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MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE

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FUNDAMENTALS OF ENGINEERING CALCULATIONS

Lecture notes
for students of specialty 274 "Motor Transport"
full-time and part-time study

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CONTENT

1 MECHANICAL TRANSMISSIONS AND THEIR MAIN LINKS (DETAILS)

1.1 General provisions

Transmissions are called devices for transmitting energy over a distance.

Transmissions (*transmissions*) are divided into: electric, pneumatic, hydraulic and mechanical (*electrical*, *pneumatic*, *hydraulic*, *mechanical transmission*).

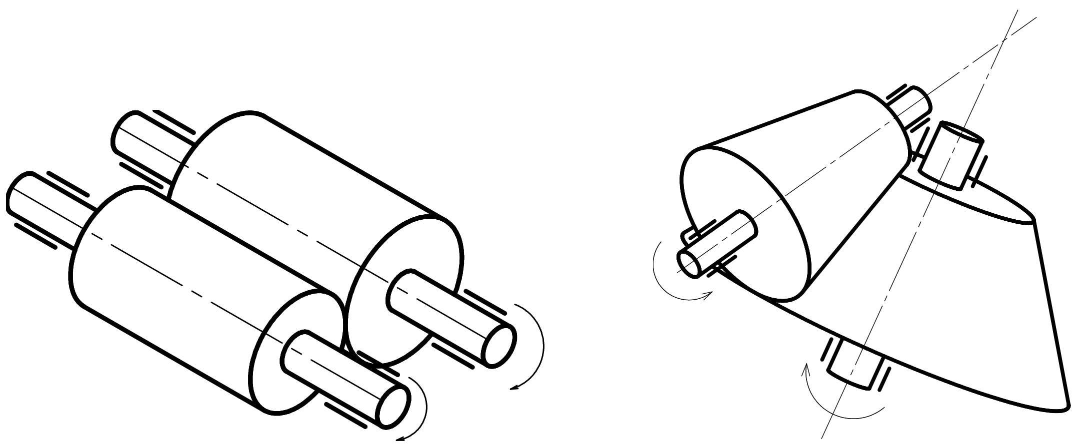
Mechanical transmission is a device for converting the parameters of the engine's motion into the parameters of the motion of the executive link of the machine drive [1, 2].

Mechanical transmissions (hereinafter simply transmissions) are divided into two classes according to the principle of action (motion transmission): friction transmissions (friction and belt); meshing (gears - cylindrical, conical, helical, rack; worm; screw-nut transmissions; chain) [1, 2, 3, 4].

The main transmission links are rollers, pulleys, gears, worms, racks, belts and chains. Other components of the transmission are shafts and shaft supports - rolling and sliding bearings, housing parts (housings, covers, etc.), spacers and rings, shaft sealing devices (cuffs, etc.) and connecting parts (pins, bolts, screws, etc.).

1.2 Friction gears

Figure 1 schematically shows examples of the simplest friction gears: cylindrical (Fig. 1, a) and conical (Fig. 1, b), the rollers of which are tightly pressed against each other using special devices (for example, spring or due to elastic deformation of the roller shafts, etc.).



and) b)

Figure 1 – Friction transmission (*friction gear*)

One of the leading skaters (*driven*), the second is the slave (*wheel*). The main advantages of such gears are simplicity of design and practically silent operation. The main disadvantages of these gears include instability of the gear ratio due to slipping during operation, large pressure forces on the shafts due to the need to press one roller against another, and relatively low power due to significant heat generation.

1.3 Passing passes

Belt drives allow you to transmit motion over distances of up to 15 m or more. These drives are simple in design and operation.

Transmissions consist of a drive pulley 1 (Fig. 2), which is usually installed on the shaft of a motor (for example, an electric motor), a belt 2, and a driven pulley 3, which transmits rotational motion to a shaft 4, which is a component of some mechanism of a machine (car, machine tool, conveyor, etc.).

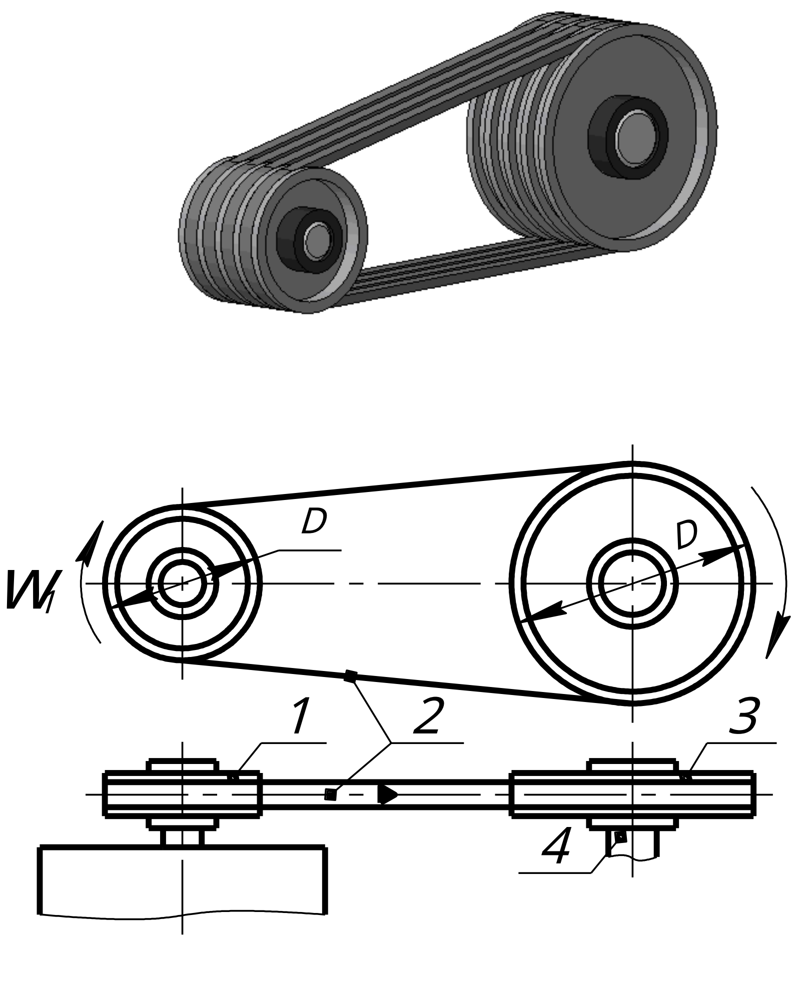


Figure 2 – Belt transmission (*belt drive*)

2 GEAR TRANSMISSIONS

2.1 General information

Gears (*toothed transmissions*) serve to transmit rotational motion from one shaft to another or to convert rotational motion into translational motion.

In mechanical engineering, gears are widely used in various mechanisms (gearboxes of machine tools, automobiles, precise movement mechanisms, dividing heads, etc.).

To transmit rotational motion (*rotating motion*) from one shaft to another, the axes of which are parallel, cylindrical gears with external (Fig. 3, a - c) and internal (Fig. 3, d) gearing are used; if the shaft axes intersect, then bevel gears with straight, oblique (tangential) and circular teeth are used, respectively (Fig. 3, e - g). In the case when the shaft axes are parallel, usually with a right angle of intersection, worm and screw gears are used (Fig. 3, i, k). Rack and pinion gears (Fig. 3, a) are used to convert rotational motion into translational motion and vice versa.

In mechanical engineering, the most common gears are those with an involute tooth profile, proposed by L. Euler in the 18th century, which has a number of significant technological advantages over other tooth profiles. In powerful gears, gears (cylindrical and bevel) with Novikov meshing (circular helical gears) are used.

Gear wheels are mounted on shafts. To transmit torque from the gears to the shaft, or vice versa, various types of connections are used - key, spline, pin, etc. Key and spline straight and involute connections are standardized.

In the meshing of two gears, the smaller of the wheels is called the pinion, and the larger is called the cogwheel.

Gear (*gear*) is a gear with a smaller number of teeth. In the case when the number of teeth of one wheel is equal to the number of teeth of the other (gear ratio $U=1$) gear is called the driving (*driving*) wheel, and driven by the wheel (*driven*).

*Leading*a wheel that receives rotational motion is called, for example from the engine, and*led*is a wheel driven by a gear (paired gear).

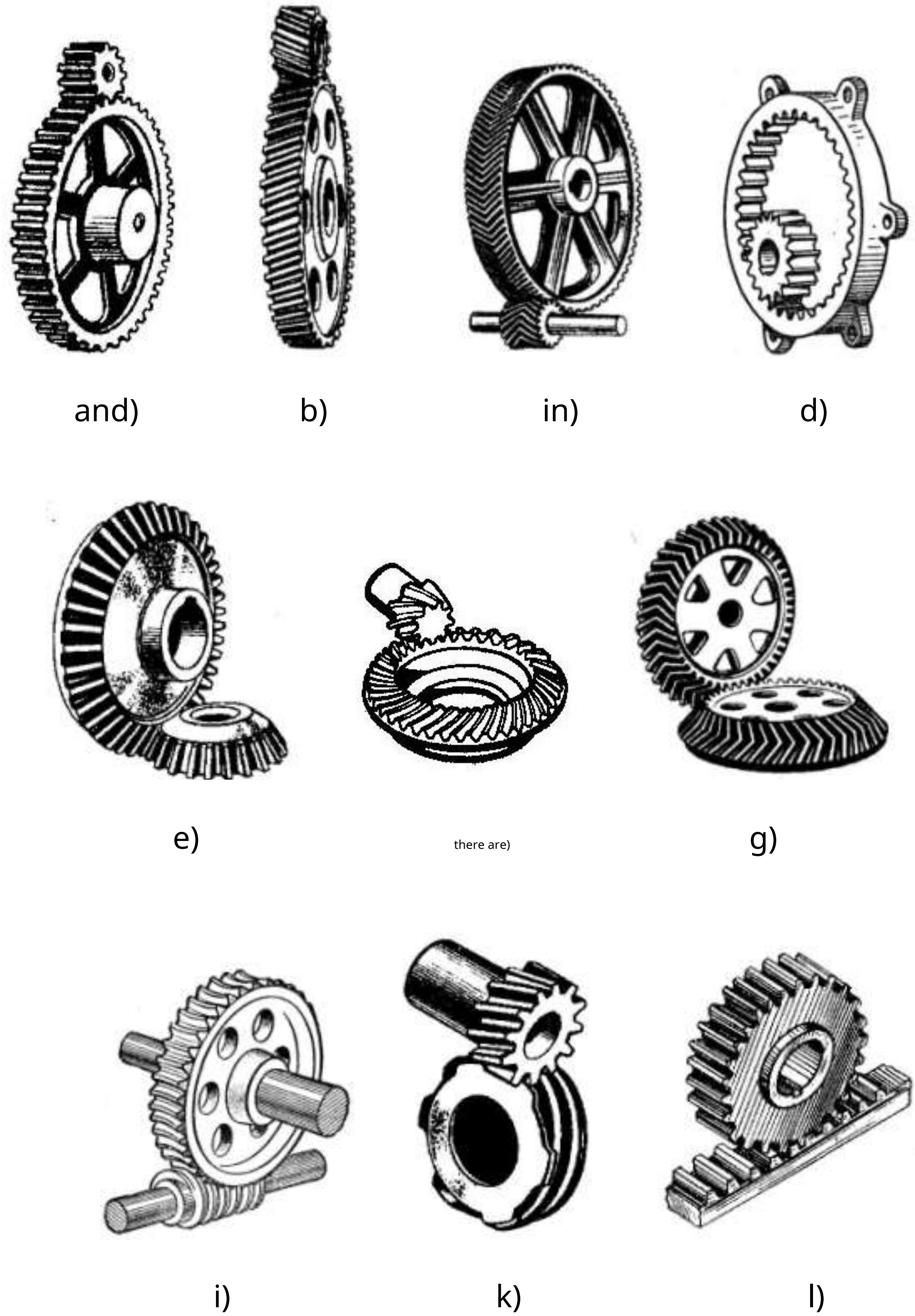


Figure 3 – Types of gears

Depending on the nature of the gearing (external or internal), the relative arrangement of the shafts, the absolute dimensions, the type of production (single, small-scale, mass), etc., gears can have a variety of designs. The most widely used in technology are cylindrical and bevel gears and worm gears, which are a kind of "symbiosis" of the screw-nut transmission (a worm is essentially a screw with a single, double or quadruple thread) and the cylindrical helical gear (a worm wheel with an arched tooth shape).

2.2 Cylindrical involute gearing

2.2.1 Elements of an involute gear

The main element (Fig. 4, a) of the involute (*involute*) of a gear wheel is a tooth (Fig. 4, b) – a protrusion, the lateral sides of which are outlined by an involute, designed to transmit motion directly by acting on the tooth of another wheel.

The surface on which the teeth are formed is called the toothed crown, and the space between the lateral surfaces of adjacent teeth, the surfaces of the peaks and bases of the depressions is called the depression.

The initial surface of an involute gear wheel is the coaxial surface on which the same surface of another wheel, which is in mesh with the first one, rolls without slipping (the main property of an involute) (Fig. 5, a). The dividing surface (*reference plane*) the wheel is divided by a tooth (*tooth*) into two parts – the head and the leg (Fig. 4, a).

For uncorrected involute gears (*non-adjusted toothed involute gear*) the dividing and initial surfaces coincide.

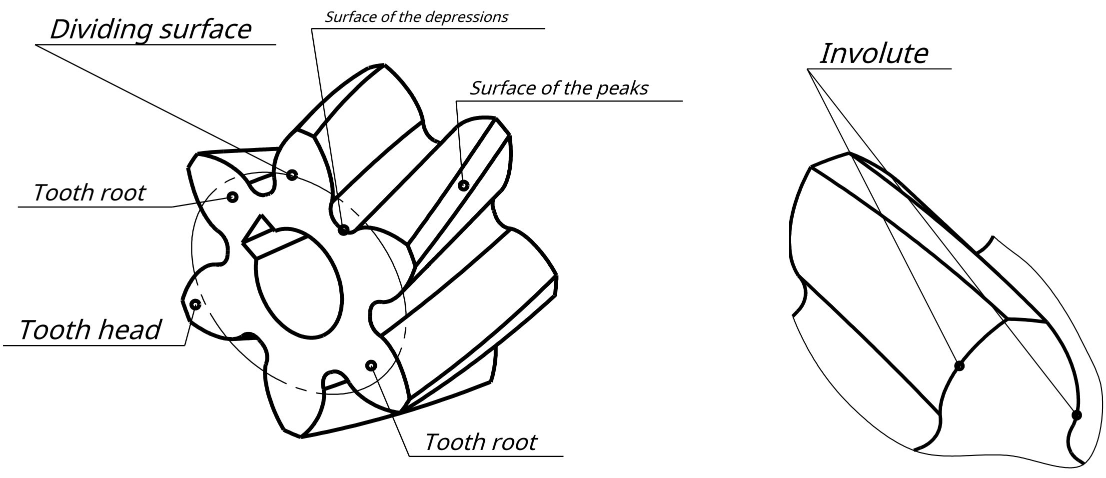


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Figure 4 – Elements (*components*) involute gear wheel

2.2.2 Basic geometric parameters of cylindrical involute gears

The main geometric parameter of gears is the module (*module*) teeth m_t – the value is proportional to the step (*pitch*) teeth p_t on the dividing cylinder of the wheel. The proportionality coefficient is π^{-1}. Thus, $m_t = p_t/\pi^{-1}$ – normalized step, where π^{-1} –normalizing factor.

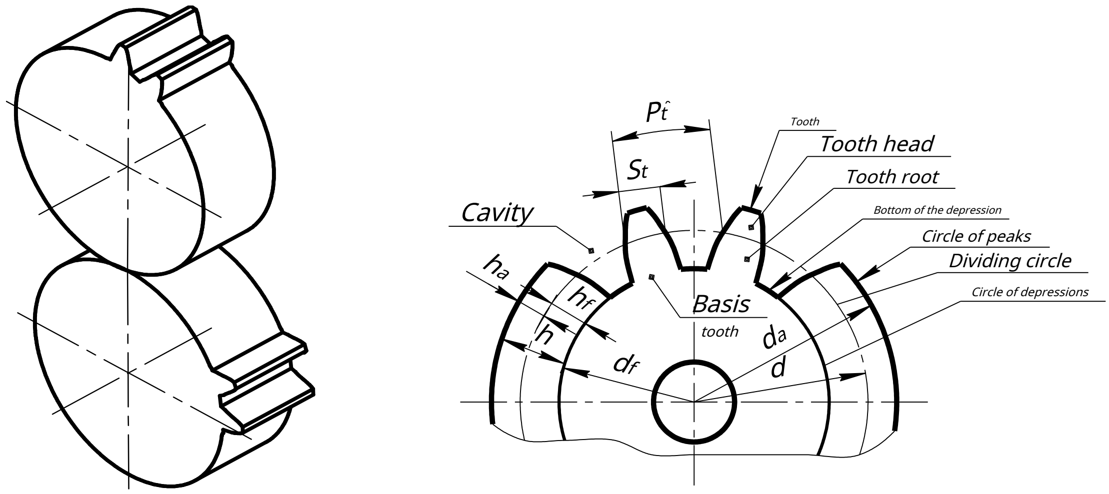

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Figure 5 – Geometric parameters of the involute gear (*of involute toothed gear*)

Circular pitch r_t is the distance between the surfaces of the same name of two of adjacent wheel teeth, measured along the circumference of the wheel pitch surface.

It is obvious that the length of the gear wheel circumference along the pitch diameter d(rice.5, b), circular step (*butt pitch*) p_t and the number of teeth (*number of teeth*) z wheels are connected by dependency $\pi d = zp_t$ from where $p_t = \pi d/z$ and $m_t = d/z$, i.e. circular modulus m_t is the fraction of dividing the diameter of the pitch wheel by number of teeth (another definition of module that is often found in the literature).

In the general case, for helical cylindrical involute gears, circular p_t, normal p_n and axial p_x steps by which corresponding modules correspond. In helical cylindrical involute gears, a circular (face) module is used m_t and normal $m_n = p_n/\pi$. Since the normal step p_n is the smallest distance along dividing cylinder between the same profile surfaces of two adjacent teeth, then $p_n = p_t \cos\beta$ and $m_n = m_t \cos\beta$, where β – the angle of inclination of the tooth line to the generator of the pitch cylinder. For spur cylindrical involute gears, circular and normal steps and their corresponding modules coincide. Since the profile (*profile*) of a helical tooth is identical to the profile of a straight tooth in a plane intersecting the helical tooth at an angle of 90° (normal plane), then in helical cylindrical gears the normal module is taken as the standard modulem_n(*main module*).

The operability of a cylindrical involute gear (hereinafter referred to as a cylindrical gear) is ensured if the circular pitches of the gear teeth$p_{t1}$and$p_{t2}$the wheels are even (Fig.6).

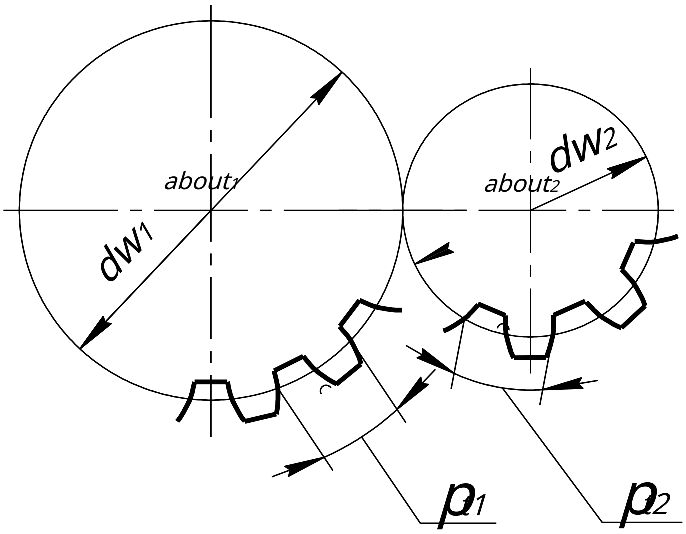


Figure 6 – Initial cylinders (*pitch cylinder*)
engagement (*gearing*)

Initial diameters(*pitch diameter*)d_{w1}andd_{w2}conjugate pair Gear pitches are the diameters of the gear surfaces that roll over each other without slipping. Pitch diameters$d_1$and$d_2$divide the teeth of the wheels into two parts – heads and legs.

Head(*addendum*)*tooth*h_a–this is the part of the tooth height that protrudes above dividing diameter.

Leg(*dedendum*)*tooth*h_f–this is the part of the tooth height that is under the pitch diameter.

Height(*tooth*)*tooth***h**-is the radial height between the circles of the protrusions and tooth cavities.

Diameter of tooth tipsd_{and}–limits the tops of the teeth.

Tooth cavity diameterd_f–passes through the base of the tooth cavities.

*Wheelbase***and**$_w$–distance between shaft axes (centers rotation of the wheels), on which the gear wheels are mounted, which are in engagement:

$$a_w=0,5(d_{w1}+d_{w2}),$$

whered_{w1}–initial gear diameter;

d_{w2}–pitch diameter (*reference diameter*) of the gear wheel.

Basic geometric parameters (*main geometric parameters*) and formulas for their calculation (*analysis formulas*) for a cylindrical spur uncorrected gear wheel (gear) are given in Table 1.

Table 1 – Geometric parameters of a cylindrical spur gear uncorrected gear wheel

Recognize-education	Parameter name	Calculation formulas
z	Number of teeth	–
m	Module	standard
h	Tooth height	$h=2,25m$
h_a	Tooth head height	$h_a=m$
h_f	Tooth root height	$h_f=1,25m$
d	Pitch diameter	$d=mz$
d_a	Diameter of tooth tips	$d_a=d+2h_a$
d_f	Tooth cavity diameter	$d_f=d–2h_f$
p	Tooth pitch	$p=m\pi$
s	Tooth thickness	$s=0,5p$
e	Width of the depression	$e=0,5p$
with	Radial clearance	$0,25m$

Design parameters (*constructive parameters*) cylindrical gears in the absence of special conditions specified in the technical task for the development of gear transmission, are determined according to recommendations established on the basis of generalization of experience in the design, manufacture and operation of gear transmissions. For example, the dimensions of the gear wheel, which are not directly related to the geometric parameters of the gearing, are found by simple formulas: hub diameter $d_m=(1,6...1,9)d_{in}$ (where d_{in} – diameter of the shaft seating surface on which install the wheel); hub length $l_m=(0,8...1,5)d_{in}$; wheel rim thickness $C_q=(0,15...0,3)b_2$ (where b_2 – wheel width; gear width $b_2=b_2+2...10$ *mm*); thickness of the wheel disk under the tooth cavities $S_q=(1,5...3)m$, chamfer (end) on the teeth $n=0,5m\cdot 45^{\circ}$. Wheel width b_2 taken as a fraction of the pitch (initial) diameter of the gear: $b_2=\psi_{bd}d_{w1}$, where ψ_{bd} – the coefficient of the width of the gear rim, which is taken in depending on the location of the wheel relative to the shaft supports and the type of heat treatment of the teeth ($\psi_{bd}=0,2...0,6$ in the case of cantilever arrangement wheels; $\psi_{bd}=0,3...1,4$ for asymmetrical wheel arrangement; $\psi_{bd}=0,4...1,6$ – for symmetrical arrangement).

Smaller coefficient values ψ_{bd} take if the teeth of the wheels have passed heat treatment according to the following types: bulk hardening (OG); high-frequency current hardening (HF); cementation (C); nitro-cementation (NTS); nitriding (A). Larger values are assigned in cases of heat treatment such as annealing (B), normalizing (N) and improving (P).

2.2.3 Sketching a spur gear uncorrected gear wheel

Before sketching a spur uncorrected cylindrical gear, it is necessary to determine the value of the modulus m.

1. To do this, first measure, for example, with a caliper, diameter of a circle d_{and} tooth tops and tooth height and count the number of teeth z. Calculation module m_p are found by the formulas:

$$m=h/2{,}25n \text{ or } m=d_a/(z-2).$$

For example, $d_{and}=75$, number of teeth $z=23$. Then

$$m=d_a/(z+2)=75/(23+2)=3 \text{ mm}.$$

2. The obtained value of the calculation module (*calculation module*) must be matched to the standard value of the gear modules, which are selected from a standard range of values:

1st row: 0.05; 0.06; 0.08; 0.1; 0.12; 0.15; 0.2; 0.25; 0.3; 0.4; 0.6; 0.8; 1; 1.25; 1.5; 2; 2.5; 3; 4; 6; 8; 10; 12; 16; 20; 25; 32; 40; 60; 80.

2nd row: 0.055; 0.07; 0.09; 0.11; 0.14; 0.15; 0.18; 0.22; 0.28; 0.35; 0.45; 0.7; 0.9; 1.125; 1.375; 1.75; 2.25; 2.75; 3.5; 4.5; 7; 9; 11; 14; 18; 22; 28; 36; 45; 70; 90.

Note: when choosing the module value, preference is given to the first row.

3. Calculate the gear wheel parameters:

pitch diameter $d=mz=3\times23=69$ *mm. diameter of the tooth tips* $d=m(z+2)=3(23+2)=75$ *mm. diameter of tooth cavities* $d=m(z-2{,}5)=3(23-2{,}5)=102{,}5$ *mm. tooth height* $h=2{,}25m=2{,}25\times3=6{,}75$ *mm.*
height of the tooth head $h_{and}=m=3$ *mm.*

tooth root height $h_f=1{,}25m=1{,}25\times3=3{,}75$ *mm.*

Other geometric dimensions of the wheel are found using the given recommendation formulas and the given diameter. d_{in} shaft surfaces and coefficient ψ_{bd} For example for $d_{in}=18$ mm and $\psi_{bd}=0,22$ (console installation of a hardened gear wheel), we get:

$b=\psi_{bd}d=0,22\cdot69=15,18$ mm; we take $b=15$ mm;
$d_m=(1,6...1.9)d_{in}=(1,6...1,9)\cdot18=28,8...34,2$ mm; we take $d_m=30$ mm; $l_m=(0.8...1,5)d_{in}=(0,8...1,5)\cdot18=14,4...27$ mm.

For gears mounted on cantilevered shafts, a single-sided hub is more appropriate (*hub*) [14], therefore, taking into account the width b gear wheel, the total length of the hub is defined as:

$l_{m\Sigma}=l_m+b=(14,4...27)+15=29,4...42$ mm; we take $l_{m\Sigma}=35$ mm;

$$n=0,5m=0,5\cdot3=1,5 \text{ mm}.$$

Calculated cylindrical spur gear (*spur wheel*) the wheel has small absolute dimensions, for this reason reducing its mass by thinning the disk is impractical, therefore the disk thickness is taken equal to the width of the gear ring, *WITH* $_d=b=15$ mm.

According to the calculated geometric parameters, a sketch or working drawing (Fig. 7) of a spur uncorrected cylindrical gear wheel is made in compliance with all the requirements of the interstate standard GOST 2.403–75 “Rules for the execution of drawings of cylindrical gears”. According to this standard, the image of the gear wheel must indicate: the diameter of the tooth tops; the width of the gear rim; the angle of the sector around the circumference of the tooth tops - for the gear sector; the dimensions of the chamfers or the radii of curvature of the blunting lines of the tooth edges. It is allowed to indicate the dimensions of the chamfers or the radii of curvature of the blunting lines in the technical requirements of the drawings; the roughness of the side surfaces of the teeth; the depth of modification - for gears with longitudinal modification of the teeth.

The gear drawing or sketch must include a table of the parameters of the gear rim, which consists of three parts, which must be separated from each other. **solid basic lines**.

The first part of the table contains the basic data for manufacturing the wheel (Fig. 7), the second part contains the data for control, and the third part contains the reference data. The second part of the table contains the data for controlling the relative position of different tooth profiles according to one of the following options [29]:

– constant chord of the tooth S_c and height to the constant chord; h_c
– length of the common normal W;
– thickness along the chord of the tooth S_y and height to the chord h_{ay}

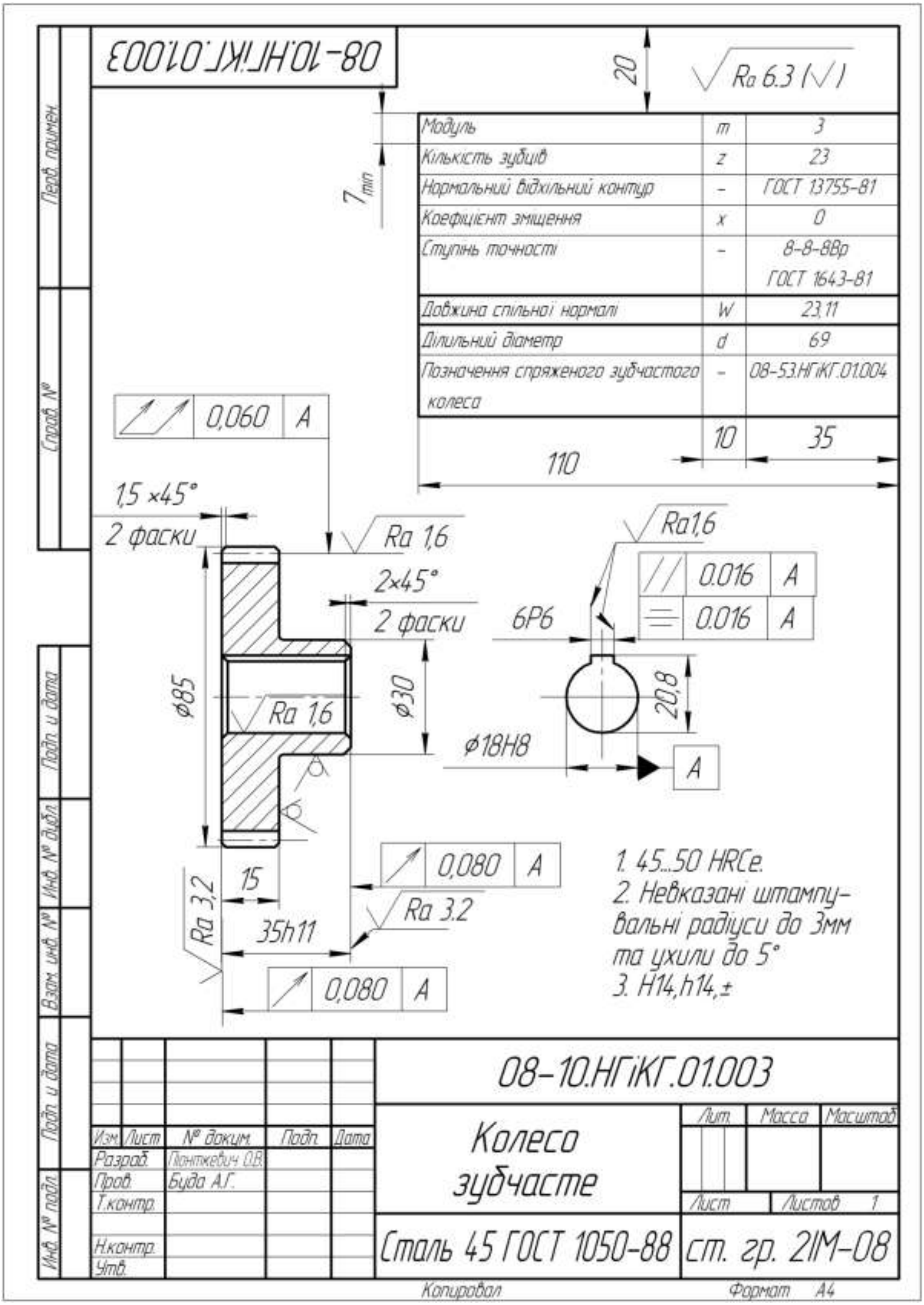


Figure 7 – Sketch of a spur gear with uncorrected cylindrical gear wheel (*spur wheel*)

If there are no special instructions in the technical specifications for the development of a gear transmission, then for the second part of the table, the second option is usually chosen and the length of the common normal for spur uncorrected cylindrical wheels is determined by the formula [1]:

$$W=mW',$$

where W' – the length of the common normal for $m=3$ mm and a given number of teeth $z=23$.

For the given example, according to Table 17 of [1], we find for $W'=7{,}7025$. Then $W=3\cdot 7{,}7025=23{,}1075$ mm $23{,}11$ mm (taking into account the points- (e.g., calipers).

The third part of the table should include: pitch diameter d; number of sector teeth – only for the toothed sector; if necessary – other reference data (for an example of the list, see GOST 2.403–75).

In educational drawings of cylindrical gears, it is recommended to provide only the drawing designation associated with a given gear from various reference data.

2.2.3.1 Execution of working drawings of gear racks

A rack-and-pinion transmission belongs to cylindrical gear or worm gears, the wheels of which have a pitch radius equal to infinity. As already noted, rack-and-pinion transmissions are designed to convert rotational motion into linear motion and vice versa (Fig. 8).

Toothed racks (*toothed racks*) can be made with spur gears (*wheels*) and helical (*helical)* both with the standard output profile and and with non-standard.

The calculation of the geometric parameters of toothed racks is performed using the formulas given in Table 2.

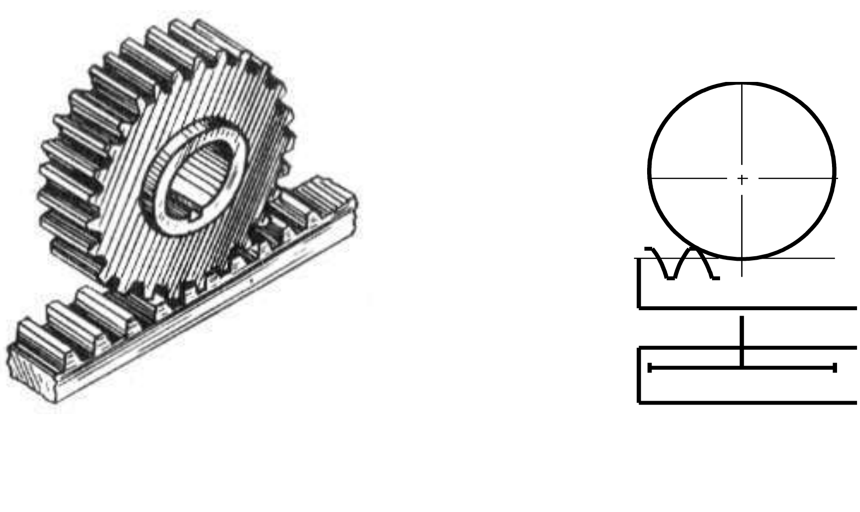


Figure 8 – Rack and pinion

Table 2 – Geometric parameters of gear racks

Marking	Parameter name	Calculation formula
α	Tooth profile angle rails	20º
β	Tooth angle	It is recommended to take $\beta \leq 20°$
m_n	Module normal	Taken as equal gear module
m_t	Main module (frontal)	$m_t = m_n / \cos\beta$
p_n	Normal step	$p_n = \pi * m_n$
p_t	End step	$p_t = p_n / \cos\beta = \pi * m_t$
h_a	Tooth head height	$h_a = m_n$
h	Tooth height (reference size)	$h = 2{,}25 * m_n$
b	Rail width	$b = (2 \ldots 10) * m_n$
b_1	Length of bevel tooth	$b_1 = b / \cos\beta$
L	Linear movement rail, which corresponds turning angle gears or worms	$L = (\gamma * p_t * z_k) / 360°$ (here z_k – number of teeth gears or number worm's activities)
γ	Wheel rotation angle, corresponding moving the rail on size L	$\gamma = L \cdot 360° / p_t z_k$

Working drawings of racks that mate with involute cylindrical gears are performed according to the interstate standard GOST 2.404–75 "Rules for the execution of rack drawings" and the requirements of the Unified System of Design Documentation (USCD).

The drawing of the gear rack (Fig. 9) must indicate: the length of the cut part of the rack; the dimensions of the chamfers or the radii of curvature of the blunting line at the edge of the teeth; the roughness of the side surfaces of the teeth.

The rack drawing also contains a rack gear parameter table, which consists of three parts, where the first part contains the basic data required for rack manufacturing, the second part contains control data, and the third part contains reference data. In the parameter table, these parts are separated from each other by main solid lines.

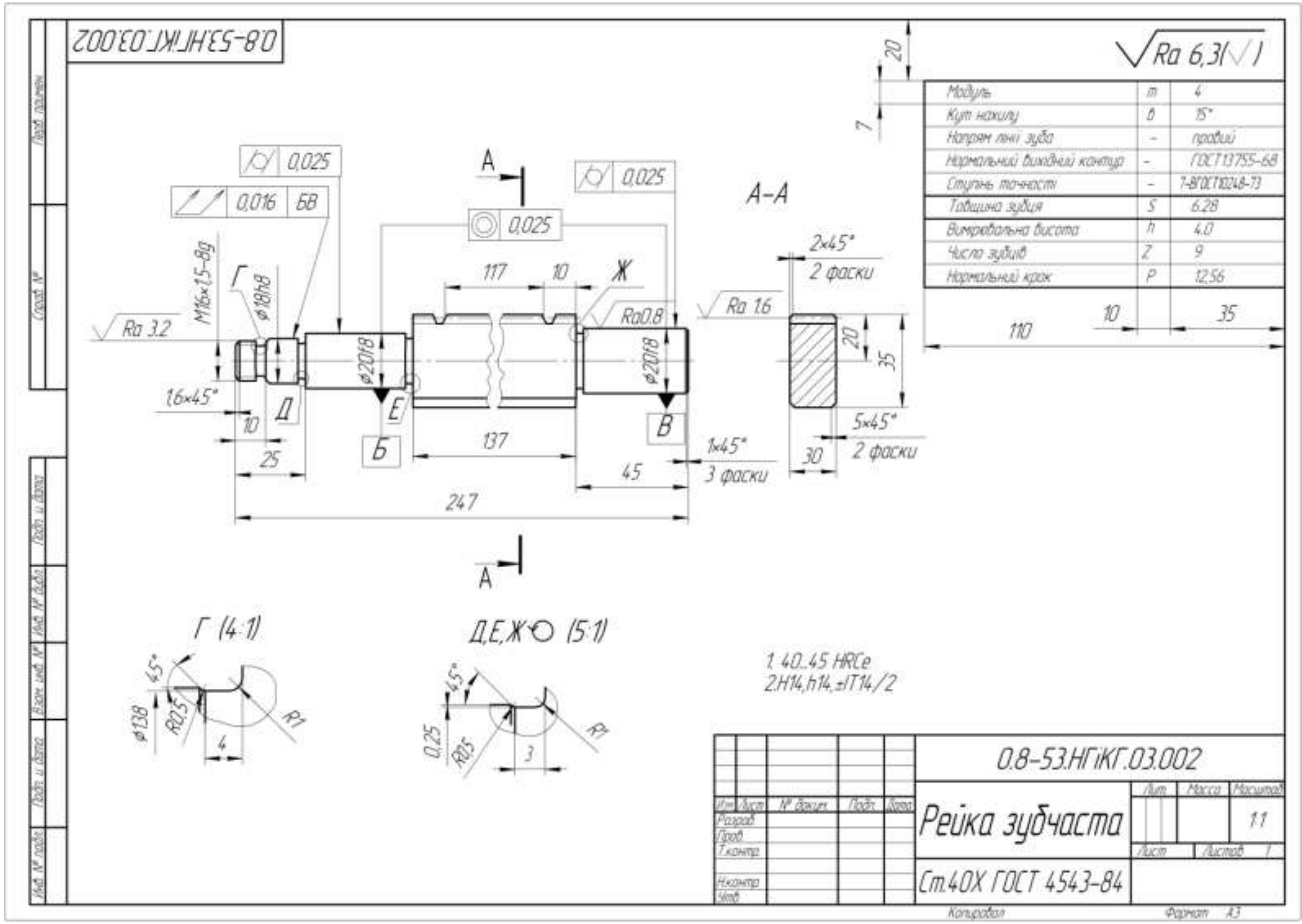


Figure 9 – Working drawing (*working drawing*) helical rack

The first part of the parameter table contains: module m; line inclination angle β bevel teeth; direction of the bevel tooth line using the inscription "Right" or "Left"; normal output contour: standard – reference to the relevant standard; non-standard with an indication of the parameters listed in GOST 2.404–75; degree of accuracy and type of mating according to the norms of lateral clearance according to the relevant standard and the designation of this standard.

The second part of the parameter table provides data for controlling the relative position of different tooth profiles, for example, tooth thickness. S_{yes} and measuring height h_{ay} or size by rollers (balls) M and roller (ball) diameter D.

If the rack is manufactured with a non-standard output contour, then in this part of the parameter table data is recorded for monitoring the standards: kinematic accuracy; smoothness of operation; contact of the teeth in the transmission; lateral clearance.

The third part of the parameter table should include: the number of rack teeth z; normal pitch; if necessary, the designation of the drawing of the conjugate gear wheel (gear) and other reference data.

Example of a working drawing of a helical rack according to normal module $m_n=4$ *mm*, by the number of teeth $z=10$ and the angle of inclination of the tooth $\beta=15^o$, shown in Figure 9.

In order to acquire skills in performing working drawings of cylindrical gear wheels, students are invited to independently calculate the geometric parameters and develop a working drawing of a gear sector, the structural diagram of which is shown in Figure 10.

Initial data for calculating geometric parameters:

$m=3$ *mm*; $z_c=25$ – number of sector teeth; $d_{in}=20$ *mm*; $\psi_{bd}=0{,}2$.

The sector teeth are spur and uncorrected. The number of sector teeth is found as the fraction of the number of teeth of the complete wheel that falls on a given sector arc.

For the given example, the number of teeth of the complete wheel is

$$z_k=360°z_c/60°=360°\cdot 25/60°=150.$$

In GOST 2.403–75 the number of teeth of a complete wheel z_k is called a number of teeth of the sector gear wheel and is included in the first part of the table of parameters of the ring gear, and the number of teeth of the sector z_c enrolled in the third part of the table.

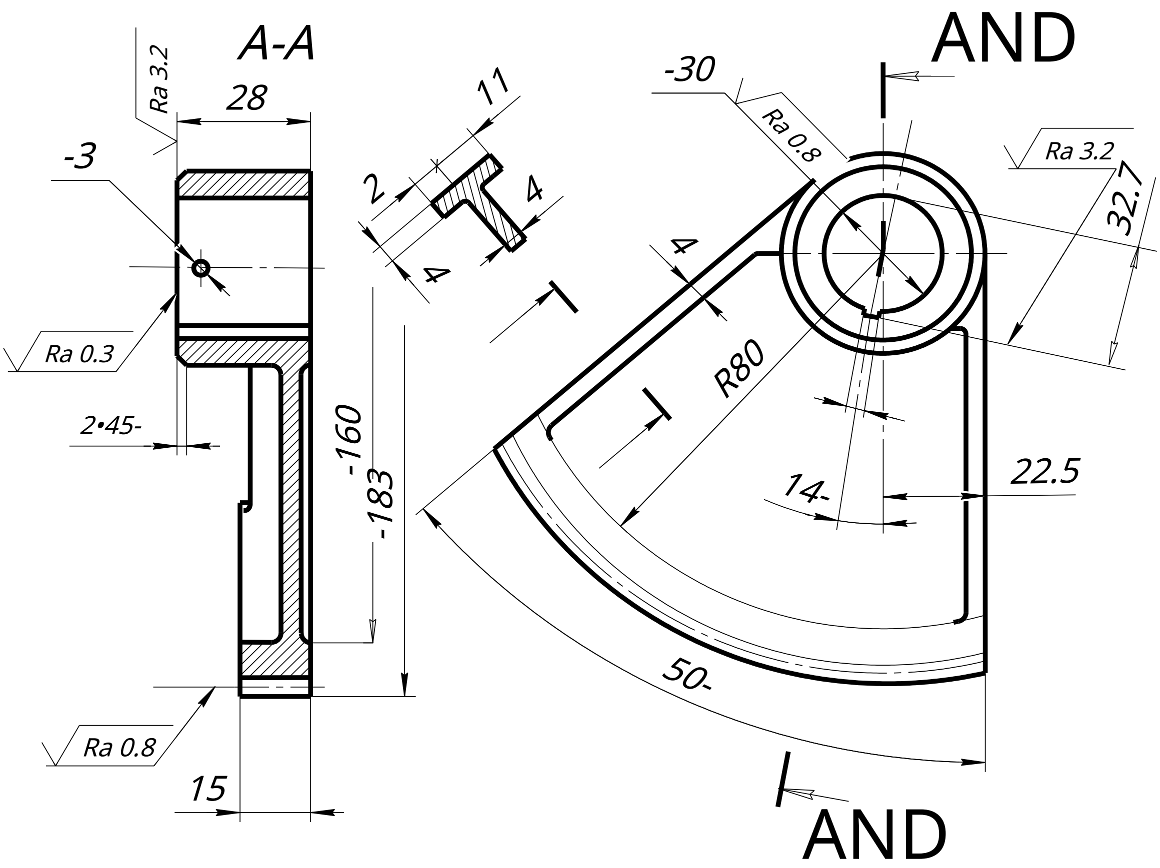


Figure 10 – Image of a spur gear sector
(*toothed quadrant*)

2.2.4 Image of gears

The conventional image of gears is established by the interstate standard GOST 2.402–68, according to which:

1) circles and generator surfaces of the tooth protrusions are shown as solid main lines, including at the point of engagement;

2) the dividing, initial and calculated circles and their generators represent dashed-dotted lines;

3) circles and generator surfaces of tooth cavities are shown in sections solid main lines, and in views – solid thin lines;

4) if the cutting plane coincides with the axes of both gears, that are in engagement, then in the section at the engagement point the tooth of the driving wheel is shown located in front of the tooth of the driven wheel (Fig. 11). In this case the radial gaps between the protrusions and depressions of the teeth that are in engagement are found as $C=c_* m_n$, where $c_*=0{,}2...0{,}25$– radial clearance coefficient;

5) teeth of gears, sprockets of chain drives and turns Worms are drawn in axial sections and cross-sections, rail teeth in transverse sections;

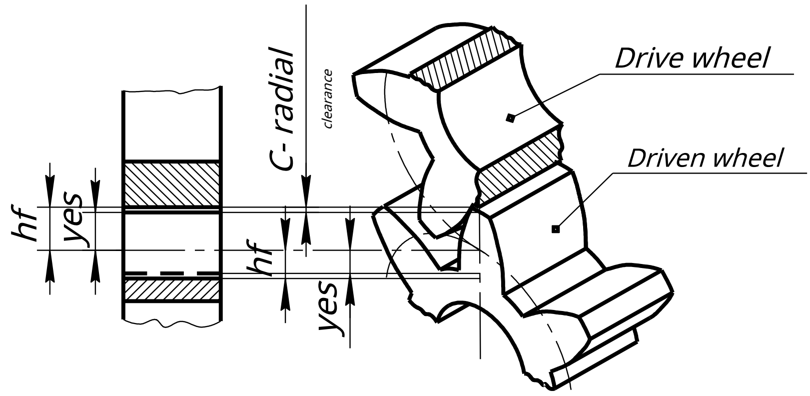


Figure 11 – Image of two teeth of cylindrical wheels in meshing

6) if you need to show the direction of the teeth of a gear wheel, rack or turns of the worm, then the image of the surface of the teeth or turns (closer to the axis) shows three solid thin lines (Fig. 12, a - c) with the corresponding slope.

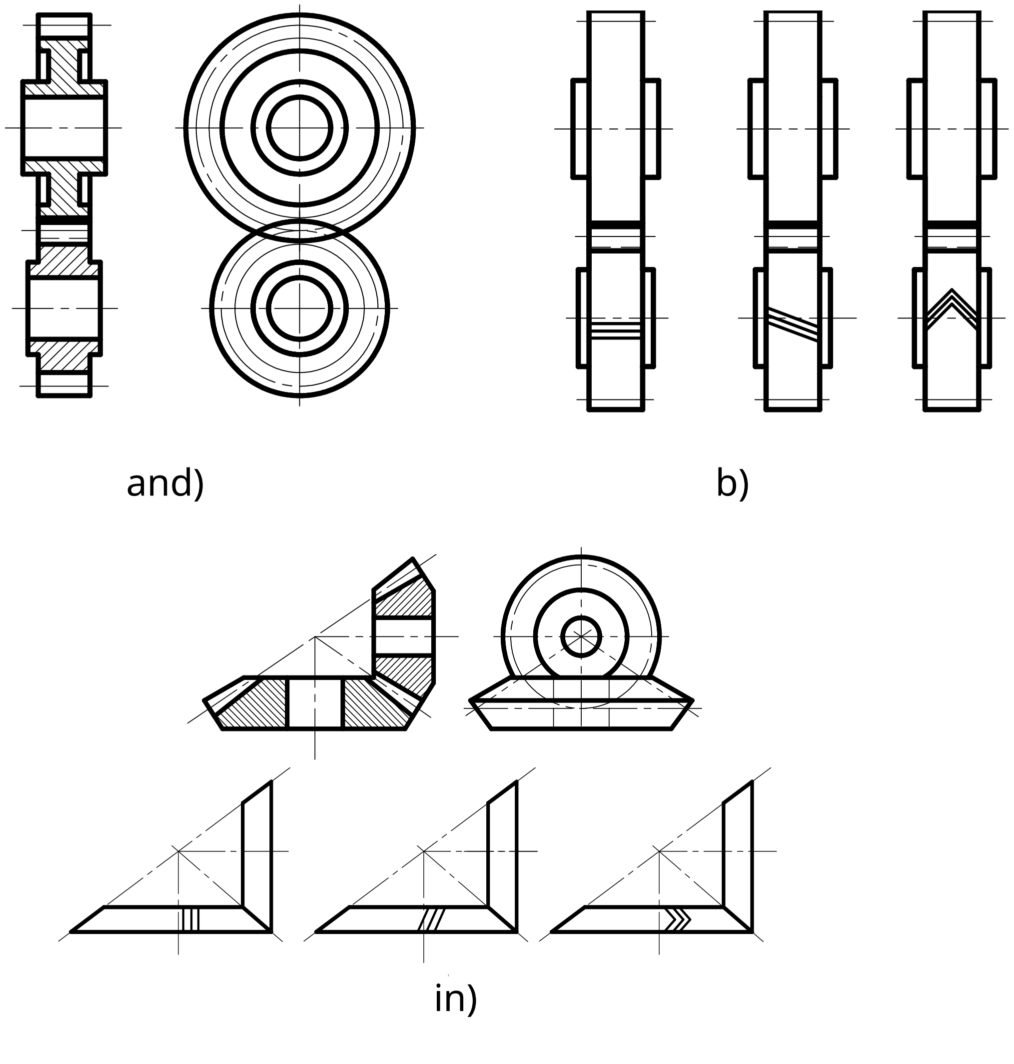
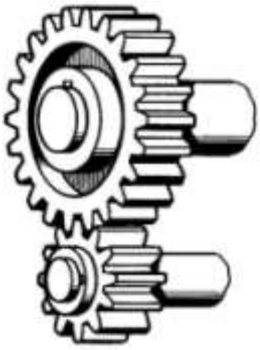


Figure 12 – Directions of teeth in engagement

2.2.5 Cylindrical gear

The transmission is formed by two meshing gears. The shaft axes are parallel to each other (Fig. 13).



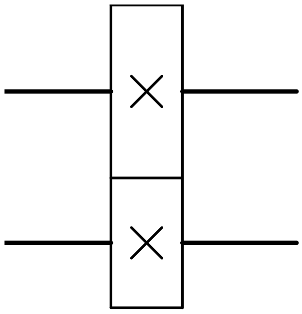


Figure 13 –Cylindrical gear (*gear transmission*)

A gear wheel that receives rotational motion from an output shaft, such as an engine (a wheel with fewer teeth) z_1), is called a gear, and a wheel with a larger number of teeth z_2 – wheel. Both wheels have the same module and the same geometric dimensions of the teeth.

Each gear is characterized by a gear ratio *and*, which for a cylindrical gear can be determined by one of the following relationships:

$$and=\omega_1/\omega_2=n_1/n_2=d_2/d_1=d\omega_2/d\omega_1=yes_2/yes_1=df_2/df_1=z_2/z_1,$$

where ω_1, ω_2; n_1 and n_2 – respectively, angular velocities and linear frequencies (min^{-1}) gear rotation (index «*1*") and wheels (index "*2*").

The initial cylinders (Fig. 14) are separated in the teeth of the heads from the legs. For uncorrected gears, the diameters of the initial diameters of the circles d_{w1} and d_{w2} (respectively, the gear and the gear wheel) coincide with the corresponding diameters of the pitch circles d_1 and d_2 (gears and cogs) wheels).

Wheelbase (*axle spacing*) a_w, which is defined as $0,5(d_{w1}+d_{w2})$, is a line AT_1AT_2, which connects the centers of rotation of the gear and the wheel. On this line there is a special point - the pole P gear and gear engagement wheel, which is the point of contact of the initial circles of the gear $d_{\omega 1}$ and wheels $d_{\omega 2}$.

Relative to the centers of circles AT_1 and AT_2 determine, respectively, the diameters of the vertices teeth d_{a1} and d_{a2} and the diameters of the depressions (*slots*) teeth d_{f1} and d_{f2}. Radial clearance *WITH* in the position of the gear and wheel as shown in the figure13, is defined as $C=c^*m_n$, where $c^*=0,25$ – radial clearance coefficient for spur gears cylindrical wheels and $c^*=0,2$ – for oblique teeth.

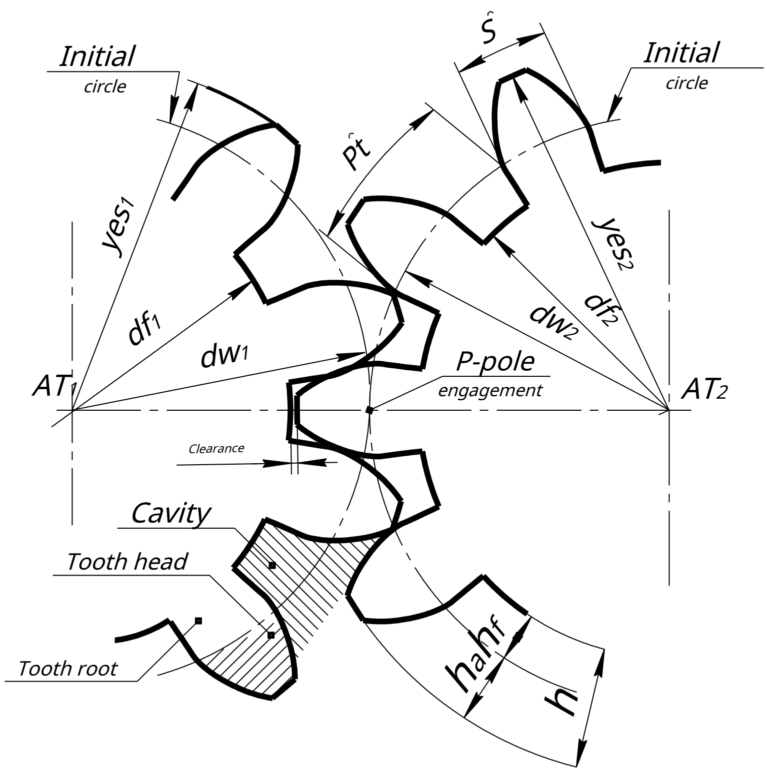


Figure 14 – Engagement (*gearing*) pairs of cylindrical wheels

To increase strength and reduce wear, the teeth are adjusted: the height of the gear head is increased by the leg, and the gear wheel is reduced. In this case, the initial circles will not coincide with the pitch circles.

2.2.6 Requirements for gear drawing

1. It is recommended to show the cylindrical gear in two images: longitudinal frontal section in place of front view and left view.

2. The initial circles of the driving and driven wheels touch each other one at a point located on the centerline. Therefore, the dividing, initial and calculation circles and their generators are depicted with dash-dotted lines.

3. At the point of engagement of the wheels, the circles of the tooth apex surfaces are depicted solid main line (see Fig. 11).

4. If the cutting plane coincides with the axes of both gears, which are in mesh, then in the section at the point of meshing the tooth of the driving wheel is shown located in front of the tooth of the driven wheel. Therefore, on the drawing, the generator of the surfaces of the vertices of the gear is shown by a solid main line, and that of the gear wheel is shown by a dashed line (see Fig. 11, 12).

5. The circles of the surfaces of the peaks and valleys of the wheels at the point of engagement are not touch, but form a radial gap (see subclause 2.2.5).

6. To depict the connection of a shaft with a gear or cogwheel keys (using spline connections, etc.) are made by making local cuts on the drawing.

7. On the assembly drawing of the gear transmission, as a rule, there is no showing chamfers, roundings on the teeth and hub, grooves for tool exit, etc.

2.2 Bevel gear

In a bevel gear, the shaft axes intersect (Fig. 15). Bevel gears come with straight, tangential, and circular (helical bevel gears) teeth. In reduction gears, the smaller of the gears is the driving gear, and the larger is the driven gear. As in cylindrical gears, the smaller of the bevel gears is called a gear, and the larger is called a wheel.

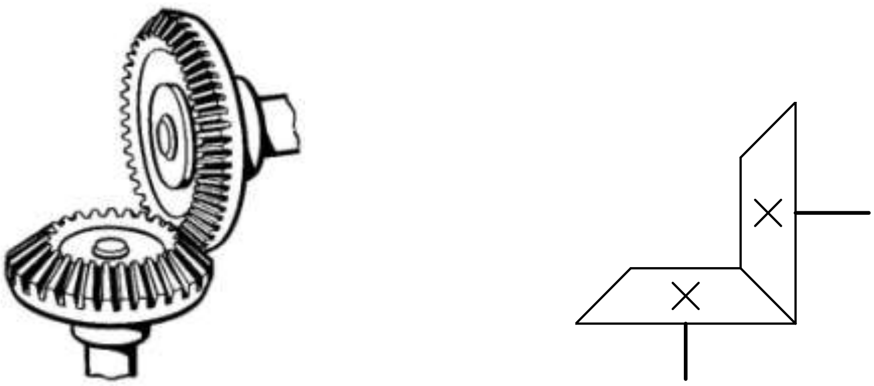


Figure 15 –Bevel gear (*toothed conical transmission*)

2.2.1 Elements of a bevel gear. Terms, definitions, designation (GOST 19325–73)

In bevel gears, all the surfaces characteristic of the wheels of these gears are cones. There are a dividing cone 2, a cone of peaks 3 and a cone of valleys 1 (Fig. 16). Designation of the angles between the axis and the generatrix of the cones: δ– angle of the dividing cone; δ_{and}– the angle of the vertex cone; δ_f– the angle of the cone of the depressions. When designing bevel gears, the angle of the tooth head is also determined. θ_a and the angle of the tooth root θ_f. The bevel gear crown is limited at the ends by two additional cones - external 6 and internal 4, the generators of which are perpendicular to the generators of the dividing cone. All sections of the bevel gear crown by planes perpendicular to the generators of the dividing cone are equal, and there can be an infinite number of such sections, therefore, when calculating bevel gears, the calculated sections are take the middle section (in the middle of the toothed crown (*0,5b*) and along the base of the dividing cone (by diameter $d_{there\ are}$).

In bevel gears, angular correction of tooth profiles is not used, but only height correction, therefore the initial and pitch cones coincide. The area of use, wheel dimensions and power of bevel gears significantly depend on the axial (in axial section) shape of the teeth of bevel wheels. According to the GOST 19326–73 standard, three axial tooth shapes are distinguished, which are denoted by Roman numerals *AND*, *II* (a, b), *III* (Fig. 17).

Form *AND* used mainly in spur and tangential bevel gears. Axial shape *II* is the main one for bevel gears with a circular tooth, and the shape *III* used in large-modulus bevel gears with circular teeth $z_{\Sigma} \geq 40$ and medium conical distances $R_m = 75 \ldots 750$ *mm* [1, 17, 27, 30].

In bevel gears with a circular tooth (preferably with an axial tooth shape *II* and *III*) the average normal modulus is taken as the standard (calculated) modulus m_{nm} (found in the middle section of the toothed crown bevel gear as $m_{nm} = p_n/\pi$, where p_n – normal step (*regular normalized*) teeth in the average cross section), if $m_{nm} < 2$ *mm*.

In cases where $m_{nm} \geq 2$ *mm*, then the calculated (standard) external end circular module is $m_{this} = d_{e2}/z_2$, here, respectively, d_{e2}-diameter the bases of the dividing cone and z_2 – the number of teeth of the bevel gear.

Circular teeth are cut with a non-modular tool, which allows bevel gears to be machined in a certain range of modules, therefore, in cases where the outer pitch diameters are standardized (d_{e2}) and cone distances R_e, the module in bevel gears can be non-standard or fractional.

One of the most important geometric parameters of a bevel gear is the outer cone distance. R_e, which is similar in meaning to the center distance and_w in a cylindrical gear transmission.

Outer cone distance (*outer cone distance*) R_e is the distance that is measured along the generator of the dividing cone from its intersection with the wheel axis to the base of the dividing cone (Fig. 16).

Distance R_e related to other geometric and kinematic parameters of bevel gears with the following dependencies

$$R_e - d_{oh}/(2\sin\delta_{and}) = m_{this} z_{and}/(2\sin\delta_{and}) = 0{,}5 m_{this} z_{\Sigma} = 0{,}5 m_{this} z_1 \sqrt{1+u^2}$$

(here *and* = *1*; *2*, respectively, bevel gears (*1*) and wheel (*2*), δ_{and} – angle the inclination of the generating cone of the wheel to its axis;

$z_{\Sigma} = \sqrt{z_1^2 + z_2^2}$ – total number of gear teeth z_1 and wheels z_2;

$u = z_2/z_1 = d_{e2}/d_{e1} = d_2/d_1$ – gear ratio, here d_2 and d_1 – diameters of bevel gears and wheels in the average cross section).

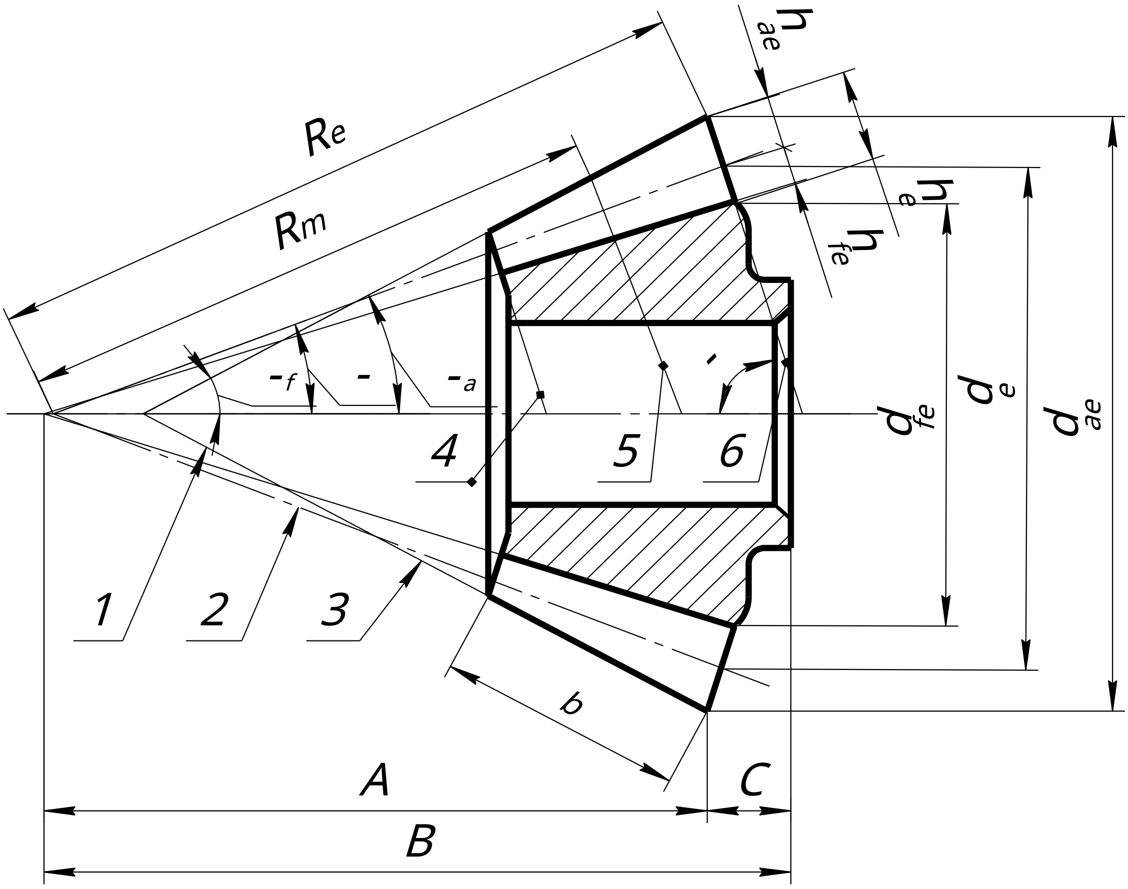


Figure 16 – Elements (*components*) bevel gear

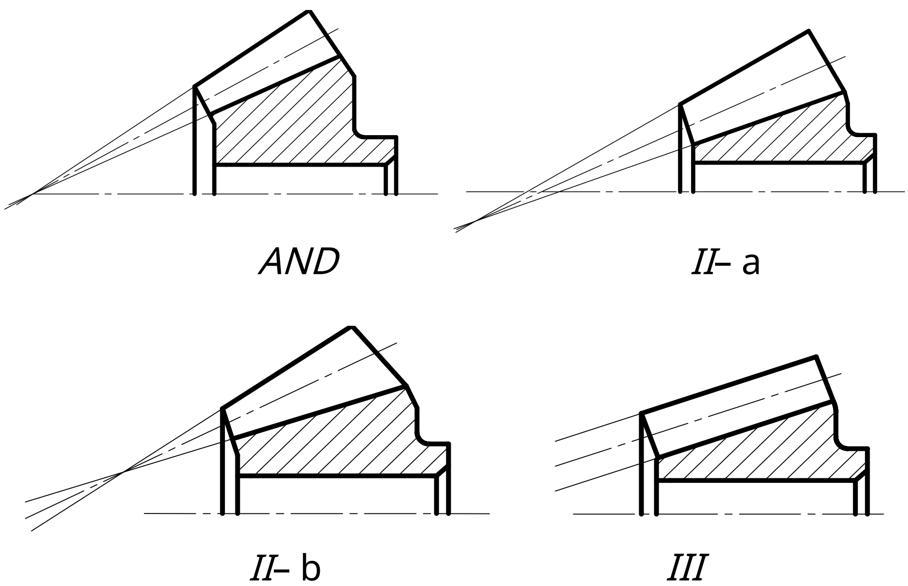


Figure 17 –Axial forms (*axial forms*) of bevel gear teeth

Width (*width)*The gear ring is defined as$b=k_{be}\cdot R_e$, where$k_{be}\leq 0{,}3$– ratio of crown width to outer taper distanceR_e
(average value$k_{be}=0{,}285$) [1, 17, 27, 30].

When calculating bevel gears, especially spur gears, the geometric parameters of bevel wheels are found in the average cross-section of the dividing cones, to which the average conical distance corresponds.

$$R_m=R_e-0{,}5b=d_{and}/(2\sin\delta_{and})=m_{this}z_{and}/(2\sin\delta_{and})=m_{tm}\cdot R_e/m_{this}$$

(herem_{tm}– average end module).

Main parameters of spur gear*and*-th (*and*=1; 2) bevel gear with axial tooth shape*AND*(orthogonal transmission – interaxial angle$\Sigma=90°$) are calculated using the formulas given in Table 3.

The design parameters of bevel gears that are not directly related to geometric parameters are determined by recommended formulas similar to the dependencies for cylindrical gears, for example: hub diameter (*nave diameter*) wheels$d_m=(1{,}6\ldots2{,}0)d_{in}$(here$d_{in}$– diameter of the shaft seating surface, on which the bevel wheel is installed); length (*length*) hub $l_m=(0{,}8\ldots1{,}5)d_{in}$; wheel rim thickness$WITH_d=(0{,}2\ldots0{,}3)b$; width of the wheel disk under the tooth depressions: on the additional cone on the side of the base of the dividing cone (end of the teeth)$S=2{,}5m_e+2$ *mm*and on the side of the top dividing cone (*reference cone*)$S'=1{,}2m_e$; chamfer of blunting the outer corners of the teeth (for any axial shape)$f_x=0{,}5m_e$(receive machining the wheel on the outer diameterd_{ae}parallel to its axis landing hole).

Working drawings of bevel gears (*toothed conic*) of the wheels are performed according to the rules established by the interstate standard GOST 2.405–75 “Rules for the execution of drawings of bevel gears”. In accordance with the requirements of this standard, the image of a bevel gear must indicate: the outer diameter of the tops of the teeth (*outer diameter of teeth*) before blunting the edge; the outer diameter of the tooth tips after blunting the edge (if necessary); the distance from the base plane to the plane of the outer wheel of the tooth tips; the angle of the tip cone (*tip angle*) of teeth; angle of the external additional cone. It is allowed to specify an additional angle to the angle of the external additional cone; width (*width*) of the gear ring. If the front end of the gear wheel is made flat-cut, then the width of the gear ring should be indicated as a reference; base distance; dimensions of the chamfers or radii of curvature of the blunting lines on the edges of the teeth (it is allowed to indicate the dimensions of these chamfers or radii in the technical requirements of the wheel drawing); position of the measuring section.

Table 3 – Parameters of a spur bevel gear

Marking	Parameter name	Estimated formula
z_{and}	Number of teeth	–
m	Average circular module	–
d_{and}	Average pitch diameter	$d_{and}=mz_{and}$
h_e	External tooth height	$h_{hey}=h_{aei}+h_{fei}=2,2m_e$
h_{aei}	External height of tooth head	$h_{oh\ my}=h_{at\ there\ are}(h_a^*=1)$
h_{fei}	External height of tooth root	$h_{fairy}=(h_a^* + C^*)m_{there\ are}$, $m_{there\ are}=1,2m_{there\ are}(WITH^*=0)$
s_{oh}	Outer circumferential thickness of the tooth	$s_{there\ are}=1,57m_{there\ are}$
d_{oh}	Outer pitch diameter	$d_{hey}=m_{there\ are}Z_{and}$
d_{aei}	Outer diameter of tooth tips	$d_{aei}=m_{there\ are}(z_{and}+2cos\delta_{and})$
d_{fei}	Outer diameter of tooth cavities	$d_{fairy}=h_{hey}+h_{aei}$
R_e	Outer cone distance	$R_e=0,5m_{there\ are}z$
R_t	Average cone distance	$R_t=R_e-0,5b$
δ_{and}	Angle of the dividing cone	$\delta_{and}=tg(\delta_1/\delta_2)=z_1/z_2$
θ_{fi}	Tooth root angle	$\delta_2=90^o-\delta_1$
θ_{ai}	Tooth head angle	$\theta_{f\ and}=arctg(h_{fei}/R_e)$
δ_{ai}	Vertex cone angle	$\theta_{ai}=arctg(h_{aei}/R_e)$
δ_{fi}	Cone angle of the depressions	$\delta_{ai}=\delta_{and}+\theta_{ai}$
A_{and}	Estimated base distance (from the top of the dividing cone to the base of the outer cone – the cone of protrusions)	$\delta_f=\delta_{and}-\theta_{fi}$

A table of the parameters of the crown gear should be placed on the working drawing (Fig. 18) or sketch of the gear wheel, which consists of three parts, separated from each other by solid main lines.

The first part of the table (Fig. 18) provides the basic data for manufacturing the wheel, the second part provides data for control, and the third part provides reference data.

The second part of the parameter table for bevel gears with a standard output contour gives the tooth dimensions in measuring section; tooth thickness along the chord $\bar{S}$ or a constant chord S_c; height to chord $\bar{h}_{and}$ or to a constant chord h_c [29].

The third part of the parameter table must include (Fig. 18): the interaxial transmission angle Σ; the module (average circular m_m – for spur gear; medium normal m_{nm} – for a wheel with tangential teeth; external circular end m_{this} – for a wheel with circular teeth); outer cone distance R_e; average cone distance R_m; average pitch diameter d; angle of the cone of the depressions (*root angle*) δ_f, outer tooth height h_e; if necessary, other reference data, e.g. external height of the tine head h_{ae} and others (GOST2.405–75).

Conventional representations of characteristic surfaces and circles in sections of bevel gears by planes perpendicular to the axes of the bevel gears are performed according to the rules of GOST 2.405–75, which are identical to the rules established for cylindrical gears (see subsection 2.2.4).

Questions for self-control

1. What are the main parameters of a cylindrical gear called?
2. What formulas are used to calculate the diameters of the pitch circle, peaks, and valleys of a gear wheel?
3. What are the geometric elements of a bevel gear called?
4. What formulas are used to calculate the elements of a bevel wheel?
5. What axial shapes of bevel gears do you know?
6. What is the difference between the conventional representations of the circles of the pitch circle, the peaks and valleys of the gear (gear wheel)?
7. What is the name of the driving wheel in gear transmissions?
8. How many parts does the ring gear parameter table consist of?
9. List the main data given in each of the three parts of the table.

10. Where is this table placed on the drawing?
11. List the main data given in each of the three parts of the table.

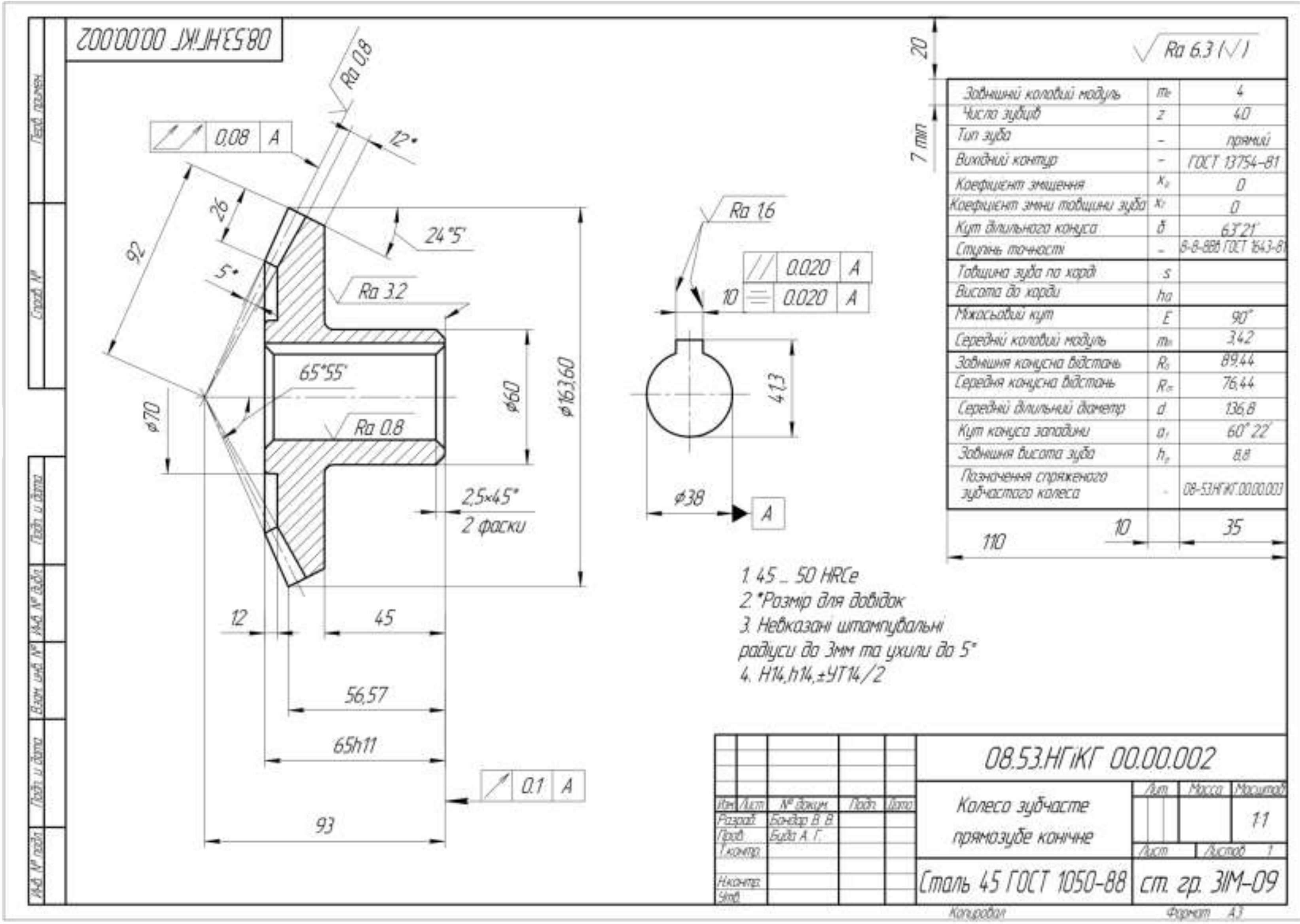


Figure 18 – Working drawing of a spur bevel gear

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2.4 Worm gear transmission

To transmit motion between shafts with intersecting axes, worm (Fig. 19), helical and hypoid gears are used. These gears are varieties of the so-called hyperboloid gear. In simple terms, a worm gear is a combination of a screw-nut gear and a cylindrical helical gear with an arched tooth shape. Most worm gears are made of a gear type, which are designed to reduce angular velocities and increase torques. The main advantage of worm gears is a large gear ratio in one stage: in power transmissions $U=8...80$, and in special cases it can reach $200...300$ and even 1000 According to GOST 18498–89, worm gears are divided into two types - cylindrical and globoidal.

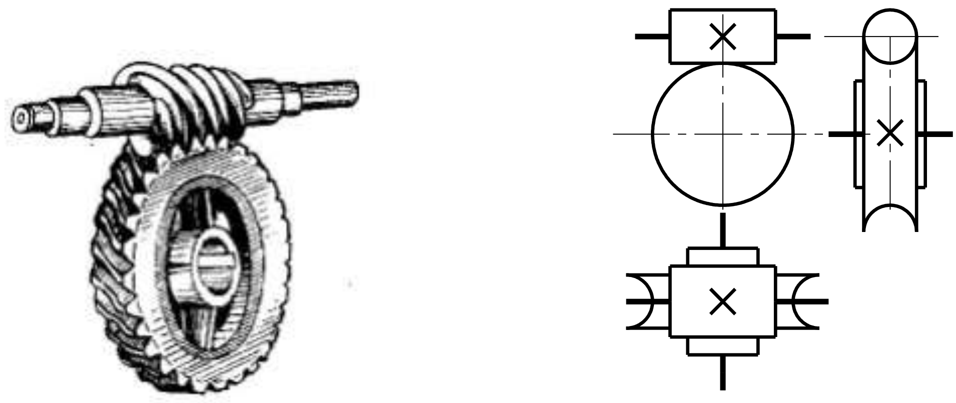

Figure 19 – Worm gear (*worm gear*)

A worm gear consists of a worm and a worm wheel. The worm (*worm*) is essentially a screw with a number of steps $z_1=1$, $z_1=2$ or $z_1=4$ and the corresponding profile of the turns in the axial and end (in the cross-section of a plane parallel to the end of the worm) sections.

By type of worm gears, worms are cylindrical and globoidal, which can be linear, when the theoretical surfaces of their turns are formed by a straight line, and non-linear, when the theoretical surfaces of their turns are formed by a curved line. Cylindrical worms can also be equal-speed, when the surfaces of the turns of different names have the same course, and different-speed in the case of unequal courses of the surfaces of the turns of different names.

Cylindrical worms are divided into Archimedean and *ZA*, involute *ZJ*, convoluted *ZN1*...*ZN3*, nonlinear *ZK1*...*ZK4* and *ZT1* and ZT2, which respectively have an Archimedean spiral profile in the end section (*of Archimedean spiral*), an involute, an elongated or shortened involute, an envelope of the generator of a cone during its helical motion relative to the worm with the axis of helical motion, which coincides with the axis of the worm and the envelope of the outer (inner) surface of the generating torus during its helical motion relative to the worm with the axis of helical motion coinciding with the axis of the worm. Worms of the type*ZT* have a concave profile in the axial cross-section, which increases the load-bearing capacity of worm gears by 30...60% compared to worm gears with conventional cylindrical worms of the type*ZA*,*ZI*,*ZN*.

Globoid worm gears, in which the dividing surface of the worm is formed by rotating a segment of the arc of the pitch circle of the paired worm wheel, which lies in the plane of its end section, which contains the centerline of the worm gear, which divides the segment of the arc of the pitch circle of the wheel in half. The dividing surface of the worm wheel is cylindrical. As a result of this design of the worm and the wheel, the length of the contact lines between the turns of the worm and the teeth of the wheel increases and the angles between the sliding velocity vector and the contact lines increase, which improves the conditions for the formation of a lubricating layer between the turns of the worm and the teeth of the wheel and increases the bearing capacity of globoid gears by 1.5 times compared to conventional ones.

The most technologically advanced are worm gears with cylindrical involute worms (*involute worms*), which determines their widespread use in technology.

Worm gears with Archimedean worms (*Archimedean worms*), despite their simplicity of manufacture (here the worm is an ordinary screw with trapezoidal thread), they are used at low speeds and loads due to significant technological difficulties in grinding such worms, which is possible only with wheels that have a complex curved profile in the axial section. End profiles (*end profiles*) of Archimedean and involute worms are shown in Figure 20.

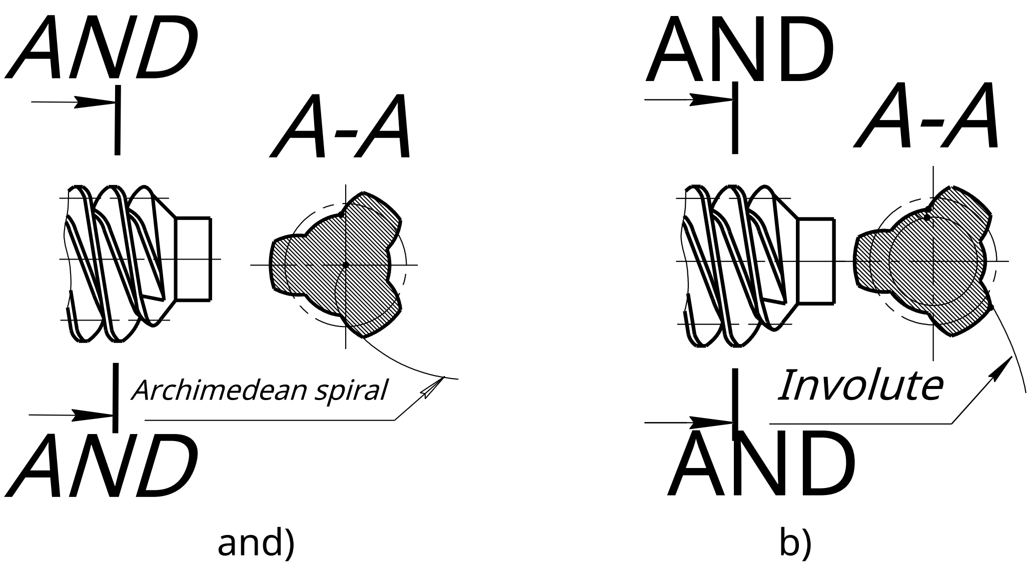

Figure 20 – End profiles of Archimedean and involute worms

2.4.1 Calculation of the geometric parameters of the worm and worm wheel

In worm gears with cylindrical worms, as in cylindrical gear gears, all surfaces are cylinders. Unlike cylindrical gear gears, the initial surfaces of the worm and worm wheel roll with high friction, which causes significant heat generation in the gear and is the reason for lower efficiency (0.7...0.9) compared to cylindrical and bevel gear gears.

As in cylindrical and bevel gears, the angle of the worm wheel tooth profile and the worm turn according to GOST 19036–81 is 20°, and for Archimedean worms this angle is determined in the axial section (α_x), in involute ones - in the normal section of the helical rack that is in mesh with the worm, in convolute ones - in the section of the plane normal to the line of the worm turn, and in nonlinear ones - this is the angle of the profile of the conical generating surface.

Standard worm gear module $m = m_{x1} \equiv m_{t2} \equiv \frac{p_{x1}}{\pi} = \frac{p_{t2}}{\pi}$, where, respectively, m_{x1} and m_{t2} are the axial and end modules, and p_{x1} and p_{t2} – axial and end steps of the worm and worm wheel.

In order to reduce the range of gear cutting tools, the pitch diameter of the worm $d_1 = mq$, where q – worm diameter coefficient – standard a quantity that is chosen in accordance with the module m according to GOST2144–76, and a number of their values are established by GOST 19672–74 (table 4).

Table 4 – Module values m depending on the coefficients q worm diameter for $z_1=1$, 2, 4

m, мм	q	m, мм	q
2	8;10;(12);12.5;16;20	(7)	(12)
2.5	8;10;(12);12.5;16;20	8	8;10;12.5;16;20
(3)	(10);(12)	10	8;10;12.5;16;20
3.15	8;10;12.5;16;20	(12)	(10**)
(3.5)	(10);(12*);(14*)	12.5	8;10;12.5;16;20
4	8;(9);10;(12*);12.5;16;20	(14)	(8***)
5	8;10;12.5;16;20	16	8;10;12.5;16
(6)	(9);(10)	20	8;10
(6.3)	8;10;12.5;14;16;20		

Notes: Values not in parentheses ( ) are preferred; only for: *$z_1=1$; **$z_1=1$ та 2; ***$z_1=2$.

Optimal value of worm diameter coefficient $q_{onm}=0,25z_2$, where z_2 – the number of teeth of the worm wheel, and the minimum $q_{min}=0,212z_2$. So narrow The limits for q are due to the fact that its increase leads to a decrease in the transmission efficiency, and its decrease leads to a decrease in the bending stiffness of the worm.

The main geometric parameters of the worm are calculated using the formulas given in Table 5.

Table 5 – Main geometric parameters of the worm

Marking	Parameter name	Calculation formula
z_1	Number of events	–
m	Module	–
q	Diameter coefficient worm	–
x	Offset coefficient	$x=\frac{a_w}{m}-0,5(q+z_2)$, $-1\leq x\leq 1$
h_{a1}	Coil head height worm	$h_{a1}=h^*_{a1}\cdot m$, ($h^*_{a1}=1$ – head height coefficient)
h_{f1}	The height of the worm's leg coil	$h_{f1}=h^*_{f1}\cdot m$, ($h^*_{f1}=1,2$ – for Archimedean, convolute and non-linear worms; $h^*_{f1}=1+2\cos\gamma$ – for involute)
h_1	Worm coil height	$h_1=h_{a1}+h_{f1}$
γ	Pitch angle worm helical line	$\gamma=arctg(z_1 \quad q)$
γ_w	Initial lift angle worm helical line	$\gamma_w=arctg\ z_1 \quad q+2x$
d_1	Pitch diameter	$d_1=mq$
d_{w1}	Initial diameter	$d_{w1}=m(q+2x)$
d_{a1}	Diameter of the tops of the turns worm	$d_{a1}=d_1+2h_{a1}=m(q+2)$
d_{f1}	Diameter of the winding cavities worm	$d_{f1}=d_1-2h_{f1}$
H_1	Worm's turn (*worm travel)*	$H_1=P_{x1}\cdot z_1=\pi mz_1$
B_1	Length of the cut part of the worm *)	$B_1\geq C_1+C_2z_2\ m$, (for x=0, and: $z_1=1\ i\ 2$, $C_2=0,06$ $z_1=4$, $C_1=12,5$; $C_2=0,09$)

*) Note: For worms that are ground and milled, the length is B_1 increased depending on the module: $m<10$ мм на 25 мм, $m=10\ldots16$ мм на $35\ldots40$ мм; $m>16$ мм на 50 мм.

Structurally, the worm is made together with the shaft. Such a part is called a worm shaft. Inserted worms are rarely made, usually for $z_1=4$.

In order to fit the worm gear into the standard center distance and facilitate the unification of body parts, worm gears can be made with a worm offset (height correction).

Displacement coefficient $-1 \leq x \leq 1$. Preference is given to positive displacement, which increases the strength of the worm wheel teeth. The offset does not affect the parameters of the worm turn, except for the length of the cut part of the worm, and for the wheel the diameters of the protrusions change d_{a2} and depressions d_{f2}. Also changes center distance transmission $a_n=0,5(d_{w1}+d_2)=0,5m(q+2x+z_2)$, where $d_{w1}=m(q+2x)$ and $d_2=mz_2$, respectively, the initial and dividing diameters worm and worm wheel.

2.4.2 Execution of working drawings of cylindrical worms and worm wheels

Drawings of cylindrical worm species *ZA*, *ZJ*, *ZN1*, *ZN2* and *ZK* and The worm wheels of worm gears coupled to them with an angular intersection of the axes of their shafts equal to 90° are performed in accordance with the requirements of the interstate standard GOST 2.406–76.

The image of a cylindrical worm must indicate the following main dimensions: diameter of the top of the turn; d_{a1} length of the cut part of the worm *in* *1*; data defining the contour of the cut part of the worm, for example, linear or angular dimensions of the chamfer, etc.; radius of curvature of the transition surface of the curve of the turn; radius ρ_{f1} of curvature of the blunting line turn or ρ_{k1} chamfer dimensions; roughness of the side surfaces of the worm.

The worm drawing (Fig. 21) contains a table of parameters consisting of three main parts: the first is the basic data of the worm; the second – data for control; the third – reference data.

The first part of the parameter table contains the following data: module *m*; number of visits z_1; type of worm – by recording by type: *ZA*, *ZJ* etc.; angle lifting the helical line of the worm's turn; the main one is for a worm γ_b of the type *ZJ* and dividing - for worms of other types; direction of the turn line - by writing "Right" or "Left"; output worm: standard - by reference to the relevant standard; non-standard - by indications of the worm parameters given in GOST 2.406–76. Degree of accuracy and type of mating according to the norms of lateral clearance according to the relevant standard and the designation of this standard.

The second part of the worm parameters provides data for controlling the relative position of different worm turn profiles according to one of the following options: pitch thickness along the turn chord S_{a1} and height to chords h_{a1}; worm size by rollers M_1 and the diameter of the measuring roller D (see GOST 2475–62 and GOST 19650–74), control size $S_{a1}=0{,}5\pi m \cos\gamma$.

In the third part of the worm parameters table, the pitch diameter of the worm and the worm turn stroke, center d_1 distance, must be specified. P_{z1} worm gear distance; worm diameter coefficient; worm turn height; a_w q number of teeth of the coupled worm wheel; main worm diameter – for h_1 z_2 worms of the type ZI; designation of a conjugate worm wheel. d_b

Example of a working drawing of a worm shaft of a worm gear with a gear ratio $u=20$, the number of worm steps $z_1=2$, module $m=4$ мм, the displacement coefficient $x=0$ and optimal worm coefficient diameter $q=0{,}25z_2=0{,}25\cdot 40=10$ (here $z_2=z_1\cdot 4=2\cdot 20=40$), shown in Figure 21 (worm of the type ZA).

The main parameters of the worm are calculated according to the formulas in Table 4, the dimensions for control S_{a1} i h_{a1} mutual position of the winding profile worms found by formulas $S_{a1}=S^* m\cos\gamma$ and $h_{a1}=h_a^* m+0{,}5S_{a1} tg\ 0{,}5 arcsin\ (S_{a1}\sin^2\gamma)/d_1$ (here $S^*=1{,}571$ – coefficient calculated thickness of the turn), given in GOST 19650–74, and the design parameters of the worm (diameter and length of the output end, diameters and lengths of the intermediate sections of the worm shaft) are determined according to the design recommendations of the works [1, 17, 27, 37]. An important note - the diameter and length of the output end of the worm shaft must necessarily be brought into line with the dimensions given in GOST 12080–66.

Worm wheels (*worm gears*) in order to save on expensive anti-friction materials (*antifriction materials*) made of non-ferrous alloys (various grades of brass and bronze), are manufactured in prefabricated units - the gear ring is made of anti-friction material, and the center is made of steel or cast iron. The following typical worm wheel designs are used: banded design (*reinforced construction*) (Fig. 22, a), in which a gear ring made of antifriction material is fitted with tension on a steel or cast iron center and is additionally fixed with threaded pins installed at the junction of the surfaces of the ring and the center; a bandaged bolted structure, in which the gear ring is made with a flange that is attached to the center with special bolts (Fig. 22, b); bimetallic (*bimetal construction*) design (the most rational for serial and mass production) (Fig. 22, c), in which the gear ring is formed by casting in a mold with a prepared center previously installed in it.

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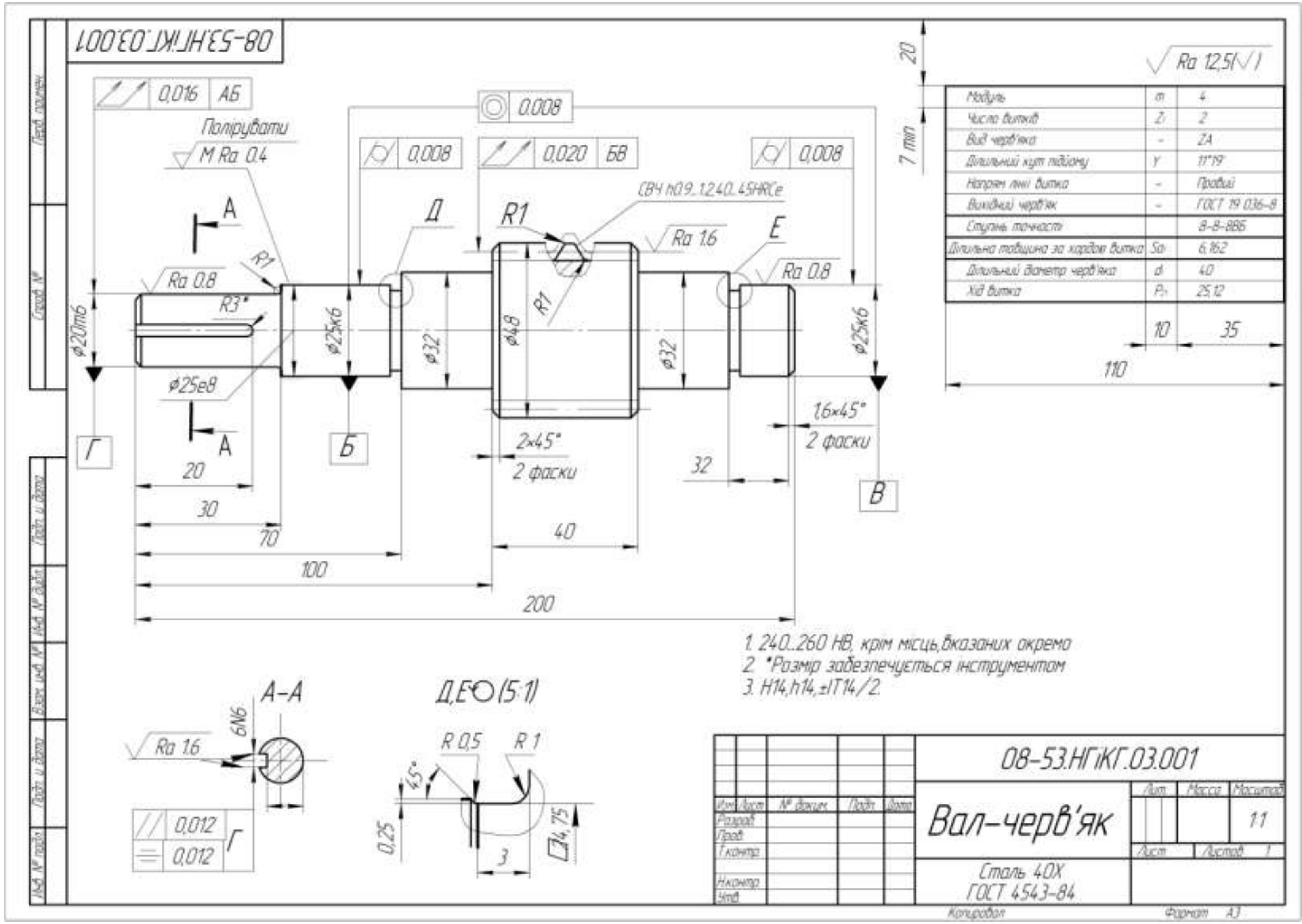
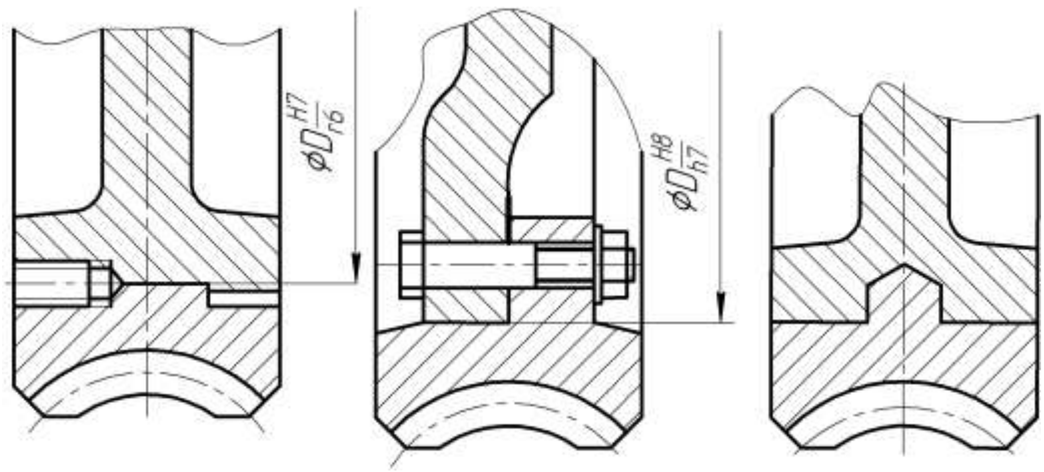


Figure 21 – Working drawing of the worm shaft

Worm wheels are made entirely of anti-friction non-ferrous alloys with a diameter of $d_{a2} \le$ 100 mm (here d_{a2} – the diameter of the tops of the wheel teeth).



and) b) in)

Figure 22 – Typical worm wheel designs

The geometric parameters of the worm gear ring are calculated using formulas in accordance with the requirements of GOST 19650–74, given in Table 6.

Table 6 – Geometric parameters of the worm gear ring wheels

Marking	Parameter name	Calculation formula
1	2	3
z_2	Number of worm wheel teeth	$z_2 = uz_1$ rounded to to the nearest whole number, after which it is specified gear ratio *you*
a_w	Wheelbase	$a_w=0,5(d_{w1}+d_2)=$ $=0,5(z_2+q+2x)m$
x	Worm displacement coefficient	$x=\frac{a_w}{m} - 0,5(z_2+q)$; $-1\le x\le 1$
m	Module	–
h_{a2}	Tooth head height	$h_{a2}=h^*_{a2}\cdot m$ (here $h^*=1$– height coefficient heads)

Continuation of table 6

1	2	3
h_{f2}	Tooth root height	$h_{f2}=(h^*_{a2}+c^*)$, ($c^*=0,2$ – radial coefficient gap)
h_2	Tooth height	$h_2= h_{f2}+ h_{a2}= 1 2m$
d_2	Worm wheel pitch diameter	$d_2=m\cdot z_2$
d_{a2}	Diameter of tooth tips worm wheel in the middle section of the width of the gear ring IN_2	$d_{a2}=d_2+2h_{a2}=m(z_2+2)$ – for x=0; $d_{a2}=m(z_2+2+2x)$ – for $x0\neq$
d_{f2}	Tooth cavity diameter worm wheel in the middle section of the width of the gear ring B_2	$d_{f2}=d_2-2h_{f2}=m(z_2-2,4)$ – for $x=0$; $d_{f2}=m(z_2\ \ 2,4+2x)$ – for $x\neq 0$ and worm gears ZA,ZN and ZK; $d_{f2}=m(z_2-2-0,4cos\gamma+2x)$ – for $x0$ and $\neq$ transmissions from worms ZJ
p	Pitch of teeth worm wheel	$p=\pi\cdot m$
in_2	Width of the worm gear ring	$в_2\leq 0,75d_{a1}$ – for $z_1=1$; 2 and $в_2\leq 0,67d_{a1}$ – for $z_1=4$
2δ	Conditional angle of engagement of the teeth of the worm wheel of the worm	$\delta=arcsin(\frac{в_2}{d_{a1}0,5m})$
$d_{aм2}$	The largest diameter of the worm wheel tooth tips	$d_{aм2}=d_2+d_1(1-cos\ \delta)\leq$ $d_{a2}+\frac{6\cdot m}{z_1+2}$ If the inequality is not fulfilled, then the diameter is determined by the right part of this inequality
R_{a2}	Radius of the notch of the tooth apex surface of the gear wheel	$R_{a2}=0,5d_1-h_{a2}$
R_{f2}	Radius of the recess surface of the worm wheel tooth cavities	$R_{f2}=0,5d_1-h_{f2}$

The design parameters of worm wheels that are not directly related to geometric parameters are determined by recommended formulas, similar to the dependencies for cylindrical and bevel gears, for example, for a banded wheel (Fig. 23) of a typical scheme (Fig. 22, a): hub diameter
$d_M = 1{,}2 \ldots 1{,}6\ d_b$ (here d_b – diameter of the shaft seating surface on which install a worm wheel); hub length $l_M = 0{,}8 \ldots 1{,}5\ c$;
$c = 1{,}2 \ldots 1{,}3\ S_ц$ – thickness of the wheel center disc; $S_ц = 1{,}2 \ldots 1{,}3\ S$ – thickness center rim; $S_0 \approx 25m$ – worm gear rim thickness wheels along the bottom of the tooth cavities in the middle section of the width of the gear ring.

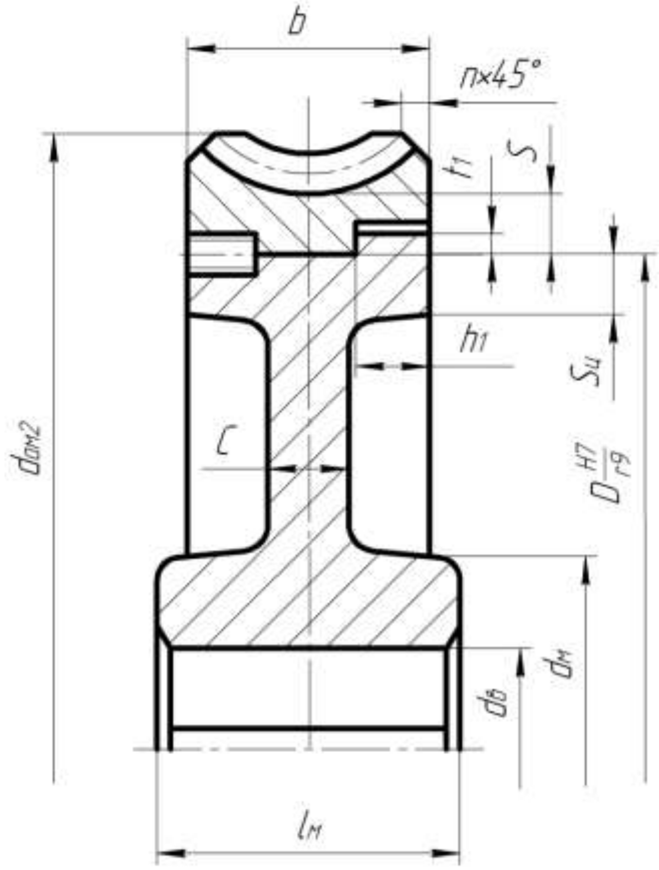


Figure 23 – Design parameters (*constructive parameters*) worm wheel

When performing working drawings of worm wheels, which, depending on the type of design (Fig. 22), can be performed as assembly drawings (see Fig. 22, a, b) or as working drawings of parts (see Fig. 22, c). The following main dimensions must be indicated on the image of the worm wheel: diameter of the tops of the teeth; largest diameter d_{a2}; width of the crown; data d_{aM2} that determine B_2 the contour of the wheel crown, for example, dimensions of the chamfer or radius of rounding of the end edges of the teeth, radii recesses (*flute radius*) the surfaces of the tops of the teeth of the wheel, etc.; the distance from the base end to the middle end plane of the wheel and, if necessary, to the center of the notch of the transition curve of the tooth ρ_{f2}; radius curvature of the tooth blunting line ρ_{k2} or chamfer dimensions; roughness lateral surfaces of the teeth.

The worm wheel parameter table, which is placed on its drawing, consists of two parts, where the first part contains the main data of the wheel necessary for its manufacture, and the second part contains reference data. These parts of the table are separated by a main solid line.

The first part of the table of parameters of the worm gear ring should include: module m, number of teeth z_2, and for a sector worm gear, the direction of the tooth line is indicated by the inscription "Right" or "Left"; worm displacement coefficient x; initial worm gear: standard – with reference to the relevant standard; non-standard – with indications of the worm parameters given in GOST 2.406–76; degree of accuracy and type of mating according to the lateral clearance norms according to the relevant standard and the designation of this standard.

The second part of the table of parameters of the worm gear crown provides the following data: center distance a_w; pitch diameter d_2 of the worm wheel; number of teeth of the sector (for a sector worm wheel); number of steps of the conjugate worm z_1; designation of the drawing of the conjugate worm; if necessary, other reference data, for example, center distance during wheel machining a_0, etc.

An example of the design of a working drawing of a worm wheel made of a bimetallic blank (see Fig. 22, c) coupled with a worm is shown in Figure 24.

Questions for self-control

1. What are the main parameters of a worm and worm wheel called?
2. What formulas are used to calculate the diameters of the pitch circle of a worm and a worm wheel, the diameters of the vertices and valleys of a worm and a worm wheel?
3. How do the drawings show conventional images of cylindrical, conical, worm wheels, a worm, and a rack?
4. In what units is the gear module expressed?
5. Are the gear teeth (wheels) shaded in the sections?
6. What teeth are there on a cylindrical gear wheel and a bevel gear?
7. What line should be used to show the gear tooth at the point of engagement?

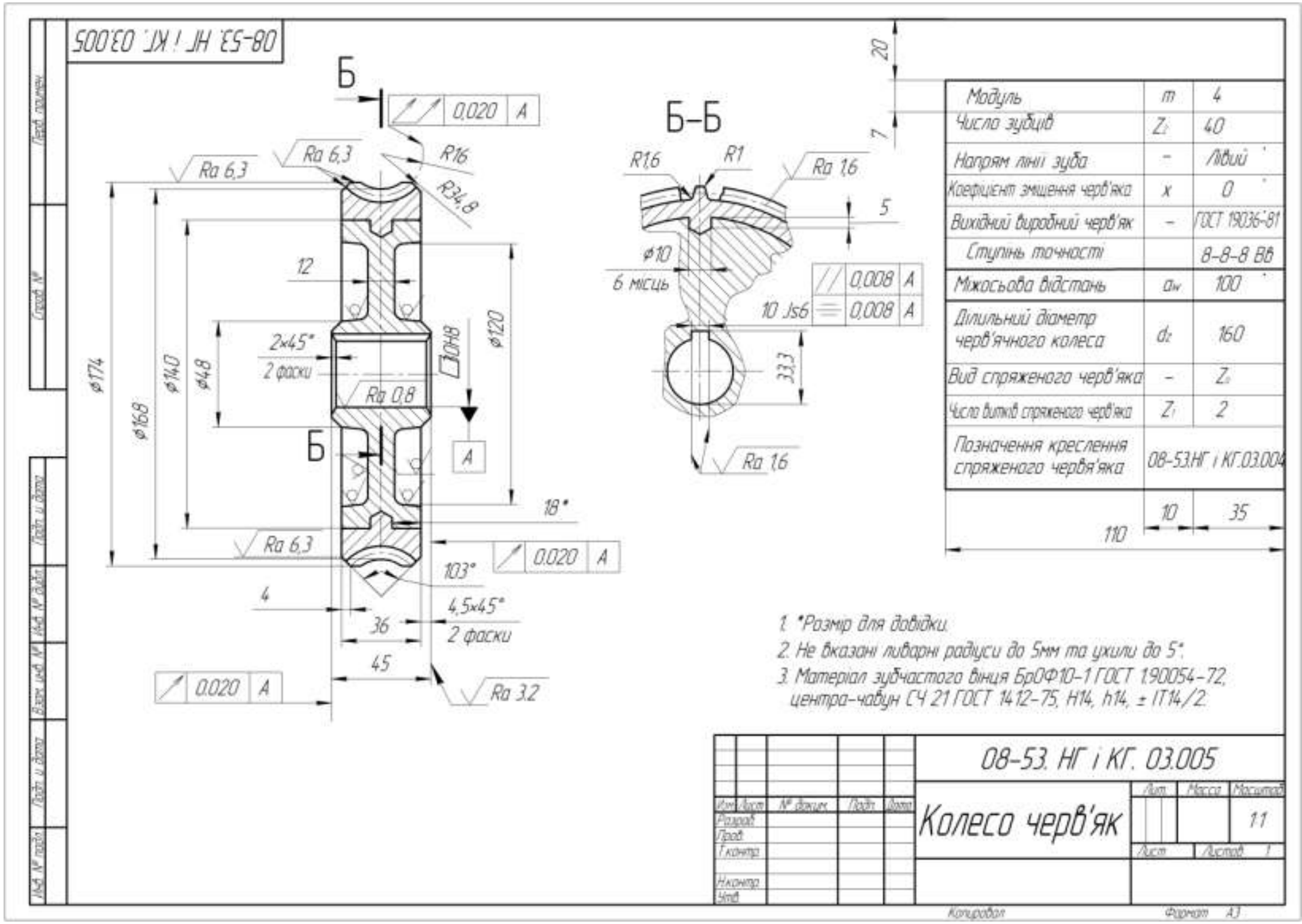


Figure 24 – Working drawing of a worm wheel (*worm gear*)

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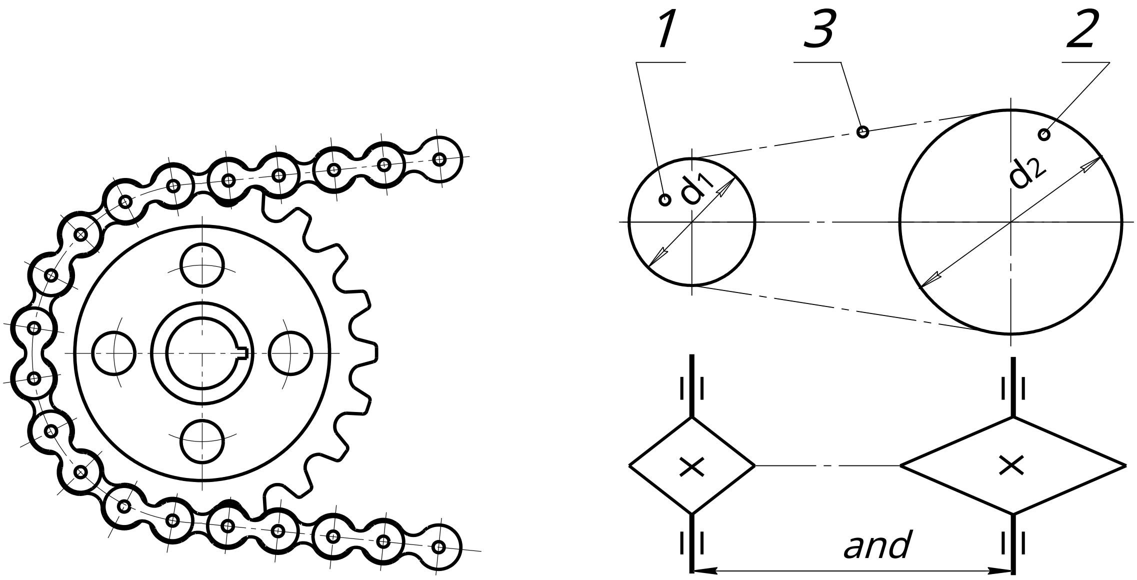
3 CHAIN TRANSMISSIONS

Chain transmission (Fig. 25, a), the kinematic diagram (Fig. 25, b) of which consists of a driving 1 and a driven 2 sprockets (*drive and driven sprockets*), and links 3, the links of which engage with the teeth of the sprockets (d_1,d_2– pitch diameters of the driving and driven sprockets,a –center distance).



and) b)

Figure 25 – Chain transmission (*chain gear*)

Chain transmissions are used at medium center-to-center distances, where the use of gear transmissions requires intermediate stages or parasitic gears, if there is a strict requirement to obtain the desired gear ratio. Chain transmissions are also used instead of belt transmissions in cases where there are requirements for transmission dimensions and the operation of mechanisms without slipping.

Chain drives are widely are used in rural household and lifting and transport vehicles, oil drilling equipment, cars, motorcycles, bicycles, etc.

Drive chains are made of bushings, roller and toothed. The main geometric characteristics of the chains are the pitch, a multiple of one inch, and the width, and the main strength characteristic is the breaking load, which is established experimentally. Roller and toothed chains are standardized. They are made, respectively, roller chains according to GOST 13568–75, and toothed chains according to GOST 13552–81.

Drive roller chains can be manufactured in single, double and multi-row configurations. The traction capacity of multi-row chains is approximately proportional to the number of rows.

3.1 Calculation of geometric parameters of chain drives

The main geometric parameters of chain drives with plate-roller and toothed chains, depending on the chain pitch, are calculated P_t using the formulas given in Table 7.

Table 7 – Basic geometric parameters of chain drives

Marking	Parameter name	Calculation formula
1	2	3
z_1	Number of teeth driving sprocket	$z_{1min}=29-2U\geq 13$ where – U gear ratio transmission
z_2	Number of teeth driven sprocket	$z_2=z_1U$
d_1	Pitch diameter driving sprocket	$d_1=P_t\ /sin(\frac{180°}{z_1})$
d_2	Pitch diameter driven sprocket	$d_2=P_t/sin(\frac{180°}{z_2})$ or $d_2=d_1U$
d_{a1}	Diameter of the peaks teeth of the driving gear asterisks	$d_{a1}=P_t\ 0{,}5+ctg(\frac{180°}{z_1})$ – for a plate-roller chain; $d_{a1}=P_tctg(\frac{180°}{z_1})$ – for toothed chain
d_{a2}	Diameter of the peaks driven teeth asterisks	$d_{a2}=d_{a1}U$
d_{f1}	Diameter of the depressions teeth of the driving gear asterisks	$d_{f1}=d_1-2r$ – for plate-roller chains, where $r=0{,}5025d_1+0{,}05$ мм (here d_1 is the diameter of the chain roller); $d_{f1}=d_1-\ 2h/cos(\frac{180°}{z_1})$ – for a toothed chain, here h – tooth height

Continuation of table 7

1	2	3
d_{f2}	Diameter tooth cavities driven sprocket	$d_{f2}=d_{f1}U$
a_{onm}	Optimal interaxle distance	$a_{onm}= 30\ldots50\ P_t$
z_3	Number of links chains	$z_{3p}=\dfrac{2a_{onm}}{P_t}+0{,}5z_\Sigma+$ $+\ z_\Sigma+\Delta z/(2\pi)^2\cdot P_t/a_{onm},$ where $z_\Sigma=z_1+z_2;\ \Delta z=z_2-z_1$ found value z_{3p}, usually, rounded to the nearest even number according to the condition $z_{3p}\geq z_3$ (index «*p*» means "calculated")
a_y	Refined interaxle distance	$a_y=0{,}25P_t\left[z_3-0{,}5z_\Sigma+\sqrt{(z_3-0{,}5z_\Sigma)^2-8\left(\dfrac{\Delta z}{2\pi}\right)^2}\right]$

An example of a working drawing of a sprocket with a plate-roller chain PR–19.05–3180 GOST 13568–75 is shown in Figure 26.

Questions for self-control

1. What types of programs do you know?
2. For what purpose are cylindrical, bevel, worm and chain gears used?
3. What are the types of tooth engagements in gears?
4. What end profiles of worms do you know?
5. Name the geometric elements of chain drives.

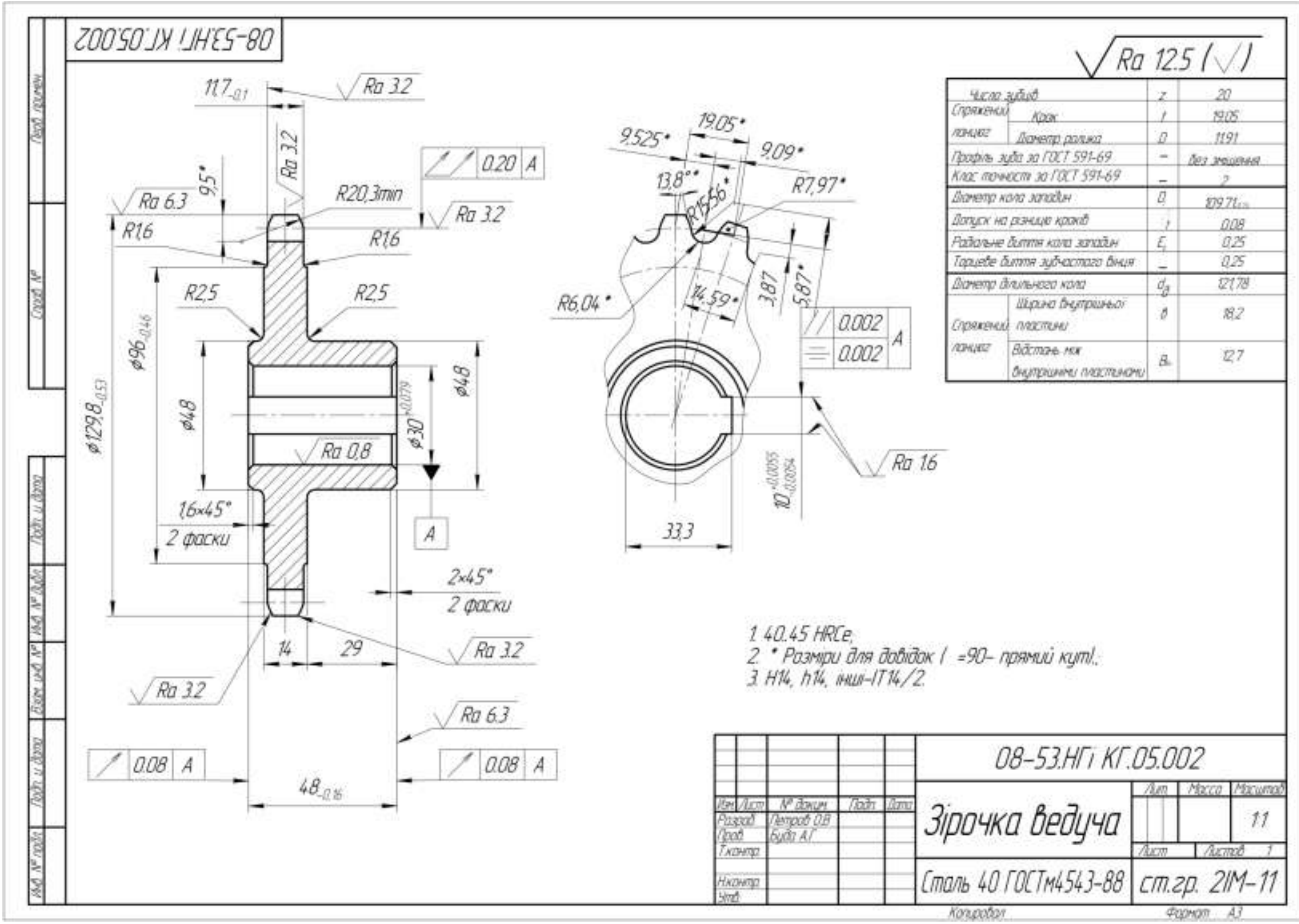


Figure 26 – Working drawing of a drive sprocket with a plate-roller chain

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4 MANUFACTURING METHODS AND MATERIALS OF CYLINDRICAL, CONICAL AND WORM GEAR

Cutting the teeth of cylindrical, bevel, and worm gears, depending on the required degree of transmission accuracy, is performed by copying or running-in (wrapping) method.

By the copying method, a depression is formed between the teeth with a tool, the cutting part of which has the shape of a depression profile of the appropriate size. The copying method is implemented in various ways: with modular, disk and finger cutters, special broaches; special grinding wheels. The wheel blank is installed in a fixture equipped with a whitening head, which, after the formation of the depression, rotates the blank to the desired angular step.

The copying method is used to produce gears for low-precision transmissions, mainly in repair shops, and also to cut large-module chevron gears. A significant disadvantage of the copying method is also the need to have a large set of tools for each module (8...26 pcs.) for the manufacture of wheels with different numbers of teeth.

Gears and worm wheels of high accuracy grades (1...9) are manufactured by the rolling (wrapping) method. During rolling, the tool and workpiece move like wheels in a gear. This movement is provided by the kinematics of the gear cutting machine, which also gives the tool additional movement, during which the cutting process takes place. Gears and worm wheels are cut by the wrapping method using worm cutters, comb-racks and gouge-gears, etc.

With the overturning method, the tooth cutting process is continuous, which provides higher accuracy and productivity compared to the copying method, and the advantage of the overturning method is the possibility of using wheels and gouges for cutting teeth (*jointers*) a rack tool with a rectilinear profile of teeth that are profiled along the contour of the generating (tool) rack.

Figure 27 shows examples of manufacturing methods: a) finger milling cutter (*worm cutter*); b) a dowser (*jointer*); c) worm milling cutter (*worm mill*); d) rail (*rack*). These methods are among the most productive running-in methods.

The teeth of cylindrical and bevel gears can also be manufactured by the rolling method (pressure treatment), in which the strength of the teeth increases by 15...20% due to cold working [17, 27, 30].

The teeth of precision cylindrical and bevel gears are ground and lapped.

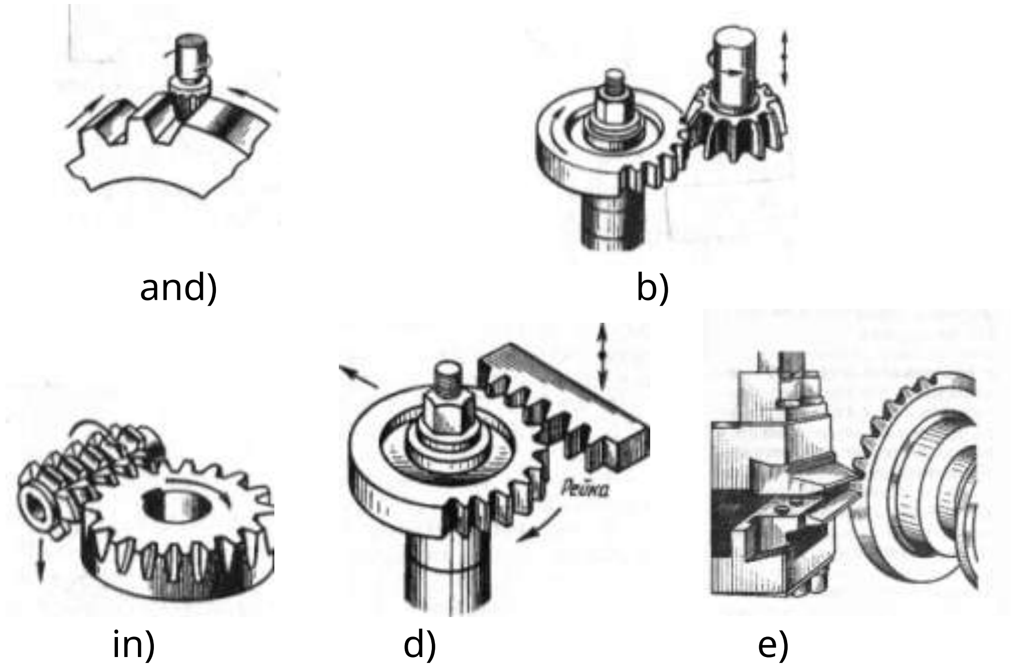


Figure 27 – Manufacturing methods (*ways of producing*) gears

In order for gears to maintain their operability during a given service life, the materials from which the wheels are made must provide the bending strength of the teeth, contact chipping of their surface layers and resistance to seizing (molecular adhesion). The main materials in mechanical engineering for the manufacture of gears of cylindrical and bevel gears are structural carbon and alloyed heat-treated steels. This is due to the fact that the permissible contact stresses in the teeth of the wheels are proportional to the hardness of their material, and the bearing capacity of the gears in terms of contact strength (endurance) is proportional to the square of the hardness [17, 27, 30]. The required hardness of the surfaces of the teeth of steel wheels is provided by the appropriate TO (thermal) or CTO (chemical-thermal treatment).

Depending on the load and requirements for the dimensions of cylindrical and bevel gears with steel wheels, the following types of thermal or chemical-thermal treatment and materials are used:

– normalization (N), which is used for lightly loaded wheels gears. Normalized wheels are made of structural carbon steels 40, 45, 50, 40X, etc.;

– improvement (P), which is applied to gear wheels, which manufactured in single and small-scale production in the absence of strict conditions for the dimensions of the transmission. The main materials are 40, 45, 40X, 50G, 30XG and others. Hardness of the tooth surfaces (200…320NV);

– volume hardening, used for medium-loaded gears. Such wheels are made of steels 45, 40X, 40XN, whose teeth are hardened to a hardness of 45...50HRCe. Currently, this type of TO is rarely used, since bulk hardening does not preserve the tough core of the tooth, which significantly reduces its bending strength;

– surface hardening by high-frequency current (MHF), which is widely used for wheels of medium-loaded gears in machine tools. The main materials are steels 40X, 40XN, 58, 45RP and others, which are subjected to TO and microwave. The hardness of the hardened layer with a thickness of 0.5...1.5 mm is usually 50...55 HRCe;

– cementation – HTO, which consists in saturating the tooth surface with carbon with subsequent hardening and low tempering. This type of HTO is used for heavily loaded wheels, gears operating under variable and shock loads. Cemented wheels are made of steels 20X, 12XNZA, 20XNM, 18X2M4MA, 18GT, 25XGT, 15XF and others. The hardness of the tooth surfaces after cementation is 58...64 HRCe.

– nitro-cementation (cyanidation) – simultaneous saturation of surfaces of steel wheel teeth with carbon and nitrogen in a gas environment with subsequent quenching and low tempering, which provides the teeth with high strength, wear resistance and resistance to seizing; the same steels are used as during carburizing. The hardness of the tooth surfaces is 60...64 HRCe.

– nitriding – HTO by saturating the surface of the teeth with nitrogen during which provides particularly high hardness and wear resistance of the tooth surfaces. The disadvantage of nitriding is the small thickness of the hardened layer, not more than 0.2...0.5 mm, which excludes grinding of the tooth surfaces and operation of nitrided wheels under shock loads due to the danger of cracking of the hardened layer and during intensive wear due to rapid abrasion of the nitrided layer. The hardness of the tooth surfaces is 550...750HV. The best nitriding is achieved by alloyed steels containing aluminum, for example 38X2KIUA, and now aluminum-free steels of grades 40XFA, 40X, 40XNA are also used, which are nitrided to lower hardness, but higher toughness;

– laser hardening, which provides hardness for unalloyed steels tooth surfaces up to 64HRCe. The advantages of this type of treatment are surface hardening, no warping, and a high degree of process automation. The disadvantage of laser hardening is its slowness.

In order to prevent seizing, which is typical for highly loaded gears, the gears of such gears are made of heat-resistant steels of grades Х3НВМ2Ф, 20Х3НВФА, 16Х3НВФМБ, etc. Such gears are also lubricated with greases with extreme pressure additives.

Large diameter gears are often made by casting from steel 35L...50L, 40HL, 30KhGSL, 50G2 and others, which are mainly subjected to tempering and normalization. Wheels of large-sized, open cylindrical and bevel gears are made from cast iron grades SCh20... SCh35, and high-strength magnesium cast irons with spheroidal graphite are also used.

Plastic gears in combination with metal ones are used in low-load gears for the purpose of noiselessness, self-lubrication or chemical resistance. The main materials are textolite of PT and PTK brands, chipboard of DSP-G brand with star-shaped at 25 °C...30 °C, polyformaldehyde resins, caprolon and phenylone, etc.

Worm gears, as already noted in section 2, are a combined type of transmission that operates with high heat generation, which requires the manufacture of one of the links of this transmission from antifriction materials. In power worm gears, the gear ring of the worm wheel is usually made of antifriction material, and the worms are made of heat-treated to significant hardness structural carbon and alloy steels of grades 40X, 40XN, 35XGSA, 20X, 18XGT, 12XNZA, 15X A, 38X2MYA, 38X2YU and otheØs, which are subjected, depending on the load and operating conditions, to various types of MOT and CTO.

The antifriction material of the worm gear ring is selected depending on the sliding speed in the transmission. V_s At the beginning of the design of the worm gear, the oriented sliding speed is determined by the formula [17, 27, 30]:

$$V_s = 4 \cdot 10^{-4} n_1 \sqrt[3]{T_2}, \text{ m/s},$$

where n_1 – worm rotation frequency, хв$^{-1}$;

T_2 – torque on the worm wheel shaft, N m.

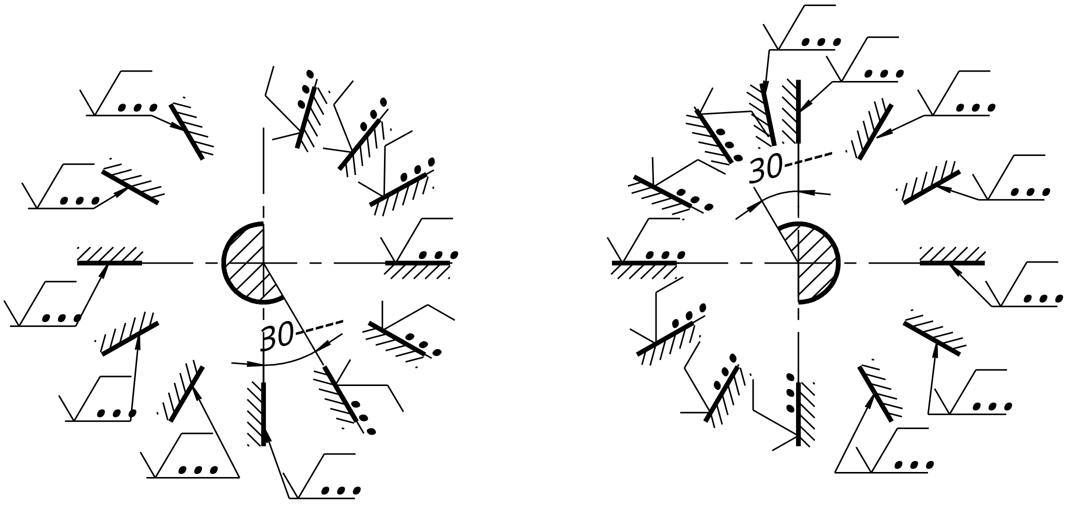
Worm wheels of low-speed worm power units are made gears entirely of cast iron of grades SCh15 and SCh20 ($V_s \leq 8$ *m/s*).

At sliding speeds $V_s \geq 4$ m/s worm gear ring are made of various grades of brass and bronze, which are connected to a center made of cast iron or steel in various ways (see Fig. 27). The following grades of bronze are used for the manufacture of crowns: Br A9ZhZL and BrAYUZhChN4L ($V_s \leq 5$ м/с); Br07Ts7S7 ($V_s \leq 8$ м/с); Br010Ø1 ($V_s \leq 12$ м/с); Br 010N1 Ø1($V_s \leq 25$ м/с).

4.1 Roughness of tooth surfaces

Roughness marks are applied directly to inclined, vertical lines of surface images in cases where the tip of the mark is shown from above and on the leader lines (Fig. 28).

The roughness of the side surfaces of the teeth (the profile of each) is shown on the line corresponding to the diameter of the pitch circle (Fig. 29). Parameter value *Ra* the tooth profile must correspond to the values *1,6* or *0,8*.



and)   b)

Figure 28 – Applying roughness marks to surfaces
details according to changes in the standard

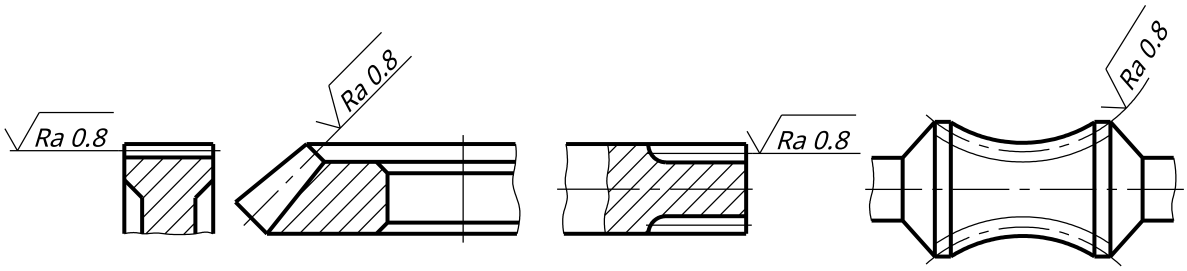


Figure 29 – Gear tooth roughness designation

5 METHODOLOGICAL RECOMMENDATIONS FOR IMPLEMENTATION

GRAPHIC TASKS

According to the course program and work plan, when studying the topic "Transmissions", students must perform: calculations of the geometric parameters of a gear transmission and construction of one of the types of gear transmission.

According to the work plans of the discipline "Descriptive Geometry, Engineering and Computer Graphics", students study the following types of gears: cylindrical, bevel and worm gears. Therefore, one of the above gears is included in the task options.

The following are the initial data for further implementation: 1.

One of the types of gear transmission (Appendices A – B).

2. Module value t, the number of teeth of the gear and the gear wheel, in accordance, z_1 and z_2; numerical values of the diameters of the shafts on which the gear and gear wheel are mounted (Appendices A – B).

3. Variant of the constructive form of the gear and wheel (Appendix D).

5.1 Calculation of geometric parameters and images cylindrical gear

For example, according to the option, the student received a variant of the type of transmission - cylindrical gear (Appendix A). Table 1 contains formulas for calculating the main parameters of gears. The symbols in Figure 31 explain the corresponding formulas in Table 7.

The sequence of execution is as follows:

1. Taking into account the initial values of the module t and the number of gear teeth z_1 and gear wheel z_2 perform calculations of pitch diameters gears $d_1=mz_1$ and gear wheel $d_2=m_{z2}$. Determine the centerline distance $a_w=0,5(d_1+d_2)$.

2. In thin lines by the projection connection in two projections (main view and left view) begin to draw the contours of the gear and the gear wheel that are in engagement. In doing so, all elements are taken into account (see table 7): the diameters of the peaks and valleys of the gear and the wheel (d_{a1} ,d_{a2} and d_{f1}, d_{f2}); diameters d_{m1}, d_{m2} and hub length l_{m1}, l_{m2} in accordance; inner rim diameters d_{o1}, d_{o2}; rim thickness δ_1, δ_2 and disk.

3. Depending on the diameters of the shafts (for the gear and wheel), select dimensions for dowels: width b and height h key (for gear shaft and wheel shaft); dimensions t_1 and t_2 for keyways (Appendix D).

Estimated length of the key l_p $_{shp}$ constructively choose in depending on the length of the gear hub and gear wheel:

$$l^p_{shp}=l_m- 5...6\ mm.$$

Next, the calculated length of the key is compared l_p $_{shp}$ with standard the value of the lengths of the keys l_{st} $_{shp}$ (recommended range is given below) and choose the closest value from the standard series:

$$l^p_{shp} \approx l_{st\ shp}$$

Length of the prismatic key l_{st} $_{shp}$ is selected from a row in mm:

6, 8, 10, 12, 14, 16, 18, 20, 22, 25, 28, 32, 36, 40, 45, 50, 50, 56, 63, 70, 80, 90.

Example of performing a graphical task for a cylindrical gear transmission

Let us consider the formulas for calculating the main and design parameters of a cylindrical gear, taken from tables 1, 7.

Output: module $t=4\ mm$; number of gear teeth $z_1=14$, number of teeth of the gear wheel $z_2=28$; diameters of shafts, respectively gears $d_{in1}=18\ mm$ and gear wheel $d_{in2}=22\ mm$.

The main and design parameters of the gear and gear wheel are calculated, which are given in Tables 8, 9.

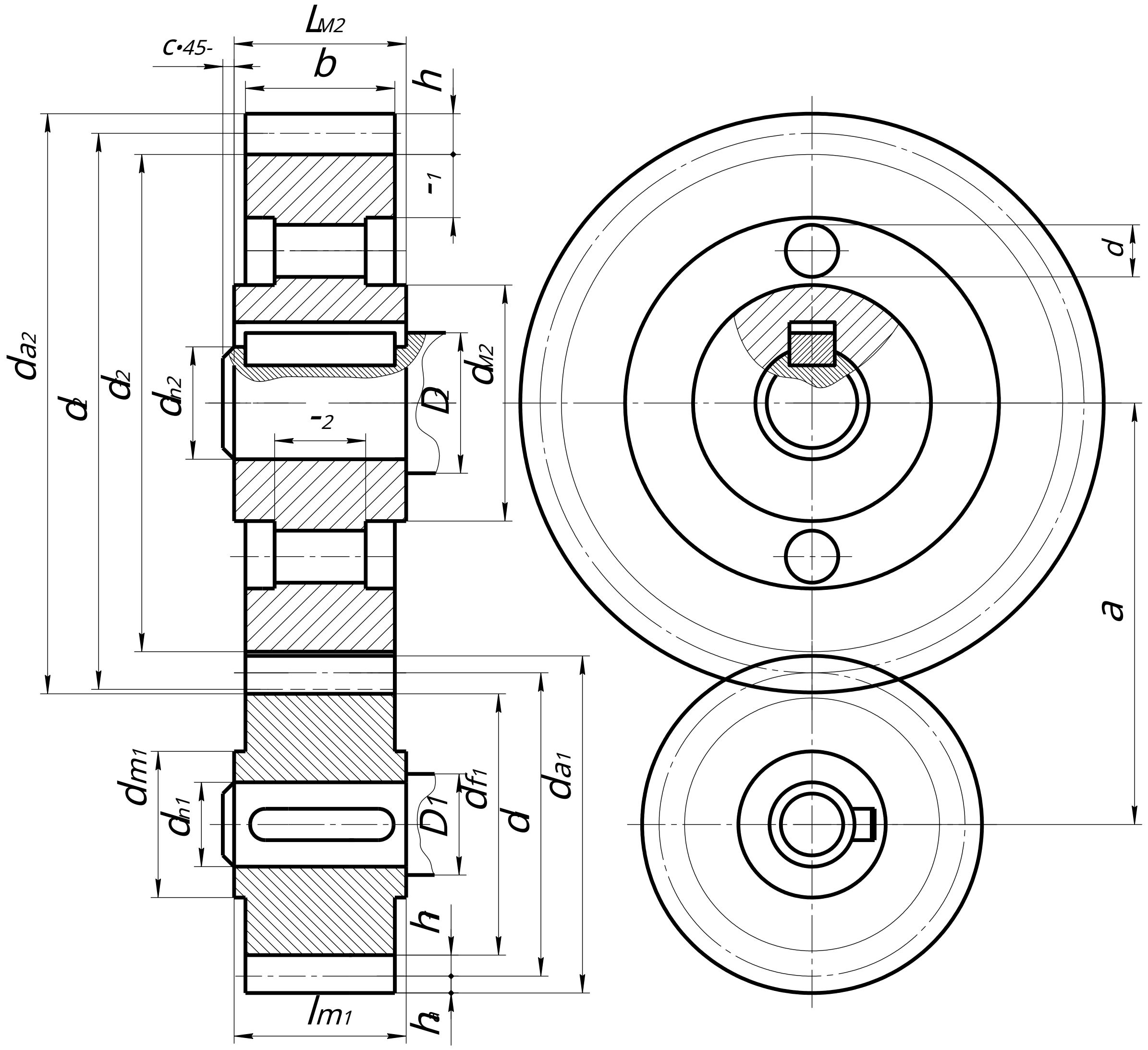


Figure 31 – General drawing for calculating geometric parameters of cylindrical gear transmission

Table 8 – Formulas for calculating the main parameters

Transmission element		Gear	Wheel
Name	Formula		
Tooth height	h=2,25m	h=9 mm	
Tooth head height	h_a=m	h_{and}=4 mm	
Tooth root height	h_f=1,25m	h_f=5 mm	
Pitch diameter	d=mz	d_1=mz_1=56 mm	d_2=mz_2=112 mm
Diameter of the peaks teeth	d_a=d+2h_a	d_{a1}=64 mm	d_{a2}=120 mm
Diameter of the depressions teeth	d_f=d–2h_f	d_{f1}=46 mm	d_{f2}=92 mm

Table 9 – Formulas for calculating design parameters

Transmission element		Gear	Wheel
Name	Formula		
Hub diameter	$d_m=1,6d_{in}$	$d_{m1}=29$ mm	$d_{m2}=35$ mm
Hub length	$l_m=1,2b$	$l_{m1}=22$ mm	$l_{m2}=27$ mm
Width of the gear crown	$b=6...7m$	$b=6m=24$ mm	
Keyway dimensions groove	By *GOST23360–78*		
Wheelbase	$a_w=0,5(d_1+d_2)=84$ mm		
Chamfer dimensions	$c=2$ mm		

*Note:*The gear ring is solid, that is, it does not contain a disk and a rim.

So, based on the initial data, the student performs certain calculations, and then graphical constructions.

As the final result of the graphic work, the student submits for verification his graphic version of the gear transmission (Fig. 32), specification (Fig. 33) and calculations.

6.2 Calculation of geometric parameters and image of a conic section transmission

Based on the initial data, the student must: use tables 3 and 8, which take into account calculations*geometric parameters*and *constructive* features of the gear and gear wheel; general drawing (Fig. 34) for a bevel gear.

The sequence of execution is as follows:

1. Taking into account the initial values of the module$t$and the number of gear teethz_1 and gear wheel$z_2$carry out calculations of gear pitch diameters $d_1=mz_1$ and gear wheel$d_2=mz_2$.Determine the angles of the dividing conesδ_1 andδ_2; external cone distancesR_{e1}andR_{e2}; width of the gear rims of the gear and the wheel$b$; external height of the tooth head and rooth_{a1},h_{f1}andh_{a2}, h_{f2} respectively, the gear and the cogwheel.

2. In thin lines, along the projection connection in two projections, begin draw the contours of the gear and the gear wheel in engagement. All elements are taken into account: the diameters of the peaks and valleys of the gear and the wheel (d_{a1},d_{a2}andd_{f1}, d_{f2}); diametersd_{m1},$d_{m2}$and the length of the hubsl_{m1},l_{m2}in accordance.

3. Depending on the diameters of the shafts (for the gear and wheel), select dimensions for dowels: width$b$and height$h$key (for gear shaft and wheel shaft); dimensions$t_1$and$t_2$for keyways.

A sample of graphic work is shown in Figures 35, 36.

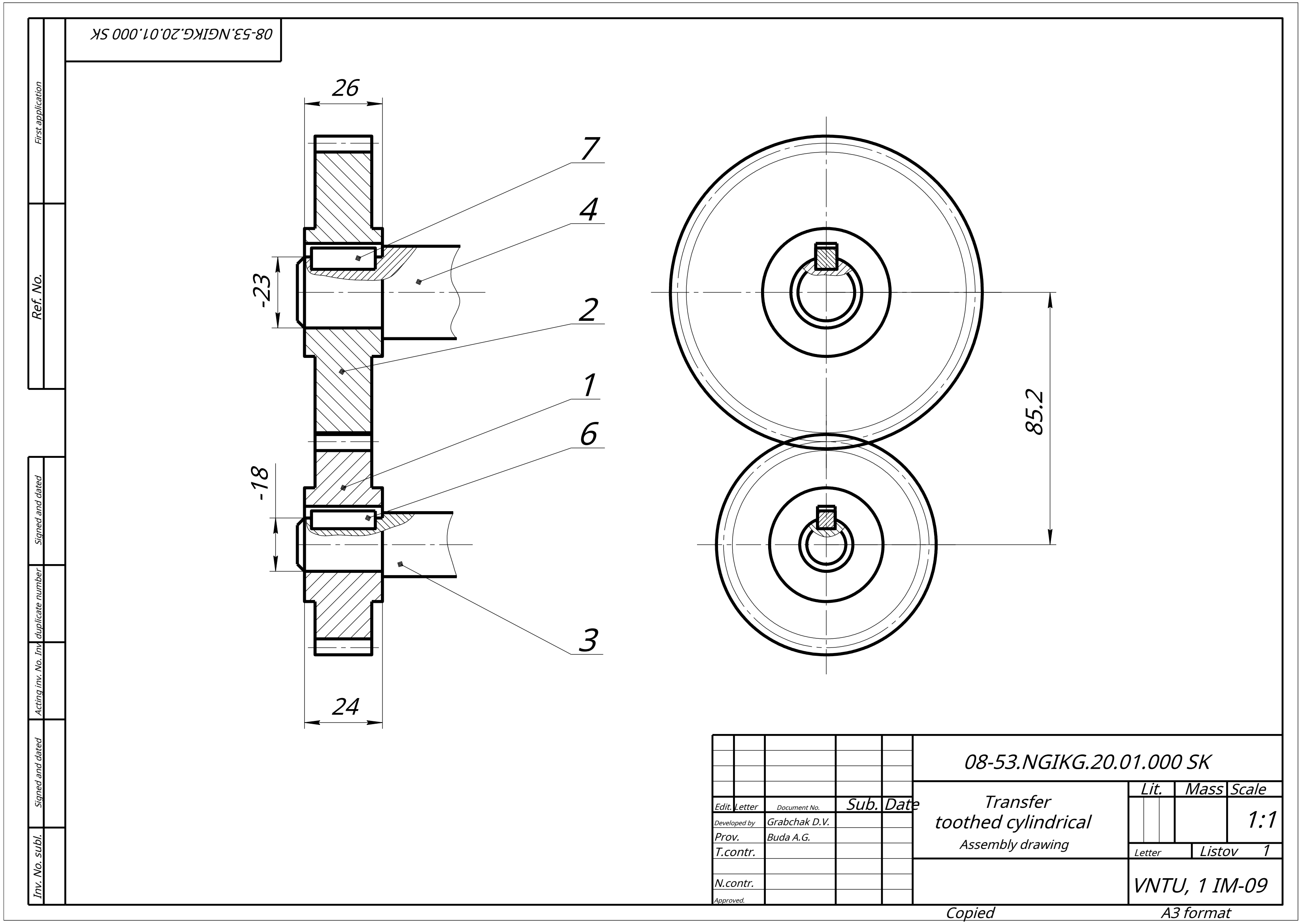


Figure 32 – Example of a cylindrical gear transmission

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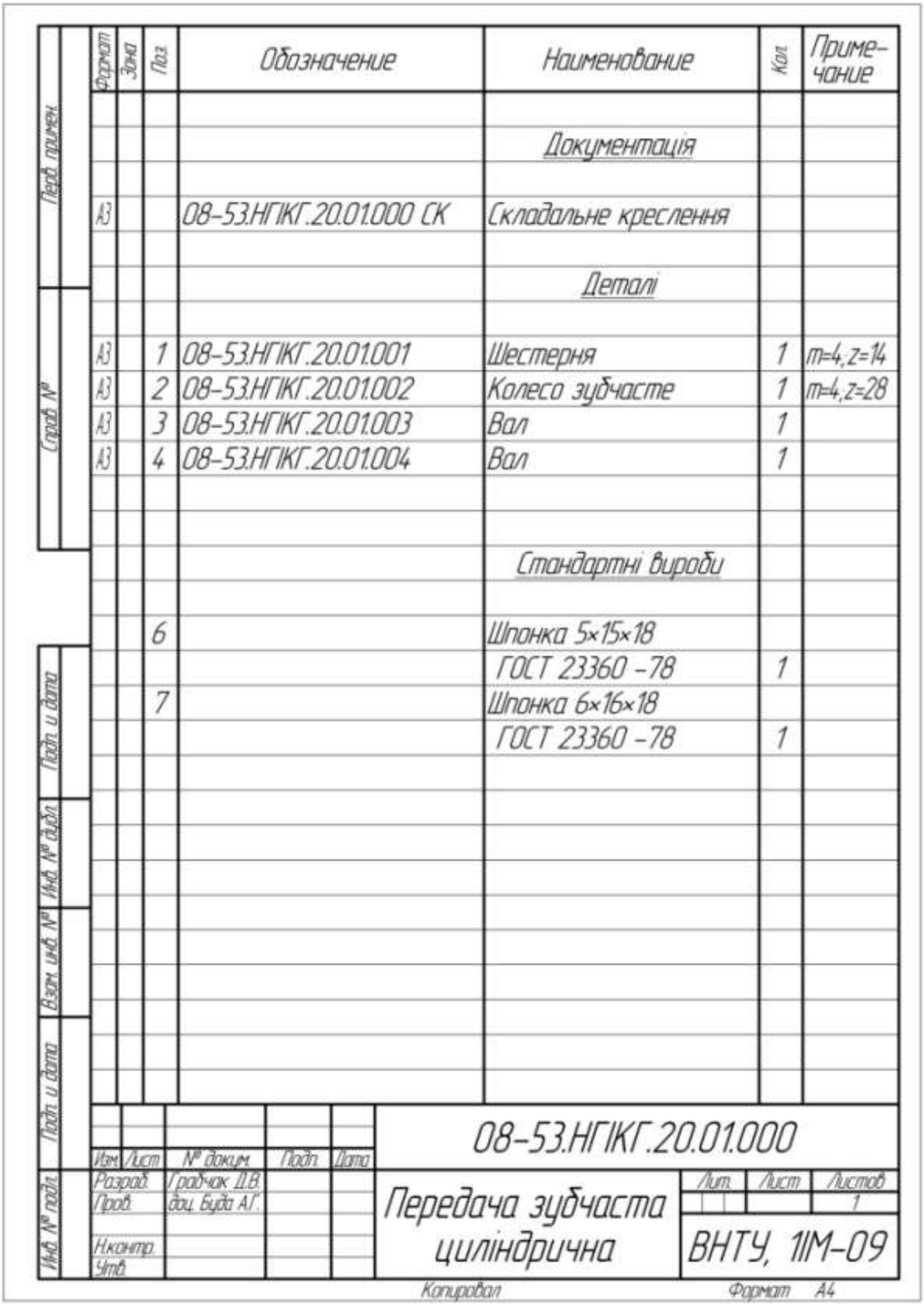

Формат	Зона	Поз.	Обозначение	Наименование	Кол.	Приме-чание
				<u>Документація</u>		
А3			08-53.НГІКГ.20.01.000 СК	Складальне креслення		
				<u>Деталі</u>		
А3		1	08-53.НГІКГ.20.01.001	Шестерня	1	m=4; z=14
А3		2	08-53.НГІКГ.20.01.002	Колесо зубчасте	1	m=4; z=28
А3		3	08-53.НГІКГ.20.01.003	Вал	1	
А3		4	08-53.НГІКГ.20.01.004	Вал	1	
				<u>Стандартні вироби</u>		
		6		Шпонка 5×15×18		
				ГОСТ 23360 –78	1	
		7		Шпонка 6×16×18		
				ГОСТ 23360 –78	1	

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Разраб.		Грабчак Д.В.		
Пров.		доц. Буда А.Г.		
Н.контр.				
Утв.				

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Figure 33 – Sample specification for a spur gear

Table 10 – Design parameters of bevel gears

Transmission element	Marking	Formula
Gear hub length	l_{m1}	$l_{m1}=1,3d_{in1}$
Outer diameter of gear hub	d_{m1}	$d_{m1}=1,7d_1$
Gear shaft diameter	d_1	$d_1=1,2d_{in1}$
Wheel hub length	l_{m2}	$l_{m2}=1,3d_{in2}$
Wheel hub outer diameter	d_{m2}	$d_{m2}=1,7d_{in2}$
Wheel shaft diameter	d_2	$d_2=1,2d_{in2}$
Thickness of the ring gear rim	$\delta=2,5m$	
Cavity	$n=2…3m$	
Radius of fillets	$R=2…3$ mm	
Chamfer dimensions	$c=2…3$ mm	
Keyway dimensions	By *GOST23360–78*	

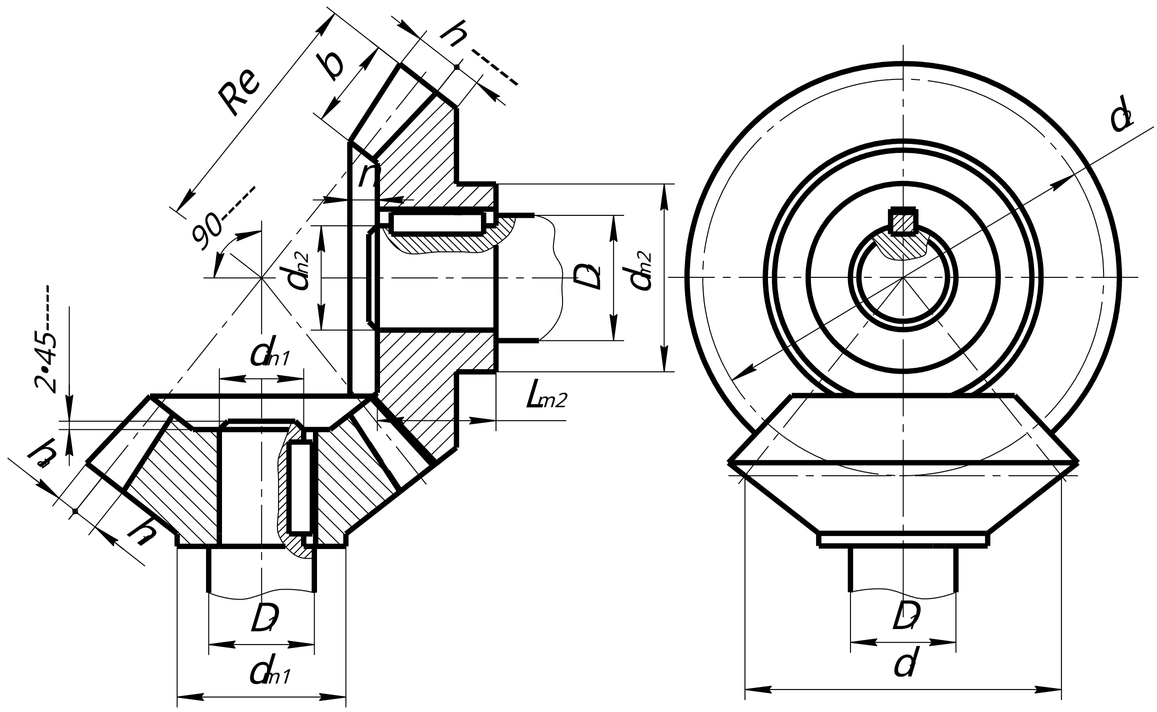


Figure 34 – General drawing for calculating geometric parameters of bevel gear transmission

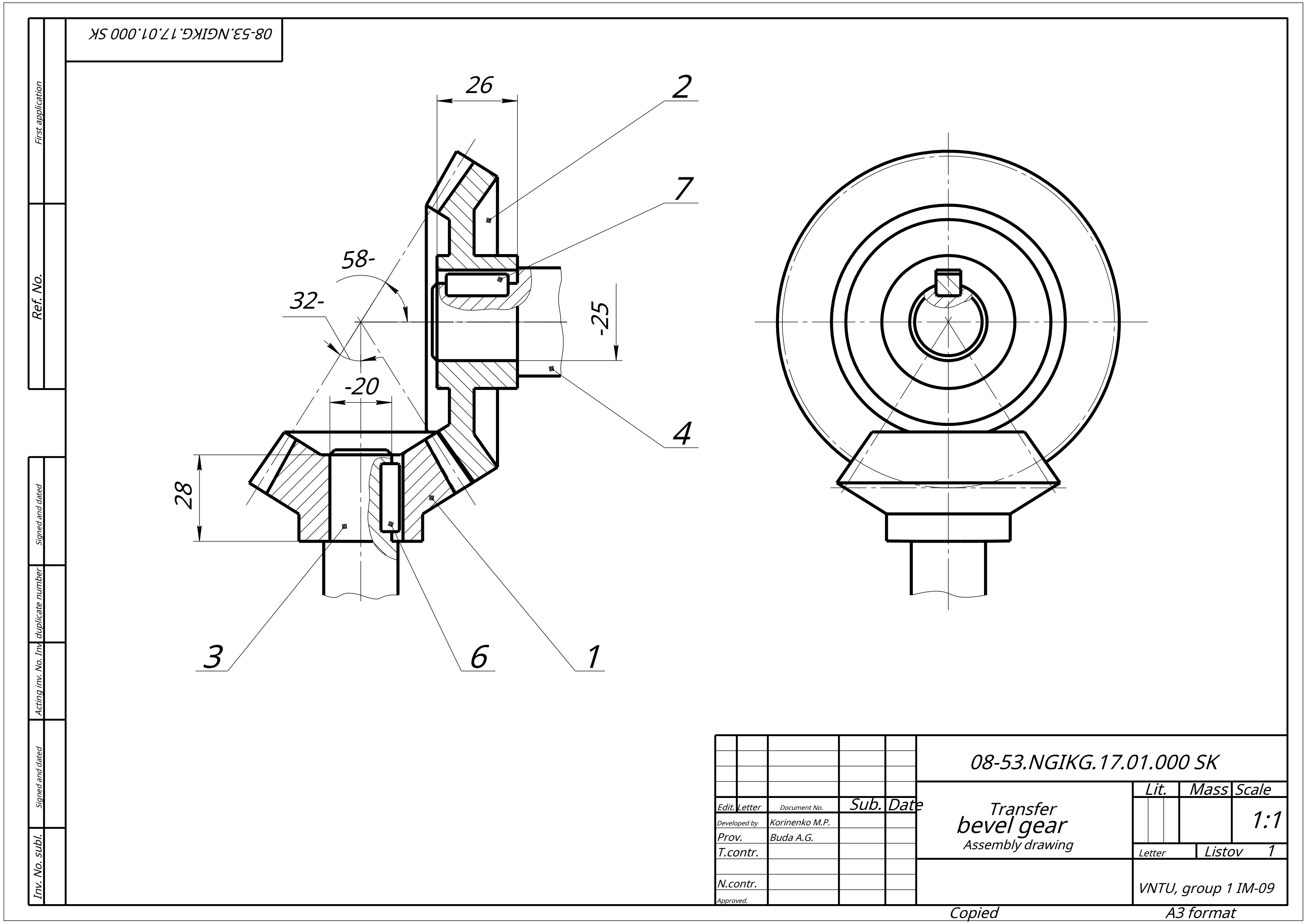
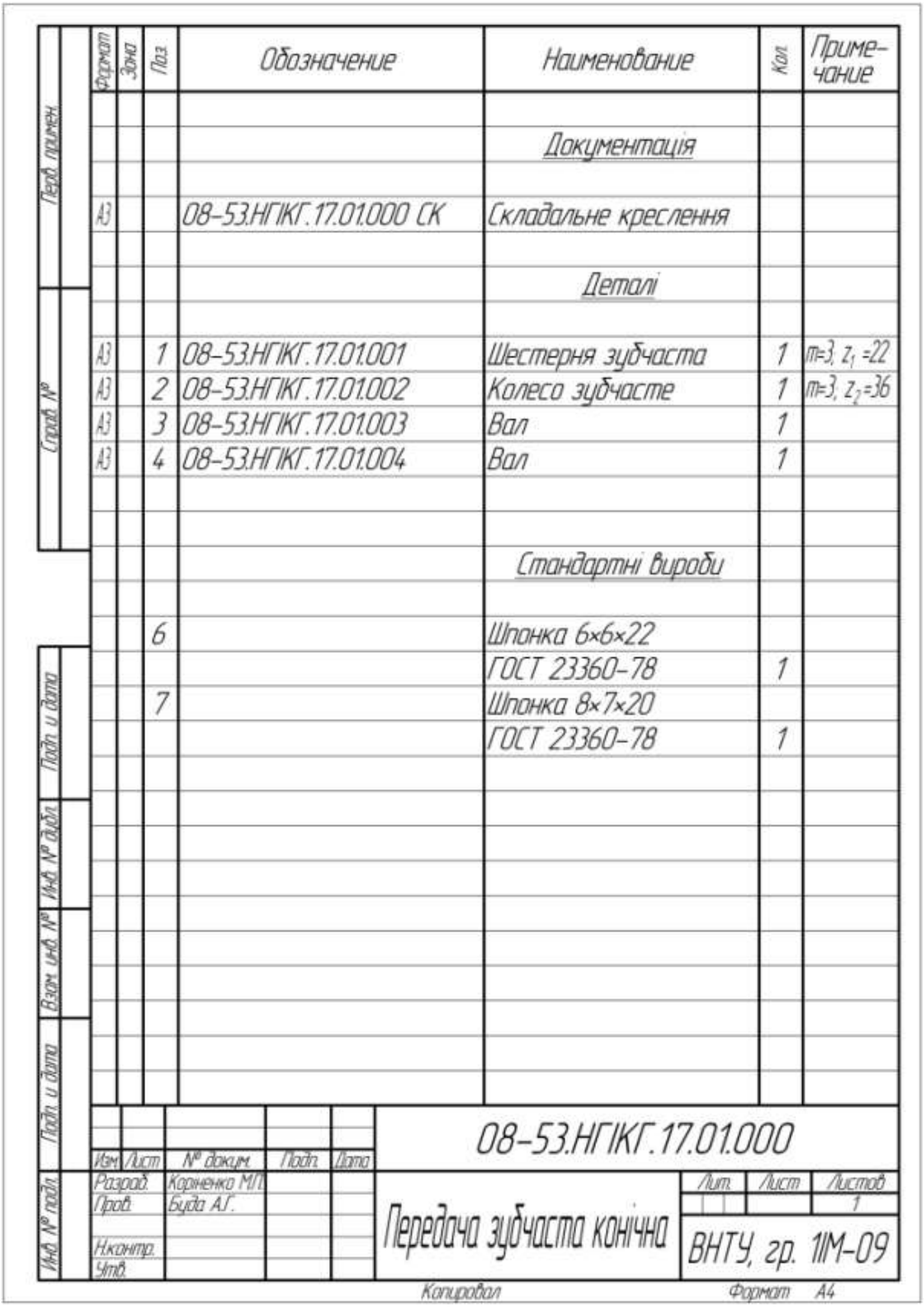


Figure 35 – Example of a bevel gear transmission



Формат	Зона	Поз	Обозначение	Наименование	Кол.	Приме-чание
				Документація		
А3			08–53.НГІКГ.17.01.000 СК	Складальне креслення		
				Деталі		
А3		1	08–53.НГІКГ.17.01.001	Шестерня зубчаста	1	m=3; z_1 =22
А3		2	08–53.НГІКГ.17.01.002	Колесо зубчасте	1	m=3; z_2=36
А3		3	08–53.НГІКГ.17.01.003	Вал	1	
А3		4	08–53.НГІКГ.17.01.004	Вал	1	
				Стандартні вироби		
		6		Шпонка 6×6×22		
				ГОСТ 23360–78	1	
		7		Шпонка 8×7×20		
				ГОСТ 23360–78	1	

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Н.контр.				
Утв.				

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		1

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Figure 36 – Sample specification for bevel gearing

5.3 Calculation of geometric parameters and images worm gear

For example, according to the option, the student received a variant of the type of transmission - worm gear and the initial values of the geometric parameters m, z_1, z_2, d_1 and d_2 (Appendix B). When drawing a worm gear, it is necessary to take into account its design features: the presence of a rim (holes in it), a disk, a hub (Appendix D).

Based on the above initial data, the student must:

– use the general drawing (Fig. 37) and introduce certain changes regarding design features for worm and worm wheel images;

– use tables 5, 6, which take into account *geometric* calculations of the worm and worm wheel, and table 9, which takes them into account *constructive* features.

The sequence of execution is as follows:

1. Taking into account the initial values of the module t and the number of worm movements q and worm wheel z perform calculations of pitch diameters worm $d_1=mq$ and worm wheel $d_2=mz$. Determine the length of the cut part of the worm and the width of the worm wheel gear ring b; external height of the tooth head and root h_{a1}, h_{f1} and h_{a2}, h_{f2} in accordance worm and worm wheel.

Table 11 – Design parameters of worm gear

Transmission element	Worm	Wheel
Width of the gear ring	–	$b_2=0,75d_{a1}$
Hub length	–	$l_{m2}=1,3d_{in2}$
Hub outer diameter	–	$d_{m2}=1,6d_{in2}$
Shaft diameter	–	$d_2=1,2d_{in2}$
Worm length	$l=1,5d_{a1}$	–
Worm shaft diameter	$D_1=0,9d_{f1}$	–
Wheelbase	$A=0,5(d_1+d_2)$	

2. In thin lines by the projection connection in two projections (main view and left view) begin to draw the contours of the worm and worm wheel that are in engagement. In doing so, all elements are taken into account: the diameters of the vertices and valleys of the worm and wheel (d_{a1}, d_{a2} and d_{f1}, d_{f2}); diameter d_{m2} and the length of the worm wheel hub l_{m2} in accordance.

3. Depending on the diameter of the shaft on which the worm wheel is mounted, GOST 23360–78 select the key size: width *in* and height *h*; dimensions t_1 and t_2 for a keyway. The length of the key is calculated, compare with the standard value and choose the closest standard value *l*.

As the final result of the graphic work, the student submits for verification his graphic version of the worm gear (Fig. 38), specification (Fig. 39) and calculations.

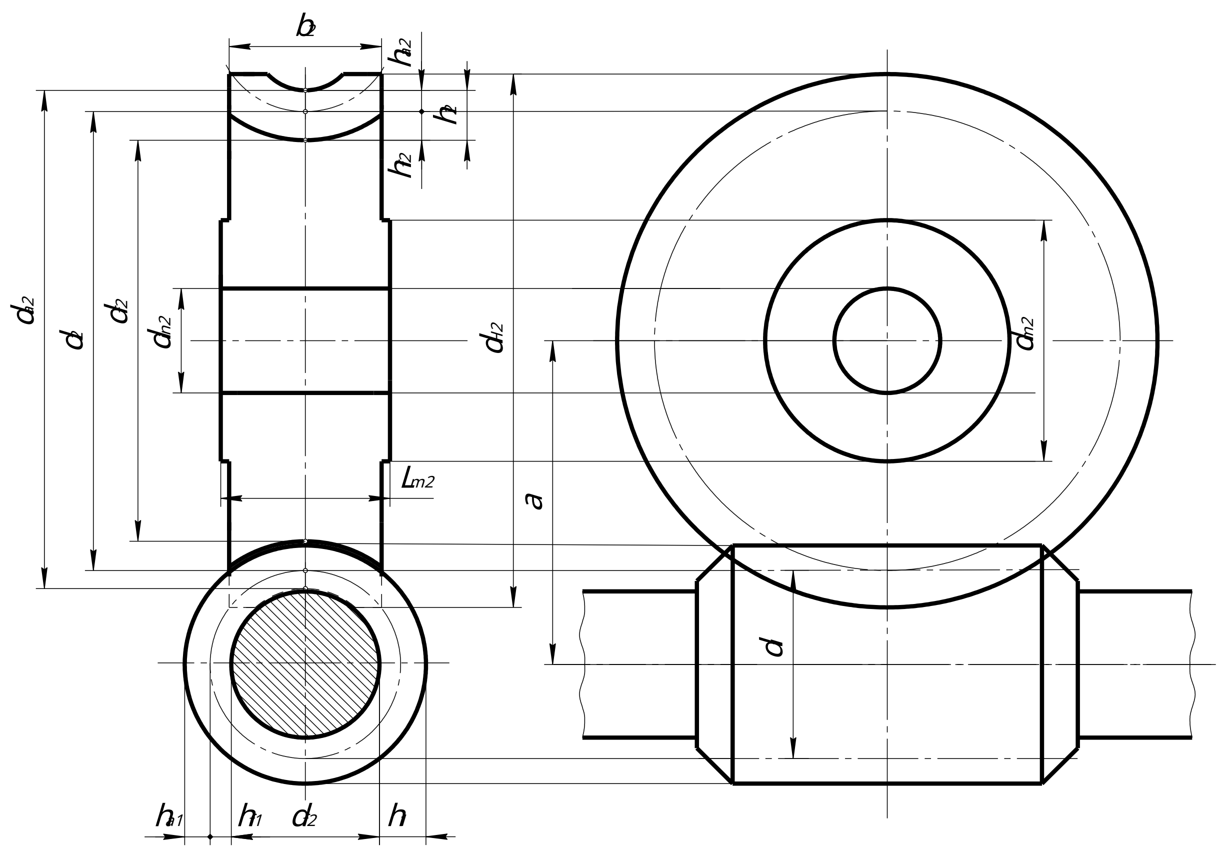
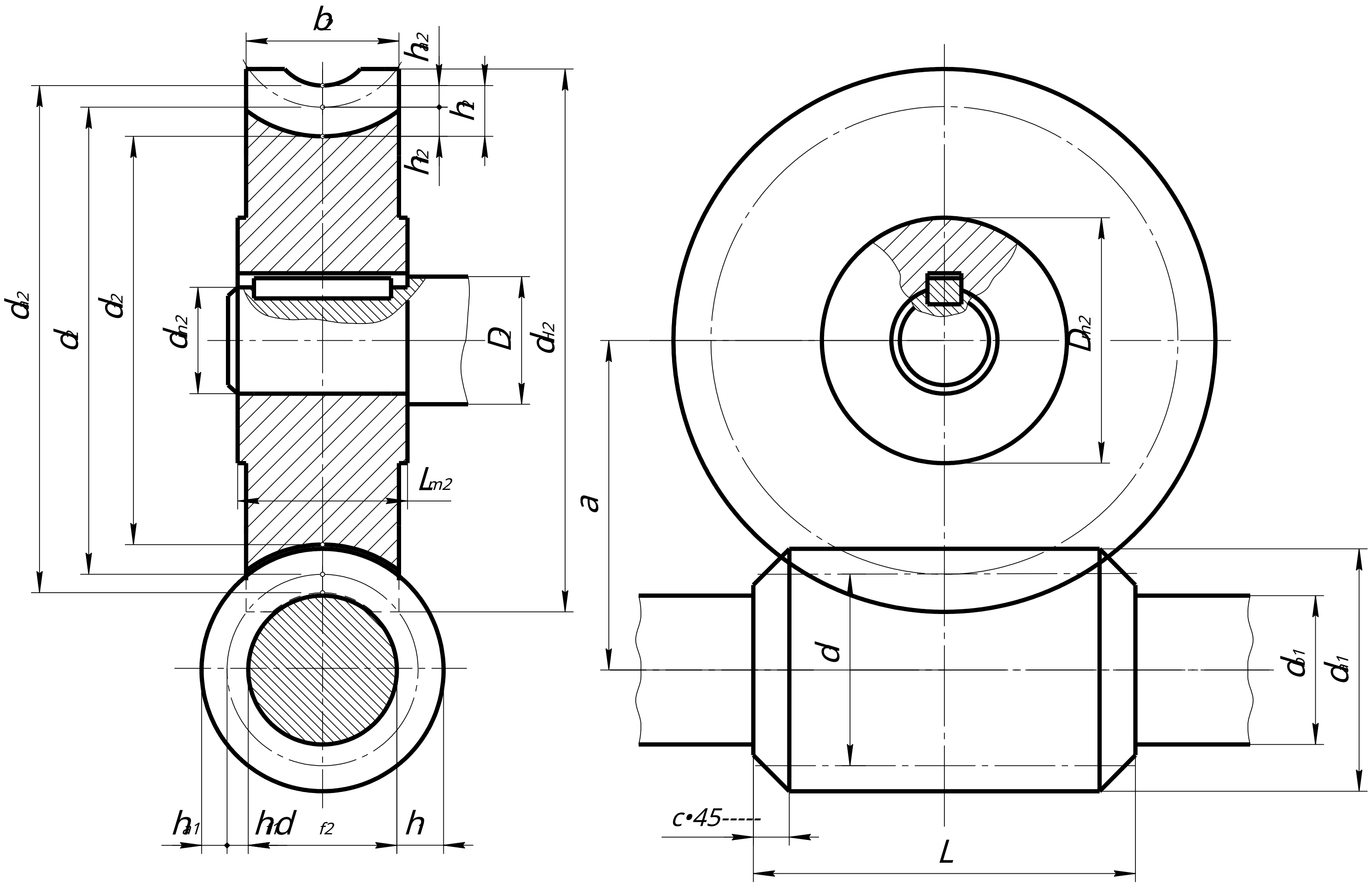


Figure 37 – General drawing for calculating geometric parameters worm gear



b)

Figure 37

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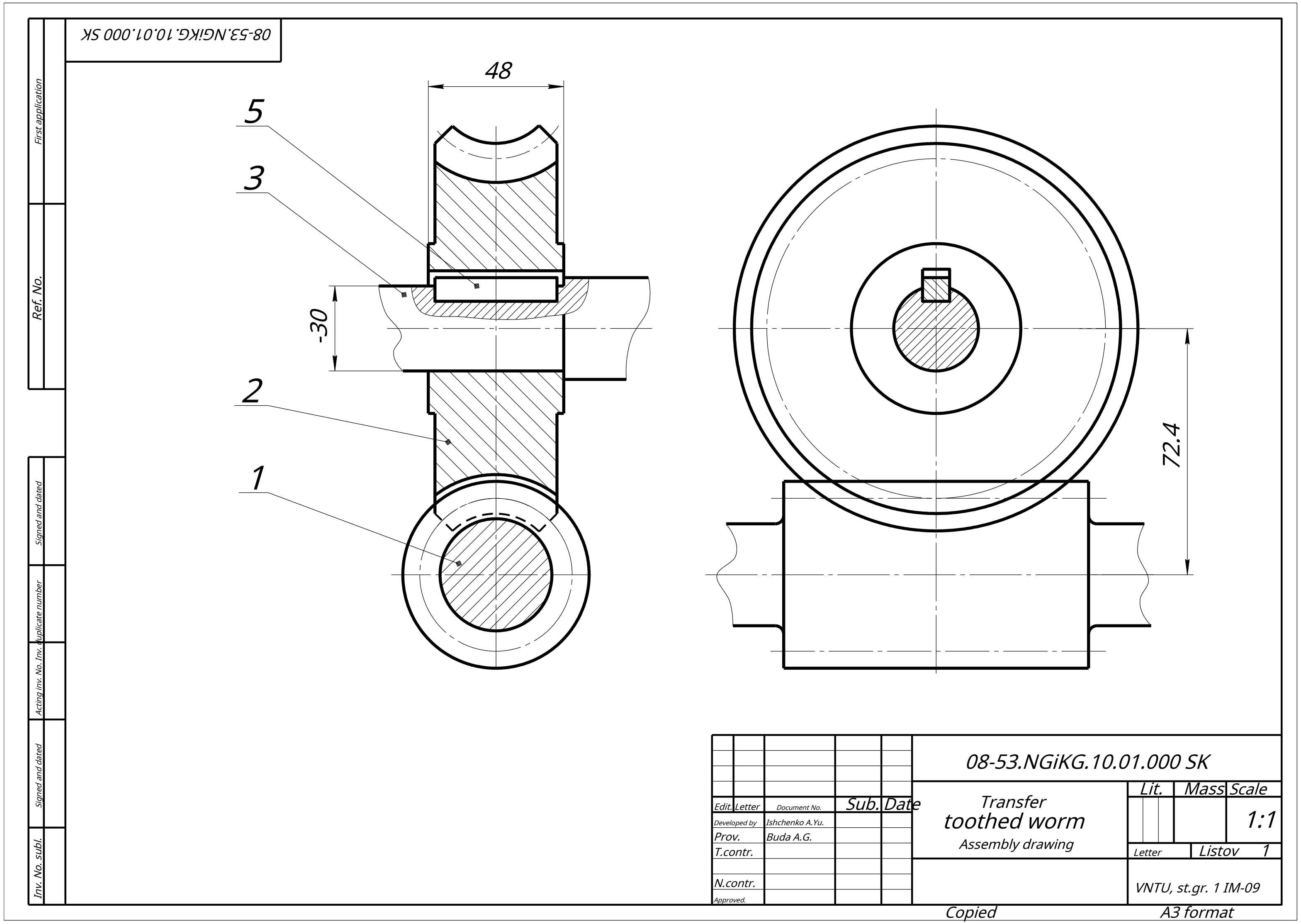

Figure 38 – Example of a worm gear transmission

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Формат	Зона	Поз.	Обозначение	Наименование	Кол.	Приме-чание
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А3			08–53.НГіКГ10.01.000 СК	Складальне креслення		
				Деталі		
А3		1	08–53.НГіКГ10.01.001	Черв'як	1	q=9, m=5
А3		2	08–10НГіКГ10.01.002	Колесо черв'ячне	1	z=20, m=5
А3		3	08–10НГіКГ10.01.003	Вал	1	
				Стандартні вироби		
		5		Шпонка 8×7×36		
				ГОСТ 23360–78	1	

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Изм.	Лист	№ докум.	Подп.	Дата	08–53.НГіКГ.10.01.000		
Разраб.		Іщенко А.Ю.			Передача зубчаста черв'ячна	Лит.	Лист / Листов 1
Пров.		Буда А.Г.				ВНТУ, гр. 1ІМ–09	
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Figure 39 – Sample specification for worm gear transmission

7 RULES FOR IMPLEMENTING KINEMATIC DIAGRAMS

7.1 Types of schemes

A diagram is a graphic design document containing a conventional graphic representation of the component parts of a product and the relationships between them GOST 2.102–68 and GOST 2.701–76.

The development of any mechanism begins with drawing a schematic diagram, which conventionally depicts the interaction and sequence of action of the product's elements.*Element*Schematics are a component part of a circuit that performs a specific function in a product and cannot be divided into components that have a separate functional purpose. Schematic drawings are used in the development of all design documents for a product during setup, adjustment, control, operation, and repair.

Depending on the nature of the elements and connections that make up the product, diagrams are divided into types, which are conventionally denoted by letters:*K* – kinematic;*G* –hydraulic;*P* –pneumatic;*E* –electric;*C* – combined*.*

The diagrams are executed on standard format sheets, without using a scale, and without strictly reflecting the actual location of the product's components.

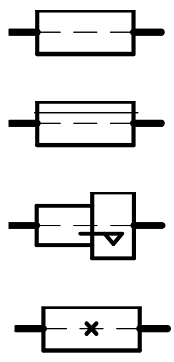
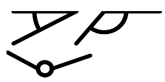
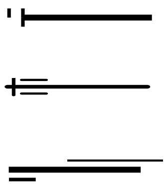
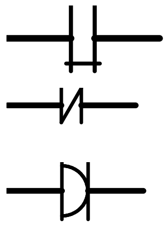
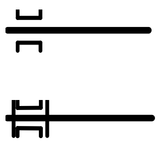
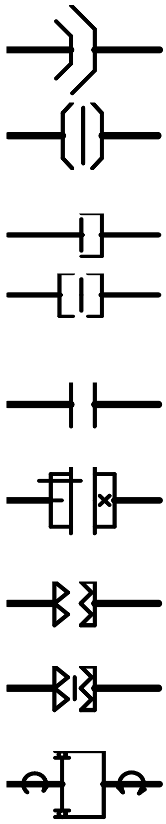
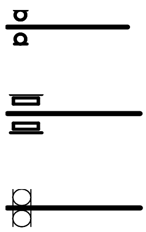
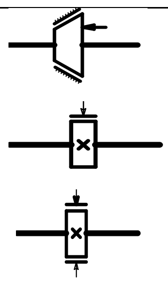
7.2 Kinematic diagrams

Performed in accordance with the requirements of the standard (GOST 2.703–86). Elements (GOST 2.770–68) that make up the product are depicted on the diagrams with conventional graphic symbols (Table 10).

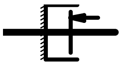
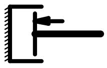
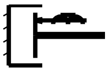
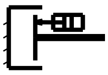
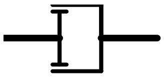
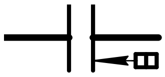
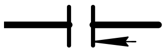
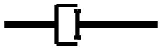
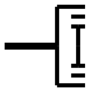
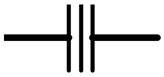
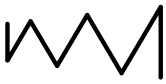
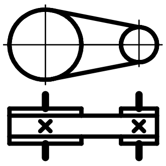
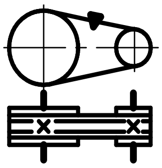
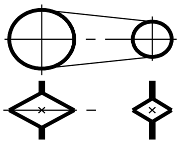
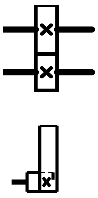
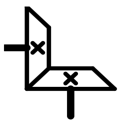
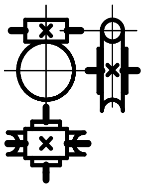
Kinematic elements include shafts, axles, bearings, couplings, brakes, pulleys, gears, worms, sprockets, gears, and so on.

On kinematic diagrams (*kinematic gearing*) for images of kinematic elements, a certain line thickness is used: solid main lines with a thickness of*s* –shafts, axles, rods, connecting rods, cranks and similar; solid thin lines, thickness*s/2* –elements depicted in the form of contour outlines, gears, worms, sprockets, pulleys, cams and the like; in solid thin lines with a thickness of*s/3* –contour of the product in which the diagram is inscribed; dashed lines with a thickness of*s/2* – kinematic relationships between conjugate links of a pair, drawn separately; double dashed lines and lines of thickness*s/2* –kinematic connections between elements or between them and the source of motion through non-mechanical (electrical) sections; triple dashed lines with a thickness of*s/2* – design relationships between elements.

Table 10 – Symbols on kinematic diagrams

Name	Marking	Name	Marking
1	2	3	4
Shaft, axle, rod, connecting rod		Part connection with shaft: – free at rotation – moving without rotation – with the help of exhaust key – deaf	
Immovable axle fixing			
Rod connections: – tough – hinged			
Bearings sliding and rolling: – radial – radial thrust one-sided – stubborn one-sided		Connecting two shafts: – deaf – elastic – hinged	
Sliding bearing: – radial – radial thrust bilateral		Couplings clutch: – conical one-sided – conical bilateral – disk one-sided – disk bilateral – frictional – friction with mounting on shaft – cam one-sided – cam bilateral – overtaking one-sided	
Rolling bearing: – radial – radial-rolling – stubborn ball one-sided			
Brakes – conical – block – striped			

Continuation of table 10

1	2	3	4
Disc brakes ordinary – electromagnetic – hydraulic – centrifugal	    	Couplings – one-sided hydraulic – one-sided – with pads – with detachable ring – bilateral	    
A screw that transmits motion Fixed nut on screws Split nut on screws	  	Springs – compression – stretching – conical	  
Transfer: – flat belt – V-belt – asterisk	  	Gear transmissions: – cylindrical (with external and internal engagement) – conical – with cylindrical worm	  

The kinematic diagrams show:

– the name of each kinematic group of elements, taking into account its main functional purpose (feed drive): a leader line drawn from the corresponding group is applied to the shelf;

– the main characteristics and parameters of kinematic elements that determine the executive movements of the working bodies of the product or its components.

Approximate list main characteristics and parameters kinematic elements are given in Table 11.

Each kinematic element depicted in the diagram is usually assigned a serial number, starting from the source of motion.

Shafts are numbered in Roman numerals, other elements in Arabic numerals (Fig. 40).

The element's serial number is placed on the shelf of the leader line. Below the shelf of the leader line, the main characteristics and parameters of the kinematic element are indicated (for gears, the number of teeth).

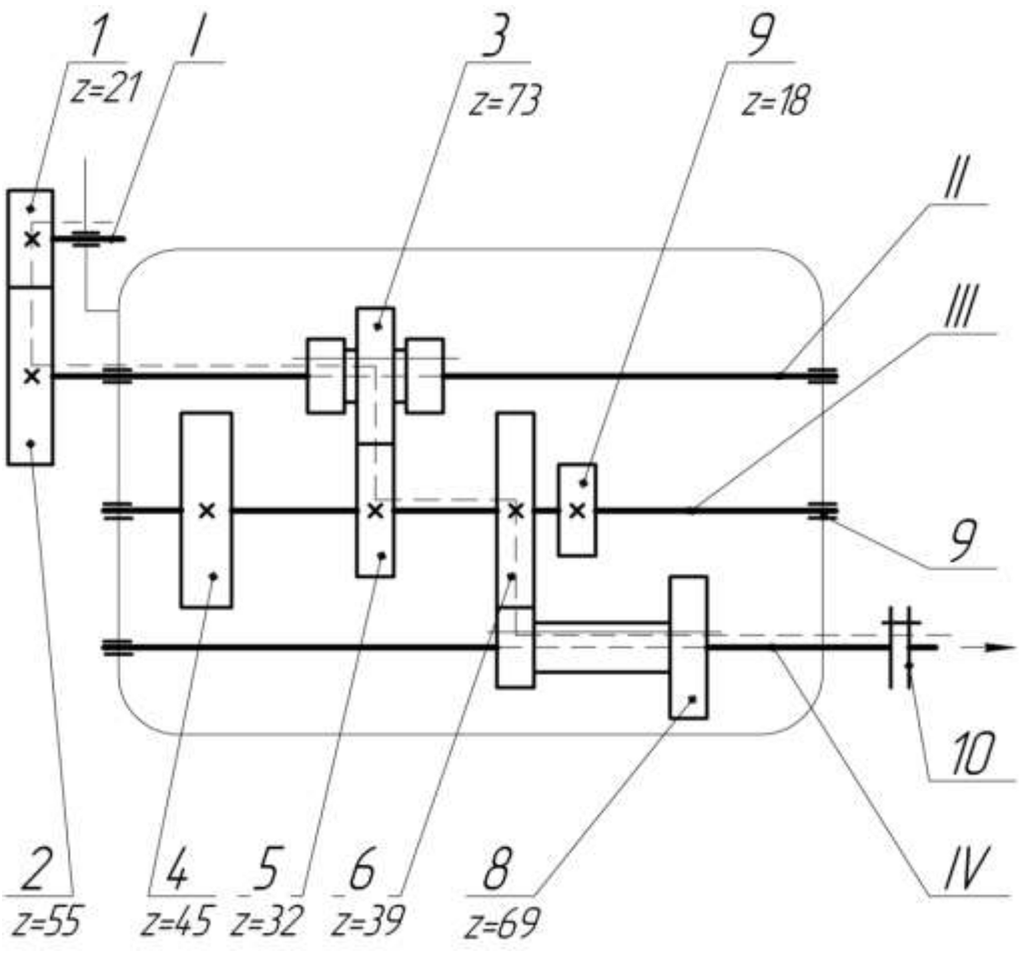


Figure 40 – Images and symbols on kinematic diagrams

The sample (Fig. 41) is of an educational nature, which does not take into account the presence of standard elements.

The list of elements of kinematic diagrams can be placed above the main inscription (if there is space on the drawing) or executed as a separate non-main design document (Fig. 42).

Table 11 – Characteristics and parameters of kinematic elements (GOST 2.703–68)

Name	Data indicated on the diagram
1. Source of movement (engine)	name, type, characteristics, characteristics of the main executive movements, range of adjustment, and the like.
2. Mechanism, kinematic group	dimensions that determine the limits of movements: the length of movement or the angle of rotation of the executive body the direction of rotation or movement of elements on which the obtaining of the specified executive movements and their consistency depend
3. Kinematic links:	
) belt pulleys transmission	diameter (for interchangeable pulleys – the ratio of the diameters of the driving pulleys to the diameters of the driven pulleys)
) gear wheel	number of teeth, module; for helical gears – direction and angle of inclination of the teeth
) toothed rack	helical rack module – direction and angle of tooth inclination
) worm	axial module, number of steps, worm type, direction of turn and worm diameter
) lead screw	helix ID, number of steps, inscription "left" – for left-hand threads
) chain sprocket transmission	number of teeth, chain pitch
) cam	Curve parameters that determine the speed and limits of movement of the pusher

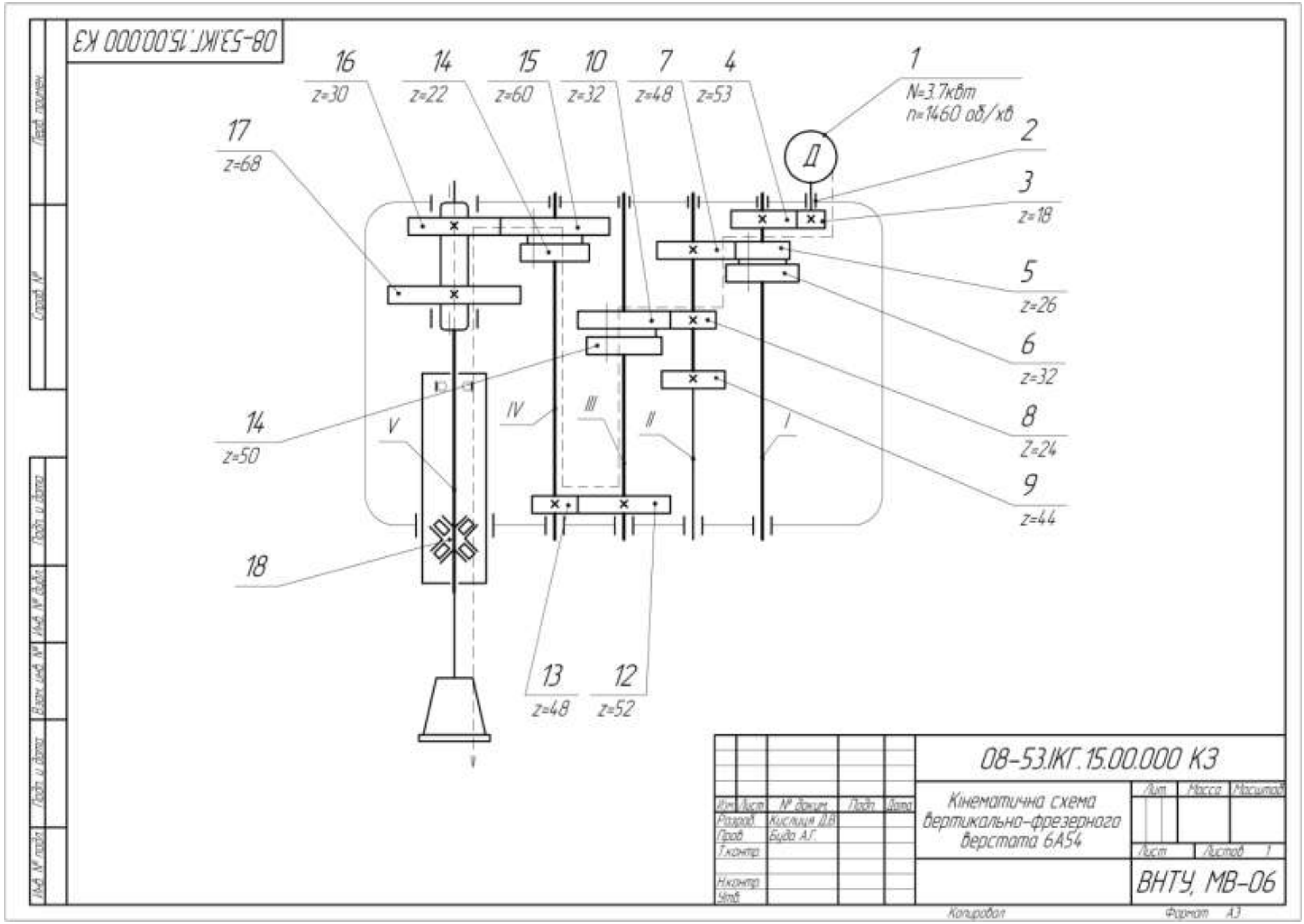


Figure 41 – Example of a kinematic diagram

Поз. познач.	Найменування	Кіл.	Примітка
1	Двигун	1	P=3,7 кВт, n=1460 об/хв
2	Підшипник радіальний	10	
3	Шестерня	1	Z=18
4	Колесо зубчасте	1	Z=53
5, 6	Блок-шестерня	1	Z=24, Z=32
7	Шестерня	1	Z=48
8	Колесо зубчасте	1	Z=44
9	Шестерня	1	Z=28
10, 11	Блок-шестерня	1	Z=32, Z=50
12	Колесо зубчасте	1	Z=52
13	Шестерня	1	Z=48
14, 15	Блок-шестерня	1	Z=22, Z=60
16	Шестерня	1	Z=30
17	Колесо зубчасте	1	Z=68
18	Підшипник роликовий	1	
19	Шпіндель	1	

Перв. примен.
Справ. №
Подп. и дата
Инв. № дубл.
Взам. инв. №
Подп. и дата
Инв. № подл.

Изм.	Лист	№ докум.	Подп.	Дата	08-53.ІКГ.15.00.000ПЕК3		
Разраб.		Кислиця Д.В.			Кінематична схема вертикально-фрезерного верстата 1336М	Лит.	Лист / Листов
Пров.		Буда А.Г.					1
Н.контр.						ВНТУ гр. МВ-06	
Утв.							

Копировал Формат A4

Figure 42 – List of elements for the kinematic diagram

UKRAINIAN-RUSSIAN-ENGLISH GLOSSARY OF SOME TERMS

Ukrainian		English
1	2	3
Antifriction material	Anti-friction material	Antifriction material
Archimedean (involute) worm	Archimedean (involute) worm	Archimedean involute worm
Archimedean spiral	Archimedean spiral	Archimedean spiral
Bandaged (bimetallic) construction	Bandage (bimetallic) construction	Reinforced (bimetallic) construction
Executive (main) link	Executive (main) link	Executive (main) link
Height (width) tooth	Height (width) tooth	Tooth depth (width)
Screw line	Screw line	Helical line
Screw transmission	Screw transmission	Screw gear
Head (stem) of a tooth	Head (stem) of a tooth	Addendum (determinant)
Diameter of the peaks (dents) of teeth	Diameter of the vertices (dental cavities) of teeth	Tooth tip (slot)diameter
Dividing diameter	Divisive diameter	Reference diameter
Diameter (length) hubs	Diameter (length) hubs	Nave (length) Diameter
Dovbach	Dolbyak	Jointer
Involute	Evolventa	Evolvent (involute)

1	2	3
Gear elements wheels	Elements of gear wheels	Components of toothed gear
Elements wheels	Elements wheels	Wheel components
Engagement	Engagement	Gearing
Leading sprocket (conducted)	Leading asterisk (slave)	Drive sprocket
Connection	Connection	Junction
External height heads (feet) of a tooth	External height heads (legs) of a tooth	Outer tooth addendum (determinant)
Outer diameter peaks (valleys) teeth	Outer diameter peaks (valleys) teeth	Outer diameter (recess) of teeth
External (middle) cone distance	External (medium) cone distance	Outer (middle) cone distance
Tooth	Tooth	Tooth
Toothed rack	Toothed rack	Toothed rack
Toothed sector	Toothed sector	Toothed quadrant
Toothed cylindrical (bevel) gear	Toothed cylindrical (bevel) gear	Toothed conic transmission
Kinematic schemes	Kinematic schemes	Kinematic diagram
Drive wheel (slave)	Drive wheel (slave)	Driving (driven) wheel
Gear wheel (wormy)	Gear wheel (wormy)	Gear wheel
Helical rack	Helical rack	Helical rack
Normal step (rationed, frontal)	Normal step (normal, end)	Regular pitch (normalize, butt)

1	2	3
Dividing angle cone	Divisor angle cone	Reference cone angle
Vertex cone angle (depression)	Cone angle peaks (valleys)	Tip angle(root angle)
Leg (head) angle tooth	Leg angle (head) of a tooth	Dedendum angle (addendum).
Chain transmission	Chain transmission	Chain gear
Materials	Materials	Materials
Hub	Hub	Nave (hub)
Mechanical transfer	Mechanical transfer	Mechanical gear (transmission
Wheelbase	Center distance	Axle spacing
Module (calculation) module)	Module (calculated module)	Module (calculation module)
Module normal (basic)	The module is normal (main)	Main Module
Unadjusted gear wheel	Uncorrected gear wheel	Non-adjusted toothed gear
Rotary (translational) motion	Rotary (progressive) movement	Rotating motion
Basic geometric (constructive) parameters	Basics geometric (constructive) parameters	Main geometric (design) parameters
Axial forms	Axial forms	Axial forms
Belt drive	Belt transmission	Belt drive
Hydraulic transmission, electric, mechanical, pneumatic	Hydraulic transmission- electrical, Czech, mechanical, pneumatic	Hydraulic, electrical, mechanical, pneumatic transmission

1	2	3
Initial cylinders	Initial cylinders	Pitch cylinder
Device, principle of operation	Device, working principle	Device, principle of operation
Notch radius	Notch radius	Flute radius
Working drawing book	Working drawing	Working drawing
Estimated base distance	Calculation base distance	Calculated base distance
Calculation geometric parameters	Calculation geometric parameters	Analysis of geometric parameters
Medium conical distance	Medium conical distance	Middle cone distance
Methods production	Manufacturing methods	Ways of producing
Terms, definitions, marking	Terms, definitions, designations	Terms, definitions, symbols
End profile	End profile	End profile
Formulas for calculation	Calculation formulas	Analysis formulas
Finger mill (wormy)	Finger cutter (worm)	Worm cutter (mill)
Friction transmission	Friction transmission	Friction gear
Worm's move	Worm's move	Worm travel
Cylindrical straight-toothed (helical-toothed) wheel	Cylindrical spur gear (helical) wheel	Spur wheel (helical)
Number of teeth (depression)	Number of teeth (depression)	Amount of teeth(slots)
Worm, wormy wheel	Worm, wormy wheel	Worm, worm gear
Gear	Gear	Gear

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Appendix A

Graphical condition and initial data for drawing a cylindrical gear

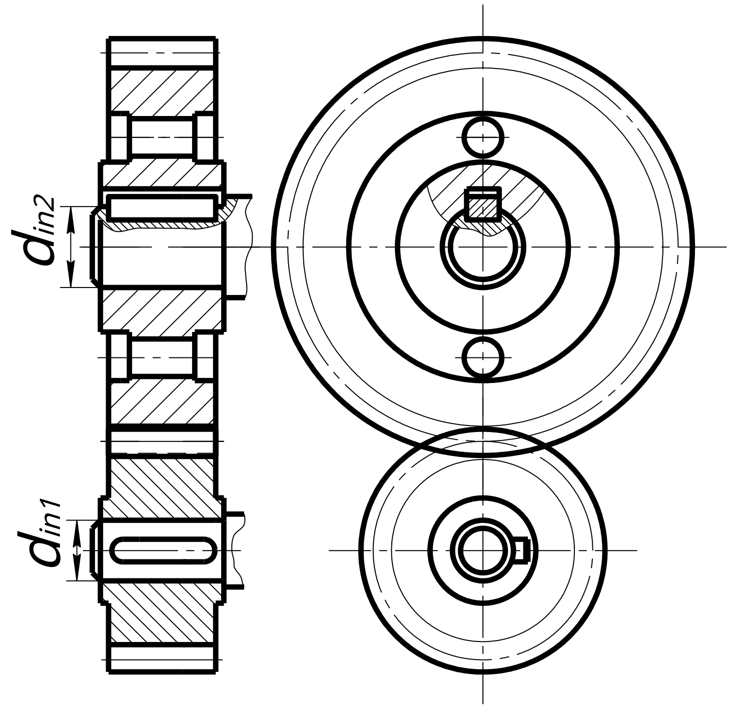


Figure A.1 – Graphical condition for drawing a cylindrical gear

Table A.1 – Input data for calculating parameters

No. var.	m	z_1	z_2	d_{in1}	d_{in2}	No. var.	m	z_1	z_2	d_{in1}	d_{in2}
1	5	20	25	25	20	16	4	18	30	22	25
2	4	20	40	25	20	17	4	20	36	22	30
3	5	15	32	25	15	18	4	15	35	20	30
4	3	25	40	20	25	19	5	16	30	25	32
5	4	25	35	25	25	20	4	20	32	22	30
6	4	20	34	22	20	21	5	16	30	25	36
7	5	18	30	25	18	22	4	15	35	20	25
8	4	15	35	20	15	23	4	18	35	24	30
9	4	20	36	25	32	24	4	20	35	25	32
10	5	16	30	25	30	25	4	18	35	20	30
11	4	20	30	20	25	26	5	18	32	25	30
12	4	20	34	20	25	27	4	25	30	20	25
13	5	16	28	25	35	28	4	20	36	20	30
14	4	22	36	25	30	29	4	18	38	20	28
15	4	20	38	22	30	30	5	18	26	25	30

Appendix B

Graphical condition and initial data for drawing a bevel gear

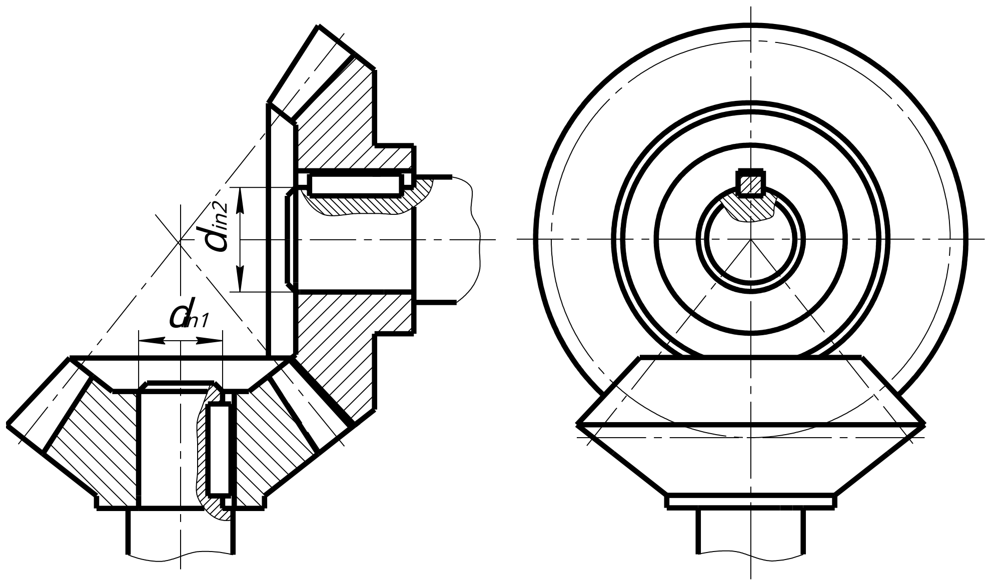


Figure B.1 – Graphical condition for drawing a bevel gear

Table A.1 – Input data for calculating parameters

No. var.	m	z_1	z_2	d_{in1}	d_{in2}	No. var.	m	z_1	z_2	d_{in1}	d_{in2}
1	4	20	35	26	35	16	4	20	35	30	35
2	4	18	30	25	30	17	4	16	32	25	30
3	4	24	35	30	40	18	5	15	25	25	30
4	4	18	32	25	36	19	4	18	36	25	35
5	5	16	30	25	40	20	5	16	30	26	35
6	5	15	30	30	35	21	4	16	25	20	30
7	4	20	32	25	30	22	5	15	28	25	30
8	5	18	28	30	40	23	4	20	35	30	36
9	4	18	28	25	30	24	5	20	30	30	30
10	5	20	30	30	40	25	4	20	40	30	40
11	4	20	36	25	35	26	4	16	40	25	46
12	5	15	30	25	36	27	5	16	32	30	35
13	4	16	32	25	30	28	4	15	25	20	25
14	5	18	36	30	40	29	5	15	26	25	30
15	4	20	40	30	40	30	4	18	36	25	35

Graphical condition and initial data before drawing a worm gear

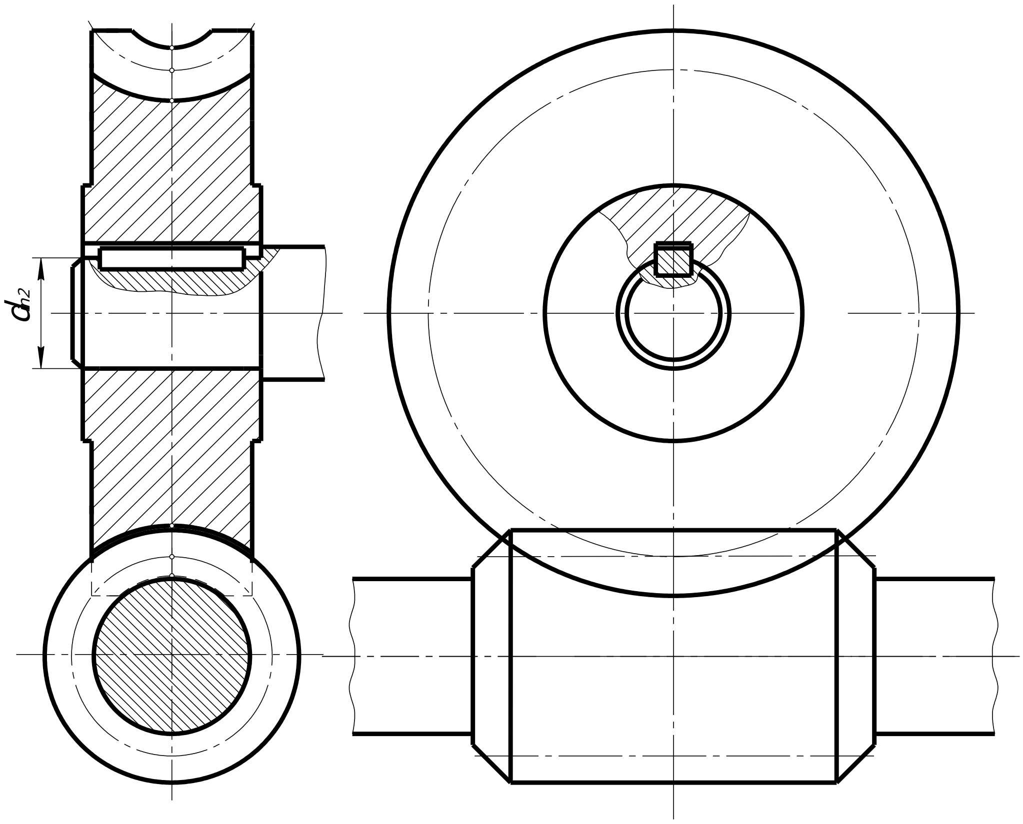


Table B.1 – Input data for calculating parameters

No. var.	m	q	z	d_{in2}	No. var.	m	q	z	d_{in2}
1	3	12	40	36	16	4	9	31	32
2	4	10	40	40	17	3.5	14	40	32
3	5	9	35	40	18	3.5	12	46	36
4	3.5	12	40	40	19	4	12	36	40
5	3	12	50	32	20	3	12	54	40
6	5	9	31	36	21	3	10	40	32
7	3	12	40	36	22	4	9	48	32
8	2.5	12	46	32	23	3	12	40	36
9	3	10	54	36	24	3	10	54	36
10	3	10	40	32	25	4	10	31	32
11	5	9	31	36	26	4	9	40	40
12	3.5	12	36	32	27	2.5	16	46	36
13	3.5	12	40	36	28	3	12	50	40
14	4	9	36	32	29	5	9	31	40
15	3	12	40	32	30	4	12	31	36

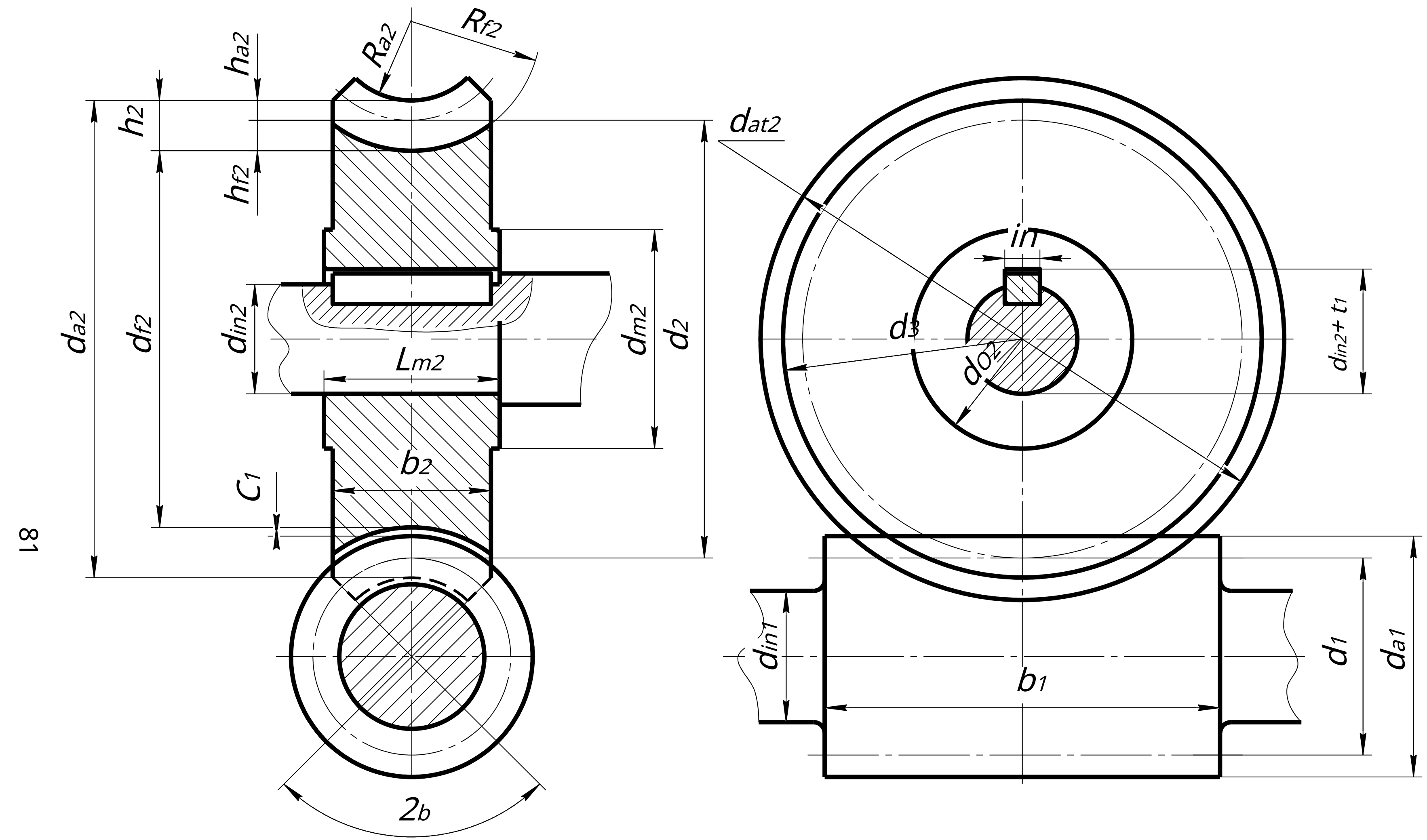
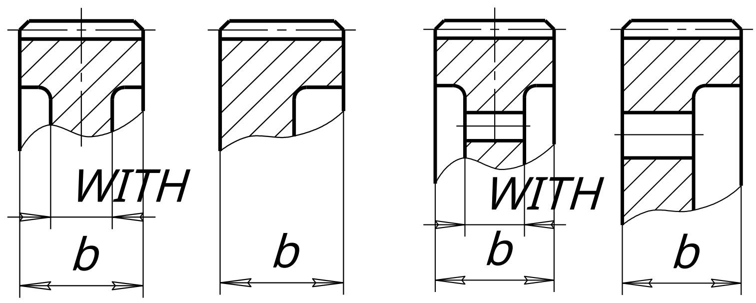


Figure B.2 – Design parameters of worm gear

Appendix D

Design features of gears



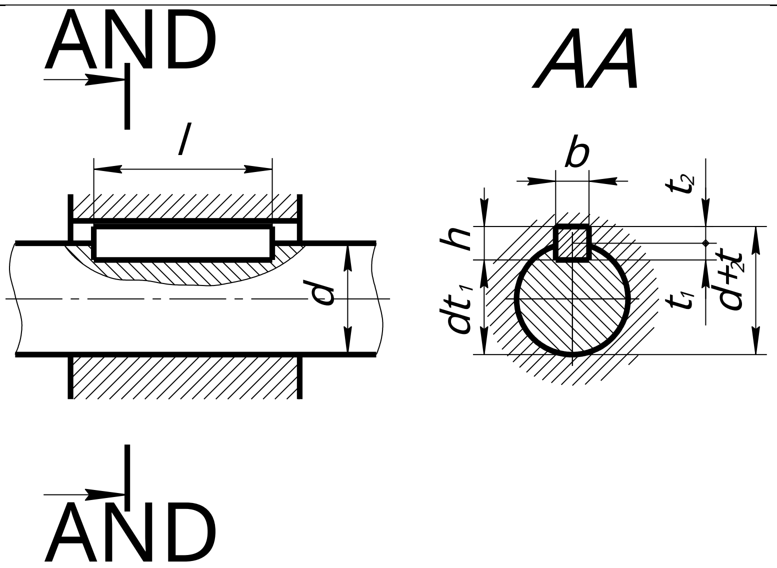
and) b) in) d)

Table D.1 – Options for selecting design features wheels

No. var.	*and*	*b*	*in*	*g*	No. var.	*and*	*b*	*in*	*g*
1	3	–	–	–	16	4	9	31	32
2	4	10	40	40	17	3.5	14	40	32
3	5	9	35	40	18	3.5	12	46	36
4	3.5	12	40	40	19	4	12	36	40
5	3	12	50	32	20	3	12	54	40
6	5	9	31	36	21	3	10	40	32
7	3	12	40	36	22	4	9	48	32
8	2.5	12	46	32	23	3	12	40	36
9	3	10	54	36	24	3	10	54	36
10	3	10	40	32	25	4	10	31	32
11	5	9	31	36	26	4	9	40	40
12	3.5	12	36	32	27	2.5	16	46	36
13	3.5	12	40	36	28	3	12	50	40
14	4	9	36	32	29	5	9	31	40
15	3	12	40	32	30	4	12	31	36

Appendix D

Table D.1 – Prismatic keys (GOST 23360–78)



Diameter shaft d	Cut keys $b \times h$	Depth keyway		Key length		
		Shaft t_1	Bushing t_2	from	to	A series of lengths l
From						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, 180, 200, 220, ..., 500.
10 to 12	4×4	2.5	1.8	8	45	
10 to 12	5×5	3.0	2.3	10	56	
12 to 17	6×6	3.5	2.8	14	70	
17 to 22						
22 to 30	8×7	4.0	3.3	18	90	
30 to 38	10×8	5.0	3.3	22	110	
38 to 44	12×8	5.0	3.3	28	140	
44 to 50	14×9	5.0	3.3	36	160	
50 to 58	16×10	6.0	4.3	45	180	
58 to 65	18×11	7.0	4.4	50	200	
65 to 75	20×12	7.5	4.9	56	220	

Keys come in three designs. Design 1 is not indicated in the key symbol.

Record

Key 8×7×40 GOST 23360-78

reads as follows: key version 1, width b=8 mm, height b=7 mm, length l =40 mm;

Key 8×7×40 GOST 23360-78

– the same key version 2.