MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE

National aerospace university «Kharkov aviation institute»

Yu. Koveza, I. Lykhosherst, S. Svitlychniy

DESIGN OF MACHINE ELEMENTS. COURSE PROJECT

Tutorial

Kharkov “KhAI” 2014

UDK 621.81.001.66(075.8)
ББК 34.44я73
K79

Посібник містить методики розрахунку основних видів механічних передач, їхніх складових елементів, а також взаємозв'язані з ними методики і рекомендації щодо розроблення конструкції. Методики розрахунку і конструювання проілюстровано покроковими інструкціями, прикладами розрахунку та конструювання. Подано зразки складальних і робочих креслень. Наведено також необхідні для розрахунку і конструювання стандарти, інформацію щодо машинобудівних матеріалів, їх термооброблення, допусків і посадок, шорсткості поверхонь, допусків форми і розташування поверхонь.

Для студентів машинобудівних вузів при виконанні курсових і дипломної робіт.

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

Koveza, Yu.

K79 Design of machine elements. Course project [Text] : tutorial / Yu. Koveza, I. Lykhosherst, S. Svitlychniy. – Kh. : National aerospace university «Kharkov aviation institute», 2014. – 180 p.

ISBN 978-966-662-344-0

The tutorial contains a methodology for calculating the main types of mechanical transfers, their components and related with them techniques and recommendations for the development of construction. Analysis and designing are illustrated step-by-step instructions, examples of calculation and design. Samples of assembly and working drawings are presented. Information on standards, engineering materials and their heat treatment, tolerances and fits, surface roughness are also given for the analysis and design.

For students of technical higher education establishments for course projects and Bachelor works.

Fig. 73. Tab. 86. References: 10 titles

UDK 621.81.001.66(075.8)
ББК 34.44я73

ISBN 978-966-662-344-0

INTRODUCTION

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

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:

− qualitative, i.e. defining components and assemblies shape, the arrangement and interaction;

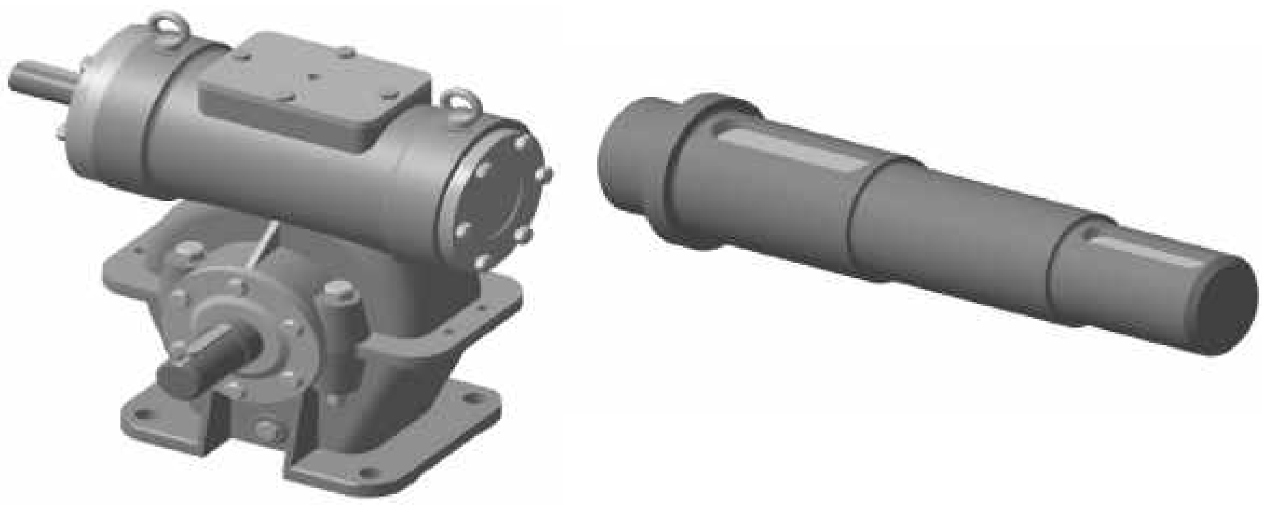
− quantitative, i.e. determining elements’ size and number. Means of quantitative design:

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

We will understand **machine** as a technical system that 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 (an assembly unit) is a set of interacting components, connected together for a common functional purpose (gearbox, coupling, bearing, Fig. I.1, a).

Element (a part, a component) is a product, which is manufactured without using assembly operations (for example, a shaft, a bolt, a nut, Fig. I.1, b).



a b

Fig. I.1

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 (Fig. I.2).

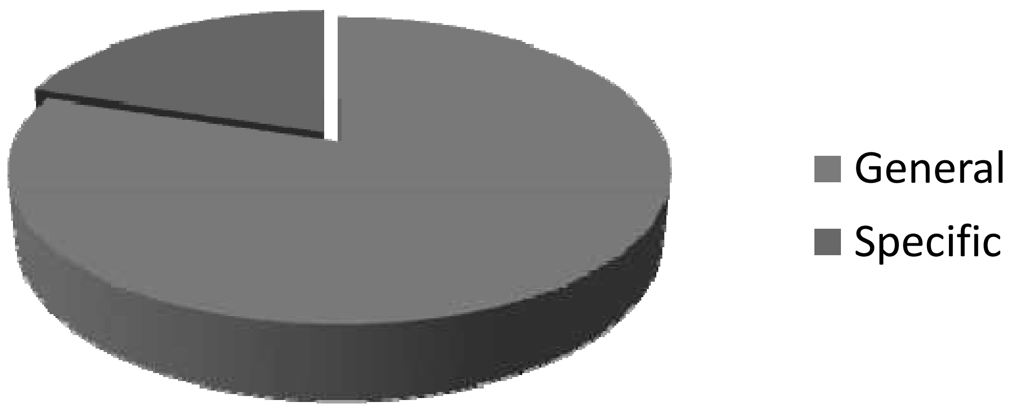


Fig. I.2

A student must be able to:

− transit mentally from a diagram/drawing to the structure and vise versa;

− size machine components and assemblies;

− select materials and standard elements using reference books;

− generate calculations and design documentation in accordance with applicable regulations;

− design parts and assemblies using the basic principles of efficient design.

During the course studying a number of difficulties occur. They are:

1. A lot of empirical equations and correction 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.

To acquire these skills and to overcome the difficulties of the course the first student's independent design work − course project − is intended.

The object of the course project is the designing of mechanical drive, consisting of the engine, gearbox and couplings.

Calculating portion includes designing and verification of gears, shafts and bearings. The results of calculations and standard elements selection are expounded in the explanatory note (25 – 30 pages).

The graphical part of the project consists of a gearbox drawing in natural scale with part list (1-2 pages A1), a drive drawing with part list (1 sheet A1) and working drawings of 2-3 elements (1 sheet A1).

1. KINEMATIC CALCULATION OF A DRIVE

The objectives of kinematical calculation is definition of the power acting to all units of a kinematic chain, and also preliminary distribution of rotational speed and the torques on all units. If necessary, other speeds and forces could be calculated.

As a source of mechanical energy in a drive is the engine, first it is necessary to select the most suitable to the task. The engine type is coordinated with the teacher, and its required characteristics – power (or the torque) and rotational speed – are calculated and selected in kinematical calculation.

1.1. Output and Input Power

Calculation of the engine power begins from the end of a kinematic chain, i.e. definition of a prime drive power is required.

Initial data for kinematical calculation are stipulated in a technical project. They are:

– at lineal prime drive – force F and speed V. Then $P_{out} = F\,V$;

– at rotary – torque T and angular speed ω: $P_{out} = T\,\omega$. In the mechanical engineering rotational speed n (RPM) is frequently used instead of angular speed. Then if to substitute T in Nm and n in RPM, power in kW equals

$$P_{out} = \frac{T_{out} n_{out}}{9550}. \qquad (1)$$

Required input power of a drive engine

$$P_{in} = P_{out} / \eta_o.$$

Then preliminary losses are estimated by means of each kinematic chain element efficiency known from the previous experience (Tab. 1.1). The total efficiency of a drive is a product of all elements efficiencies

$$\eta_o = \eta_I \eta_{II} \eta_C \eta_{RB} \dots .$$

Table 1.1

Kinematic chain element		**Efficiency η**
Spur gear		0,96…0,98
Bevel gear		0,94…0,97
Worm gear	$z_1 = 1$	0,68…0,72
	$z_1 = 2$	0,73…0,78
	$z_1 = 4$	0,78…0,84
Planetary gear		0,96…0,98
Harmonic drive		0,80…0,94
A pair of roller bearings – η_{RB}		0,990…0,995
Coupling – η_C		0,98…0,99

If the standard engine is required, it is selected from corresponding catalogues with engine power P_{en} more than P_{in}. Parameters of the most widespread and simple asynchronous engines are provided in Appendix A.

1.2. Engine Shaft Rotational Speed

Shaft rotational speed is also important engine characteristic, and the drive configuration, its sizes and cost in many respects depend on it.

Prime drives' rotational speed is usually known. To choose an engine rotational speed (if it is not set in a technical project) it is necessary to calculate the transfer ratio of a drive:

$$i_d = n_{en} / n_{out} \tag{2}$$

for various n_{en} and whenever possible to choose, first, realized in the set scheme of a drive, and secondly, optimum from the point of view of weight, dimensions, costs, etc. The industry manufactures standard electric asynchronous engines with some levels of power with synchronous rotational speeds ***3000, 1500, 1000, 750, 600 rpm*** (Fig. 1.1).

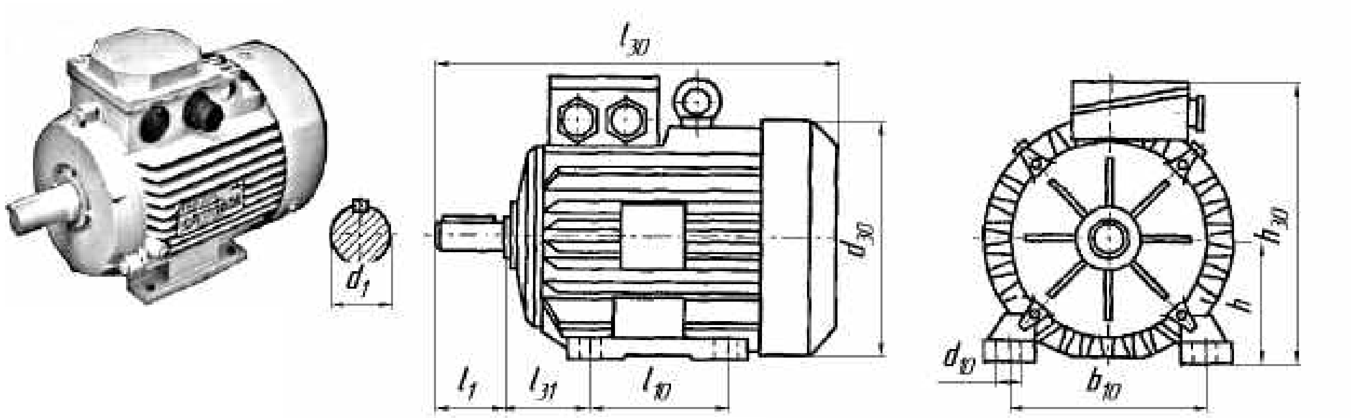


Fig. 1.1

To select proper one it is convenient to tabulate (Tab. 1.2) parameters of one level power engines with various n.

Table 1.2

P_{en}, kW	$n_{en\ async}$, rpm	Dimensions, mm	$i_d = n_{en} / n_{out}$

To obtain dimensions (see Tab. A2, Appendix A) each engine code should be found according to standard power and rotational speed (see Tab. A1, appendix A).

The engine is selected after comparison calculated i_d with the limit transfer ratio i_{lim} for the given gearbox scheme: $i_{lim} = u_{limI}\, u_{limII}\, u_{limIII} \ldots$, where $u_{lim\ i}$ – limit transfer number of i step (recommended values are shown in Tab. 1.3).

If some engines satisfy to a condition $\boldsymbol{i_d} < \boldsymbol{i_{lim}}$ one with smaller dimensions (it means that its weight is less) is usually selected.

Let us notice, that the asynchronous engine shaft rotates with nominal asynchronous RPM smaller than synchronous one. Therefore for the further calculations it is necessary to accept $\boldsymbol{n_{en}} = \boldsymbol{n_{\mathrm{en\ async}}}$ and recalculate transfer ratio (see formula (2)).

Table 1.3

Stage type		$\boldsymbol{u}$
Spur		3…6
Bevel		1…3,5
Planetary $\boldsymbol{AI}$		7…16
Planetary $\boldsymbol{A\bar{I}}$		3…9
Worm	z_1=1	28…50
	z_1=2	14…40
	z_1=4	8…30
Harmonic		80…320

1.3. Gearbox Input/Output Parameters

According to selected engine power and RPM torques (Tab. 1.4) and rotational speeds (Tab. 1.5), acting at input/output shafts, may be obtained.

Table 1.4

Parameter	**Formula**
Output torque $\boldsymbol{T_{out}}$, N m	Is given or $\boldsymbol{T_{out} = 9550\frac{P_{out}}{n_{out}}}$
Input torque $\boldsymbol{T_{in}}$, N m	$\boldsymbol{T_{in} = \frac{T_{out}}{i_d \eta_o}}$

Table 1.5

Shaft	**Formula**
Input	$\boldsymbol{n_{in} = n_{en}}$
Output	$\boldsymbol{n_{out} = n_{en} / i_d}$ or is given

1.4. Examples

Example 1

Task: calculate gearbox kinematical parameters with given asynchronous engine 4A132S6.

Initial data

Parameter	Value
Engine power P, kW	5,5
Engine synchronous rotational speed $\boldsymbol{n_{en}}$, rpm	1 000
Engine asynchronous rotational speed $\boldsymbol{n}_{\mathbf{en\ async}}$, rpm	965
Gearbox transfer ratio $\boldsymbol{i_d}$	3,8
Service life, h	20 000
Gearbox type	one-stage gearbox with spur straight teeth

Engine dimensions (see Fig. 1.1)

Dimensions		Signification	Value, mm
Overall	Height	$\boldsymbol{h_{30}}$	350
	Width	$\boldsymbol{d_{30}}$	302
	Length	$\boldsymbol{l_{30}}$	480
Conjunctive to gear box	Axis height	$\boldsymbol{h}$	132
	Shaft diameter	$\boldsymbol{d_1}$	38
	Shaft end length	$\boldsymbol{l_1}$	80
Conjunctive to base plate	Distance to first hole	$\boldsymbol{l_{31}}$	89
	Hole diameter	$\boldsymbol{d_{10}}$	12
	Distances between holes	$\boldsymbol{b_{10}}$	216
		$\boldsymbol{l_{10}}$	140

Input and output torques

Parameter		Formula and value
Input torque $\boldsymbol{T_{in}}$, N m		$T_{in} = 9550\frac{P_{en}}{n_{en}} = 9550\frac{5,5}{965} = 54,4$
Efficiency	spur stage	$0,97$
	coupling	$0,99$
	bearings	$0,98$
	general	$\eta_o = \eta_I \eta_C \eta_{RB} = 0,94$
Output torque $\boldsymbol{T_{out}}$, N m		$T_{out} = T_{in} i_d \eta_o = 54,4 \cdot 3,8 \cdot 0,94 = 194,4$

Rotational speeds of gearbox shafts

Shaft	Formula	Value
Input n_{in}, rpm	$n_{in} = n_{en}$	***965***
Output n_{out}, rpm	$n_{out} = n_{en} / i_d = 965 / 3{,}8$	***254***

Example 2

Task: select asynchronous engine for a gearbox with data as follows:

Initial data

Parameter	Value
Output torque T_{out}, Nm	220
Output n_{out}, rpm	300
Gearbox type	one-stage gearbox with bevel gears

Output and input power

Parameter		Formula and value
Output power, kW		$P_{out} = \frac{T_{out} n_{out}}{9550} = \frac{220 \cdot 300}{9550} = 6{,}91$
Efficiency	Bevel gearing	***0,96***
	Input coupling	***0,99***
	Output coupling	***0,99***
	Total	$\eta_o = \eta_B \eta_{CI} \eta_{CO} = 0{,}94$
Input power, kW		$P_{in} = P_{out} / \eta_o = 6{,}91 / 0{,}94 = 7{,}35$

Nearest bigger standard value of an engine power is 7,5 kW.

Engine shaft rotational speed

To compare some engines with the same standard power more than calculated let us use appendix A.

According to standard engine power (7,5 kW) in Tab. A1 we find engine code and asynchronous RPM. Then, according to the code, we take overall dimensions d_{30}, l_{30} and h_{30} from Tab. A2 and put down these parameters into Tab. 1.6.

Table A1

Power, kW	Synchronous rotational speed, rpm							
	3000		1500		1000		750	
	Code	n_{nom}	Code	n_{nom}	Code	n_{nom}	Code	n_{nom}
0,75	4A71A2	2840	4A71B4	1390	4A80A6	915	4A80B8	700
1,1	4A71B2	2810	4A80A4	1420	4A80B6	920	4A90L8	700
1,5	4A80A2	2850	4A80B4	1415	4A90L6	935	4A100S8	700
2,2	4A80B2	2850	4A90L4	1425	4A100S6	950	4A100L8	700
3	4A90L2	2840	4A100S4	1435	4A100L6	955	4A112M8	700
4	4A100S2	2880	4A100L4	1430	4A112M6	950	4A132S8	720
5,5	4A100L2	2880	4A112M4	1445	4A132S6	965	4A132M8	720
7,5	4A112M2	2900	4A132S4	1455	4A132M6	870	4A160S8	730

Table A2

Type	h	d_{30}	l_1	l_{30}	d_1	l_{10}	l_{31}	d_{10}	b_{10}	h_{30}
100L	100	235	60	392	28	140	63	12	160	263
112M	112	260	80	452	32	140	70	12	190	310
132S	132	302	80	480	38	140	89	12	216	350

Table 1.6

P, kW	Code	$n_{en\ async}$, rpm	Dimensions, mm	$i_d = n_{en} / n_{out}$
7,5	4AM112M	2900	260x452x310	9,67
7,5	4AM132S	1455	302x480x350	4,85
7,5	4AM132M	870	302x530x350	2,9
7,5	4AM160S	730	358x624x430	2,43

Bevel gearbox transfer ratio should be from 1 to 3,5 (see Tab. 1.3). Two engines satisfy this condition: with $n_{en\ async}$ = 870 and 730 rpm. First of them has smaller dimentions, that is why it is preferred. So, transfer ratio is $i = 2,9$ and input torque $T_{in} = \frac{T_{out}}{i_d \eta_o} = \frac{220}{2,9 \cdot 0,94} = 80,7$ Nm.

Other engine dimensions are shown in Tab. 1.7.

Table 1.7

Type	h	d_{30}	l_1	l_{30}	d_1	l_{10}	l_{31}	d_{10}	b_{10}	h_{30}
4AM132M	132	302	80	530	38	178	89	12	216	350

2. CALCULATION OF TOOTH GEARINGS

Tooth gearings are widespread in mechanical engineering thanking to small power losses, high bearing ability, reliability and durability.

Teeth's load character is difficult enough. Useful forces and additional external and internal ones load teeth. Warps and deformations of elements of transfers, discrepancy of manufacturing lead to concentration of loadings on separate pointes of contact lines.

In existing design procedures of tooth gearings, in particular, in GOST 21354 «Transfers gear cylindrical involute external toothing. Calculation on durability», the various additional factors acting on working power of tooth gearings, are considered by means of using of corresponding empirical factors.

The executed calculation corresponds to the above-stated standard with some simplifications, which are not influencing essentially on results.

2.1. The Basic Terms and Definitions

Designations of the basic geometrical parameters of cylindrical wheels of external and internal gearings are shown in Fig. 2.1. Symbols of these parameters and the formulae for their calculation see also in Tab. 2.1.

Table 2.1

Parameter	Symbol	Formula
Profile angle of basic rack	α	$\alpha = \alpha_n = 20^\circ; 25^\circ; 28^\circ$
Profile angle of basic rack in transverse cross-section	α_t	$tg\ \alpha_t = tg\ \alpha / \cos\beta$
Pressure angle in transverse cross-section	α_{tw}	$inw\alpha_{tw} = \dfrac{2X_\Sigma tg\,\alpha_t}{z_1 + z_2} + inv\,\alpha_t$
Pressure angle in normal cross-section	α_{nw}	$tg\ \alpha_{nw} = tg\ \alpha_{tw}\ \cos\beta$
Pitch helix angle	β	$\beta \approx 9\ldots20^\circ$ (for herringbone gears $\beta \le 45^\circ$)
Module in normal cross-section	m_n (m)	Standardized
Transverse module (circular module in the plane of rotation of the gear)	m_t	$m_t = m / \cos\beta$
Pitch center distance	a	$a = 0{,}5m_t(z_2 \pm z_1)$
Center distance	a_w	$a_w = a \cos\alpha_t / \cos\alpha_{tw}$
Shift factor	x_Σ	$x_\Sigma = x_1 + x_2$. If $x_\Sigma = 0$ $\alpha_{tw} = \alpha_t$, $a_w = a$, $d_w = d$
Pitch diameter	d	$d = m_t z$
Operating pitch diameter	d_w	$d_w = d\cos\alpha_t / \cos\alpha_{tw}$

The end of Tab. 2.1

Parameter	Symbol	Formula
Base diameter	$d_в$	$d_в = d \cos \alpha_t$
Equivalent teeth number	z_V	$z_V = z / \cos^3 \beta$
Transverse contact ratio	ε_α	$\varepsilon_\alpha = \left[1{,}88 - 3{,}2\left(\frac{1}{z_1} + \frac{1}{z_2}\right)\right] \cos \beta$
Axial contact ratio	ε_β	$\varepsilon_\beta = \frac{b_w \sin \beta}{\pi m_n}$

The basic geometrical parameters of bevel gearings are shown on Fig. 2.2, their symbols and formulae in case of straight teeth (the axial form of teeth I (Fig. 2.3, a)) are resulted in Tab. 2.2, parameters for axial forms II (Fig. 2.3, b) and III (Fig. 2.3, c) – look in work [2].

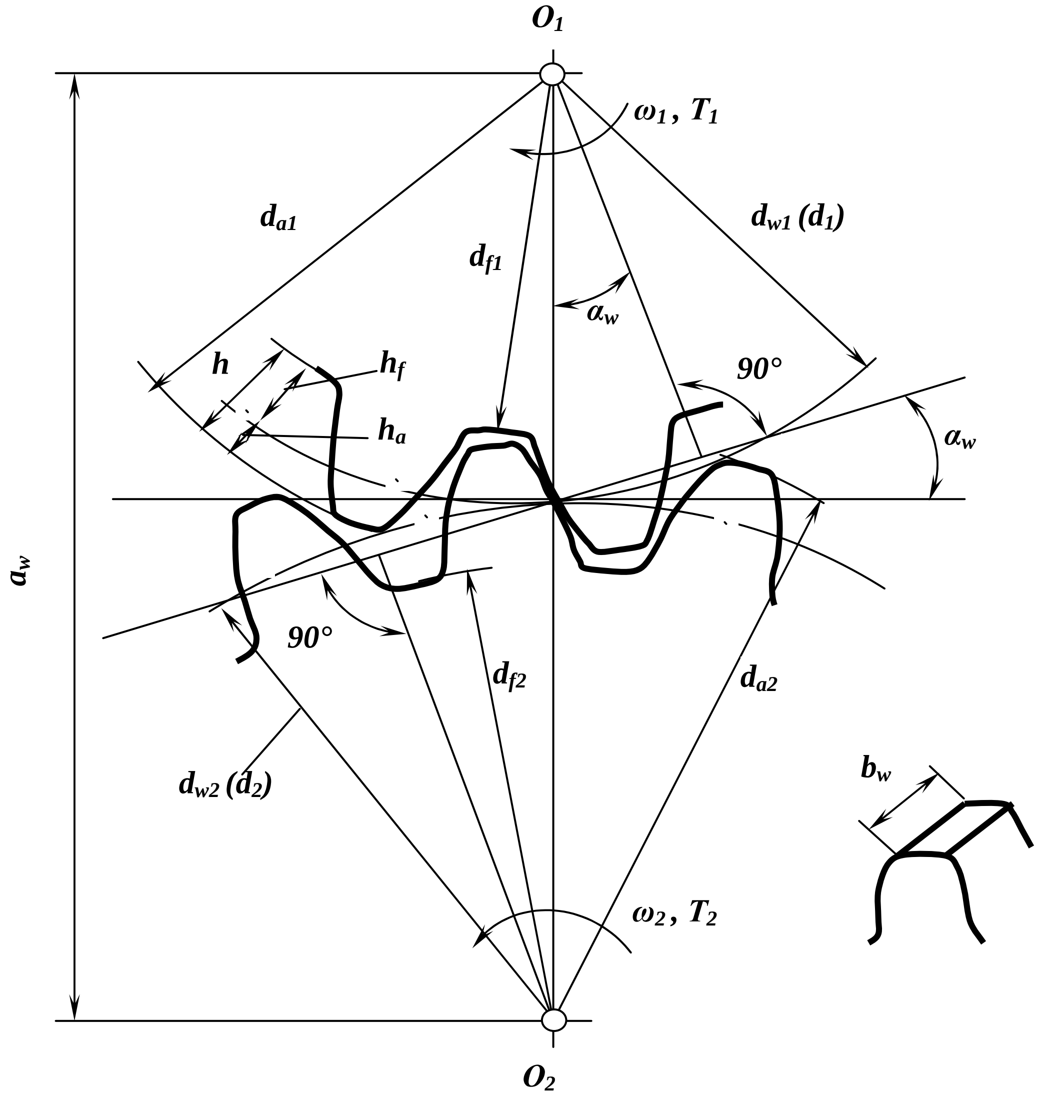


Fig. 2.1

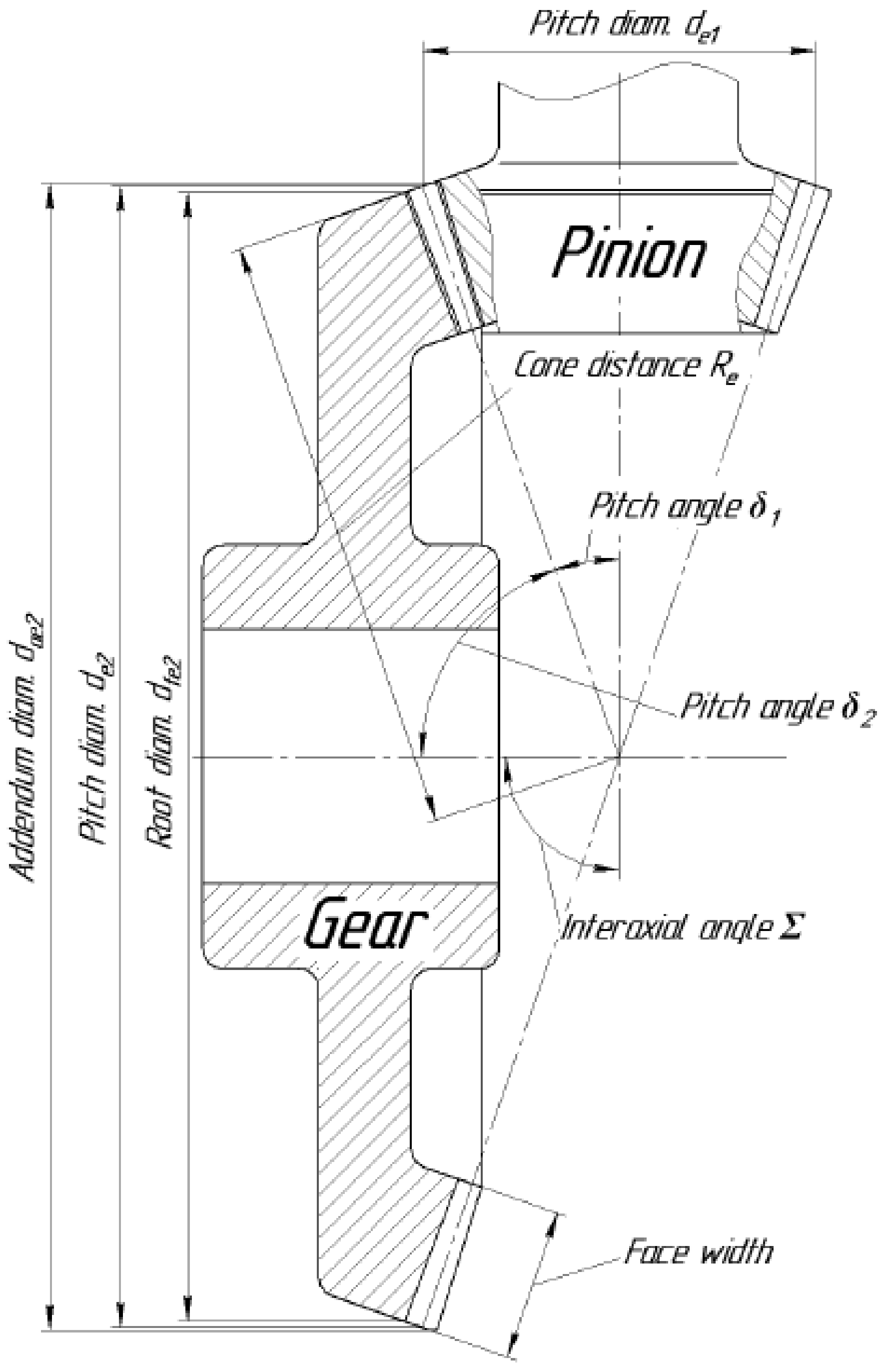


Fig. 2.2

Table 2.2

Parameter	Symbol	Formula
Interaxal angle	Σ	Due to design reasons
Outer pitch module	m_e	Standardized
Outer pitch diameter	d_e	$d_e = m_e z$
Number of teeth of a plane gear	Z_c	$z_c = \sqrt{z_1^2 + z_2^2}$

End of Tab. 2.2

Parameter	Symbol	Formula
Cone distance	R_e	$R_e = \dfrac{d_{e1}\sqrt{u^2+1+2u\cos\Sigma}}{2\sin\Sigma}$ when $\Sigma = 90°$ $R_e = 0{,}5\, m_e\, z_c$
Mean cone distance	R_m	$R_m = R_e - 0{,}5b$
Mean pitch module	$m = m_m$	$m_m = m_e\, R_m / R_e$
External transverse pitch module	m_{te}	$m_{te} = \dfrac{2R_e}{z_c}$
Mean normal module	m_n	$m_n = \dfrac{2R_m\cos\beta_n}{z_c}$
Half angle of the pitch cone	δ_1, δ_2	$tg\ \delta_1 = \sin\Sigma/(u+\cos\Sigma)$; when $\Sigma = 90°$ $tg\ \delta_1 = z_1/z_2$, $\delta_2 = \Sigma - \delta_1$
Mean pitch diameter	d	$d = m_m z$
Number of teeth of equivalent plane gear	Z_E	$z_E = z / \cos\delta$

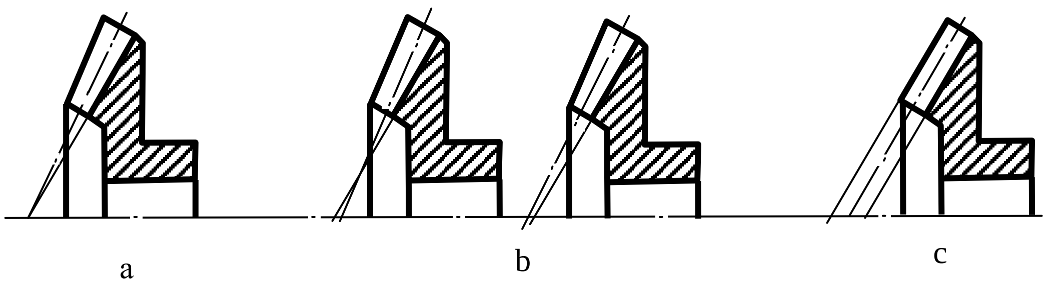


Fig. 2.3

2.2. Introduction

The calculation **objectives** are to obtain the pinion and gear dimensions to be able operate during required service life at the given load without fatigue damages.

Generally, calculation on durability includes:

– the describing of initial data;

– calculation of allowable stresses;

– calculation of the gearing basic geometrical parameters: the module, pitch diameters of gears, center distance, face width (at check calculation these parameters are known) and rated of the standardized parameters to standard values;

– check calculation on contact and bending endurance, on durability under the maximum (peak) loadings, on jamming (if necessary).

After calculations, one makes out the working documentation according to requirements of the state standards.

2.3. Initial Data

The parameters required for check calculation are presented in Tab. 2.3 (the asterisk notes sizes, which at design of new transfers define from design calculation).

Table 2.3

Parameter		**Way to determine**
Gearing type		According to given drive scheme
Power P, kW or torque on pinion T_1, Nm		According to given data
Service life L_h, h		According to given data
Pinion speed n_1, rpm		Due to kinematic calculation
Transfer number u		For gearing under design from kinematic calculation, for real gearing $u = z_2 / z_1$. It should be: $u \le 6$ for straight spur gear; $u \le 10$ for helix spur gear; $u \le 3{,}5$ for bevel gear
Number of teeth	pinion z_1	See subsection 2.3.1
	gear z_2	If u is given – $z_2 = z_1 u$
Module m*$^{)}$, mm		See subsection 2.3.7
Face width, mm	pinion b_1*$^{)}$	From strength calculation
	gear b_2*$^{)}$	
Shift factor	pinion x_1	[1, 2]
	gear x_2	
Helix angle β		See subsection 2.3.6
Degree of accuracy (GOST 1643–81)		See subsection 2.3.2
Grade of surface roughness Ra, mkm		See subsection 2.3.4
Steel	pinion	See subsection 2.3.3
	gear	
Heat treatment	pinion	
	gear	
Surface hardness	pinion	
	gear	

2.3.1. Number of Teeth

Gear wheel number of teeth considerably effects on transfer parameters. The error in gearing, dynamic load, vibration and speed of sliding in final points of gearing decrease with increasing in number of teeth at same diameters of the meshed gears, the transfer ratio and degree of accuracy. As a result, loading ability of gearing raises and friction losses and the weight of the metal translated in a shaving at cutting of teeth decrease.

However, teeth bend durability decreases with module reduction.

In the general mechanical engineering, gear wheels with number of teeth from 10 to 70 are usually used. For pinions of spur and bevel gears at active surface hardness more than 350 HB it is characteristic z_1 = 15…30. In the majority high-speed heavy-load external gears z_1 = 25...45, internal – z_1 = 81…127.

For machine under design, it should accept $z_1 > 17$. For the approached calculation it is possible to use dependence $z_1 = (17 + 0{,}003n_1)$.

2.3.2. Accuracy of Tooth Gearings

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

For spur, bevel and hypoid tooth gearings 12 degrees of accuracy from the first to the twelfth as its decrease are established.

Degrees of accuracy of tooth gearings of various machines see in Tab. 2.4.

Table 2.4

Gear transfer branch	Degree of accuracy
Aviation gearboxes, turbo engines and generators	4…7
Cars	6…9
Gearboxes for general purpose and transport machines	7…10

Accuracy degrees as a first approximation define depending on peripheral speed (Tab. 2.5).

Table 2.5

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

For bevel transfers at the specified peripheral speeds accuracy has to be accepted one degree higher.

2.3.3. Selection of Materials and Type 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 caп 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 oп 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–nickeI steel, grade 15XΦ chrome–vanadium steel, grade 18XГТ 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 38XMЮA (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.

Phenolic laminated toothed wheels cannot stand up to considerable loads because of their comparatively low resistance to wear and to contact effects. Phenolic laminated plastics are usually used to make one wheel of a pair, the other being manufactured from cast iron or steel with the hardness HB > 250. Since the transmitted load is determined by the plastic wheel this combination proves effective only in individual cases, for example, when a gear operating at high peripheral velocity must be made noiseless without increasing the accuracy of its manufacture.

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 Tab. 2.6.

Table 2.6

Steel	Type of treatment	Limit stresses, MPa		Hardness	
		$\sigma_в$	$\sigma_т$	of surface, HRC	of core, HB
40ХН	Tempering	850	600	230…300	
	Surface hardening	1600	1400	48…54	260…300
40Х	Tempering	850	550	230…260	
	Surface hardening	1000	850	45…55	260…280
30ХГСА		1080	830		280…320
12ХН3А	Casehardening	1000	850	58…63	260…400
12Х2Н4А		1200	1000	60…65	280…400
20Х2Н4А		1400	1200	60…65	300…400
38ХМЮА	Nitriding	1000	850	60	350

2.3.4. Roughness of Gears

The roughness of teeth working surfaces depends on degree of accuracy of a gear (Tab. 2.7).

Table 2.7

Degree of accuracy	Roughness of				
	hole		working surfaces		top land
	7	8	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_z 20	R_a 5	R_a 2,5	R_z 20

Roughness of face surfaces of a gear rim and hub – R_a 2,5, the surfaces of the gear which has been not interfaced to other details, – R_z 40…120.

2.3.5. Design Load

Teeth are loaded not only useful, but also additional forces. For the account of these forces design load factor $\boldsymbol{K}$ is used

$$\boldsymbol{K} = \boldsymbol{K_A}\ \boldsymbol{K_\upsilon}\ \boldsymbol{K_\beta}\ \boldsymbol{K_\alpha}. \tag{2.1}$$

Then tangential force on pitch diameter

$$\boldsymbol{F_t K} = \frac{\boldsymbol{2T_1 K}}{\boldsymbol{d_1}},$$

where $\boldsymbol{T_1}$ is the torque on a pinion.

In expression (2.1) $\boldsymbol{K_A}$ is an external dynamic factor (Tab. 2.8).

Table 2.8

Engine's condition of operation	**Driven machine's condition of operation**			
	Uniform	**With low ununiformity**	**With medium ununiformity**	**With high ununiformity**
Uniform	1,00	1,25	1,50	1,75
With low ununiformity	1,10	1,35	1,60	1,85
With medium ununiformity	1,25	1,50	1,75	2,00 and higher
With high ununiformity	1,50	1,75	2,00	2,25 and higher

More perfect way of the account of external dynamic load is usage of operating cyclogram. Then factor $\boldsymbol{K_A}$ is not necessary, and

$$\boldsymbol{K = K_v\, K_\beta\, K_\alpha}, \tag{2.2}$$

where $\boldsymbol{K_v}$ is the factor considering internal dynamic loading; $\boldsymbol{K_\beta}$ is the factor considering non-uniformity of load distribution on length of contact lines; $\boldsymbol{K_\alpha}$ is the factor considering distribution of loading between teeth, being in gearing (in design calculations it is possible to accept equal 1).

In a before resonance zone when the condition $\boldsymbol{vz_1 < 1000}$ for spur gear and $\boldsymbol{vz_1 < 1400}$ for helix transfers is satisfied (here $\boldsymbol{v}$ – peripheral speed, m/s), $\boldsymbol{K_v}$ is found under formula

$$\boldsymbol{K_v = 1 + \frac{w_v b_w}{F_t K_A}}. \tag{2.3}$$

The internal dynamic forces characterised by specific dynamic load $\boldsymbol{w_v}$, appear owing to errors in the basic both pitch and a profile of an active surface of teeth, deformation of gears and the elements connected to them.

Specific dynamic load

$$\boldsymbol{w_v = \delta\, g_0\, v\sqrt{\frac{a_w}{u}}}, \tag{2.4}$$

where $\boldsymbol{v}$ is a peripherial speed, m/s; $\boldsymbol{a_w}$ is a center distance, mm; $\boldsymbol{u}$ is a transfer number; $\boldsymbol{\delta_H}$, $\boldsymbol{\delta_F}$ are the factors which consider influence of a profile tip relief and a kind of teeth (Tab. 2.9); $\boldsymbol{g_0}$ is the factor considering influence of a pitch difference of a pinion and a gear (Tab. 2.10).

Table 2.9

Hardness	**Tooth type**		**δ**
Less 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

Table 2.10

Module *m*, mm	**Degree of accuracy**					
	4	**5**	**6**	**7**	**8**	**9**
	$\frac{g_0}{w_{v\,max},\ N/mm}$					
less 3,55	1,7	2,8	3,8	4,7	5,6	7,3
	32	85	160	240	380	700
3,55…10	2,2	3,1	4,2	5,3	6,1	8,2
	53	105	194	310	410	880
more 10	2,7	3,7	4,8	6,4	7,3	10,0
	98	150	250	450	590	1050

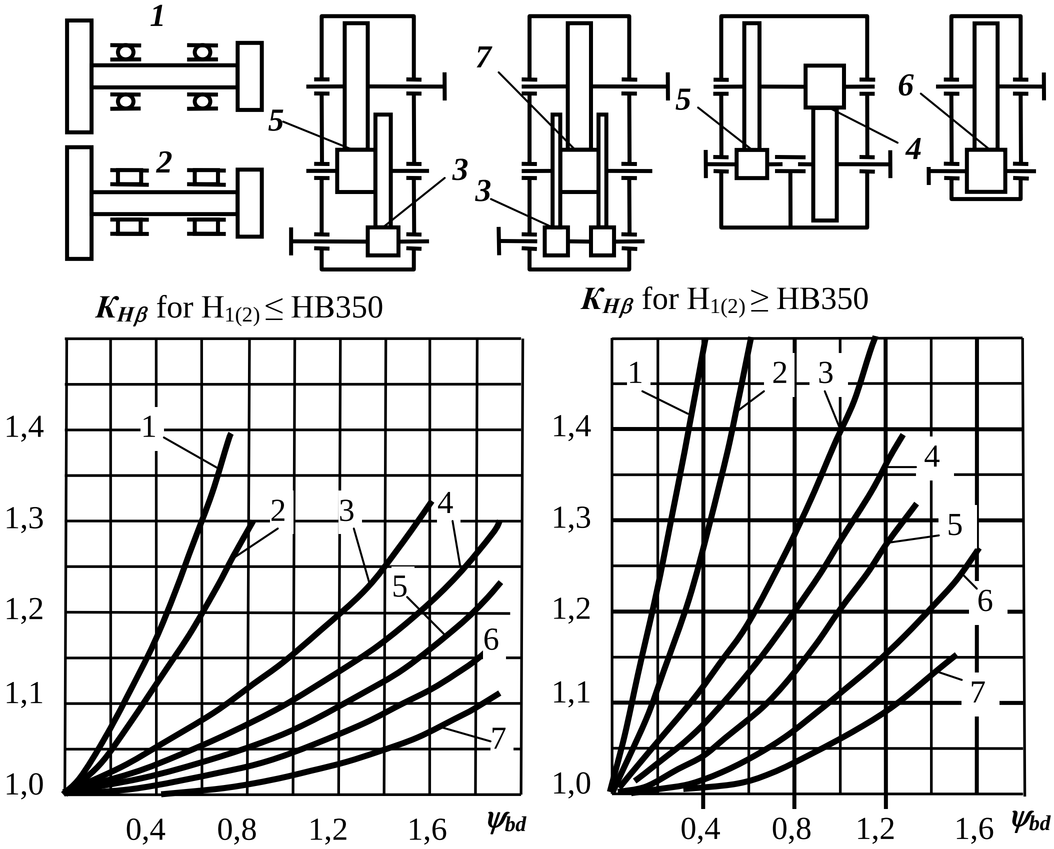
If calculated value w_{Hv} exceeds limit w_{vmax} (see Tab. 2.10), it should be accepted equal to limit value.

The factor K_β is the relation of the greatest specific load to its average. It considers non-uniformity of load distribution on length of the contact line, arising owing to elastic deformations of shaft, a warp and deterioration of bearings. Load concentration is directly proportional to corners of a shaft warp under the interfaced gears and teeth face width.

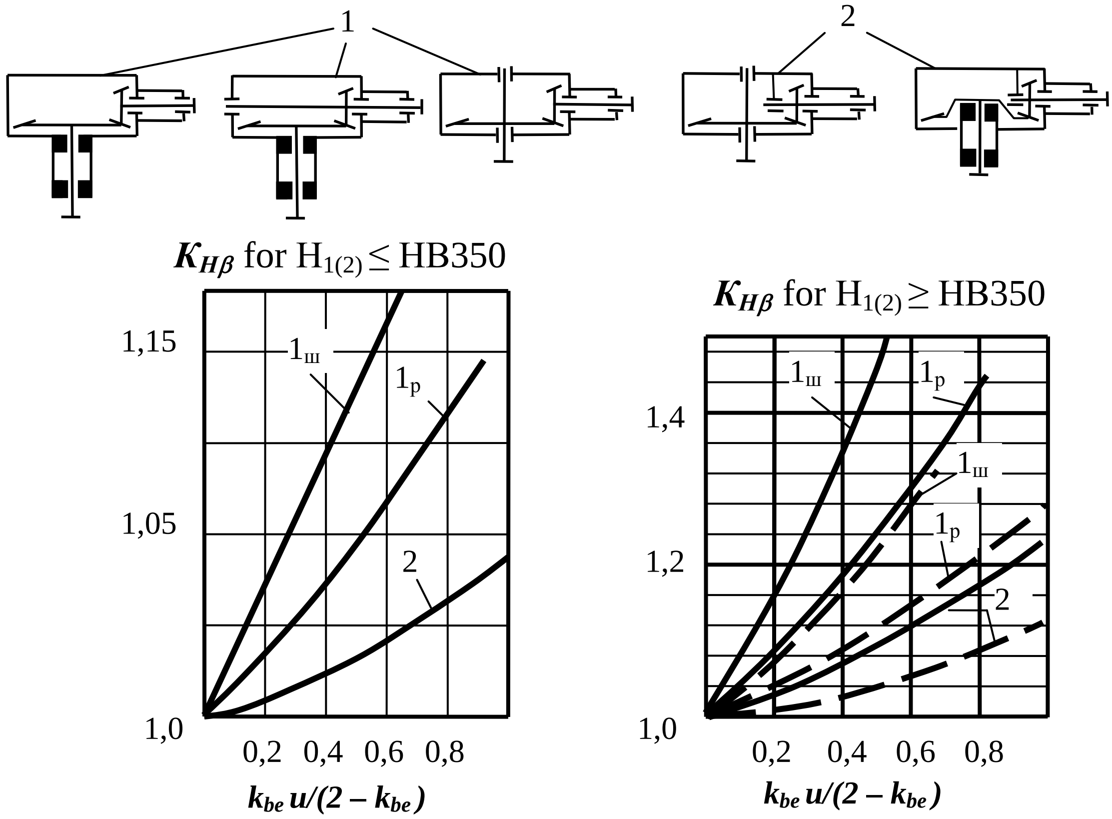
In designing and approximate calculations, it is possible to accept value $K_{H\beta}$ approximately, by means of graphs (Fig. 2.4).

For bevel transfers with an interaxal angle 90° approximate values $K_{H\beta}$ depending on parameter $k_{be}\, u\ /\ (2 - k_{be})$ can be defined under the graphs shown in Fig. 2.5.

In Fig. 2.5 curves with an index "ш" correspond to shaft on ball bearings, with an index "р" – on roller, continuous lines – to transfers with straight teeth, dotted – to transfers with spyral or helix teeth.



a Fig. 2.4 b



a b

Fig. 2.5

At calculations with computer, it is possible to use the approximations shown in Tab. 2.11.

Table 2.11

Figure	Line	Parameters	Formula
2.4, b	2	$x = \psi_{bd}$ $y = K_{H\beta}$	$y = 1{,}81x + 0{,}207/x$
	3		$y = 0{,}969 + 0{,}318x + 0{,}09x^2$
	4		$y = e^{(0{,}261x - 0{,}022/x)}$
	5		$y = e^{(0{,}1713/x + 0{,}4165\ \ln x)}$
	6		$y = 0{,}868 + 0{,}132e^x - 0{,}107x$
2.5, b (continuous curves)	1ш	$x = k_{be}u/(2 - k_{be})$ $y = K_{H\beta}$	$y = 0{,}8964 + 0{,}0071/x + 1{,}08x$
	1р		$y = e^{(0{,}4098x + 0{,}000257/x)}$
	2		$y = 1/(1 - 0{,}1666x)$

Design load factors at the specified calculation on contact K_H and bending K_F endurance differ from each other and look like

$$K_H = K_{Hv} K_{H\beta}, \qquad K_F = K_{Fv} K_{F\beta}.$$

With engineering accuracy it is possible to consider $K_H = K_F = K$.

2.3.6. Helix Angle

Approximately 97 % of spur gears in aviation gearboxes and drives have straight teeth.

In the general mechanical engineering approximately 30 % transfers are helical because they have more loading ability and work smoothness, less overall dimensions and weight. The main disadvantage of such transfers is the axial forces, which increase with growth of a helix angle β. Therefore, frequently herringbone gearing is preferable because axial forces are counterbalanced in them.

Usually helix angle is β = 8…18°, in herringbone gears β = 28…45 °. Moreover, the condition should be satisfied

$$\varepsilon_\beta = \frac{b_w \sin\beta}{\pi\, m_n} \geq 1{,}1\,,$$

where ε_β – factor of axial overlapping, m_n – the normal module.

For power bevel transfers with a spyral teeth the corner $\beta \approx 35°$ is the most widely spread.

2.3.7. Module

The module is one of the main parameters of 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. In helical transfers it is considered circumferential $\boldsymbol{m_t}$ and normal $\boldsymbol{m_n}$ modules, in straight bevel meshing – external $\boldsymbol{m_e}$, in bevel gears with a spyral teeth – external $\boldsymbol{m_{te}}$ or mean $\boldsymbol{m_m}$ module.

At a module selection it is considered the following:

– with module increase at identical diameters of wheels bend durability of teeth grows;

– 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 training, the tooth is calcinated for all thickness);

– with module reduction metal losses decrease.

Values of modules are standardised in a range from 0,05 to 100 mm. The interval 1…10 mm, consisting of two rows is most common:

1st row	1	1,25	1,5	2	2,5	3	4	5	6	8	10
2nd 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.

As a rule the normal module is standardised in helical transfers, circumferential at an external end face – in bevel with teeth of the form I (see Fig. 2.3, a), normal on the middle of face width of a gear – in wheels with teeth of forms II (see Fig. 2.3, b) and III (see Fig. 2.3, c).

Value of the module should be co-ordinated with face width of a gear, thus it is necessary to provide following proportions: $\boldsymbol{b/m \le 15}$ for a spur transfer and $\boldsymbol{b/m_{te} \le 10}$ – for the bevel one.

2.4. Allowable Stresses

2.4.1. Contact Stresses

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

$$[\sigma]_H = \frac{\sigma_{H\,lim} Z_N}{S_H} Z_R Z_v Z_L Z_X \,, \tag{2.5}$$

where σ_{Hlim} is the limit of contact endurance of the teeth surfaces, corresponding to 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. In designing calculation it is accepted

$$Z_R\, Z_v\, Z_L\, Z_X = 0{,}9.$$

Limit of contact endurance of teeth surfaces σ_{Hlim} is calculated according to formulae shown in Tab. 2.12.

Table 2.12

Type of termal treatment	**Mean surface hardness**	**Steel**	**σ_{Hlim}, MPa**
Annealing, normalizing, tempering	≤ 350 HB	Carbon and alloy	2 HB + 70
Volume hardening	38...50 HRC		17 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 larger the hardness, the larger allowable stresses and load capacity. However, in some cases achievement of limiting hardness is very expensive for manufacture of such gears. Besides, extremely high hardness is useful in combination with a viscous core, in other case teeth become fragile and the main reason 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 concrete enterprise and the tasks.

The minimum safety factor is: for gears with homogeneous structure of a material (after such kinds of heat treatment, as improvement, volume training) S_{Hmin} = 1,1; for gears with surface hardening of teeth (casehardening, nitriding) S_{Hmin} = 1,2.

For the transfers which failure is caused with heavy consequences, values of the minimum factors of safety factor need to be increased to 1,25 and 1,35.

The method of the durability factor definition Z_N (formula (2.5)) see in Tab. 2.13.

Table 2.13

Parameter	**The method of definition**
1. Durability factor Z_N	$$Z_N = \sqrt[m]{\frac{N_{H\,lim}}{N_K}}$$ 0,75 < Z_N < 1,8 in case of surface hardening of teeth; 0,75 < Z_N < 2,6 for homogeneous structure of a material
2. The base number of cycles of stress $N_{H\,lim}$	$N_{Hlim} = 30\,(HB)^{2,4} \le 120 \cdot 10^6$ Correlation of hardness numbers HRCЭ, HB and HV is shown on Fig. 2.6. For computer calculation next approximation could be used: $H_{HB} = H_{HRC} / 0,102$
3. Number of cycles of stress during service life N_K	At constant load $N_K = 60cnL_h$

Parameter	The method of definition
4. Number of meshing at one revolution ***c***	According to gearing scheme (see Fig. 2.7)
5. Rotational speed ***n***, rpm	According to initial data
6. Service life L_h, h	According to initial data
7. Root index ***m***	If $N_K > N_{H\,lim}$ $m = 20$ If $N_K < N_{H\,lim}$ $m = 6$

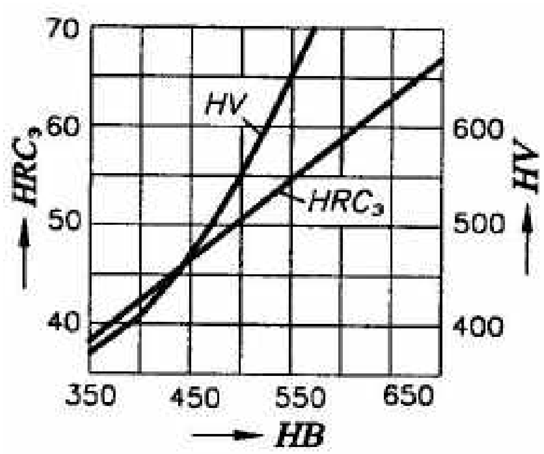


Fig. 2.6

For subsequent calculations next allowable contact stress should be accepted:

– for **straight spur** transfers – smaller from values for a pinion and a gear;

– for **helical** and **herringbone** transfers

$$[\sigma]_H = 0{,}45\left([\sigma]_{H1} + [\sigma]_{H2}\right) \qquad (2.6)$$

at condition

$$[\sigma]_{H\,min} < [\sigma]_H < 1{,}25\,[\sigma]_{H\,min},$$

where $[\sigma]_{Hmin}$ – smaller from calculated above stresses $[\sigma]_{H1}$ and $[\sigma]_{H2}$.

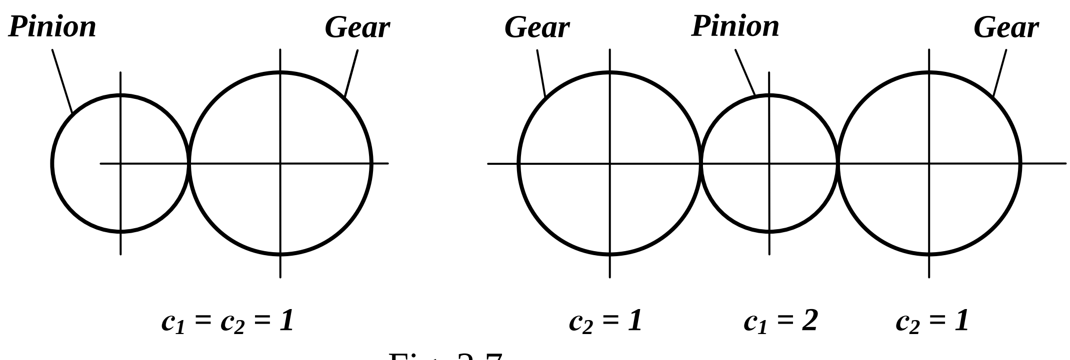


Fig. 2.7

2.4.2. Allowable bending stresses

Allowable bending stresses

$$[\sigma]_F = \frac{\sigma_{F\,lim}}{S_F} Y_N Y_R Y_z\,. \qquad (2.7)$$

The expression (2.7) parameters are described in Tab. 2.14.

Table 2.14

Parameter	The method of definition
The bend endurance limit corresponding to base number of stress cycles σ_{Flim}, MPa	It is obtained depending on a material and method of thermal treatment (Tab. 2.15)
Safety factor S_F	
Durability factor Y_N	$Y_N = \sqrt[q_F]{\dfrac{N_{F\ lim}}{N_K}} \geq 1$, (if $N_K > N_{Flim}$ $Y_N = 1$)
The base number of cycles of stress $N_{F\ lim}$	$N_{F\ lim} = 4 \cdot 10^6$
Root index q_F	$q_F = 6$ for gears with homogeneous structure of a material $q_F = 9$ for nitrited and case–hardened gears with an ungrinding interjacent surface
The factor considering a method of billet production Y_z	For forging and forming $Y_Z = 1$, for rolled metal $Y_Z = 0,9$, for casting $Y_Z = 0,8$
Factor considering a roughness of interjacent surface Y_R	For a surface roughness no more $R_z = 40$ microns $Y_R = 1$. For polishing depending on a method of thermal hardening it is accepted: $Y_R = 1,05$ at cementation, nitrocementation, nitriding; at RF induction hardening when the tempered layer repeats hollow outlines between teeth $Y_R = 1,2$ at normalisation and improvement; at chardening when the tempered layer is distributed on all section of a tooth

Table 2.15

Heat treatment	**Steel**	**Surface hardness**	**σ_{Flim}, MPa**	**S_F**
Case–hardening	20ХН, 12ХН2, 15ХГНТА	57…63 HRC	950	1,55
	18ХГТ, 30ХГТ, 20Х		820	1,55
Nitro–hardening	25ХГМ	57…63 HRC	1000	1,55
	25ХГТ, 35ХГТ, 35Х		750	1,55
Normalization, tempering	40, 45, 40Х, 40ХН	180…350 HB	1,75 HB	1,7
Volume hardening	40Х, 40ХН, 40ХФА	45…55 HRC	580	1,7
Surface hardening (hardened layer repeats depth contour)	40Х, 35ХМ	48…58 HRC	680	1,7
	40ХН, 40ХН2МА		580	
Surface hardening (all tooth section is hardened)	40Х, 35ХМ		480	1,7
	40ХН, 40ХН2МА		580	
Nitriding	38Х2Ю, 40Х2НМА	700…950 HV	$12HRC_{core}$ + +290	1,7

Allowable bending stresses compare with corresponding acting stresses.

2.4.3. Allowable Stress for Calculation on Strength under the Maximum (Peak) Loading

Allowable contact stress at the maximum loading depends on a method of heat treatment and character of change of hardness on depth of a tooth.

For gears:

– after normalisation, improvement or through hardening with low tempering $[\sigma]_{Hmax} = 2{,}8\sigma_T$;

– case-hardened or tempered on a contour $[\sigma]_{Hmax} = 44\ HRC$;

– nitrated $[\sigma]_{Hmax} = 3\ HV$.

Allowable bending stress for the treatment conditions shown in Tab. 2.16, defines as

$$[\sigma]_{F\,max} = \frac{\sigma_{FSt}}{S_{FSt}}. \qquad (2.8)$$

In expression (2.8) σ_{FSt} is the limit bending stress at the maximum load, MPa, it is defined on Tab. 2.16; S_{FSt} is the safety factor, for steels and the heat treatment specified in Tab. 2.16, $S_{FSt} = 1{,}75$.

Table 2.16

Heat treatment		**Steel**	**Hardness of surface**	**Hardness of core**	**σ_{FSt}, MPa**
Normalization, tempering		Carbon and alloy	200…350 HB		6,5 HB
Volume hardening		With nikel content *>1 %*	48…52 HRC		2500
		Another alloy			2250
Surface hardening	contour	With nikel content *>1 %*	48…54 HRC	24…30 HRC	2200
		Another alloy			1800
	transparent	Another alloy	48…52 HRC		2500
		With nikel content *>1 %*			2250
Nitriting		Alloy without aluminium	550…850 HV	24…30 HRC	1800
Nitrocasehardening		Alloy with molybdenum	56…60 HRC	32…45 HRC	2500
		Another alloy		27…45 HRC	2200
Casehardening		With nikel content *>1 %*	56…62 HRC	27…43 HRC	2800
		Another alloy	54…60 HRC	30…43 HRC	2000

2.5. Design Calculation

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

At design calculation basical geometrical parameters should be obtained. In particular, approximate value of a pinion pitch diameter and than all the others could be found.

2.5.1. Spur Transfers

Approximate value of a pinion pitch diameter, mm,

$$d_{w1} = K_d \sqrt[3]{\frac{T_1 \, K_{H\beta}}{\psi_{bd} [\sigma]_H^2} \frac{u_{12} + 1}{u_{12}}}, \qquad (2.9)$$

where K_d is the auxiliary factor (for straight transfers $K_d = 770$, for helical and herringbone K_d = 675), T_1 is the torque on a pinion, Nm.

At constant load

$$T_1 = 9550\frac{P_1}{n_1},$$

where P_1 is the rated power on a pinion, kW; n_1 is the pinion rotational speed, rpm; u_{12} is the tooth gearing transfer number, $u_{12} = z_2/z_1$; ψ_{bd} is the relative face width of a tooth, $\psi_{bd} = b/d_{w1}$ (Tab. 2.17).

Table 2.17

Gear location relatively of bearings	Relative face width according to surface hardness	
	less 350 HV at least for one gear	more 350 HV for both gears
Symmetrically	0,8…1,4	0,4…0,9
Asymmetrically	0,6…1,2	0,3…0,6
Cantileverly	0,3…0,4	0,2…0,3

The note: bigger values are accepted for constant loads.

A center distance a_w, mm, can be defined also from a condition of contact durability

$$a_w = K_a(u+1)\sqrt[3]{\frac{T_1 K_{H\beta}}{\psi_{ba} u_{12} [\sigma]_H^2}}, \qquad (2.10)$$

where K_a is the auxiliary factor (for straight transfers $K_a = 495$, for helical and herringbone $K_a = 430$); ψ_{ba} is the relative width of a tooth, $\psi_{ba} = b/a_w$.

Factors ψ_{bd} and ψ_{ba} are correlated among themselves by the next dependence $\psi_{bd} = 0{,}5\ (u_{12} + 1)\ \psi_{ba}$. 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.

Than other geometrical parameters of transfer are obtained by such way:

for straight transfers:

– define the module

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

and round it to the nearest standard value m_{st} (see subsection 2.3.7);

– calculate number of teeth of a wheel $z_2 = z_1 u_{12}$. If z_2 – is not integer it rounded to the nearest integer and specify transfer number of a stage $u_{12} = z_2 / z_1$;

– calculate pitch diameters of wheels as $d_{w1} = m_{st} z_1$ and $d_{w2} = m_{st} z_2$;

– define face width $b_w = \psi_{bd}\, d_{w1}$ and round it till the normal size;

for helical transfers:

– define transverse module $m_t = d_{w1} / z_1$;

– assume helix angle β (see subsection 2.3.6) and calculate the normal module $m_n = m_t \cos\beta$, then round it to the nearest standard value (see subsection 2.3.7);

– recalculate transverse module $m_t = m_n / cos\beta$ and external pitch diameter $d_w = m_t z$;

– obtain face width $b_w = \psi_{bd}\, d_{w1}$;

– check condition $\varepsilon_\beta = \dfrac{b_w \sin\beta}{\pi\, m_n} \geq 1{,}1$. If it don't satisfy change angle β, face width b_w or normal module m_n.

2.5.2. Bevel Transfers

Pitch diameter at an external face (see Fig. 2.2)

$$d_{e_1} = K_d \sqrt[3]{\frac{T_1\, K_H \sin\Sigma}{\vartheta_H\,(1 - k_{be})\,k_{be}\,[\sigma]_H^2\, u}}, \tag{2.11}$$

where K_d is the auxiliary factor $K_d = 1013$; θ_H is the factor considering the form of a tooth. For straight wheels $\theta_H = 1$, for spiral teeth transfers $\theta_H = 1{,}5$; k_{be} is the relative face width $k_{be} = b/R_e$ = 0,2… 0,3 (b is the face width, R_e is the external cone distance (see Fig. 2.2)); K_H is the load factor (see subsection 2.3.5).

At design calculation factor $K_{H\beta}$ depending on $k_{be}u / (2 - k_{be})$ could be assumed by means of Fig. 2.5, and K_{Hv} – depending on degree of accuracy, hardness of surfaces – from Tab. 2.18 (the top values – for straight teeth, bottom – for helical one).

Table 2.18

Degree of accuracy	Surface hardness	K_{Hv}				
		Peripherial 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

Than module at the back face $m_{te} = d_{e1} / z_1$ (for straight teeth – m_e) should be obtained.

The value of the module should be assumed as the nearest standard value and further:

– define number of teeth of a gear $z_2 = z_1 u_{12}$. If z_2 is not integer it is rounded to the nearest integer and transfer number of a step is recalculated $u_{12} = z_2 / z_1$;

– recalculate pitch diameters

$$d_{e1} = m_{te} z_1;\ d_{e2} = m_{te} z_2;$$

– find number of teeth of flat gear

$$z_c = \sqrt{z_1^2 + z_2^2}$$

(calculate value to within three signs after a comma);

– calculate external cone distance

$$R_e = 0{,}5 m_{te} z_c\ ;$$

– define face width $b = k_{be} R_e$ and round one to the integer;

– recalculate $k_{be} = b/R_e$ and check conditions

$$k_{be} \le 0{,}3,\ b/m_{te} \le 10.$$

If necessary change face width b, the module m_{te} or numbers of teeth of a pinion z_1 and a wheel z_2, keeping the set transfer number u_{12}.

2.6. Check Calculation

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? It should satisfy all of next conditions (Tab. 2.19).

Table 2.19

Criterion	**Load conditions**	
	teeth surface	**bending**
Stress	$\sigma_H \le [\sigma]_H$ $\sigma_{Hmax} \le [\sigma]_{Hmax}$	$\sigma_F \le [\sigma]_F$ $\sigma_{Fmax} \le [\sigma]_{Fmax}$

2.6.1. Calculation on Contact Durability

2.6.1.1. Calculation of Spur Transfers on Contact Endurance

Contact stress in a gearing pole is

$$\sigma_H = Z_E Z_H Z_\varepsilon \sqrt{\frac{F_t K_H}{b_w d_w} \frac{u+1}{u}} \le [\sigma]_H\ . \qquad (2.12)$$

Load factor K_H is defined according to the recommendations stated in part 2.3.5, other parameters are described in Tab. 2.20.

Table 2.20

Parameter	The method of definition
The factor considering mechanical properties of materials of interfaced gears Z_E, MPa$^{-0,5}$	$$Z_E = \sqrt{\dfrac{1}{\pi\left(\dfrac{1-\nu_1^2}{E_1}+\dfrac{1-\nu_2^2}{E_2}\right)}}$$ For steels $E = 2{,}1\cdot 10^5$ MPa, $Z_E = 190$
The factor considering the form of teeth interfaced surfaces in a pole of gearing Z_H	$$Z_H = \frac{1}{\cos\alpha_t}\sqrt{\frac{2\cos\beta}{tg\alpha_{tw}}}$$ without shift $Z_H = 2{,}5$
The factor considering total length of contact lines Z_ε	If $\varepsilon_\beta \geq 1$, $Z_\varepsilon = \sqrt{\dfrac{1}{\varepsilon_\alpha}}$, For straight teeth $Z_\varepsilon = \sqrt{\dfrac{(4-\varepsilon_\alpha)}{3}}$
Tangentional force on pitch cylinder F_t, H	$$F_t = \frac{2000\ T_1}{d_1}$$ without shift $d = d_w$

It would be better, if a relative deviation of contact stress

$$\varepsilon = \frac{\sigma_H - [\sigma]_H}{[\sigma]_H}\cdot 100\ \%$$

is in limits $-5 \leq \varepsilon \leq 3$, i.e. underloading should not be more 5 %, and overload – 3 % ($[\sigma]_H$ it is defined under the recommendations stated in subsection 2.4.1).

If the value ε lays out of the specified range in limits ± 20 % it is necessary to change face width b_w, maintaining recommended value ψ_{bd} (see Tab. 2.17) (at positive ε increase b_w to the nearest normal value, at negative ε – reduce) and to recalculate $K_{H\beta}$, K_{Hv}, K_H and σ_H. At a bigger deviation it is necessary to change the module m (at positive ε increase it to next standard value, at negative ε – reduce), and then to count pitch diameters d_{w1} and d_{w2}, center distance a_w, factors $K_{H\beta}$, K_{Hv}, K_H and actual stress σ_H. Calculation is finished, when ε will get to the recommended region.

2.6.1.2. Calculation of Bevel Transfers on Contact Endurance

Let us assume, that tangential force is applied to the tooth middle, i.e. at mean pitch diameter d_m (see Fig. 2.2), which for a gear is equal

$$d_{m1} = d_{e1}(1-0{,}5k_{be}).$$

Therefore

$$F_t = \frac{2000T_1}{d_{m1}}.$$

Contact stress in a pole of gearing is

$$\sigma_H = 32240\sqrt{\frac{T_1\, K_{H\beta} K_{Hv} K_{H\alpha} \sin \Sigma}{d_{e_1}^3 (1 - k_{be}) k_{be}\, u\, \vartheta_H}},$$

where $K_{H\beta}$, K_{Hv} are obtained as it is described in subsection 2.3.5. It should be noted that in the formula (2.4) instead of a_w it is necessary to substitute value $d_{m1}\,(u+1)$.

It is allowable, that a relative deviation of contact stress

$$\varepsilon = \frac{\sigma_H - [\sigma]_H}{[\sigma]_H} 100\,\%$$

should be like in case of spur gears in limits $-5 \le \varepsilon \le 3$ ($[\sigma]_H$ it is defined according to the recommendations stated in subsection 2.4.1).

If the value ε is outside of the specified region, it is necessary to change face width b, maintaining recommended value k_{be} (see subsection 2.5.2) and then to recalculate $K_{H\beta}$, K_{Hv}, K_H and σ_H. Calculation is finished, when ε will get to the required interval.

2.6.1.3. Calculation on Contact Strength at Action of the Maximum Load

At action of maximum torque T_{max} for the given service life contact stress σ_{Hmax} should not exceed the lowest allowable stress $[\sigma]_{Hmax}$ (see subsection 2.4.3).

Acting stress

$$\sigma_{H_{max}} = \sigma_H \sqrt{\frac{T_{max}}{T_{nominal}}}.$$

2.6.2. Calculation at Bend Durability

2.6.2.1. Calculation of Spur Transfers

The bend endurance is necessary for prevention of a fatigue break of teeth and is defined **for each of gears** by comparison of local bend stress operating in dangerous section on a transitive surface and allowable stress (see subsection 2.5.1):

$$\sigma_{F\,(1,2)} \le [\sigma]_{F\,(1,2)}.$$

Local stress

$$\sigma_F = \frac{F_t}{b_w m} K_F Y_{FS} Y_\beta Y_\varepsilon\,, \qquad (2.13)$$

where b_w is the face width, mm, is defined either from designing calculation, or from dimensions of existing transfer; F_t is the tangential force (see Tab. 2.20); K_F is the load factor (see subsection 2.3.5); other parameters are described in Tab. 2.21.

Table 2.21

Parameter	The method of definition
Factor, considering the tooth form and stress concentration Y_{FS}	$Y_{FS} = 3{,}47 + \frac{13{,}2}{z_v} - 29{,}7\frac{x}{z_v} + 0{,}092x^2$, where x – summary shift factor; z_v – equivalent number of teeth. For spur gears $z_v = \frac{z}{\cos^3\beta}$ For bevel: straight teeth $z_v = \frac{z}{\cos\delta}$, others $z_v = \frac{z}{\cos\delta\cos^3\beta}$
The factor considering a tooth helix angle Y_β	$Y_\beta = 1 - \varepsilon_\beta\frac{\beta}{120} \geq 0{,}7$
Factor considering teeth overlapping Y_ε	For straight teeth $Y_\varepsilon = 1$ For helix teeth at $\varepsilon_\beta < 1$ $Y_\varepsilon = 0{,}2 + \frac{0{,}8}{\varepsilon_\alpha}$, if $\varepsilon_\beta \geq 1$ $Y_\varepsilon = \frac{1}{\varepsilon_\alpha}$

2.6.2.2. Calculation of Bevel Transfers

Local stress for straight bevel gears equals

$$\sigma_{F_1} = \frac{2280T_1K_{F\beta}K_{Fv}Y_{FS_1}}{bd_{e_1}m_e(1-k_{be})}.$$

For wheels with spiral teeth

$$\sigma_{F_1} = \frac{1500T_1K_{F\alpha}K_{F\beta}K_{Fv}Y_{FS_1}}{bd_{e_1}m_e(1-k_{be})\cos\beta_n}.$$

Factor Y_{FS} should be increased by 20 % in comparison with obtained in Tab. 2.21.

2.6.2.3. Calculation of Teeth Strength at Maximum Bend Load

Local stress at the simplified calculations is

$$\sigma_{F_{max}} = \sigma_F\frac{T_{max}}{T_{nom}}.$$

2.7. Gear Design

Key parameters of gears – width, diameter, the module, number of teeth – are found at calculation on durability. The dimensions of other elements of a wheel are obtained during their designing.

The form of gears defines depending on a way of manufacturing of billet (forging, punching, moulding, cutting from a round bar). Under condition of series and mass manufactures of wheels, billets are produced by means of stamping in bilateral stamps. Thus, it is necessary to provide radiuses of a rounding off not less than 5 mm and stamping biases 5 … 7 $^{\circ}$.

2.7.1. Gear Dimensions

Elements of a gears' design and their designation are shown on Fig. 2.8. Relations between the sizes of elements of spur and bevel wheels are resulted in Tab. 2.22, 2.23 (parameters of a bevel wheel profile with the form of a tooth I see in Tab. 2.2). Smaller values of length are selected for pressed-on to a shaft gears. In case of mass gear production the length of a hub should be assumed equal to face width.
For weight reduction of gears (especially in flying machines), a thickness of walls of a disk, a rim and a hub is reduced on 20...30 %.

When the difference in the diameters of a pinion and a shaft is small (***d_a <2d***), teeth are directly cut on a shaft (shaft-gear wheel).

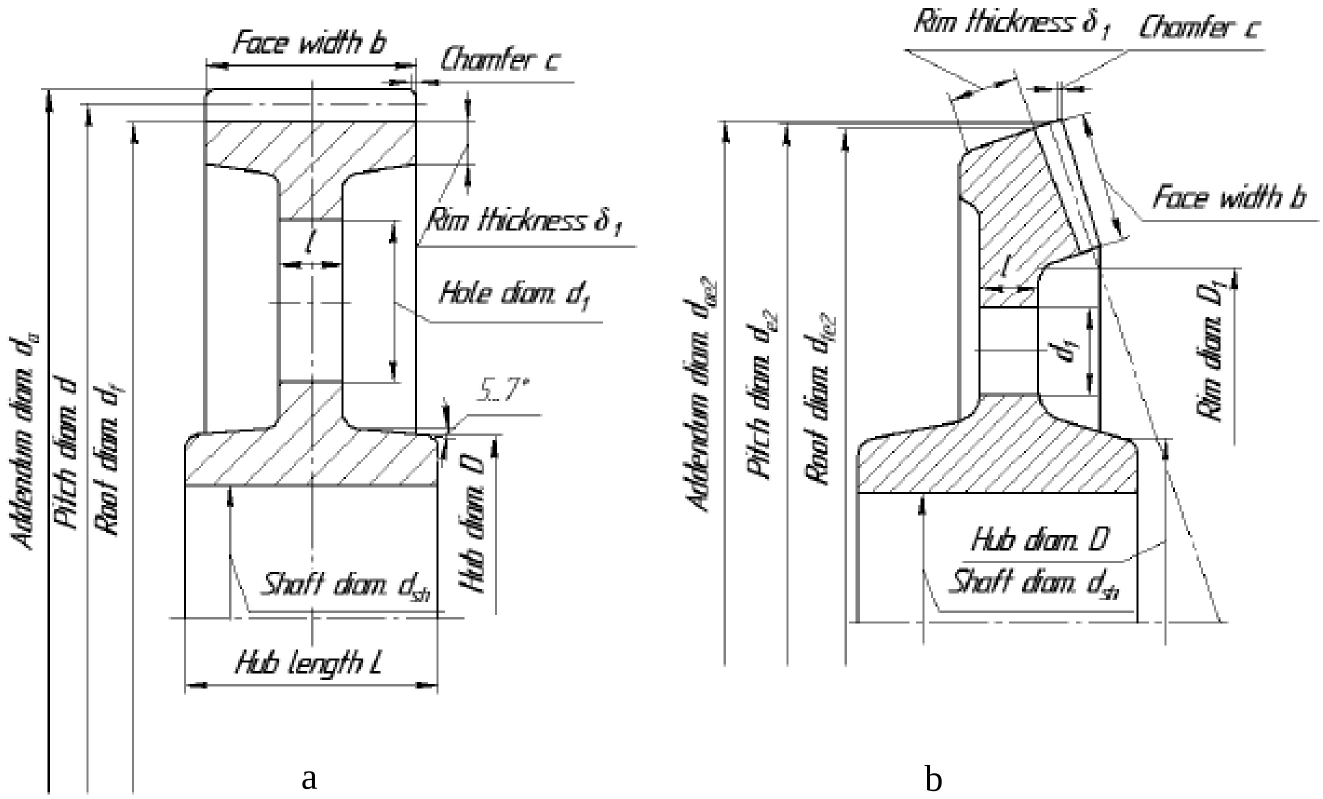


Fig. 2.8

Table 2.22

Parameter, mm	Symbol	The method of definition
Outside diameter	$d_{a(1,2)}$	$d_{a_{1,2}} = d_{w_{1,2}} + 2m$
Root diameter	$d_{f_{1,2}}$	$d_{f_{1,2}} = d_{w_{1,2}} - 2{,}5m$
A gear face width*	b_2	$b_2 = b_w$ from check calculation (part 2.6)
A pinion face width *	b_1	$b_1 = b_w + (1...2)m$
A shaft hole diameter*	d_{sh}	$d_{sh} = \sqrt[3]{\dfrac{1000\,T_2}{0{,}2[\tau]}}$, $[\tau] = 30…50$ MPa, T_2 in Nm
A hub diameter*	D	$(1{,}5...1{,}7)\ d$
A rim diameter *	D_1	For spur gears $D_1 = d_f - 2\delta_1$, For bevel – geometrically
A hub length*	L	$(0{,}7...1{,}8)\ d_{sh}$
A rim thickness*	δ_1	$(2{,}5...4)\ m$ (more than 10 mm)
A disk thickness *	l	$(0{,}2...0{,}3)\ b$
The holes' diameter* (4 – 6 holes)	d_1	$\approx \dfrac{D_1 - D}{3}$
Chamfer*	c	$0{,}5\ m$

* Calculated values should be rounded to nearest standard value.

Table 2.23

Parameter	Symbol	The method of definition
Addendum factor	h_a^*	$h_a^* = \cos\beta_n$
Clearance factor	c^*	$c^* = 0{,}2$
External addendum height, mm	h_{ae}	$h_{ae} = m_e\,(h_a^* + x\cos\beta_n)$
External dedendum height, mm	h_{fe}	$h_{fe} = m_e\,(h_a^* + c^* - x\cos\beta_n)$
External height of teeth, mm	h_e	$h_e = h_{ae} + h_{fe}$
External outside diameter of teeth, mm	d_{ae}	$d_{ae} = d_e + 2\,h_{ae}\cos\delta$
Addendum angle	θ_f	$tg\theta_f = h_{fe} / R_e$
Angle of root cone	δ_f	$\delta_f = \delta - \theta_f$

2.8. Examples

2.8.1. Example 1

Initial data for calculation of spur straight transfer are represented in Tab. 2.24.

Table 2.24

<table>
<tr><th colspan="2">Parameter</th><th>Symbol</th><th>Value</th></tr>
<tr><td colspan="2">Gears’ function</td><td>–</td><td>One-stage gearbox</td></tr>
<tr><td colspan="2">Operation mode</td><td></td><td>Continual</td></tr>
<tr><td colspan="2">Service life, h</td><td>L_h</td><td>20 000</td></tr>
<tr><td colspan="2">Power on a pinion, kW</td><td>P</td><td>5,5</td></tr>
<tr><td colspan="2">Type and code of engine</td><td>–</td><td>Electrical asynchronous 4A132S6, $\frac{T_{max}}{T_{ном}} = 2,2$</td></tr>
<tr><td colspan="2">A pinion rotational speed, rpm</td><td>n_1</td><td>965</td></tr>
<tr><td colspan="2">Transfer number</td><td>u</td><td>3,8</td></tr>
<tr><td colspan="2">A pinion number of teeth</td><td>z_1</td><td>22</td></tr>
<tr><td rowspan="2">Shift factor</td><td>pinion</td><td>x_1</td><td>0</td></tr>
<tr><td>gear</td><td>x_2</td><td>0</td></tr>
<tr><td colspan="2">Helix angle</td><td>β</td><td>0</td></tr>
<tr><td colspan="2">Degree of accuracy</td><td>–</td><td>7–B</td></tr>
<tr><td colspan="2">Teeth surface roughness, mkm</td><td>Ra</td><td>1,25</td></tr>
<tr><td rowspan="2">Steel</td><td>pinion</td><td>–</td><td rowspan="2">40XH</td></tr>
<tr><td>gear</td><td>–</td></tr>
<tr><td rowspan="2">Billet</td><td>pinion</td><td>–</td><td rowspan="2">Forging</td></tr>
<tr><td>gear</td><td>–</td></tr>
<tr><td rowspan="2">Teat treatment</td><td>pinion</td><td rowspan="2">–</td><td rowspan="2">Surface hardening</td></tr>
<tr><td>gear</td></tr>
<tr><td rowspan="2">A mean surface hardness</td><td>pinion</td><td>H_{O1}</td><td>50 HRC</td></tr>
<tr><td>gear</td><td>H_{O2}</td><td>45 HRC</td></tr>
</table>

Let us define a gear teeth number:

$$z_2 = z_1 u = 22 \cdot 3{,}8 = 83{,}6.$$

Rounding it to the nearest integer value (84), we will recalculate transfer number and we will calculate rotational speed of a gear:

$$u_{12} = \frac{z_2}{z_1} = \frac{84}{22} = 3{,}82, \quad n_2 = \frac{n_1}{u_{12}} = \frac{965}{3{,}82} = 254 \text{ rpm.}$$

Allowable stresses

Calculation of the allowable contact stresses is shown in Tab. 2.25 (see subsection 2.4.1).

Table 2.25

Parameter	Definition and value
The base number of cycles of stress $N_{H\,lim}$	$N_{Hlim1} = 30(HB_1)^{2,4} = 30(50/0{,}102)^{2,4} = 85{,}9 \cdot 10^6$ $N_{Hlim2} = 30(HB_2)^{2,4} = 30(45/0{,}102)^{2,4} = 66{,}7 \cdot 10^6$ (aproximation $H_{HB} = H_{HRC} / 0{,}102$ is used)
Number of meshing at one revolution c	According to gearing scheme $c_1 = c_2 = 1$
Stress cycle number during service life N_K	$N_{к1} = 60 n_1 c_1 L_h = 60 \cdot 965 \cdot 20\,000 = 1\,158 \cdot 10^6$ $N_{к2} = 60 n_2 c_2 L_h = 60 \cdot 253{,}9 \cdot 20\,000 = 304{,}7 \cdot 10^6$
Root index m	$N_{K1} > N_{H\,lim1}$ and $N_{K2} > N_{H\,lim2}$ that's why $m = 20$
Durability factor Z_N	$Z_{N1} = \sqrt[m]{\frac{N_{H\,lim\,1}}{N_{K1}}} = \sqrt[20]{\frac{85{,}9}{1158}} = 0{,}878$ $Z_{N2} = \sqrt[m]{\frac{N_{H\,lim\,2}}{N_{K2}}} = \sqrt[20]{\frac{66{,}7}{304{,}7}} = 0{,}927$
Contact endurance limit σ_{Hlim}, MPa	$\sigma_{Hlim1} = 17HRC_{Э1} + 200 = 17 \cdot 50 + 200 = 1050$ $\sigma_{Hlim2} = 17HRC_{Э2} + 200 = 17 \cdot 45 + 200 = 965$
Safety factor S_H	$S_H = 1{,}2$
Allowable contact stresses $[\sigma]_H$, MPa (see formula (2.1))	$[\sigma]_{H1} = \frac{1050 \cdot 0{,}878}{1{,}2} 0{,}9 = 692$ $[\sigma]_{H2} = \frac{965 \cdot 0{,}927}{1{,}2} 0{,}9 = 671$ Let us assume the smallest value ***671*** MPa

Calculation of allowable bend stresses is represented in Tab. 2.26 (see subsection 2.4.2) for pinion and gear material steel 40X.

Table 2.26

Parameter	Definition and value
Limit of bend endurance σ_{Flim}, MPa	$\sigma_{Flim1} = \sigma_{Flim2} = 580$ (see Tab. 2.15)
Safety factor S_F	$S_{F1} = S_{F2} = 1{,}7$ (see Tab. 2.15)
The base number of cycles of stress $N_{F\,lim}$	$N_{F\,lim} = 4 \cdot 10^6$
Durability factor Y_N	$N_{K1} > N_{K2} > N_{F\,lim}$ so $Y_N = 1$
The factor considering a method of billet production Y_z	For forging $Y_Z = 1$
Factor considering a roughness of interjacent surface Y_R	For $R_z = 1{,}25$ μm $Y_R = 1$
Allowable bend stresses $[\sigma]_F$, MPa	$[\sigma]_F = \dfrac{\sigma_{F\,lim}}{S_F} = \dfrac{580}{1{,}7} = 354$

Allowable **contact** stress at check of strength under the maximum load for the tempered on a contour gears

$$[\sigma]_{Hmax2} = 44\ HRC_{э2} = 44 \cdot 45 = 1980\ MPa.$$

Allowable stress for check **bending** strength under the maximum load we will define under the formula (2.4) and Tab. 2.16 for the given conditions (surface hardening on a contour, the alloyed steel with the nickel content less than 1 %):

$$[\sigma]_{F\,max} = \frac{\sigma_{FSt}}{S_{FSt}} = \frac{1800}{1{,}75} = 1068 \text{ MPa.}$$

Design calculation

Design calculation is shown in Tab. 2.27.

Table 2.27

Parameter	Definition and value
Torque on a pinion T_1, Nm	$T_1 = 9550\ \frac{5{,}5}{965} = 54{,}4$
Relative face width ψ_{bd}	$\psi_{bd} = 0{,}6$ (see Tab. 2.17)
Factor considering non-uniformity of load distribution on contact line $K_{H\beta}$	According to gearbox scheme and ψ_{bd} value $K_{H\beta} = 1{,}04$ (see Fig. 2.4)
Auxiliary factor K_d, MPa$^{1/3}$	For straight spur gear $K_d = 770$
A pinion pitch diameter d_{w1}, mm	$d_{w1} = K_d \sqrt[3]{\dfrac{T_1 K_{H\beta}}{\psi_{bd} [\sigma]_H^2} \dfrac{u+1}{u}} = 770 \sqrt[3]{\dfrac{54{,}4 \cdot 1{,}04}{0{,}6 \cdot 671^2} \dfrac{4{,}82}{3{,}82}} = 49{,}5$

End of Table 2.27

Parameter	Definition and value
Calculated module m, mm	$m = \frac{d_{w1}}{z_1} = \frac{49,5}{22} = 2,25$
Standard module, mm	$m = 2,5$ (see subsection 2.3.7)
Pitch diameters d_w, mm	$d_{w1} = m\, z_1 = 2,5 \cdot 22 = 55$ $d_{w2} = m\, z_2 = 2,5 \cdot 84 = 210$
Center distance a_w, mm	$a_w = \frac{m(z_1 + z_2)}{2} = \frac{2,5(22+84)}{2} = 132,50$

Check calculation

Check calculation on **contact endurance** is resulted in Tab. 2.28 (methods of parameters' definition are described in subsection 2.6.1).

Table 2.28

Parameter	Definition and value
Overlapping factor ε_α	$\varepsilon_\alpha = 1,88 - 3,2\left(\frac{1}{z_1} + \frac{1}{z_2}\right) = 1,88 - 3,2\left(\frac{1}{24} + \frac{1}{91}\right) = 1,71$
The factor considering total length of contact lines Z_ε	For straight teeth ($\beta = 0^\circ$) $Z_\varepsilon = \sqrt{(4-\varepsilon_\alpha)/3} = \sqrt{(4-1,711)/3} = 0,87$
The factor considering mechanical properties of materials of interfaced wheels Z_E, MPa$^{-0,5}$	For steel gears $Z_E = 190$
The factor considering the form of teeth interfaced surfaces in a pole of gearing Z_H	When $\alpha_t = \alpha_{tw} = 20°$ and without shift $Z_H = \frac{1}{\cos\alpha_t}\sqrt{\frac{2}{tg\alpha_{tw}}} = \frac{1}{\cos 20^\circ}\sqrt{\frac{2}{tg 20^\circ}} = 2,49$
Tangentional force on pitch cylinder F_t, N	$F_t = \frac{2000\ T_1}{d_{w1}} = \frac{2000 \cdot 54,4}{55} = 1\ 979$
Peripheral speed v, m/s	$v = \frac{\pi d_{w1} n_1}{60\ 000} = \frac{3,14 \cdot 55 \cdot 965}{60000} = 2,79$
Face width b_w, mm	$b_w = \psi_{bd} d_{w1} = 0,6 \cdot 55 = 33$

End of Tab. 2.28

Parameter	Definition and value
External dynamic factor K_{HA}	For uniform operating condition K_{HA} = 1 (see subsection 2.3.5)
Check on closeness to resonance	$vz_1 = 2{,}79 \cdot 22 \approx 61 < 1000$ – – before resonance zone
Factor considering influence of a profile tip relief and a kind of teeth δ_H (see Tab. 2.11)	For straight teeth with surface hardness more than 350 HV and without a profile tip relief $\delta_H = 0{,}14$
Factor considering influence of a pitch difference of teeth of a pinion and a wheel g_0 (see Tab. 2.12)	For 7 degree of accuracy and $m = 2{,}5$ $g_0 = 4{,}7$
Specific dynamic load w_{Hv}, N/mm (see Tab. 2.12)	$w_{Hv} = \delta_H g_0 v \sqrt{\frac{a_w}{u}} = 0{,}14 \cdot 4{,}7 \cdot 2{,}79 \sqrt{\frac{132{,}5}{3{,}82}} = 10{,}82$ $10{,}82 < w_{vmax} = 240$
Internal dynamic factor K_{Hv}	$K_{Hv} = 1 + \frac{w_{Hv} b_w}{F_t K_{HA}} = 1 + \frac{10{,}82 \cdot 33}{1979 \cdot 1} = 1{,}18$
Load factor K_H	$K_H = K_{Hv} K_{H\beta} = 1{,}18 \cdot 1{,}04 = 1{,}23$
Contact stress σ_H, MPa	$\sigma_H = 190 \cdot 2.49 \cdot 0{,}87 \sqrt{\frac{1979 \cdot 1{,}23}{33 \cdot 55} \frac{4{,}82}{3{,}82}} = 542$
Relative stress ε	$\varepsilon = \frac{\sigma_H - [\sigma]_H}{[\sigma]_H} = \frac{542 - 671}{671} \cdot 100\,\% = -19{,}2\,\%$, i. e. underloading

Let us reduce face width to 22 mm ($\psi_{bd} = 0{,}4$, that is admissible, see Tab. 2.17) and recalculate

$$K_{Hv} = 1 + \frac{w_{Hv} b_w}{F_t K_{HA}} = 1 + \frac{10{,}82 \cdot 22}{1\,979 \cdot 1} = 1{,}12$$

and load factor

$$K_H = K_{HV} K_{H\beta} = 1{,}12 \cdot 1{,}04 = 1{,}17.$$

Then contact stress in a gearing pole

$$\sigma_H = 190 \cdot 2.49 \cdot 0{,}87 \sqrt{\frac{1979 \cdot 1{,}17}{22 \cdot 55} \frac{4{,}82}{3{,}82}} = 647 \text{ MPa}$$

and a relative deviation of stress

$$\varepsilon = \frac{\sigma_H - [\sigma]_H}{[\sigma]_H} = \frac{647 - 671}{671} \cdot 100\% = -3{,}6\ \%,$$ that is admissible.

Calculation of teeth on **bending endurance** is shown in Tab. 2.29.

Table 2.29

Parameter	Definition and value
Factor, considering the tooth form and stress concentration Y_{FS}	In case of absence of shift for straight gears $z_{v1,2} = z_{1,2}$ $Y_{FS1} = 3{,}47 + 13{,}2 / z_1 = 3{,}47 + 13{,}2 / 22 = 4{,}07$ $Y_{FS2} = 3{,}47 + 13{,}2 / z_2 = 3{,}47 + 13{,}2 / 84 = 3{,}63$
The factor considering a tooth helix angle Y_β	For straight teeth ($\beta = 0^\circ$) $Y_\beta = 1 - \varepsilon_\beta \dfrac{\beta}{120} = 1$
Factor considering teeth overlapping Y_ε	For straight teeth $Y_\varepsilon = 1$
Load factor K_F	$K_F = K_H = 1{,}16$
Local bending stress σ_F, MPa	$\sigma_{F1} = \dfrac{F_t}{b_w m} K_F Y_{FS1} = \dfrac{1\,979}{22 \cdot 2{,}5} 1{,}17 \cdot 4{,}07 = 170 \le [\sigma]_{F2} = 354$ $\sigma_{F2} = \dfrac{F_t}{b_w m} K_F Y_{FS2} = \dfrac{1\,979}{22 \cdot 2{,}5} 1{,}17 \cdot 3{,}63 = 152 \le [\sigma]_{F2} = 354$

Let us check up gearing on **strength at action of the maximum loading**:

– define actual contact stress (see subsection 2.6.1.3) and compare it with the allowable one:

$$\sigma_{H\,max} = \sigma_H \sqrt{T_{max} / T_{ном}} = 647\sqrt{2{,}2} = 959 < [\sigma]_{H\,max} = 1980 \text{ MPa;}$$

– calculate actual bending stress separately for a pinion and a gear and compare them with corresponding allowable stress (see subsection 2.6.2.3):

$$\sigma_{F\,max\,1} = \sigma_{F1}(T_{max} / T_{ном}) = 170 \cdot 2{,}2 = 375 < [\sigma]_{F\,max} = 1068 \text{ MPa,}$$

$$\sigma_{F\,max\,2} = \sigma_{F2}(T_{max} / T_{ном}) = 152 \cdot 2{,}2 = 334 < [\sigma]_{F\,max\,2} = 1068 \text{ MPa.}$$

A pinion and a gear dimensions

Calculation of a pinion and a gear dimensions see in Tab. 2.30.

Table 2.30

Parameter	Definition and value, mm
Outside diameter d_a	$d_{a_1} = d_{w_1} + 2m = 55 + 2 \cdot 2{,}5 = 60$ $d_{a_2} = d_{w_2} + 2m = 210 + 2 \cdot 2{,}5 = 215$
Root diameter d_f	$d_{f_1} = d_{w_1} - 2{,}5m = 55 - 2{,}5 \cdot 2{,}5 = 48{,}750$ $d_{f_2} = d_{w_2} - 2{,}5m = 210 - 2{,}5 \cdot 2{,}5 = 203{,}750$
A gear face width b_{w_2}	$b_{w_2} = b_w = 22$

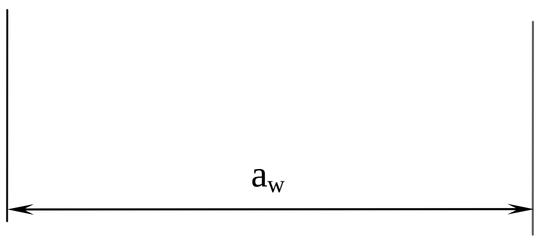
End of Table 2.30

Parameter	Definition and value, mm
Pinion face width b_{w_1}	$b_{w_1} = b_w + (1...2)m = 22 + 2 \cdot 2,5 = 27$
Torque on a gear T_2, Nm	$T_{out} = T_1 i\ \eta_o = 54,4 \cdot 3,8 \cdot 0,94 = 194,4$
Shaft hole diameter	$d = \sqrt[3]{\dfrac{1000T_2}{0,2[\tau]}} = \sqrt[3]{\dfrac{1000 \cdot 194,4}{0,2 \cdot 50}} = 26,88 \approx 28$
Hub diameter	$(1,5...1,7)\ d = 1,5 \cdot 28 = 42$
Hub length	$L = (0,7...1,8)\ d = 1,2 \cdot 28 = 33,6 \approx 34$
Rim thickness	$(2,5...4)\ m = 4 \cdot 2 = 8$, increased to 10
Disk thickness	$(0,2...0,3)\ b = 0,3 \cdot 28 \approx 10$
Rim diameter	$D_1 = d_{f2} - 2\delta_1 = 177 - 2 \cdot 10 = 157$
The holes' diameter	$d_1 = (0,25...0,3)(D_1 - D) = 0,25\ (157 - 42) \approx 30$
Chamfer	$0,5\ m = 0,5 \cdot 2,5 = 1,25$

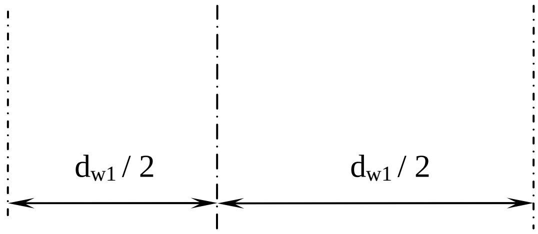
Order of spur gearing design

To design spur gearing, it is necessary to know:
b – face width of a gear;
a_w – stage center distance;
d_{w1}, d_{w2} – pinion and gear pitch diameters (according to the strength calculation);
d_{a1}, d_{a2} – pinion and gear addendum diameters (for straight teeth transfer without shift $d_a = d_w + 2\ m$);
d_{f1}, d_{f2} – pinion and gear root diameters (for straight teeth transfer without shift $d_f = d_w - 2,5\ m$).

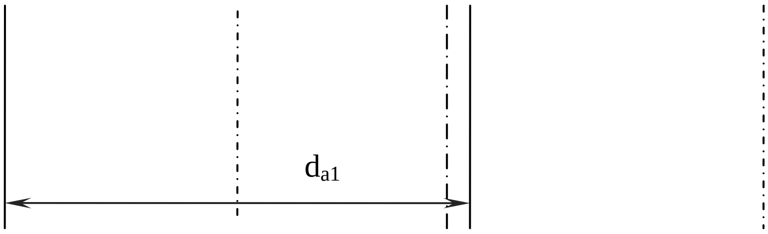
1. Draw two parallel straight lines at the center distance a_w (blue line is a pinion axis, green line is a gear one):



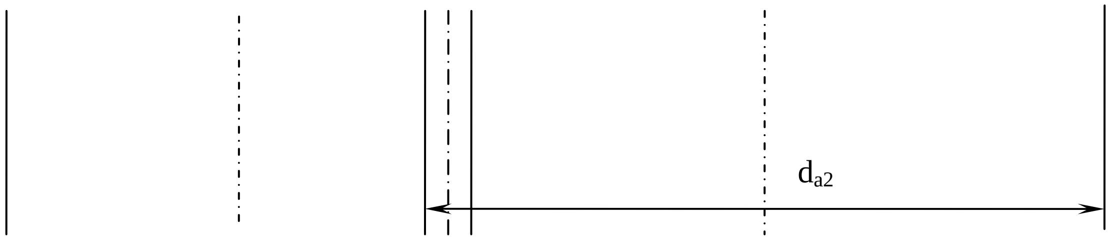
2. Draw a parallel line between axis lines at the distance $\boldsymbol{d_{w1} / 2}$ from the pinion axis or $\boldsymbol{d_{w2} / 2}$ from the gear one:



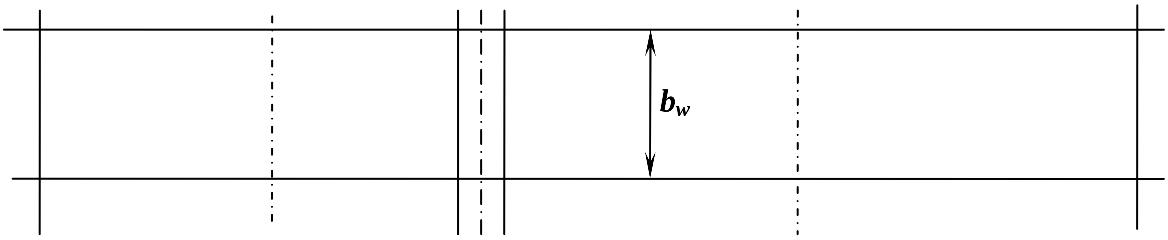
3. Draw two parallel lines at the distance $\boldsymbol{d_{a1}}$ simmetrically about the pinion axis:



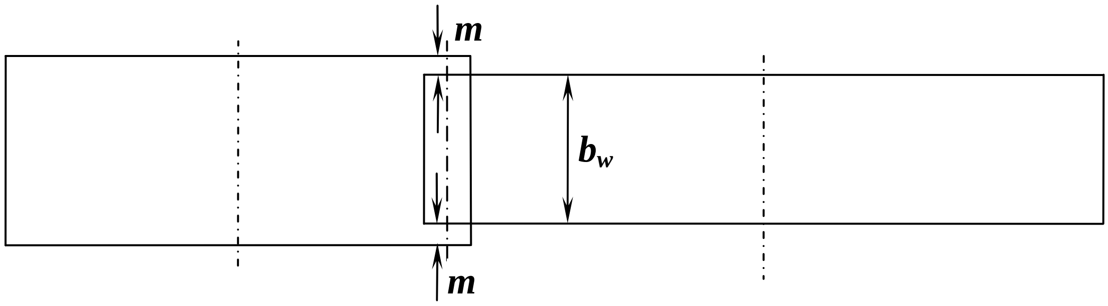
4. Draw two parallel lines at the distance $\boldsymbol{d_{a2}}$ simmetrically about the gear axis:



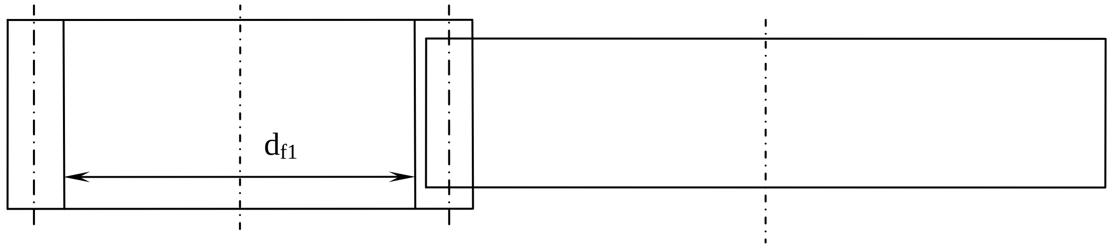
5. Draw two parallel straight lines perpendicularly to the existing lines at the distance $\boldsymbol{b_w}$:



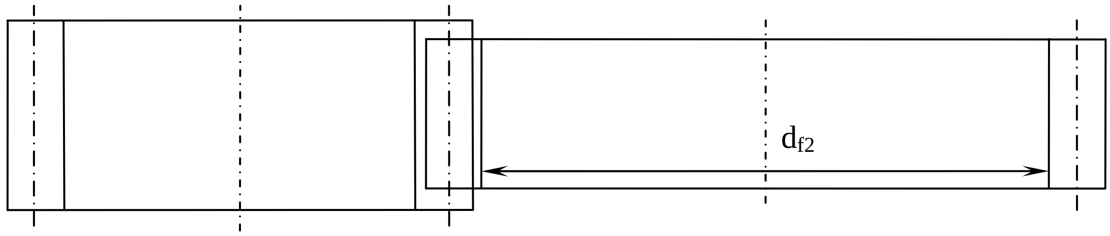
6. Generate both gear and pinion contoures (it is recommended to increase pinion width relatively gear with one module value on each side):



7. Draw two parallel lines at the distance d_{f1} simmetrically about the pinion axis:



8. Draw two parallel lines at the distance d_{f2} simmetrically about the gear axis:



9. Design the wheels like in Fig. 2.9 (see parameters in Tab. 2.30):

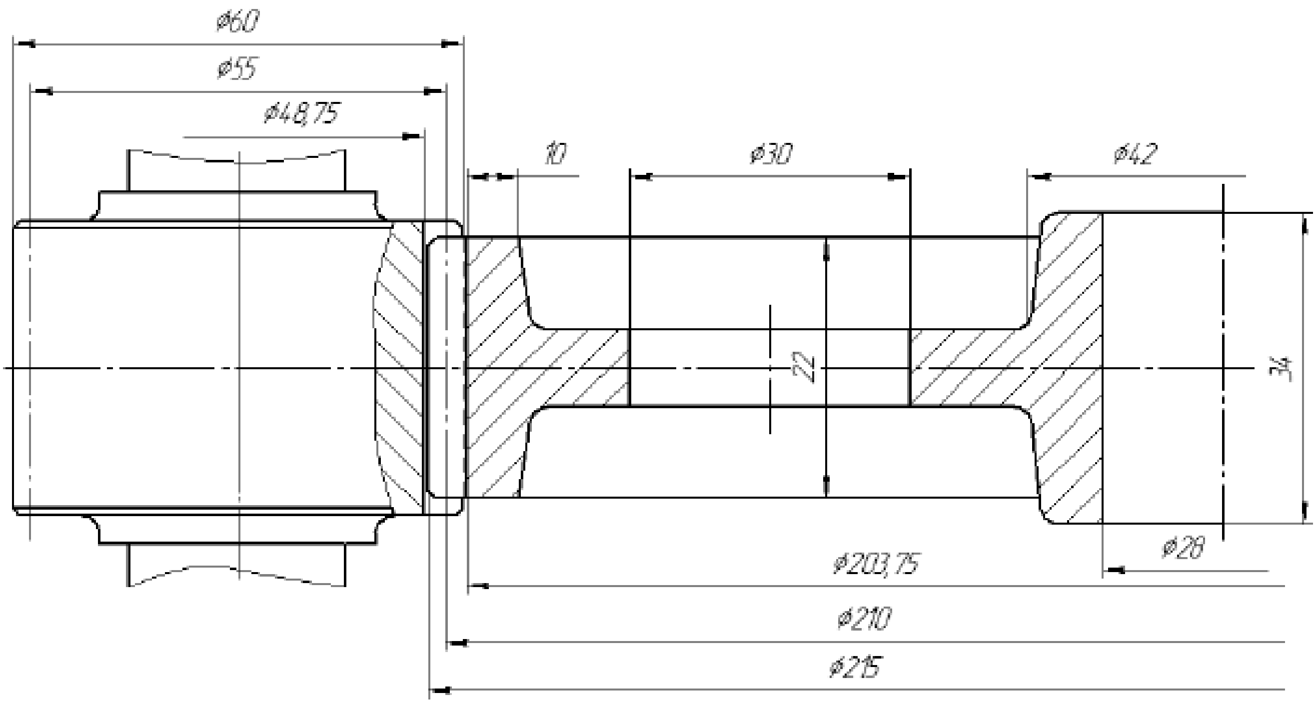
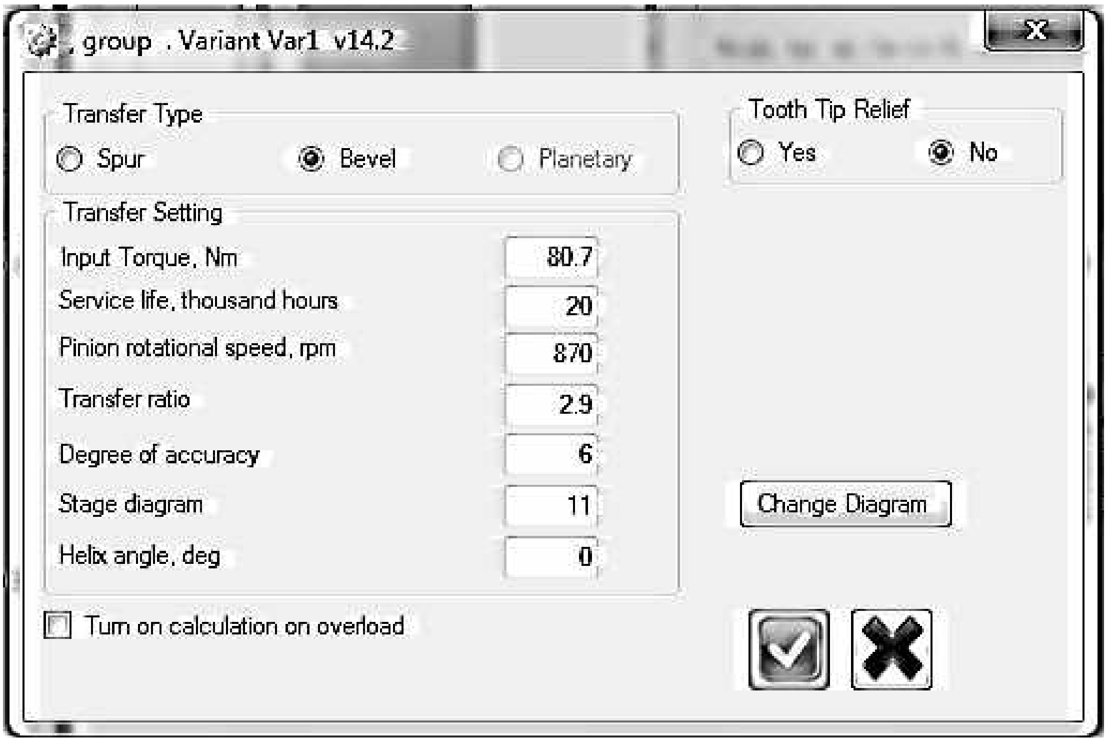


Fig. 2.9

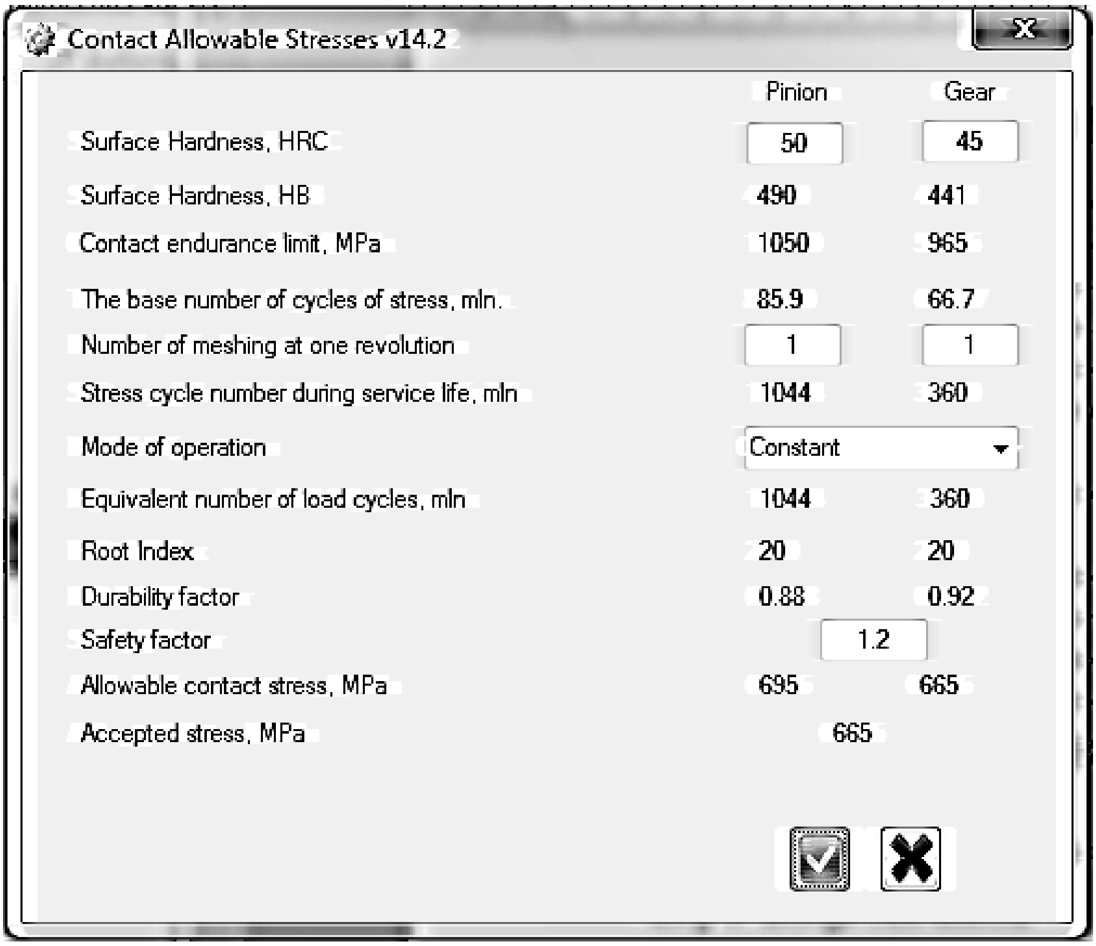
Example 2

Calculation of bevel gearing by means of a program is described below.

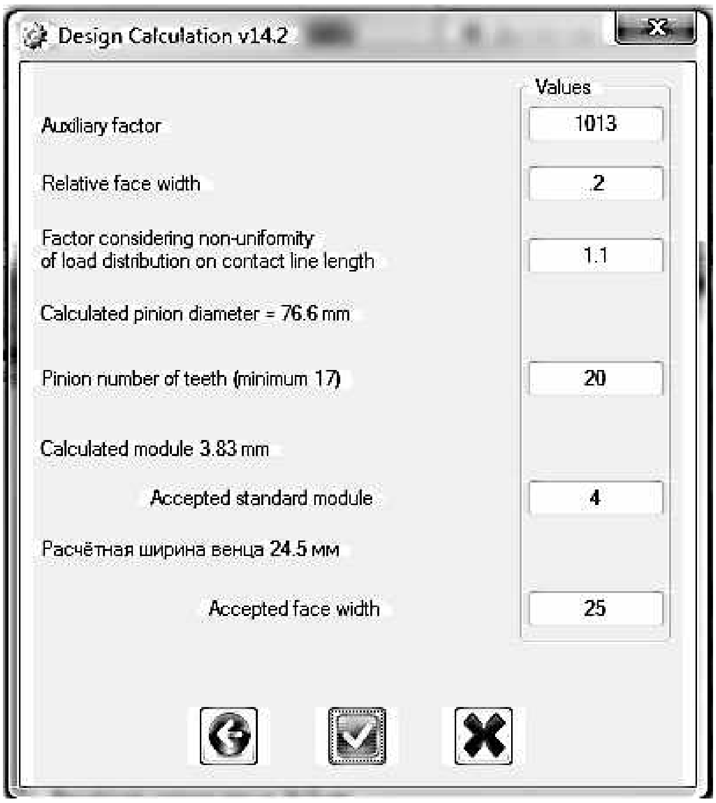
1. Fulfill the first form with initial data (see subsection 1.4, Example 1) after authorization:



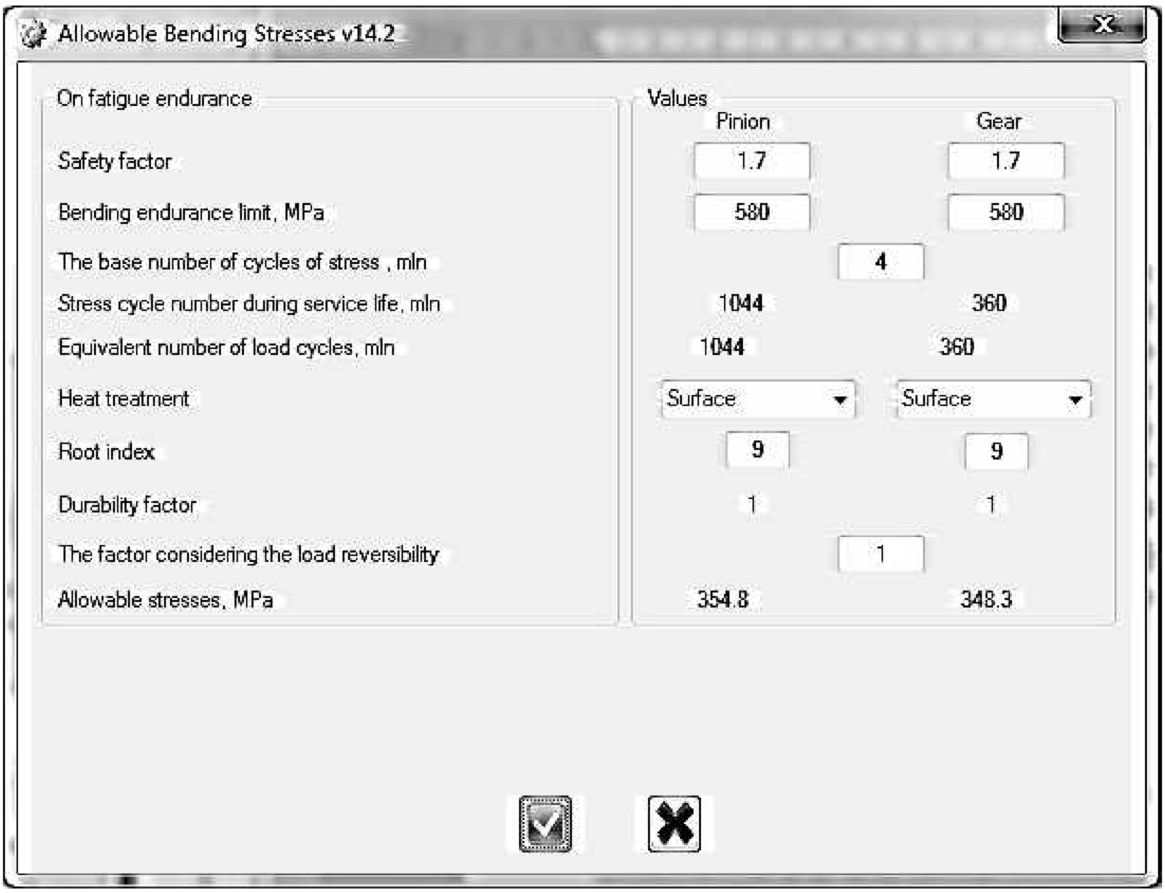
2. Fill allowable contact stresses form:



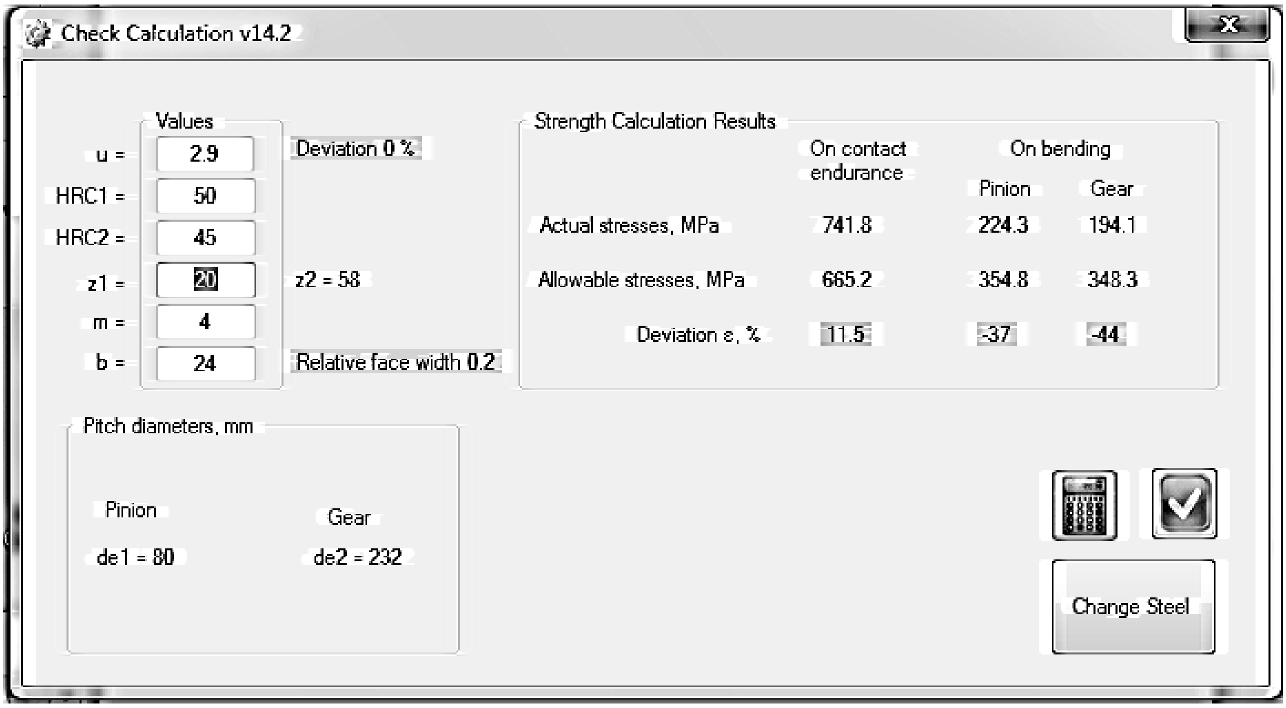
3. Fill design calculation form:



4. Fill allowable bending stresses form:

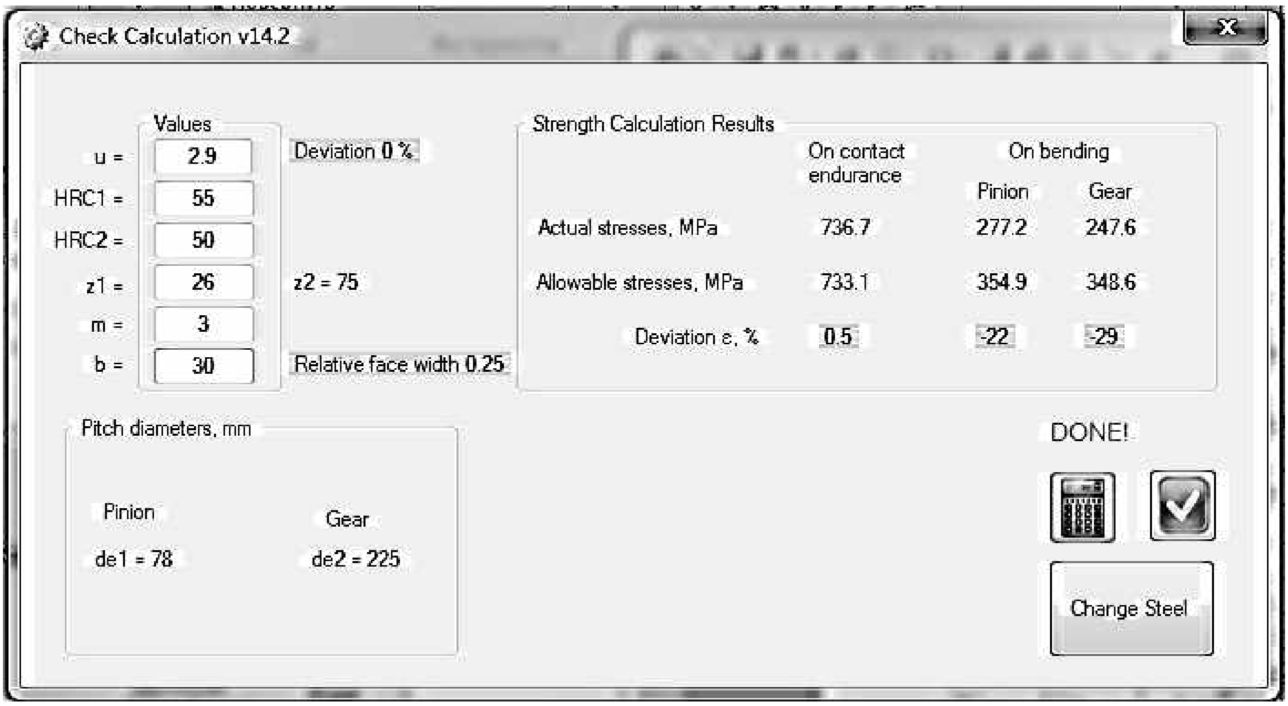


5. Check results of the verify calculation:



As the first result shows, there is overload on contact endurance (red colour cell, 11,5 %). To avoid overload some parameters were changed: pinion hardness increase from 50 to 55 HRC; gear hardness increase from 45 to 50 HRC; number of teeth increases from 20 to 26; face width increases from 25 to 30 mm (relative face width less 0,3); the module decrease from 4 to 3.

Finally, deviation on contact endurance is about zero and less than zero on bending endurance.



Results are shown in the table below:

Pinion pitch diameter, mm	75,7
Pinion number of teeth z_1	20
Module m, mm	3,78
Calculated face width b_w, mm	25,3

Internal dynamic load factor K_{Hv}	1,21
Load distribution non-uniformity factor $K_{H\beta}$	1,36
Load factor K_H	1,67
Actual contact stress σ_H, MPa	737
Relative contact stress ε	**0,5**
Actual peak contact stress σ_{Hmax}, MPa	

N_F	0,88	
Load distribution non-uniformity factor $K_{F\beta}$	1,35	
Load factor K_F	1,66	
Actual bending stress σ_F, MPa	277	248
Relative bending stress ε	**-21,9**	**-29,0**
Actual peak bending stress σ_{Fmax}, MPa		

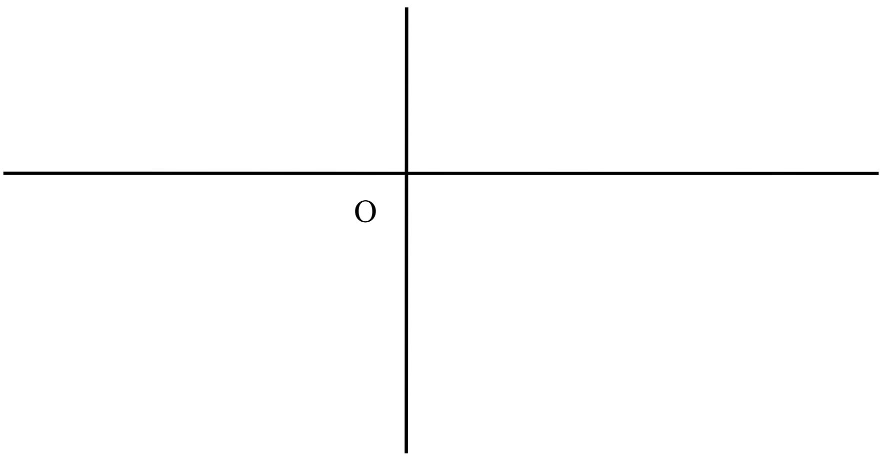
Final Parameters

Bevel			pinion	gear
Transfer number u	2,90	Number of teeth z	26	75
Standard module m, mm	3,00	Rotational speed n, rpm	870	300
Cone distance R_e, mm	119,068	Addendum diam., mm	83,669	226,965
Face width b, mm	30,0	Pitch diam., mm	78,00	225,00
Relative face width	0,25	Root diam., mm	71,197	222,642
b/m	10,00	Mean pitch diam., mm	68,174	196,655
□		Pitch cone angle, deg	19,12	70,88

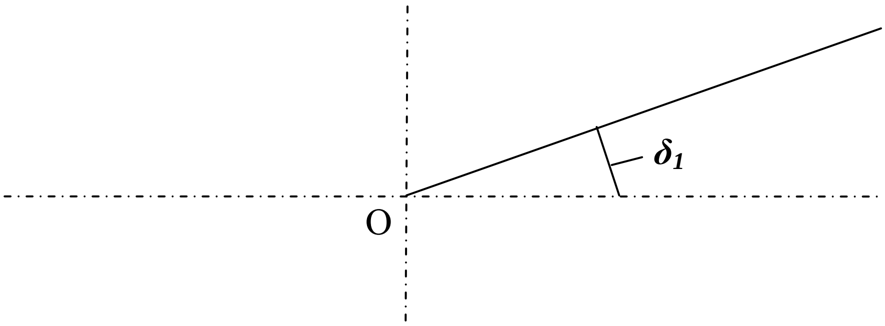
Order of bevel gearing design

To design bevel gearing, it is necessary to know (see in table above):
$\boldsymbol{R_e}$ – cone distance;
b – face width of a gear;
$\boldsymbol{\delta_1, \delta_2}$ – pinion and gear pitch cone angles;
$\boldsymbol{d_{ae1}, d_{ae2}}$ − pinion and gear addendum diameters at an external face (for straight teeth orthogonal transfer without shift $\boldsymbol{d_{ae} = d_e + 2\, m \cos \delta}$);
$\boldsymbol{d_{fe1}, d_{fe2}}$ − pinion and gear root diameters at an external face (for straight teeth orthogonal transfer without shift $\boldsymbol{d_{fe} = d_e - 2{,}4\, m \cos \delta}$).

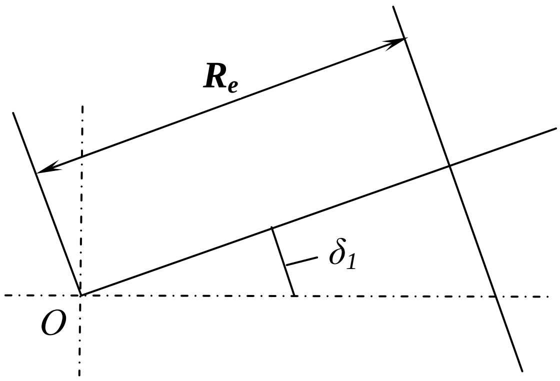
1. Draw two straight lines under an interaxal angle of transfer (for orthogonal transfer − 90°). Let the horizontal line will be a pinion axis, vertical – a gear axis:



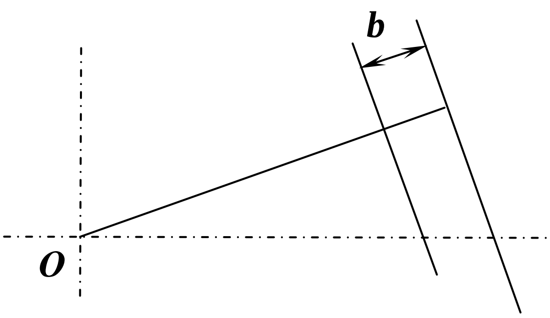
2. Draw a semi-line from a point O under an angle δ_1 to a pinion axis:



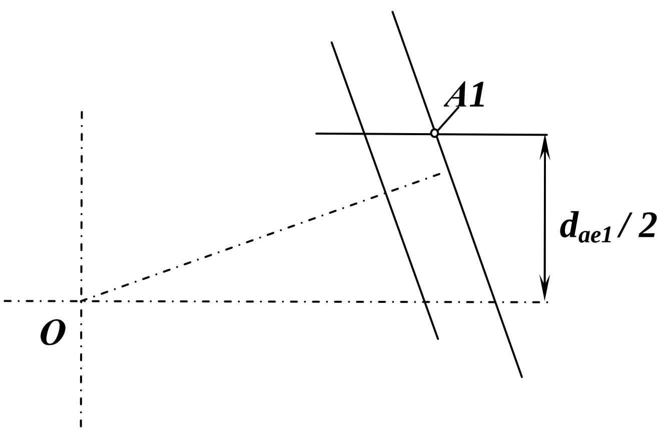
3. Draw a straight line perpendicularly to the semi-line on distance R_e from the center:



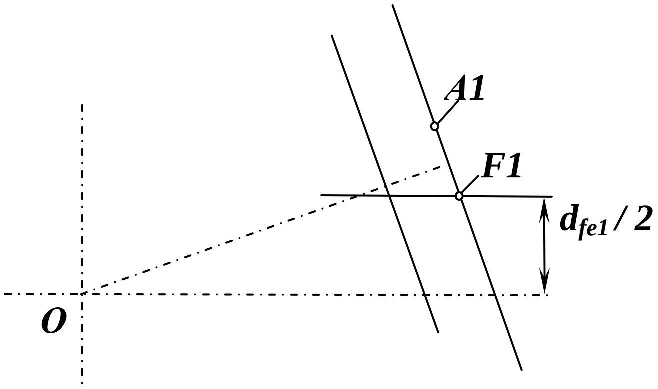
4. Draw a parallel straight line on distance b closer to centre O:



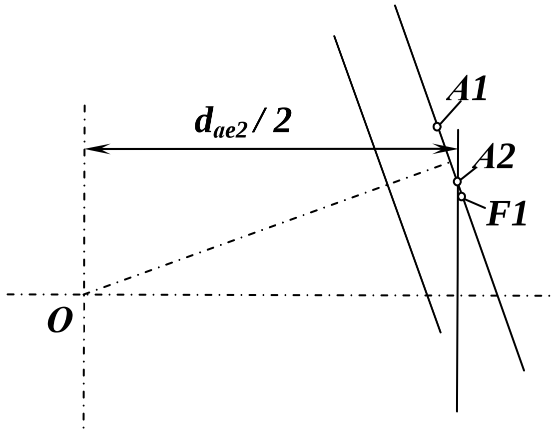
5. Draw a parallel to a pinion axis on distance $d_{ae1}/2$ (point $A1$):



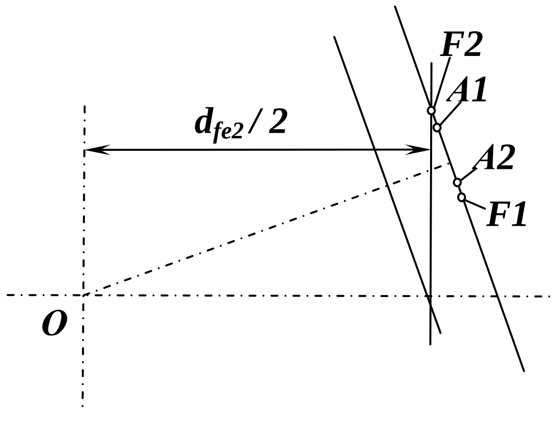
6\. Draw a parallel to a pinion axis on distance $\boldsymbol{d_{fe1}/2}$ (point ***F1***):



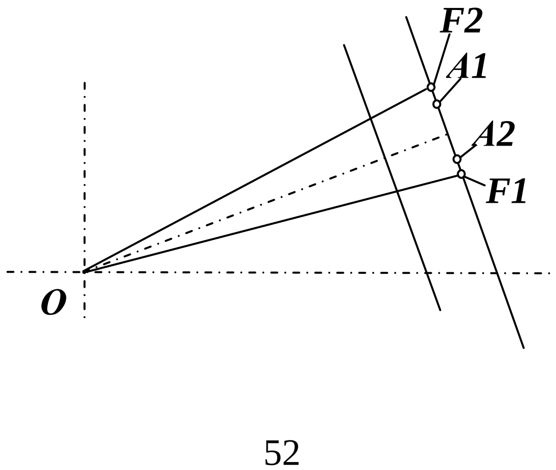
7\. Draw a parallel to a gear axis on distance $\boldsymbol{d_{ae2}/2}$ (point ***A2***):



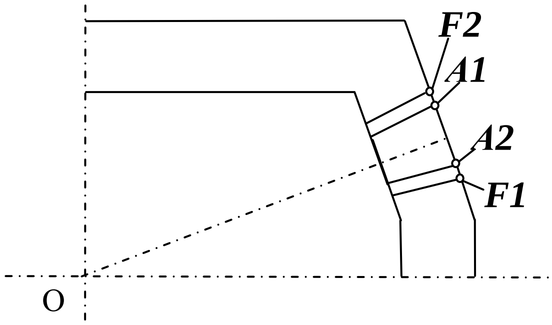
8\. Draw a parallel to a gear axis on distance $\boldsymbol{d_{fe2}/2}$ (point ***F2***):



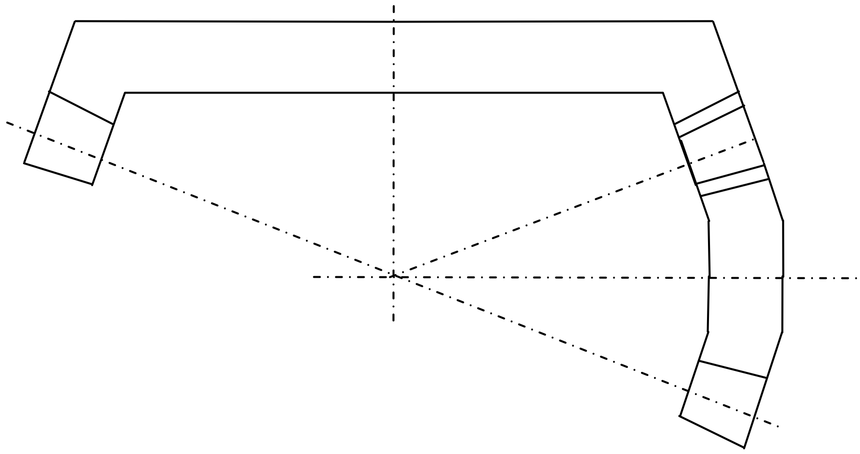
9\. Connect the points ***F1*** and ***F2*** to the centre ***O***:



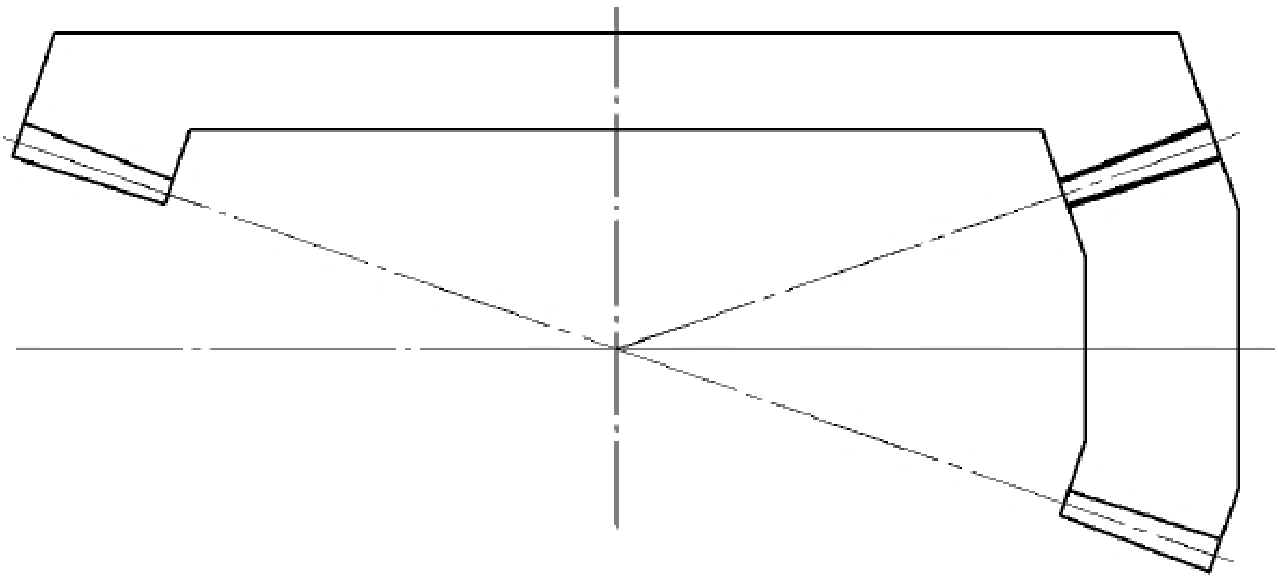
10. Draw lines parallel to ***OF1*** and ***OF2*** from points ***A2*** and ***A1*** correspondingly:



11. The generated profiles of wheels mirror to display concerning their axes:



Draw a sketch of gearing calculated above according to the order:



Than, we can design the gears' disks and hubs. Additional parameters are shown in Tab. 2.31, a sketch according to these sizes – in Fig. 2.10.

Table 2.31

Parameter	**Formula**	**Value, mm**	
		pinion	**gear**
Torque, Nm	See subsection 1.4, Example 2	80,7	220
Allowable stress ***[τ]***, MPa	See Tab. 2.22	50	
A shaft diameter	$d = \sqrt[3]{\dfrac{1000T}{0{,}2[\tau]}}$	20,1	28
A hub diameter	***(1,5...1,7) d = 1,5 · 28***	***X***	42
A hub length	***(0,7...1,8) d = 1,2 · 28***	***X***	34
A rim thickness	***(2,5...4) m = 3 · 3 ≈***	***X***	10
A disk thickness	***(0,2...0,3) b = 0,3 · 30 ≈***	***X***	10
A rim diameter	geometrically	***X***	215
The holes' diameter	from design reasons	***X***	32
Chamfer	***0,5 m***	1,5	

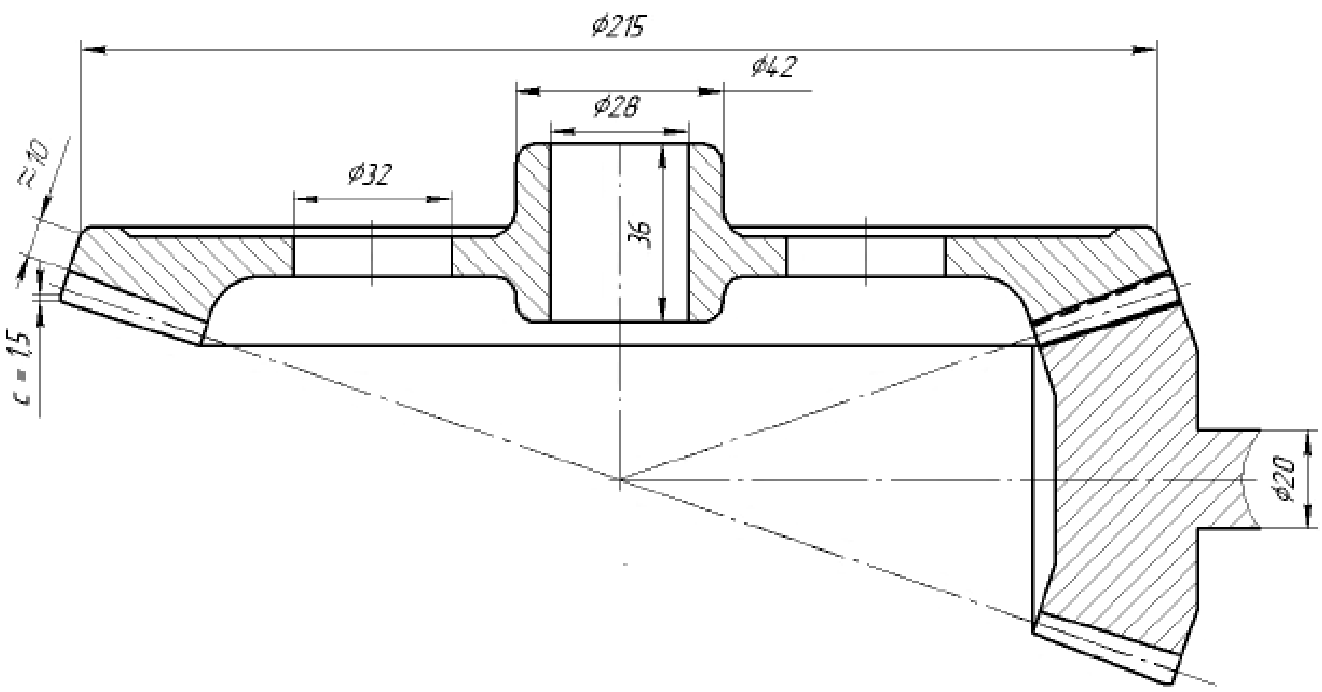


Fig. 2.10

3. SHAFTS CALCULATION AND DESIGN

Shaft is an element intended for transfer of the rotating moment along the axis and for maintenance of rotating elements. It is also subjected to action of cross-section forces and the bending moments.

Calculation on durability of shaft is realized in two stages: design and check. During design calculation, a shaft material is chosen taking into account loads and rotational speed and than shaft dimensions are defined. After that a shaft (or an axle) design is developed. Demanded verify calculation is carried out for the chosen design.

Depending on load character and operational requirements a shaft is checked on durability under the influence of short-term overloads, on fatigue endurance, on rigidity, on twisting oscillations, etc.

3.1. Design Calculation and Shaft Designing Rules

Before design calculation power torque ***T***, Nm, and rotational speed ***n***, rpm, are usually known.

To define a preliminary shaft diameter ***d***, mm, only torsion calculation (without accounting bending stresses) with lowered allowable stress is used:

$$d = \sqrt[3]{\frac{1000T}{0{,}2(1-\alpha^4)[\tau_t]}}, \tag{3.1}$$

where $[\tau_t]$ is conditional allowable torsion stress, MPa, which is usually accepted 20...50 MPa for shafts of the general mechanical engineering gearboxes and 70...90 MPa – for shafts of aviation ones.

After preliminary definition of a shaft diameter it should be rounded to standard value.

Then shaft should be designed: surfaces under hub, necks for bearings, thrust fillets, key ways, turnings for an exit of the tool, etc. should be formed.

Thus it is necessary to keep of the following **design rules**:

1. *The shaft form should be approached to the form of a body of equal resistance to a bending.* It means that the greatest diameter should be in those places where the maximum bending and twisting moments operate. As a rule, it is somewhere in the middle of a shaft. In other words, diameter of a shaft should decrease stepwise from the middle to the ends (see Fig. 3.1) that allows satisfying the second rule also.

2. *Each element should pass to its seat without tightness.* For this purpose it is necessary to provide a separate diameter for each element (gear, bearings etc.).

If a key is provided on the shaft end, it is desirable that the elements mounted more deeply on a shaft pass 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 (one point does not provide fixing, and at three and more systems become statically indefinable; of course, long multi-support shafts exist, but are not

considered here). It means that two radial bearings, taking both radial and axial forces and a combination radial with thrust or radial-thrust beating should be mounted.

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 under a circlip on a shaft.

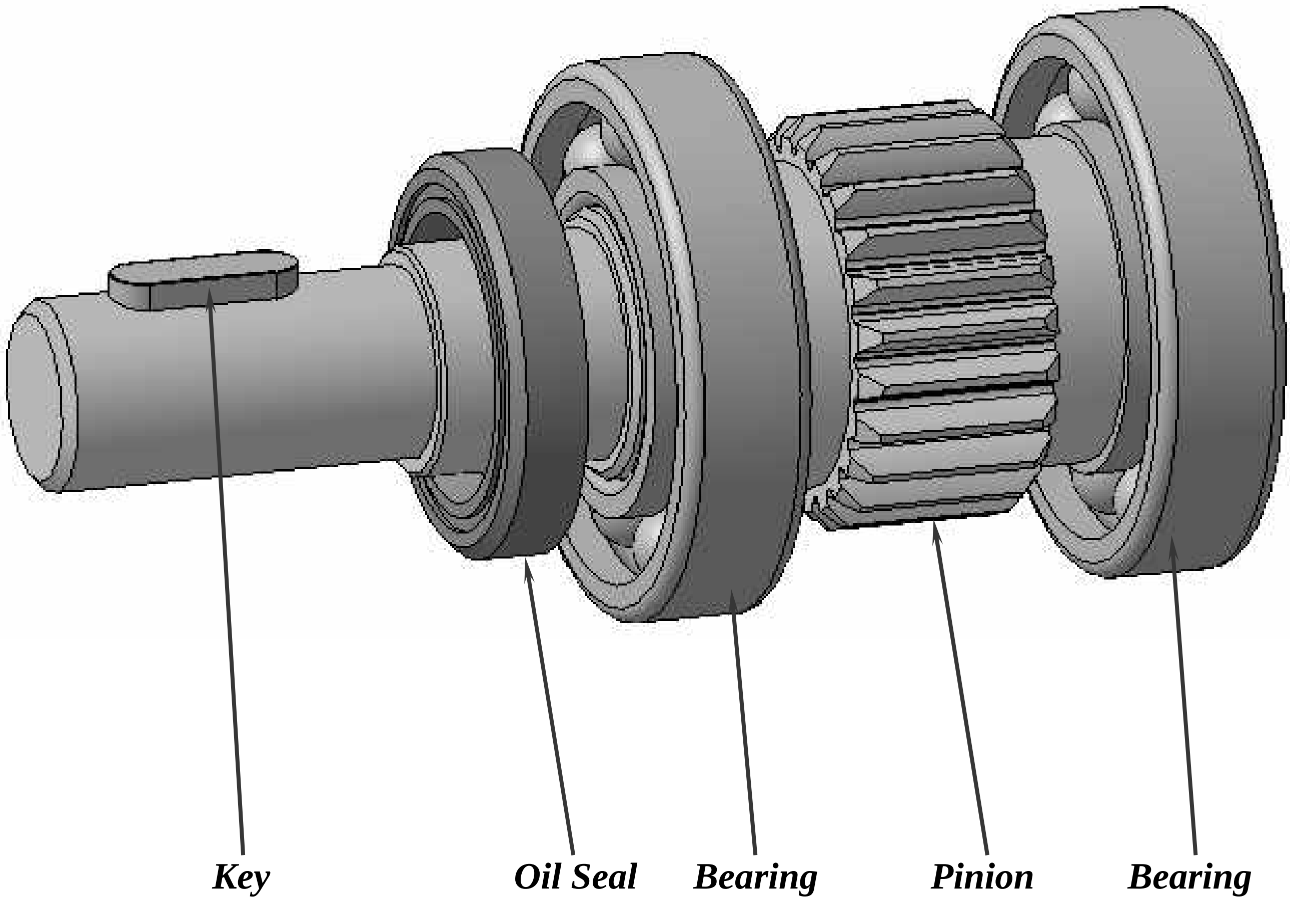
4. *The length of a neck of a shaft should be corresponded to the length of a contact surface of an element.* Moreover, its length can be decreased on chamfer or rounding length.

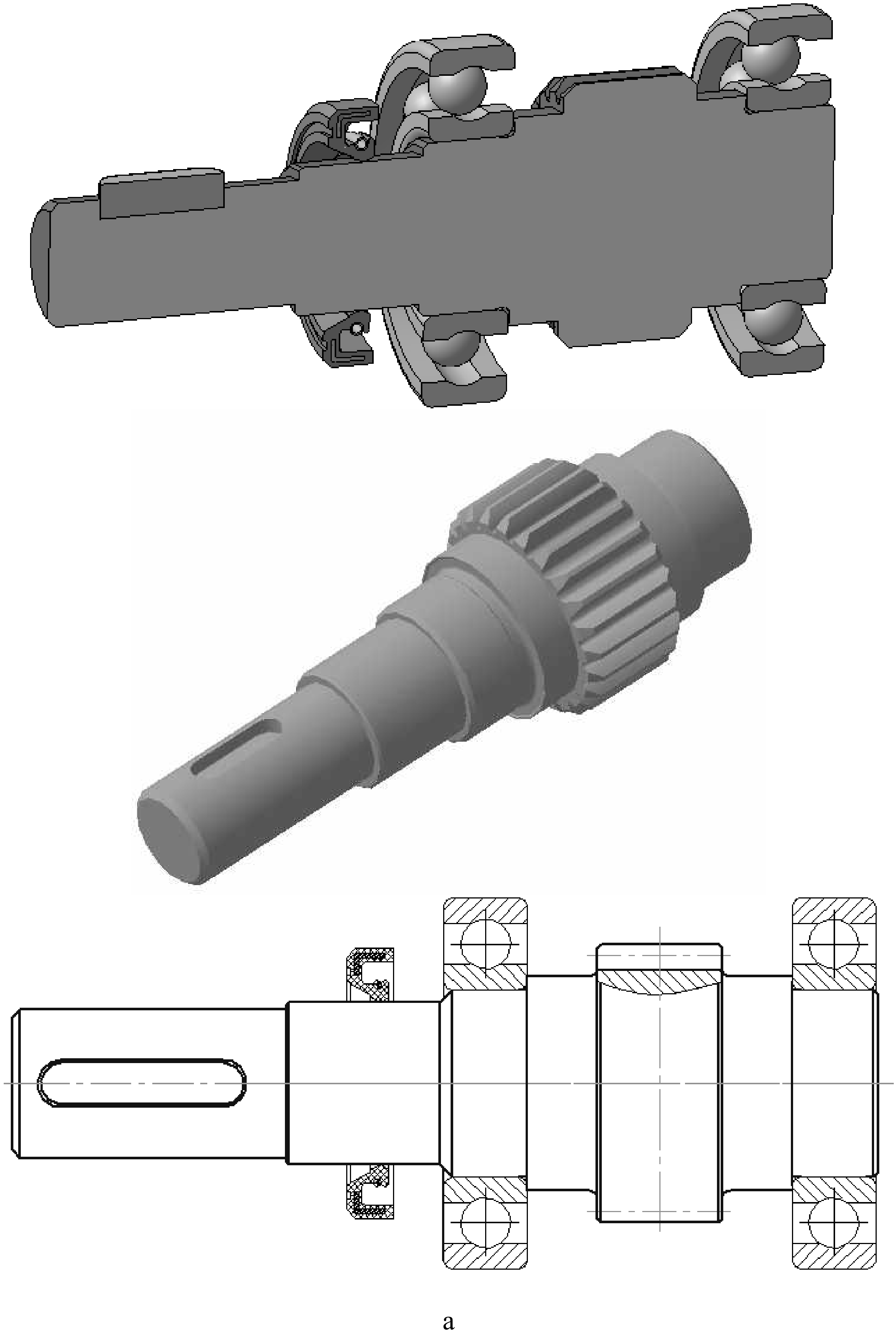
5. *Make chamfers* on all connected surfaces (under bearings, wheels, couplings, etc.), because an element assembling is rather complicated without chamfer. The recommended chamfer sizes are:

Shaft diameter d, mm	16…20	20…30	30…50	50…100
Chamfer, mm	1	1,5	2	3

6. *Diameters of the input/output shaft end should be corresponded with diameters of attached units* (for example, with standard coupling diameter). If these sizes strongly differ, it is necessary either to change shaft diameter, or to use transitive bushes.

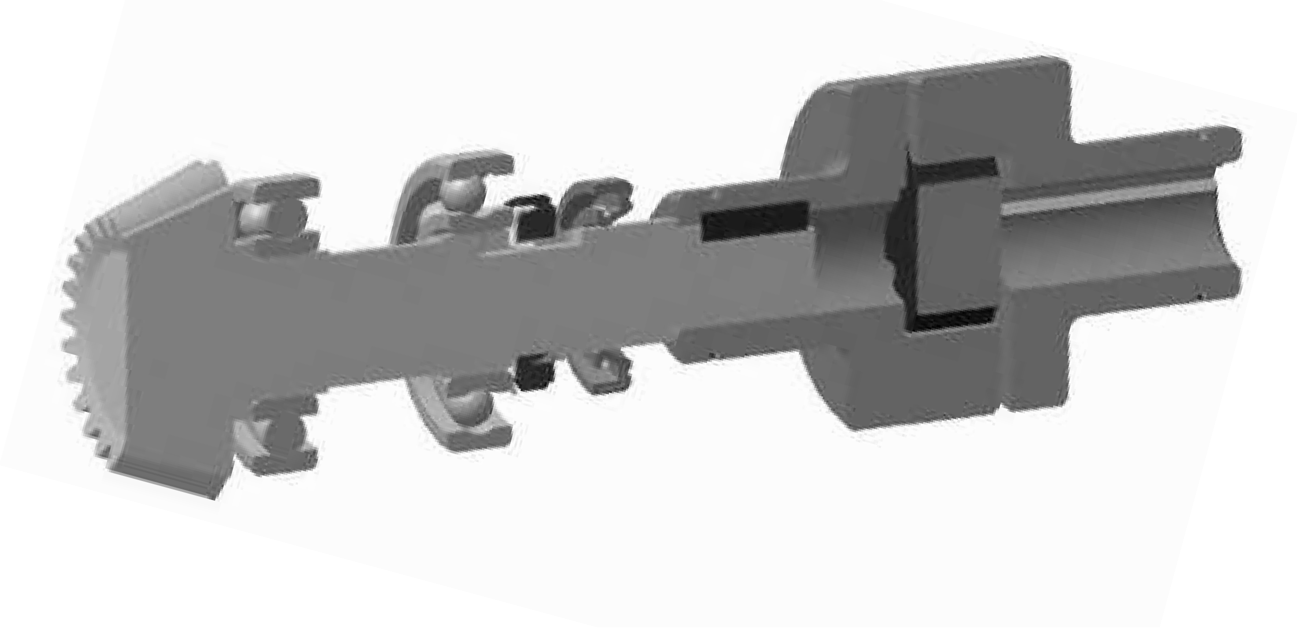
According to these rules a spur gearbox input shaft, for example, should look like in Fig. 3.1, a, for bevel gearbox – like in Fig. 3.1, b, an output shaft for both spur or bevel gearbox – like in Fig. 3.2. Therefore, elements connected to each shaft (a coupling, an oil seal, a bearing) should be selected to design the shaft surfaces.

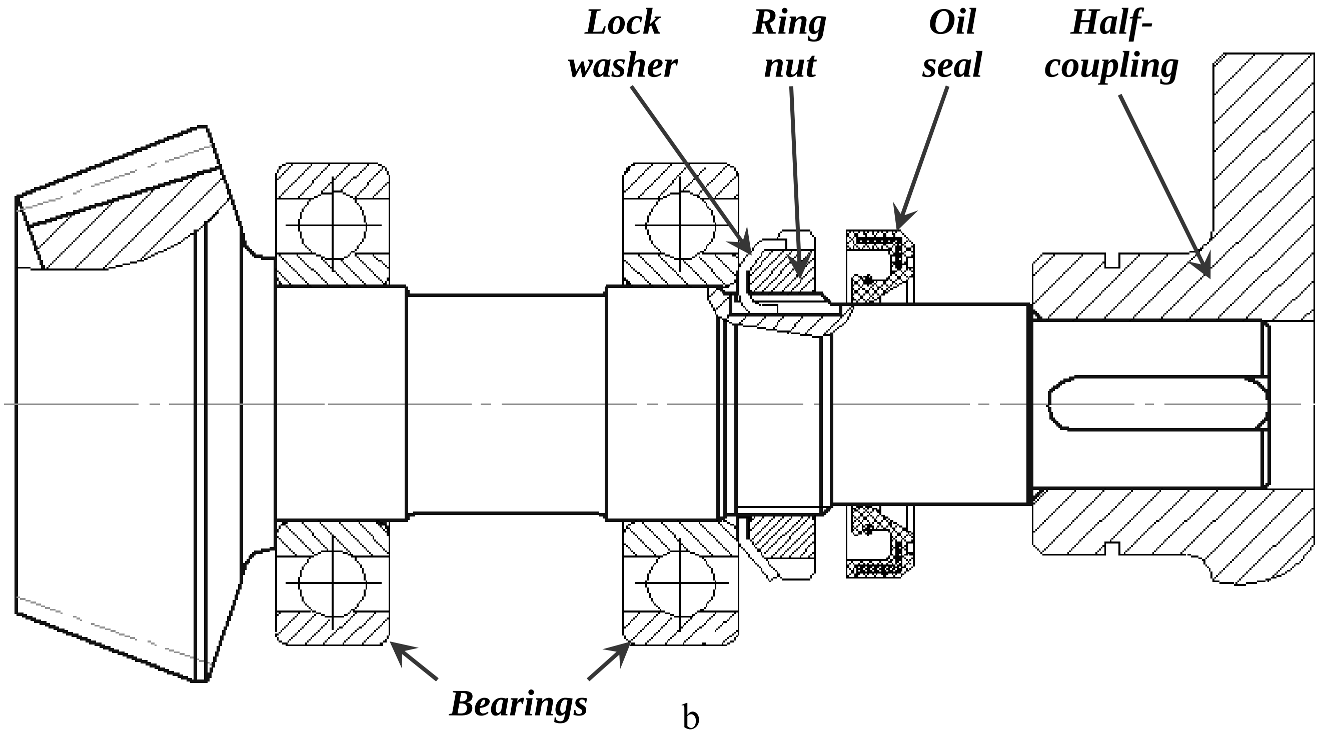




a

Fig. 3.1





b

End of Fig. 3.1

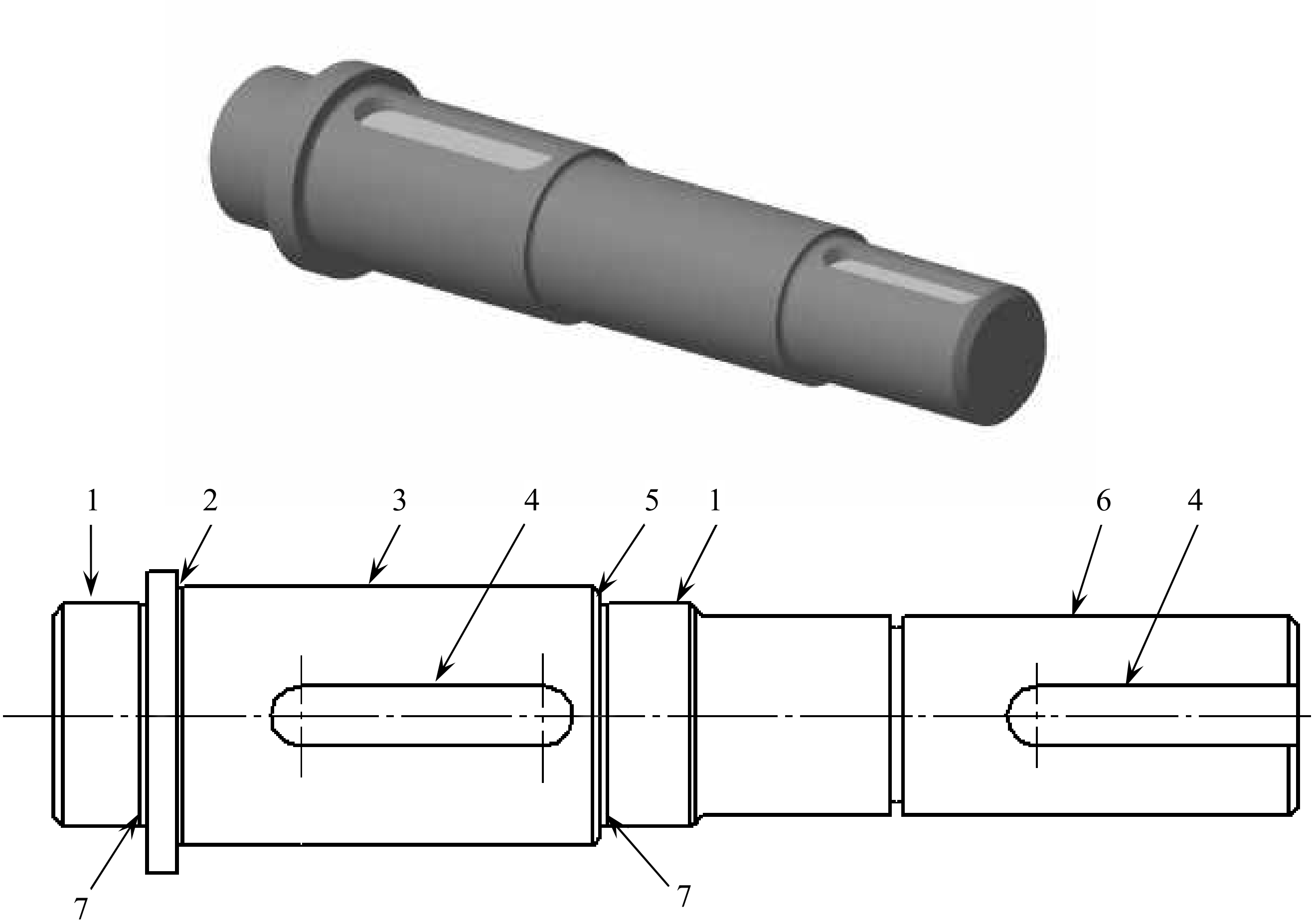


Fig. 3.2. An output shaft design:
1 – neck for bearings; 2 – thrust filet for a gear; 3 – neck for a gear;
4 – keyseats; 5 – thrust filet for a bearing; 6 – neck for half-coupling;
7 – grooves

3.2. Input Coupling Selection

Clutches and couplings are the mechanisms used for transferring torque between shafts and to compensate their misalignment.

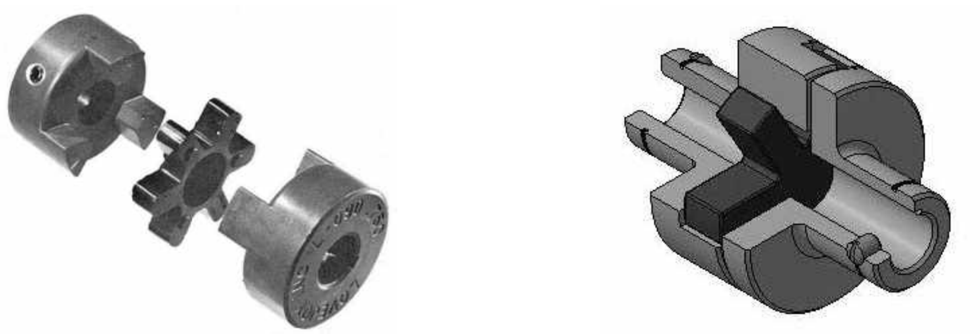
Standard couplings are selected on the base of a torque and the connecting shafts diameters. In addition, rotational speed should not exceed allowable one.

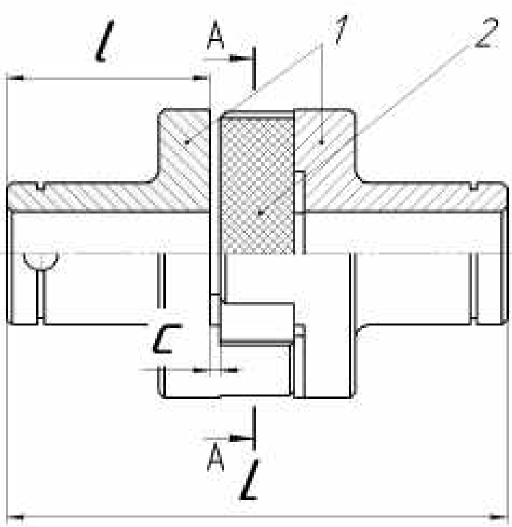
The main coupling characteristic is a nominal torque $\boldsymbol{T_p = k_p T}$, where $\boldsymbol{k_p = 2...2{,}5}$ is an overload factor considering an operating mode.

To connect an electric motor shaft and a gearbox one jaw compression coupling with flexible spider insert (GOST 14084-76*, Fig. 3.3) is recommended to use. Advantages of this type relatively to other elastic ones are smaller overall dimensions and possibility to connect shafts with different diameters.

The coupling consists of two half-couplings and an elastic element – a flexible spider insert with four or six beams.

The main coupling parameters and dimensions are shown in Appendix B.





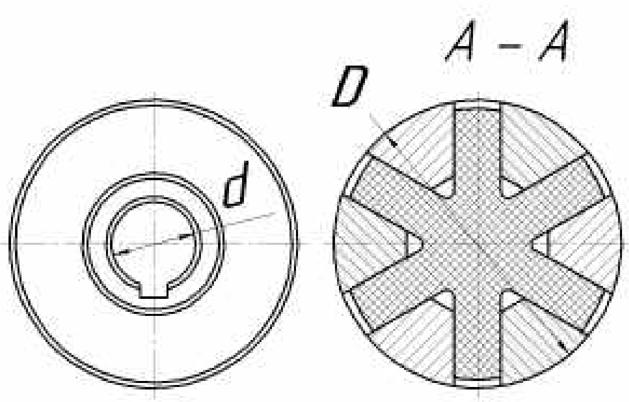


Fig. 3.3. Coupling with flexible spider insert:
1 – hubs (half-couplings); 2 – insert

As variant, so called flexible coupling with rubber-bushed studs (Fig. 3.4) may be used. It is usually used for general purpose to decrease dynamic loads in connection of coaxial shafts in case of torque transmission from 6,3 to 16 000 Nm and for shafts with diameter from 9 to 160 mm.

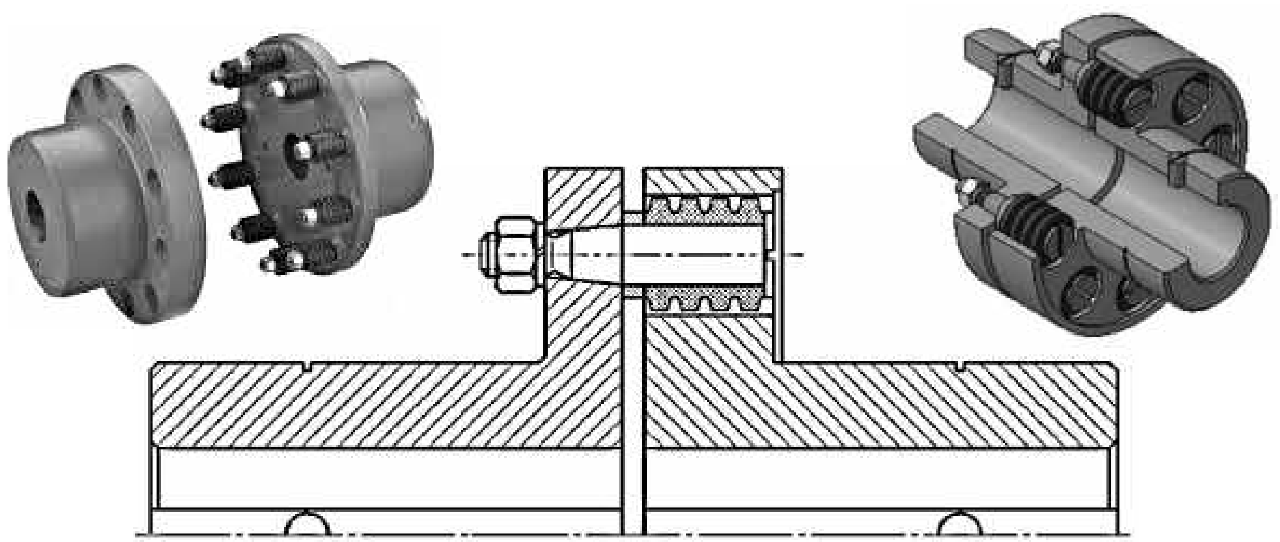


Fig. 3.4

Couplings are produced of two types: one with cylindrical bore and another with conical bore. Half-couplings of each type are made of two types: one for long shaft ends and another for short shaft ends.

The main coupling dimensions are shown in Appendix B.

Example 1

Let us consider spur straight teeth gearbox calculated above. The input nominal torque ***T*** = 54,4 Nm (see subsection 1.4, Example 1) and engine overload factor equals 2,2. Thus

$$T_p = k_p T = 2{,}2 \cdot 54{,}4 = 119{,}7 \text{ Nm}.$$

Let us select flexible coupling with rubber-bushed studs. According to calculated T_p we should select coupling with nominal torque 125 Nm (see Table B1, Appendix B). But the engine shaft diameter is 38 mm (see initial data) and this coupling series the biggest diameter is 28 mm. In such case, we have to take coupling with nominal torque 250 Nm with one end hole diameter 36 mm and another the smallest one 32 mm (see fragment of Table B1 below).

<table>
<tr><th rowspan="2">Nominal torque T, Nm</th><th rowspan="2">d</th><th rowspan="2">D</th><th colspan="2">L</th><th colspan="2">l</th><th rowspan="2">D₁</th><th rowspan="2">The number of studs</th><th rowspan="2">RPM_{max}</th><th colspan="2">Max shafts' misalignment</th></tr>
<tr><th>Long end</th><th>Short end</th><th>Long end</th><th>Short end</th><th>radial Δ_r, mm</th><th>angular Δ_α, deg</th></tr>
<tr><td rowspan="2">125</td><td>25
28</td><td rowspan="2">125</td><td rowspan="3">165</td><td>89</td><td>60</td><td>42</td><td rowspan="2">90</td><td rowspan="2">4</td><td rowspan="2">4 600</td><td rowspan="4">0,3</td><td rowspan="4">1</td></tr>
<tr><td>25
28</td><td rowspan="2">125</td><td>80</td><td>60</td></tr>
<tr><td rowspan="2">250</td><td>32
36</td><td rowspan="2">140</td><td rowspan="2">110</td><td rowspan="2">85</td><td rowspan="2">105</td><td rowspan="2">6</td><td rowspan="2">3 800</td></tr>
<tr><td>40
45</td><td>225</td><td>175</td></tr>
</table>

Selected coupling dimensions

<table>
<tr><th colspan="2">Parameter</th><th>Value</th></tr>
<tr><td rowspan="2">Diameters, mm</td><td>Engine shaft d₁</td><td>38</td></tr>
<tr><td>Gearbox input shaft</td><td>32</td></tr>
<tr><td rowspan="2">Conjunctive length l, mm, for half-coupling connected to</td><td>engine</td><td>85</td></tr>
<tr><td>gear box</td><td>85</td></tr>
<tr><td rowspan="2">Overall dimensions, mm</td><td>Diameter D</td><td>105</td></tr>
<tr><td>Total length L</td><td>125</td></tr>
</table>

Example 2

Let us select coupling for the drive with bevel gearbox calculated above (see subsection 1.4, Example 2, Fig. 2.10). The input nominal torque $\boldsymbol{T}$ = 80,7 Nm and engine overload factor equals 2,2. Thus

$$T_p = k_p T = 2{,}2 \cdot 80{,}7 = 177{,}5 \text{ Nm}.$$

According to calculated T_p we should select coupling with nominal torque 250 Nm. The engine shaft diameter is 38 mm (see Tab. 1.7). In such case, we can take one end hole diameter 38 mm and another the smallest one 32 mm (see fragment of Table B2 below).

Nominal torque T, Nm	d	D	L, Type I	L, Type II	l, Type I	l, Type II	c, mm	Maximum rotational speed, rpm	Maximum shaft axes misalignment: radial Δ_r, mm	Maximum shaft axes misalignment: angle Δ_α, deg
250,0	32 35	135	191	147	80	58	3,0	1500	0,4	1
	36		251	195	110	82				
	(38) 40 (42) 45		256	200						

3.3. Output Coupling Selection

To connect the output shaft and a driven machine many coupling types can be used. In a case of great load and necessity to compensate every kind of operating misalignment, as well as unavoidable misalignments when installing coupled machines curved-tooth gear coupling (GOST 5006-94) is recommended (Fig. 3.5).

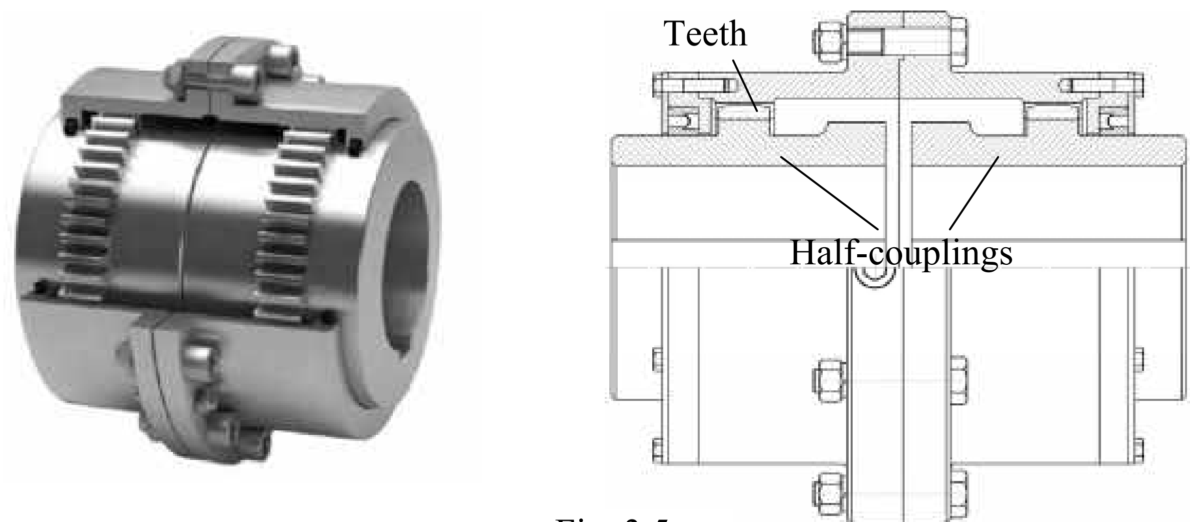


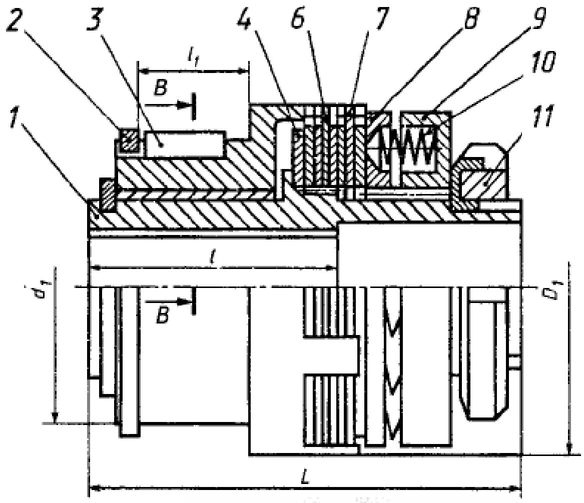
Fig. 3.5

This type of coupling transmits a torque through internal and external teeth. The operating principle of the gear couplings relies on engagement of the external teeth of the hub with the internal toothing of the sleeve, or bell, which allows the transmission of torque between the flanges. The relative misalignment is compensated by axial sliding of the internal gear teeth on the external gear teeth.

The coupling is made of hardened and tempered steel, fully machined and fastened by high strength bolts, to provide higher transmission capacity in relation to its size/weight.

The main coupling parameters and dimensions are shown in Appendix C.

Under the high shock loads, a slipping clutch or coupling or both should be used (Fig. 3.6). Therefore, slipping clutches protect machines against overload.



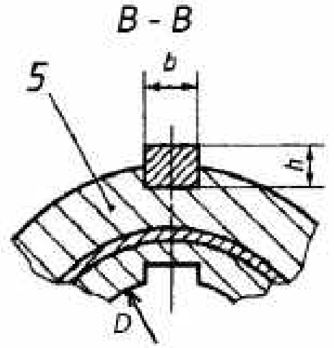


Fig. 3.6. A slipping clutch:
1 – driving hub; 2 – split ring; 3 – key; 4 – ring;
5 – driven hub; 6 – driving plate; 7 – driven plate;
8 – press plate; 9 – spring frame; 10 – spring;
11 – nut and stopper

The most common design uses a clutch plate that is clamped between the driving and driven plates by spring pressure that can be adjusted. When excessive load causes the driven member to slow, the clutch plate surfaces slip, allowing reduction of the transmitted torque. When the overload is removed, the drive is taken up automatically. Switches can be provided to cut off current supply to the driving motor when the driven shaft slows to a preset limit or to signal a warning or both.

Safety multiple-disc slipping coupling (GOST 15622-96) is used to connect a gearbox with a driven device because its advantages are: small dimensions, simple design and possibility to connect shafts with different diameters.

The slip or overload torque is calculated with taking 150 per cent of the normal running torque.

The main coupling parameters and dimensions are shown in Appendix C.

Example 1

Let us select curved-tooth gear coupling for spur straight teeth gearbox output shaft. The output nominal torque $\boldsymbol{T}$ = 194,4 Nm (see subsection 1.4, Example 1). Thus

$$\boldsymbol{T_p = k_p T = 2{,}2 \cdot 194{,}4 = 427{,}7} \text{ Nm}.$$

Look at Table C1 in Appendix C:

Torque T_p, Nm	**Driving shaft diam. *d*, mm**	**Outer diam. D_1, mm**	**Driven shaft diam. d_1, mm**	**Overall length *L*, mm**	**Driving end *l*, mm**	**Driven end l_1, mm**	**Driven key cross-section, mm**	
							h	***b***
1 000	40	145	90	180	80 110	48	9	14

Calculated output shaft diameter is 28 mm (see Tab. 2.30), but according to the table above, the smallest one should be 40 mm, therefore we assume this value for the output shaft coupling end.

Example 2

Let us select safety multiple-disc slipping coupling for bevel gearbox output shaft. The output nominal torque $\boldsymbol{T}$ = 220 Nm (the given data). Thus

$$\boldsymbol{T_p = k_p T = 1{,}5 \cdot 220 = 330} \text{ Nm}.$$

Look at Appendix C, Table C2:

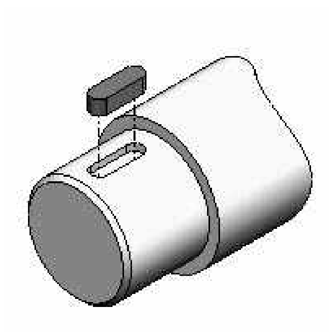
Torque T_p, Nm	**Driving shaft diam. *D*, mm**	**Outer diam. D_1, mm**	**Driven shaft diam. d_1, mm**	**Overall length *L*, mm**	**Driving end *l*, mm**	**Driven end l_1, mm**	**Driven key cross-section, mm**	
							h	***b***
400	38 40, 42, 45, 48	145	90	180	80 110	48	9	14

Calculated output shaft diameter is 28 mm (see Tab. 2.31), but according to the table above, the smallest one should be 38 mm, therefore we assume this value for the output shaft end.

3.4. Key selection

Keys are used to attach parts to shafts in order to transmit rotational power to gears. From many key types we will consider the most widespread and simple Pratt & Whitney key with rectangular cross-section (GOST 23360-78, Appendix D).

The standard key size ***b*** x ***h*** is selected depending on a shaft diameter ***d*** (Fig. 3.7). The size of section ***b*** x ***h*** provides the full-strength on bearing stress and a shear one that is why key working length is usually obtained from a calculation on bearing.



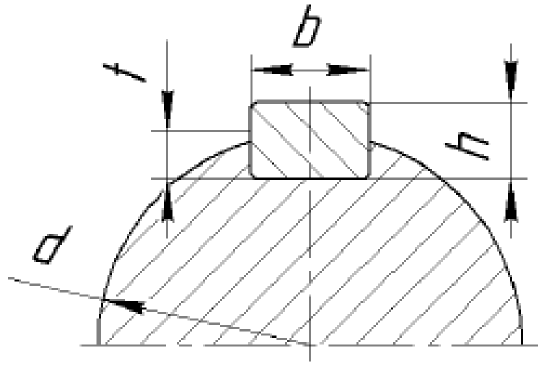


Fig. 3.7

The strength condition on bearing at uniform load distribution along a key operating length

$$\sigma_b = \frac{2000T_p}{d\,l\left(h-t\right)} \le \left[\sigma_b\right],$$

Whence the key length equals

$$l = \frac{2000\,T_p}{0{,}4d\,h\left[\sigma_b\right]},$$

where $[\sigma_b]$ is the allowable bearing stress for the weakest material of a shaft, a hub and a key. GOST demands, that key materials should ensure UTS $\sigma_в \ge 590$ MPa (for example, steel 45, steel 40Г, steel 40X, etc.).

As keys work in rather heavy conditions, it is recommend to accept the allowable bearing stress a little below that are shown in corresponding tables. For motionless connections with transitive fits the allowable bearing stress should be $[\sigma_b] = 80 \ldots 120$ MPa and less.

There are three types of the keys (see Appendix D). At connection on transitive fits or with a tightness (which more often is used in drives), it is desirable to use Pratt & Whitney keys with one or two round ends, i.e. types 1 or 3. In this case key width ***b*** should be added to the calculated key length and rounded to standard value (Fig. 3.8).

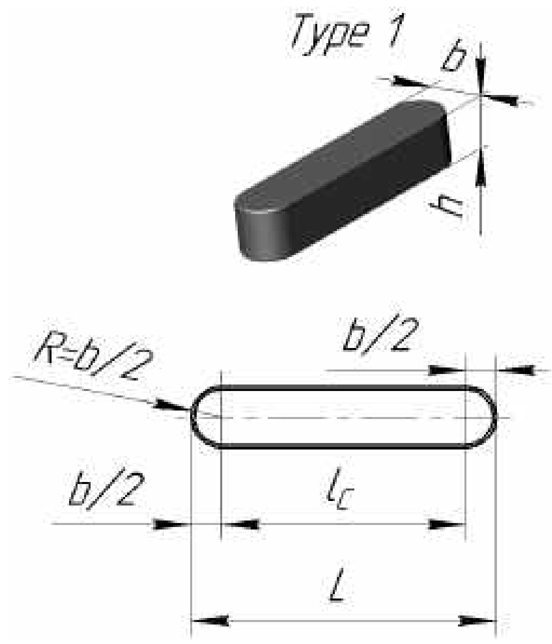


Fig. 3.8

The chosen standard length should be co-ordinated with recommended one, a shaft diameter and a gear face width. Relation ***L/d*** is recommended 0,8…1,5, because the bigger a key length the bigger non-uniformity of load. If the key length exceeds recommended ***L/d*** it is possible to increase a shaft diameter or select bigger key section size and recalculate length.

If total standard length less than minimum one, it has to be taken as minimum standard value.

Example 1

Let us calculate and select a key for connection of the spur gearbox input shaft and the input half-coupling. The cross-section dimensions ***b*** x ***h*** according to the shaft diameter ***d*** = 32 mm see in Appendix D:

Shaft diameter		b	h	Key length		Keyway depth	
from	to			from	to	shaft	hub
22	30	8	7	18	90	4	3,3
30	38	10	8	22	110	5	3,3
38	44	12	8	28	140	5	3,3

Let us use steel 45 and $[\sigma_b] = 80$ MPa. Thus

$$l = \frac{2000T}{0{,}4d\,h[\sigma_b]} = \frac{2000 \cdot 54{,}4}{0{,}4 \cdot 32 \cdot 8 \cdot 80} = 13{,}3 \text{ mm.}$$

We use type 1 key, its total length equals $L = l + b = 13{,}3 + 10 = 23{,}3$ mm. This value is not standard (see Appendix D), therefore we must coordinate it with standard length and select the next bigger value 25 mm.

To connect the output shaft and the half-coupling, a key with cross-section dimensions according to the shaft diameter ***d*** = 40 mm ***b*** x ***h*** = 12 x 8 mm may be provided.

If we use steel 45 and $[\sigma_b] = 80$ MPa

$$l = \frac{2000T}{0{,}4d\,h[\sigma_b]} = \frac{2000 \cdot 194{,}4}{0{,}4 \cdot 40 \cdot 8 \cdot 80} = 37{,}97 \text{ mm.}$$

If we use type 1, the total length equals $L = l + b = 38 + 12 = 50$ mm. This value is standard.

Example 2

Let us calculate and select a key for connection of the input shaft and the input half-coupling. The cross-section dimensions ***b*** x ***h*** according to the shaft diameter ***d*** = 32 mm see in Appendix D:

Shaft diameter		b	h	Key length		Keyway depth	
from	to			from	to	shaft	hub
22	30	8	7	18	90	4	3,3
30	38	10	8	22	110	5	3,3
38	44	12	8	28	140	5	3,3

We can use recommended by standard steel 45 (conditional bearing allowable stress is $[\sigma_b] = 80$ MPa). Thus,

$$l = \frac{2000T}{0{,}4d\,h[\sigma_b]} = \frac{2000 \cdot 80{,}7}{0{,}4 \cdot 32 \cdot 8 \cdot 80} = 19{,}7 \text{ mm.}$$

We use type 1 key, its total length equals $\boldsymbol{L = l + b = 19{,}7 + 10 = 29{,}7}$ mm. This value is not standard (see Appendix D), therefore we must coordinate it with standard length and select the next bigger value 32 mm.

Let us calculate a key for connection of the output shaft and the coupling selected above. The cross-section dimensions $\boldsymbol{b}$ x $\boldsymbol{h}$ are found in Appendix D according to the shaft diameter $\boldsymbol{d}$ = 38 mm: $\boldsymbol{b}$ x $\boldsymbol{h}$ = 10 x 8 mm.

Let us use the same steel 45 ($[\sigma_b] = 80$ MPa). Thus

$$l = \frac{2000T}{0{,}4d\,h[\sigma_b]} = \frac{2000 \cdot 220}{0{,}4 \cdot 38 \cdot 8 \cdot 80} = 45 \text{ mm.}$$

If we use type 1, the total length equals $\boldsymbol{L = l + b = 45 + 10 = 55}$ mm. This value is not standard (see Appendix D), therefore we must coordinate it with standard length and select the closest bigger value 56 mm.

3.5. Preliminary Bearings' Selection

At this design stage, we cannot define load on bearings, therefore we have to choose them from geometrical reasons: on a shaft diameter in a place of their installation, the previous experience and easy assemblage.

According to rule 3 it is necessary to mount two bearings on a shaft, capable to take both radial, and axial load. As axial force in spur gearing is absent and in bevel gearing is insignificant, it is possible to use the most simple ball deep-groove bearings of a light or middle series.

According to rule 2 bearing should be mounted on the diameter and freely pass to the its place. A key is usually located on a bearing way, therefore the diameter has to be standard (usually multiple 5) and bigger that a shaft together with the key (Fig. 3.9).

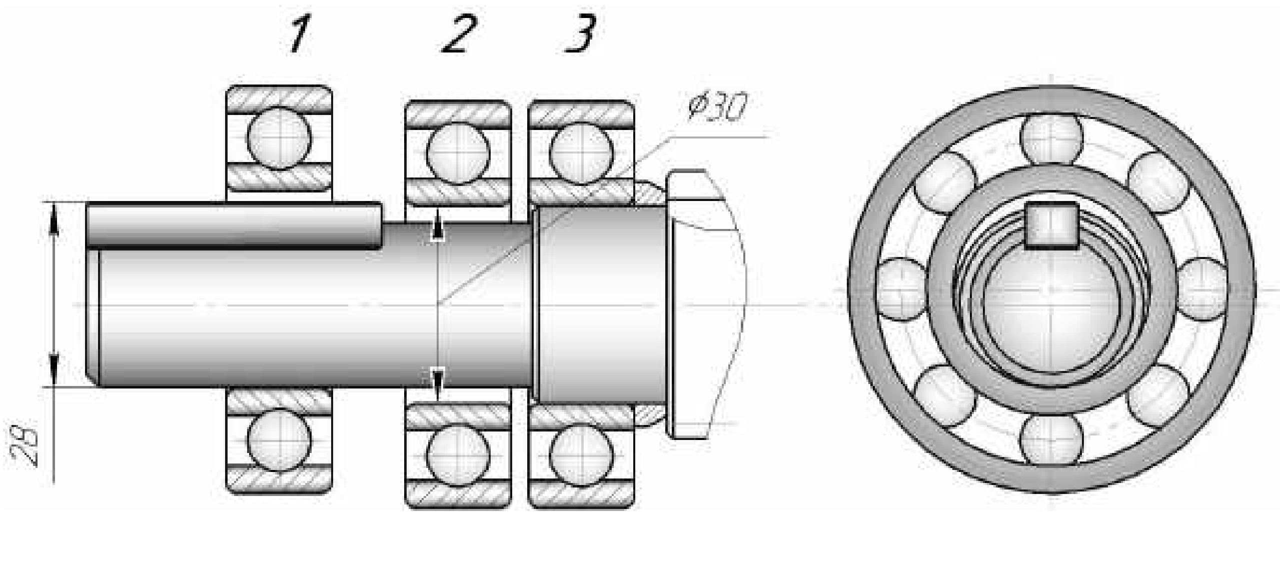


Fig. 3.9

Example 1

Let us select bearings for the spur gearbox input shaft. The smallest bearing bore diameter equals $\boldsymbol{d_{sm} = d + (h - t)} = 32 + (8 - 5) = 35$ mm (see Fig. 3.7, 3.9). Thus, we may select light series bearing 208 with the next nearest standard diameter 40 mm (see Appendix E).

In the same way for output shaft $\boldsymbol{d_{sm} = d + (h - t)} = 40 + (8 - 5) = 43$ mm, therefore bearings light series 209 with standard bore diameter 45 mm may be used.

d	Code	D	B	r	Load capacity, N		Minimum d_a	Maximum D_a
					dynamic C	static C_0		
40	1000908	62	12	1	12 200	6 920	44	58
	108	68	15	1	16 800	9 300	45	63
	7000108	68	9	1	13 300	7 800	42	66
	208	80	18	3	32 000	17 800	46	73
	308	90	23	1	41 000	22 400	48	81
	408	110	27	2	63 700	36 500	49	97
45	1000909	68		1	14 300	8 130	49	64
	109	75	16	1	21 200	12 200	50	70
	7000109	75	10	1	15 600	9 300	49	71
	209	85	19	1	33 200	18 600	52	78
	309	100	25	1	52 700	30 000	53	91
	409	120	29	1	76 100	45 500	54	107

Example 2

Let us select for the input shaft light series bearings № 208 with bore diameter 40 mm (shaft diameter under the key is 32 mm, key height over shaft is 3 mm). Light series bearings № 209 with bore diameter 45 mm (shaft diameter under the key is 38 mm, key height over shaft is 4 mm) may be selected for the output shaft.

So we assume bearings as follows

Parameter	Symbol	Value, mm, for shaft	
		input	output
Code	–	208	209
Series	–	light	light
Inner diameter	$\boldsymbol{d}$	40	45
Outer diameter	$\boldsymbol{D}$	80	85
Width	$\boldsymbol{B}$	18	19
Fillet minimum diameter	$\boldsymbol{d_{amin}}$	46	52
Static load capacity, N	$\boldsymbol{C_0}$	32 000	33 200
Dynamic load capacity, N	$\boldsymbol{C}$	17 800	18 600

3.6. Selection of a Cover

To close support unit hole a cover (a bearing cap, an end plug) is used. The cover is housing for the oil seal, that is why it is also standardized and coordinated with both oil seal and bearing dimensions.

The bearing covers are recognized on the outer bearing diameter ***D*** (Fig. 3.10). According to this diameter and a shaft diameter d_{sh} all other dimensions are selected (Appendix F).

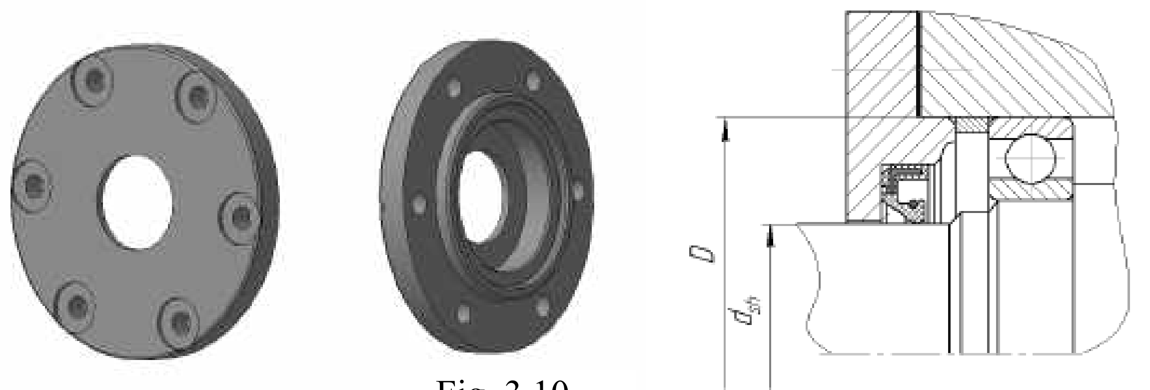


Fig. 3.10

For example, the bearing 208 has outer diameter 80 mm, therefore we have to consider corresponding size into the first column (see table from Appendix F below).

D	D_S	D_1	D_2	D_3	D_4	D_5	D_6	d	d_1	d_2	H	h	l	n	b_1	c	r
80	25	100	120	72	80	26	42	9	15	20	18	6	3	6	11,0	1,0	0,6
	30					31	52										
	35					36	58										
	40					41	60										
	45					46	65										
	50					51	70										
85	35					36	58										
	45					46	65										

It is seen that may be a number of shaft diameter variants (see column D_S): 25, 30…50 mm. What variant should we select?

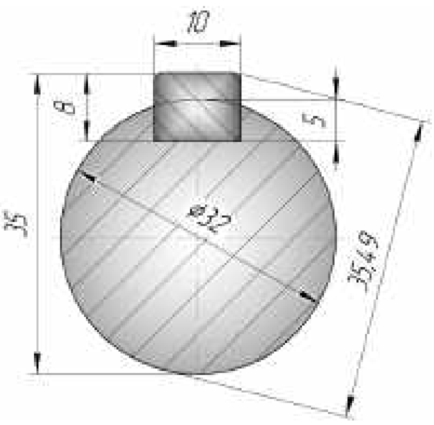


Fig. 3.11

First, we have to ensure passing of the cover over the previously selected key (the maximum size is 35,49 mm, see Fig. 3.9, 3.11). To satisfy this requirement, the hole diameter for the shaft (column D_5) should be bigger or equal 36 mm. Moreover, shaft diameters 25 and 30 are not suitable as they less than coupling end shaft diameter.

At the same time, we cannot realize diameters bigger than 40 because previously selected bearings' diameter is 40 mm and it is impossible to mount bearing on its seat if shaft size around a bearing bigger than bearing bore diameter.

Thus, the only variant is 35 mm (it is marked in the table above). So, we select covers as follows:

Parameter	Symbol	Value, mm, for shaft	
		input	output
Bearing outer diameter	$\boldsymbol{D}$	80	85
Bearing inner diameter	$\boldsymbol{d_b}$	40	45
Coupling end diameter	$\boldsymbol{d_c}$	32	Spur □ 40; bevel □ 38
Shaft diameter	$\boldsymbol{D_S}$	35	45
Hole	$\boldsymbol{D_5}$	36	46
Cover outer	$\boldsymbol{D_2}$	120	120
Oil seal outer	$\boldsymbol{D_6}$	58	65
Total height	$\boldsymbol{H}$	18	18
Rim height	$\boldsymbol{h}$	6	6
Diameter of a hole for bolt	$\boldsymbol{d}$	9	9
Counterbore diameter	$\boldsymbol{d_2}$	20	20

3.7. Selection of an Oil Seal

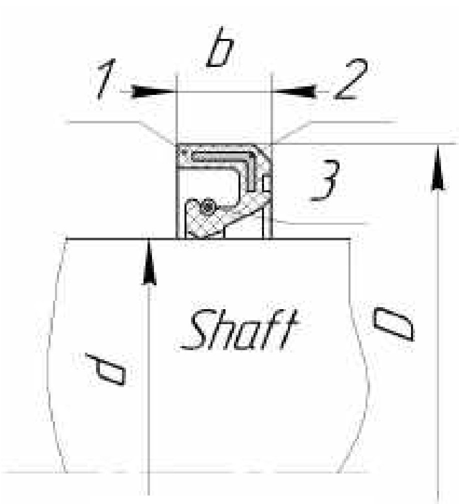


Fig. 3.12

The bearings and the mating surfaces in gearboxes with gears peripheral velocity less than 15 m/s are usually lubricated with liquid oil by immersing the teeth in an oil bath and so-called “an oil fog”. To avoid the oil leakage and protect a gearbox against dust and dirt special sealing devices are used.

The sealing simplest types are a stuffing box seal and oil seal, but the first of them is limited to a peripheral velocity in contact point of 3…5 m/s, that is why the oil seal is more preferable.

A typical oil seal (Fig. 3.12) consists of rubber body 1 with metallic core 2 and spring ring 3 to press operating edge to a shaft surface.

The oil seals are stantardized and selected from a table according to a shaft diameter. The main parameters are: shaft diameter $\boldsymbol{d}$, the oil seal outer diameter $\boldsymbol{D}$ and width $\boldsymbol{b}$.

Preferably to select the shaft diameter $\boldsymbol{d}$ under oil seal different from surrounding ones. As standard cover is selected this size (see column $\boldsymbol{D_S}$ in Appendix F) and oil seal outer diameter $\boldsymbol{D}$ (see column $\boldsymbol{D_6}$ in Appendix F) are also determinated.

Any way, it should be selected bigger than coupling end shaft diameter and less than a standard bearing bore diameter.

In our examples the input shaft diameter (end connected to coupling) equals 32 mm, a standard bearing bore diameter equals 40 mm, a standard cover shaft diameter is 35 mm, therefore let us take from Appendix G an oil seal with ***d* = 35** mm, ***D* = 58** mm, ***b* = 10** mm.

The covers for output shafts have standard bearing bore diameter 45 mm therefore let us take from Appendix G an oil seal with ***d* = 45** mm, ***D* = 65** mm and ***b* = 10** mm.

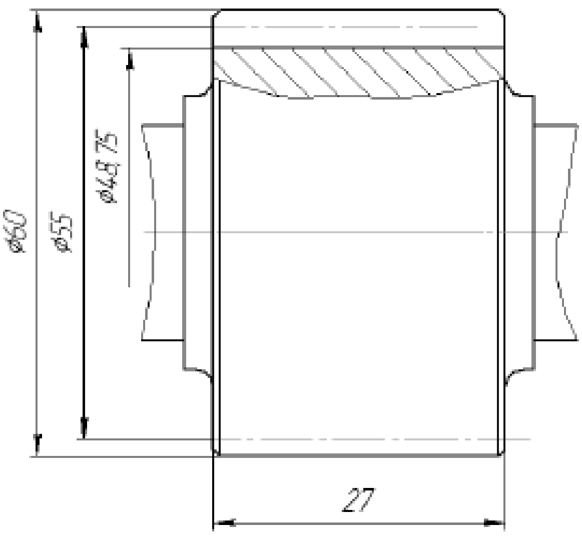
d	D	b
26	45	7
28	50	10
30	52	10
32	52	10
35	58	10
36	58	10
38	58	10
40	60;62	10
42	62	10
45	65	10

3.8. Designing of the Input Shaft

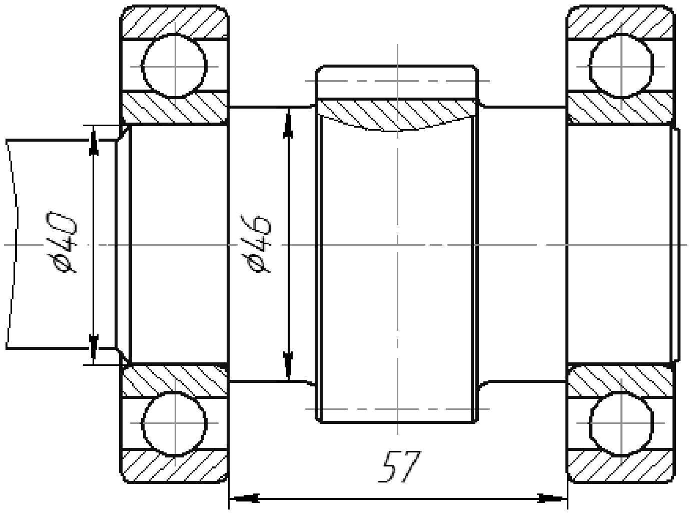
Now, when all elements are selected, we may form the shaft.

3.8.1. Straight Teeth Spur Gearbox

According to Fig. 2.9 pinion is made as the same element with an input shaft. It means that we can continue shaft designing using the obtained shape (it is turned on right angle outer clockwise relatively to Fig. 2.9).



1. Add selected bearings. To form the thrust fillets for bearings we use standard fillet minimum diameter $\boldsymbol{d_{amin}}$ 46 mm (see subsection 3.5).



To obtain bearing places in the axial direction the following recommendations should be taken into account: the rotating parts must not contact the frame. Therefore, gaps between respective surface of a rotating part and casing surface must be provided.

Gap value has to be equal 5…7, let us assume 6 mm.

Gear hub has the biggest inner axial dimension – 40 mm. Additionally, thrust fillet for the bearing and the gear has to be ensured (axial size 3…5 mm, let us take 5).

Thus, total distances between inner frame surfaces and between bearing faces equal 40 + 5 + 2 · 6 = 57 mm.

2. Define external outline of the frame flange. The total frame of gearbox is detachable, consists of two parts (frame and frame cover), bolted together. To find external outline of the casing the flange width must be determined (Fig. 3.13). Here $\boldsymbol{\delta}$ is a frame wall thickness which can be calculated as follows:

$$\boldsymbol{\delta = 1{,}12\sqrt[4]{T_{out}}}\ ,$$

but at least 6 mm because the frames of general-purpose gearboxes are made of casting from cast iron.

Bolt diameter to attach frame to cover should be equal

$$\boldsymbol{d_{fc} = \sqrt[3]{2T_{out}} \geq 10}\ \text{mm}.$$

Bolts diameter attaching frame to cover near the bearing arrangement may be assigned next available standard value (i.e. if $\boldsymbol{d_{fc}}$ = 8 mm, then diameter should be 10 mm).

Flange width to attach frame to cover can be determined using the formula:

$$\boldsymbol{S = k + \delta + x}.$$

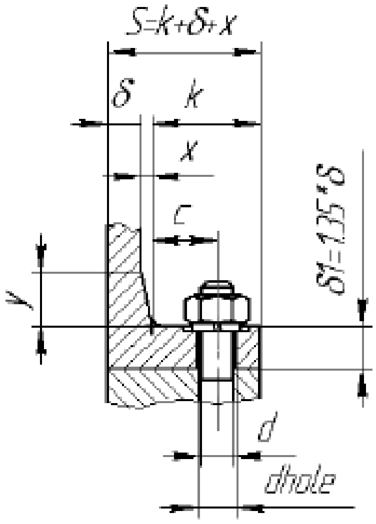


Fig. 3.13

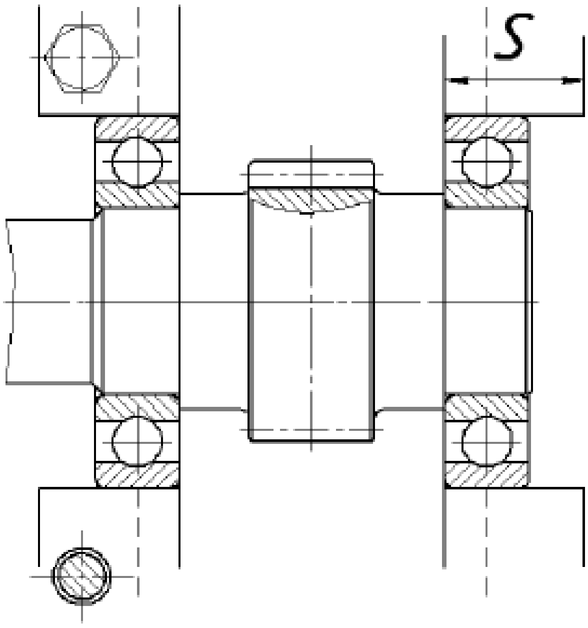
Dimensions ***k*** and ***x*** see in Tab. 3.1 and 3.2.

Table 3.1

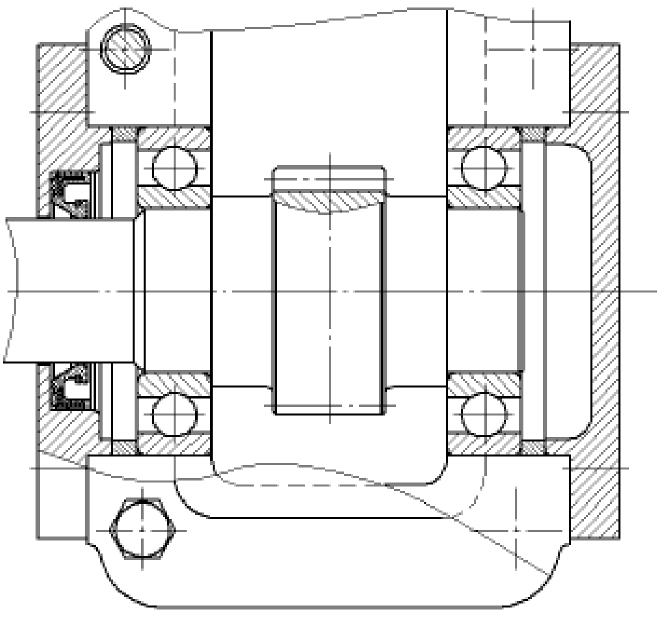
d	M6	M8	M10	M12	M14	M16	M18	M20	M22	M24	M27	M30
k	20	24	28	32	36	40	44	48	52	56	62	68
c	12	14	16	18	20	22	24	26	28	30	34	37

Table 3.2

δ	**x**	**y**	**r**
6…10	2…3	5x	3
10…15	3…5		4
15…20	4…5		5



3. Locate the bearing caps (covers) at their places. If a bearing does not touch with cover, specific spacer ring can be provided.



4. To obtain final view, shape the shaft coupling end according to selected data (see subsection 3.2, 3.3)(Fig.3.14).

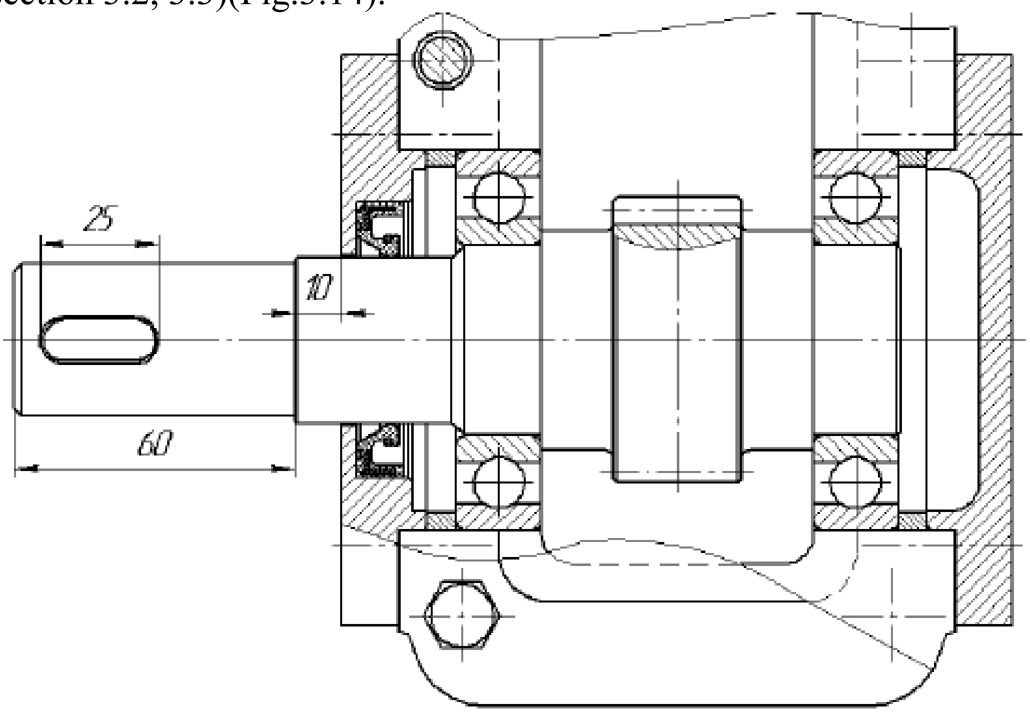
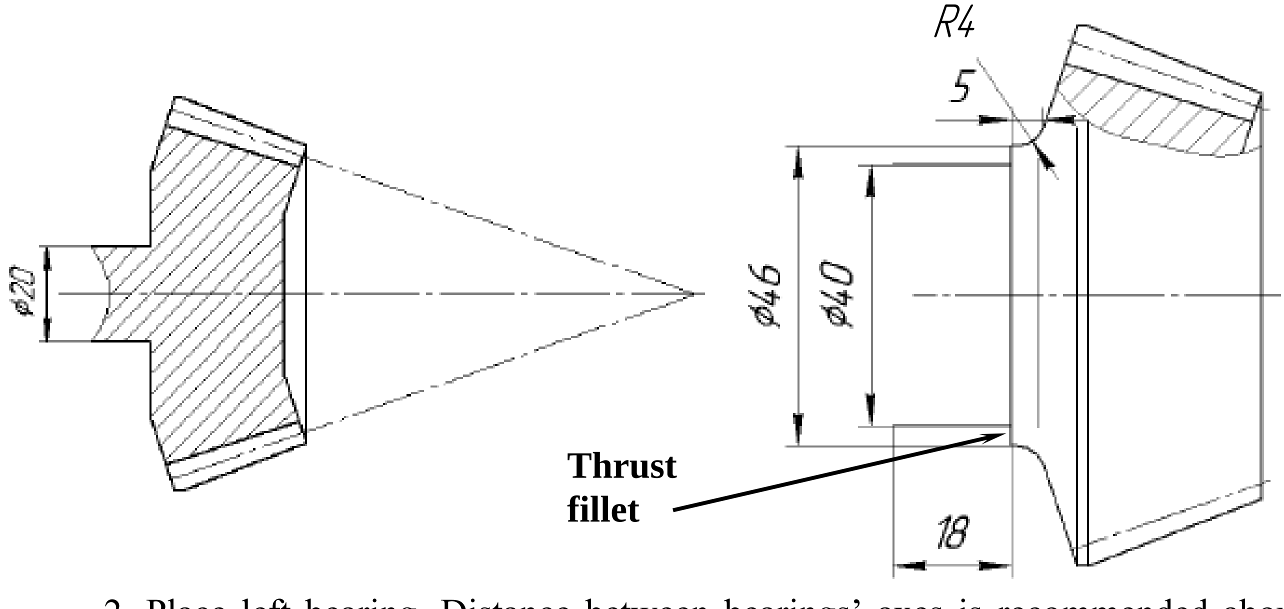


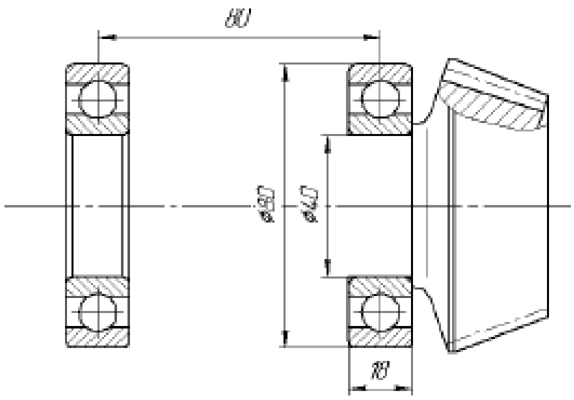
Fig. 3.14

3.8.2. Straight Teeth Bevel Gearbox

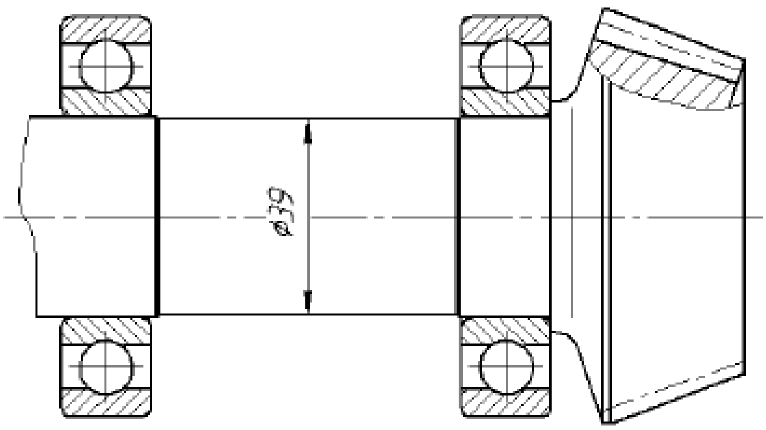
1. Redraw the pinion (see Fig. 2.10) according to selected bearings: increase shaft diameter from 20 to 40 mm, generate thrust fillet to retain right bearing in axial direction (minimum diameter $\boldsymbol{d_{amin}}$ 46 mm see subsection 3.5) and round transition from outer cone to a cylinder surface.



2. Place left bearing. Distance between bearings' axes is recommended about ***2d***, where ***d*** is a bore diameter.

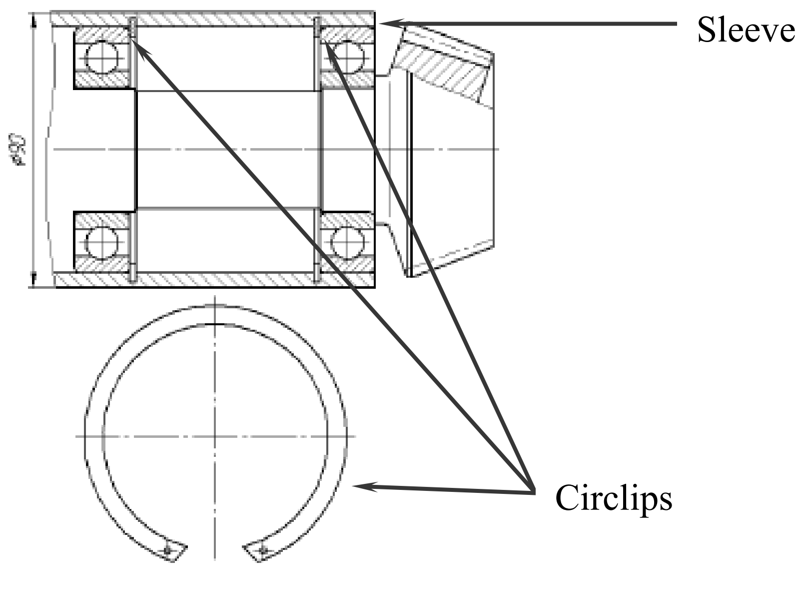


3. A shaft diameter between bearings should be some less for right bearing easy mounting.

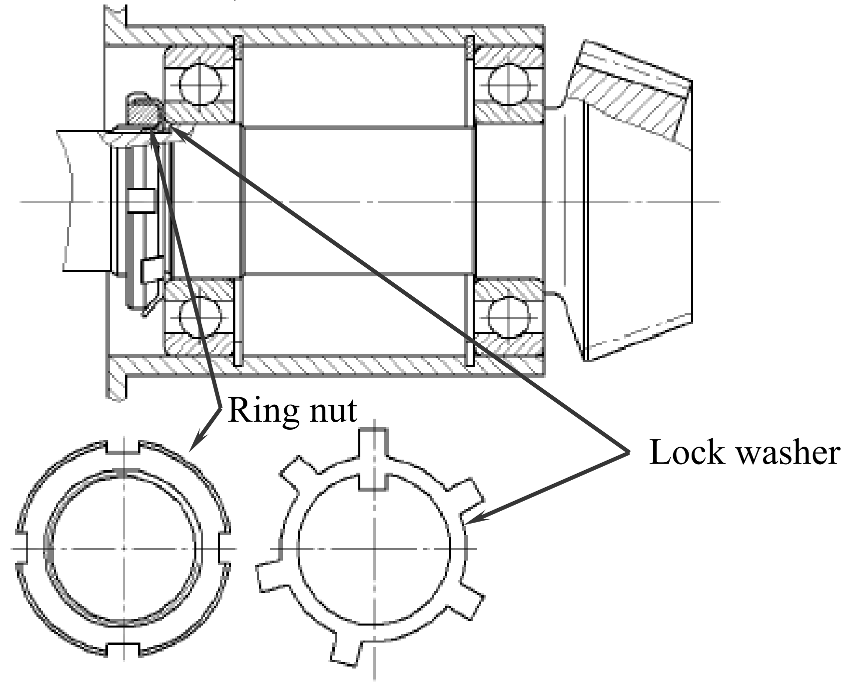


4. To obtain correct gears' contact the input shaft with the bearing units should shift along its axis in the required direction. A very effective method to solve this problem is to assemble the shaft with pinion into a sleeve, which is then installed into the gearbox housing.

Each of the supports retains the shaft in one axial direction. Two fix points on the sleeve inner diameter are the circlips and third point is the thrust fillet on the shaft. The circlips' base dimension is bearing outer diameter (see Appendix I).

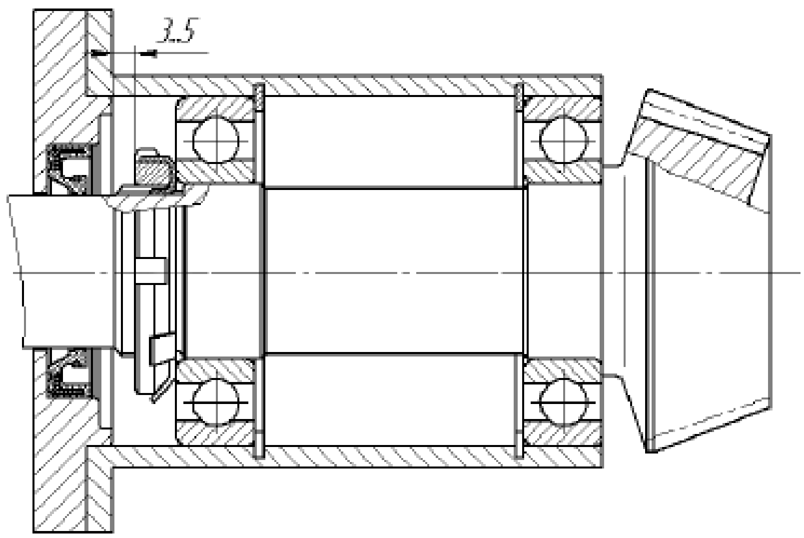


5. A ring (or lock) nut and a locking washer are used to retain the shaft in right-hand axial direction. It is a simple method of ensuring that lock nuts remain securely fastened. The central tab locates in a key-way in the shaft and the nut is fitted and tightened to the correct torque, the washer will now have one of its tabs aligned with one of the slots around the lock nut, this tab is folded over into the slot.



The nut and the washer are standardized and nut thread major diameter is coordinated with a bearing bore diameter (see Appendix H). Clearance into the bearings is also adjusted by means of a ring nut with a tab washer.

6. Close sleeve hole by a cover.



7. Finally, draw a cylinder end for a coupling. To fix the bevel gears correctly and to align the apexes of the pitch cones all covers of the bearing units and the sleeve rest on mounting shims. Their thickness is very small (less 1 mm), so shims are shown conditionally in assembly drawings.

Finally, the input shaft unit looks as in Fig. 3.15.

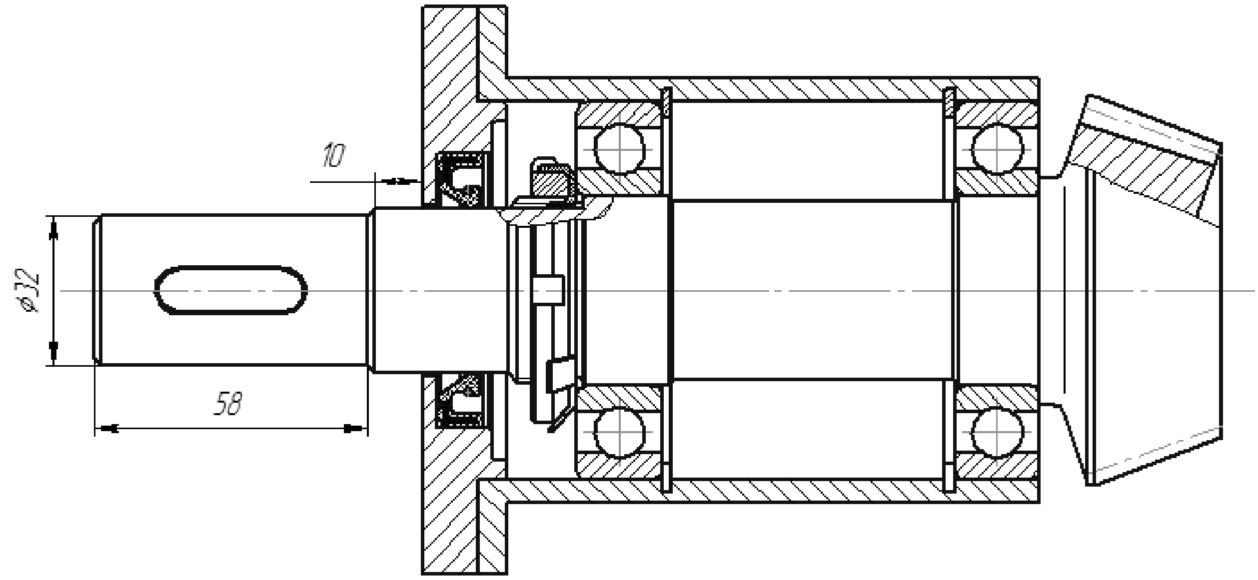


Fig. 3.15

3.9. Design of the Output Shaft

An output shaft differs the input one and should approximately look like in Fig. 3.1. All elements connected to the shaft (a coupling, an oil seal and the bearings) were selected, therefore we may form the shaft surfaces.

3.9.1. Straight Teeth Spur Gearbox

1. Let us begin from a hub.

When we know bearing dimensions we can calculate and select a key to connect the output shaft and the gear hub. First, the hub diameter 28 mm was selected (see Fig. 2.9), but according to rule 2 it cannot be less 45 mm (the bearings bore diameter). Therefore, inner hub diameter ought to be increased up to 46 mm.

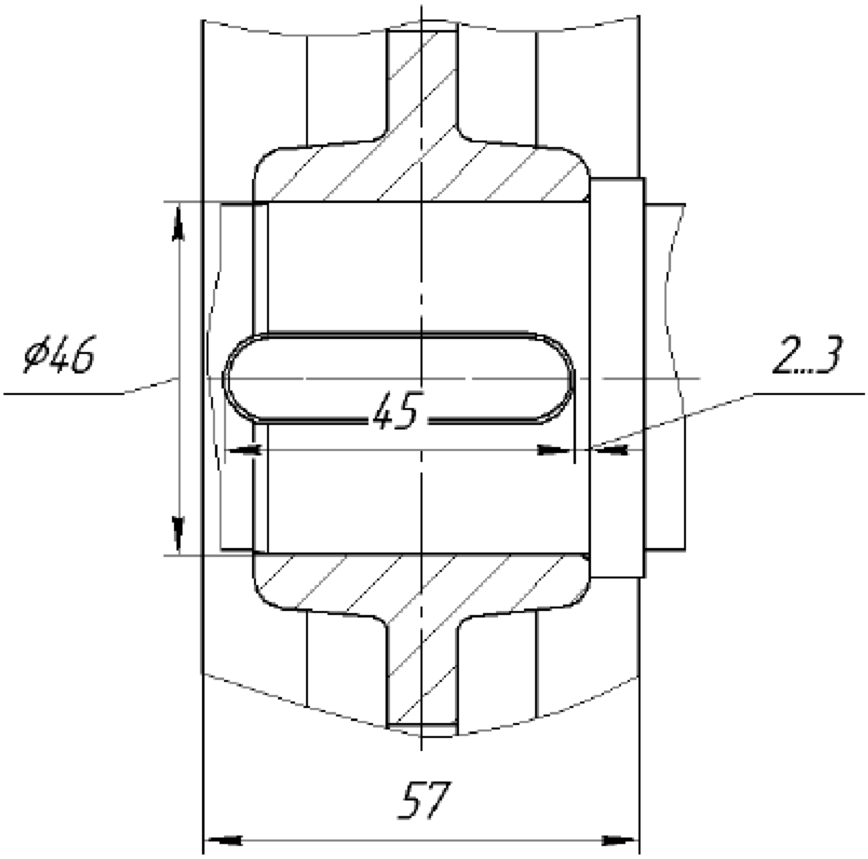
Due to standard a key cross-section dimensions ***b*** x ***h*** should be 14 x 9 mm (see Appendix D). But if two keyseats are placed in the one shaft it is recommended to use the same cross-section dimensions ***b*** x ***h*** = 12 x 8 mm according to the smallest diameter (as for the output shaft end). In such case both keyseats are machined during one manufacturing operation and high level of accuracy can be achieved.

Thus,

$$\boldsymbol{l = \frac{2000T}{0{,}4d\,h[\sigma_b]} = \frac{2000 \cdot 194{,}4}{0{,}4 \cdot 46 \cdot 8 \cdot 80} = 33}\ \text{mm.}$$

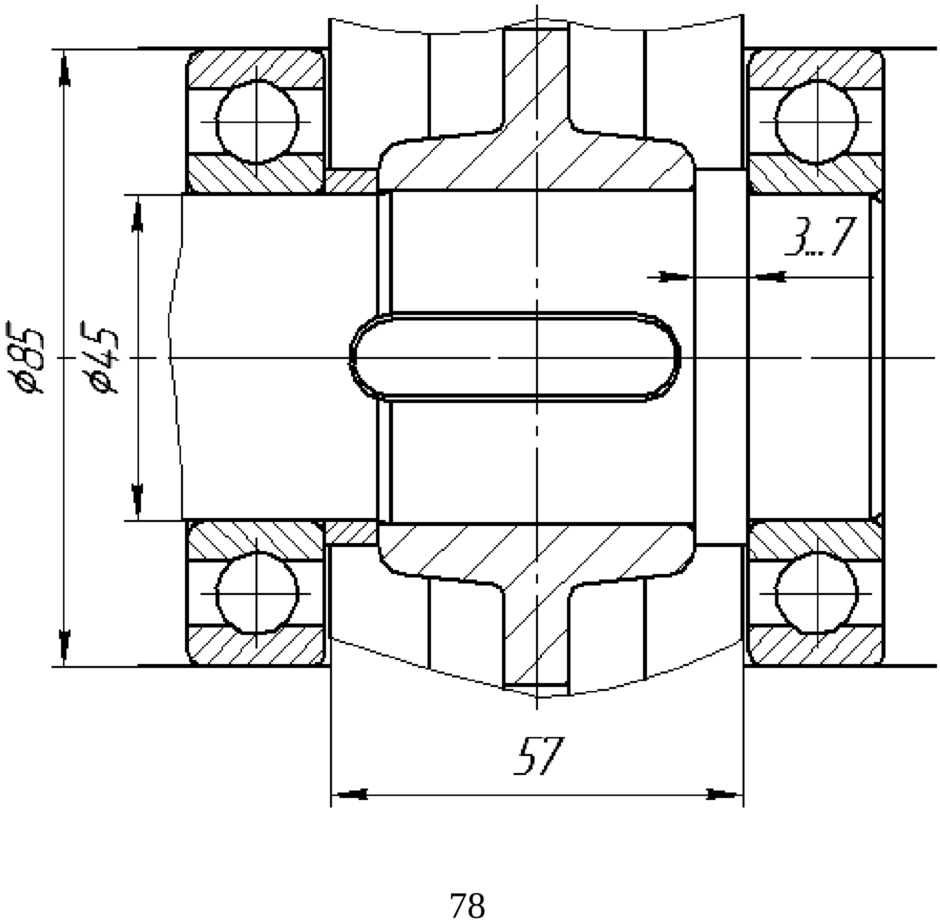
If we use type 1, the total length equals $\boldsymbol{L = l + b = 33 + 12 = 45}$ mm. This value is standard, but in such case hub length is too big (gear face width equals 28 mm, $\boldsymbol{L/d}$ = 45/28 = 1,6 > 1,5). The simplest way is to select key type 3, thus $\boldsymbol{L = l + b/2 = 33 + 6 = 39}$ mm, standard value 40 mm.

Thus, we change previous drawn hub dimensions (turned sketch):

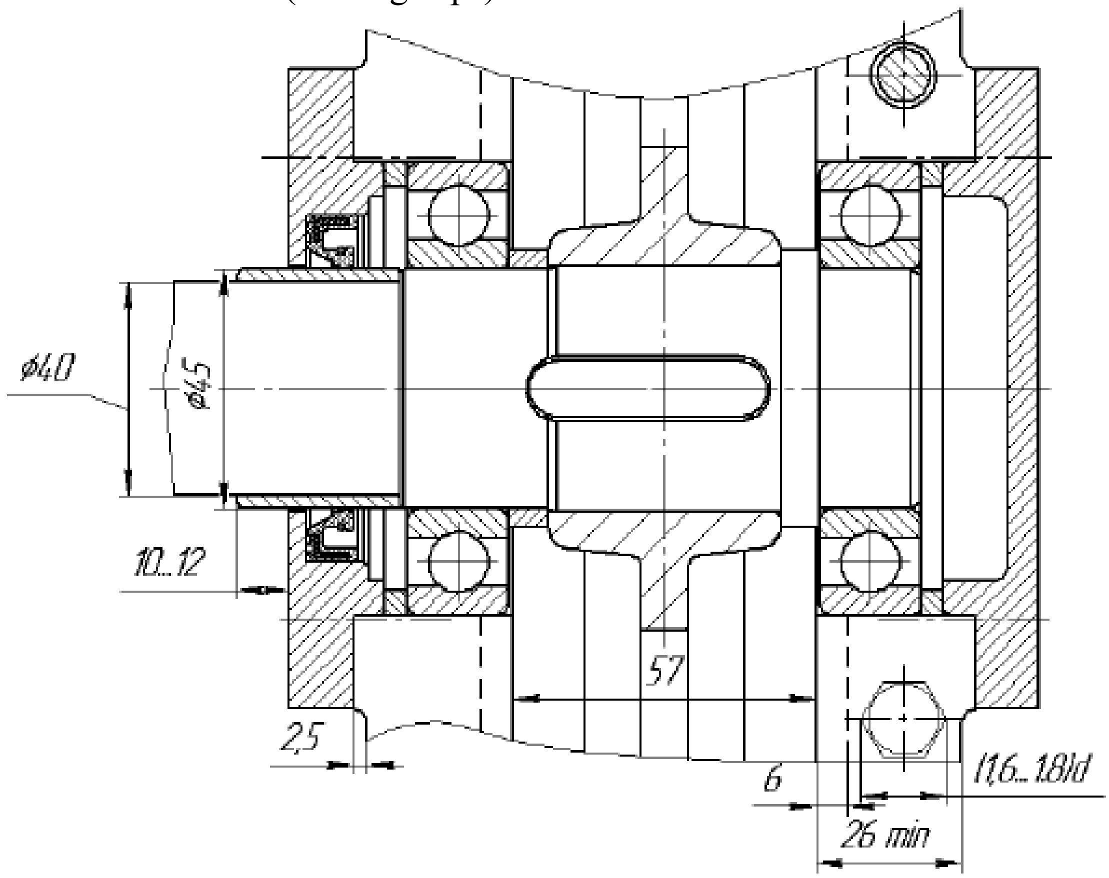


To avoid superposition of stress concentrations in the shaft clearance 2…3 mm between the key seat and the shoulder is provided.

2. Place the bearings. We know their dimensions (see subsection 3.5) and can locate their into support seats to ensure flat inner vertical wall.



3. Add the covers (bearing caps).



4. Draw the output end. Finally, the output shaft unit looks like on Fig. 3.16.

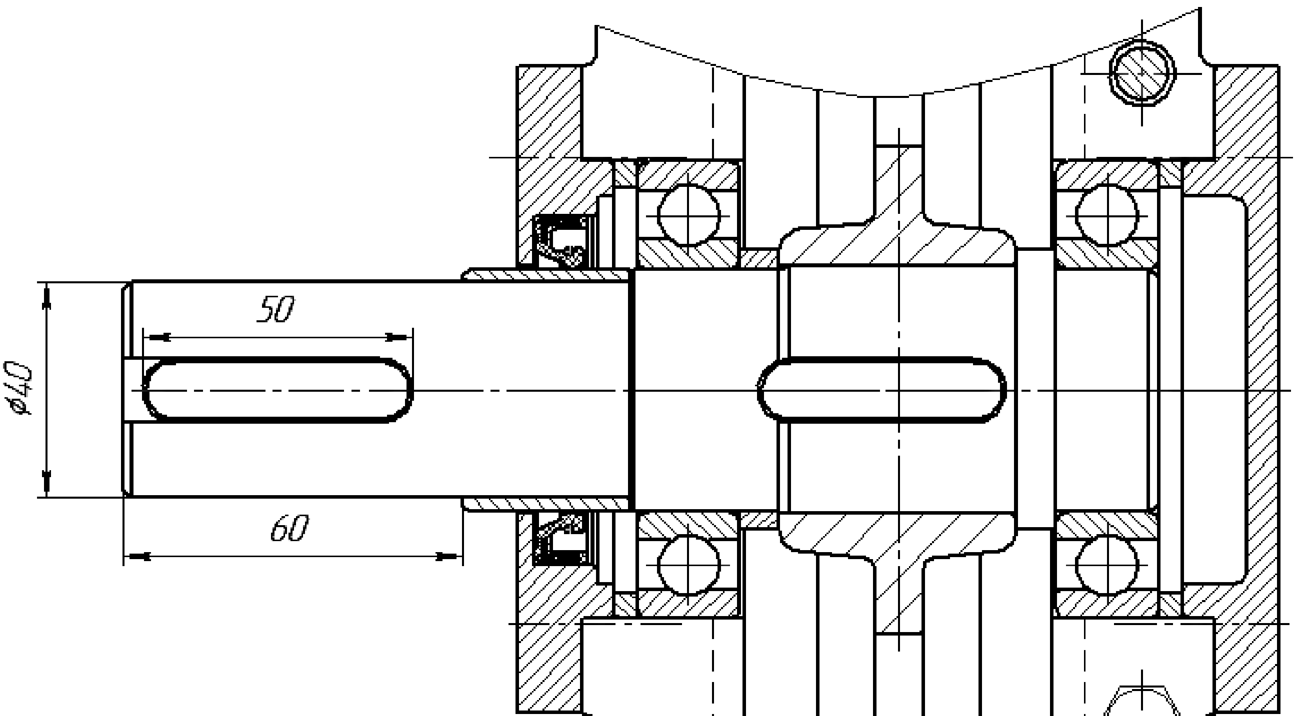


Fig. 3.16

3.9.2. Straight Teeth Bevel Gearbox

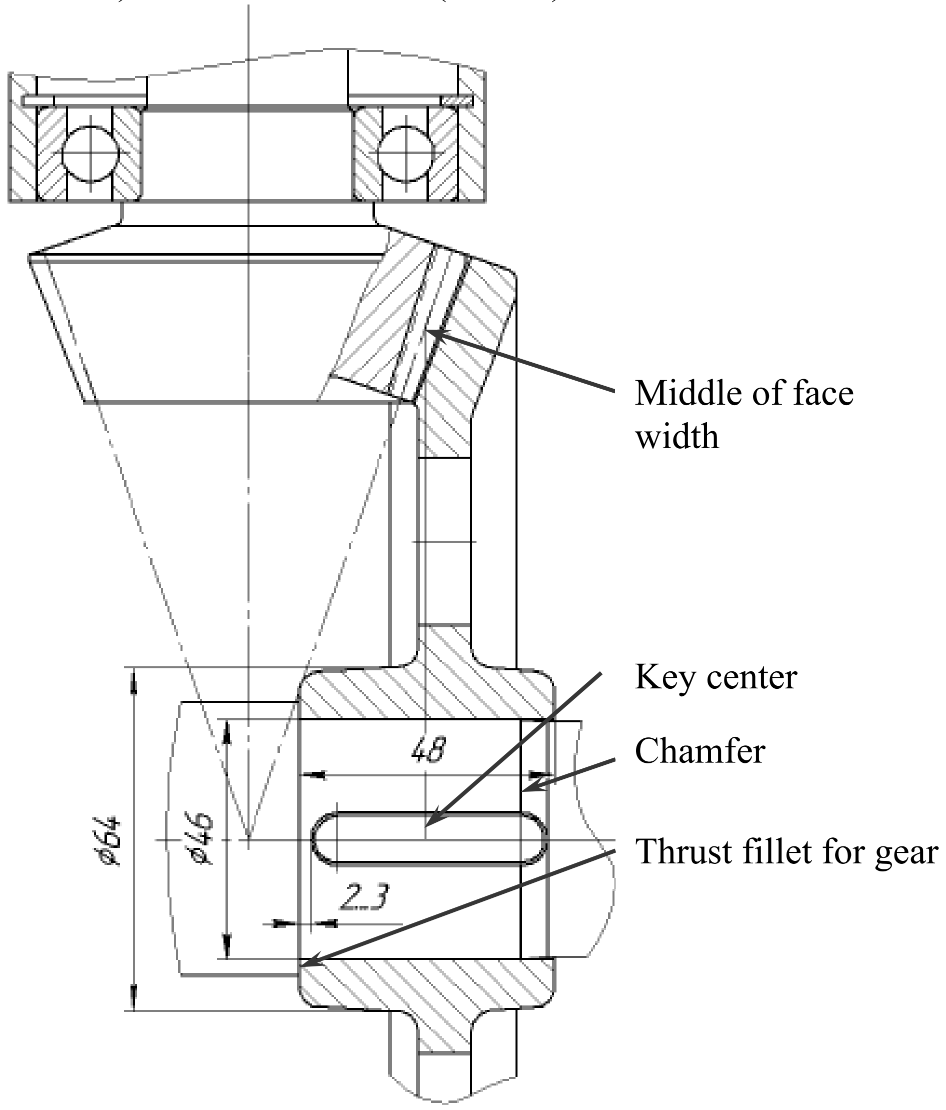
1. Let us begin from a gear hub. We selected bearings 209, and a gear should be between the bearings and at the separate diameter bigger than bearing bore one: $\boldsymbol{d_g = d_b + (1\ldots2)}$ mm = 45 + 1 = 46 mm (if $\boldsymbol{d_b}$ is even, preferable value is 2 mm).

To obtain a hub length it is necessary to calculate length of a gear-shaft key. It is recommended to use two keys on one shaft with the same smaller cross-section, therefore width and height should be like size of the key between the output shaft and coupling $\boldsymbol{b}$ x $\boldsymbol{h}$ = 10 x 8 mm. Thus (allowable stress is accepted the same):

$$\boldsymbol{l = \frac{2000T}{0{,}4d\,h[\sigma_b]} = \frac{2000\cdot 220}{0{,}4\cdot 46\cdot 8\cdot 80} = 37}\ \text{mm.}$$

The total length equals $\boldsymbol{L = l + b = 37 + 10 = 47}$ mm. This value is not standard (see Appendix D), therefore we must coordinate it with standard length and accept the closest value 48 mm.

We changed a gear dimension comparison with calculated above (see Fig. 2.10): increased a hub length (new value 48 mm, previous – 36 mm), a hole diameter (46 instead of 28 mm) and a hub diameter (64 mm).



Pay attention that:

− the key centre lies on the same line with the middle of the gear face width. Thus, we minimize additional bending moment into the page plane;

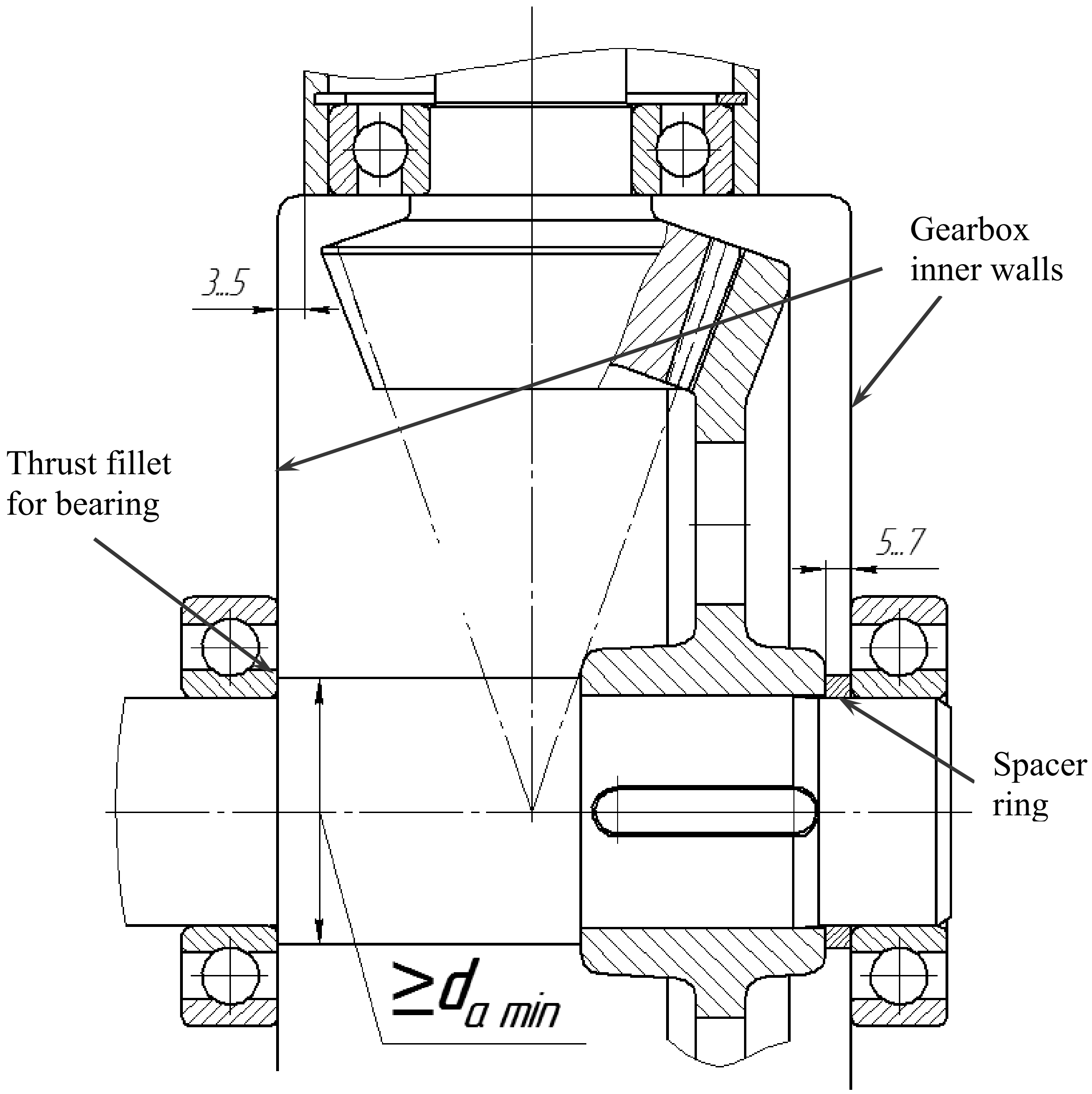
− clearance (2…3 mm) between the key seat and the shoulder decreases stress concentrations;

− a chamfer (or a conical ramp section) on the shaft makes assembly easier.

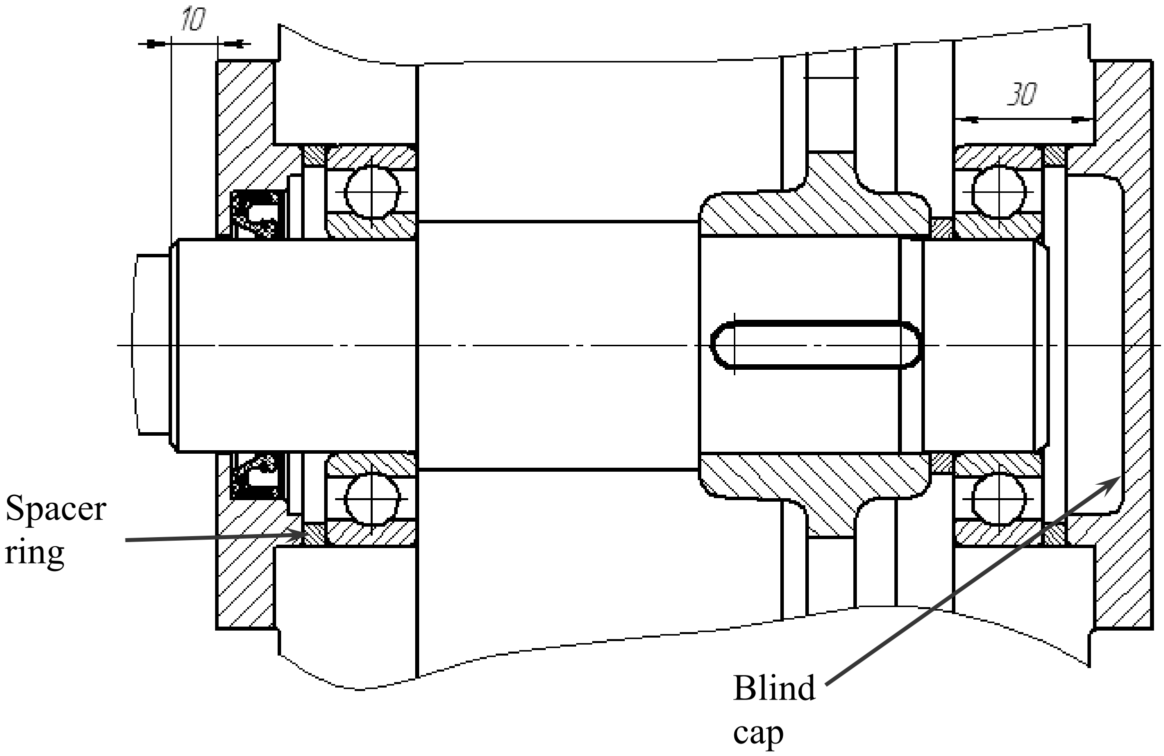
2. Place selected bearings. The distance between bearings is obtained geometrically to be enough to locate a pinion plus 3…7 mm at each side to avoid contact between rotating elements and immovable ones.

The shaft diameter between the gear hub and the left bearing has to be equal or bigger than minimum thrust fillet diameter $\boldsymbol{d_{amin}}$ (standard value, see subsection 3.5).

To retain the gear at its place a spacer ring is used.



3. Place the bearing caps: one of them (left) with recess for oil seal ($\boldsymbol{d = 45}$ mm, $\boldsymbol{D = 65}$ mm and $\boldsymbol{b = 10}$ mm) and another – without a hole (so called blind cap):



As the first approximation horizontal flange width is provided 30 mm.

4. We know output coupling shaft diameter ***d*** = 38 mm and conjunctive length ***l*** = 80 mm; key length ***L*** = 56 mm.

Finally, the output shaft unit looks like in Fig. 3.17.

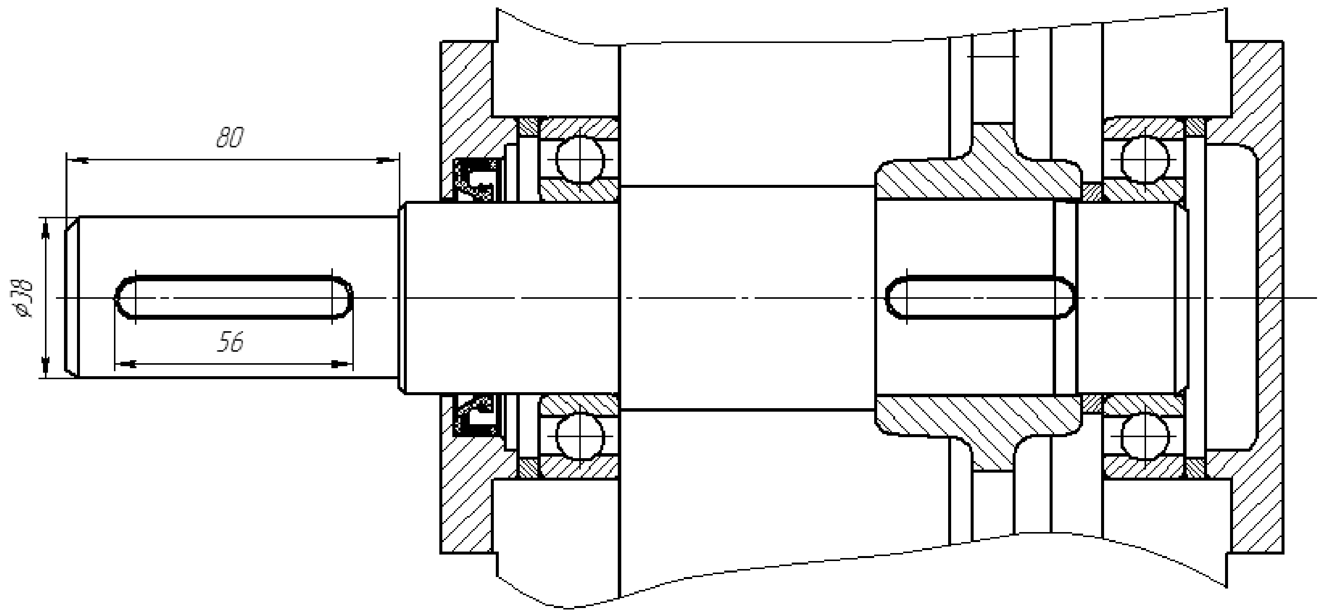


Fig. 3.17

4. CALCULATION OF BEARINGS ON DYNAMIC LOAD CAPACITY

4.1. General Statements

The type of bearings' capacity calculation depends on a ring rotational speed. Bearings are selected on static load capacity if external load is motionless or rotates slow ($n \le 1$ rpm). At ring rotational speed $n > 10$ rpm, bearings are calculated on dynamic load capacity. The bearings working at ring rotational speed $n > 10$ rpm and under sharply variable load, also it is necessary to check on static load capacity.

Bearings type and the positioning scheme are selected preliminary for a both 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

$$P^p L = C^p, \tag{4.1}$$

where P is the equivalent load, N; L is the service life in millions revolutions; C is the required dynamic load capacity, N; p is the exponent: for ball bearings p = 3, for roller bearings p =10/3.

It is possible to define dynamic load capacity from equation (4.1) if P and L are known:

$$C = \left(\frac{L}{a_1 a_{23}} \right)^{\frac{1}{p}} P, \tag{4.2}$$

and then select a bearing which satisfies condition

$$C \le C_c, \tag{4.3}$$

where C_c is the bearing dynamic load capacity taken from the catalogue or calculated on empirical dependences.

Also on the base of known P and C_c a preliminary selected bearing service life can be calculated as

$$L = a_1 a_{23} \left(\frac{C_к}{P} \right)^p, \tag{4.4}$$

and then calculated value should be compared with required one: $L_c \ge L_{req}$.

Service life in millions revolutions L and in hours L_h are connected by equation

$$L = \frac{60 n L_h}{10^6}, \tag{4.5}$$

where n is the rotating ring rotational speed, rpm.

In formulae (4.2) and (4.4) a_1 is the factor considering of reliability (Tab. 4.1).

Table 4.1

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 an lubricant film in contacts;

3) the same, as in 2), but at manufacturing of rings and rolling elements from electroslag or vacuum steel.

Values of factor a_{23} are shown in Tab. 4.2.

Table 4.2

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 under the following formulae:

– for the radial

$$P = V F_r K_B K_T; \tag{4.6}$$

– for the thrust

$$P = F_a K_B K_T; \tag{4.7}$$

– for radial-thrust

$$P = (XVF_r + YF_a) K_B K_T, \tag{4.8}$$

where F_r and F_a are the radial and axial loads on a bearing, N; V is the rotation factor (at rotation of an internal ring about a load vector V = 1, external V = 1,2); K_B is the safety factor considering load non-uniformity (in case of light load K_B = 1, with medium non-uniformity K_B = 1,3…1,8, with heavy load K_B = 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 (Tab. 4.3).

If condition (4.3) is not satisfied or the service life is too small, it is necessary:

– select bearings of heavier series or other bearing type;

– or increase a shaft diameter;

– or 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.

Table 4.3

Contact angle α, ^{o}	**Relative load F_a/C_o**	e*	$\frac{F_a}{VF_r} \le e$		$\frac{F_a}{VF_r} > e$	
			X	$Y = \frac{1-X}{e}$	X	$Y = \frac{1-X}{e}$
Ball radial deep groove						
	0,014	0,19				2,30
	0,028	0,22				1,99
	0,056	0,26				1,71
	0,084	0,28				1,55
0	0,110	0,30	1	0	0,56	1,45
	0,170	0,34				1,31
	0,280	0,38				1,15
	0,420	0,42				1,04
	0,560	0,44				1,00
Ball radial-thrust						
	0,014	0,30				1,81
	0,029	0,34				1,62
	0,057	0,37				1,46
	0,086	0,41				1,34
12	0,110	0,45	1	0	0,46	1,22
	0,170	0,48				1,13
	0,290	0,52				1,04
	0,430	0,54				1,01
	0,570	0,54				1,00
	0,015	0,38				1,47
	0,029	0,40				1,40
	0,058	0,43				1,30
	0,087	0,46				1,23
15	0,120	0,47	1	0	0,44	1,19
	0,170	0,50				1,12
	0,290	0,55				1,02
	0,440	0,56				1,00
	0,580	0,56				1,00
20		0,57			0,43	1,00
25		0,68			0,41	0,87
30	-	0,8	1	0	0,39	0,76
35		0,95			0,37	0,66
40		1,14			0,35	0,57
Tapered roller						
-	-	1,5tanα	1	0	0,4	0,4ctanα

*For ball radial deep groove bearing intermediate value $e = 0{,}518\left(\frac{F_a}{C_0}\right)^{0{,}24}$

The sequence of the radial and axial load factor calculation is shown in Fig. 4.1.

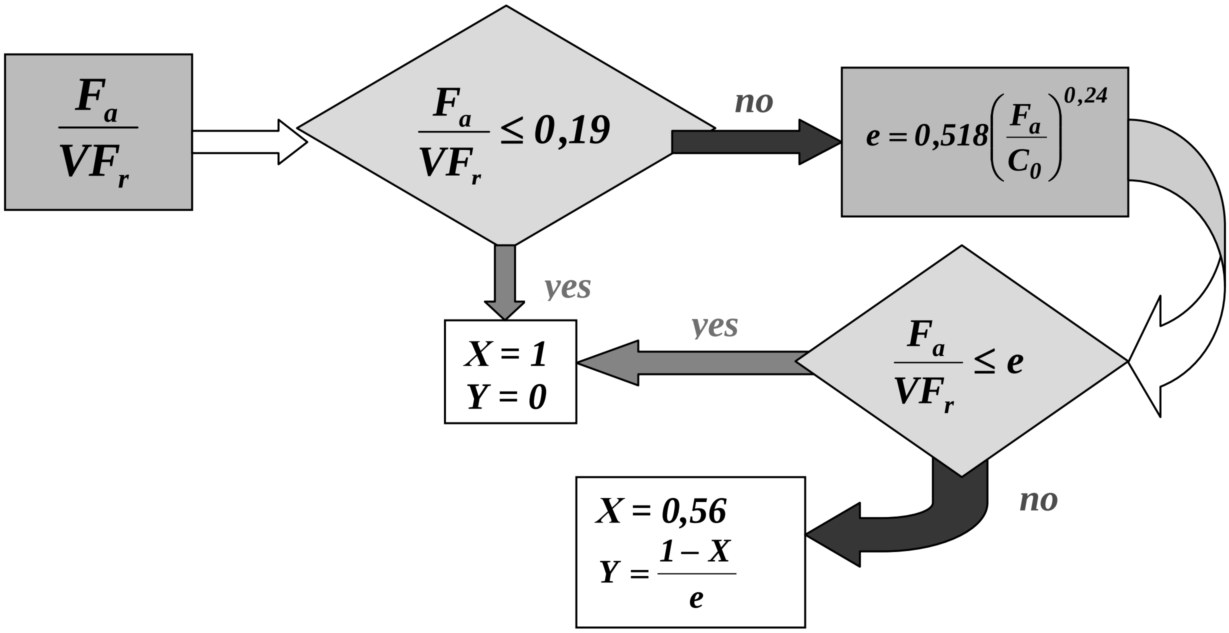


Fig. 4.1

4.2. Calculation of Loading on Bearings

To define an equivalent load $\boldsymbol{P}$ the radial and axial loads $\boldsymbol{F_r}$ and $\boldsymbol{F_a}$ should be obtained. Each of them is a total reaction in a support: $\boldsymbol{F_r}$ is a reaction in a plane perpendicular to the longitudinal shaft axis, $\boldsymbol{F_a}$ – along this axis.

The forces acting on shaft and causing reactions $\boldsymbol{F_r}$ and $\boldsymbol{F_a}$, 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.

For designing of tooth gearings' support, a normal force $\boldsymbol{F_n}$ and force of friction $\boldsymbol{F_f}$ should be known. A friction force in gearing is usually neglected, because the friction factor is rather small. Normal force $\boldsymbol{F_n}$ is resolved on three components: a radial $\boldsymbol{F_r}$; a peripheral $\boldsymbol{F_t}$; an axial $\boldsymbol{F_a}$.

Normal force components can be calculated by following formulae:

– for spur straight teeth gears:

$$F_t = \frac{2T_1}{d_{w1}}; \quad F_r = F_t \tan \alpha_w; \quad F_a = 0; \tag{4.9}$$

– for spur helical gears:

$$F_t = \frac{2T_1}{d_{w1}}; \quad F_r = \frac{F_t \tan \alpha_w}{\cos \beta}; \quad F_a = F_t \tan \beta; \tag{4.10}$$

– for straight teeth bevel gears:

$$F_t = \frac{2T_1}{d_{m1}}; \quad F_r = F_t \tan \alpha_w \cos \delta; \quad F_a = F_t \tan \alpha_w \sin \delta \; ; \quad (4.11)$$

where T_1 is the torque on the driving shaft; d_{w1} is the pinion pitch diameter; α_n is the pressure angle (the most common value without shift $\alpha_w = 20°$); β is the teeth helix angle ($\beta = 7…20^o$; for strength teeth $\beta = 0$); δ_1 is the pinion pitch cone angle ($\delta_1 = atan\,(z_1 / z_2)$); 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 (Tab. 4.4).

Table 4.4

Coupling type	Force F'_C (T is a transferring torque, Nm)
Flexible coupling with rubber-bushed studs	$2000T/D_1$ (D_1 is a 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 maximum loaded.

4.3. 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. 4.2).

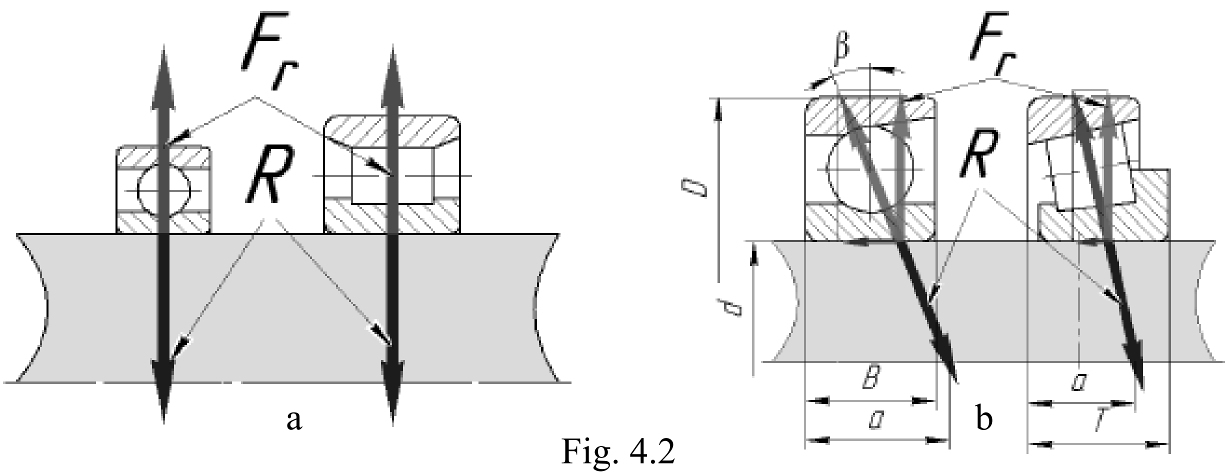


Fig. 4.2

For radial bearings, this point is in the middle of their width (Fig. 4.2, a), and for radial-thrust ball and roller the distance ***a*** between this point and a bearing’s outer ring face (Fig. 4.2, b) can be defined graphically or analytically:

- for the angular contact single row ball bearings

$$\boldsymbol{a = 0{,}5\ [B + 0{,}5\ (d + D)\ tg\ \alpha];}$$

- for tapered roller bearings

$$\boldsymbol{a = 0{,}5\ [T + e\ (d + D)/3].}$$

Ring width ***B***, assembly height ***T***, axial load factor ***e***, contact angle ***α***, diameters ***d*** and ***D*** are shown in the catalogues.

Shift of a radial reaction applying point changes a distance between supports ***L'*** (Fig. 4.3) for the same distance between their middles ***L***.

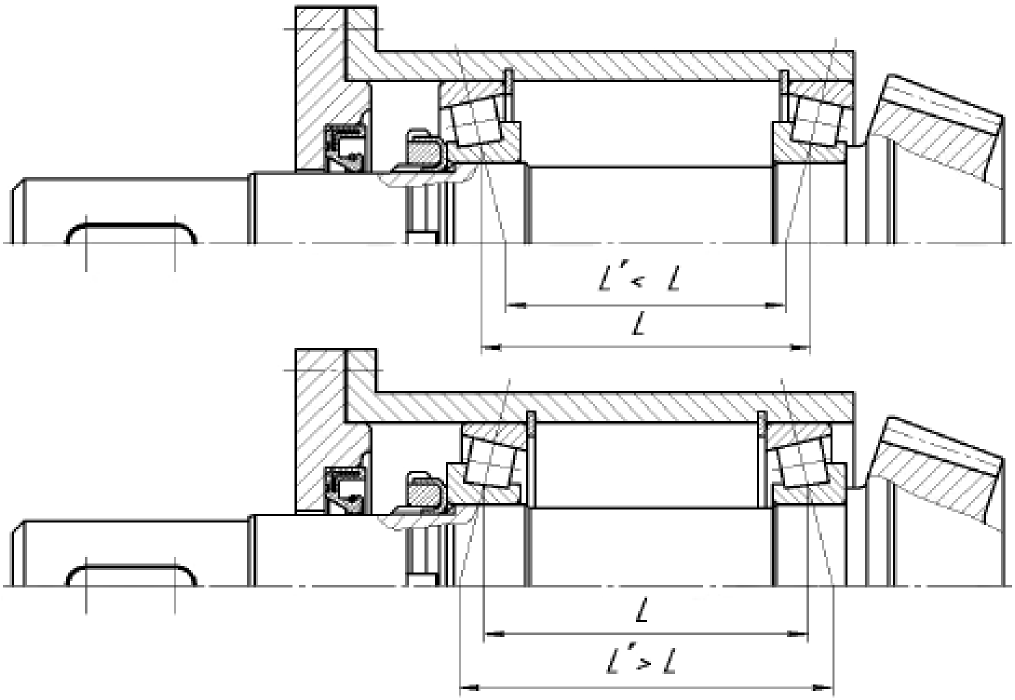


Fig. 4.3

Radial reactions define from the equilibrium equation: the sum of the external force moments and the reaction moments is equal to zero.

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 $\boldsymbol{R_a}$ is equal to the actual external axial force $\boldsymbol{F_a}$. The force is taken by that bearing which fixes a shaft axial shift at one or two directions.

Let us consider a number of typical cases of gearing shaft loading (shaft of other mechanisms, for example worm gear, it is possible to reduce to similar schemes). ***Attention! For correct usage formulae below it is necessary to reduce a considered shaft to the same order of supports and gears arrangement.***

1. A cantilever bevel pinion is placed on a shaft (Fig. 4.4) (to define the components $\boldsymbol{F_r}$, $\boldsymbol{F_t}$ and $\boldsymbol{F_a}$ see formulae (4.11)).

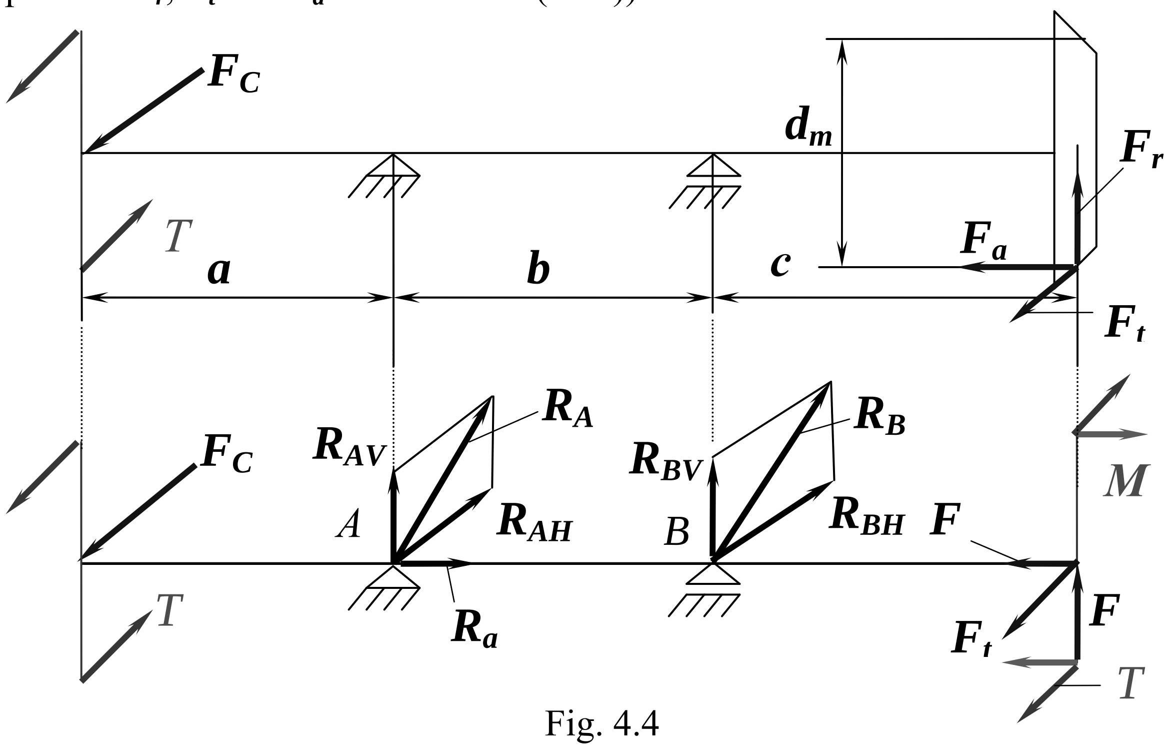


Fig. 4.4

Let us write equilibrium condition for two perpendicular planes (it is conventionally assumed that plane where forces $\boldsymbol{F_r}$ and $\boldsymbol{F_a}$ act on is named «vertical»; analogically «horizontal» is a plane where force $\boldsymbol{F_t}$ acts on):

– the sum of the moments about point A in vertical plane

$$\sum_{i=1}^{N} M_{iA} = R_{BV}\, b - M + F_r (b + c) = 0\ ;$$

in horizontal plane

$$\sum_{i=1}^{N} M_{iA} = R_{BH}\, b - F_t (b + c) + F_C\, a = 0\ ;$$

– the sum of the moments about point B in vertical plane

$$\sum_{i=1}^{N} M_{iB} = -R_{AV}\, b - M + F_r c = 0\ ;$$

in horizontal plane

$$\sum_{i=1}^{N} M_{iB} = -R_{AH}\, b - F_t c + F_C\, (a + b) = 0\ ;$$

where the moment caused by the axial force $\boldsymbol{F_a}$ equals $M = F_a \dfrac{d_m}{2}$.

Thus the radial reactions

$$R_{AV} = \frac{F_r c - M}{b};\ R_{AH} = \frac{F_C (a + b) - F_t c}{b};\ R_A = \sqrt{R_{AV}^2 + R_{AH}^2}\ ;$$

$$R_{BV} = \frac{M - F_r (b + c)}{b};\ R_{BH} = \frac{F_t (b + c) - F_C a}{b};\ R_B = \sqrt{R_{BV}^2 + R_{BH}^2}\ ;$$

and the axial reaction $R_a = F_a$.

2. A helical spur gear and coupling are placed on a shaft (in case of bevel straight teeth gear all formulae below are the same) (Fig. 4.5).

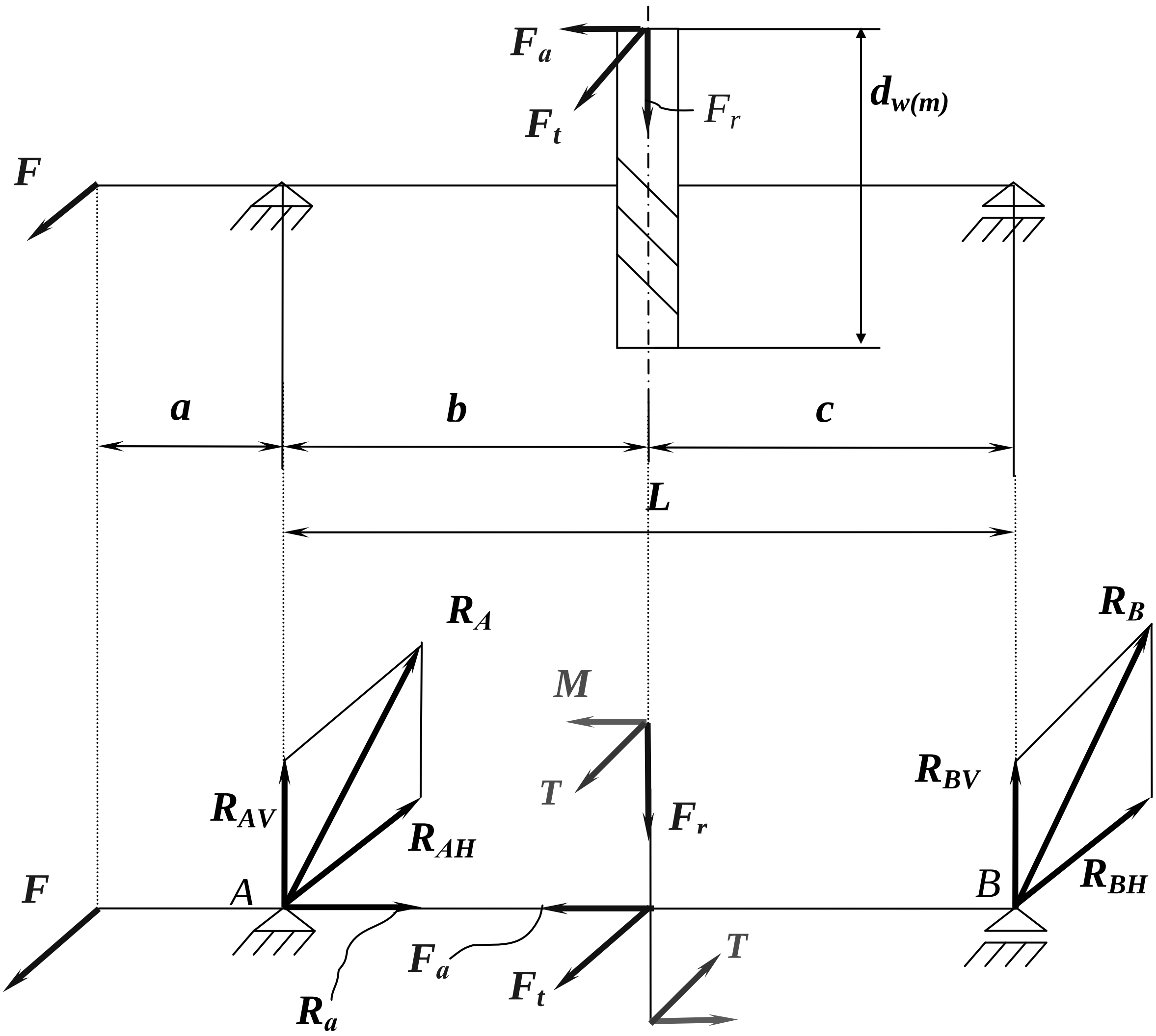


Fig. 4.5

A moment cased by the axial force $\boldsymbol{F_a}$ equals $\boldsymbol{M = F_a \dfrac{d_{w(m)}}{2}}$.

The radial reactions are

$$\boldsymbol{R_{AV} = \frac{F_r c + M}{L}};\quad \boldsymbol{R_{AH} = \frac{F_t c + F_C (L + a)}{L}};\quad \boldsymbol{R_A = \sqrt{R_{AV}^2 + R_{AH}^2}};$$

$$\boldsymbol{R_{BV} = \frac{F_r b - M}{L}};\quad \boldsymbol{R_{BH} = \frac{F_t b - F_C a}{L}};\quad \boldsymbol{R_B = \sqrt{R_{BV}^2 + R_{BH}^2}}.$$

The axial reaction is $\boldsymbol{R_a = F_a}$.

In case of **spur straight teeth** gear $\boldsymbol{F_a = 0}$ and $\boldsymbol{M = 0}$.

3. Two helical spur gears are placed on a shaft (one or both of gears may be bevel straight teeth gear) (Fig. 4.6).

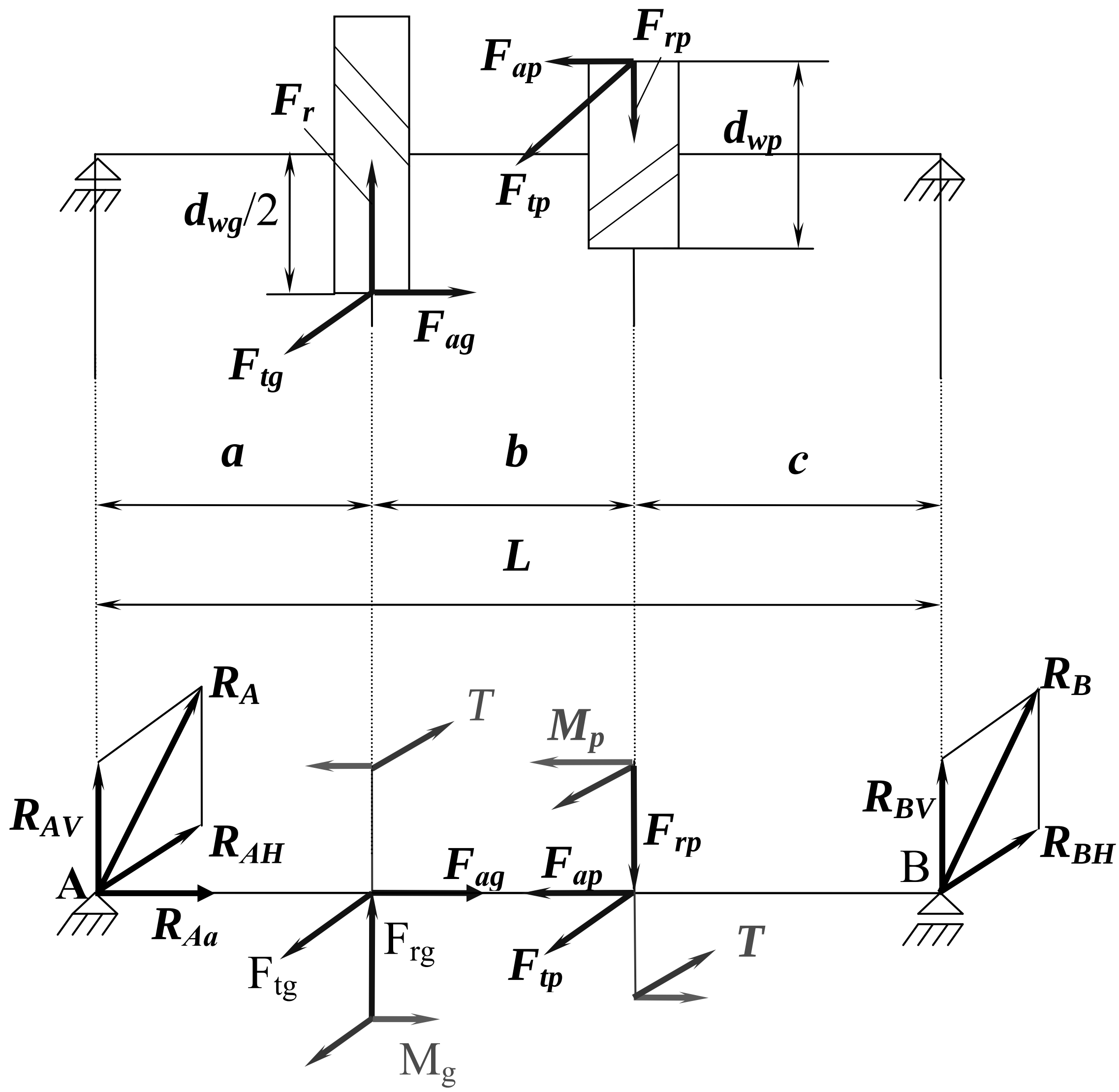


Fig. 4.6

In such case

$$M_g = F_{ag}\frac{d_{wg}}{2};\ \ M_p = F_{ap}\frac{d_{wp}}{2}\ \ R_a = F_{ap} - F_{ag};$$

$$R_{AV} = \frac{F_{rp}c - F_{rg}(b+c) + M_p + M_g}{L};\ \ R_{BV} = \frac{F_{rp}(a+b) - F_{rg}a - M_p - M_g}{L};\ \ R_A = \sqrt{R_{AV}^2 + R_{AH}^2};$$

$$R_{AH} = \frac{F_{tg}(b+c) + F_{tp}c}{L};\quad R_{BH} = \frac{F_{tg}a + F_{tp}(a+b)}{L};\quad R_B = \sqrt{R_{BV}^2 + R_{BH}^2}.$$

In any gear is **spur straight teeth** gear, its axial force $F_{ag(p)} = 0$ and $M_{g(p)} = 0.$

4. A helical spur or bevel gear and two couplings are placed on one shaft (Fig. 4.7):

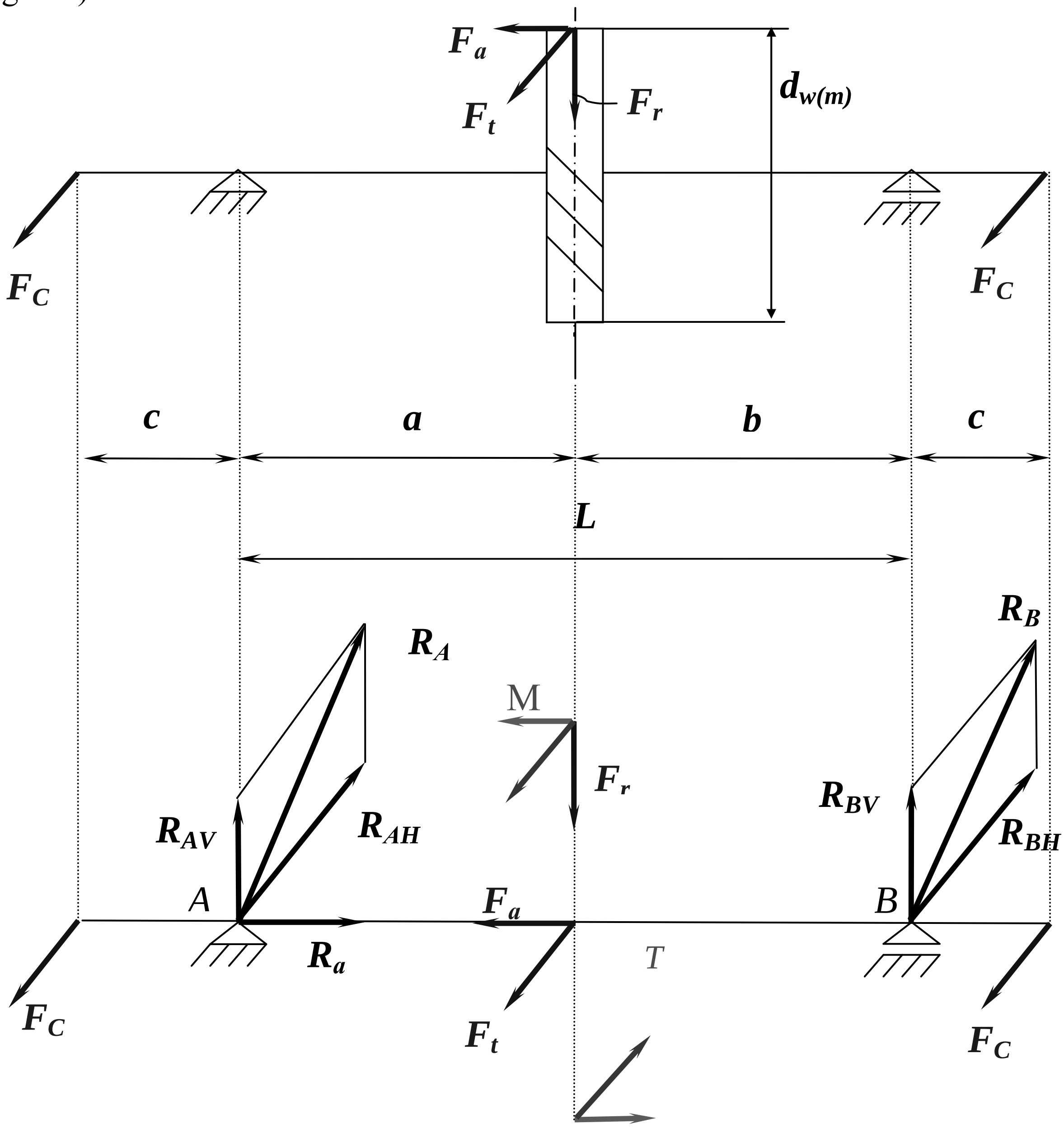


Fig. 4.7

A moment cased by the axial force F_a, equals $M = F_a \dfrac{d_{w(m)}}{2}$;

The radial reaction components and reactions are

$$R_{AV} = \frac{F_r b + M}{L}; \quad R_{AH} = F_C + F_t \frac{b}{L}; \quad R_A = \sqrt{R_{AV}^2 + R_{AH}^2};$$

$$R_{BV} = \frac{F_r a - M}{L}; \quad R_{BH} = F_C + F_t \frac{a}{L}; \quad R_B = \sqrt{R_{BV}^2 + R_{BH}^2}.$$

The axial reaction is $R_a = F_a$.

In case of **spur straight teeth** gear $F_a = 0$ and $M = 0$.

4.4. Examples

Example 1

Check the bearings selected for the input shaft of spur gearbox described above. Initial data see in Tab. 4.5.

Table 4.5

Parameter		Symbol	Value
The torque, N m		T	54,4
Rotational speed, rpm		n	965
The distances between	coupling and support A	a	84
	support A and pinion	b	38
	pinion and support B	c	38
Pressure angle		α_w	20
Preliminary bearing code			208
Bearing dynamic load capacity, N		C	32 000
Bearing static load capacity, N		C_0	17 800

Distances between acting forces and reactions applying points a, b, c are measured in the drawing (Fig. 4.8).

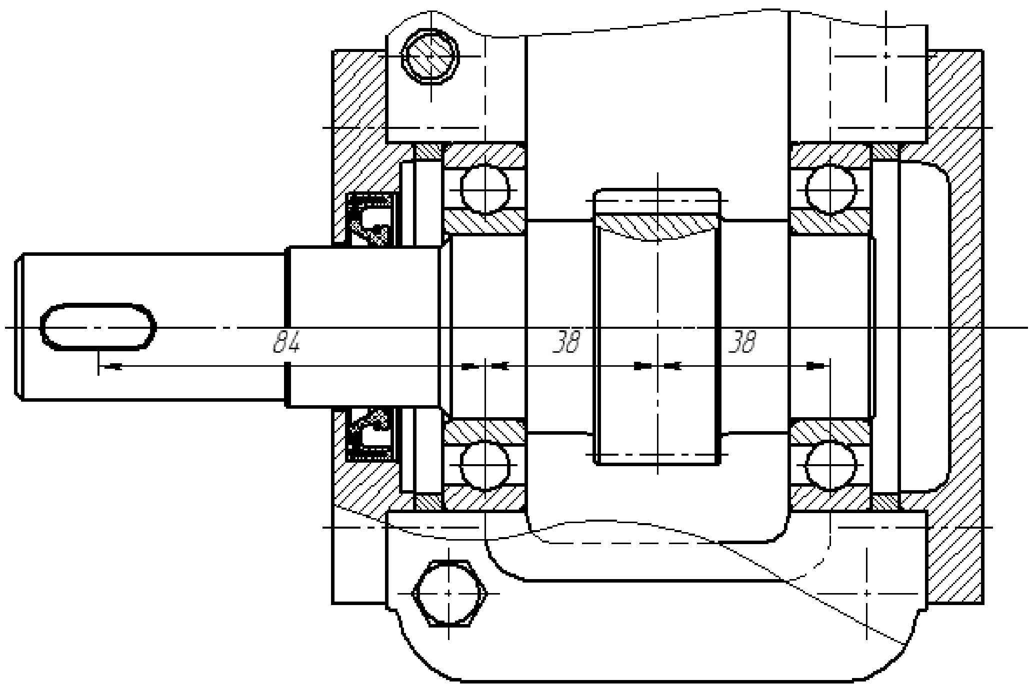


Fig. 4.8

Calculating scheme looks like in Fig. 4.5, but axial force $F_a = 0$ and $M = 0$. Calculation results are shown in Tab. 4.6 (distance $L = b + c = 38 + 38 = 76$ mm).

Table 4.6

Parameter		Formula	Value
The peripheral force F_t, N		From gearing calculation (see Tab. 2.28)	1 979
The radial force F_r, N		$F_r = F_t \tan\alpha_w = 1\,979 \cdot \tan 20^\circ$	720
The clutch force F_C, N		see Tab. 4.4: $F_C = 0{,}1 \cdot \frac{2000T}{D_1} = 0{,}1 \cdot \frac{2000 \cdot 54{,}4}{105}$	103,6
Support A radial reaction components, N	Vertical	$R_{AV} = F_r c/L = \frac{720 \cdot 38}{76}$	360
	Horizontal	$R_{AH} = \frac{F_t c + F_C(L+a)}{L} = \frac{1979 \cdot 38 + 103{,}6(76+84)}{76}$	1 206
Support B radial reaction components, N	Vertical	$R_{BV} = F_r b/L = \frac{720 \cdot 38}{76}$	360
	Horizontal	$R_{BH} = \frac{F_t b - F_C a}{L} = \frac{1979 \cdot 38 - 103{,}6 \cdot 84}{76}$	876
Support A radial reaction, N		$R_A = \sqrt{R_{AV}^2 + R_{AH}^2} = \sqrt{360^2 + 1206^2}$	1 259
Support B radial reaction, N		$R_B = \sqrt{R_{BV}^2 + R_{BH}^2} = \sqrt{360^2 + 876^2}$	947

Thus, support A radial load higher than support B one. According to design and technological reasons, we use in both supports the same bearings, therefore bearing in support A should be calculated on dynamic load capacity. In such case, bearing radial load F_r in equation (4.8) is the support A radial load: $F_r = R_A = 1\,259\ N$.

Let us assume:

– a reliability factor a_1 for normal reliability 0,9 $a_1 = 1$ (see Tab. 4.1);

– for usual operating conditions and ball bearings factor a_{23} equals 0,8 (see Tab. 4.2);

– the safety factor considering load non-uniformity for medium non-uniformity $K_B = 1{,}3$;

– the temperature factor $K_T = 1$ because operating temperature should be less 100 °C (we will use an oil lubricant and transferring power relatively small).

Service life in million revolutions according to formula (4.5)

$$L = \frac{60nL_h}{10^6} = \frac{60 \cdot 965 \cdot 20\,000}{10^6} = 1\,158,$$

where L_h is the given service life in hours (given data).

Thus, equivalent load

$$P = VF_r K_B K_T = 1 \cdot 1\,259 \cdot 1{,}3 \cdot 1 = 1\,634 \text{ N}.$$

For ball bearings an exponent $p = 3$, so finally the dynamic load capacity

$$C = \left(\frac{L}{a_1 a_{23}}\right)^{\frac{1}{p}} P = \left(\frac{1\,158}{1 \cdot 0{,}8}\right)^{1/3} 1\,634 \;= 18\,513 \text{ N}.$$

Strength condition (4.3) is satisfied: ***18 513 < 32 000*** N, therefore we can use previously selected bearings.

Coordinate final decision about bearing code with teacher.

Example 2

Check the bearing selected for the input shaft of bevel gearbox described above. Initial data see in Tab. 4.7.

Table 4.7

Parameter		Symbol	Value
The torque, N m		T	80,7
Rotational speed, rpm		n	870
Pinion pitch angle, deg		δ_1	19,03
The distances between	coupling and support A	a	86
	support A and support B	b	80
	support B and pinion	c	33
Pressure angle		α_w	20
Preliminary bearing code			208
Bearing dynamic load capacity, N		C	32 000
Bearing static load capacity, N		C_0	17 800

Distances between acting forces and reactions applying points ***a***, ***b***, ***c*** are measured in the drawing (Fig. 4.9).

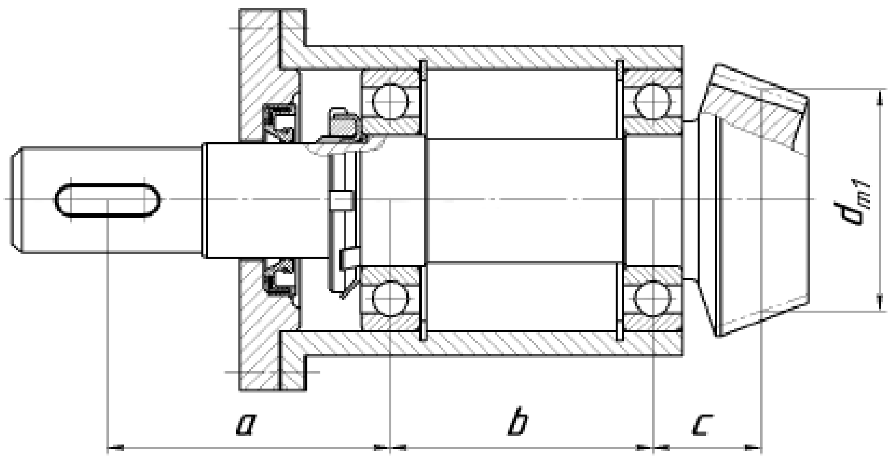


Fig. 4.9

Calculating scheme looks like in Fig. 4.4. Calculation results are shown in Tab. 4.8.

Table 4.8

Parameter		Formula	Value
The peripheral force F_t, N		From gearing calculation (Fig. 4.10)	2 367
The radial force F_r, N		$F_r = F_t \tan\alpha_w \cos\delta_1 = 2366 \cdot \tan 20^\circ \cdot \cos 19{,}03$	818
The axial force F_a, N		$F_a = F_t \tan\alpha_w \sin\delta_1 = 2366 \tan 20^\circ \sin 19{,}03$	269
The bending moment M, Nmm		$M = F_a \frac{d_{m1}}{2} = 269 \frac{68{,}22}{2}$	9 176
The clutch force F_C, N		see Tab. 4.4: $F_C = 0{,}1 \cdot 50\sqrt{T} = 0{,}1 \cdot 50\sqrt{80{,}7}$	45
Support A radial reaction components, N	Horizontal	$R_{AH} = \frac{F_C(a+b) - F_t c}{b} = \frac{45(86+80) - 2366 \cdot 33}{80}$	- 883
	Vertical	$R_{AV} = \frac{F_r c - M}{b} = \frac{818 \cdot 33 - 9176}{80}$	223
Support B radial reaction components, N	Horizontal	$R_{BH} = \frac{F_t(b+c) - F_C a}{b} = \frac{2366(80+33) - 45 \cdot 86}{80}$	3 294
	Vertical	$R_{BV} = \frac{M - F_r(b+c)}{b} = \frac{9176 - 818(80+33)}{80}$	-1 041
Support A radial reaction, N		$R_A = \sqrt{R_{AV}^2 + R_{AH}^2} = \sqrt{223^2 + 883^2}$	938
Support B radial reaction, N		$R_B = \sqrt{R_{BV}^2 + R_{BH}^2} = \sqrt{1041^2 + 3294^2}$	3455

Negative values of R_{AH} and R_{BV} mean that these components act in opposite direction.

Load distribution non-uniformity factor $K_{H\beta}$	1,1
Pinion pitch diameter, mm	75,7
Pinion number of teeth z_1	20
Module m, mm	3,78
Calculated face width b_w, mm	25,3

Tangential force F_t, N	2367
Internal dynamic load factor K_{Hv}	1,21
Load distribution non-uniformity factor $K_{H\beta}$	1,36
Load factor K_H	1,67
Actual contact stress σ_H, MPa	737
Relative contact stress ε	**0,5**
Actual peak contact stress σ_{Hmax}, MPa	

h		
N_F	0,88	
Load distribution non-uniformity factor $K_{F\beta}$	1,35	
Load factor K_F	1,66	
Actual bending stress σ_F, MPa	277	248
Relative bending stress ε	**-21,9**	**-29,0**
Actual peak bending stress σ_{Fmax}, MPa		

Final Parameters				
Bevel			*pinion*	*gear*
Transfer number u	2,90	Number of teeth z	26	75
Standard module m, mm	3,00	Rotational speed n, rpm	870	300
Cone distance R_e, mm	119,068	Addendum diam., mm	83,669	226,965
Face width b, mm	30,0	Pitch diam., mm	78,00	225,00
Relative face width	0,25	Root diam., mm	71,197	222,642
b/m	10,00	Mean pitch diam., mm	68,174	196,655
□		Pitch cone angle, deg	19,12	70,88

Fig. 4.10

Thus, support B radial load higher than support A one. According to design and technological reasons, we use in both supports the same bearings, therefore bearing in support B should be calculated on dynamic load capacity. In such case, bearing radial load $\boldsymbol{F_r}$ in equation (4.8) is the support B radial load: $\boldsymbol{F_r = R_B = 3\ 455}$ N.

Let us assume:

– a reliability factor $\boldsymbol{a_1}$ for normal reliability 0,9 $\boldsymbol{a_1} = 1$ (see Tab. 4.1);

– for usual operating conditions and ball bearings factor $\boldsymbol{a_{23}}$ equals 0,8 (see Tab. 4.2);

– the safety factor considering load non-uniformity for medium non-uniformity $\boldsymbol{K_B} = 1{,}3$;

– the temperature factor $\boldsymbol{K_T} = 1$ because operating temperature should be less 100 ^{o}C (we will use an oil lubricant and transferring power relatively small).

Service life in million revolutions according to formula (4.5)

$$\boldsymbol{L = \frac{60nL_h}{10^6} = \frac{60 \cdot 870 \cdot 20\,000}{10^6} = 1044},$$

where $\boldsymbol{L_h}$ is the given service life in hours (given data).

Let us determine radial and axial load factors $\boldsymbol{X}$ and $\boldsymbol{Y}$. First, calculate

$$\boldsymbol{\left(\frac{F_a}{C_0}\right) = \frac{269}{17\ 800} = 0{,}015 > 0{,}014}\ .$$

According to Fig. 4.1 define (inner ring rotates about a load vector, therefore rotation factor $\boldsymbol{V} = 1$)

$$\boldsymbol{e = 0{,}518\left(\frac{F_a}{C_0}\right)^{0,24} = 0{,}518 \cdot 0{,}015^{0,24} = 0{,}189} \text{ and } \boldsymbol{\frac{F_a}{VF_r} = \frac{269}{1 \cdot 3455} = 0{,}07 < e}\ .$$

Pay attention, that here F_r is **support B radial load, not radial force in the gearing!**

Thus, $X = 1$ and $Y = 0$.

Thus, equivalent load

$$P = (XVF_r + YF_a)\, K_B K_T = (1\cdot 1\cdot 3455 + 0\cdot 269)\cdot 1{,}3\cdot 1 = 4492 \text{ N.}$$

For ball bearings exponent $p = 3$, so finally the dynamic load capacity

$$C = \left(\frac{L}{a_1 a_{23}}\right)^{\frac{1}{p}} P = \left(\frac{1044}{1\cdot 0{,}8}\right)^{1/3} 4492 = 49\,080 \text{ N.}$$

Strength condition (4.3) is not satisfied: $49\ 080 > 32\ 000$ N, therefore we must change something. There are some ways:

– change previous selected bearing (Fig. 4.11, a). Look in Appendix E: we can take bearing #408 with $C = 63\ 700$ N and outer diameter $D = 110$ mm (Fig. 4.11, b);

– increase distance between supports, for example, up to 100 mm (Fig. 4.12, a). Thus, recalculated reaction components $R_{BH} = 2\ 631$ N, $R_{BV} = 833$ N, support B reaction $R_B = 2\ 760$ N, equivalent load $P = 3\ 588$ N and dynamic load capacity $C = 39\ 200$ N. In such case we may use bearing #308 ($C = 41\ 000$ N; $D = 90$ mm);

– increase shaft diameter up to 45 mm (Fig. 4.12, b) and use bearing #309 ($C = 52\ 700$ N; $D = 100$ mm).

Coordinate final decision about bearing code with teacher.

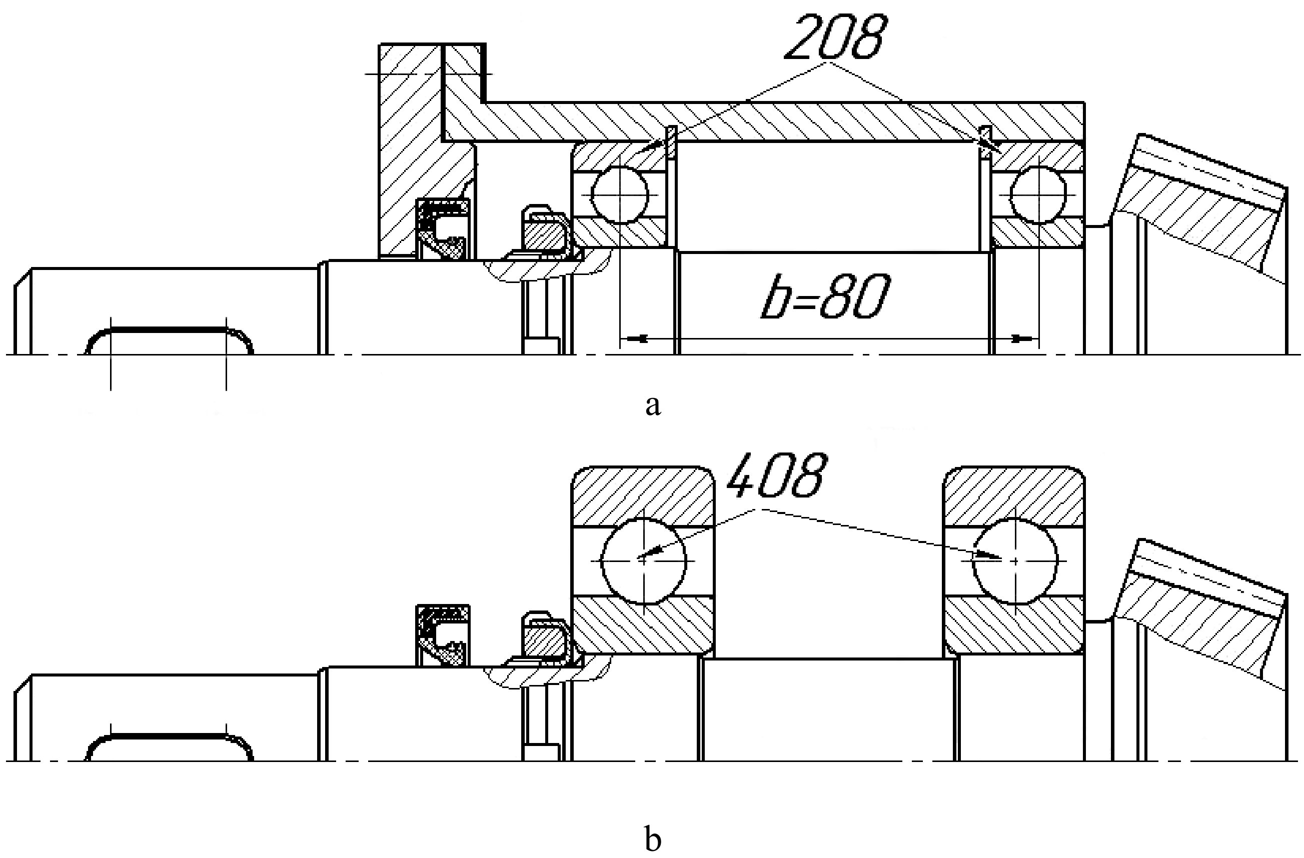
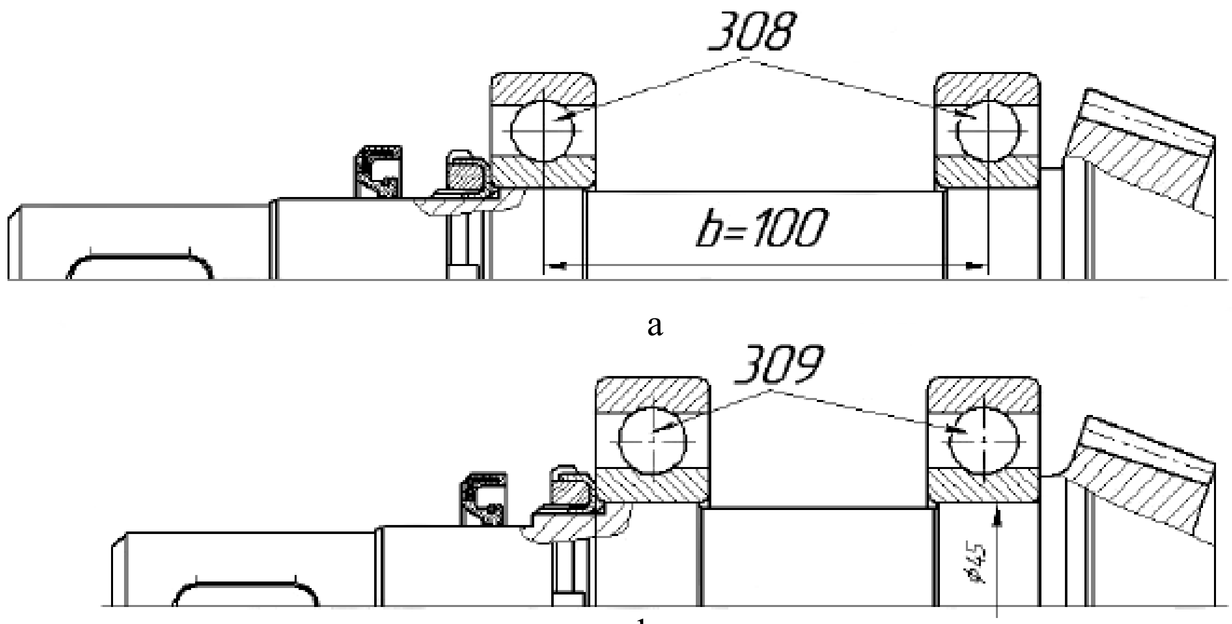


Fig. 4.11



a

b

Fig. 4.12

Example 3

Check the bearing selected for the bevel gearbox output shaft described above (see initial data in Tab. 4.9).

Distances between acting forces and reactions applying points ***a***, ***b***, ***c*** are measured in the drawing (Fig. 4.12). Pay attention that drawing was turned around vertical axis according to the calculation scheme.

Table 4.9

<table>
<tr><th colspan="2">Parameter</th><th>Symbol</th><th>Value</th></tr>
<tr><td colspan="2">The torque, N mm</td><td>T</td><td>220</td></tr>
<tr><td colspan="2">Rotational speed, rpm</td><td>n</td><td>300</td></tr>
<tr><td colspan="2">Pinion pitch angle, deg</td><td>δ_2</td><td>70,97</td></tr>
<tr><td rowspan="3">The distances between</td><td>support A and gear</td><td>a</td><td>34</td></tr>
<tr><td>gear and support B</td><td>b</td><td>92</td></tr>
<tr><td>support B and coupling</td><td>c</td><td>82</td></tr>
<tr><td colspan="2">Pressure angle</td><td>α_w</td><td>20</td></tr>
<tr><td colspan="2">Preliminary bearing code</td><td></td><td>209</td></tr>
<tr><td colspan="2">Bearing dynamic load capacity, N</td><td>C</td><td>33 200</td></tr>
<tr><td colspan="2">Bearing static load capacity, N</td><td>C_0</td><td>18 600</td></tr>
</table>

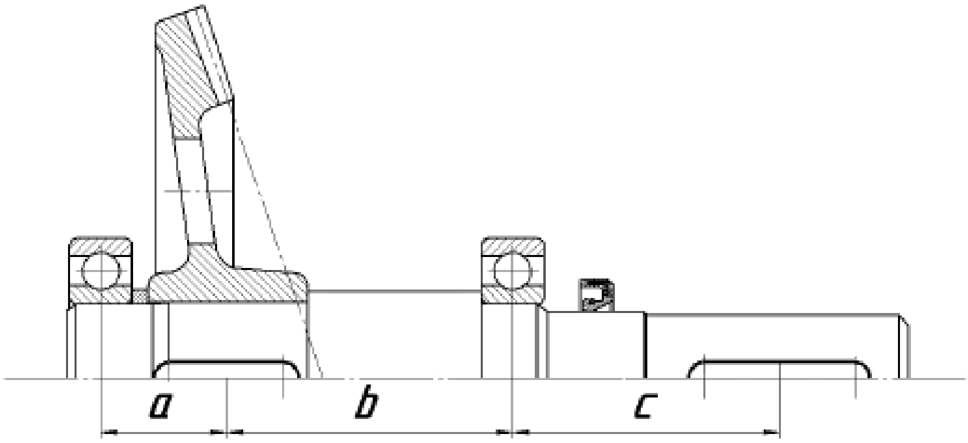


Fig. 4.12

Calculating scheme looks like mirrored to the Fig. 4.5, therefore equations for reaction components **are changed**. Calculation results are shown in Tab. 4.10.

Table 4.10

Parameter		Formula	Value
The peripheral force $\boldsymbol{F_t}$, N		From gearing calculation (the same with the input shaft)	2 366
The radial force $\boldsymbol{F_r}$, N		$F_r = F_t \tan\alpha_w \cos\delta_2 =$ $= 2367 \cdot \tan 20^\circ \cdot \cos 70{,}88$	269
The axial force $\boldsymbol{F_a}$, N		$F_a = F_t \tan\alpha_w \sin\delta_2 =$ $= 2367 \tan 20^\circ \sin 70{,}88$	818
The bending moment $\boldsymbol{M}$, Nmm		$M = F_a \dfrac{d_{m2}}{2} = 818 \dfrac{196{,}7}{2}$	80 490
The clutch force $\boldsymbol{F_C}$, N		see Tab. 4.4: $F_C = 0{,}1 \cdot 2000T / d = 0{,}1 \cdot 2000 \cdot 220 / 38$	1 158
Support A radial reaction components, N	Vertical	$R_{AV} = \dfrac{F_r b + M}{L} = \dfrac{269 \cdot 92 + 80490}{34 + 92}$	973
	Horizontal	$R_{AH} = \dfrac{F_t b - F_C c}{L} = \dfrac{2366 \cdot 92 - 1158 \cdot 82}{34 + 92}$	835
Support B radial reaction components, N	Vertical	$R_{BV} = \dfrac{F_r a - M}{L} = \dfrac{269 \cdot 34 - 80490}{126}$	2 550
	Horizontal	$R_{BH} = \dfrac{F_t a + F_C (L + c)}{L} =$ $= \dfrac{2366 \cdot 34 + 1158 \cdot (126 + 82)}{126}$	-566
Support A radial reaction, N		$R_A = \sqrt{R_{AV}^2 + R_{AH}^2} = \sqrt{223^2 + 883^2}$	1 282
Support B radial reaction, N		$R_B = \sqrt{R_{BV}^2 + R_{BH}^2} = \sqrt{1041^2 + 3294^2}$	2 612

Negative value of R_{BV} means that this component has opposite direction.

Thus, support B radial load higher than support A one. According to design and technological reasons, we use in both supports the same bearings, therefore bearing in support B should be calculated on dynamic load capacity. In such case, bearing radial load F_r in equation (4.8) is the support B radial load: $F_r = R_B = 2\,612$ N.

Let us assume:

– a reliability factor a_1 for normal reliability 0,9 $a_1 = 1$ (see Tab. 4.1);

– for usual operating conditions and ball bearings factor a_{23} equals 0,8 (see Tab. 4.2);

– the safety factor considering load non-uniformity for medium non-uniformity $K_B = 1{,}3$;

– the temperature factor $K_T = 1$ because operating temperature should be less 100 °C (we will use an oil lubricant and transferring power relatively small).

Service life in million revolutions according to formula (4.5)

$$L = \frac{60nL_h}{10^6} = \frac{60 \cdot 300 \cdot 20\,000}{10^6} = 360\ .$$

Radial load factor $X = 1$ and axial one $Y = 0$ because the axial force F_a loads support A only.

Rotation factor $V = 1$ because vector of the radial load ($F_r = R_B$) is permanently directed relatively to the rotating bearing inner ring.

Thus, equivalent load

$$P = VF_r K_B K_T = 1{\cdot}2612{\cdot}1{,}3{\cdot}1 = 3396\ \text{N}.$$

For ball bearings an exponent $p = 3$, so finally the dynamic load capacity

$$C = \left(\frac{L}{a_1 a_{23}}\right)^{\frac{1}{p}} P = \left(\frac{360}{1 \cdot 0{,}8}\right)^{1/3} 3396 = 26024\ \text{N}.$$

Strength condition (4.3) is satisfied: $26\,024 < 33\,200$ N, therefore we may use previous selected bearings.

5. SHAFT CHECK CALCULATIONS

5.1. Calculation of Shaft Static Strength

To prevent the plastic deformations at the big short-term overloads a shaft static strength calculation is carried out. Strength condition has a form

$$S = \frac{\sigma_y}{\sigma_E} \geq [S] = 1{,}5...2{,}5\ , \tag{5.1}$$

where $\sigma_E = \sqrt{\sigma^2 + 4\tau^2}$ is the equivalent stress, σ and τ are the acting normal and shear stresses in a dangerous section; σ_y is the yield stress of a shaft material (Tab. 5.1); $[S]$ is the allowable safety factor.

Table 5.1

Steel	Hardness, HB	σ_B (σ_{UTS})	σ_Y	σ_{-1}	τ_{-1}	ψ_σ	ψ_τ
		MPa					
45	200	610	280	250	150	0	0
40X	200	630	330	310	180	0,10	0,05
12XH3A	260	950	700	420	210	0,10	0,05
12X2H4A	300	1100	850	500	250	0,05	0, 10
18XΓT	330	1150	950	520	280	0,15	0,10
30XΓT	320	1150	950	520	310	0,15	0,10
40XH	240	820	650	360	210	0,10	0,05

Actual stresses are:

$$\sigma = \sigma_b + \sigma_t = \frac{M_c}{W_0} + \frac{F_a}{A}, \quad \tau = \frac{T_c}{W_\rho}, \tag{5.2}$$

where $M_c = M_\Sigma k$ is the bending moment in dangerous section; $T_c = Tk$ is the torque in the same section; k = *2...3* is the overload factor; W_0 is the section modulus, $W_0 = 0{,}1d^3(1-\alpha^4)$; W_0 is the polar section modulus, $W_\rho = 0{,}2d^3(1-\alpha^4)$; F_a is the axial force; A is the cross-section area.

Diameters of various sections of a shaft were defined during a shaft designing. To find dangerous sections and calculate the moments acting in these sections, the design diagram would be drown.

The most typical cases of shaft loading are described below. The forces represented in these drawings are transfered in static zero to an axis of a shaft rotation separately for vertical and horizontal planes. It should be noted, that names of planes («vertical» and «horizontal») in which gearing forces operate, are conditional, as the real plane depends on a relative gears location in space. However, these planes always remain mutually perpendicular, and the total bending moment in any section of a shaft

$$M_\Sigma = \sqrt{M_V^2 + M_H^2} = \sqrt{M_y^2 + M_z^2}\ . \tag{5.3}$$

Let us draw bending and twisting moment diagrams (the compression caused by axial force F_a, is not shown).

Distances between support and points of the applying of external forces should be measured on the drawing.

1. A spur helical (or straight teeth bevel) gear and coupling are mounted on a shaft (gear locates between supports, coupling is cantilever) (Fig. 5.1) (calculation of forces and reactions see part 4).

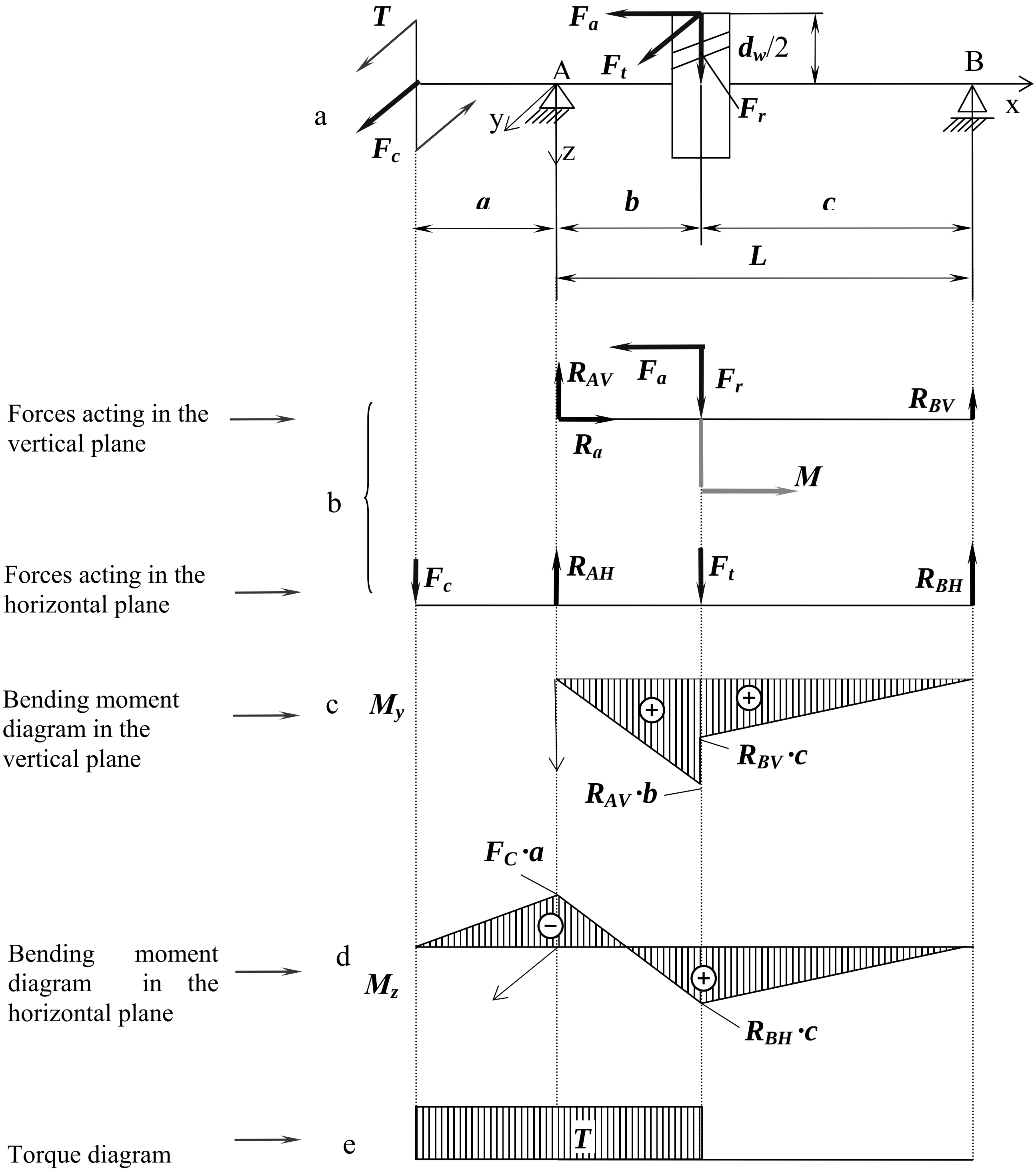


Fig. 5.1

In cases of straight teeth spur gear $M = F_a \dfrac{d_w}{2} = 0$ and step will not be on bending moment diagram M_y (Fig. 5.1, c).

2. A bevel gear and half of coupling are mounted on a shaft cantilever (Fig. 5.2).

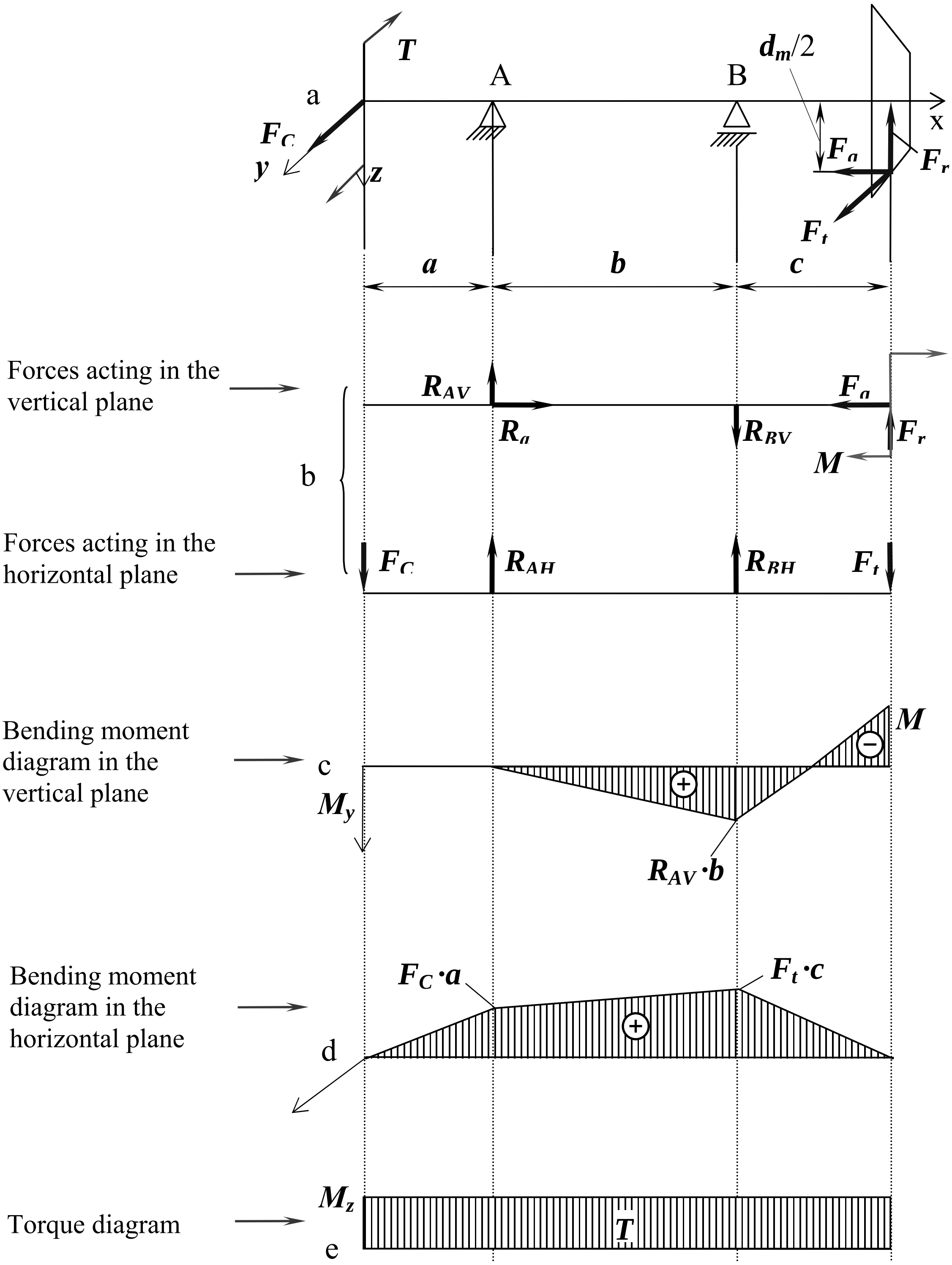


Fig. 5.2

In cases of straight spur gear, these bending and twisting moment diagrams and formulae could be also used, $M = F_a \frac{d_w}{2} = 0$ and step there will not be on bending moment diagram M_y (Fig. 5.2, c).

3. Two cylindrical helix gears are mounted on a shaft between supports (Fig. 5.3).

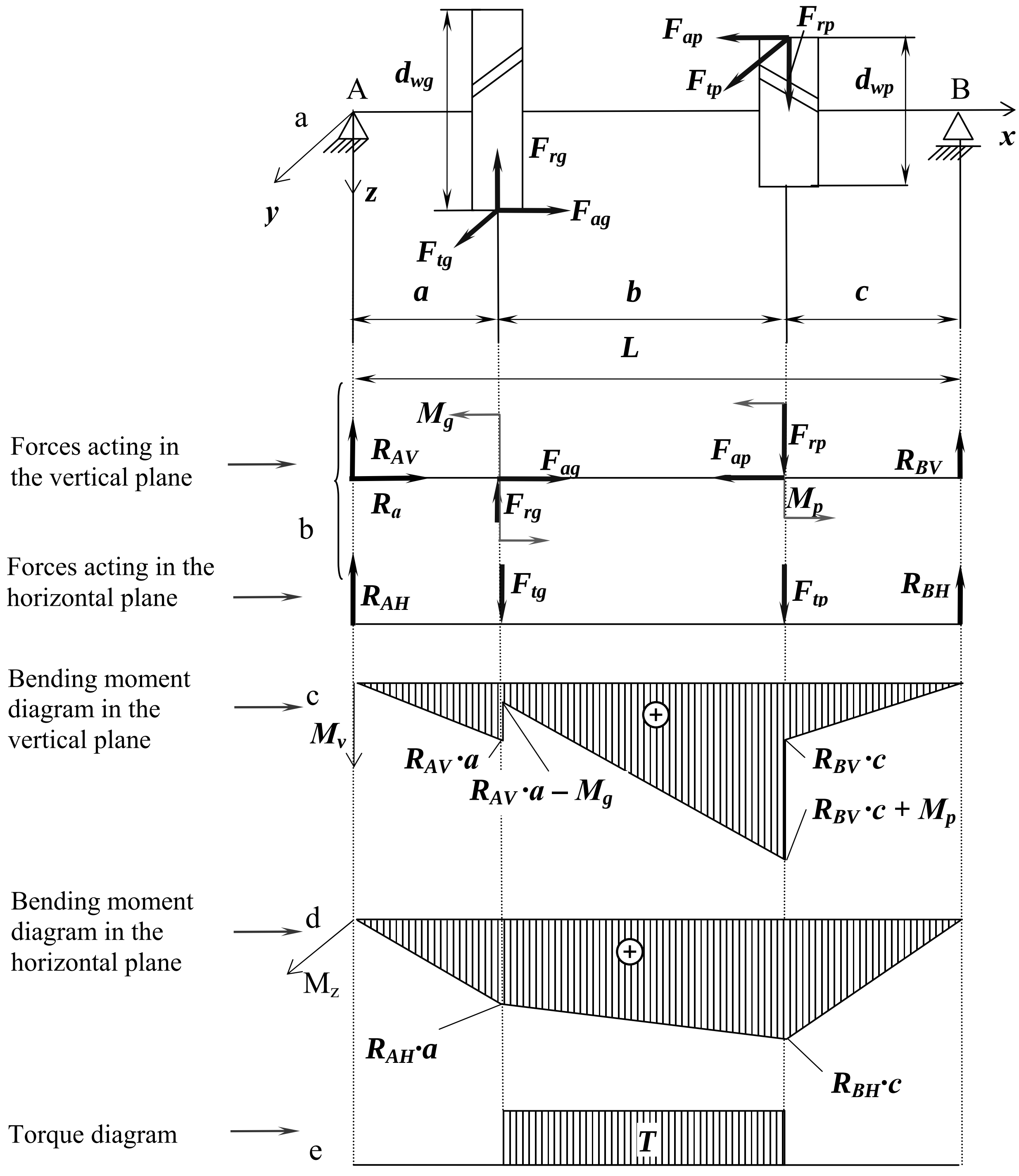


Fig. 5.3

In cases of straight gear, these bending and twisting moment diagrams and formulae could be also used, $M = F_a \dfrac{d_w}{2} = 0$ and sudden changes there will be no on bending moment diagram M_y (Fig. 5.3, c).

4. A spur helical (or straight teeth bevel) gear and two couplings are mounted on a shaft (gear locates between supports, couplings are cantilever) (Fig. 5.4).

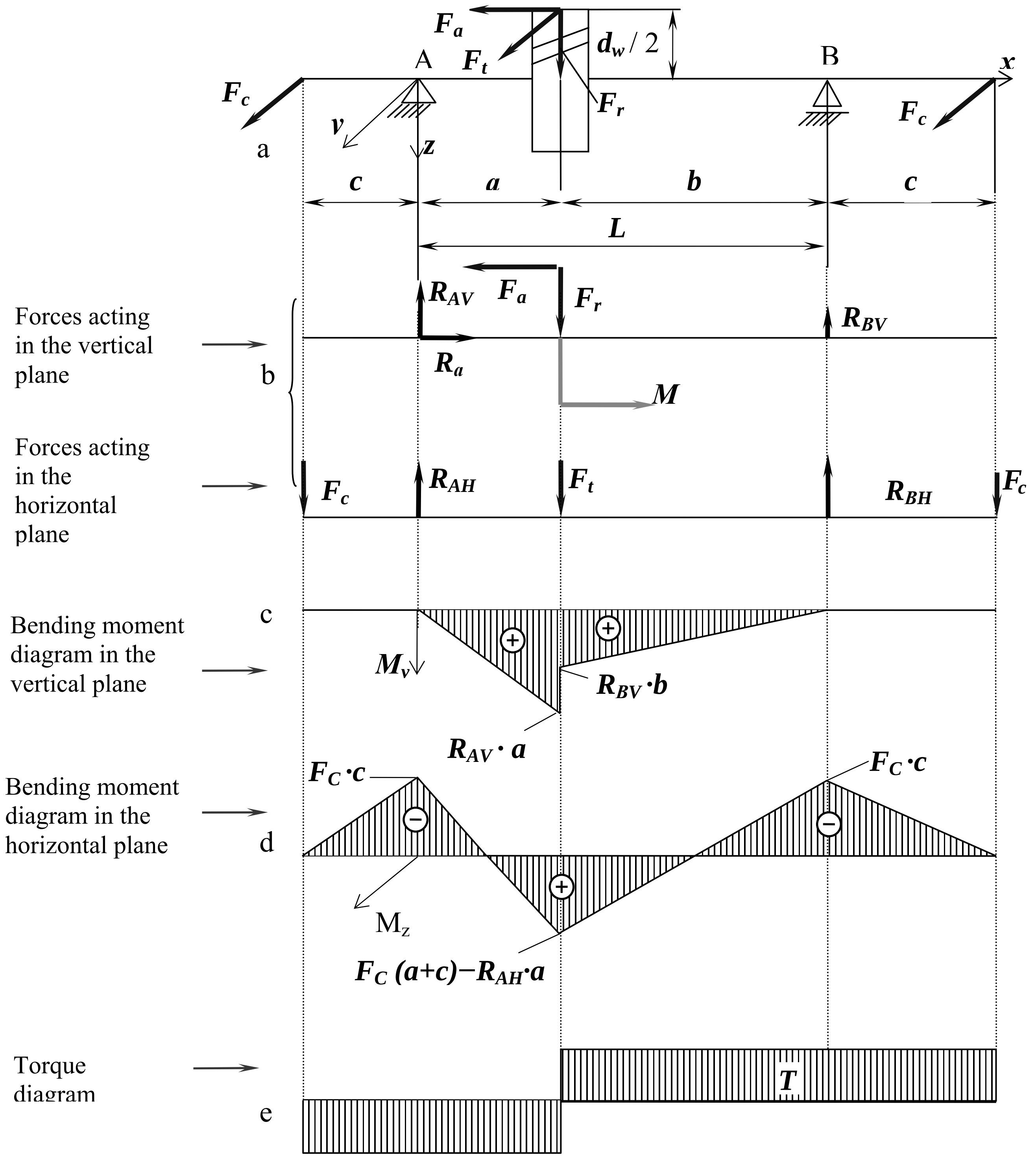


Fig. 5.4

In cases of straight teeth spur gear $M = F_a \dfrac{d_w}{2} = 0$ and step will not be on bending moment diagram M_y (Fig. 5.4, c).

5.1.1. Examples

Example 1

Check the described above spur gearbox input shaft.

As for the calculating scheme and design diagrams on Fig.5.1, there are two potential dangerous sections: at support A (marked 1) and under the gear (marked 2). Let us find an actual safety factor in these sections, taking into account dimensions according to Table 4.5 and the actual forces and the reactions according to Table 4.6.

As input shaft is pinion-shaft and steel 40X was selected for pinion, so shaft material is also steel 40X with yield stress 330 MPa (see Table 5.1). Calculated values are shown in Tab. 5.2.

Table 5.2

Parameter	Formula	Value for cross-section	
		1	2
Horizontal maximum bending moment M_H, Nmm	$M_{H1} = F_C a = 103{,}6 \cdot 84$	8 702	
	$M_{H2} = R_{BH} c = 876 \cdot 38$		33 280
Vertical maximum bending moment M_V, Nmm	$M_{V1} = 0$	0	
	$M_{V2} = max\{ R_{AV} b; R_{BV} c \}$ (In our case $b = c$, $R_{AV} = R_{BV}$)		13 680
Total bending moment M_Σ, Nmm	$M_{\Sigma i} = \sqrt{M_{Vi}^2 + M_{Hi}^2}$	8 702	35 980
Shaft diameter, mm	d	40	46
Overload factor	k	2,2	
Overloading bending moment M_P, Nmm	$M_{ci} = M_{\Sigma i} k$	19 145	79 150
Overloading torsion moment T_P, Nmm	$T_c = T\,k = 54\,400 \cdot 2{,}2$	119 680	
Bending stress σ, MPa	$\sigma_i = \dfrac{M_{ci}}{W_{0i}}$	3,0	8,1
Shear stress τ, MPa	$\tau_i = \dfrac{T_c}{W_\rho} = \dfrac{T_c}{0{,}2 d_i^3}$	9,4	6,1
Equivalent stress σ_E, MPa	$\sigma_{Ei} = \sqrt{\sigma_i^2 + 4\tau_i^2}$	18,9	14,7
Actual safety factor S	$S_i = \dfrac{\sigma_y}{\sigma_{Ei}}$	17,4	22,4

Example 2

Check the described above bevel gearbox input shaft.

As for the calculating scheme and design diagrams on Fig.5.2, there is one potential dangerous section: at support B. Let us find an actual safety factor in this section, taking into account dimensions according to Tab. 4.7 and the actual forces and the reactions according to Tab. 4.8.

Let us assume steel 40X with yield stress 330 MPa as the shaft material (see Tab. 5.1). Calculated values are shown in Tab. 5.3.

Table 5.3

Parameter	Formula	Value
A horizontal maximum bending moment $\boldsymbol{M_H}$, Nmm	$M_H = F_t c = 2366 \cdot 33$	78 078
A vertical maximum bending moment $\boldsymbol{M_V}$, Nmm	$M_V = M - F_r c = 9176 - 818 \cdot 33$	-17 818
Total bending moment $\boldsymbol{M_\Sigma}$, Nmm	$M_\Sigma = \sqrt{M_V^2 + M_H^2} = \sqrt{78078^2 + 17818^2}$	80 085
Shaft diameter, mm	d	40
Overload factor	k	2,2
Overloading bending moment $\boldsymbol{M_P}$, Nmm	$M_c = M_\Sigma k = 80085 \cdot 2{,}2$	176 190
Overloading torsion moment $\boldsymbol{T_P}$, Nmm	$T_c = T\,k = 80700 \cdot 2{,}2$	177 540
Bending stress $\boldsymbol{\sigma}$, MPa	$\sigma = \frac{M_c}{W_0} + \frac{F_a}{A} = \frac{176190}{0{,}1 \cdot 40^3} + + \frac{269}{0{,}78 \cdot 40^2} =$ $= 27{,}53 + 0{,}21$	27,74
Shear stress $\boldsymbol{\tau}$, MPa	$\tau = \frac{T_c}{W_\rho} = \frac{177540}{0{,}2 \cdot 40^3}$	13,87
Equivalent stress $\boldsymbol{\sigma_E}$, MPa	$\sigma_E = \sqrt{\sigma^2 + 4\tau^2} = \sqrt{27{,}74^2 + 4 \cdot 13{,}87^2}$	39,2
Actual safety factor $\boldsymbol{S}$	$S = \frac{\sigma_y}{\sigma_E} = \frac{280}{39{,}2}$	7,1

Thus, actual safety factor is too greater than allowable.

5.2. Calculation on Fatigue Endurance

Calculation on endurance (the basic calculation) takes place in the form of checking of safety factor at summary action of bending and torsion

$$S = \frac{S_\sigma S_\tau}{\sqrt{S_\sigma^2 + S_\tau^2}} \geq [S] = 1{,}5...2{,}5\,, \qquad (5.4)$$

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 of a shaft material accordingly at bending and torsion with a reversed stress cycle (see Tab. 5.1); $\psi_\sigma = \dfrac{2\sigma_{-1} - \sigma_0}{\sigma_0}$ and $\psi_\tau = \dfrac{2\tau_{-1} - \tau_0}{\tau_0}$ are the factors characterizing material sensitivity for alternating stresses (see Tab. 5.1);

σ_a and τ_a are the actual amplitude stress: $\sigma_a = \dfrac{\sigma_{max} - \sigma_{min}}{2}$ (τ_a define similarly);

σ_m and τ_m are the actual mean stress: $\sigma_m = \dfrac{\sigma_{max} + \sigma_{min}}{2}$ (τ_m define similarly).

Shear stress changes proportionally to load change. 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) for the bending stress and repeated for the torsion stress (Fig. 5.5).

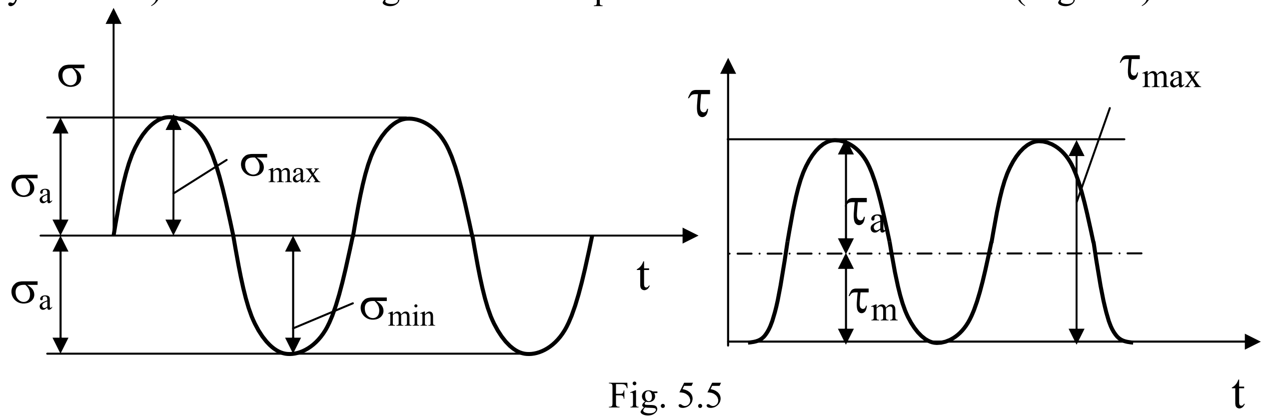


Fig. 5.5

According to the assumed cycles, amplitude and mean stresses are:

$$\sigma_a = M_\Sigma / W_0;\ \tau_a = \tau_m = 0{,}5\ \tau = \frac{1}{2}\frac{T}{W_\rho}.$$

If there is an axial force,

$$\sigma_m = \frac{4F_a}{\pi d^2 (1 - \alpha^2)}.$$

Theoretical stress-concentration factors $K_{\sigma D}$ and $K_{\tau D}$ considering influence of all factors on resistance of bending fatigue and torsion are calculated by means of formulae:

$$K_{\sigma D} = (K_{\sigma} / \varepsilon + K_F - 1) / K_v ; \; K_{\tau D} = (K_{\tau} / \varepsilon + K_F - 1) / K_v ,$$

where ε is the absolute size factor of the of cross-section (Tab. 5.4); K_F is a surface roughness factor (Tab. 5.5); K_V is a factor of influence of the hardening, entered for shaft with superficial hardening (Tab. 5.6); K_σ and K_τ are a fatigue normal and shear stress-concentration factors.

Reference data for various types of concentrators (Fig. 5.6) are shown in Tab. 5.7–5.10.

Table 5.4

Strength state	**Material**	**Absolute size factor ε depending on shaft diameter, mm**							
		15	**20**	**30**	**40**	**50**	**70**	**100**	**200**
Bend	Carbon steel	0,95	0,92	0,86	0,85	0,81	0,76	0,70	0,61
Bend and torsion	Alloy steel	0,87	0,83	0,77	0,73	0,70	0,65	0,59	0,52

Table 5.5

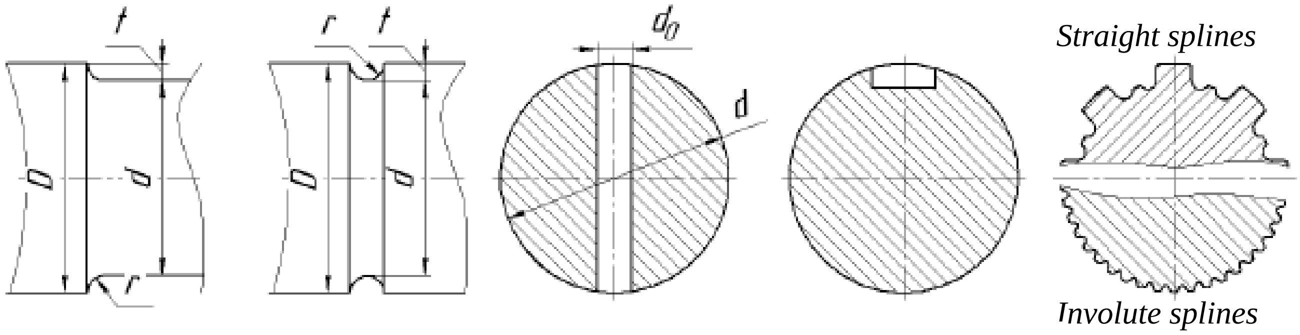
Surface finishing	**A surface roughness R_a, μm**	**A surface roughness factor K_F at σ_{UTS}, MPa**		
		400	**600**	**1200**
Polishing	0,32…0,08	1	1	1
Turning	2,5…0,32	1,05	1,1	1,25
Rough machining	20…5	1,2	1,25	1,5

Table 5.6

Type of hardening	**Hardening factor K_V for specimen***	
	without stress concentrator	**with stress concentrator** **
Surface hardening for carbon and alloy steels	1,2…1,5	1,5…2,5
Nitriding at layer depth 0,1…0,4 mm	1,1…1,15	1,3…2,0
Case-hardening at layer depth 0,2…0,6 mm	1,1…1,5	1,2…2,0
Rolling by rollers for carbon and alloy steels	1,1…1,25	1,3…1,8
Blowing by grit for carbon and alloy steels	1,1…1,2	1,1…1.5

*Based on experience with d = 30...40 mm specimen

**The larger stress-concentration the larger value



a b c d

Fig. 5.6

Table 5.7 (see Fig. 5.6, a)

t/ r	***r/ D***	***K_σ* at *σ_{UTS}*, MPa**				***K_τ* at *σ_{UTS}*, MPa**			
		500	**700**	**900**	**1200**	**500**	**700**	**900**	**1200**
1	0,01	1,35	1,40	1,45	1,50	1,30	1,30	1,30	1,30
	0,02	1,45	1,50	1,55	1,60	1,35	1,40	1,40	1,40
	0,03	1,65	1,70	1,80	1,90	1,40	1,45	1,45	1,50
	0,05	1,60	1,70	1,80	1,95	1,45	1,45	1,50	1,55
	0,10	1,45	1,55	1,65	1,85	1,40	1,40	1,45	1,50
2	0,01	1,55	1,60	1,65	1,70	1,40	1,40	1,45	1,45
	0,02	1,80	1,90	2,00	2,15	1,55	1,60	1,65	1,70
	0,03	1,80	1,95	2,05	2,25	1,55	1,60	1,65	1,70
	0,05	1,75	1,90	2,00	2,20	1,55	1,66	1,65	1,75
3	0,01	1,90	2,00	2,10	2,20	1,55	1,60	1,65	1,75
	0,02	1,95	2,10	2,20	2,40	1,60	1,70	1,75	1,85
	0,03	1,95	2,10	2,25	2,45	1,65	1,70	1,75	1,90
5	0,01	2,10	2,25	2,35	2,50	2,22	2,30	2,40	2,60
	0,02	2,15	2,30	2,45	2,65	2,10	2,15	2,25	2,40

Table 5.8 (see Fig. 5.6, d)

***σ_{UTS}*, MPa**	***K_σ***			***K_τ***		
	For splines	**For keyway**	**For thread**	**For splines**		**For keyway**
				straight	**involute**	
500	1,45	1,60	1,80	2,25	1,43	1,40
600	1,55	1,75	1,95	2,35	1,46	1,50
700	1,60	1,90	2,20	2,45	1,49	1,70
800	1,65	2,05	2,30	2,55	1,52	1,90
900	1,70	2,20	2,45	2,65	1,55	2,00
1000	1,72	2,30	2,60	2,70	1,58	2,20

Table 5.9 (see Fig. 5.6, b)

Strength state	σ_{UTS}, **MPa**	**Theoretical normal K_σ and shear K_τ stress-concentration factors for shaft with turning at relationship r / d**								
		0	**0,02**	**0,03**	**0,05**	**0,10**	**0,01**	**0,02**	**0,03**	**0,05**
Bend (K_σ)		***t / r = 0,5***					***t / r = 1***			
	500	2,00	1,85	1,75	1,65	1,50	2,15	2,05	1,95	1,85
	700	1,10	1,95	1,85	1,75	1,55	2,25	2,15	2,10	1,95
	900	2,20	2,05	1,95	1,90	1,60	2,40	2,30	2,20	2,10
	1200	2,30	2,20	2,10	2,05	1,75	2,60	2,50	2,35	2,25
		t/r = 2					***t/r = 5***			
	500	2,40	2,25	2,15	–	–	2,45	2,35	–	–
	700	2,50	2,40	2,30	–	–	2,65	2,50	–	–
	900	2,70	2,50	2,40	–	–	2,80	2,65	–	–
	1200	2,90	2,70	2,60	–	–	3,05	2,85	–	–
Torsion (K_τ)	500	1,70	1,60	1,50	1,40	1,20	–	–	–	–
	700	1,90	1,75	1,65	1,50	1,25	–	–	–	–
	900	2,10	1,95	1,80	1,65	1,30	–	–	–	–
	1200	2,40	2,20	2,05	1,80	1,40	–		–	–

Table 5.10 (see Fig. 5.6, c)

σ_{UTS}, **MPa**	**Theoretical normal K_σ and shear K_τ stress-concentration factors for shaft with hole**		
	K_σ at d_0 / d		**K_τ at d_0 / d = 0,05…0,25**
	0,05…0,15	**0,15…0,25**	
≤700	2,00	1,80	1,75
900	2,15	1,90	1,90
≥1000	2,30	2,10	2,00

5.2.1. Examples

Example 1

Let us consider bevel gearbox input shaft as the most loaded.

To calculate a shaft on fatigue endurance we must define actual safety factors on both normal bend stress (Tab. 5.11) and shear stresses (Tab. 5.12) in the dangerous cross-section located at the support B (see Fig. 5.2). From previous calculation (see Tab. 5.3), we know total bending moment M_Σ = 80 085 Nmm, torsion moment T = 80 700 Nmm and shaft diameter d = 40 mm.

Table 5.11

Parameter	Formula	Value
Endurance limit at a bend σ_{-1}, MPa	For steel 40X (see Tab. 5.1)	310
Factor characterizing material sensitivity for alternating stresses ψ_σ		0,1
Absolute size factor of the of cross-section ε	At diameter 40 mm (see Tab. 5.4)	0,73
Surface roughness factor K_F	For turning surface (see Tab. 5.5)	1,15
Factor of influence of the hardening K_V	For blowing by grit for alloy steel (see Tab. 5.6)	1,2
Fatigue normal stress-concentration factor K_σ	For shoulder near the bearing (see Tab. 5.7, t/r = 3, r/D = 0,022)	≈ 2,0
Theoretical normal stress-concentration factor $K_{\sigma D}$	$K_{\sigma D} = \dfrac{\dfrac{K_\sigma}{\varepsilon} + K_F - 1}{K_V}$	2,41
Actual amplitude normal stress σ_a, MPa	$\sigma_a = \dfrac{M_\Sigma}{0{,}1d^3}$	12,5
Axial force F_a, N	F_a	269
Mean normal stress σ_m, MPa	$\sigma_m = \dfrac{4F_a}{\pi d^2}$	0,2
Actual safety factor on normal stress S_σ	$S_\sigma = \dfrac{\sigma_{-1}}{K_{\sigma D}\sigma_a + \psi_\sigma \sigma_m}$	10,3

Table 5.12

Parameter	Formula	Value
Endurance limit at a bend τ_{-1}, MPa	For steel 40X (see Tab. 5.1)	180
Factor characterizing material sensitivity for alternating stresses ψ_τ		0,05
Absolute size factor of the of cross-section ε	At diameter 40 mm (see Tab. 5.4)	0,73
Surface roughness factor K_F	For turning surface (see Tab. 5.5)	1,15
Factor of influence of the hardening K_V	For blowing by grit for alloy steel (see Tab. 5.6)	1,2
Fatigue shear stress-concentration factor K_τ	For shoulder near the bearing (see Tab. 5.7, ***t/r*** = 3, ***r/D*** = 0,022)	1,70
Theoretical shear stress-concentration factor $K_{\tau D}$	$K_{\tau D}=\dfrac{\dfrac{K_\tau}{\varepsilon}+K_F-1}{K_V}$	2,07
Actual amplitude shear stress τ_a, MPa	$\tau_a=\dfrac{T}{2\cdot 0{,}2d^3}$	3,2
Mean shear stress τ_m, MPa	$\tau_m=\tau_a$	3,2
Actual safety factor on shear stress S_τ	$S_\tau=\dfrac{\tau_{-1}}{K_{\tau D}\tau_a+\psi_\tau\tau_m}$	26,4

The integral safety factor of fatigue endurance at joint action of bending and torsion is calculated under the formula

$$S=\frac{S_\sigma\,S_\tau}{\sqrt{S_\sigma^{\,2}+S_\tau^{\,2}}}=\frac{10{,}28\cdot 26{,}4}{\sqrt{10{,}28^2+26{,}4^2}}=9{,}58\,.$$

This one is more than allowable value $[\,S\,]=1{,}5...2{,}5.$

Example 2

Let us calculate the actual safety factor for intermediate shaft of two stages helical spur gearbox (Tab. 5.13, Fig. 5.7).

Table 5.13

Parameter		Symbol	Value
The torque on shaft, N mm		$\boldsymbol{T}$	250 000
Rotational speed, rpm		$\boldsymbol{n}$	800
The distances between	support A and gear	$\boldsymbol{a}$	40
	gear and pinion	$\boldsymbol{b}$	40
	pinion and support B	$\boldsymbol{c}$	50
Total length		$\boldsymbol{L = a + b + c}$	130
Shaft diameter, mm	under gear	$\boldsymbol{d}$	40
	under pinion		46
Pitch diameters of	pinion	$\boldsymbol{d_{wp}}$	70
	gear	$\boldsymbol{d_{wg}}$	140
Helix angle		$\boldsymbol{\beta}$	11°
Pressure angle		α_w	20°
Material			Steel 40X

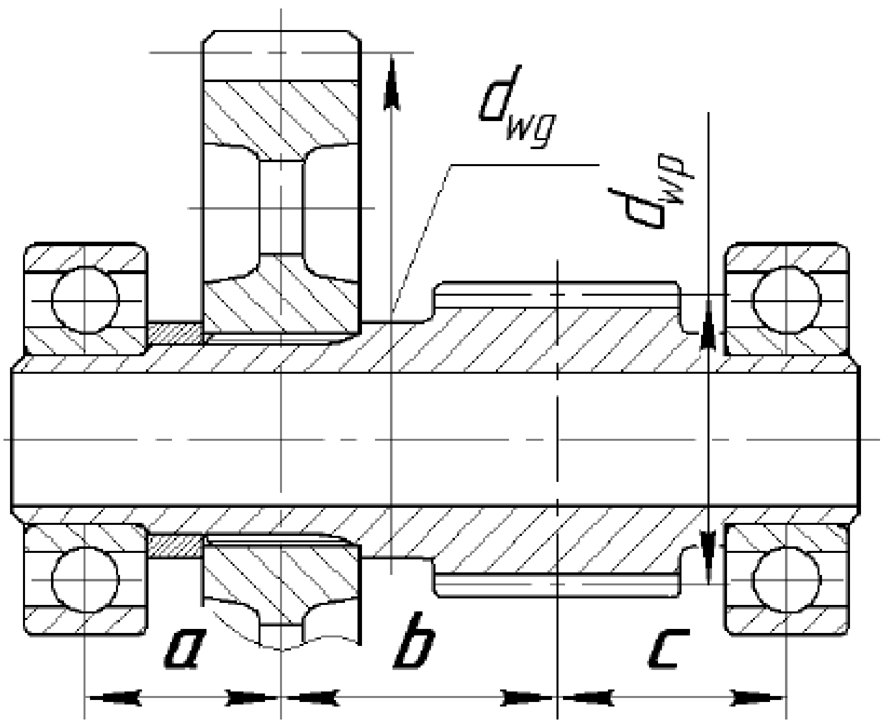


Fig. 5.7

First, the acting forces should be found (Tab. 5.14).

Table 5.14

Parameter		Formula		Value
The peripheral force F_t, N		$F_t = \frac{2T}{d_w}$	pinion	7143
			gear	3571
The radial force F_r, N		$F_r = \frac{F_t \tan\alpha_w}{\cos\beta}$	pinion	2647
			gear	1323
The axial force F_a, N		$F_a = F_t \tan\beta$	pinion	1388
			gear	694
The actual bending moment M, Nm		$M_p = F_a \frac{d_w}{2}$	pinion	48,58
			gear	48,58
Support A reactions, N	Horizontal	$R_{AH} = \frac{F_{tp}c + F_{tg}(b+c)}{L}$		5220
	Vertical	$R_{AV} = \frac{F_{rp}c - F_{rg}(b+c) + M_p + M_g}{L}$		850
Support B reactions, N	Horizontal	$R_{BH} = \frac{F_{tg}a + F_{tp}(a+b)}{L}$		5494
	Vertical	$R_{BV} = \frac{F_{rp}(a+b) - F_{rg}a - M_p - M_g}{L}$		474

Calculated above forces are represented in the diagram and transferred in static zero to an axis of shaft rotation separately for vertical and horizontal planes. The shaft should be considered as a beam on two supports (as shown in Fig. 5.3). Bending and twisting moment diagrams are the same with Fig. 5.3.

At a choice of dangerous sections of a shaft, we consider values of the bending moments and torques, the each parts' diameters and presence of concentrators (keyway, groove, holes, the turnings etc.). The bending and twisting moment diagrams analysis shows that both sections (located in the middle of a gear hub and in the middle of a pinion) should be checked up because the first of them has lower diameter and the second is effected by larger total moment. Calculations on stresses are shown in Tab. 5.15.

Table 5.15

Parameter	Formula	Value	
		under gear	under pinion
Maximum bending moment acting on a horizontal plane M_H, Nmm	$M_H = R_{AH}\,a$ $M_H = R_{BH}\,c$	208800	274700
Maximum bending moment acting on a vertical plane M_V, Nmm	$M_V = R_{AV}a$ $M_V = R_{BV}c + M_p$	34000	72300
Total bending moment M_Σ, Nmm	$M_\Sigma = \sqrt{M_V^2 + M_H^2}$	211550	284100

End of Tab. 5.15

Parameter	Formula	Value	
		under gear	under pinion
Shaft diameter, mm	d	40	46
Overload factor	$k = T_{max}/T_{nom}$	2,2	
Overloading bending moment M_P, Nmm	$M_P = kM_\Sigma$	465 410	625 020
Overloading torsion moment T_P, Nmm	$T_P = kT$	550 000	
Bending stress σ, MPa	$\sigma_u = \frac{M_P}{W_o} = \frac{M_P}{0,1d^3}$	72,7	64,2
Shear stress τ, MPa	$\tau = \frac{T_P}{W_\rho} = \frac{T_P}{0,2d^3}$	43,0	28,3
Equivalent stress σ_E, MPa	$\sigma_E = \sqrt{\sigma_u^2 + 4\tau^2}$	112,6	85,6
Actual safety factor S	$S = \sigma_T / \sigma_E$	4,4 > 1,5	5,8 > 1,5

To calculate a shaft on fatigue endurance we must define actual safety factors on both normal bend stress (Tab. 5.16) and shear stress (Tab. 5.17) in the dangerous cross-section located in the middle of a gear hub.

Table 5.16

Parameter	Formula	Value
Endurance limit at a bend σ_{-1}, MPa	For steel 40X (see Tab. 5.1)	320
Factor characterizing material sensitivity for alternating stresses ψ_σ		0,1
Absolute size factor of the of cross-section ε	At diameter 40 mm (see Tab. 5.4)	0,73
Surface roughness factor K_F	For turning surface (see Tab. 5.5)	1,13
Factor of influence of the hardening K_V	For blowing by grit for alloy steel (see Tab. 5.6)	1,2
Fatigue normal stress-concentration factor K_σ	For splines (see Tab. 5.8)	1,95
Theoretical normal stress-concentration factor $K_{\sigma D}$	$K_{\sigma D} = \frac{\frac{K_\sigma}{\varepsilon} + K_F - 1}{K_V}$	2,3
Actual amplitude normal stress σ_a, MPa	$\sigma_a = \frac{M_\Sigma}{0,1d^3}$	33,0
Axial force F_a, N	$F_a = F_{ap} - F_{ag}$	694
Mean normal stress σ_m, MPa	$\sigma_m = \frac{4F_a}{\pi d^2}$	0,55
Actual safety factor on normal stress S_σ	$S_\sigma = \frac{\sigma_{-1}}{K_{\sigma D}\sigma_a + \psi_\sigma \sigma_m}$	4,2

Table 5.17

Parameter	Formula	Value
Endurance limit at a bend τ_{-1}, MPa	For steel 40X (see Tab. 5.1)	200
Factor characterizing material sensitivity for alternating stresses ψ_τ		0,05
Absolute size factor of the of cross-section ε	At diameter 40 mm (see Tab. 5.4)	0,73
Surface roughness factor K_F	For turning surface (see Tab. 5.5)	1,13
Factor of influence of the hardening K_V	For blowing by grit for alloy steel (see Tab. 5.6)	1,2
Fatigue shear stress-concentration factor K_τ	For splines (see Tab. 5.8)	1,74
Theoretical shear stress-concentration factor $K_{\tau D}$	$K_{\tau D} = \dfrac{\dfrac{K_\tau}{\varepsilon} + K_F - 1}{K_V}$	2,0
Actual amplitude shear stress τ_a, MPa	$\tau_a = \dfrac{T}{2 \cdot 0{,}2d^3}$	9,76
Mean shear stress τ_m, MPa	$\tau_m = \tau_a$	9,76
Actual safety factor on shear stress S_τ	$S_\tau = \dfrac{\tau_{-1}}{K_{\tau D}\tau_a + \psi_\tau \tau_m}$	9,57

The integral safety factor of fatigue endurance at joint action of bending and torsion is calculated under the formula

$$S = \frac{S_\sigma\, S_\tau}{\sqrt{S_\sigma^{\,2} + S_\tau^{\,2}}} = \frac{4{,}2 \cdot 9{,}57}{\sqrt{4{,}2^2 + 9{,}57^2}} = 3{,}81\,.$$

This one is more than allowable value $[\,S\,] = 1{,}5...2{,}5.$

6. GENERATING OF THE GEARBOX ASSEMBLY DRAWING

6.1. Frame Designing

After shafts' designing and calculation, we can create a frame. An example of the spur gearbox assembly is shown in Fig. 6.1, a, the bevel gearbox see in Fig. 6.1, b.

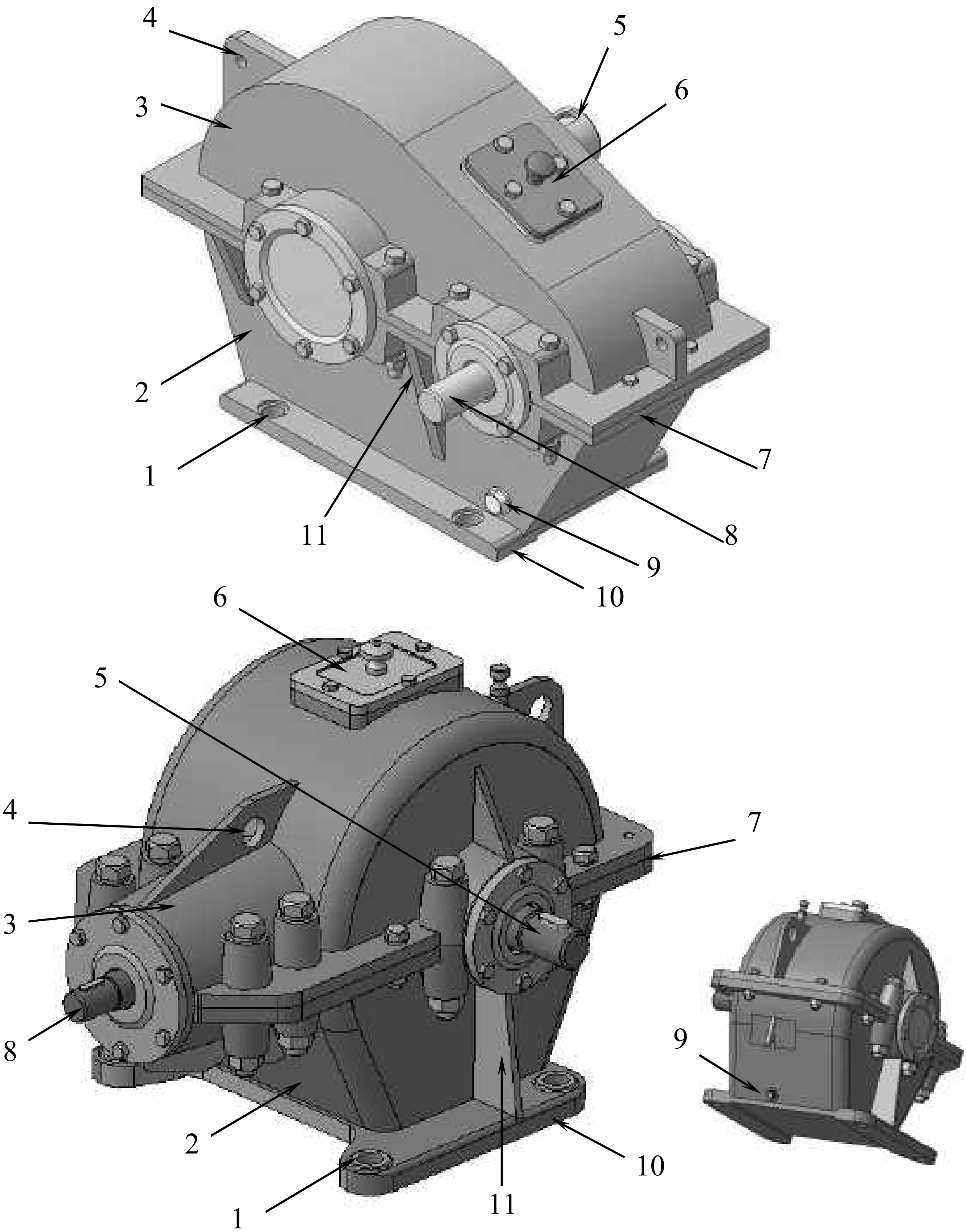


Fig. 6.1. Gearbox frame design:
1 – mounting bolt hole; 2 – case; 3 – cover; 4 – lifting eye; 5 – output shaft; 6 – inspection window; 7 – flange; 8 – input shaft; 9 – oil drain; 10 – foot; 11 – rib

The gearboxes of split type with radial assembly are usually used. Gears and bearings are mounted on the shafts separately outside, assembled radially in the gearbox and top cover 3 (see Fig. 6.1) is bolted in its position over the case 2.

The gearbox case executes the following functions:

– ensure input 8 and output 5 shaft relative positions and orientation;

– support shafts 8 and 5 and translating acting forces to the foot 10;

– ensure stiffness of casing due to enough wall thickness and ribs 11;

– save gearbox inner space from outer dirt penetration and lubricant leakage by means of covers (as 3) and seals;

– ensure containing of the sufficient amount of lubricant and sometimes taking part in lubrication process. The holes for filling with oil and oil draining 9 and oil level measuring devices should be provided;

– dissipate heat generated by gear friction;

– provide a safety and noise barrier;

– provide convenient access to internals for inspection and maintenance;

– Aesthetic benefits;

– fix gearbox on a baseplate by bolts through holes 1 in foot 10.

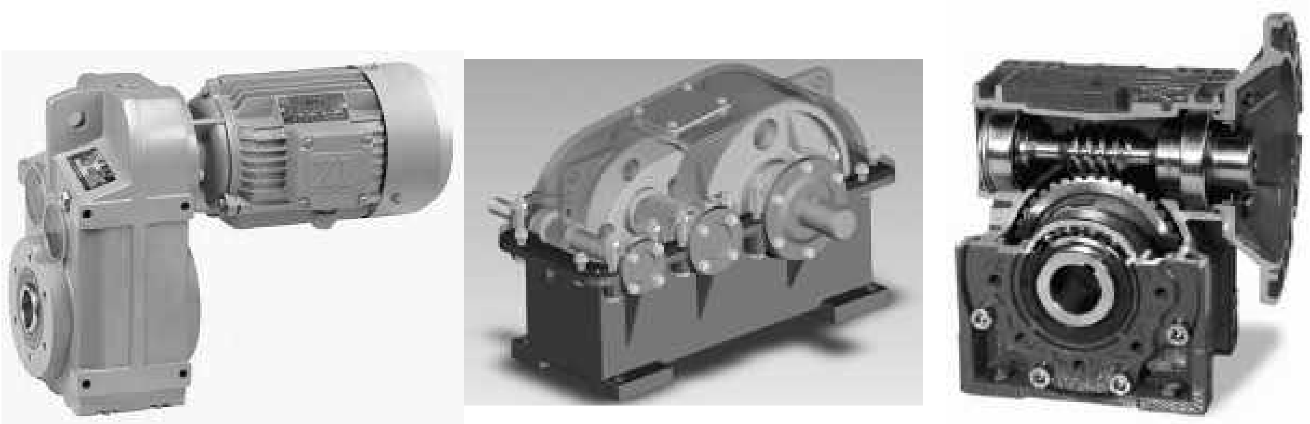
The normal method of fixing an enclosed gearbox is to mount it on a rigid horizontal baseplate designed to absorb vibration. There are a number of variations as listed below:

– foot mounted on vertical surfaces (Fig. 6.2, a);

– foot mounted below horizontal surface (Fig. 6.2, b);

– flange mounted onto the prime mover (Fig. 6.2, c).

When using a gearbox in a non-standard mounting position the lubrication system should be checked for suitability.



a b c

Fig. 6.2

The large gearbox casings are generally castings from cast iron or steel. Cast iron is a rigid material with excellent vibration damping properties.

To decrease weight, gearboxes used for the transmissions in vehicles are often made from cast aluminum or magnesium alloys. The tiny gearbox units are made from a variety of materials including cast zinc alloys.

To shape spur gearbox, let us combine designed above the input (see Fig. 3.14) and the output shafts (see Fig. 3.16) according to Fig. 2.9. See result in Fig. 6.3. Here shafts are placed on both sides of the gearbox, but their arrangement may be different according to project task.

Pay attention that two cylindrical pins were added. Their function is to guarantee coinciding of the bearings' axes and their seats' axes during seats' machining and the gearbox assembling. Accurate fixation is achieved with two pins located on the diagonal of the flange. Pin diameter 20-30 % less than the diameter of the clamping bolts.

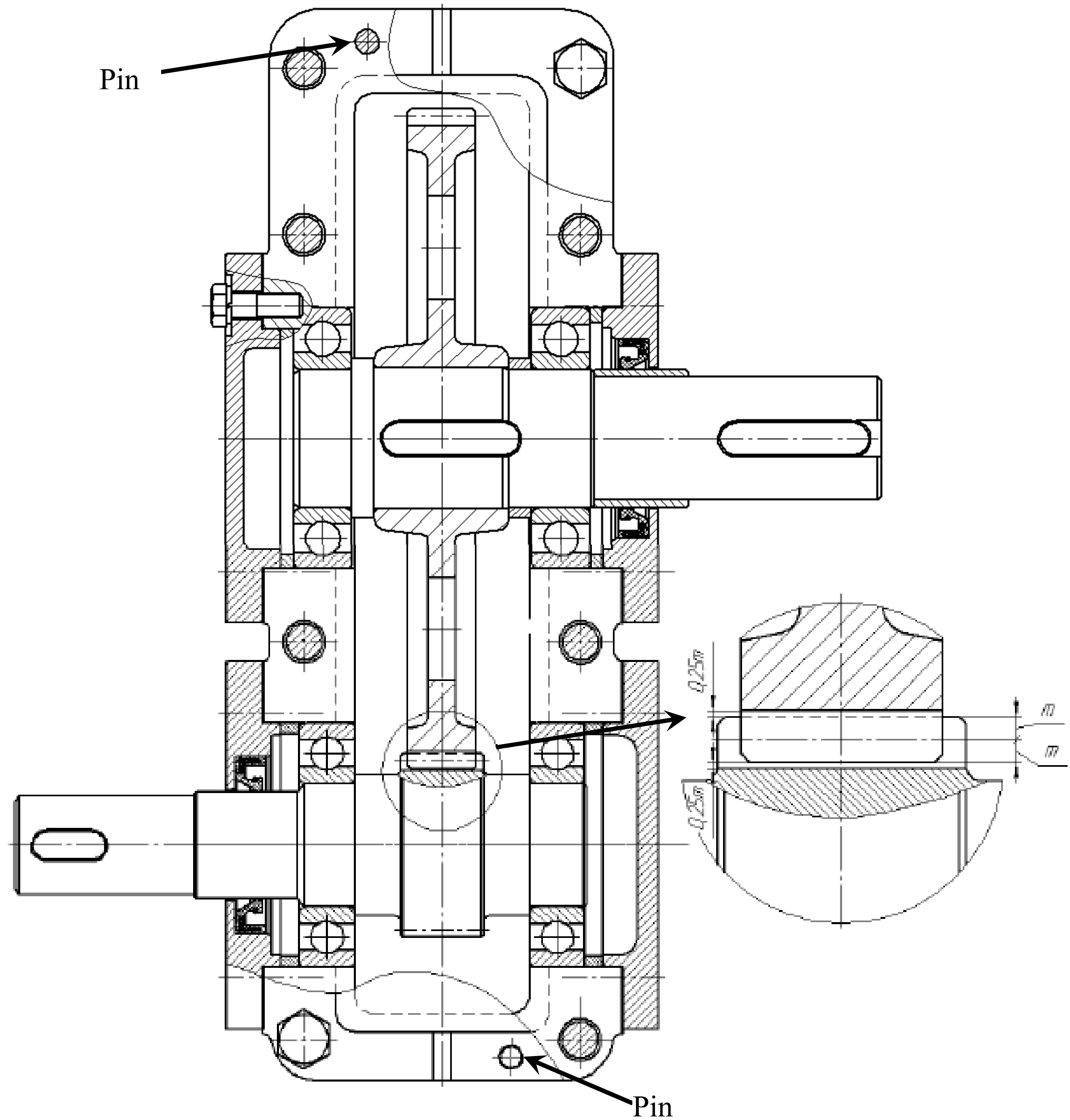


Fig. 6.3

Let us obtain the gearbox second view.

1. Project wheels pitch and addendum diameters (Fig. 6.4, the drawing was turned counterclockwise). As a result, the pitch diameters have to touch each other in a point on the centerline.

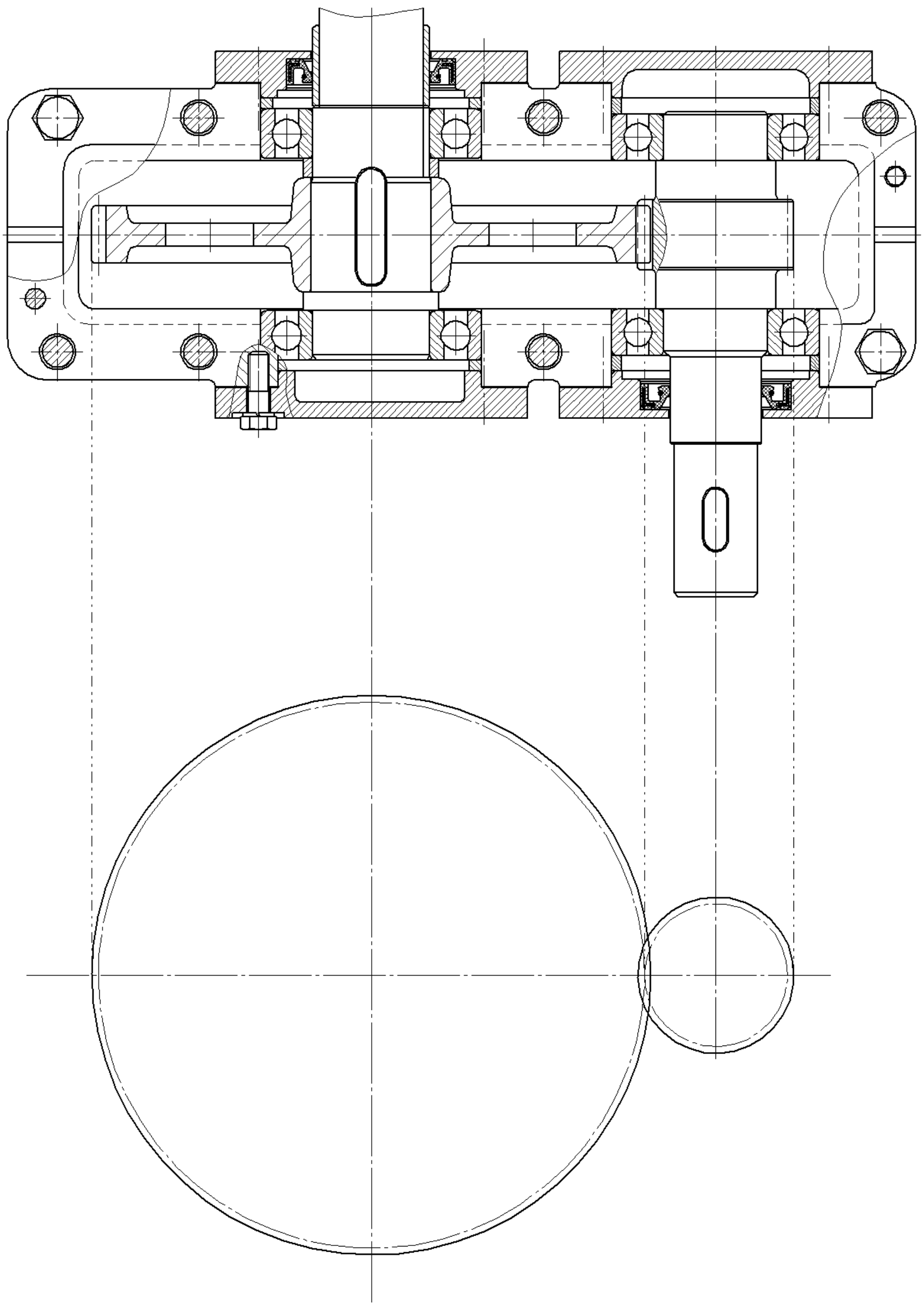


Fig. 6.4

2. Draw the frame around the wheels (Fig. 6.5). Wall thickness is recommended more than 6 mm for casting, a gap 5…7 mm between rotating wheels and the closest frame element should also be ensured.

The minimum depth of dipping to an oil bath $\boldsymbol{H_G}$ has to be equal tooth total height (2,25$\boldsymbol{m}$, where $\boldsymbol{m}$ is a module), the maximum is 1/3 of a gear radius.

The bath depth $\boldsymbol{H_B}$ should be enough to hold sufficient lubricant volume. The volume equals $\boldsymbol{V} = \boldsymbol{H_B}\,\boldsymbol{W_B}\,\boldsymbol{L_B}$, where $\boldsymbol{W_B}$ is the oil bath width, $\boldsymbol{L_B}$ is the bath length (see Fig. 6.5). At the same time, the volume is recommended 0,3…0,5 liter per 1 kW of transferred power. In our case $\boldsymbol{V}$ = 0,4 · 5,5 = 2,2 *l*. It is known, that 1 liter includes 10^6 mm^3. Thus

$$\boldsymbol{H_B = \frac{V}{W_B L_B} = \frac{2{,}2 \cdot 10^6}{70 \cdot 300} \approx 100}\ \text{mm}.$$

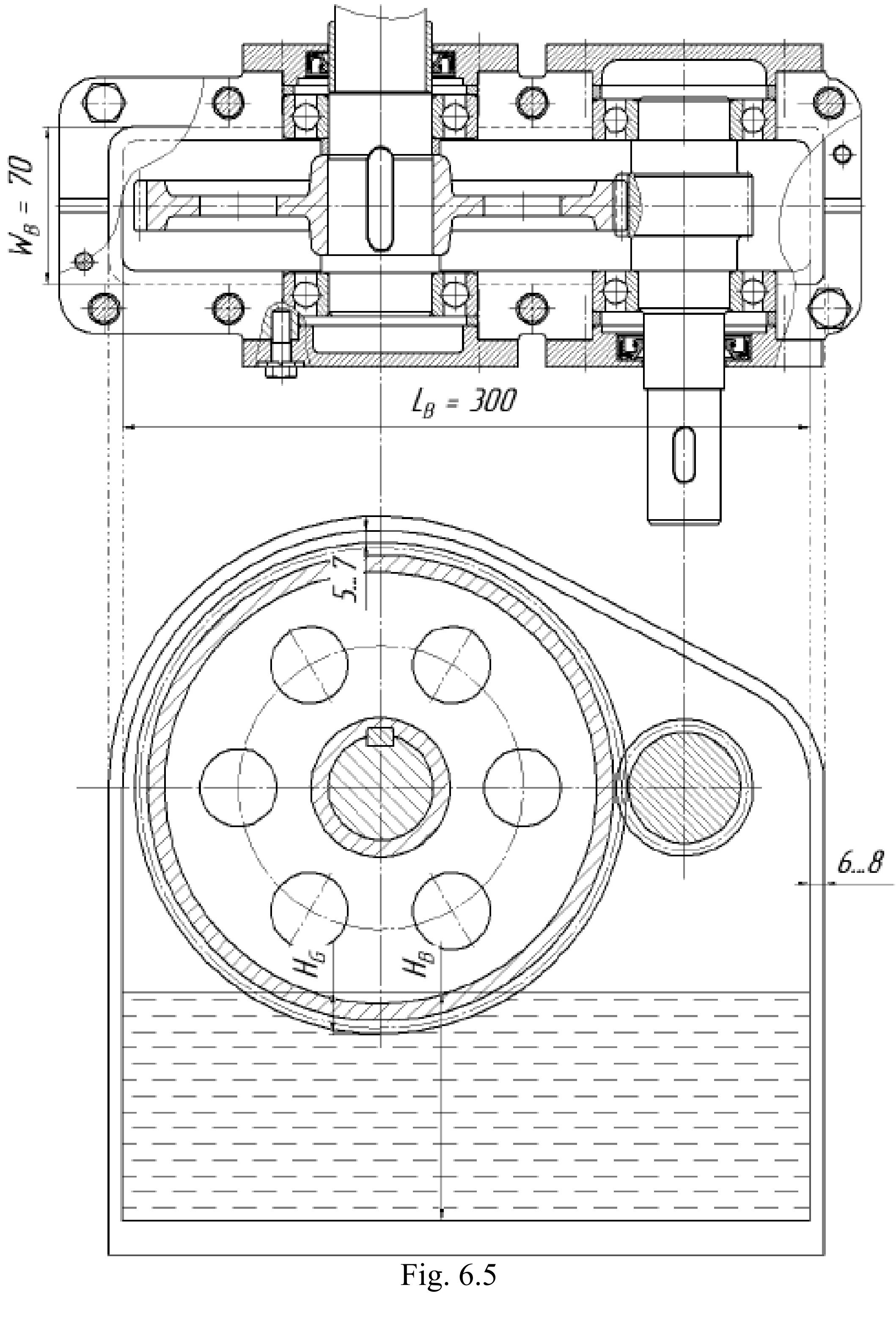


Fig. 6.5

3. Shape the flanges (Fig. 6.6). Flange thickness H_F is recommended $(1\ldots1,5)d_b$, where d_b is the diameter of the bolts, clamping case and cover.

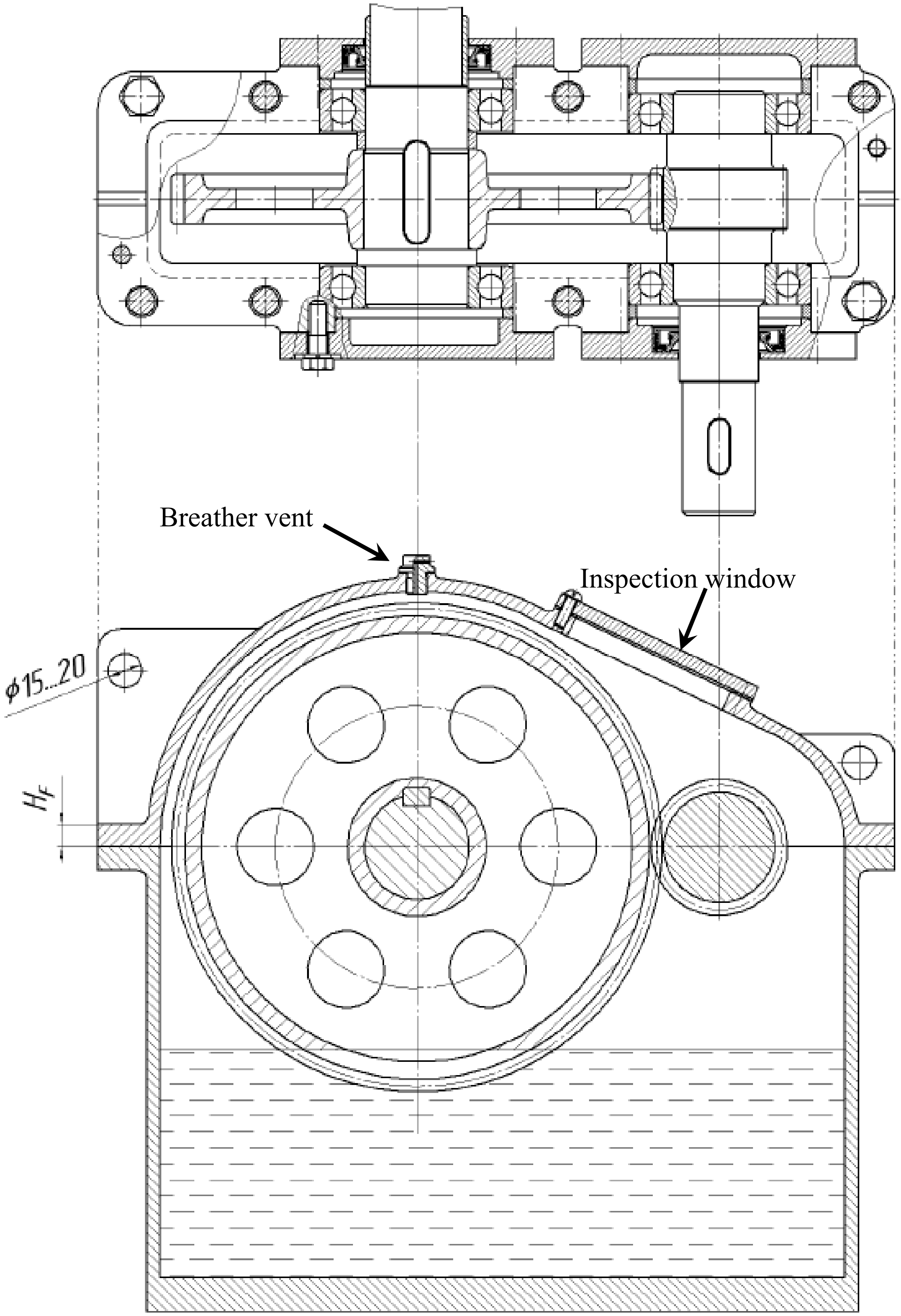


Fig. 6.6

4. Place bearing caps and bolts around the flanges (Fig. 6.7).

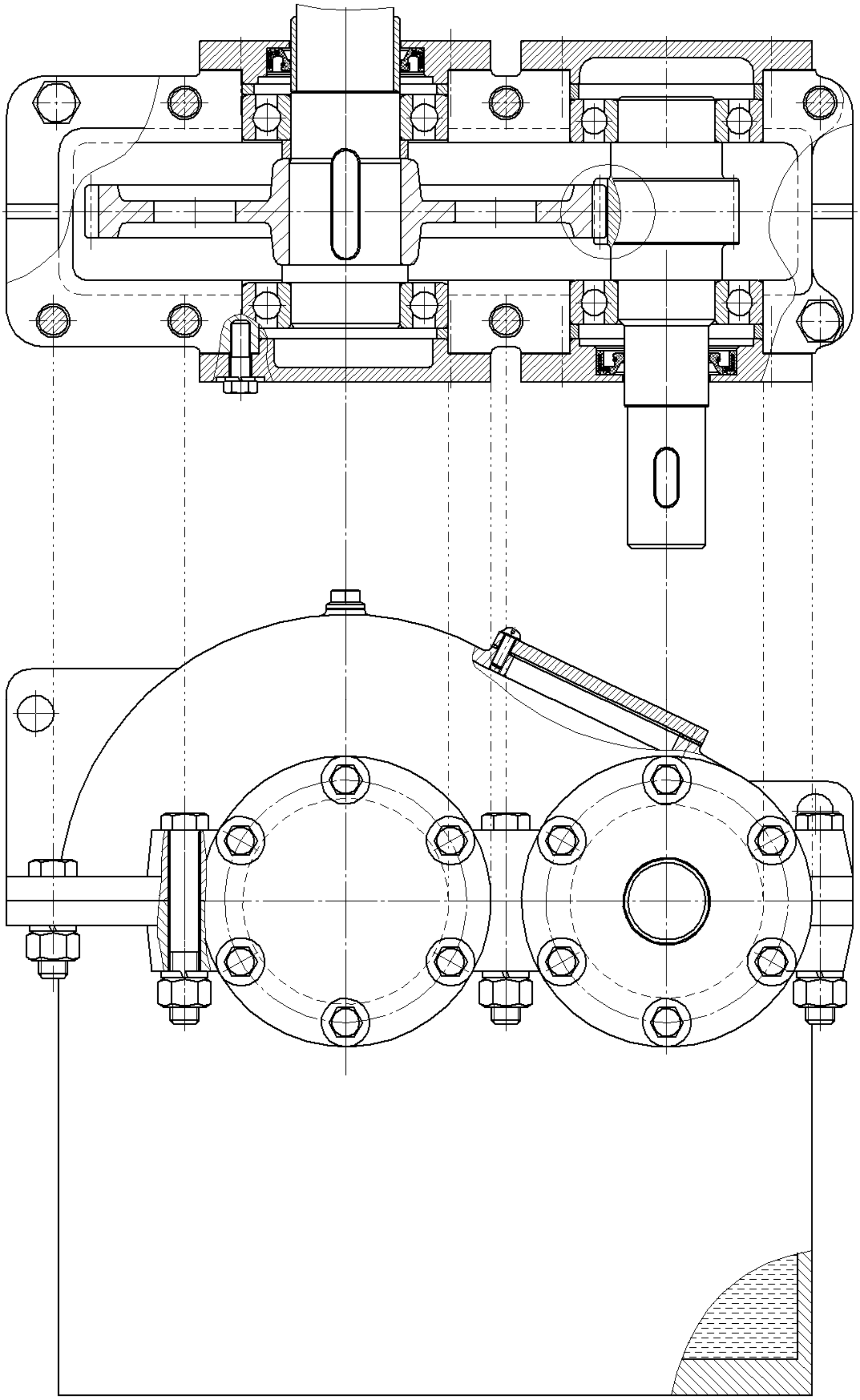


Fig. 6.7

5. Add oil bleed plug to close oil drain hole, oil dipstick to measure lubricant level, ribs to increase frame stiffness and holes for bolts to retain gearbox on baseplate (Fig. 6.8).

The rib thickness is usually made the same like a frame wall thickness, i.e. for cast iron 6…8 mm.

Bolt diameter may be calculated as

$$\boldsymbol{d_F = \sqrt[3]{4T_{out}} = \sqrt[3]{4 \cdot 194{,}4} = 9{,}2 \approx 10},$$

but should not be smaller 12 mm. Diameter of a hole for fixing bolt is standard, it has to be 1 mm bigger than standard bolt diameter.

Oil bleed plug thread major diameter is recommended 16…20 mm.

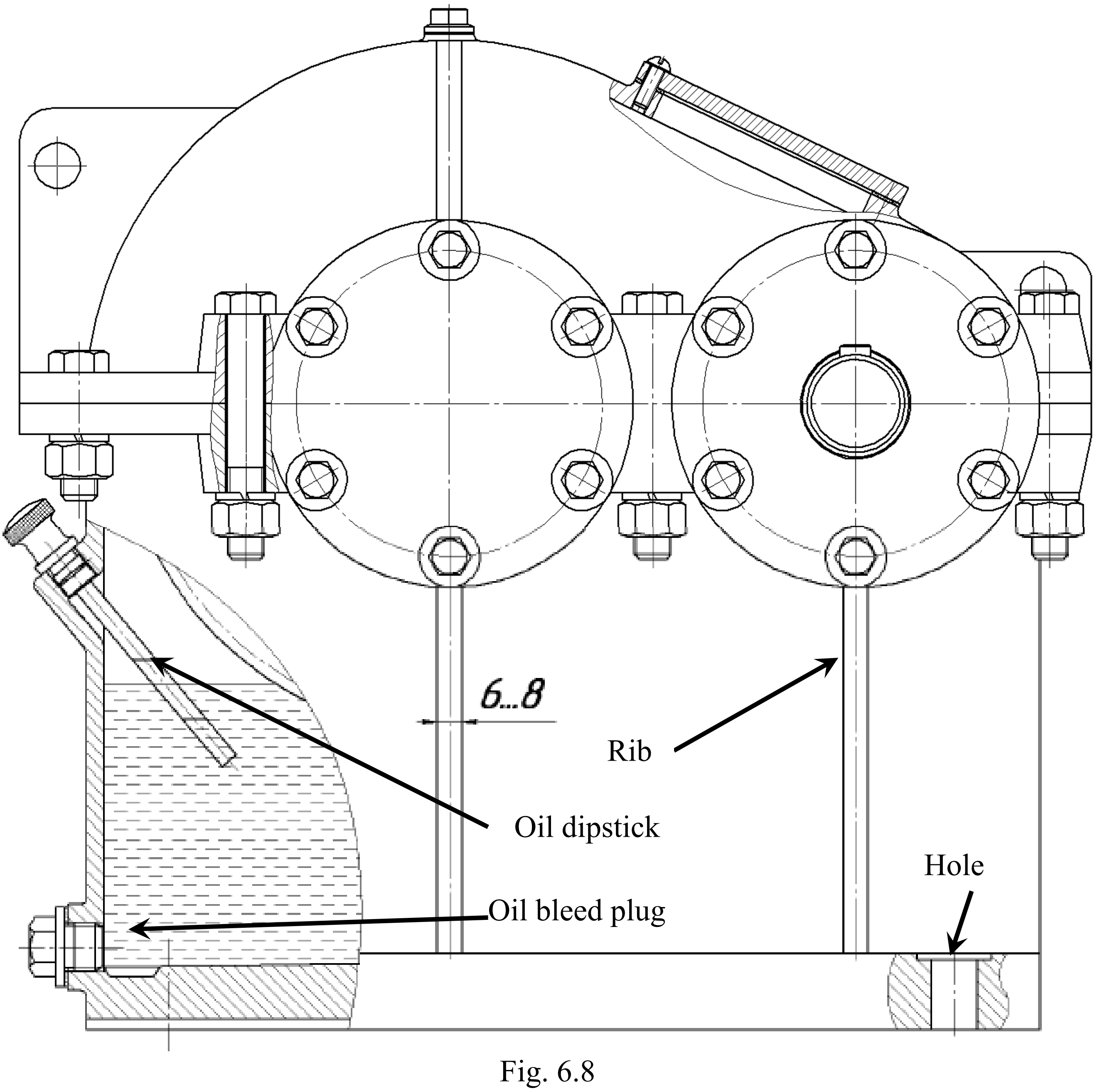


Fig. 6.8

In the same way, we can obtain the bevel gearbox frame (Fig. 6.9 and 6.10).

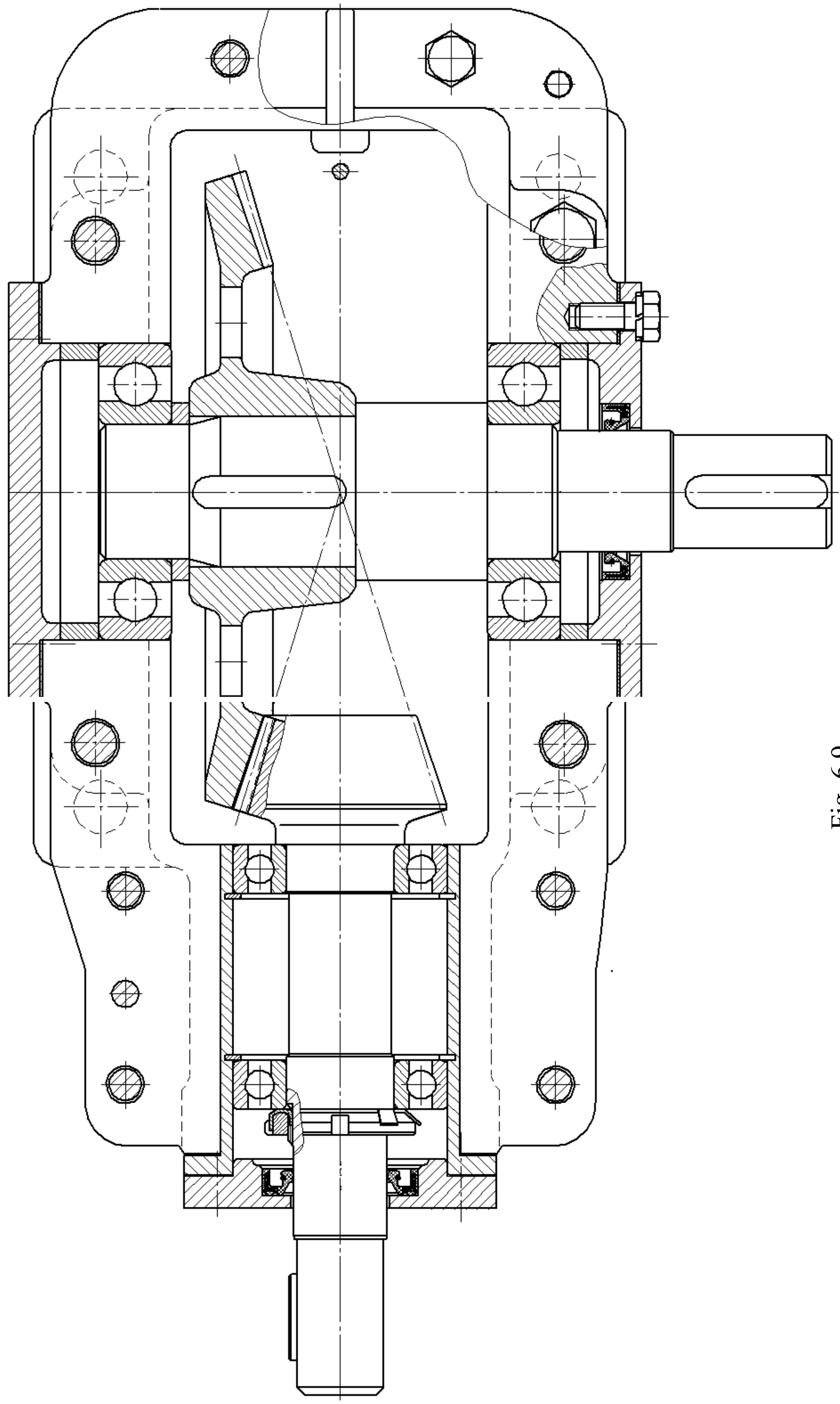


Fig. 6.9

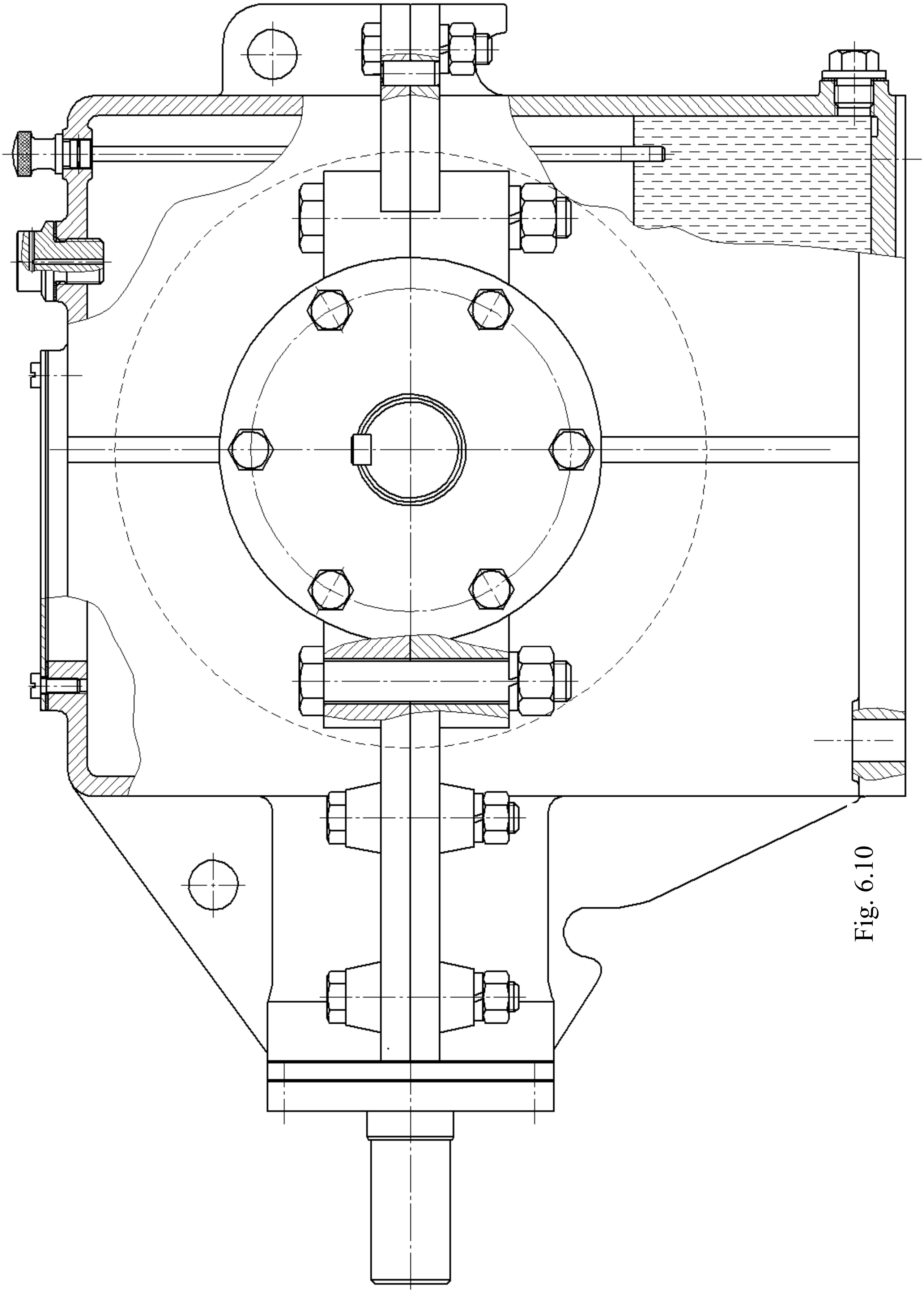


Fig. 6.10

6.2. Dimensions in Assembly Drawing

Three types of dimension are usually designated in assembly drawings:

– overall: total height, width and length of unit as a whole;

– fits: nominal size and tolerances' combination for surfaces mating according to the drawing;

– conjunctive: those that are used to joint a unit to the other parts (driven machine, coupling, baseplate, foundation, etc.) or to describe unit elements orientation and position in space (center distance, height of an axis over a foot, etc.).

6.2.1. Selection of Fit in General

When designing the fit itself, it is recommended to follow several principles:

– design a fit in a hole basis system (preferably) or in a shaft basis system;

– use hole tolerances greater or equal to the shaft tolerance;

– tolerances of the hole and shaft should not differ by more than two grades.

Depending on the mutual position of tolerance zones of the coupled parts, three **types** of fit can be distinguished:

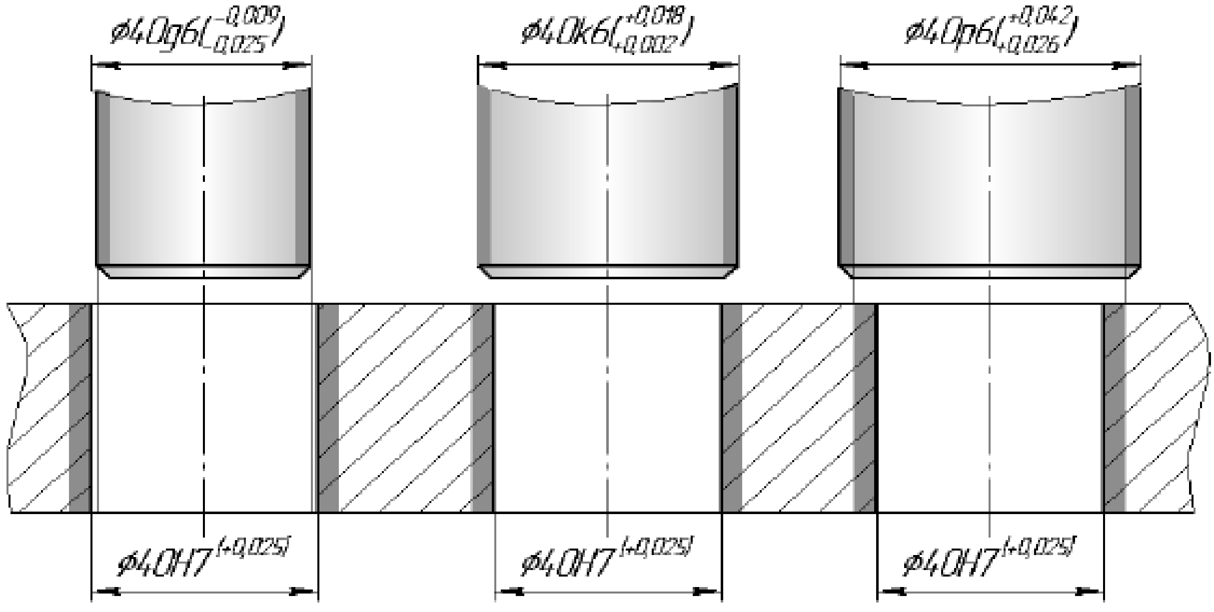
– **clearance fit** (Fig. 6.11, a) is a fit that always enables a clearance between the hole and shaft in the coupling. The lower limit size of the hole is greater or at least equal to the upper limit size of the shaft, i.e. the shaft is always smaller than the hole;

– **transition fit** (Fig. 6.11, b) is a fit where (depending on the actual sizes of the hole and shaft) both clearance and interference may occur in the coupling. Tolerance zones of the hole and shaft partly or completely interfere, the maximum clearance is positive and the minimum clearance is negative. Transition fits are used only for locating a shaft relative to a hole, where accuracy is important but either a clearance or an interference is permitted;

– **interference fit** (Fig. 6.11, c) is a fit always ensuring some interference between the hole and shaft in the coupling. The upper limit size of the hole is smaller or at least equal to the lower limit size of the shaft, i.e. the shaft is always larger in diameter than the hole – parts must be assembled by pressure or heat expansion.

A sufficient fit can be selected in Tab. 6.1 (preferred fits are in bold). The list of recommended fits given here is for information only and cannot be taken as a fixed listing. The enumeration of actually used fits may differ depending on the type and field of production, local standards and national usage and last but not least, depending on the plant practices.

Properties and field of use of some selected fits are described in Tab. 6.1. When selecting a fit it is often necessary to take into account not only constructional and technological views, but also economic aspects. Selection of a suitable fit is important particularly in view of those measuring instruments, gauges and tools, which are implemented in the production. Therefore, follow proven plant practices when selecting a fit.



a b c

Fig. 6.11

Table 6.1

Fits	Features	Field of application
Clearance fits		
H11/a11, **H11/c11**, H11/c9, H11/d11, A11/h11, **C11/h11**, D11/h11	Fits with great clearances with parts having great tolerances	Pivots, latches, fits of parts exposed to corrosive effects, contamination with dust and thermal or mechanical deformations
H9/C9, H9/d10, **H9/d9**, H8/d9, H8/d8, **D10/h9**, D9/h9, D9/h8	Running fits with greater clearances without any special requirements for accuracy of guiding shafts	Multiple fits of shafts of production and piston machines, parts rotating very rarely or only swinging
H9/e9, **H8/e8**, H7/e7, **E9/h9**, E8/h8, E8/h7	Running fits with greater clearances without any special requirements for fit accuracy	Fits of long shafts, e.g. in agricultural machines, bearings of pumps, fans and piston machines
H9/f8, H8/f8, **H8/f7**, **H7/f7**, F8/h7, **F8/h6**	Running fits with smaller clearances with general requirements for fit accuracy	Main fits of machine tools. General fits of shafts, regulator bearings, machine tool spindles, sliding rods
H8/g7, **H7/g6**, **G7/h6**	Running fits with very small clearances for accurate guiding of shafts. Without any noticeable clearance after assembly	Parts of machine tools, sliding gears and clutch disks, crankshaft journals, pistons of hydraulic machines, rods sliding in bearings, grinding machine spindles

End of Tab. 6.1

Fits	Features	Field of application
H11/h11, H11/h9	Slipping fits of parts with great tolerances. The parts can easily be slid one into the other and turn	Easily demountable parts, distance rings, parts of machines fixed to shafts using pins, bolts, rivets or welds
H8/h9, H8/h8, H8/h7, **H7/h6**	Sliding fits with very small clearances for precise guiding and centering of parts. Mounting by sliding on without use of any great force, after lubrication the parts can be turned and slid by hand	Precise guiding of machines and preparations, exchangeable wheels, roller guides
Transition fits		
H8/j7, **H7/js6**, H7/j6, J7/h6	Tight fits with small clearances or negligible interference. The parts can be assembled or disassembled manually	Easily dismountable fits of hubs of gears, pulleys and bushings, retaining rings, frequently removed bearing bushings
H8/k7, **H7/k6**, K8/h7, **K7/h6**	Similar fits with small clearances or small interferences. The parts can be assembled or disassembled without great force using a rubber mallet	Demountable fits of hubs of gears and pulleys, manual wheels, clutches, brake disks
H8/p7, H8/m7, H8/n7, H7/m6, **H7/n6**, M8/h6, N8/h7, **N7/h6**	Fixed fits with negligible clearances or small interferences. Mounting of fits using pressing and light force	Fixed plugs, driven bushings, armatures of electric motors on shafts, gear rims, flushed bolts
Interference fits		
H8/r7, **H7/p6**, **H7/r6**, **P7/h6**, R7/h6	Pressed fits with guaranteed interference. Assembly of the parts can be carried out using cold pressing	Permanent coupling of gears with shafts, hubs of clutch disks, bearing bushings
H8/s7, H8/t7, **H7/s6**, H7/t6, **S7/h6**, T7/h6	Pressed fits with medium interference. Assembly of parts using hot pressing. Assembly using cold pressing only with use of large forces	Permanent coupling of gears with shafts, bearing bushings
H8/u8, H8/u7, H8/x8, H7/u6, U8/h7, U7/h6	Pressed fits with big interferences. Assembly using pressing and great forces under different temperatures of the parts	Permanent couplings of gears with shafts without keys, flanges

6.2.2. Fit Selection for Bearings

Fit selection for bearings has a number of specific features. The main requirements are:

1. Adjust shaft and frame dimensions to standard bearing.
2. Ensure fixed conjunction for a rotating bearing race, or 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.

Bearing diameter tolerances are marked with letter ***L*** for the inner race and ***l*** for the outer one in combination with the degree of accuracy: ***L0***, ***L6***, ***l0***, ***l6***. Tolerances zones of the races are lower the nominal size (Fig. 6.12). This state should be taken into account when the fit is selected (Tab. 6.2).

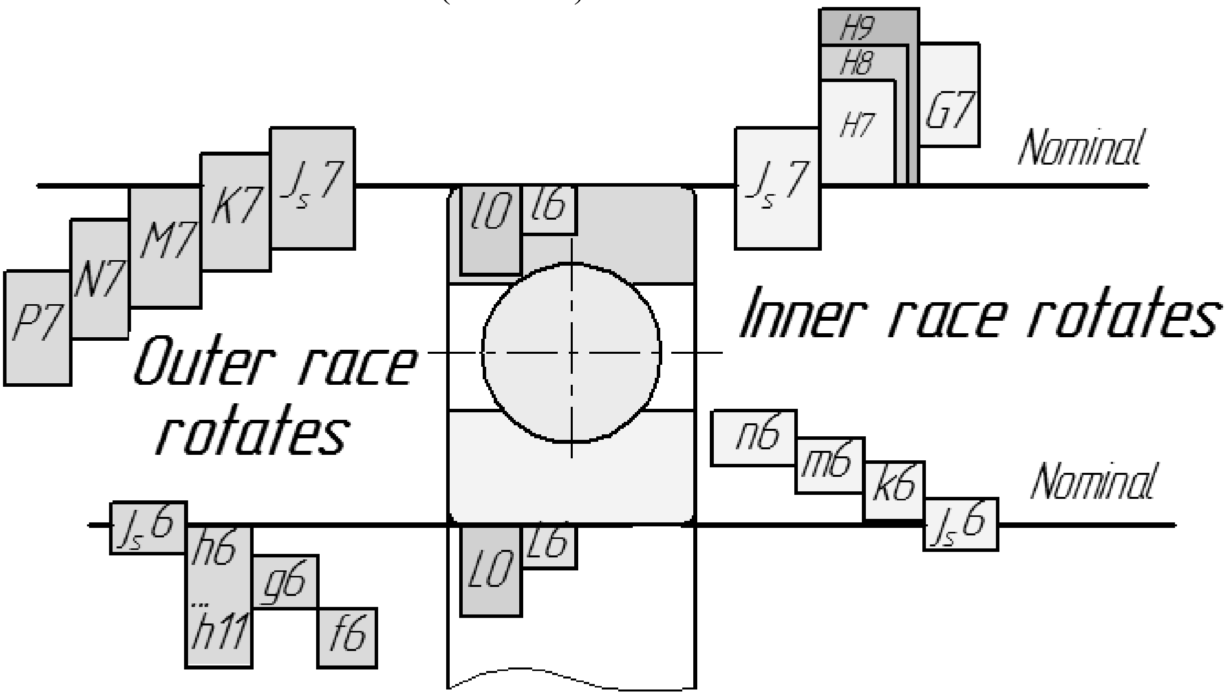


Fig. 6.12

Table 6.2

Rotating race	Fits for shaft	Fits for frame
Inner	L0/k6; L0/m6; L0/n6 L6/k6; L6/m6; L6/n6	H7/l0 H7/l6
Outer	L0/h6; L0/g6; L0/js6 L6/h6; L6/g6; L6/js6	M7/l0; N7/l0 M7/l6; N7/l6

See an example of the gearbox assembly drawing with dimensions in Fig. 6.13 and 6.14.

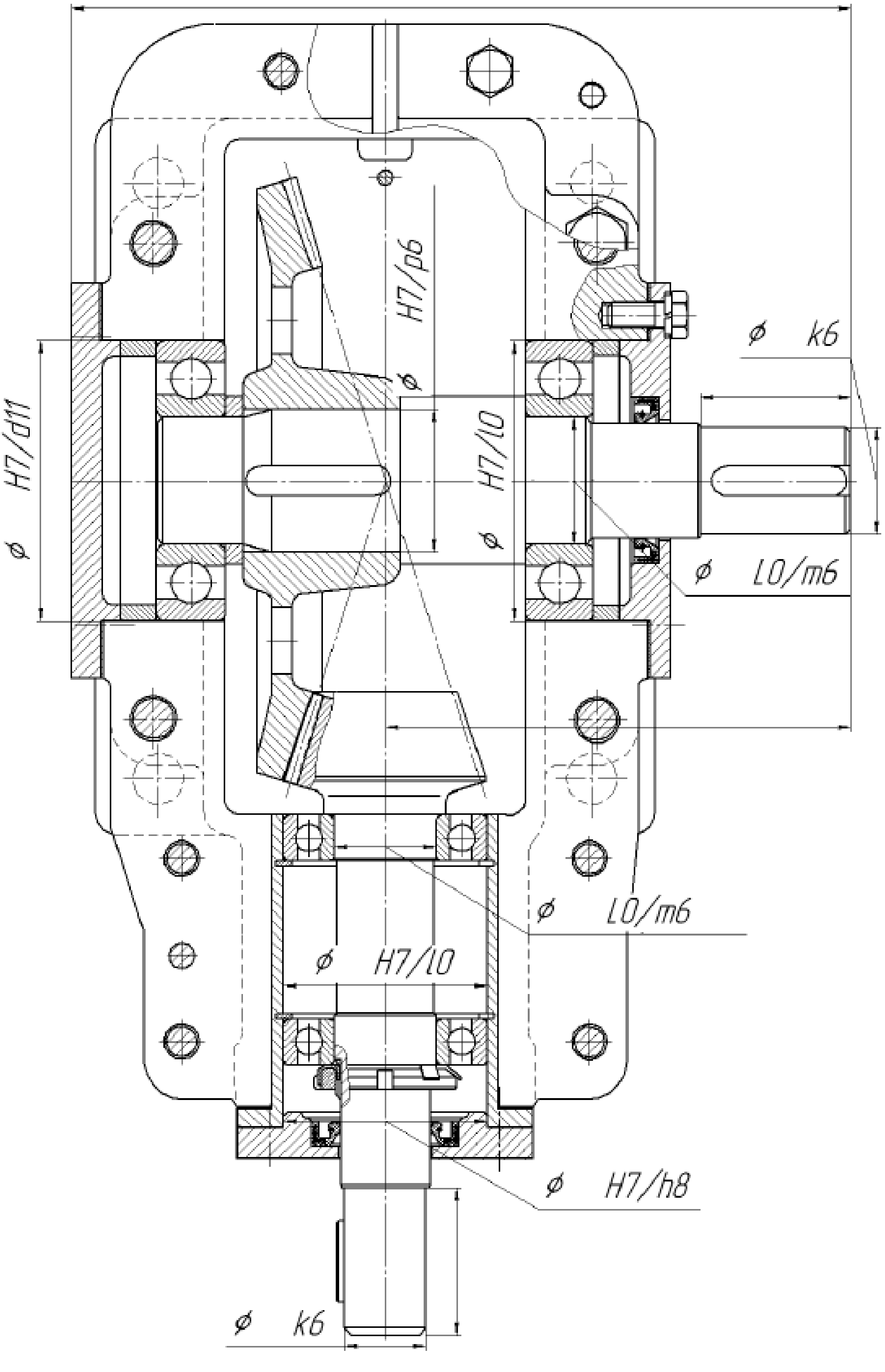


Fig. 6.13

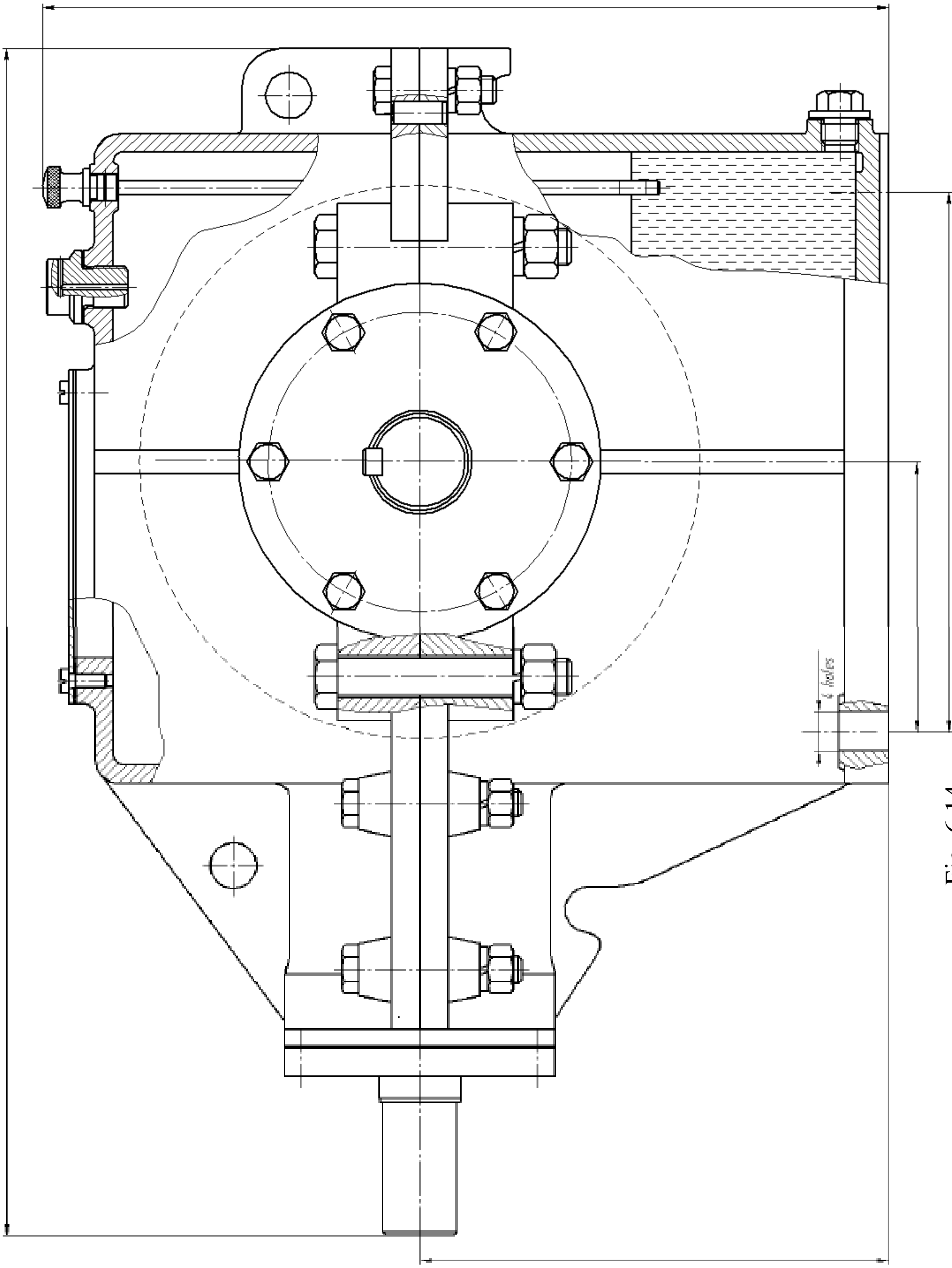


Fig. 6.14

7. GENERATING OF ELEMENT WORKING DRAWING

7.1. General States

Drawings of the individual parts are called piece part drawings, part drawings, or detail drawings. Detail drawings contain all of the necessary information to manufacture any specific part being created for a product or design. The information provided on detail drawings includes:

– all necessary drawing views and information to fully define the part shape. The number of an element projections, sections and species should be sufficient to make the element clear as a whole. On the main view, part should be placed in a position where most of its surfaces will be processed on, or a its axis is parallel to the drawing stamp;

– dimensions that can be specified in a drawing;

– tolerances specified in a drawing could be clearly understood;

– the material for the manufactured part;

– any general or specific notes including heat treatment, painting, coatings, hardness, pattern number, estimated weight, and surface finishes, such as maximum roughness.

The number of dimensions, tolerances, tolerances of form and arrangement of surfaces and their roughness should be **minimal** and **sufficient** for the manufacturing and part inspection.

Working drawings may be on more than one sheet and may contain written instructions called specifications.

Working drawing quality influences on cost and quality of designed machine as a whole.

7.2. Dimensions and Tolerances

Parts of a machine are designed in order to make a function. The working parts have a definite relationship with each other: free rotation, free longitudinal movement, clamping action, permanent fixed position. Precision is the degree of accuracy necessary to ensure the functioning of a part as intended.

There are two types of surfaces (Fig. 7.1):

1) non-mating parts – left either in their original rough-cast form or after draft machining;

2) mating parts – are machined to proper smoothness and must be accurate and at correct position relatively to each other.

It is principally impossible to produce machine parts with absolute dimensional accuracy. In fact, it is not necessary or useful. It is quite sufficient that the actual dimension of the part is found between two limit dimensions and a permissible deviation is kept with production to ensure correct functioning of engineering products.

The required level of accuracy of production of the given part is then given by the dimensional tolerance which is prescribed in the drawing. The production accuracy is prescribed with regards to the functionality of the product and to the economy of production as well.

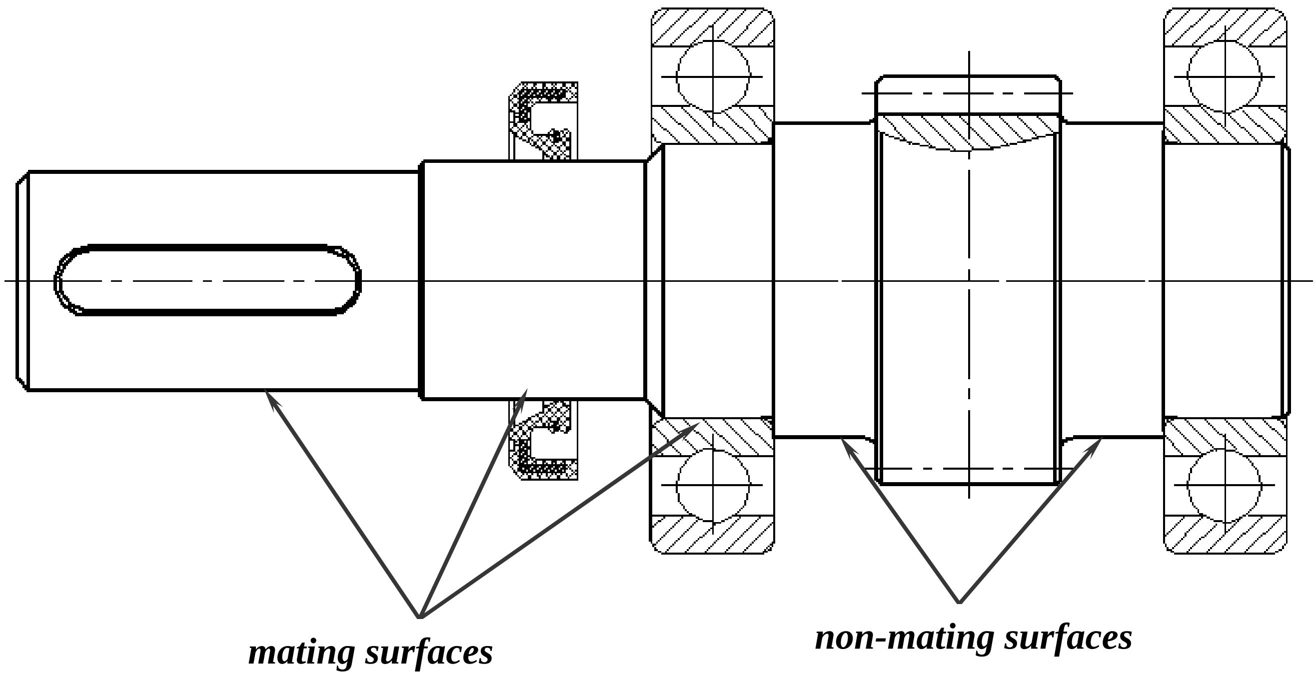


Fig. 7.1

In an assembly process the degree of «clearance» or «tightness» desired between mating parts is important. So greater manufacturing precision is required for these surfaces. However because of impossibility to make a distance to an absolute size some variation must be allowed.

Tolerance is the allowable variation for any given size in order to achieve a proper function.

The system of tolerances and fits ISO can be applied in tolerances and deviations of smooth parts and for fits created by their coupling. It is used particularly for cylindrical parts with round sections. Tolerances and deviations in this standard can also be applied in smooth parts of other sections. Similarly, the system can be used for coupling (fits) of cylindrical parts and for fits with parts having two parallel surfaces (e.g. fits of keys in grooves).

The term "**shaft**", used in this standard has a wide meaning and serves for specification of all outer elements of the part, including those elements which do not have cylindrical shapes. Also, the term "**hole**" can be used for specification of all inner elements regardless of their shape.

Basic size is the size whose limit dimensions are specified using the upper and lower deviations. In case of a fit, the basic size of both connected elements must be the same.

The tolerance of a size is defined as the difference between the upper and lower limit dimensions of the part. In order to meet the requirements of various production branches for accuracy of the product, the system ISO implements 20 grades of accuracy. Each of the tolerances of this system is marked "IT" with attached grade of accuracy (IT01, IT0, IT1...IT18).

Field of use of individual tolerances of the system ISO:

IT01 to IT6	For production of gauges and measuring instruments
IT5 to IT12	For fits in precision and general engineering
IT11 to IT16	For production of semi-products
IT16 to IT18	For structures
IT11 to IT18	For specification of limit deviations of non-tolerated dimensions

Note: When choosing a suitable dimension it is necessary to also take into account the used method of machining of the part in the production process.

The tolerance zone is defined as a zone limited by the upper and lower limit dimensions of the part. The tolerance zone is therefore determined by the amount of the tolerance and its position related to the basic size. The position of the tolerance zone, related to the basic size (zero line), is determined in the ISO system by a so-called basic deviation. The system ISO defines 28 classes of basic deviations for holes and shafts. These classes are marked by capital letters for holes (A, B, C, ... ZC) and by lower case letters for shafts (a, b, c, ... zc) (Fig. 7.2). The tolerance zone for the specified dimensions is prescribed in the drawing by a tolerance mark, which consists of a letter marking of the basic deviation and a numerical marking of **the tolerance grade** (e.g. H7, H8, D5, h7, h6, g5, etc.).

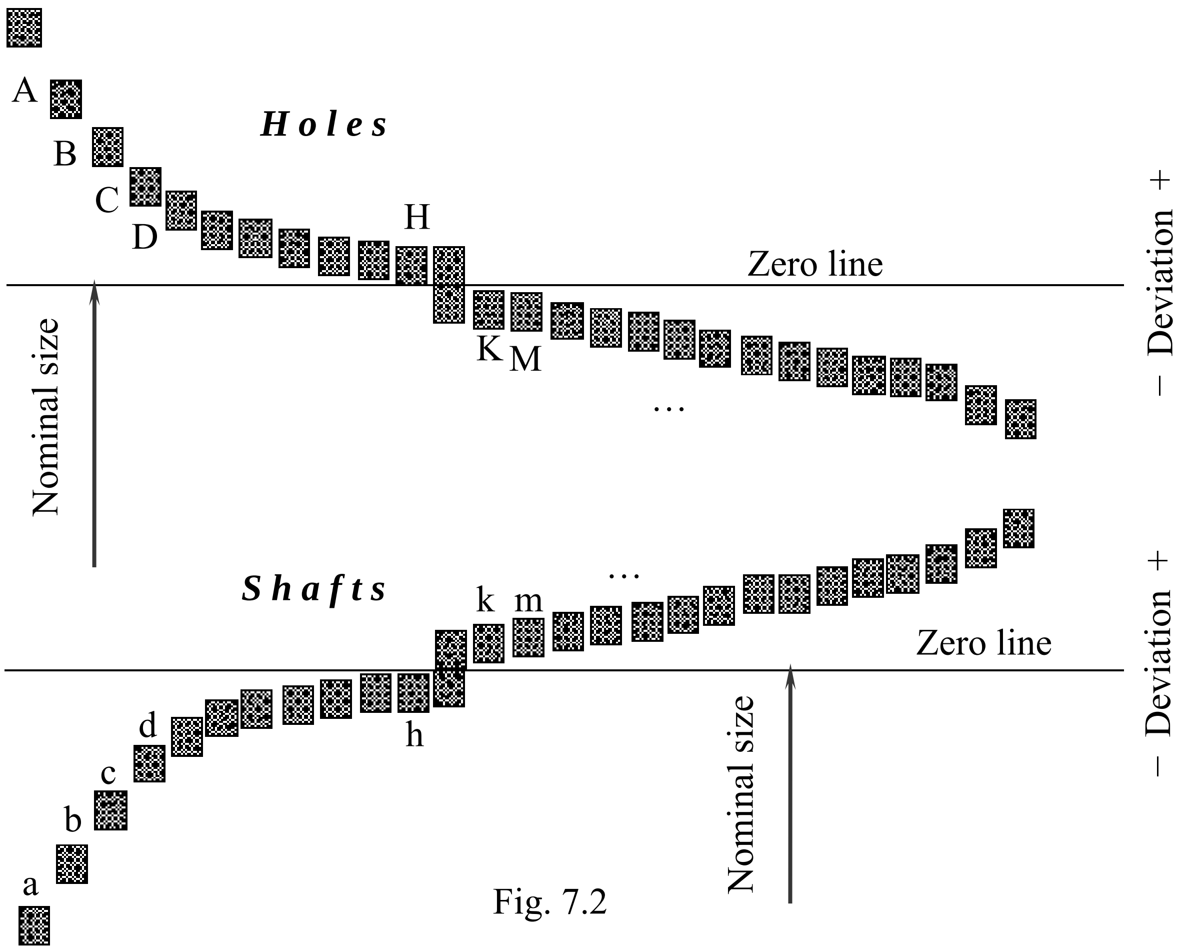


Fig. 7.2

Though the general sets of basic deviations (marked by letter) and tolerance grades (IT1 ... IT18) can be used for prescriptions of tolerance zones by their mutual combinations, in practice only a limited range of tolerance zones is used. An overview of tolerance zones for general use can be found in the following tables (Tab. 7.1 for shafts, Tab. 7.2 for holes; preferred zones are in bold). The tolerance zones not included in these tables are considered as special zones and their use is recommended only in technically well-grounded cases.

Table 7.1

Basic deviations	**Tolerance grades**									
	5	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**
a					a9	a10	a11	a12	a13	
b					b9	b10	b11	b12	b13	
c				c8	c9	c10	c11	c12		
d	d5	d6	**d7**	**d8**	**d9**	**d10**	**d11**	d12	d13	
e	e5	**e6**	**e7**	**e8**	**e9**	**e10**				
f	f5	**f6**	**f7**	**f8**	**f9**	f10				
g	g5	**g6**	**g7**	**g8**	g9	g10				
h	**h5**	**h6**	**h7**	**h8**	**h9**	**h10**	**h11**	**h12**	**h13**	**h14**
js	**js5**	**js6**	**js7**	**js8**	**js9**	**js10**	**js11**	**js12**	**js13**	**js14**
k	k5	**k6**	**k7**	**k8**	**k9**	**k10**	**k11**	**k12**	**k13**	
m	m5	**m6**	**m7**	m8	m9					
n	n5	**n6**	**n7**	n8	n9					
p	p5	**p6**	**p7**	**p8**	p9	p10				
r	r5	**r6**	**r7**	**r8**	r9	r10				
s	s5	**s6**	**s7**	**s8**	s9	s10				
t	t5	**t6**	**t7**	t8						
u	u5	**u6**	**u7**	**u8**	u9					

Table 7.2

Basic deviations	Tolerance grades									
	5	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**
A					A9	A10	A11	A12	A13	
B				B8	B9	B10	B11	B12	B13	
C				C8	C9	C10	C11	C12	C13	
D		**D6**	**D7**	**D8**	**D9**	**D10**	**D11**	**D12**	**D13**	
E	E5	**E6**	**E7**	**E8**	**E9**	**E10**				
F	F5	**F6**	**F7**	**F8**	**F9**	F10				
G	G5	**G6**	**G7**	**G8**	G9	G10				
H	**H5**	**H6**	**H7**	**H8**	**H9**	**H10**	**H11**	**H12**	**H13**	**H14**
JS	**JS5**	**JS6**	**JS7**	**JS8**	**JS9**	**JS10**	**JS11**	**JS12**	**JS13**	**JS14**
K	K5	**K6**	**K7**	**K8**						
M	M5	**M6**	**M7**	**M8**	M9	M10				
N	N5	**N6**	**N7**	**N8**	**N9**	N10	N11			
P	P5	**P6**	**P7**	**P8**	**P9**	P10				
R	R5	**R6**	**R7**	**R8**	R9	R10				
S	S5	**S6**	**S7**	**S8**	S9	S10				
T	T5	**T6**	**T7**	**T8**						
U	U5	**U6**	**U7**	**U8**	U9	U10				

There are some standard fundamental rules that need to be applied:

1. **All dimensions** must have a tolerance. Every feature on every manufactured part is a subject to variation, therefore, the limits of allowable variation must be specified. Plus and minus tolerances may be applied directly to dimensions or applied from a general tolerance block or general note.

Tolerances are specified in a drawing using toleranced dimensions. For those dimensions that are not specifically toleranced, a general tolerance note is used. The only exceptions are for dimensions marked as minimum, maximum, stock or reference (reference dimension is usually used for information purposes only, it does not govern production or inspection operations).

2. Dimensioning and tolerancing shall completely define the **nominal** geometry and **allowable** variation. Measurement and scaling of the drawing is not allowed except in certain cases.

3. Engineering drawings define the requirements of finished (complete) parts. Every dimension and tolerance required to define the finished part shall be shown on the drawing. If additional dimensions would be helpful, but are not required, they may be marked as reference.

Dimensions should be applied to features and arranged in such a way as to represent the function of the features. Additionally, dimensions should not be subject to more than one interpretation.

Descriptions of manufacturing methods should be avoided. The geometry should be described without explicitly defining the method of manufacture.

If certain sizes are required during manufacturing but are not required in the final geometry (due to shrinkage or other causes) they should be marked as non-mandatory.

All dimensioning and tolerancing should be arranged for maximum readability and should be applied to visible lines in true profiles.

7.3. Roughness

In accordance with standard values in these ranges is provided from the series: 12.5; 10; 8; 6.3; 5; 4; 3,2; 2.5; 2; 1,6; 1.25; 1,0; 0.8; 0.63.

Working surface roughness of the **gear wheels** is shown in Tab. 2.7.

For the hub inner hole the following values of the roughness parameter $\boldsymbol{R_a}$ are appointed: for shaft diameter less than 80 mm – 2,5...3,2; 80 mm and more – 3,2...6,3.

For top land surface and base side face surfaces, $\boldsymbol{R_a}$ is selected depending on the accuracy of the product and may be equal 1,6...6,3.

For gear wheel free surfaces – $\boldsymbol{R_a}$ 6,3...12,5.

Working surface roughness of a **shaft** and housing at the **bearing seats** is shown in Tab. 7.3.

Table 7.3

Fitting surfaces	Bearing degree of accuracy	Parameter R_a, μm	
		Bore diam. < 80 mm	Bore diam. > 80 mm
Shaft	0 and 6	1,25	2,5
	5 and 4	0,63	1,25
Hole in a frame	0 and 6	1,6…1,25	3,2…2,5
	5, 4, 2	1,25…0,63	2,5…1,25
Thrust fillet on a shaft or in a frame	0 and 6	2,5	2,5
	5, 4, 2	1,25	2,5

Shaft surface roughness under a **rubber oil seal** depends on the shaft peripheral velocity: up to 3 m/s $\boldsymbol{R_a}$ = 0,8…1,25; in case of 3…5 m/s $\boldsymbol{R_a}$ = 0,32…0,63; more than 5 m/s – $\boldsymbol{R_a}$ = 0,1…0,25. To obtain such roughness the shaft surface has to be polished.

Roughness of the other surfaces see in Tab. 7.4.

Table 7.4

Typical surface	Roughness parameter, μm
Hole for a fastener Grooves Gearbox footing	R_z 80
Machined free surfaces of the shaft end faces Machined surfaces for a fastener/washer head	R_z 40
Shaft and bush abutting surfaces (not centering) Metric thread working surfaces	R_z 20
Keyseat and keyway parallel faces	R_a 3,2
Keyseat bottom	R_a 6,3
Keyway bottom	R_a 2,5
Shaft thrust fillet for gear – for hub diam. < 80 mm – for hub diam. > 80 mm	 R_a 1,6 R_a 3,2
Oil seal seats in the bearing caps – flat – cylinder	 R_a 12,5 R_a 3,2
Bearing cap fit cylinder and thrust face	R_a 2,5

7.4. Geometrical Tolerances

A design model is an idealized representation of a part design. However, the design model by itself does not fully define the design. Due to imperfections in manufacturing and inspection processes, physical parts never match the design model exactly. An important aspect of a design is to specify the amount the part features may deviate from their theoretically exact geometry (as defined in the design model).

Geometric dimensioning and tolerancing, often referred to as GD&T, is a symbolic language used on engineering drawings and models to define the allowable deviation of feature geometry. The language of GD&T consists of dimensions, tolerances, symbols, definitions, rules, and conventions that can be used to precisely communicate the functional requirements for the location, orientation, size, and form of each feature of the design model. Thus, GD&T is an exact language that enables designers to “say what they mean” with regard to their design models. Production can then use the language to understand the design intent and inspection looks to the language to determine set up requirements.

In order for any language to be effective, it must be based on a common standard. ASME and ISO both have standards related to the application and use of GD&T. The ASME standards are generally used by U.S. companies while the ISO standards are commonly used by European companies. Within each of those organizations there are several standards that are applicable to the topic of GD&T. Furthermore, each standard is periodically revised, with each version of the standard identified by the year of its acceptance.

The main types of tolerance, their symbols and short descriptions are represented in Tab. 7.5.

Table 7.5

Type of tolerance	**Geometric characteristics**	**Symbol**	**Description**
Form	Circularity (roundness)	○	Describes the condition on a surface of revolution (cylinder, cone, sphere) where all points of the surface intersected by any plane
Form	Cylindricity	⌭	Describes a condition of a surface of revolution in which all points of a surface are equidistant from a common axis
Form	Planarity (flatness)	⏥	Is the condition of a surface having all elements in one plane
Form	Straightness	—	A condition where an element of a surface or an axis is a straight line
Location	Concentricity	◎	Describes a condition in which two or more features, in any combination, have a common axis
Location	Positional Tolerance	⌖	Defines a zone within which the axis or center plane of a feature is permitted to vary from true (theoretically exact) position
Location	Symmetry	⌯	Is a condition in which a feature (or features) is symmetrically disposed about the center plane of a datum feature
Orientation	Angularity	∠	Is the condition of a surface, axis, or centerplane, which is at a specified angle from a datum plane or axis
Orientation	Parallelism	∥	Is the condition of a surface, line, or axis, which is equidistant at all points from a datum plane or axis
Orientation	Perpendicularity	⊥	Is the condition of a surface, axis, or line, which is 90 deg. From a datum plane or a datum axis
Run-out	Circular run-out	↗	Is the composite deviation from the desired form of a part surface of revolution through on full rotation (360 deg) of the part on a datum axis.
Run-out	Total run-out	⌰	Is the simultaneous composite control of all elements of a surface at all circular and profile measuring positions as the part is rotated through 360

GD&T information in a drawing is placed into Feature Control Frame. It is a rectangular box containing the geometric characteristics symbol, and the form, runout or location tolerance. If necessary, datum references and modifiers applicable to the feature or the datums are also contained in the box. 

Datum feature is the actual component feature used to establish a datum. A datum is a virtual ideal plane, line, point, or axis. A datum feature is a physical feature of a part identified by a datum feature symbol and corresponding datum feature triangle, e.g., . These are then referred to by one or more datum references which indicate measurements that should be made with respect to the corresponding datum feature.

See below recommended values of:

– the spur gear outer (addendum) diameter d_a allowable radial runout (Tab. 7.6);

– the spur gear face width ***b*** allowable face runout (Tab. 7.7);

– the spur gear hub allowable face runout (Tab. 7.8);

– the bevel gear allowable addendum cone runout (Tab. 7.9);

– the allowable runout of a shaft thrust fillet fixing a gear wheel (Tab. 7.10);

– the allowable runout of a shaft thrust fillet fixing a bearing (Tab. 7.11).

Table 7.6

Degree of accuracy	**Module *m*, mm**	**Pitch diameter *d*, mm**			
		< 125	**125…400**	**400…800**	**800…1600**
		Allowable radial runout, µm			
5	1,0…3,5	16	22	28	32
	3,5…6,3	16	25	32	36
	6,3…10	20	28	36	40
6	1,0…3,5	25	36	45	50
	3,5…6,3	28	40	50	56
	6,3…10	32	45	55	63
7	1,0…3,5	36	50	63	71
	3,5…6,3	40	56	71	80
	6,3…10	45	63	80	90
8	1,0…3,5	45	63	80	90
	3,5…6,3	50	71	90	100
	6,3…10	56	80	100	112

Table 7.7

Teeth contact degree of accuracy	**Per 100 mm of the pitch diameter for the face width, mm**			
	< 30	**30…55**	**55…110**	**110…160**
	Allowable face runout, µm			
6	24	13	7	6
7	32	15	9	9
8	58	25	15	10
9	86	42	24	12

Table 7.8

Kinematic degree of accuracy	**Hub bore diameter, mm**		
	< 50	**50…80**	**> 80**
	Allowable face runout, µm		
6 and 7	20	30	40
8 and 9	30	40	50

Table 7.9

Degree of accuracy	**Mean normal module m_n, mm**	**Mean pitch diameter d_m, mm**		
		< 125	**125…400**	**400…800**
		The allowable runout for addendum cone, µm		
5	1,0…3,5	16	22	28
	3,5…6,3	18	25	32
	6,3…10	20	28	36
6	1,0…3,5	25	36	45
	3,5…6,3	28	40	50
	6,3…10	32	45	56
7	1,0…3,5	36	50	63
	3,5…6,3	4	56	71
	6,3…10	45	63	80
8	1,0…3,5	45	63	80
	3,5…6,3	50	71	90
	6,3…10	56	80	100

Table 7.10

Shaft diameter, mm	**Degree of accuracy**			
	6	**7**	**8**	**9**
	The allowable runout, µm			
16…25	6	10	16	25
25…40	8	12	20	30
40…63	10	16	25	40
63…100	12	20	30	50

Table 7.11

Shaft diameter, mm	The allowable runout, µm					
	Radial			Axial		
	Bearing degree of accuracy					
	0	6	5	0	6	5
< 50	0,5 IT (IT – shaft diameter tolerance)		0,25 IT	20	10	7
50…100				25	12	8

7.5. Table of Parameters

Table of parameters (TOP) is placed in the upper right corner of the drawing (Fig. 7.3). The form and content of the table are standard.

TOP for all gear types consists of three sections separated by the main lines. The number of rows in each section is different according to the type of gears.

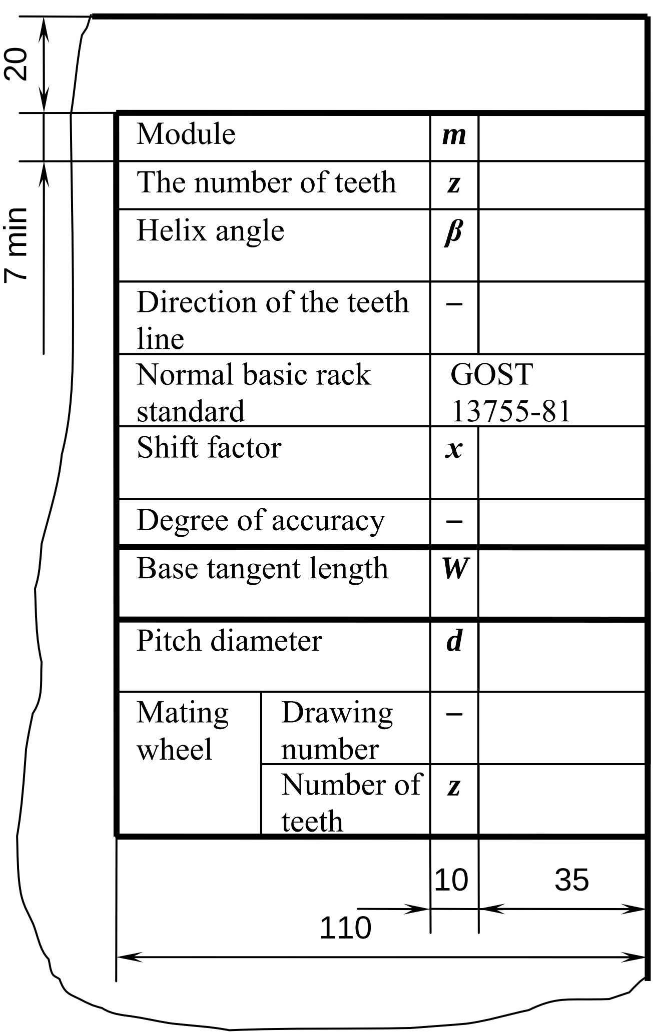


Fig. 7.3

7.5.1. TOP for Spur Gears

The first section of the TOP includes the basic data for teeth cutting:

a) the module ***m*** (for helical gears □ normal module m_n);

b) the number of teeth ***z***;

c) the angle of tooth inclination ***β*** (for helical gears only);

d) direction of the teeth line (tooth helix direction is indicated by the words «Right» or «Left»; for spur gears rows c) and d) are excluded from the TOP);

e) the normal basic rack standard (GOST 13755-81);

f) shift factor ***x*** (if it is absent the digit «0» is marked) ;

g) the degree of accuracy, type of coupling and standard backlash standard (e.g., GOST 1643-81).

In the second portion of the TOP data to inspect profile accuracy are provided. There may be:

– the constant tooth chord $\overline{S}_c$ and its height $\overline{h}_c$;

– the base tangent length W. Gear teeth thickness is determined by measuring length across a certain number of teeth;

– chordal tooth thickness $\overline{S}_y$ and its height $\overline{h}_{ay}$;

– dimension over pins (DOP) or balls M and their diameter D.

Calculation of these parameters is considered below.

In the third portion next parameters are shown:

a) pitch diameter d;

b) other reference data.

7.5.2. Calculation of the Constant Chord and its Height for Spur Wheels

The constant chord is independent of the number of teeth and this is a useful measure where a large number of gears have the same modular pitch and pressure angle. The tooth thickness $\overline{S}_c$ is measured at the points of contact at a dimension $\overline{h}_c$ from the tip of the tooth.

The tooth thickness theoretical values at the constant chord (Fig. 7.4)

$$\overline{S}_c^* = (\frac{\pi}{2} cos^2 \alpha + x \sin 2\alpha) m ,$$

where m is the module (for helical gears m_n); x is the shift coefficient; α is the angle of the tooth profile of the basic rack in the normal section. When α = 20 °

$$\overline{S}_c^* = (1{,}387 + 0{,}643x) m ,$$

for wheels without shift

$$\overline{S}_c^* = 1{,}387 m .$$

The actual thickness of the tooth at a constant chord, mm, marked on the drawings, is

$$\overline{S}_c = (\overline{S}_c^* - E_{s\bar{c}}) - T_{\bar{c}} ,$$

where $E_{s\bar{c}}$ is the smallest deviation tooth thickness at constant chord (Tab. 7.12); $T_{\bar{c}}$ is the thickness tolerance along the constant chord (Tab. 7.13).

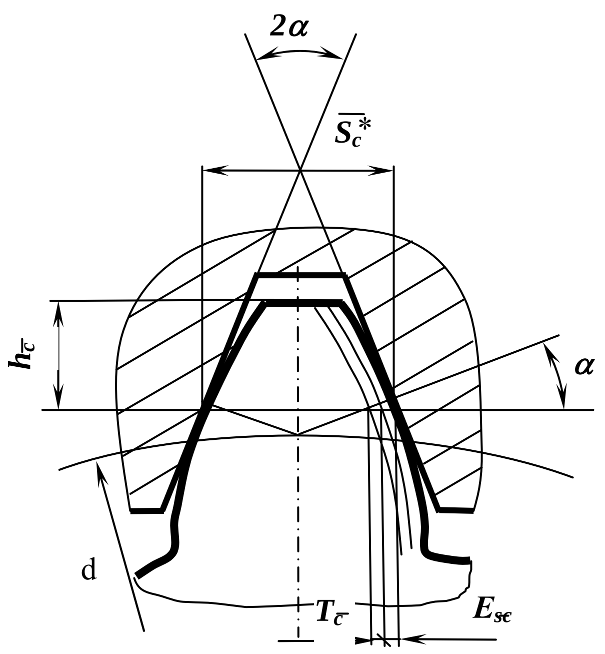


Fig. 7.4

Table 7.12

Type of coupling	Degree of accuracy	Pitch diameter, mm						
		< 80	80… …125	125… …180	180… …250	250… …315	315… …400	400… …500
		The tooth thickness smallest deviation at constant chord, μm						
C	3–6	55	60	70	80	90	100	110
	7	60	70	80	90	100	120	140
	8	70	80	90	100	120	140	140
	9	70	90	100	120	140	140	140
B	3–6	90	100	120	140	160	160	180
	7	100	120	140	140	180	180	200
	8	100	120	140	160	180	200	220
	9	120	140	160	180	200	220	250
A	3–6	140	160	180	200	250	250	300
	7	150	180	200	220	250	300	350
	8	160	200	220	250	300	350	350
	9	180	200	250	300	300	350	350

Table 7.13

Type of coupling	Type of tolerance	Addendum diameter allowable radial runout, μm							
		16… …20	20… …25	25… …32	32… …40	40… …50	50… …60	60… …80	80… …100
		Thickness tolerance along the constant chord, μm							
H, E	h	30	35	40	45	50	70	70	90
D	d	40	45	50	60	70	70	100	120
C	c	50	60	70	70	90	100	140	160
B	b	60	70	70	90	100	140	140	180
A	a	70	80	100	120	140	140	180	220

Tooth addendum to constant chord with $\boldsymbol{\alpha} = 20°$

$$\bar{h}_c = 0{,}5(d_a - d - 0{,}364\bar{S}_c^*),$$

where d_a is the tooth addendum diameter, d is the pitch diameter. For gears without shift height of constant chord equals $\bar{h}_c = 0{,}748m$.

7.5.3. Calculation of the Base Tangent Length

The base tangent length W is a distance between two heteronymic side surfaces of the teeth AB (Fig. 7.5) along the tangent to the base circle. Measurement of the base tangent length has the advantage over constant chord: it is not required more precise manufacturing of the gears on the outer diameter.

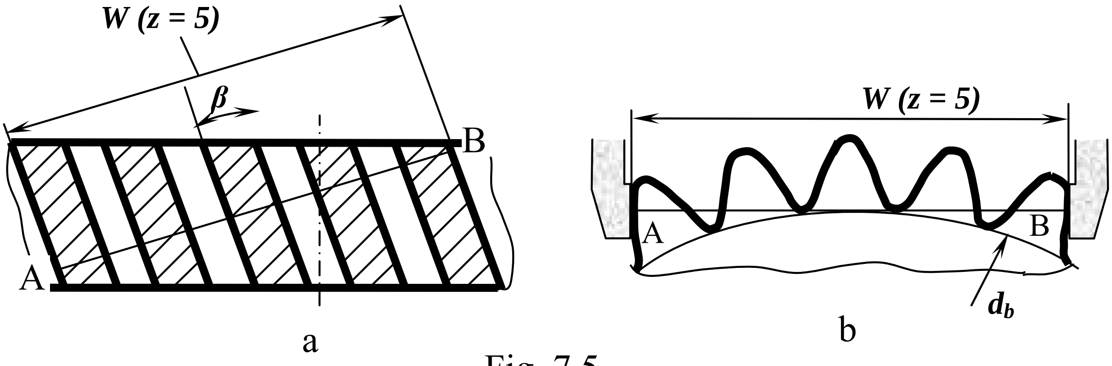


Fig. 7.5

For spur gears (Fig. 7.5, b) without shift it equals ***W*** = ***mW '***, with the shift – ***W*** = ***m (W' + 0,684 x)***, where ***W '*** is the base tangent length in case of module 1 mm.

The values ***W '*** and the terminal number of teeth z_n depending on the number of gear teeth ***z*** is given in Tab. 7.14.

Table 7.14

z	z_n	*W '*	*z*	z_n	*W '*	*z*	z_n	*W '*
18		7,6324	79	9	26,1995	140		47,7180
19		7,6464	80		29,1657	141		47,7320
20		7,6604	81		29,1797	142	16	47,7460
21	3	7,6744	82		29,1937	143		47,7600
22		7,6884	83		29,2077	144		47,7740
23		7,7024	84	10	29,2217	145		50,7410
24		7,7165	85		29,2357	146		50,7550
25		7,7305	86		29,2497	147		50,7690
26		10,6966	87		29,2637	148		50,7830
27		10,7106	88		29,2777	149	17	50,7970
28		10,7246	89		32,2438	150		50,8110
29		10,7386	90		32,2558	151		50,8250
30	4	10,7526	91		32,2678	152		50,8390
31		10,7666	92		32,2798	153		50,8530
32		10,7806	93	11	32,2918	154		53,8192
33		10,7946	94		32,3038	155		53,8332
34		10,8086	95		32,3158	156		53,8470
35		13,7748	96		32,3278	157		53,8610
36		13,7888	97		32,3398	158	18	53,8750
37		13,8028	98		35,3320	159		53,8890
38	5	13,8168	99		35,3360	160		53,9030
39		13,8308	100	12	35,3501	161		53,9170
40		13,8448	101		35,3641	162		53,9310
41		13,8588	102		35,3781	163	19	56,8973
42		13,8728	103		35,3921	164		56,9113

End of Tab. 7.14

z	z_n	W'	z	z_n	W'	z	z_n	W'
43	5	13,8868	104	12	35,4060	165	19	56,9250
44	6	16,8530	105		35,4200	166		56,9390
45		16,8670	106		35,4340	167		56,9530
46		16,8810	107		35,4480	168		56,9670
47		16,8950	108		35,4620	169		56,9810
48		16,9090	109	13	38,4283	170		56,9950
49		16,9230	110		38,4423	171		57,0090
50		16,9370	111		38,4563	172	20	56,8973
51		16,9510	112		38,4703	173		56,9113
52		16,9650	113		38,4840	174		56,9250
53	7	19,9311	114		38,4980	175		56,9390
54		19,9452	115		38,5120	176		56,9530
55		19,9592	116		38,5260	177		56,9670
56		19,9732	117		38,5400	178		56,9810
57		19,9872	118	14	41,5064	179		56,9950
58		20,0012	119		41,5204	180		57,0090
59		20,0152	120		41,5344	181	21	63,0537
60		20,0292	121		41,5485	182		63,0677
61		20,0432	122		41,5620	183		63,0810
62	8	23,0093	123		41,5766	184		63,0950
63		23,0233	124		41,5900	185		63,1090
64		23,0373	125		41,6040	186		63,1230
65		23,0513	126		41,6180	187		63,1370
66		23,0653	127	15	44,5846	188		63,1510
67		23,0794	128		44,5985	189		63,1650
68		23,0934	129		44,6126	190	22	66,1319
69		23,1074	130		44,6260	191		66,1450
70		23,1214	131		44,6400	192		66,1590
71	9	26,0875	132		44,6540	193		66,1730
72		26,1015	133		44,6680	194		66,1870
73		26,1155	134		44,6820	195		66,2010
74		26,1295	135		44,6950	196		66,2150
75		26,1435	136	16	47,6628	197		66,2290
76		26,1575	137		47,6768	198		66,2430
77		26,1715	138		47,6908	199	23	69,2100
78		26,1855	139		47,7010	200		69,2240

For helical gears (see Fig. 7.4, a) ***W '*** is used according to the same table as for spur gears, but the ***W'*** is calculated by the conditional number of teeth $z_y = z \cdot k$ (***k*** values are presented in Tab. 7.15).

Table 7.15

β	k	β	k	β	k	β	k
8°	1,0288	11°40′	1,0613	15°20′	1,1088	19°	1,1730
8°20′	1,0309	12°	1,0652	15°40′	1,1139	19°20′	1,1797
8°40′	1,0333	12°20′	1,0688	16°	1,1192	19°40′	1,1866
9°	1,0359	12°40′	1,0728	16°20′	1,1244	20°	1,1936
9°20′	1,0388	13°	1,0768	16°40′	1,1300	20°20′	1,2010
9°40′	1,0415	13°20′	1,0810	17°	1,1358	20°40′	1,2084
10°	1,0446	13°40′	1,0853	17°20′	1,1415	21°	1,2160
10°20′	1,0477	14°	1,0896	17°40′	1,1475	21°20′	1,2239
10°40′	1,0508	14°20′	1,0943	18°	1,1536	21°40′	1,2319
11°	1,0543	14°40′	1,0991	18°20′	1,1598	22°	1,2401
11°20′	1,0577	15°	1,1039	18°40′	1,1665	22°20′	1,2485

As a rule, the conditional number of teeth z_y is not an integer. Therefore, a correction W_u is introduced. It depends on the fractional part $(z_y - z_y ')$:

$$W_u = 0{,}0149\ (z_y - z_y '),$$

where $z_y '$ is the integer part of the conditional number of teeth. Then the base tangent length at α = 20 ° is $W = m\ (W ' + W_u + 0{,}684\ x)$.

7.5.4. TOP for Bevel Gears

In the drawings of all types of bevel gears TOP is disposed similar to the spur wheels. The number of rows in each section depends on the type of teeth (Tab. 7.16). Dimensions of rows and columns are the same as in spur wheel TOP (see Fig. 7.3).

Table 7.16

Middle normal module	–	–	m_n
Outer normal module	–	m_{ne}	–
Outer transverse module	m_e	–	–
The number of teeth	z	z	z
Tooth type	Straight	Tangential	Spiral
Tooth axial type	–		
Helix angle	–	β_{ne}	β_n
Tooth direction	–	Right (Left)	Right (Left)
Middle basic rack	GOST 13754-81	GOST 13754-81	GOST 16202-81
Shift factor	x_e	x_{ne}	x_n
Tooth thickness factor	x_τ	x_τ	x_τ

End of Tab. 7.16

Pitch angle	δ	δ	δ
Degree of accuracy			
Chordal thickness	$\overline{S}_{ce}$	$\overline{S}_{ce}$	$\overline{S}_{c}$
Chordal addendum	$\overline{h}_{c}$	$\overline{h}_{c}$	$\overline{h}_{c}$
Interaxial angle	Σ	Σ	Σ
Middle transverse module	m_m	–	–
Middle normal module	–	m_n	–
Outer transverse module	–	–	m_{te}
Cone distance	R_e	R_e	R_e
Middle cone distance	R_m	R_m	R_m
Mean pitch diameter	d_m	d_m	d_m

Notes:
1. Do not include to the TOP items marked as «–».
2. Indicate shift factors (for straight teeth gears it is the external circular x_e, for wheels with tangential teeth it is the outside normal x_{ne}, with spiral – the middle normal x_n) and the tooth thickness factor with the appropriate sign. In the case of absence it has to be equal «0».

7.5.5. A Bevel Gear Chordal Thickness and Chordal Addendum Calculation

To check a bevel straight tooth profile accuracy a chordal thickness $\overline{S}_{ce}$ and chordal addendum $\overline{h}_{ce}$ are usually obtained (Fig. 7.6). When $\alpha = 20^{\circ}$ and a shift coefficient $x < 0{,}4$ the chordal thickness equals $\overline{S}_{ce} = 0{,}883\ S_{e(1,2)}$, the height

$$\overline{h}_{ce(1,2)} = h_{ae(1,2)} - 0{,}1607\ S_{e(1,2)},$$

where $S_{e(1,2)}$ is the circular thickness: for a pinion

$$S_{e1} = (0{,}5\pi + 2x_1 \tan\alpha)m_e = (1{,}571 + 0{,}728x_1)m_e,$$

for a gear $S_{e2} = \pi m_e - S_{e1}$; h_{ae1} and h_{ae2} are a pinion and a gear addendum:

$$h_{ae1} = m_e\left(h_a^* + x_1\right),\ h_{ae2} = m_e\left(h_a^* - x_1\right);$$

$h_a^* = 1{,}0$ is the tooth addendum factor. When $u < 2{,}5$, $x_{\tau 1}$ equals zero; for transmission with $u > 2{,}5$ values are given in handbooks.

For the bevel gears with spiral teeth the constant chord and addendum height are obtained in the middle cross-section of the gear face width.

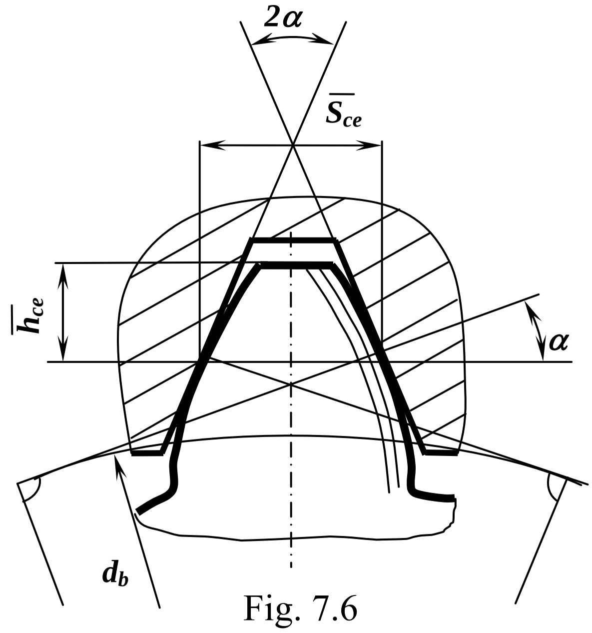


Fig. 7.6

The constant chord

$$\overline{S}_{c_{(1,2)}} = 0{,}883\,S_{n_{(1,2)}},$$

where the normal tooth thickness in the middle section

$$S_{n1} = (0{,}5\pi + 2x_{n1}\tan\alpha_n + x_{\tau 1})m_n,$$

$$S_{n_2} = \pi m_n - S_{n_2}.$$

Height to a constant chord

$$\overline{h}_{c_{(1,2)}} = h_{a_{(1,2)}} - 0{,}1607\,S_{n_{(1,2)}}.$$

Addendum height in the middle section may be found as follows: for pinion – $h_{am_1} = m_n\left(h_a^* + x_{n_1}\right)$, for gear – $h_{am_1} = 2h_a^* m_n - h_{am_1}$.

7.6. Technical Requirements

Technical requirements are placed above the main stamp in the form of a column as wide as the main stamp. The text has to be written from the top down. Points should be numbered consecutively. Each point begins with a red line. Do not write the title "Technical Requirements".

Technical requirements are combined into homogeneous characteristics and placed in the following sequence:

1. Requirements for material, heat treatment, etc. Example: "Hardness HB 220...240", "Teeth cementation depth h 1,0...1,3, case-hardening to HRC 56...60."

2. Dimensions, tolerances in size, shape, arrangement of surfaces and other information are not shown in the drawing. Example: «Unspecified tolerated dimensions are: holes – H14, shafts – h14, others – ± IT14 / 2».

3. Surface quality requirements, coating and treatment.

4. Other requirements: about quietness; balancing, etc.

Information about reference dimension should be placed above technical requirements as follows: «*Reference dimensions».

7.7. Examples

Spur gears

In accordance with GOST 2.403-75 dimensions as follows must be specified in the spur gear detail drawing (Fig. 7.7):

− the outer (addendum) diameter d_a (diameter tolerances see in Tab. 7.17, the allowable radial runout see in Tab. 7.6);

− the face width of the gear $\boldsymbol{b}$ (tolerances are according to h10…h12; the allowable face runout see in Tab. 7.7);

− the hub bore diameter $\boldsymbol{D_b}$ (tolerance is according to H7, cylindricity tolerance equals 0,3 diameter tolerance);

− the length of the hub $\boldsymbol{L_H}$ (if it is different from the face width $\boldsymbol{b}$, the hub length is the free size; allowable runout is shown in Tab. 7.8);

− the keyway dimensions:

− the depth equals $\boldsymbol{D_b} + \boldsymbol{t_1}$ ($\boldsymbol{t_1}$ size see in Appendix D);

− the width tolerance is JS9 (size 16JS9 in Fig. 7.7);

− the allowable misalignment of the keyway faces relative to the its axis, it equals 0,5 IT9 (IT9 for the size of 16 mm is 43 microns, so 0,5 IT9 = 0,5 • 43 ≈ 22 μm);

− the allowable asymmetry, it equals 2 IT9 (in case been shown 2IT9 = 2 • 43 = = 86 microns).

In the drawing of the pinion-shaft, the allowable radial and face runout for bearings seats should be indicated (see Tab. 7.11).

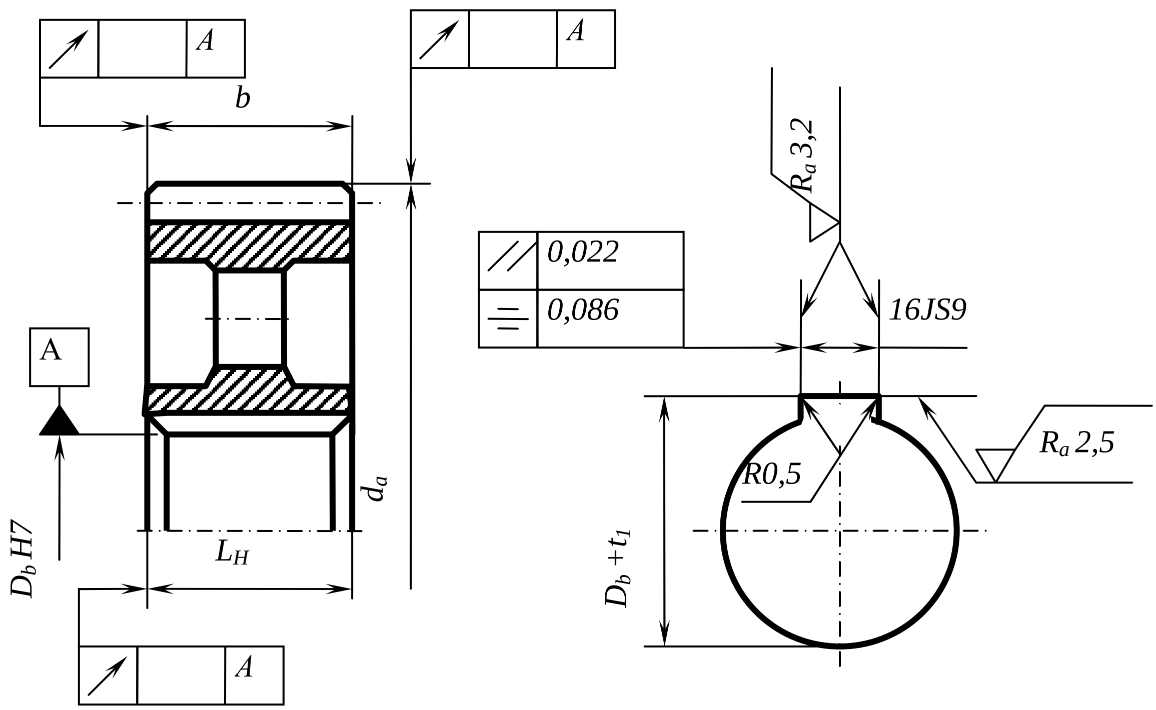


Fig. 7.7

Table 7.17

Type of coupling	Degree of accuracy			
	6	7	8	9
C	h6	h7	h8	h9
B, A	h7	h8	h9	h12

Examples of spur gear working drawings see in Fig. 7.8.

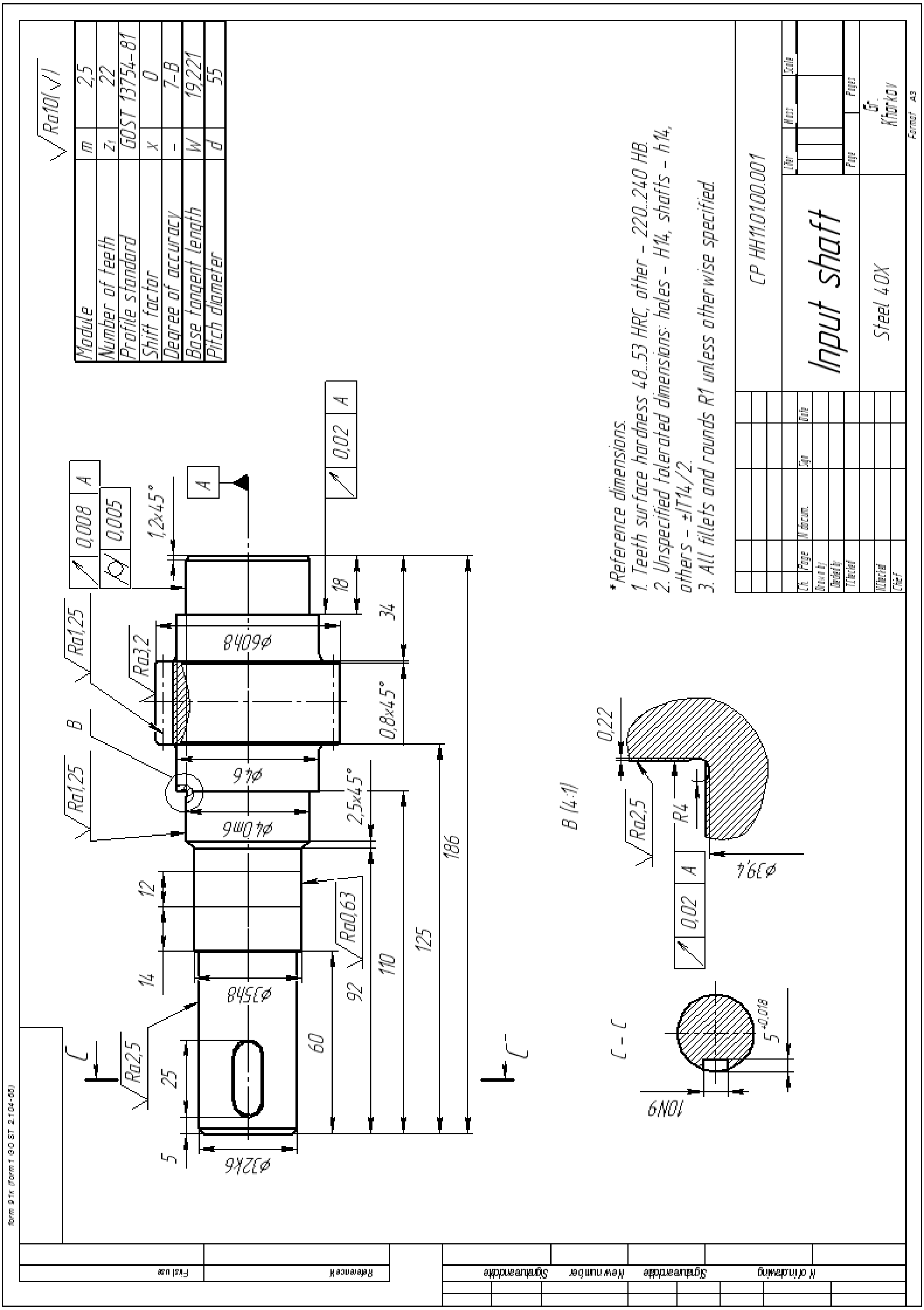

Ra10(√)
Module m 2,5
Number of teeth z1 22
Profile standard GOST 13754-81
Shift factor x 0
Degree of accuracy - 7-B
Base tangent length W 19,221
Pitch diameter d 55
*Reference dimensions.
1. Teeth surface hardness 48...53 HRC, other – 220...240 HB.
2. Unspecified tolerated dimensions: holes – H14, shafts – h14, others – ±IT14/2.
3. All fillets and rounds R1 unless otherwise specified.
CP HH11.01.00.001
Input shaft
Steel 40X
Gr. Kharkov
B (4:1)
C – C
ø60h8
ø46
ø40m6
ø35h8
ø32k6
ø39,4
10N9
1,2x45°
0,8x45°
2,5x45°
186
125
110
92
60

Fig. 7.8

Bevel gears

Most of the dimensions and tolerances of bevel gears is determined by tables provided for spur gears. Standard dimensions shown in Fig. 7.9, a are:

1. Addendum diameter at the main outer cone d_{ae} and d_{ae}'. In case of spur bevel wheels the outer diameter of the teeth is used as the meter base. For 7th degree of accuracy and B type of coupling d_{ae} tolerance is h6.

Diameter d_{ae}' is free size, it is obtained by blunting the sharp edge (chamfer size is 0,5m). The diameter d_{ae}' is machined after the teeth cutting and checking.

In case of wheels with spiral and tangential teeth they are measured in the measuring mean cross-section normal to the face width direction (see Fig. 7.9, b). The accuracy of the outer diameter can be reduced in this case, it is prescribed by h12.

2. Axial dimensions are measured from the basic face. Dimensions C and L are reference, size A is free.

3. Additional external cone angle (90 - δ °), this angle tolerance is ± 15′... ± 30′.

4. Pitch cone angle tolerance is ± 5′... ± 10′, allowable runout for addendum cone, μm, − according to Tab. 7.9.

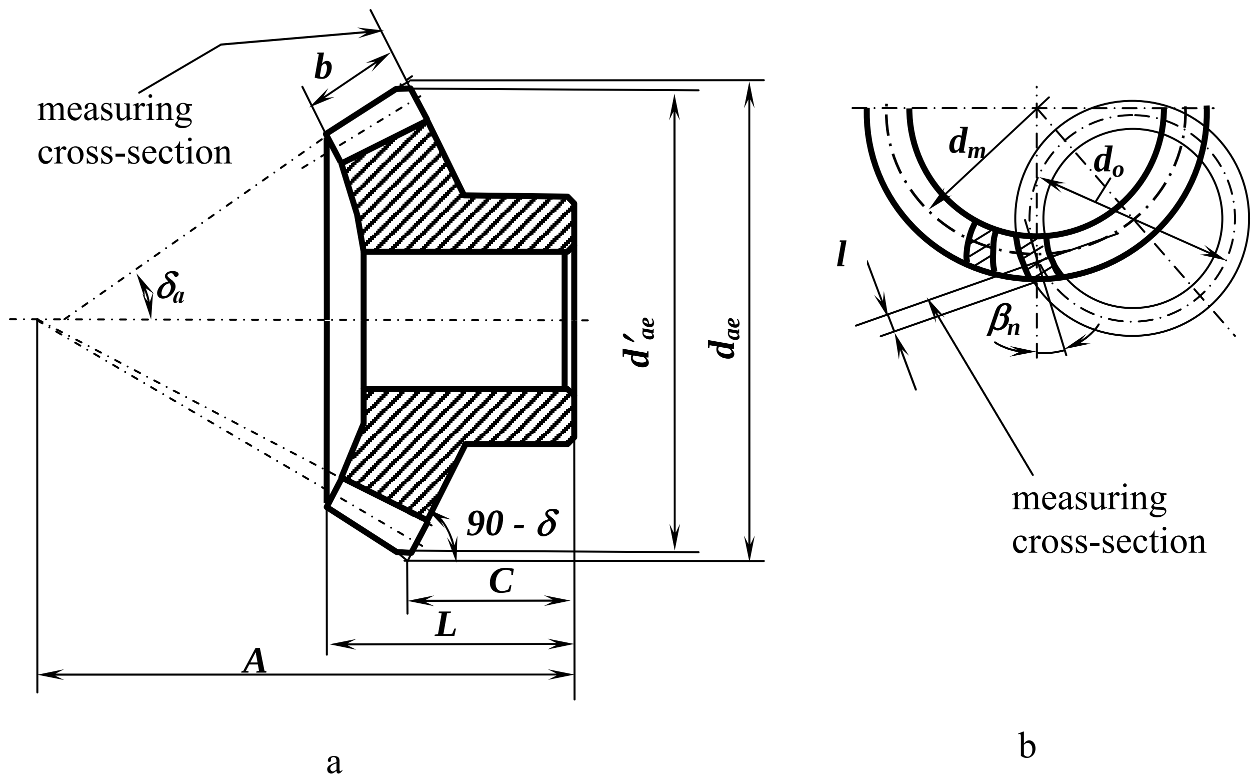
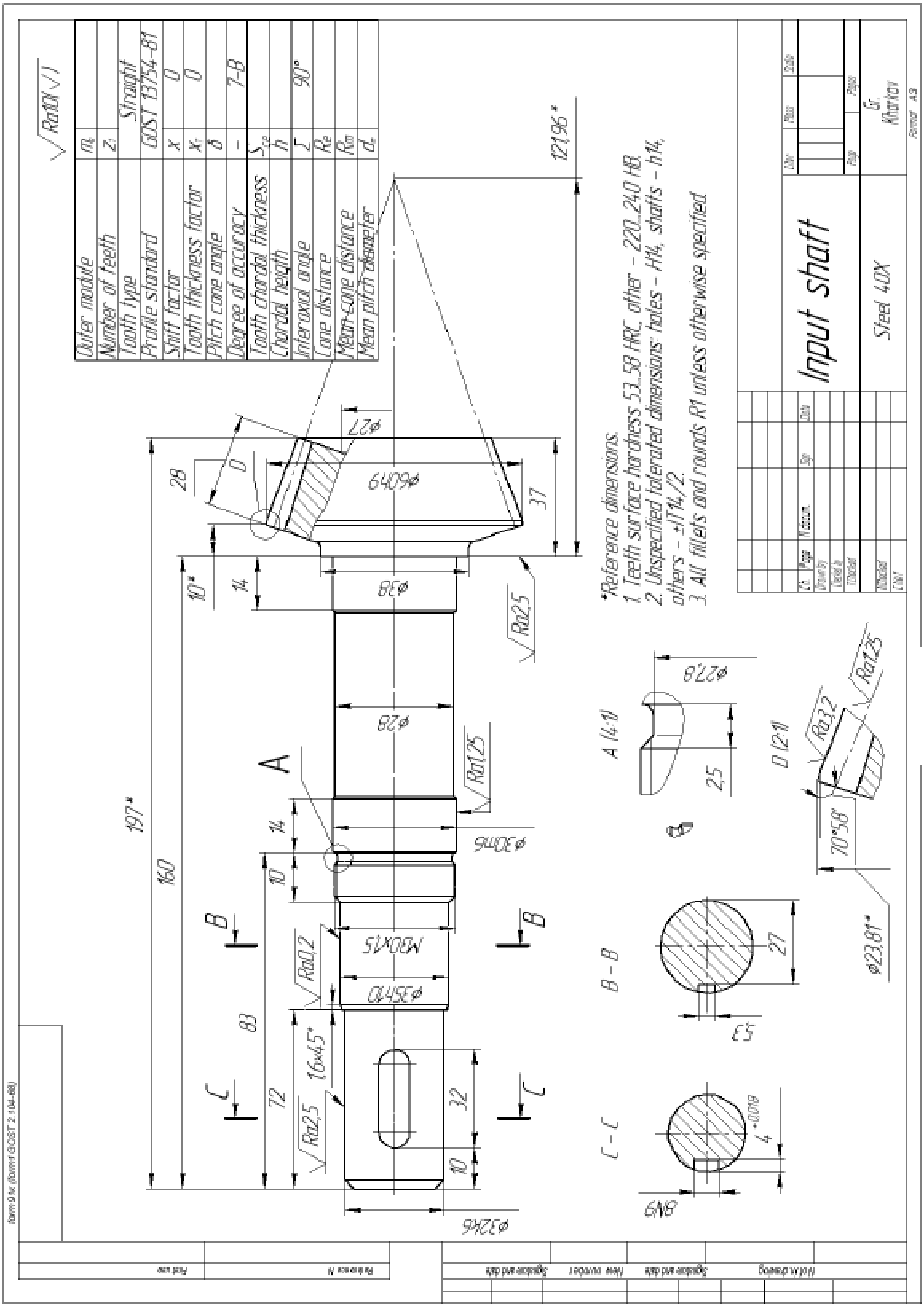


Fig. 7.9

Measuring cross-section to control the accuracy of the teeth in the spur bevel wheels is placed tangentially to the outer additional cone (see Fig. 7.9, a).

In wheels with circular and tangential teeth, metering section passes through the middle of the face width: $l = b / 2$.

Examples of straight teeth bevel pinion shaft working drawings see in Fig. 7.10.



Ra10 (√)
Outer module | m_e
Number of teeth | z_1
Tooth type | Straight
Profile standard | GOST 13754-81
Shift factor | x | 0
Tooth thickness factor | x_τ | 0
Pitch cone angle | δ
Degree of accuracy | – | 7-B
Tooth chordal thickness | S_{ce}
Chordal heigth | h
Interaxial angle | Σ | 90°
Cone distance | R_e
Mean-cone distance | R_m
Mean pitch diameter | d_m
*Reference dimensions.
1. Teeth surface hardness 53...58 HRC, other – 220...240 HB.
2. Unspecified tolerated dimensions: holes – H14, shafts – h14, others – ±IT14/2.
3. All fillets and rounds R1 unless otherwise specified.
Input shaft
Steel 40X
Gr. Kharkov
A (4:1)
D (2:1)
B – B
C – C

Fig. 7.10

Appendix A

ASYNCHRONOUS ENGINES

Table A.1

Engine code and nominal rotational speed n_{nom}, rpm, according to its synchronous rotational speed

Power, kW	Synchronous rotational speed, rpm							
	3000		1500		1000		750	
	Code	n_{nom}	Code	n_{nom}	Code	n_{nom}	Code	n_{nom}
0,75	4A71A2	2840	4A71B4	1390	4A80A6	915	4A80B8	700
1,1	4A71B2	2810	4A80A4	1420	4A80B6	920	4A90L8	700
1,5	4A80A2	2850	4A80B4	1415	4A90L6	935	4A100S8	700
2,2	4A80B2	2850	4A90L4	1425	4A100S6	950	4A100L8	700
3	4A90L2	2840	4A100S4	1435	4A100L6	955	4A112M8	700
4	4A100S2	2880	4A100L4	1430	4A112M6	950	4A132S8	720
5,5	4A100L2	2880	4A112M4	1445	4A132S6	965	4A132M8	720
7,5	4A112M2	2900	4A132S4	1455	4A132M6	870	4A160S8	730

The main dimensions, mm

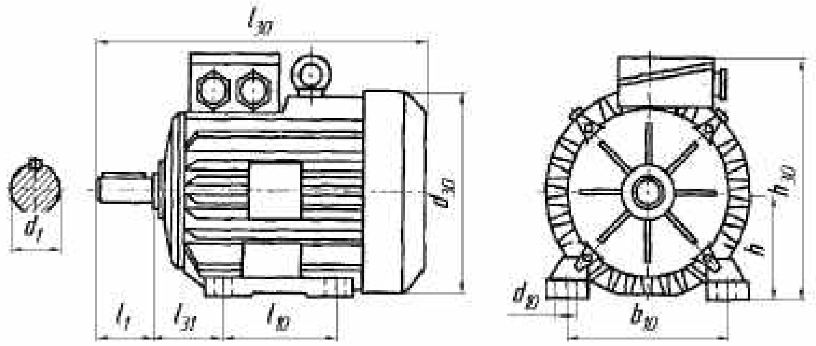
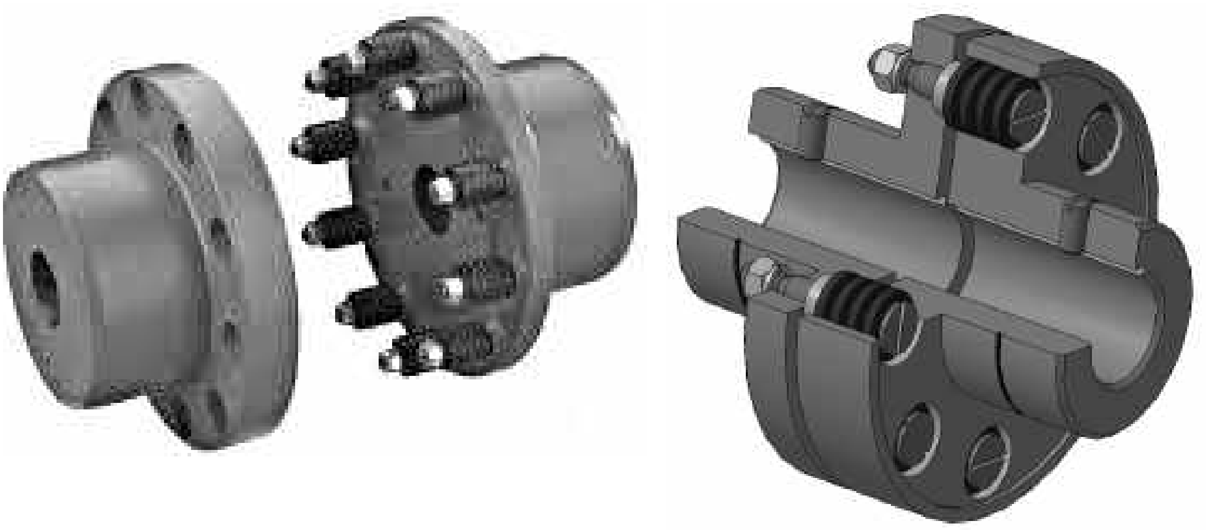


Table A.2

Type	*h*	d_{30}	l_1	l_{30}	d_1	l_{10}	l_{31}	d_{10}	b_{10}	h_{30}
71A,B	71	170	40	285	19	90	45	7	112	201
80A	80	186	50	300	22	100	50	10	125	218
80B	80	186	50	320	22	100	50	10	125	218
90L	90	208	50	350	24	125	56	10	140	243
100S	100	235	60	362	28	112	63	12	160	263
100L	100	235	60	392	28	140	63	12	160	263
112M	112	260	80	452	32	140	70	12	190	310
132S	132	302	80	480	38	140	89	12	216	350
132M	132	302	80	530	38	178	89	12	216	350
160S	160	358	110	624	42	178	108	15	216	370

Appendix B

FLEXIBLE COUPLINGS



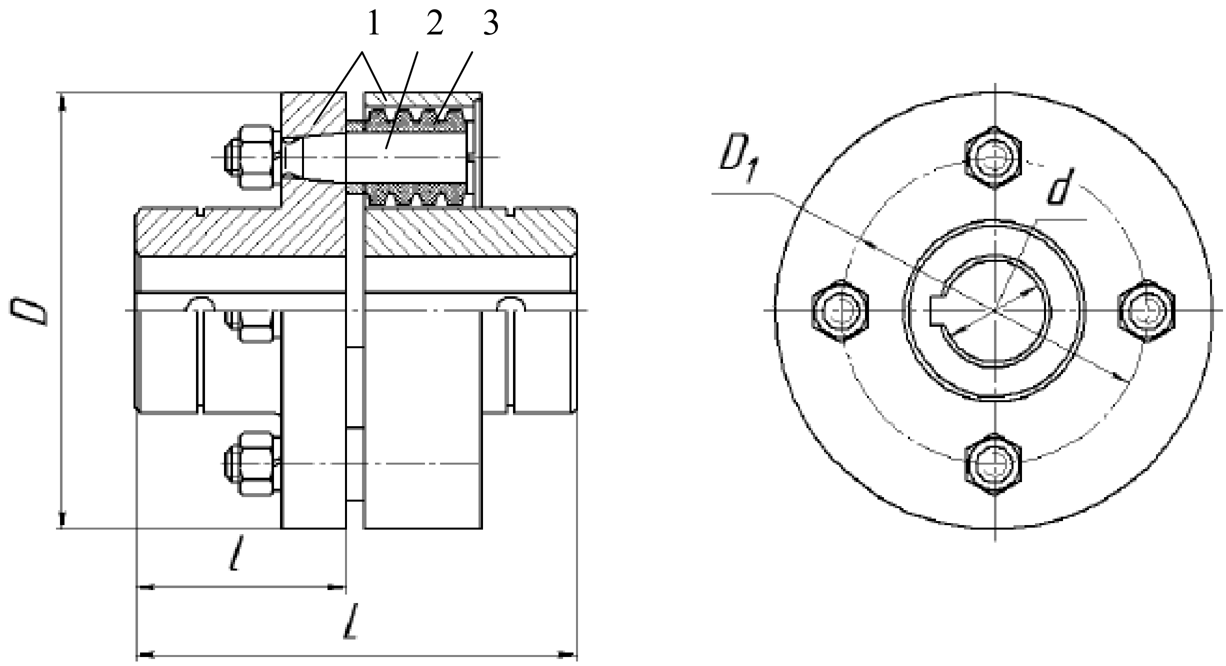


Fig. B.1. Flexible coupling with rubber-bushed studs (GOST 21424-75*):
1 – hubs (half-coupling); 2 – pin; 3 – rubber bush

Code: name of coupling – nominal torque – first diameter– second diameter – GOST
Example: Flexible coupling 125 – 25 – 30 GOST 21424-75*

Table B.1

Parameters of the Flexible Coupling with Rubber-Bushed Studs

Nominal torque T, Nm	*d*	*D*	*L* Long end	*L* Short end	*l* Long end	*l* Short end	D_1	The number of studs	RPM_{max}	Max shafts' misalignment: radial Δ_r, mm	Max shafts' misalignment: angular Δ_α, deg
16	12	75	63	53	30	25	50	4	7 600	0,2	1,5
	14	75	63	53	30	25	50	4	7 600		
	16	75	83	59	40	28	50	4	7 600		
31,5	16	90	84	60	40	28	63	4	6 350		
	18	90	84	60	40	28	63	4	6 350		
	19	90	84	60	40	28	63	4	6 350		
63	20	100	104	76	50	36	71	4	5 700		
	22	100	104	76	50	36	71	4	5 700		
	24	100	125	88	50	36	71	4	5 700		
125	25	120	125	89	60	42	90	4	4 600	0,3	1
	28	120	125	89	60	42	90	4	4 600		
	30	120	165	121	80	58	90	4	4 600		
250	32	140	165	121	80	58	105	6	3 800		
	35	140	165	121	80	58	105	6	3 800		
	36	140	165	121	80	58	105	6	3 800		
	38	140	165	121	80	58	105	6	3 800		
	40	140	225	169	110	82	105	6	3 800		
	42	140	225	169	110	82	105	6	3 800		
	45	140	225	169	110	82	105	6	3 800		
500	40	170	225	169	110	82	130	8	3 600		
	42	170	225	169	110	82	130	8	3 600		
	45	170	225	169	110	82	130	8	3 600		
1000	50	220	226	170	110	82	160	10	2 860	0,4	
	55	220	226	170	110	82	160	10	2 860		
	56	220	226	170	110	82	160	10	2 860		
	60	220	286	216	140	105	160	10	2 860		
	63	220	286	216	140	105	160	10	2 860		
	65	220	286	216	140	105	160	10	2 860		
	70	220	286	216	140	105	160	10	2 860		

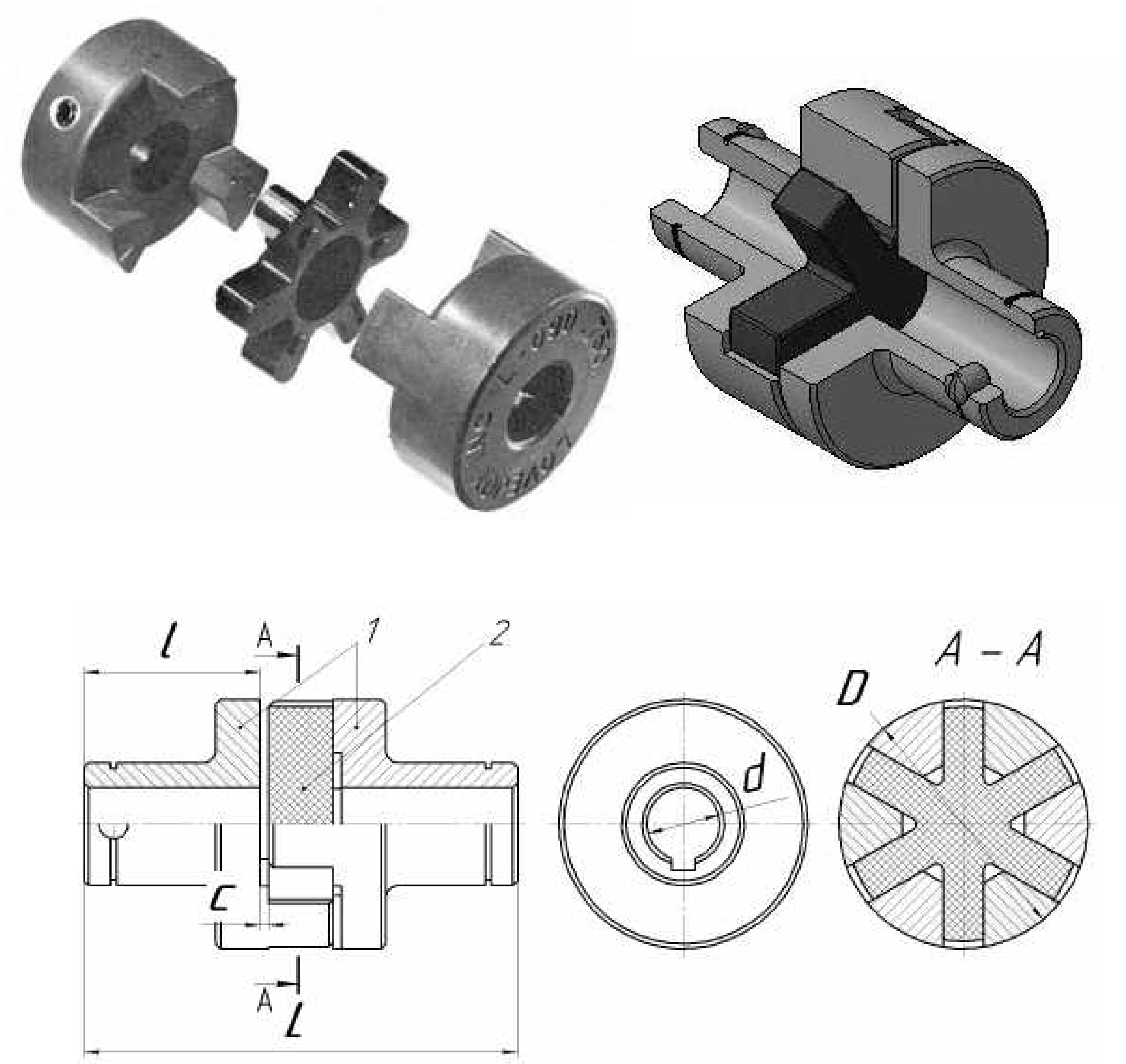


Fig. B.2. Jaw compression coupling with flexible spider insert (GOST 14084-76*):
1 – hubs (half-coupling); 2 – flexible spider insert

Code:
name of coupling – nominal torque – type – first diameter– second diameter – GOST

Example: Jaw compression coupling 125 – I – 25 – 36 GOST 14084-76*

Table B.2

Parameters of the Jaw Compression Couplings with Flexible Spider Insert

<table>
<tr><th rowspan="3">Nominal torque T, Nm</th><th rowspan="3">d</th><th rowspan="3">D</th><th colspan="2">L</th><th colspan="2">ℓ</th><th rowspan="3">c, mm</th><th rowspan="3">Maximum rotational speed, rpm</th><th colspan="2">Maximum shaft axes misalignment</th></tr>
<tr><th colspan="4">Type</th><th rowspan="2">radial Δ_r, mm</th><th rowspan="2">angular Δ_α, deg</th></tr>
<tr><th>I</th><th>II</th><th>I</th><th>II</th></tr>
<tr><td rowspan="3">25,0</td><td>14</td><td rowspan="3">63</td><td>81</td><td>71</td><td>30</td><td>25</td><td rowspan="12">3,0</td><td rowspan="3">3 500</td><td rowspan="5">0,2</td><td rowspan="5">1,5</td></tr>
<tr><td>16</td><td>101</td><td>77</td><td>40</td><td>28</td></tr>
<tr><td>18
(19)
20</td><td>121</td><td>93</td><td>50</td><td>36</td></tr>
<tr><td rowspan="2">31,5</td><td>16</td><td rowspan="2">71</td><td>101</td><td>77</td><td>40</td><td>28</td><td rowspan="2">3 000</td></tr>
<tr><td>18
(19)
20
22</td><td>121</td><td>93</td><td>50</td><td>36</td></tr>
<tr><td rowspan="2">63,0</td><td>22
25</td><td rowspan="2">85</td><td>128</td><td>100</td><td>50</td><td>36</td><td rowspan="2">2 250</td><td rowspan="4">0,3</td><td rowspan="7">1</td></tr>
<tr><td>28</td><td rowspan="2">148</td><td rowspan="2">112</td><td rowspan="2">60</td><td rowspan="2">42</td></tr>
<tr><td rowspan="2">125,0</td><td>25</td><td rowspan="2">105</td><td rowspan="2">2 000</td></tr>
<tr><td>28
30
32
35
36</td><td>188</td><td>144</td><td rowspan="2">80</td><td rowspan="2">58</td></tr>
<tr><td rowspan="3">250,0</td><td>32
35</td><td rowspan="3">135</td><td>191</td><td>147</td><td rowspan="3">1 500</td><td rowspan="3">0,4</td></tr>
<tr><td>36</td><td>251</td><td>195</td><td rowspan="2">110</td><td rowspan="2">82</td></tr>
<tr><td>(38)
40
(42)
45</td><td>256</td><td>200</td></tr>
</table>

Appendix C

COMPENSATING AND SAFETY COUPLINGS

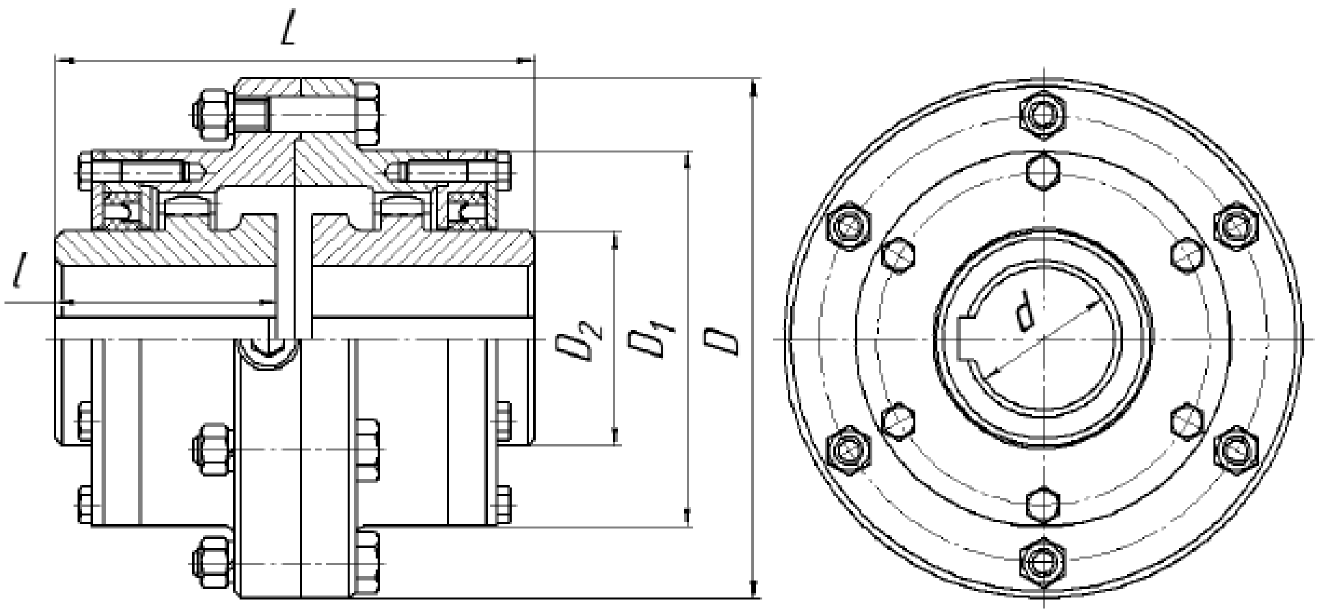


Fig. C.1. Compensating curved-tooth gear coupling (GOST 5006-94)

Table C.1

Parameters of the Compensating Curved-Tooth Gear Couplings

Nominal torque *T*, Nm	**d**	**D**	**D_1**	**D_2**	**l**	**L_{max}**	**RPM_{max}**	**Module, mm**	**The number of teeth**
1 000	40	145	105	60	82	174	5 400	2,5	30
1 600	55	170	125	80			4 800		38
2 500	60	185	135	85	105	220	4 500	3,0	36
4 000	65	200	150	95			3 720		40
6 300	80	230	175	115	130	270	3 300		48
10 000	100	270	200	145	165	340	2 820		56
16 000	120	300	230	175		345	2 400	4,0	48
25 000	140	330	260	200	200	415	2 100		56
40 000	160	410	330	230			1 740	6,0	46
63 000	200	470	390	290	240	500	1 200		56

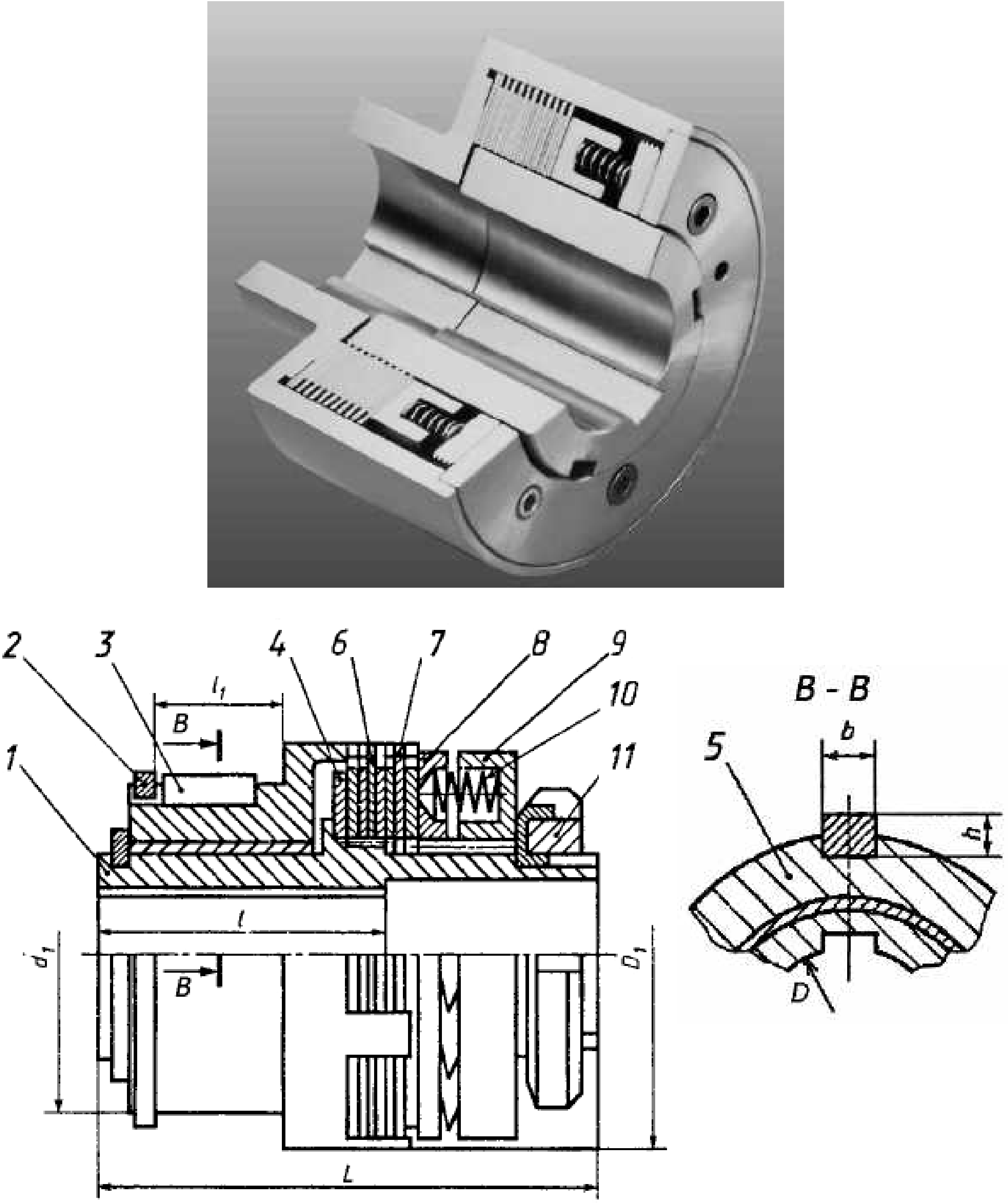


Fig. C.2. Safety multiple-disc slipping clutch (GOST 15622-96):
1 – driving hub; 2 – ring; 3 – press plate; 4 – spring frame; 5 – driven hub;
6 – driving plate; 7 – driven plate; 8 – spring; 9, 10 – nut and stopper;
11 – split ring; 12 – key

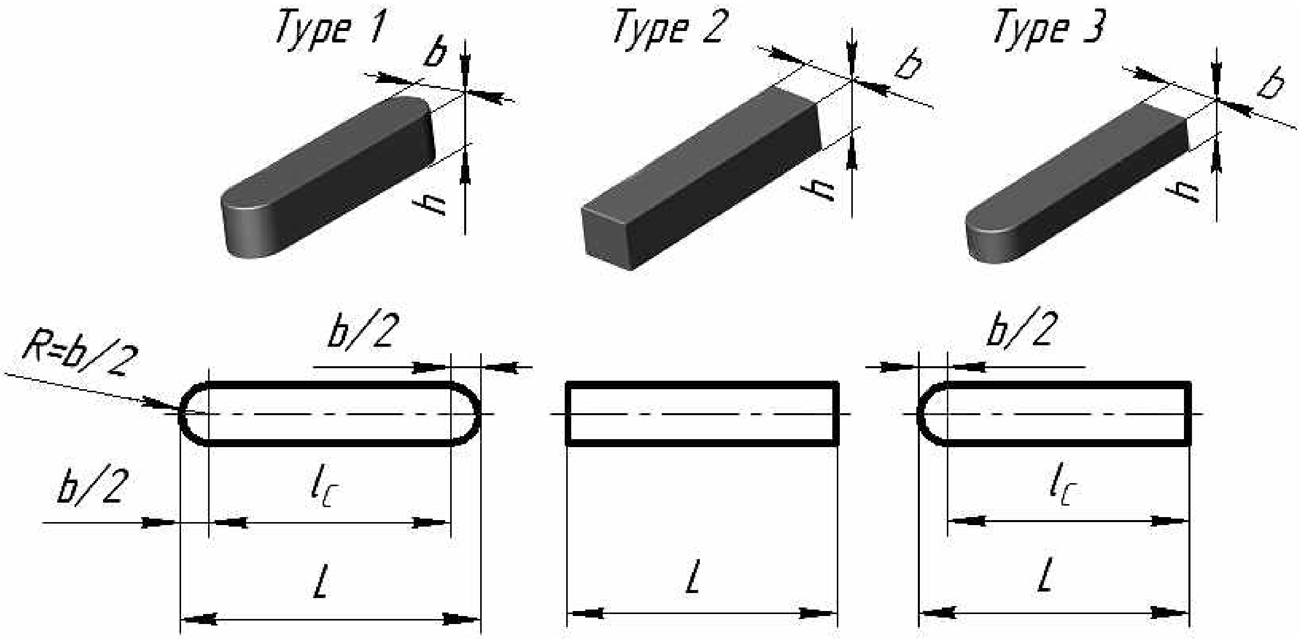
Table C.2

Parameters of the Safety Multiple-Disc Slipping Clutch

Torque T_p, Nm	**Driving shaft diam. D**	**Outer diam. D_1**	**Driven shaft diam. d_1**	**Overall length L**	**Driving end l**	**Driven end l_1**
40	18	70	45	95	40	24
	20				50	
	22					
63	20, 22, 24	85	55	120	50	28
	25				60	
100	24	95	65	125	50	32
	25, 28				60	
	30				80	
160	28	100	70	150	60	36
	30, 32				80	
250	36, 38	120		160	80	42
	40				110	
400	38	145	90	180	80	48
	40, 42, 45, 48				110	
630	45, 48, 50, 53, 55	155	95	240	110	56
1000	50, 53, 55, 56	170	120	270	110	67
	60, 63				140	

Appendix D

KEYS (GOST 23360-78)



Dimensions, mm

Shaft diameter		b	h	Key length		Keyway depth	
from	to			from	to	shaft	hub
6	8	2	2	6	20	1,2	1
8	10	3	3	6	36	1,8	1,4
10	12	4	4	8	45	2,5	1,8
12	17	5	5	10	56	3	2,3
17	22	6	6	14	70	3,5	2,8
22	30	8	7	18	90	4	3,3
30	38	10	8	22	110	5	3,3
38	44	12	8	28	140	5	3,3
44	50	14	9	36	160	5,5	3,8
50	58	16	10	45	180	6	4,3
58	65	18	11	50	200	7	4,4
65	75	20	12	56	220	7,5	4,9
75	85	22	14	63	250	9	5,4
85	95	25	14	70	280	9	5,4
95	110	28	16	80	320	10	6,4
110	130	32	18	90	360	11	7,4

Standard length: 6, 8, 10, 12, 14, 16, 18, 20, 22, 25, 28, 32, 36, 40, 45, 50, 56, 63, 70, 80, 90, 100, 110, 125, 140, 160, etc.

Appendix E

BEARINGS

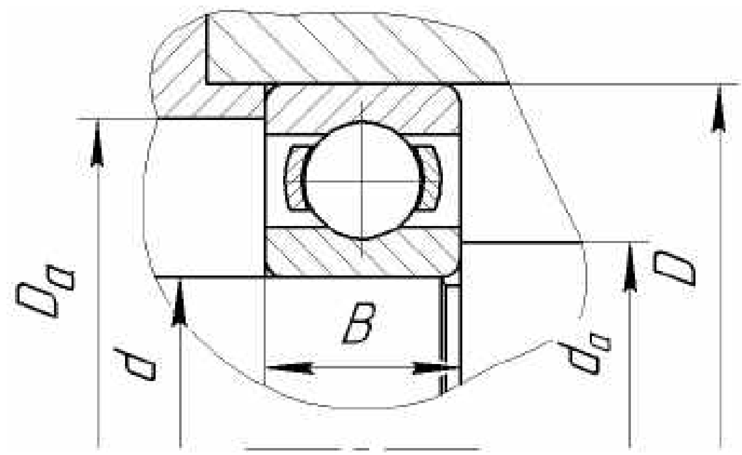


Fig. E.1. A single row deep groove ball bearing (GOST 8338-75)

Table E.1

Parameters of the Single Row Deep Groove Ball Bearings

d	Code	D	B	r	Load capacity, N		d_a, min	D_a, max
					dynamic C	static C_0		
10	1000900	22	6	0,5	3 340	1 350	12	20
	100	26	8	0,5	4 620	1 960	12	24
	200	30	9	1	5 900	2 650	14	26
	300	35	11	1	8 060	3 750	14	31
12	1000801	21	5	0,5	1 430	700	14	19
	1000901	24	6	0,5	3 390	1 350	14	22
	101	28	8	0,5	5 070	2 240	14	26
	7000101	28	7	0,5	5 070	2 240	14	26
	201	32	10	1	6 890	3 100	16	28
	301	37	12	1,5	9 750	4 650	17	31
15	1000802	24	5	0,5	1 560	830	17	22
	1000902	28	7	0,5	3 480	1 480	17	26
	7000102	32	8	0,5	5 590	2 500	17	30
	202	35	11	1	7 800	3 550	19	31
	302	42	13	1,5	11 400	5 400	20	36
17	1000903	30	7	0,5	3 640	1 650	19	28
	7000103	35	8	0,5	6 050	2 800	19	33
	203	40	12	1	9 560	4 500	21	36
	303	47	14	1,5	13 500	6 650	22	41
	403	62	17	2	22 900	11 800	24	53

Table E.1 (continued)

d	Code	D	B	r	Load capacity, N		Minimum d_a	Maximum D_a
					dynamic C	static C_0		
20	1000904	37	9	0,5	6 550	3 040	22	35
	104	42	12	1	9 360	4 500	24	39
	204	47	14	1,5	12 700	6 200	25	42
	304	52	15	2	15 900	7 800	26	45
	404	72	19	2	30 700	16 600	28	48
25	1000805	37	7	1	3 120	1 980	27	35
	1000905	42	9	0,5	7 320	3 680	27	40
	7000105	47	8	0,5	7 610	4 000	29	43
	105	47	12	1	11 200	5 600	29	44
	205	52	15	1,5	14 000	6 950	30	47
	305	62	17	2	22 500	11 400	31	55
	405	80	21	2,5	36 400	20 400	33	70
30	1000906	47	9	0,5	7 590	4 000	32	45
	7000106	55	9	0,5	11 200	5 850	32	53
	106	55	13	1,5	13 300	6 800	35	50
	206	62	16	1,5	19 500	10 000	35	57
	306	72	19	2	28 100	14 600	36	65
	406	90	23	2,5	47 000	26 700	38	80
35	1000807	47	7	1	4 030	3 000	37	45
	1000907	55	10	1	10 400	5 650	39	51
	7000107	62	9	0,5	12 400	6 950	37	60
	107	62	14	1,5	15 900	8 500	40	57
	207	72	17	2	25 500	13 700	42	65
	307	80	21	2,5	33 200	18 000	43	71
	407	100	25	2,5	55 300	31 000	43	90
40	1000908	62	12	1	12 200	6 920	44	58
	7000108	68	9	0,5	13 300	7 800	42	66
	108	68	15	1,5	16 800	9 300	45	63
	208	80	18	2	32 000	17 800	46	73
	308	90	23	2,5	41 000	22 400	48	81
	408	110	27	3	63 700	36 500	49	97
45	1000909	68	12	1	14 300	8 130	49	64
	7000109	75	10	1	15 600	9 300	49	71
	109	75	16	1,5	21 200	12 200	50	70
	209	85	19	2	33 200	18 600	52	78
	309	100	25	2,5	52 700	30 000	53	91
	409	120	29	3	76 100	45 500	54	107

End of Tab. E.1

d	Code	D	B	r	Load capacity, N		Minimum d_a	Maximum D_a
					dynamic C	static C_0		
50	7000110	80	10	1	16 300	10 000	54	76
	110	80	16	1,5	21 600	13 200	55	75
	210	90	20	2	35 100	19 800	57	83
	310	110	27	3	61 800	36 000	60	99
	410	130	31	3,5	87 100	52 000	63	116
55	1000911	80	13	1,5	16 000	10 000	60	75
	7000111	90	11	1	17 000	11 700	59	86
	111	90	18	2	28 100	17 000	62	84
	211	100	21	2,5	43 600	25 000	64	91
	311	120	29	3	71 500	41 500	64	111
	411	140	33	3,5	100 000	63 000	68	126
60	1000912	85	13	1,5	16 400	10 600	65	80
	7000112	95	11	1	18 600	12 400	64	91
	112	95	18	2	29 600	18 300	68	88
	212	110	22	2,5	52 000	31 000	68	101
	312	130	31	3,5	81 900	48 000	71	118
	412	150	35	3,5	108 000	70 000	73	146
65	1000913	90	13	2	17 400	11 900	70	85
	7000113	100	11	1	19 000	13 100	69	96
	113	100	18	2	30 700	19 600	72	93
	213	120	23	2,5	56 000	34 000	73	111
	313	140	33	3,5	92 300	56 000	76	128
	413	160	37	3,5	119 000	78 000	78	164
70	7000114	110	13	1	22 200	15 300	74	106
	114	110	20	2	37 700	24 500	77	103
	214	125	24	2,5	61 800	37 500	78	116
	314	150	35	3,5	104 000	63 000	81	138
	414	180	42	4	143 000	105 000	93	157
75	1000915	105	16	1,5	24 300	16 800	80	100
	115	115	20	2	39 700	26 000	82	108
	215	130	25	2,5	66 300	41 000	83	121
	315	160	37	3,5	112 000	72 500	86	148
	415	190	45	4	153 000	114 000	98	166
80	116	125	22	2	47 700	31 500	87	118
	216	140	26	3	70 200	45 000	90	129
	316	170	39	3,5	124 000	80 000	91	158
	416	200	48	4	163 000	125 000	105	176

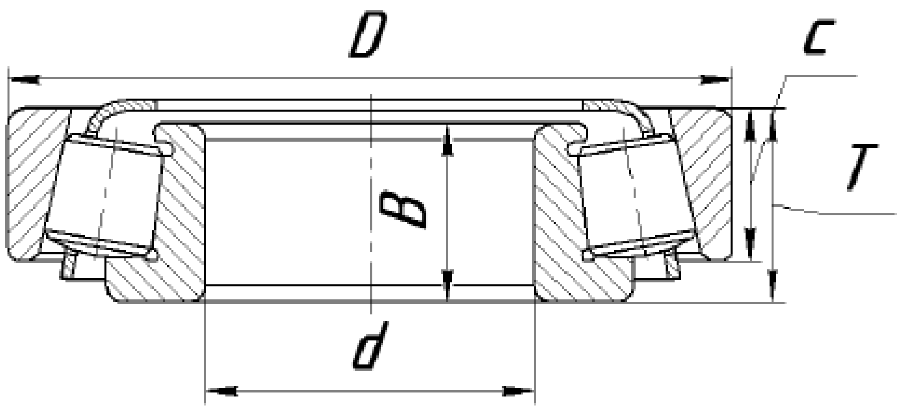


Fig. E.2. A tapered roller bearing (GOST 27365-87)

Table E.2

Parameters of the Tapered Roller Bearings

d	Code	D	B	T	c	Load capacity, kN		r	e	Y	Y_0	n_{max}, rpm	
						C	C_0						
17	7203	40	12	13,3	11	14,0	9,0	2	0,3	2	1,1	9 000	13 000
	7203A	40	12	13,3	11	17,9	12,0	2	0,4	1,7	0,9	9 000	13 000
20	7204	47	14	15,3	12	21,0	13,0	2	0,4	1,7	0,9	8 000	11 000
	7204A	47	14	15,3	12	26,0	16,6	2	0,4	1,7	0,9	8 000	11 000
	7304	52	16	16,3	13	26,0	17,0	2	0,3	2	1,1	8 000	11 000
	7304A	52	15	16,3	13	31,9	20,0	2	0,3	2	1,1	8 000	11 000
25	7205	52	15	16,3	13	24,0	17,5	2	0,4	1,7	0,9	7 500	10 000
	7205A	62	15	16,3	13	29,2	21,0	2	0,4	1,6	0,9	7 500	10 000
	7305	62	17	18,3	15	33,0	23,2	2	0,4	1,7	0,9	6 700	9 000
	7305A	62	17	18,3	15	41,8	28,0	2	0,3	2	1,1	6 000	9 000
30	7206	62	16	17,3	14	31,6	22,0	2	0,4	1,6	0,9	6 300	8 500
	7206A	62	16	17,3	14	38,0	25,5	2	0,4	1,6	0,9	6 300	8 500
	7506	62	21	21,3	17	36,0	27,0	2	0,4	1,6	0,9	6 300	8 500
	7506A	62	20	21,3	17	47,3	37,0	2	0,4	1,6	0,9	6 300	8 500
	7306	72	19	20,8	17	43,0	29,5	2	0,3	1,8	1	5 600	7 500
	7306A	72	19	20,8	16	52,8	39,0	2	0,3	1,9	1,1	5 600	7 500
35	7207	72	17	18,3	15	38,5	26,0	2	0,4	1,6	0,9	5 300	7 000
	7207A	72	17	18,3	15	48,4	32,5	2	0,4	1,6	0,9	5 300	7 000
	7507	72	23	24,2	20	53,0	40,0	2	0,4	1,7	1	8 300	7 000
	7307	80	21	22,8	18	54,0	38,0	3	0,3	1,4	1	5 000	6 700
	7307A	80	21	22,8	18	68,2	50,0	3	0,3	1,9	1,1	5 000	6 700

Table E.2 (continued)

d	Code	D	B	T	c	Load capacity, kN		r	e	Y	Y_0	n_{max}, rpm	
						C	C_0						
40	7208	80	20	19,8	16	46,5	32,5	2	0,4	1,6	0,9	4 800	6 300
	7208A	80	18	19,8	16	58,3	40,0	2	0,4	1,6	0,9	4 800	6 300
	7508	80	24	24,8	20	56,0	44,0	2	0,4	1,6	0,9	4 800	6 300
	7308	90	23	25,3	20	66,0	47,5	3	0,3	2,2	1,2	4 500	6 000
45	7209	85	19	20,8	16	50,0	33,0	2	0,4	1,5	0,8	4 500	6 000
	7209A	85	19	20,8	16	62,7	50,0	2	0,4	1,5	0,8	4 500	6 000
	7509	85	24	24,8	20	60,0	46,0	2	0,4	1,4	0,8	4 500	6 000
	7509A	85	23	24,8	19	74,8	60,0	2	0,4	1,5	0,8	4 500	6 000
	7309	100	26	27,3	22	83,0	60,0	3	0,3	2,2	1,2	4 000	5 300
50	7210	90	21	21,8	17	56,0	40,0	2	0,4	1,6	0,9	4 300	6 600
	7210A	90	20	21,8	17	70,4	65,0	2	0,4	1,4	0,8	4 300	5 600
	7510	90	24	24,8	20	62,0	54,0	2	0,4	1,4	0,8	4 300	5 600
	7510A	90	23	24,8	19	76,5	64,0	2	0,4	1,4	0,8	4 300	5 600
	7310	110	29	29,3	23	100,0	75,5	3	0,3	1,9	1,1	3 600	4 800
	7310A	110	27	29,3	23	117,0	90,0	3	0,4	1,7	0,9	3 600	4 800
55	7211	100	21	22,8	18	65,0	46,0	3	0,4	1,5	0,8	3 800	5 000
	7511	100	25	26,8	21	80,0	61,0	3	0,4	1,7	0,9	3 800	5 000
	7511A	100	25	26,8	21	99,0	80,0	3	0,4	1,5	0,8	3 800	5 000
	7311	120	29	31,5	25	107,0	81,5	3	0,3	1,8	1	3 200	4 300
60	7212	110	23	23,8	19	78,0	58,0	3	0,4	1,7	0,9	3 400	4 500
	7212A	110	22	23,8	19	91,3	70,0	3	0,4	1,5	0,8	3 400	4 500
	7512	110	28	29,8	24	94,0	75,0	3	0,4	1,5	0,8	3 400	4 500
	7512A	110	28	29,8	24	120,0	100,0	3	0,4	1,5	0,8	3 400	4 500
	7312	130	31	33,5	27	128,0	96,5	4	0,3	2	1,1	3 000	4 000
	7312A	130	31	33,5	26	161,0	120,0	4	0,4	1,7	0,9	3 000	4 000
65	7513	120	31	32,8	27	119,0	98,0	3	0,4	1,6	0,9	3 000	4 000
	7513A	120	31	32,8	27	142,0	120,0	3	0,4	1,5	0,8	3 000	4 000
	7311A	120	29	31,5	25	134,0	110,0	3	0,4	1,7	0,9	3 200	4 300
	7313	140	33	36	28	146,0	112,0	4	0,3	2	1,1	2 600	3 600
	7313A	140	33	36	28	183,0	150,0	4	0,4	1,7	0,9	2 600	3 600

End of Tab. E.2

d	Code	D	B	T	c	Load Capacity, kN		r	e	Y	Y_0	n_{max}, rpm	
						C	C_0						
70	7214	125	26	26,3	21	96,0	82,0	3	0,4	1,6	0,9	3 000	4 000
	7214A	125	24	26,3	21	119,0	89,0	3	0,4	1,4	0,8	3 000	4 000
	7514	125	31	33,3	27	125,0	101,0	3	0,4	1,6	0,9	2 800	3 800
	7314	150	37	38	30	170,0	137,0	4	0,3	1,9	1,1	2 400	3 400
	7314A	150	35	38	30	209,0	170,0	4	0,4	1,7	0,9	2 400	3 400
75	7215	130	26	27,3	22	107,0	84,0	3	0,4	1,6	0,9	2 800	3 800
	7215A	130	25	27,3	22	130,0	100,0	3	0,4	1,4	0,8	2 800	3 800
	7515	130	31	33,3	27	130,0	108,0	3	0,4	1,5	0,8	2 600	3 600
	7515A	130	31	33,3	27	157,0	130,0	3	0,4	1,4	0,8	2 600	3 600
	7315	160	37	40	31	180,0	148,0	4	0,3	1,8	1	2 200	3 200
	7315A	160	37	40	31	229,0	185,0	4	0,4	1,7	0,9	2 200	3 200
80	7216	140	26	28,3	22	112,0	95,2	3	0,4	1,4	0,8	2 400	3 400
	7216A	140	26	28,3	22	140,0	114,0	3	0,4	1,4	0,8	2 400	3 400
	7516	140	33	35,3	28	143,0	126,0	3	0,4	1,5	0,8	2 400	3 400
	7516A	140	33	35,3	28	176,0	155,0	3	0,4	1,4	0,8	2 400	3 400
85	7217	150	28	30,6	24	130,0	109,0	3	0,4	1,4	0,8	2 200	3 200
	7217A	150	28	30,5	24	165,0	134,0	3	0,4	1,4	0,8	2 200	3 200
	7517	150	36	38,5	30	162,0	141,0	3	0,4	1,6	0,9	2 200	3 200
	7517A	150	36	38,5	30	201,0	180,0	3	0,4	1,4	0,8	2 200	3 200
	7317	180	41	44,5	35	230,0	195,0	4	0,3	1,9	1,1	1 900	2 800
90	7218	160	31	32,5	26	158,0	125,0	3	0,4	1,6	0,9	2 000	3 000
	7218A	160	30	32,5	26	183,0	150,0	3	0,4	1,4	0.8	2 000	3 000
	7518	160	40	42,5	34	190,0	171,0	3	0,4	1,6	0,9	2 000	3 000
	7318	190	43	46,5	36	250,0	201,0	4	0,3	1,9	1	1 800	2 600

Appendix F

BEARING COVERS

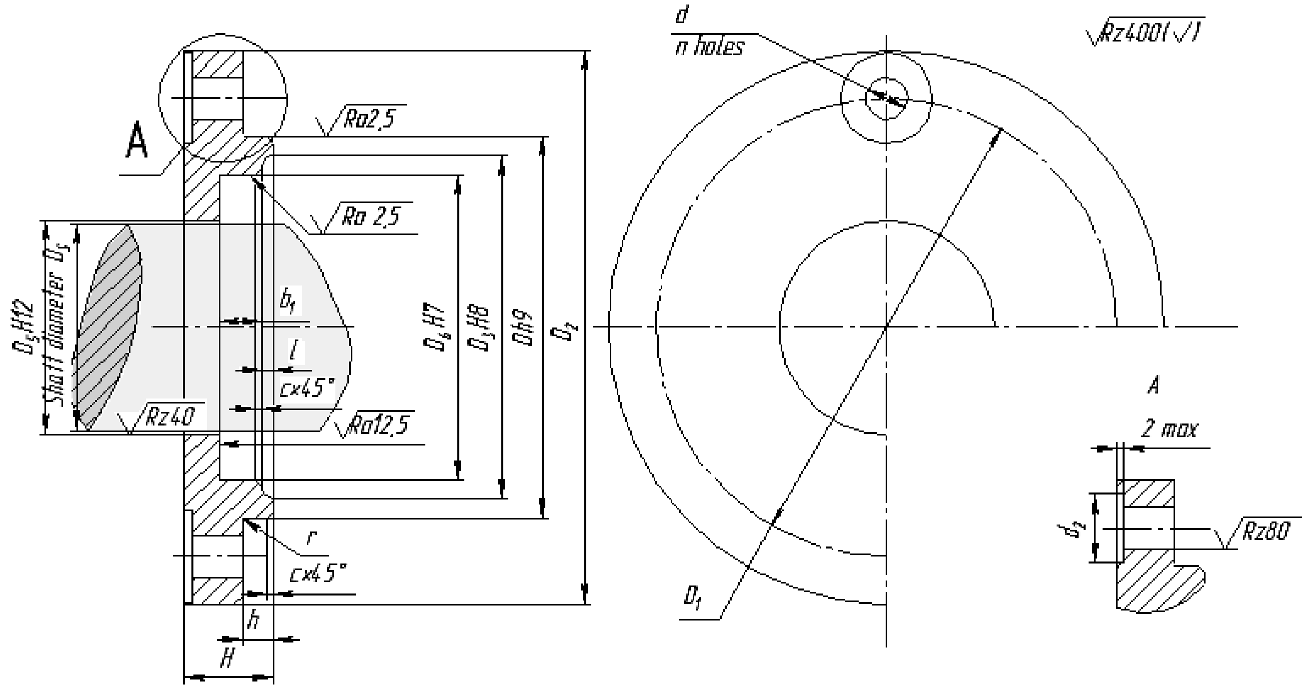


Fig. F.1. Dimensions of a bearing cap (GOST 18512-73)

Table F.1

Dimensions of the Bearing Caps, mm

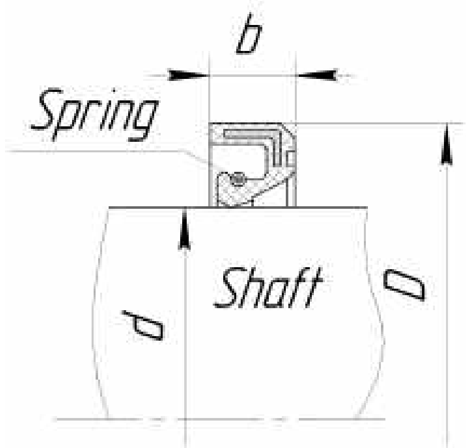
<table>
<tr><th>D</th><th>D_S</th><th>D1</th><th>D2</th><th>D3</th><th>D4</th><th>D5</th><th>D6</th><th>d</th><th>d2</th><th>H</th><th>h</th><th>l</th><th>n</th><th>b1</th><th>c</th><th>r</th></tr>
<tr><td rowspan="2">40</td><td>15</td><td rowspan="4">54</td><td rowspan="4">70</td><td rowspan="4">31</td><td rowspan="4">40</td><td>16</td><td>20</td><td rowspan="15">7</td><td rowspan="15">14</td><td rowspan="5">15</td><td rowspan="15">5</td><td rowspan="18">2</td><td rowspan="18">4</td><td rowspan="5">8,0</td><td rowspan="4">0,6</td><td rowspan="18">0,6</td></tr>
<tr><td>17</td><td>18</td><td>32</td></tr>
<tr><td rowspan="2">42</td><td>12</td><td>13</td><td>28</td></tr>
<tr><td>15</td><td>16</td><td>30</td></tr>
<tr><td rowspan="2">47</td><td>17</td><td rowspan="2">60</td><td rowspan="2">78</td><td rowspan="2">38</td><td rowspan="2">47</td><td>18</td><td>32</td><td rowspan="14">1,0</td></tr>
<tr><td>20</td><td>21</td><td>40</td><td rowspan="13">17</td><td rowspan="13">11,0</td></tr>
<tr><td rowspan="2">52</td><td>20</td><td rowspan="2">65</td><td rowspan="2">82</td><td rowspan="2">44</td><td rowspan="2">50</td><td>21</td><td>40</td></tr>
<tr><td>25</td><td>26</td><td>42</td></tr>
<tr><td>55</td><td>25</td><td>75</td><td>95</td><td>48</td><td>50</td><td>26</td><td>42</td></tr>
<tr><td rowspan="2">60</td><td>25</td><td rowspan="6">78</td><td rowspan="6">95</td><td rowspan="6">52</td><td rowspan="6">60</td><td>26</td><td>42</td></tr>
<tr><td>30</td><td>31</td><td>52</td></tr>
<tr><td rowspan="4">62</td><td>20</td><td>21</td><td>40</td></tr>
<tr><td>25</td><td>26</td><td>42</td></tr>
<tr><td>30</td><td>31</td><td>52</td></tr>
<tr><td>32</td><td>33</td><td>52</td></tr>
<tr><td rowspan="3">68</td><td>30</td><td rowspan="3">84</td><td rowspan="3">105</td><td rowspan="3">58</td><td rowspan="3">68</td><td>31</td><td>52</td><td rowspan="3">9</td><td rowspan="3">20</td><td rowspan="3">6</td></tr>
<tr><td>35</td><td>36</td><td>58</td></tr>
<tr><td>40</td><td>41</td><td>60</td></tr>
</table>

End of Tab. F.1

<table>
<tr><th>D</th><th>D_S</th><th>D_1</th><th>D_2</th><th>D_3</th><th>D_4</th><th>D_5</th><th>D_6</th><th>d</th><th>d_2</th><th>H</th><th>h</th><th>l</th><th>n</th><th>b_1</th><th>c</th><th>r</th></tr>
<tr><td rowspan="6">72</td><td>20</td><td rowspan="8">90</td><td rowspan="8">110</td><td rowspan="6">62</td><td rowspan="8">72</td><td>21</td><td>40</td><td rowspan="23">9</td><td rowspan="23">20</td><td rowspan="8">17</td><td rowspan="23">6</td><td rowspan="6">2</td><td rowspan="8">4</td><td>8,0</td><td rowspan="23">1,0</td><td rowspan="39">0,6</td></tr>
<tr><td>25</td><td>26</td><td>42</td><td rowspan="18">11,0</td></tr>
<tr><td>30</td><td>31</td><td>52</td></tr>
<tr><td>35</td><td>36</td><td>58</td></tr>
<tr><td>38</td><td>39</td><td>58</td></tr>
<tr><td>45</td><td>46</td><td>65</td></tr>
<tr><td rowspan="2">75</td><td>35</td><td rowspan="2">64</td><td>36</td><td>58</td><td rowspan="33">3</td></tr>
<tr><td>45</td><td>46</td><td>65</td></tr>
<tr><td rowspan="6">80</td><td>25</td><td rowspan="8">100</td><td rowspan="8">120</td><td rowspan="8">72</td><td rowspan="8">80</td><td>26</td><td>42</td><td rowspan="11">18</td><td rowspan="31">6</td></tr>
<tr><td>30</td><td>31</td><td>52</td></tr>
<tr><td>35</td><td>36</td><td>58</td></tr>
<tr><td>40</td><td>41</td><td>60</td></tr>
<tr><td>45</td><td>46</td><td>65</td></tr>
<tr><td>50</td><td>51</td><td>70</td></tr>
<tr><td rowspan="2">85</td><td>35</td><td>36</td><td>58</td></tr>
<tr><td>45</td><td>46</td><td>65</td></tr>
<tr><td rowspan="4">90</td><td>30</td><td rowspan="7">110</td><td rowspan="7">130</td><td rowspan="7">80</td><td rowspan="7">92</td><td>31</td><td>52</td></tr>
<tr><td>40</td><td>41</td><td>60</td></tr>
<tr><td>50</td><td>51</td><td>70</td></tr>
<tr><td>55</td><td>56</td><td>80</td><td>21</td><td>13,6</td></tr>
<tr><td rowspan="3">95</td><td>40</td><td>41</td><td>60</td><td rowspan="2">18</td><td rowspan="2">11,0</td></tr>
<tr><td>50</td><td>51</td><td>70</td></tr>
<tr><td>60</td><td>62</td><td>85</td><td>21</td><td>13,6</td></tr>
<tr><td rowspan="4">100</td><td>35</td><td rowspan="4">120</td><td rowspan="4">145</td><td rowspan="4">90</td><td rowspan="4">100</td><td>36</td><td>58</td><td rowspan="16">11</td><td rowspan="16">24</td><td rowspan="16">23</td><td rowspan="16">8</td><td rowspan="2">11,0</td><td rowspan="16">1,6</td></tr>
<tr><td>45</td><td>46</td><td>65</td></tr>
<tr><td>55</td><td>56</td><td>80</td><td rowspan="2">13,6</td></tr>
<tr><td>65</td><td>67</td><td>90</td></tr>
<tr><td rowspan="3">105</td><td>45</td><td rowspan="8">130</td><td rowspan="8">155</td><td rowspan="8">95</td><td rowspan="8">110</td><td>46</td><td>65</td><td>11,0</td></tr>
<tr><td>55</td><td>56</td><td>80</td><td rowspan="2">13,6</td></tr>
<tr><td>70</td><td>72</td><td>95</td></tr>
<tr><td rowspan="5">110</td><td>40</td><td>41</td><td>60</td><td rowspan="2">11,0</td></tr>
<tr><td>50</td><td>51</td><td>70</td></tr>
<tr><td>60</td><td>562</td><td>85</td><td rowspan="3">13,6</td></tr>
<tr><td>70</td><td>72</td><td>95</td></tr>
<tr><td>75</td><td>77</td><td>100</td></tr>
<tr><td rowspan="4">115</td><td>50</td><td rowspan="4">140</td><td rowspan="4">165</td><td rowspan="4">105</td><td rowspan="4">120</td><td>51</td><td>70</td><td>11,0</td></tr>
<tr><td>65</td><td>67</td><td>90</td><td rowspan="3">13,6</td></tr>
<tr><td>75</td><td>77</td><td>100</td></tr>
<tr><td>80</td><td>82</td><td>105</td></tr>
</table>

Appendix G

OIL SEALS (GOST 8752-79)



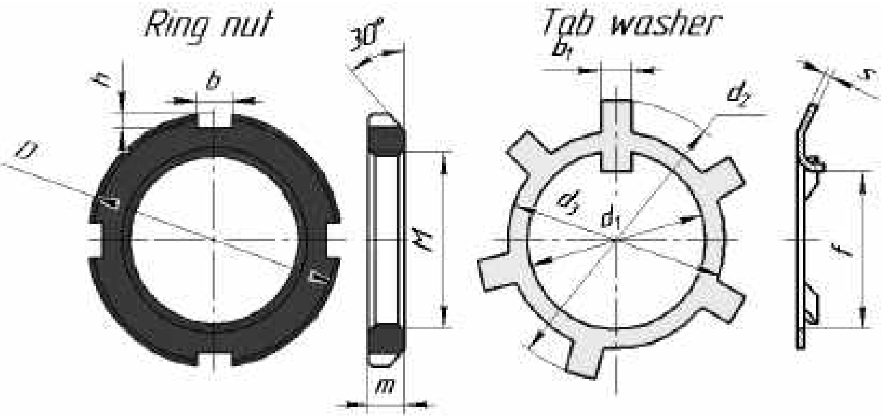
Dimensions, mm

d	D	b
10	26	7
11	26	7
12	28	7
13	28	7
14	28	7
15	30;32	7
16	30;35	7
17	32	7
18	35	7
19	35	7
20	35	7
20	40	10
21	40	10
22	40	10
24	40	7
25	42	7
26	45	7
28	50	10
30	52	10
32	52	10
35	58	10
36	58	10
38	58	10
40	60;62	10

d	D	b
42	62	10
45	65	10
48	70	10
50	70	10
52	75	10
55	80	10
56	80	10
58	80	10
60	85	10
63	90	10
65	90	10
70	95	10
71	95	10
75	100	10
80	105	10
85	110	12
90	120	12
92	120	12
100	125	12

Appendix H

RING NUTS & LOCKING WASHERS



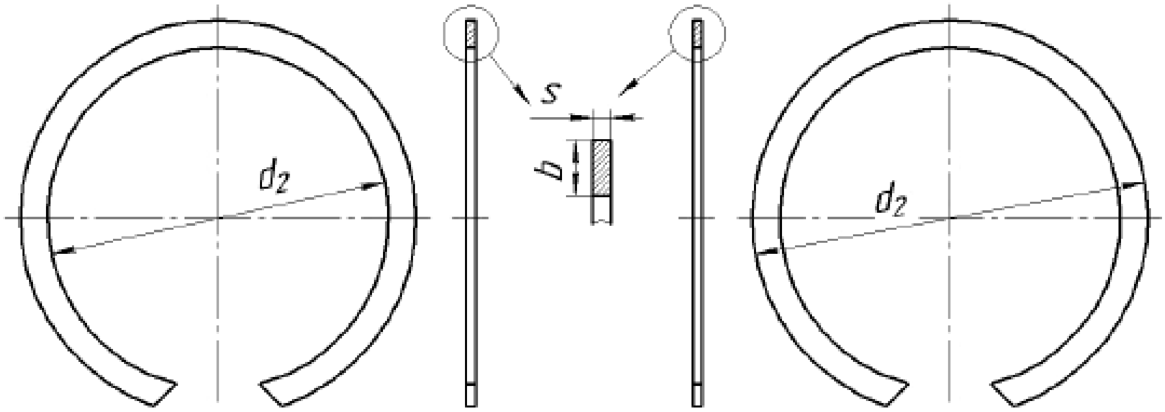
GOST 11871-88 GOST 11872-89

Dimensions, mm

Shaft diam.	Thread	Nut							Washer					
d	M	f_1	b_1	D	m	b	h	z	b_1	f	s	d_1	d_2	d_3
20	18x1,5	15	5,3	32	8	6	2,0	4	4,8	15	1,0	18,5	34	24
	20x1,5	17	5,3	32	6	6	2,0	4	4,8	17	1,0	20,5	37	27
25	24x1,5	21	5,3	42	10	6	2,5	4	4,8	21	1,0	24,5	44	33
30	27x1,5	24	5,3	45	10	6	2,5	4	4,8	24	1,0	27,5	47	36
	30x1,5	27	5,3	45	7	6	2,5	4	4,8	27	1,0	30,5	50	39
35	33x1,5	30	6,3	52	10	8	3,0	4	5,8	30	1,6	33,5	54	42
40	39x1,5	36	6,3	60	10	8	3,0	4	5,8	36	1,6	39,5	62	48
45	42x1,5	39	6,3	65	10	8	3,0	4	5,8	39	1,6	42,5	67	52
	45x1,5	42	6,3	63	8	8	3,0	6	5,8	42	1,6	45,5	72	56
50	48x1,5	45	8,3	75	12	8	3,5	6	7,8	45	1,6	48,5	77	60
55	52x1,5	49	8,3	80	12	10	3,5	6	7,8	49	1,6	52,5	82	65
60	60x2	57	8,3	80	8	10	4,0	6	7,8	57	1,6	61,0	92	75
65	64x2	61	8,3	95	12	10	4,0	6	7,8	61	1,6	65,0	97	80
70	68x2	65	10,0	100	15	10	4,0	6	9,5	65	1,6	69,0	102	85

Appendix I

CIRCLIPS



Outer (GOST 13940-86)

Inner (GOST 13941-86)

Dimensions, mm

<table>
<tr><th>Shaft diam.</th><th colspan="3">Outer</th><th colspan="3">Inner</th><th></th><th>Shaft diam.</th><th colspan="3">Outer</th><th colspan="3">Inner</th></tr>
<tr><th>d</th><th>d_2</th><th>s</th><th>b</th><th>d_2</th><th>s</th><th>b</th><th></th><th>d</th><th>d_2</th><th>s</th><th>b</th><th>d_2</th><th>s</th><th>b</th></tr>
<tr><td>20</td><td>18,2</td><td rowspan="11">1,2</td><td rowspan="5">3,2</td><td>21,8</td><td rowspan="2">1,0</td><td rowspan="2">2,0</td><td></td><td>50</td><td>45,8</td><td rowspan="8">2,0</td><td rowspan="8">6,0</td><td>54,2</td><td rowspan="13">1,7</td><td>4,0</td></tr>
<tr><td>22</td><td>20,2</td><td>23,8</td><td></td><td>52</td><td>47,8</td><td>56,2</td><td rowspan="12">5,0</td></tr>
<tr><td>23</td><td>21,1</td><td>24,9</td><td rowspan="13">1,2</td><td rowspan="9">2,5</td><td></td><td>54</td><td>49,8</td><td>58,2</td></tr>
<tr><td>24</td><td>22,1</td><td>25,9</td><td></td><td>55</td><td>50,8</td><td>59,2</td></tr>
<tr><td>25</td><td>23,1</td><td>26,9</td><td></td><td>56</td><td>51,8</td><td>60,2</td></tr>
<tr><td>26</td><td>24,0</td><td rowspan="6">4,0</td><td>28,0</td><td></td><td>58</td><td>53,8</td><td>62,2</td></tr>
<tr><td>28</td><td>25,8</td><td>30,2</td><td></td><td>60</td><td>55,8</td><td>64,2</td></tr>
<tr><td>29</td><td>26,8</td><td>31,2</td><td></td><td>62</td><td>57,8</td><td>66,2</td></tr>
<tr><td>30</td><td>27,8</td><td>32,2</td><td></td><td>65</td><td>60,8</td><td rowspan="9">2,5</td><td rowspan="5">7,0</td><td>69,2</td></tr>
<tr><td>32</td><td>29,5</td><td>34,5</td><td></td><td>68</td><td>63,6</td><td>72,5</td></tr>
<tr><td>34</td><td>31,4</td><td>36,5</td><td></td><td>70</td><td>65,6</td><td>74,5</td></tr>
<tr><td>35</td><td>32,2</td><td rowspan="9">1,7</td><td rowspan="9">5,0</td><td>37,8</td><td rowspan="4">3,2</td><td></td><td>72</td><td>67,6</td><td>76,5</td></tr>
<tr><td>36</td><td>33,0</td><td>38,8</td><td></td><td>75</td><td>70,6</td><td>79,5</td></tr>
<tr><td>37</td><td>34,0</td><td>39,8</td><td></td><td>78</td><td>73,5</td><td rowspan="4">8,0</td><td>82,5</td><td rowspan="6">2,0</td><td rowspan="6">6,0</td></tr>
<tr><td>38</td><td>35,0</td><td>40,8</td><td></td><td>80</td><td>75,0</td><td>85,5</td></tr>
<tr><td>40</td><td>36,5</td><td>43,5</td><td rowspan="5">1,7</td><td rowspan="5">4,0</td><td></td><td>82</td><td>77,0</td><td>87,5</td></tr>
<tr><td>42</td><td>38,5</td><td>45,5</td><td></td><td>85</td><td>79,5</td><td>90,5</td></tr>
<tr><td>45</td><td>41,5</td><td>48,5</td><td></td><td>88</td><td>82,5</td><td rowspan="2">3,0</td><td rowspan="2">8,5</td><td>93,5</td></tr>
<tr><td>46</td><td>42,5</td><td>49,5</td><td></td><td>90</td><td>84,5</td><td>95,5</td></tr>
<tr><td>48</td><td>44,5</td><td>51,6</td><td></td><td></td><td></td><td></td><td></td><td></td><td></td><td></td></tr>
</table>

REFERENCES

1. Spotts, M. F. Design of Machine Elements / by Merhyle Franklin Spotts Terry E. Shoup, Lee Emrey Hornberger. Pearson Prentice Hall, 2004. – 928 p.
2. Gupta, J. K. A Textbook of Machine Elements/ by R. S. Khurmi, J. K. Gupta. Eurasia publishing house. – 2005. – 1230 p.
3. Dobrovolsky, V. Machine elements. M.: Peace Publishers, 1977. – 606 p.
4. Lingaiah, K. Machine Design Databook. Second Edition / by K. Lingaiah. McGraw-Hill. – 2004. – 1296 p.
5. Mechanical Engineer's Handbook. Edited by Dan B. Marghitu. – Academic Press. – 2001. – 865 p.
6. Основы расчета и конструирования деталей и механизмов ЛА [Текст]: учеб. пособ. для втузов / Н. А. Алексеева, Л. А. Бонч-Осмоловский, В. В. Волгин и др.; под ред. В. Н. Кестельмана. Г. И. Рощина. – М.: Машиностроение, 1989. – 456 с.
7. Анурьев, В. И. Справочник конструктора–машиностроителя: в 3 т. [Текст] – 9-е изд. перераб. и доп.; под ред. И. Н. Жестковой. М.: Машиностроение, 2006.
8. Доценко, В. Н. Проектирование зубчатых передач [Текст]: учеб. пособ. / В. Н. Доценко, Ю. В. Ковеза. – Х.: Нац. аэрокосм. ун-т им. Н. Е. Жуковского «ХАИ», 2009. – 99 с.
9. Кузьминов, Ф. Ф. Подбор муфт. Конструкция, основы расчета. [Текст]: учеб. пособ. по курс. и диплом. проект. /Ф. Ф. Кузьминов, С. И. Пшеничных. – Х.: Нац. аэрокосм. ун-т им. Н. Е. Жуковского «ХАИ», 2006. – 103 с.
10. Киркач, Н. Ф. Расчет и проектирование деталей машин [Текст]: учеб. пособ. / Н. Ф. Киркач, Р. А. Баласанян. – Х.: Основа, 1991. – 276 с.

TABLE OF CONTENT

Навчальне видання

Ковеза Юрій Володимирович
Лихошерст Іван Григорович
Світличний Сергій Петрович

КОНСТРУЮВАННЯ ДЕТАЛЕЙ МАШИН. КУРСОВЕ ПРОЕКТУВАННЯ

(Англійською мовою)

Редактор С. П. Опаріна
Технічний редактор Л. О. Кузьменко

Зв. план, 2014
Підписано до друку 14.07.2014
Формат 60x84 1/16. Папір офс. № 2. Офс. друк
Ум. друк. арк. 10. Обл.-вид. арк. 11,25. Наклад 200 пр. Замовлення 249.
Ціна вільна

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