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National aerospace university «Kharkov aviation institute»

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SCREW-NUT MECHANISMS

Tutorial

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D74

Розглянуто основні механізми з передачею гвинт-гайка з тертям ковзання. Викладено методики розрахування та проектування найпростіших гвинтових механізмів, а також наведено їхні розрахункові схеми. Подано типові конструктивні рішення та порядок виконання розрахунків, а також довідкову інформацію про механічні характеристики матеріалів, параметри тертя та підшипники.

Для студентів машинобудівних вузів.

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D74 Screw-nut mechanisms [Text] : tutorial / V. G. Dorofeyev, V. N. Dotsenko, Yu. V. Koveza. – Kh. : National aerospace university «Kharkov aviation institute», 2011. – 40 p.

The tutorial considers general types of screw-nut mechanisms with sliding threads with the position of simple mechanism calculation and design methods as well as engineering solutions. Tutorial material is the data of practical importance including friction features, mechanical properties of the most commonly used production materials, typical thread and bearings dimensions.

For students of technical higher education establishments.

Fig. 33. Tab. 6. References: 6 titles.

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Introduction

The screw-nut transfer (or so called power screws) is employed both to transform rotary motion into lineal one and to produce large axial forces. Power screws are used in actuators of flight control systems (flap, slat, stabilizer drives, etc.), to lift weights (jacks) as well as in various technological processes (screw presses, bending presses, drive screws of machine tools, etc.). Power screws have a number of essential advantages: simple design and manufacturing; no additional equipment; high reliability; move smoothly and accurately; ensure self-locking.

Among disadvantages, we would like to mention: low efficiency; wear susceptibility; low speed; non-uniform thread loading.

Topic 1 SCHEMES OF MECHANISMS WITH POWER SCREWS

1.1. Jacks

The simple screw jack (Fig. 1.1) consists of the case or a frame 5, the cargo screw 4, a cup 1 for bearing lifted cargo, a nut 3 and handle 2 for screw rotation. In telescopic jacks (Fig. 1.2) two screws, screwed in each other are used. Such devices allow to carry out at rather small sizes lifting on the big height, therefore they are widely applied to air field service and to operate in the constrained conditions.

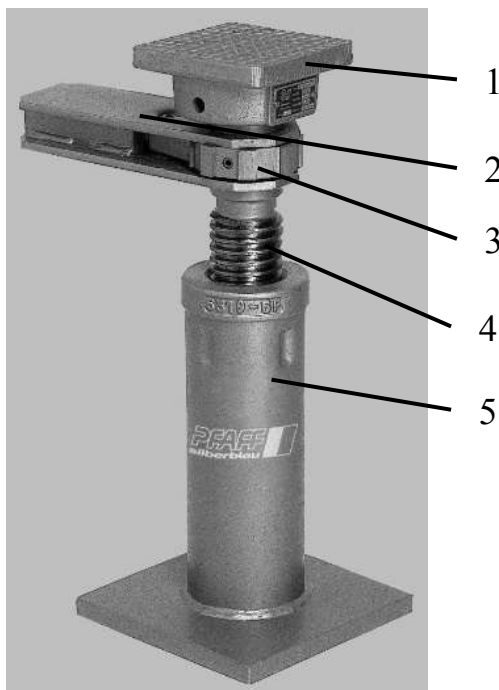


Fig. 1.1. Screw jack



Fig. 1.2. Telescopic jack

Sometimes handles are equipped with ratchets (Fig. 1.3) in order to replace rotary motion of the handle either with swinging motion or with mechanical (worm, bevel) transfer (Fig. 1.4) to reduce applied forces.

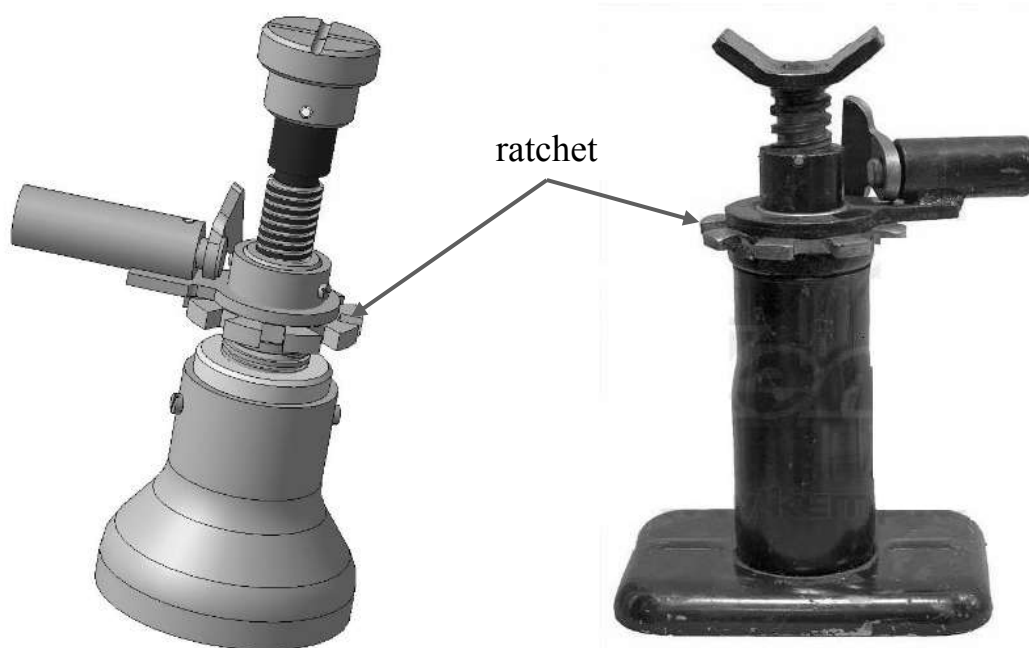


Fig. 1.3. Screw jack with ratchet



a



b

Fig. 1.4. Jacks with mechanical transfer:
a – worm; b – bevel

In portable jacks, drives actuated by the electric motor with a wave or planetary gearbox or a hydromotor (Fig. 1.5) are used along with handles. These me-

chanisms have the same basic elements as jacks with a hand drive; however, they are set in motion by an electric motor or a pump station.



Fig. 1.5. Screw jacks with a hydromotor

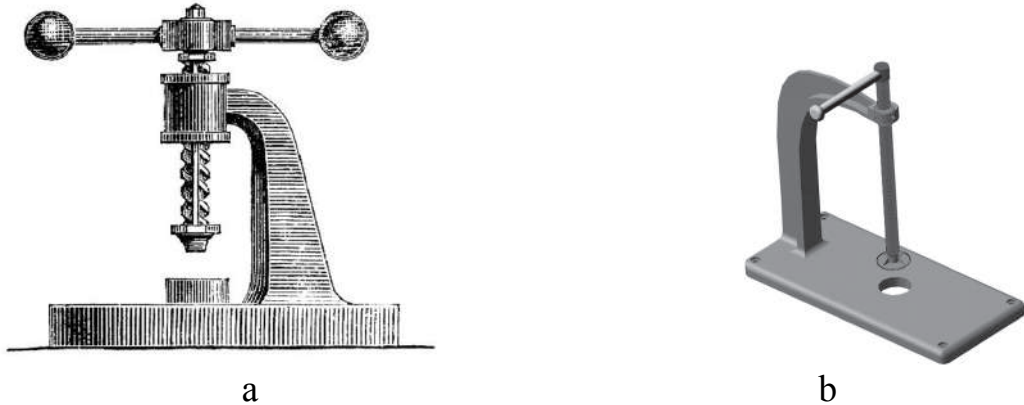
Some jacks consist of lever and screw mechanisms (Fig. 1.6) in which vertical rectilinear motion of a boot is achieved through gearing teeth cut in levers *a* and *b*.



Fig. 1.6. Jack with lever and screw mechanisms

1.2. The presses

Simple screw presses with a hand drive (Fig. 1.7) are used in repair shops to assemble and disassemble units, to produce elements by bending and punching as well as to mold products or fasten elements. Screw presses not only consist of the same elements as jacks but also can have hand or mechanical drive.



a

b

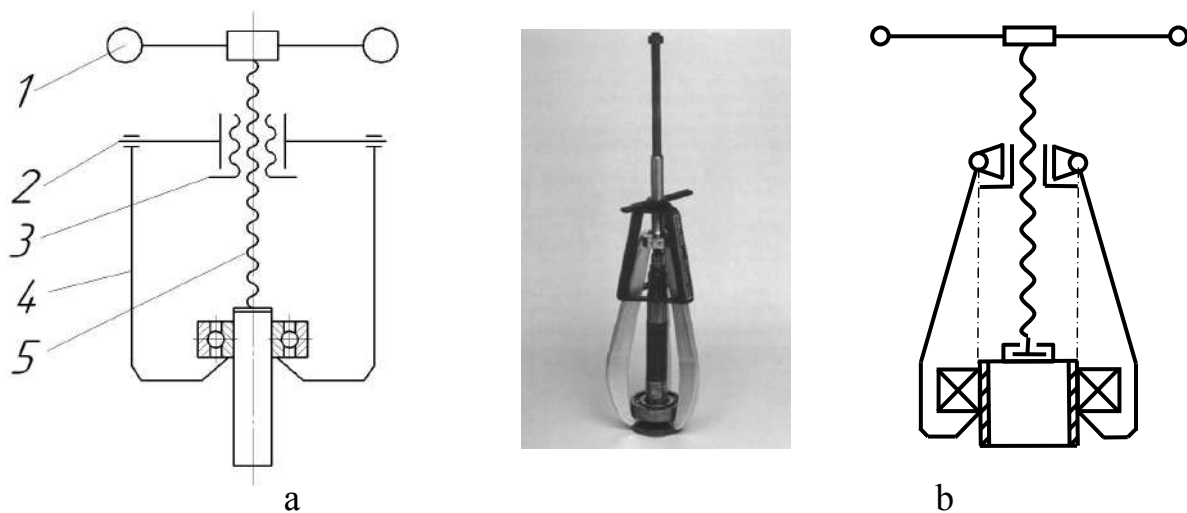
Fig. 1.7. Schemes of screw presses:

a – with one column and flat plate; b – with one column and hole

1.3. Screw bearing removers

Screw bearing removers (Fig. 1.8) are intended to dismantle units which elements are connected with tightness. Usually they include rolling bearings, various plugs, disks, pivots and other parts tightened to a shaft or into the hole of a case.

Removers with cross-arms (see Fig. 1.8, a) consist of two cross-arms lengthening a case 2, a screw 5, grips 4, a nut 3 and a handle 1.



a

b

Fig. 1.8. Screw bearing removers:
a – with a cross-arms; b – with turning grips

Removers with turning grips consist of the same elements as those described above but have axles instead of cross-arms for grips to turn about. In such removers, axes are located along the same line as the points of the axial force application that is why grips are free from additional bending moment.

Usually there are two or three grips.

To prevent dismantled bearings from destruction, an internal ring of the bearing should be captured in general case, and an external one – when removing the bearing from the case.

1.4. Mechanisms of flight control systems

In flight control systems, screw-nut transfers are organized according to two basic patterns:

1) with large displacements, the transfer consists of a rotating screw and a driven nut which moves translationally (Fig. 1.9). With this pattern, bilateral aerodynamic force acts on the case 1 which is rigidly connected to the nut 2. Then axial force is transferred to the screw 3 and through the case of the gearbox 4 to the plane power units. Mechanical energy is produced by the engine and transferred to the screw by transmission shafts;

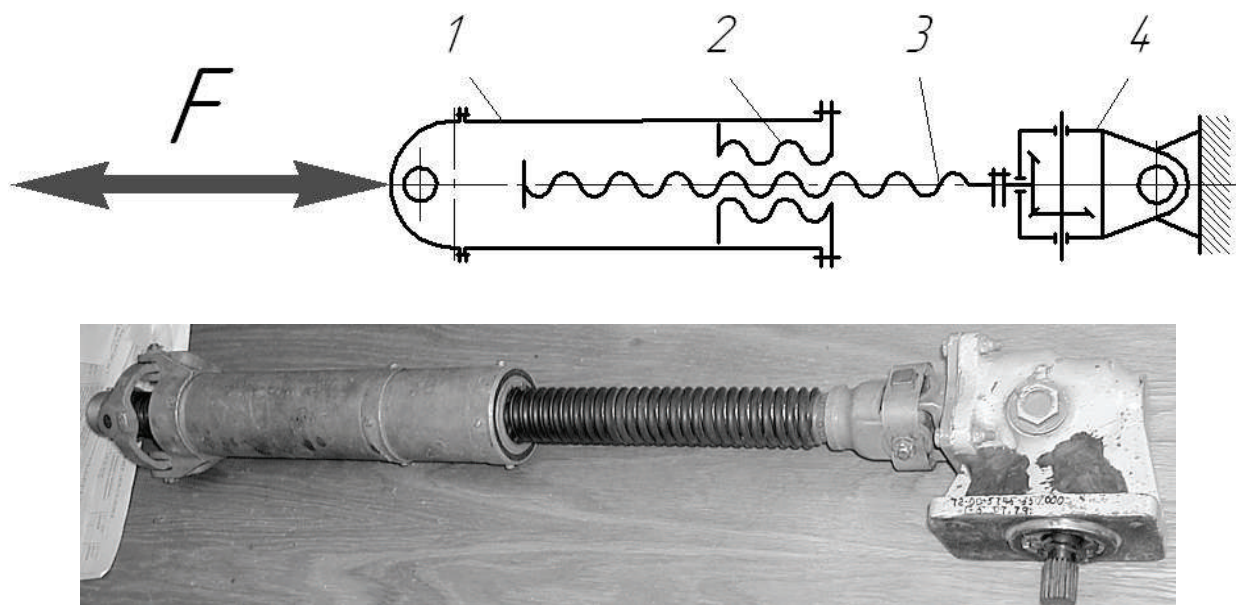


Fig. 1.9. Converter with a rotating screw

2) with small displacements and large axial forces, the transfer consists of a rotating nut and a driven screw which moves translationally (Fig. 1.10). In this case, axial force acts through the eye 1 directly on the screw 2, then through the nut 3 and on the case 4 as well as adjustment points. The nut is driven by the engine connected to it through shafts, clutches, and a tooth gearing.

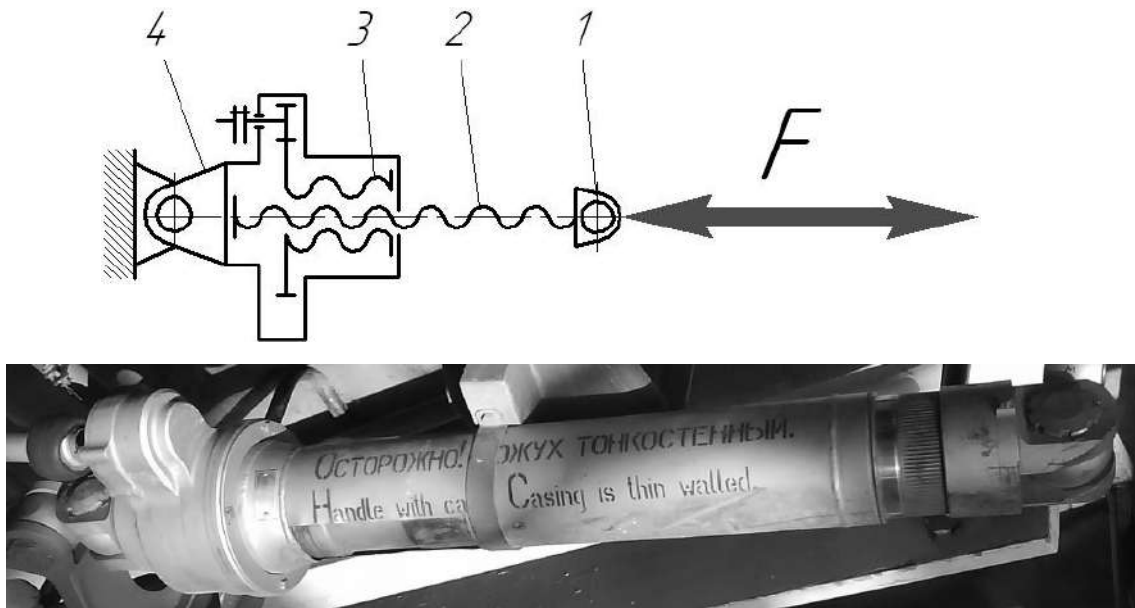


Fig. 1.10. Converter with a rotating nut

Topic 2 THREADS USED IN POWER SCREWS

2.1. Types of threads

In sliding power screws square, trapezoidal (Acme) and buttress threads are mainly applied. Square thread (Fig. 2.1) cannot be made by milling and grinding, one are not standardised and have no application in batch manufacturing.

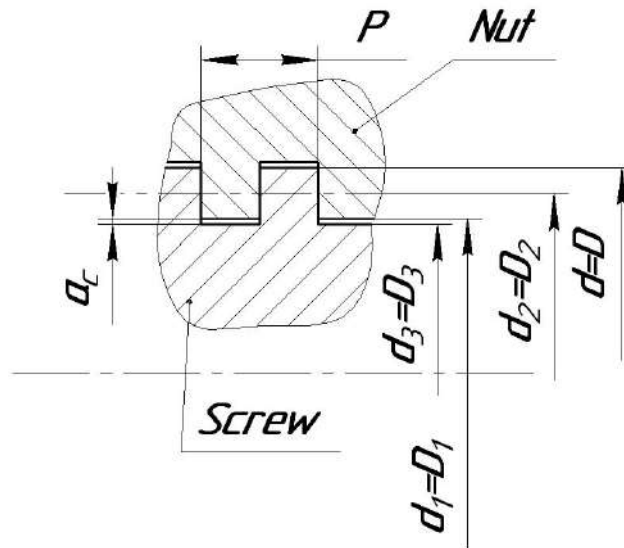


Fig. 2.1. Square thread

Depending on diameter, trapezoidal (ГОСТ 9484) (Fig. 2.2, a) and buttress (ГОСТ 10177) (Fig. 2.2, b) threads are divided into three rows according to

application preferences. The first row should be preferred to the second one, and the second row should be preferred to the third one.

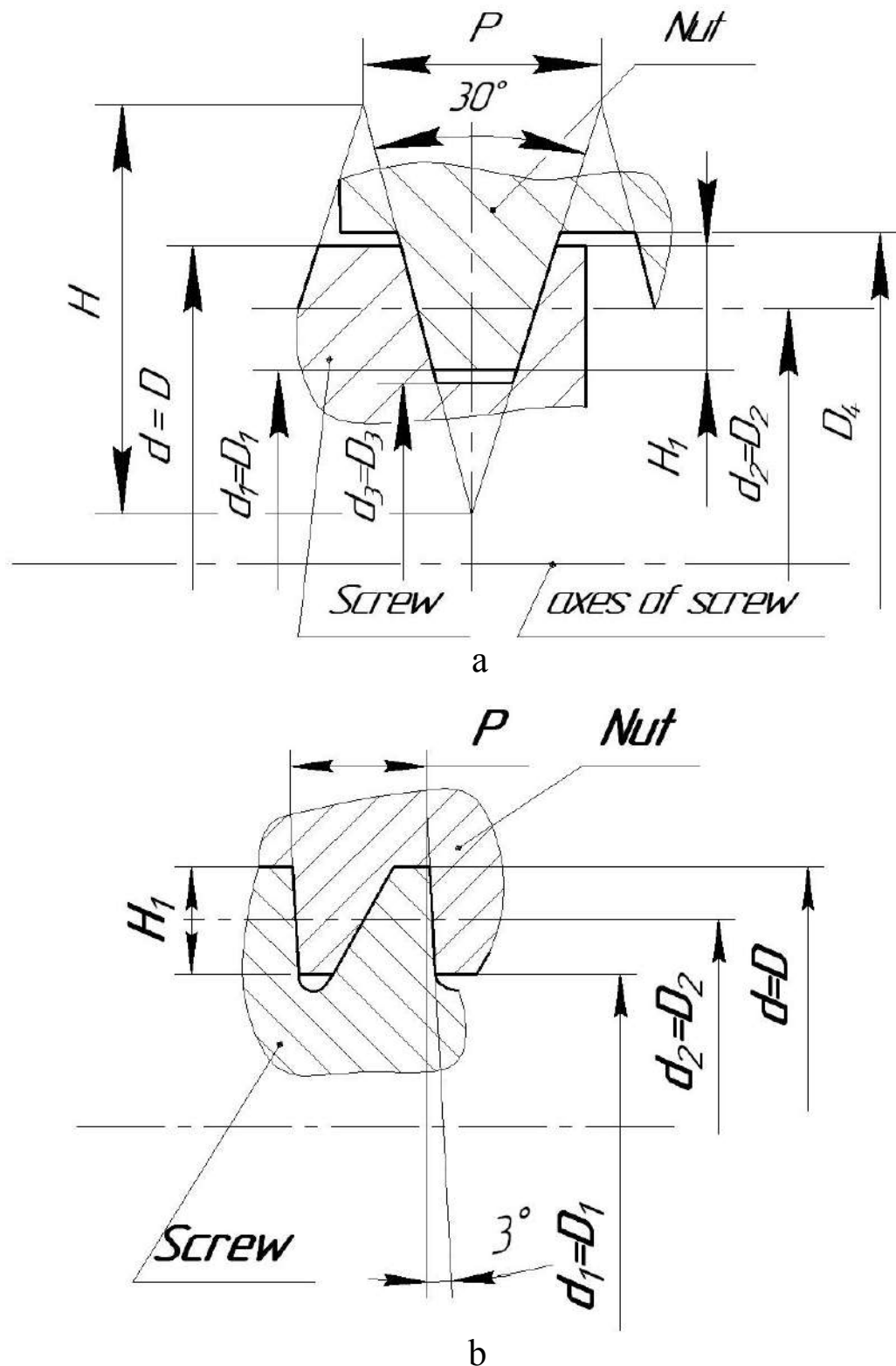


Fig. 2.2. Trapezoidal (a) and buttress (b) threads

After comparing thicknesses at the roots of different threads (see Fig. 2.2) we see that shear and bending strengths of a buttress thread are greater than those

of a trapezoidal one and, especially, square one. Besides, because of a larger radius of the root, buttress threads have a lower concentration factor and higher fatigue strength. For these reasons, buttress threads have found wide application in mechanisms with unidirectional loading. When the thread takes two-directional forces, symmetric trapezoidal threads are used.

2.2. The basic geometrical parameters of a thread

Basic geometrical parameters of a thread are:

1. Diameters. Threads have an external, or major, diameter (D, d), a pitch, or mean, diameter (D_2, d_2), an internal, or minor, diameter of a thread (D_1, d_1), and an internal diameter of a screw root (d_3). Screw diameters are designated with small letters, and nut diameters – with capital ones. Nominal values of diameters with the same name are equal. On the surface of an imagined cylinder with a mean diameter d_2 , the width of screw and nut threads are identical.

Threads are specified according to major diameters, that's why it is considered a nominal, or basic, size.

2. Profile is a view of a crest and a thread root in the plane of its longitudinal axial section. An apex angle is an angle between adjacent lateral faces of a thread. For a square thread, it is equal 0° , for a trapezoidal one – to 30° , for a metric one – to 60° . Buttress threads are asymmetrical, that's why their active face angle is 3° and that corresponding to inactive one – to 30° . A thread profile is also characterised by the initial triangle H altitude and the H_1 profile working height (see Fig. 2.1, 2.2).

3. A thread pitch P is a distance between average points of the nearest same-name lateral faces of a thread profile lying in the same axial plane and to the same side of the thread axis. It is plotted along the line parallel to the thread axis.

A standard thread is designated by a combination of a conventional letter code (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 by S 32x5.

Numerical values of thread's geometrical parameters are shown in the appendix.

2.3. Efficiency

As any mechanism, threads have characteristics reflecting its degree of sophistication and ability to operate in certain working conditions.

In power screws, the work of friction forces is characterised by a universal parameter, efficiency,

$$\eta = \frac{\tan \psi}{\tan(\psi + \rho')}, \quad (2.1)$$

where $\psi = \operatorname{arctg} \frac{t}{\pi d_2}$ is a helix angle of a mean screw line;

$t = iP$ is a thread throw;

d_2 is a pitch diameter;

$\rho' = \operatorname{arctg} \frac{f}{\cos \alpha'}$ is a reduced angle of friction;

f is a coefficient of friction between a screw and a nut;

α' is a working profile angle (for a metric threads it is equal to 30° , for trapezoidal ones – to 15° , for buttress ones – to 3°);

P is a pitch;

i is a number of starts.

Intensively working power screws should be carried out with the greatest possible efficiency which, at identical pitch diameters, increases with growing ψ and P .

As we see from (2.1), the smaller the reduced angle of friction, the higher the efficiency, with the angle decreasing together with the decrease of α' and the drop of the coefficient of friction. Therefore, under other equal conditions, power screws with square (a half apex angle of 0°) and buttress (a working profile angle of 3°) threads have higher efficiency than those with trapezoidal threads (a half apex angle of 15°).

However, this difference is little and does not exceed 5%, that's why we can neglect this angle when calculating the reduced angle of friction in **trapezoidal** and **buttress** threads:

$$\rho' = \operatorname{arctg} \frac{f}{\cos \alpha'} \approx \operatorname{arctg} f \quad (2.2)$$

To lower the coefficient of friction, antifrictional materials with high surface quality as well as lubricants are used. Coefficients of friction for a steel screw and a bronze nut with poor lubrication lie in the range 0,12...0,15.

2.4. Self-locking

Under certain conditions, threads possess the property of self-locking which is sometimes useful. 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.

As we know from the theory of machines and mechanisms, the self-locking occurs at $\psi < \rho'$, therefore, the helix angle of a screw line is limited and the efficiency of a self-locking screw pair does not exceed 0,5.

In drives which do not require self-locking (for example, with power screws of flap drives), a bigger pitch or two-start threads are used.

2.5. The torque required to lift the loads by power screws

Total frictional torque for V-thread at pitch diameter is

$$T_t = F \frac{d_2}{2} \operatorname{tg}(\psi + \rho'), \quad (2.3)$$

where F is from now on a given external axial load.

In (2.3), a share of a useful resistance is represented by a variable ψ , and a share of a resistance to friction in a thread – by ρ' .

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

Topic 3 SCREW CALCULATION

3.1. Screw materials

Carbon and alloyed steels (35, 45, 40X, 40XH, etc.) are usually used for manufacturing screws. They have good mechanical properties ($\sigma_y \geq 450$ MPa), good machinability and high surface hardness (to HRC 45 and more) after case-hardening (see appendix 1).

3.2. The dangerous section

The diagram of operating forces and the torques for typical variants is shown in Fig. 3.1.

Linear force distribution on threads is based on 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 .

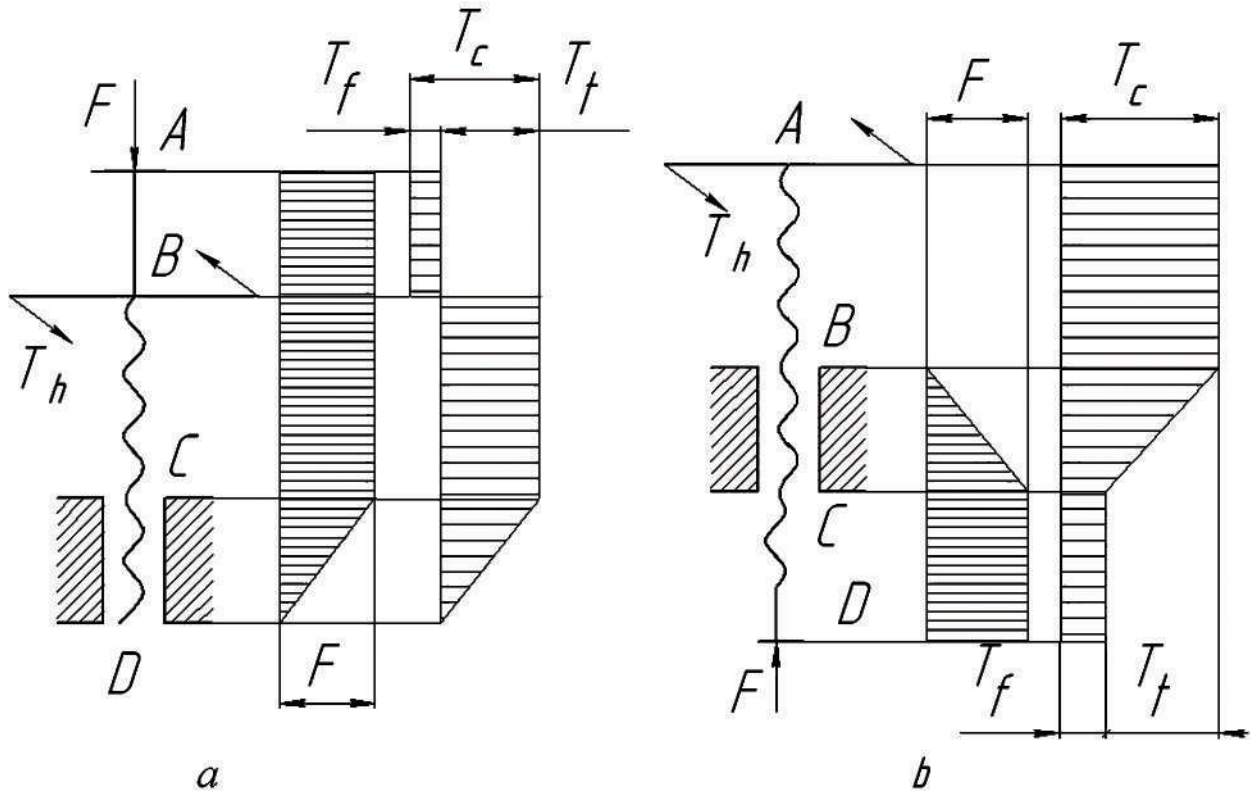


Fig. 3.1. Force and torques diagrams occurring in the typical screws:
a – jacks; b – removers and presses

3.3. Design calculation

In calculation, screw diameter is determined proceeding from the following conditions: compression strength under the action of longitudinal force, screw stability, and wear resistance of a thread. The last condition, as a rule, is the most dangerous, that is why preliminary sizes are usually found with respect to it, and other two are used for verifying calculation.

Wear resistance exhibited by working surfaces of thread coils limits specific stresses between screw and nut coils so that it is necessary to provide a lubricant between them:

$$p = \frac{F}{A} = \frac{4F}{\pi(d^2 - d_1^2)} \approx \frac{F}{\pi d_2 H_1 z} \leq [p] \quad (3.1)$$

whence

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

Here d_2 is a pitch diameter; H_1 is an effective height of a thread profile;

z is a number of threads; $\psi_H = H_n/d_2$ is a nut height factor (for lead screws, accept it of 1,2...2,5); H_n is a nut height; $\psi_h = H_1/P$ is a standard height factor of a thread (for trapezoidal threads it is equal to 0,5, for buttress ones – to 0,75); P is a pitch; $[p]$ is an allowable bearing pressure which depends on a specific pair of materials (Table 3.1).

Table 3.1

Allowable bearing pressures

Materials	Allowable bearing pressure $[p]$, MPa (N/mm ²)
Steel to cast iron	5...6
Steel to tinless bronze	5...7
Steel to steel	7...15
Steel to tin bronze	10...15

After the value of diameter d_2 is calculated, we should chose a standard thread (with a nominal diameter d and a pitch P) for which d_{2st} is greater than that calculated (see appendix 2). As a rule, there are several combination variants of threads and pitches. We should take into account that it is desirable for the efficiency to increase together with the pitch growing (see (2.1)) but undesirable for the driving moment (see (2.2)) to grow too. So, we better choose an average pitch value.

3.4. Revised calculation

The self-locking condition $l^\circ < \rho' - \psi$ and the efficiency (see (2.1)) should be checked when conducting revised calculations.

Combined maximum normal stresses (according to the third theory of failure) are calculated by

$$\sigma_{max} = \sqrt{\sigma^2 + 4\tau^2} \leq \varphi[\sigma], \quad (3.2)$$

where $\sigma = F/A$ is a compression stress in a dangerous cross-section;

$A = \frac{\pi}{4} d_1^2 (1 - \alpha^2)$ is the cross-section area of the screw with diameter d_1 . 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 %) as the screw cross-section is not perfectly round (Fig. 3.2);

T is a torque in a dangerous section (see (2.2));

$W_\rho = \frac{\pi d_1^3}{16} (1 - \alpha^4)$ is a polar section modulus of the screw;

$\tau = \frac{T}{W_\rho}$ is a maximum shear stress;

$\alpha = d_h / d_3$ in the case of a hollow screw (for solid section $\alpha = 0$);

d_h is the internal diameter of the hollow screw cross-section;

$[\sigma] = \frac{\sigma_y}{n}$ is an allowable stress;

σ_y is an yield stress of a screw production material (see appendix 1);

$[n]$ is a design safety factor (for screws of jacks $[n] = 4...6$, for screws of removers and presses $[n] = 2...3$);

φ is a reduction factor of the basic allowable stress in the case of compressed screw, for tension screws, $\varphi = 1$. It is taken from Table 3.2 depending on a material of which a screw is made and the actual flexibility of a screw

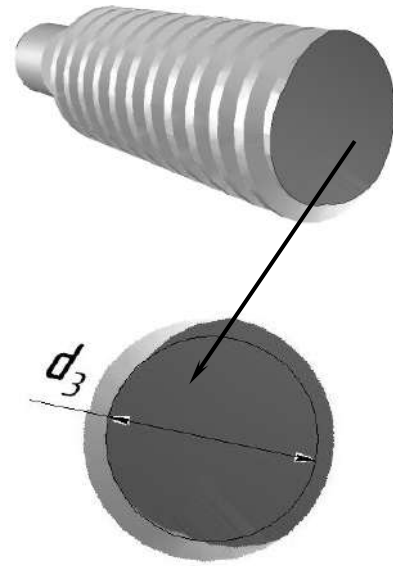


Fig. 3.2. Screw cross-section

$$\lambda = \frac{\nu l}{i_{min}} = \frac{\nu l}{\theta d_1}. \quad (3.3)$$

Here l is a maximum effective length of the screw (is usually given); i_{min} is an inertia radius of the screw cross-section, is equals to θd_1 ; the factor θ depends on completeness of the screw cross-section: for solid section, $\theta = 0,25$; for hollow ones

α	0,5	0,6	0,7	0,8	0,9
θ	0,28	0,29	0,305	0,317	0,34

$\nu = 0,5...2$ – is a reduction factor of the effective screw length. For different screw end fixing, values ν are specified in Fig. 3.3.

Sliding supports at $\frac{l}{d_1} \leq 2$ as well as 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.

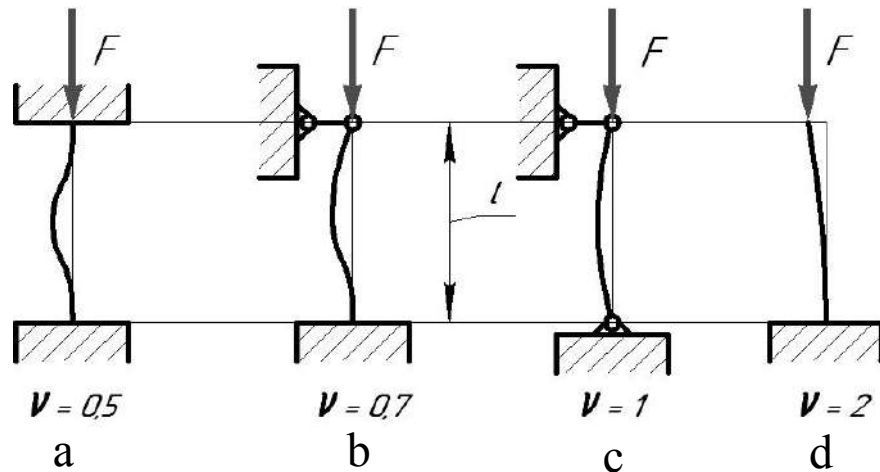


Fig. 3.3. Reduction factors of the effective screw length

Table 3.2

Reduction factor of basic allowable stress

Flexibility λ	Steel			Duralumin Д16-Т
	carbon		alloyed	
	C < 0,4%	C > 0,4%		
40	0,92	0,9	0,9	0,7
50	0,89	0,84	0,83	0,57
60	0,86	0,8	0,78	0,45
70	0,81	0,74	0,71	0,35
80	0,75	0,69	0,66	0,27
90	0,69	0,59	0,54	0,21
100	0,6	0,5	0,45	0,17
110	0,52	0,43	0,39	0,14
120	0,45	0,38	0,33	0,12

If the condition 3.2 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.

Topic 4 NUT CALCULATION

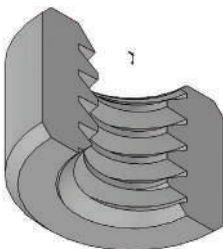


Fig. 4.1. Nut

Nuts are turned in a form of cylindrical bush (Fig. 4.1) and pressed or screwed into the case. A nut can be assembled according to one of the patterns presented in Fig. 4.2.

The screw-nut friction coefficient depends on a nut production material (see

Table 4.1) and a lubricant, therefore a nut is usually made of materials which in combination with a steel screw prove a low friction coefficient and good wear resistance. Such materials include tin and tinless bronze, brass, cermet, and aluminum alloys.

Under compressing loads, the best force distribution on coils is reached in variants "a" and "b" when nut and screw coils are compressed. In the case of a tension screw, variant "d" is preferred as nut coils are also extended.

Nut calculation includes determination of a shear and bearing strength of a thread, external dimensions (H_n , D), and other elements dimensions (a thrust fillet Δ , fixing elements, etc.).

Shear strength of a nut coil is greater than that of a screw one (with identical materials) because of a longer coil involute at the diameter D . However, in a steel-to-bronze pair, the material of which a nut is made usually has worse mechanical properties. That is why mechanical properties of nut coils should be checked.

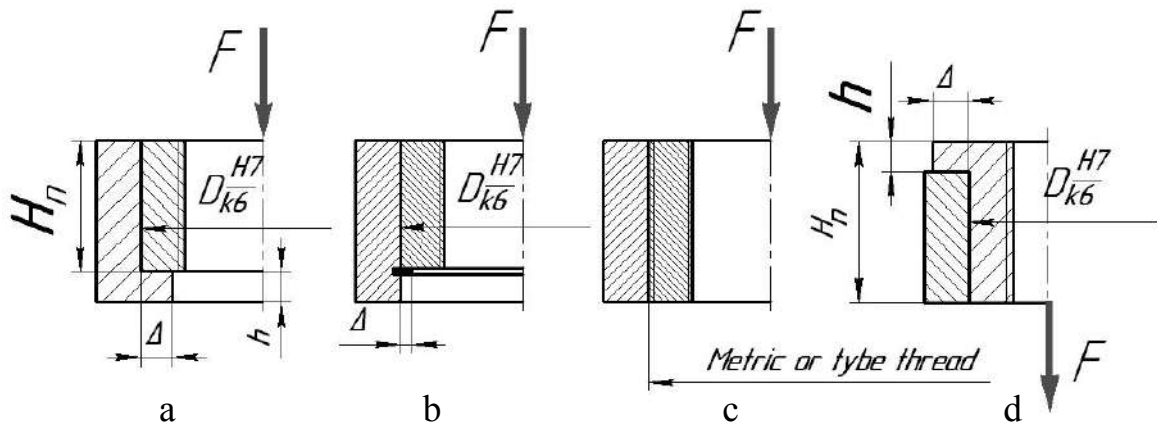


Fig. 4.2. Types of nut assembling:

- a – a fillet on a case;
- b – a ring used as a fillet;
- c – a nut fixed by thread;
- d – a fillet on a nut

A nut is calculated at following assumptions:

- axial force F is uniformly distributed among threads;
- helix angle is so small, that each coil can be considered like a flat circular rings.

A number of threads is determined on the wear resistance (3.1):

$$z = \frac{4F}{[p]\pi[d^2 - d_1^2]} \quad (4.1)$$

It should be smaller than or equal to 10 for all the coils to be loaded. It is also undesirable to use less than 5 coils because of the insufficient screw centering.

If the design height of nut $H_n = zP$ is smaller than $(1...1,25)d_2$ it should be chosen from the range of $(1,2...2)d$ (greater values are recommended for sim-

ple jacks because a nut should resist considerable bending moments caused by eccentric loads).

A nut thread suffer a shear deformations on cylindrical surfaces with area of

$$A_s = z \pi D_4 h_0 = z \pi D_4 m P$$

(Fig. 4.3) and bearing deformation on surfaces with

$$A_b = \frac{z \pi}{4} [d^2 - (d_1)^2] = 0,78 z [d^2 - (d_1)^2].$$

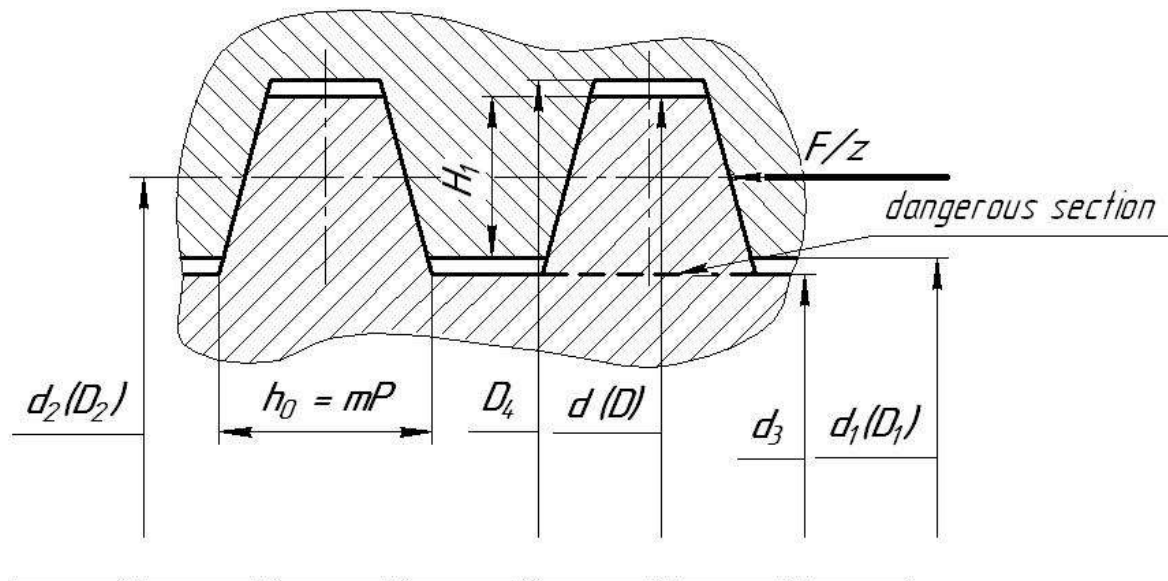


Fig. 4.3. Dimensions of thread coils

In order to simplify calculations, a coil is conditionally cut across the helix line, evolved and represented like a cantilever with a square cross-section in a place of fixation (Fig. 4.4). The length of a rectangle is equal to the length of a circle with diameter D , and its width – to the height mP (for buttress thread, $m = 0,74$; for trapezoidal one, – $0,64$).

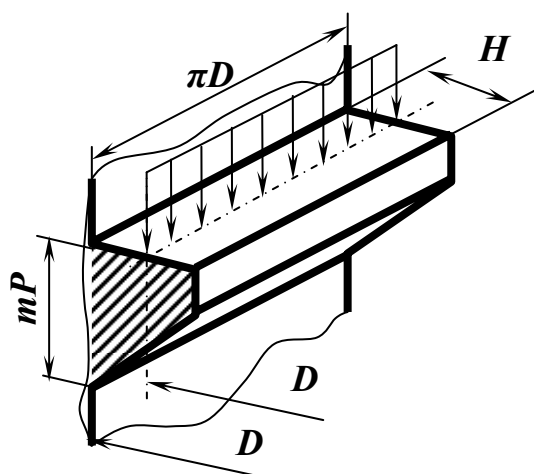


Fig. 4.4. Thread coil scheme for calculation on shear

The equation of a shear strength of a single loaded coil (F/z) takes the following form (we can neglect the difference between d and D_4)

$$\tau_s = \frac{F}{zA} = \frac{F}{z \pi d m P} \leq [\tau_s]. \quad (4.2)$$

For the purposes of design calculations, allowable shear stresses are determined from catalogues or from empirical equations:

$$[\tau_s] \approx 0,6 [\sigma_b], [\sigma_b] = (0,15 \dots 0,3) \sigma_B. \quad (4.3)$$

For values σ_B for different bronzes, see Table 4.1.

Table 4.1

The nuts' materials		
Material	GOST	σ_B , MPa (N/mm ²)
Tin bronze E = 80000 MPa, HB = 60		
БрОФ 10-1	613-79	300
БрОЦ 4-3	"	250
БрОЦСН 3-7-5-1	"	180
БрОЦС 3-5-5	"	180
Tinless bronze E = 120000 MPa, HB = 100		
БрАЖН 10-4-4	493-79	650
БрАЖН 10-4-4	"	600
БрАЖМц10-3-1	"	600
БрАЖМц 10-3-1	"	500
БрАЖ 9-4	"	500
БрАЖ 9-4	"	400

Note that A stands for aluminium, Ж stands for iron, Мц stands for manganese, Ф stands for phosphorus, and σ_u is an average ultimate strength in tension.

External nut diameter is chosen proceeding from design reasons:

$$D_n = d + (2 \dots 4) P.$$

To determine dimensions of a thrust fillet Δ (see Fig. 4.2, a, b, d), use the bearing strength condition of a nut production material (which is usually less strong than that of a case) under the action of force F

$$\sigma_{br} \cong \frac{F}{\pi D_n \Delta} \leq [\sigma_{br}], \quad (4.5)$$

where

$$[\sigma_{br}] \approx 2[\sigma_b].$$

The height of a thrust fillet h (see Fig. 4.2, a, b, d) is determined from a bending condition under the action of force F . Like in the calculation of a thread coil, a ring thrust fillet is conditionally cut and evolved in a line. Thus,

- remember that the force is applied at the maximum distance from the dangerous section, i.e. at the edge of an internal hole;
- neglect friction in the nut surface.

Subject to the assumptions made above, we can obtain the thrust fillet height from a bending strength

$$\sigma_b = \frac{M_b}{W_x} \leq [\sigma_b]. \quad (4.6)$$

For square section

$$W_x = \frac{bh^2}{6} = \frac{\pi Dh^2}{6} \quad \text{and} \quad h = \frac{M_b}{W_x} = \sqrt{\frac{6F\Delta}{\pi D[\sigma_b]}}.$$

Note that in this formula we should substitute allowable stress $[\sigma_b]$ of the thrust fillet production material.

If a nut is mounted in the case according to "a" (see Fig. 4.2) $[\sigma_b]$ is taken for the case production material, and in the case "d" – for that of nuts.

To avoid nut displacement in the case, it is not only press-fitted but also fixed with steel pins (Fig. 4.5, a) or special adjusting screws (Fig. 4.5, b) with cylindrical or conic shafts.

Dimensions of these elements could be determined on the basis of a pin shear along the cross-section of a diameter d_0 and lateral surface bearing with an area $A = d_0 l$ under the action of a screw pair torque:

$$\tau_s = \frac{F}{A_s} = \frac{2T_t}{D} \frac{4}{\pi d_0^2} \leq [\tau_s], \quad (4.7)$$

$$\sigma_{br} = \frac{F}{A_{br}} = \frac{2T_t}{D l d_0} \leq [\sigma_{br}],$$

where $[\tau_s]$ is an allowable shear stress of a pin production material (as a rule, pins are made of constructional, carbon, or alloy steels 45, 65Г) which is taken from the reference materials; $[\sigma_{br}]$ is an allowable bearing stress of a nut production material because it is usual weaker than that of a pin.

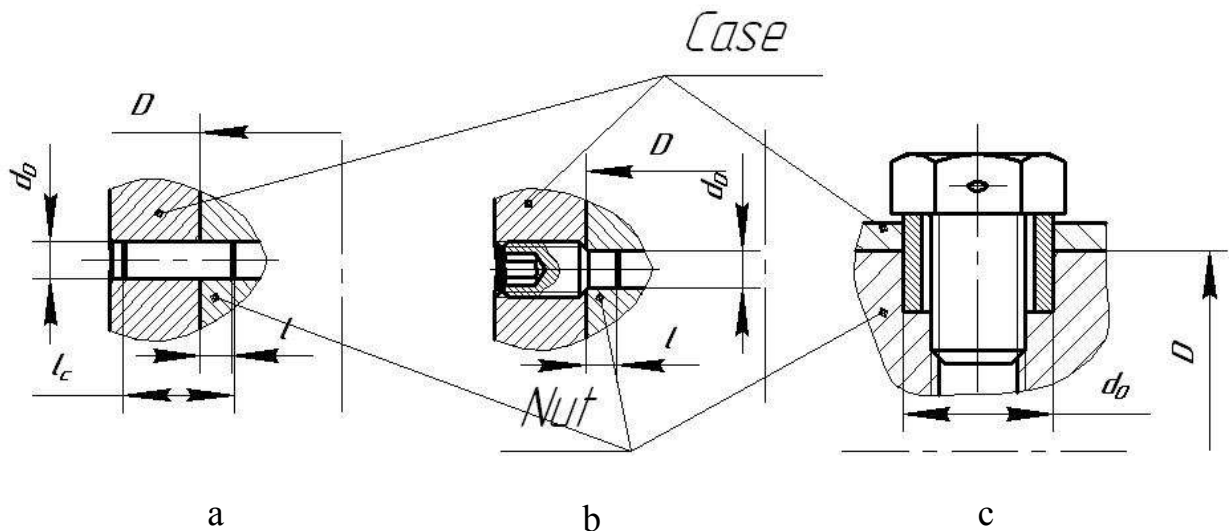


Fig. 4.5. Fixing nut by means of:
a – pin; b – screw; c – bush

At first, we calculate the diameter of a fixing element d_{op} from the first formula of (4.7) and then chose the nearest (greater) standard value (for example, in accordance with ГOCT 3128-70, 8878-93, etc.). Length l is calculated from the second formula of (4.7) with the standard value of d_{θ} . At last, we add a case wall thickness (usually 6...8 mm) to the length and chose the nearest (greater) standard length of a pin (screw) l_c .

Topic 5 PIVOT CALCULATION

Rotating screws carry axial loads by means of pivots. Pivots can be flat (Fig. 5.1, a), ring (Fig. 5.1, b), spherical (Fig. 5.1, c), and have a fitting place for the thrust ball bearing (Fig. 5.1, d).

Diameter of flat and ring pivot is determined from the wear resistance condition of a friction surface and depends on allowable bearing pressure $[p]$.

Diameter of a flat pivot

$$d = \sqrt{\frac{4F}{[p]\pi}} \quad (5.1)$$

Diameter of a ring pivot

$$d = \sqrt{\frac{4F}{\pi(1-\alpha^2)[p]}} \quad (5.2)$$

where $\alpha = 0,5...0,7$ is a fullness factor of a pivot bearing surface, $\alpha = d_{in} / d_{ex}$ (see Fig. 5.1, b).

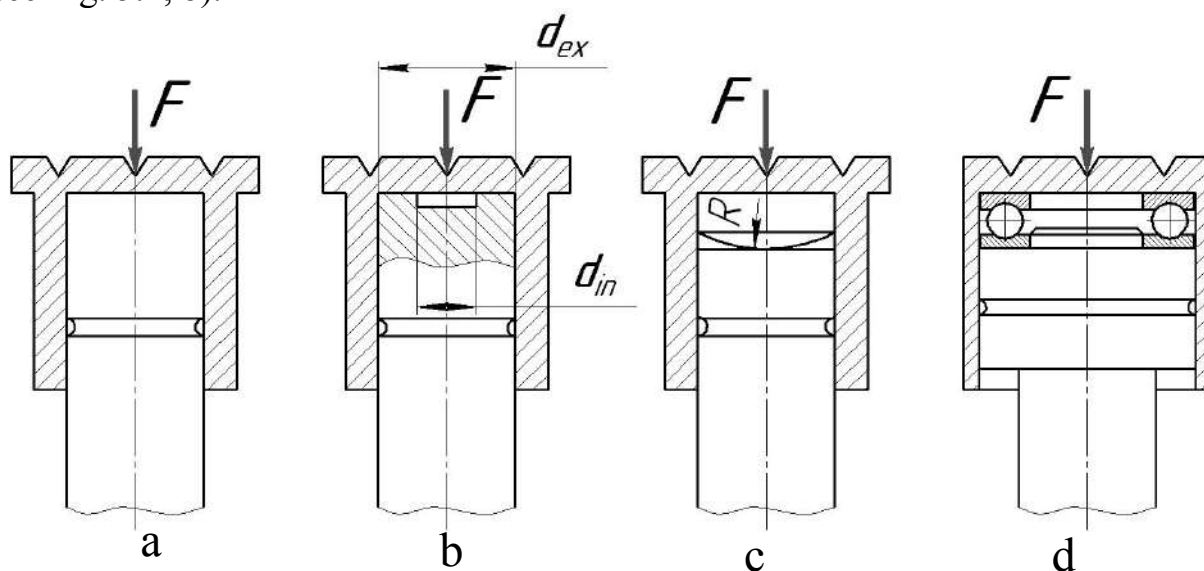


Fig. 5.1. Types of pivot:
a – flat; b – ring; c – spherical; d – with a ball bearing

Allowable bearing pressure $[p]$ depends on the pair materials, surface hardness and relative speed. Its values can be taken from Table 3.1.

In a spherical pivot (one element of the pair is flat), the sphere radius is determined on the basis of the Hertz contact strength equation

$$\sigma_H = 0,388 \sqrt[3]{F \frac{E^2}{R^2}} \leq [\sigma_H], \quad (5.3)$$

where R is a sphere radius; $[\sigma_H]$ is an allowable contact stress, MPa (N/mm²). For high-carbon or the alloyed steels with surface hardness H_{HRC} more than 45 HRC, we should take $[\sigma_H] = (30 \dots 80) H_{HRC}$.

For standard thrust ball bearings in mechanisms with a hand drive, static load-carrying capacity is calculated by

$$C_0 = SF, \quad (5.4)$$

where $S = 1,2 \dots 1,5$ is a safety factor; C_0 is a basic static load capacity (N) which is the basis for the thrust bearing to be taken from the catalogue (see appendix 3) with the greater static load capacity than that calculated. The thrust bearing parameters include a symbolic notation, internal d and external D diameters, and height H .

For different pivots, the friction moment is found as follows:

– for a flat pivot

$$T_f = \frac{2}{3} F f \frac{d_p}{2}, \quad (5.5)$$

where d_p is a pivot diameter;

– for a ring pivot

$$T_f = \frac{1}{3} F f \frac{d_{ex}^3 - d_{in}^3}{d_{ex}^2 - d_{in}^2}. \quad (5.6)$$

For a spherical pivot, the friction moment is determined from (5.5) in which $d_p/2$ is substituted by the radius of a contact area a , calculated by

$$a = 0,88 \sqrt[3]{\frac{2FR}{E}}. \quad (5.7)$$

The coefficient of friction f for a steel pair can be taken from Table 2.1.

In a ball bearing, the friction moment is calculated from

$$T_f = F f' \frac{d}{2}, \quad (5.8)$$

where d is a diameter of a bearing bore, mm; $f' = 0,003 \dots 0,006$ is the reduced coefficient of friction.

Topic 6 CALCULATION OF HANDLES

In small devices, a hand drive is usually used.

The required length of a handle is calculated taking into account a mean worker's force $Q = 200 \dots 250$ N:

$$l = \frac{T_t + T_f}{Q} + 100 \text{ mm},$$

where the moment $T_C = T_t + T_f$ is measured in Nmm. The obtained value is approximated to the nearest (greater) normal dimension according to GOST 6636-69.

Then the handle material and its allowable bending stress are chosen. Diameter of a handle is calculated on the basis of a bending strength condition like for a cantilever (Fig. 6.1):

$$\sigma_b = \frac{M}{W_x} = \frac{32QL}{\pi d_h^3} \leq [\sigma_b], \quad d_h = \sqrt[3]{\frac{QL}{0,1[\sigma_b]}}$$

Allowable stress $[\sigma_b]$ depends on the handle production material (e.g., steel CT. 5, 20, 45) and taken from appendix 1.

The handle diameter should not be smaller than 10 mm by reasons of ergonomics; so increase it, if necessary, by replacing a solid rod with a pipe with larger diameter which satisfies strength conditions.

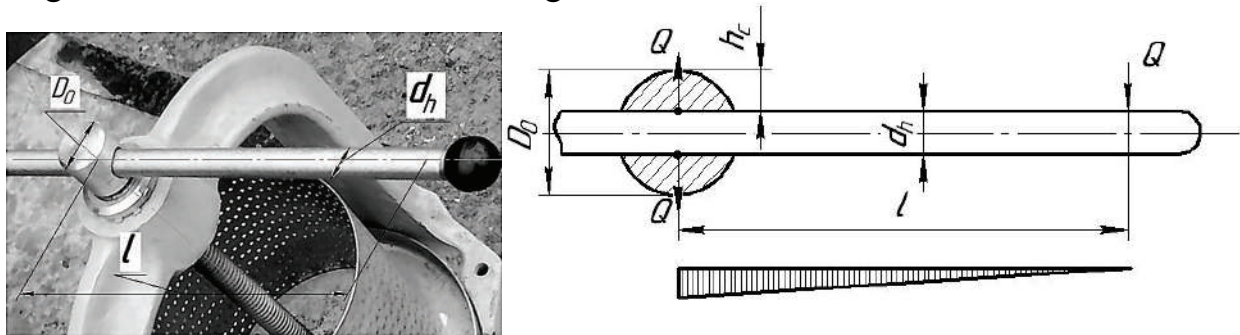


Fig. 6.1. A calculation scheme of handle

For D_0 to be large enough, a wall $h_c = (D_0 - d_h)/2$ (see Fig. 6.1) should be more than 3...4 mm thick.

Topic 7 CALCULATION OF CASE ELEMENTS

7.1. Calculation of cases of jacks

Jack cases intended to resist axial loads smaller than 5 kN are usually manufactured from light alloys (AJI-7, AJI-9, MJI-9, etc.); and those intended to

resist greater loads ($> 5\,000\text{ N}$) - from cast irons (CT 15, CT 20) or cast steels (20JI, 45I). Thus, if the form is filled normally with a fused metal, its walls δ should be not less than 6 mm thick.

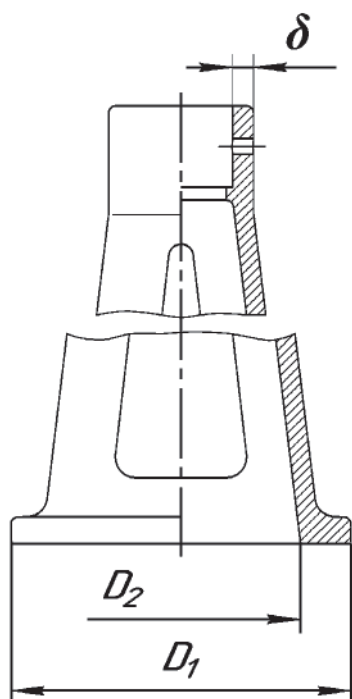


Fig. 7.1. A jack's case

The minimum external diameter of the case (together with a nut) is

$$d_{ex} = D_n + 2(\delta + 2).$$

It is preferable to make the case conic (a cone 1:10), with a square or round basic plate. To decrease the mass of a jack, notches are made in the case (Fig. 7.1).

The external sizes of a jack plate D_1 (the size D_2 could be determined by the drawing) are determined from bearing strength condition for a material on which the jack is placed. Usually it is soft (boards, cross ties) with an allowable bearing stress $[\sigma_{br}] = 1,2 \dots 2\text{ MPa}$.

For round bases,

$$\sigma_{br} = \frac{F}{A_{br}} = \frac{4F}{\pi(D_1^2 - D_2^2)} \leq [\sigma_{br}];$$

for square basis

$$\sigma_{br} = \frac{F}{A_{br}} = \frac{F}{(D_1^2 - \frac{\pi}{4}D_2^2)} \leq [\sigma_{br}].$$

Calculated values D_1 should be approximated to the nearest (greater) normal one.

7.2. Removers' cases calculation

Remover case elements include a case and grips. Grips can freely turn round axles (see Fig. 1.8, b).

Case elements of a remover with turning grips (Fig. 7.2) can be calculated along the following pattern:

1. Choose the materials for axle (usually normalized or tempered steel 45, 65Г), case (cast iron or cast steel 20JI, 40JI), and grip production (tempered carbon steel 35, 45) and a thickness a . Usually cases are made by moulding, therefore it is expedient for the form walls to be not less than 6 mm thick if the form is filled normally with a metal. For $b = 2a$, define the axle diameter d on the basis of the bending strength

$$\sigma_b = \frac{M}{W_x} = \frac{32F(a+b)}{2X\pi d^3} \leq [\sigma_b] .$$

Approximate it to the nearest (greater) standard size and check the axle on the shear

$$\tau_s = \frac{F}{A_s} = \frac{F}{iX} \frac{4}{\pi d^2} \leq [\tau_s] ,$$

where X is a number of grips; i is a number of shear cross-sections, $[\sigma_b]$, $[\tau_s]$ are allowable bending and shear stresses of an axle production material (see appendix 1).

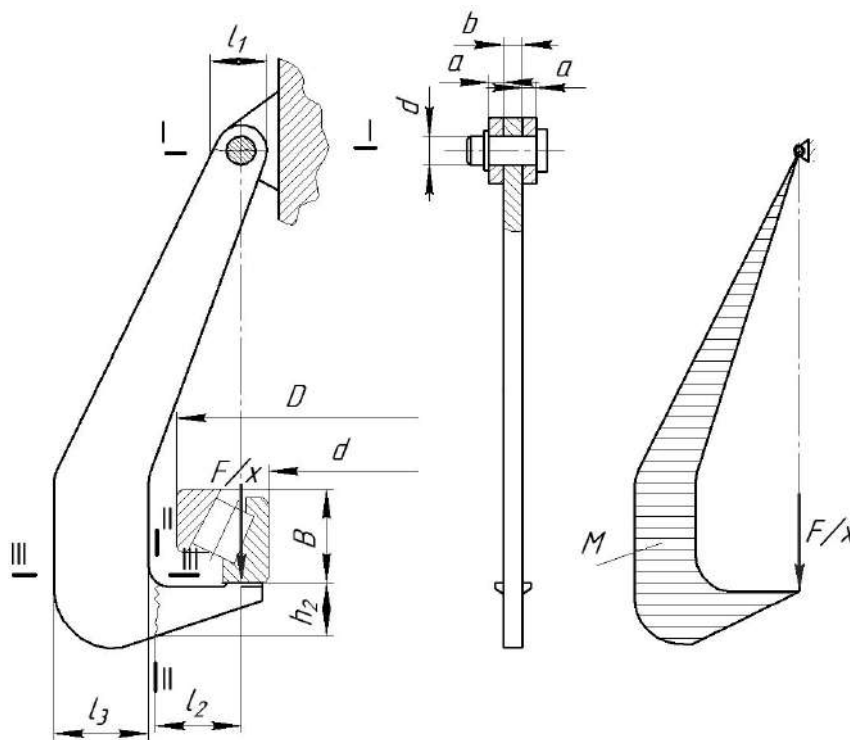


Fig. 7.2. Calculation of remover with turning grips

2. Check the size a on a bearing strength condition

$$\sigma_{br} = \frac{F}{A_{br}} = \frac{4F}{2Xad} \leq [\sigma_{br}] ,$$

where $[\sigma_{br}]$ is an allowable bearing stress of the case production material (see appendix 1).

3. Find the size l_1 of the dangerous section with hole I-I on the basis of the tension strength (here $[\sigma_t]$ is an allowable tensile stress of grip production material (see appendix 1):

$$\sigma_t = \frac{F}{A_t} = \frac{F}{X(l_1 - d)b} \leq [\sigma_t]; \quad l_1 = d + \frac{F}{Xb[\sigma_t]}$$

4. Define the size h_2 of section II–II from a bend strength condition:

$$\sigma_b = \frac{6Fl_2}{Xbh_2^2} \leq [\sigma_b]; \quad h_2 = \sqrt{\frac{6Fl_2}{Xb[\sigma_b]}}$$

where $[\sigma_b]$ – allowable bending stress of grip material (see appendix 1).

The size l_2 is defined from a drawing of a remover.

For this purpose, write out the following sizes of the removed bearing from the catalogue: internal diameter d , external diameter D , and width H or B to put them into the drawing.

5. Finally, calculate the size of the section III-III:

$$\sigma_b = \frac{F}{Xbl_3} + \frac{6F(l_2 + 0,5l_3)}{Xbl_3^2} \leq [\sigma_b];$$

$$l_3 = \frac{2F + \sqrt{4F^2 + 6Fl_2[\sigma_b]Xb}}{X[\sigma_b]b}$$

7.3. Calculation of press cases

Press cases can be made by moulding, welding or their combination. In Fig. 7.3 we see the case calculation pattern of a one-rack press of the cast type. It includes the following steps.

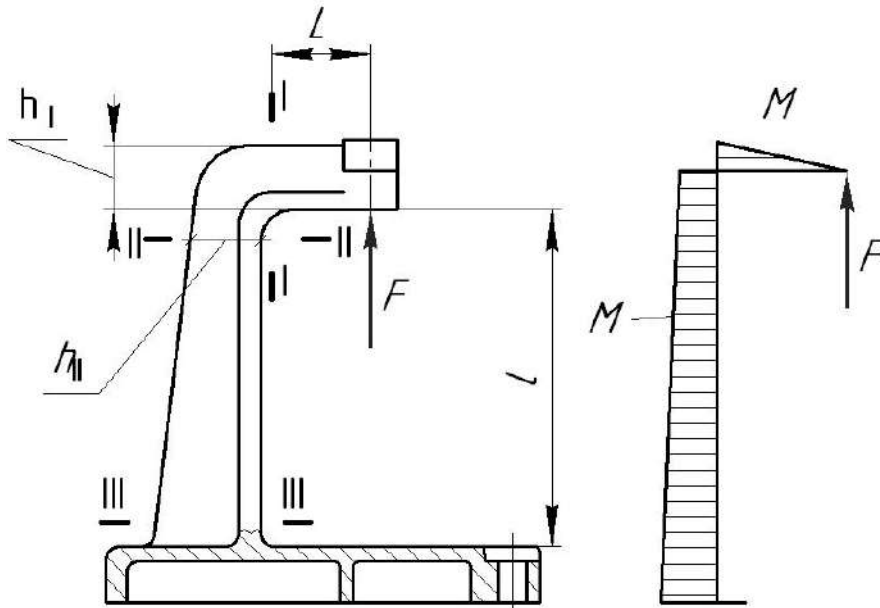


Fig. 7.3. Calculation of one-rack press case

1. Choose the case production material (cast iron or cast steel 20Л, 40Л) and set a profile of the section I-I in relative units (e.g. a T-section in Fig. 7.4 has minimal $W_x = 0,029 h^3$). Define the characteristic size $h = h_I$ on the basis on bending strength

$$\sigma_b = \frac{M}{W_x} = \frac{FH}{0,029h^3} \leq [\sigma_b] ,$$

where $[\sigma_b]$ is an allowable bending stress of the case production material (see appendix 1).

2. Accept the size $h_{II} \approx 1,1h_I$ and check section II-II on strength taking into account stress of a bend and a tension/compression:

$$\sigma_t = \frac{F(H+a)}{W_t} + \frac{F}{A} \leq [\sigma]$$

and

$$\sigma_c = \frac{F(H+a)}{W_c} - \frac{F}{A} \leq [\sigma] , \quad (7.6)$$

where σ_t , σ_c – stress of a tension and compression accordingly; W_t and W_c – the flexural section modulus for the tension and compressed fibres. In a case of T-section with parameters according to Fig. 7. 4 (and only for this section!) the flexural section modulus accordingly are $W_t = 0,067h^3$, $W_c = 0,029h^3$.

Calculation of section III-III is spent similarly to calculation of section II-II.

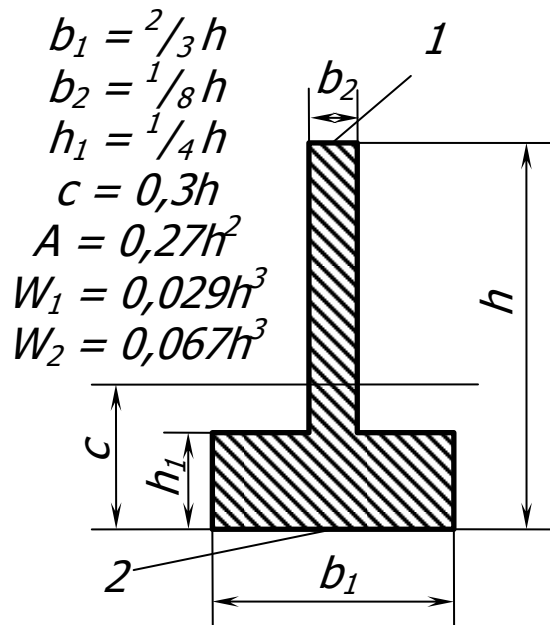


Fig. 7.4. The T-section

Topic 8 DESIGNING OF MECHANISMS WITH THE SCREW-NUT TRANSMISSION

Calculation of mechanisms should be carried out simultaneously with design drawing as many sizes necessary for calculation it is possible to define only from the drawing. At the same time step-by-step drawing of a design in the course of calculation serves as check of this calculation. The wrong result is usually shown in proportionality infringement. Some sizes of elements of mechanisms at designing are not counted, and accept according to experience of designing of similar constructions.

Various elements and fragments of designs which can be applied depend-

ing on the scheme of the mechanism and conditions of its work are shown on figures stated below. As a result of configuration of fragments of designs the mechanism design as a whole, corresponding to the given scheme is created.

On Fig. 8.1 cast and welded cases of screw jacks are drawn.

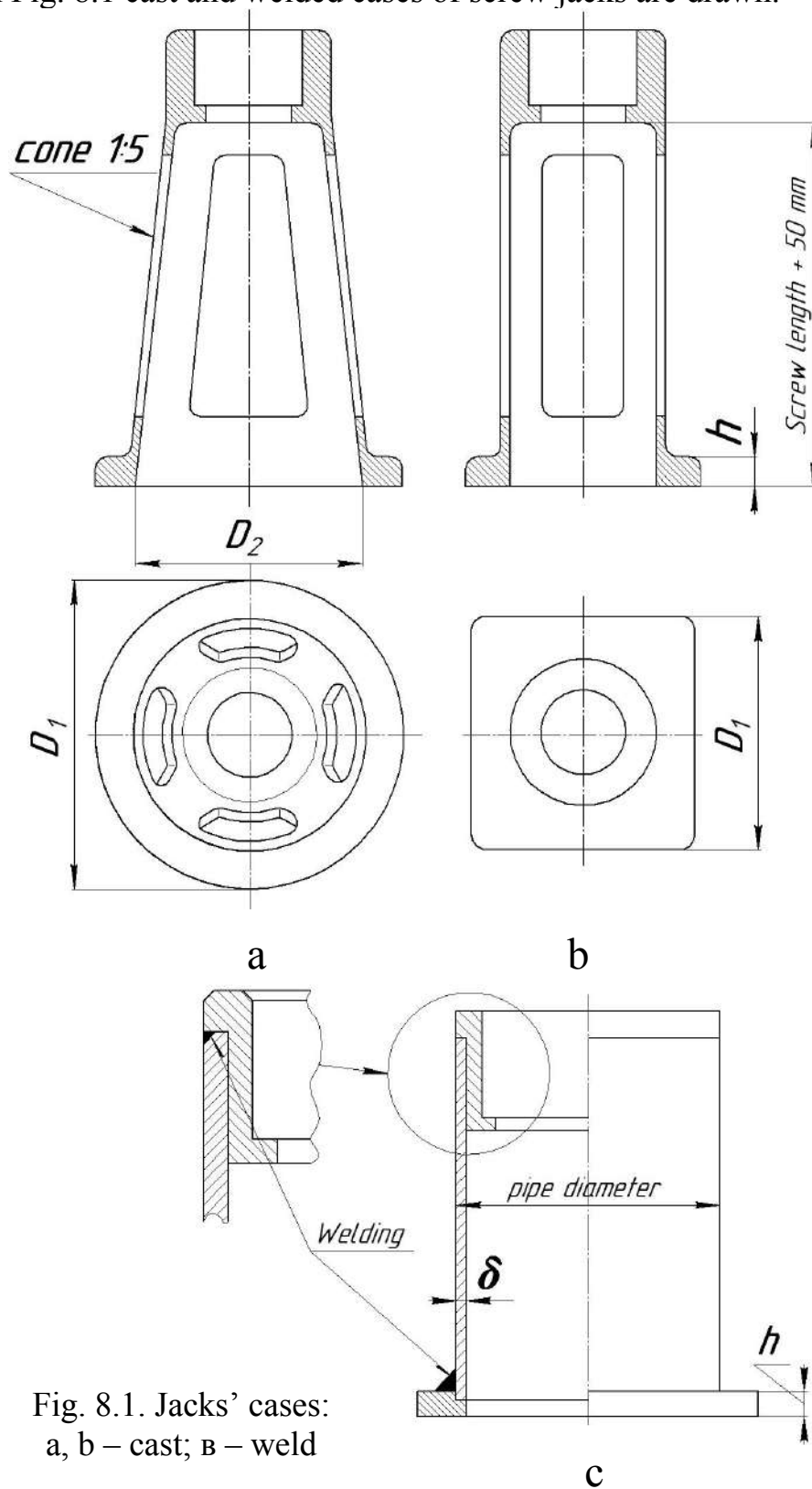


Fig. 8.1. Jacks' cases:
a, b – cast; c – weld

Cast designs of cases (see Fig. 8.1, a, b) could be manufactured from aluminum alloys (АЛ-4, АЛ-9); cast iron (СЧ15, СЧ18) and steel (20Л, 25Л). Easy alloys are used for jacks with load capacity to 10 000 N, for jacks' load capacity more than 30 000 N – cast steel, between these values any of the listed materials can be used.

The case form can be conic or cylindrical with a ring or square basic plate.

On Fig. 8.1, c the welded design is made of a steel pipe which diameter gets out any way, and a thickness of a wall δ is proved by calculation on compression. Into one end of the pipe the transitive bush is welded, and on the other – a support surface. The nut can be screwed on a pipe thread directly, without the transitive bush.

Pay attention to that both welded seams do not transfer axial load, they serve only for fixing of elements in demanded position.

On Fig. 8.2 support units of power screws (a cup, a crown, a pillow, boots) different types (a – a flat pivot, b – a spherical pivot, c – the non-standard bearing, d, f, g – standard) are represented.

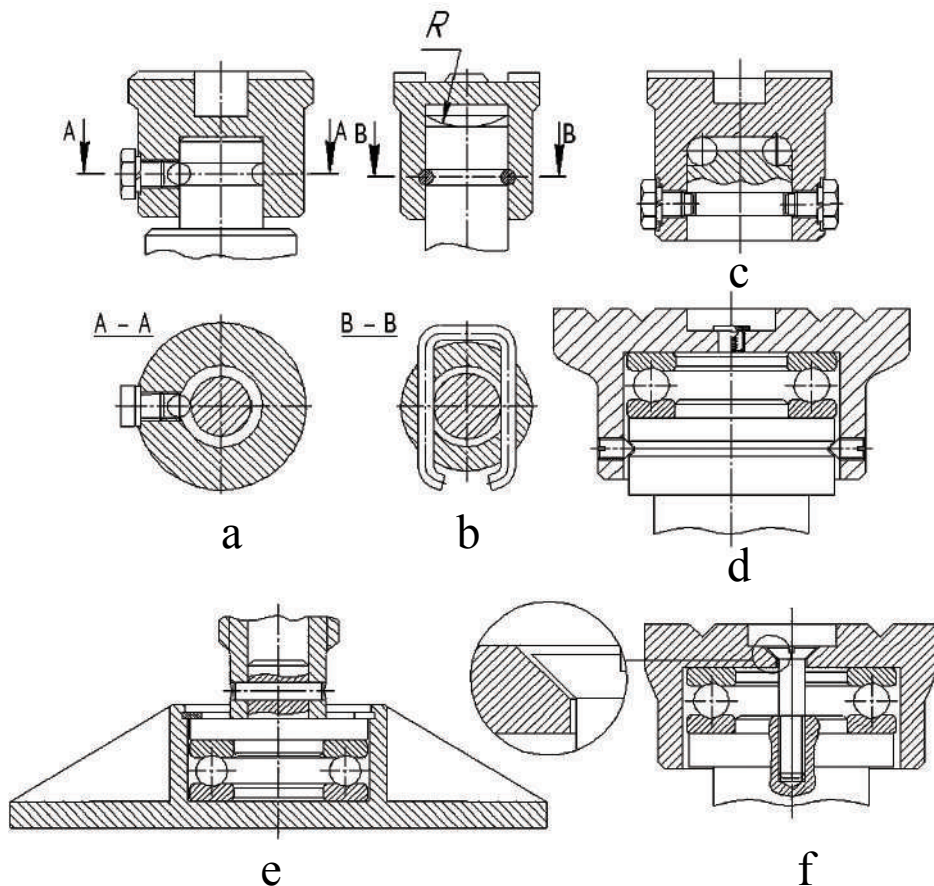


Fig. 8.2. Devices for supporting screws:

- a – a ball fixing flat pivot; b – a spherical staple fixing pivot; c – the non-standard bearing, fixing by adjusting screws with a cylindrical end; d – the standard bearing, fixing by adjusting screws with a conic end; e – the standard bearing, fixing by a lock cutting ring; f – the standard bearing, fixing by the screw

In all designs locks – the devices to avoid dismantling of unit – are provided. There are many types of locks, such as: a bent soft wire staples (Fig. 8.2, b); screws with cylindrical (Fig. 8.2, c) or conic (Fig. 8.2, d) end; lock cutting rings (Fig. 8.2, e); the usual screws twisted in an end face of the lead screw (Fig. 8.2, f). In a design represented on Fig. 8.2, a, the ball serves as the lock. For lubrication of rubbing surfaces nipples and seals are sometimes provided in a design.

Screw moving limiters are shown on Fig. 8.3.

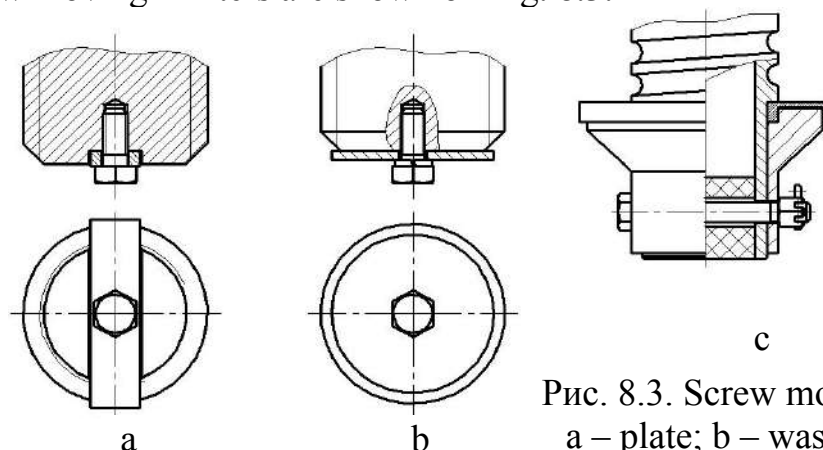


Рис. 8.3. Screw moving limiters:
a – plate; b – washer; c – stop

They are necessary to prevent full the screw unscrewing. In drawings variants with a plate (Fig. 8.3, a), a washer (Fig. 8.3, b) and a fixed stop (Fig. 8.3, c) are presented. Last design often uses in power screw of airplane control mechanisms.

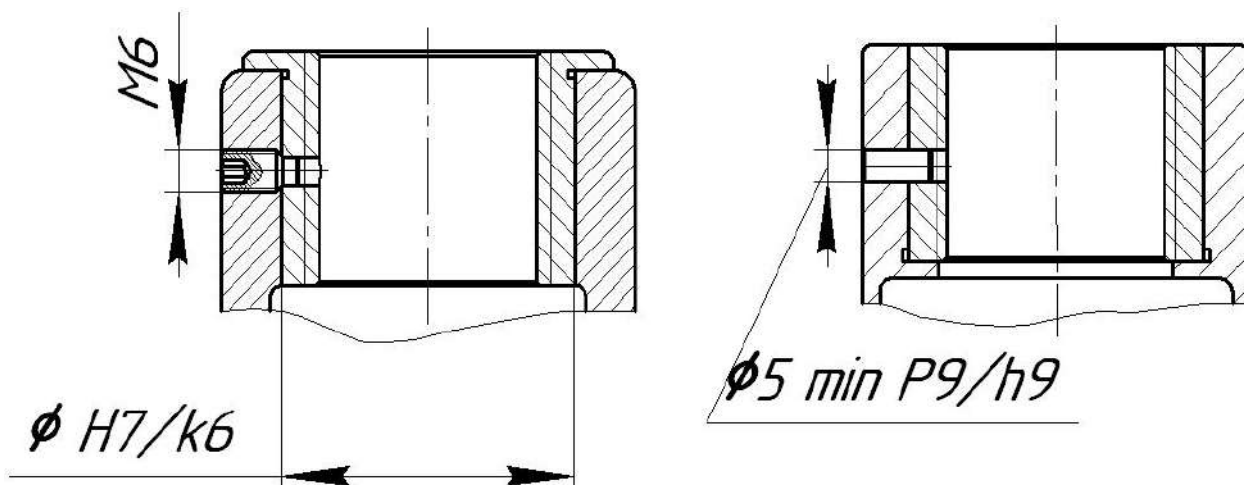


Fig. 8.4. Nut fixing in a case:
a – by adjusting screw; б – by pin

Nuts are usually demountable bushes pressed or screwed in the case (see Fig. 2.6). To prevent nut's turning and to use easier fit various fixing devices like screws (Fig. 8.4, a), pins (Fig. 8.4, b) etc. are provided.

On Fig. 8.5 variants of connection of handle and knob are shown.

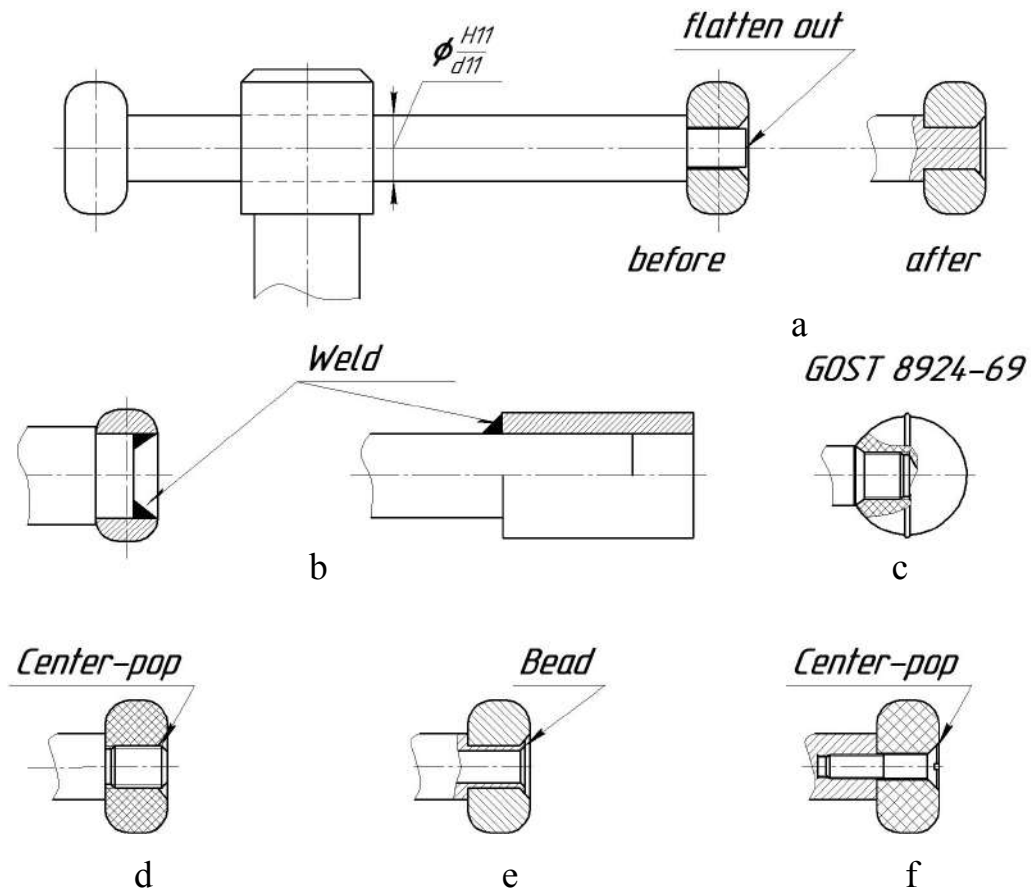


Fig. 8.5. Handle and knob connections:
 a – by flatten; b – by weld; c, d – by thread; e – by bead; f – by screw

In a variant "a" a handle end (made, as a rule, from a soft material) is flattened out by hammer impacts on an end face. Being deformed, handle metal fills a gap between a knob and the handle and a conic space. As a result the knob is fixed in an axial direction, and from a turn too.

In cases "b" knobs are welded either in hollow or outside.

The variant "c" shows the standard handle and the plastic spherical knob according to GOST 8924–69. In this case knob screwed on a handle end. In a variant «d» the non-standard handle and knob are connected by means of thread too.

On Fig. 8.5, e the way of connection in that case when the handle is a pipe is shown. In this case a thin pipe end is beaded.

Probably also fastening by means of the usual screw with a countersunk head (Fig. 8.5, f), and many other variants.

For power screws of the big load capacity the handle often extends by means of a pipe selected in a place. If such possibility is provided, the handle should be pressed at the one end and without the knob.

On Fig. 8.6 versions of removers are represented: with flat rotary grips (Fig. 8.6, a) and short cantilever grips with cylindrical or square (Fig. 8.6, b) sections. In the first design standard axles in accordance with GOST 9650–80 serve as axes of grips' rotation, the nut is fixed in the case by a cylindrical pin

(GOST 3128–70). In the second variant (see Fig. 8.6, b) grips move in a radial direction and tightening screws fix them in demanded position.

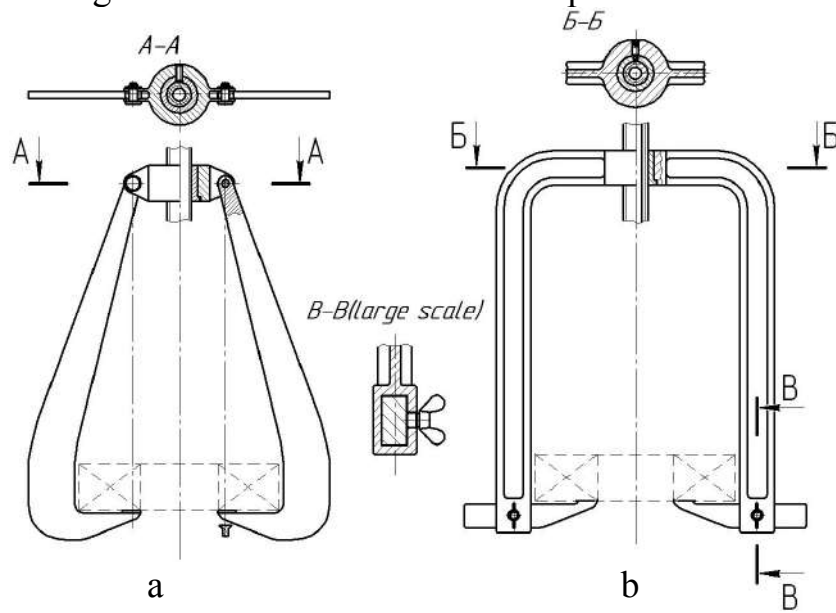


Fig. 8.6. Cases of bearing removers:
 a – with a rotary grips; b – with movable grips

On Fig. 8.7, a cast case design of one-rack press is presented.

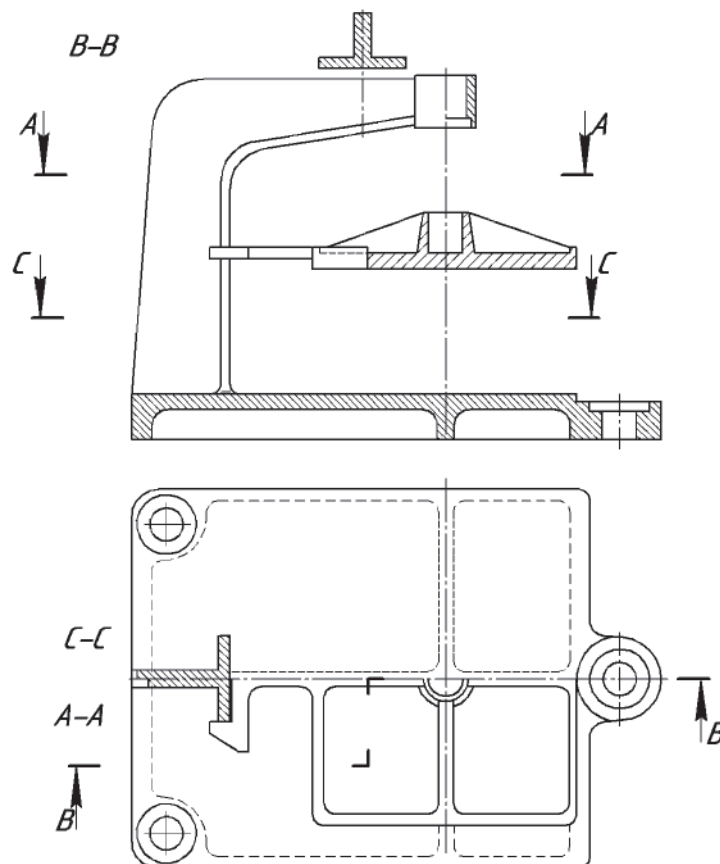


Fig. 8.7. Cast case of one-rack press

In this variant the movable plate is supplied by the ledges sliding on directing rack. The rack has T-section with unequal height in upper and lower sections. The basis is a continuous plate with rigidity edges on the bottom. In some cases in a plate do hole for shaft from which the bearing is removed (see Fig. 1.7).

THE ORDER OF DESIGN

1. **Definition of initial data.** Basic data at designing of power screw mechanisms are the axial force F acting on the screw, and a screw length. The technical project can define and other parameters. The sizes of the removed bearing necessary for calculation are specified in catalogues.

If screw and nut materials are not given student must choose them. Materials for a sliding power screws see in appendix 1 and Table 4.1.

2. Screw calculation:

- draw a force and torque distributions along a screw axis (see Fig. 3.1) and find dangerous section;

- choose, if it is not given, thread type;

- define diameter of the screw on a wear resistance condition (see formula (3.1));

- choose the nearest standard thread which has diameter d_2 more than calculated one. The sizes of the buttress and trapezoidal threads see appendix 2;

- write out code of the chosen thread;

- check the self-locking condition $\psi + l < \rho'$ and efficiency of the screw-nut transmission (see expression (2.1)). Efficiency could not exceed 50 % and it is undesirable to have it less than 20 %. If these conditions are not satisfied choose another thread dimensions;

- check the screw on strength in dangerous section under the formula (3.2). To calculate the torsion moment in dangerous section (see Fig. 3.1) it is necessary to know T_t (the torque of screw pair, see formula (2.2)) and T_f (the friction moment in a pivot, when type of a pivot is accepted);

- calculate diameter of a sliding pivot (flat or ring) under formulas (5.1) and (5.2), and then the friction moment under formulas (5.5) and (5.6) (allowable bearing pressure are presented in Table 3.1);

- or calculate a sphere radius of a spherical pivot from the contact strength condition on the formula (5.3) (allowable contact stress $[\sigma]_H$ depends on hardness of contacted surfaces which should be not less HRC 45), then find the friction moment in a pivot under formulas (5.5);

- or select the standard bearing for the pivot with a rolling bearing according to static load capacity is found from the formula (5.4) and determinate a friction moment under the formula (5.8).

It is possible to use section 5 recommendations to design pivot.

3. Nut calculation and designing:

- choose a nut design (see Fig. 4.2);
 - define threads number and nut height (see the formula (4.1));
 - check nut threads (see the formula (4.2)) in dangerous section;
 - define external diameter of a nut body (see the formula (4.4)) and, at necessity, its thrust fillet size Δ and height (see subsection 4 and Fig. 4.4);
 - choose a type of mechanical fastening of a nut to the case (see Fig. 4.5) and find diameter and quantity of fix elements (see formulae 4.7).
- draw the full-scale sketch of the screw-nut pair and coordinate it with the instructor.

It is possible to use section 5 recommendations to design of screw, pivot, nut and fastening elements.

4. Calculation and handle designing:

- define length of the handle;
 - choose a handle material (see appendix 1);
 - define diameter of the handle from a bend strength condition in dangerous section (see subsection 6), then round it to normal, corresponding to the standard size of rod or pipe.
- design the handle and knob under section 5 recommendations and draw them to the sketch.

5. Definition of efficiency of the mechanism:

- define efficiency under the formula

$$\eta = \frac{A_U}{A_S} = \frac{FPi}{2\pi(T_t + T_f)},$$

where A_U is a useful work for one turn; A_S – the spent work.

The mechanism efficiency should differ from the efficiency of a power screw defined under the formula (2.1) a little. At the big difference of these efficiencies it is necessary to change a design of a pivot for reduction of losses by a friction.

6. Case designing:

- choose a design of the case of the mechanism under section 5 recommendations, by means of albums of designs, etc., then check it on strength according to the subsection 7 recommendations;
- calculate and draw case elements on the same sketch, coordinating their sizes with the sizes of other elements. The sketch would be coordinated with the instructor.

7. Documentation:

- draw the whole mechanism on the basis of the coordinated sketch. To make the design clear additional views, cuts and projections are provided in the drawing. Coordinate number of additional views with the instructor.

Such dimensions should be put down on the drawing:

a) overall mechanism dimensions and distance between extreme positions of moving parts (for example, a screw operation movement);

b) the connecting sizes of elements with which the mechanism contacts to surrounding elements: diameters and center distances of holes for mechanism fastening, dimensions of a pivot, of a support part, etc.;

c) the sizes on which there is a connection of elements at assembly operations: fitting of bearing to shaft and in the case, nuts in the case, fastening elements in a nut and the case, pins, axles, etc.; threads.

– put technical requirements and a product technical characteristic in the drawing over the basic inscription. In technical requirements specify:

a) lubricating elements' numbers, a way and type of lubricant;

b) data on product colouring.

– in a technical characteristic put down the given data and main parameters of the designed mechanism: efficiency, force on the handle (for products with a mechanical drive – required capacity), etc.

The text part consists of an explanatory note and the specification.

Mechanical properties and allowable stresses of some materials

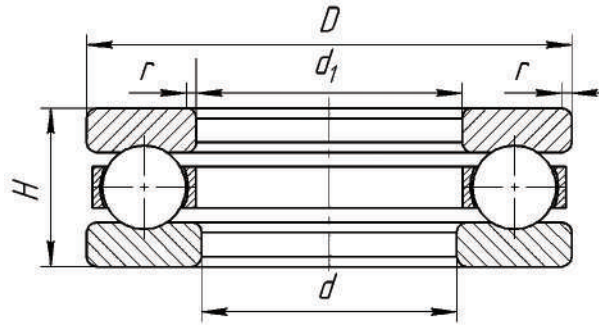
Steel	Heat treatment	σ_y	Allowable stresses, MPa				
			tensile-compression [σ_{tc}]	bending [σ_b]	bearing [σ_{br}]	shear [τ_s]	torsion [τ_t]
Ст. 5	N	275	165	200	250	100	125
Ст. 6	N	315	195	230	290	115	145
20Л	N	220	90	110	135	50	63
45Л	N	320	125	150	190	70	90
35 ХГЦЛ	N	340	140	170	210	75	100
20	N	250	140	170	210	85	105
35	N	320	180	210	270	110	135
	T	380	210	260	520	130	160
	C	650	330	400	500	200	250
45	T	450	240	290	360	145	185
	C	650	300	360	450	185	230
40X	T	650	270	320	400	160	200
	C	1100	440	530	670	270	330
40XH	N	460	260	310	390	155	190
	C	1000	410	490	620	250	310
65Г	T	700	300	360	450	185	230
	C	1250	500	600	760	300	380
30 ХГСА	T	850	370	440	550	220	280
	C	1300	510	620	760	310	390
ШХ15	C	1700	740	890	1100	440	550
Cast iron	σ_u		tensile [σ_t]	bending [σ_b]	compression [σ_c]	torsion [τ_t]	
СЧ 15	150		33	60	145	40	
СЧ 20	200		45	73	165	45	

N – normalizing; T – tempered; C – casehardening

The dimensions of standard threads

buttress				trapezoidal			
Diameters, mm, in order to increasing d_2			Pitch P , mm	Diameters, mm, in order to increasing d_2			Pitch P , mm
d_2	d_1	d		d_2	d_1	d	
8,50	7	10	2	12,5	11,0	14	3
10,50	9	12	2	13,0	12,0	14	2
12,50	11	14	2	14,0	12,0	16	4
14,50	13	16	2	15,0	14,0	16	2
16,00	10	22	8	16,0	14,0	18	4
16,50	15	18	2	17,0	16,0	18	2
18,00	12	24	8	18,0	14,0	22	8
18,25	14,5	22	5	18,0	16,0	20	4
18,50	17	20	2	19,0	18,0	20	2
20,00	14	26	8	19,5	17,0	22	5
20,25	16,5	24	5	20,0	16,0	24	8
22,00	16	28	8	20,5	19,0	22	3
22,25	18,5	26	5	21,0	20,0	22	2
22,50	15	30	10	21,5	19,0	24	5
24,25	20,5	28	5	22,0	18,0	26	8
24,50	17	32	10	22,5	21,0	24	3
25,50	21	30	6	23,0	22,0	24	2
27,50	23	32	6	23,5	21,0	26	5
27,75	25,5	30	3	24,0	20,0	28	8
28,50	21	36	10	24,5	23,0	26	3
29,75	27,5	32	3	25,0	20,0	30	10
31,50	27	36	6	25,0	24,0	26	2
32,50	25	40	10	25,5	23,0	28	5
35,50	31	40	6	26,5	25,0	28	3
				27,0	22,0	32	10
				27,0	24,0	30	6
				27,0	26,0	28	2
				28,5	27,0	30	3
				29,0	24,0	34	10
				29,0	26,0	32	6
				30,5	29,0	32	3
				31,0	26,0	36	10
				31,0	28,0	34	6
				32,5	31,0	34	3
				33,0	28,0	38	10
				33,0	30,0	36	6
				34,5	31,0	38	7
				34,5	33,0	36	3
				35,0	30,0	40	10
				35,0	32,0	38	6

Basic load capacity and ball thrust bearing dimensions



Bearing No	Dimensions, mm					Load capacity, kN	
						dynamic	static
	d	D	H	r	d ₁	C	C ₀
8100	10	24	9	0,5		10,0	11,8
8101	12	26	9	0,5		10,4	12,9
8102	15	28	9	0,5		10,6	14,0
8103	17	30	9	0,5		11,4	16,6
8104	20	35	10	0,5	35	15,0	22,4
8105	25	42	11	1,0	42	18,0	30,0
8106	30	47	11	1,0	47	19,0	33,5
8107	35	52	12	1,0	52	20,0	39,0
8108	40	60	13	1,0	60	27,0	53,0
8109	45	65	14	1,0	65	28,0	58,5
8201	12	28	11	1,0	14	13,2	16,0
8202	15	32	12	1,0	17	16,6	20,8
8204	20	40	14	1,0	40	22,4	32,0
8205	25	47	15	1,0	47	28,0	42,5
8206	30	52	16	1,0	52	25,5	40,0
8207	35	62	18	1,5	62	35,5	57,0
8208	40	68	19	1,5	68	46,5	83,0
8209	45	73	20	1,5	73	39,0	67,0
8305	25	52	18	1,5	52	34,5	46,5
8306	30	60	21	1,5	60	38,0	55,0
8307	35	68	24	1,5	68	50,0	75,0
8308	40	78	26	1,5	78	61,0	95,0
8309	45	85	28	1,5	85	75,0	118,0
8405	25	60	24	1,5	60	45,5	57,0
8406	30	70	28	1,5	70	69,5	95,0
8407	35	80	32	2,0	80	76,5	106,0
8408	40	90	36	2,0	90	96,5	143,0
8409	45	100	39	2,0	100	122,0	186,0

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(Англійською мовою)

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