Fig. I.3

II. Size machine components and assemblies. It means:

- being able to visualize flow of energy through the structure and loads distribution;
- being able to determine types of loads on the elements and decide on safety conditions for them;
- knowing the right sequence of major components' calculation;
- knowing applicable load cases and applicability and limits of calculation methods.

In common designs, a number of components to be sized averages up to 20 %, and in all the rest are deal with the next requirement.

III. Select materials and standard elements using reference books, i.e.

- knowing where everything is;
- being able to use the standard tables;
- knowing and being able to find the governing parameters for the rest of them;
- being able to “read” coding of the common components.

IV. Generate calculations and design documentation in accordance with applicable regulations:

- being able to work in any of the graphics programs;

- being able to competently and reasonably choose specifications for parts manufacturing, tolerances and fits for general joints;
- being able to provide a technical description of the product being designed;
- competently describe and justify engineering decisions.

V. Design parts and assemblies using the basic principles of efficient design. They may be written as follows 10 precepts:

1. **Squeeze out maximum possible** – subdue construction increased economic benefit, defined primarily useful impact machine, its durability and running costs for the entire period of use of the machine.

2. **Don't seek strength in weight.** Provide high strength parts and machine parts in general imparting rational forms with the best use of the material, the use of high-strength materials and the introduction of a hardening treatment.

3. **Think of humans.** This means:

- ensure the safety of operating personnel,
- prevents accidents by maximum automation of working operations,
- introduce closed mechanisms and installation of safety barriers;
- facilitate the maintenance of the machine and its contents;
- focus management and control bodies as possible in one place, easy to view and manipulate;
- do accessible and convenient for exploring the sites and mechanisms requiring periodic inspection.

4. **KISS principle.** KISS is an acronym for "Keep it simple, stupid" as a design principle noted by the U.S. Navy in 1960. The phrase has been associated with aircraft engineer Kelly Johnson, lead engineer at the Lockheed Skunk Works (creators of the Lockheed U-2 and SR-71 Blackbird spy planes, among many others).

While popular usage has translated it for decades as, "Keep it simple, stupid", Johnson translated it as, "Keep it simple stupid" (no comma), and this reading is still used by many authors. There was no implicit meaning that an engineer was stupid; just the opposite.

The KISS principle states that most systems work best if they are kept simple rather than made complicated; therefore, simplicity should be a key goal in design and unnecessary complexity should be avoided.

An alternative view - "Make everything as simple as possible, but not simpler." - is attributed to Albert Einstein, although this may be an editor's paraphrase of a lecture he gave.

5. **Save on everything, but keep the essential.** Seeking, as a rule, to the cheapness of manufacture is not limited to the cost of manufacturing of key components; perform such details from quality materials used for their production processes, providing the greatest increase in reliability and durability.

6. **Don't re-invent the bicycle.** Use an experience full of the structures, related experience, and, where necessary, on the profile and remote engineering industries.

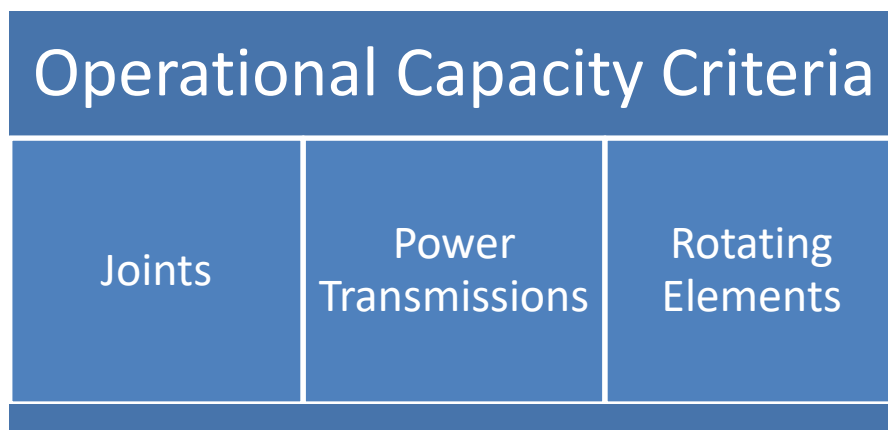
7. **Make beautifully.** A machine must have simple, slim and sleek exterior shape, comply with the requirements of technical aesthetics, improve the finish machines. Remember that ugly plane will not fly!

8. **Don't put all your eggs in one basket.** Book a duplicated or vital functions of the machine.

9. **Don't expect mercy from the Nature.** Expected the worst, even fantastic event in determining the unknown parameters and loads, because the reality is going to get worse.

10. **Memento more.** Design of parts and assemblies and their resources should not be eternal, these should be commensurate with the resources of the whole machine. Remember those who will disassemble your machine after its resource.

The course consists of three big parts – “Joints”, “Power transmissions” and “Rotating elements” – and section “Operational Capacity Criteria” which unites all these parts:



The division into parts is rather arbitrary, since some sections are considered in several of them. So, for example, power screws have common approaches in “Power Transmissions” and in “Joints”.

Methodical literature for the course is located here:



Tutorials

Part 1. OPERATIONAL CAPACITY CRITERIA

A machine under design should satisfy various requirements outlined in specifications. These are its output capacity, its cost – initial and operational, its weight and expected (guaranteed) service life. In certain cases, additional requirements are made concerning overall dimensions and transportability, uniformity of rotation, noiseless operation, simple and easy control, etc.

Some of these requirements are so essential that the normal operation of a machine becomes impossible if they are not satisfied; these requirements are regarded therefore as the criteria of operating capacity.

The main are:

- strength;
- stiffness;
- wear resistance;
- heat resistance;
- vibration resistance.

1.1. Strength

Strength is ability of assembly, part or element to withstand a load without destruction. It is considered by load type as a constant (permanent, static) (Fig. 1.1, *a*) and a cyclic one (Fig. 1.1, *b*), by location as a volume (Fig. 1.1, *c*) and a surface strength (Fig. 1.1, *d*).



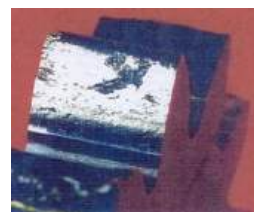
a



b



c



d

Fig. 1.1

Usually strength is the main criterion, i.e. the task of providing the necessary strength is to determine the size and forms of machine parts, excluding the possibility of an unacceptably large residual strain, premature breakage and surface damage.

To determine or to show the dependence between the actual and the limit strength, we can denote the strength condition. Strength condition is inequality, which shows relationship between the actual load, stress, deformation or safety factor and their allowable value. For example:

$$\sigma \leq [\sigma] ; S \geq [S].$$

Safety factor is a ratio between limit stress and maximum actual value:

$$S = \frac{\sigma_{lim}}{\sigma_{max}},$$

where σ_{lim} is a limit stress (for ductile metals – the yield stress σ_y ($\sigma_{0,2}$); for brittle materials – the ultimate stress σ_{UTS} (Fig. 1.2).

The allowable stress may be obtained as follows:

$$[\sigma] = \frac{\sigma_{lim}}{[S]},$$

where $[S]$ is the allowable safety factor which is usually known from experience and provided by handbook tables.

Let us consider strength conditions for the simplest deformations:

- tension-compression;
- bearing;
- bending;
- shear;
- torsion.

1. **TENSILE/COMPRESSIVE** normal stresses are developed in a specimen when two forces act along one line in opposite directions (Fig. 1.3).

If $l_{ts} > l_0$ it is a tension, if $l_{ts} < l_0$ – compression.

Strength condition is

$$\sigma_{tc} = \frac{F}{A_0} \leq [\sigma_{tc}].$$

For hollow circular cross-section area is

$$A_0 = \frac{\pi d_{ex}^2 (1 - \alpha^2)}{4} \approx 0,78 d_{ex}^2 (1 - \alpha^2),$$

where hollowness factor $\alpha = d_{in} / d_{ex}$ (see Fig. 1.3), d_{ex} is external specimen diameter before loading.

For rectangular cross-section area $A_0 = bh$.

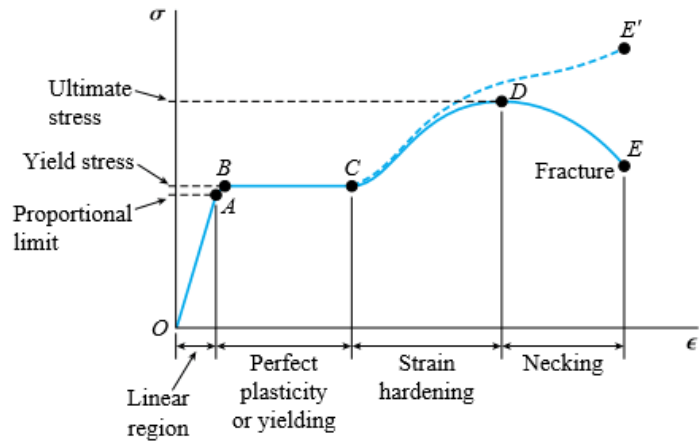


Fig. 1.2

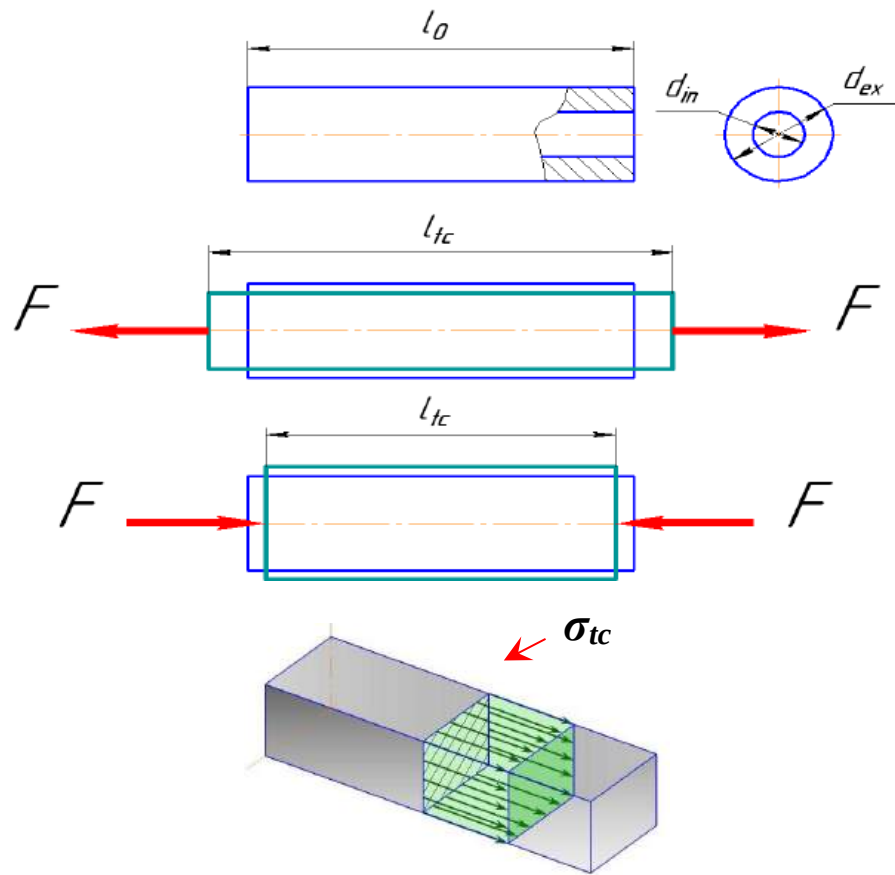


Fig. 1.3

2. **BEARING** stress is caused by one part acting directly onto another (Fig. 1.4). Bearing stress is a normal compressive stress and is equal to the bearing force divided by the bearing area.

In typical connections (by bolts, pins or rivets) we are mainly interested in the average bearing stress which is the bearing force divided by the projected area of the fasteners.

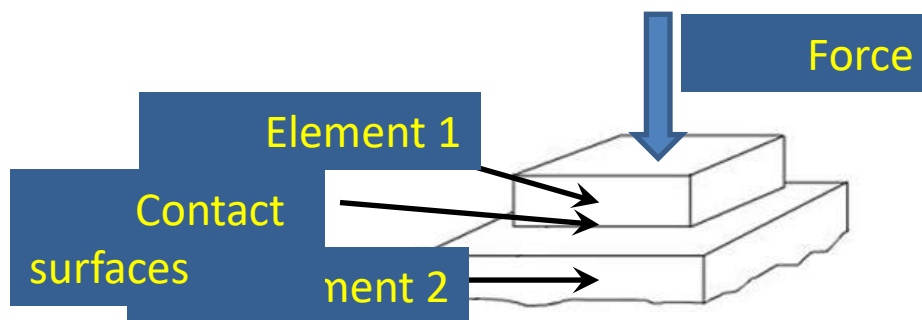


Fig. 1.4

Strength condition is

$$\sigma_b = \frac{F}{A_0} \leq [\sigma_b].$$

Here A_0 is the projected area before loading.

For example, let us consider riveted joint (Fig. 1.5). In this case the projected area of the fastener is $A_0 = dh$ instead of actual half-circumferential area $A = \pi dh / 2$ (Fig. 1.6).

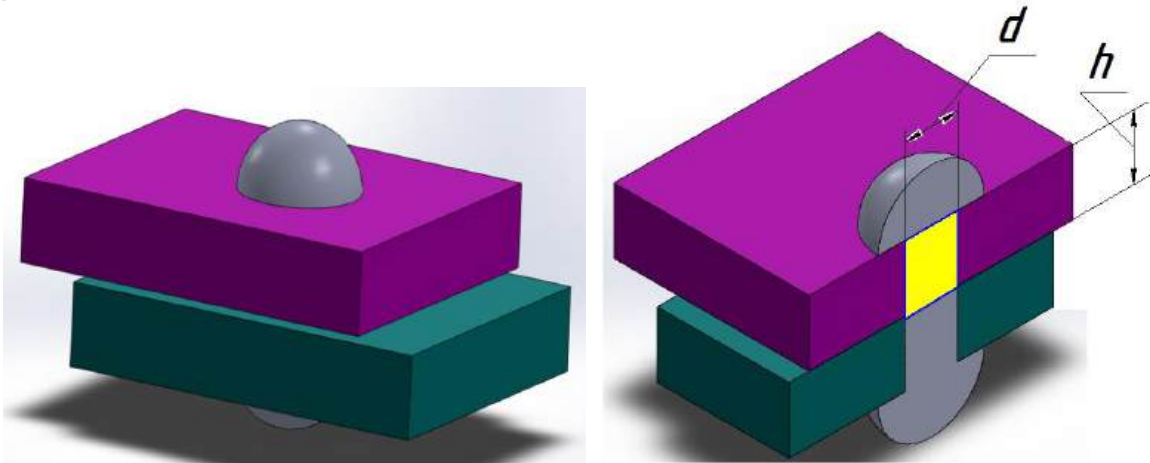


Fig. 1.5

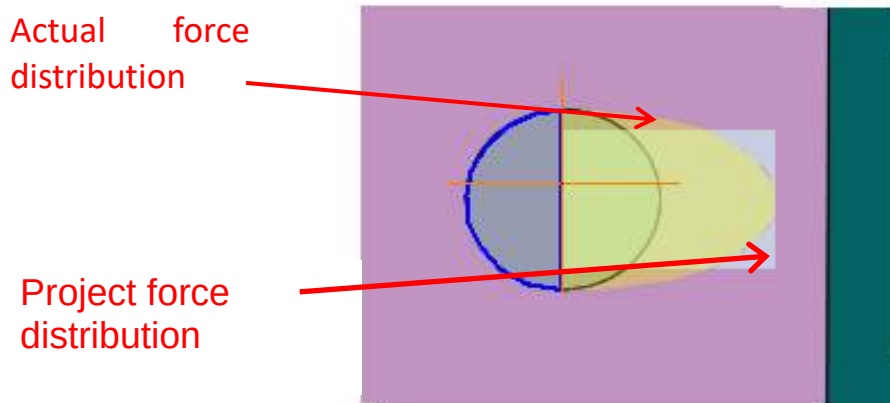


Fig. 1.6

3. The average **SHEAR** stress is defined as a force, which action line is parallel to a shear plane, divided by its cross-sectional area (Fig. 1.7). Strength condition is:

$$\tau_s = \frac{F}{A_0} \leq [\tau_s].$$

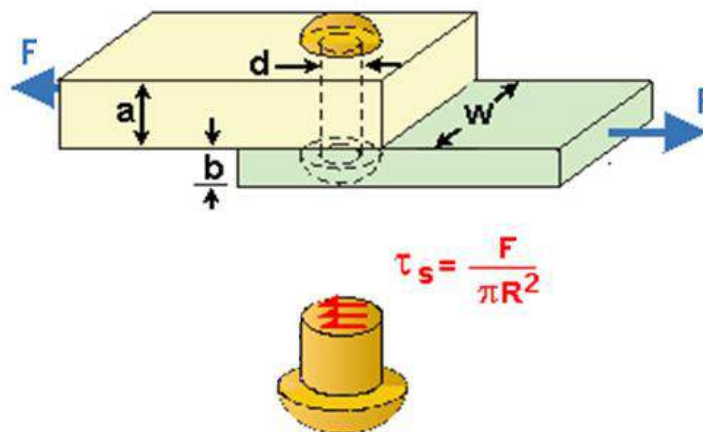


Fig. 1.7

4. **BENDING** normal stresses are developed in a beam when an external load is applied or has a component which is perpendicular to the longitudinal axis of the beam (Fig. 1.8).

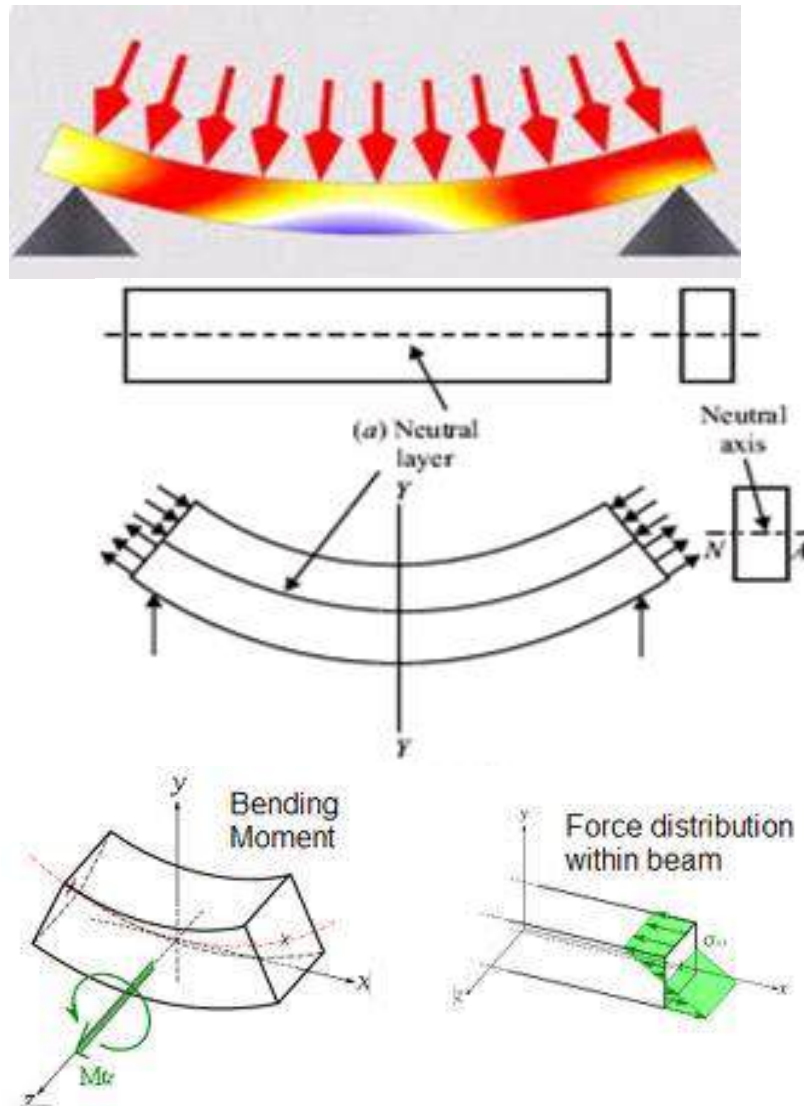


Fig. 1.8

5. Bending normal stress equals to bending moment appearing due to the external load divided by cross-section modulus. Strength condition is:

$$\sigma_b = \frac{M}{W_0} \leq [\sigma_b].$$

Section modulus: for tubular cross-section $W_0 = \frac{\pi d^3 (1 - \alpha^4)}{32} \approx 0,1 d^3 (1 - \alpha^4)$;

for rectangular cross-section (Fig. 1.9): $W_0 = \frac{bh^2}{6}$.

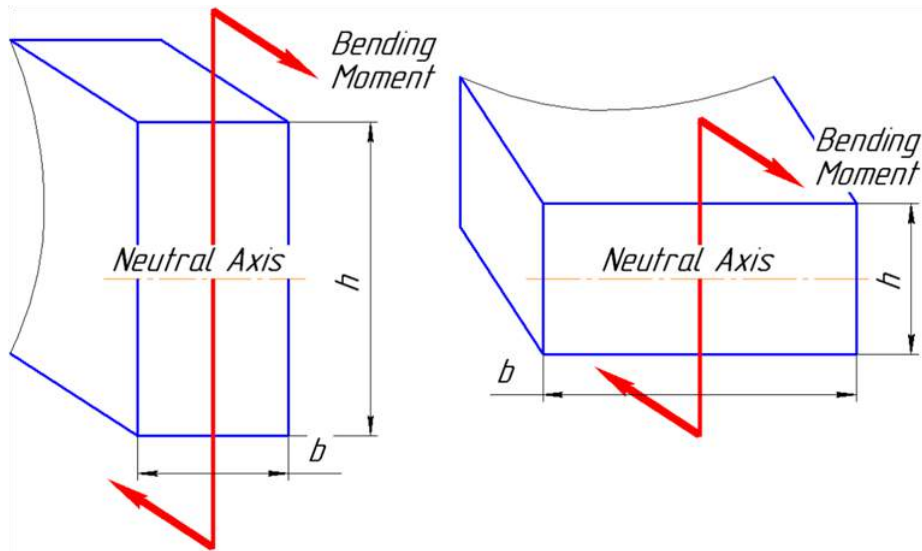


Fig. 1.9

6. **TORSION** develops shear stresses in cross-sectional area of a specimen (Fig. 1.10). Strength condition is:

$$\tau_{tr} = \frac{T}{W_{\rho}} \leq [\tau_{tr}].$$

Polar section modulus for tubular cross-section is

$$W_{\rho} = \frac{\pi d^3 (1 - \alpha^4)}{16} \approx 0,2 d^3 (1 - \alpha^4).$$

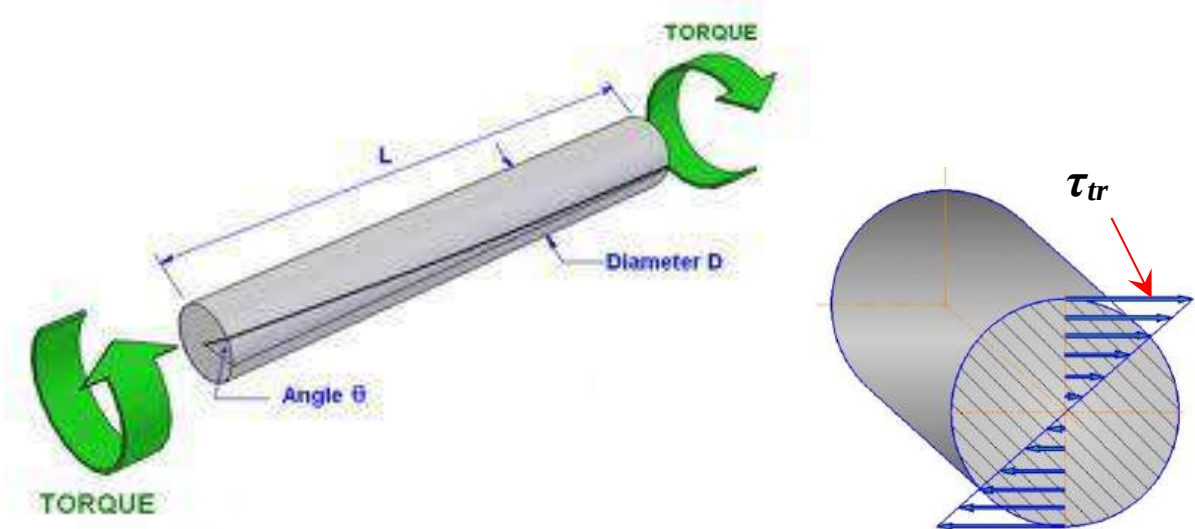
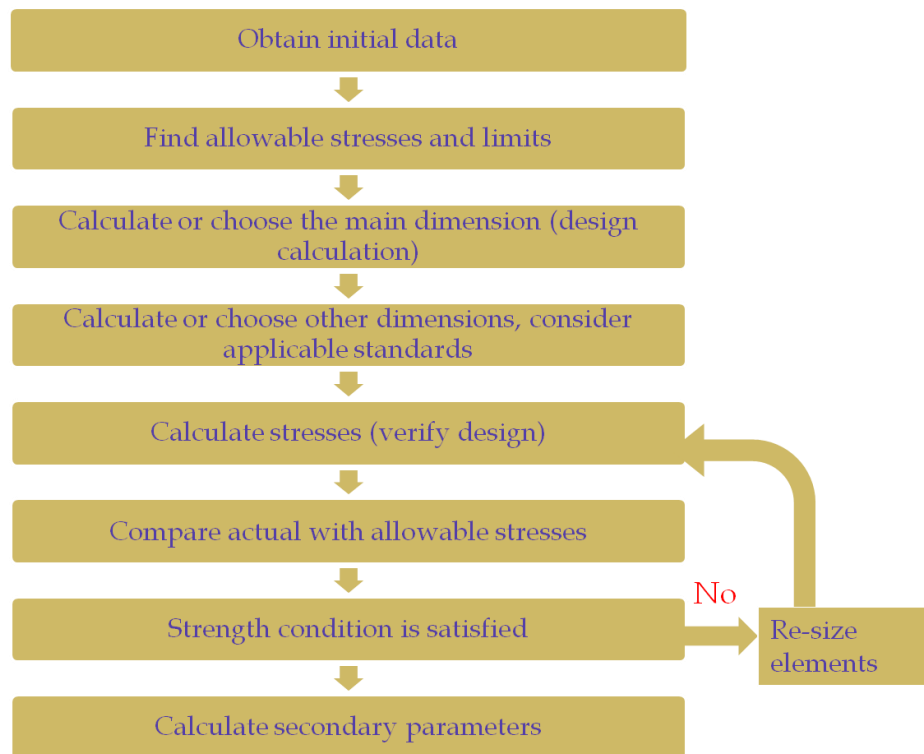


Fig. 1.10

Sequence of Quantitative Design



Fatigue Analysis

Many of the machine parts (such as axles, shafts, springs, gear teeth etc.) are subjected to variable or alternating loads (also known as fluctuating or fatigue loads).

Calculation of endurance (sometimes the main calculation) takes place in the form of checking a safety factor at a summary action of bending and torsion:

$$S = \frac{S_{\sigma} S_{\tau}}{\sqrt{S_{\sigma}^2 + S_{\tau}^2}} \geq [S],$$

where S_{σ} is the actual safety factor for normal stress (without considering torsion); S_{τ} is the actual safety factor for shear stress (without considering bending, compression or tension):

$$S_{\sigma} = \frac{\sigma_{-1}}{K_{\sigma D} \sigma_a + \psi_{\sigma} \sigma_m}, \quad S_{\tau} = \frac{\tau_{-1}}{K_{\tau D} \tau_a + \psi_{\tau} \tau_m}.$$

The MAIN CYCLE PARAMETERS are (Fig. 1.11):

1. Mean or average stress

$$\sigma_m = (\sigma_{min} + \sigma_{max})/2.$$

2. Amplitude stress

$$\sigma_a = (\sigma_{max} - \sigma_{min})/2.$$

3. Stress ratio r is the algebraic ratio of minimum stress to maximum stress

$$r = \sigma_{min} / \sigma_{max}.$$

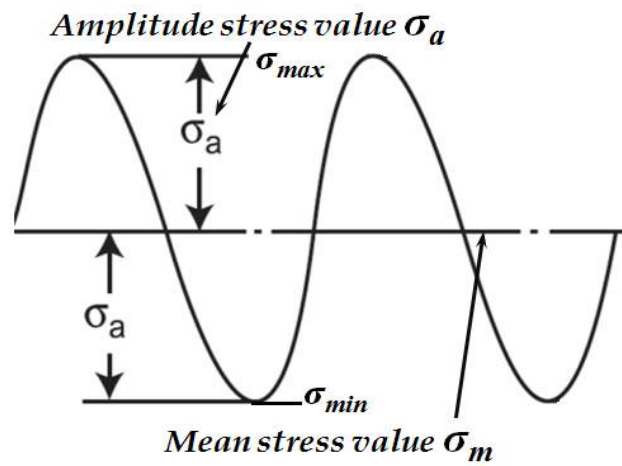


Fig. 1.11

In case of a REVERSED CYCLE (Fig. 1.12) $r = -1$; $\sigma_m = 0$; $|\sigma_{min}| = \sigma_{max} = \sigma_a$.

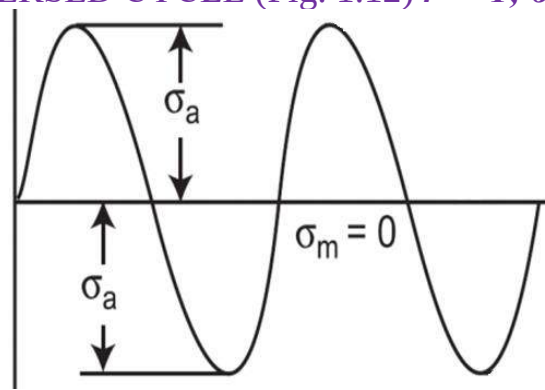


Fig. 1.12

In case of a REPEATED CYCLE (Fig. 1.13) $r = 0$; $\sigma_m = \sigma_a$; $\sigma_{min} = 0$.

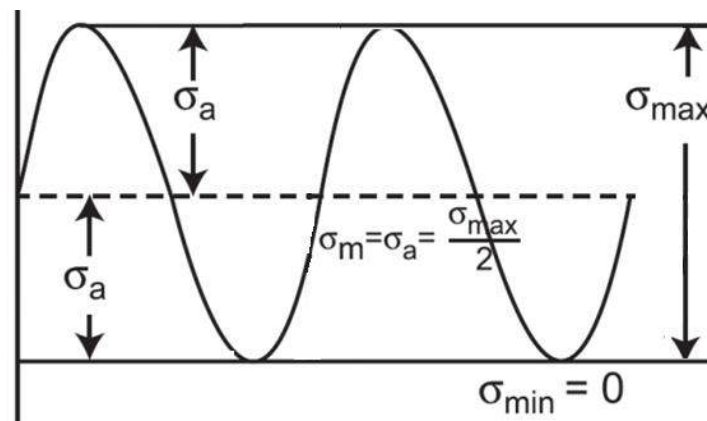


Fig. 1.13

In case of a FLUCTUATING CYCLE (Fig. 1.14) $r > 0$;

$$\sigma_m = (\sigma_{min} + \sigma_{max})/2;$$

$$\sigma_a = (\sigma_{min} - \sigma_{max})/2.$$

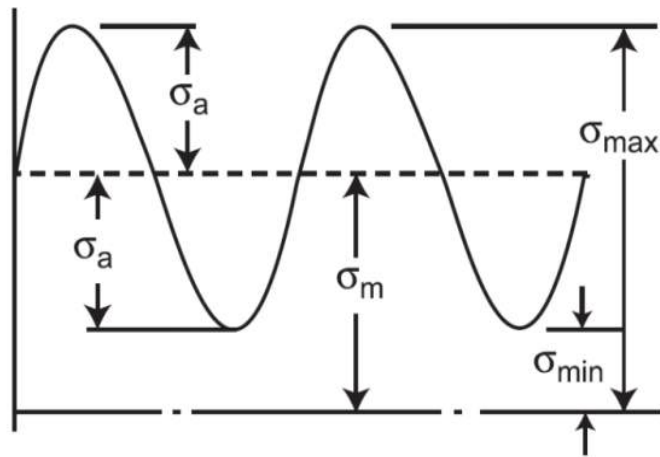


Fig. 1.14

1.2. Stiffness

Stiffness C is the property of a structure to resist deformation under applied force:

$$C = \frac{EA}{l_0} = \frac{1}{\lambda}.$$

The reciprocal factor of stiffness is the flexibility λ :

$$\lambda = \frac{1}{C} = \frac{l_0}{EA}.$$

In case of TENSION/COMPRESSION specimen elongation is:

$$\delta = \frac{Fl_0}{EA} = \frac{F}{C} = F\lambda \text{ and } F = C\delta \Rightarrow C = \tan \alpha \text{ (Fig. 1.15).}$$

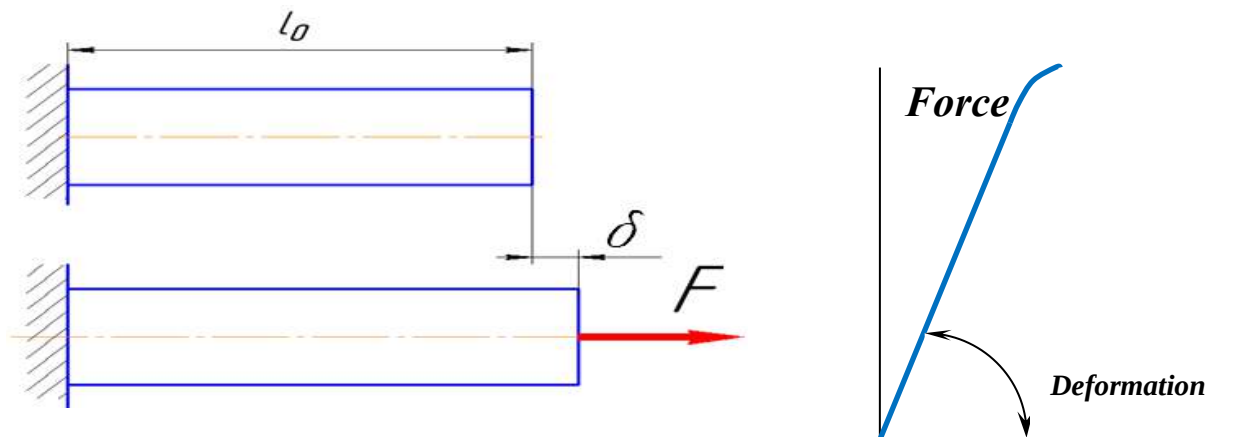


Fig. 1.15

Beam deflection on BENDING (Fig. 1.16) is

$$y = \frac{Fl^3}{KEJ} \leq [y],$$

where K depends on load and relative location of supports; J is the moment of inertia.

Angle of deformation slope θ can be obtained from the Strength of Materials equations.

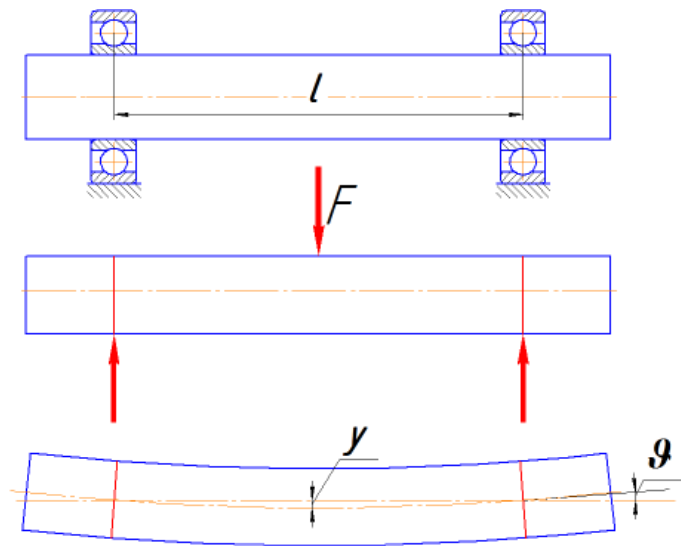


Fig. 1.16

The angle of twist on TORSION (Fig. 1.17) is

$$\varphi = \frac{Tl}{GJ} \leq [\varphi],$$

where G is the modulus of rigidity; J is the polar moment of inertia (the second moment).

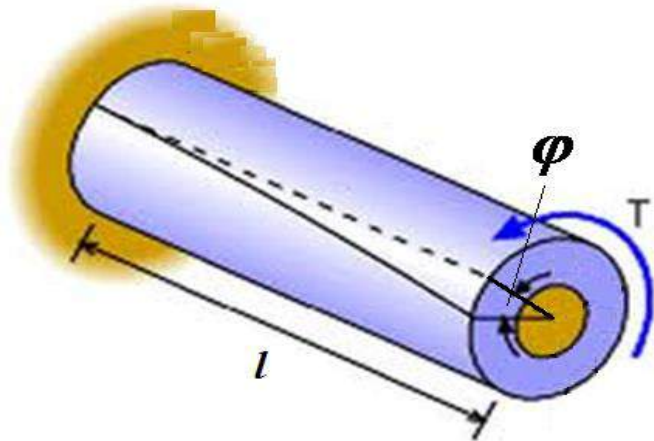


Fig. 1.17

Ways to increase stiffness are:

- reduce moment arms of bending or twisting forces;
- add more supports;
- use more effective cross-sections to resist bending or torsion (Table 1.1);
- reduce part's length.

Table 1.1

Cross-section	Bending		Torsion	
	W	J	W	J
Round solid	0,14	0,08	0,28	0,16
Rectangular solid	0,235	0,167	0,175	0,115
Round tubular	0,39	0,36	0,78	0,72
T-section	1,14	2,08		

1.3. Wear resistance

Wear is the process of gradual change in the surface proportions of a component due to friction. The three stages of typical wear process are (Fig. 1.18):

1. Running-in period.
2. Steady-state wear period (normal operation).
3. Wear out period (intensive wear).

Ways to increase wear resistance are:

- decrease load;
- lubricate rubbing surfaces;
- use antifriction materials to prevent rapid wear of the most expensive elements;
- use heat and thermo-chemical treatment (hardening of a surface layer, nitriding, etc.) and various coatings.

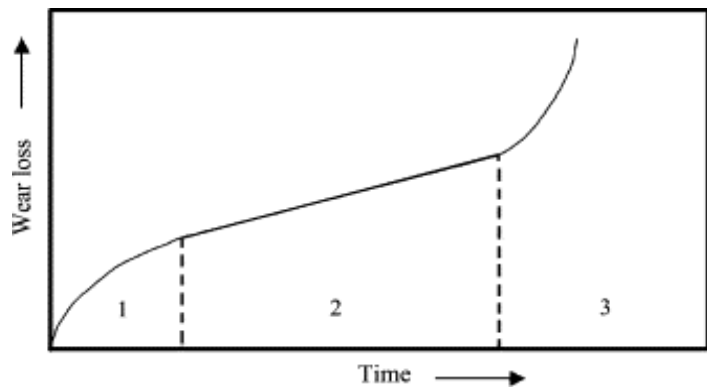


Fig. 1.18

1.4. Heat resistance

Heat resistance is the property or ability of component to resist the deteriorating caused by high temperatures. Temperature rising results in: decreasing in accuracy, reducing strength, deteriorating of lubricant.

Ways to increase heat resistance are:

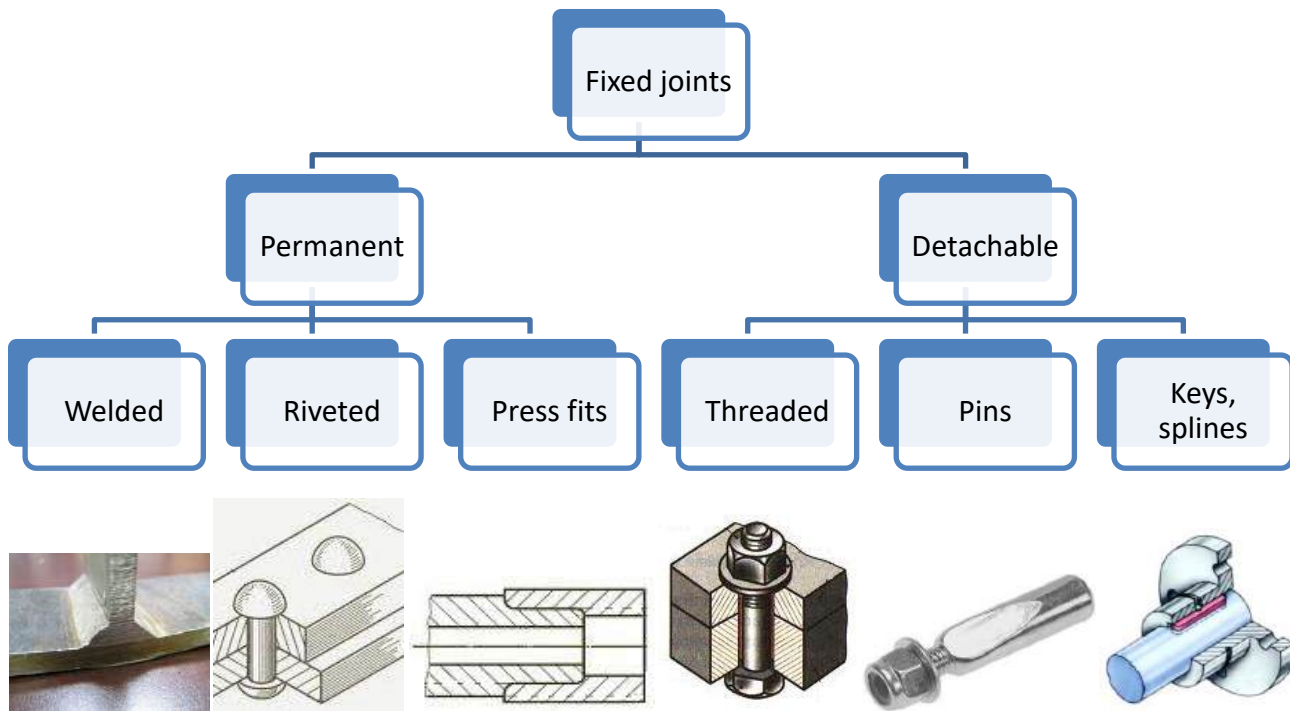
- use special liquid lubricants formulated for higher pressure;
- use cooling;
- reduce internal factors, which stimulate heat generating;
- select materials of construction with similar or very low coefficients of linear expansion.

Part 2. JOINTS

Joint is a part of a machine, which is used to connect another mechanical part or mechanism. Mechanical joints may be temporary or permanent. Most types are designed to be disassembled when required. Mechanical joints are much cheaper and are usually bought ready assembled.

Joints can be sliding or fixed. The latter we will consider.

Fixed joints are subdivided into permanent and detachable:



Permanent joints do not allow a unit to be disassembled without destroying the connected components. Such joints are usually used in locations where disassembling is not required during service life.

Permanent joints can be made by mechanical methods (riveting, expanding, interference fits) – and by physic-chemical adhesion – welding, soldering, adhesive bonding.

Detachable joints allow the disassembly without damaging the fastened elements and connecting components. They are used when disassembling is needed during service life.

The most common type of detachable joints is the threaded joints.

2.1. Threaded Joints

Thread is the prominent spiral part of a screw; a ridge of uniform section in the form of a helix on alternating helical grooves and ridges of regular profile formed on the surface of a part.

Main advantages are:

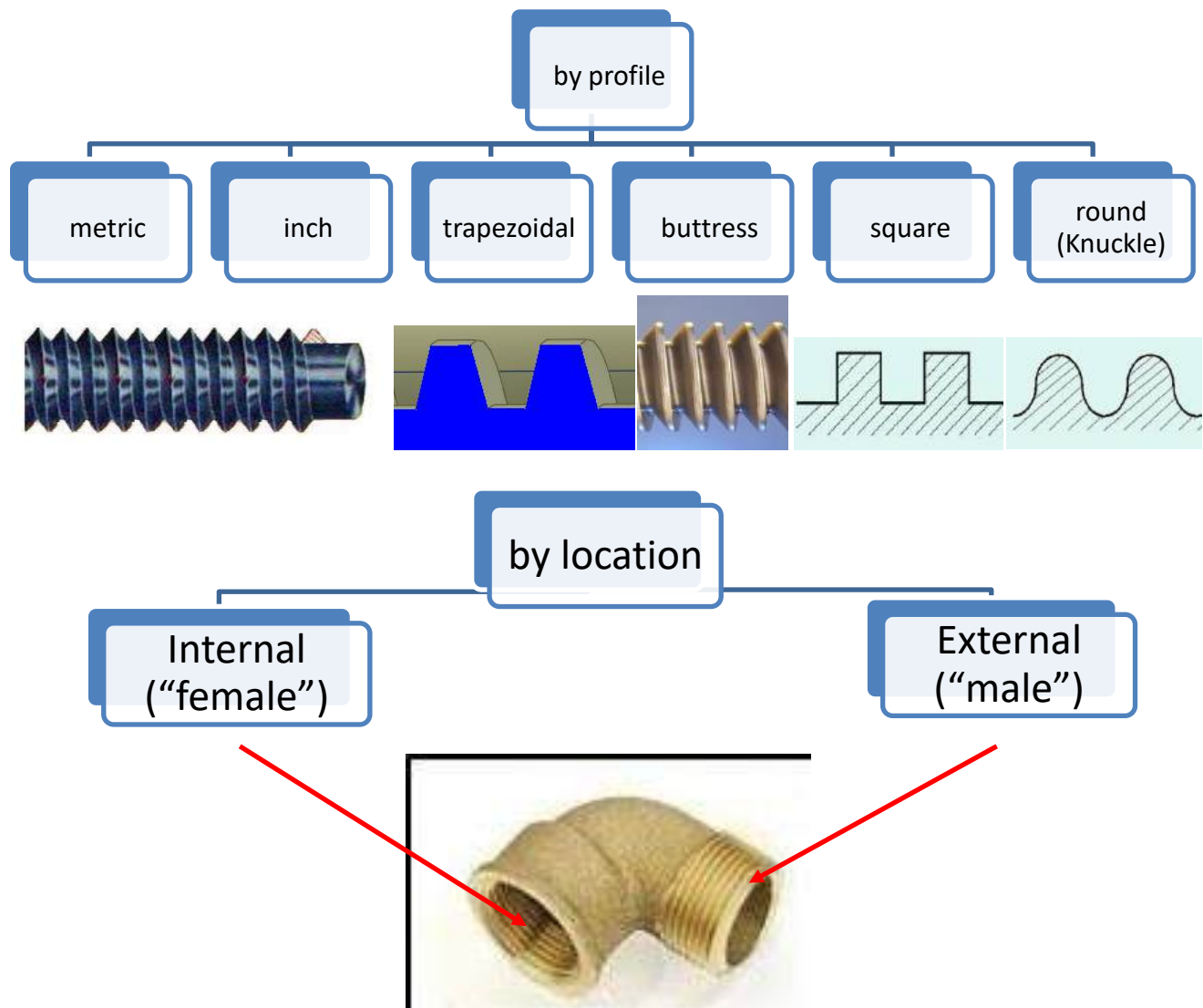
- high reliability;
- convenient assembling/disassembling;
- simple design;
- ability to adapt various operating conditions;
- relatively cheap production due to high level of standardization and efficient manufacturing processes.

Disadvantages are:

- the number of stress concentration points on threaded surfaces. Special methods of improving their fatigue endurance are required;
- uneven load distribution among threads;
- relatively big axial size.

2.1.1. Classification of the Threads

Threads may be classified as follows:



by base surface

cylindrical

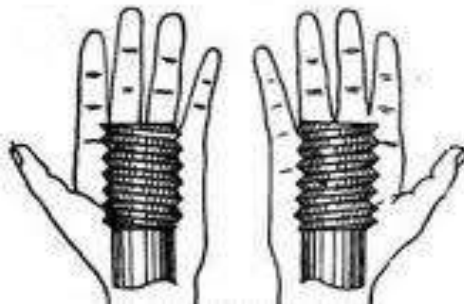
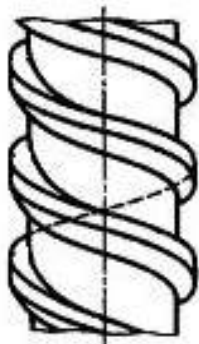
tapered



by screw line direction

Left-hand

Right-hand



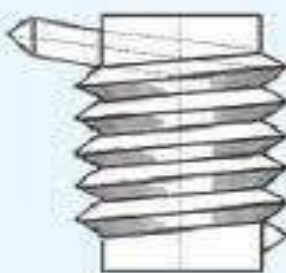
by number of starts

single

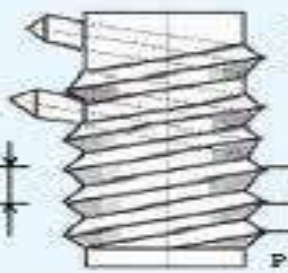
double

triple

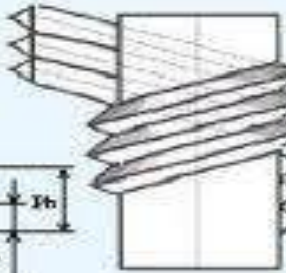
etc.



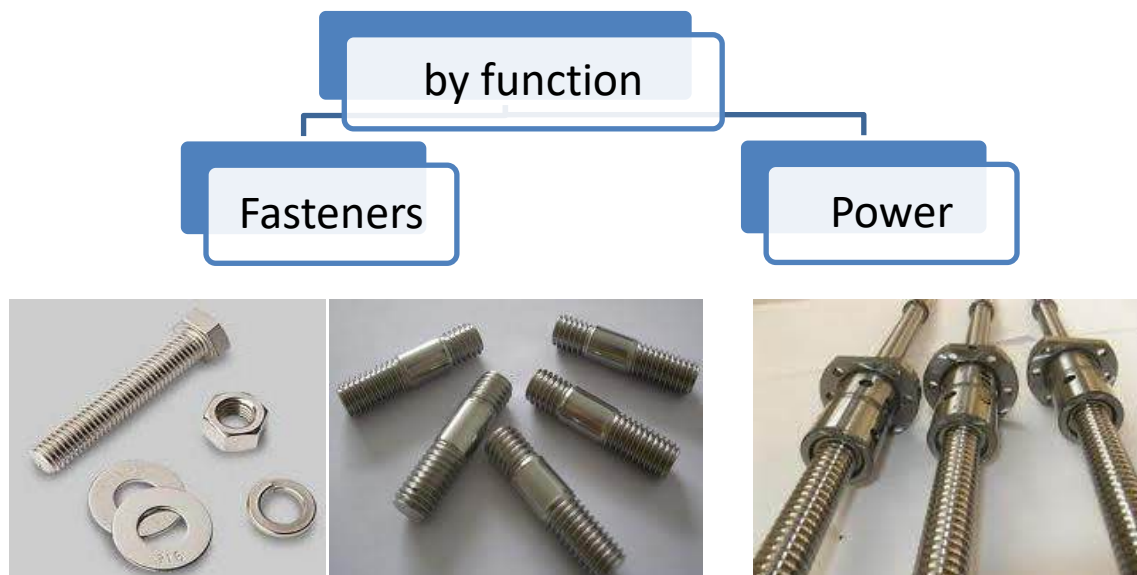
$P_h = P$



$P_h = 2P$



$P_h = 3P$



2.1.2. Types of Fasteners

Bolts

A bolt (Fig. 2.1) is a screwed cylinder fastener with a head designed to be used with a nut.

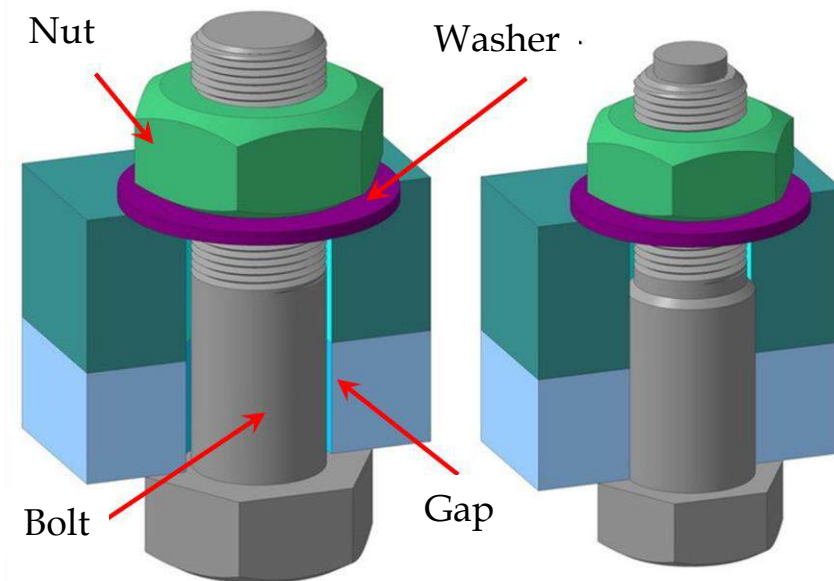


Fig. 2.1

The bolt joint located in a hole with a gap is more convenient but have great weight.

Cap screws

Cap screws (Fig. 2.2) are used to hold two parts together without a nut since one part has a threaded cylindrical hole. The other part is drilled with an oversized hole so that the cap screw could pass through it freely.



Spring washer

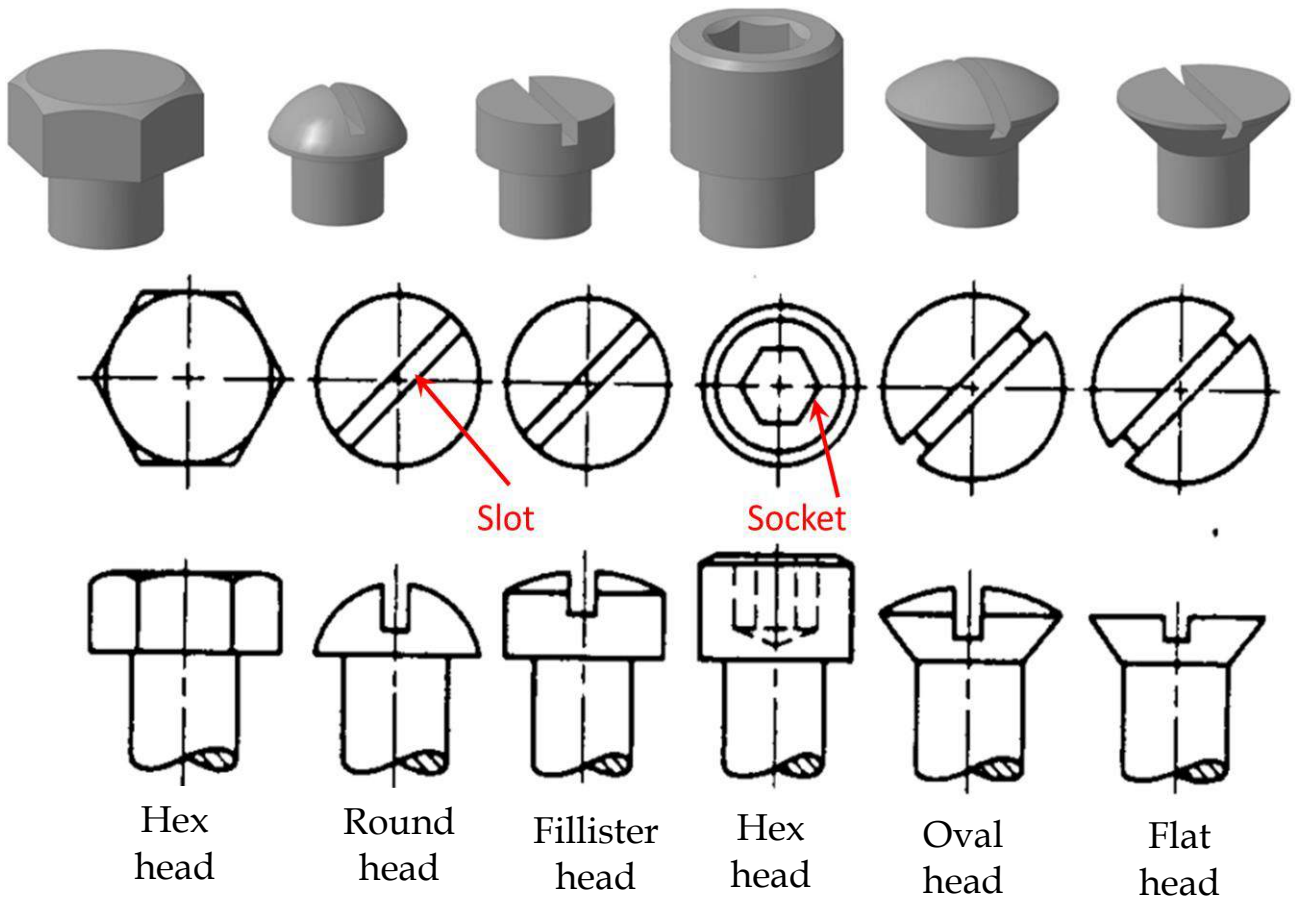
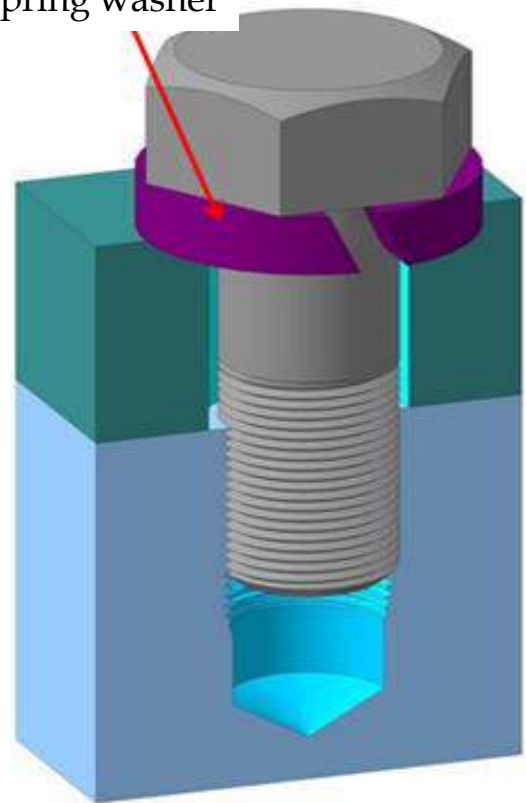


Fig. 2.2

Studs

It is a rod threaded on both ends (Fig. 2.3).



Fig. 2.3

The thread may be extended along the complete length of the rod (so called all-thread) (Fig. 2.4).



Fig. 2.4

Studs may have a drive slot in one end to facilitate installing the stud (Fig. 2.5).

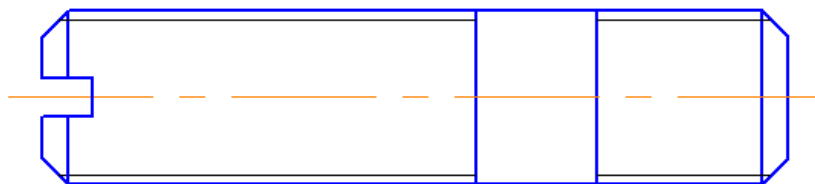


Fig. 2.5

Nuts

A nut is a type of fastener with a threaded hole (Fig. 2.6). Nuts are almost always used opposite a mating bolt to fasten a stack of parts together. In applications where vibration or rotation may cause a nut loose, various locking mechanisms may be

provided: adhesives, safety pins, nylon inserts. The most common shape is hexagonal, for similar reasons as the bolt head - 6 sides give a good granularity of angles for a tool to approach. Other specialized shapes exist for certain needs, such as wing nuts for finger adjustment and captive nuts for inaccessible areas.

Normal nut thickness is about $0,8 d$, but may be $(0,5 \dots 1,6) d$.

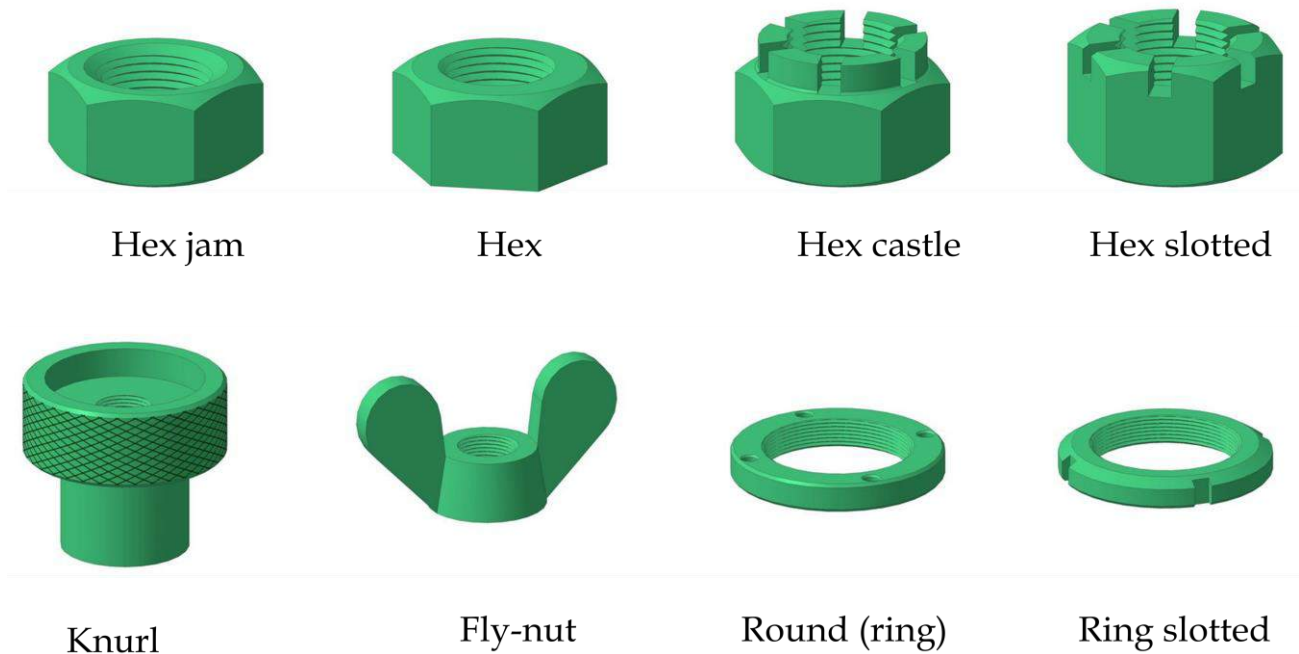


Fig. 2.6

Set screws

Set screws (Fig. 2.7) are generally used to secure an object within or relatively to another object (i.e. to fix parts on the shaft). The set screw passes through a threaded hole in outer object and is tightened against the inner object to prevent it from moving relatively to the outer object.

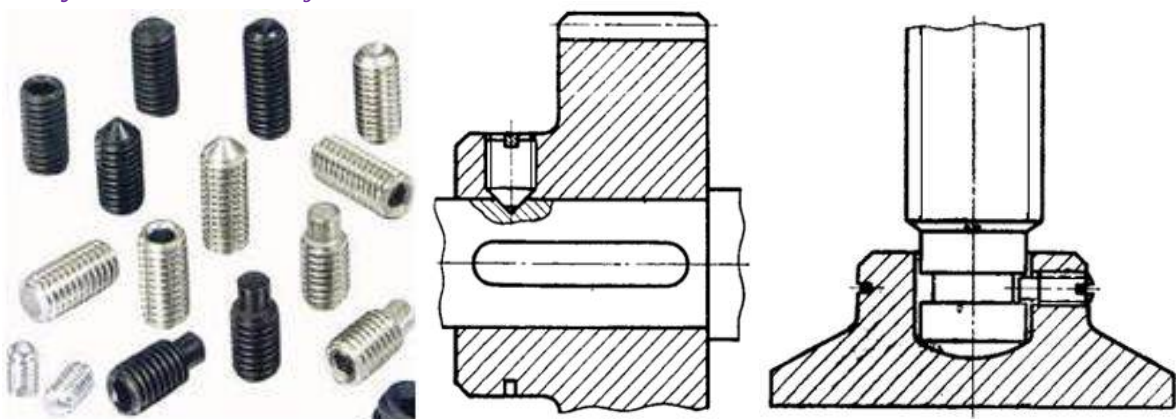


Fig. 2.7

Set screws are available in combinations of points and heads.

The shaft against which the set is tightened may have a flat surface – dog or flat point would be most effective (Fig. 2.8).

The cup point gives good friction when applied to a round shaft.

The cone point screw is used for permanent setting of parts. These set screws offer the highest holding capability, due to the deep penetration of the cone point.

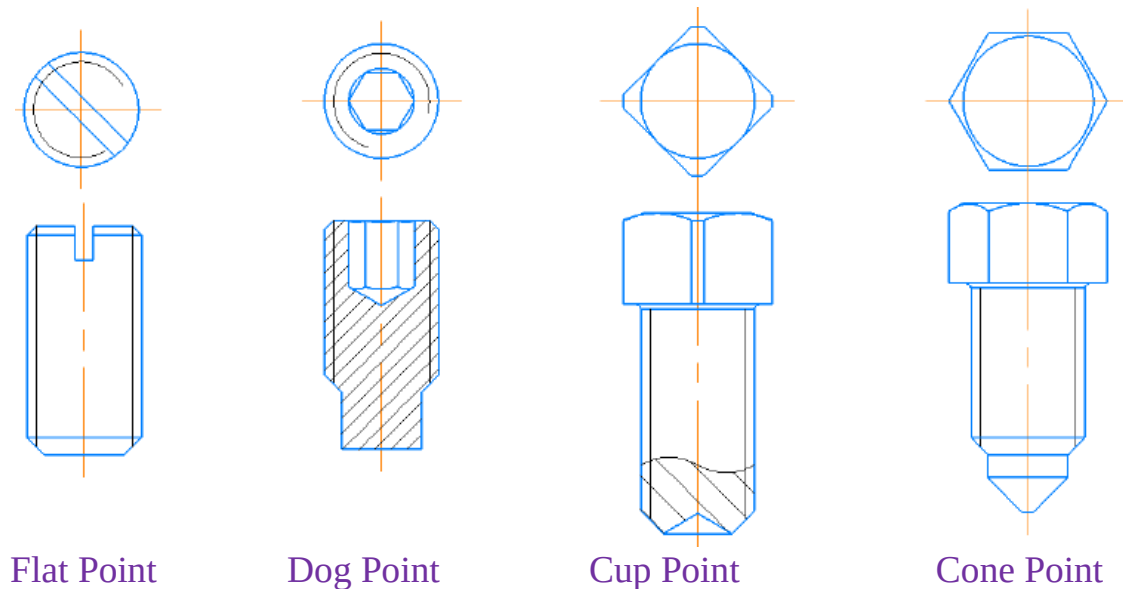


Fig. 2.8

2.1.3. Thread Parameters and Terminology

The outline of the grooves and ridges on a plane passing through the axis of the part is called the thread profile. The principal parameters of both external and internal threads are shown in Figs 2.9, 2.10 and Table 2.1.

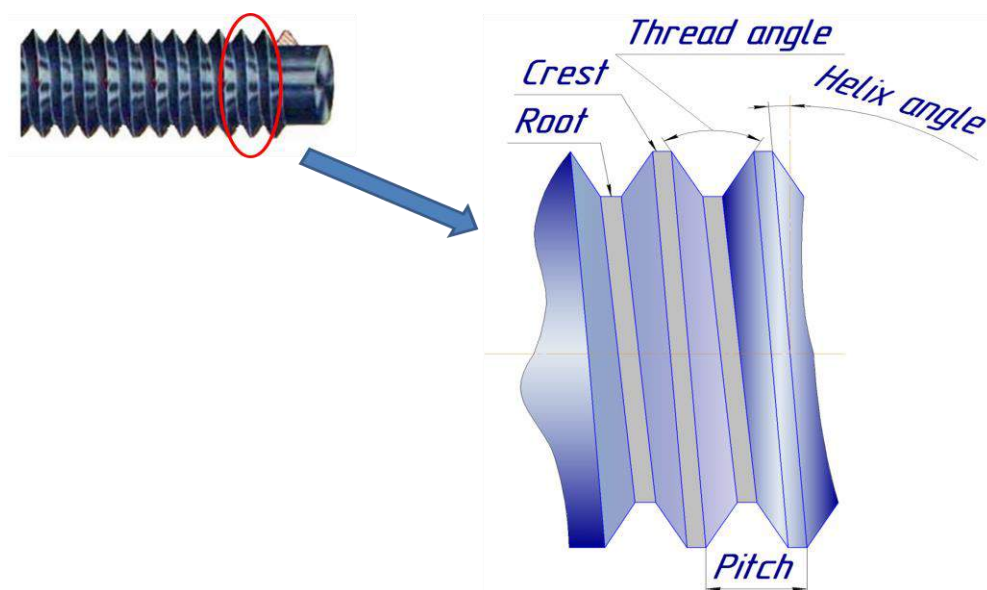


Fig. 2.9

Table 2.1

Parameter	Signification for thread	
	Internal (nut)	External (bolt)
Major diameter	D	d
Pitch diameter	D_2	d_2
Minor diameter	D_1	d_1
Root diameter	D_3	d_3
Pitch	P	
Thread height	H	
Effective thread height	H_1	
Thread angle	α	
Helix angle	ψ	

Notes:

1. Small letters designate screw diameters, capital letters – nut diameters.
2. Nominal diameters with the same name are equal for a screw and a nut.

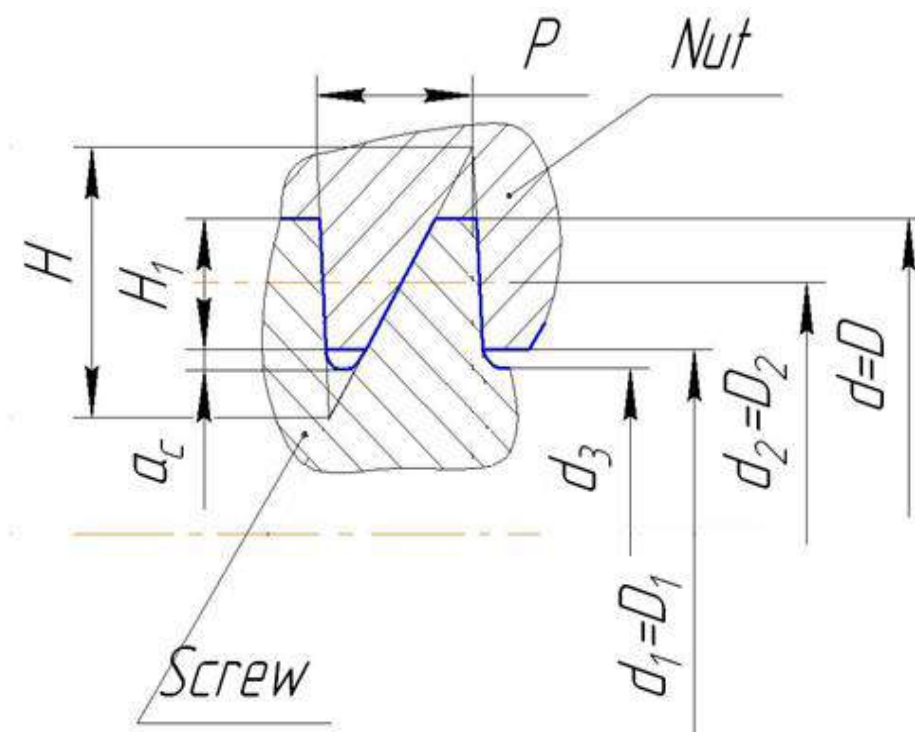


Fig. 2.10

Thread angles for different types of thread are shown in Fig. 2.11.

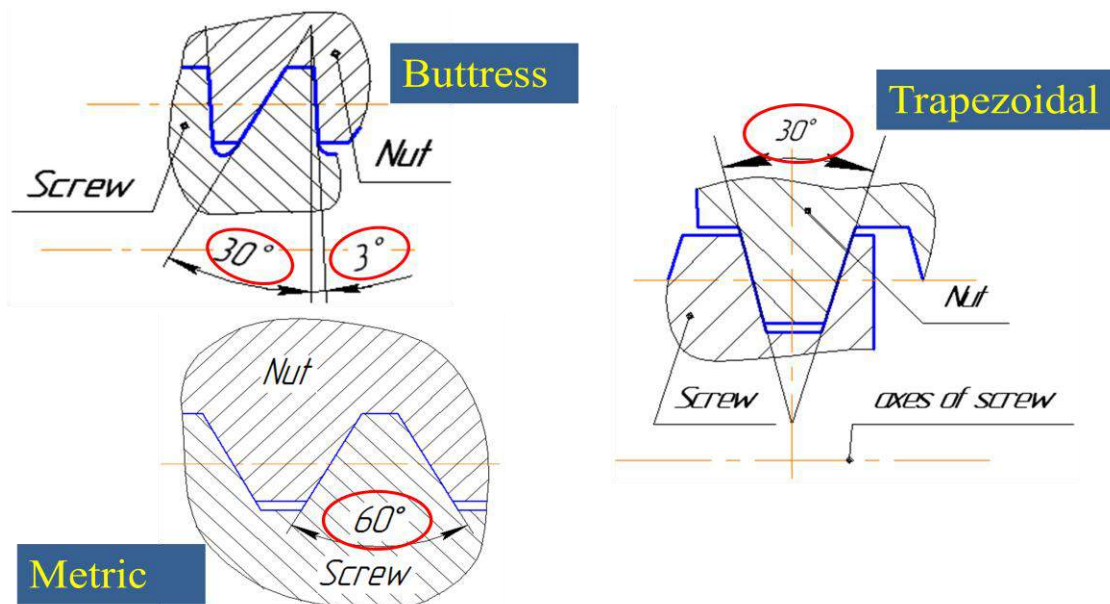


Fig. 2.11

Pitch P (see Figs 2.9 and 2.10) is a distance between the two nearest corresponding points on the thread profile.

An **external (major) diameter D (d)** is a nominal, or basic, thread size. Together with a pitch, it is used for thread standard signification.

Pitch (mean) diameter D_2 (d_2) is the diameter of an imaginary cylinder passing through the threads at the points where the thread width of the screw is equal to the thread width of the nut.

D_1 (d_1) is a **internal (minor) thread diameter**. It is the smallest nut diameter.

d_3 is a diameter of a screw root. It is the smallest screw diameter.

Effective height H_1 is a distance between the screw and the nut threads' crests:

$$H_1 = \psi_h P,$$

where $\psi_h = 0,5$ for trapezoidal; $\psi_h = 0,541$ for metric; $\psi_h = 0,75$ for buttress.

Thread height H is a distance between theoretical vertices of a thread profile: $H = 1,866P$ for trapezoidal; $H = 0,866P$ for metric; $H = 1,587911P$ for buttress.

Root depth a_c is a distance between theoretical vertices of a thread profile.

$a_c = 0,117767P$ for buttress; $a_c = 0,07217P$ for metric; for trapezoidal it depends on the pitch:

P	a_c
2...5	0,25
6...12	0,5
>14	1

Power screws may contain a single screw thread (single start thread) or multiple screw threads. For a multiple start screw, the screw displacement would be measured between threads associated with a given start (Fig. 2.12). Multiple start screws are often

used to provide load-carrying redundancy since each start (or thread) would be sized to carry the maximum load.

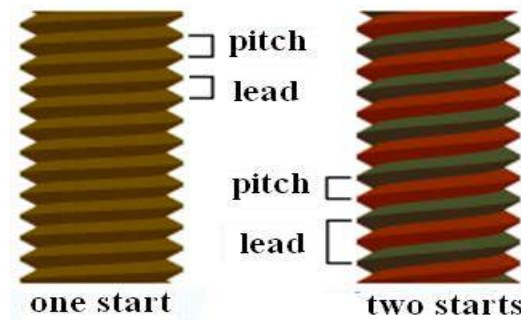


Fig. 2.12

Lead is the distance a nut travels (linearly) during a full single screw revolution: $\text{Lead} = \text{pitch} \times \text{number of starts}$.

Since actuators are usually specified through a linear (nut) velocity, the chosen lead (or lead angle) will have implications for motor and gear train design. For example, a motor will have a certain speed (rad/sec), which when translated through a gearbox may or may not drive the nut at the required velocity (in/sec). Hence, the motor characteristics, gearing through any gearbox arrangement and the screw lead must be compatible to achieve desired overall performance. Lead may also affect accuracy. A larger lead (or lead angle) will require finer rotational position control to maintain accurate nut position control. A smaller lead angle will require a larger screw rotation to produce the same output translation.

A standard thread is designated as a combination of a conventional letter code (“M” for metric, “Tr” for trapezoidal and “S” for buttress ones), major diameter, an “x” sign and a pitch. For example, a buttress thread with a diameter of 32 mm and pitch of 5 mm is designated as S 32x5.

2.1.4. Self-locking Condition

Due to high frictional losses, power screws are not very efficient but generally self-locking and the property of self-locking is sometimes useful. In self-locking power screws, the load cannot lower itself without an external effort. This feature is generally used to hold loads. For example,

- when lifting weights with the help of jacks with self-locking threads there is no need to keep the handle constantly by means of any intentionally provided clamps (let along by hand), because the load will not go down under the action of the weight,
- it is not necessary to keep the press screw used to compress elements stuck together tightened.

Let us obtain thread self-locking condition.

A screw can be viewed as a narrow inclined plane wrapped around a cylinder. If we conditionally cut a thread across the helix line and evolve it we see the inclined plane like in Fig. 2.13. We can obtain helix angle from the triangle:

$$\tan \psi = \frac{Pn}{\pi d_2}$$

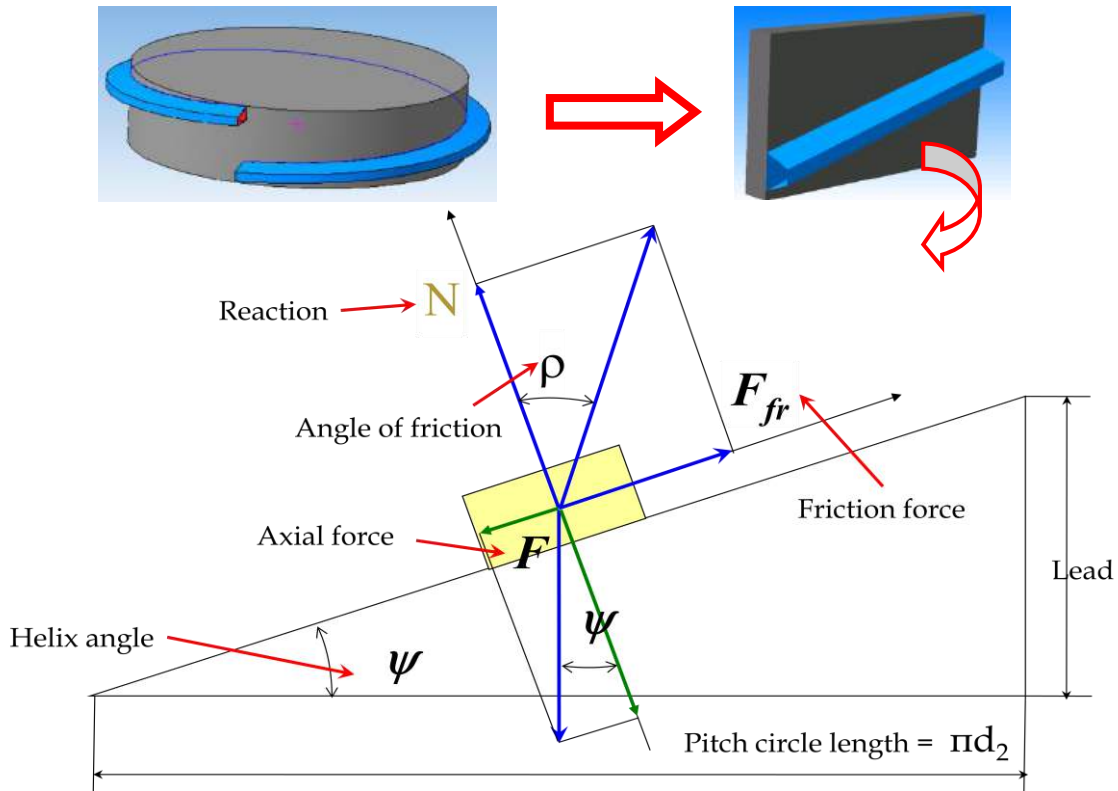


Fig. 2.13

Let us consider thread element equilibrium condition. It is clear that the element keeps unmovable under any axial load if friction force F_{fr} produced by the load F is greater than the force F component

$$F_{fr} \geq F \sin \psi .$$

Friction force

$$F_{fr} = Nf = F \cos \psi f$$

and self-locking condition may be re-written as

$$F \cos \psi f > F \sin \psi .$$

Tangent of the angle of friction is $\tan \rho = \frac{F_{fr}}{N} = \frac{N f}{N} = f$. If we replace friction coefficient f with $\tan \rho$ and divide both left and right parts by $\cos \psi$, we will obtain

$$\tan \rho > \tan \psi .$$

If thread profile angle is not equal to 0, reduced friction angle (Fig. 2.14)

$$\rho' = \tan^{-1} \frac{f}{\cos \beta}$$

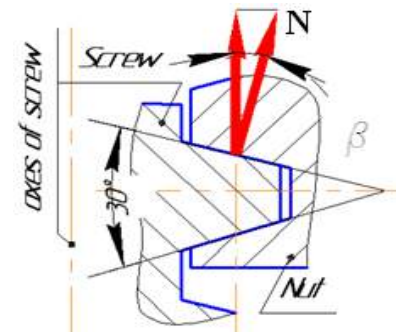


Fig. 2.14

should be used instead of ρ (in case of buttress thread $\cos 3^\circ$ very close to 1 and for engineering practice we may neglect the difference between ρ and ρ' ; for trapezoidal thread this difference is about 3,5 %).

Finally, self-locking condition may be written as follows

$$\rho' > \psi + 1^\circ.$$

Here 1° is used like a safety factor.

2.1.5. Total Frictional Torque

In most cases, it is needed to know the driving torque required to lift the loads by power screws. To do this, let us consider equilibrium equation like in previous case (Fig. 2.15):

$$F_{fr} + F \sin \psi = F_d \cos \psi.$$

Here F_d is a driving force:

$$F_d = \frac{2T_t}{d_2},$$

where T_t is the total frictional torque which should be applied to overcome axial force F . Taking into account equations obtained above, we can write

$$F \cos \psi \tan \rho + F \sin \psi = \frac{2T_t}{d_2} \cos \psi;$$

$$F (\tan \rho + \tan \psi) = \frac{2T_t}{d_2}.$$

$$\text{Finally } T_t = F \frac{d_2}{2} \tan(\psi + \rho').$$

Here a part of a useful resistance is represented by a variable ψ , and a part of a resistance to friction in a thread – by ρ' .

In case of moving down

$$T_t = F \frac{d_2}{2} \tan(\rho' - \psi).$$

In most cases (especially with regularly operating transfers with a hand drive), it is desirable to reduce the torque. To do this, ψ should be reduced (for example, choose a smaller pitch).

2.1.6. Efficiency

Efficiency is important in rotary/ linear motion transformation such as provided by power screws. There is a large contact area between male and female threads in power screws and this results large frictional losses during the operation.

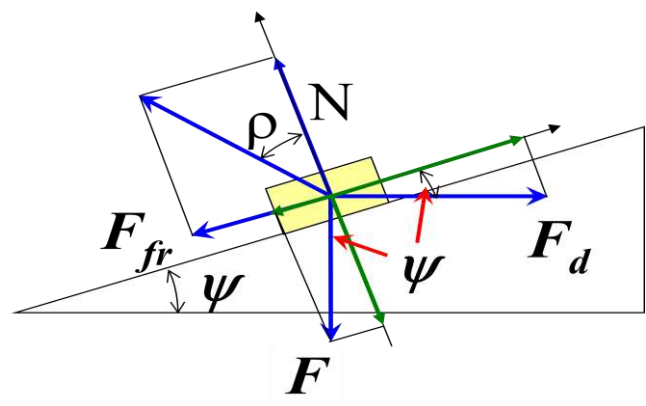


Fig. 2.15

The transmission efficiency of the power screw can be expressed as

$$\eta = \frac{A_u}{A_d}.$$

Let us consider diagrams in Fig. 2.15. Here the force F_d moves elementary part of thread from the lower end to the upper end of the inclined plane. The output useful work A_u is lifting the load F on the lead height (lead equals to pitch P for one-start thread): $A_u = FP$.

The input work A_d includes rotating the screw by means of a driving force F_d along the pitch diameter length:

$$A_d = \pi d_2 F_d = \pi d_2 \frac{2T_t}{d_2} = 2\pi T_t = 2\pi F \frac{d_2}{2} \tan(\psi + \rho') = \pi d_2 F \tan(\psi + \rho').$$

Thus

$$\eta = \frac{FP}{\pi d_2 F \tan(\psi + \rho')}.$$

We know, that $\frac{P}{\pi d_2} = \tan \psi$, so finally

$$\eta = \frac{\tan \psi}{\tan(\psi + \rho')}.$$

The helix angle characterizes lead effect on thread efficiency (useful work) and angle of friction – influence of the harmful friction. It is clear, that high efficiencies demand either very low friction or big helix angle.

Coefficients of friction about 0.1...0.2 may be expected for common materials under conditions of ordinary service and lubrication: steel on steel for fasteners, and steel on bronze or cast iron in the case of power screws. Specific coatings and surface treatments can reduce friction significantly.

Increasing of the helix angle causes losing of the self-locking. The limit helix angle $\psi = \rho$ (see self-locking condition above), in such case the maximum efficiency is

$$\eta = \frac{\tan \psi}{\tan(\psi + \rho')} = \frac{\tan \psi}{\tan(2\psi)} \approx \frac{\psi}{2\psi} = 0,5.$$

This result means that self-locking thread efficiency does not exceed 50 %.

2.1.7. Load Distribution

Various studies on thread loading have established that the shear stress is not evenly distributed across the threads. The first thread could withstand nearly all the load if the thread materials were very hard and did not yield. However, because of material yielding there is some distribution of the load. It is known that for a typical nut the percentages of the load taken by consecutive threads are about 34 %, 23 %, 16 %, 11 %, 9 %, 7 % and the last – less 1 % (Fig. 2.16).

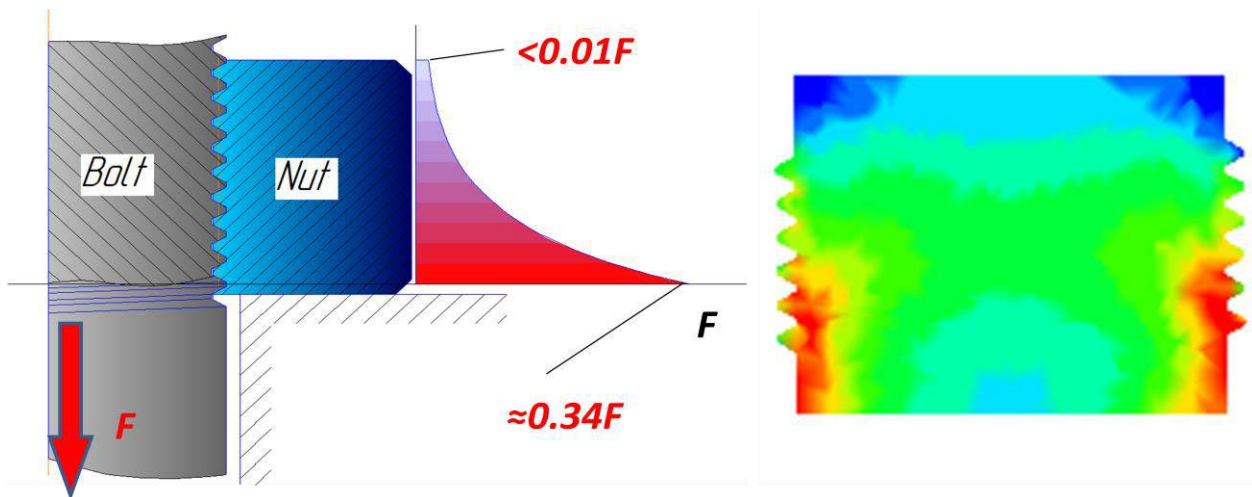


Fig. 2.16

Moreover, bolt and an ordinary nut have deformations in opposite directions (Fig. 2.17).

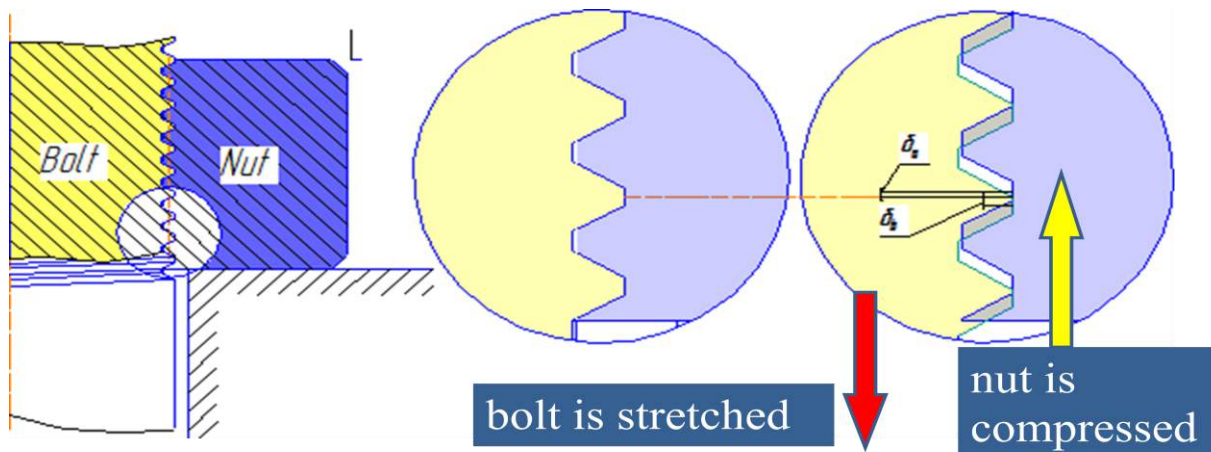


Fig. 2.17

It is preferable that the screw fails rather than the internal or external thread strips. The length of the screw engagement should therefore be sufficient to carry the full load and necessary to break the screw without the threads stripping.

A very simple rule that can be applied for that vast majority of applications is that a thread length of 80 % of the screw diameter (standard nut height) is sufficient for ensuring that the screw will fail in tension before the female thread (nut) fails in thread stripping (assuming the screw and nut are similar materials). Generally, a female thread is stronger than a male one in shear in a case of the same length of engagement and the same material. That is why it is useless to increase the number of nut threads (usually there are about 5-6). Thus, the fastener manufacture sizes the length of the nut to ensure the screw will fail before the nut.

If a screw fastens into a tapped hole (like studs or cap screws), a check of the depth of thread engagement is required. The following rules are suggested for arriving

at reasonable lengths of thread for steel screws used with screwed holes in different materials (here d is the nominal diameter of the thread):

- for steel a length of thread engagement of at least $1 \times d$;
- for cast iron, brass or bronze – at least $1,5 \times d$;
- for aluminum, zinc or plastics – at least $2 \times d$.

Equalizing of non-uniformity of the load distribution can be:

- by using very accurate threads;
- by using ductile materials for the components;
- by specific nut design;
- by thread profile modification;
- etc.

Equalizing by Specific Nut Design

Stiffness of the most loaded threads decreases and the load shifts to the upper fillets. Such tension nuts (Fig. 2.18) improve the endurance of bolted joints up to 30 %.

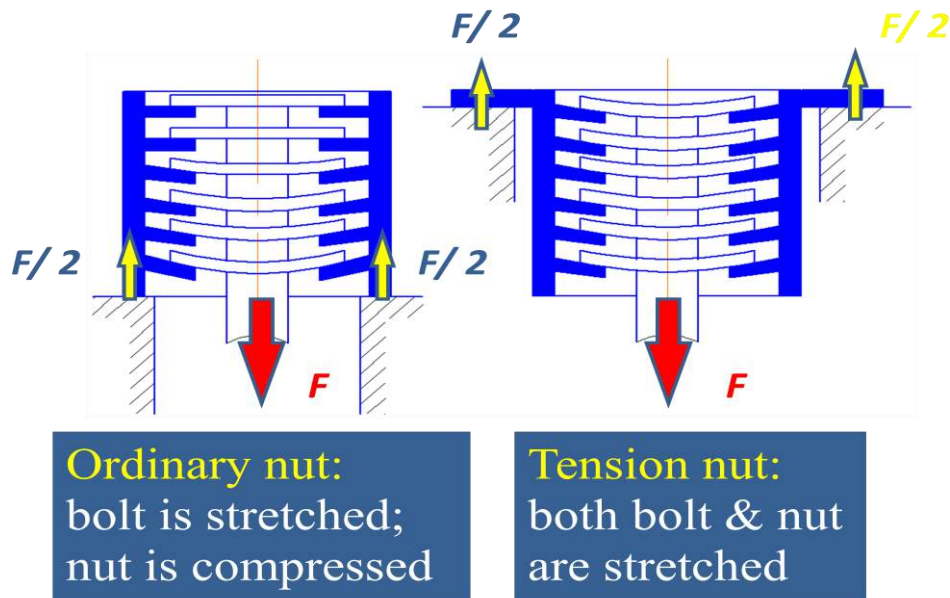


Fig. 2.18

Another way to decrease stiffness of the most loaded threads is to cut their as it is shown in Fig. 2.19. In such case the threads cross-sectional area is less and deformation increases involving in work upper threads.

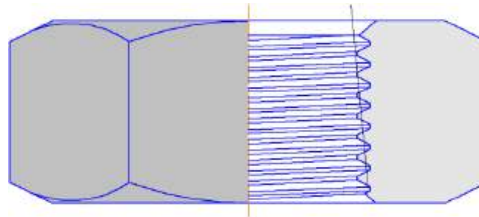


Fig. 2.19

Equalizing by Profile Modification

One from a lot of alternative thread forms is called Spirallock. It adds a special 30° wedge ramp at the root of the female thread which mates with standard 60° male-thread fasteners (Fig. 2.20).

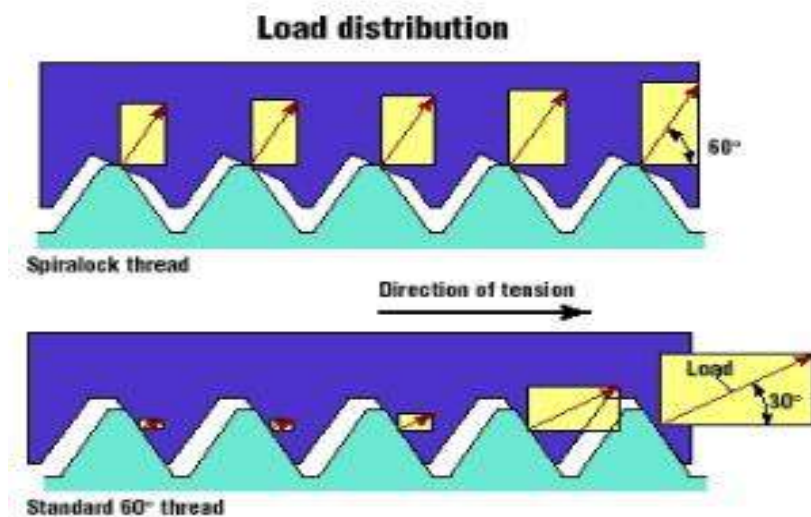


Fig. 2.20

A continuous-spiral-line contact along the length of the thread engagement more evenly spreads axial force (Fig. 2.21).

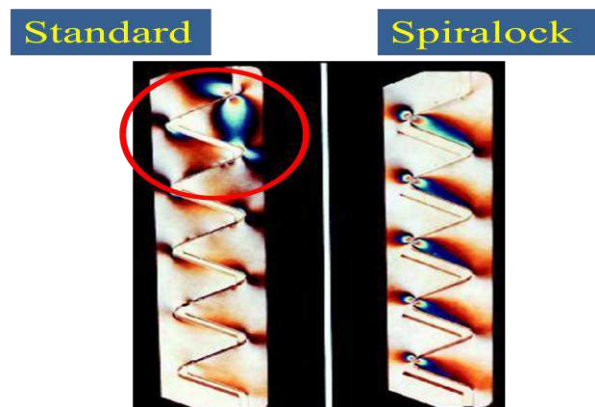


Fig. 2.21

2.1.8. Thread Revisé Calculation

Threaded parts carry mainly axial loads.

Let us assume:

1. The joint operates symmetrically and the load is considered to act along its axis.
2. The design area in the bolt threaded portion is taken as the area of a circle in d_1 diam.
3. The load is distributed uniformly between the threads.

Taking into account these assumptions, we can check:

- shear stresses along line **ab** (Fig. 2.22):

$$\tau = \frac{F_a}{\pi d_1 K_m P Z} \leq [\tau];$$

$$[\tau] = (0,3 \dots 0,6) \sigma_y;$$

- normal bearing stresses on the effective thread surface (**ae**):

$$\sigma_{br} = \frac{F_a}{\pi d_2 H_1 Z K_m} \leq [\sigma_{br}]; [\sigma_{br}] = 0,8 \sigma_y,$$

where $K_m = 0,87$ for V-thread (metric or inch).

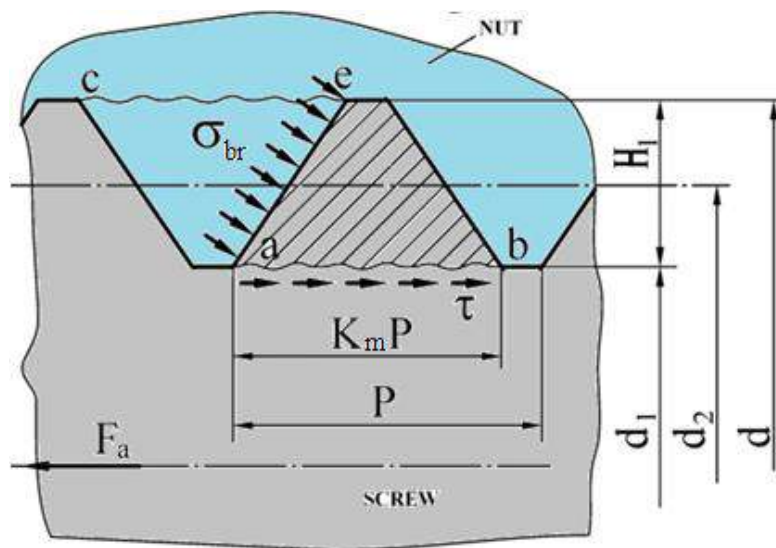


Fig. 2.22

2.1.9. Calculation of Bolt Loaded by Axial Forces

Prestressed bolt joints are the majority of bolt connections used in practice. These joints are loaded with great internal axis force (mounting prestressing) during assembly. This prestressing provides the necessary force bond of contact surfaces of the connected materials. A correctly prestressed joint forms guaranteed force closing in the contact surfaces and unchanged mutual position of the connected parts during operation.

The mounting prestressing of a bolt joint thus includes two basic functions. In case of connections loaded in the plane of the connected parts, the prestressing meets

the requirement of shear loading capacity on a base of friction forces; in case of connections loaded along the axis of the bolt, it meets the requirement of compactness or tightness of the connection resp.

The size of a screwed fastener is first established by calculating the tensile load to be withstood by the screw and selecting a suitable screw to withstand the tensile load with the appropriate factor of safety or preload.

The calculation is designed for a geometrical design and strength check of a prestressed bolt joint, loaded by static or cyclic loading resp., acting both in the axis of the bolt and in the plane of the connected parts.

From the above-mentioned, the individual steps which are necessary for successful design of a prestressed bolt connection result:

- selection of a suitable connection bolt;
- calculation of the necessary prestressing;
- strength check of the designed joint.

This calculation is meant for the design of a connection with one bolt. If you need to calculate bolt connections with more bolts, first it is necessary, according to general principles to specify the maximum load falling on one (identically or the most loaded) bolt. Let us assume that all bolts carry the same axial load (Fig. 2.23) and the

force is given: $F = \frac{F_{\Sigma}}{Z}$.

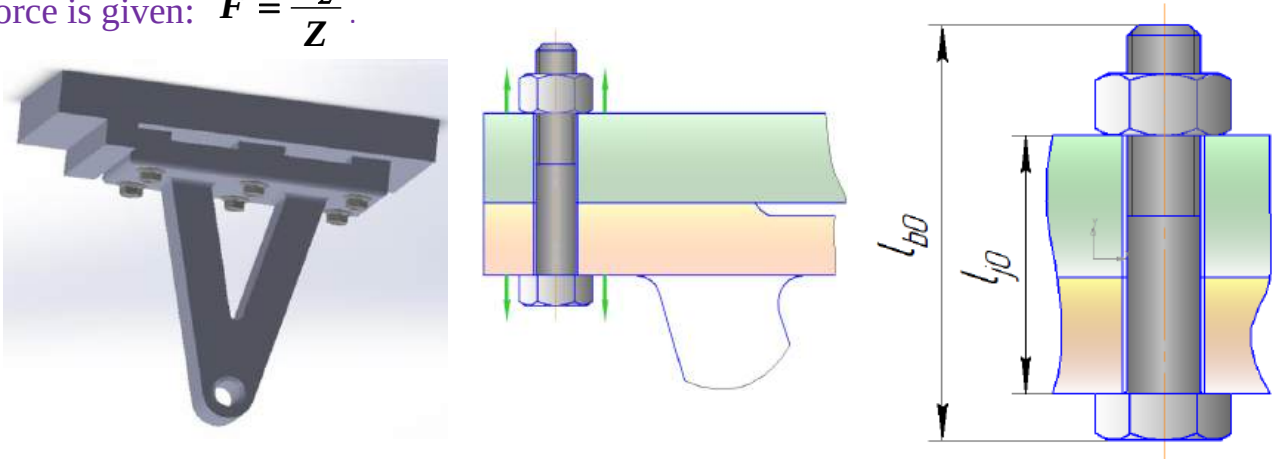


Fig. 2.23

Let us designate initial bolt and parts (joint) dimensions as l_{b0} and l_{j0} resp.

As a nut is rotated, the bolt elongates on Δ_b and at the same time, the clamped parts are squeezed on Δ_j . Internal forces within the bolt resists bolt extension, and tension force F_p (bolt preload) is generated (Fig. 2.24).

To visualize the loading within bolted connections, the bolt joint diagram is used (Fig. 2.25). If we lay off the forces along the Y-axis and the deformations along the X-axis then the stiffness is characterized by the tangent of the angle between the straight line and the X-axis (we consider elastic region, thus dependence is a straight line). For bolt:

$$F_p = c_b \Delta_b \Rightarrow c_b = \frac{F_p}{\Delta_b} = \tan \alpha_b,$$

for joint:

$$c_j = \frac{F_p}{\Delta_j} = \tan \alpha_j.$$

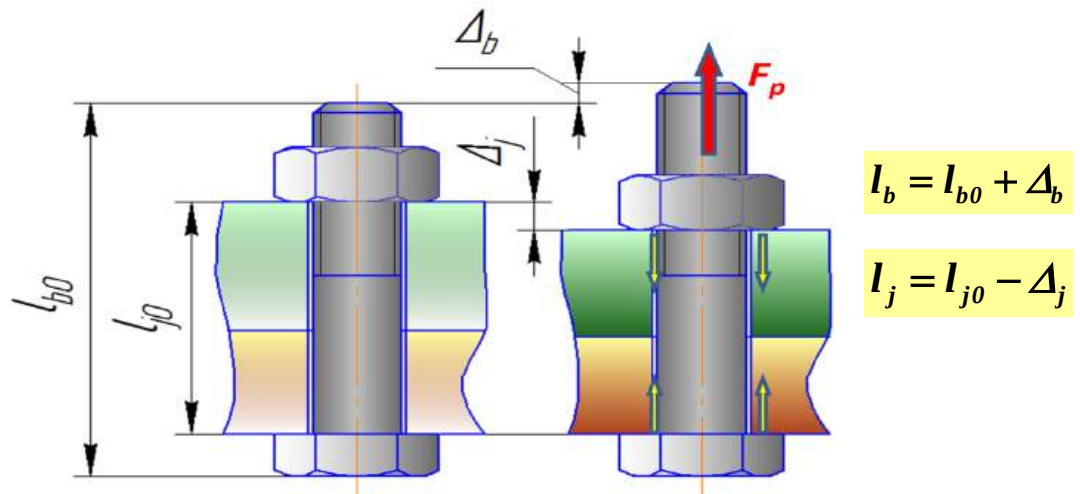


Fig. 2.24

Line OA corresponds to the bolt deformation Δ_b at the preload force F_p . Respectively OB is the joint compression line and point B X-coordinate is equal to Δ_j . The basic joint diagram is formed by moving the compression line of the joint to the right side so as point B coincides with A. A triangle is formed. The compression of the joint by the preload force F_p is equal to the bolt preload tension:

$$F_p = c_j \Delta_j = c_b \Delta_b.$$

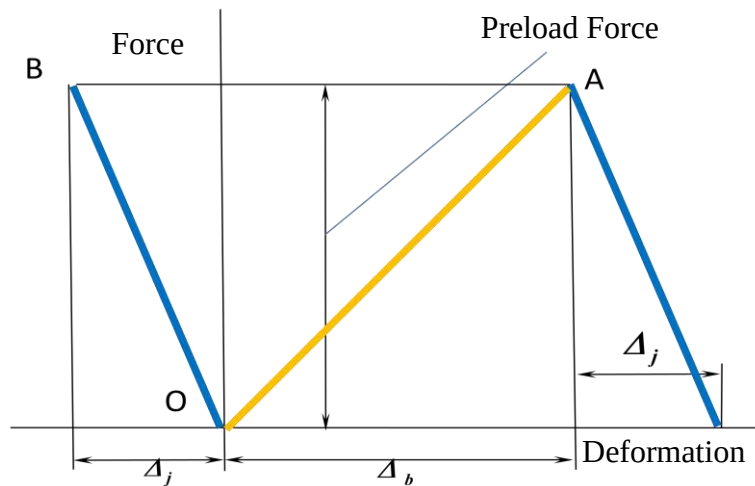


Fig. 2.25

The ratio between deformation of the bolt Δ_b and summary deformation of the clamped parts may be represented by the ratio of their particular stiffnesses. This ratio is named as a joint constant and may be expressed as follows:

$$\chi = \frac{\Delta_j}{\Delta_b + \Delta_j} = \frac{c_b}{c_b + c_j} = \frac{\lambda_j}{\lambda_B + \lambda_j}.$$

After applying of an axial operational force F_{ex} to the joint (Fig. 2.26), action of the external axial force is distributed between the bolt and the clamped parts of the joint and may be determined using the bolt-joint diagram (Fig. 2.27). Force F_z loads bolt additionally, force ΔF decompresses joint parts. These forces are calculated as follows: $F_z = \chi F_{ex}$, $\Delta F = F_{ex} - F_z = F_{ex}(1 - \chi)$. Finally, at the external action the compression of the clamped parts is reduced and the bolt elongates at value δ :

$$l_b = l_{b0} + \Delta_{bf} = l_{b0} + \Delta_b + \delta; \quad l_j = l_{j0} - \Delta_{jf} = l_{j0} - \Delta_j + \delta.$$

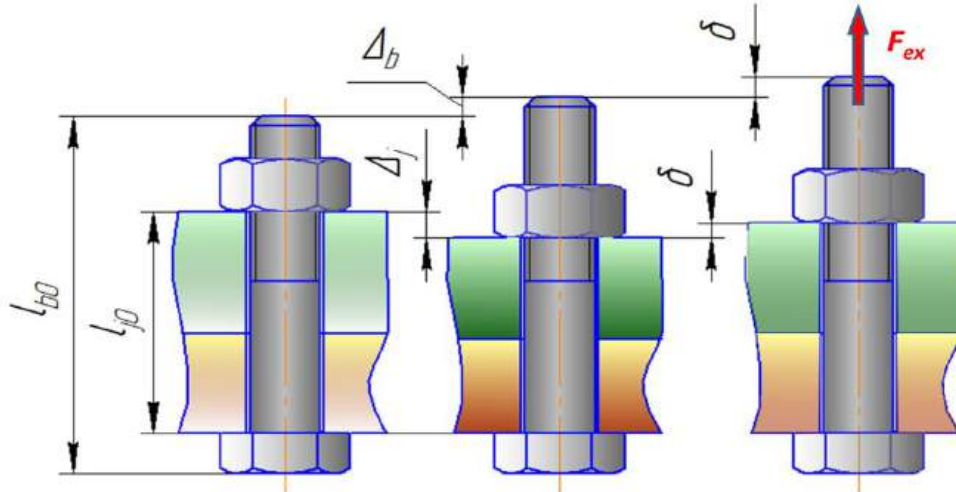


Fig. 2.26

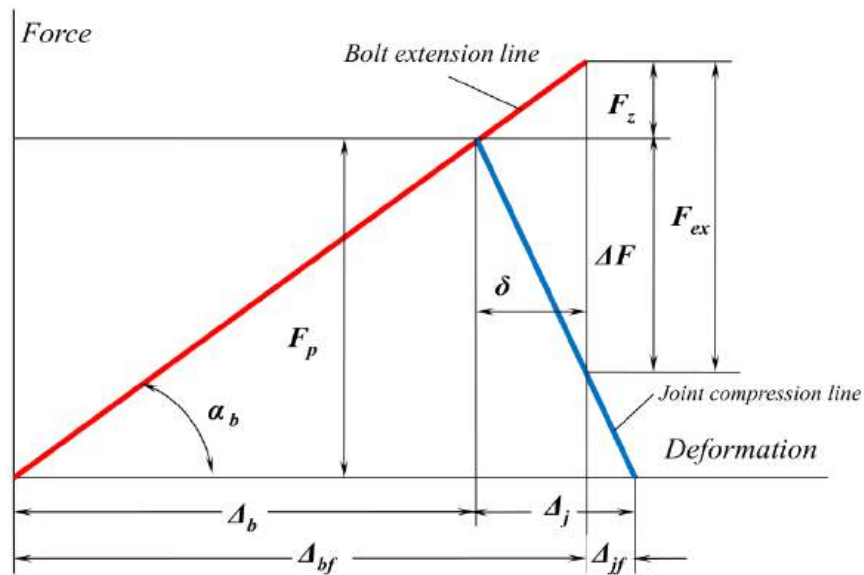


Fig. 2.27

2.1.10. Preload Force Calculation

Calculation should answer two questions:

1. What force value should be applied to avoid the joint opening under external load?
2. What bolt diameter can take both the preload and the external forces?

Condition $|\Delta_j| \geq \delta$ should be satisfied to avoid gap between parts of the joint. In other words, $|\Delta_{jf}| \geq 0$ or $F_p > \Delta F$. It means that the minimum value of the prestressing force is $F_{pmin} = \Delta F$.

Thus, taking into account the instability of a tight

$$F_p = KF(1 - \chi),$$

where K is a safety factor (at constant loading $K = 1,3 \dots 1,5$; at altering loading $K = 1,4 \dots 4$).

The average joint constant χ (joint stiffness factor) is equal to $0,1 \dots 0,4$ for metal components connection without gasket. It is obtained as

$$\chi = \frac{c_b}{c_b + c_j} = \frac{c_b \delta}{c_b \delta + c_j \delta} = \frac{F_Z}{F_Z + \Delta F} = \frac{F_Z}{F_{ex}}.$$

2.1.11. Bolt Design Calculation

For the purpose of design and strength calculation, the maximum internal axis force acting on the bolt should be determined. Total force in a bolt after external loading is $F_b = F_p + F_Z = KF_{ex}(1 - \chi) + F_{ex}\chi$.

A bolt minor diameter is found from the tensile strength condition:

$$d_1 = \sqrt{\frac{4\beta F_b}{\pi[\sigma]}},$$

where $\beta = 1,3$ is a factor, taking into account the influence of the torsion of a bolt by the friction torque of thread.

2.1.12. Bolt Revise Calculation

At a static load strength condition in a form of safety factor calculation is usually used:

$$S_{st} = \frac{\sigma_y}{\sigma_{max}} \geq [S_{st}] = 1,5 \dots 2,5.$$

The maximum acting stress is

$$\sigma_{max} = \frac{F_b}{A_1} = \frac{F_p + \chi F}{A_1},$$

where A_1 is a conditional bolt cross section area $A_1 = \pi d_1^2 / 4 = 0,78d_1^2$.

Endurance safety factor is also calculated and compared with allowable value:

$$S_\sigma = \frac{\sigma_{-1}\epsilon_\sigma}{K_\sigma \sigma_a} \geq [S_\sigma] = 1,5 \dots 2,5,$$

where ϵ is the factor taking into account the influence of the bolt's absolute size at its limit of endurance; σ_{-1} is the endurance limit at a reversed cycle; K_σ is the effective fatigue stress concentration factor.

Stress amplitude in a bolt is $\sigma_a = \frac{F_z}{2A_l} = \frac{\chi F}{2A_l}$.

2.1.13. Bolt Stiffness

To determine correct prestressing force, we have to find the actual values of the bolt and the joint stiffness.

Stiffness of each bolt portion (Fig. 2.28)

$$c_{bi} = \frac{E_b A_{bi}}{l_{bi}}$$

and total one is $\frac{1}{c_b} = \sum \frac{1}{c_{bi}}$.

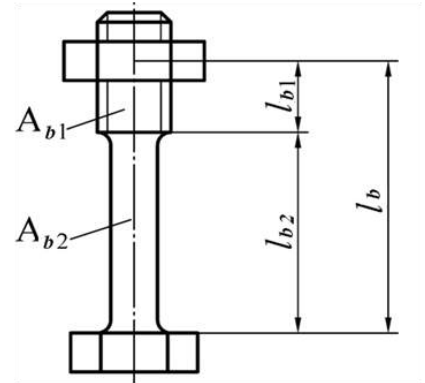


Fig. 2.28

2.1.14. Joint Stiffness

It was investigated that the force is transmitted from the bolt head and the nut to the joint parts along so called “cones of pressure” or “cones of influence” (Fig. 2.29, a).

It is presumed and experimentally shown that cones’ generators are inclined to the bolt axis at angle $\alpha = 26...30^\circ$ and diameter of the smaller basis of the cone is equal to the standard Width Across Flats (wrench size) S (Fig. 2.29, b).

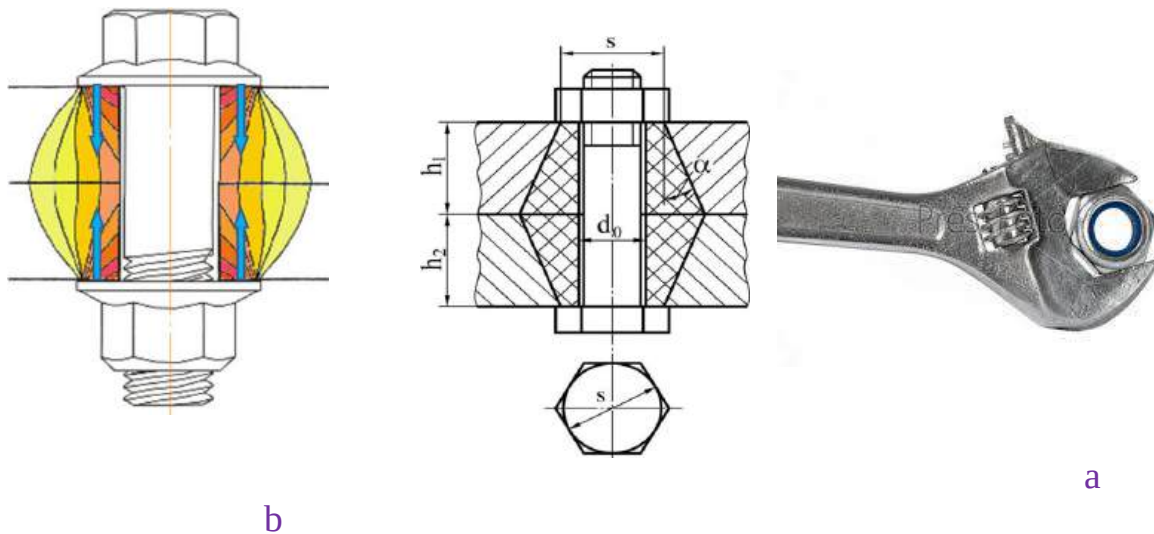


Fig. 2.29

The joint stiffness can be defined as

$$\frac{1}{c_j} = \frac{h_1}{E_1 A_E} + \frac{h_2}{E_2 A_E}.$$

To simplify the calculation, cones of influence are replaced with cylinders with diam. D_E and d_0 which axial sectional areas A_E are equal to the areas of the cones with- in the same sections (Fig. 2.30).

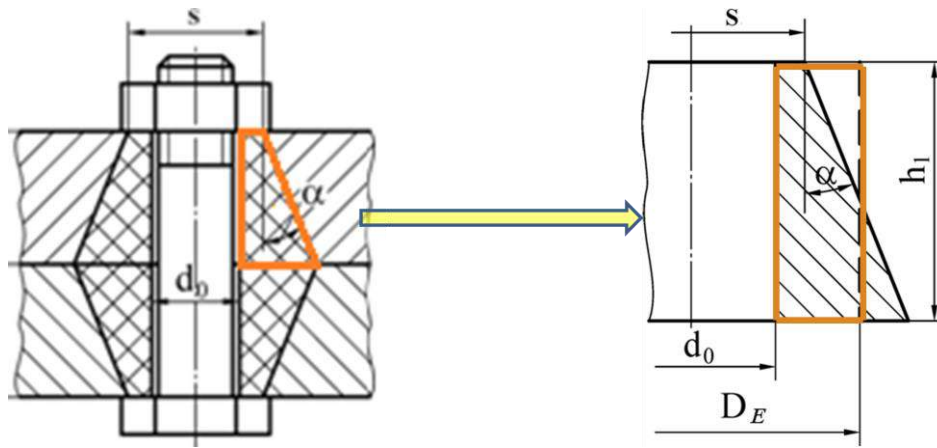


Fig. 2.30

The effective area A_E of the equivalent cylinder's cross-section with external diameter D_E

$$A_E = \frac{\pi}{4} (D_E^2 - d_0^2),$$

where external diameter (Fig. 2.31)

$$D_E = S + 2 \frac{1}{2} AB = S + h \cdot \tan \alpha = S + 0,5h.$$

and internal diameter (it is standard value) $d_0 = d + 1$.

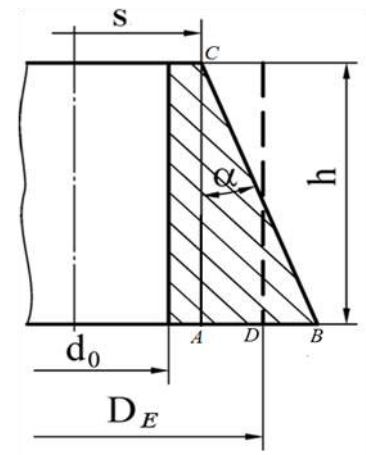


Fig. 2.31

2.1.15. Influence of the Operational Temperature on Prestressing of the Connection

If the connected parts and the bolt are made of materials with the same thermal expansivity, heating of all parts of the connection to the same temperature has no noticeable effects on force conditions in the connection.

However, some bolt connections get to a situation where heat from the heated parts is transferred to the bolt with difficulties, so that its temperature is lower than the temperature of adjacent parts. This increases the operational prestressing of the connection and thus also loading of the connecting bolt. Operational prestressing is also changed if the bolt and the connected part have different thermal expansivities. In both these cases, the correct design must include effects of operational temperature in the calculation of the connection.

2.1.16. Typical Bolt Failures

Bolt failures occur due to:

- overload;
- lack of locking mechanism;
- metal fatigue;

- improper torque;
- improper design or manufacture;
- corrosion;
- evidence handling.

Approximately 10...15 % cases of bolted joint failures usually initiate by axial force near the bolt head (Fig. 2.32, point 1), 15...20 % in zone of fillet between bolt shank and the thread portion, 60...70 % – by fatigue cycles at the root of the first bolt thread near the bearing surface of nut (Fig. 2.32, point 3), and this failure is encouraged by high stress concentration at the thread root.

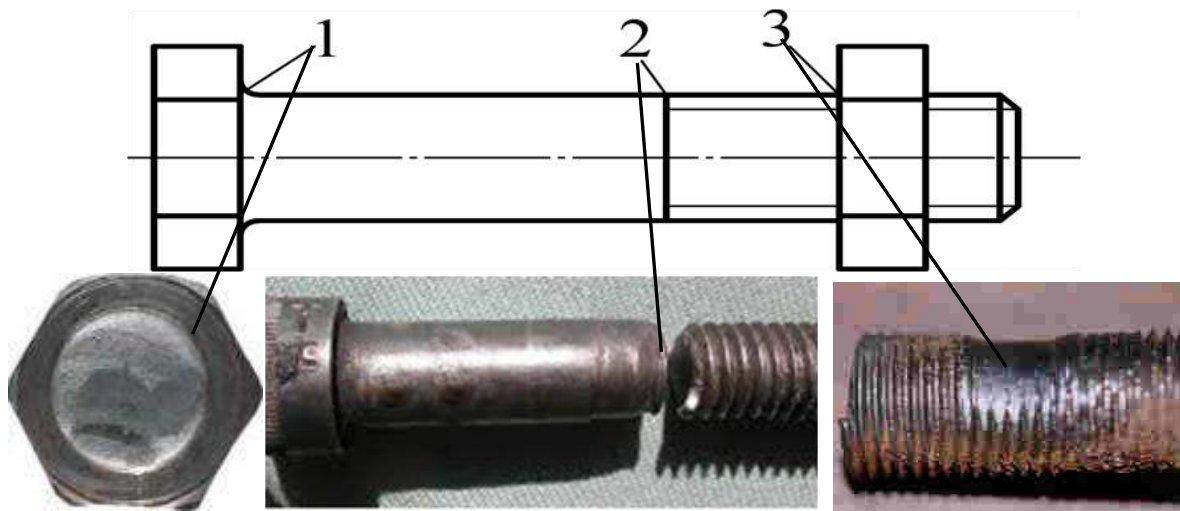


Fig. 2.32

2.1.17. Methods of Improving of the Threaded Joint Endurance

They are:

Design	Bolt stiffness decreasing
	Connected parts' stiffness increasing
	Decreasing of stress concentration on a bolt surface
	Prestressing of joint
	Usage of specific nut design
	Usage of high strength material
Technological	Thread rolling instead of cutting
	Thermo-chemical treatment
	Running rollers and shot blasting

Influence of Bolt Stiffness on Bolt Fatigue Endurance

In a case of low stiffness bolt with a high stiffness joint the bolt will only take a small proportion of the applied force (Fig. 2.33). The joint stiffness increasing causes the same result.

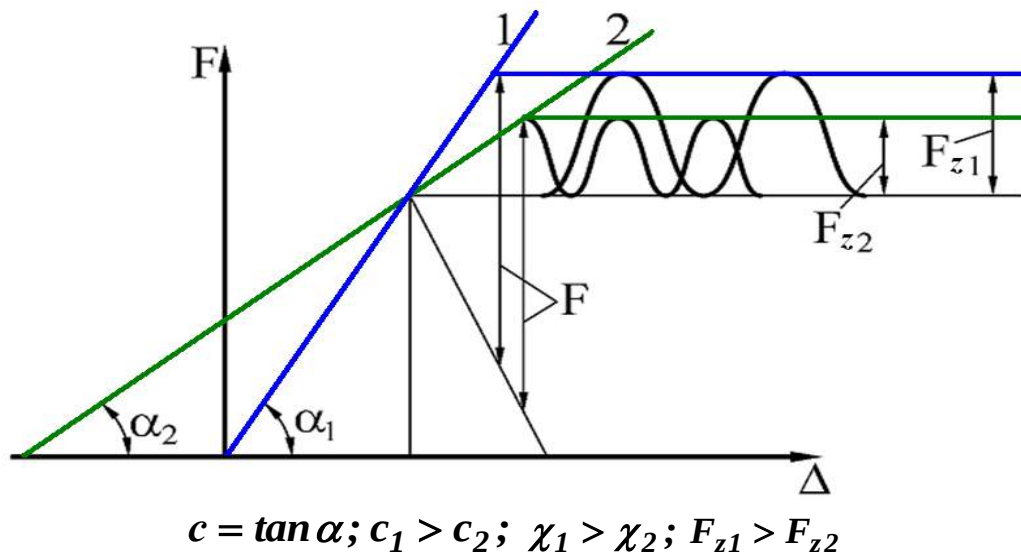


Fig. 2.33

Decreasing of Stress Concentration

Stress concentration can be decreased by the usage of improved thread profiles, another design of the imperfect-thread region and of transition zone between the shank and the head (Fig. 2.34).

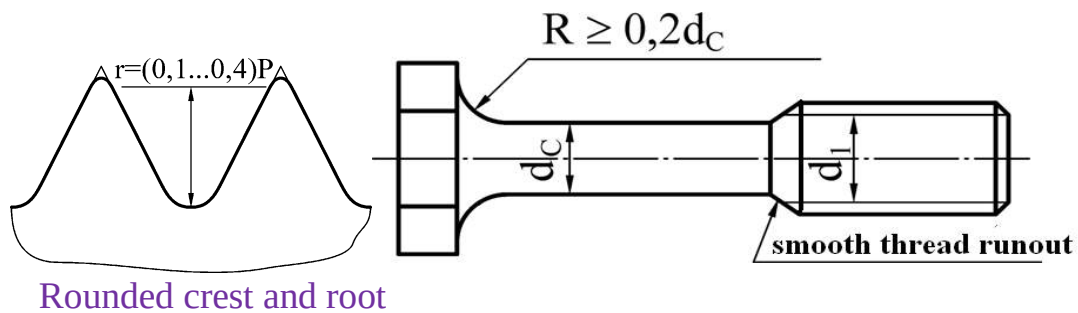


Fig. 2.34

Influence of Prestressing on Bolt Fatigue Endurance

It is known that the smaller σ_a is at the same σ_{max} the greater are the number of load variations required to cause the joint to fail, i.e., such a joint has a longer service life. The effect of the pre-load is to greatly reduce the magnitude of the alternating force F_a (force amplitude without preload F_{a1} much more than preloaded one F_{a2} , Fig. 2.35) and stress in the bolt.

2.1.18. Calculation of Group Bolt Joint (gap is not allowable)

Let us consider a bracket is fixed on a base-plate by means of 6 bolts. The joint is loaded by force F , directed at angle α to the joint plane (Fig. 2.36). Horizontal component F_x creates moment $M = F_x h$ and the vertical force F_y evenly loads all bolts.

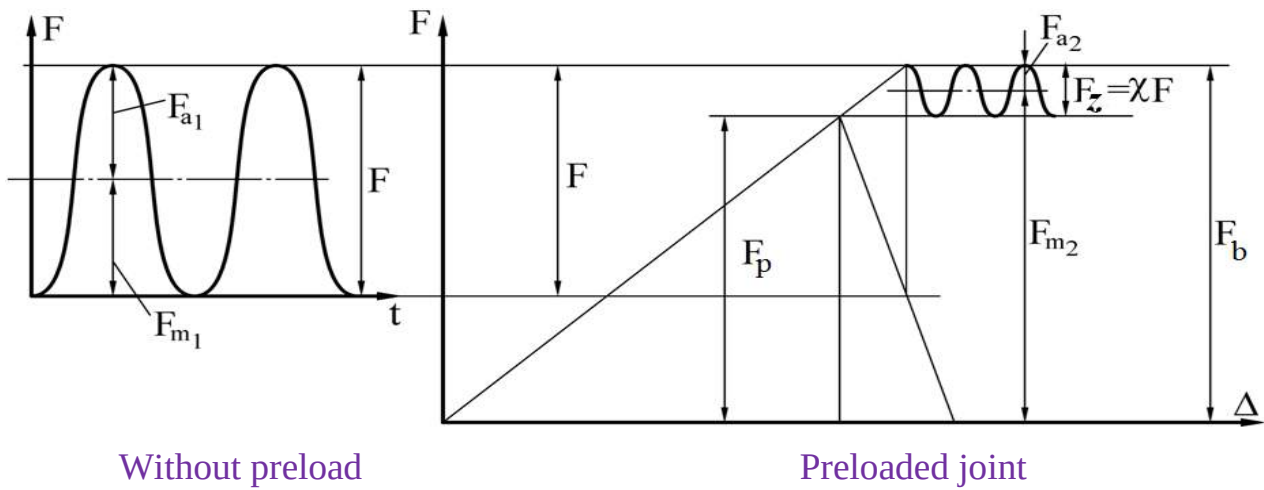


Fig. 2.35

Let us assume that all bolts are made from the same material, they have the same diameter and physical properties. Additionally, the bolts are placed symmetrically about vertical plane where force F lays.

To avoid shear stresses into the bolts a lug K is used.

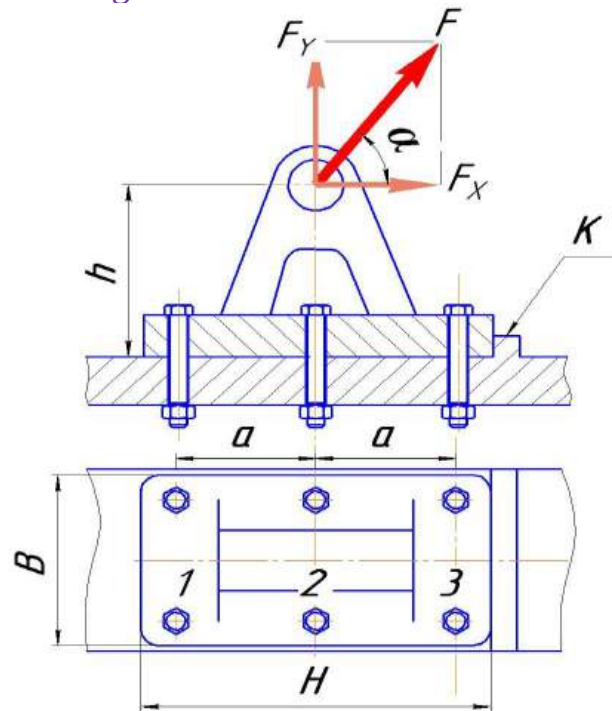
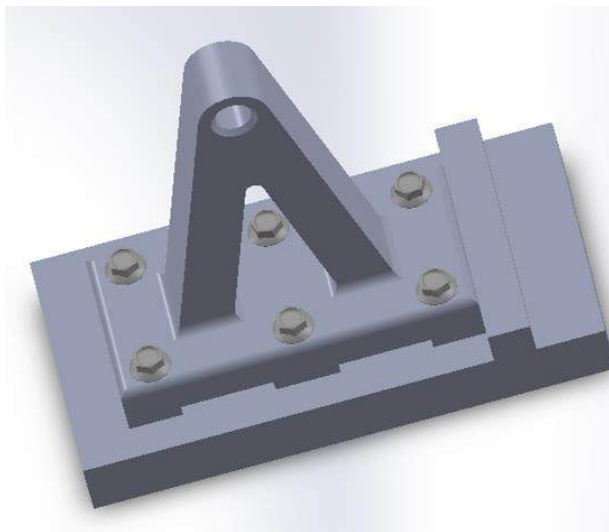


Fig. 2.36

To avoid the joint opening bolts should be pre-tightened with force F_p . After preloading, the joint acts as a solid body in which:

- force F_p creates compressing stresses into the joint (Fig. 2.37): $\sigma_p = \frac{F_p}{A_j} Z$;
- the component F_y produces tensile stress: $\sigma_y = \frac{F_y}{A_j}$;

– the overturning moment compresses one half of the joint and unloads other one:

$$\sigma_M = \frac{F_x h}{W}$$

Here $A_j = BH$ is the joint cross-section area and the section modulus is $W_j = \frac{BH^2}{6}$.

The external force will not open the joint if the summary compressive stress in the most loaded zone is more that tensile one:

$$\sigma_\Sigma = \sigma_p - (\sigma_y + \sigma_M) > 0.$$

Thus, the prestressing force

$$F_p = \frac{\sigma_p A_j}{Z} = \frac{\sigma_p BH}{Z} = k \left(\frac{F_y}{BH} + \frac{6F_x h}{BH^2} \right) \frac{BH}{Z} = k \left(\frac{F_y}{Z} + \frac{6F_x h}{ZH} \right),$$

where k is a safety factor, $k = 1, 2 \dots 1,5$; Z is the number of bolts.

We can find the forces acting in the bolts as

$$F_{b1} = F_p + F_{1ex}\chi; \quad F_{b2} = F_p + F_{2ex}\chi; \quad F_{b3} = F_p - F_{3ex}\chi.$$

The overturning moment creates pair of forces F_M , which can be obtained from the equilibrium condition: $\Sigma M = F_x h - NF_M 2a = 0$, where N is number of rows (in our case $N = 2$).

The external forces

$$F_{1ex} = \frac{F_x h}{N \cdot 2a} + \frac{F_y}{Z}; \quad F_{2ex} = \frac{F_y}{Z}; \quad F_{3ex} = \frac{F_x h}{N \cdot 2a} - \frac{F_y}{Z}.$$

When we know the forces acting in the bolts, the minor diameter of the most loaded bolt # 1 can be obtained from tensile strength condition (see Bolt Design Calculation).

2.1.19. The Fasteners Fixing Methods

Practice has shown that the threaded joints can reliably operate at variable loads with locking only. Two basic methods are used in fixing: positive (by special parts, see below) and frictional (create additional friction between the threads and/or between nut and body, like spring washer, nylon insert lock nut, check nut etc.)

The word safetying is a term universally used in the aircraft industry. Briefly, safetying is defined as: "Securing by various means any nut, bolt, turnbuckle etc., on the aircraft so that vibration will not cause it to loosen during operation." Frictional

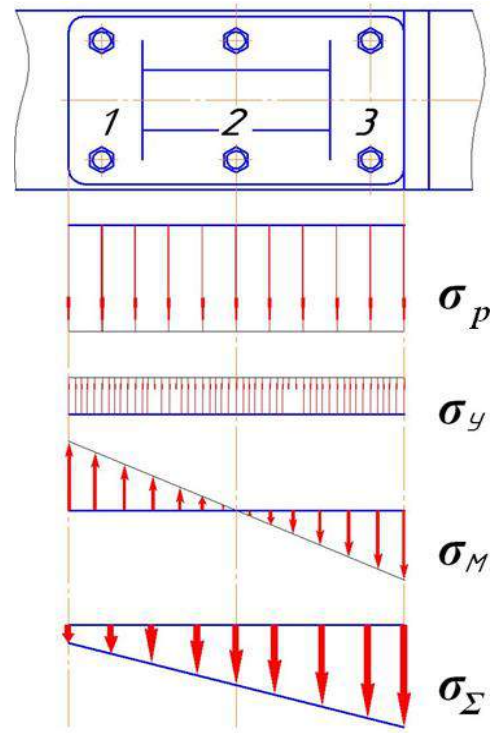


Fig. 2.37

locking is less reliable, so in all crucial joints and inside the machine only the positive locking is used.

A positive locking device is a device used in conjunction with a fastener in order to lock the fastener by meshing. The following positive locking devices are widely used [1]:

- a split beam nut and a split pin;
- a castellated nut and a split pin;
- a hex nut or cap screw and a tab washer;
- a hex nut or cap screw and a lock plate;
- safety wiring with various types of fasteners.

Tab washers

Tab (lock) washers (Fig. 2.38) are a very simple positive locking but effective and reliable method of fastening. An internal tab fits into the keyway of a shaft and an outer tab is bent over into one of the slots located on the outside diameter of the ring nut. In some cases, one or more of the tabs are pre bent, the remaining tab or tabs being bent into position after assembly.

Ideal for use in harsh environments, tab washers effectively lock a part into place in applications requiring extreme heat conditions or heavy vibrations. Standard sizes allow using of a tab washer with all types of nuts, bolts or fastening systems.

Another type of tab washer (Fig. 2.39) takes the form of plain round washers with one or more tabs and notches, which can be bent to provide a mechanical lock between a nut or bolt and the component being clamped to.

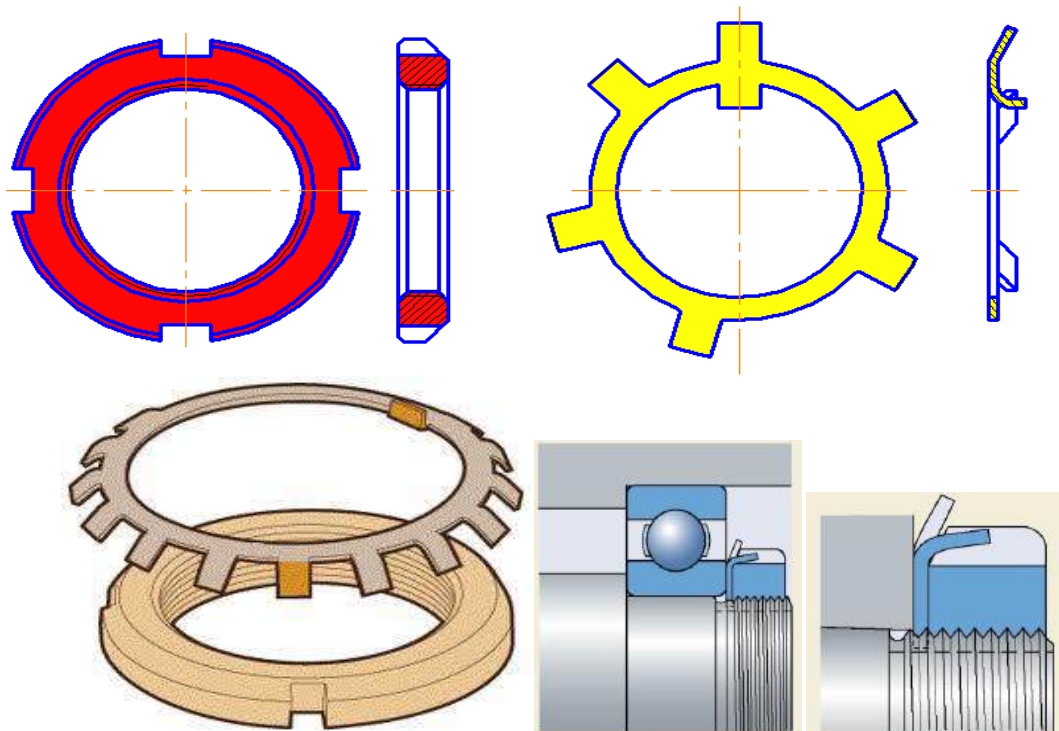


Fig. 2.38

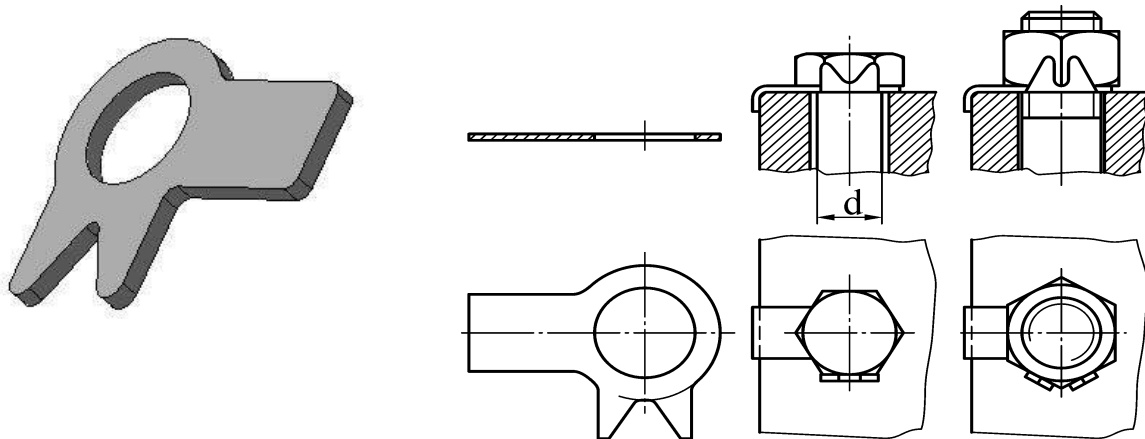


Fig. 2.39

Locking Wire

Safety wire or lock-wire (Figs 2.40, 2.41) is a type of positive locking device that prevents fasteners from loosening or falling out due to vibration and other forces. The presence of safety wiring may also serve to indicate that the fasteners have been properly tightened.

Safety wire is available in a variety of gauges and materials, depending on the application. In aircraft and racing applications, stainless steel wire is used, such as in 0.8 mm diameter. Typically, the wire is threaded through a hole drilled into a fastener or part, then twisted and anchored to a second fastener or part, then twisted again.

There are a few techniques for different applications. The wire itself maintains tension and remains in place by being twisted around itself and attached to the fastener to be secured on one end and an anchor point (which could be another fastener) on the other end. Since safety wire is made of a malleable alloy, it retains its shape after being bent, rather than springing back to its original shape. This property allows it to remain locked around an object, such as when it is passed through a small hole on a fastener, looped back upon itself and then twisted. The same process is then repeated around the anchor point, which could be another fastener. Since it remains twisted instead of unraveling, it acts as a fixed loop and will not back out without considerable force (greater than the stresses which it is intended to counter) being applied.



Fig. 2.40



Fig. 2.41

Safety wire is not reusable, thus it can be cut apart in order to remove it easily when the fastener is to be opened.

Advantages and disadvantages. Although many systems purport to be more efficient than installing traditional safety wire, an advantageous by-product of the twisting method of installing safety wire is that it leaves a highly visible and easily inspectable indication that the fasteners in question are in fact properly secured.

In addition, safety wire twisting is a standard, non-proprietary technique, and tools and materials can be easily found, cheaply purchased, and mixed with other brands while still working properly (provided of course that all components are used properly and with the proper types of complimentary components and tools, if not brands).

The primary disadvantage of traditional safety wire is the time it requires to install properly when securing fasteners, although technicians who use it often can implement it fairly quickly. It also leaves behind waste products when ends are clipped off or when it is cut off of secured fasteners that need to be removed during maintenance, resulting in sharp metal bits that can easily damage soft materials or injure skin. However, the amount of waste product is relatively small, it is non-toxic, and the hazard can be mitigated altogether if technicians properly dispose of any waste product.

Split Pins

Split pins are among the cheapest and most widely used locking fasteners for low speed, low stress parts. Split pins are very common as they are easy to use and effective. They are usually made from ‘soft’ steel or bronze, brass, aluminum and they come in a range of sizes.

The split pin typically consists of a flat, round or half round pin bent double with a pronounced loop at the closed end (Fig. 2.42). After a part has been put in place, the pin is passed through a normally drilled hole in the shaft and one or both its “legs” bent back to prevent it working loose while the loop on the closed end stops it from falling through the hole. If the part has to be removed, the bent pin legs are simply straightened and the

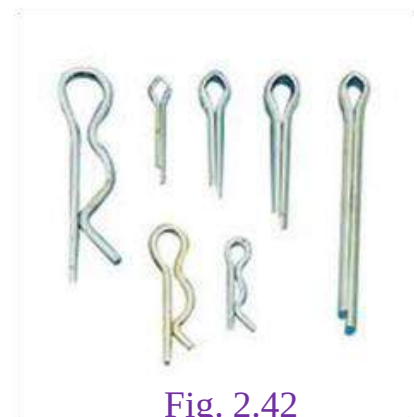


Fig. 2.42

split pin removed.

There are different cross section profiles available such as round, flat or half round. The split pin is also available with different end profiles including bevel ends, miter ends, and extended prong pins all of which are suited to specific uses.

A washer can be used to reduce friction between the split pin and the wheel. Split pins are strong but after time they can wear and break.

One of the well known uses of the split pin is the locking mechanism for the castellated nuts commonly used on automobile hubs (Fig. 2.43). These lightweight nuts lock the hubs in place and exert an appropriate amount of torque on the wheel bearings. They feature a series of raised tabs along their upper edges reminiscent of castle battlements. The stub axle has a hole drilled through it, and once the nut is tightened to the correct torque, the split pin is inserted through the hole between the tabs on the nut. The ends are then bent backward to lock the pin and prevent the castellated nut from loosening.

Unfortunately, the legs will only survive being bent once or twice before they break off. For this reason, split pins should only be used once and then discarded; this is quite acceptable when considering the low cost of these versatile little fasteners.



Fig. 2.43

Nut Lock Plates

Self-locking nut plates (Fig. 2.44) are pressed into the nut plate, eliminating the problem of dropping or losing hardware during installation. The nut plate becomes a captive part eliminating loose pieces, adding stability, and saving valuable time. Fixed nuts are designed for blind mounted applications where it is impractical to use hex nuts or where it is desirable to have an attached nut for ease of maintenance.

A plate nut, also known as a nut plate, anchor nut or anchor plate, is a stamped sheet metal nut that is usually riveted to a workpiece. They have a long tube that is internally threaded and a plate with two clearance holes for rivets. The most popular versions have two lugs and they exist as fixed anchor nuts and as floating anchor nuts.^[2] The latter allows the nut to move slightly and so enlarges the positioning tolerances of the mounted parts. They were originally developed for the aerospace industry, but are now also common in automotive racing. These nuts are made up of variety of soft and

hard materials. The choice of material depends on environment to which nut is subjected. Soft materials like copper or brass are used when nut is used in electrical application. Hard materials are used when nut is subjected to high stress environment. Sometimes stainless steel or nickel plated nuts are used in order to increase corrosion resistance.

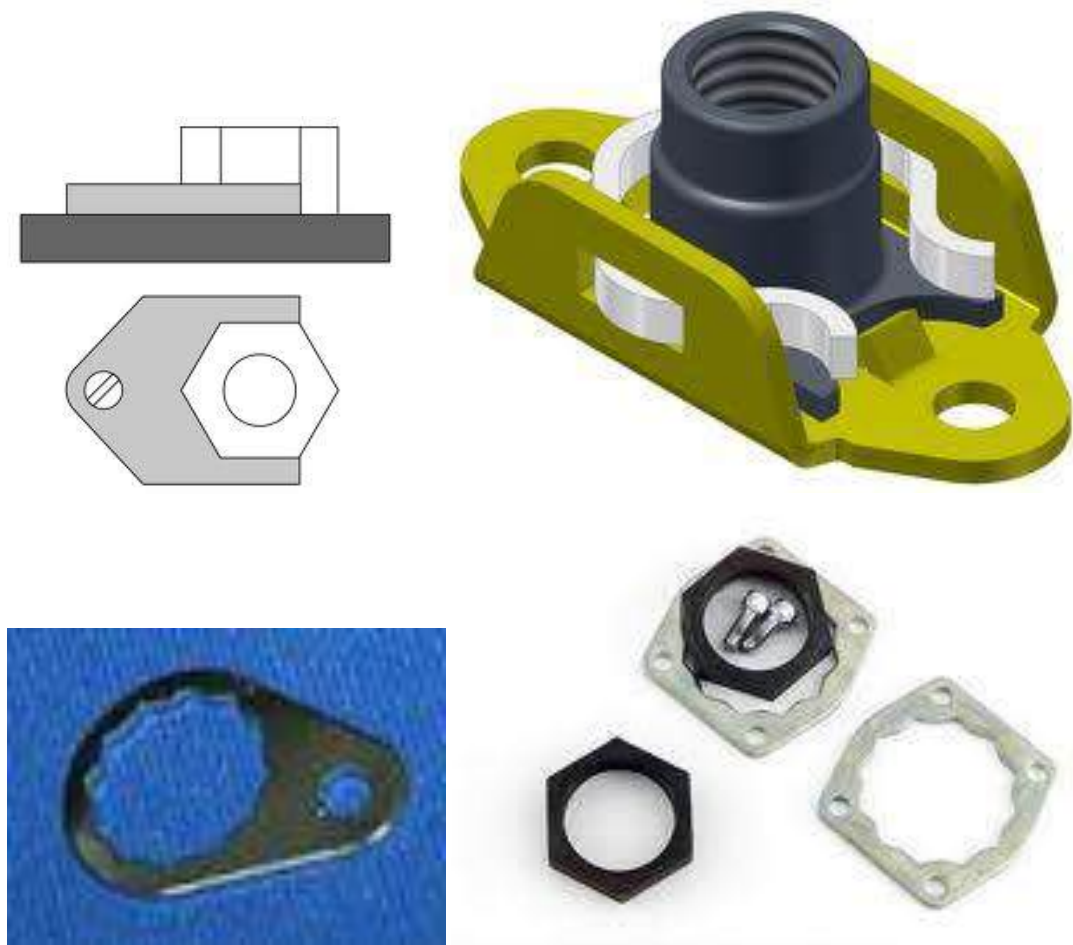


Fig. 2.44

Spring Washers

Spring washers, sometimes called disc springs, are a subtype of washers (Fig. 2.45). They lend their mechanical capabilities to the unique profile of the material: when subject to a load, the irregularities of the washer compress with a proportionate resistance to return to their pre-deflected shape. Spring washers are employed in applications where assemblies need a part to take up play, eliminate rattle, maintain assembly tension, compensate for expansion or contraction in materials after assembly, or to absorb intermittent shock loads and provide a controlled reaction under dynamic loads.

Deflecting under a load, a spring washer is very much like a compression spring. When under stress, the raised irregularities of the disc spring flatten to a predetermined degree. Spring washers are either under static or dynamic load. In a static load application, the basic function of the washer is to retain load and the elastic load rating of the washer material may be exceeded. For a dynamic load, the washer functions

as a flexing spring and the yield limit must not be exceeded or permanent disfiguration will occur.



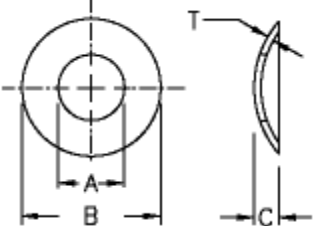
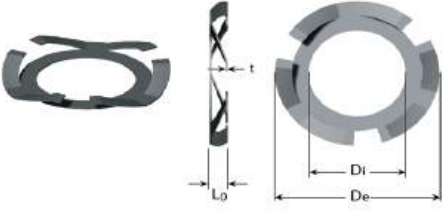
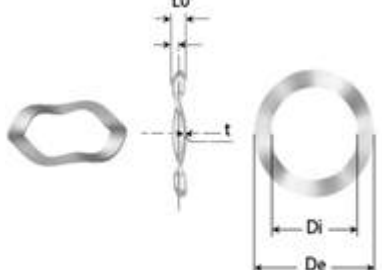
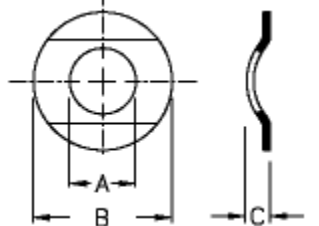
Fig. 2.45

Spring washers undergo elastic deformation which is reversible and is determined by the material of construction. If the washer is overstressed or near the end of its cycle life, it will experience plastic deformation where some of its deflection is irreversible; this is the first sign of washer fatigue.

There are six primary instances of spring washers (Table 2.2). Other types may exist as a specialty option through manufacturers.

Table 2.2

Belleville washers can support high loads with small deflections. The load and deflection capability is dependent on height/thickness ratio. These are common in thermal expansion applications.	
Crescent washer is meant for lighter loads and produces a small deflection. There is a uniform spring rate throughout the washer's deflection. This is used in flexible, load-cycling applications.	

Dome washers have a very high load capacity with a small overall deflection. While similar in appearance to a crescent washer, a dome washer has ground curves to create a flatter load-bearing surface.	
Finger washers are notable for their split construction with protruding flanges. They combine the flexibility of a crescent washer with the distributed load points of a wave washer. They are common for noise and vibration dampening instances.	
Wave washers offer moderate load capacity and deflection, and are typically used as cushions or spacers. These have multiple waves within the washer.	
Single wave washers are similar to crescent washers but have flattened load points on the washer. This is meant to minimize abrasion against sensitive surfaces.	

Steel is the most common material for spring washers due to its ease of process and its high tensile strength. Chrome vanadium alloy is a higher quality steel with a good through-hardening capability; it is used to make very thick spring washers. Specialty washers offering corrosion resistance include steel-nickel alloys. High-temperature applications incorporate chrome-molybdenum into the material structure to create washers that can operate over 400°C.

Copper alloys offer non-magnetic yet electrically conductive washers, along with some corrosion resistance. These alloys have lower strength values than steel.

Nickel-chrome and nickel-chrome-cobalt alloys have very high tensile strengths with very low elastic ratios. These washers will have more set if overloaded, but they have extreme longevity and corrosion resistance. They can also operate in a large range of temperatures, from absolute zero to over 700°C. These expensive alloys are difficult to manufacture.

The tension generated by the deflection of the **Regular helical spring split lock washer** (Fig. 2.46) meets the needs of most commercial applications. It is, therefore,

the most popular type spring lock washer in terms of usage and production. Historically, helical spring lock washers have been among the most widely used anti-loosening assembly components on the market. They have been specified in thousands of assembly applications because they work – and work extremely well.

They enhance the security of general industrial assemblies in the following ways:

- apply greater bolt tension per unit of applied torque;
- provide hardened bearing surfaces to create more uniform torque control;
- provide more uniform load distribution through controlled radii (section) cutoff;
- provide protection against looseness resulting from vibration and corrosion;
- optimum locking device for use in applications with hardened faying or bearing surfaces.

The “Split” in the helical spring lock washer absorbs initial driving torque and visually closes under nominal bolt loading. When tension in the assembly is reduced and loosening occurs, it provides resistance to the back-off rotation of the screw (Fig. 2.47).

The spring rate, which is developed by the final deformation, is very high and provides a reactive load that is equivalent to a significant increase in effective bolt length.

The clamped components are compressed and, in their inherent effort to expand to their original form, they set up an opposing force. It is these two forces that create the dynamics of a tight assembly.

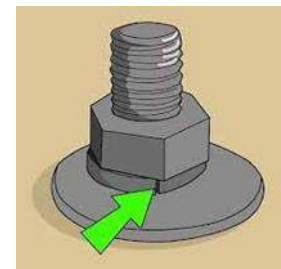


Fig. 2.46

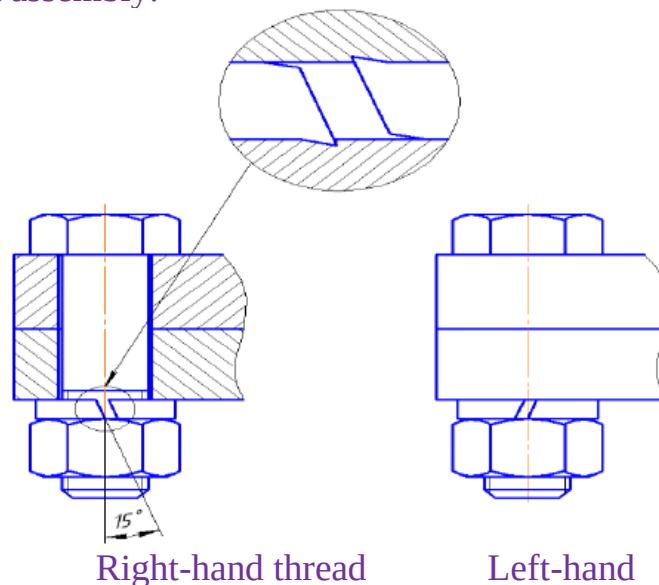


Fig. 2.47

Nylon Insert Lock Nut

Locknut types (Fig. 2.48) are available, which utilize deformed threads or a nylon locking element, much like a nyloc nut. These locking plate nuts are used when the nut is subjected to constant changes in environment or to vibration. The nylon insert expands and stops vibrations of the nut and acts as a locking arrangement.



Fig. 2.48

Check Nut

A check nut (Fig. 2.49) is used to ensure that a nut, once installed, will stay in place. Check nuts, also called lock nuts, are usually thin and used in instances where there is a high probability of the first nut shaking or vibrating loose. Check nuts are designed and used to prevent the first nut from becoming loose and potentially causing damage or injury when using a device. Lock nuts, or check nuts, were designed to replace lock washers, which did not work as well at preventing loosening over time from movement and vibration.

To install a check nut properly, it is necessary to have the stud or bolt the check nut is being applied to and one or two correct sized wrenches. Next, the user must take the check nut in hand.

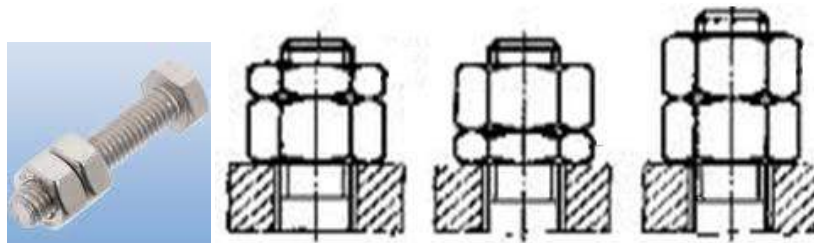


Fig. 2.49

To install, the user must slide the check nut over the shaft of the bolt or stud and turn it in a clockwise direction until the lock nut sits just against the other nut. It will be clear when it has done this because the user will feel some resistance and will no longer be able to turn it by hand.

2.1.20. A Bolt Designation

There are a lot of bolt designation systems. GOST variant includes information as follows: Bolt (*deg. of acc.*) *M* (*Diam.*) x (*Pitch*).(*Tolerance*) x (*Length*).(*Strength Code*).(*Material*).(*Coating Code*) *Standard*.

For example: Bolt B M12x1,5.6g x 60. 56.40X.00 GOST 7796. Here:

– B is the degree of accuracy (default A is not shown);

- M is the type of thread – metric;
- 12 mm is the major diam.;
- 1,5 mm is the pitch – fine, coarse pitch is not shown);
- 6g is the tolerance of the pitch diam.;
- 60 mm is the bolt length (head height is not included);
- 58 is the Strength Code: 5.6 (see below);
- material: steel 40X;
- coating code: 00 (see in standard).

Some of widely available grades of metric bolts are:

Grade	Minimum UTS, MPa	σ_y , MPa	Material
4.6	$4 \cdot 100 = 400$	$400 \cdot 0,6 = 240$	Cт. 10
5.6	$5 \cdot 100 = 500$	$500 \cdot 0,6 = 300$	Steel 30, 35
8.8	$8 \cdot 100 = 800$	$800 \cdot 0,8 = 640$	med. carbon
10.9	$10 \cdot 100 = 1000$	$1000 \cdot 0,9 = 900$	alloy steel, med. carbon steel, Q & T*, or med. C steel with addi- tives Q & T
12.9	$12 \cdot 100 = 1200$	$1200 \cdot 0,9 = 1100$	alloy steel, Q & T Steel 40XH2MA, 35XГCA

*Quenched and Tempered Low-Alloy Steel.

2.2. Key Joints

Key is a piece of mild steel inserted between a shaft and a hub (Fig. 2.50) used for connection a rotating machine element to a shaft. It prevents relative rotation between the two parts and enables torque transmission.

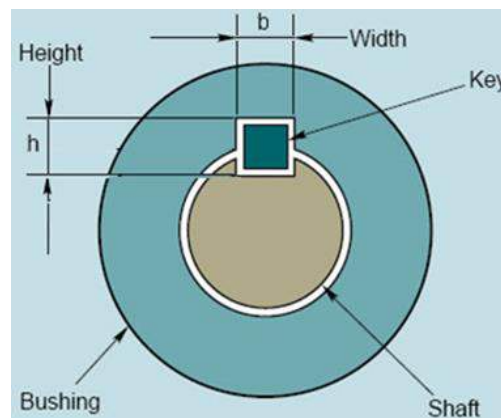


Fig. 2.50

The recess machined in a shaft or hub to accommodate the key is called a keyway. Keyways can be milled horizontally (Fig. 2.51, *a*) or vertically (Fig. 2.51, *b*). Keys are made of steel, in order to withstand the considerable shear and compressive stresses caused by the torque transmit.

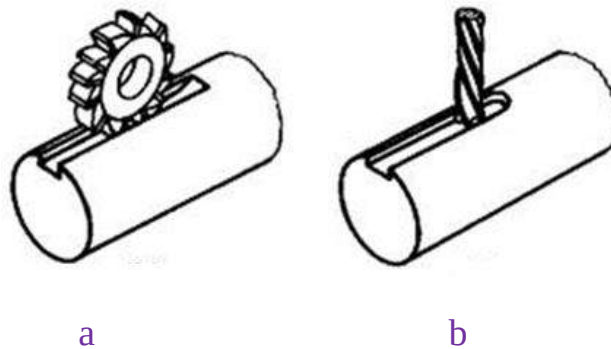


Fig. 2.51

Advantages	Disadvantages
• simple design and convenient assembling-disassembling • high shaft alignment • standardization	• considerable stress concentration • tendency to turning out of keyway • low load capacity • low rotation speed • interference fit is required

2.2.1. Classification

There are two basic types of keys:

- *saddle keys*, which are sunk into the hub only. These keys are suitable only for light duty, since they rely on a friction drive alone;
- *sunk keys*, which are sunk into the shaft and into the hub for half their thickness in each. These keys are suitable for heavy duty, since they rely on positive drive.

Parallel keys (Fig. 2.52) are the most widely used. They have a square or rectangular cross-section. Square keys are used for smaller shafts and rectangular faced keys are used for shaft diameters over 170 mm or when the wall thickness of the mating hub is small. Set screws often accompany parallel keys to lock the mating parts into place. The keyway is a longitudinal slot in both the shaft and mating part.

Feather key is a type of parallel keys and is used when the hub is required to slide along the shaft. It is lightly fitted or secured by means of screws in the shaft keyway, and is made to slide in the hub keyway.

Rectangular and square keys can be parallel or tapered with a basic taper of 1:100 to prevent sliding. These keys are used for heavy-duty applications. It is advised to use square keys for assembly-drawing solutions.

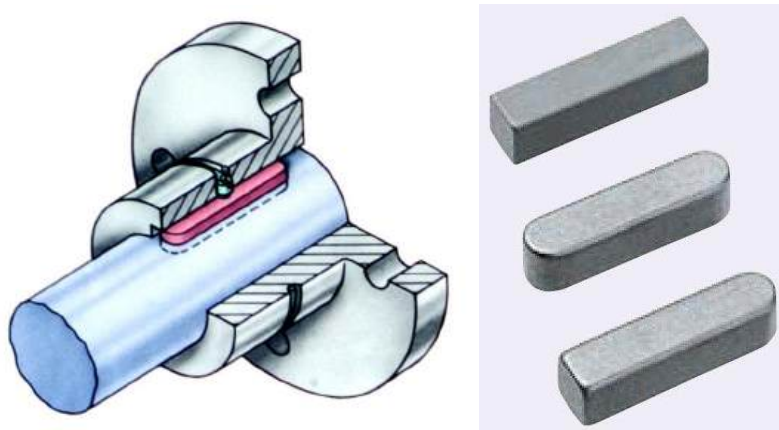


Fig. 2.52

Gib heads key (Fig. 2.53) are sometimes provided on taper keys to facilitate their withdrawal. It is usually used where removal of the key is required frequently.

The tapered key is tapered only on the side that engages the hub. The keyway in the hub has a taper that matches that of the tapered key. The purpose of the taper is to secure the key itself, as well as, to firmly engage the shaft to the hub without the need for a set screw. The problem with taper keys is that they can cause the center of the shaft rotation to be slightly off of the mating part.

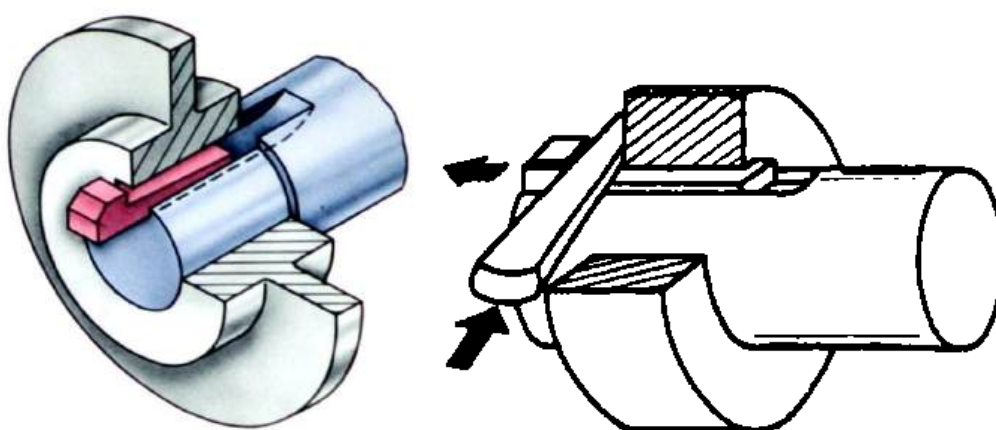
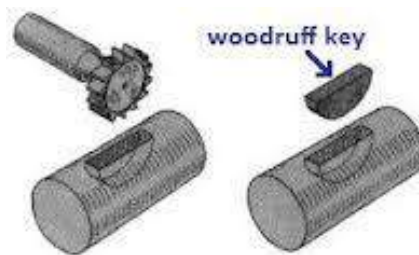
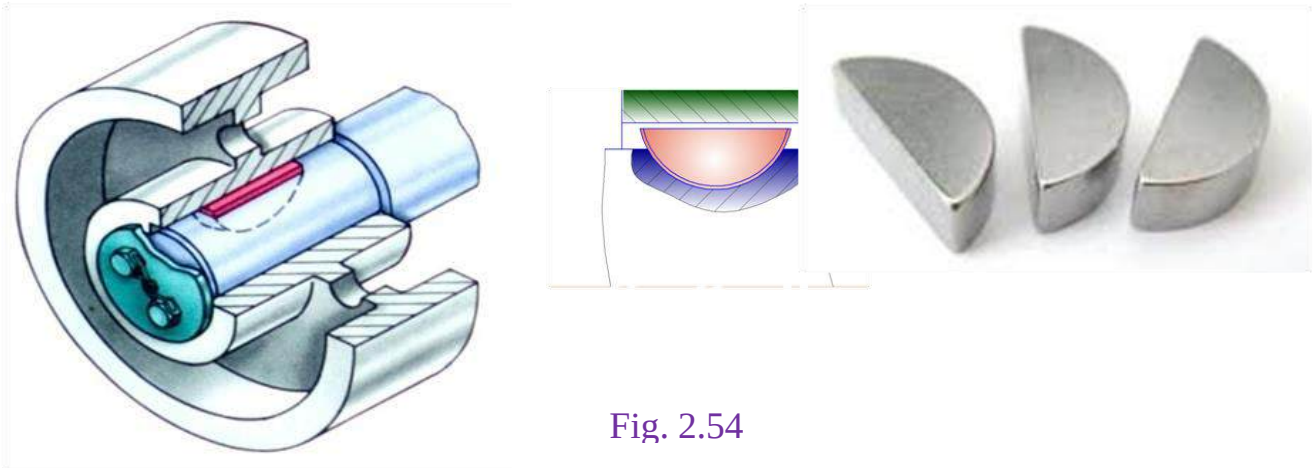


Fig. 2.53

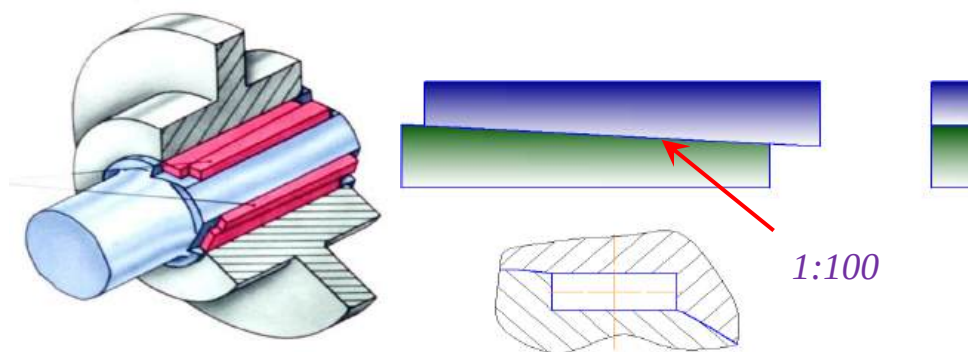
Woodruff keys (Fig. 2.54) are semicircular shaped, such that, when installed, leave a protruding Table The keyway in the shaft is a semi-circular pocket, the mating part is a longitudinal slot. They are used to improve the concentricity of the shaft and the mating part, which is critical for high-speed operation.

The main advantage of the Woodruff key is that it eliminates milling a keyway near shaft shoulders, which already have stress concentrations. Moreover, a woodruff key is capable of tilting in a recess milled out in the shaft by a cutter having the same curvature as the disc from which the key is made (Fig. 2.55).



Common applications include machine tools, automotive applications, snow-blowers and marine propellers.

Tangential keys (Fig. 2.56) are fitted in pair at right angles. Each key is to withstand torsion in one direction only. These are used in applications with large heavy-duty shafts.



2.2.2. Parallel Key Calculation

Keys are subjected to considerable bearing, shear and bending stresses (Fig. 2.57).

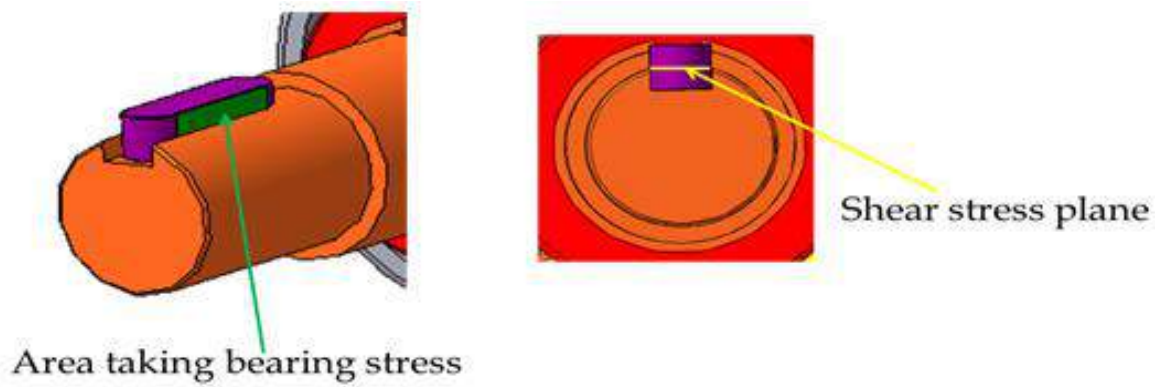


Fig. 2.57

In designing a key bending, mounting and fitting stresses acting on a key are neglected. Also, it is assumed that:

- the distribution of forces along the length of the key is uniform;
- the tangential force F acts tangent to the surface of the shaft and it is applied

at the outside shaft diameter d (Fig. 2.58): $F = \frac{2T}{d}$. This force produces both shear and compression stresses in the key.

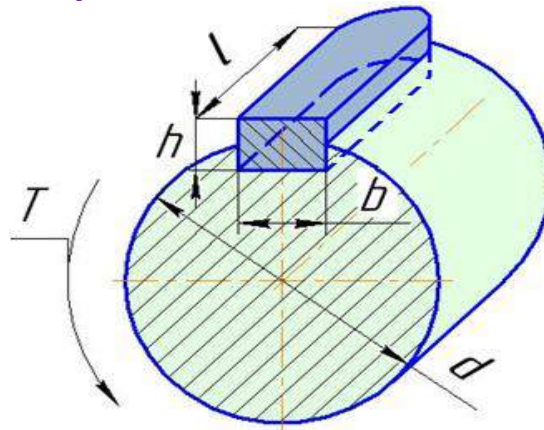


Fig. 2.58

All key types are standardized. The standard key dimensions b and h are usually selected according to a shaft diameter and are provided to be equally strong in shearing and bearing. That is why key strength condition is bearing only:

$$\sigma_b = \frac{F}{A_b} \leq [\sigma_b], \text{ where } A_b = 0,4hl.$$

Pay attention that design length l is the length of the key parallel faces and the second dimension equals to $0,4h$ because the contact height is less than half of total height h due to chamfers on both edges.

$$\text{Thus, } \sigma_b = \frac{2T}{d \cdot 0,4h} \leq [\sigma_b] \text{ and the design length } l \text{ is obtained as } l = \frac{2T}{d \cdot 0,4h[\sigma_b]}.$$

Conditionally reduced allowable bearing stress $[\sigma_b] = 80...120$ MPa is used because it is very difficult to take into account all acting factors.

2.2.3. Design Rules

Usually rules as follows should be taken into account:

1. If installation of two or more keys on one shaft is required, the keyseats should be placed on the same line (Fig. 2.59). In this case, standard allows to use the keys with the same cross sections corresponding to the smaller diameter.
2. L/d ratio should be equal $0.8...2$. The greater L/d the greater non-uniformity of stress distribution.
3. The element placed on the key should be free to pass over the shaft before the work surfaces (up to the point where parallel faces start, Fig. 2.60).
4. To avoid superposition of stress concentrations the clearance between keyseat and thrust fillet should be ensured (Fig. 2.61).

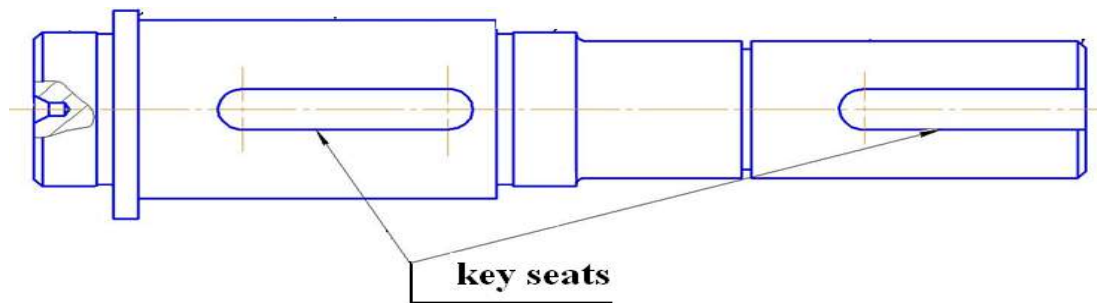


Fig. 2.59

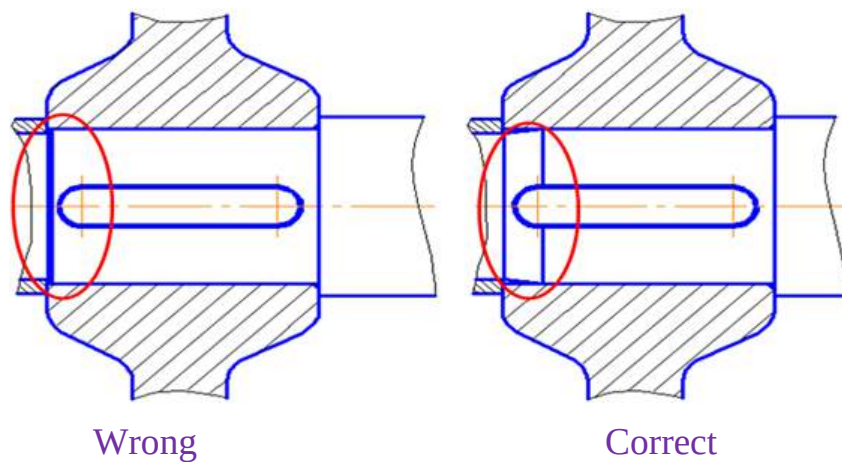


Fig. 2.60

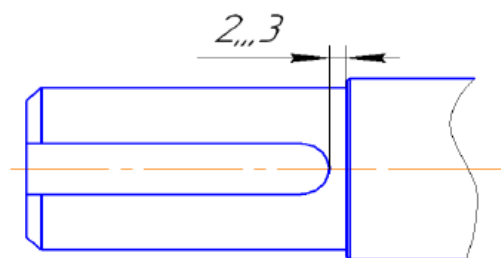


Fig. 2.61

2.3. Splines Joints

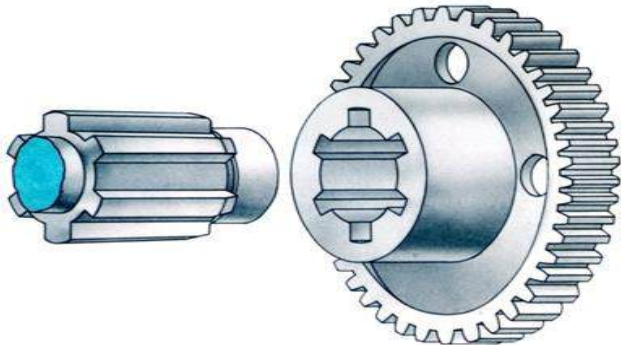
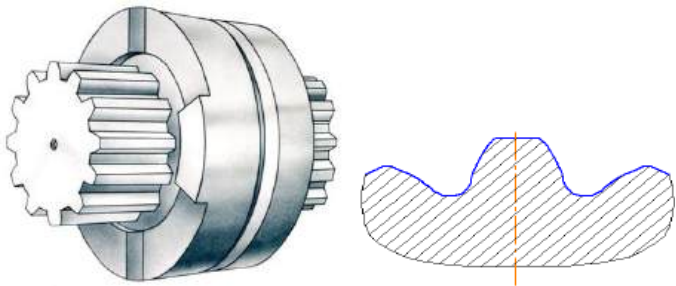
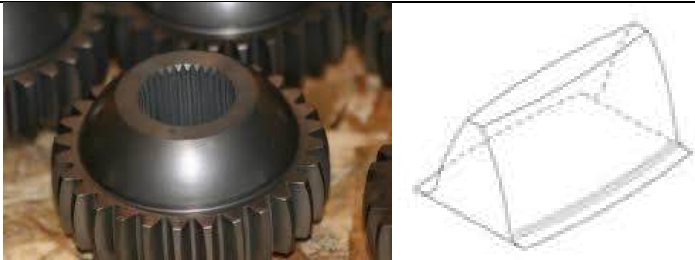
Splines are “keys” which are integral with the shaft. The splined shafts are used when:

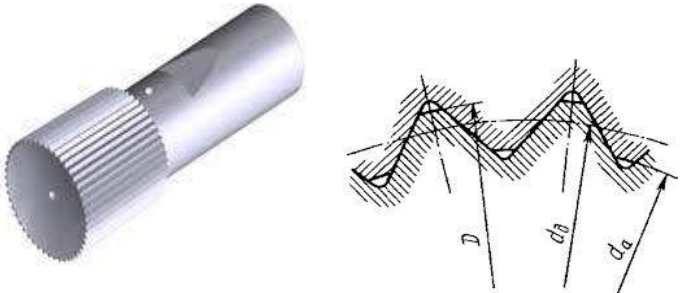
- the force to be transmitted is large in proportion to the size of the shaft;
- high rotational speed;
- axial movement is needed.

Advantages	Disadvantages
• significantly lower stress concentration • greater load capacity • ability to operate at high speeds	• more expensive and difficult machining • uneven load distribution between splines

2.3.1. Types of Splines

The most widely spread types are:

<i>Parallel key spline</i> The sides of the equally-spaced grooves are parallel in both directions, radial and axial	
<i>Involute spline</i> The sides of the equally-spaced grooves are involute, as with an involute gear, but not as tall. The curves increase strength by decreasing stress concentrations.	
<i>Crowned splines</i> Here the sides of the equally-spaced grooves are usually involute, but the male teeth are modified (rounded) to allow for misalignment	

<i>Serrations</i> The sides of the equally-spaced grooves form a "V". These are used on small-diameter shafts	 The photograph shows a shaft with a section of serrations. The diagram to the right shows a cross-section of the serrations with labels: D for the outer diameter, d_h for the hub diameter, and d_a for the addendum diameter.
<i>Helical splines</i> The equally-spaced grooves form a helix about the shaft. The sides may be parallel or involute. This can either minimize stress concentrations for a stationary joint under high load, or allow for rotary and linear motion between the parts	 The photograph shows a helical spline shaft and a matching hub. The shaft has a helical pattern of grooves, and the hub has a corresponding helical pattern of teeth.

2.3.2. Centering

The method depends on the required accuracy of fitting hub onto shaft and the load conditions. For parallel splines see Fig. 2.62, for involute – Fig. 2.63.

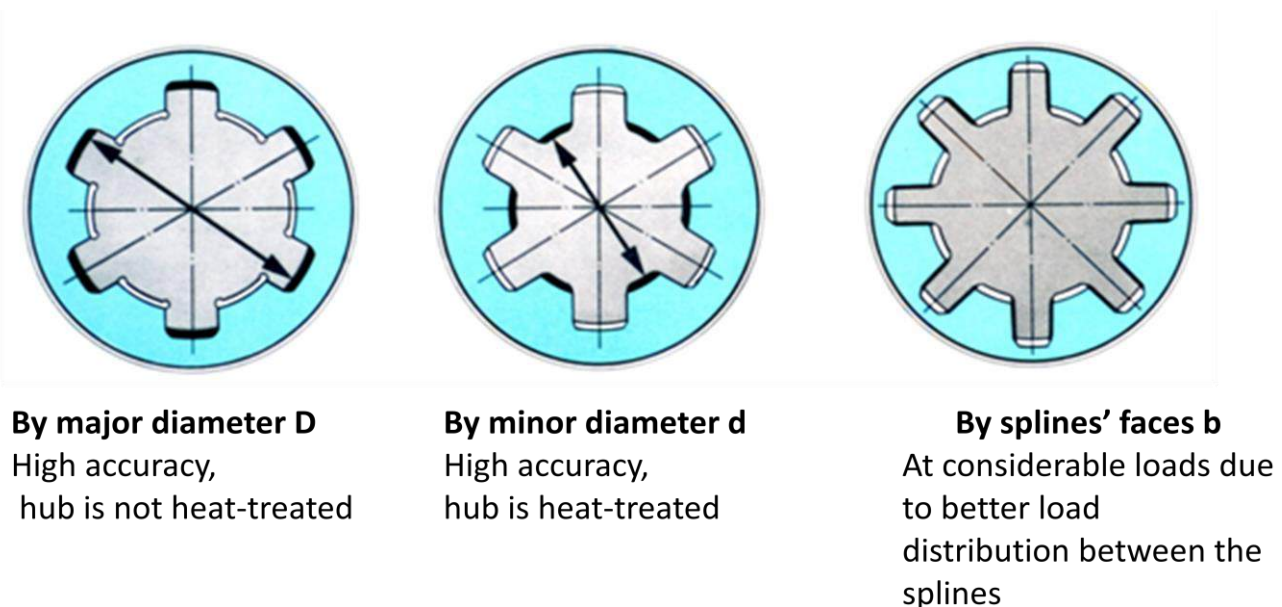
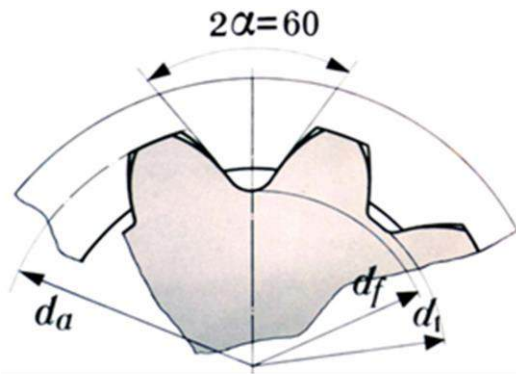
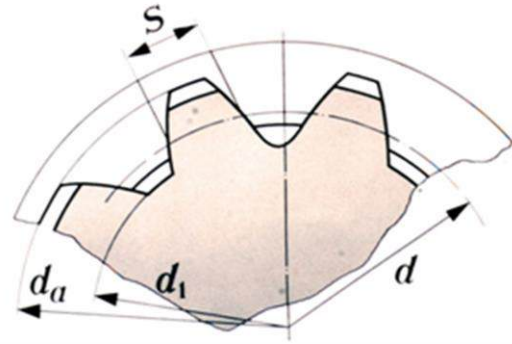


Fig. 2.62



By major diameter D



By splines' faces
Hub self-aligns under load.
This is the preferred method

Fig. 2.63

2.3.3. Calculation

On bearing strength condition:

$$\sigma_b = \frac{F_t}{A_b} = \frac{2T}{d_m z h l \psi} \leq [\sigma_b], \quad \sigma_b = 80 \dots 160 \text{ MPa.}$$

For straight-sided splines (Fig. 2.64):

pitch diameter: $d_m = \frac{D+d}{2}$;

height: $h = \frac{D-d}{2} - 2f$;

non-uniformity factor: $\psi = 0.75 \dots 0.85$.

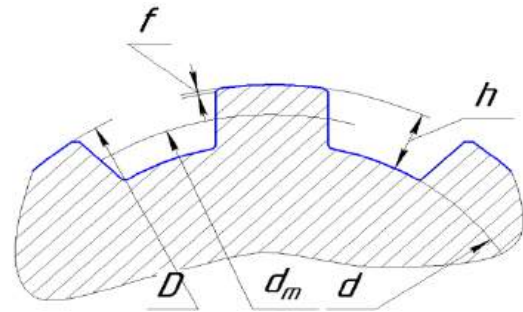


Fig. 2.64

For involute splines (Fig. 2.65):

pitch diameter: $d_m = mz$;

height $h \approx 0.9m$.

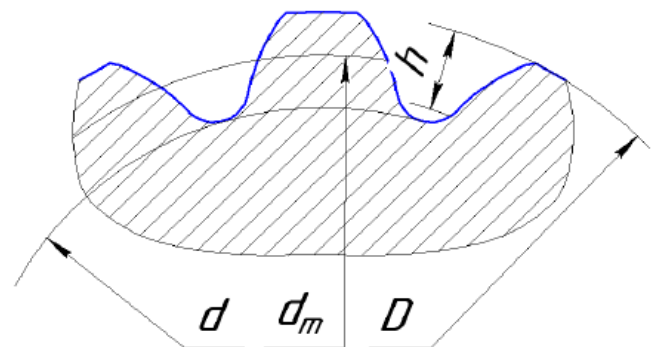


Fig. 2.65

Module m , the number of teeth z and diameters d and D are standardized. They should be selected from the tables according to standards and the major diameters.

When there is relative axial motion in a splined connection, the side pressure on the splines should be limited to about 7 MPa.

2.4. Rivet Joints

Rivets are considered to be permanent fasteners. Riveted joints are therefore similar to welded and adhesive joints. When considering the strength of riveted joints similar calculations are used as for bolted joints.

Rivets have been used in many large scale applications including aviation, ship-building, boilers, pressure vessels, bridges and buildings etc. In recent years there has been a progressive move from riveted joints to welded, bonded and even bolted joints. But it is the most common method used to fasten or join aluminum alloys in aircraft construction and repair. In the wide-body aircraft about 75% of all connections are made with rivets (Fig. 2.66). For example, aircraft IL-86 has up to 1.5 million rivets.

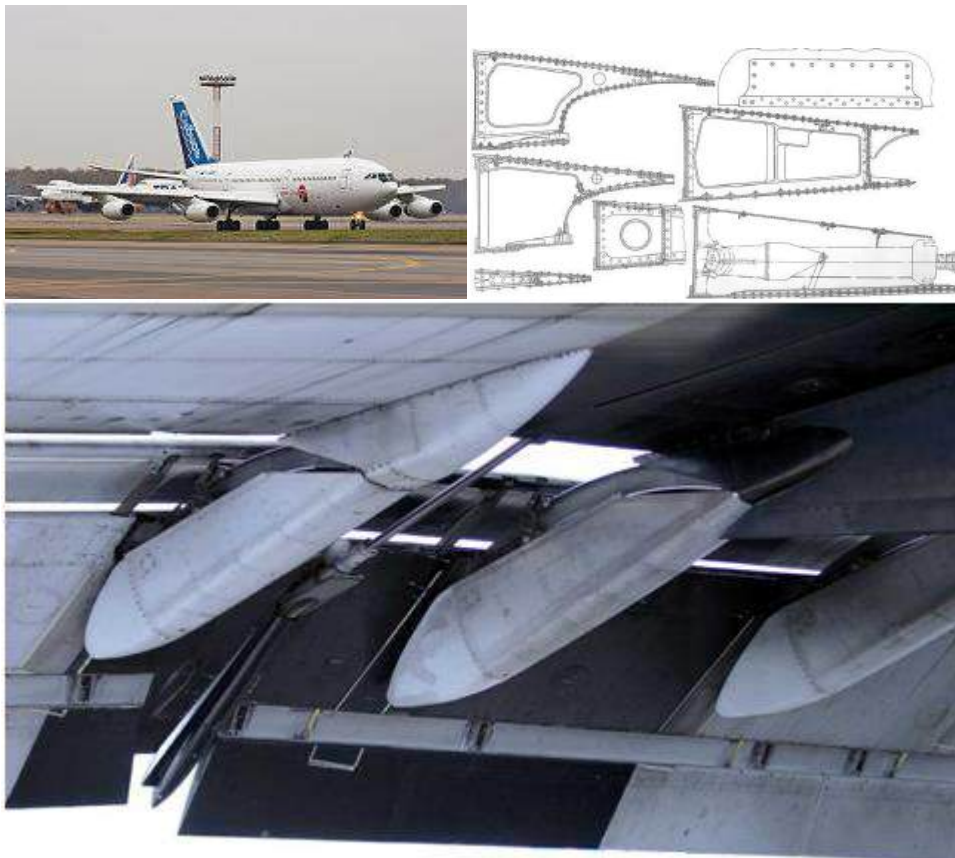


Fig. 2.66

There are strict standards and codes for riveted joints used for aviation and structural/pressure vessels engineering but the standards are less rigorous for using riveted joints in general mechanical engineering.

A rivet is a cylindrical body called a shank with a head (Fig. 2.67).

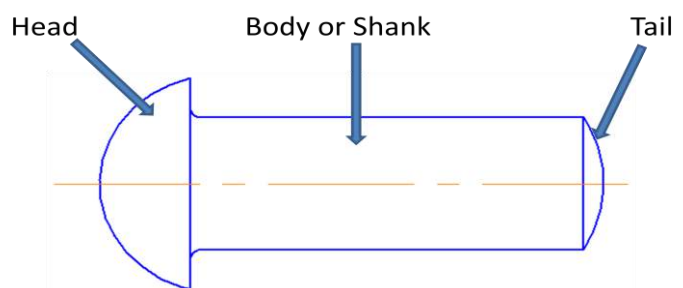


Fig. 2.67

A rivet is inserted into a punched or drilled hole, passing through two clamped plates to be attached (Fig. 2.68). The factory head is supported by backup die whilst a shop head is upset, or bucked (i.e., formed) on the other end of the shank using a forming die, a hammer or a special shaped tool. The plates are thus permanently attached.

The diametrical clearance provided for hot rivets is about 1,5 mm max. For cold rivets very tight fits are often provided by using reamed holes.

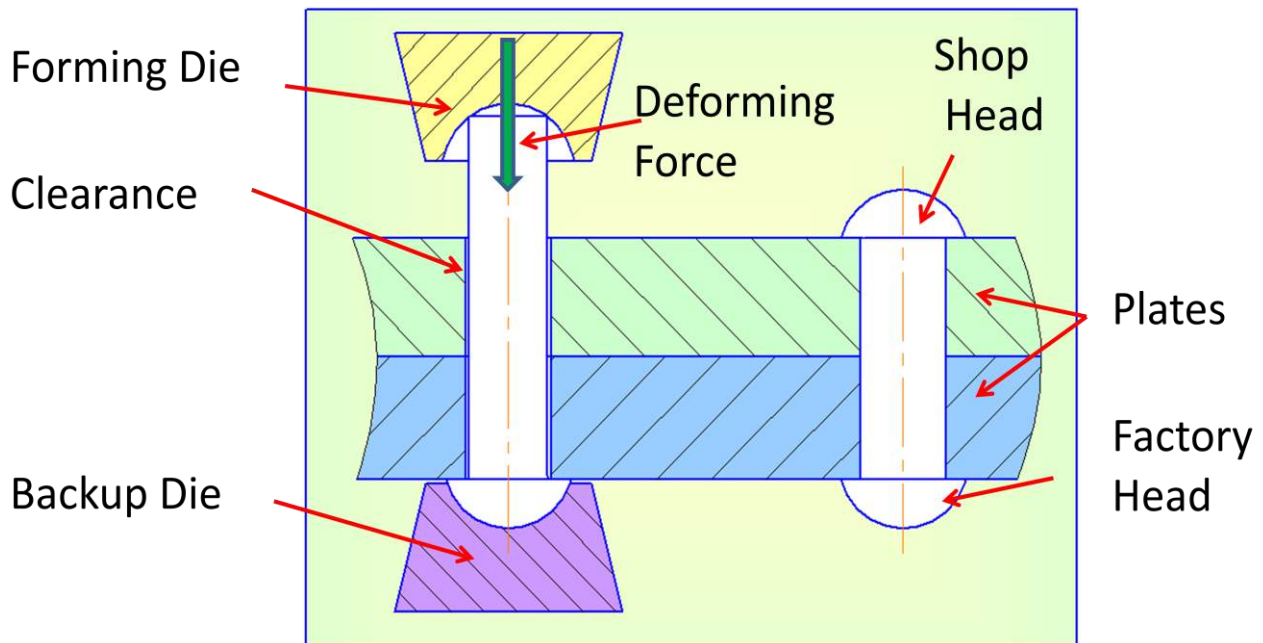
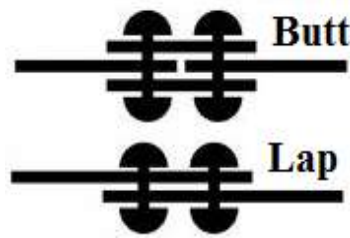


Fig. 2.68

Advantages	Disadvantages
• Stability of the joint quality • Ability to connect elements, which is unacceptable heating (heat-treated, precision) • Satisfactory operation under varying loads and vibration • Ability to connect elements (materials) that are difficult to weld, and heterogeneous, including metals and non-metals	• Compared to the welded joints of metal loss greater than 15...20% • Process inefficient and quite noisy • Increased cost of production

2.4.1. Classification

Riveted joints may be classified as follows:

By purpose • Tight (hermetic) • Strong • Tight -strong	
By deforming temperature • Hot (a rivet is heated before bucking. When a hot rivet cools it contracts imposing a compressive stress on the plates) • Cold (without heating - cold rivets can be used for smaller sizes) • Mixed (only tail end is heated)	
By sheets relative position • Butt (single strap; double strap) • Lap	

2.4.2. Types

A lot of rivet modifications is known.

Round (button) head rivets (Fig. 2.69, a) are used in the interior of the aircraft, except where clearance is required for adjacent members. The roundhead rivet has a deep, rounded top surface. The head is large enough to strengthen the sheet around the hole and, at the same time, offer resistance to tension.

Flat head rivet (Fig. 2.69, b), like the roundhead rivet, is used on interior structures. It is used where maximum strength is needed and where there isn't sufficient clearance to use a roundhead rivet. It is seldom, if ever, used on external surfaces.

Brazier head rivet (Fig. 2.69, c) has a head of large diameter, which makes it particularly adaptable for riveting thin sheet stock (skin). The brazier head rivet offers only slight resistance to the airflow, and because of this factor, it is frequently used for riveting skin on exterior surfaces, especially on aft sections of the fuselage and empennage. It is used for riveting thin sheets exposed to the slipstream. A modified brazier head rivet is also manufactured; it is simply a brazier head of reduced diameter.

Universal head rivet (Fig. 2.69, *d*) is a combination of the roundhead, flathead, and brazier head. It is used in aircraft construction and repair in both interior and exterior locations. When replacement is necessary for protruding head rivets - roundhead, flathead, or brazier head - they can be replaced by universal head rivets.

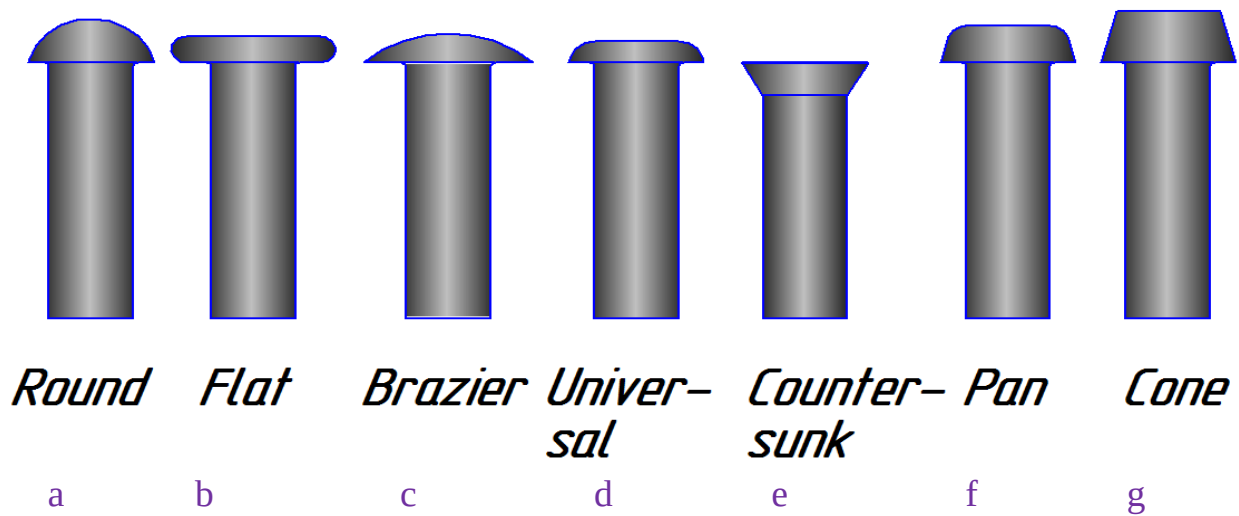


Fig. 2.69

Countersunk head rivet (Fig. 2.69, *e*) is flat topped and beveled toward the shank so that it fits into a countersunk or dimpled hole and is flush with the material's surface. The angle at which the head slopes may vary from 78° to 120° . The 100° rivet is the most commonly used type. These rivets are used to fasten sheets over which other sheets must fit. They are also used on exterior surfaces of the aircraft because they offer only slight resistance to the slipstream and help to minimize turbulent airflow. Shank may be solid or tubular.

2.4.3. Rivet materials

Rivets for mechanical and structural applications are normally made from ductile (low carbon) steel. For applications where weight, corrosion, or material constraints apply, rivets can be made from copper (+alloys) aluminum (+alloys), monel etc. The material used for the majority of aircraft solid shank rivets is aluminum alloy.

Mild steel rivets are used for riveting steel parts. The corrosion resistant steel rivets are for riveting corrosion resistant steels in firewalls, exhaust stack brackets, and similar structures.

Monel rivets are used for riveting nickel-steel alloys. They can be substituted for those made of corrosion resistant steel in some cases.

The use of copper rivets in aircraft repair is limited. Copper rivets can be used only on copper alloys or nonmetallic materials such as leather. If a copper rivet is inserted into an aluminum alloy structure, two dissimilar metals are brought in contact with each other. Remember, all metals possess a small electrical potential. Dissimilar metals in contact with each other in the presence of moisture cause an electrical current

to flow between them and chemical byproducts to be formed. Principally, this results in the deterioration of one of the metals.

Metal temper is an important factor in the riveting process, especially with aluminum alloy rivets. Aluminum alloy rivets have the same heat-treating characteristics as aluminum alloy stock. They can be hardened and annealed in the same manner as aluminum. The rivet must be soft, or comparatively soft, before a good head can be formed.

2.4.4. Rivet Joint Failure

A rivet joint may fail due to:

- shearing through one section of the rivet (single shear);
- shearing through two sections of the rivet (double shear);
- compressive bearing failure of the rivet;
- shearing of the plate(s) being joined;
- bearing failure of the plate(s) being joined;
- tearing of the plates between the rivets.

2.4.5. Strength Calculation

The plate loads are withstood by the rivets. In practice the loads are generally withstood by friction between the plates under the compressive force of the contracted rivets. The calculations provided below are simplified but provide relatively conservative joint strength value.

In designing of rivet joints, it is convenient to simplify the process by making the following assumptions:

- a rivets fail in either pure compression or pure shear;
- the shear stress is evenly distributed across the rivet section;
- the bearing stress is evenly distributed across the projected area of the rivet;
- the tensile stress is uniform across the plate area between the rivets.

The rivet shear calculation is according to the limit force: $F_0 = \frac{\pi d^2}{4} [\tau]_{sh} \leq F_{lim}$, where F_0 is a force acting on single rivet (Fig. 2.70), F_{lim} is found in handbook tables.

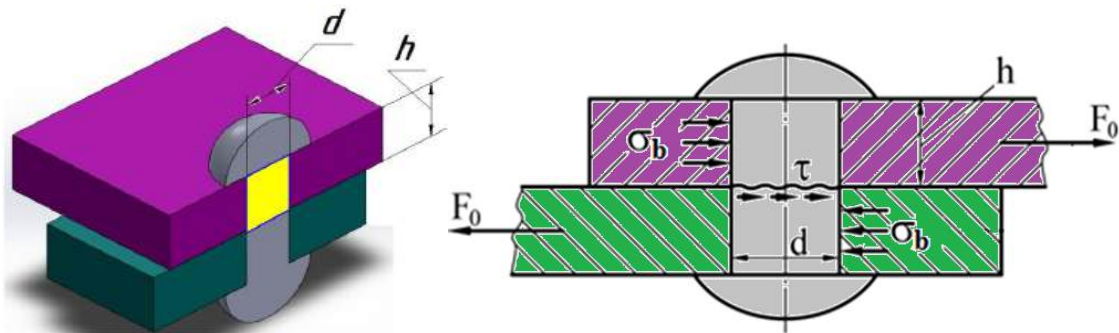


Fig. 2.70

Bearing strength condition is $\sigma_b = \frac{F}{dh} \leq [\sigma_b]$, where allowable stress is $[\sigma]_b \approx (0,8...1)\sigma_y$. The rivet tensile stress is approximately equal to the yield stress of the rivet material.

Rivets are initially sized with nominal diameters $(1,2...1,4)h$. A rivet has the same strength on shear and bearing if $d \approx 2,5h$.

The selection of the number of rivets used for a joint and the array is simply to ensure the maximum strength of the rivets and the plates.

In cross-section $a - a$

$$\sigma_t = \frac{F_0}{A} = \frac{F_0}{h(t-d)} \leq [\sigma_t],$$

in cross-sections $m - m$ and $n - n$

$$\tau = \frac{F_0}{2hl_{m-m}} \leq [\tau]_{sh},$$

$$[\tau]_{sh} \approx (0,5...0,6)\sigma_y.$$

A rivet and a sheet are equal in strength if pitch $t \approx (3...6)d$ (Fig. 2.71) and the distance between rows is also greater than $3d$.

It is important that the rivets are not positioned too close to the side of the plate or the edge of a plate. Distances $m-a$ and $m-m$ should be greater than $1,5d$.

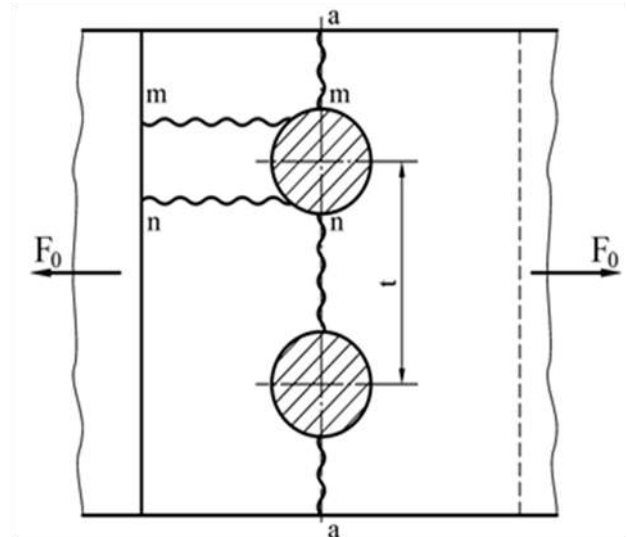


Fig. 2.71

2.4.6. Load Distribution into Flank Seam

Flank seam is parallel to the direction of loading. In a stressed state the load is stepwise transmitted from one plate to another and the acting forces are not identical in different zones, common for both strips (Fig. 2.72).

In the cross-section at a rivet is placed (Fig. 2.72, b), the strip **I** will be loaded by force F_i^I , and the strip **II** by the force F_i^{II} , with the equation $F_i^I + F_i^{II} = F$ true for each cross-section.

Under the load described above, changes in length for the strips **I** and **II** (Δ_1 and Δ_3 , Fig. 2.72, a) will be proportional to the strips' stiffness. The greater the difference in elongation of the corresponding segments of strips **I** and **II**, the greater maximum stresses in the weld seam at its start and end. In the middle of the seam, this difference will be minimal and the middle rivets are underloaded.

It is investigated that with six rivets in a seam the end rivets are overloaded by 26 % relative to the mean load. That is why it is recommended the limit number of rivets in seam no more six.

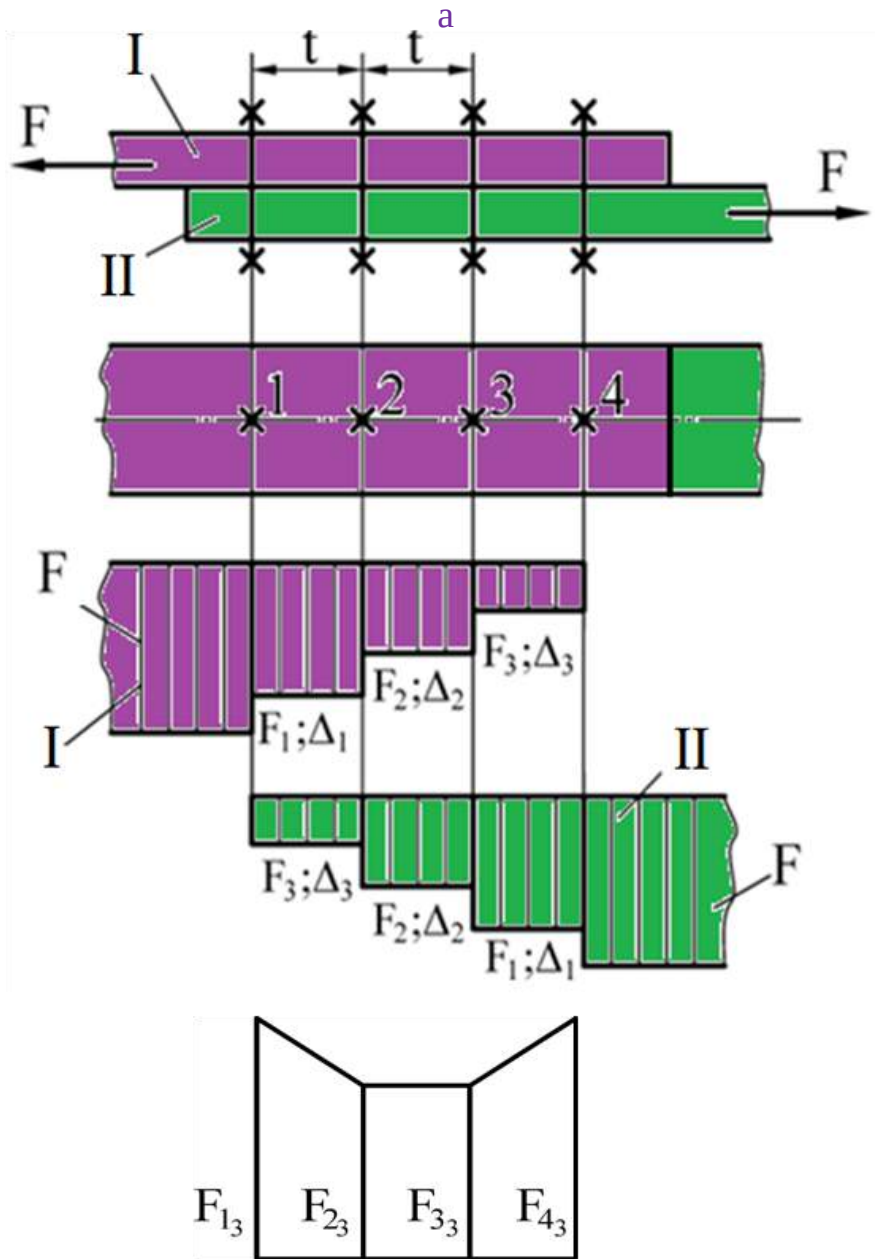
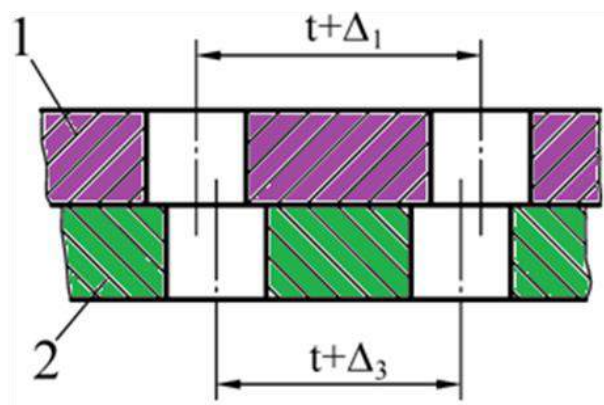


Fig. 2.72

To uniform force distribution along seam, plates with different stiffness may be used (Fig. 2.73).

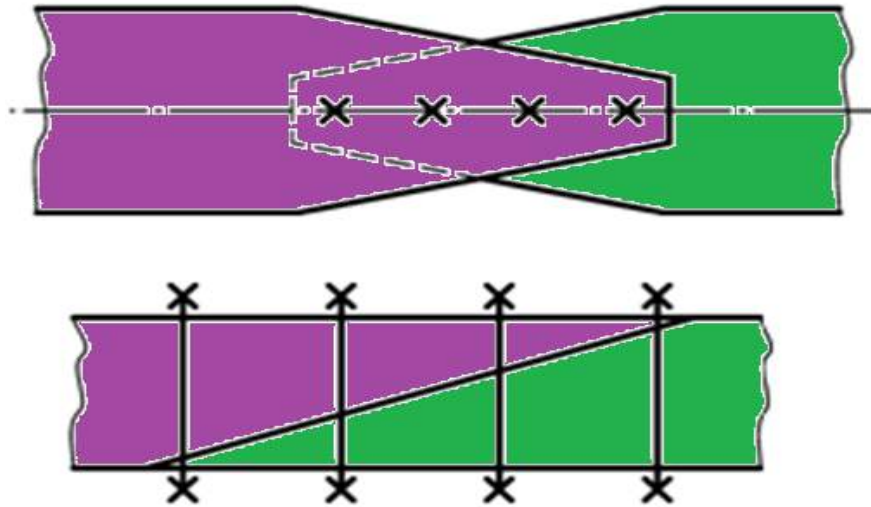


Fig. 2.73

In such case, the bigger force acts on portion with big stiffness and vice versa. Thus deformations caused by these forces will be approximately equal.

2.4.7. Design Rules

Usually rules as follows should be taken into account:

1. The hole misalignment considerably weakens rivets, that is why holes into the plates should be machined together. In important cases the holes should be reamed and rivets should be set with tension.
2. Avoid close places. Riveting tool must be free to come to the installation rivets. The minimum distance from the first rivet to the wall should be more than $2d$.
3. With a large number of rivets is desirable to put them in a checkerboard pattern.

2.5. Welded Joints

It is a permanent joint formed by the melting of the parts. This is the main form of permanent joints in engineering.

Welding is used not only for joining, but also to make themselves parts.

During the welding heat developed by any source (electric arc, gas, friction etc) melts the surfaces of the joined parts to a depth 2...5 mm (depth of penetration). As the result a molten pool is formed between them and an alloy connects the parts after cooling (Fig. 2.74).

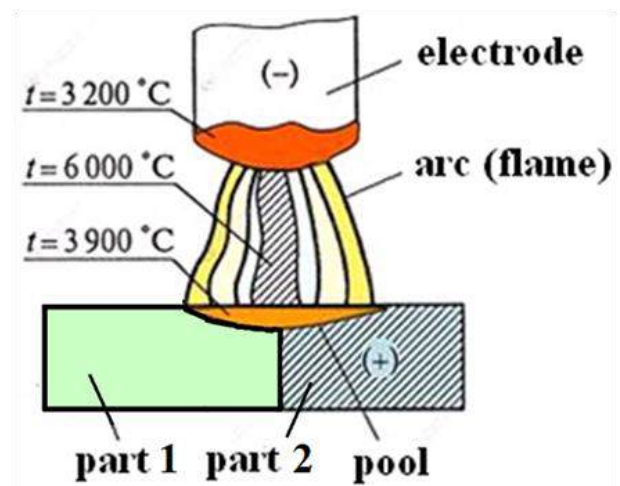


Fig. 2.74

Advantages

- Metal saving up to 20% compared to the cast and riveted joints
- High efficiency of the process
- Low cost of the equipment
- Noiseless process
- Improved design forms
- Less allowance for processing

Disadvantages

- The instability of weld quality
- Low vibration strength
- Not all materials are able to weld
- Not all materials and components allow heating

2.5.1. Classification

1. By source of heat: thermal (arc, electroslog), thermo-mechanical (contact, diffusion), mechanical (cold, frictional).
2. By melting type (Fig. 2.75): electric (manual, semi-automatic, automatic, submerged arc), gas, spot, rolling.
3. By purpose: strong, tight-strong.
4. By place: lap (Fig. 2.76), butt (Fig. 2.77, Fig. 2.78).
5. By cross-section (Fig. 2.79): isosceles triangle, triangle, curvilinear triangles.

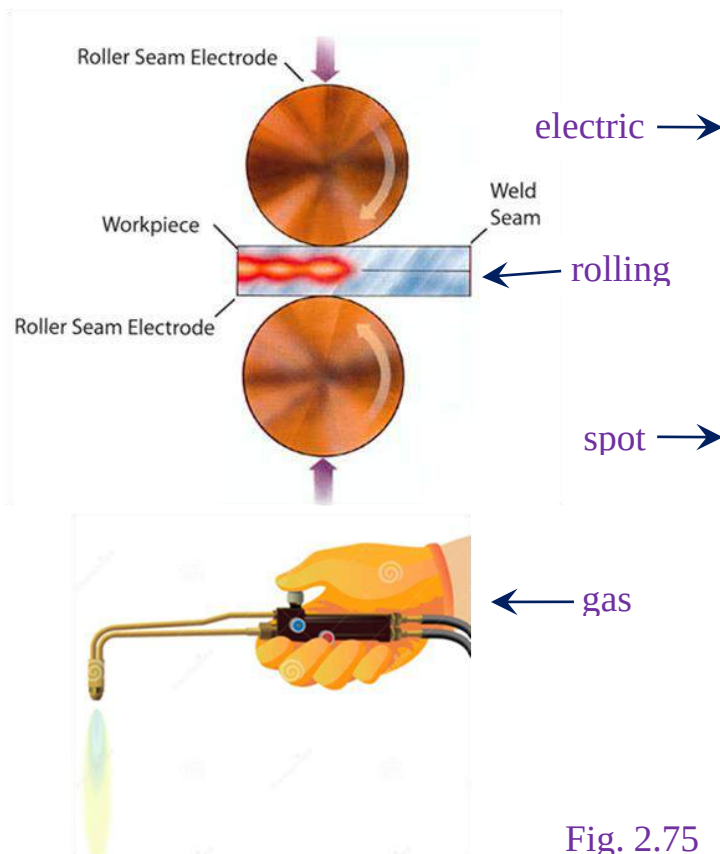
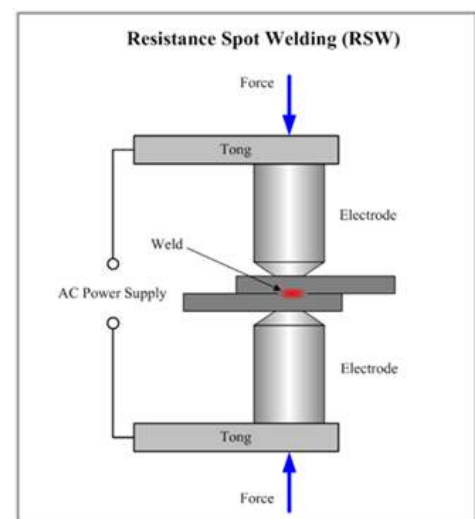
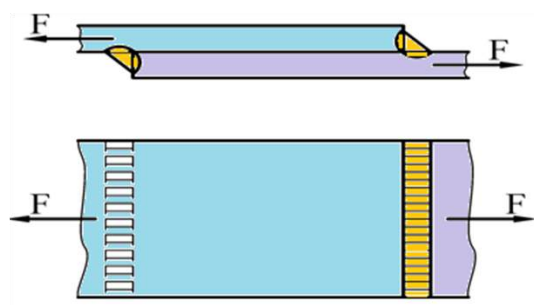
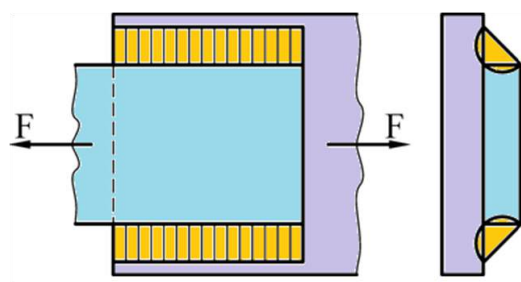


Fig. 2.75

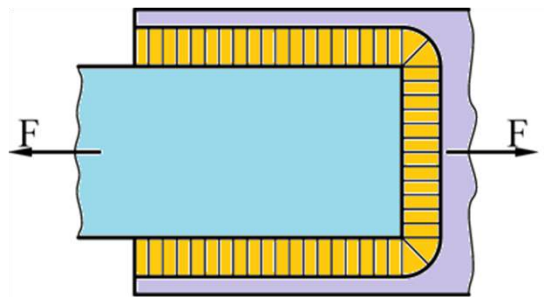




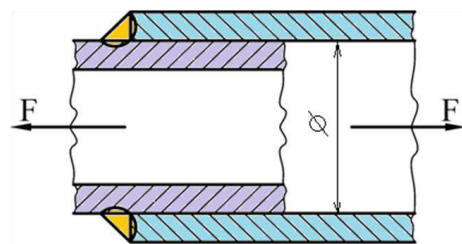
End seam



Flank seam

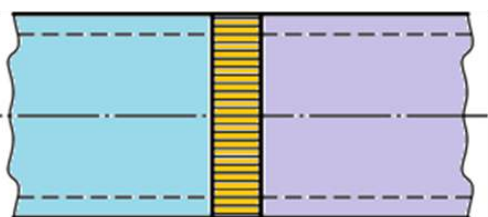
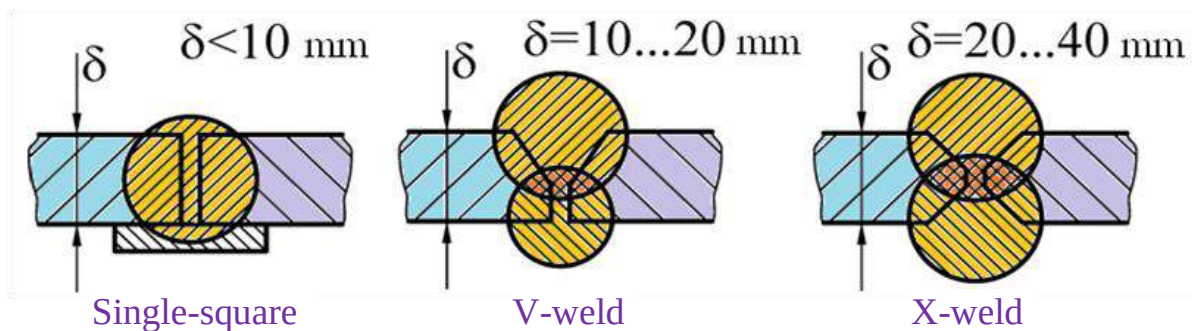


Compound seam



Ring seam

Fig. 2.76



Round

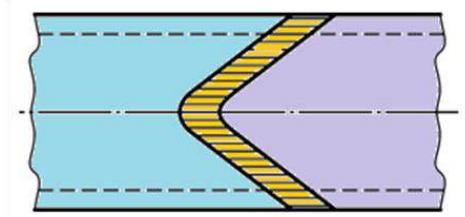
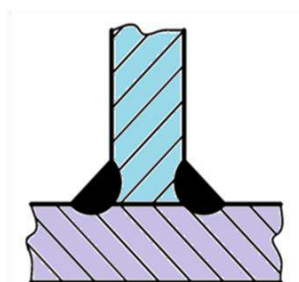
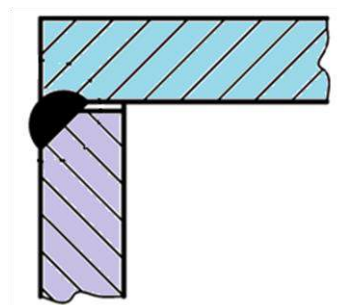


Fig. 2.77

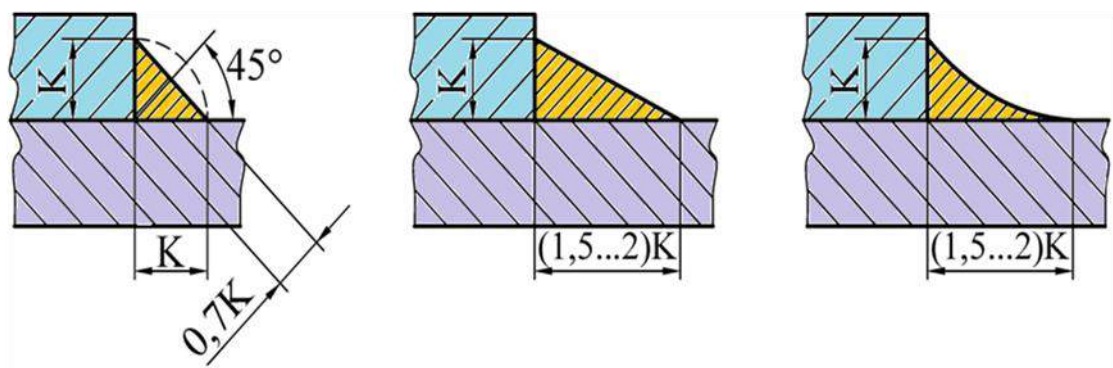


Tee



Corner

Fig. 2.78



Normal – isosceles triangle

Improved – triangle

Improved – curvilinear triangle

Fig. 2.79

2.5.2. Strength Calculation

For alloyed and high-strength steels weld are calculated on failure loads:

$$F = \varphi \sigma_{UTS} A \leq F_{lim}, \quad \varphi = 0,8.$$

For low-carbon steels – at reduced allowable stress calculated relative yield stress:

$$\sigma' = (0.87 \dots 1)[\sigma],$$

$$\tau' = (0.5 \dots 0.7)[\sigma],$$

$$[\sigma] = \frac{\sigma_y}{(1.4 \dots 1.8)}.$$

Allowable stress depends on the type of electrode and allowable stress for the base material.

In a case of *butt weld seam* the stresses in the heat affected zone is calculated as for a solid material. For example, for seam shown in Fig. 2.80 which is loaded by force and bending moment, stresses are calculated as

$$\sigma = \frac{F}{hb} + \frac{6M}{bh^2} \leq [\sigma'] .$$

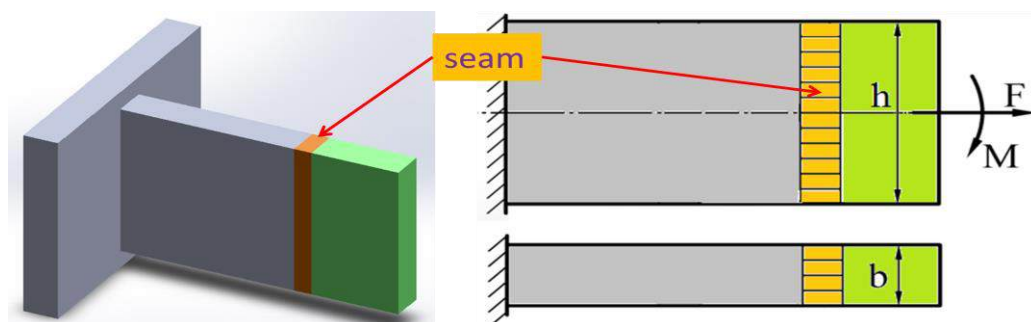


Fig. 2.80

The main failure mode in case of a *flank seam* is a cut on the right angle bisecting the site cross-section of the joint (Fig. 2.81).

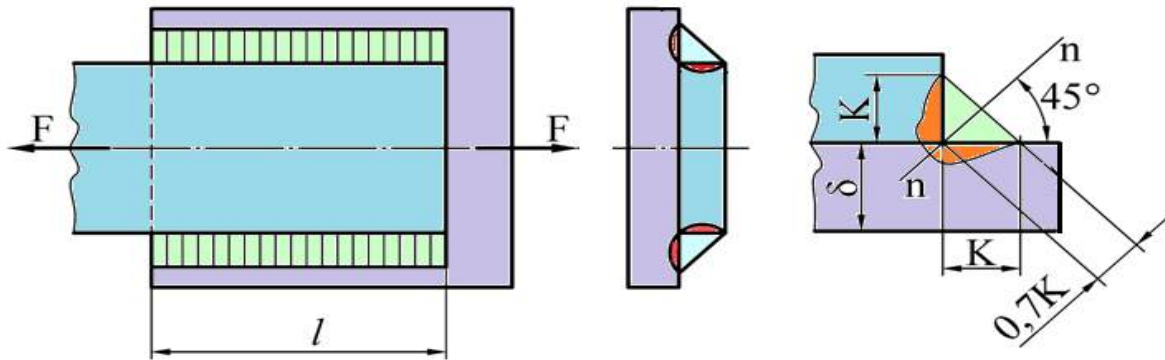


Fig. 2.81

For calculation purposes, we assume that stresses are identical in any section of the fillet, that is

$$\tau = \frac{F}{0,7Kl_s} \leq 0,8\tau_{UTS}, \tau_{UTS} = 0,6\sigma_{UTS}$$

where K is the fillet leg size, l_s is the design length of the weld seam, in case shown in Fig. 2.81 $l_s = 2l$.

Real stress distribution along the weld seam is significantly different from the one assumed.

Let's review the elements of the weld fillet in a stressed state. Since the load is gradually transmitted from one welded component (strip) to another, the action forces are not identical in different zones, common for both strips (Fig. 2.82).

In the cross-section at the distance x from the weld start (Fig. 2.82, a), the strip *I* will be loaded by force F_x^I , and the strip *II* by the force F_x^{II} , with the equation $F_x^I + F_x^{II} = F$ true for each cross-section.

Let us consider a dx segment of each welded strip. Under the load described above, it will extend and get to the place a_1b_1 for the strip *I* and a_2b_2 for the strip *II* (Fig. 2.82, b). Thus, changes in length (Δ_1 and Δ_2) of the segment ab for the strips *I* and *II* will be

$$\Delta_1 = \frac{F_x^I dx}{EA_1}, \quad \Delta_2 = \frac{F_x^{II} dx}{EA_2},$$

where A_1 and A_2 are the cross-sectional areas of the strips *I* and *II*.

The difference in elongation of the corresponding segments of the strips *I* and *II* will be

$$b_1'b_2 = \Delta_1 - \Delta_2.$$

The greater the difference in elongation of the corresponding segments of strips *I* and *II*, the greater maximum stresses in the weld seam at its start and end. In the middle of the seam, this difference will be minimal (Fig. 2.83). So will be the stresses. That is why a flank seam length should be less than $l_{max} = 60K$.

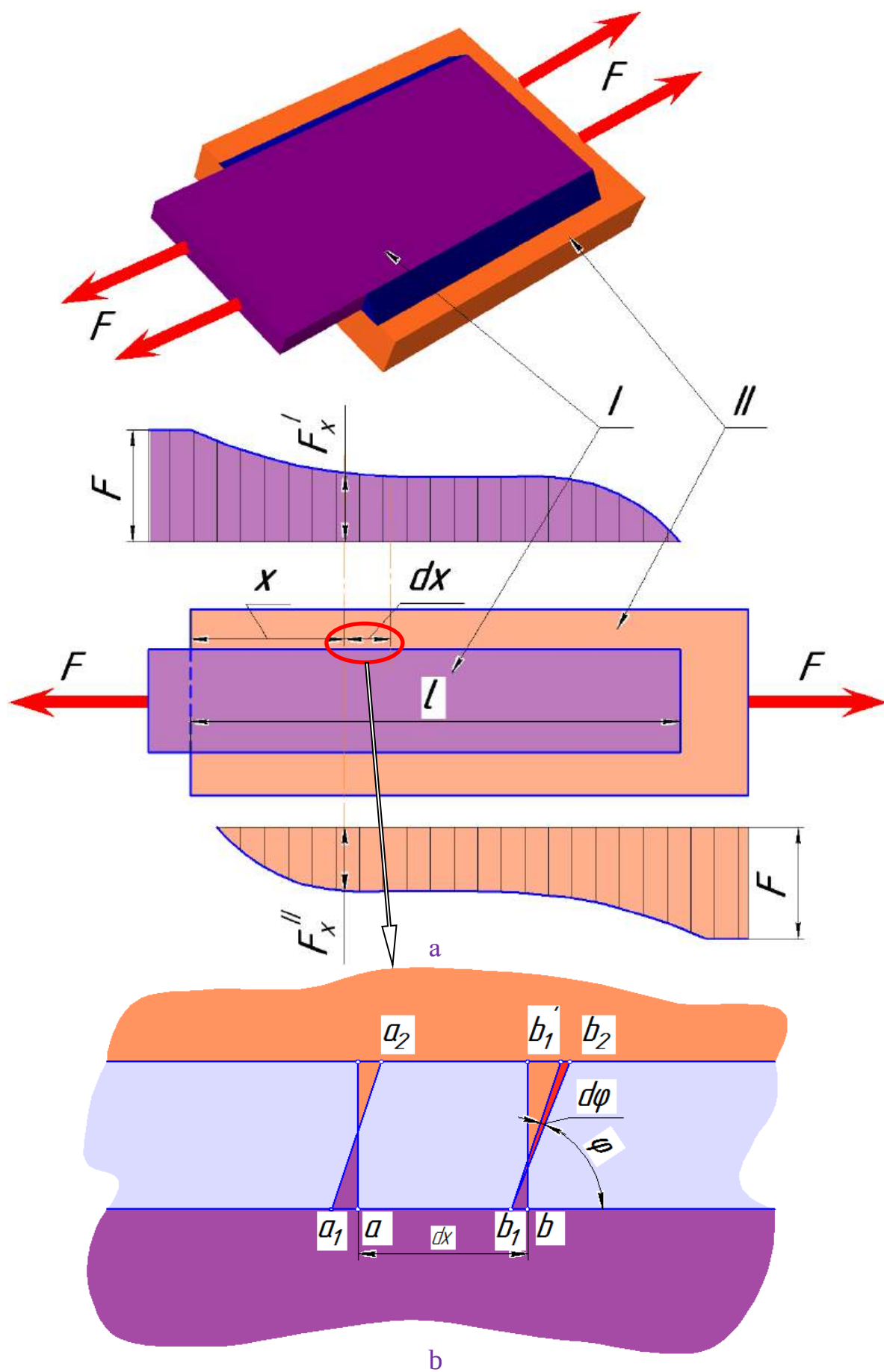


Fig. 2.82

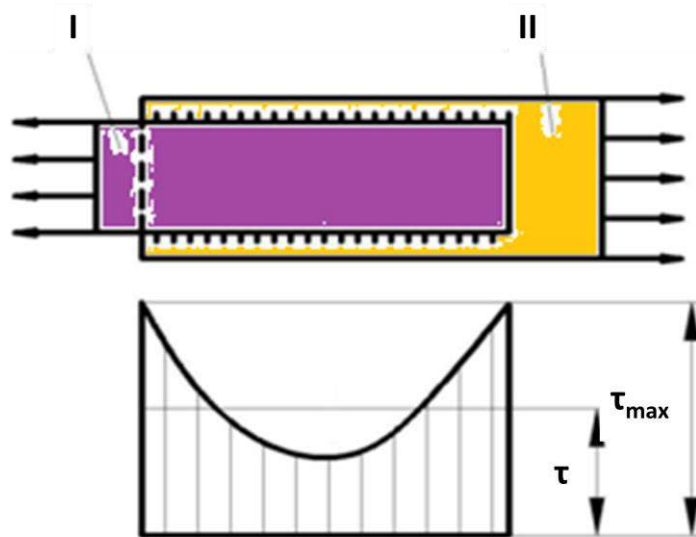
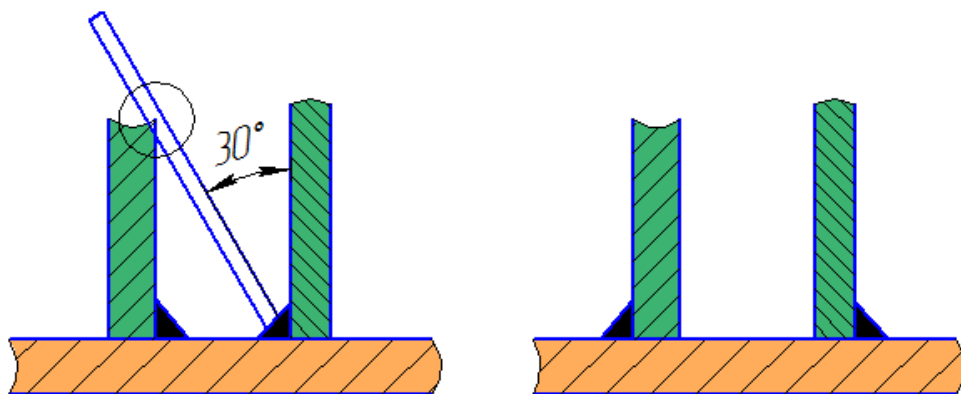


Fig. 2.83

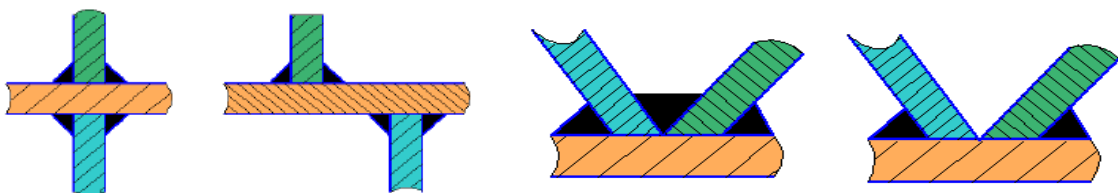
2.5.3. Design Rules

Usually rules as follows should be taken into account:

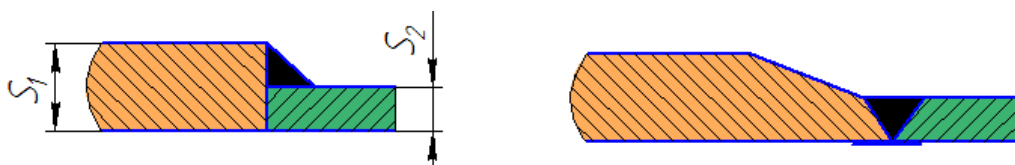
1. Provide an electrode convenient access to the welding zone:

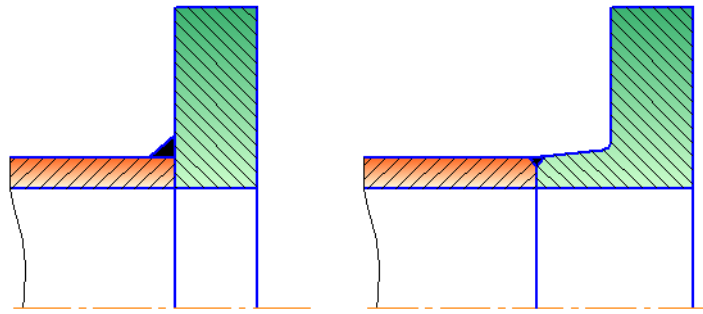


2. Avoid seam superposition:

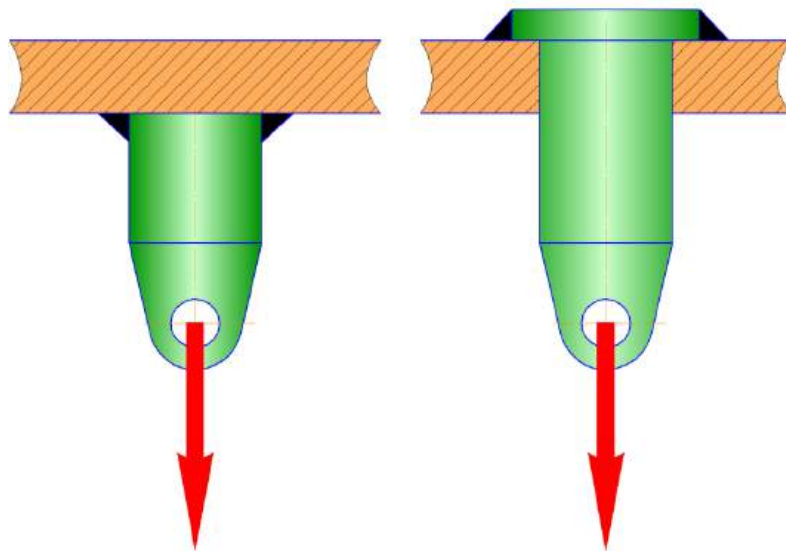


3. Avoid welding of a massive part with thin one ($S_1/S_2 < 3$):

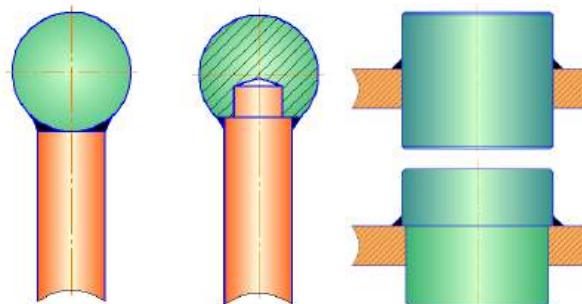




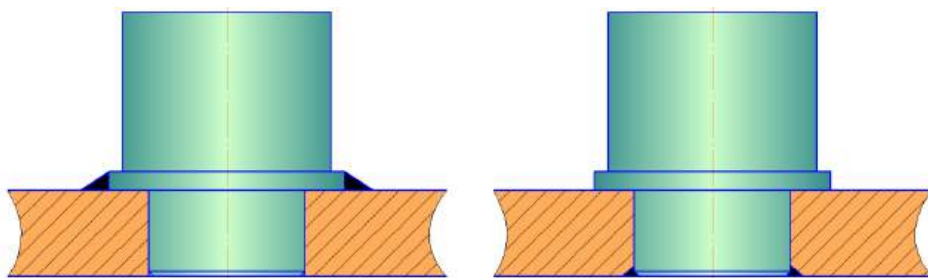
4. Unload welds, apply loads on the material solid portions:



5. Provide mutual fixation of the parts to avoid welding equipment using:



6. Locate the heat-treated surfaces far from the welding zone. Machine surfaces after welding process:



Part 3. MECHANICAL TRANSFERS

General Characteristic

There are two main reasons why mechanical transfers are widely used in practice. First of them is necessity to deliver mechanical power from a suitable location of a power supplier to a place where the power is utilized. In such meaning mechanical transfers are the mechanisms serving for energy transporting on distance usually with transformation of speeds and the rotating moments (torques).

Some transfers, connected sequentially or in parallel, form transmission. In British English, the term transmission refers to the whole drive-train, including clutch, gearbox, prop shaft (for rear-wheel drive), differential, and final drive shafts (Fig. 3.1). In American English, however, the term refers more specifically to the gearbox alone.

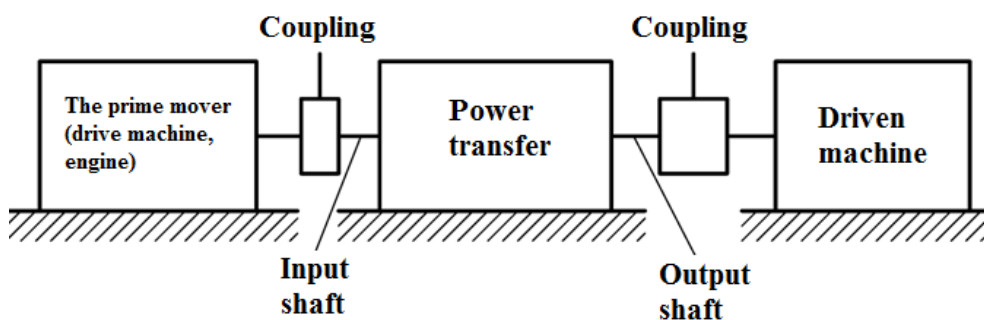


Fig. 3.1

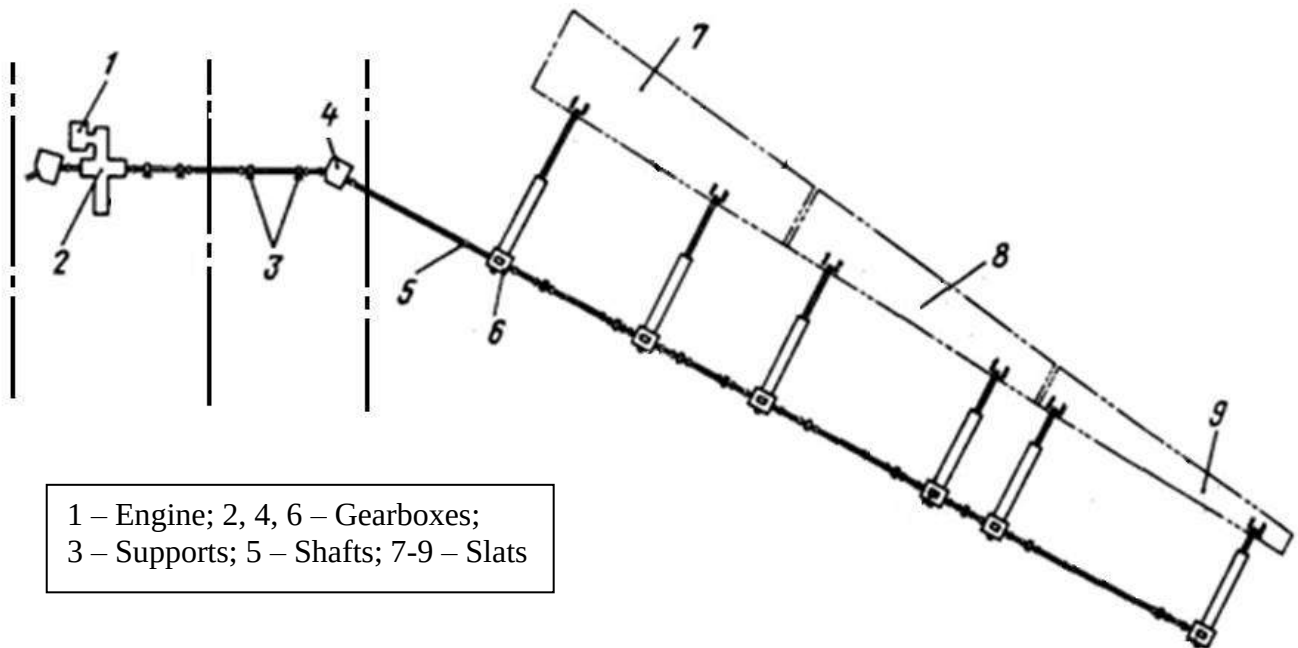
Second application of mechanical transfers is caused by aspiration to reduce the overall dimensions and weight of a drive. Usually, low rotational speed for various operations is needed. In such case, if an engine and a driven machine are connected directly, the engine shaft has to rotate slowly. There is a relationship between a torque and a transferred capacity

$$P = T\omega,$$

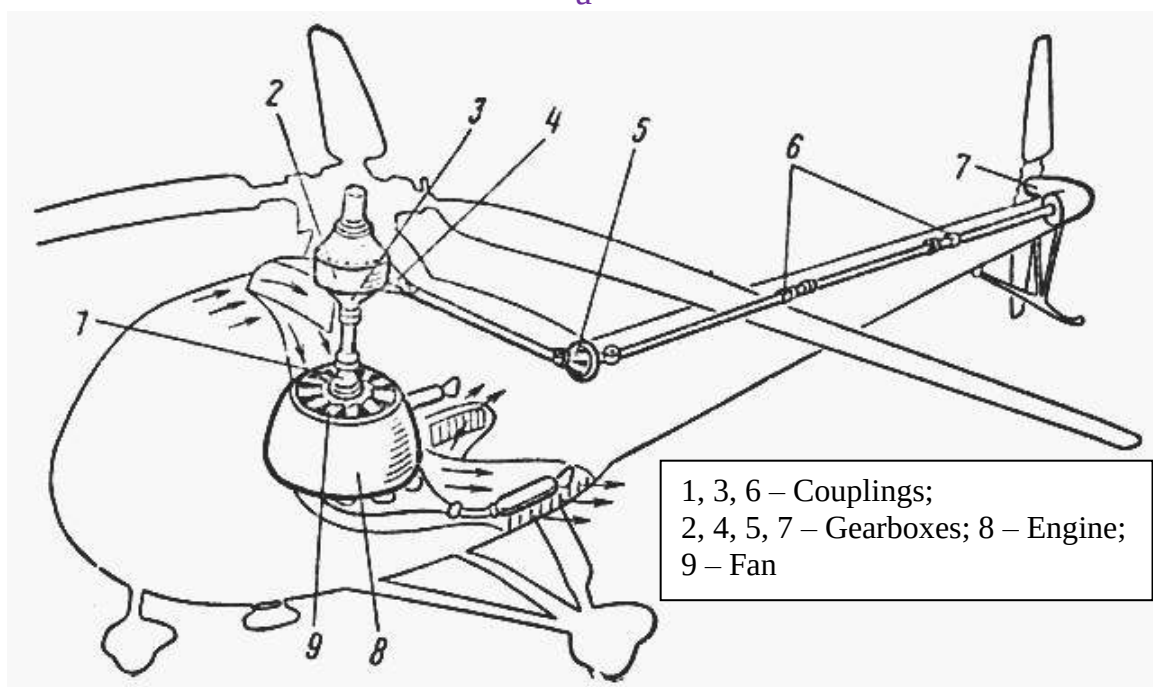
where P is power; T is torque; ω is an angular speed.

Engine overall dimensions are proportional to the torque and in case of low rotational speed and the same transferred power engine will be massive. It is rational to apply high-speed engines and mechanical transfers with the big transfer ratio and efficiency. In this case drive specific parameters on weight and overall dimensions will be minimal.

The typical examples are a slat control system (Fig. 3.2, a) and a wing sweep control system (Fig. 3.2, b).



a



b

Fig. 3.2
82

Classification

By the mode of transmission:

- friction (friction drives, belting);
- meshing (gearing, chain, power screws, toothed belt).

By function:

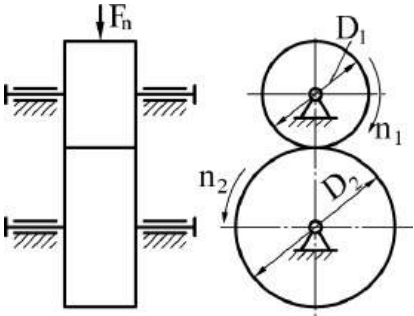
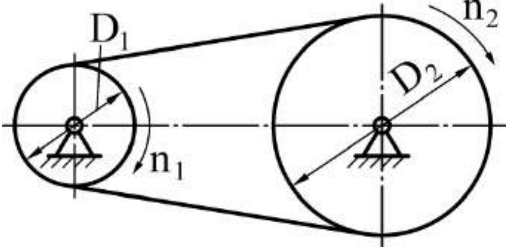
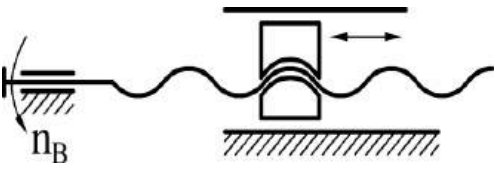
- power (significant loads);
- kinematic (high accuracy).

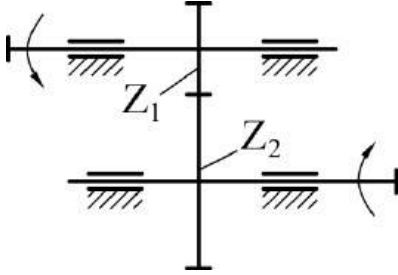
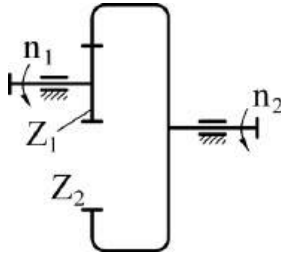
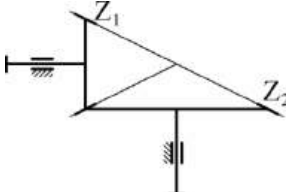
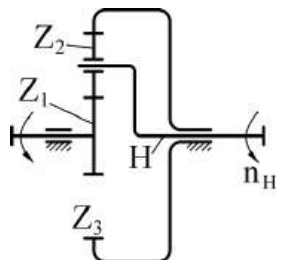
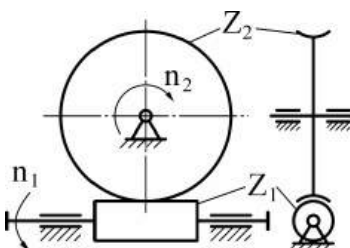
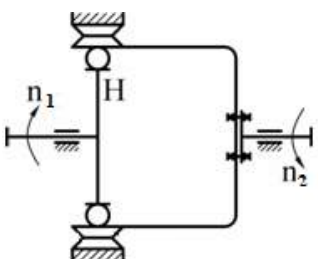
By transfer ratio:

- to reduce velocity;
- to multiply velocity;
- with a variable transfer ratio;
- with a constant transfer ratio.

The Main Types of Mechanical Transfers

They are:

Transfer type	Scheme	Transfer ratio	Efficiency
Friction gears		< 6	$\eta=0.92\dots0.98$
Belt drive		< 6	$\eta=0.9\dots0.96$
Power screws		-	Sliding: $\eta= 0.2\dots0.6$; Ball: $\eta= 0.9\dots0.96$

Transfer type	Scheme	Transfer ratio	Efficiency
External tooth gearing		$\leq 6,3$	$\eta = 0.97 \dots 0.99$
Internal tooth gearing		≤ 6	$\eta = 0.97 \dots 0.99$
Bevel		$\leq 3,5$	$\eta = 0.95 \dots 0.98$
Planetary		$3 \dots 8$	$\eta = 0.95 \dots 0.98$
Worm		$10 \dots 80$	At $Z_1 = 1$ $\eta = 0.7 \dots 0.75$ At $Z_1 = 2$ $\eta = 0.75 \dots 0.82$ At $Z_1 = 4$ $\eta = 0.87 \dots 0.92$
The wave gear		$80 \dots 320$	$\eta = 0.8 \dots 0.94$

The Main Parameters

At designing such parameters are usually known: input P_1 and/or output P_2 power; input T_1 and/or output T_2 torques; input n_1 and/or output n_2 rotational speeds; transfer ratio i .

The basic power and kinematic parameters of the mechanical transfers are:

1. Efficiency: $\eta = \frac{P_2}{P_1}$.

2. Losses: $\psi = \frac{P_1 - P_2}{P_1} = 1 - \eta$.

3. Torque: $T = \frac{P}{\omega} = 9550 \frac{P}{n}$ (here P in kW; n in rpm; T in Nm).

4. Transfer (velocity) ratio: $i = \frac{\omega_1}{\omega_2} = \frac{n_1}{n_2}$.

5. Transfer number (for one stage transfer only!): $u = \frac{Z_2}{Z_1} = \frac{d_2}{d_1}$.

The transfer number characterizes the ratio of the geometrical sizes and is always positive. The transfer ratio may be negative in a case when directions of input and output shafts rotation are different.

At $i > 1$ ($n_1 > n_2$) transfer is called as a reducer, at $i < 1$ ($n_1 < n_2$) it is called as the multiplier. The majority of mechanical transfers are reversible, i.e. can work in a reducer and multiplier mode. However, some types of transfers (worm, planetary, screw, etc.) are known, which transfer movement in one direction only, i.e., are irreversible.

Kinematic and power transfers are distinguished. Kinematic transfers usually transform movement with the required accuracy and do not take considerable loadings. Power transmissions transfer the great torques and capacities. Sometimes increased requirements concerning accuracy of movement are also demanded, for example in robots and machine tools.

There are transfers with constant or the variable (adjustable) transfer ratio. Transfer with regulated transfer ratio is called as a variator (Fig. 3.3). Direct regulation allows optimize technological process and machine works.

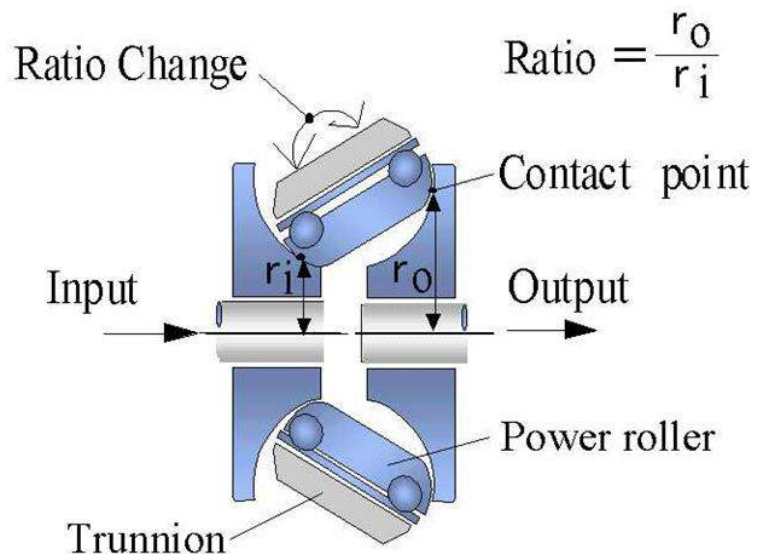
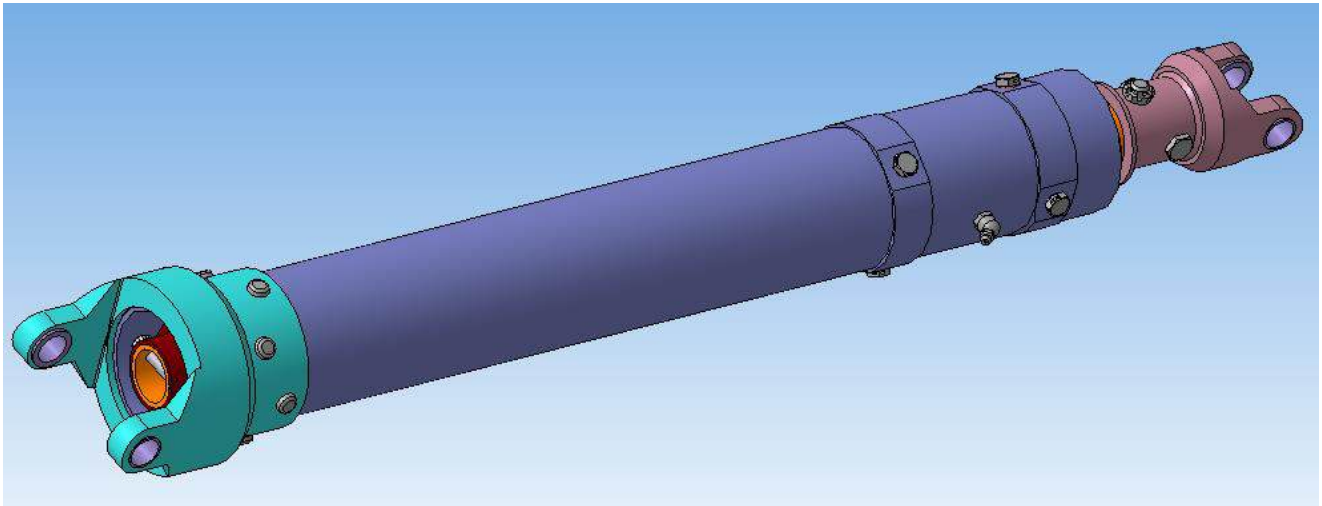


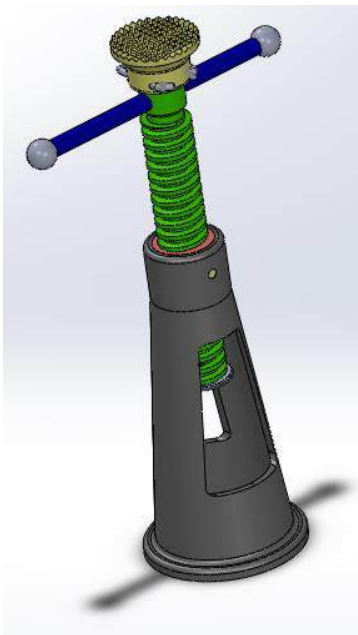
Fig. 3.3

3.1. Power Screws

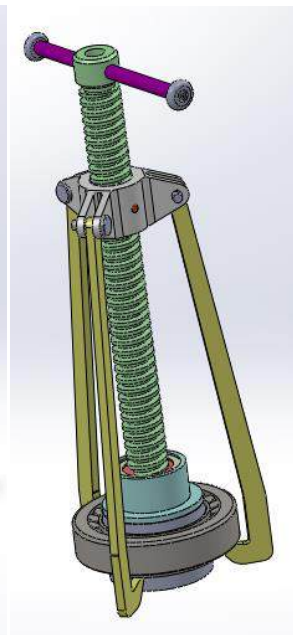
The power screws are employed both to transform rotation motion into linear and to produce large axial forces. Power screws are used in actuators of flight control systems (flaps, slats, stabilizers, drivers, etc, Fig. 3.4, *a*), to lift weights (jacks, Fig. 3.4, *b*), as well as various technological processes: bearing removers (Fig. 3.4, *c*), screw presses (Fig. 3.4, *d*), bending presses, drive screws of machine tools etc.



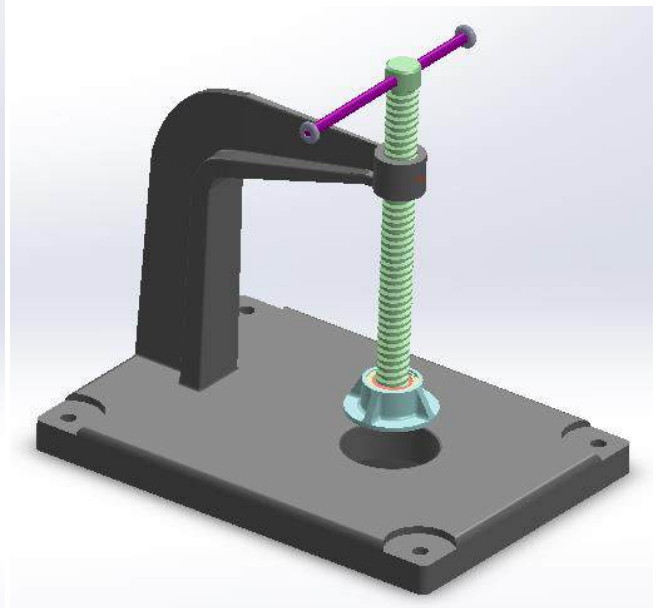
a



b



c



d

Fig. 3.4

Some different combinations are available:

- nut is fixed, screw rotates and moves;
- screw is fixed, nut rotates and moves;

- screw rotates, nut moves;
- nut rotates, screw moves.

Two latest types are used in aviation in high-lift controls. In the first case (Fig. 3.5) transmission consists of screw 3, which is rotated by gearbox 4, and driven nut 2 with frame 1, which moves translationally. Such devices are used to produce significant axial displacements.

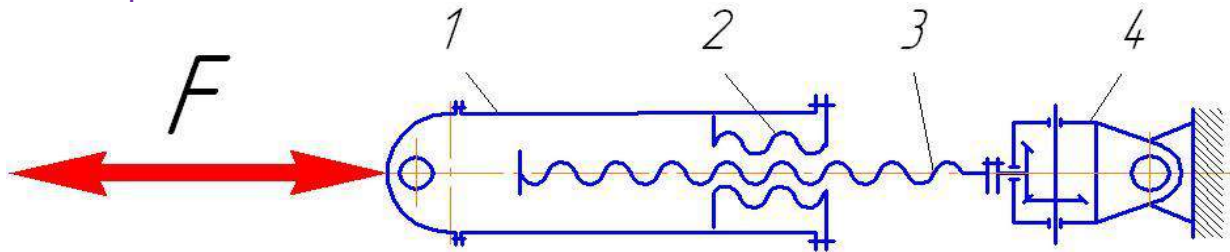


Fig. 3.5

Second variant is shown in Fig. 3.6. Here the transfer consists of fixed in axial direction rotating nut 3 and a movable driven screw 2. Axial force acts through eye 1 directly on screw 2, then on case 4 as well as adjustment points. Nut 3 is unloaded from axial force and rotates screw only. This scheme is realized when small displacements and large axial forces are needed.

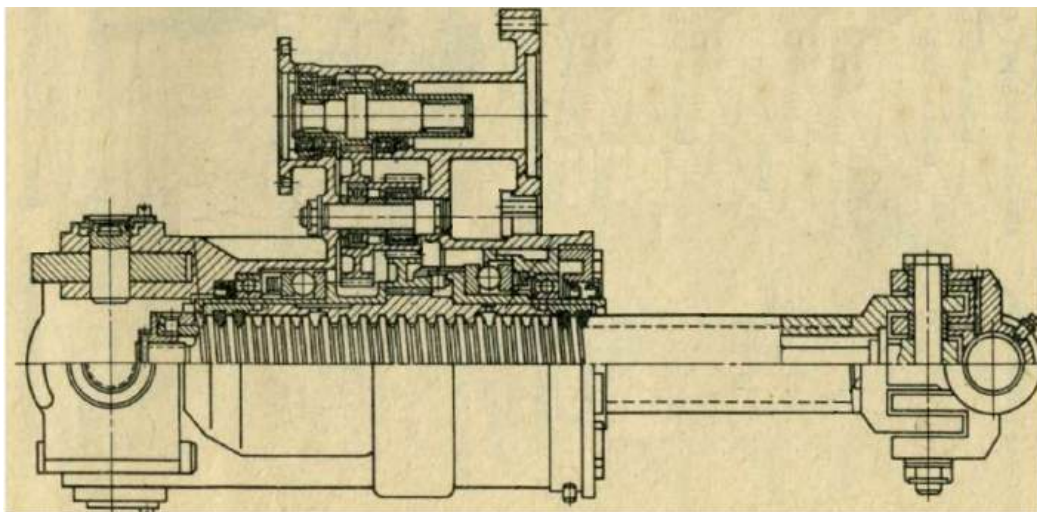
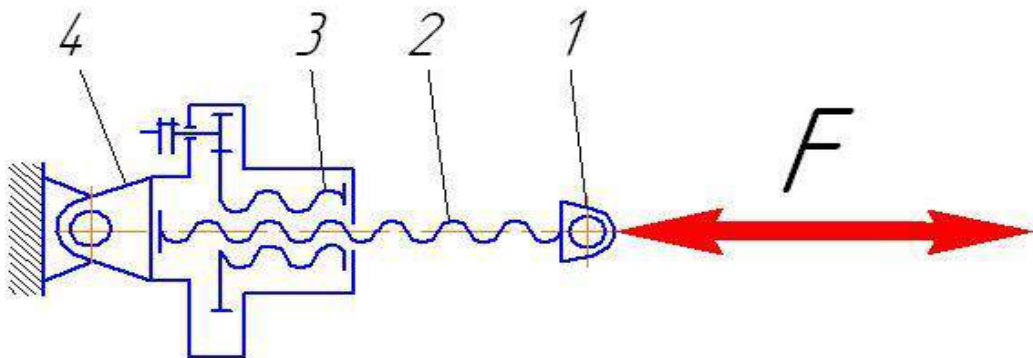


Fig. 3.6

3.1.1. Power Screw Thread Profiles

Trapezoidal GOST 9484-81 (ACME)

The type has symmetrical profile (Fig. 3.7) so it is used when a thread takes a **bilateral** load. American variant ACME has profile angle 29° . It is characterized by low efficiency due to the friction between the screw and nut. Because of the friction, an trapezoidal (ACME) thread screw is not easily back drivable and may be self-locking.

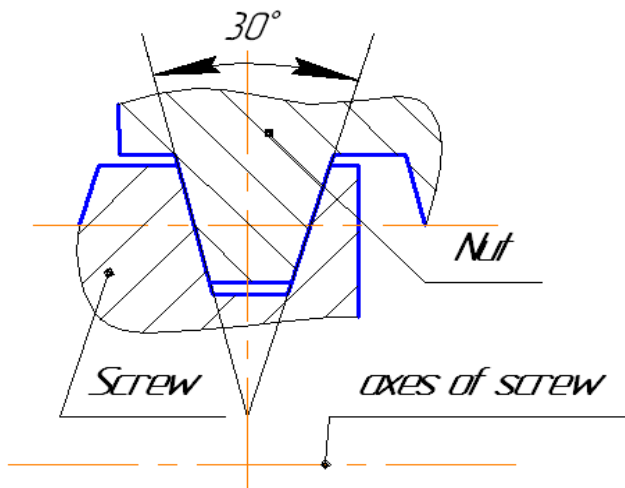


Fig. 3.7

Buttress GOST 10177-82

Here shear and bending strengths are greater than those of a trapezoidal one, it has a lower concentration factor and higher fatigue strength. For these reasons buttress threads (Fig. 3.8) have found wide application in mechanisms with **unilateral** loads.

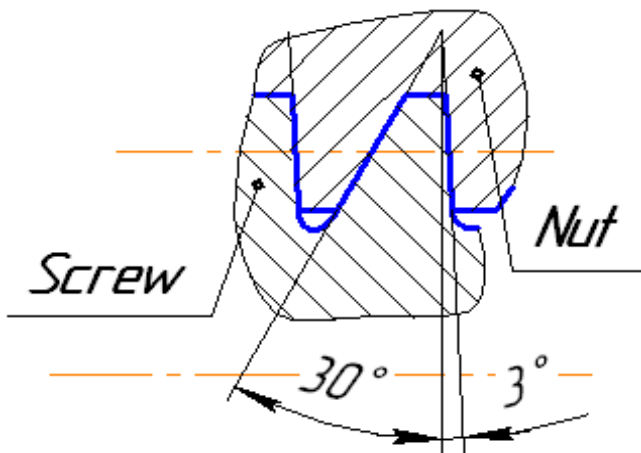


Fig. 3.8

Square

This type (Fig. 3.9) has the highest efficiency but shear and bending strengths are lower than those of a trapezoidal and buttress ones, it cannot be milled and polished, axial clearance cannot be adjusted by tightening a split nut. It is not standardized.

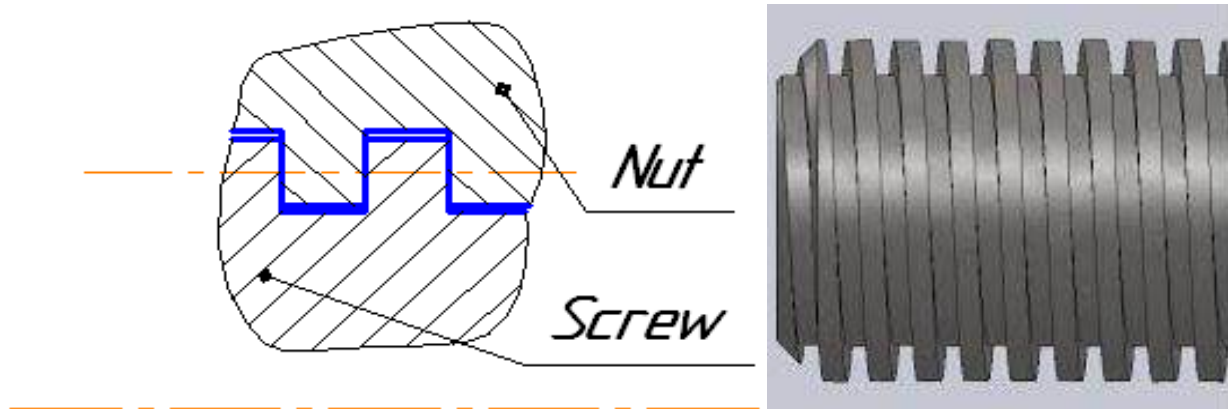


Fig. 3.9

Ball Screw (Ball Transfer)

This transfer (Fig. 3.10) uses rolling contact instead of sliding one. Rolling friction provides a low coefficient of friction and much higher efficiency (up to 90 %). Ball transfers are widely used in high speed applications like aircraft control mechanisms, machine tools, robots and others. Because of the low friction, ball screws are easily back-drivable. Consequently most ball screw actuators contain brake devices to prevent back-drive. It is not standardized.

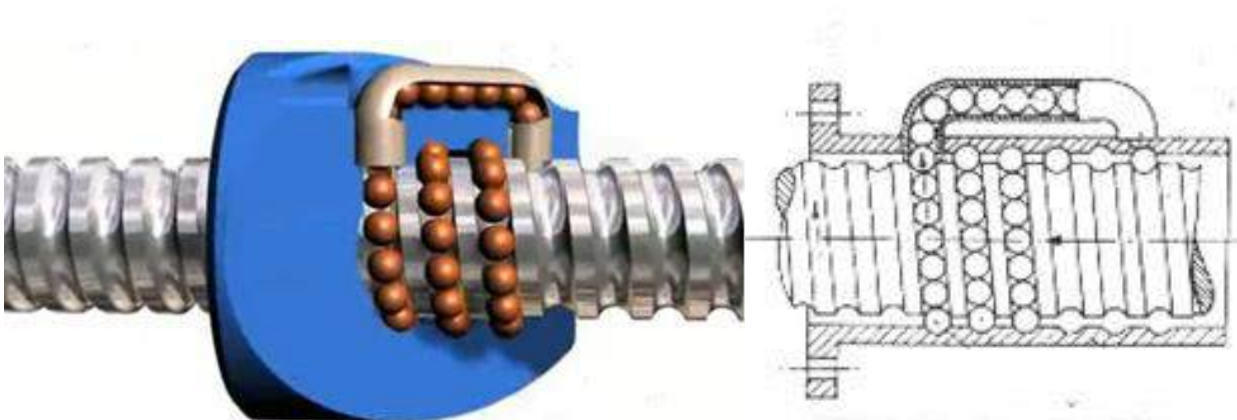


Fig. 3.10

In aerospace applications (like flap, slat and stabilizer actuators), two the most common power screws are the trapezoidal (ACME) thread and the ball screw design.

3.1.2. Sliding Contact Threads

Advantages	Disadvantages
• big transfer and force ratio • move smoothly and accurately • simple design and manufacturing • high reliability • no additional equipment • can be self-locking • standardization	• low efficiency • rapid wear • non-uniform thread loading • low rotation speed

Design Calculation

The dimensions of the thread are obtained according to conditions:

1. Wear resistance limits specific pressure between screw and nut so that it's necessary to provide a lubricant layer between them:

$$p = \frac{F}{A} = \frac{4F}{z\pi(d^2 - d_1^2)} \approx \frac{F}{\pi d_2 H_1 z} \leq [p].$$

Allowable pressure $[p]$ depends on a screw and nut materials (Table 3.1). Carbon and alloyed steels (35, 45, 40X, 40XH, etc.) are usually used for manufacturing of screws. They have good mechanical properties ($\sigma_y \geq 450$ MPa), good machinability and high surface hardness (up to HRC 45 and more) after casehardening.

A nut is usually made of materials which in combination with a steel screw ensure a low friction coefficient and good wear resistance. Such materials include tin and tinless bronze, brass, cermets, and aluminum alloys.

Table 3.1

Materials	Allowable pressure, MPa
Steel to cast iron	5...6
Steel to tin bronze	10...15
Steel to tinless bronze	5...7

Three unknown parameters d_2 , H_1 and z are present here. To obtain pitch diameter we multiply denominator on P/P and d_2/d_2 :

$$p = \frac{F}{\pi d_2 H_1 z} = \frac{F}{\pi d_2 H_1 z \frac{P}{P} \frac{d_2}{d_2}} \leq [p].$$

Let us use two dimensionless factors:

$$\psi_h = \frac{H_1}{P}; \psi_H = \frac{zP}{d_2} = \frac{H_n}{d_2},$$

where ψ_h is standard value equals to 0,75 for buttress thread and 0,5 for trapezoidal one; $H_n = zP$ is a nut height; ψ_H is known from experience $\psi_H = 1,2 \dots 2,5$.

Thus:

$$d_2 = \sqrt{\frac{F}{\pi \psi_h \psi_H [p]}}.$$

2. Tension/compression of screw:

$$\sigma = \frac{F}{A} = \frac{4\beta F}{\pi d_1^2 (1 - \alpha^2)} \leq [\sigma],$$

where allowable stress $[\sigma] = \frac{\sigma_y}{[S]}$.

Safety factor $[S] = 2..5$ depends on required unit reliability; factor β takes into account twisting deformation of the screw by applied torque, $\beta = 1,3$.

Notice that d_1 is not the smallest diameter: it is greater than the internal one of a screw root d_3 . However, even calculation based on d_1 lead to the growth of safety factor (by 20 ... 30 %) because the screw cross-section is not perfectly round (Fig. 3.11).

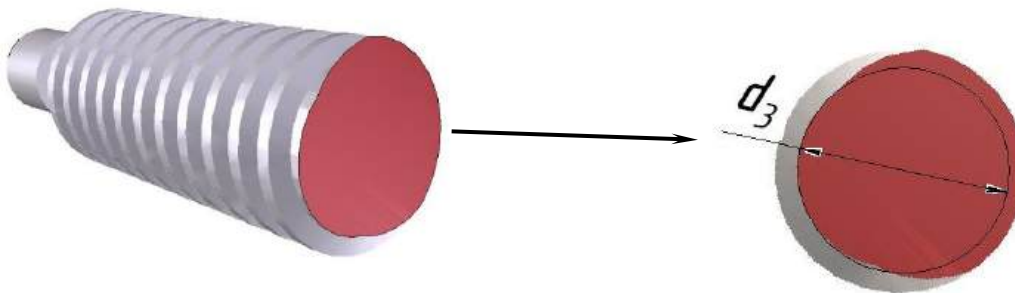


Fig. 3.11

3. Screw buckling strength (longitudinal stability) can be obtained as follows:

$$S = \frac{F_{cr}}{F} \geq [S] \text{ or } \lambda = \frac{\nu l}{i_{min}} = \frac{\nu l}{\theta d_1} \leq [\lambda],$$

where F_{cr} is the critical force (compressive load on screw):

$$F_{cr} = \frac{\pi^2 EJ}{(\nu l)^2},$$

λ is the slenderness ratio, for power screws it approximately equals 100; l is the screw unsupported length; ν is the screw effective length factor, its value depends on type of the screw support.

For a different screw end fixing, values ν are specified in Fig. 3.12. Rolling supports with one bearing are equivalent to a pivot joint, and the nuts fixed in the case are considered to be fixed end; therefore figure "d" ($\nu = 2$) is applicable to a simple screw

jack, figure "c" ($\nu = 1$) – to air-field ones, and figure ($\nu = 0,7$) – to screws of removers and presses.

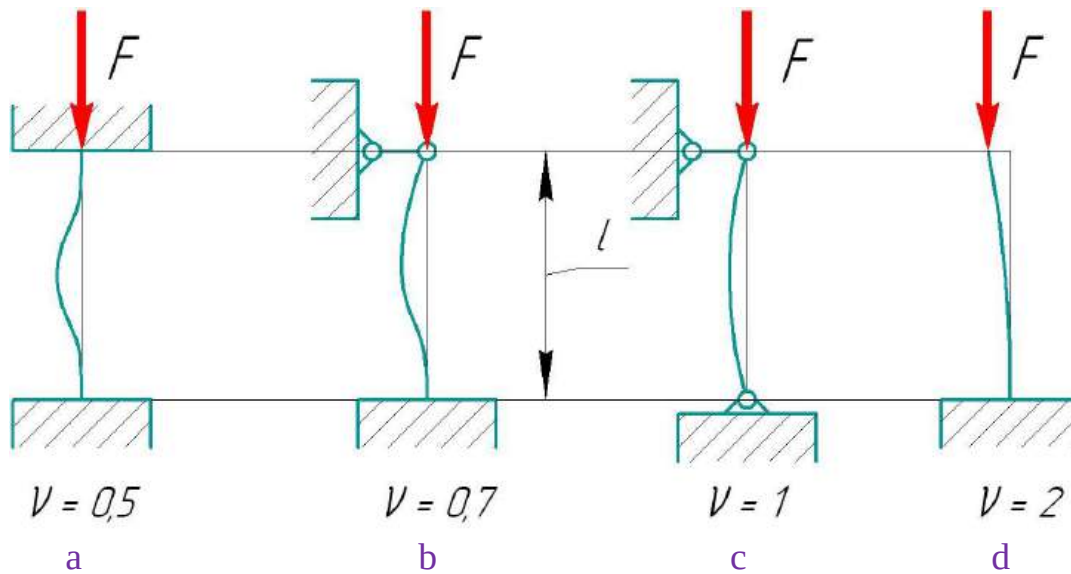


Fig. 3.12

After the value of diameter d_2 and d_1 are calculated, we should chose a standard thread (with a nominal diameter d and a pitch P) for which d_{2st} and d_{1st} are greater that calculated. As a rule, there are several combinations of threads and pitches. We should take into account that the bigger pitch the bigger efficiency and bigger the driving moment too. Therefore, we better choose an average pitch value.

Revise Calculation

To guarantee reliable unit operation the actual stresses in the most dangerous section should be found. The diagram of operating forces and the torques for typical variants is shown in Fig. 3.13 (a – for jacks; b – for removers and presses). Linear force distribution on threads is based on the axial force and the torque diagrams.

In the considered cases, the screw resists compression, longitudinal bending and torsion. The screw represented in figure "a" is loaded stronger than that in "b" as in interval BC it bears full load F and suffers large twisting moment T_f .

Combined maximum normal stresses are calculated as:

$$\sigma_{max} = \sqrt{\sigma^2 + 4\tau^2} \leq [\sigma],$$

where compression and shear stresses in dangerous cross-section are:

$$\sigma = \frac{F}{A}; A = \frac{\pi}{4} d_1^2 (1 - \alpha^2),$$

$$\tau = \frac{T}{W_\rho}; W_\rho = \frac{\pi d_1^3}{16} (1 - \alpha^4).$$

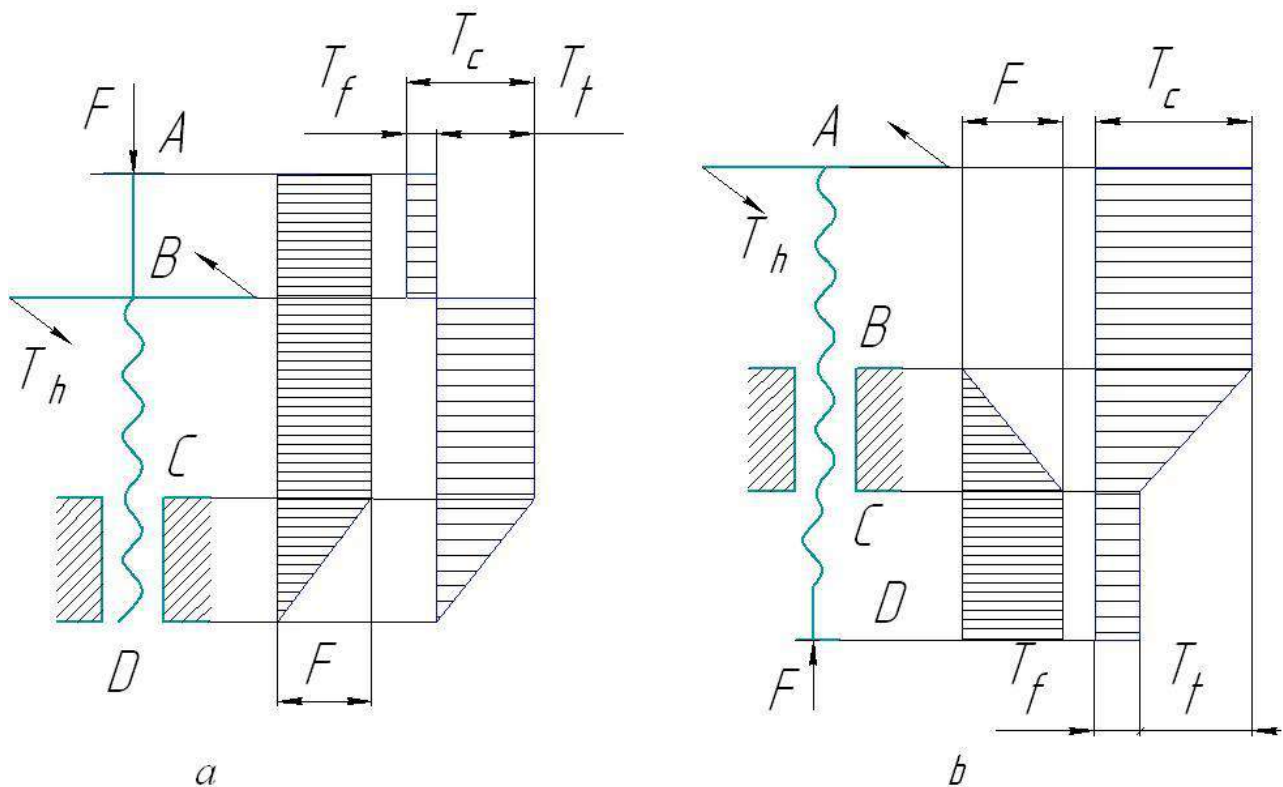


Fig. 3.13

Combined compression and buckling strength conditions may be employed as:

$$\sigma_{max} = \sqrt{\sigma^2 + 4\tau^2} \leq \varphi[\sigma],$$

where φ is the basic allowable stress reduction factor in the case of compressed screw. It is taken from the suitable tables depending on the screw material and the actual screw slenderness ratio:

$$\lambda = \frac{vl}{\theta d_1},$$

where the factor θ depends on hollowness of the screw cross-section:

$\alpha = d_{in}/d_{out}$	0,5	0,6	0,7	0,8	0,9
θ	0,28	0,29	0,305	0,317	0,34

If the strength condition is not satisfied, it is necessary to accept a standard thread with a bigger nominal diameter and then to check up whether the condition is fulfilled.

3.1.3. Ball Screws

Ball screw assemblies have a lot of balls that transfer the load between the nut and screw (Fig. 3.14). This has the effect of reducing ball screw friction and increasing efficiency.

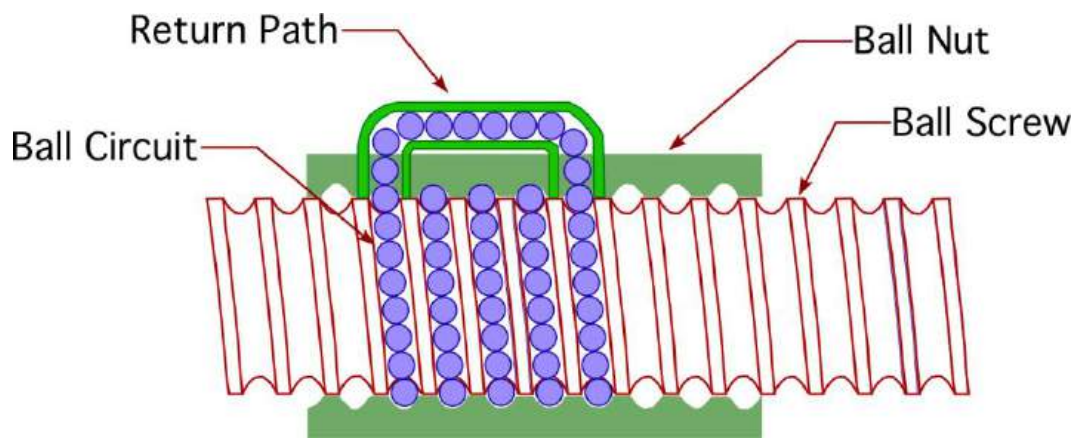


Fig. 3.14

Advantages

- high efficiency
- high load capacity
- high rigidity
- high displacement accuracy
- low wear and long service life

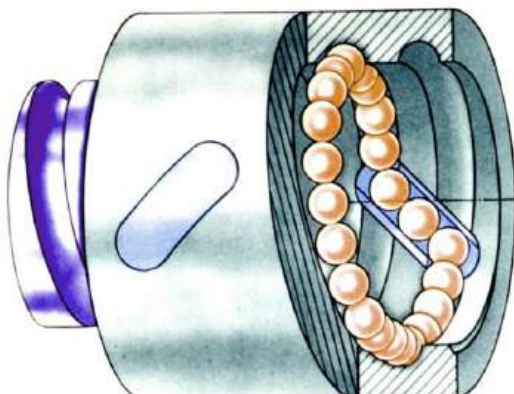
Disadvantages

- high price
- design and manufacture complications
- big radial size
- isn't self-locking

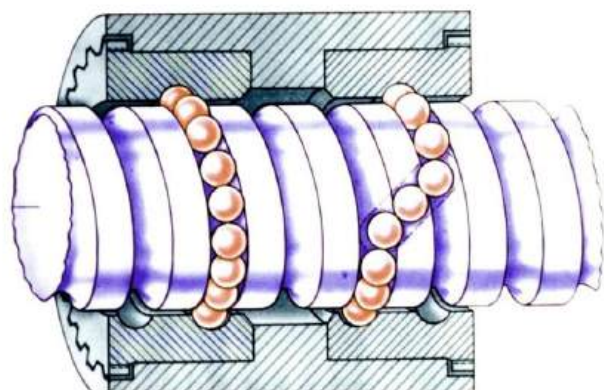
Ball Return Systems

All ball screws have a ball return path. As a ball screw rotates, friction between the balls and threads tend to push the ball bearings in one direction. A return path is thus added to all ball screws to allow the balls to cycle from one end of the thread back to the other end. Without a return path there may also be a tendency for a ball screw to bind. Two groups of power screw ball return systems are usually considered.

1. Balls move from one thread to previous through the groove between them:



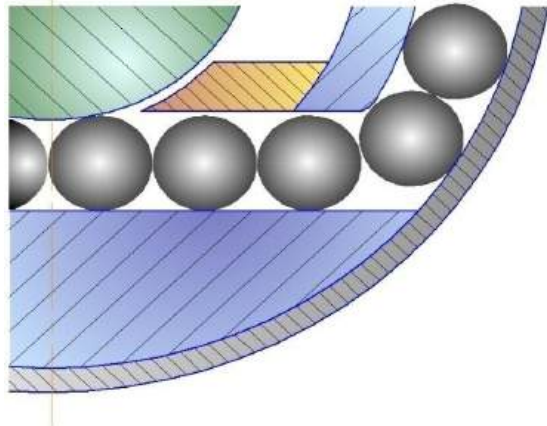
Groove in the nut



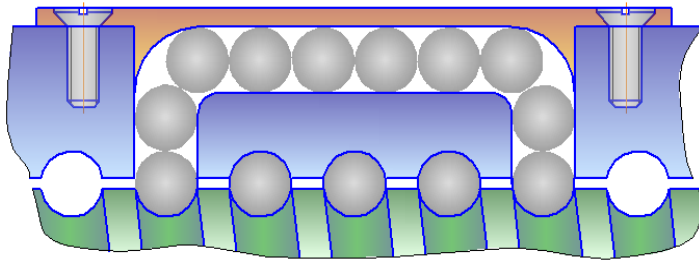
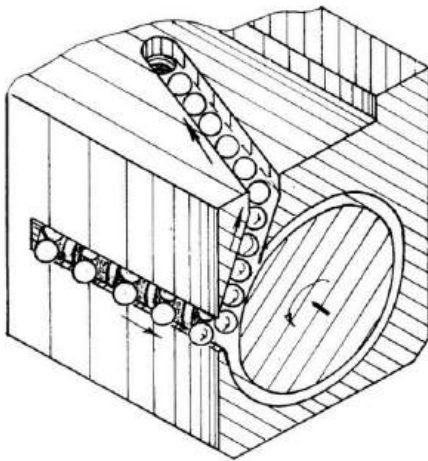
Groove on the screw

2. Balls are removed from the end of raceway with use of small pick up finger.
Then they move to the start of the raceway thread through:

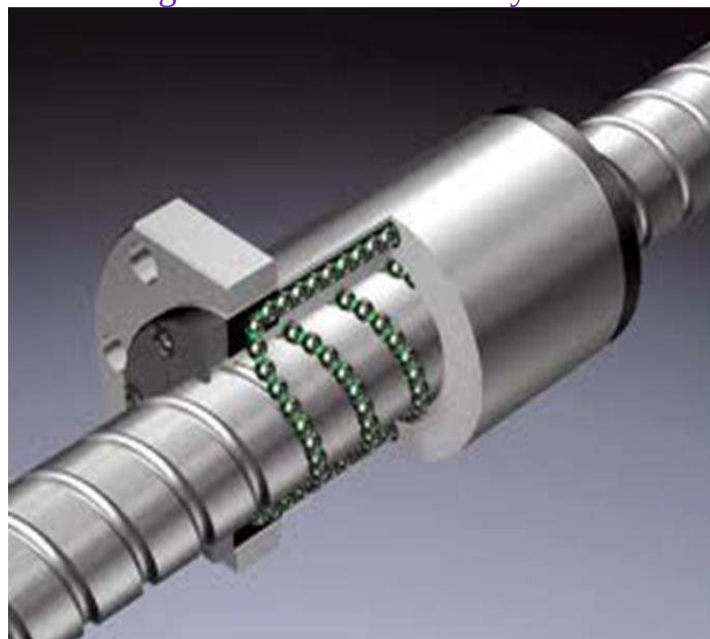
- the external bend tube:



- the groove machined on external nut surface:



- the hole drilled through the nut or screw body:



A circuit is the complete path for a set of balls. The circuit consists of the path between the screw and nut threads as well as the return path. For a given start within a ball screw/nut assembly, there may be one or more circuits. Multiple circuits are desirable for redundancy. In this case, if one circuit should fail such that all of the balls were lost, the other circuit would maintain operation of the ball screw. In aerospace, it is unlikely that a two start screw would have two circuits for per start (four ball circuits total) as this amount of redundancy is not generally required. However, for a single start screw the use of a two ball circuit design should be evaluated from a redundancy perspective.

Thread Parameters

Some of the ball screw thread parameters are the same as for any another thread (Fig. 3.15):

- mean diam. d_2 ;
- screw outside diam. d ;
- screw inside diam. d_1 ;
- nut inside diam. D ;
- nut outside diam. D_1 .

But some of them are specific:

- ball diam. D_b ,
- ball radius r_b ;
- groove radius r_g ;
- contact angle α_c .

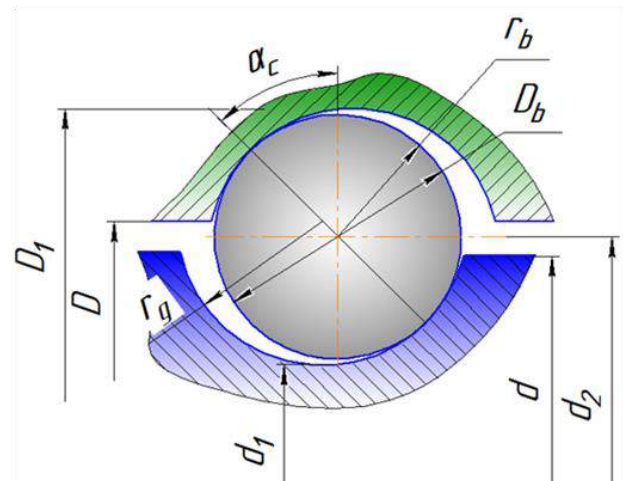


Fig. 3.15

Thread Profiles

As shown above, a ball screw assemblies have a lot of balls that transfer the load between the nut and the screw. Balls running between flat surfaces (rectangular profile, see Fig. 3.16) have small contact patches with each other and cannot transmit high loads. Thus ball-screw tracks usually have a curvature perpendicular to the balls' direction of movement.

The simplest type of not flat profiles is a semicircular arc with constant curvature (round, see Fig. 3.16), where balls run on one side of the track only. The balls are smaller than the arc forming the track ($r_b / r_g = 0,95 \dots 0,97$) and the optimum contact angle α_c is 45° . If the ball fits exactly into the track curvature, excessive sliding occurs between ball and track (sometimes referred to as differential slip).

The thread form in which the bearing balls ride is an ogival shape formed from two arcs of the same radius with offset centers. This form is also referred to as a gothic arch (see Fig. 3.16). It consists of at least two radii slightly larger than the ball used, but offset so the raceway touches the ball on both sides. Such thread profiles allow independent control of track conformity and contact angle, and are also suitable for four-point contact.

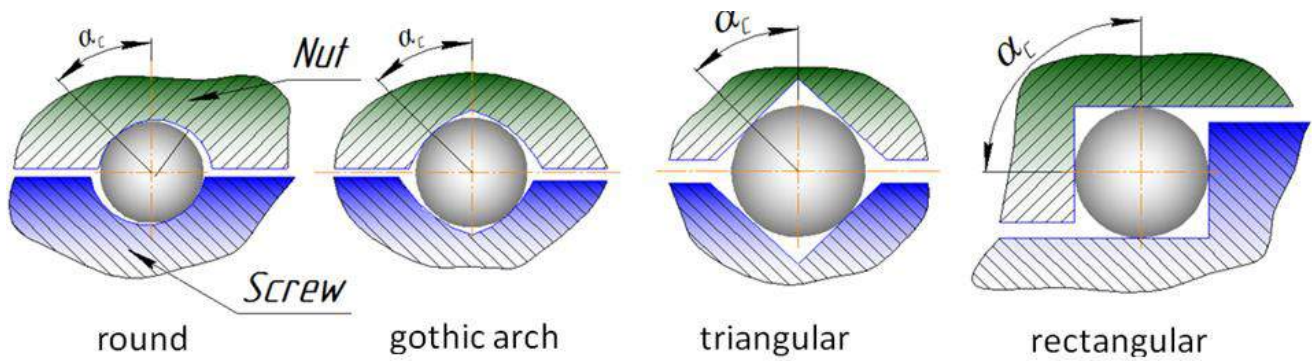


Fig. 3.16

Screw Material

Stainless steel is the usual choice for the aerospace ball power screws and nuts. Surface or case hardening of thread surfaces (increasing the hardness of ball screw and ball nut wear surface) are common in power screws to maximize wear characteristics, provide good surface stability and reduce backlash. The material is heat treated to achieve a desired hardness range (more than HRC 30). In some cases, the Rockwell hardness between a screw and a nut will be different.

To work at the highest efficiency, the running surface finish should be as smooth as possible. Surface plating of power screws surfaces is not a good idea because of the potential for flaking of the finish, which will lead to increased and accelerated wear on the surfaces.

The balls are generally hardened steel ball bearings manufactured to ball bearing specifications. The controlling specifications for the ball bearing should be reviewed for compatibility with the load and life expectancy for the balls. Additionally, each circuit will have a fixed number of balls such that the balls completely fill up the circuit with minimal freeplay in the circuit.

Backlash (Freeplay)

A power screw will always contain a specification for backlash or freeplay. Backlash or freeplay is determined by applying a small load in one direction and then measuring the distance the nut moves when applying the same load in the opposite direction. In measuring freeplay, the load should be small enough so that structural deflections are not occurring (a few percent of maximum load is usually sufficient).

Freeplay can lead to flutter when a power screw is connected to a flight surface. Also, freeplay is a measure of control accuracy. Anti-backlash nuts can be used to reduce freeplay. However, use of anti-backlash nuts on horizontal stabilizer and flap actuators is not used in practice.

Hardening is commonly used to ensure the freeplay requirements are maintained in service. Another means used in manufacturing to maintain tight freeplay requirements

is to machine the screw and nut to tighter tolerances and then use lapping techniques to open up the freeplay to the correct requirement.

Design Calculation

To determine mean diameter d_2 working surfaces contact strength condition (Hertz formula) is used:

$$\sigma_H = \alpha_\rho \sqrt[3]{\frac{F_n E^2}{\rho_{max}^2}} \leq [\sigma_H] = 2500 \dots 4000 \text{ MPa},$$

where F_n is the force normal to the contact surfaces:

$$F_n = \frac{F}{Z \lambda \sin \alpha_c \cos \psi} \leq [F_n];$$

Z is the number of balls in contact; λ is the non-uniformity factor, $\lambda = 0,85$; α_c is the contact angle; α_ρ is the auxiliary factor depending on ratio between maximum and minimum design radii in a contact point (Fig. 3.17):

$$\rho_{max} = \frac{D_b r_g}{2r_g - D_b}; \quad \rho_{min} = \frac{D_b d_1}{2(D_b \cos \alpha_c + d_1)}.$$

Thus, mean diameter d_2

$$d_2 = \sqrt{\frac{4(1 - K_r)^2 E^2 F \alpha_\rho^3}{K_W^2 [\sigma_H]^3 Z \lambda \sin \alpha_c \cos \psi}}.$$

Here two dimensionless auxiliary factors depending on thread geometrical parameters are used:

$$K_r = \frac{D_b}{2r_g} = 0,945 \dots 0,972;$$

$$K_W = \frac{D_b}{d_2} = 0,106 \dots 0,15.$$

To solve this problem, let us assume at the base of experience next mean values: $K_r = 0,97$, $K_W = 0,145$, $Z = 100$, $\alpha_\rho = 1,5$, $\psi = 4^\circ$, $\alpha_c = 45^\circ$, $E = 2,1 \cdot 10^5$ MPa. In such way we can reduce the formula as follows:

$$d_2 = 21\,000 \sqrt{\frac{F}{[\sigma_H]^3}}.$$

If a ball race surface hardness is under 62 HRC, allowable stress should be multiplied by factor K_T :

HRC	29	35	40	45	49	54	58	62
K_T	0,38	0,415	0,5	0,6	0,69	0,79	0,89	1

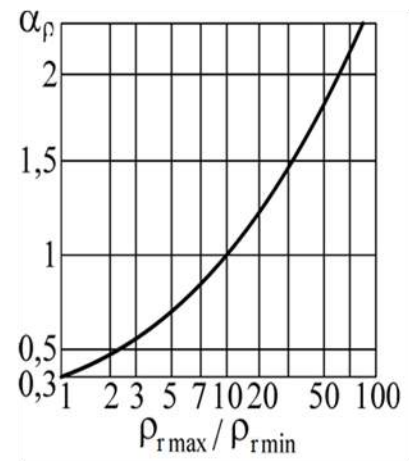


Fig. 3.17

After the value of diameter d_2 has calculated, we should chose a recommended thread parameters (nominal diameter d_2 , pitch P , ball diameter D_b , etc.) for which d_2 is bigger that calculated.

Then the number of balls

$$Z = \frac{F}{[F_n] \lambda \sin \alpha_k \cos \psi} \leq 120$$

and minimum number of threads

$$N_{min} = \frac{Z D_b \cos \psi}{\pi d_2}$$

may be obtained.

According to calculated N_{min} , the total number of threads should be selected:

N_{min}	1,5	2,5	3,5	4
N_{total}	3	4,5	5,5	6

and the number of balls should be re-calculated:

$$Z = \frac{N_{total} \pi d_2}{D_b \cos \psi} \leq 120 .$$

If this condition is not satisfied, mean diameter d_2 and ball diameter D_b should be increased.

Efficiency

Ball screw-nut efficiency is obtained according to the known formula

$$\eta = \frac{\tan \psi}{\tan(\psi + \rho')},$$

but design angle of friction is calculated as

$$\rho' = a \tan \frac{k}{D_b \cos \alpha_c},$$

where coefficient of rolling friction

$k = 0,01 \dots 0,015$ mm (Fig. 3.18).

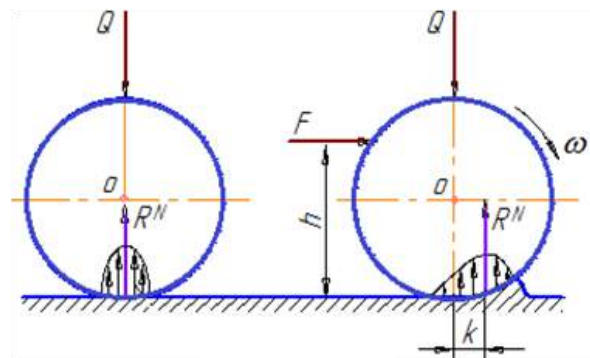


Fig. 3.18

Verify Calculation

When thread dimensions are known, stresses acting in screw may be calculated like in case of sliding thread according to the combined both compression and bulking strength condition:

$$\sigma = \sqrt{\sigma^2 + 4\tau^2} \leq \varphi[\sigma]$$

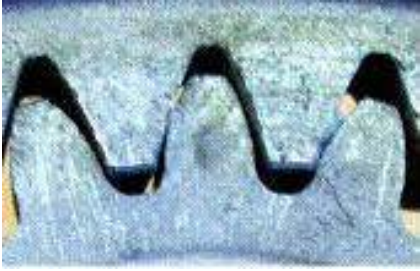
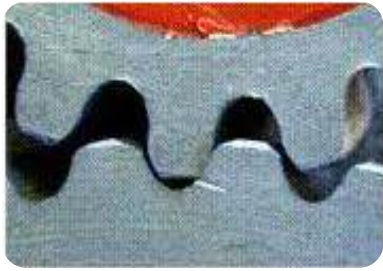
(all designations see above).

3.2. Gearing

Gearing is a three-link mechanism, in which two elements are moving gears, forming a fixed link of rotational or translational kinematic pair.

3.2.1. Classification

By teeth profile

Involute	Round	Cycloidal
		

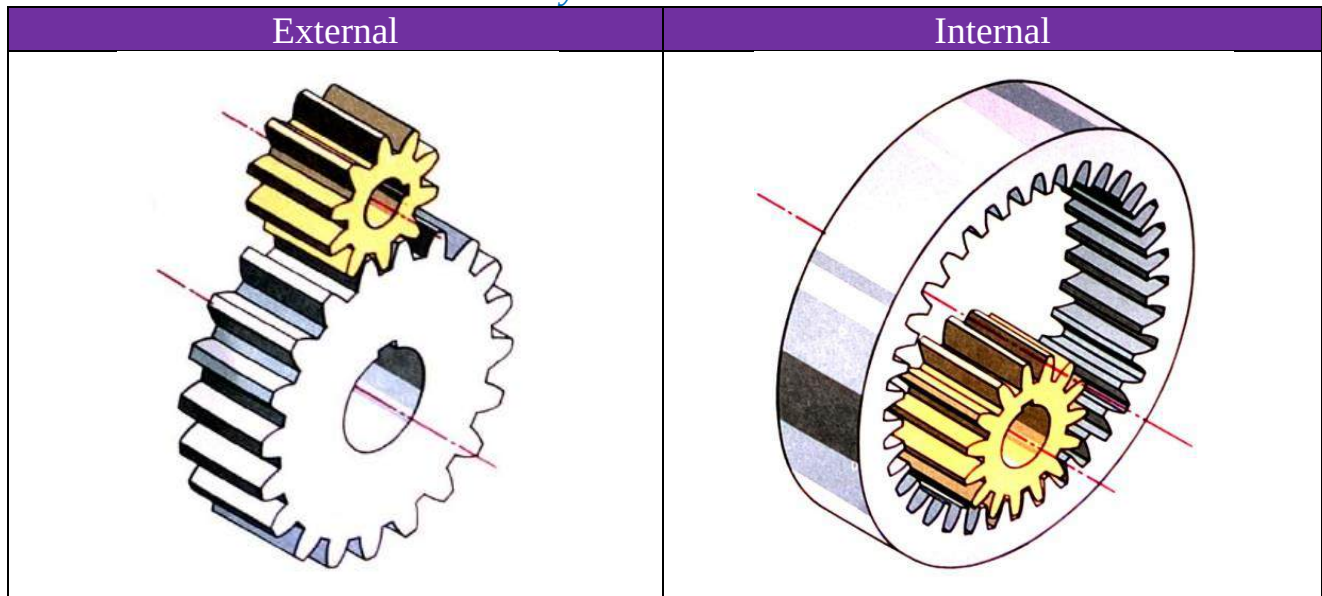
By teeth type

Spur (straight)	Helical	Double helical (herringbone, chevron)	Round (curved)
			

By housing

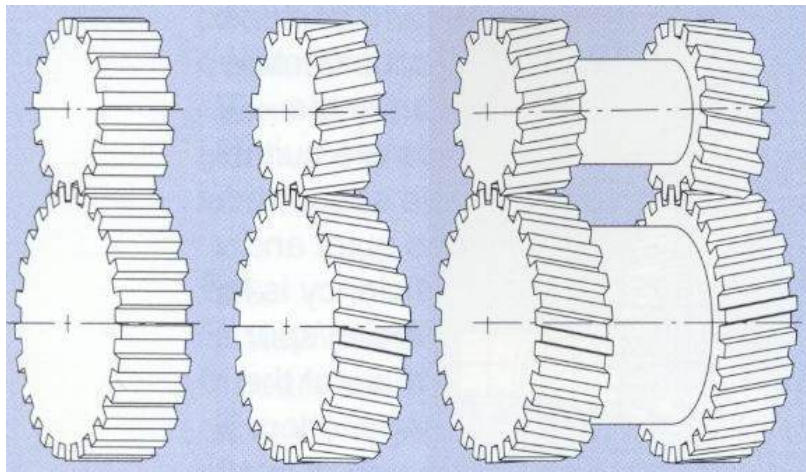
Closed	Open
	

By teeth location



By shaft axis location

With parallel axes:



With intersected axes (bevel):



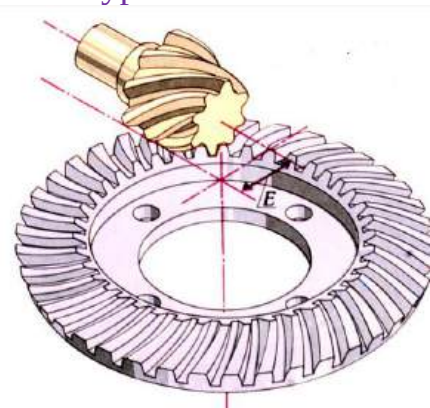
With axes in the different planes:
Screw



Worm



Hypoid



Advantages

- High reliability and load capacity
- Long service life
- High efficiency – up to 0,99
- Stable transfer ratio

Disadvantages

- Manufacture requires special equipment and tools
- Inevitable errors in teeth machining cause vibration and noise during operation

3.2.2. Terminology

The main parameters are shown in Fig. 3.19.

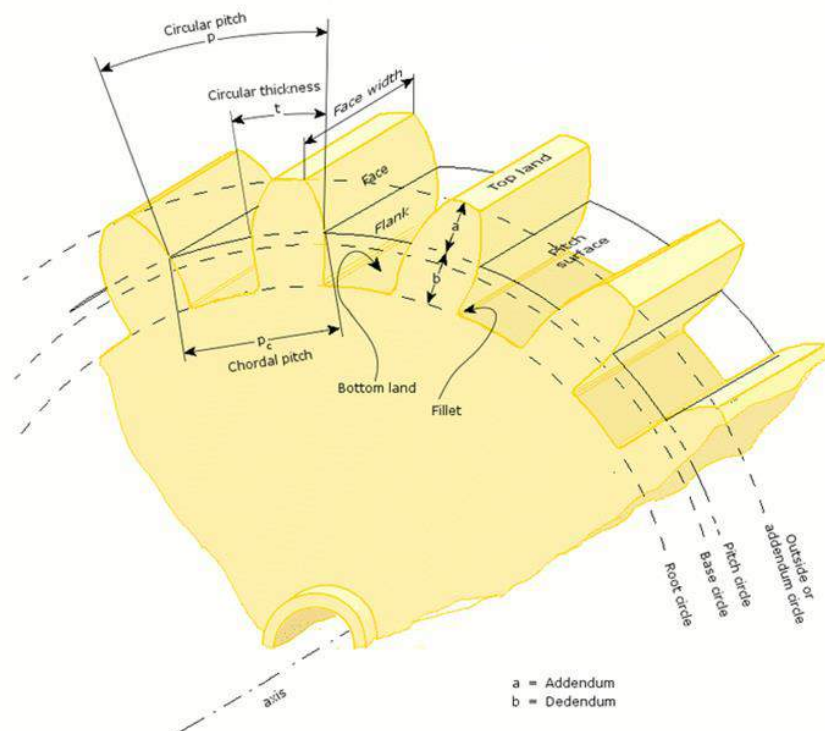


Fig. 3.19

3.2.3. Teeth Operation Conditions

When teeth contact each other, load is transferred first by one pair, then by two pairs of teeth. It means that close to the addendum and to the root, when two pairs of teeth contact, load two times less that close to the pole, where the only pair of teeth operates (Fig. 3.20).

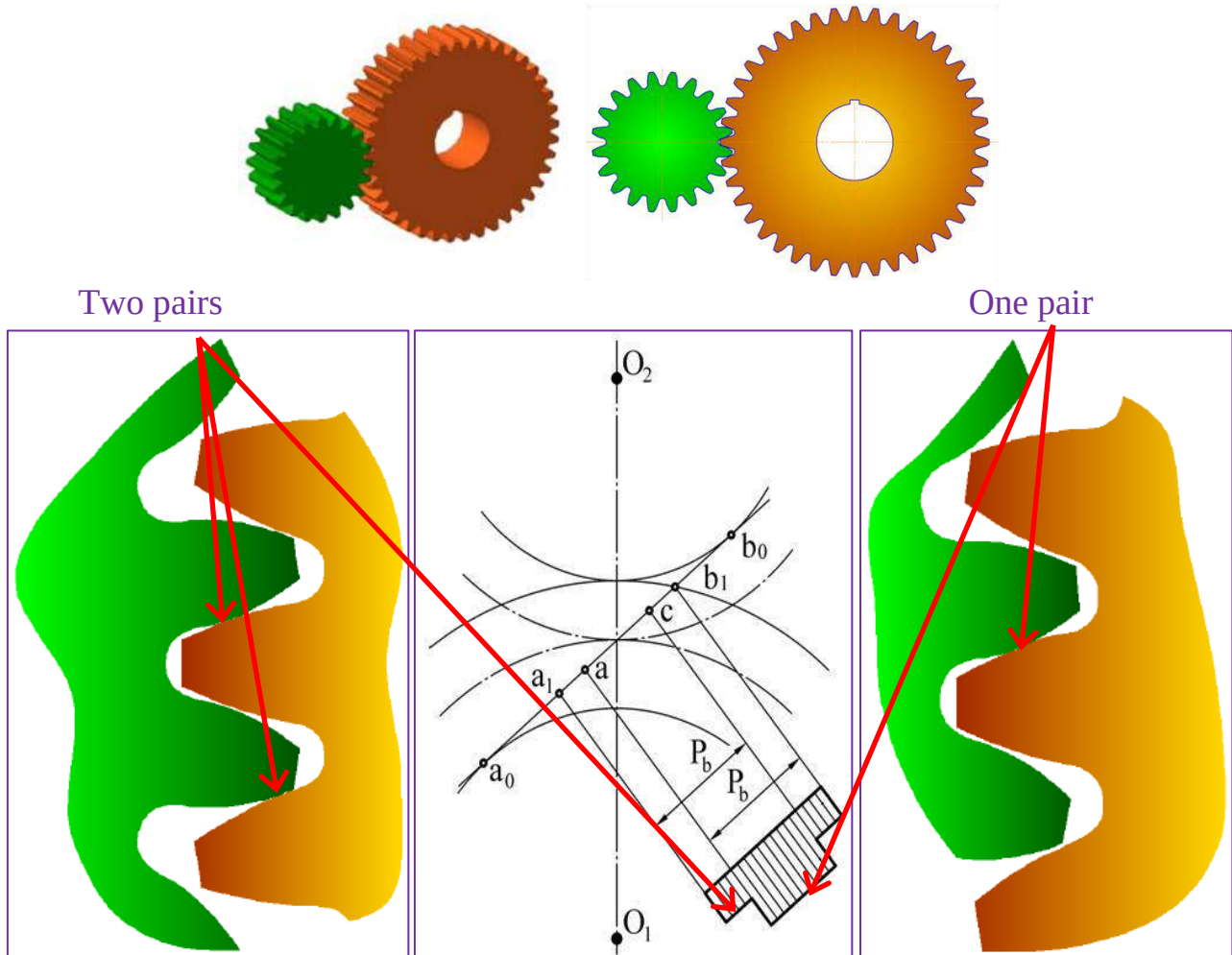


Fig. 3.20

Acting forces cause contact and bending stresses (Fig. 3.21). Friction force exists but it is neglected in calculations.

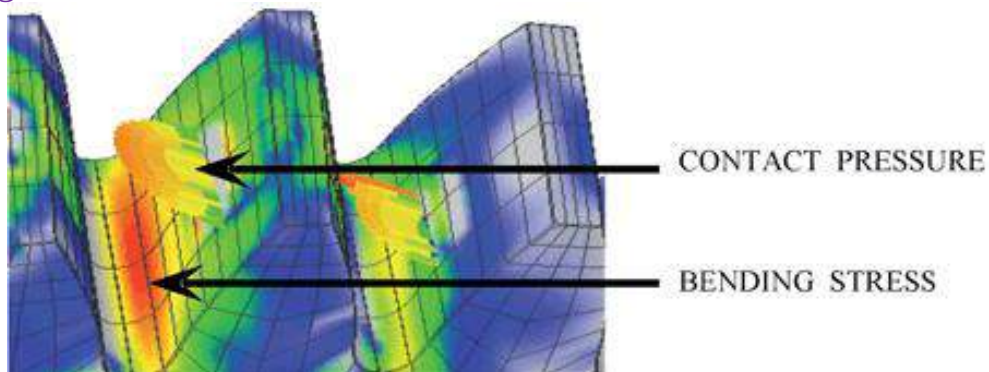


Fig. 3.21

Acting forces and stresses are variable. Each tooth is loaded periodically, as minimum one time during one revolution (Fig. 3.22).

4. In addition to the payload an engagement is loaded by:

- the external dynamic loads which are generated by uneven loading of a drive or/and driven machine;
- the internal dynamic loads caused by the inaccuracy of manufacturing and shafts' deflections.

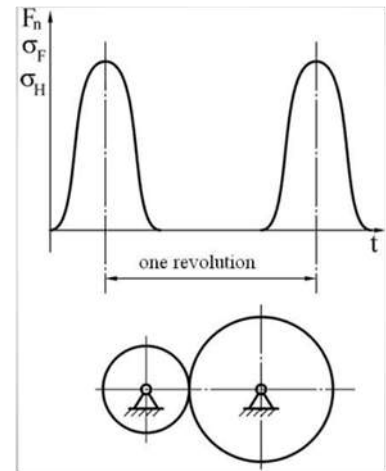
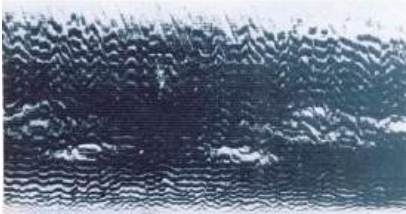


Fig. 3.22

3.2.4. Teeth Damages

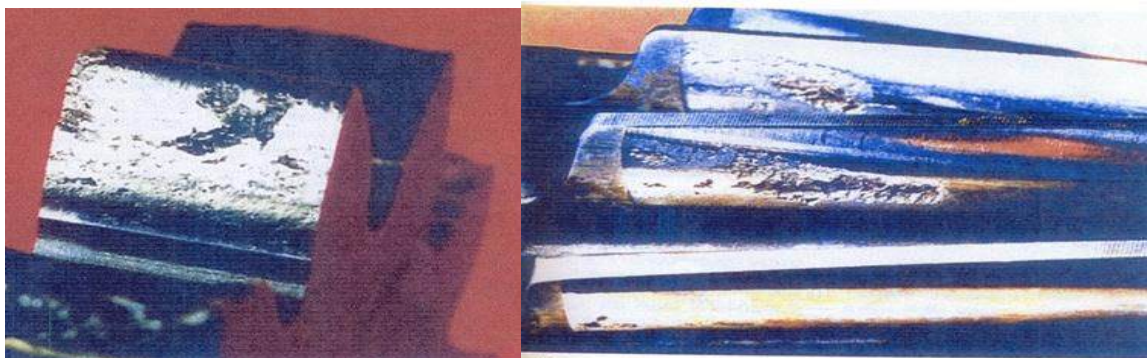
Plastic Flow

Plastic flow of tooth surface is the result of high contact stress under rolling sliding action. Surface deformation takes place due to yielding of surface or subsurface material:

Rippling	Plastic deformation	Cold flow
		

Pitting

Pitting is a surface fatigue failure. It occurs due to repeated loading of tooth surface and the contact stress exceeding the surface fatigue strength of the material. Material in the fatigue region is removed and a pit is formed:



Wear

The wear is a kind of tooth damage wherein layers of metal are removed from the surface. Consequently, tooth thins down and gets weakened:



Some other kinds of damages are: abrasive (fatigue, teeth penetration, by particles); corrosive (chemical, fretting); adhesive (polishing moderate severe).

Tooth Fracture

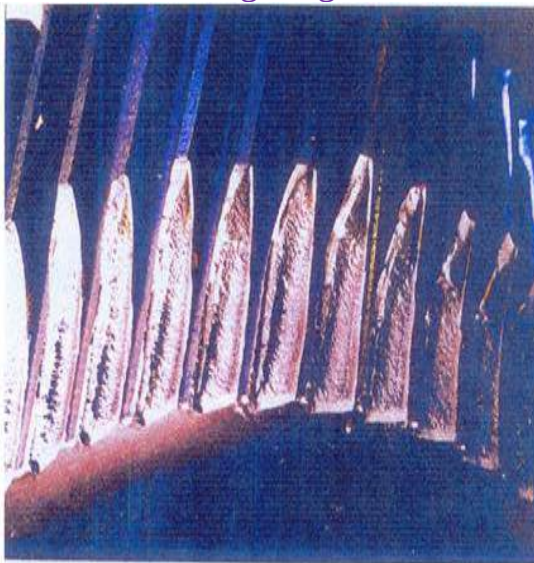
Tooth breakage is the result of:

- high overloads of either impact or static force,
- low-cycle fatigue caused by repeated overloads,
- high cycle fatigue of the material due to multiple repeated loads.

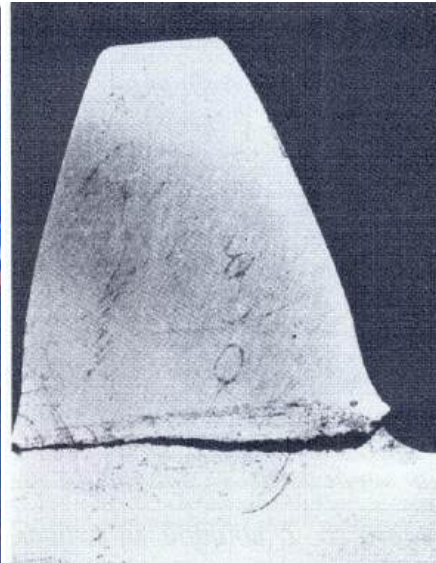
Some types of teeth fracture:

Overload breakage	Low cycle fatigue
	Pick overload
Fatigue breakage	High cycle fatigue
	Bending fatigue

Bending fatigue



Overload



3.2.5. Design Load

To take into account the additional forces design load factor K is used:

$$F_p = KF,$$

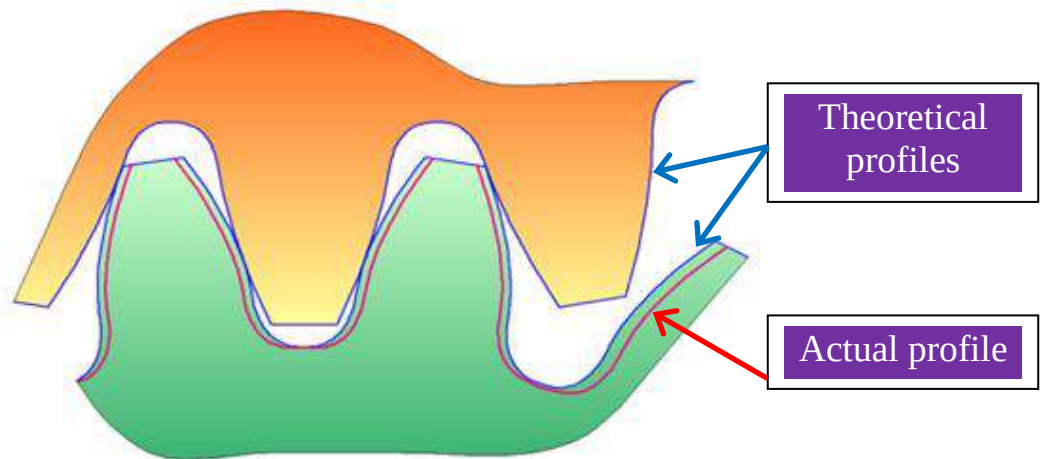
$$K = K_V K_\beta K_\alpha K_A,$$

where K_V is the factor considering internal dynamic load; K_β is the factor considering non-uniformity of load distribution along the contact lines; K_α is the factor considering the load distribution between teeth; K_A is the factor considering external dynamic load.

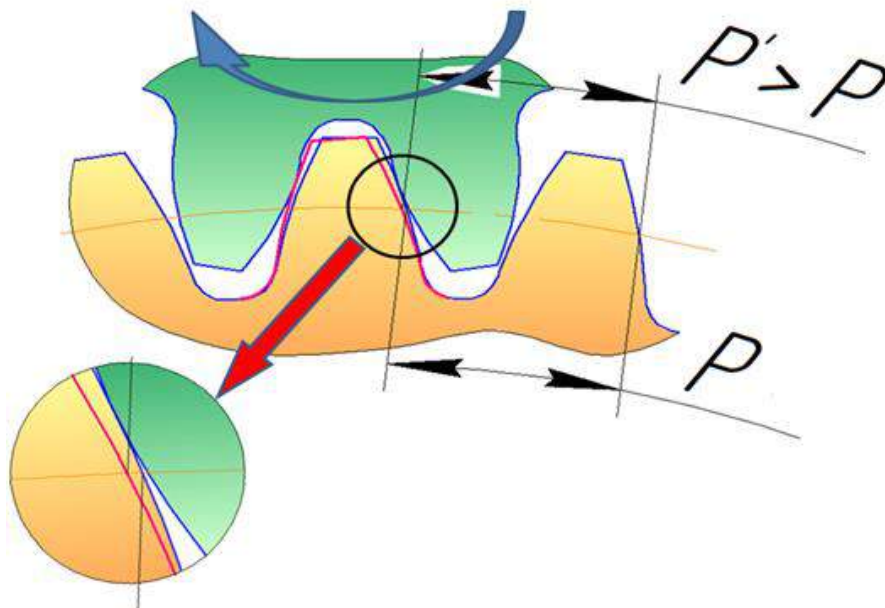
Factor K_V

The internal dynamic force is characterized by specific dynamic load w_v , which appears due to:

- pitch value deviation because the teeth profile is machined according to assumed degree of accuracy and each dimension has suitable tolerance:



- deformation of teeth under the load:



When the condition $VZ_1 < 1000$ for spur gear and $VZ_1 < 1400$ for helical transfers is satisfied (far from resonance zone) K_V is found as follows

$$K_V = 1 + \frac{W_v b_w}{F_t},$$

where V is the tangential velocity, m/s; Z_1 is the pinion number of teeth; b_w is the gear face width, mm; F_t is the tangential force on a pith diameter, N; W_v is the specific dynamic load, N/mm.

Specific dynamic load characterizes the intensity of dynamic process and can be calculated by means of follow empirical equation:

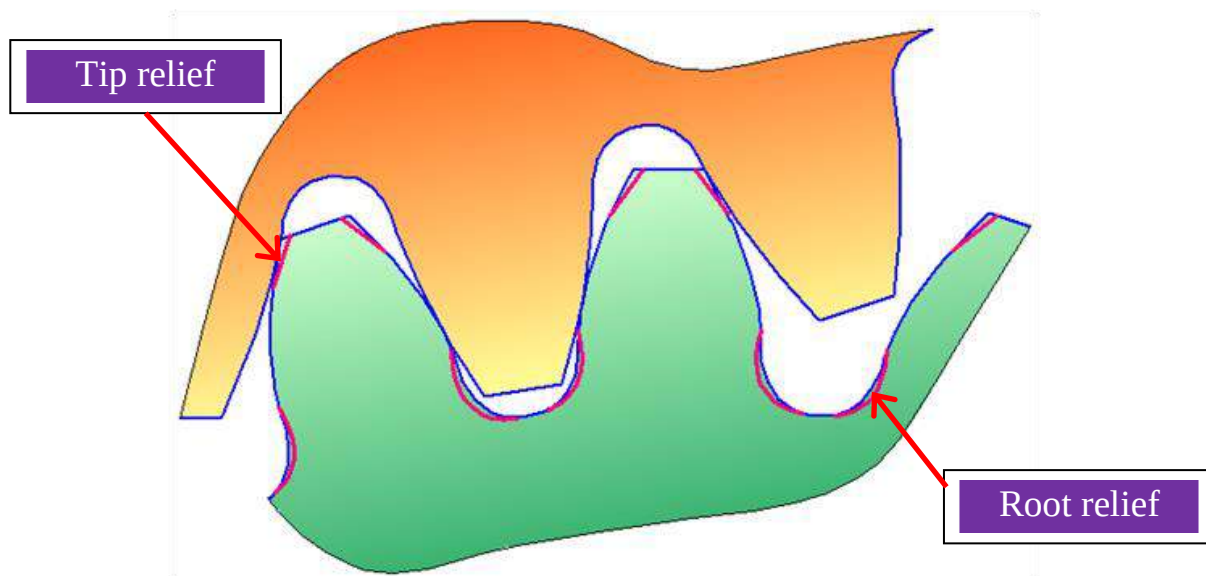
$$W_v = \delta g_0 V \sqrt{\frac{a_w}{u}},$$

where a_w is the center distance, mm; u is the transfer ratio; δ_H is the factor which considers influence of a profile tip relief and a kind of teeth (Table 3.2); g_0 is the factor considering influence of a pinion and a gear pitch difference (Table 3.3).

Table 3.2

Hardness	Tooth type		δ
< 350 HV	straight	without tip relief	0,06
		with tip relief	0,04
	helix		0,02
More than 350 HV	straight	without tip relief	0,14
		with tip relief	0,10
	helix		0,04

To decrease harmful noise and vibration, the most effective method is “tooth profile modification”, which is by tip relief or root relief for modifying geometry profile of tooth to regulate transmission error:



It ensures smooth sliding of the teeth into and out of contact without impacts and compensates misalignment, errors in manufacture, and deflections under load.

Table 3.3

Module m , mm		Degree of accuracy					
		4	5	6	7	8	9
< 3,55	g_0	1,7	2,8	3,8	4,7	5,6	7,3
	W_{Vmax} , N/mm	32	85	160	240	380	700
3,55...10	g_0	2,2	3,1	4,2	5,3	6,1	8,2
	W_{Vmax} , N/mm	53	105	194	310	410	880
> 10	g_0	2,7	3,7	4,8	6,4	7,3	10,0
	W_{Vmax} , N/mm	98	150	250	450	590	1050

If calculated value exceeds limit W_{Vmax} (see Table 3.3), it should be accepted equal to limit value.

Factor K_β

The factor K_β is the relation of the greatest specific load to its average (Fig. 3.23):

$$K_\beta = \frac{W_{max}}{W_m}$$

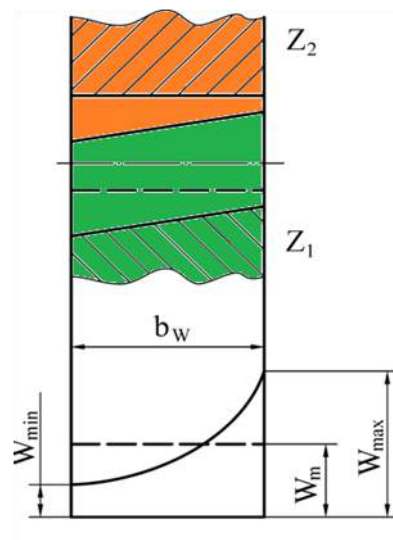
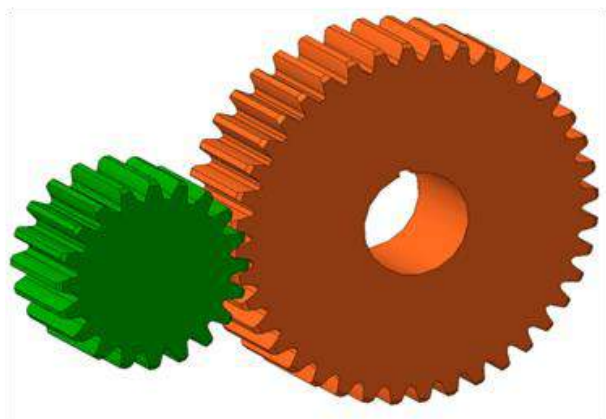


Fig. 3.23

It considers non-uniformity of load distribution on length of the contact line. It is known that the load is not distributed uniformly over the line of contact because during the gear operation its parts are deformed and change their mutual position. This is also due to inaccuracies in the machining of wheels, shafts and housing and to assembly errors.

The value of W_{max} depends on many factors whose effect cannot be accurately calculated. The most important are as follows:

- the location of the wheels on the shaft relatively to the bearings – the bending effect on load concentration is negligible when wheels are fitted near the shaft middle (Fig. 3.24, a);

- the relative length of the shafts – the greater the ratio between shaft length and diameter, the more the shaft deflection and asymmetrically fitted wheels are misaligned (Fig. 3.24, b);

- the relative face width of the pinion $\psi_{bd} = \frac{b_w}{d_{w1}}$ – the larger this ratio, the greater the relative shift of the tooth section along the width of the misaligned wheels and the W_{max} will be greater. That is why in general engineering $\psi_{bd} = \frac{b_w}{d_{w1}} = 0,4...0,9$, in aviation $\psi_{bd} = 0,2...0,5$.

- the total rigidity of the teeth – the higher the rigidity of the mating teeth, the greater load concentration.

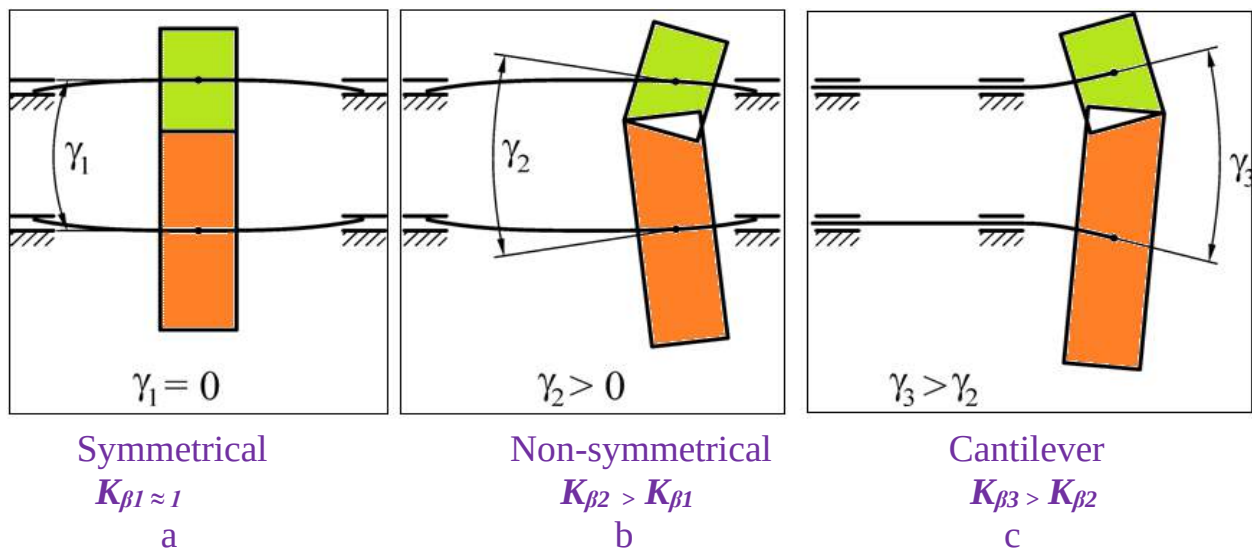


Fig. 3.24

To reduce load concentration the bearings of the shaft should be preferably located near the wheels and symmetrically relatively to them. When the wheels are arranged asymmetrically, their shafts should be more rigid. Pinions and wheels fixed cantilever (Fig. 3.24, c) should be narrower and mounted as close to the bearing as possible.

Under design and approximate calculations, it is possible to accept value $K_{H\beta}$ approximately, by means of graphs (Fig. 3.25). It is seen, that the load concentration depends on teeth face width b_w , surface hardness HB (HRC, HRV) and gear location.

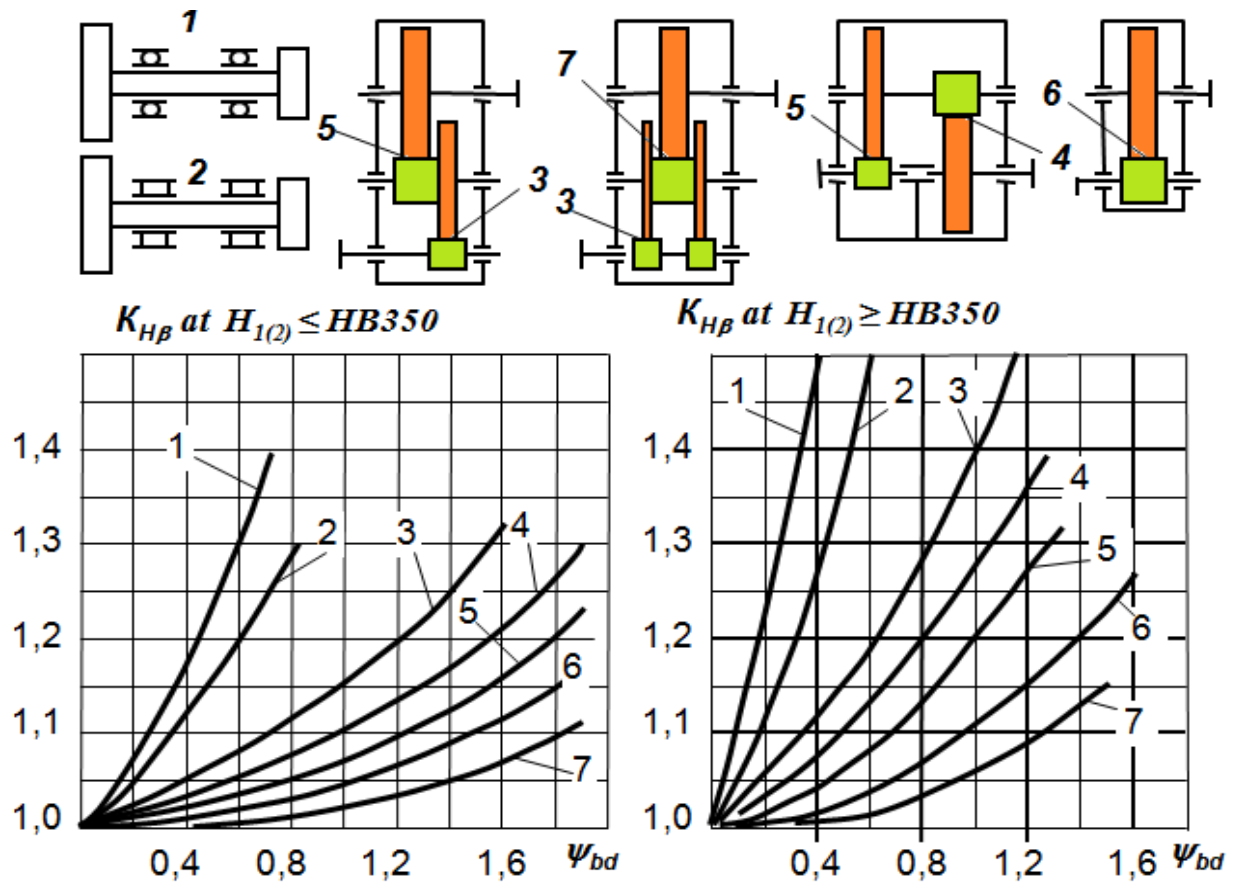


Fig. 3.25

Factor K_A

It takes into account both a prime driver and a driven machine operating non-uniformity (Table 3.4).

Table 3.4

Engine's operation condition	K_A for driven machine's operation condition			
	Uniform	With low non-uniformity	With medium non-uniformity	With high non-uniformity
Uniform	1,00	1,25	1,50	1,75
Low non-uniformity	1,10	1,35	1,60	1,85
Medium non-uniformity	1,25	1,50	1,75	> 2,00
High non-uniformity	1,50	1,75	2,00	> 2,25

3.2.6. Accuracy

Accuracy, kinematics and smoothness of transfer work are broken by manufacturing errors of a circular pitch and the form of a profile of the teeth. Errors in a direction of teeth together with a shaft deflection lead to non-uneven load distribution on tooth length.

Production errors are unavoidable due to:

- deviation in pitch, profile, tooth direction;
- radial run-out of the gear ring;
- deviation from parallelism and misalignment of the gear axis;
- center distance variation;
- etc.

These errors result in:

- increasing noise;
- rotational un-accuracy of the driven wheel;
- failure of precision and smooth tooththing;
- torsional vibration;
- etc.

For spur, bevel and hypoid tooth gearings 12 standard degrees of accuracy from the first to the twelfth as its decrease are provided (see examples in Table 3.5).

Table 3.5

Gear transfer branch	Degree of accuracy
Aviation gearboxes, turbo engines and generators	4...7
Cars	6...9
Gearboxes for general purposes	7...10

Accuracy degrees as a first approximation define depending on peripheral speed:

Peripheral speed, m/s	Degree of accuracy								
	3	4	5	6	7	8	9	10	11
Spur gears with straight teeth	35	25	15	15	10	6	3	1	0,5
Spur gears with helix teeth	70	45	30	25	15	10	5	2	1

Degrees of accuracy provide norms of tolerances as follows:

Tolerances	Regulates	Measures
Kinematic	Difference between the actual and nominal rotational angles of the driven gear wheel per one pinion revolution	Radial run-out of the gear ring, center distance variation per one pinion revolution in gearing with etalon one, etc.
Smooth operation	Rotary speed fluctuations taking place many times per one wheel revolution	Center distance variation per one tooth, deviation in profile, etc.
Teeth contact	Teeth adjacency in the mounted gearing and the degree of load distribution in contact lines	Deviation in tooth direction, deviation from parallelism and misalignment of the contact lines, total teeth contact area, etc.

The gearing type of coupling is also regulated. Type of coupling does not depend on the accuracy tolerance and includes norms of the teeth non-working surface backlash (side clearance) and the tolerance on this dimension (Fig. 3.26).

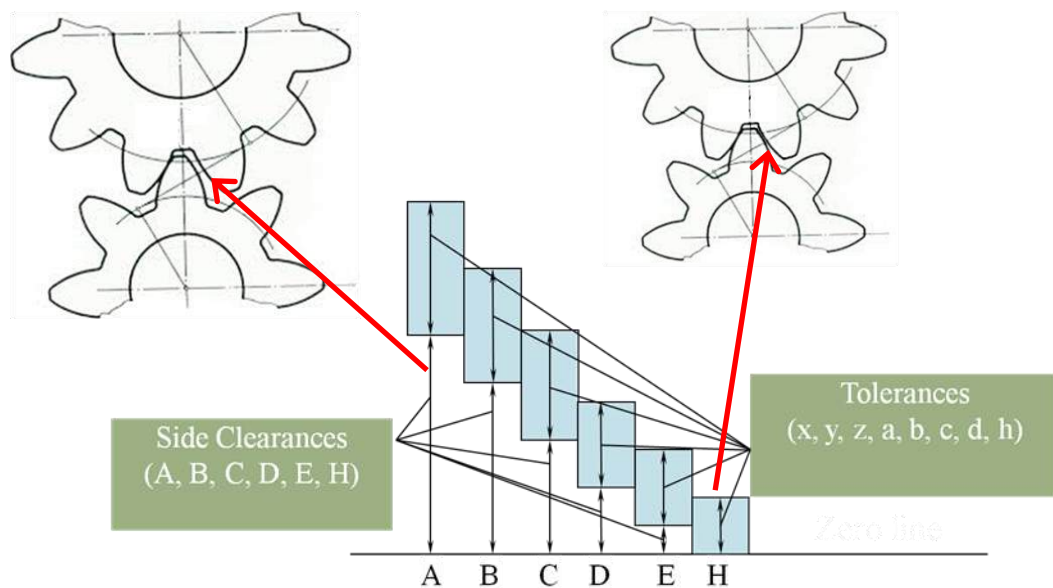


Fig. 3.26

Side clearance (Fig. 3.27) is the distance between the teeth side faces, which determines the free rotation of one of the gear wheels by a fixed another one.

Side clearance is required:

- to avoid teeth seizing in the gearing as a result of their expansion at the working temperature;
- to provide a space for lubricant;
- to provide free wheel rotation.

An individual gear does not have backlash. Backlash is only present when one gear mates with another. The backlash of a gear set is dependent upon tooth thickness, run-out, tooth geometry of each member in mesh, and their operating center distance.

Type of coupling B provides the minimum side clearance, which prevents the heat jamming of steel gears at the temperature difference between gears and housing up to 25° C.

Designation

The gears degree of accuracy is designated by means of digits and letters in this order: Kinematic norm – Smoothness norm – Contact norm – Type of coupling – Backlash tolerance.

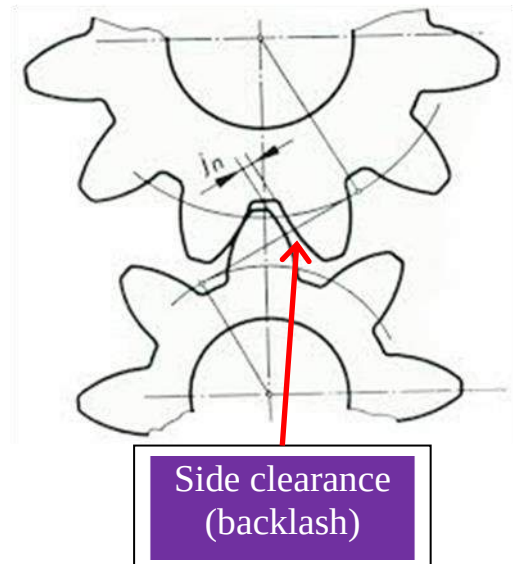


Fig. 3.27

Examples

8-7-6-B a	Grade 8 according to the standards of kinematic accuracy; grade 7 according to the standards of drive operation smoothness; grade 6 according to the standards of teeth contact; type of coupling B and tolerance a
7-B	All grades are the same and equal 7; type of coupling B and tolerance b

3.2.7. Gears’ Materials and Types of Chemical and Heat Treatment

The material for the manufacture of a designed toothed wheel should be so selected as to make it possible to cut and finish the teeth with the required accuracy and to ensure sufficient beam strength under the action of varying and impact load, adequate strength of the tooth surfaces and high resistance to abrasion.

Toothed wheels are ordinarily made from steel and cast iron, which can be used to produce wheels of any proportions. These materials are easily cast, especially cast iron, while steel has a good forge ability.

The tendency towards a reduction in size, the transmission of large loads in one assembly and at higher speed has stimulated the wide use of steel wheels. A great variety of grades of steel and the possibility of obtaining various properties by means of heat treatment allow the most favorable combination of required properties.

Carbon steel with 0,35 to 0,50 % carbon content has proved highly efficient for medium loads. It possesses sufficient strength and hardness and good machinability (steel grades 35, 40, 45, 50: Ukrainian standard designation is Steel 35...Steel 50; US analogue designation 1035...1050).

As a rule, these steels are used in a normalized or tempered state. The size of the wheel cross-section materially affects the mechanical properties obtained after heat treatment. This is explained by the fact that, as the size of the cross-section increases, the cooling rate diminishes, and, if it drops below some critical point, there will be full hardening.

The size of cross-section is much less decisive in wheels made from alloy steel, which makes it possible to obtain better mechanical properties for large wheels. Of all alloy steels used for the manufacture of toothed wheels preference is given to grades Steel 40X (US analogue designation 5140) and Steel 40XH (US analogue designations 3135, 3140H).

Finish cutting of such toothed wheels is done after final heat treatment; the hardness of the gear face surfaces is therefore $HB < 350$ and more frequently $HB < 320$.

In order to increase the load-carrying power of toothed wheels and reduce the overall dimensions of the gear, use is made of toothed wheels whose surface hardness is above $HB = 350$ achieved by through or surface hardening, casehardening, cyaniding and nitriding.

In a through hardening of carbon and alloy steels with 0.35...0.5 % carbon content the hardness obtained depends on the carbon content of the steel and can reach $HRC = 50...65$.

For important gears, when overloads and hammering effect are anticipated, steel grades 40XH and 40XHMA (US analogue designations 4340, 9840) are used, and, more frequently, 40X. The maximum core hardness for carbon steel should never exceed $HRC = 45$, and for alloy nickel and molybdenum steel – $HRC = 50$, since otherwise impact strength drops sharply.

The disadvantage of through hardening consists in the considerable warping of the wheels and a reduction in the toughness the tooth core which reduces tooth resistance to bending under impact load. This shortcoming can be avoided by surface hardening which ensures adequate strength of the surface and retains the toughness of unhardened metal. This method of heat treatment can be used to advantage in large wheels and is especially effective in wheels with large cross-sections, since with a conventional method of hardening it allows the use of carbon steel instead of alloy steel. The surface hardness obtained amounts to $HRC = 51...57$.

Casehardening is another means of increasing the surface hardness of the teeth while retaining proper core toughness. For casehardening use is made of steel with 0.1 to 0.2 % carbon content, 0.15 % being the most suitable. Casehardened alloy steels possess high abrasion resistance. The hardness of the casehardening surface reaches values HRC about 56...63.

Carbon steel of grades 15 and 20 subjected to casehardening is comparatively rarely used for making toothed wheels: the metal under the hard case resists poorly both surface and beam bending stresses, uneven hardening causes unequal surface hardness while at higher loads the hard case separates.

Alloy chromium steel, of grades 15X (US analogue designations 5015, 5115) and 20X (US analogue designations 5117, 5120, 5120H), ensures better quality of the wheels, reduced warping and greater core strength.

When overloads or impact loads are anticipated (gear wheels of automotive vehicles, aircraft reduction gears, etc.), and when the impact resilience and plastic properties of the core are of special importance, use is made of grade 12XH3A (US analogue designations 3415) chrome-nickel steel, grade 15XΦ chrome–vanadium steel, grade 18XTT chrome-manganese-titanium steel and other alloy steels.

Nitrided and cyanided wheels are extremely effective as they warp little, making subsequent grinding unnecessary. Wheels with internal toothing are very often nitrided. For such wheels steel grade 38XMIOA (US analogue designations J24056) is employed.

The hard nitrided case is rather thin (0.1...0.3 mm); therefore wheels with nitrided teeth are employed for steady loads in well–lubricated reduction gears to preclude or reduce abrasive wear.

Sometimes toothed wheels are made from fabric laminated and veneer laminated phenolic plastics, artificial leather and fibre. A phenolic laminated pinion should have a smaller width than its metal counterpart. Otherwise, the edges of the metal wheel teeth will damage the surface of the phenolic laminated teeth.

Mechanical characteristics of some most widespread steels are shown in Table 3.6.

Table 3.6

Steel	Type of treatment	Limit stresses, MPa		Hardness	
		σ_{UTS}	σ_y	of surface, HRC	of core, HB
40XH	Tempering	850	600	230...300	
	Surface hardening	1600	1400	48...54	260...300
40X	Tempering	850	550	230...260	
	Surface hardening	1000	850	45...55	260...280
30XГCA		1080	830		280...320
12XH3A	Casehardening	1000	850	58...63	260...400
12X2H4A		1200	1000	60...65	280...400
20X2H4A		1400	1200	60...65	300...400
38XMЮA	Nitriding	1000	850	60	350

3.3. Spur Gears

Spur gears or straight-cut gears are the simplest type of gear. They consist of a cylinder or disk with the teeth projecting radially, and although they are not straight-sided in form, the edge of each tooth is straight and aligned parallel to the axis of rotation. These gears can be meshed together correctly if they are fitted to parallel shafts only.

A profile is one side of a tooth in a cross section between the outside circle and the root circle (Fig. 3.28). Usually a profile is the curve of intersection of a tooth surface and a plane or surface normal to the pitch surface, such as the transverse, normal, or axial plane.

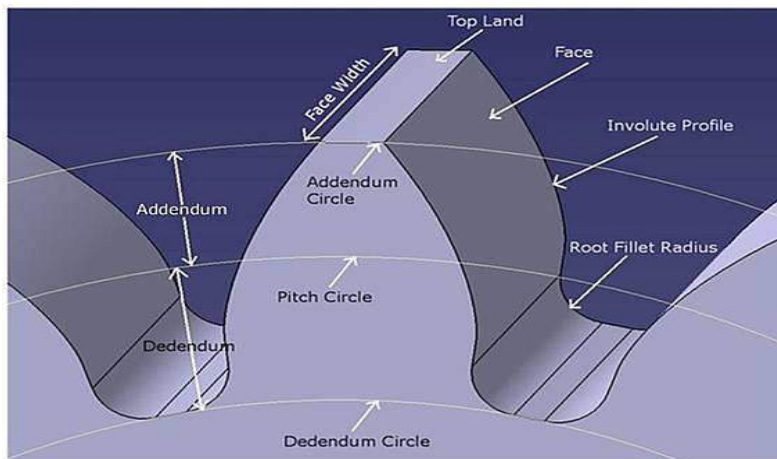


Fig. 3.28

An involute (also known as evolvent) is a curve obtained from given circle (base circle) by attaching an imaginary taut string to the base circle and tracing its free end as it is wound onto that given circle (Fig. 3.29).

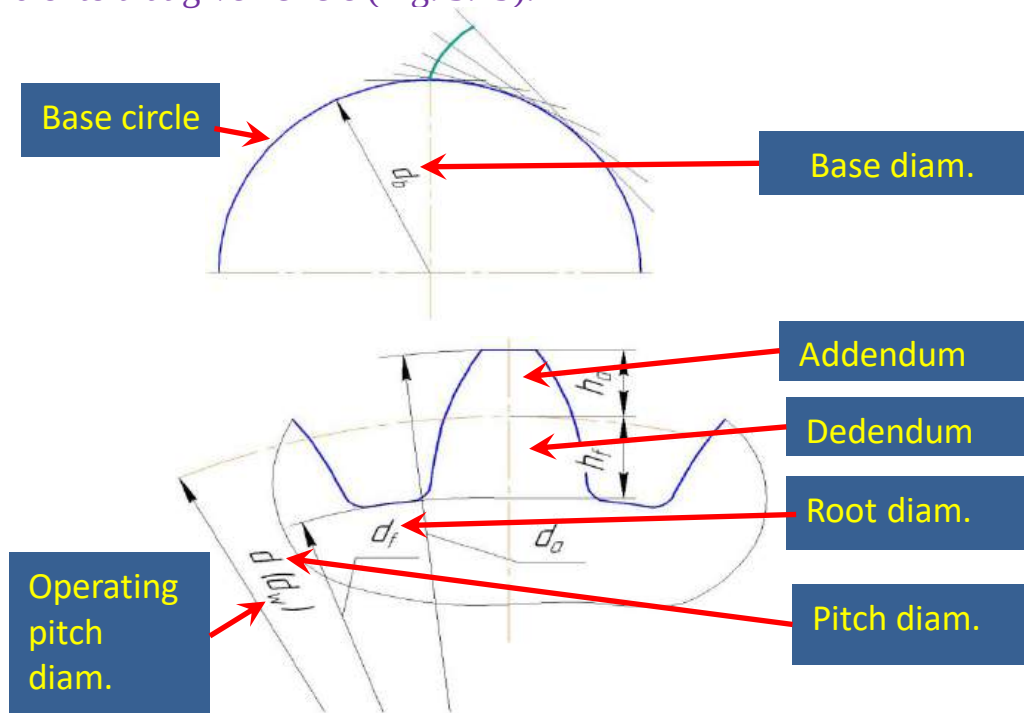


Fig. 3.29

Center distance (operating) is the shortest distance between non-intersecting axes. It is measured along the mutual perpendicular to the axes, called the line of centers (Fig. 3.30).

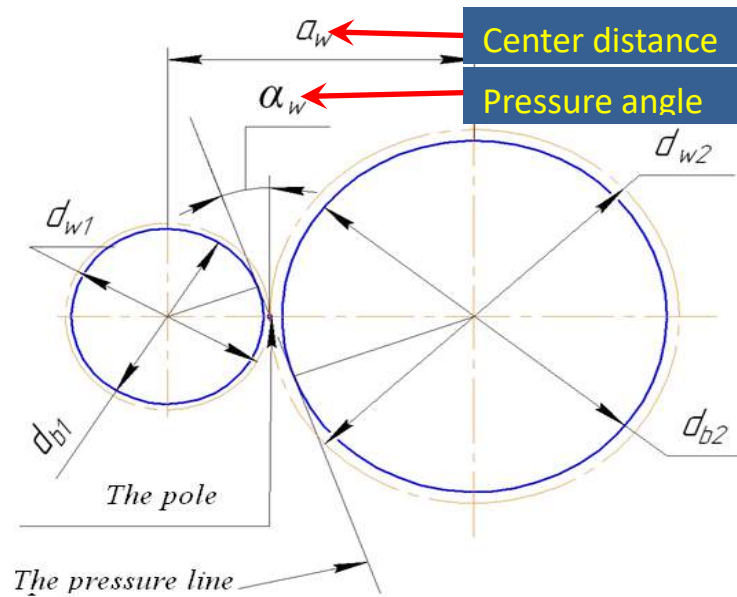


Fig. 3.30

Pressure angle is the angle at the pole between the pressure line, which is normal to the tooth surface and is the common tangent to base circles, and the plane perpendicular to the center distance. The pressure angle gives the direction of the normal to the tooth profile.

The main transfer geometrical parameters are represented in Table 3.7.

Table 3.7

Parameter	Formula
Module (standard)	m
Profile angle of basic rack (standard)	$\alpha = 18, 20, 25, 28^\circ$
Pitch diameter	$d = mz$
Operating pitch diameter	$d_w = d \cos \alpha / \cos \alpha_w$
Base diameter	$d_b = d \cos \alpha$
Addendum diameter	$d_a = d + 2m$
Root diameter	$d_f = d - 2,5m$
Center distance	$a_w = \frac{m(z_1 + z_2)}{2}$
Shift factor	$x_\Sigma = x_1 + x_2.$ If $x_\Sigma = 0$ $\alpha_{tw} = \alpha_t, a_w = a, d_w = d$

The module is one of the main parameters of a transfer. In straight spur gears, the module m means the size proportional to a step on a pitch circle in a plane, which is perpendicular to the longitudinal axis of a gear.

At the module selection the reasons are considered:

- at the same wheel diameter bend durability of teeth grows with module increasing;
- module reduction complicates technological process of heat treatment as it is difficult to create a hard surface and a viscous core (it is impossible to execute superficial machining because the tooth is calcinated as a whole);
- the less module the less metal losses.

Values of modules are standardized in a range from 0,05 to 100 mm. The interval 1...10 mm is widely used; it consists of two rows:

1 st row	1	1,25	1,5	2	2,5	3	4	5	6	8	10
2 nd row	1,125	1,375	1,75	2,25	2,75	3,5	4,5	5,5	7	9	

It would be better to prefer the first row to the second.

Modules from 2 to 8 mm are applied for heavy-load aviation tooth gearings, and about 40 % of gears have the module 3...5 mm.

Value of the module should be coordinated with face width of a gear, thus it is necessary to provide for a spur transfer $b/m \leq 15$.

3.3.1. The Acting Forces

The force between two meshing involute gear teeth is always directed along the pressure line. The spur gear's transmission force F_n , which is normal to the tooth surface, can be resolved into the tangential component, F_t , and the radial component, F_r :

$$F_t = F_n \cos \alpha_w = \frac{2T_1}{d_{w1}} = \frac{2T_2}{d_{w2}};$$

$$F_{r1} = F_{r2} = F_n \sin \alpha = F_t \operatorname{tg} \alpha_w.$$

The directions of the forces, acting on the gears, are shown in Fig. 3.31.

3.3.2. Allowable Contact Stresses

The allowable contact stresses, which are not causing dangerous contact fatigue damages, are calculated **for a pinion and a gear separately**:

$$[\sigma]_H = \frac{\sigma_{H \lim B} Z_N}{S_H} Z_R Z_v Z_L Z_X,$$

where $\sigma_{H \lim B}$ is the limit of contact endurance of the teeth surfaces, corresponding to the base number of cycles; S_H is the minimum safety factor; Z_N is the durability factor; Z_R is the factor considering influence of an initial roughness of interfaced surfaces; Z_v is the factor considering influence of peripheral speed; Z_L is the factor considering influence of

lubricant viscosity; Z_X is the factor considering the size of a gear; S_H is the minimum safety factor.

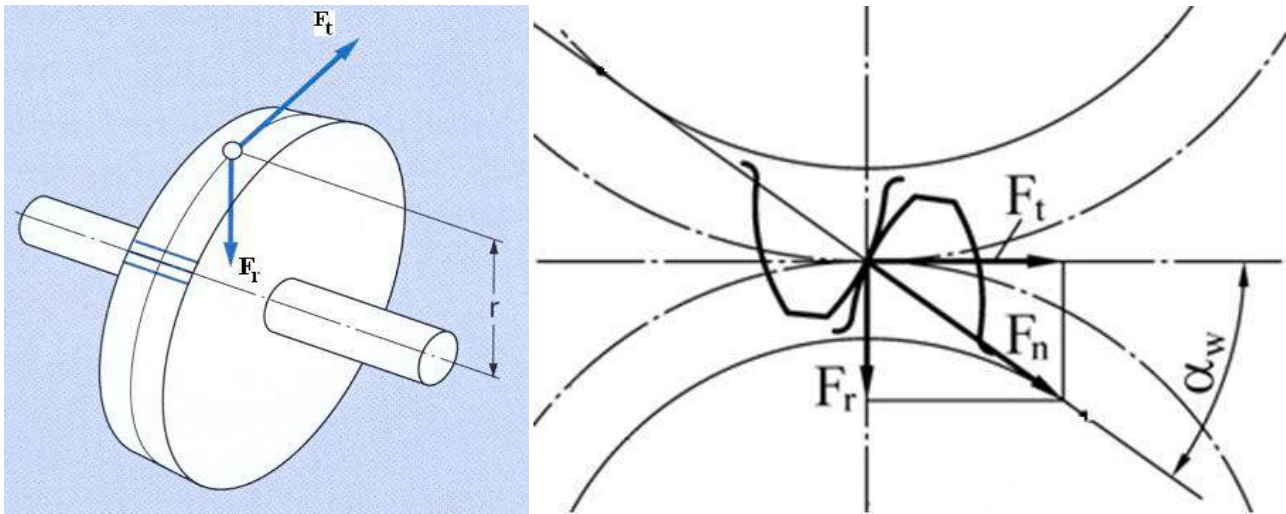


Fig. 3.31

Limit of contact endurance of teeth surfaces $\sigma_{H \lim B}$ is calculated according to formulae shown in Table 3.8.

Table 3.8

Type of thermal treatment	Mean surface hardness	Steel	σ_{ulimB} , MPa
Annealing, normalizing, tempering	≤ 350 HB	Carbon and alloy	$2 \text{ HB} + 70$
Volume hardening	38...50 HRC		$17 \text{ HRC} + 200$
Surface hardening	40...56 HRC		
Casehardening, nitriding	56...65 HRC	Alloy	23 HRC
Cyaniding	550...750 HV		1050

It is clearly, that the greater the hardness, the greater allowable stresses and load capacity. However, in some cases achievement of limiting hardness is very expensive. Besides, extremely high hardness is useful in combination with a viscous core, in other case teeth become fragile and the main cause of their damage is the break. Highly rigid teeth are much worse break-in.

So, hardness of working surfaces should be appointed with the account of economic and operational expediency, each time correlating possibilities of the enterprise and the tasks.

The minimum safety factor S_H for gears with homogeneous structure (after such kind of heat treatment as tempering, volume hardening) equals to 1,1; for gears with surface hardening – 1,2; for transfers which failure may cause grave consequences the value of minimum safety factor should be increased to 1.25...1.35.

The fatigue limit dependence on the number of load cycles is shown by durability factor Z_N . Experiments show that stress-cycle diagrams (Fig. 3.32) have two typical areas: a left area ($N_K < N_{HlimB}$ and exponent $m = 6$) that is sloping and a right area ($N_K > N_{HlimB}$ and $m = 20$) that is horizontal (for carbon steels) or that has a slight tilt towards the cycle axis (for alloy steels).

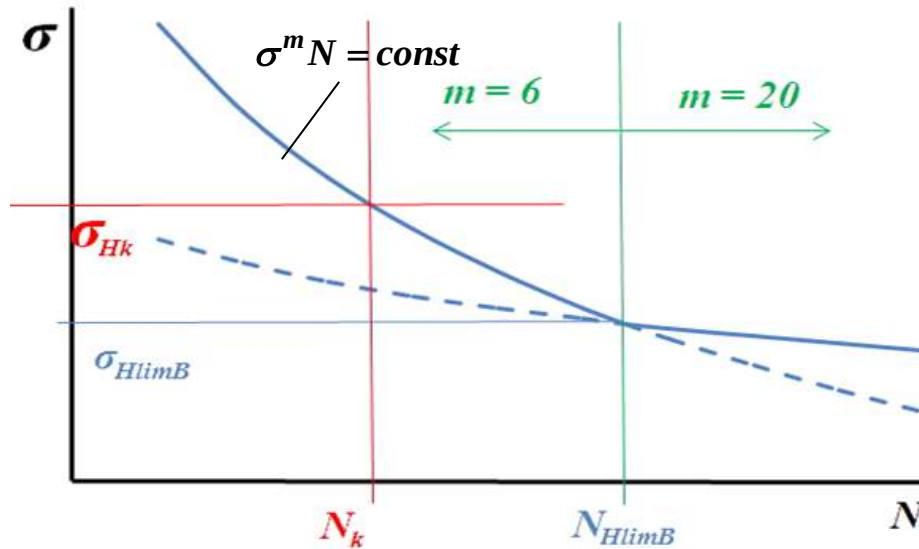


Fig. 3.32

Here the number of load cycles during service life $N_K = c60L_h n$, where n is the rotational speed, rpm; L_h is the service life, h; c is the number of meshing at one revolution (Fig. 3.33).

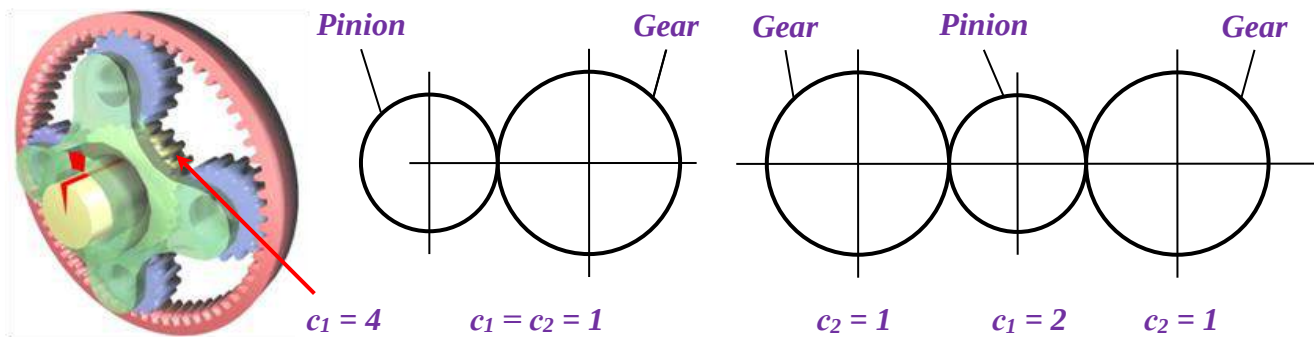


Fig. 3.33

If $N_K < N_{HlimB}$,

$$Z_N = \sqrt[m]{\frac{N_{HlimB}}{N_K}} > 1$$

and allowable stresses may be increased in Z_N times.

If $N_K > N_{HlimB} \rightarrow Z_N < 1$ and allowable stresses should be decreased.

The durability factor is limited: $0,75 < Z_N < 1,8$ in case of teeth surface hardening and $Z_N < 2,6$ in case of homogeneous structure of a material (after tempering, normalizing etc.).

The factor considering influence of an initial roughness of mated surfaces Z_R for both pinion and gear depends on the worst roughness (Table 3.9). The roughness of teeth working surfaces depends on degree of accuracy of a gear (Table 3.10).

Table 3.9

Roughness parameter R_a	Z_R
1,25...0,63	1
2,5...1,25	0,95
40...10	0,9

Table 3.10

Degree of accuracy	Roughness of			
	hole	working surfaces		top land
		spur	bevel	
6	R_a 0,63	R_a 0,63	R_a 0,63	R_a 0,63
7	R_a 1,25	R_a 1,25	R_a 0,63	R_a 1,25
8	R_a 2,5	R_a 2,5	R_a 1,25	R_a 2,5
9	—	R_a 5	R_a 2,5	R_z 20

The factor Z_v takes into account influence of the peripheral speed: the higher values of circumferential velocity the better conditions for the forming of a reliable oil layer between the contact teeth areas. Thus, the allowable stress increases (Fig. 3.34).

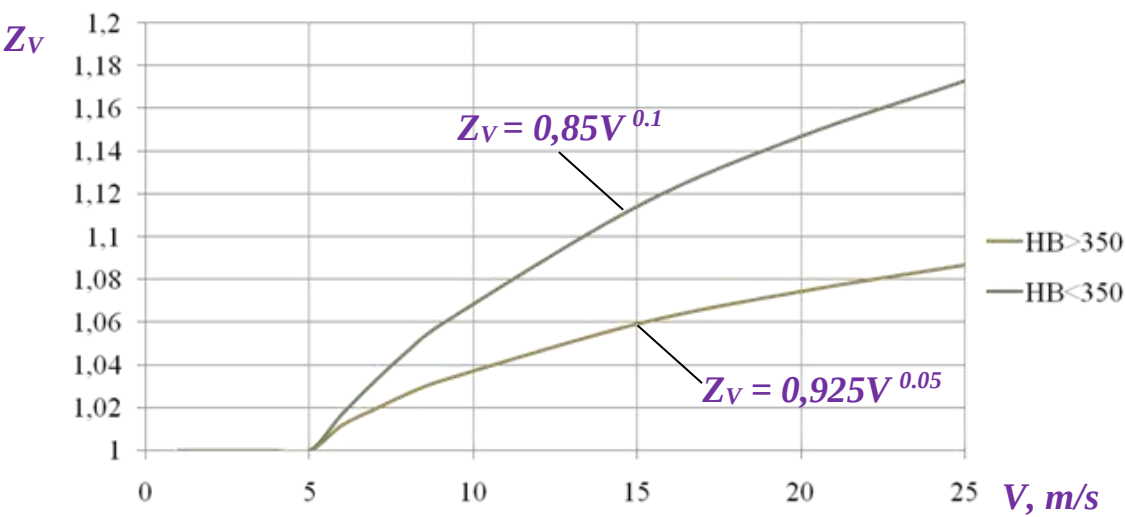


Fig. 3.34

For subsequent calculations, lower from a pinion and a gear allowable contact stresses should be accepted.

3.3.3. Allowable Bending Stresses

Allowable bending stresses are **separately** calculated for both pinion and gear as follows:

$$[\sigma]_F = \frac{\sigma_{F \lim B}}{S_F} Y_N Y_R Y_Z Y_A,$$

where Y_N is the durability factor; Y_R is the factor taking into account the influence of the roughness of fillet surface between the teeth; Y_Z is the factor considering a method of billet manufacturing; Y_A takes into account the influence if the double-sided load (reversing gear).

Bending endurance limit $\sigma_{F \lim B}$ and safety factor S_F corresponding to the base numbers of stress cycles $N_{F \lim B}$ is determined from the tables depending on a material and method of heat treatment (Table 3.11).

Table 3.11

Heat treatment	Steel	Surface hardness	$\sigma_{F \lim B}$, MPa	S_F
Case-hardening	20XH, 12XH2, 15XГHTA	57...63 HRC	950	1,55
	18XГТ, 30XГТ, 20X		820	1,55
Nitro-hardening	25XГМ	57...63 HRC	1000	1,55
	25XГТ, 35XГТ, 35X		750	1,55
Volume hardening	40X, 40XH, 40XΦA	45...55 HRC	580	1,7
Surface hardening (hardened layer repeats root contour)	40X, 35XM	48...58 HRC	680	1,7
	40XH, 40XH2MA		580	
Surface hardening (all tooth sections are hardened)	40X, 35XM		480	1,7
	40XH, 40XH2MA		580	
Nitriding	38X2Ю, 40X2HMA	700...950 HV	12HRC _{core} + +290	1,7

Durability factor Y_N

$$Y_N = \sqrt[q_F]{\frac{N_{F \lim B}}{N_{FE}}},$$

where $N_{F \lim B}$ is the base number of stress cycles; for steel gears $N_{F \lim B} = 4 \cdot 10^6$; root index $q_F = 6$ for gears with homogeneous structure of material and $q_F = 9$ for nitride and case-hardened gears with ungrinding root surface;

N_{FE} is the equivalent number of stress cycles (see above).

The durability factor is limited:

- in case of $q_F = 9$: $1 < Y_N < 2,5$;
- in case of $q_F = 6$: $1 < Y_N < 4$.

The factor taking into account the influence of the roughness of fillet surface between the teeth is:

- $Y_R = 1$ in the case of a surface roughness parameter $R_z \leq 40$;
- $Y_R = 1,05 \dots 1,2$ in the case of polishing.

The factor considering a method of billet production:

- for forging and forming $Y_Z = 1$,
- for rolled metal $Y_Z = 0,9$,
- for casting $Y_Z = 0,8$.

The factor taking into account the influence of the double-sided load Y_A :

- in the case of one-side load $Y_A = 1$,
- in the case of reversal load $Y_A = 0,75$ for gears with hardness more than 45 HRC.

3.3.4. Allowable Contact Stresses under Varying Loading Conditions

Aviation engines and units operate under varying loading conditions. A gearing load is sometimes characterized by the moment sequence diagram, where the acting torques (loads, powers) during the service life are represented. As it is seen from a real flying cycle (Fig. 3.35, where 1, 7 – ground idle power; 2 – take off; 3, 5 – hovering flight; 4 – cruising; 6 – landing), the maximal loads action is short-term and gear calculation according to these loads leads to the unjustified increasing of the gears size and weight. That is why it is very useful to replace actual loads with equivalent but lower ones.

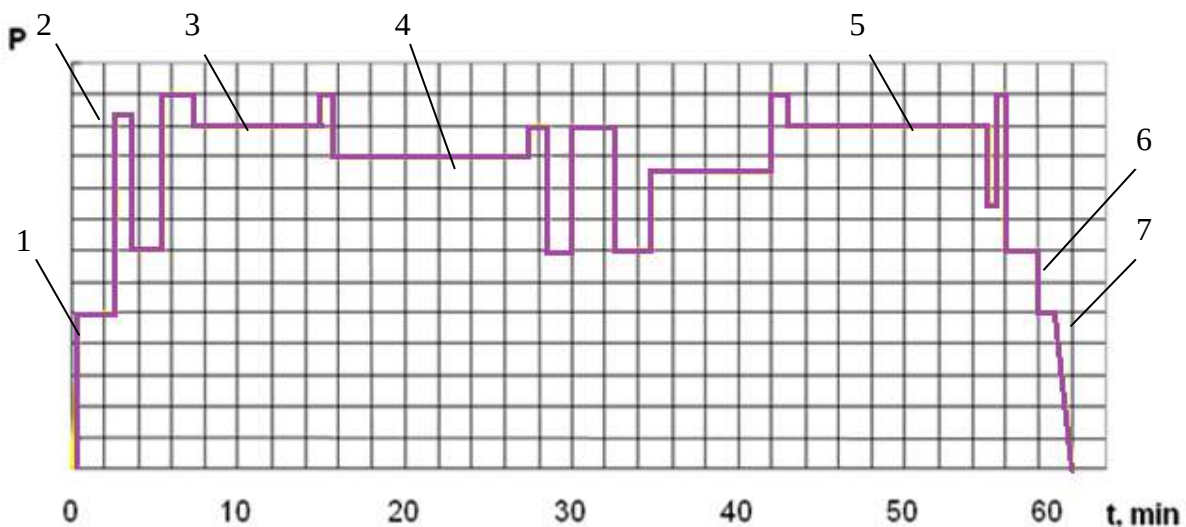


Fig. 3.35

Let us transfer a real flying cycle (see Fig. 3.35) to sequence diagram in decreasing order (Fig. 3.36).

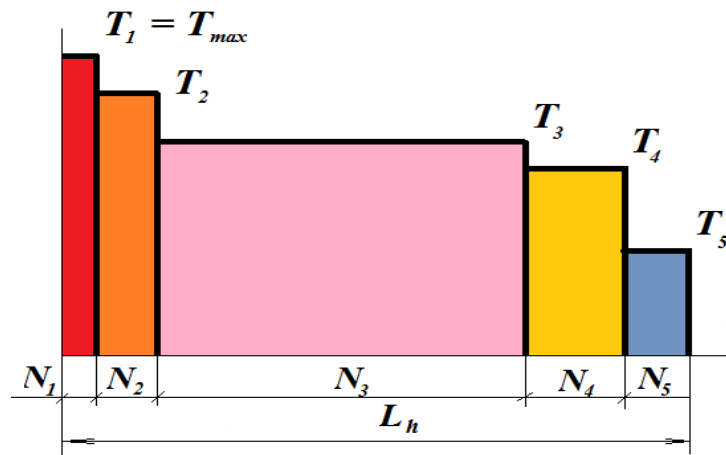


Fig. 3.36

Let us assume that:

- the maximum possible operating torques are short-term ($< 3\%$ of service life) and not taken under consideration in fatigue resistance calculations;
- fatigue damages are accumulated in proportion to the appropriate relative life (so called hypothesis of lineal summing up of fatigue damages);
- the actual varying loading conditions are changed into equivalent constant conditions when the elements reach the same degree of fatigue damage.

Thus, actual sequence diagram may be replaced with equivalent diagram where the equivalent torque T_E or equivalent number of cycles N_E is used (Fig. 3.37).

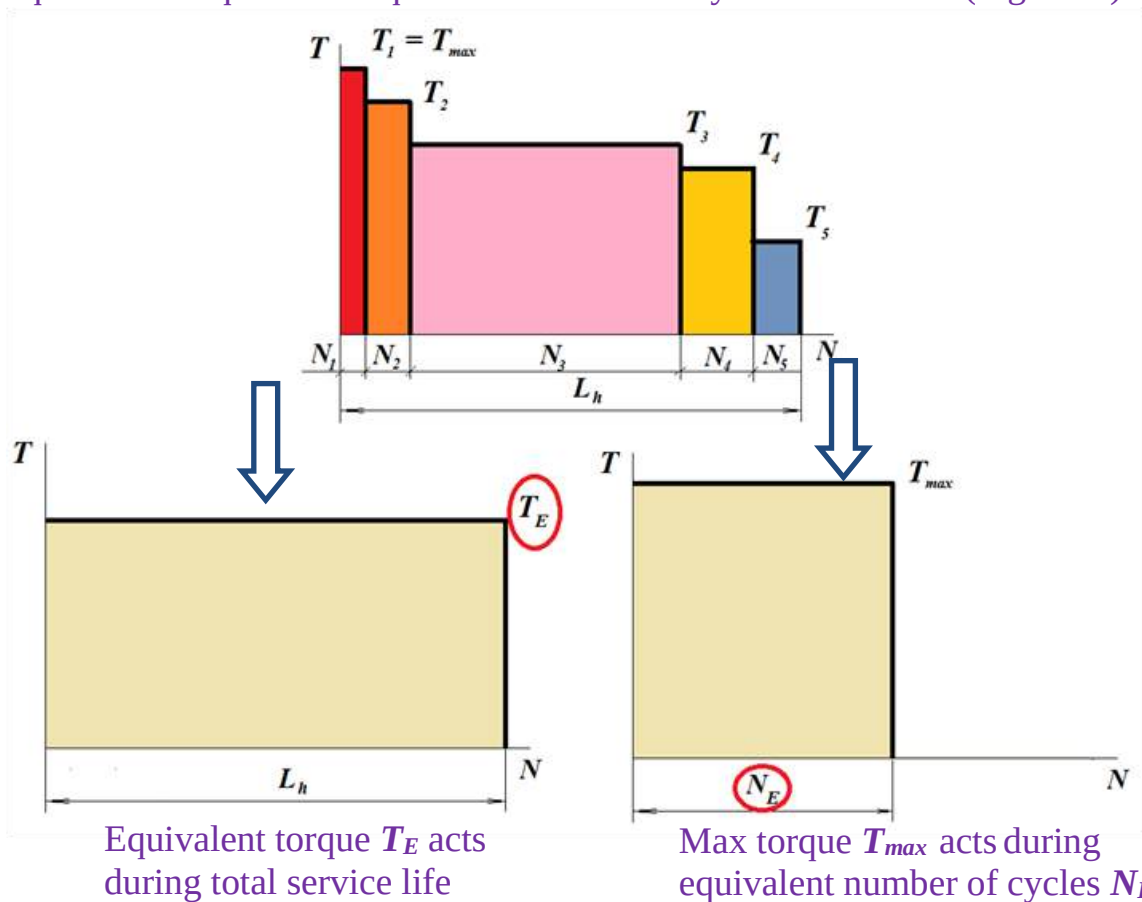


Fig. 3.37

The equivalent contact and bending numbers of cycles are determined as

$$N_{HE} = 60c \sum \left(\frac{T_k}{T_{max}} \right)^3 n_k L_{hk},$$

$$N_{FE} = 60c \sum \left(\frac{T_k}{T_{max}} \right)^q n_k L_{hk}.$$

Based on statistical analysis of different machines the wide variety of loads sequence diagrams can be brought to a few typical ones (Fig. 3.38).

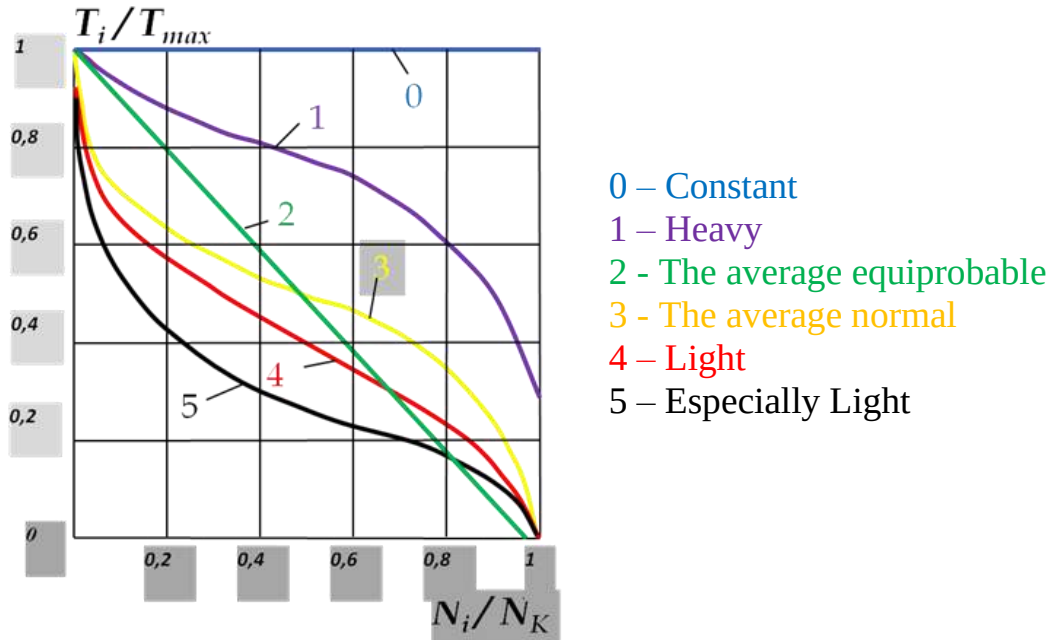


Fig. 3.38

The usage of typical rates considerably simplifies equivalent cycles' calculation:

$$N_{HE} = \mu_H N_K,$$

$$N_{FE} = \mu_F N_K,$$

where coefficients μ_H and μ_F are taken from the Table 3.12.

Table 3.12

Load type number	Loading condition	μ_H	Values of μ_F at	
			$q_F = 6$	$q_F = 9$
0	Constant	1,000	1,000	1,000
1	Heavy	0,500	0,300	0,200
2	The average equiprobable	0,250	0,143	0,100
3	The average normal	0,180	0,065	0,040
4	Light	0,125	0,038	0,016
5	Especially light	0,063	0,013	0,004

3.3.5. Design Calculation

The objectives of design calculation are a preliminary definition of the gears' dimensions and then check calculation should be done because many essential factors are not considering at the first stage.

At design calculation, basic geometrical parameters should be obtained. In particular, it may be approximate value of a pinion pitch diameter and then all the others could be found:

$$d_{w1} = K_d \sqrt[3]{\frac{T_1 K_{H\beta}}{\psi_{bd} [\sigma]_H^2} \frac{u_{12} + 1}{u_{12}}},$$

where K_d is the auxiliary factor (for straight teeth $K_d = 770$), T_1 is a pinion torque, Nm:

$$T_1 = 9950 \frac{P_1}{n_1},$$

P_1 is power on a pinion, kW; n_1 is the pinion rotational speed, rpm; $K_{H\beta}$ is the factor considering non-uniformity of load distribution along the contact lines (see subsection 3.2.5); u_{12} is the transfer ratio; $\psi_{bd} = b/d_{w1}$ is the relative face width of a tooth (see subsection 3.2.5 and Table 3.13).

Table 3.13

Bearings location relatively to the gear	Relative face width according to surface hardness	
	less 350 HV at least for one gear	more 350 HV for both gears
Symmetrical	0,8...1,4	0,4...0,9
Asymmetrical	0,6...1,2	0,3...0,6
Cantilever	0,3...0,4	0,2...0,3

The note: bigger values are accepted for constant loads.

A centre distance a_w , mm, can be also defined on a base of contact durability condition:

$$a_w = K_a (u + 1) \sqrt[3]{\frac{T_1 K_{H\beta}}{\psi_{ba} u_{12} [\sigma]_H^2}},$$

where K_a is the auxiliary factor (for straight teeth $K_a = 495$); ψ_{ba} is the relative width of a tooth, $\psi_{ba} = b/a_w$ and

$$\psi_{ba} = \frac{2\psi_{bd}}{u + 1}.$$

It is recommended to accept following values ψ_{ba} : 0,1; 0,125; 0,16; 0,2; 0,25; 0,315; 0,4; 0,5; 0,63; 0,8; 1,0; 1,25.

Then the module is obtained

$$m_p = \frac{d_{w1}}{z_1}$$

and rounded to the nearest standard value m_{st} .

The module can also be found from the bending strength condition:

$$m = K_m \sqrt[3]{\frac{T_1 K_{F\beta} Y_{FS1}}{\psi_{bd} z_1^2 [\sigma]_F}}; \quad K_m = 14,$$

where K_m is the auxiliary factor (for straight teeth $K_m = 14$); $K_{F\beta}$ is the factor considering non-uniformity of load distribution along the contact lines; Y_{FS} is the factor considering the tooth form and stress concentration (see below).

3.3.6. Check Calculation

It was noted in subsection 3.2.3 that the acting forces cause contact and bending stresses. Too many essential factors influence on a transfer operating ability and they are not considered at the first stage (design calculation). That is why check calculation should be done after preliminary gear design.

If a transfer geometry is known, the objective of the check calculation is to answer the question: could the gears with such dimensions operate at given load during required service life?

In any case the actual stresses should satisfy contact and bending strength conditions as follows: $\sigma_H \leq [\sigma]_H$, $\sigma_F \leq [\sigma]_F$.

Revise Calculation on Contact Durability

The strength calculation of the contact surfaces is based on limiting the maximum surface compressive stress according to the Hertz formula

$$\sigma_H = Z_E \sqrt{\frac{W_n}{\rho_{red}}} \leq [\sigma_H],$$

which deals with a classic contact problem. The solution of the contact problem is given in the theory of elasticity.

The following assumptions underlie this solution (Fig. 3.39):

1. The materials of the contacting bodies are homogeneous and isotropic.
2. The teeth contact is like contact of two cylinders. The cylinders radii equal to the teeth radii of curvature in the pole.
3. Cylinders are indefinite and load is even distributed along the contact line.
4. The area of contact is very small as compared with the surfaces of the contact bodies.
5. The lubrication and friction are absent.

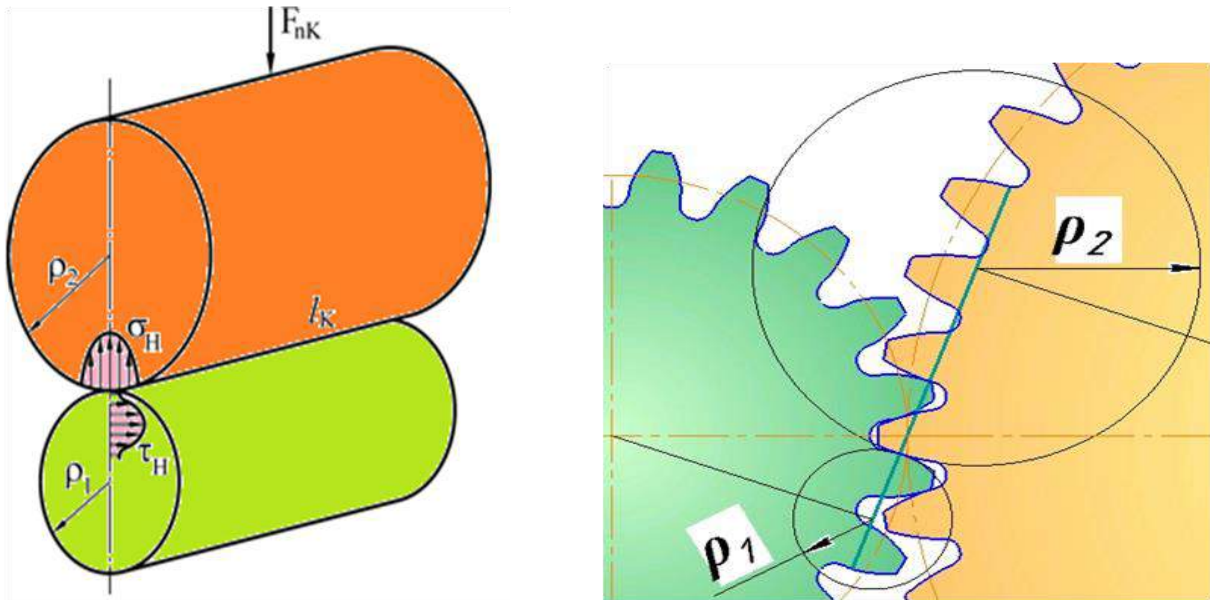


Fig. 3.39

To obtain contact stress, Hertz' formula is modified to apply it to the teeth of mating wheels. After replacing its values by the parameters of a toothed gear we can write

$$\sigma_H = Z_E Z_H Z_\varepsilon \sqrt{\frac{F_t K_H u + 1}{b_w d_w u}},$$

where Z_E is the factor considering mechanical properties of gear materials,

$$Z_E = \sqrt{\frac{1}{\pi \left(\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right)}}$$

Z_H is the factor considering the form of the teeth in pole of gearing:

$$Z_H = \frac{1}{\cos \alpha_t} \sqrt{\frac{2}{\tan \alpha_{tw}}},$$

Z_ε is the factor considering total length of contact lines. For straight teeth:

$$Z_\varepsilon = \sqrt{\frac{(4 - \varepsilon_\alpha)}{3}},$$

ε_α is a transverse contact ratio. For straight teeth without shift

$$\varepsilon_\alpha = \left[1,88 - 3,2 \left(\frac{1}{z_1} + \frac{1}{z_2} \right) \right].$$

Revise Calculation on Bending Durability

The bending endurance is necessary for prevention of a fatigue break of teeth. It is defined for each of gears by comparison of local bending stress operating in dangerous section and allowable stress.

Assumptions (Fig. 3.40):

1. A tooth is considered as a cantilever beam, loaded by the normal force evenly distributed along the tooth.

2. The dangerous section passes through the root of tooth and straight lines held an angle 30° to the axis of the tooth.

3. For exact wheels (6th degree of accuracy and higher) force is applied at the top of addendum.

The bending fatigue strength is calculated in point A (Figs 3.40, 3.41) as it locates on the stretched fibres and the fatigue limit in tension is less than in compression. Under static load more dangerous is point B, because stresses are greater at this point.

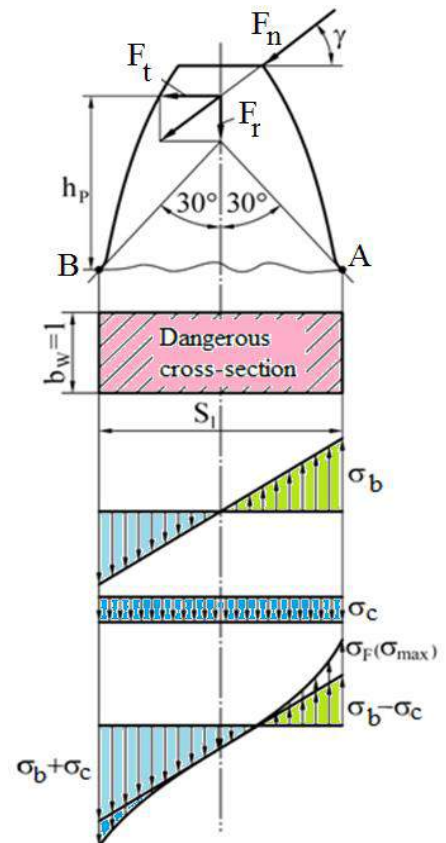


Fig. 3.40

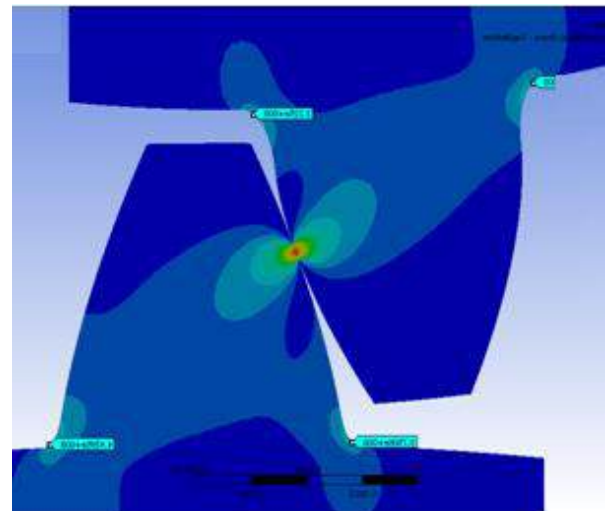
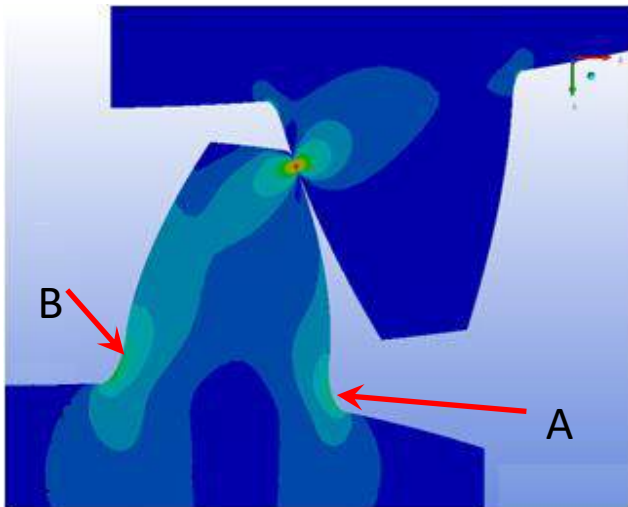


Fig. 3.41

The local bending stresses:

$$\sigma_F = \frac{F_t K_F Y_{FS}}{b_w m} \leq [\sigma]_F,$$

where Y_{FS} is a factor considering the tooth form and stress concentration. It may be either calculated

$$Y_{FS(1,2)} = 3,47 + 13,2/z_{(1,2)}$$

or taken from the tables.

3.4. Specific Aspects of Helical Gear Calculations

If noise is the main design criteria (e.g., car gearbox, machine tools etc.) and you can compromise on power, you should use the helical gears (Fig. 3.42). Helical gears are normally used for high speed, high power mechanical systems.

Helical wheels are more expensive than straight-tooth wheels of the same accuracy. Therefore, it is pointless to take an excessively small inclination, because at a small β the smoothness of contact does not noticeably increase. On the other hand, an excessive helix angle will generate a higher axial force and larger dimensions of the supports. Moreover, sometimes additional thrust bearing is needed so the size of the gear is increased.

In practice, therefore, the helix angle of helical teeth is taken from 8 to 20°. Herringbone wheels have a helix angle of $\beta < 40^\circ$. In such case the axial forces on both sides of the herringbone wheel are mutually equalized (if the gears have been manufactured in a proper way) and are not transmitted to the bearings.

The principal methods employed in designing straight-tooth gears are also valid for helical gears. Below we'll consider the specific features of design which are due only to the geometry of helical wheels.



Fig. 3.42

Advantages

- Higher load capacity comparing to straight teeth gear
- Silent and smooth operation
- Smaller size and weight

Disadvantages

- Axial forces increases with the helix angle increasing
- More complicated manufacturing

3.4.1. General Dependences of Non-Shifted Gears

The proportions of helical wheels are found from the same formulae as for straight-tooth wheels (Table 3.14) if we assume that their geometrical parameters refer to circumferential cross-section, but the standard parameters of the mating helical wheel elements are considered as applied to the normal cross-section (Fig. 3.43).

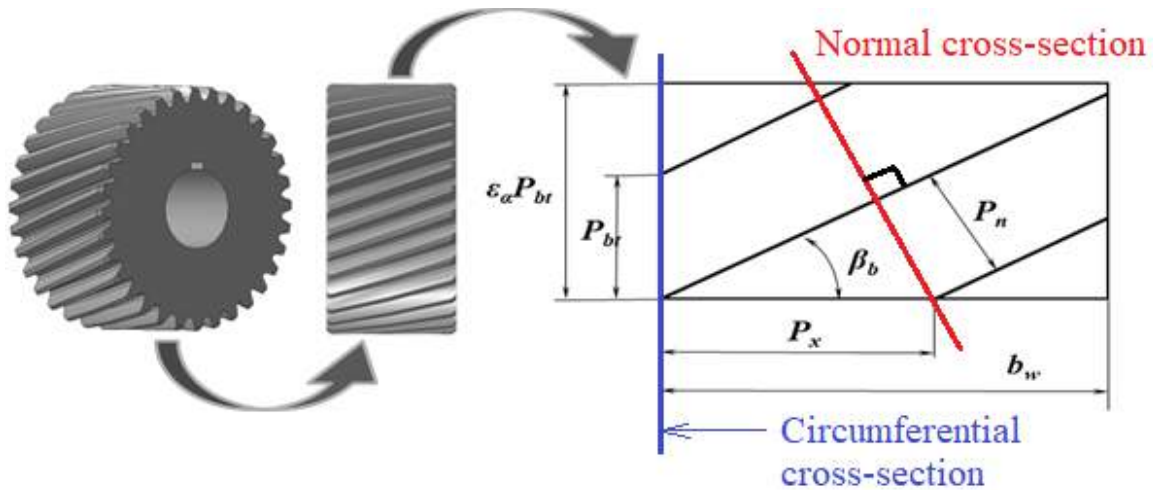


Fig. 3.43

Table 3.14

Parameters	Formulae
Profile angle or basic rack in transverse cross-section	$\alpha_t = \arctan\left(\frac{\tan\alpha}{\cos\beta}\right)$
Pressure angle in normal cross-section	$\alpha_{nW} = \arctan(\tan\alpha_t \cos\beta)$
Pitch helix angle	7..20° in helical gears 28..40° in herring bone gears
Base helix angle	$\beta_b = \arcsin(\sin\beta \cos\alpha)$
Module (m_n – normal, regularly, due to technological features, standard; m_t – transverse module)	$m_t = \frac{d}{z} = \frac{m_n}{\cos\beta}$
Pitch centre distance	$a = 0.5m \frac{(z_2 \pm z_1)}{\cos\beta}$
Pitch diameter	$d = m_n z / \cos\beta$
Transverse contact ratio	$\varepsilon_\alpha = \left[1.88 - 3.2 \left(\frac{1}{z_1} + \frac{1}{z_2}\right)\right] \cos\beta$
Axial contact ratio	$\varepsilon_\beta = \frac{b_w}{p_x} = \frac{b_w \sin\beta}{p_n} = \frac{b_w}{\pi m_n} \sin\beta$
Overall bridging coefficient	$\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$
Equivalent teeth number	$z_V = \frac{z}{\cos^3\beta}$

3.4.2. Active Forces

The full load on a tooth F_n (Fig. 3.44) acts normally to the teeth surfaces in the normal plane. Since in the plane of action the teeth are inclined at an angle β , the vector F_n is likewise inclined, relative to the end-face, at an angle β (the effect of friction is neglected). Let us resolve this force into three components: the peripheral force F_t , the radial force F_r and the axial force F_a .

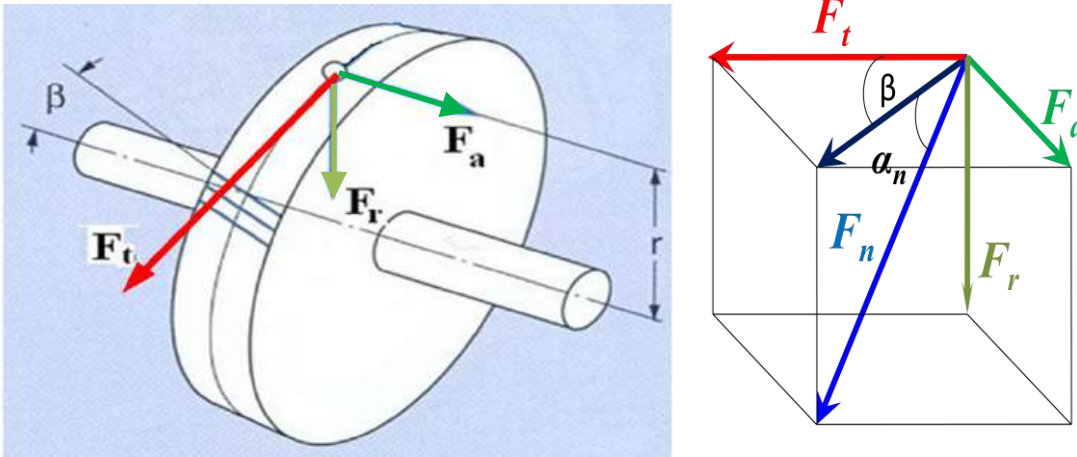


Fig. 3.44

The magnitude and direction of the peripheral and radial forces are determined in the same way as in the case of a straight-tooth gear. The axial force will be $F_a = F_t \tan \beta$, where β is the helix angle on the pitch cylinder.

The direction of the axial force depends on the directions of rotation and inclination of the teeth. The forces when the teeth are inclined to the right on a driving wheel rotating counterclockwise are shown in Fig. 3.45. When the direction of the teeth inclination or that of rotation is changed, the direction of the axial force is reversed.

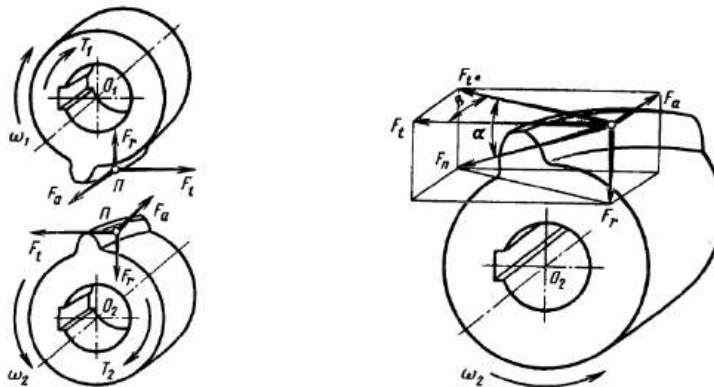


Fig. 3.45

In a helical gear, even if the wheels between the bearings are arranged symmetrically, the loads carried by the bearings will be different, because the reaction from the moment $F_a r$ (see Fig. 3.44) on one bearing is directed in the same way as the

reaction due to the radial force F_r while on the second bearing it has an opposite direction. The axial force should be taken by the bearing carrying the smaller radial load.

The tangential, axial and normal forces can be found from the relations as follows:

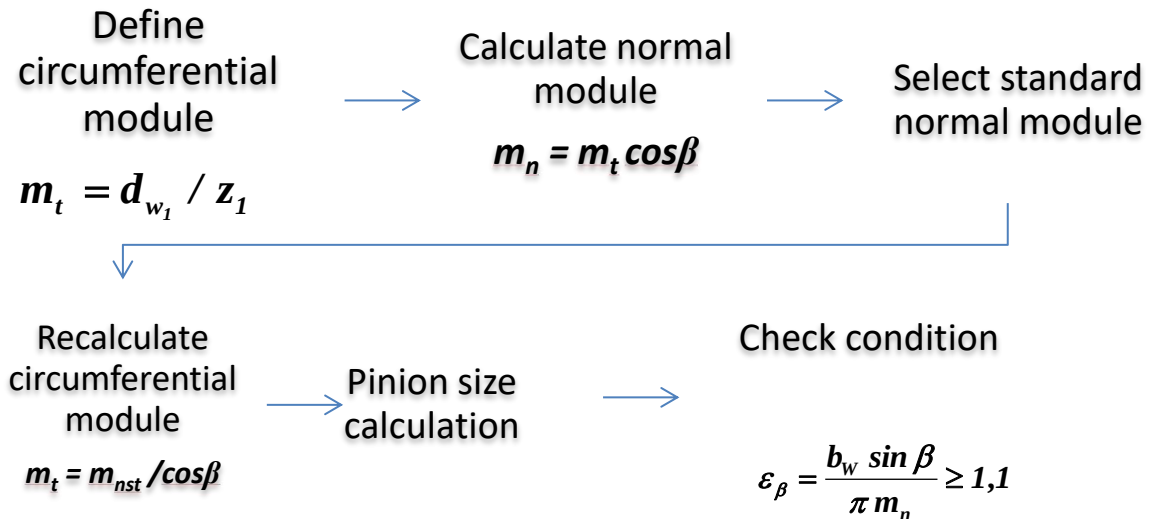
$$F_t = \frac{2T_1}{d_{w1}} = \frac{2T_2}{d_{w2}}; F_r = F \operatorname{tg} \alpha_w = \frac{F_t \operatorname{tg} \alpha_w}{\cos \beta}; F_a = F_t \operatorname{tg} \beta.$$

These forces are taken as initial for calculating the strength of helical and herringbone wheels.

3.4.3. Design Calculation

To calculate the pinion pitch diameter d_{w1} , the center distance a_w or module m the same formulae are used (see subsection 3.3.5), the only difference is auxiliary factor: $K_d = 675$, $K_a = 430$ and $K_m = 11,2$.

It is recommended design calculation sequence as follows:



3.4.4. Check Calculation

Since helical and herringbone gears operate more smoothly than straight-tooth gears and values of the dynamic load are approximately twice smaller.

The equations in subsection 3.3.6 must be modified as follows:

– the coefficient taking into account the shape of the teeth contact surfaces in the pole:

$$Z_H = \frac{1}{\cos \alpha_t} \sqrt{\frac{2 \cos \beta}{\operatorname{tg} \alpha_{tw}}};$$

– the coefficient taking into account the total length of the contact lines when $\varepsilon_\beta \geq 1,1$

$$Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}};$$

– the transverse ratio for uncorrected helix gears

$$\varepsilon_\alpha = \left[1.88 - 3.2 \left(\frac{1}{z_1} + \frac{1}{z_2} \right) \right] \cos \beta;$$

– acting bending stress:

$$\sigma_F = \frac{F_t K_F Y_{FS}}{b_W m} Y_\beta Y_\varepsilon \leq [\sigma]_F,$$

where

$$Y_{FS} = 3.47 + \frac{13.2}{z_v} - 29.7 \frac{x}{z_v} + 0.092 x^2; \quad Z_v = \frac{z}{\cos^3 \beta};$$

the factor considering a tooth helix angle:

$$Y_\beta = 1 - \varepsilon_\beta \frac{\beta}{120} > 0.7,$$

and factor considering teeth overlapping:

$$Y_\varepsilon = \frac{1}{\varepsilon_\alpha}.$$

3.5. Specific Aspects of Bevel Gear Calculations

A bevel gear has teeth cut into a conical surface. Two such gears mesh together to transmit power between two shafts with the intersecting axes. The shafts connecting is generally at 90° (Fig. 3.46, a) but the angle between the shafts can be any except zero or 180 degrees. Sometimes one shaft drives a bevel gear which is mounted on a through shaft resulting in two output shafts (Fig. 3.46, b).

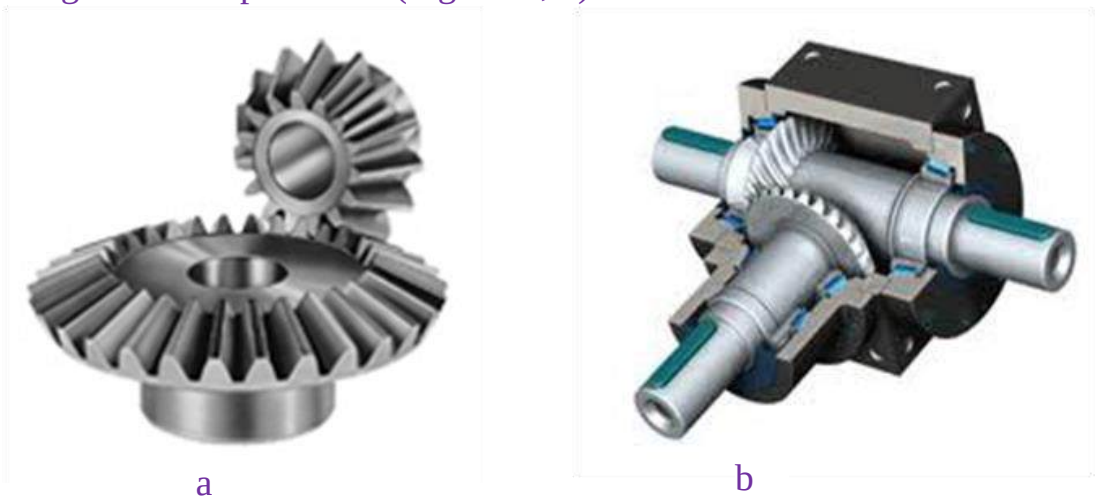


Fig. 3.46

Straight bevel gears have been traditionally used on transport aircraft in high-lift control systems (see gearboxes 4 and 6 in Fig. 3.2, a), in helicopter transmissions (Fig. 3.47) and in automobile gear differential (Fig. 3.48).

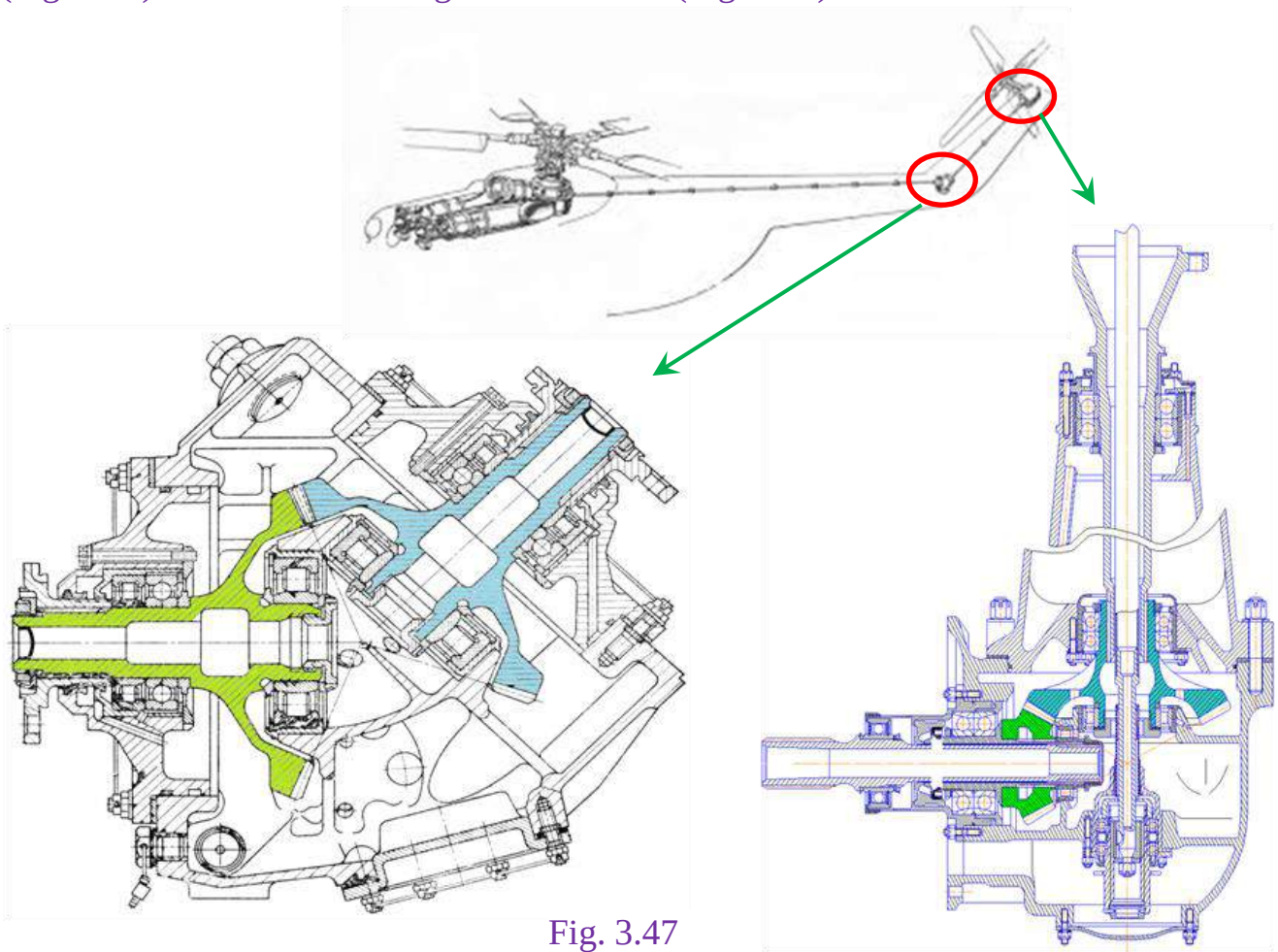


Fig. 3.47

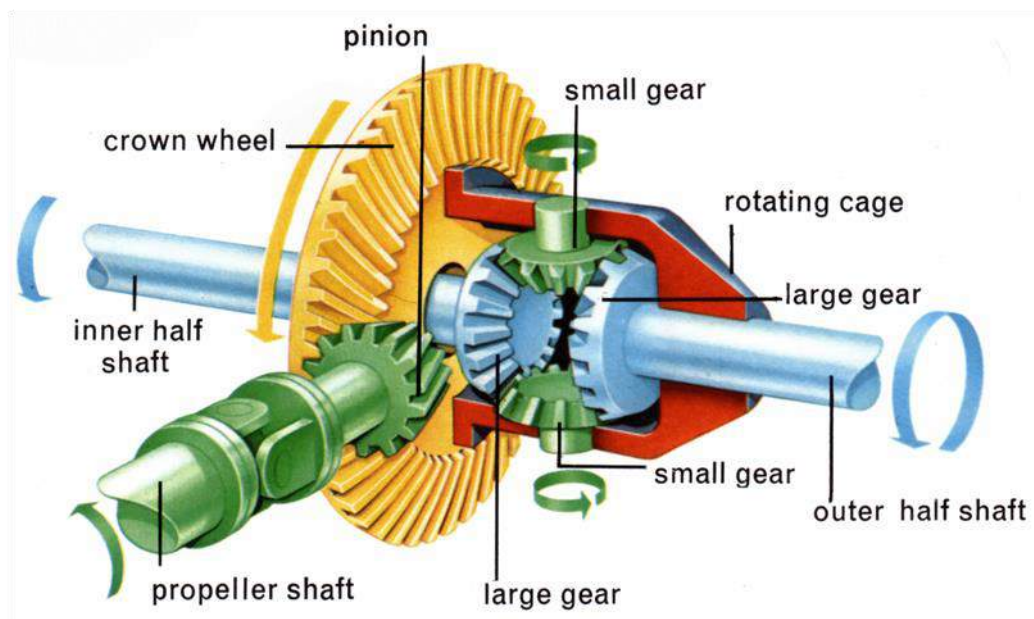


Fig. 3.48

Generally, this power is transferred at a 1:1 ratio with relation to speed. However, these gear drives are capable of increasing or reducing speed depending on the gear ratio used. Bevel gears with equal numbers of teeth and shaft axes at 90 degrees are called mitre gears.

The basic methods for designing spur gears can also be applied to bevel gears. Below we'll consider only the ways of deriving design formulae and their specific features due to the geometry of bevel wheels.

Advantages	Disadvantages
– Ability to change transmission direction	– Axial forces – Large size and weight – Must be precisely mounted – More complicated production

3.5.1. Types of Teeth

They are:

Straight tooth These gears have a conical pitch surface and straight teeth tapering towards an apex. Straight bevel drives are suitable for lower input speeds.	
Tangential (Oblique) tooth They have higher load capacity and operate smoother than straight tooth gears but less than spiral ones. They are rarely used.	
Spiral tooth Spiral bevel drives are suitable for higher input speeds due to their quietness and smoother gear meshing action. They are also generally able to handle higher power but are more expensive than straight bevels.	

3.5.2. The Main Dimensions

They are shown in Fig. 3.49 and named as follows:

Diameters	addendum	d_{ae}
	pitch	d_e
	root	d_{fe}
	pitch in mean cross-section	d_m
Distances	cone	R_e
	mean cone	R_m
	face width	b
Angles	the pitch cone	δ
	interaxial	Σ

Note: subscript “e” means that this parameter deals with outside gear face.

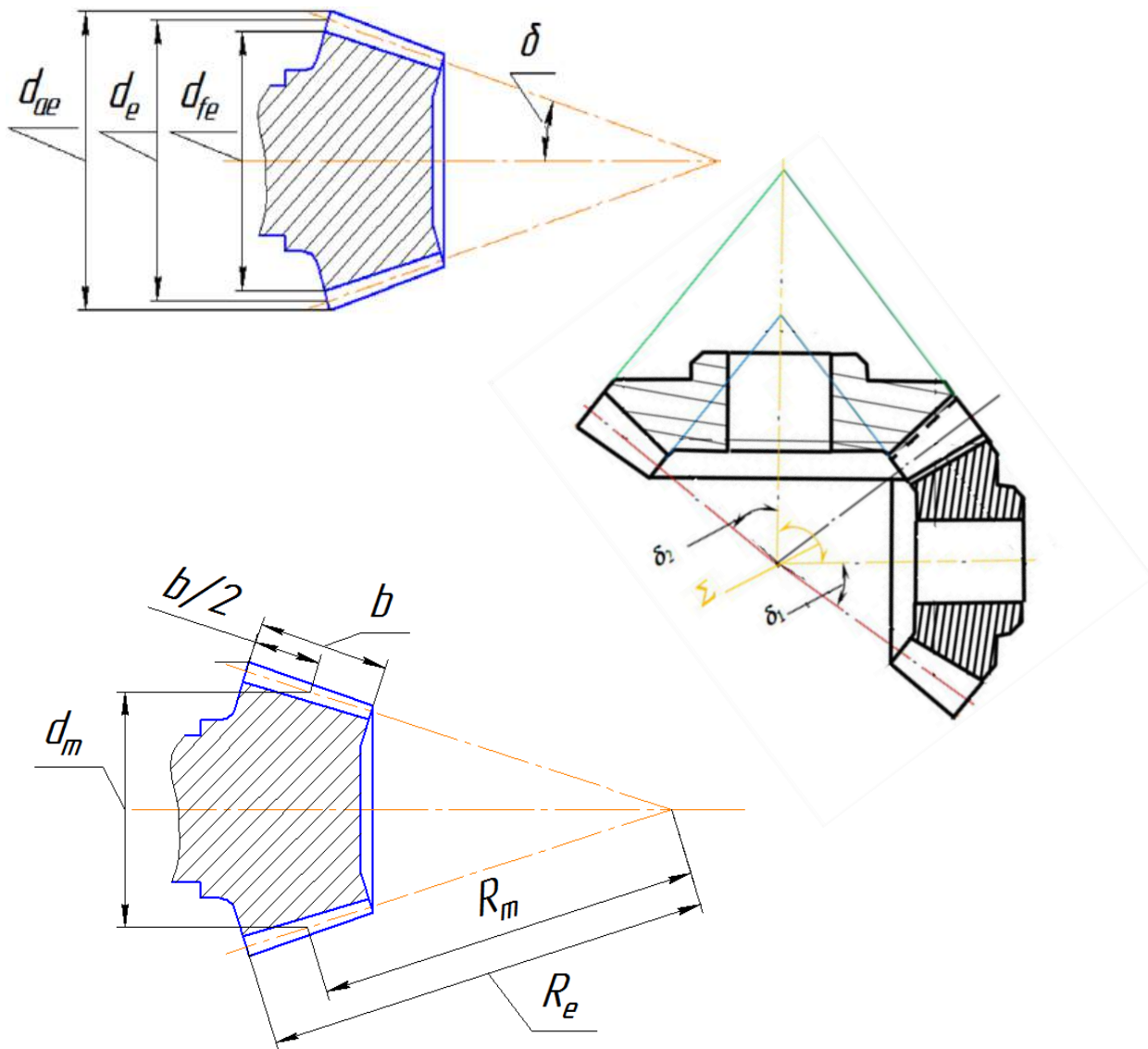


Fig. 3.49

3.5.3. General Equations of Non-shifted Gears

They are:

Parameters	Formulae
Pitch diameter of a back face	$d_e = m_e z$
Cone distance	$R_e = \frac{d_e \sqrt{u^2 + 1 + 2u \cos \Sigma}}{2 \sin \Sigma}$
Mean cone distance	$R_m = R_e - 0.5b$
Mean circular module	$m_m = \frac{m_e R_m}{R_e}$
Pitch cone angle	$\operatorname{tg} \delta_1 = \frac{\sin \Sigma}{(u + \cos \Sigma)}$
Equivalent straight-spur wheel teeth number	$z_E = \frac{z}{\cos \delta}$
Addendum diameter	$d_{ae} = d_e + 2m \cos \delta$
Outer root diameter on a back face	$d_{fe} = d_e - 2.4m \cos \delta$

3.5.4. Tooth Shapes

Geometry of a bevel gearing consists of a pair of truncated cones (foot and head cones) and a pitch cone between them. According to positions of the foot, pitch and head cones three types of gears are recognized:

Type I (Fig. 3.50, *a*) – proportionally decreasing tooth, pitch and foot conical surfaces have a common apex. It is the only suitable shape for straight teeth.

Type II (Fig. 3.50, *b*) – tooth thickness by the pitch cone grows proportionally with the distance to the apex, the root and the pitch cones do not coincide, the foot cone apex is shifted to create a constant width of the teeth gap, so the width of the root bottom is equal. It is used in spiral and tangential teeth.

Type III (Fig. 3.50, *c*) – Constant height of teeth. Surfaces of all cones are parallel.

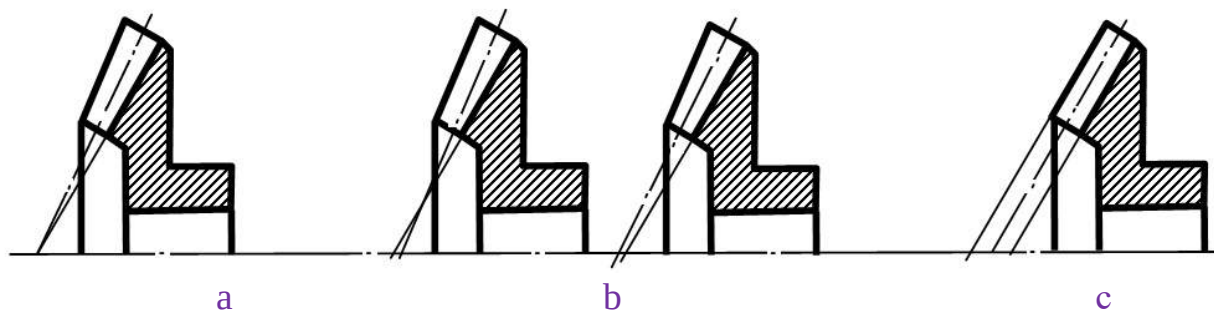


Fig. 3.50

3.5.5. Acting Forces

It is assumed that the acting force is applied at the mean pitch diameter. Axial force is usually directed opposite to the cone apex (Fig. 3.51).

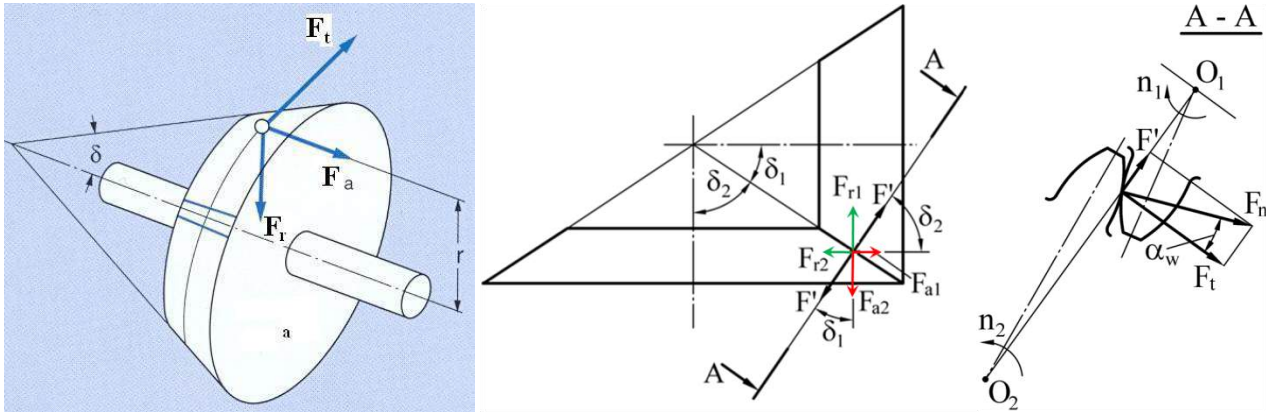


Fig. 3.51

The acting forces are obtained as follows (see Fig. 3.51):

- tangential $F_t = \frac{2T}{d_m}$;
- radial $F_r = F_t \operatorname{tg} \alpha_w \cos \delta$;
- axial $F_a = F_t \operatorname{tg} \alpha_w \sin \delta$.

In the case of interaxial angle $\Sigma = 90^\circ$: $F_{a1} = F_{r2}$; $F_{r1} = F_{a2}$.

3.5.6. Design Calculation

1. The pinion pitch diameter on the outside face is obtained as:

$$d_{e1} = K_d^3 \sqrt{\frac{T_1 K_{H\beta} \sin \Sigma}{\theta_H (1 - k_{be}) k_{be} u [\sigma]_H^2}},$$

where the auxiliary factor $K_d = 1013$; k_{be} is the relative face width:

$$k_{be} = \frac{b}{R_e} \leq 0,3;$$

θ_H is the factor considering the form of the tooth. For straight wheels it is equals 1, for spiral teeth transfers – 1,5.

2. Tangential module on the outside face is:

$$m_{te} = 14^3 \sqrt{\frac{T_1 K_{Fv} K_{F\beta} Y_{FS1}}{\psi_{bm} z_1 [\sigma]_{F1} \theta_F}},$$

where the relative face width $\psi_{bm} = b/m < 10$.

At design calculation factor $K_{H\beta}$ depending on $k_{be}u / (2 - k_{be})$ could be assumed from Fig. 3.52, and K_{Hv} depending on degree of accuracy, hardness of surfaces – from Table 3.15 (the top values – for straight teeth, bottom – for helical one).

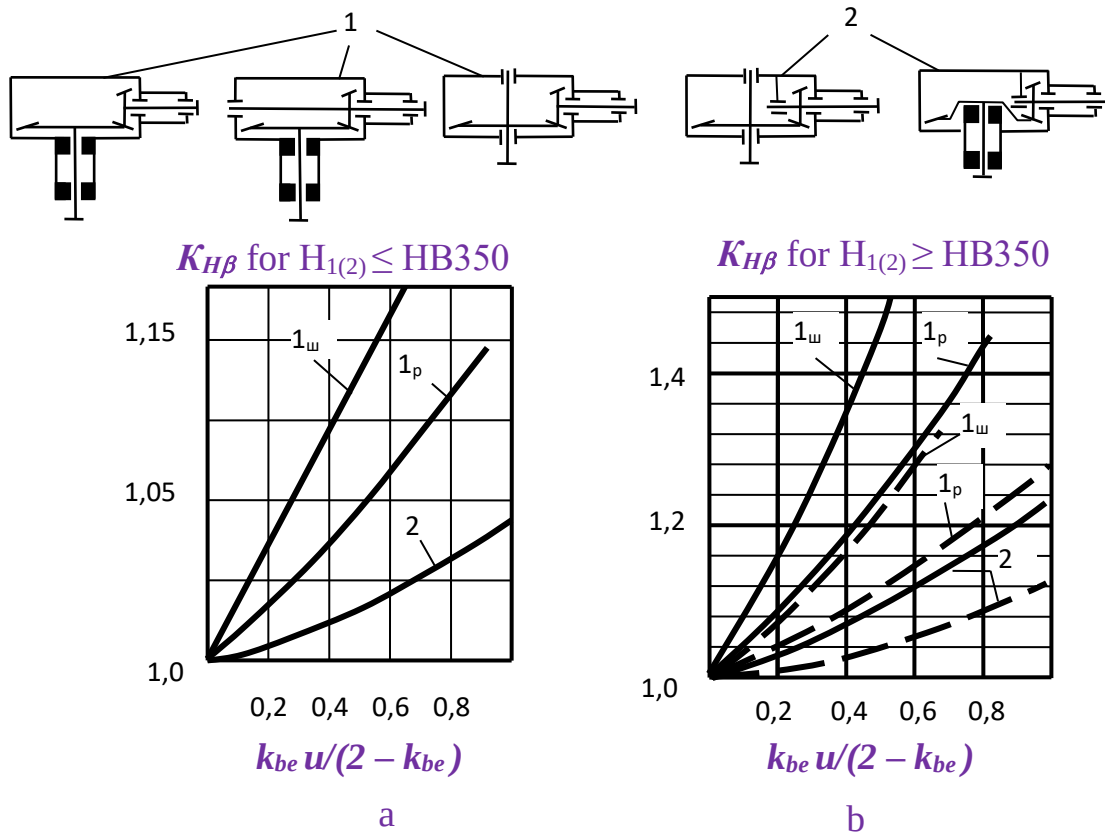


Fig. 3.52

Table 3.15

Degree of accuracy	Surface hardness	K_{Hv}				
		Peripheral speed v , m/s				
		1	5	10	15	20
6	H_1 and $H_2 > 350$ HB	1,02	1,10	1,20	1,30	1,40
		1,01	1,06	1,08	1,12	1,16
	H_1 or $H_2 \leq 350$ HB	1,03	1,16	1,32	1,48	1,64
		1,01	1,06	1,13	1,19	1,26
7	H_1 and $H_2 > 350$ HB	1,02	1,12	1,25	1,37	1,5
		1,01	1,05	1,10	1,15	1,20
	H_1 or $H_2 \leq 350$ HB	1,04	1,20	1,40	1,60	1,80
		1,02	1,08	1,16	1,24	1,32
8	H_1 and $H_2 > 350$ HB	1,03	1,15	1,30	1,45	1,60
		1,01	1,06	1,12	1,18	1,24
	H_1 or $H_2 \leq 350$ HB	1,05	1,24	1,48	1,72	1,96
		1,02	1,10	1,19	1,29	1,38

It is recommended design calculation sequence as follows:

- define circumferential module: $m_{te} = d_{e1}/z_1$;
- calculate number of teeth of flat gear: $z_c = \sqrt{z_1^2 + z_2^2}$;
- calculate cone distance: $R_e = 0,5 m_{te} z_c$;
- calculate pinion size: $b = k_{be} R_e$;
- check conditions: $k_{be} \leq 0,3$; $b/m_{te} \leq 10$;
- calculate other geometrical parameters.

3.5.7. Check Calculations

The design of a bevel gear is based on the assumption that its load-carrying capacity equals that of an equivalent spur gear when the width of its wheels is the same as that of the bevel wheels.

It is assumed that tangential force is applied at the mean pitch diameter d_m

$$d_{m1} = d_{e1} (1 - 0,5k_{be})$$

and tangential force:

$$F_t = \frac{2000T_1}{d_{m1}}.$$

Acting contact stress is

$$\sigma_H = 32240 \sqrt{\frac{T_1 K_{H\beta} K_{Hv} K_{H\alpha} \sin \Sigma}{d_{e1}^3 \theta_H k_{be} (1 - k_{br}) u}}.$$

Local bending stresses for straight bevel gears are represented in the formula:

$$\sigma_{F1,2} = \frac{2280 T_{1,2} K_{F\beta} K_{Fv} Y_{FS1,2}}{b d_{e1,2} m_e (1 - k_{be})}.$$

3.6. Worm Transfers

3.6.1. The base characteristics

Worm transfer is the transmission with axes that do not intersect (Fig. 3.53). Worm gears are used to transmit small and medium capacity, usually up to 50 kW.

The worm gear operates on the same principle as a screw. The gear is made of two parts: the worm wheel and the worm screw are connected to the shafts.

The worm gear looks much like a typical spur gear; it is round and covered in teeth. The worm is a screw without a head.



Fig. 3.53

Advantages
– Large ratio in a single pair: $i = 8 \dots 80$ – Smooth and quiet operation, low vibration – High kinematic accuracy

Disadvantages
– The relatively low efficiency – Increased wear and a tendency to jam – The need for non-ferrous metals – High precision mounting

3.6.2. Worm tooth profiles

Worms are similar to the Archimedean screw. They have a straight profile in axial section. In the front section of the coils are outlined Archimedean spiral (Fig. 3.54, *a*; front section – Fig. 3.55, *a*).

These worms are easy to make, if there is no need for them to grinding. Therefore, they are used in low-speed, not heavily loaded gears. **Archimedean** worms are marked ZA.

Convolute worms (Fig. 3.54, *b*; front section – Fig. 3.55, *b*) have a straight profile in the normal section (B-B), and in the front - delineated extended or shortened involute. They are more technologically than the Archimedean worms, they are easier to cut and grind cutters. Convolute worms are marked as ZN.

Involute worms (Fig. 3.54, *c*; front section – Fig. 3.55, *c*) are the helical gears with a small number of teeth, which is equal to a worm number of starts, and very large angle of the teeth inclination. They have a straight profile in the cross-section parallel to the axis. Profile outlined in the front section is an involute. Involute worms are marked as ZI.

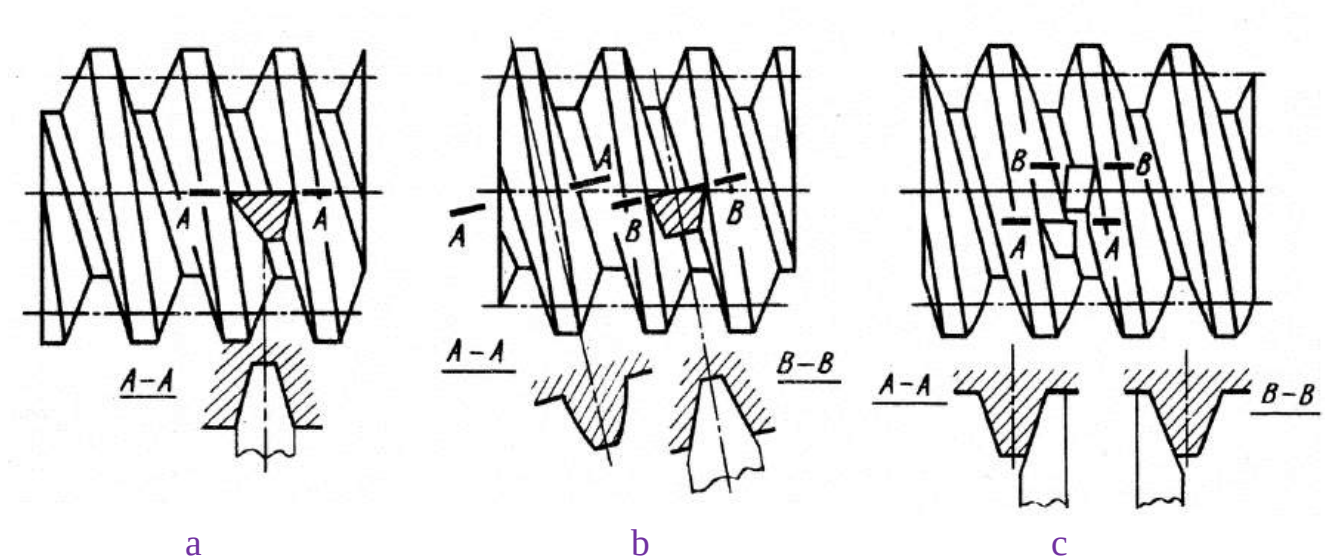


Fig. 3.54

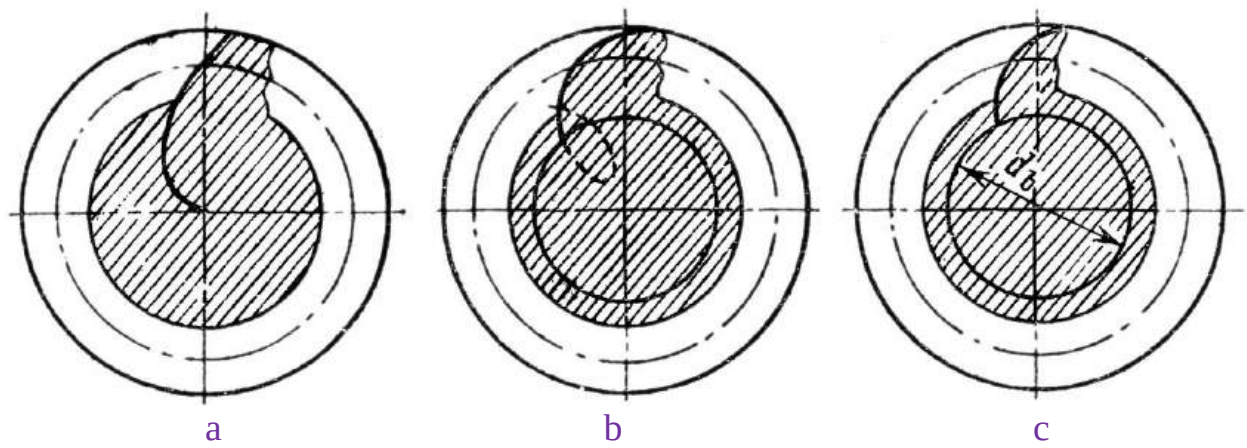


Fig. 3.55

3.6.3. Worm and Gear Types

There are two worm types: cylindrical (Fig. 3.56, *a*) and hour-glass (globoid) (Fig. 3.56, *b*). The first of them is simpler, the second type has load capacity much greater.

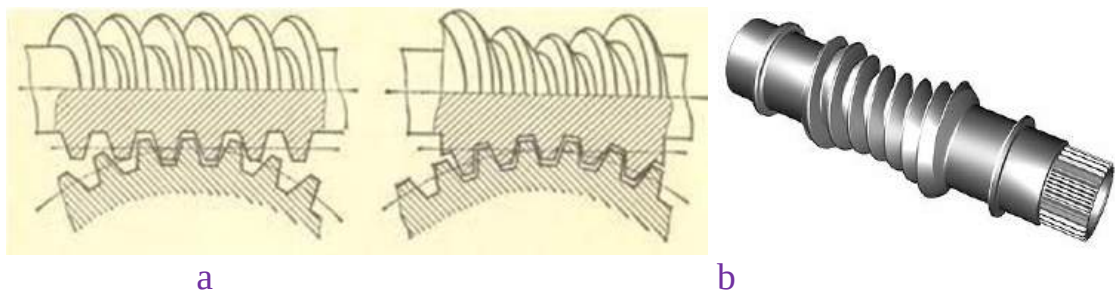


Fig. 3.56

Various types of interaction are shown in Fig. 3.57.

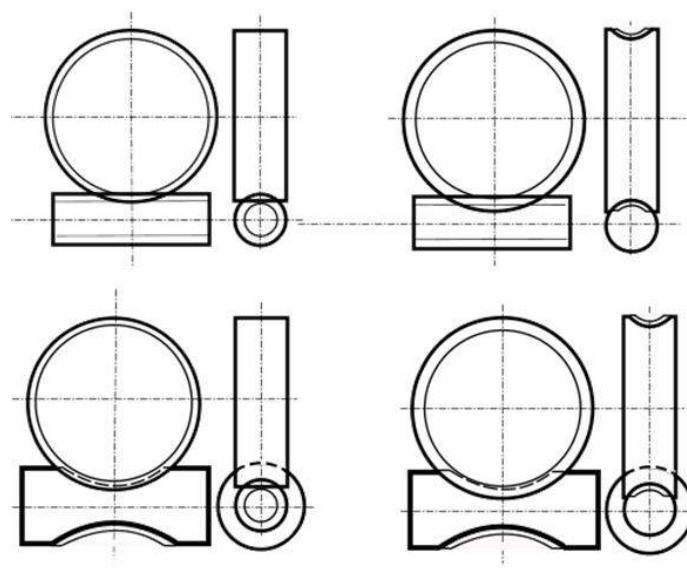


Fig. 3.57

3.6.4. Basic parameters

The main are:

- Z_1 is the worm number of starts, standard $Z_1 = 1, 2, 4$;
- $m = P_x / \pi$ is the module in axial cross-section;
- $q = d_1 / m$ is the worm diameter factor;
- $u = Z_2 / Z_1$ is the transfer ratio.

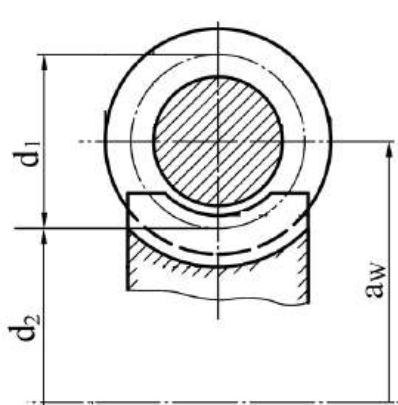
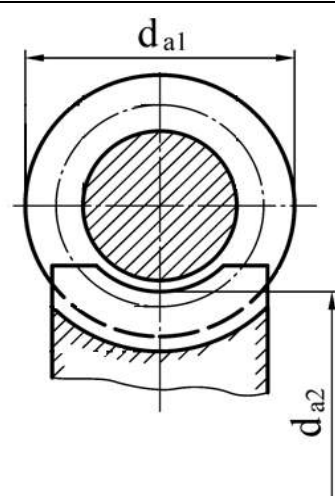
The worm number of starts depends on transfer ratio:

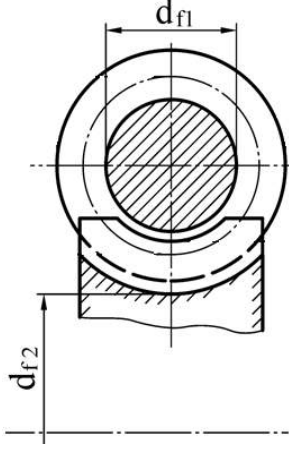
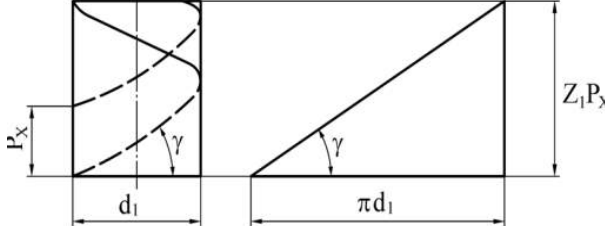
u	≤ 14	14...30	≥ 30
Z_1	1	2	4

To ensure required worm stiffness parameter q should be coordinated with module m as follows:

m	3	6	12
q	12.5...14	9...14	8;10

The main geometric parameters are:

Name	Formula	Sketch
Pitch diameter of worm and gear	$d_{w1} = m q$ $d_{w2} = m Z_2$	
Center distance	$a_w = 0,5m (q + z_2)$	
Addendum diameter of worm and gear	$d_a = d + 2 m$	

Name	Formula	Sketch
Root diameter of worm and gear	$d_f = d - 2,4 m$	
A helix angle	$\gamma = \text{atan } Z_1 / q$	

3.6.5. Acting forces

Axial load on worm is opposite to tangential load on wheel (Fig. 3.58):

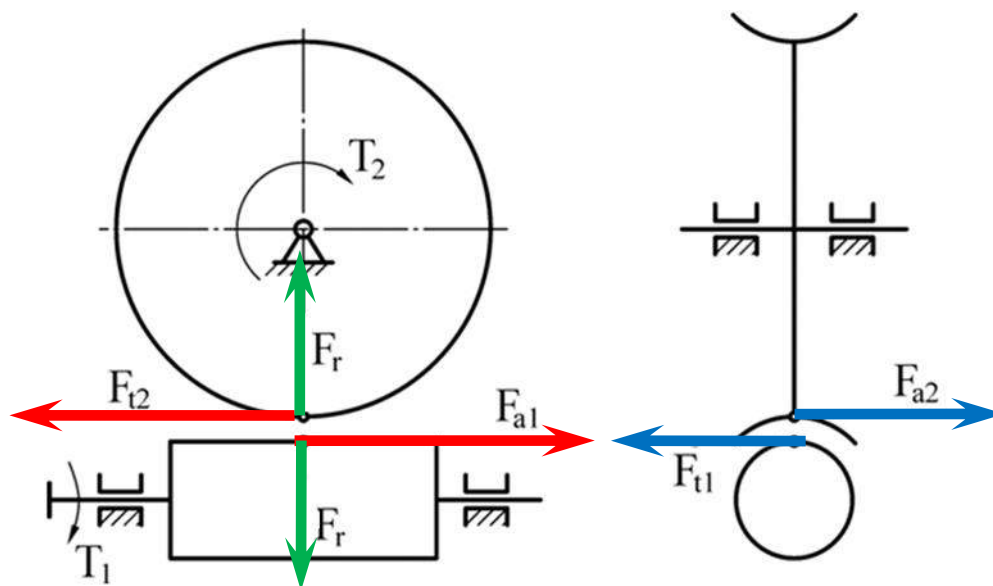


Fig. 3.58

$$F_{a1} = F_{t2} = \frac{2T_2}{d_{w2}}.$$

Radial load on both parts $F_{r1} = F_{r2} = F_{t2} \tan \alpha$.

Tangential load on the worm is opposite to the axial load on the wheel:

$$F_{a2} = F_{t1} = \frac{2T_1}{d_{w1}}.$$

3.6.6. Efficiency

In a case of driving worm formula to calculate transfer efficiency looks as that of power screw:

$$\eta = \frac{\tan \gamma}{\tan(\gamma + \rho')}.$$

If the gear is a driving part

$$\eta = \frac{\tan(\gamma - \rho')}{\tan \gamma}.$$

At the design calculation efficiency may be taken from experience:

Z_1	1	2	4
η	0.7...0.75	0.75...0.82	0.82...0.9

3.6.7. Allowable stresses

Limit of contact stress is $[\sigma]_H = [\sigma]_{HO} Z_N$, where durability factor is

$$Z_H = \sqrt[8]{\frac{10^7}{N_{HE}}}.$$

Limit of bending stress is $[\sigma]_F = [\sigma]_{FO} Y_N$, where durability factor is

$$Y_N = \sqrt[9]{\frac{10^6}{N_{FE}}}.$$

Allowable stresses are calculated by empirical formulae according to the gear material:

$V_s, \text{ m/s}$	≤ 2	2...5	≥ 5
Material group	III (antifriction cast iron)	II (tinless bronze)	I (tin bronze)
$[\sigma]_H$ at $HB \leq 350$	175 - 35 V_s	250 - 25 V_s	0,7 $\sigma_{UTS} Z_N c_V$
$[\sigma]_H$ at $HRC \geq 45$	200 - 35 V_s	300 - 25 V_s	0,9 $\sigma_{UTS} Z_N c_V$
$[\sigma]_F$	0,12 $\sigma_B Y_N$	(0,08 σ_{UTS} + 0,25 σ_y) Y_N	

$V, \text{ m/s}$	1	2	3	4	5	6	7	8
c_v	1,33	1,21	1,11	1,02	0,95	0,88	0,83	0,8

3.6.8. Design calculation

Define the number of starts taking into account that $z_2 = z_1 u$ and $z_2 \geq 28$.

Calculate the center distance: $a_w = 610 \sqrt[3]{\frac{T_2 K_H}{[\sigma_H]^2}}$.

Calculate the module: $m = \frac{2a_w}{q + z_2}$.

Select standard module.

Select standard worm factor q .

Calculate worm and gear parameters.

3.6.9. Check calculation

Acting contact stress is obtained as

$$\sigma_H = \frac{15000}{d_{w2}} \sqrt{\frac{T_2 K_H}{d_{w1}}} \leq [\sigma_H].$$

Acting bending stress is obtained as

$$\sigma_F = 0,7 Y_F \frac{F_{t2}}{b_2 m} \leq [\sigma_F],$$

where

Z_v	20	50	150	300
Y_F	1,98	1,45	1,27	1,24

and the equivalent wheel number of teeth $Z_v = Z_2 / \cos^3 \gamma$.

3.6.10. Heat calculation

The heat generated due to friction must be dissipated in order to avoid both overheating of the drive and lubricating oil. The maximum temperature of the mineral lubricant should not exceed 80°C.

The quantity of generated heat is

$$Q_{gen} = 1000(1 - \eta)P_1,$$

where P_1 is the transmitted power; η is the efficiency.

The heat generated must be dissipated through the lubricating oil to the gear box housing and then to the atmosphere.

The heat dissipated may be estimated by formula

$$Q_{dis} = K_T(t_1 - t_0)A,$$

where A is the total area of the housing; t_1 is the housing surface temperature and t_0 is the ambient air temperature; K_T is the heat transfer coefficient.

To increase heat dissipating capacity next parameters should be increased:

- area of the housing A (Fig. 3.59) by ribs;
- temperature difference between the housing surface t_1 and surrounding air t_0 ;
- the heat transfer coefficient K_T .

The average value of heat transfer coefficient for natural circulation (convective heat transfer)

$$K_T \approx 7 \dots 10 \frac{\text{kcal}}{\text{m}^2 \cdot \text{hour} \cdot \text{deg}}.$$

To increase heat transfer coefficient forced air cooling may be used (Fig. 3.60). In such case

$$K_T = 18 \dots 24 \frac{\text{kcal}}{\text{m}^2 \cdot \text{hour} \cdot \text{deg}}.$$

Another way is to use lubricating under pressure (Fig. 3.61).



Fig. 3.59

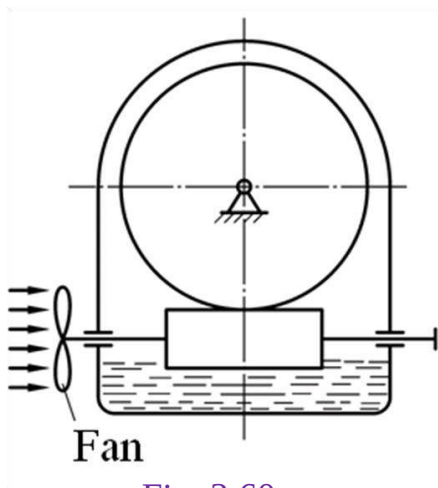


Fig. 3.60

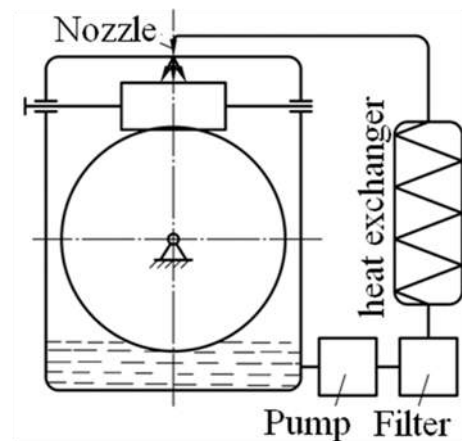


Fig. 3.61

3.7. Wave Transmission (Harmonic Drive)

It is a special type of mechanical gear system that can improve certain characteristics compared to traditional gearing systems such as helical gears or planetary gears.

The simplest strain wave transmission consists of three main elements (Fig. 3.62): circular spline (or rigid wheel) 1, flexible spline (or flexible wheel) 2 and wave generator 3.

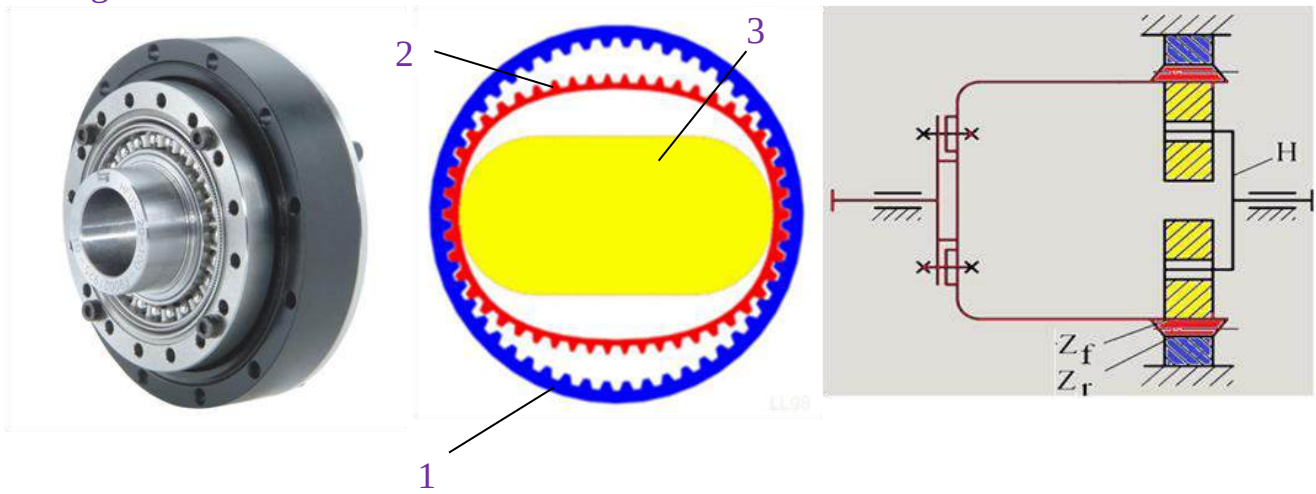


Fig. 3.62

The thin-walled flexible wheel is made as a pipe, and it is named flexible because during work it resiliently becomes deformed by wave generator.

One of wheels, rigid or flexible, joints to an output shaft, and then the second wheel is fixed (Fig. 3.63). If both wheels are movable, the transmission is differential.

Transmission is frictional if wheels are without teeth, if teeth are present, then transmission – geared.

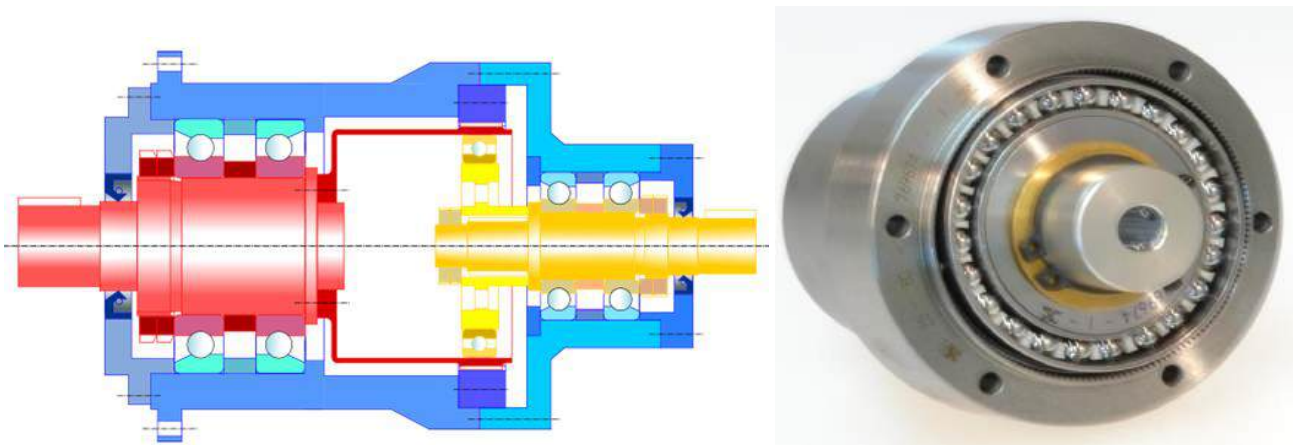


Fig. 3.63

A flexible wheel transforms from round to elliptic in the place where a generator is located.

On the ellipse major axis (A, Fig. 3.64) teeth of flexible and rigid wheels mesh fully, on the ellipse minor axis (C) teeth contact is absent; on intermediate areas (B, D) there is the incomplete teeth meshing.

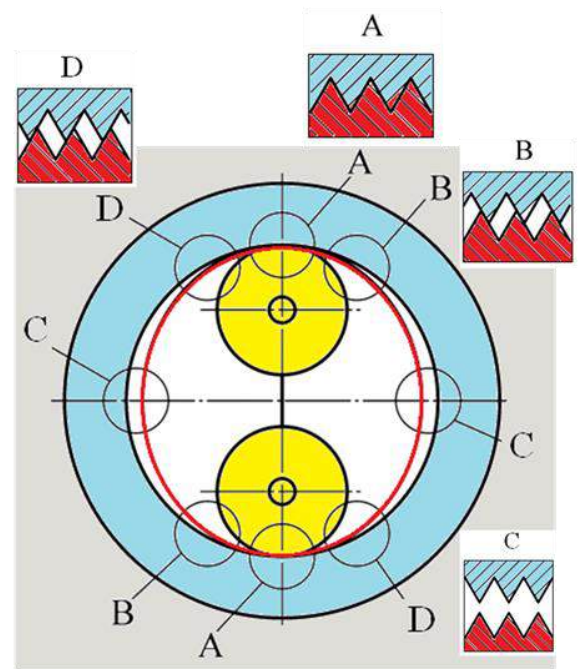
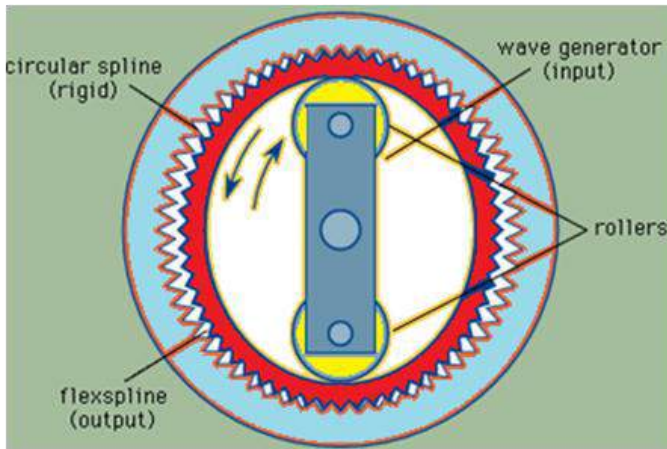


Fig. 3.64

Advantages

- small size and weight due to the large load capacity, because of the multi-gearing (20 ... 40 % of teeth are in contact at the same time);
- large transfer ratio for a small number of parts, for flexible metal wheels $i = 80 \dots 320$;
- the relatively high efficiency (0.8...0.94);
- transmission of motion through wall (Fig. 3.65);
- can work as a reducer and a multiplier;
- high kinematic accuracy and smoothness

Disadvantages

- high tension of main elements
- reduced torsional rigidity

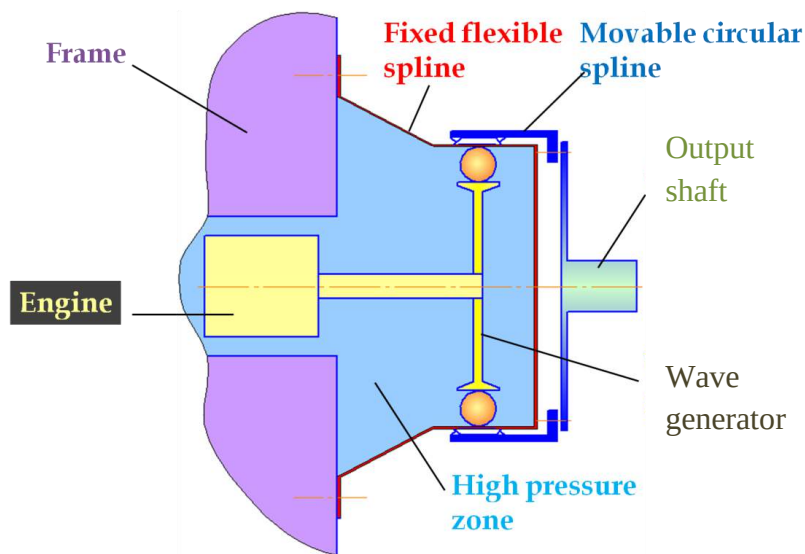


Fig. 3.65

3.7.1. Wave Generator Types

Cam

A cam generator consists of the elliptic cam 1 (Fig. 3.66) and bearing with flexible rings 2 and 3 pressed on the cam and the balls 4 with a cage (separator). The form of cam corresponds to the accepted form of the flexible wheel deformation.

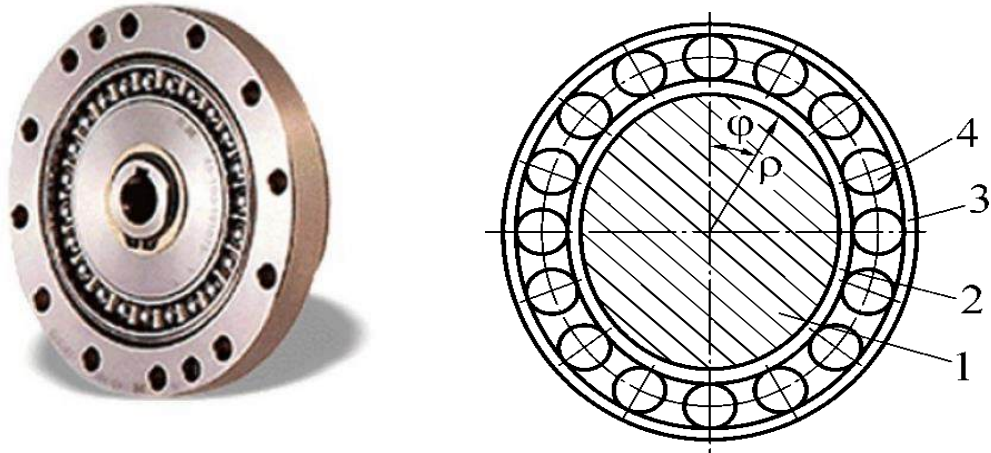


Fig. 3.66

Disk

Disk generator contains two disks 1 and 2 (Fig. 3.67) with a diameter D_d set to rotate on bearings fixed on an eccentric shaft 3. To reduce the contact stresses intermediate ring 4 is located between the disk and the flexible wheel.

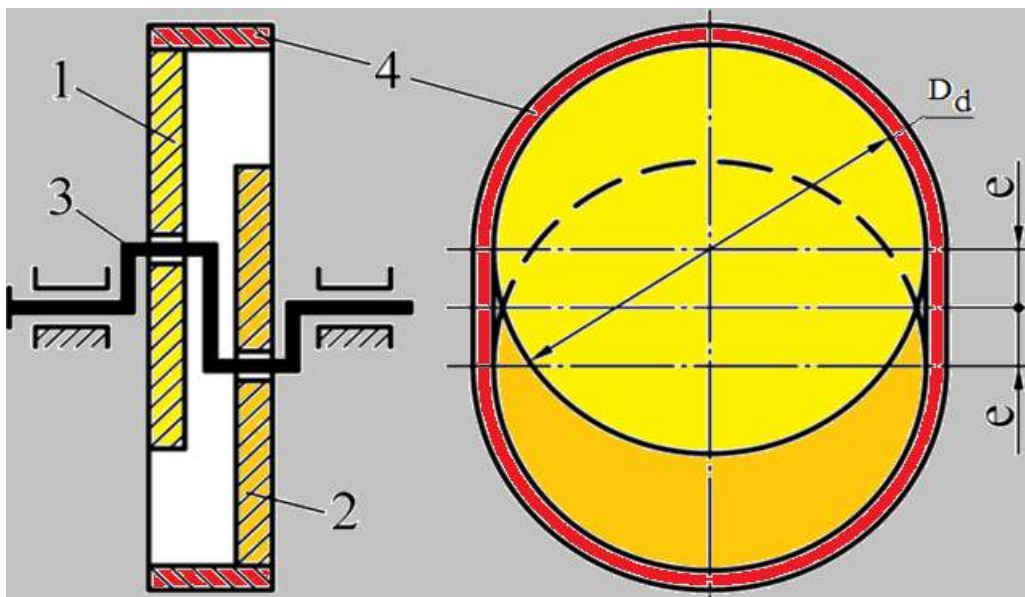


Fig. 3.67

Rollers

Disk generator includes a number of rollers (usually 2...4) which are placed on carrier h (Fig. 3.68). This type ensures free deformed form for flexible spline.

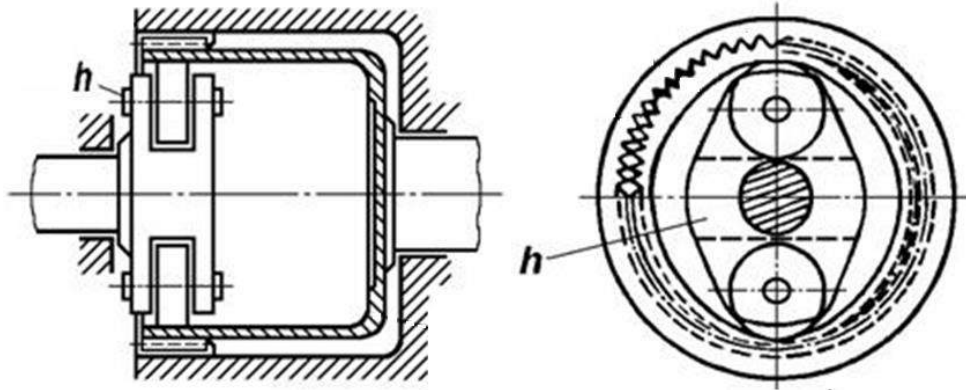


Fig. 3.68

3.7.2. The basic parameters

They are (Fig. 3.69): D_b is the bearing outer diameter; b_f is the flexible wheel face width; C is the clearance; h is the addendum and dedendum height; h_1 is the flexible wheel thickness under the teeth; h_{sh} is the shell thickness.

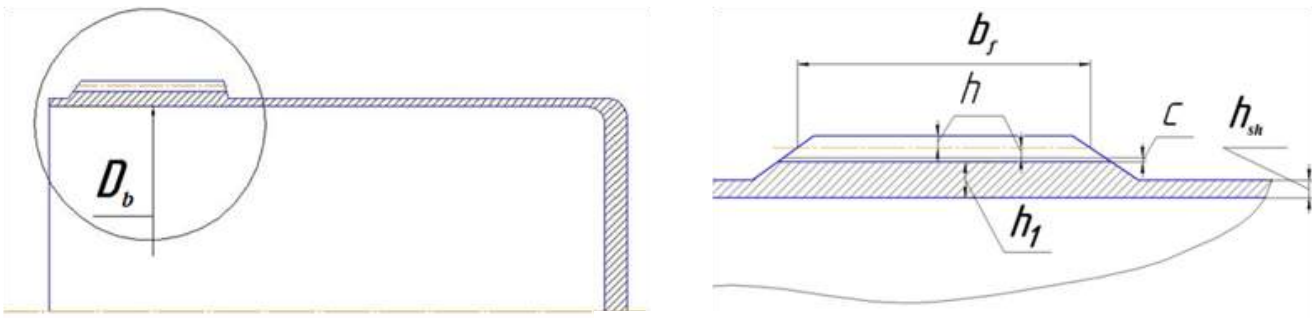


Fig. 3.69

3.7.3. Kinematics

The difference of the number of teeth is accepted proportionally to the number of waves:

$$Z_r - Z_f = \pm K_Z \nu,$$

where ν is the wave quantity; $K = 1, 2, 3$ (usually 1).

The numbers of teeth of the flexible and circular splines are defined from transfer ratio formulae. For two-wave transmission with an external and fixed circular spline

$$i = \frac{Z_f}{Z_r - Z_f} = -\frac{Z_f}{2} \Rightarrow Z_f = |2i|, Z_r = Z_f + 2.$$

When flexible spline is fixed

$$i = \frac{Z_r}{Z_r - Z_f} = \frac{Z_r}{2} \Rightarrow Z_r = 2i, Z_f = Z_r - 2.$$

3.7.4. Design calculation

The wave gearing theory is based on elastic dynamics and utilizes the flexibility of metal.

Bearing outer diameter should be found due to equation

$$D_b = 28,4 \sqrt[3]{\frac{T_2}{\psi_{bd}[p]}},$$

where allowable pressure

$$[p] = 25 \left(\frac{5000}{L_h} \right)^{0,2} K_i K_f,$$

where K_i is the coefficient considering transfer ratio influence

$$K_i = \left(\frac{i-10}{80} \right)^{0,45},$$

K_f is the coefficient considering the wave generator influence: $K_f = 1$ at the cam, $K_f = 0,85$ at the disk generator.

$$\text{Relative face width } \psi_{bd} = \frac{b_f}{D_b} = 0,2 \dots 0,22.$$

3.7.5. Check calculation

Stresses which act in the dangerous cross-section in the inner rim face (Fig. 3.70) are:

1. Bending in tangential direction (reversed cycle):

$$\sigma_t = \sigma_{ta} = \frac{8Eh_1W_0}{d_f^2},$$

where d_f is the pitch diameter; W_0 is the maximum flexible wheel deformation.

2. Shell tension under the teeth (repeated cycle):

$$\sigma_{ten} = \frac{0,9T_2}{d_f b_f h_1}, \quad \sigma_a = \sigma_m = \frac{\sigma_{ten}}{2},$$

where b_f is the face width; T_2 is the output torque.

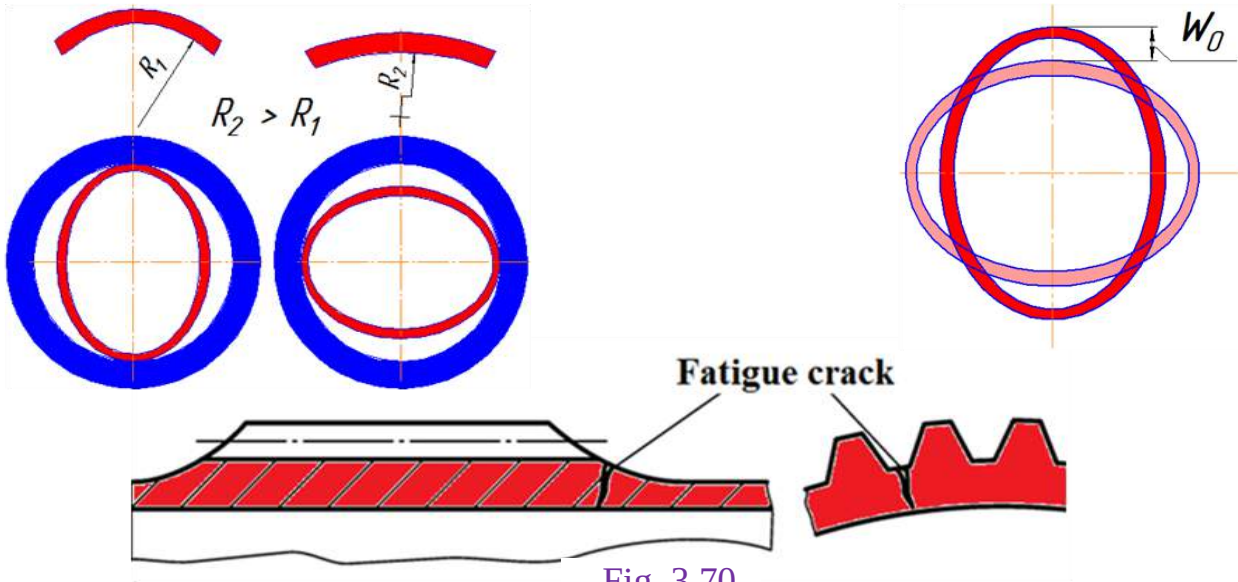


Fig. 3.70

3. Shear due to the torsion:

$$\tau = \frac{0,52 E h_1^{0,5} W_0}{d_f^{1,5}} + \frac{T_2}{d_f^2 h_1}.$$

The total amplitude stress is

$$\sigma_{a\Sigma} = K_{wg} \left(\sigma_t + \frac{\sigma_{ten}}{2} \right),$$

where $K_{wg} = 1$ in case of cam generator; $K_{wg} = 1,1$ for disk generator.

The amplitude and mean shear stresses are

$$\tau_a = 0,45 \tau_{max}, \tau_m = 0,55 \tau_{max}.$$

The sum safety factor

$$S = \frac{S_\sigma S_\tau}{\sqrt{S_\sigma^2 + S_\tau^2}} \geq [S] = 1,3 \dots 1,8,$$

where S_σ is the actual safety factor on normal stress:

$$S_\sigma = \frac{\sigma_{-1}}{(K_\sigma \sigma_a + \psi_\sigma \sigma_m)},$$

S_τ is an actual safety factor on shear:

$$S_\tau = \frac{\tau_{-1}}{(K_\tau \tau_a + \psi_\tau \tau_m)},$$

where σ_{-1} and τ_{-1} are the limits of endurance accordingly at bend and torsion with a reversed stress cycle; ψ_σ , ψ_τ are the factors characterizing material sensitivity for alternating stresses; K_σ and K_τ are a fatigue normal and shear stress-concentration factors:

$$K_\sigma = 1,5 \dots 2; \quad K_\tau = 0,8 K_\sigma.$$

Part 4. SHAFTS & AXLES

4.1. Terminology

Shaft is an element to transfer the rotating moment along its axis and to support rotating elements (Fig. 4.1, *a*). It also is loaded by the cross-section forces and the bending moments.

An axle is a central shaft for rotating a wheel or a gear (Fig. 4.1, *b*). On wheeled vehicles, the axle may be fixed to the wheels, rotating with them, or fixed to its surroundings, with the wheels rotating around the axle.

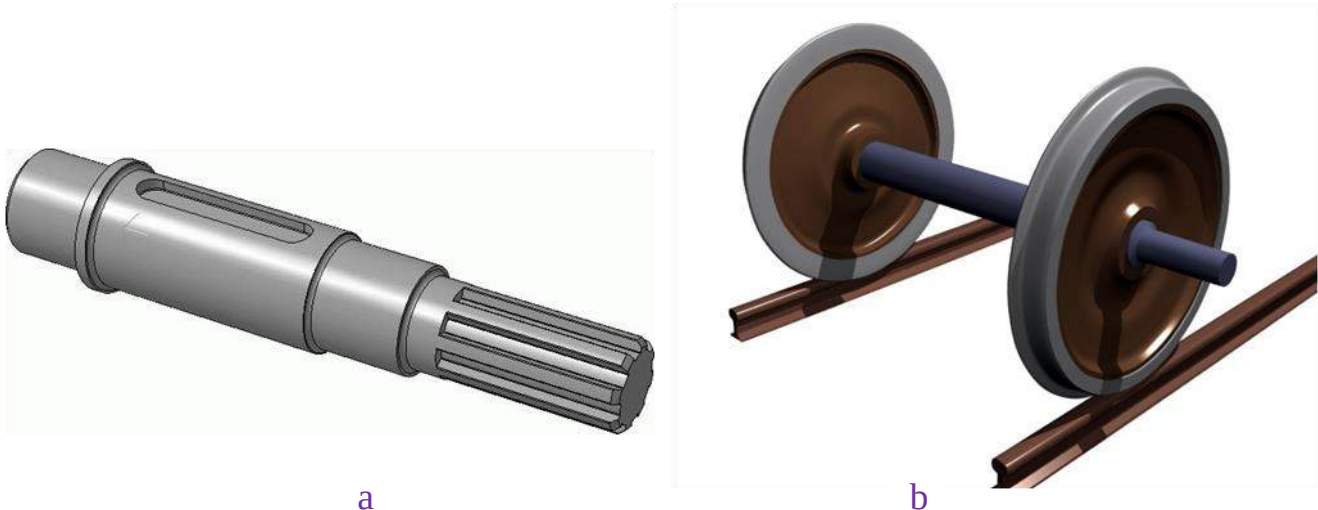


Fig. 4.1

Train wheels are affixed to a straight axle, such that both wheels rotate in unison. This is called a wheelset.

An automobile may use a longitudinal shaft to deliver power from an engine/transmission to the other end of the vehicle before it goes to the wheels. A pair of short drive shafts is commonly used to send power from a central differential, transmission, or transaxle to the wheels.

Axles are an integral component of a wheeled vehicle. In a live-axle suspension system, the axles serve to transmit driving torque to the wheel, as well as to maintain the position of the wheels relative to each other and to the vehicle body. The axles in this system must also bear the weight of the vehicle plus any cargo. A non-driving axle, such as the front beam axle in Heavy duty trucks and some two wheel drive light trucks and vans, will have no shaft. It serves only as a suspension and steering component. Conversely, many front wheel drive cars have a solid rear beam axle.

In rear-wheel-drive cars and trucks, the engine turns a driveshaft also called a propeller shaft or tail shaft which transmits rotational force to a drive axle at the rear of the vehicle. The drive axle may be a live axle, but modern automobiles generally use a split axle with a differential.

A dead axle, also called lazy axle, is not part of the drivetrain but is instead free-rotating. The rear axle of a front-wheel drive car may be considered a dead axle.

Many trucks and trailers use dead axles for strictly load-bearing purposes. A dead axle located immediately in front of a drive axle is called a pusher axle. A tag axle is a dead axle situated behind a drive axle. They are also found on semitrailers, farm equipment, and certain heavy construction machinery serving the same function. On some vehicles (such as motorcoaches), the tag axle may be steerable. In some designs the wheels on a lazy axle only come into contact with ground when the load is significant thus saving wear.

Some dump trucks and trailers are configured with lift axles (also known as airlift axles or drop axles), which may be mechanically raised or lowered. The axle is lowered to increase the weight capacity, or to distribute the weight of the cargo over more wheels, for example to cross a weight restricted bridge. When not needed, the axle is lifted off the ground to save wear on the tires and axle and to increase traction in the remaining wheels. Lifting an axle also alleviates lateral scrubbing of the additional axle in very tight turns, allowing the vehicle to turn more readily. In some situations removal of pressure from the additional axle is necessary for the vehicle to complete a turn at all.

Drive shafts have been used on motorcycles almost as long as there have been motorcycles. As an alternative to chain and belt drives, drive shafts offer relatively maintenance-free operation and long life. A disadvantage of shaft drive on a motorcycle is that gearing or a Hobson's joint or similar is needed to turn the power 90° from the shaft to the rear wheel, losing some power in the process. On the other hand, it is easier to protect the shaft linkages and drive gears from dust, sand and mud.

On a power-driven ship, the drive shaft, or propeller shaft, usually connects the transmission inside the vessel directly to the propeller, passing through a stuffing box or other seal at the point it exits the hull. There is also a thrust block, a bearing to resist the axial force of the propeller. As the rotating propeller pushes the vessel forward, any length of drive shaft between propeller and thrust block is subject to compression, and when going astern to tension. Except for the very smallest of boats, this force isn't taken on the gearbox or engine directly.

Cardan shafts are also often used in marine applications between the transmission and either a propeller gearbox or waterjet.

The drive shaft has served as an alternative to a chain-drive in bicycles for the past century, never becoming very popular.

A spindle is a short rotation shaft. It is originated from the round tapering stick on a spinning wheel, on which the thread is twisted. Spindles are used in all machine tools such as the small drive shaft of lathe or the spindle of drilling machine.

A countershaft is a secondary shaft, which is driven by the mainshaft and from which the power is supplied to the machine component. Often, the countershaft is driven from the main shaft by means of spur or helical gears and thus rotate 'counter' to the direction of mainshaft. Countershafts are used in multi-stage gearboxes.

A jackshaft is an auxiliary or intermediate shaft between two shafts that are used in transmission of power. Its function is the same as that of countershaft.

A lineshaft consists of number of shafts connected in axial direction by means of couplings. They were popular in workshops using group drive, driven by a single electric motor. Number of pulleys is mounted on a lineshaft and power is transmitted to individual machines by different belts. However, nowadays individual drives have replaced group drive, making lineshaft obsolete.

By the shape of the geometric axis of the shafts are:

- direct;
- eccentric (crankshaft, cam);
- flexible;
- rollers.

By load:

- bending and torsion;
- torsion only.

By moving ability:

- movable;
- fixed.



Wrist is a section of the shaft or axis lying in supports. They are divided into spines, neck and heels.

Spine is a wrist situated on the end of shaft or axle and usually transfer only radial load.

Neck is a wrist situated in the middle of shaft or axle.

Bearing is a support for spine or neck.

Spines and necks can be cylindrical, conic and spherical. Cylindrical are commonly used.

4.2. Materials

Material of shafts and axels must be strong, well-handled and have high modulus of elasticity. Carbon and alloyed steels are commonly used. For shaft and axels without heat treatment Steel 5 and Steel 6 are used, for shafts and axels with heat treatment Steel 40 and Steel 40X are used. High-speed shafts are made from Steel 20, Steel 20X, Steel 12XHCA. Wrists of such shafts are cemented in order to increase its wear resist.

Shafts and axels are worked out on lathes with following grinding of wrists and landing surfaces.

4.3. Shafts and Axels Calculations

The order of calculation consists of stages as follows:

1. Design calculation – it includes shaft diameter calculating.
2. Round to standard.
3. Design shaft profile.
4. Calculate the support reactions.

5. Plot the diagram of bending and torsion moments.
6. Calculate the strength, rigidity, etc.

Shaft Design Calculation

Shaft design calculation is done for static strength for the following preliminary calculation of shaft diameter. Before design calculation torque T , Nm, and rotational speed n , rpm, are usually known.

Bending moments can be determined only after design of shaft axial length. Moreover, when design of the shaft is known we can specify places of stress concentration: fillets, keyways etc. That is why shaft design calculation is accomplished under condition of existence of torque only.

Shaft diameter is obtained on torsion strength condition:

$$\tau_t = \frac{T}{W_\rho} = \frac{T}{0,2d^3(1 - \alpha^4)} \leq [\tau_t],$$

$$d = \sqrt[3]{\frac{T}{0,2(1 - \alpha^4)[\tau_t]}},$$

where $[\tau_t] = 20 \dots 50$ MPa in general engineering, $[\tau_t] = 60 \dots 90$ MPa in aviation.

Calculated diameter is rounded to standard diameter: 10; 11; 12; 14; 16; 18; 20; 22; 25; 28; 32; 36; 40; 45; 50; 56; 63, etc.

Final diameters and shape is defined after bearing selection when dimensions of wrists are known.

Axle Design Calculation

Axels are supporting elements, that's why they are loaded by bending forces only. Axle design calculation on static strength is the same as for girder with the rocket bearing using simple methods of Strength Materials.

Axle diameter is obtained on bending strength condition:

$$\sigma_b = \frac{M}{W} = \frac{M}{0,1d^3(1 - \alpha^4)} \leq [\sigma_b],$$

$$d = \sqrt[3]{\frac{M}{0,1(1 - \alpha^4)[\sigma_b]}}.$$

Check Calculation

1. Determine the distances between the supports and the points of external forces application at the draft drawing (Fig. 4.2).

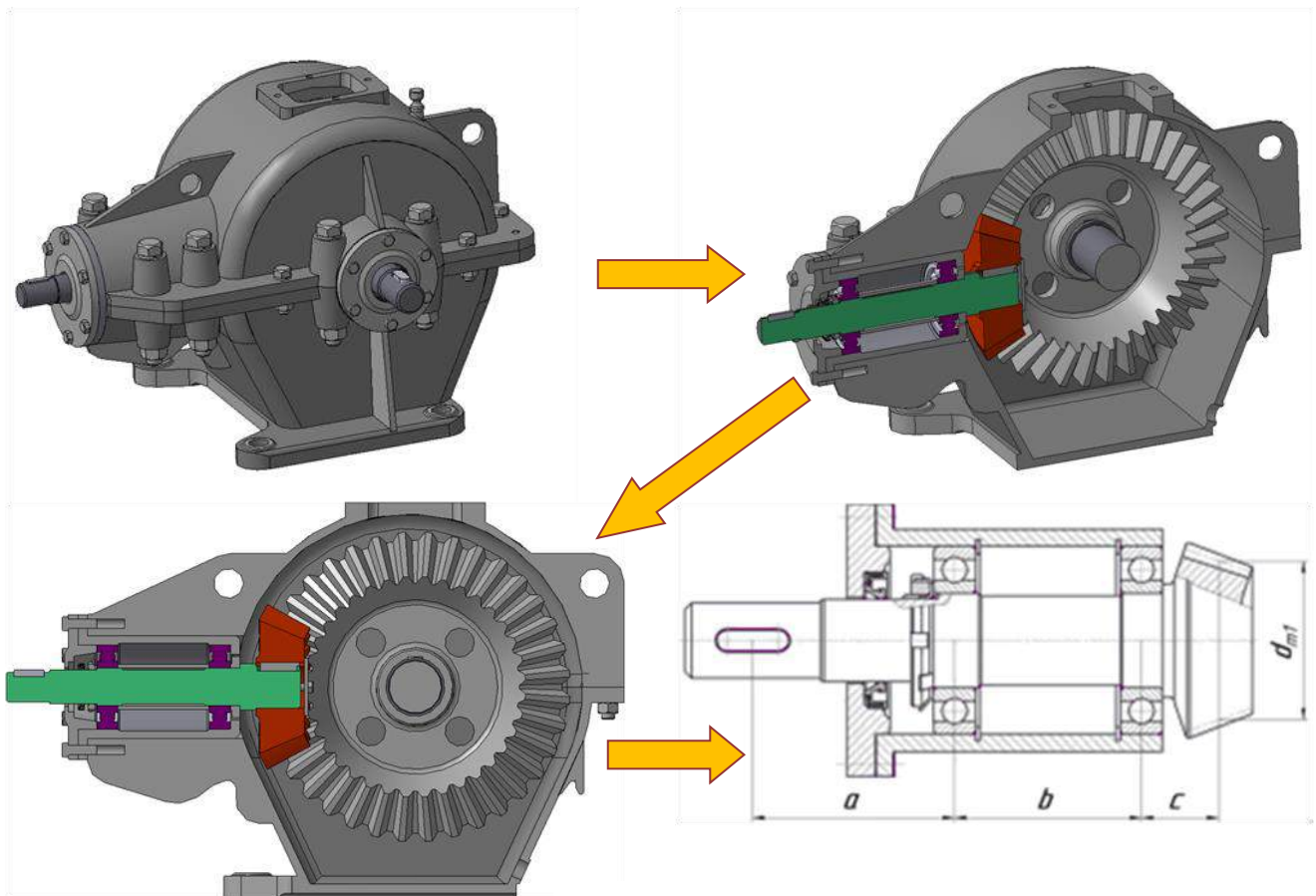


Fig. 4.2

2. Replace drawing with diagram (Fig. 4.3).

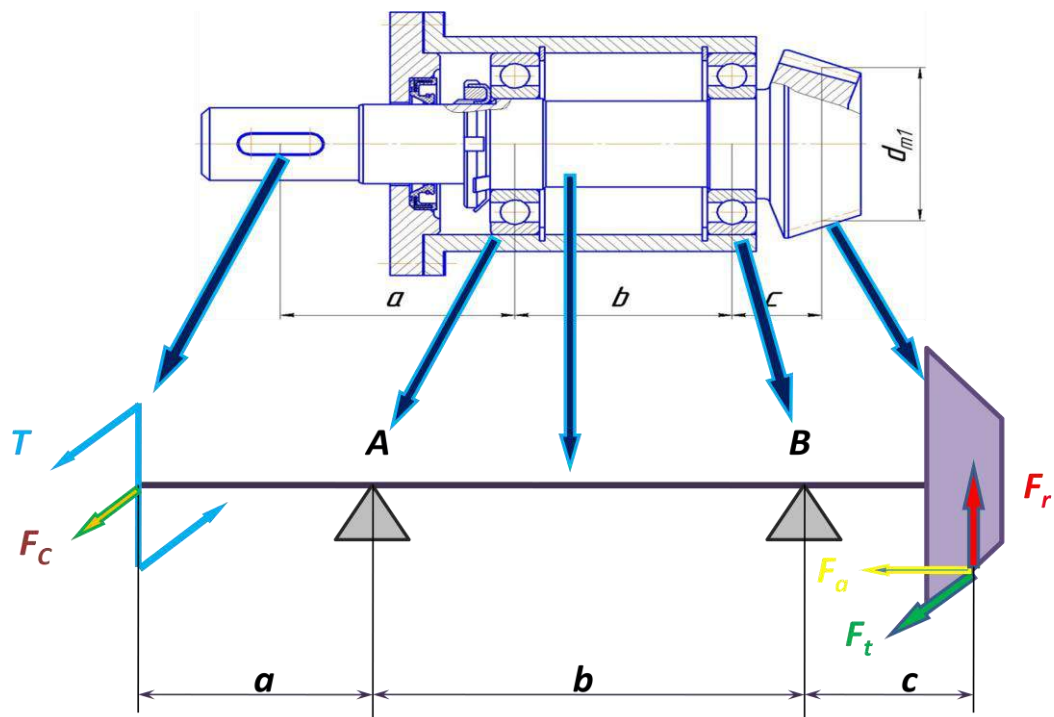


Fig. 4.3

3. Transfer the represented in the diagram acting forces to an axis of a shaft. Place the forces separately in vertical and horizontal planes (names of planes – vertical and horizontal – where forces operate in are conditional).

Support reactions are determined by methods of theoretical mechanics: equilibrium condition in moments form is considered:

$$\sum_{i=1}^N M_{F(i)B} = 0; \sum_{i=1}^N M_{F(i)A} = 0.$$

Reactions in the vertical plane are obtained as (Fig. 4.4):

$$\sum_{i=1}^N M_{iA} = R_{BV}b - M + F_r(b+c) = 0 \Rightarrow R_{BV} = \frac{M - F_r(b+c)}{b},$$

$$\sum_{i=1}^N M_{iB} = -R_{AV}b - M + F_rc = 0 \Rightarrow R_{AV} = \frac{F_rc - M}{b}.$$

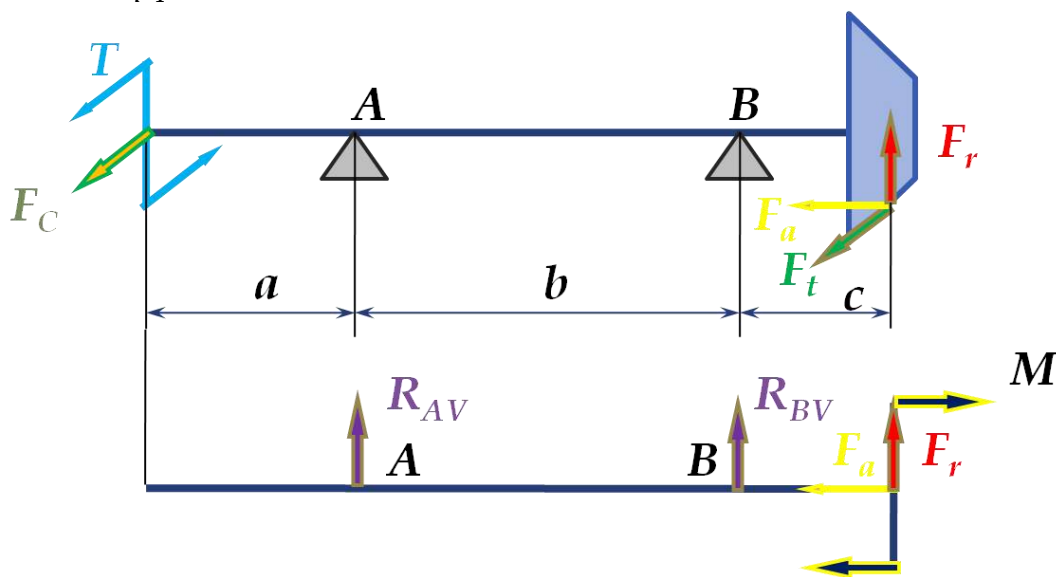


Fig. 4.4

Reactions in the horizontal plane are obtained as (Fig. 4.5):

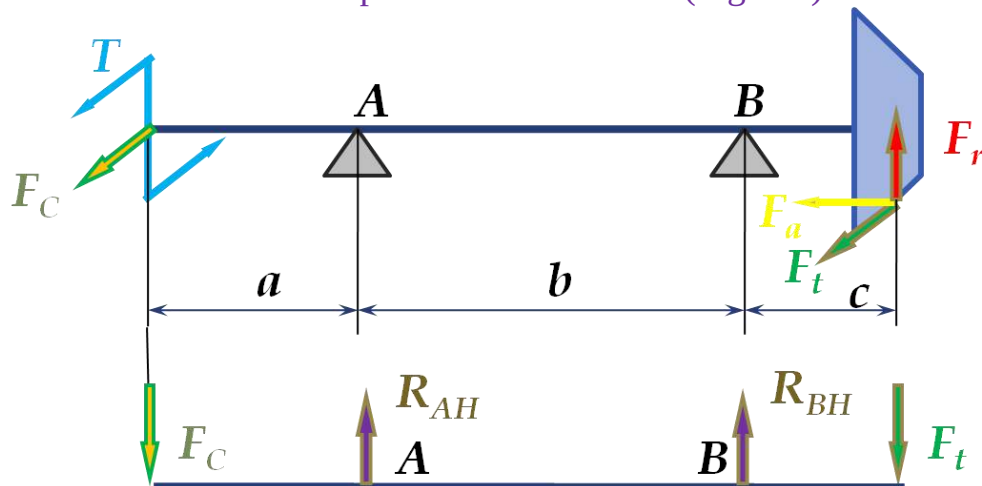


Fig. 4.5

$$\sum_{i=1}^N M_{iB} = -R_{AH}b - F_t c + F_C(a+b) = 0 \Rightarrow R_{AH} = \frac{F_C(a+b) - F_t c}{b};$$

$$\sum_{i=1}^N M_{iA} = R_{BH}b - F_t(b+c) + F_C a = 0 \Rightarrow R_{BH} = \frac{F_t(b+c) - F_C a}{b}.$$

4. Graph bending moments and torque diagrams for both vertical and horizontal planes (Fig. 4.6).

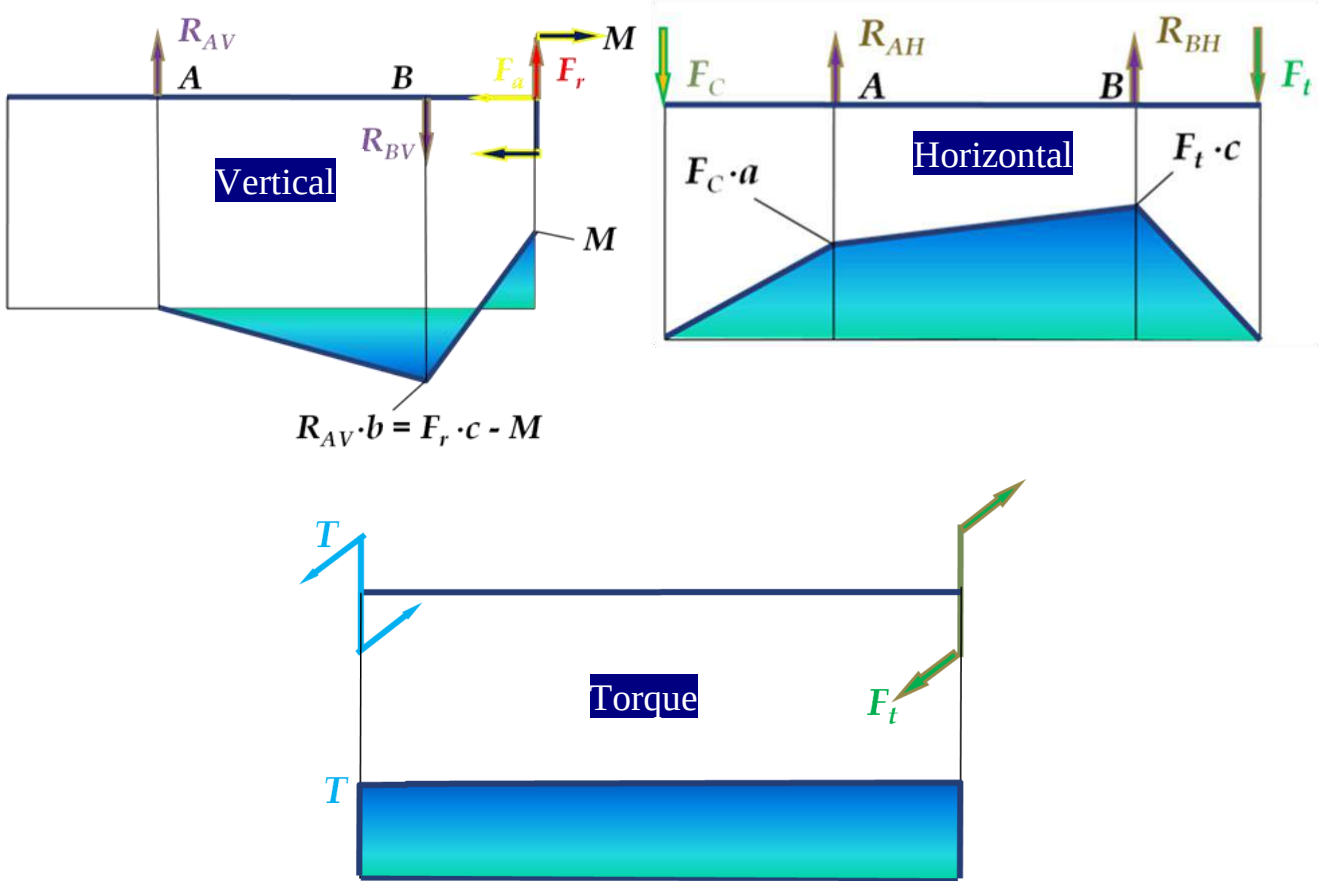


Fig. 4.6

5. Determine the dangerous sections (support B, see Fig. 4.6).

6. Calculate total moments into the section:

$$M_{\Sigma} = \sqrt{M_V^2 + M_H^2}.$$

7. To prevent remnant plastic deformation at a shaft overload safety factor is calculated:

$$S = \frac{\sigma_y}{\sigma_E} \geq [S] = 1,5...2,5; \sigma_E = \sqrt{\sigma^2 + 4\tau^2}.$$

The overload is estimated by overload coefficient $K = 1.5...3$. Thus

$$\sigma = \frac{M_{\Sigma} K}{W_x} + \frac{F_a}{A}; \tau = \frac{TK}{W_{\rho}}.$$

Check Calculation on Fatigue Endurance

The safety factor of fatigue strength under the combined action of normal and shear stresses is calculated:

$$S = \frac{S_\sigma S_\tau}{\sqrt{S_\sigma^2 + S_\tau^2}} \geq [S] = 1,5 \dots 2,5,$$

where S_σ is the actual safety factor on normal stress (without the torsion account):

$$S_\sigma = \frac{\sigma_{-1}}{(K_{\sigma D} \sigma_a + \psi_\sigma \sigma_m)};$$

S_τ is the actual safety factor on shear stress (without the bend account):

$$S_\tau = \frac{\tau_{-1}}{(K_{\tau D} \tau_a + \psi_\tau \tau_m)};$$

σ_{-1} and τ_{-1} are the limits of endurance accordingly at bending and torsion with a reversed stress cycle;

ψ_σ , ψ_τ are the factors characterizing material sensitivity for alternating stresses:

$$\psi_\sigma = \frac{2\sigma_{-1} - \sigma_0}{\sigma_0}, \quad \psi_\tau = \frac{2\tau_{-1} - \tau_0}{\tau_0};$$

$K_{\sigma D}$ and $K_{\tau D}$ are the fatigue normal and shear stress-concentration factors:

$$K_{\sigma D} = \frac{\frac{K_\sigma + K_F - 1}{\varepsilon}}{K_V}, \quad K_{\tau D} = \frac{\frac{K_\tau + K_F - 1}{\varepsilon}}{K_V};$$

ε is the absolute size factor of the cross-section (table parameter): the bigger absolute dimensions the smaller ε and the smaller safety factor;

K_F is a surface roughness factor (table parameter): the better surface the bigger safety factor; K_V is the factor of influence of the hardening (table parameter).

K_σ and K_τ are the normal and shear fatigue stress concentration factors for various types of the stress concentrators (Fig. 4.7).

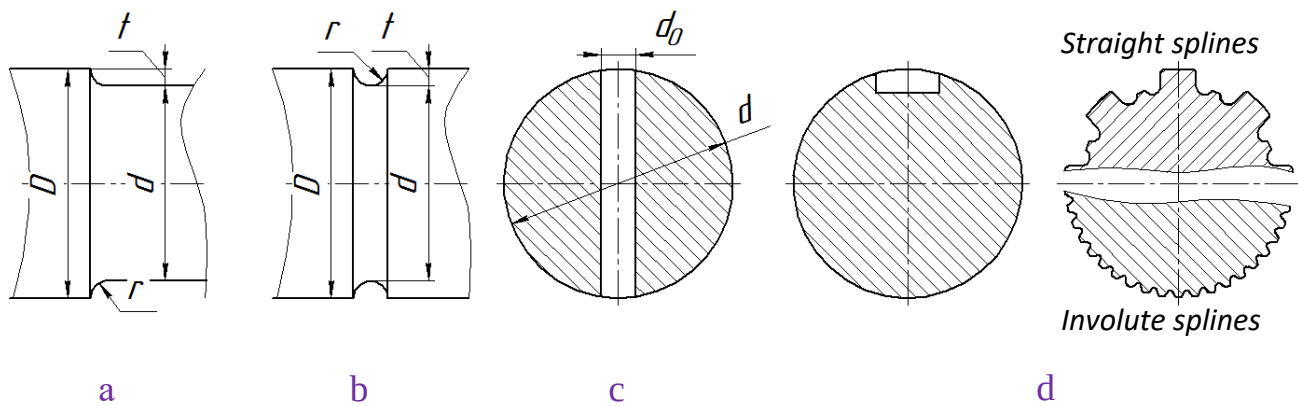


Fig. 4.7

In most cases it is difficult to clarify the valid load cycle under actual operating conditions, therefore calculation carries out conditionally on a rated load, and a cycle of stress is assumed as reversed (symmetric, Fig. 4.8, *a*, *b*) for the bending stress and repeated (Fig. 4.8, *c*) for the torsion stress.

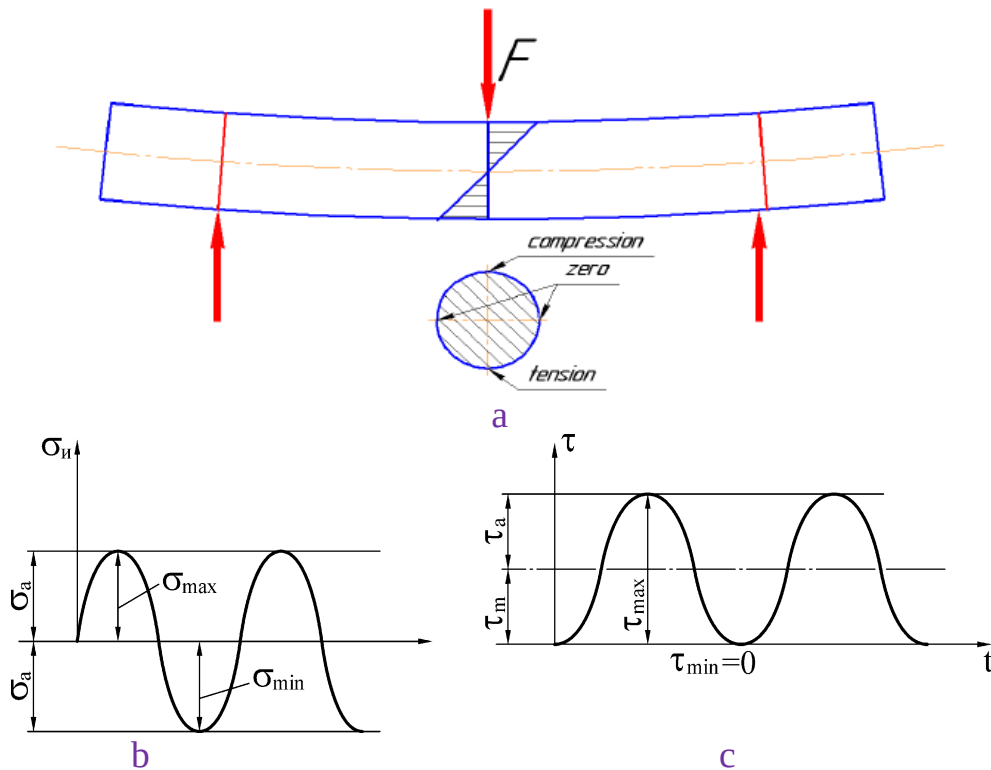


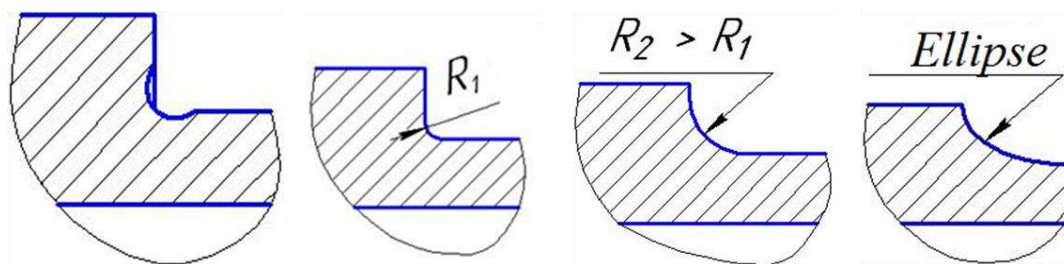
Fig. 4.8

In this case: $\sigma_a = \frac{M_z}{W_x}$; $\sigma_m = \frac{F_a}{A}$; $\tau_a = \tau_m = \frac{1}{2} \frac{T}{W_\rho}$.

Actions to Increase Fatigue Endurance

Design ways are:

- use fine thread pitch 1,5 mm with the length 8...12 mm;
- use involute splines instead of keys or straight splines;
- provide smooth transitions from one size to another. The bigger fillet radius the lower stress concentration. The best variant is the elliptical form:



Technological ways are:

- hardening of the seating surfaces run-in rollers or balls, blasting shot. In this case, the limit of endurance can be increased two times;
- thermo-chemical treatment: carburizing, nitriding, carbonitriding, etc.;
- decreasing of surface roughness: micro stress concentrators of machining decrease.

4.4. Critical Frequency Calculation

Periodically repeating impulses of the external forces cause the load fluctuations in rotating elements. When the frequency of the external forces coincides with the natural frequency of the system comprising the shaft and parts fitted to it the vibration amplitude sharply increases (the resonance phenomenon occurs).

Rotation speed corresponding to the resonance is called critical. Excessive vibration of the shaft indicates that its speed is close to the critical point. If the shaft continues operation in these conditions inevitable fail will happen. That is why it is necessary to know the critical points.

Let us consider shaft cross-section fluctuations on the elementary example when the fluctuating system includes a solid section shaft with an unbalanced disk fitted to it. (Fig. 4.9, a).

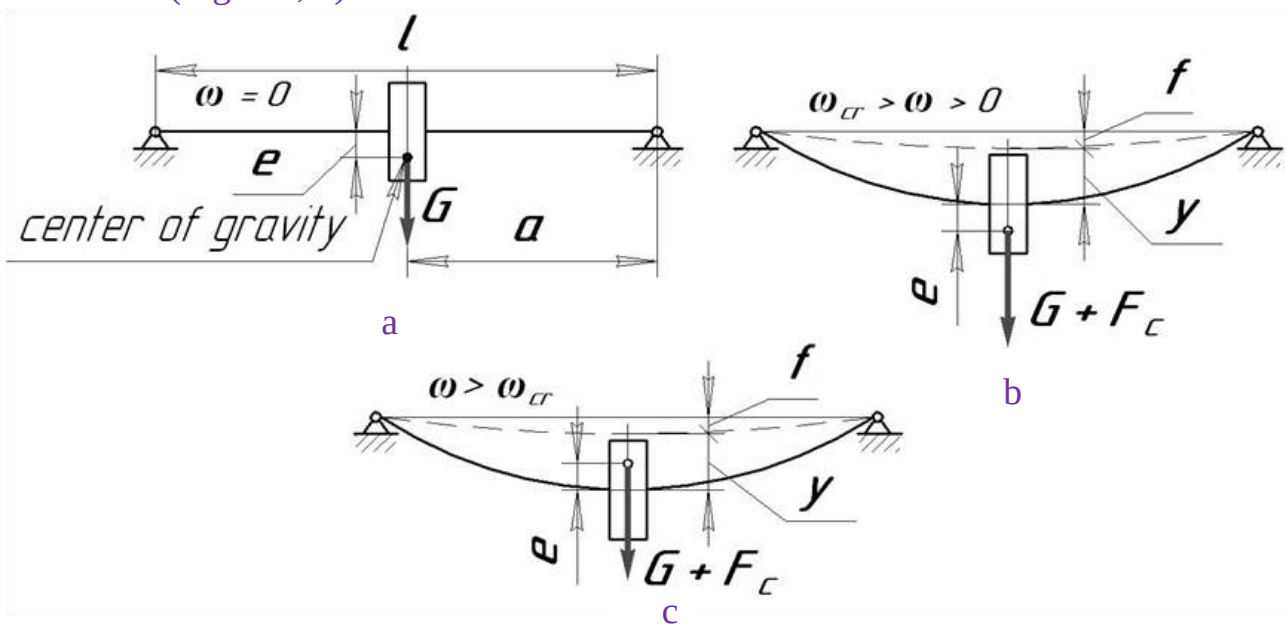


Fig. 4.9

The given system can be considered as a one-mass rotor with bending rigidity c , mounted on the absolutely rigid supports.

The weight of rotor G bends the shaft on value f (a static deflection, see Fig. 4.9, b). Usually the effect of the rotor weight is neglected.

At the influence of centrifugal force during rotation with the angular velocity ω the additional (dynamic) shaft deflection y appears (see Fig. 4.9, b). The centrifugal force caused by displacement of the centre of gravity from an axis of a shaft rotation is

$$F_c = m(e + y)\omega^2, \quad (4.1)$$

where m is the rotor mass, e is the disk centre of gravity displacement from a geometrical axis of rotation (eccentricity).

This force is the cause of the rotor's difficult movement, so called direct synchronous precession, when the elastic line rotates in the direction of a shaft rotation about its geometrical axis with angular velocity ω . As centrifugal force changes the direction, the shaft makes harmonious fluctuations.

Force F_c is counterbalanced by the reaction equals to the system bending rigidity c multiply deformation y , that is

$$m(e + y)\omega^2 = cy.$$

Thus

$$y = \frac{m\omega^2 e}{c - m\omega^2}.$$

The shaft angular velocity can reach value at which $(c - m\omega^2) \rightarrow 0$ then $y \rightarrow \infty$ (Fig. 4.10) and the resonance occurs. In this case critical angular speed

$$\omega_{cr} = \frac{\pi n_{cr}}{30} = \sqrt{\frac{c}{m}}.$$

Critical speed is

$$n_{cr} = \frac{30}{\pi} \sqrt{\frac{c}{m}}.$$

A shaft does not fail suddenly in the resonance zone because there are some resistances (bearing reactions, friction etc.) proportionally to the vibration amplitude. If rotational speed further increases above critical one, the centre of gravity moves on the other side of an elastic line (see Fig. 4.9, c) and equation (4.1) should be rewritten as

$$F_c = m(y - e)\omega^2.$$

Hence, in the zone $n > n_{cr}$ the dynamic deflection is found from dependence

$$y = \frac{m\omega^2 e}{m\omega^2 - c} = \frac{e}{1 - \frac{c}{m\omega^2}} = \frac{e}{1 - \left(\frac{\omega_{cr}}{\omega}\right)^2}.$$

Thus, that at increase in angular speed over critical the dynamic deflection decreases, which is a so-called self-centering of a shaft phenomenon. Therefore, if the shaft passes as quickly as possible through the

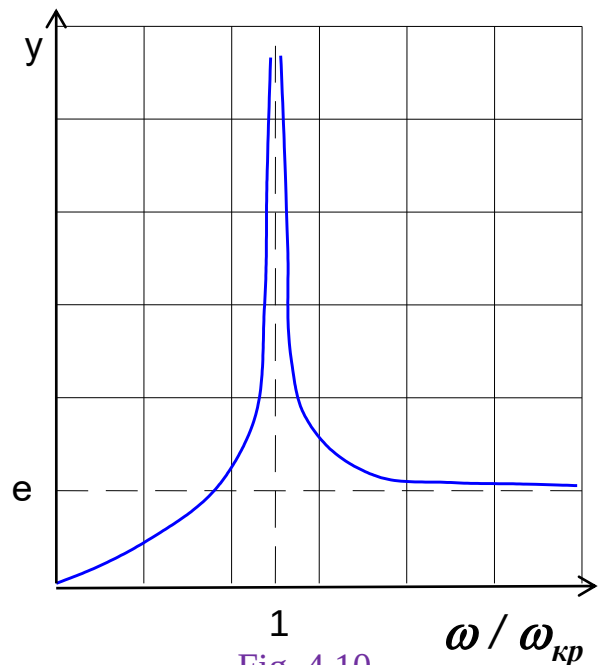


Fig. 4.10

zone of the critical speed, its rotation becomes stable. The shaft deflection has the limit value: if $\omega \rightarrow \infty$ $y \rightarrow e$ (Fig. 4.10).

The shaft with operation speed over critical is called as flexible. At high speeds (20 000 rpm and higher) flexible shafts make not only simple and cheap designs but are the only possible solution.

4.5. Design Rules

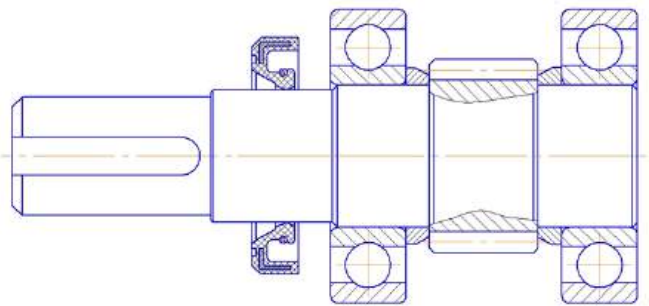
1. *A shaft form should be approached to the form of a body of equal resistance to bending.*

It means that the greatest diameter should be in that place where the maximum bending and twisting moments operate. As a rule, it is somewhere in the middle of a shaft. In other words, shaft diameter should decrease stepwise from the middle to the ends.



2. *Each element should pass till its place start without a tightness.*

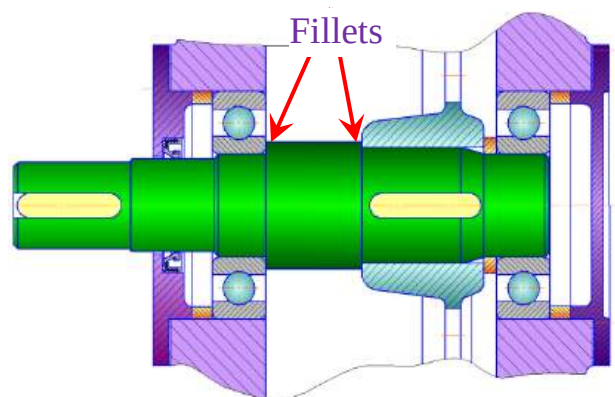
For this purpose it is necessary to provide a separate diameter for each element (a gear, the bearings etc.). If at the shaft end a key is used, it is desirable that the elements mounted more deeply on a shaft passes over a key without its dismantling.



3. *The shaft should be fixed in radial and axial directions.*

For this purpose, two points of a support in a radial direction and in each of axial directions should be provided on the shaft. It means that two radial bearings, taking radial and axial forces, or a combination of radial with thrust or radial-thrust bearing should be provided.

Radial forces are transferred by a shaft neck to a bearing holder. To transfer the axial forces it is necessary to provide fillets or grooves with a split ring on a shaft.



4. *Make chamfers on all connected surfaces (under bearings, wheels, couplings, etc.), because an element assembling is rather complicated without chamfer. The recommended chamfer sizes are:*

d, mm	16...20	20...30	30...50	50...100
c, mm	1	1,5	2	3

5. The shaft neck dimensions (diameter and length) should be corresponded to dimensions of the contact surfaces of the standard elements.

Diameters of the input/output shaft end of should be corresponded with diameters of attached units (for example, with standard coupling diameter). These diameters are taken from tables. If the size strongly differ, it is necessary either to change shaft diameter, or to use transitive bushes.

For the design of axial length of the shaft and various steps in the shaft, the axial length of hub of sprocket, pulleys, bearings, wheels, etc. are determined.

Part 5. BEARINGS

Bearings are the support devices to support the rotating parts in space, enabling them to rotate or swing, and to take acting on them loads.

Rolling (antifriction) bearings and sliding are distinguished. In the sliding bearings rotating part interacts with the stationary surface directly or through the layer of lubricant. In a rolling bearing – through the rolling elements. Antifriction bearings are standardized and produced centrally and in large numbers.

5.1. Rolling Bearings

Advantages	Disadvantages
– small start friction losses	– high rigidity due to the small contact area of operating elements
– high reliability against seizure	– large radial size
– simple servicing	– high vibrating activity
– relatively low cost of standard sizes	– limited service life especially at high loads and speeds
– lower consumption of lubricating materials	– different service life under the same conditions

5.1.1. Design of Rolling Bearings

Rolling (antifriction) bearings operate due to rolling resistance and utilize balls or rollers for rotation.

The simplest antifriction bearing (Fig. 5.1) consists of the outer ring, the separator (cage), which separates rolling elements, the rolling elements (balls or rollers), and the inner ring. Cheap bearings may have no separator. However, this element can be very important to prevent rubbing contact between balls and sidewalls.

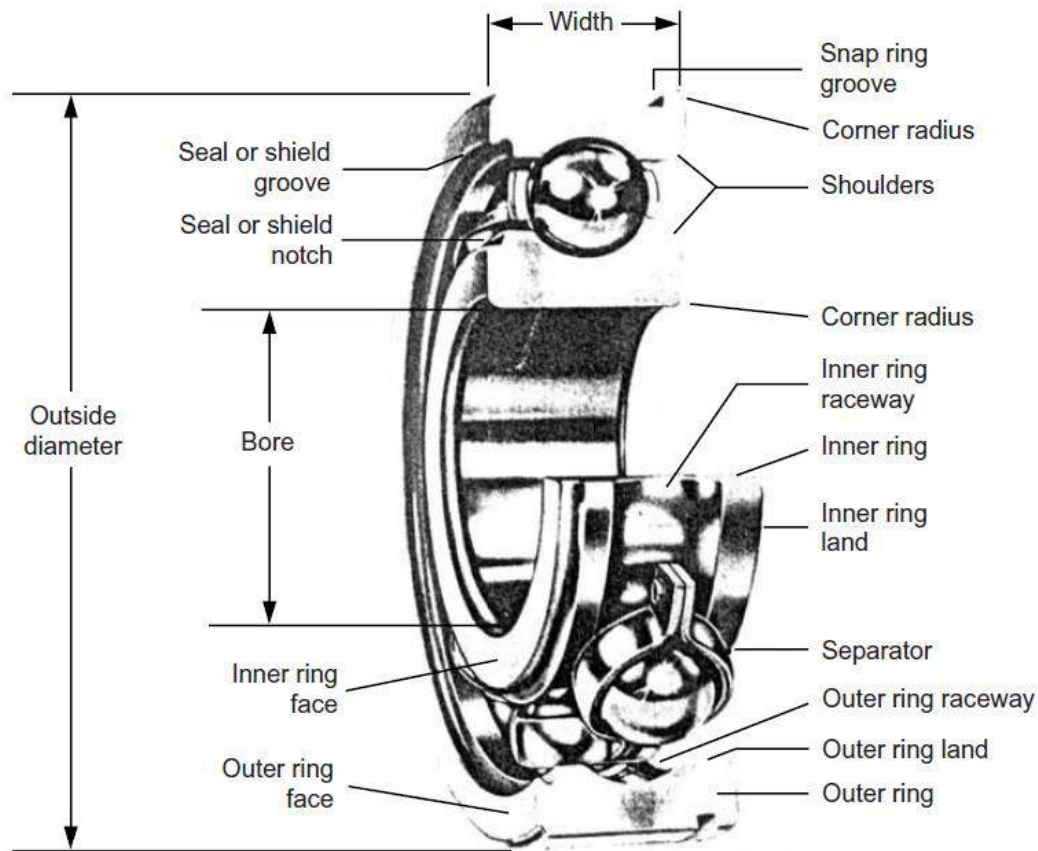


Fig. 5.1

The bearing industry uses different materials for the production of the various bearing components. The bearing materials are processed to achieve desirable properties to maximize bearing performance and life.

The bearing rings, balls and rollers are made of chromium steel or chrome-nickel steel, etc. For example, steel SAE 52100 is an excellent general purpose bearing steel. Due to its excellent hardness and wear resistance, it exhibits good fatigue life in rolling element bearings. However, the corrosion resistance of chrome steel is poor because of the low chromium content. The surfaces of the bearings must be protected with a coating of rust inhibitor or oil to stop oxidation.

Bearing components are heat-treated to hardness HRC 62...65, ground, and thoroughly polished.

Bearings made of chrome steel can function at temperatures up to 120°C, and upon heat treatment, they can even go as high as 220°C. However, one disadvantage of chrome steel bearings is the low chromium content that makes them prone to corrosion.

The separators are made of sheet steel (press formed cages), bronze, aluminum alloys (cast cages) or fabric-based laminated plastics.

5.1.2. Classification

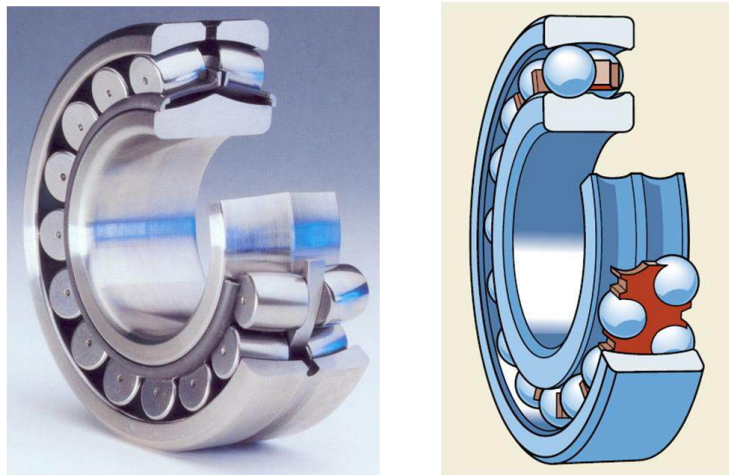
By Rolling Elements



By Load Direction



By Self-Alignment



self-alignment

Bearings of different series have the same bore diameter and differ in the outside one, width, the number of revolutions per minute, the permissible radial or axial load, and the operational factor (Table 5.1).

Changing in dimensions of bearings of different series with a constant bore diameter d is shown in Fig. 5.2.

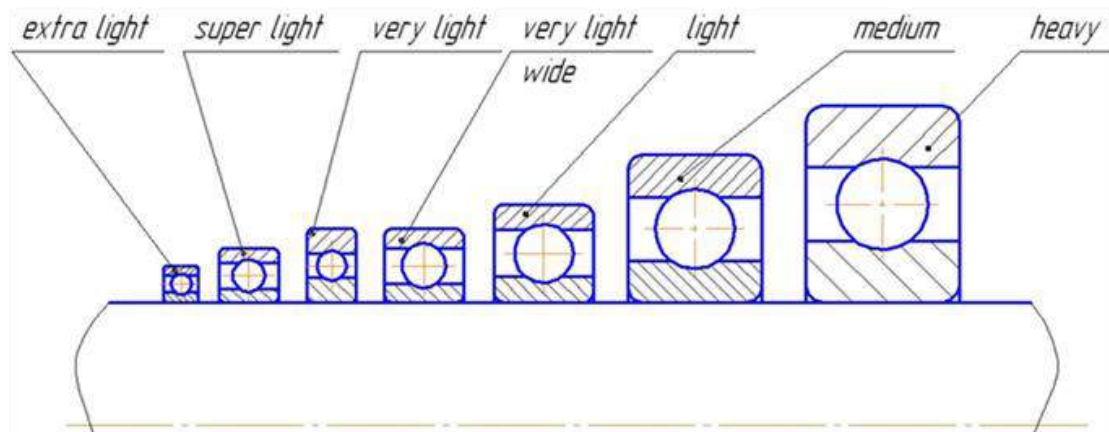


Fig. 5.2

Table 5.1

Series	Code	Outside diam. D , mm	Width B , mm	Dynamic load capacity C , κN	Relative price	Weight, kg
Very light	110	80	16	21,6	1	0,255
Light	210	90	20	35,1	0,55	0,460
Medium	310	110	27	61,8	0,85	1,07
Heavy	410	130	31	87,1	1,3	1,9

5.1.3. Description of Most Widely Used Bearing Types

1. *Deep groove ball bearing* (Fig. 5.3) is the mass type of bearings. It mainly takes radial load but can operate under bilateral axial load. It works well at ring axes angular misalignment up to $\frac{1}{4}^\circ$. It has relatively high operating speed and is cheaper in comparison with the other types.

Balls are inserted into grooves by moving inner ring to an eccentric position. They are separated after loading, with the separator installed. Sometimes manufacturers use a filling notch in inner and outer rings to enable the greater number of balls to be inserted, thus increasing the bearing capacity. Rated thrust load of the bearing decreases as balls tend to bump against the edge of the notch when thrust loads are applied.

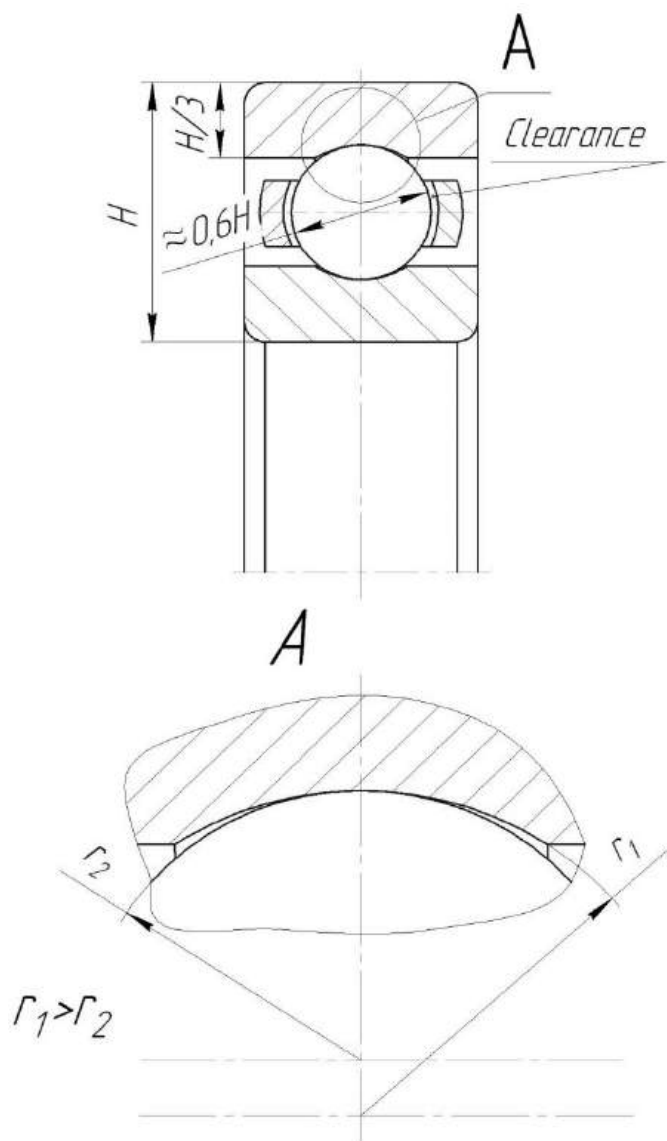


Fig. 5.3

2. *Self-aligning double-row ball bearings* (Fig. 5.4) take radial load at considerable ring axes angular misalignment ($2...3^\circ$) due to manufacturing and mounting errors as well as considerable shaft deflection under loading.

Bearings can be fixed in bilateral direction and take small axial load. They are self-aligning because of their spherical outer ring raceway.

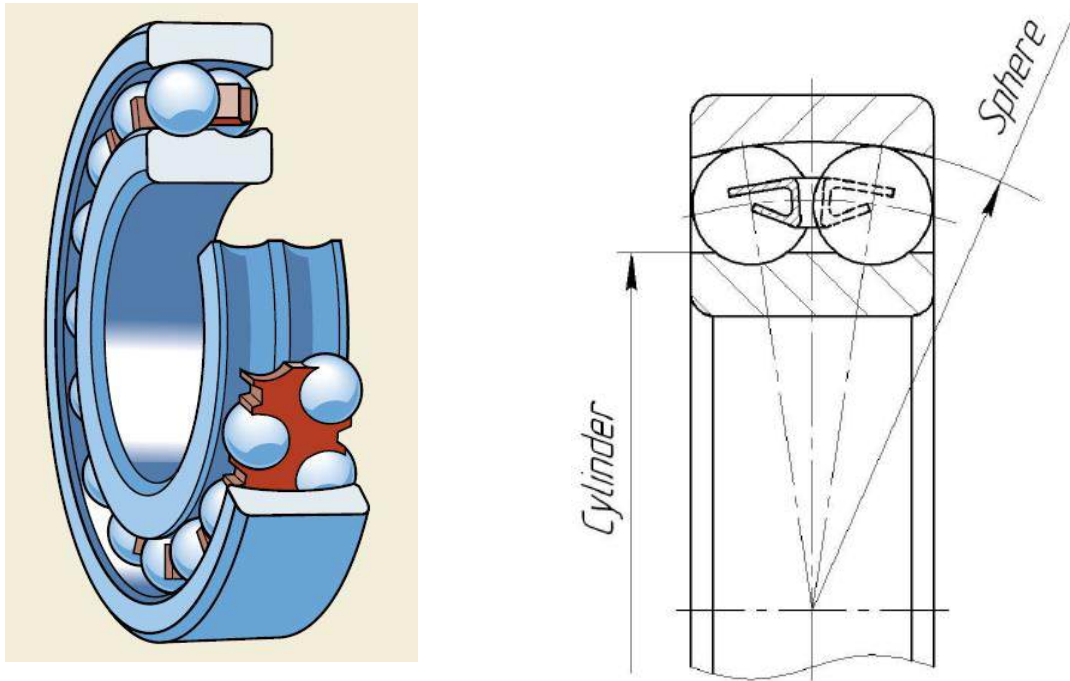


Fig. 5.4

3. *Cylindrical roller single-row bearing* (Fig. 5.5) usually takes only radial loads and sometimes random axial ones. Its load capacity exceeds load capacity of a deep groove ball bearing of identical shaft diameter in 1.7 times. It can be disassembled in an axial direction. Some axial displacement of rings is allowed; therefore, it is recommended for using at considerable shaft temperature deformations or in case of axial shaft self-installation (Fig. 5.5 *a, b*). Ring axes angular misalignment is not allowed.

To fix shafts in one axial direction, bearings with one shoulder (Fig. 5.5, *c*) are used. Bearings with a shoulder and an additional demountable washer (Fig. 5.5, *d*) are needed for bilateral fixing.

4. *Needle roller bearing* (Fig. 5.6) is used to take considerable radial loads without axial ones at constrained radial sizes and the tangential velocity up to 5 m/s. Long cylinder rollers (so called needles) have diameter of 1,6...5 mm and 5...10 times greater length. There are bearings with two rings (Fig. 5.6, *a*) but at very small dimensions, types with one ring (Fig. 5.6, *b*) can be used as well as without rings.

In cageless needle bearings, total peripheral clearance (between the first and the last needles) is 1...2 mm while radial clearance between needles and raceways is considerably larger than in ball and roller bearings and approximately equals the radial clearance of sliding-contact bearings of the same diameter.

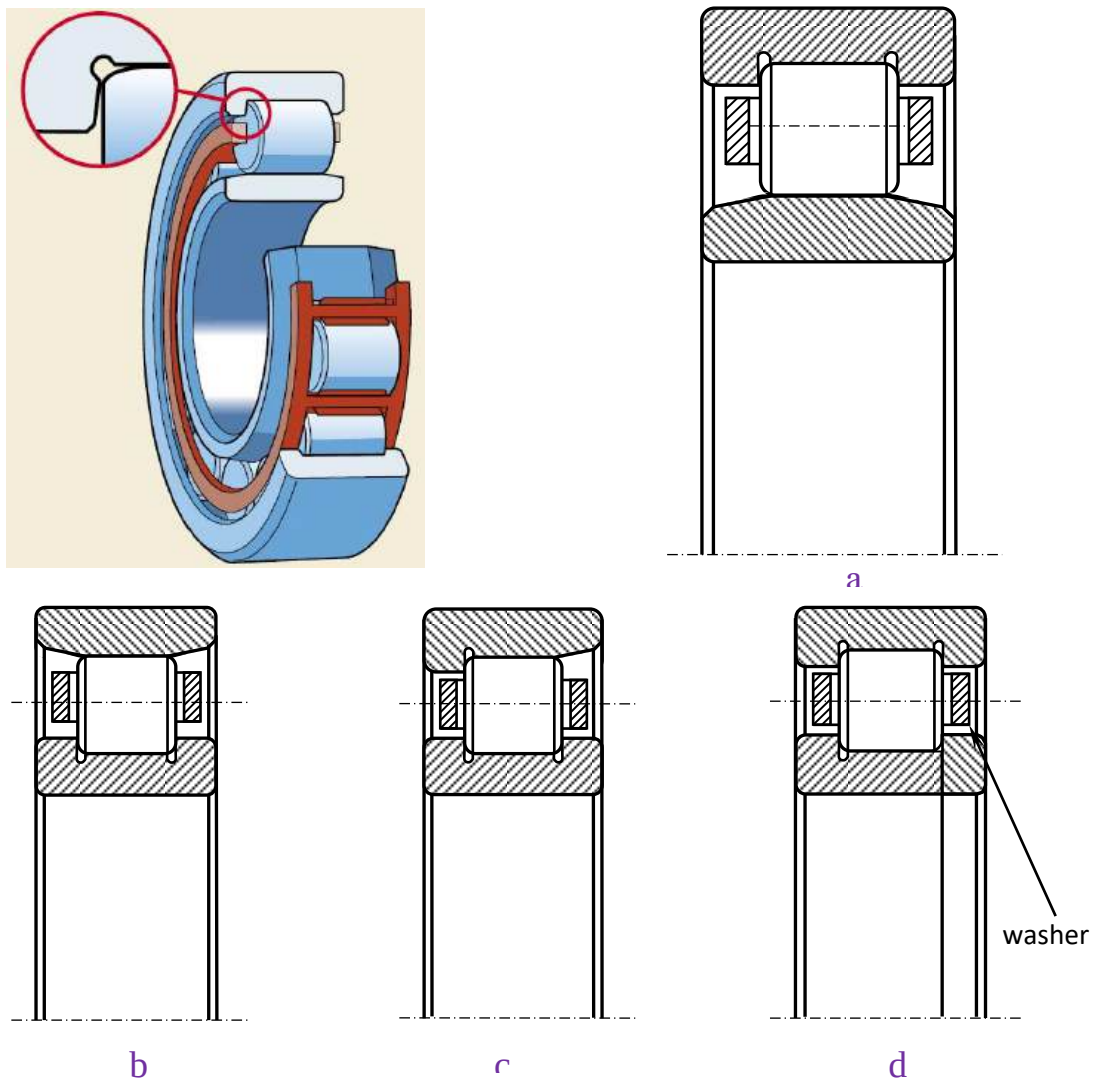


Fig. 5.5

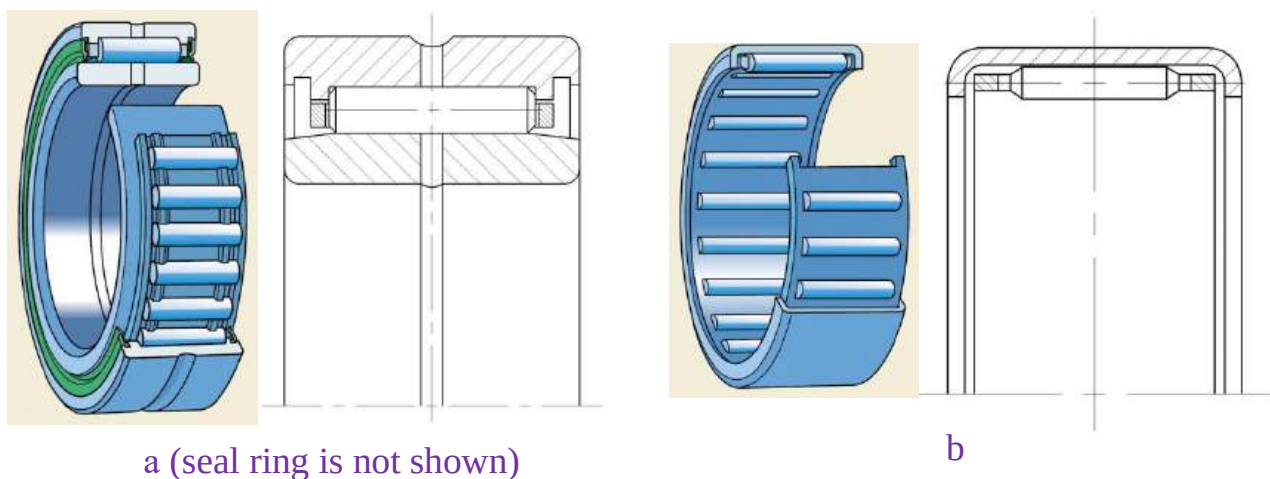


Fig. 5.6

5. *Spherical double-row roller bearing* (Fig. 5.7) can take especially large radial loads at considerable misalignment of rings ($2...3^\circ$). Bearings possess high operational parameters but are the most difficult in manufacturing. They are self-aligning.

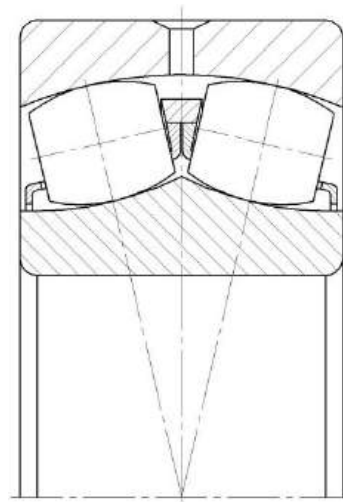


Fig. 5.7

6. *Angular contact single-row ball bearings* (Fig. 5.8) are designed so that contact between their races and bearing balls occurs at some angle relative to the bearing operating axis. That is why this type can operate with heavy unilateral thrust loads and moderate radial loads. Bearings are made with standard contact angles between balls and rings $\beta = 12^\circ, 26^\circ, 36^\circ$. To transfer bilateral axial loads, two bearings are mounted on a shaft.

Radial load capacity about 30...40% exceeds one of radial single-row ball bearings with the same shaft diameter. Angular contact single-row ball bearings are applied at average and high rotational speeds.

Angular contact bearings are assembled with a preload applied between the inner race, the bearing ball, and the outer race. It minimizes or removes gaps between bearing balls and inner and outer races.

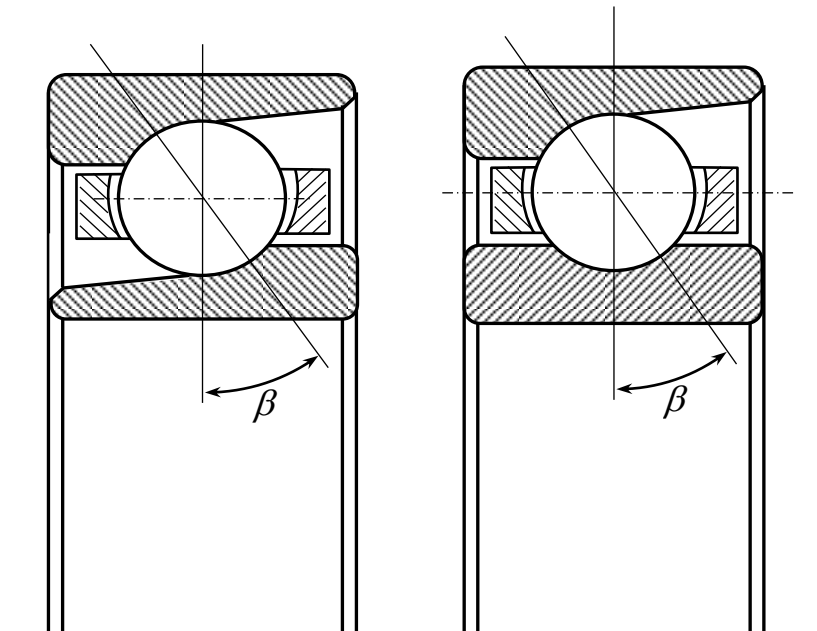


Fig. 5.8

7. Angular single-row ball bearings with three- or four-point contact (Fig. 5.9) carry very large radial and bilateral axial loads. Radial load capacity is more than 1,5 times bigger the load capacity of deep groove bearing. To assemble this bearing, one of rings is cut. Contact angles are $\beta = 12...35^\circ$, at large axial load, $\beta = 35...45^\circ$.

Bearing possess high rapidity (tangential velocity up to 20 m/s).

8. Tapered roller bearings (Fig. 5.10) are widely used in mechanical engineering. They carry considerable radial and unilateral axial loads at low and average roller tangential velocity (up to 15 m/s). Radial load capacity is 1,9 times more than that of deep groove ball bearings. This bearing type is convenient in assembling and dismantling but requires adjusted backlashes and wear clearance compensation.

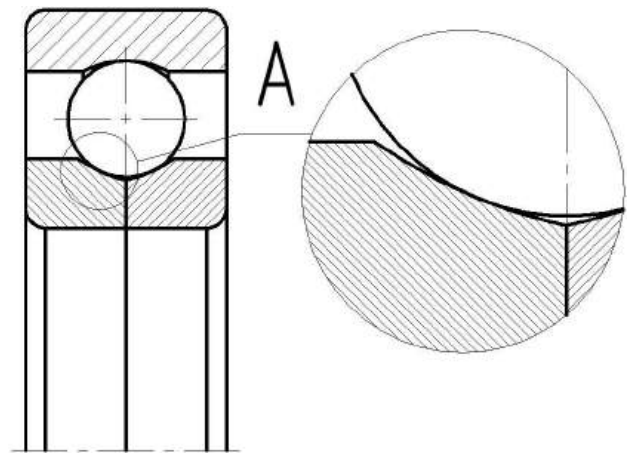


Fig. 5.9

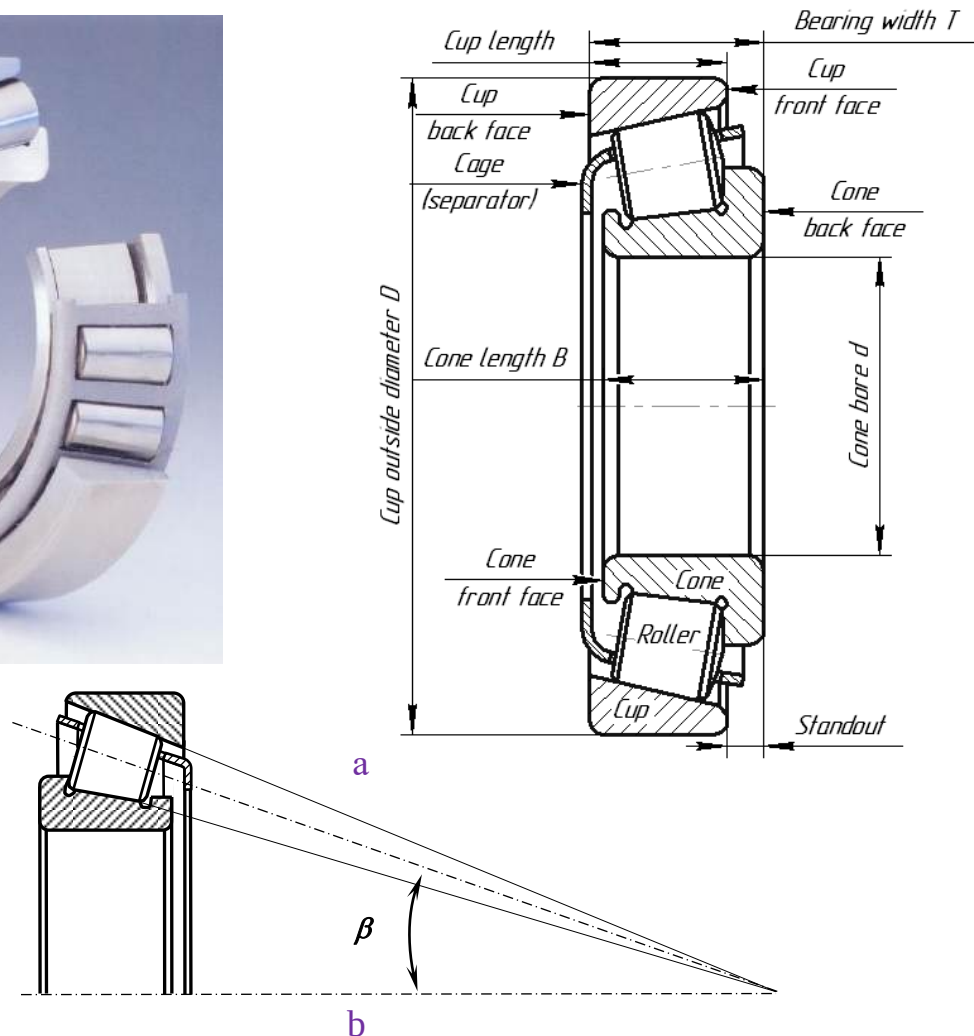


Fig. 5.10

To ensure pure rolling, cone vertexes of rollers, cup, and cone raceways should coincide and be located on the bearing axis (Fig. 5.10, *b*). A contact angle (half of the angle at a cone vertex of a cup raceway) is $\beta = 10...16^\circ$, sometimes $25...30^\circ$. Roller angle of the taper is about $1,5...2^\circ$.

9. *Thrust ball bearings* (Fig. 5.11) can take axial loads only. Single thrust ball bearings (Fig. 5.11, *a*) operate at an unilateral loading (shaft fitting diameter d a little smaller than d_1), double ones (Fig. 5.11, *b*) at the bilateral loads.

It works well at tangential speeds up to 10 m/s. At higher values, great centrifugal forces and gyroscopic moments act on balls and cause considerable friction.

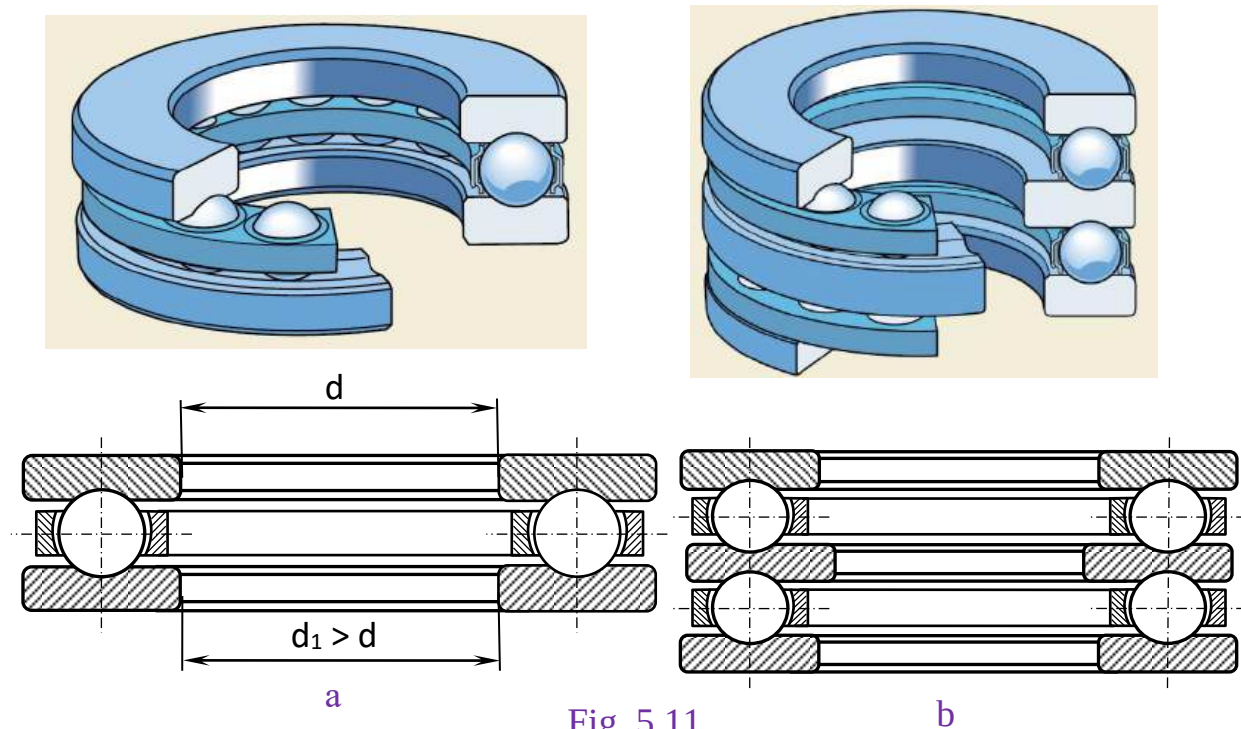


Fig. 5.11

5.1.4. Accuracy

Degree of accuracy depends on the bearing rotation speed and operational conditions (the first of them is required reliability). Usually bearings of the 0 (normal) degree are used on general-purpose shafts.

Cost of bearings essentially increases with growing degree of accuracy.

According to GOST 520-71, following classes of bearings accuracy are produced (in order of accuracy increasing): 0, 6, 5, 4 and 2:

Degree	0	6	5	4	2
Radial play, μm	20	10	5	3	2,5
Roughness R_a , μm	1,25	0,63	0,63	0,32	0,32
Tolerance, μm	23	16	9	7	5

5.1.5. Designation

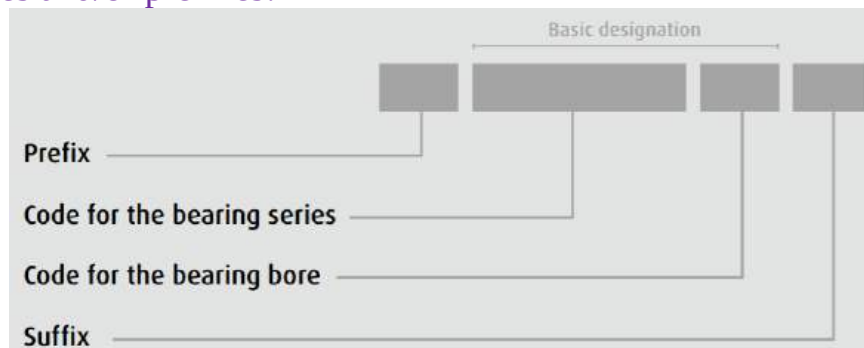
Bearing and manufacturer's codes are marked at ring faces:



Order codes for bearings are made up of a combination of letters and numbers. The code consists of the basic part and (in necessary cases) the additional one.

The manufacturers' basic designation is digital and contains up to 7 figures (GOST 3189-75). Code figures are defined by the places numbered from right to left (Table 5.2). Each figure denotes a certain feature of the bearing.

This alphanumeric combination denotes the type, size and structure of the bearing. Within each order code, a distinction is made between the basic designation and any suffixes and/or prefixes:



The basic designation indicates the bearing type and the bore size. These basic designations are defined in the relevant ISO standard. Prefixes and suffixes denote special modifications. The use of prefixes and suffixes is only partially standardized. Suffixes differ the most, with the various bearing manufacturers using different codes for certain modifications. In some cases, the same bearings use brands' different supplementary designations.

Zeros to the left of the last significant figure are rejected and not marked.

Degree of bearing accuracy is indicated by the figure preceding the bearing code and separated from it by a dash.

The diagram below shows how the order codes are made up. There should be spaces separating the individual sections of an order code from one another.

Table 5.2

Figure place in code (from right to left)	Figure meaning	Table
7 6 5 4 3 2 1	Shaft (bore) diameter	5.3
7 6 5 4 3 2 1	Bearing outside diameter series	5.4
7 6 5 4 3 2 1	Bearing type	5.5
7 6 5 4 3 2 1	Design features	-
7 6 5 4 3 2 1	Bearing width series	5.4

Table 5.3

Bore diameter d , mm	Signification
1...9	One figure is used and it shows actual size d , mm
10	00
12	01
15	02
17	03
20...495	d divided by 5 (04; 05; 06...99)

Table 5.4

Outside diameter series	3 rd figure	Width series	Signification	Series combination	
			7 th figure	3 rd figure	7 th figure
Extra light	8	Normal	1	8	1
		Wide	2	8	2
		Super wide	3	8	3
	9	Narrow	7	9	7
		Normal	1	9	1
		Wide	2	9	2
		Super wide	3, 4, 5, 6	9	3, 4, 5, 6
	Super light	1	Narrow	7	1
Normal			0	1	0
Wide			2	1	2
Super wide			3, 4, 5, 6	1	3, 4, 5, 6
7		Narrow	7	7	7
		Normal	1	7	1
		Wide	2	7	2
		Super wide	3	7	3
Light	2 or 5	Narrow	0	2	0
		Normal	1	2	1
		Wide	0	5	0
		Super wide	3	2	3

End of Table 5.4

Outside diameter series	3 rd figure	Width series	Signification	Series combination	
			7 th figure	3 rd figure	7 th figure
Medium	3 or 6	Narrow	0	3	0
		Normal	1	3	1
		Wide	0	6	0
		Super wide	3	3	3
Heavy	4	Narrow	0	4	0
		Wide	2	4	2

Table 5.5

Bearing type	4 th figure
Deep groove ball	0
Self-aligning two rows ball	1
Cylindrical roller single row	2
Spherical two rows roller	3
Needle roller	4
Roller bearing with wound rollers	5
Angular contact single row ball	6
Tapered roller	7
Thrust ball	8
Thrust roller	9

5.1.6. Kinematics

The relative motions of a cage or a separator, the rollers, and the races of cylindrical rolling element bearings are important to understand their performance. The relative velocities in a ball bearing are somewhat more complex than those in roller bearings, for that reason the ball bearing is used as an example here to develop approximate expressions for relative velocities.

The linear velocity of the cage and the roller can be obtained by the average of linear velocities of inner and outer races (Fig. 5.12):

when inner race rotates: $V_{01} = \frac{V_1}{2}$, when outer race rotates: $V_{02} = \frac{V_2}{2}$.

The linear velocity of the inner race and the outer race are obtained from the angular velocity of inner and outer races respectively:

when inner race rotates: $V_1 = \omega \frac{D_1}{2}$, when outer race rotates: $V_2 = \omega \frac{D_2}{2}$.

It is clear that $D_2 > D_1$ and $V_2 > V_1$. Thus, in the case when the outer race rotates, centrifugal load will be higher and wear will be more rapid than in case when the inner race rotates.

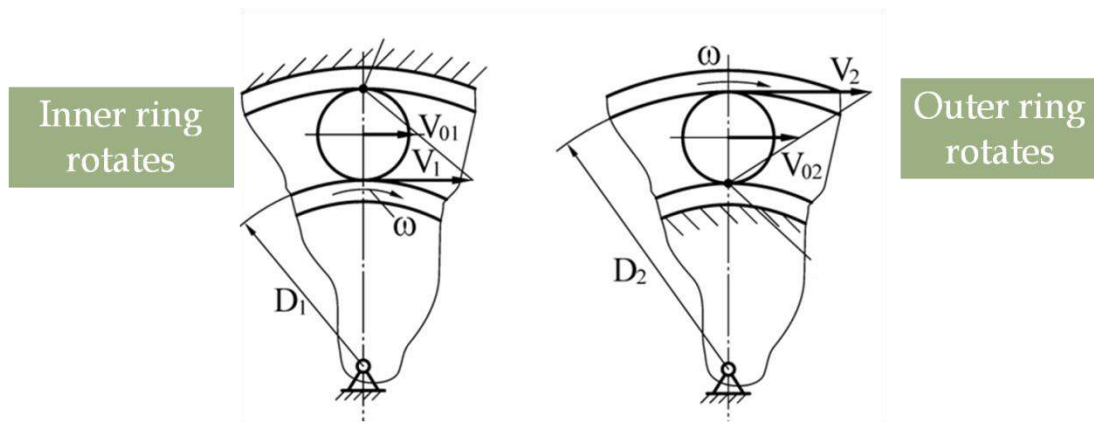


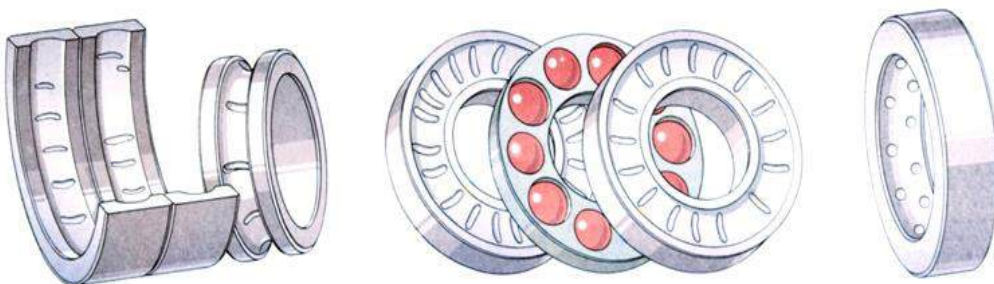
Fig. 5.12

5.1.7. Types of Failures

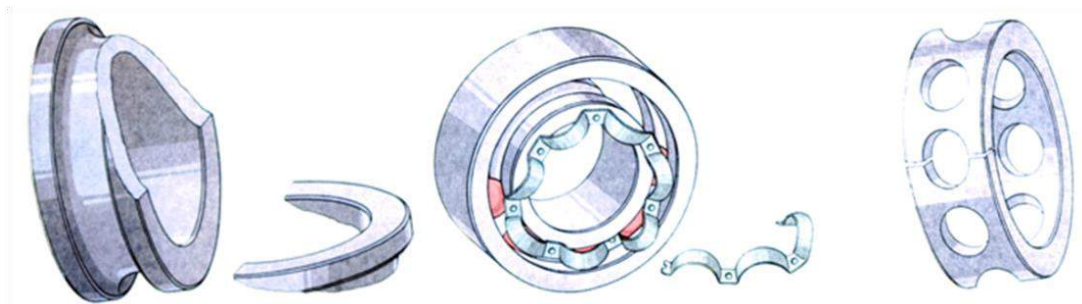
The Working Surfaces Pitting



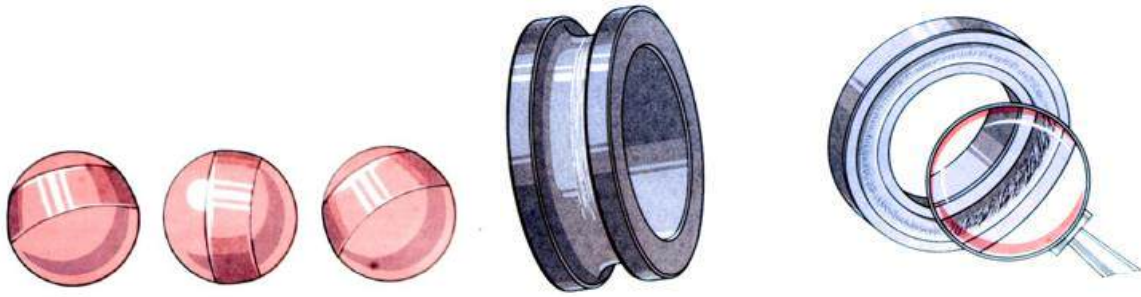
The Dents on Working Surfaces



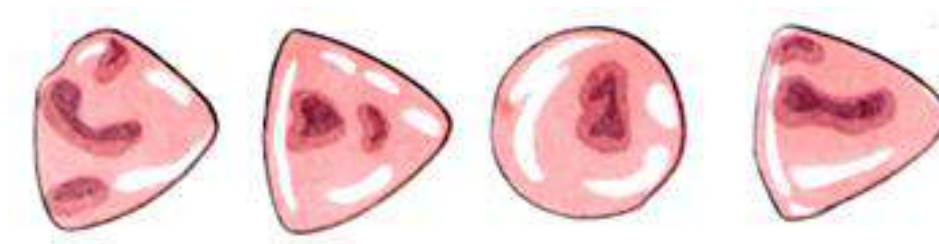
Ring and Cage Break



The Working Surfaces Wear



The Rolling Elements Heat Deformation



5.1.8. Calculation on Static Load Capacity

The bearings are calculated on a static load at $n < 1$ rpm. Static load capacity C_0 is a force, which causes total remnant deformation of rolling elements and rings

$$\delta_{sum} \leq 0,0001D_W,$$

where D_W is the rolling element diameter.

Condition should be satisfied: $P_0 \leq C_0$, where C_0 is found from a catalogue.

Equivalent static load: $P_0 = X_0 F_r + Y_0 F_a$, where F_r and F_a are the radial and the axial forces; X_0 and Y_0 are the radial and axial load factors (in catalogues).

The required load capacity of a thrust bearing is $C_0 = F_a S$, where $S = 1,2 \dots 1,5$ is the safety factor.

5.1.9. Calculation of Bearings on Dynamic Load Capacity

The type of bearing capacity calculation depends on the ring rotational speed. Bearings are selected on static load capacity if external load is motionless or rotates slow ($n \leq 1$ rpm). At the ring rotational speed $n > 10$ rpm bearings are calculated on dynamic load capacity. The bearings working at the ring rotational speed $n > 10$ rpm and under sharply variable load, also it is necessary to check on the static load capacity.

The dynamic load capacity is a force that is taken by bearing during one million revolutions without fatigue damages.

Bearing type and the positioning scheme are preliminary selected for the shaft supports. Bearings of one type and one size according to the load capacity of the most loaded support are usually used for general-purpose gearboxes. If it is impossible to decide, what support is more loaded, both supports are calculated in parallel up to the values of equivalent load and then more loaded support is selected.

Calculation is based on the known equation of fatigue curve (Fig. 5.13)

$$P^p L = C^p, \quad (5.1)$$

where P is the equivalent load, N; L is the service life in million revolutions; C is the required dynamic load capacity, N; p is the exponent: for ball bearings $p = 3$, for roller bearings $p = 10/3$.

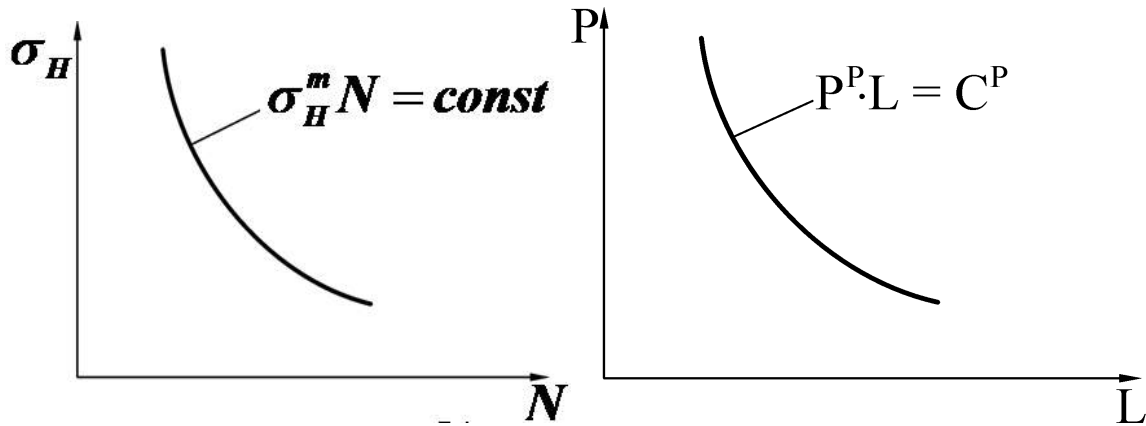


Fig. 5.13

It is possible to define dynamic load capacity from equation (5.1) if P and L are known:

$$C = \left(\frac{L}{a_1 a_{23}} \right)^{\frac{1}{p}} P, \quad (5.2)$$

and then select a bearing which satisfies condition

$$C \leq C_c, \quad (5.3)$$

where C_c is the bearing dynamic load capacity taken from a catalogue or calculated on the empirical dependences.

Also on the base of known P and C_c a preliminary selected bearing service life can be calculated as

$$L = a_1 a_{23} \left(\frac{C_c}{P} \right)^p, \quad (5.4)$$

and then calculated value should be compared with required one: $L_c \geq L_{req}$.

Service life in million revolutions L and in hours L_h are connected by equation

$$L = \frac{60nL_h}{10^6}, \quad (5.5)$$

where n is the rotating ring rotational speed, rpm.

In formulae (5.2) and (5.4) a_1 is the factor considering of reliability:

Reliability	0,90	0,95	0,96	0,97	0,98	0,99
a_1	1	0,62	0,53	0,44	0,33	0,21

The factor a_{23} considers bearings' material, lubricant and operation conditions:

1) usual conditions;

2) without bearing axes shift and with a lubricant film in contacts;

3) the same, as in 2), but at manufacturing of rings and rolling elements electroslog or vacuum steel is used.

Values of factor a_{23} are shown in Table 5.6.

Table 5.6

Bearing type	Conditions		
	1	2	3
Ball bearings (except spherical)	0,7...0,8	1,0	1,2...1,4
Cylindrical roller bearings and spherical ball one	0,5...0,6	0,8	1,0...1,2
Tapered roller bearings	0,6...0,7	0,9	1,1...1,3
Spherical roller bearings	0,3...0,4	0,6	0,8...1,0

Equivalent load P for various types of bearings can be defined by the following formulae:

– for the radial bearings

$$P = VF_r K_S K_T, \quad (5.6)$$

– for the thrust bearings

$$P = F_a K_S K_T, \quad (5.7)$$

– for radial-thrust bearings

$$P = (XVF_r + YF_a) K_S K_T, \quad (5.8)$$

where F_r and F_a are the radial and axial loads on a bearing, N; V is the rotation factor ($V = 1$ in case when the internal ring rotates about a load vector, $V = 1,2$ – external, Fig. 5.14); K_S is the safety factor considering load non-uniformity (in case of light load $K_S = 1$, with medium non-uniformity $K_S = 1,3...1,8$, with heavy load $K_S = 2...3$); K_T is the temperature factor ($K_T = 1$ at $t < 125$ °C and $K_T = 1,05; 1,1; 1,25; 1,4$ accordingly at $t = 125, 150, 200$ and 250 °C); X, Y are the factors of radial and axial load (they can be obtained from catalogues according to bearing type and axial load factor e).

If condition (5.3) is not satisfied or the service life is too small, it is necessary:

– select bearings of heavier series or other bearing type;

– increase a shaft diameter;

– use two identical radial bearings in one support. In such case it may be considered as one two-row bearing and support dynamic radial load capacity for ball bearings should be assumed as $C_\Sigma = 1,625C$, for roller $C_\Sigma = 1,714 C$; equivalent load and factors' X and Y values are defined as for two-rows bearings.

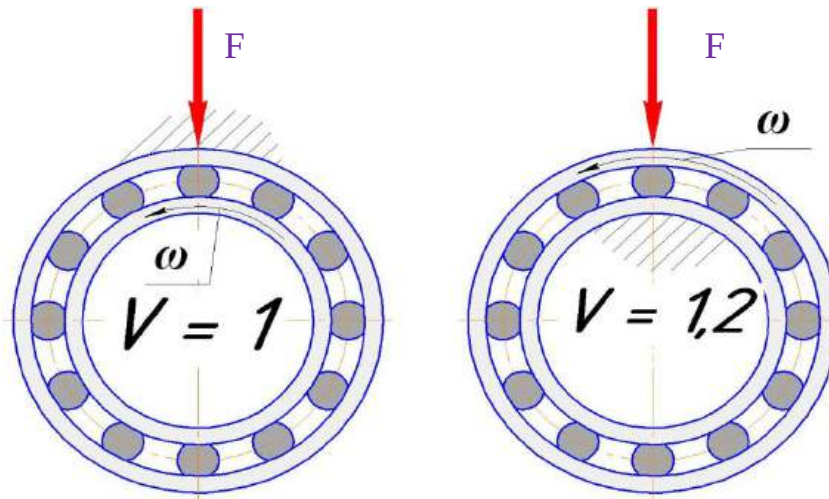


Fig. 5.14

5.1.10. Calculation of Loading on Bearings

To define the equivalent load P the radial F_r and the axial F_a loads should be obtained. Each of them is the total reaction in a support: F_r is a reaction in a plane perpendicular to the longitudinal shaft axis, F_a – along this axis (Fig. 5.15).

The forces acting on shaft and causing reactions F_r and F_a , are rather various by the nature. It can be the forces arising in a tooth gearing, inertia forces, the gyroscopic moments, the forces from a compressor or a turbine, the weight etc. It is accepted that the values, the directions and the points of applying of all external forces are known. If any force direction is not known (for example, from coupling) the most dangerous variant for bearings operating is considered. Thus, the possible error leads to increasing of reliability.

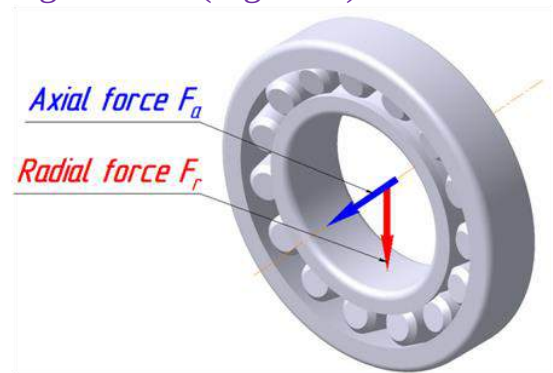


Fig. 5.15

For designing of tooth gearings' support, a normal force F_n and force of friction F_f should be known. A friction force in gearing is usually neglected, because the friction factor is rather small. Normal force F_n is resolved on three components: a radial F_r ; a peripheral F_t ; an axial F_a .

Normal force components can be calculated by following formulae:

– for spur straight teeth gears:

$$F_t = \frac{2T_1}{d_{w1}}; \quad F_r = F_t \tan \alpha_w; \quad F_a = 0; \quad (5.9)$$

– for spur helical gears:

$$F_t = \frac{2T_1}{d_{w1}}; \quad F_r = \frac{F_t \tan \alpha_w}{\cos \beta}; \quad F_a = F_t \tan \beta; \quad (5.10)$$

– for straight teeth bevel gears:

$$F_t = \frac{2T_1}{d_{m1}}; \quad F_r = F_t \tan \alpha_w \cos \delta; \quad F_a = F_t \tan \alpha_w \sin \delta; \quad (5.11)$$

where T_1 is the torque on the driving shaft; d_{m1} is the pinion pitch diameter; α_w is the pressure angle (the most common value without gear correction $\alpha_w = 20^\circ$); β is the teeth helix angle ($\beta = 7...20^\circ$; for strength teeth $\beta = 0$); δ_1 is the pinion pitch cone angle ($\delta_1 = \arctan(z_1/z_2)$); d_{m1} is the bevel pinion mean pitch diameter.

It is assumed, that components of normal force are applied in the middle of a gear face width (for bevel gears – at the mean pitch diameter d_m).

Value of a coupling force F_C depends on accuracy of manufacturing and assembling of the shafts and a coupling. Usually $F_C = (0,1...0,2) F'_C$, where F'_C is the transferring peripheral force. It depends on a coupling type (Table 5.7).

Table 5.7

Coupling type	Force F'_C (T is a transferring torque, Nm)
Flexible coupling with rubber-bushed studs	$2000T/D_1$ (D_1 is diameter where studs are located, mm)
Curved-tooth gear coupling	$2000T/mz$ (m and z are the coupling module (mm) and number of teeth)
Jaw compression coupling with flexible spider insert	$50\sqrt{T}$
Safety multiple-disc slipping couplings	$2000T/d$ (d is a driving shaft diam., mm)

The direction of this force is unknown, therefore it is usually assumed that a coupling force F_C direction is the same with a peripheral force F_t . In this case, bearings are loaded with the maximum forces.

5.1.11. Definition of Reactions

For calculation of radial and axial reactions in supports, a shaft is represented as a beam on one movable pivoting support and one motionless pivoting support. It is assumed, that **radial reactions** are applied in points where a shaft axis crosses the line perpendicular to the middle of the outer ring raceway (Fig. 5.16).

For radial bearings, this point is in the middle of their width (Fig. 5.16, *a*) and distance between supports is the distance between centers of bearings (Fig. 5.17).

For radial-thrust ball and roller the distance a between this point and a bearing's outer ring face (Fig. 5.16, *b*) can be defined graphically or analytically:

– for the angular contact single row ball bearings

$$a = 0,5 [B + 0,5 (d + D) \operatorname{tg} \alpha];$$

– for tapered roller bearings

$$a = 0,5 [T + e (d + D)/3].$$

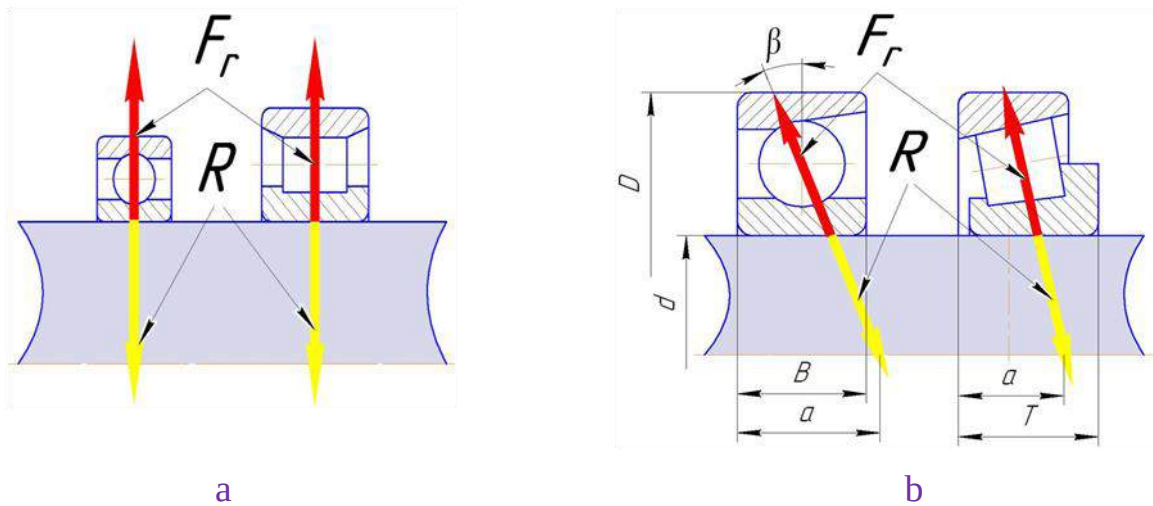


Fig. 5.16

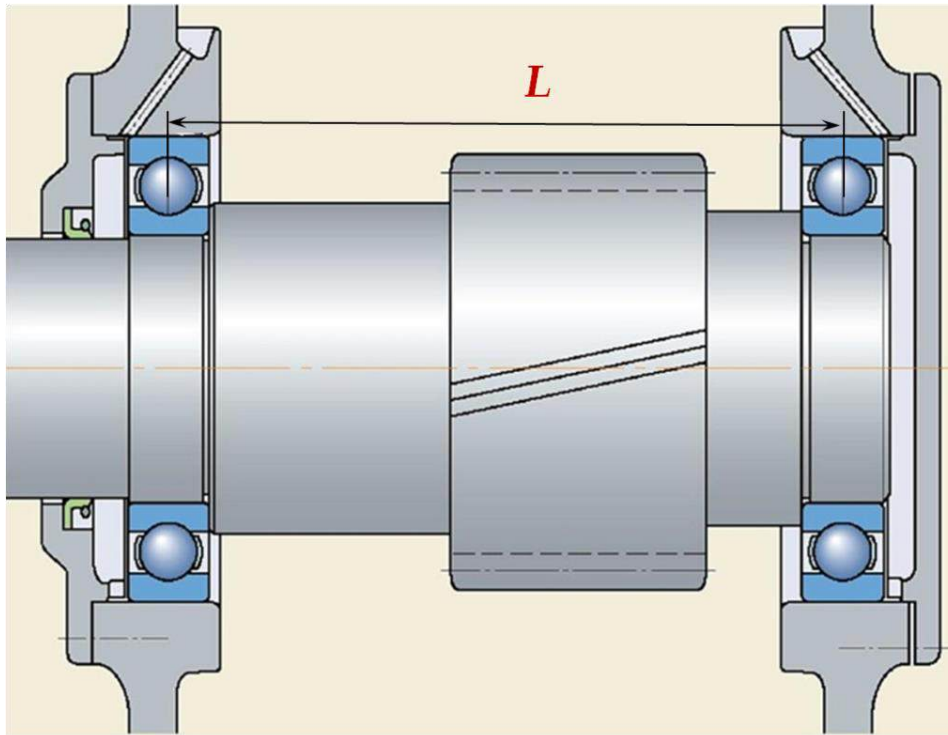


Fig. 5.17

Ring width B , assembly height T , axial load factor e , contact angle α , diameters d and D are shown in the catalogues.

The distance between supports L' (Fig. 5.18) at the same distance between their middles L can be changed by the shifting of the radial reaction applying point.

Radial reactions are defined from the equilibrium equation: the sum of the external moments and the reaction moments is equal to zero (see subsection 4.3).

Values and directions of **the axial reactions** depend not only on values and directions of external loads, but also on the support scheme, types of bearings and on a preliminary tightness. At shaft on two radial ball bearings, the axial reaction R_a is equal

to the actual external axial force F_a . The force is taken by that bearing which fixes a shaft axial shift at one or two directions.

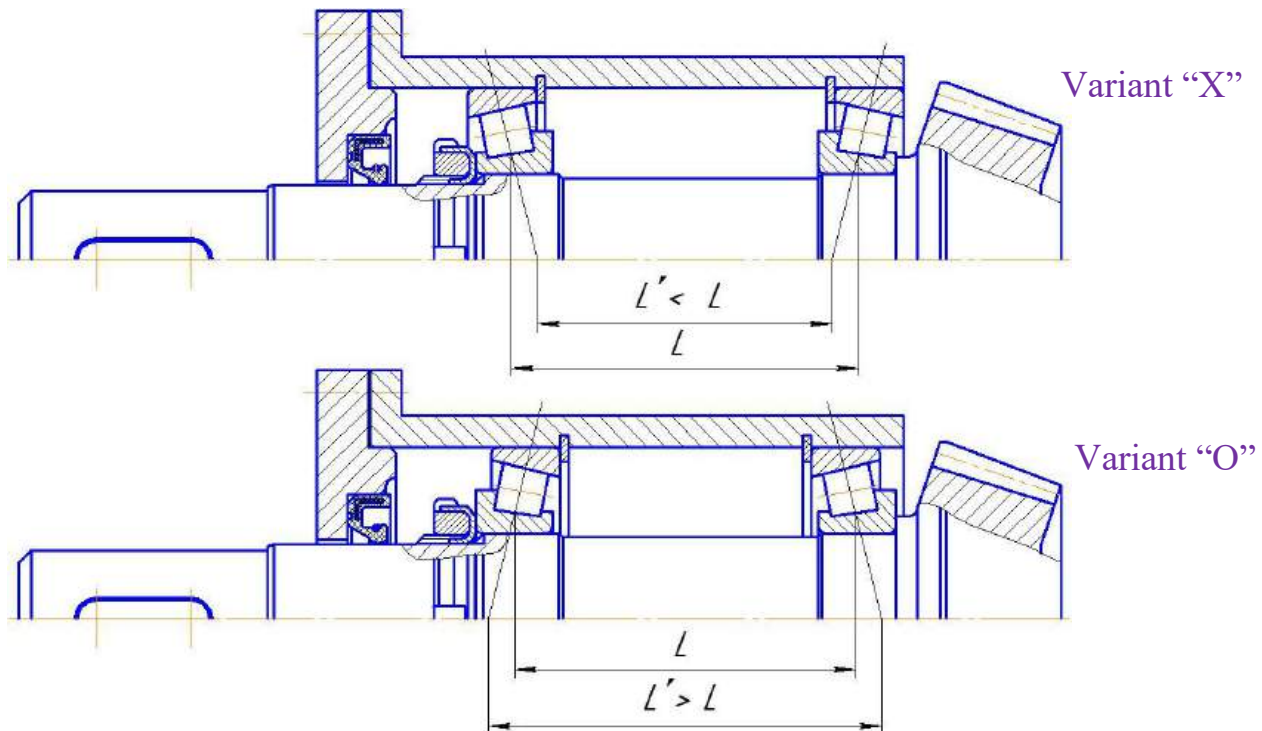


Fig. 5.18

5.1.12. The Limit Speed

The bearings rapidity is limited due to:

- increasing temperature, resulting in reduced lubricant resistance and increasing the risk of rolling elements and rings overheating;
- decreasing durability and strength of separators;
- accelerated fatigue wear associated with a large number of load cycles per time unit.

To estimate bearings rapidity parameters

$$n_{lim} \leq [n_{lim}] \text{ and } d_m n \leq [d_m n]$$

are used. Here $[n_{lim}]$, rpm, and $[d_m n] = (4,5...6,5) \cdot 10^5$, mm·rpm, are taken from the tables.

To increase bearings rapidity:

- use high precision bearings;
- reduce friction losses in the bearings;
- reduce the diameter of the balls, move to lighter series;
- use massive cages made of bronze, brass, aluminum alloys, ceramics;
- improve the lubrication of bearings.

5.1.13. Bearings Installation and Support Types

Ways to fix a bearing on a shaft are shown in Fig. 5.19.

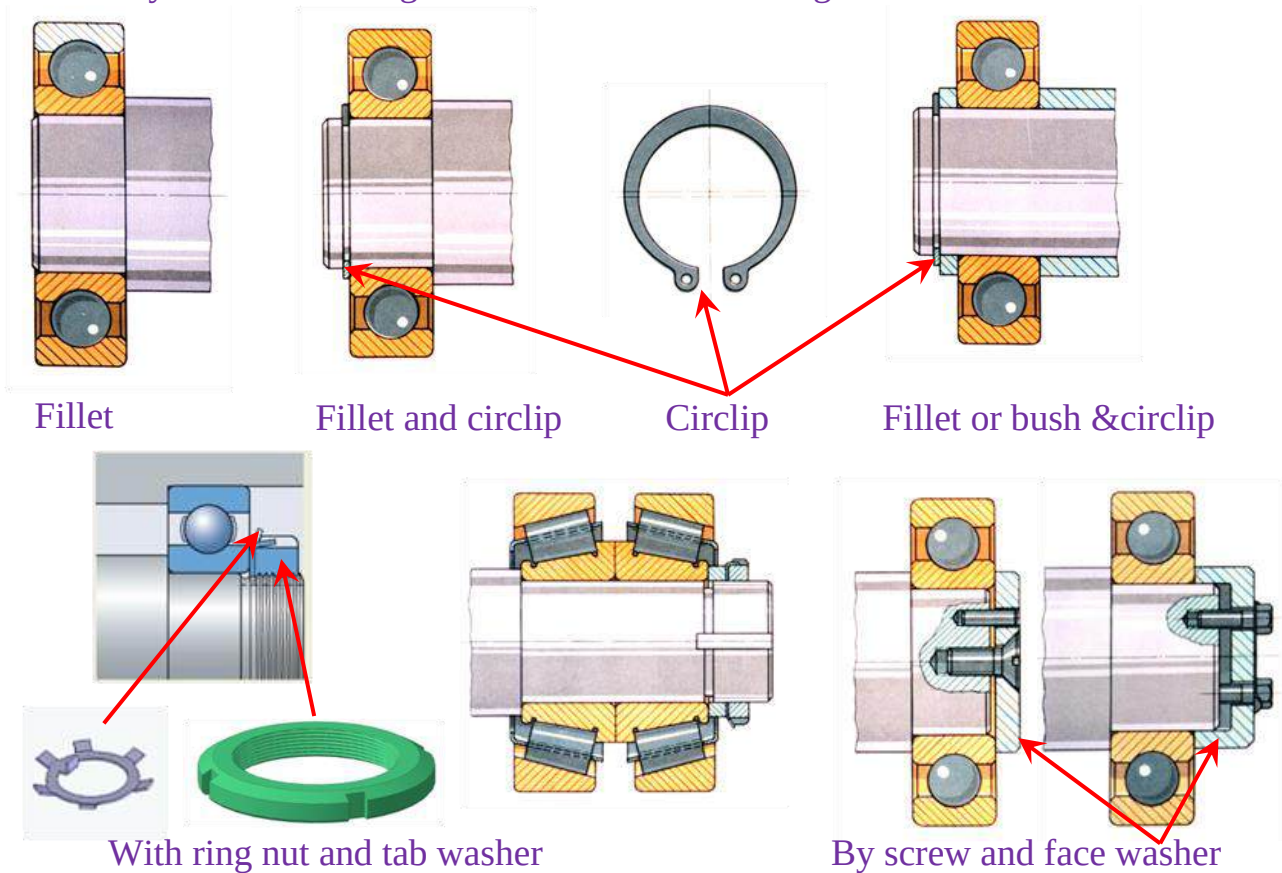


Fig. 5.19

Fixing + Floating Supports

The first (fixing) support takes radial and bilateral axial load (Figs 5.20, 5.21), the second one (“floating”) – the radial only (Figs 5.20, 5.22). This scheme works for shafts of any length.

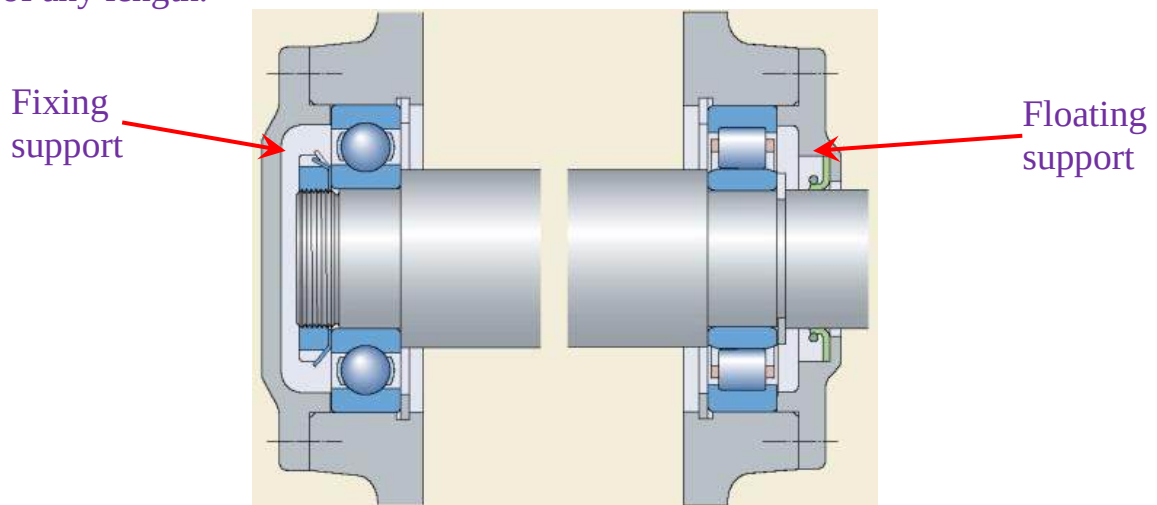


Fig. 5.20
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In a fixing support (Fig. 5.21) both the bearing inner and outer races are fixed on a shaft and in housing in axial and radial direction.

In a floating support (Fig. 5.22) the bearing inner race is fixed on a shaft in axial and radial direction, but its outer race can shift (except for some radial roller bearing types).

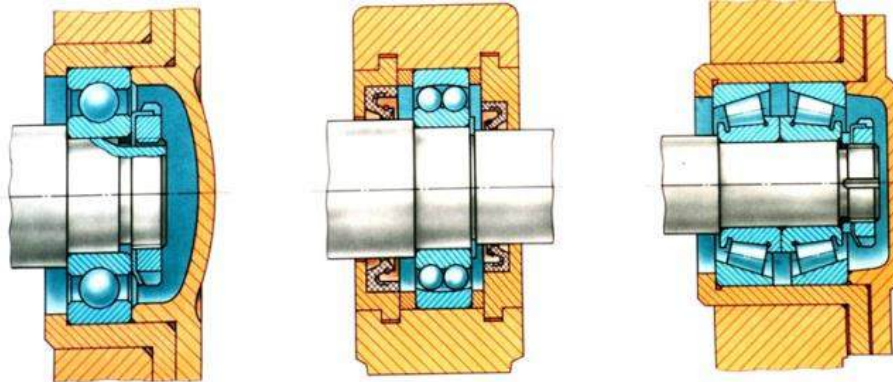


Fig. 5.21

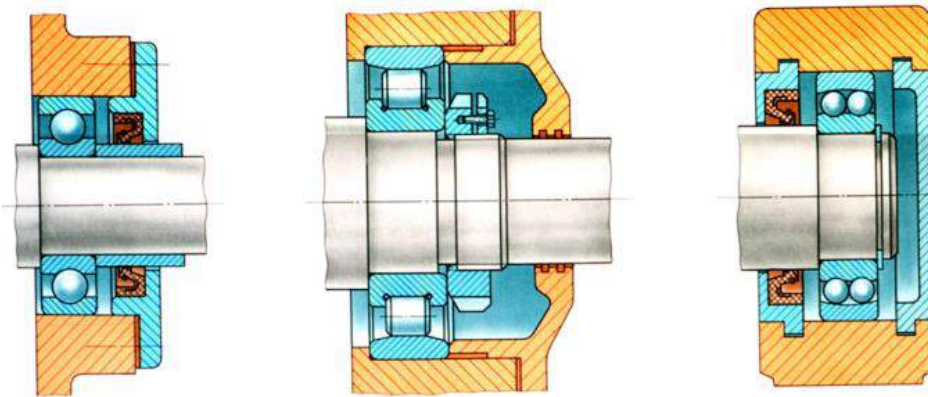


Fig. 5.22

Thrust Type

The scheme works for short shafts (usually with the distance between the supports up to 350 mm) each of the supports takes both radial and unilateral axial load (Figs 5.23, 5.24).

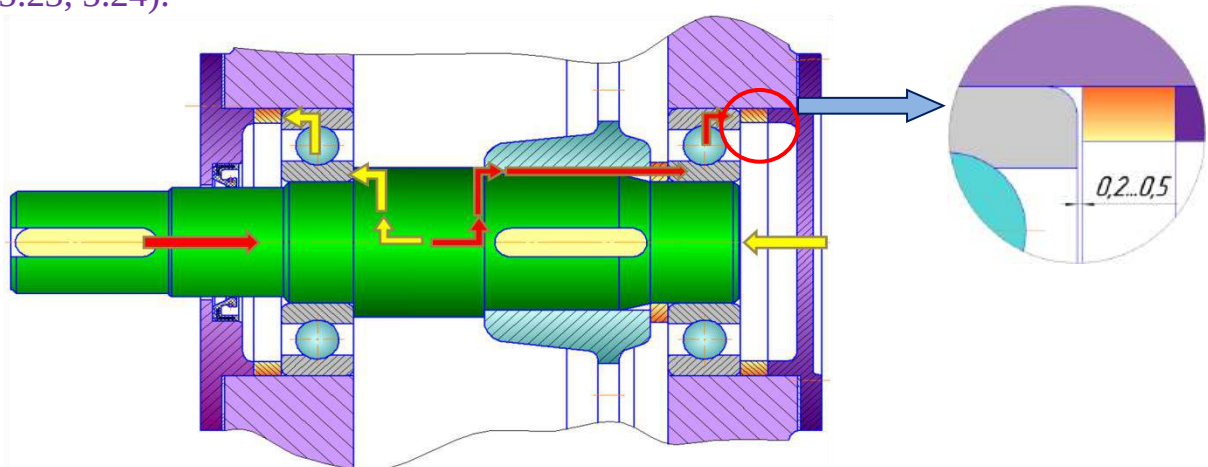


Fig. 5.23

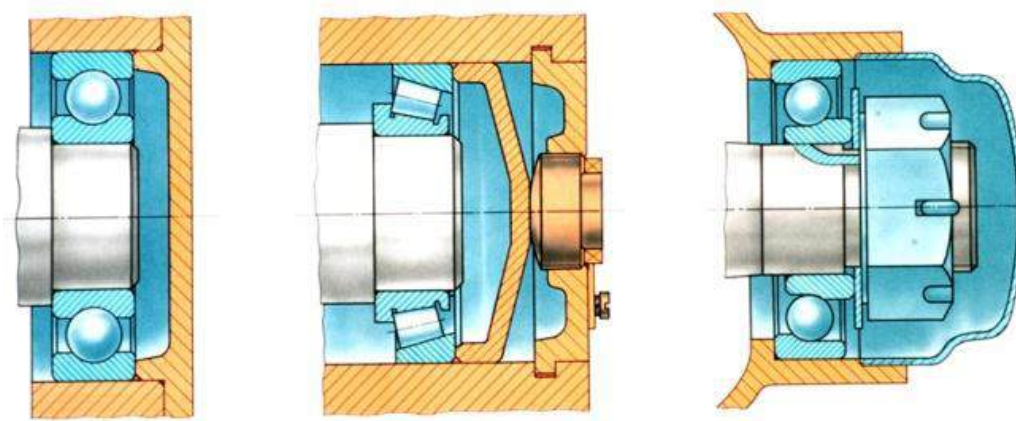


Fig. 5.24

To prevent bearing jams due to the shaft's heat extension and load, one of the supports has a 0,2...0,5 mm gap between its outer race and housing element like a cover (see Fig. 5.23). For deep groove bearing gap is adjusted by metal gaskets (mounting shims) between a cover and a frame, for radial-thrust types gap is usually adjusted by nut (see Fig. 5.24).

Stretching Type

The scheme (Fig. 5.25) works for short shafts (usually with the distance between the supports up to 350 mm). Heat expansion increases gap between outer race and housing.

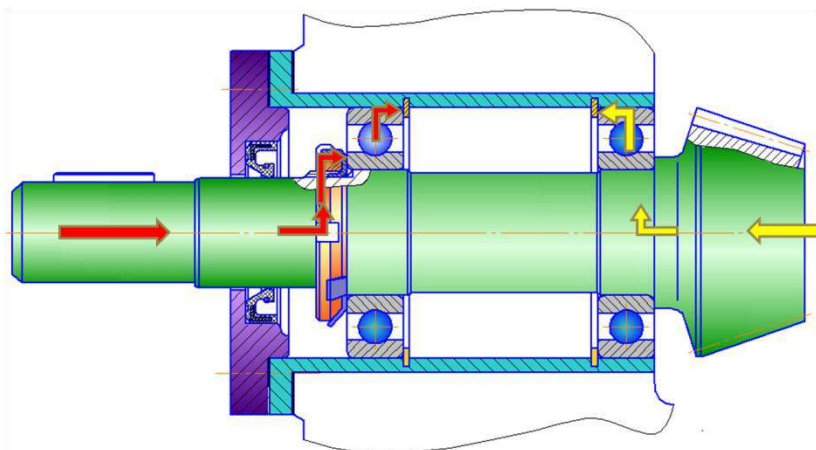


Fig. 5.25

5.1.14. Fit Selection

Type of the bearings installation depends on the load type, the direction and intensity of the active loads.

There are three types of bearing load: local, circulatory and oscillating.

The loading is called **local** when the resulting radial load F_r affecting the bearing is constantly applied to the same part of the raceway (the race is not moving relatively to the active load or both the race and the load participate in the same rotational process, Fig. 5.26).

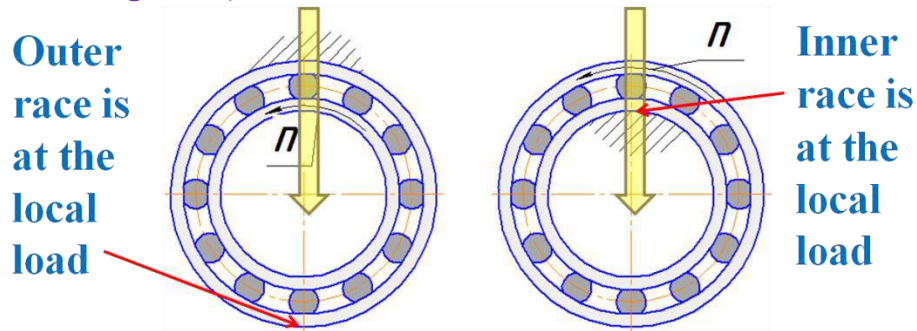


Fig. 5.26

Circulatory load (Fig. 5.27). The resulting radial load is transferred by the rolling elements along all the raceway length, and as the result to the whole shaft surface or frame one (the race is rotating relatively to the constantly directed radial load F_r or the load rotates relating to a static race).

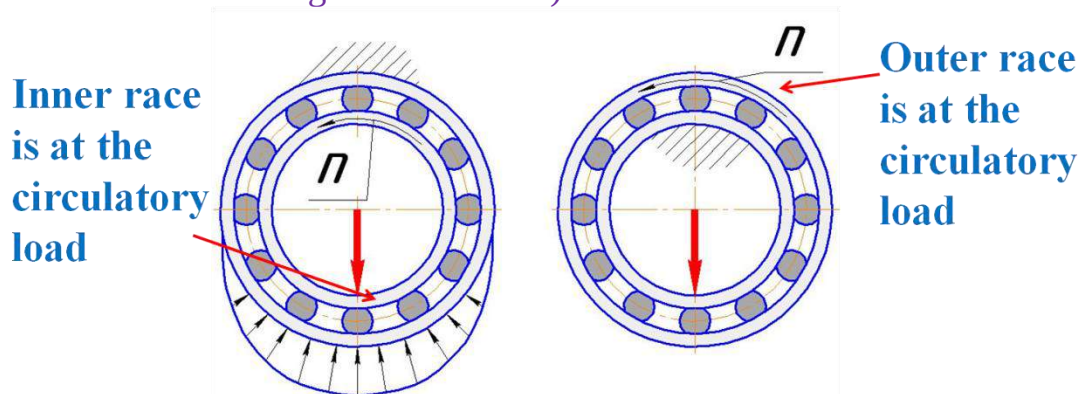


Fig. 5.27

Oscillating collar load (Fig. 5.28) takes place when both constant in its direction F_r and the rotating F_c ($F_r > F_c$) simultaneously affect the fixed bearing race. Their resultant makes a periodical oscillatory move that is symmetrical relatively to F_r 's direction.

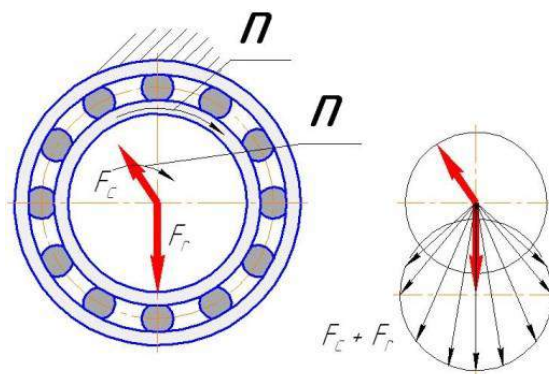


Fig. 5.28

Requirements to the fit selection:

1. Adjust shaft and frame dimensions to standard bearing.
 2. Ensure fixed conjunction for a rotating bearing race, a small gap or very slight tightness for a fixed one.
 3. The stronger load the tighter fit. The installations should be less tight for high-speed mechanisms.
 4. Roller bearings should be tightened greater than ball bearings. Radial-thrust bearings are tighter than radial. Large bearings are tighter than medium and small ones.
- The bearing diameter tolerances are marked with letter *L* for the inner race (Fig. 5.29) and *I* for the outer one in combination with the degree of accuracy: *L0*, *L6*, *I0*, *I6*.

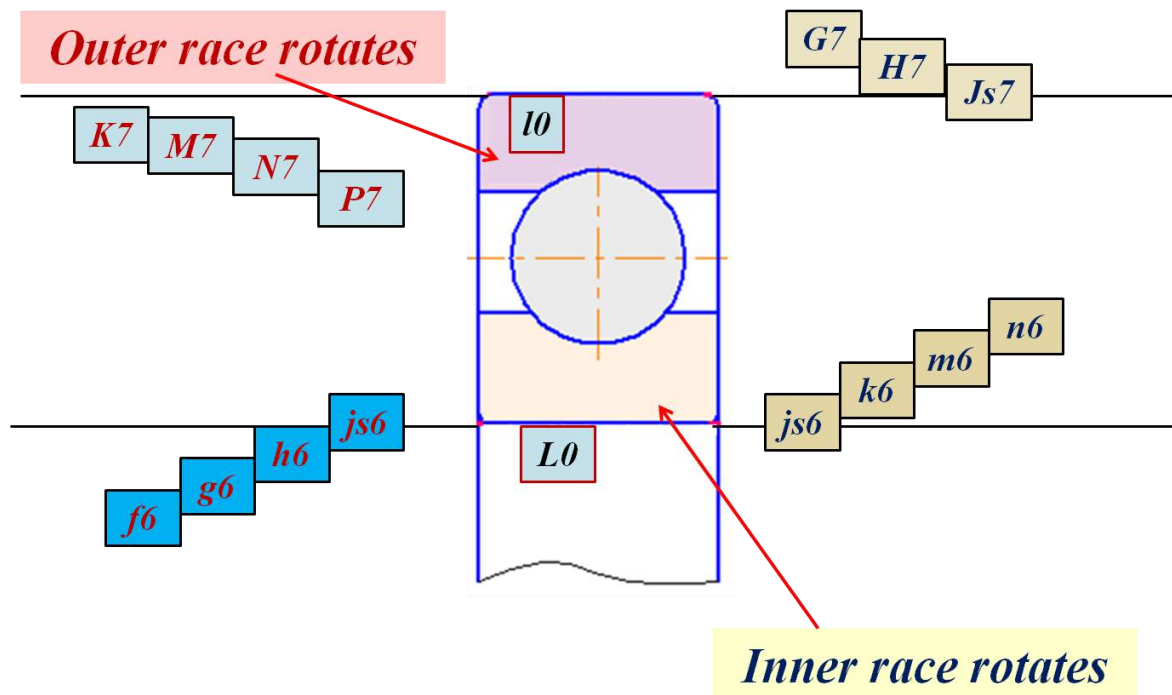


Fig. 5.29

Rotating part	Recommended fits	
	for shaft	for frame
Shaft	<i>L0/k6</i> <i>L6/k6</i>	<i>H7/I0</i> <i>H7/I6</i>
Frame	<i>L0/h6</i> <i>L6/h6</i>	<i>M7/I0</i> <i>M7/I6</i>

5.2. Sliding Bearings

Sliding bearing identifies a bearing in which a movable component, generally a shaft, a shaft extension or a guide rail, slides on top of the surface of stationary bushing, a bearing seat or a sliding strip.



Advantages

- at high angular velocities and constant load without frequent start-stops have smaller dimensions
- the only option for split support (crankshafts)
- unlimited service life (with lubricant)
- ability to work in an aggressive environment
- smaller radial dimensions
- reliability in high-speed drives due to the stability of the parameters

Disadvantages

- difficult start: liquid supply pressure or installing bearings, disconnected at the operating speed, is required
- supply and drain fluid systems are required
- efficiency lower than that of the rolling bearings
- high demands on the alignment of the shaft and the bearing, which increases the cost of production
- use of non-ferrous metals
- relatively large axial dimensions

5.2.1. Sliding Bearings Preferred Use

1. Split bearings for crankshafts.
2. Bearings for high-speed shafts, if the service life of rolling bearings is not sufficient.
3. Supports for high accurate and smooth rotation.
4. Bearings for large shaft diameters (like turbine).
5. Bearings with small diametric dimensions.
6. Bearings are subjected to shock and vibration.
7. High-speed gas bearings.

5.2.2. Comparison of Bearing Types

	Fluid Film Bearing	Sliding Bearing	Rolling Bearing
Startup Friction	0,25	0,1...0,15	0,002
Running Friction	0,001	0,05...0,1	0,001
Velocity Limit	High	Low	Medium
Load Limit	High	Low	High
Life Limit	Unlimited	Wear	Fatigue
Lubrication requirements	High	Low/None	Low
Cost	High	Low	Medium

5.2.3. Classification of Sliding Bearings

By Load Direction

The surface of the shaft, which transmits radial loads, is called a journal, and bearing surface is a radial bearing. The surface of the shaft, which transmits axial loads, is called the runner, and bearing surface – thrust collar (Fig. 5.30).

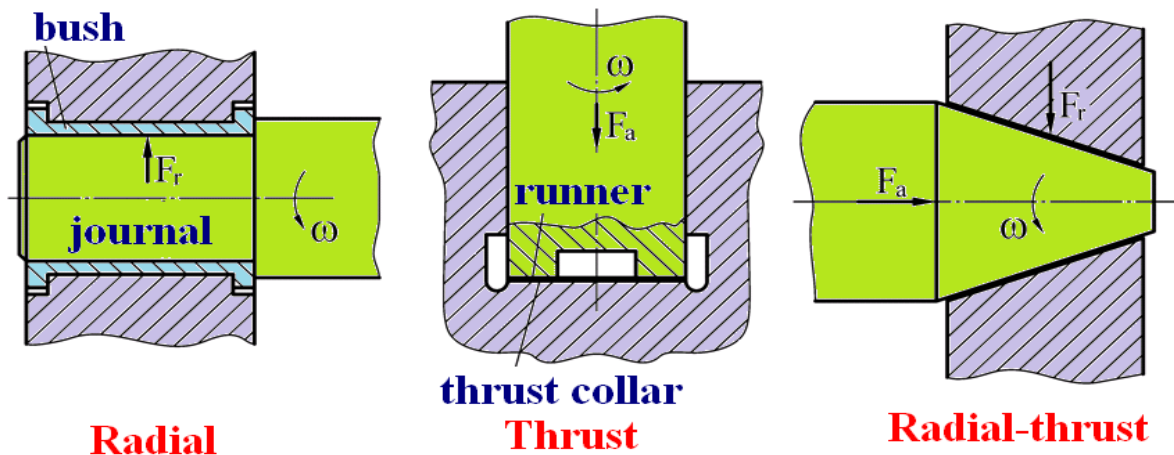


Fig. 5.30

By the Bush Design

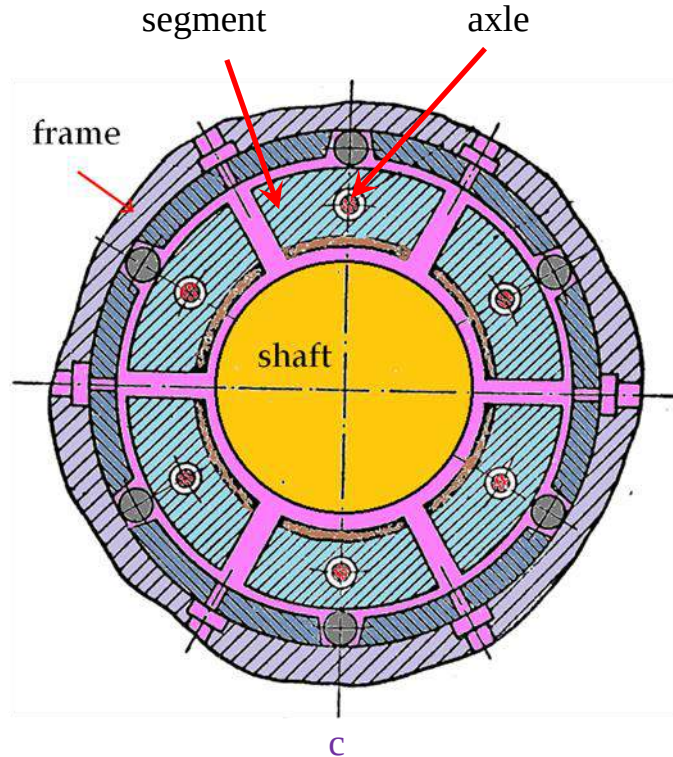
1. Monolithic of bronze, iron, ceramics, metal ceramics, plastics (Fig. 5.31, a).
2. Filling in the surface with babbitt layer about 1 mm (Fig. 5.31, b).
3. Self-aligning: segments can be rotated about an axis (Fig. 5.31, c).



a



b



c

Fig. 5.31

5.2.4. Materials

The usual general requirements for sliding bearing bush materials are:

- low friction coefficient with the material of the shaft (usually steel);
- durability;
- fatigue strength under pulsating loads;
- high thermal conductivity;
- good running-in;
- good wetting with oil.

The most popular material for bearings is *metal* such as:

- babbitt which is an alloy of tin and lead. It's used at high speeds and pressure up to 20 MPa. The main disadvantages are poor fatigue resistance and low heat resistance (up to 110 °C);
- bronze which is a cooper-based alloy with tin, lead, beryllium, etc. used at high speeds and pressures up to 30 MPa. Disadvantage is increased wear.
- aluminum alloys. They have high anti-friction, good thermal conductivity and resistance to fatigue. They are the most promising materials.

In contrast to metal which has solid structure *metal-ceramic* is a mixture of metal powders (copper or iron) with graphite powder, lead and other additives. It is made by pressing and sintering at high temperature. Metal-ceramic has porous structure that can operate at poor lubrication after impregnation with hot oil.

Non-metallic may be plastic, such as Teflon (PTFE), nylon, caprolon, rubber, graphite materials, solid wood.

Lubricants

1. *Liquid* lubricant is based on mineral oil. It is preferable for high rotational frequency, but also applicable for low rotational speed. The type of lubrication oil depends on the particular application. By increasing the adhesion and smoothing of the sliding surfaces, additives of molybdenum sulfide or graphite can improve the lubrication properties. It may be:

- industrial oils;
- transmission oils.

2. *Grease*. Grease-application is to be applied in case of low rotational frequency, pendulousness and impact loading, or if float friction is not within reach. High-class sliding bearing fat should be used exclusively. Lubrications with solid additives exceeding a proportion of 2% are not recommendable, as they are likely to cause wastage ahead of time. They are:

- solid oil is liquid mineral oil solidified by soda-base soap;
- konstalin is solidified by lime-base soap;
- litol is solidified by lithium soap.

The main lubricant characteristic is viscosity, it is fluid's resistance to flow (shear stress) at given temperature. Dynamic and kinematic viscosities are distinguished. The dynamic viscosity criterion is based on the of the Newton's formula

$$F = \mu A \frac{dV_x}{dy},$$

where F is the resistance force (force is needed to shift liquid layers); A is the layer's areas; V_x is the relative motion speed; μ is the dynamic viscosity, typical units Pa·s, Poise, P.

Kinematic viscosity is the dynamic viscosity divided by the density at the same temperature (typical units cm²/s, Stokes, St):

$$\nu = \frac{\mu}{\rho}.$$

Other properties of liquid oils are:

- oily (oil's ability to create a film on a solid surface);
- flash point (the temperature at which the vapor over the liquid will ignite upon exposure to an ignition source);
- pour point (the temperature below which the liquid loses its flow features).

5.2.5. Types of Friction

Three lubricant regimes occur in a sliding bearing. To describe the frictional characteristics of a liquid lubricant the Stribeck curve is used (Fig. 5.32). It usually spans the Boundary, Mixed and Hydrodynamic regimes.

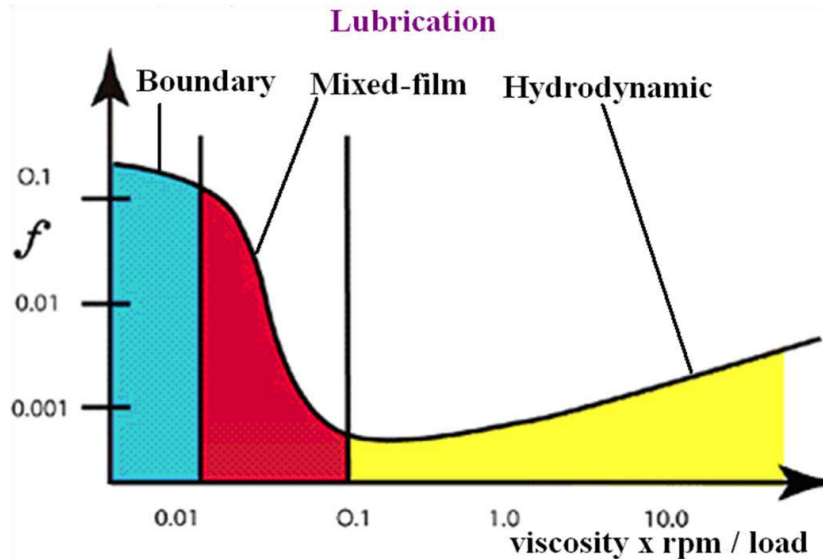


Fig. 5.32

Boundary (Fig. 5.33, a). Bearing surfaces are in contact with each other; during the friction interpenetration and deformation of asperities are taking place. During this phase friction is very high (coefficient of friction $f = 0,1 \dots 0,5$) and wear takes place.

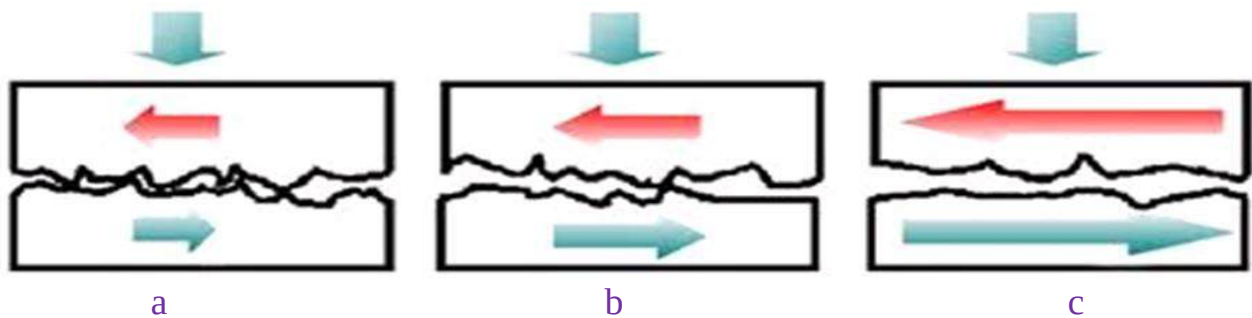


Fig. 5.33

Mixed (Fig. 5.33, b). Surfaces are covered with a layer of liquid thickness of about 0,1microns. If the condition $h > R_{z1} + R_{z2}$ is not satisfied, then in the bearing are both fluid and boundary friction. With increasing speed the sliding surfaces are partially separated by the lubricant. Coefficient of friction $f = 0,01 \dots 0,1$. As a result – medium friction and medium wear.

Hydrodynamic (Fig. 5.33, c). At higher speeds surfaces are separated from each other by a layer of lubricant (the hydrodynamic fluid film), asperities are not touching (liquid layer h is more than the sum of surface asperity heights: $h > R_{z1} + R_{z2}$).

In this phase the lowest value of the friction coefficient and wear are achieved $f = 0,001 \dots 0,005$.

5.2.6. The Sliding Bearings Calculation at Boundary and Mixed Friction

Boundary and mixed types of friction characterize any sliding bearing during start and stop operation when the shaft speed is low.

The calculation is based on two criteria:

1. Allowable pressure, MPa: $p = \frac{F_r}{d l} \leq [p]$.

Here F_r is a radial load, N; d, l – in mm (Fig. 5.34).

For internal combustion engines $[p] = 4 \dots 20$ MPa.

2. The product of pressure and peripheral speed

$$pV = \frac{F_r}{d l} \frac{\pi d n}{60000} \leq [pV],$$

n in rpm; V in m/s.

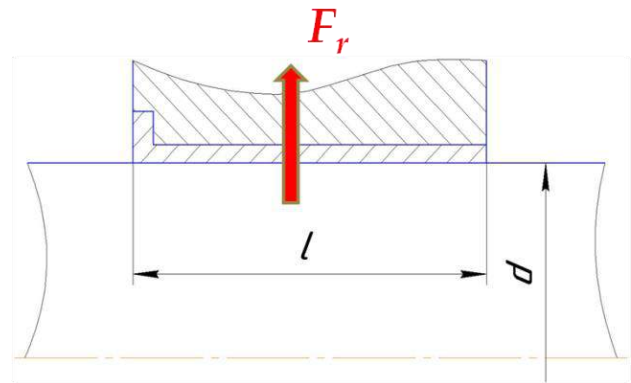


Fig. 5.34

5.2.6. The Sliding Bearings Calculation at Hydrodynamic Friction

To exist in long time layer of lubricant between the frictional surfaces it must be pressurized. If this pressure is created by the journal rotation, the bearing is called hydrodynamic (Fig. 5.35, a), if by the pump – hydrostatic (Fig. 5.35, b).

The calculation is based on the hydrodynamic theory of lubrication under these assumptions:

- no oil leakage through the faces of the bearing;
- fluid satisfies Newton's law.

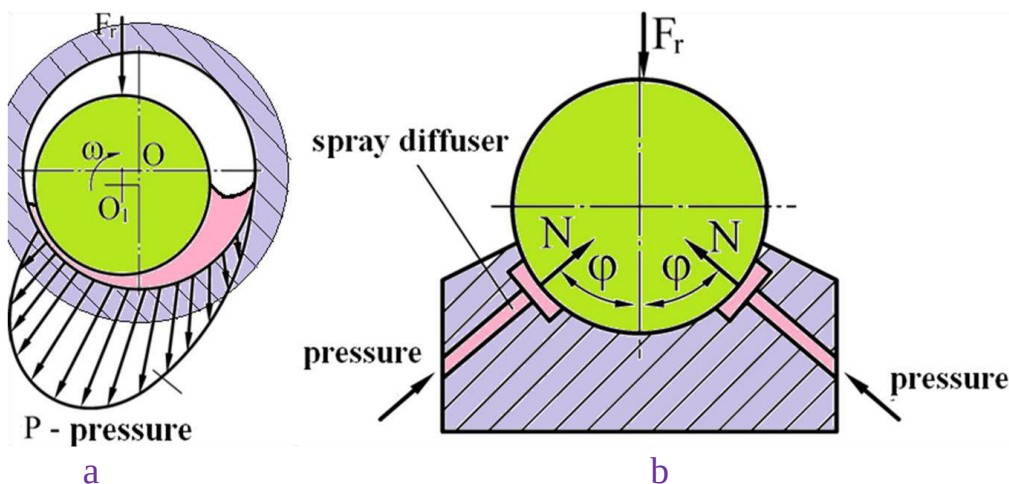


Fig. 5.35

Shear stresses into the oil layers

$$\tau = \mu \frac{dV_x}{dy}.$$

From the equilibrium condition of the selected oil volume (Fig. 5.36) we can write

$$(P + dP)dy - Pdy = (\tau + d\tau)dx - \tau dx.$$

After reducing we obtain the pressure gradient:

$$\frac{dP}{dx} = \frac{d\tau}{dy}.$$

Taking into account Newton's Law we can write

$$\frac{d^2 V_x}{dy^2} = \frac{1}{\mu} \frac{dP}{dx}.$$

Integrating twice, we obtain velocity

$$V_x = V \frac{y}{h} + \frac{y}{2\mu} (y - h) \frac{dP}{dx}.$$

Liquid flow rate through any cross-section

$$q_h = \int_0^h V_x dy = V \frac{h}{2} - \frac{h^3}{12\mu} \frac{dP}{dx}.$$

In cross-section h_m , where $\frac{dP}{dx} = 0$, $q_m = V \frac{h_m}{2}$.

From the condition of the flow continuity

$$q_h = q_m.$$

Reynolds equation is obtained

$$\frac{dP}{dx} = 6\mu V \frac{h - h_m}{h^3}.$$

Reynolds equation characterizes the change in fluid (oil) pressure in the x direction. The equation shows that the bearing load capacity of the oil film increases with increasing viscosity and decreasing the gap.

Rewriting the Reynolds equation in polar coordinates and convert it, we obtain the capacity of the bearing oil layer, i.e. radial load, a layer of oil can carry:

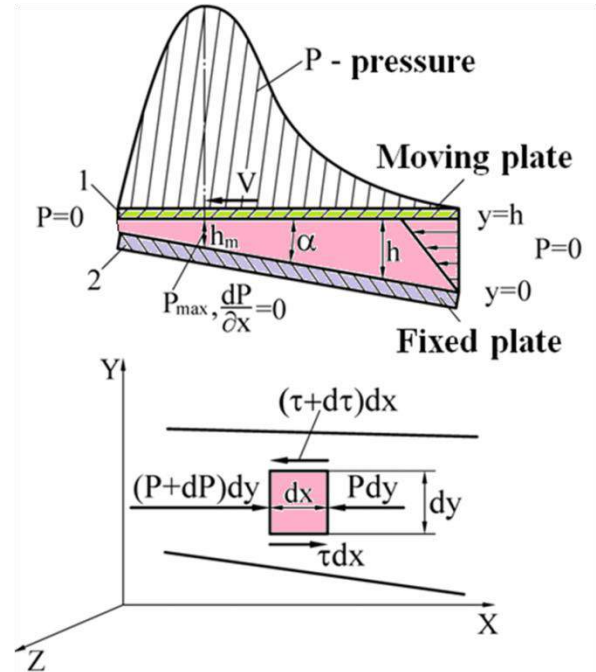


Fig. 5.36

$$F_r = \frac{\mu \omega}{\psi^2} l d \Phi_r.$$

The bearing dimensionless load factor

$$\Phi_r = f\left(\frac{l}{d}, \chi\right)$$

is taken from the tables.

The relative eccentricity is $\chi = \frac{e}{(\delta/2)}$, the relative clearance $\psi = \frac{\delta}{d}$, where δ is the diametrical clearance; e is the eccentricity (Fig. 5.37).

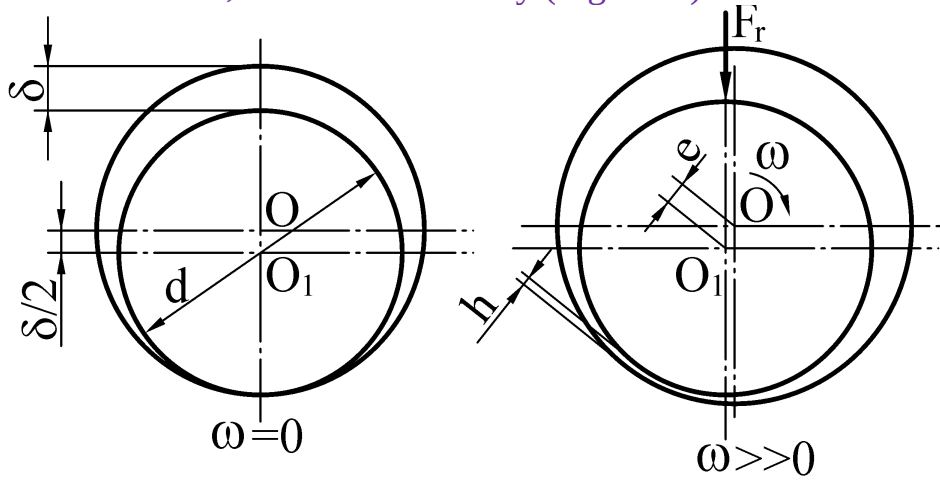


Fig. 5.37

5.2.7. Hydrostatic Bearings

In hydrostatic bearing (HSB) pressure between a shaft and a bearing is created by oil supplied by a pump. Shaft without rotation floats on a hydrostatic cushion. HSB can support large loads without journal rotation and provide large (accurate and controllable) direct stiffness as well as damping (energy dissipation) coefficients.

The load capacity and direct stiffness of hydrostatic bearings do not depend on fluid viscosity, thus making them ideal rotor support elements in process fluid pumps. Current applications intend to replace oil lubricated bearing with hybrid bearings to improve efficiency with shorten rotor spans and less mechanical complexity. Current cryogenic liquid turbopumps implement hydrostatic bearings enabling an all-fluid film bearing technology with very low number of parts and no DN limit operation.

The hydrostatic stiffness is of unique importance for the centering of high-precision milling machines, gyroscopes, large arena movable seating areas, telescope bearings, and even cryogenic fluid turbo pumps for rocket engines.

HSB may be half- (Fig. 5.38, a) or full coverage (Fig. 5.38, b).

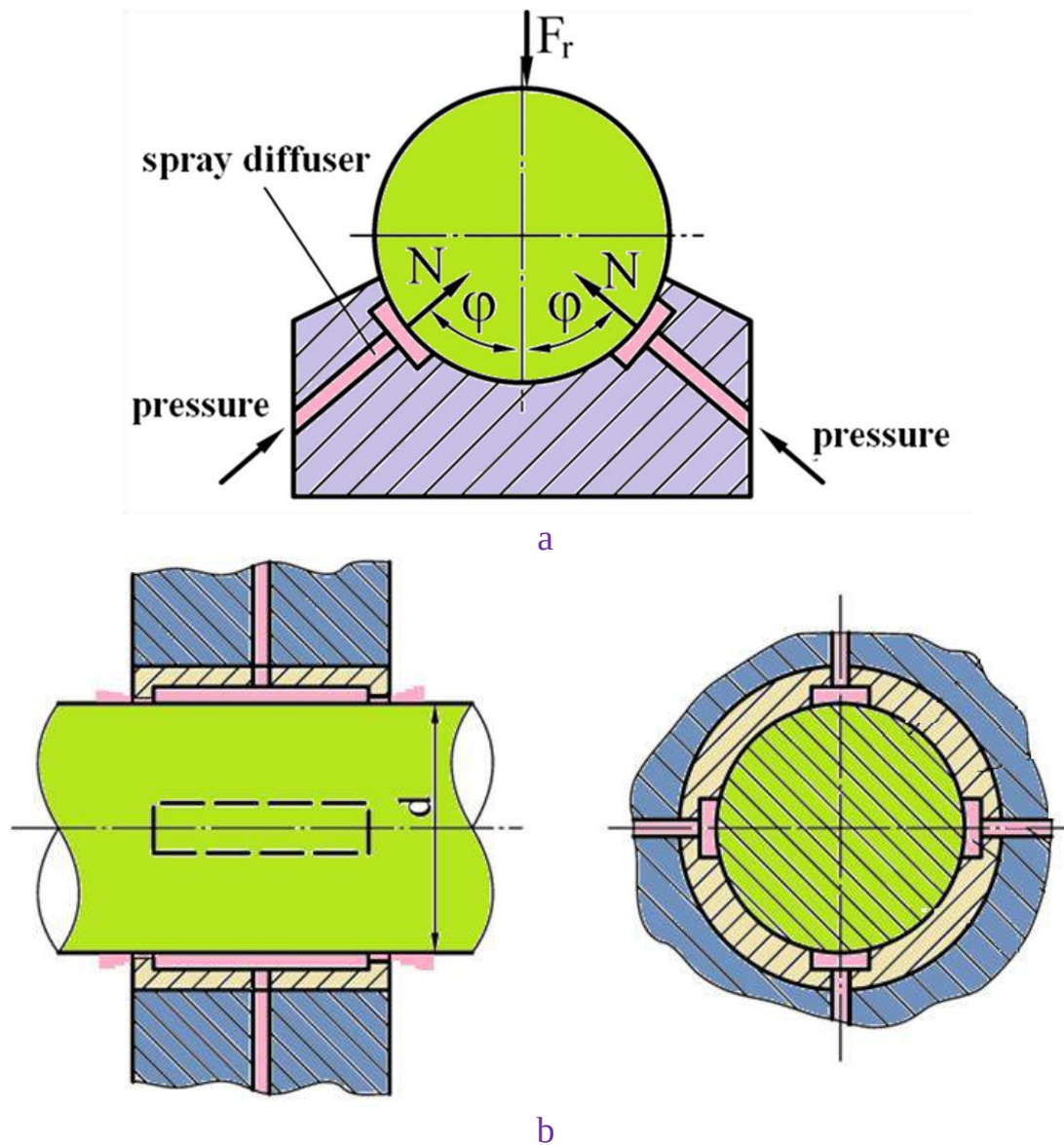


Fig. 5.38

Advantages

- the coefficient of friction is very small, when starting off is close to zero
- due to relatively big clearance manufacturing precision is less
- provides high accuracy of rotation due to bearing self-adjustment
- practically no wear

Disadvantages

- the complexity and high cost of the oil system
- reduced reliability as pressure diffusers are often clogged up

Part 6. CLUTCHES & COUPLINGS

Clutches and couplings are the mechanisms used for transferring torque between shafts and to compensate their misalignment. One shaft is typically attached to a motor or other power unit (the driving member) while the other shaft (the driven member) transfers power for mechanical work:



A coupling is a device used to connect two shafts together at their ends for the purpose of transmitting power. Couplings do not normally allow disconnection of shafts during operation.

A clutch is a mechanical device that engages and disengages the power transmission.

Installation errors, load-caused deformation and shaft's heat extension lead to mutual displacement of the shafts.

Types of displacement (Fig. 6.1):

1. Axial displacement Δl .
2. Radial displacement Δr .
3. Angle displacement $\Delta \alpha$.

A combination of the above-mentioned displacement types is called a misalignment.

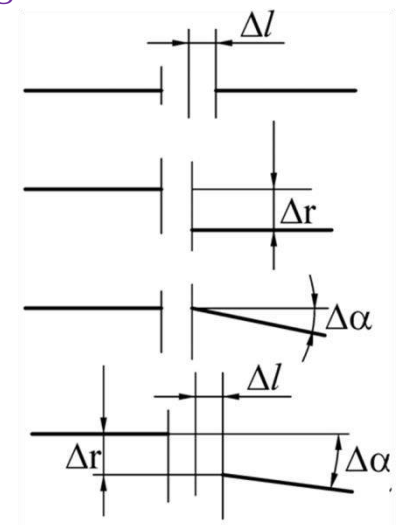
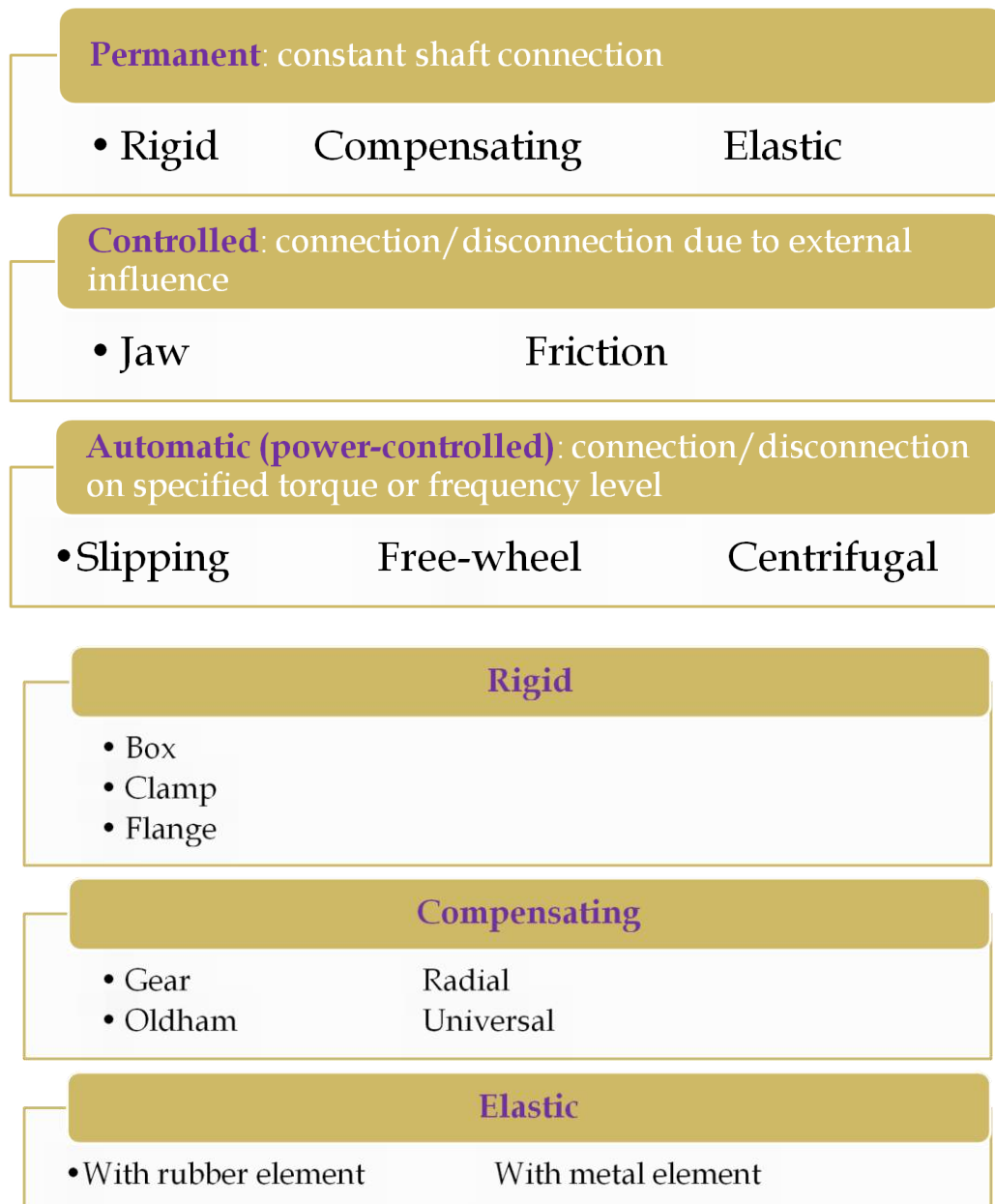


Fig. 6.1

6.1. Classification

Clutches and couplings are classified by physical structure as mechanical, electric, pneumatic and hydraulic. Here we consider mechanical devices only.

They are:



6.2. Permanent Couplings

Rigid Box Coupling

A rigid coupling is a unit of hardware used to join two shafts within a motor or mechanical system. It may be used to connect two separate systems, such as a motor and a generator, or to repair a connection within a single system. A rigid coupling may also be added between shafts to reduce shock and wear at the point where the shafts meet.

When joining shafts within a machine, mechanics can choose between flexible and rigid couplings. While flexible units offer some movement and give between the

shafts, rigid couplings are the most effective choice for precise alignment and secure hold. By precisely aligning the two shafts and holding them firmly in place, rigid couplings increase the service life of the machine.

These rigid couplings are available in two basic designs to fit the needs of different applications. Sleeve-style couplings (Fig. 6.2) are the most affordable and easiest to use. They consist of a single tube of material with an inner diameter that's equal in size to the shafts. The sleeve slips over the shafts so they meet in the middle of the coupling.

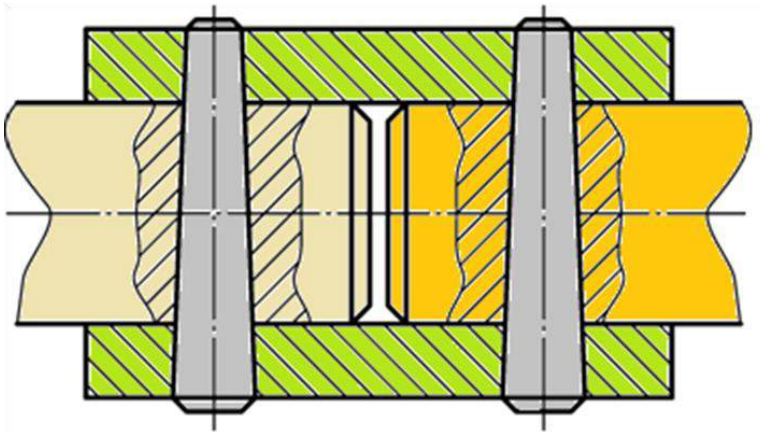


Fig. 6.2



A series of set screws can be tightened so they touch the top of each shaft and hold them in place without passing all the way through the coupling.

Clamped or compression rigid couplings (Fig. 6.3) come in two parts and fit together around the shafts to form a sleeve. They offer more flexibility than sleeved models, and can be used on shafts that are fixed in place. They generally are large enough so that screws can pass all the way through the coupling and into the second half to ensure a secure hold.



Fig. 6.3

Advantages	Disadvantages
– simplicity – average load capacity – small radial size	– absence of compensating features – weakness of the pins

Flanged Rigid Coupling

Flanged rigid couplings (Fig. 6.4, a, b) are designed for heavy loads or industrial equipment. They consist of short sleeves surrounded by a perpendicular

flange. One half-coupling is placed on each shaft so the two flanges line up face to face. A series of screws or bolts can then be installed in the flanges to hold them together. Because of their size and durability, flanged units can be used to bring shafts into alignment before they are joined together. Rigid couplings are used when precise shaft alignment is required; shaft misalignment will affect the coupling's performance as well as its life.

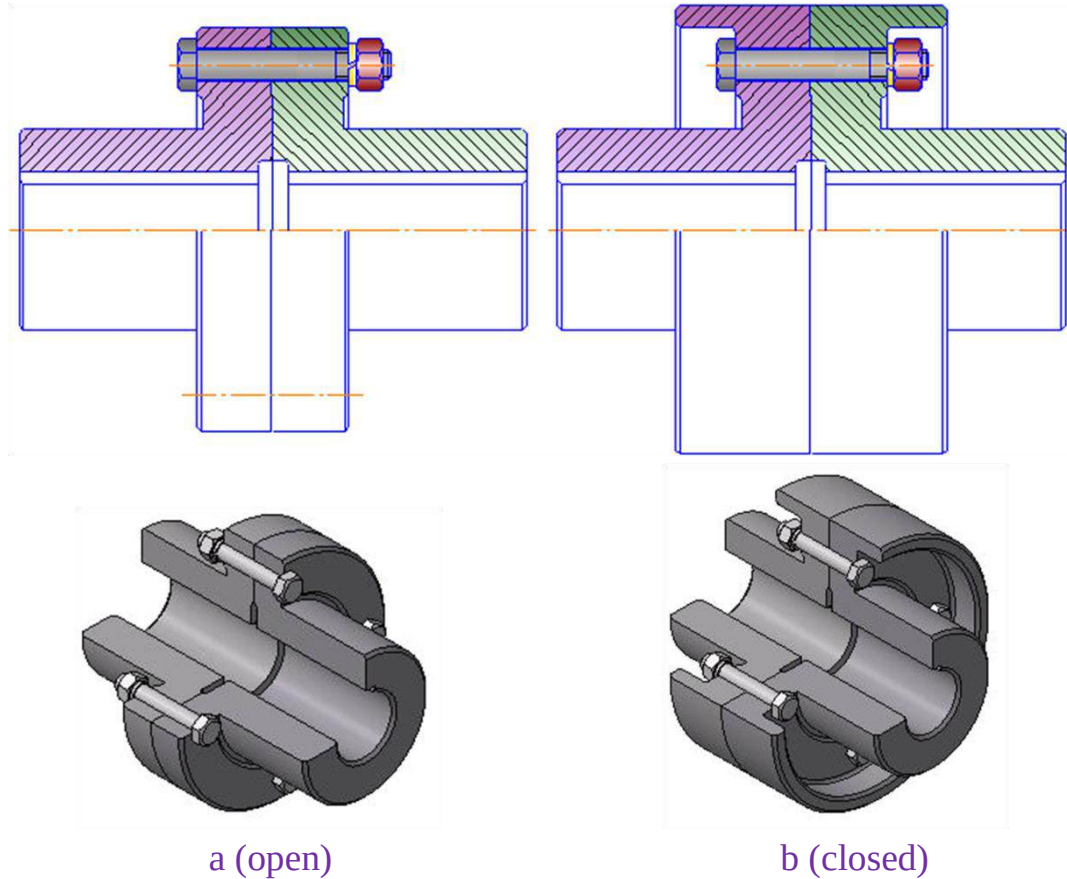


Fig. 6.4

Torque is transmitted either through resistance forces to the shear of “clean” (installed without gaps) bolts or through friction force between flanges (bolts are installed with a gap). If the bolts are installed without gaps, they are designed for shear and bearing, if with a gap – for tension caused by prestressed force. In the first case the weight is less and bearing capacity is approximately 5 times higher if size is same. In the second case, production and installation is much cheaper and simpler.

Advantages	Disadvantages
– simplicity – great load capacity – using additional collars can compensate axial displacement	– absence of other compensating features – large radial size

This type is often used for connecting parts of long composite shafts, e.g. in helicopter transmission (Fig. 6.5).

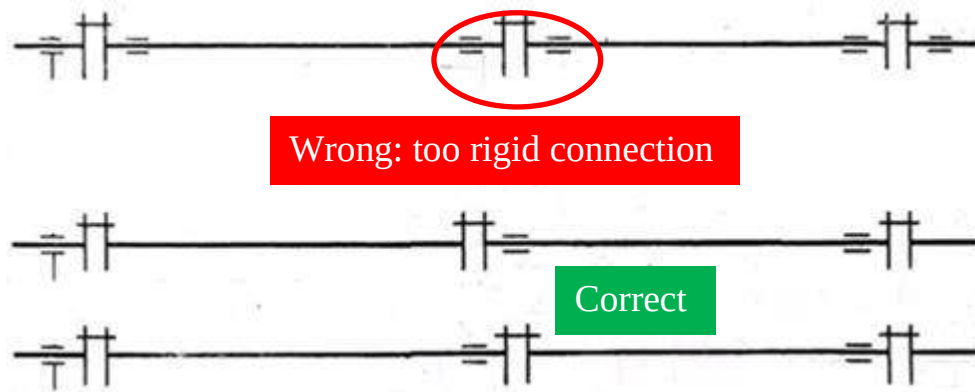


Fig. 6.5

Oldham (Compensating Jaw-Disc) Coupling

The Oldham coupling (named by John Oldham (1779–1840), Irish engineer) comprises two hubs 1 and 3, connected to the respective shafts, and an intermediate disc 2 (Fig. 6.6). The torque is transmitted between driving member and intermediate member, and between intermediate member and driven member by means of two orthogonal sliding connections. It allows the connection of shafts with a parallel misalignment between their axes without inducing nonuniformity of rotation of the driven shaft and without exerting high loads on the shaft bearings.

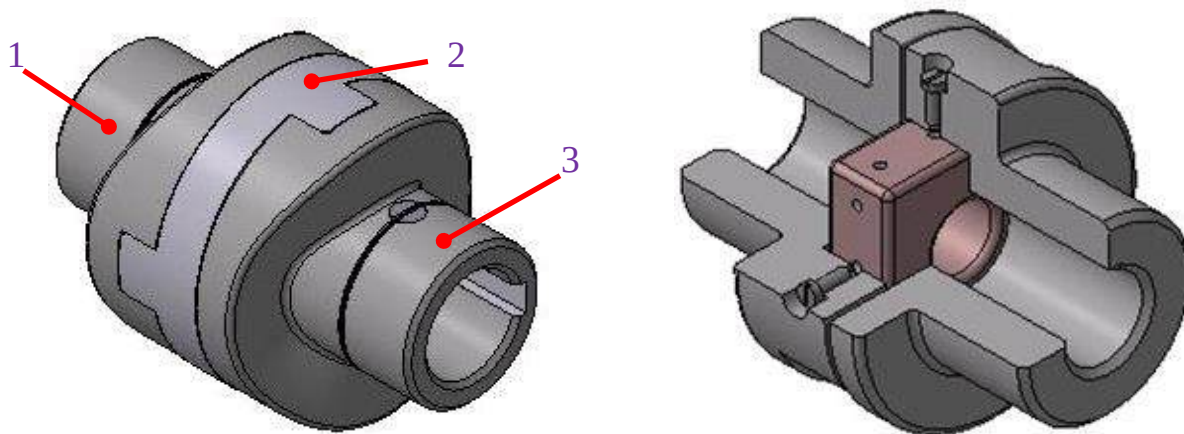


Fig. 6.6

It compensates for misalignment of connected shafts by means of limited sliding between the hub surfaces and their counterpart surfaces on the intermediate member. The sliding has a cyclical character. Such a motion pattern is not conducive to good lubrication, since at the ends of the relative travel, where the sliding velocity is zero, a metal-to-metal contact is very probable. The stoppages are associated with increasing friction coefficients, close to the static friction values.

However, this ingenious design finds only an infrequent use, usually for noncritical, low-speed applications.

Advantages	Disadvantages
– great load capacity – ability to compensate axial and radial displacement	– inevitable sliding – short service life

Compensating Curved-Tooth Gear Coupling

A gear coupling is a mechanical device for transmitting torque between two shafts that are not collinear. It consists of a flexible joint fixed to each shaft. The two joints are connected by a third shaft, called the spindle (Fig. 6.7).

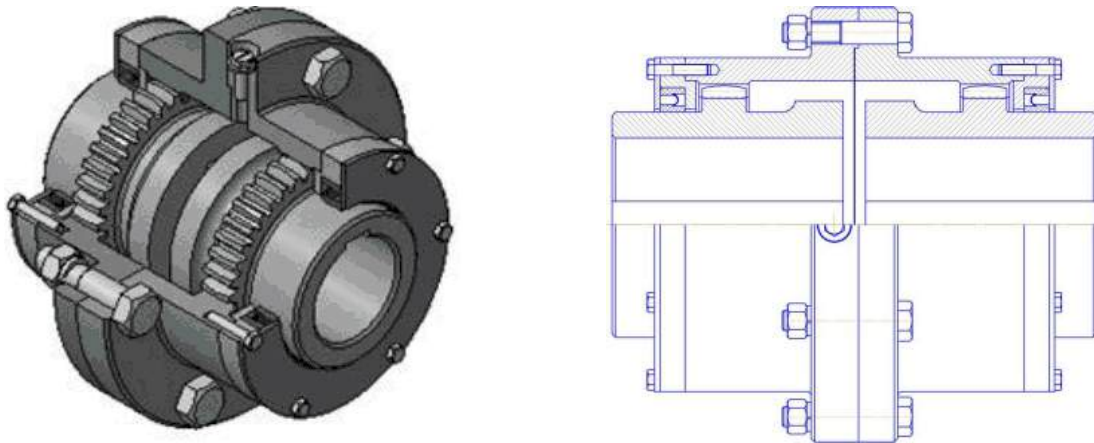


Fig. 6.7

Each joint consists of a 1:1 gear ratio internal/external gear pair. The tooth flanks and outer diameter of the external gear are crowned to allow for angular displacement between the two gears (Fig. 6.8). Mechanically, the gears are equivalent to rotating splines with modified profiles. They are called gears because of the relatively large size of the teeth.

The gear teeth have high backlash to allow for angular misalignment. The excess backlash can contribute to vibration.

Gear couplings are generally limited to angular misalignments, i.e., the angle of the spindle relative to the axes of the connected shafts, of 4...5°.

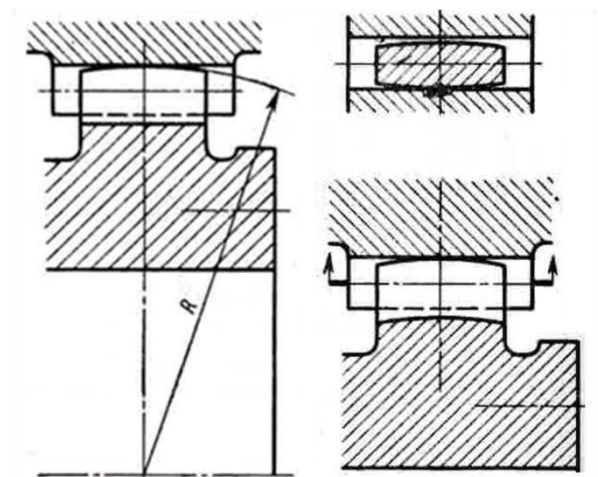


Fig. 6.8

Single joint gear couplings (Fig. 6.9) are also used to connect two nominally coaxial shafts. In this application the device is called a gear-type flexible, or flexible coupling. The single joint allows for minor misalignments such as installation errors and changes in shaft alignment due to operating conditions. These types of gear couplings are generally limited to angular misalignments of $1/4$ - $1/2^\circ$.

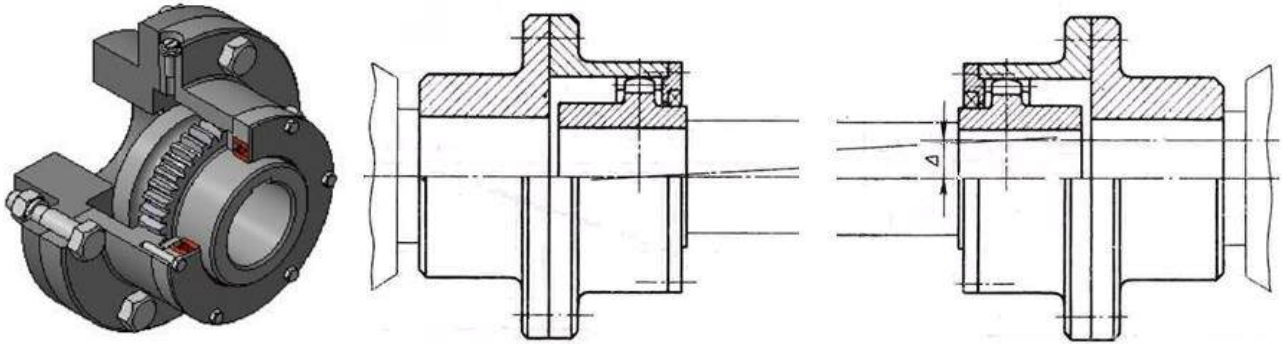


Fig. 6.9

Advantages	Disadvantages
– great load capacity – ability to compensate any misalignment	– large size – lubricant is required – great power losses

Presence of the clearance changes the contact area significantly increases the peak contact pressures and further increasing the wear rate. This leads to a rapid increase of the backlash, unless the initial (design) contact pressures are greatly reduced. Such non-uniform contact loading also results in very poor lubrication conditions in the stick-slip motion. As a result, friction coefficients and power losses in gear and Oldham couplings are quite high.

The frequent stoppages and direction reversals of the forces lead to the high noise levels generated by Oldham coupling described above and gear couplings. A gear coupling can be the noisiest component of a large power-generation system.

Compensating Universal (Cardan)

A universal coupling or U-joint, or Cardan joint (named by Gerolamo Cardano (1501-1576), a Renaissance mathematician, physician), or Hooke's joint is a joint in a rigid rod that allows the rod to 'bend' in any direction, and is commonly used in shafts that transmit rotary motion. It consists of a pair of hinges located close together, oriented at 90° to each other, connected by a cross shaft (Fig. 6.10). The limit on torque density in universal joints is due to the limited cross sections of the cross and yoke.

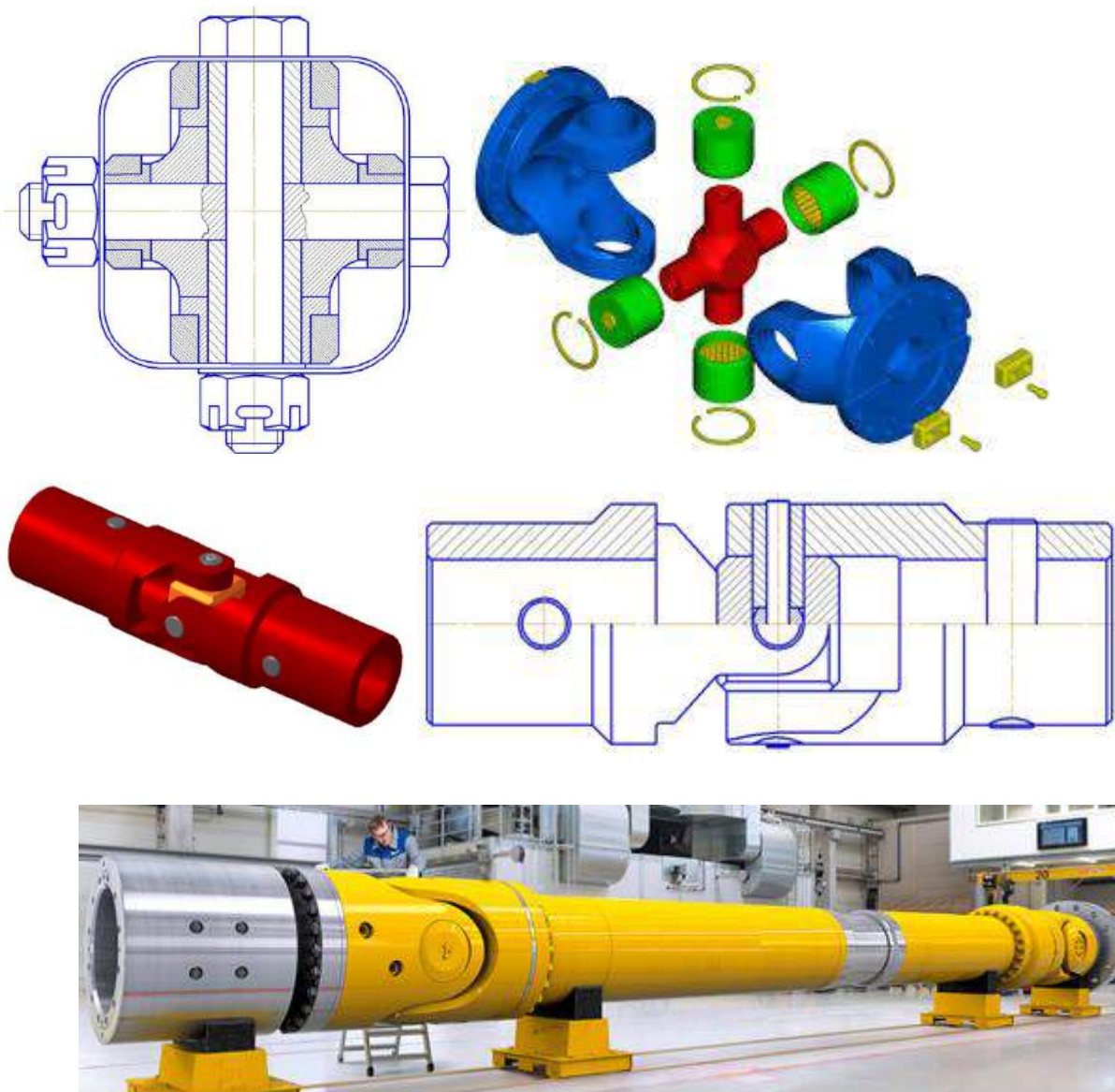


Fig. 6.10

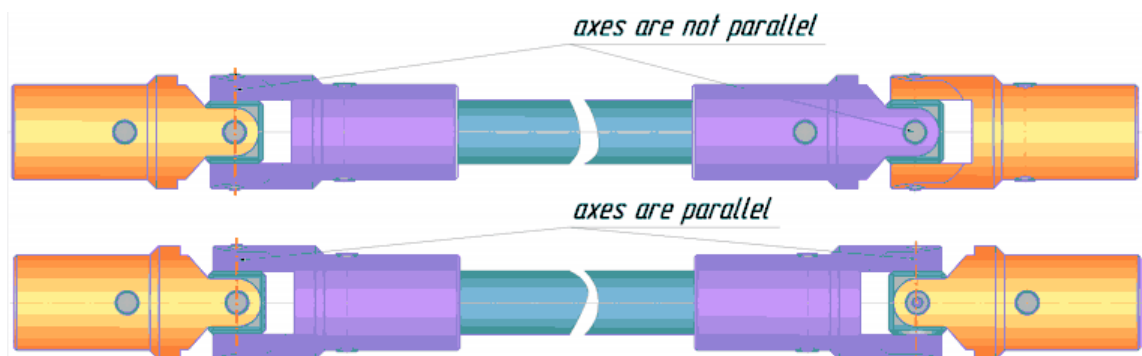
Advantages	Disadvantages
– great load capacity – ability to compensate angular misalignment up to 45°	– non-uniformity of the driven shaft rotation – cannot compensate radial misalignment

The Cardan joint suffers from one major problem: even when the input drive shaft axle rotates at a constant speed, the output drive shaft axle rotates at a variable speed, thus causing vibration and wear. The variation in the speed of the driven shaft depends on the configuration of the joint, which is specified by α – the bend angle of the joint, or angle of the axles with respect to each other, with zero being parallel or straight through. According to the driven shaft rotation uniformity couplings are

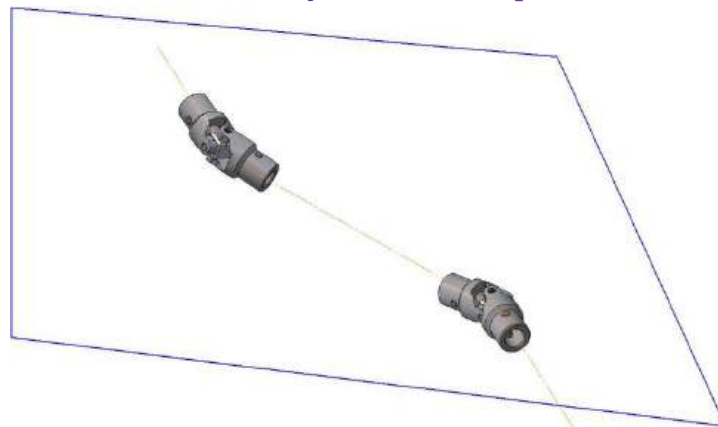
divided to synchronous (angular speed of the connected shafts is equal) and asynchronous.

Angular velocities are synchronized if:

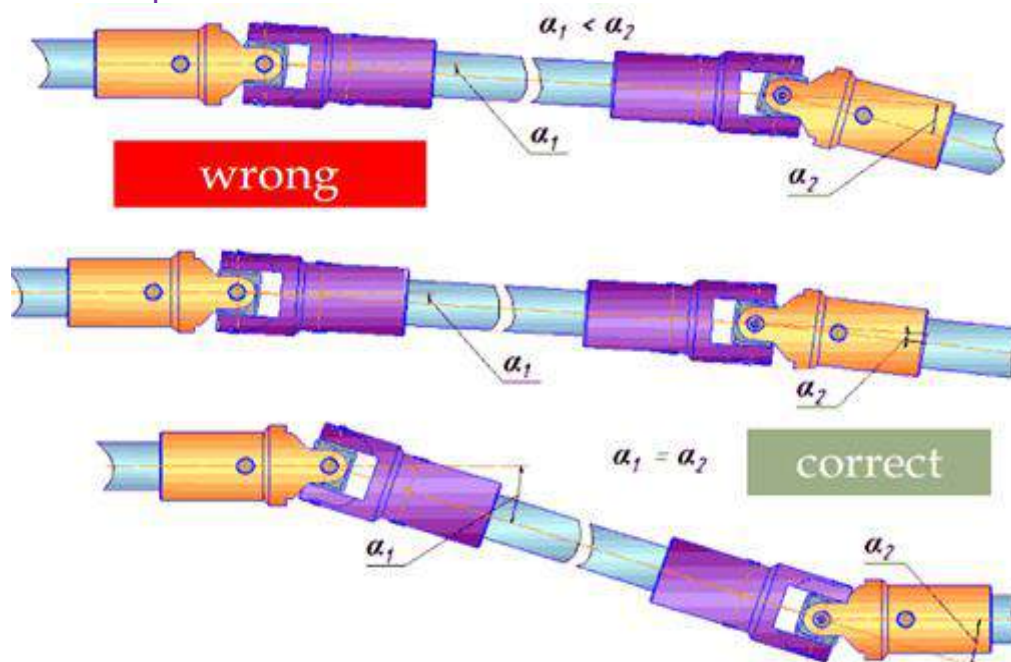
- the intermediate shaft pin axes are parallel:



- axes of the connected shafts lay in the same plane:



– angles between the first and the intermediate shafts, the intermediate and the second should be equal:



There should be an element to compensate axial deformations (splines, floating shaft etc.) in kinematic chain with Universal couplings (Fig. 6.11).

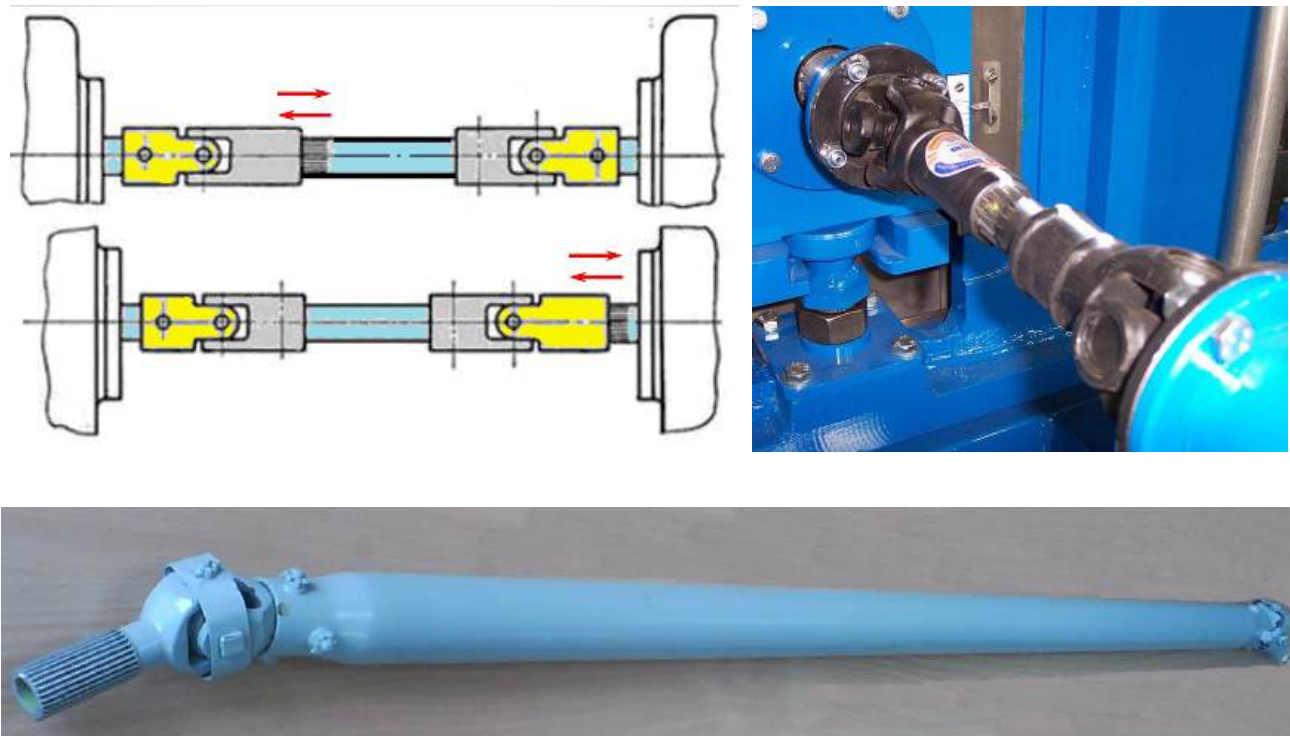


Fig. 6.11

Elastic Clutches

Couplings contain a metallic or non-metallic elastic element to soft shocks and prevent dangerous vibration (Fig. 6.12). Flexible couplings are used to transmit torque from one shaft to another when the two shafts are slightly misaligned. Flexible couplings can accommodate varying degrees of misalignment up to 3° and some parallel misalignment. In addition, they can also be used for vibration damping or noise reduction.

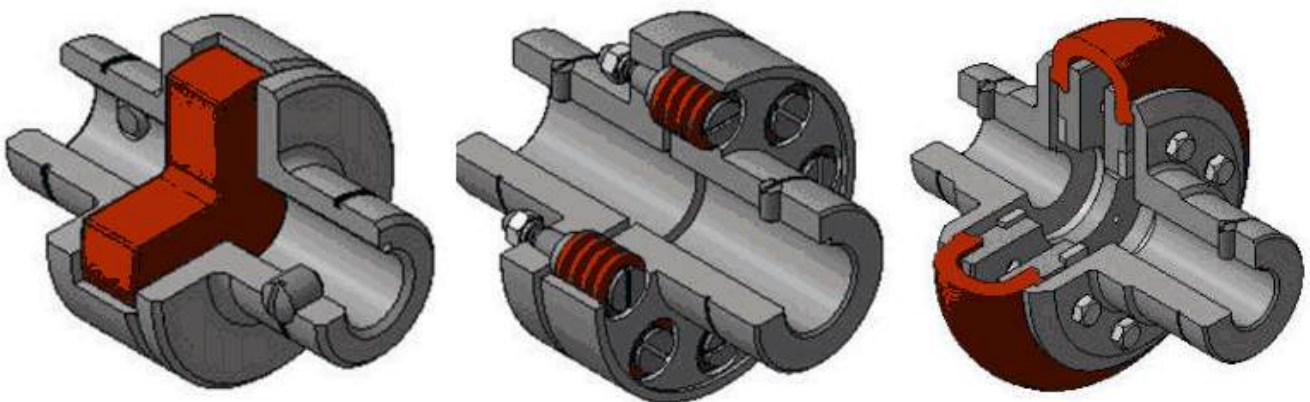


Fig. 6.12

Elastic with Spider Element

One of the most widely applied types of flexible couplings is an elastomeric design known as the jaw coupling. Jaw couplings generally are recommended for continuous-duty electric motor-driven machinery, pumps, gearboxes, etc. Jaw designs usually are not recommended for engine-driven or frequent start-stop-reversing applications because of backlash (the amount of free hub movement allowed by the spacing between the jaws and spider legs).

This design is characterised by two hubs, each having two or more thick, stubby protrusions around their perimeters, called jaws, pointing toward the opposing hub (Fig. 6.13). These jaws mesh loosely when the two hubs are brought together. Filling the gaps between the jaws are blocks of elastomeric material or rubber, usually molded into a single asterisk-shaped element called a "spider".

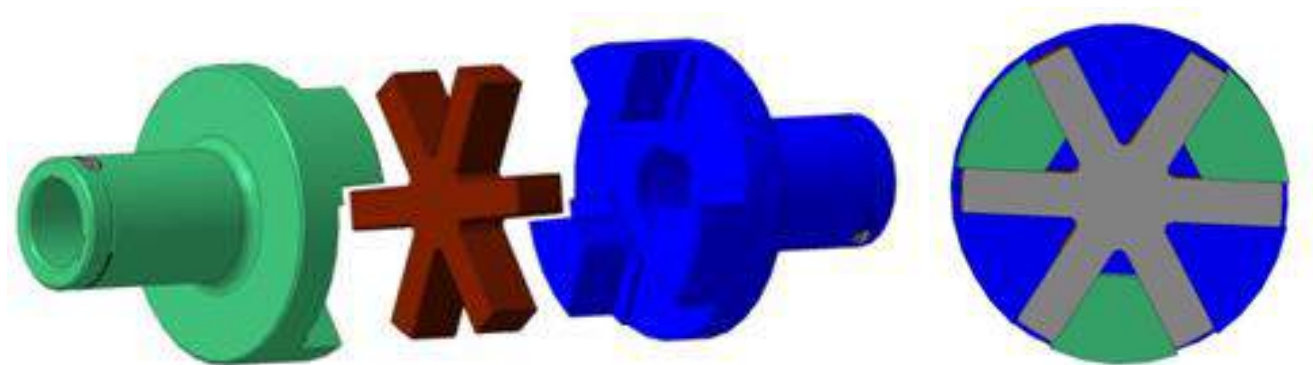


Fig. 6.13

The spider is the key determinant of the torque rating of each jaw coupling. It also can make a significant difference in the coupling's response to vibration, temperature, chemicals, misalignment, high RPM, space limitations and ease of installation or removal.

Angular shaft misalignment is typically limited 1° and tolerates up to 0,5 mm of parallel misalignment.

Advantages	Disadvantages
– compactness – simplicity of production and maintenance – insulates electricity	– low compensating ability – low load capacity (less 400 Nm)

Elastic Coupling with Toroidal Jacket

It is standardized, it has a high load capacity, the ability to reduce dynamic loads due to low torsional stiffness and high damping capacity, high compensating and electrical insulating properties. It usually has convex rubber covering – in the form of a car tire (Fig. 6.14, *a*), but in some versions, a concave jacket is used (Fig. 6.14, *b*).

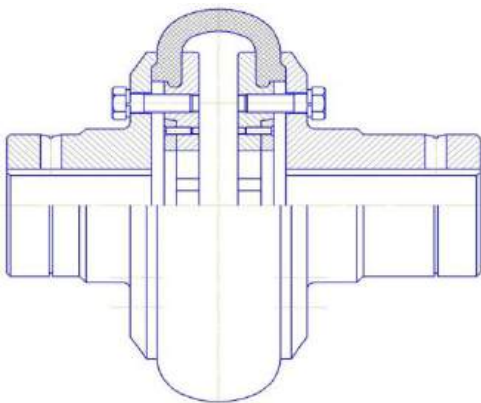
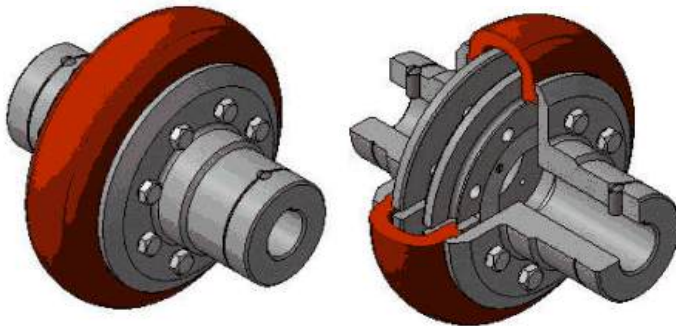
The elastic element is installed and fixed along the perimeter of the disks with screws and additional rings. The casing is made of rubber or rubber reinforced with a special cord-house. The rubber-cord casing is more difficult to manufacture than the rubber one, but its service life is several times longer than the rubber one. When the coupling is loaded, the elastic element works in torsion. This design gives the coupling a high energy consumption and significant compensating capabilities (shaft radial displacement is $\Delta r \approx 2...6$ mm, the angular displacement $\Delta \alpha \approx 2...6^\circ$, the twist angle $5...30^\circ$).

Advantages

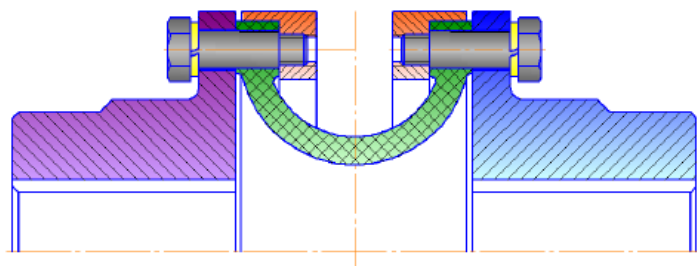
- greater load capacity (up to 40 kNm)
- sufficient compensatory features
- insulates electricity

Disadvantages

- more expensive and complicate
- shafts suffer influence of extra axial forces
- large diametrical size



Convex covering



Concave covering

Fig. 6.14
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Elastic Box-Pin Coupling

It comprises two fully machined steel flanges which are coupled via a series of drive pins and flexible elements (Fig. 6.15). The coupling elements have the ability to absorb shock loads and dampen vibration, structure is simple, easy to manufacture, replacement of flexible components is more convenient: do not move the two halves coupling. The material of the elastic element (pin) is generally made of nylon, which has the ability to compensate the offset of the two axes.

Allowances for axial and parallel misalignment of couplings depend on the number of bolts in the flexible element and operating speed. Axial and parallel misalignment are in inverse proportion; in other words, when one increases, the other decreases. Therefore, the two should be taken into consideration concurrently. It is only applicable to the medium transmission shaft speed.

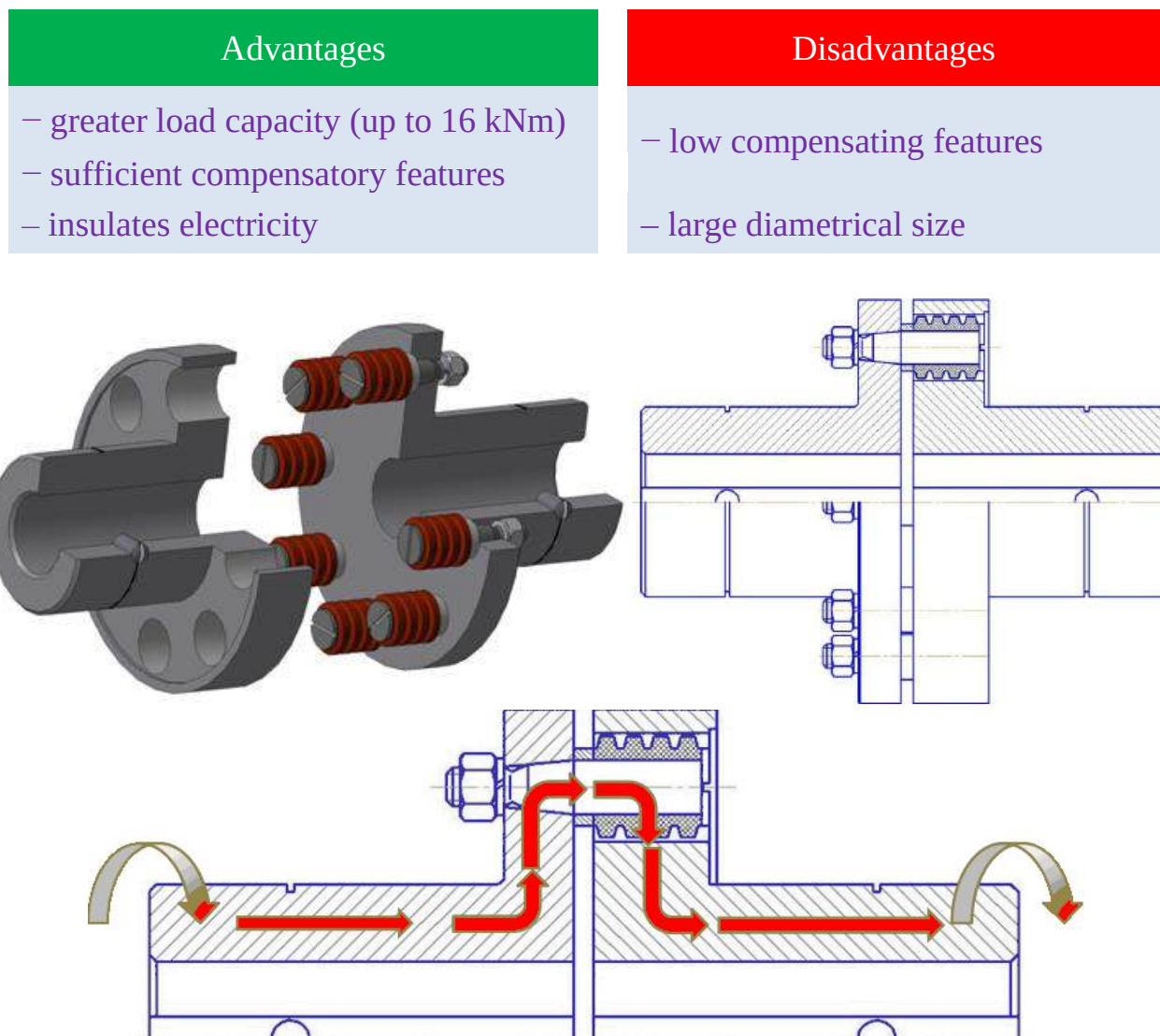


Fig. 6.15

6.3. Controlled Clutches

Connect or disconnect shafts depending on control action. They are divided to synchronous and asynchronous (frictional) with mechanical, electric, pneumatic or hydraulic switching.

Jaw Clutch

Jaw clutches are generally used for moderate speeds where a relatively inexpensive device is required for positively connecting or disconnecting two rotating components and where engagements and disengagements are not too frequent.

Square jaw construction (Fig. 6.16) is used where engagement and disengagement in motion and under load is not necessary. Engagement is only possible when the two halves are in position, so that jaws of the shifter half can enter the mating pockets of the stationary half. Square jaws will transmit power when rotating in either direction.



Fig. 6.16

Spiral jaw clutches are made either right or left hand. A right hand clutch drives when turning clockwise as viewed from the plain end of the driving half. Spiral jaws permit shifting more readily, and are sometimes used where the clutch must be engaged or disengaged when in motion. Such applications should be limited to low speeds and moderate loads. Spiral jaws will transmit power in one direction only and are furnished right or left hand, as per your requirements.

Advantages	Disadvantages
– greater load capacity (up to 16 kNm) – small size and weight	– frequent switch-on lead to instable operation – shaft setup requires accuracy

Tooth shape:

– rectangular (Fig. 6.17, *a*) – for high-load mechanisms. Operation quality of reverse load is low, switch-on is difficult but no axial force of tightness and has regular contact when capacity is partly used.



Fig. 6.17

– triangular (“mouse tooth”) – for light load (Fig. 6.17, *b*). Angle is $\alpha = 30...45^\circ$. Turns on fast and easily, but requires some axial force of tightness, teeth quantity is 15...60.

– trapezoidal (Figs 6.16, 6.18) – for sufficient load and high rotational frequencies. Angle is $\alpha = 3...5^\circ$, jaw quantity is 3...15.

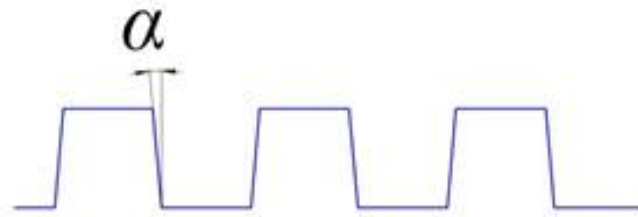


Fig. 6.18

Frictional Clutch

A friction clutch has its principal application in the transmission of power of shafts and machines which must be started and stopped frequently. The force of friction is used to start the driven shaft from rest and gradually brings it up to the proper speed without excessive slipping of the friction surfaces. In a vehicle, the clutch is used to connect the drive from the engine to the transmission and the driving wheels. Divide into normally turned on and normally turned off.

It may be noted that:

1. The contact surfaces should develop a frictional force that may pick up and hold the load with reasonably low pressure between the contact surfaces.

2. The heat of friction should be rapidly dissipated and tendency to grab should be at a minimum.

3. The surfaces should be backed by a material stiff enough to ensure a reasonably uniform distribution of pressure.

The different types of friction clutches are: disk or plate clutches (single or multiple disc clutches) and cone clutches.

A single disc or plate clutch consists of a clutch plate whose both sides are faced with a friction material (Fig. 6.19). It is mounted on the hub which is free to move axially along the splines of the driven shaft. The pressure plate is mounted inside the clutch body which is bolted to the flywheel. Both the pressure plate and the flywheel rotate with the engine crankshaft or the driving shaft. The pressure plate pushes the clutch plate towards the flywheel by a set of strong springs which are arranged radially inside the body. The three levers are carried on pivots suspended from the case of the body. These are arranged in such a manner so that the pressure plate moves away from the flywheel by the inward movement of a thrust bearing. The bearing is mounted upon a forked shaft and moves forward when the clutch pedal is pressed.



Fig. 6.19

A multiple disc (Fig. 6.20) clutch may be used when a large torque is to be transmitted. The inside discs are fastened to the driven shaft to permit axial motion (except for the last disc). The outer discs are fastened to the housing which is keyed to the driving shaft. The multiple disc clutches are extensively used in motor cars, machine tools etc.

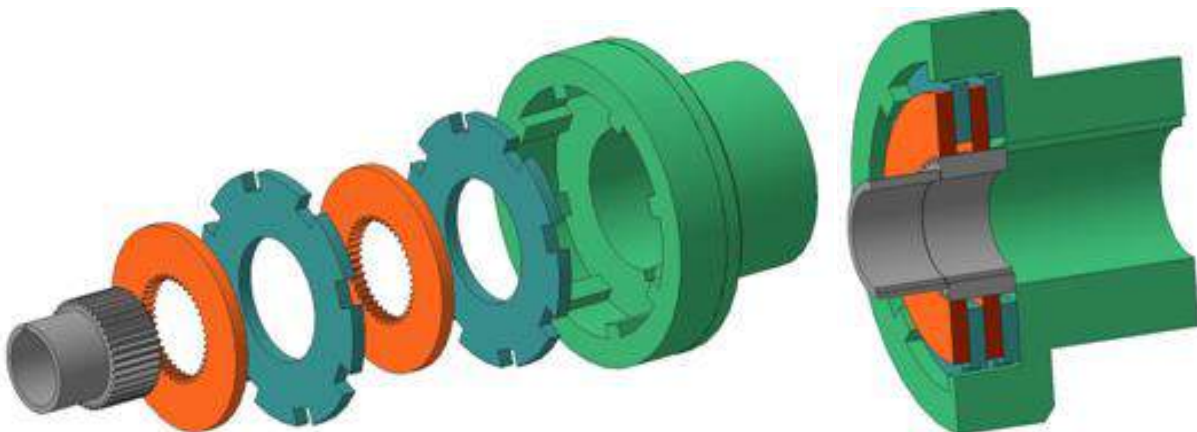


Fig. 6.20

A cone clutch transfers a higher torque than plate or disk clutches of the same size due to the wedging action and increased surface area. Cone clutches are generally now only used in low peripheral speed applications although they were once common in automobiles and other combustion engine transmissions. Small cone clutches are used in synchronizer mechanisms in manual transmissions.

The cone clutch consists of: one pair of friction surfaces – green female cone 1 (Fig. 6.21)) and blue male one; shaft 2: male cone is sliding on splines; friction element 3: usually on female cone, here on male cone; spring 4: brings the male cone back after using the clutch control; clutch control 5: separating both cones by pressing; rotating direction 6: both directions are possible.

In the clutch, the driver keyed to the driving shaft by a sunk key and has an inside conical surface or face which exactly fits into the outside conical surface of a driven. The driven member resting on the feather key in the driven shaft, may be shifted along the shaft by a forked lever provided, in order to engage the clutch by bringing the two conical surfaces in contact. Due to the frictional resistance set up at this contact surface, the torque is transmitted from one shaft to another. In some cases, a spring is placed around the driven shaft in contact with the hub of the driven. This spring holds the clutch faces in contact and maintains contact surfaces of the clutch may be metal to metal contact, but more often the driven member is lined with some material like wood, leather, cork or asbestos etc. The material of the clutch faces depends upon the allowable normal pressure and the coefficient of friction.

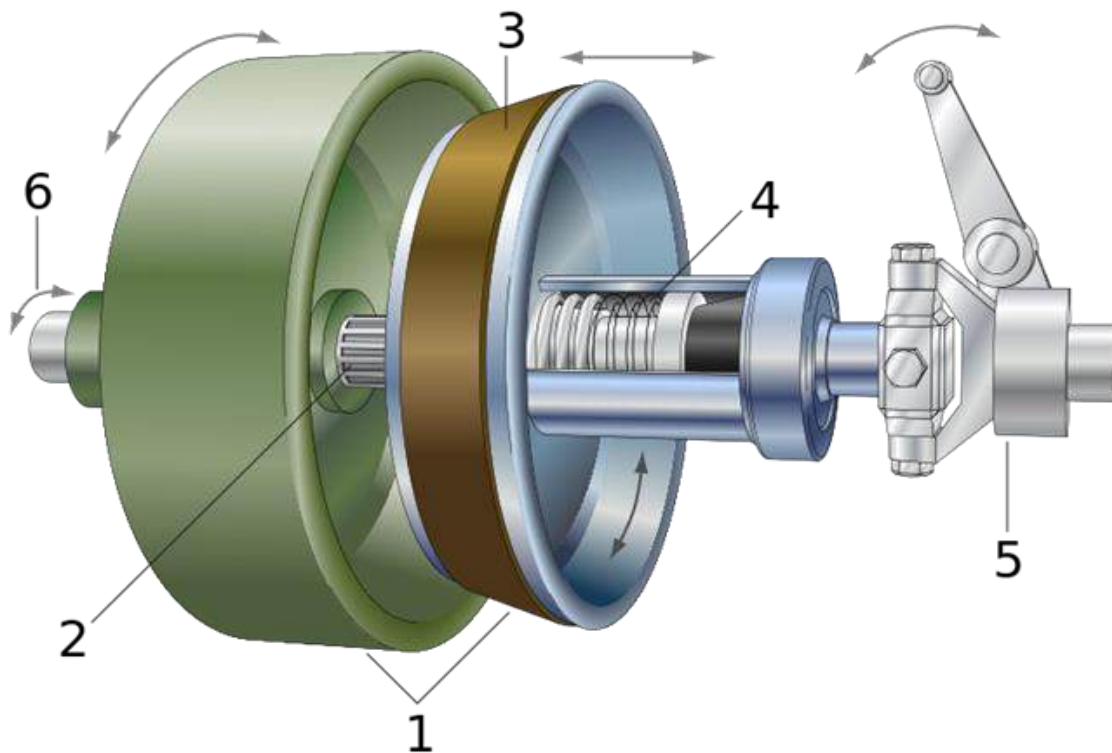


Fig. 6.21

Various materials have been used for the disc-friction facings. Modern clutches typically use a compound organic resin with copper wire facing or a ceramic material. A typical coefficient of friction used on a friction disc surface is 0.35 for organic and 0.25 for ceramic. Ceramic materials are typically used in heavy applications such as racing or heavy-duty hauling, though the harder ceramic materials increase flywheel and pressure plate wear.

Advantages	Disadvantages
– smooth turn on/off – can operate as safety clutch	– less power transmission ability at the same size – fast wear

6.4. Automatic Clutches

Breaks kinematic connection when the driven shaft's speed exceeds the driving one's. Automatic clutches provide:

- or reliable drive-line protection against damage that can occur during a mechanical overload;
- or switch on/off kinematic joint according to preliminary adjusted rotational speed.

Free-Wheel Clutch

In mechanical or automotive engineering, a **freewheel** or **overrunning clutch** is a device in a transmission that disengages the driveshaft from the driven shaft when the driven shaft rotates faster than the driveshaft. An overdrive is sometimes mistakenly called a freewheel, but is otherwise unrelated.

The condition of a driven shaft spinning faster than its driveshaft exists in helicopter autorotation mode, in most bicycles when the rider holds his or her feet still, no longer pushing the pedals. In a fixed-gear bicycle, without a freewheel, the rear wheel would drive the pedals around.

An analogous condition exists in an automobile with a manual transmission going down hill or any situation where the driver takes his or her foot off the gas pedal, closing the throttle; the wheels want to drive the engine, possibly at a higher RPM. In a two-stroke engine this can be a catastrophic situation: as many two stroke engines depend on a fuel/oil mixture for lubrication, a shortage of fuel to the engine would result in a shortage of oil in the cylinders, and the pistons would seize after a very short time causing extensive engine damage.

When driving shaft rotates clockwise (Fig. 6.22) torque is transferred by rollers caught in the wedge gap between the master and slave drives. When rotating in the

opposite direction or speed of the driven shaft more that speed of the drive one rollers move forward and connection is lost.

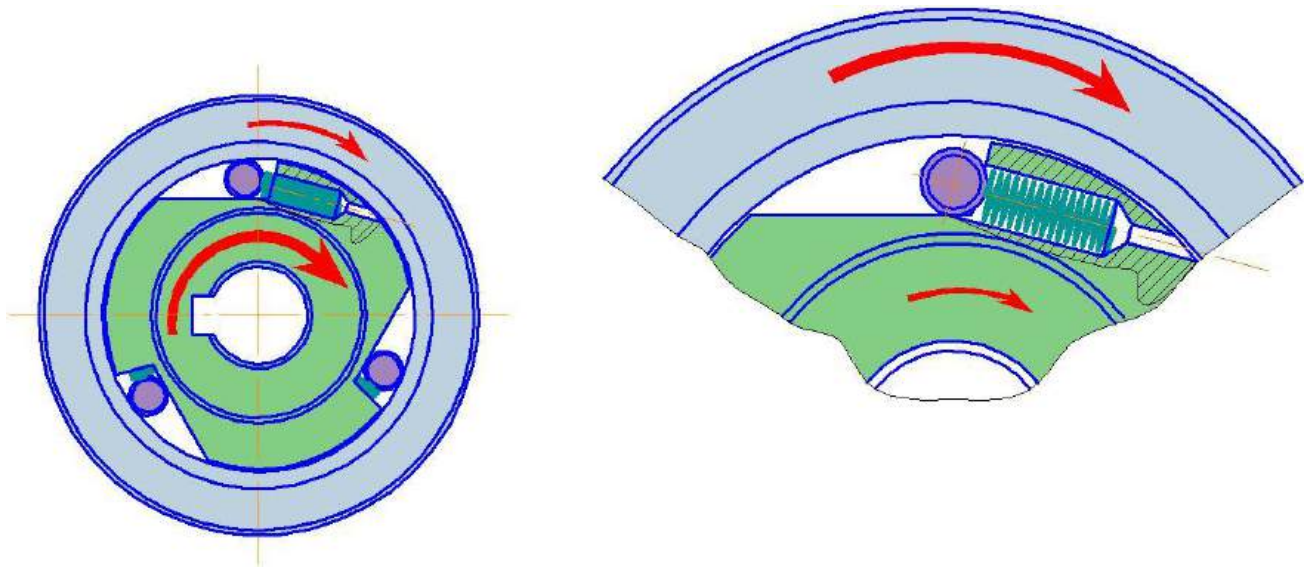


Fig. 6.22

Safety Clutches with a Ruining Element

When torque reaches a set limit, the pin (Fig. 6.23) is destroyed and the connection is lost. To restore torque transmission the element should be replaced.

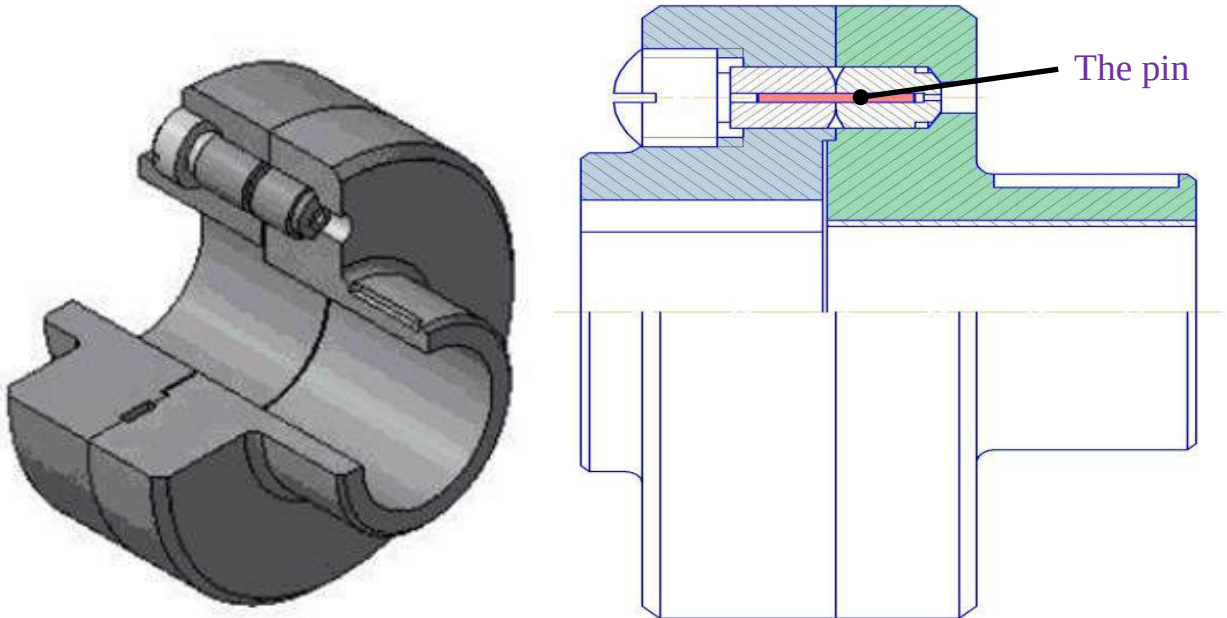


Fig. 6.23

Safety Frictional Clutch

Also known as a slip clutch, this device allows a rotating shaft to slip when higher than normal resistance is encountered on a machine (Fig. 6.24). When torque

reaches a set limit, the discs skid and the connection is lost. It is restored as torque level goes down. An example of a safety clutch is the one mounted on the driving shaft of a large grass mower. The clutch yields if the blades hit a rock, stump, or other immobile object, thus avoiding a potentially damaging torque transfer to the engine, possibly twisting or fracturing the crankshaft.

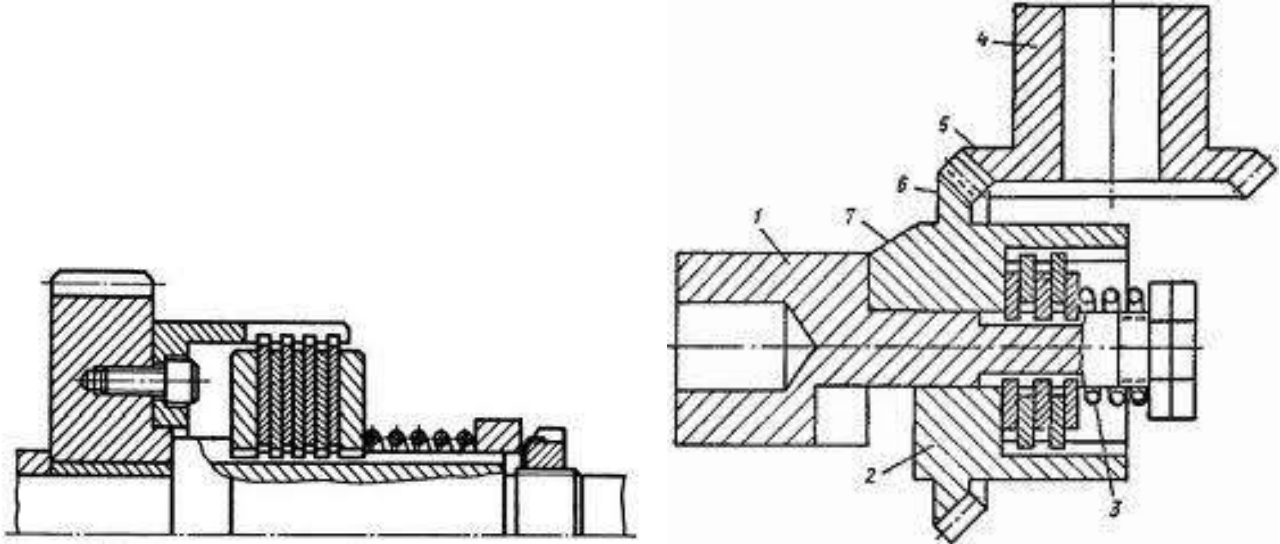


Fig. 6.24

Centrifugal Clutch

The centrifugal clutches are usually incorporated into the motor pulleys. It consists of a number of shoes on the inside of a rim of pulley 1 (Fig. 6.25). The outer surfaces of shoes 2 are covered with a friction material. These shoes, which can move radially in guides, are held against the boss (or spider) on the driving shaft by means of springs 3. The springs exert a radially inward force which is assumed constant. The mass of the shoe, when revolving, causes it to exert a radially outward force (i.e centrifugal force F_c).

The magnitude of this centrifugal force depends upon the speed at which the shoe is revolving. A little consideration will show that when the centrifugal force is less than spring force, the shoe remains in the same position as when the driving shaft was stationary, but when the centrifugal force is equals to the spring force, the shoe is just floating. When the centrifugal force exceeds the spring force, the shoe moves outward and comes into contact with the driven member and presses against it. The friction force F_f with which the shoe presses against the driven member is difference of the centrifugal force and the spring force. The increase of speed causes the shoe to press harder and enables more torque to be transmitted.

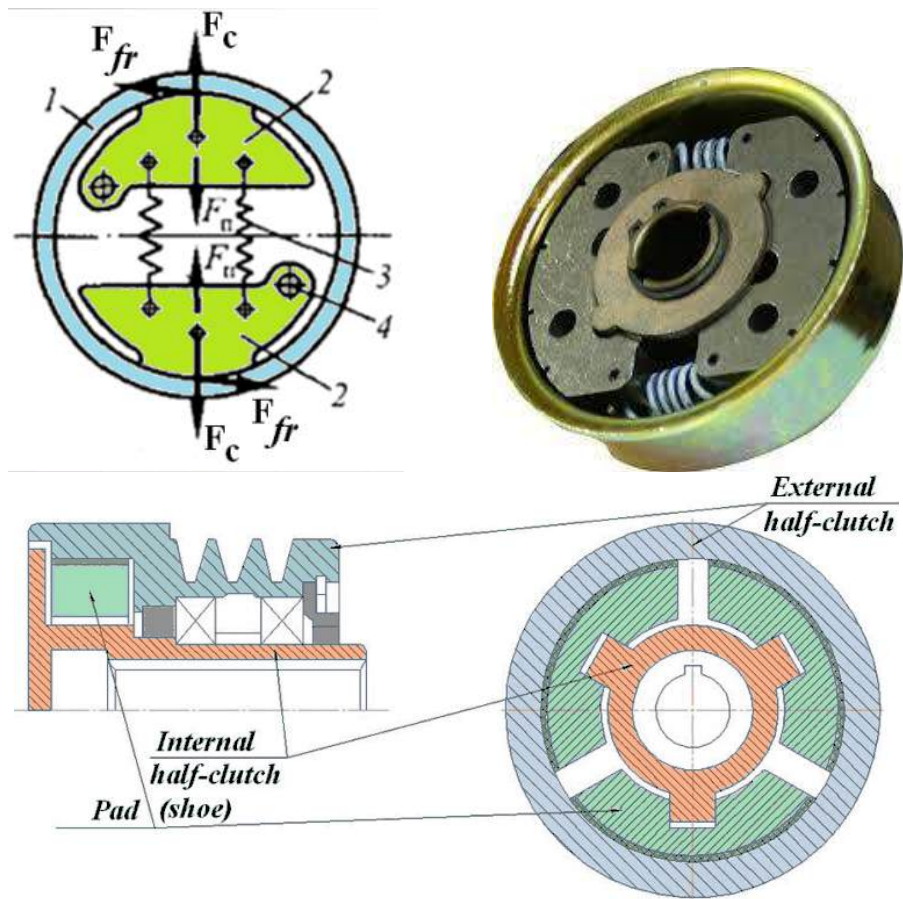


Fig. 6.25

REFERENCES

- Dobrovolsky, A. V. Machines elements / A. V. Dobrovolsky. – M., 1977.
- Dorofeyev, V. G. Screw-nut mechanisms [Text] : tutorial / V. G. Dorofeyev, V. N. Dotsenko, Yu. V. Koveza. – Kharkiv : National Aerospace University «Kharkiv Aviation Institute», 2011. – 40 p.
- Dorofeyev, V. G. Laboratory works on the course "Design of Machines and Mechanisms" [Text] : tutorial / V. G. Dorofeyev, Yu. V. Koveza. – Kharkiv : National Aerospace University «Kharkiv Aviation Institute», 2012. – 47 p.
- Gupta, J. K. A Textbook of Machine Elements/ R. S. Khurmi, J. K. Gupta. – Eurasia publishing house, 2005. – 1230 p.
- Kalpakjin, Serope. Manufacturing Engineering and Technology / Serope Kalpakjin. – Illinois Institute of Technology, 1990. – 1220 p.
- Koveza, Yu. Design of machine elements. Course project [Text] : tutorial / Yu. Koveza, I. Lykhosherst, S. Svitlychniy. – Kharkiv : National Aerospace University «Kharkiv Aviation Institute», 2014. – 180 p.
- Mechanisms : A Mechanical Designer's Workbook / Ed. in Chief J. E. Shigley, C. R. Mischke. – New York : McGraw-Hill Publishing Company, 1986. – 241 p .
- Shigley, Joseph E. Power Transmission Elements / Joseph E. Shigley, Charles R. Mischke. – University of Michigan, 1986. – 207 p.
- Spotts, M. F. Design of Machine Elements / Merhyle Franklin Spotts, Terry E. Shoup, Lee Emrey Hornberger. – Pearson Prentice Hall, 2004. – 928 p.
- Threaded joints : manual / A. Y. Cherniavskiy, K. P. Msallam, Z. A. Pogorelova ; Min. of Education and Science, Youth and Sports of Ukraine, Nat. Aerospace Univ. named after N. Ye. Zhukovskiy «Kharkiv Aviation Inst.». – Kharkiv : National Aerospace University «Kharkiv Aviation Institute», 2011. – 56 p.

Навчальне видання

**Доценко Володимир Миколайович
Ковеза Юрій Володимирович**

ДЕТАЛІ МАШИН І ОСНОВИ КОНСТРУЮВАННЯ

(Англійською мовою)

За редакцією Ю. В. Ковези
Технічний редактор А. М. Ємленінова

Зв. план, 2021

Підписано до видання 30.08.2021

Ум. друк. арк. 12,4. Обл.-вид. арк. 14. Електронний ресурс

Видавець і виготовлювач

Національний аерокосмічний університет ім. М. Є. Жуковського
«Харківський авіаційний інститут»
61070, Харків-70, вул. Чкалова, 17
<http://www.khai.edu>

Видавничий центр «ХАІ»
61070, Харків-70, вул. Чкалова, 17
izdat@khai.edu

Свідоцтво про внесення суб'єкта видавничої справи
до Державного реєстру видавців, виготовлювачів і розповсюджувачів
видавничої продукції сер. ДК № 391 від 30.03.2001