MINISTRY OF EDUCATION THE REPUBLIC OF BELARUS
Belarusian National Technical University

Department «Machine Science and Machine Parts»

MECHANISM THEORY AND MACHINE PARTS

Textbook

Approved by the Educational and metodological association in mechanical engineering equipment and technologies for the students of speciality 1-36 01 01 «Technology of mechanical engineering»

Minsk 2021

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ББК 34.41 я7
Т33

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Т33 **Mechanism** theory and machine parts: textbook / A. T. Skoybeda [and etc.]. – Minsk : BNTU, 2021. – 41 p.
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This textbook is intended for technical universities students performing laboratory works in such disciplines as "Machine parts", "Fundamentals of design and machine parts", "Theory of machines and mechanisms", "Technical mechanics", "Applied mechanics". The textbook gives the brief theoretical information, the descriptions of laboratory installations, the methods of work, the recommendations for making reports and the control questions.

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Laboratory Work № 1

STRUCTURAL ANALYSIS OF PLANAR MECHANISMS

Purpose of the work

To master the skills in drawing structural schemes of mechanisms and making structural analysis.

Basic Safety Rules

Study the mechanisms carefully, avoid them falling on the floor.

How to do lab work

For the first mechanism:

1) study the mechanism, set its purpose (for converting movements), select the position of the mechanism, which clearly shows the relative location of the links;
2) draw a structural diagram of the mechanism, using the symbols of links and pairs;
3) number all links, mark kinematic pairs with capital letters of the Latin alphabet, fill in the table of kinematic pairs in your report;
4) count the number of moving links and kinematic pairs. Determine the degree of movability of the mechanism;
5) mark the input link with an arrow;
6) decompose the mechanism into structural groups (Assur groups) and 1 class mechanisms. Draw each group separately, state its class, and type;
7) write a structural formula for the mechanism.

For the second mechanism:

1) study the mechanism, set its purpose (for converting movements), select the position of the mechanism, which clearly shows the relative location of the links;
2) draw a structural diagram of the mechanism, using the symbols of links and pairs;

3) number all links, mark kinematic pairs with capital letters of the Latin alphabet, fill in the table of kinematic pairs in your report;

4) count the number of movable links and kinematic pairs, de-ter-mine the degree of movability of the mechanism;

5) if the mechanism does not coincide with the number of input links, then identify the links that create passive bonds or excess degrees of freedom, and draw a diagram of the mechanism, re-moving such links, and recalculate the degree of movement of the mechanism.

Theoretical information

Basic definitions and symbols

A link is one or more rigidly connected to each other solid bodies that are part of the mechanism.

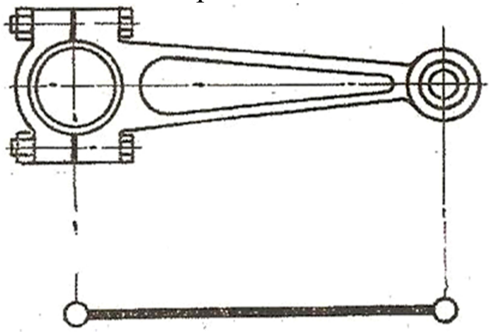


Fig. 1.1

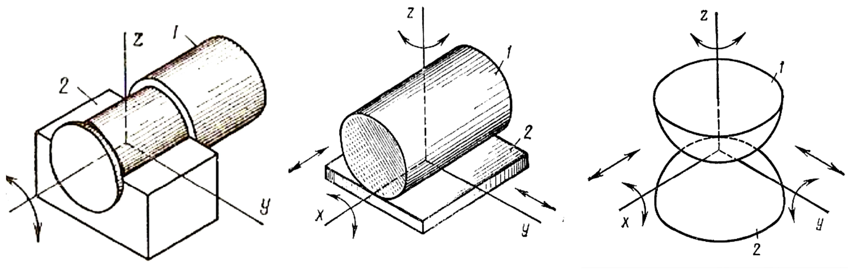
Links in the diagram do not show their structur-al form, but only deter-mine the positions of kin-ematic pairs and geomet-ric features of the links. So, for example, when drawing a connecting rod (fig. 1.1) on the diagram, you do not need to draw all the details that it consists of, but it is important to note only the positions of the bushing axes and the rigid connection between them.

Initial Link: Link in respect of which the generalized coordi-nate is prescribed.

A kinematic pair is a connection of two links that allows their relative motion. The links of this pair can touch each other with surfaces, lines and points, called elements of the kinematic pair.

If the elements are points or lines, then the pairs are called higher (fig. 1.2, *b*, *c*), and if the surfaces are lower (fig. 1.2, *a*).

Examples of the most common kinematic pairs are shown in tab. 1.1.

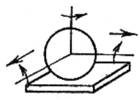
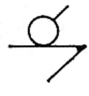
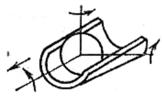
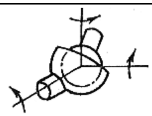
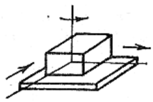
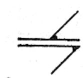


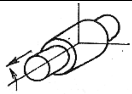
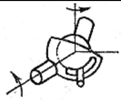
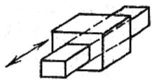
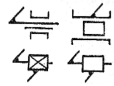
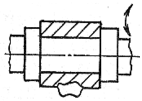
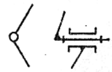
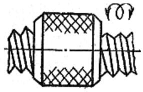
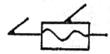
a *b* *c*

Fig. 1.2

Table 1.1

Kinematic pairs

Pair class	Number of bonds	Number of degrees of freedom	Pair name	Figure	Symbol
1	1	5	Ball-plane		
2	2	4	Ball-cylinder		
3	3	3	Spherical		
3	3	3	Planar contact pair (sandwich pair)		

4	4	2	Cylindrical		
4	4	2	Spherical with pin		
5	5	1	Prismatic (sliding joint)		
5	5	1	Revolute		
5	5	1	Helical (screw pair)		

Generalized Coordinate: Coordinate that represents a set of independent variables determining the positions of the links of a mechanism with respect to the frame.

Degree Of Freedom (Mobility) Of A Mechanism is the number of independent parameters (generalized coordinates) that uniquely determine the position of the mechanism links relative to the frame. Generalized coordinates can be either the angular coordinates of the links, or the linear coordinates of the points of the link. Each link before forming kinematic pairs with other links has 6 degrees of freedom in space (3 linear movements along the coordinate axes and 3 rotational relative to them). If you connect links with kinematic pairs, the relative movement of the links will be limited by the imposed bonds. Each pair of the 1st class imposes one bond, the 2nd class imposes two bonds, and so on. So the degree of spatial mechanism mobility can be calculated as

$$W = 6n - 5p_5 - 4p_4 - 3p_3 - 2p_2 - p_1,$$

where p_5, p_4, p_3, p_2, p_1 – the number of 5, 4, 3, 2, 1 class kinematic pairs.

In plane motion, each link have no more than three degrees of freedom, and pairs impose only two or one bonds, so the degree of planar mechanism mobility can be calculated as

$$W = 3n - 2p_5 - p_4.$$

This formula can be transformed into $W = 3n - 2p_L - p_H$, where p_L, p_H – numbers of lower and higher kinematic pairs.

Basic generation principle of planar mechanisms

Every planar mechanism can be generated by the successive joining of simple structural groups (or ***Assur groups***) to the 1 class mechanism.

1 class mechanisms consist of 1 input link, 1 kinematic pair of 5 class and frame (fig. 1.3).

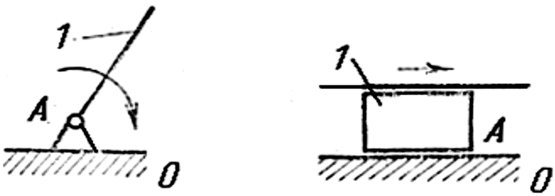


Fig. 1.3

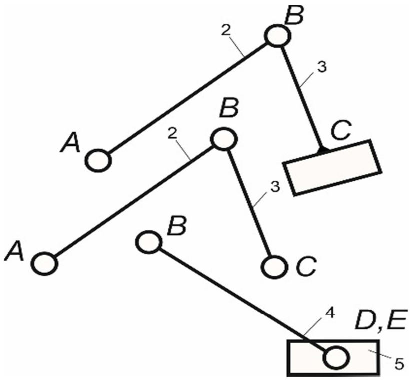


Fig. 1.4

The simplest Assur groups are shown in the fig 1.4. They consist of 2 links and 3 lower pairs.

Assur groups have movability $W = 3n - 2p_L - p_H = 0$, being connected to the frame. That is why they can consist of 2 links and 3 lower pairs or 4 links and 6 lower pairs or 6 links and 9 lower pairs. Otherwise the movability will not be zero.

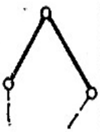
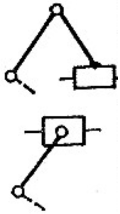
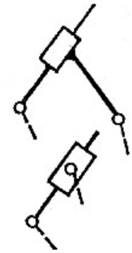
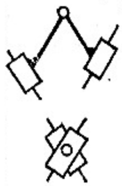
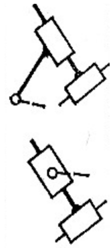
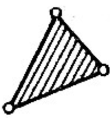
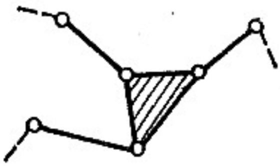
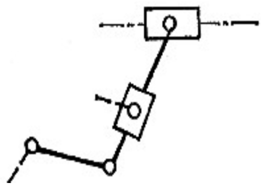
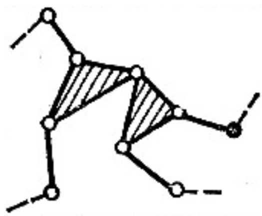
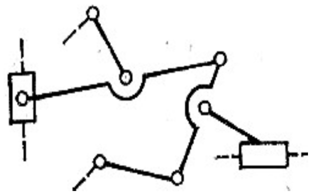
Assur group doesn’t change mechanism movability, being connected to it. Assur groups are characterized by it class (see tab. 1.2 below).

The group class is determined by the highest grade of closed loop.

The class of loop – number of kinematic pairs in the loop. See tab. 1.2 for details.

Table 1.2

Assur groups

<table>
<tr><th>Class</th><th>Closed chain (loop)</th><th colspan="5">Assur groups</th></tr>
<tr><td rowspan="2">II</td><td></td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>Type</td><td>1</td><td>2</td><td>3</td><td>4</td><td>5</td></tr>
<tr><td rowspan="2">III</td><td rowspan="2"></td><td colspan="3"></td><td colspan="2"></td></tr>
<tr><td colspan="3"></td><td colspan="2"></td></tr>
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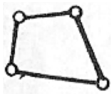
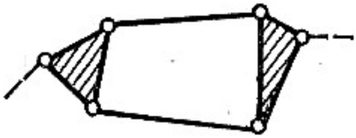
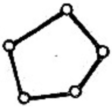
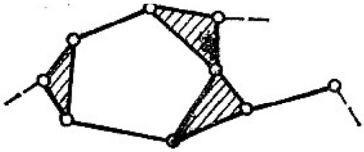
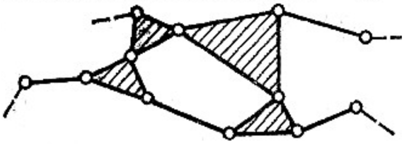
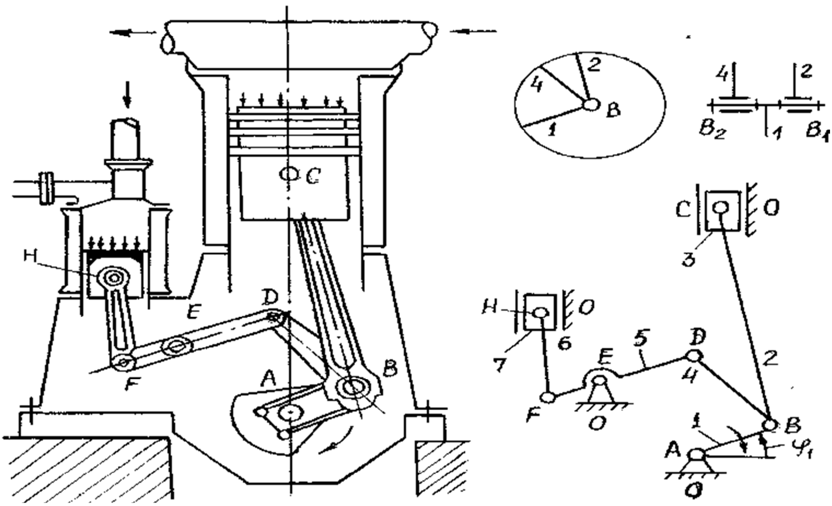
IV		
V		
		

The order of joining Assur groups is described by the structural formula of the mechanism. For example, for the mechanism (fig. 1.5, *b*), the structure formula has the following form:

I(0,1) → II(2,3); I(0,1) → II(4,5) → II(6,7)



a *b*

Fig. 1.5

BELARUSIAN NATIONAL TECHNICAL UNIVERSITY

Department «Machine Science and Machine Parts»

Laboratory Work № 1

STRUCTURAL ANALYSIS OF PLANAR MECHANISMS

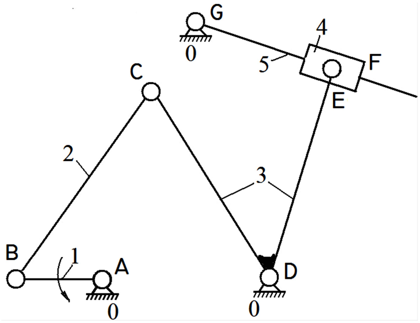
Purpose of the work:

To master the skills in drawing structural schemes of mechanisms and making structural analysis.

The work was performed by:_____________________
Group:_____________________
The work was accepted by:_____________________

Minsk ____

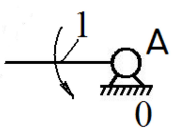
Structural diagram



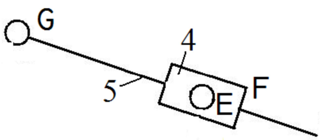
Kinematic pair	№ of links	Pair name
A	*0–1*	Revolute
B	*1–2*	Revolute
C	*2–3*	Revolute
D	*3–0*	Revolute
E	*3–4*	Revolute
F	*4–5*	Prismatic
G	*5–0*	Revolute

The number of the movable links $n = 5$.
The number of lower kinematic pairs $p_L = 7$.
The number of higher kinematic pairs $p_H = 0$.
Movability of mechanism.

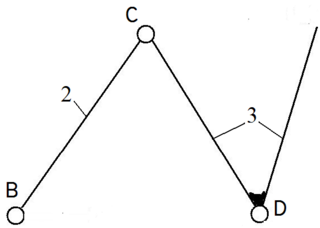
Decomposition of the mechanism into Assur groups with indication of their class and type



Mechanism of 1 class



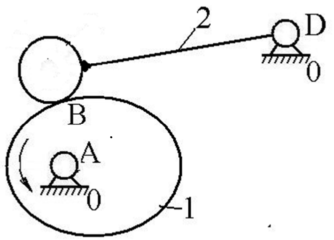
Assur group of 2 class, 1 type



Assur group of 2 class, 3 type

Structural formula $I\ (\ 0,1\ ) \rightarrow II\ (\ 2,\ 3\ ) \rightarrow II\ (\ 4,5\ ).$

Structural diagram
with higher kinematic pair



Kinematic pair	№ of links	Pair name
A	*0–1*	Revolute
B	*1–3*	Higher
C	*2–3*	Revolute
D	*2–0*	Revolute

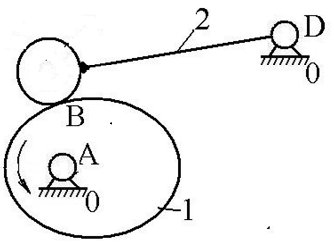
The number of the movable links $n = 3$.

The number of lower kinematic pairs $p_L = 3$.

The number of higher kinematic pairs $p_H = 1$.

Movability of mechanism $W = 3n - 2p_L - p_H = 3 \cdot 3 - 2 \cdot 3 - 1 = 2$.

Roller 3 is installed to reduce friction in the higher kinematic pair. It creates an extra degree of freedom and must be removed by a hard connection to the link 2.



Kinematic pair	№ of links	Pair name
A	*0–1*	Revolute
B	*1–2*	Higher
D	*2–0*	Revolute

The number of the movable links $n = 2$. The number of lower kinematic pairs $p_L = 2$. The number of higher kinematic pairs $p_H = 1$.

Movability of mechanism $W = 3n - 2p_L - p_H = 3 \cdot 2 - 2 \cdot 2 - 1 = 1$.

Questions for Control

1. What is the degree of freedom (mobility) of a mechanism?
2. Write down the formula for calculating the degree of mobility of spatial mechanisms and explain the parameters included in it.
3. Write down the formula for calculating the degree of mobility of flat mechanisms and explain the parameters included in it.
6. Formulate the basic generation principle of planar mechanisms.
7. What does a 1 class mechanism consist of?
8. What kinematic chain is called the Assur group?
9. What combination of links and kinematic pairs takes place in Assur groups?
10. How is the class and type of Assur group determined?
11. Give examples of Assur groups of the 2nd class.
12. Give examples of Assur groups of the 3rd class.
14. What formula shows the structure of the mechanism?

Laboratory Work № 2

PARAMETER DETERMINATION OF TWO-STAGE HELICAL REDUCTION GEAR

Purpose of the work

1. Studying the gearbox design and basic requirements for its assembly.
2. Determining the main parameters of reduction gear.
3. Dimensions of gears and gear trains.

Basic Safety Rules

1. While nuts unscrewing, the reduction gear should not move along the table.
2. Before measuring the gear parameters, the assembly unit (shaft with gears and bearings) must be placed on special supports.

Construction and description of PM-250 reduction gear (fig. 2.1)

Reduction gear is a mechanism consisting of gear or worm gears made in the form of a separate assembly unit and designed to reduce angular speed and therefore to increase the torque of the driven shaft compared to the driving shaft.

The pinions are usually made integrally with the shaft, from rolled stock. In small reduction gears the gears are forged, in large gears are cast.

The gear frame is made detachable along the plane in which all shaft axes are. This provides for a convenient assembly of the gearbox when each shaft is assembled in advance with parts mounted thereon.

The lower part of the frame (gear casing) is connected with the upper one (gear cover) by bolts and two pins fixing the relative position of the frame. The casing and cover are provided with ribs to increase rigidity.

The bearings mounted on a single shaft are usually of the same size. This makes it possible to obtain different assembly options.

The gears are installed on shafts with interference or using key (spline) joints, that is why the assembly units is assembled under press.

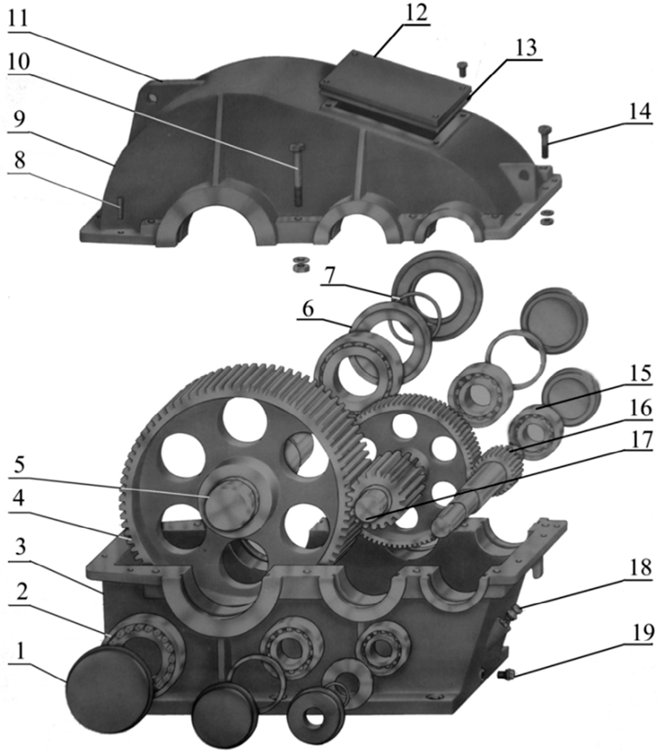


Fig. 2.1. Two-stage helical reduction gear:
1 – cast-in bearing cap; *2* – single-row radial ball bearing (or tapered roller bearing); *3* –gear casing; *4* – gear of low-speed shaft; *5* – driven (output)or low-speed shaft; *6* – ring; *7* – adjusting ring; *8* – pin; *9* – gear cover; *10* – tightening bolt; *11* – eye; *12* – oil hole cover; *13* – gasket; *14* – bolt for attachment of cover and casing; *15* – single-row radial ball bearing (or tapered roller bearing); *16* – driving (input or high-speed) shaft; *17* – intermediate shaft; *18* – oil-level gauges; *19* – oil drain plug

The outer rings of bearings are mounted in the gear casing on the landing providing a small clearance that allows a ring to be turned in operating because of which new sections of the race come into contact with bodies of swing. Besides, the clear ancefa-cilitates the movement of rings necessary for adjusting the tightness of bearings. The bearing caps may be cast-in or flange. The cast-in caps are easy for using as they provide access to separate bearings for survey without dismantling the whole reduction gear. Flange caps make the reduction gear construction simpler and reduce the weight of reduction gear.

The seals in cast-in caps prevent the falling of mechanical particles into bearings and the internal cavity of the reduction gear also do not allow the outflow of oil. Lip-type reinforced seals are more reliable and durable.

The gear cover has a special hole for filling oil and which may be used for viewing. This hole is covered by cover 12, which has the vent channel. This channel is intended to make the pressure in the case and the atmospheric pressure equal. At the base of gear casing 3 there is an oil draining hole closed by plug 19.

The escape of oil on the connector is prevented by covering the connector planes with spirit varnish or liquid glass with the subsequent bolt tightening. The application of gaskets is not allowed because while tightening the bolts the deformations of bearing rings and the breakage of landings are possible. To raise the case cover and the gear reduction eyes 11 are used.

How to do lab work

1. Determine the main, overall and mounted dimensions of the reduction gear (fig. 2.2).

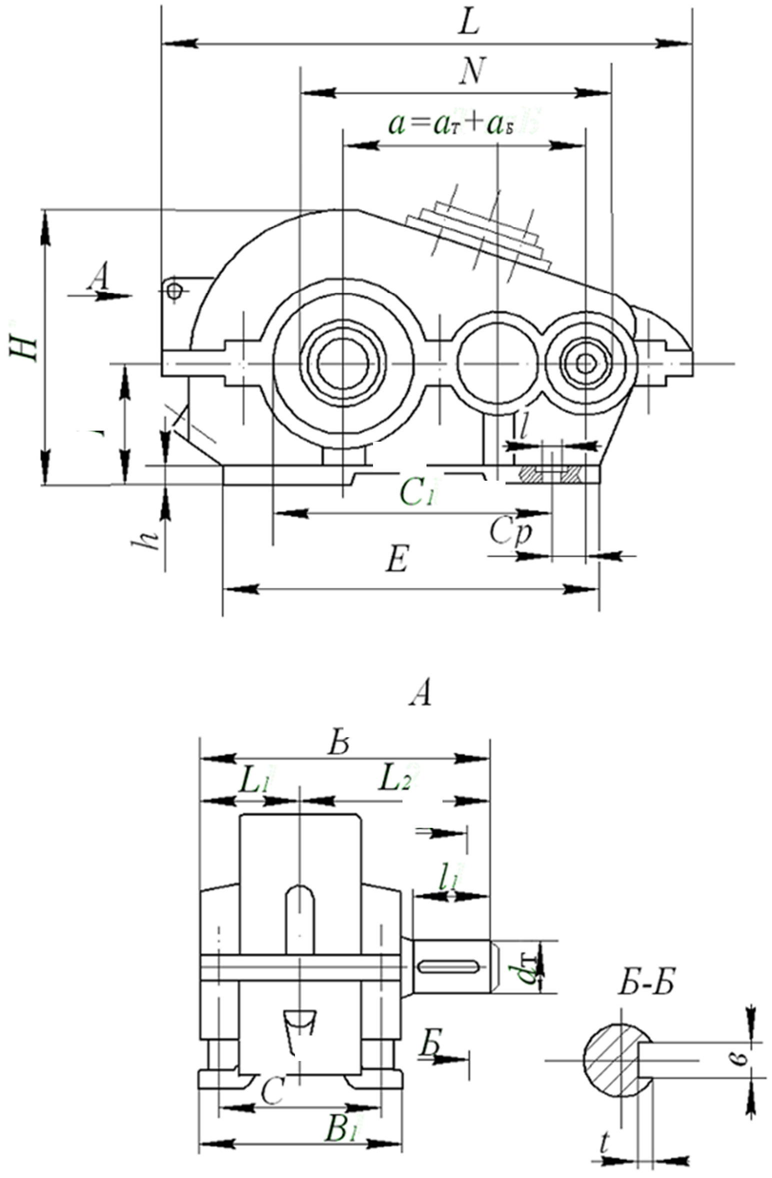


Fig. 2.2. Main dimensions of reduction gear

Put down the results of measurements into tab. 2.2.

The overall dimensions include length, height and width of reduction gear, including the protruding ends of input and output shafts. These dimensions are taken into account when arranging the reduction gear in the drive device.

Mounting dimensions mean the distances and the location of surfaces of reducer connection relative to other elements. These include the dimensions of the mounting plane by which the reduction gear is mounted on the plate or frame, hole sizes for bolts to fix the reduction gear, shaft end sizes (ends of input and output shafts) and the dimensions determining their arrangement relative to each other and relative the mounting plane.

Centre distance (see fig. 2.2):

$$a = N - \frac{d_{Б}}{2} - \frac{d_{Т}}{2},$$

where $d_{Б}$, $d_{Т}$ – are diameters of input and output shaft ends.

2. How to disassemble the reduting gear:

a) unscrew the plug (19) and drain the oil;

b) unscrew the nuts, remove the bolts and remove the cover (9).

c) remove cast-in bearing caps and adjusting rings;

d) remove the shafts with the parts, mounted on it;

e) remove cast-in bearing through caps;

f) familiarize yourself with the design and purpose of all gear parts.

3. Determine the gear parameters. Put down the results of measurements and calculations into tab. 2.3:

a) count the number of pinion and gear teeth of high-speed and low-speed stages (refer to fig. 2.1);

b) measure outside gear and pinion diameters (d_a) – (fig. 2.3);

c) measure the face width of the gears (b);

d) determine the direction of tooth contact line for the gear (left or right);

e) determine the helix angle of tooth contact line (β).

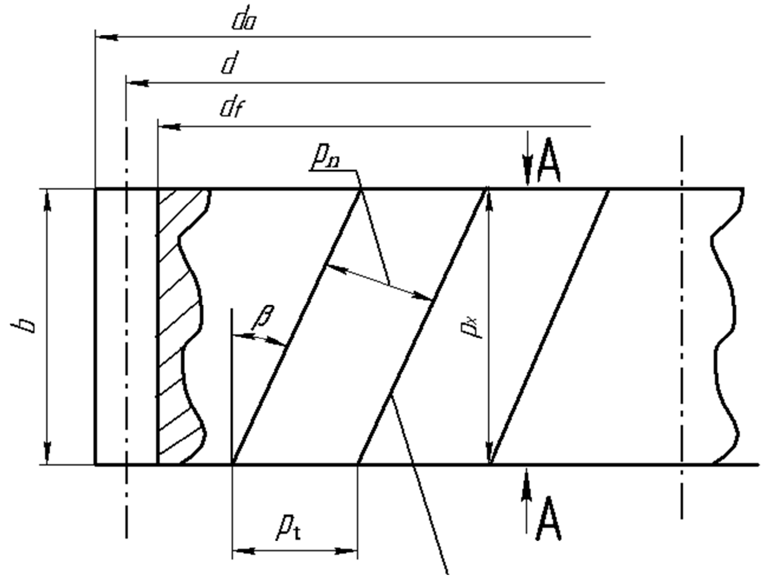


Fig. 2.3. To determine the size of gears:

$p_n = \pi \cdot m_n$ – normal pitch;

$p_t = \pi \cdot m_t$ – circular pitch;

$p_x = p_n \cdot \sin\beta$ – axial pitch

4. The reducing gears assembled in the reverse order.

Theoretical information, required to perform the work

The helical gears have normal m_n, transverse (circular) m_t and axial m_x modules. Module of a gear is a linear quantity which is smaller than the pitch by factor of π: $m = p / \pi$. ($m_t = p_t / \pi$; $m_n = p_t / \pi$; $m_x = p_x / \pi$).

m_n and m_t are constrained: $m_n = m_t \cdot \cos\beta$.

In spur gearing, circular and normal pitches and modules coincide.

β is the helix angle of tooth contact line on the reference cylinder.

β usually ranges from 8 to 18° (sometimes running into 22°).

Normal module is standardized. Two standard module series are recommended:

Series 1: 1; 1.25; 1.5; 2; 2.5; 3; 4; 5; 6; 8;10;12…

Series 2: 1.125; 1.375; 1.75; 2.25; 2.75; 3.5; 4.5; 5.5; 7,0; 9,0; 11…

Preference should be given to series 1.

According to international standards, the principal geometric relationships for uncorrected gearing are listed below (fig. 2.4).

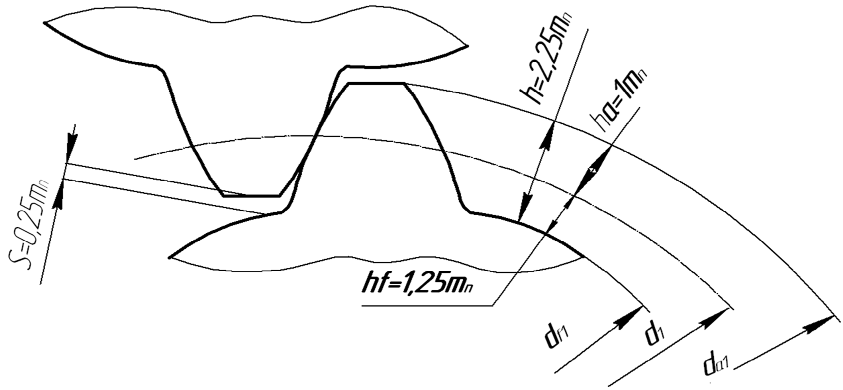


Fig. 2.4

Addendum $h_a = m_n$; dedendum $h_f = 1,25m_n$; whole depth of teeth $h = 2,25m_n$.

Reference diameter (d) = Pitch diameter (d_w) (if $x_1 = x_2 = 0$)

x – the tool displacement factor.

$$d_1 = d_{w1} = m_t z_1 = \frac{m_n}{\cos\beta} z_1;$$

$$d_2 = d_{w2} = m_t z_2 = \frac{m_n}{\cos\beta} z_2.$$

Outside (addendum circle) pinion and gear diameters:

$$d_{a1} = d_1 + 2m_n; \qquad d_{a2} = d_2 + 2m_n.$$

Center distance:

$$a_w = \frac{d_{w1} + d_{w2}}{2} = \frac{d_{a1} - 2m_n + d_{a2} - 2m_n}{2} = \frac{d_{a1} + d_{a2} - 4m_n}{2},$$

from here

$$m_n = \frac{d_{a1} + d_{a2} - 2a_w}{4}.$$

The calculated m_n is consistent with the standard.

Center distance:

$$a_w = \frac{d_1 + d_2}{2} = \frac{m_n}{2\cos\beta}\left(z_1 + z_2\right),$$

from here

$$\cos\beta = \frac{m_n(z_1 + z_2)}{2a_w}.$$

The value of β calculated from this equation is consistent with those given in the tab. 2.1.

Table 2.1

Angles of tooth line inclination on dividing cylinder

a_w	100		125		150	
m_n	z_Σ	β	z_Σ	β	z_Σ	β
1,00	198	8°06 34	247	8°53 06	297	8°06 34
1,25	158	9°04 07	198	8°06 34	237	9°04 07
1,50	132	8°06 34	165	8°06 34	198	8°06 34
1,75	113	8°36 09	141	9°14 55	169	9°39 21
2,00	99	8°06 34	123	10°15 47	148	9°22 00
2,25	99	8°06 34	110	8°06 34	132	8°0634
2,50	99	8°06 34	99	8°06 34	118	10°2834
3,00	99	8°06 34	99	8°06 34	99	8°06 34

Tool displacement factor:

$$x_1 = \frac{d_{a1}}{2m_n} - \frac{z_1}{2\cos\beta} - 1;$$

$$x_2 = \frac{d_{a2}}{2m_n} - \frac{z_2}{2\cos\beta} - 1.$$

Outside (addendum circle) pinion and gear diameters (for corrected gearing):

$$d_a = m_n\left(\frac{z}{\cos\beta} + 2 + 2x\right).$$

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Department «Machine Science and Machine Parts»

Laboratory Work № 2

PARAMETER DETERMINATION OF TWO-STAGE HELICAL REDUCTION GEAR

Purpose of the work:

1. Studying the gearbox design and basic requirements for its assembly.
2. Determining the main parameters of reduction gear.
3. Dimensions of gears and gear trains.

The work was performed by:______________________
Group:______________________
The work was accepted by:______________________

Minsk ____

General view of the reduction gear

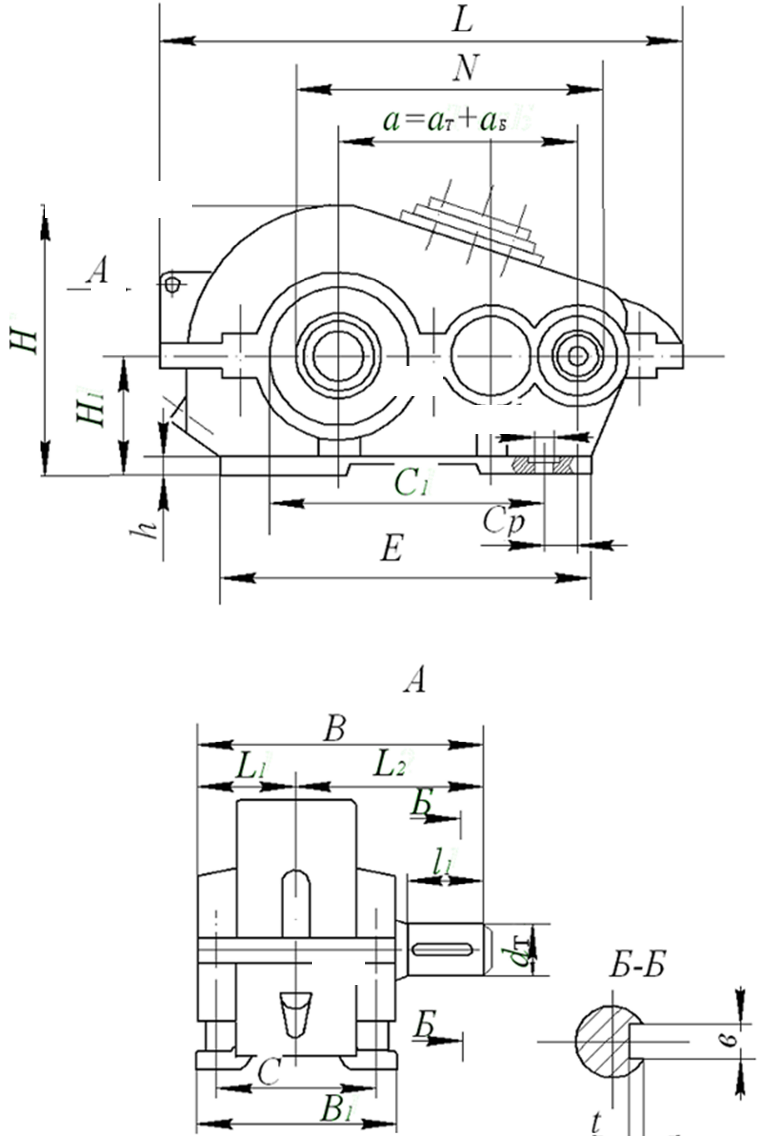


Table 2.2

The overall and mounting dimensions of reduction gear

Size designation	H	L	B	H_1	h	C_0	C_1	C	E	B_1	L_1	L_2	$a_{wб}$	$a_{wт}$	$d_б$	$d_т$	N
Dimensions, mm																	

Table 2.3

Values of gears and gear trans

Parameter	Formula	Values			
		Fast stage		Slow stage	
		Calcu-lation	Measu-rement	Calcu-lation	Measu-rement
1	2	3	4	5	6
Gear type					
Centre distance a_w, mm					
Number of pinion teeth Z_1, Z_3					
Number of gear teeth Z_2, Z_4					
Outside pinion diameters d_{a1}, d_{a3}, mm					
Outside gear diameters d_{a2}, d_{a4}, mm					
Gear face width b_2, b_4, mm					
Direction of tooth contact line for the gears (*L* – left; *R* – right)					
Helix angle of tooth contact line β					
Gear ratio, $U = Z_{gear} / Z_{pinion}$					

End of tab. 2.3

1	2	3	4	5	6
Gear width factor, $\psi_{ba} = b / a_w$					
Circular module m_t, mm					
Normal module m_n, mm					
Reference pinion diameter d_1, d_3, mm					
Reference gear diameter d_2, d_4, mm					
Circular pitch P_t, mm					
Normal pitch P_n, mm					
Total transmission ratio i					

Control questions

1. Purpose, device and classification of gearboxes.
2. Structures of sealing devices.
3. Ways to lubricate gears and gearbox bearings.
4. Why is the pinion in the gearbox wider than the gear by 4…6 mm?

NOTE

1. Measuring tool: metal line 0–500 mm; Calipers (trammel) 0–250 mm with division value 0,05 mm.

2. Determine the Centre distance a_w.

The Centre distance cannot be determined by direct measurement, so the measurement is as follows:

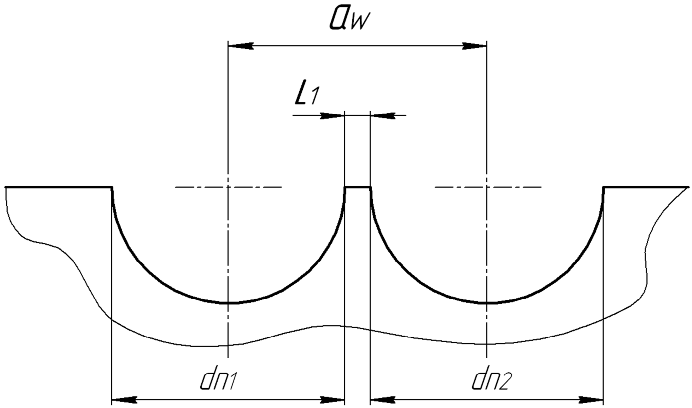
2.1. Disassembly of reduction gear box.

2.2. Measure external diameters of bearing rings of high-speed and intermediate shaft, d_{b1} and d_{b2} by the rod.

2.3. Measure the bridge between the L1 holes.

2.4. The Centre distance of the first stage of the reduction gear box is calculated by:

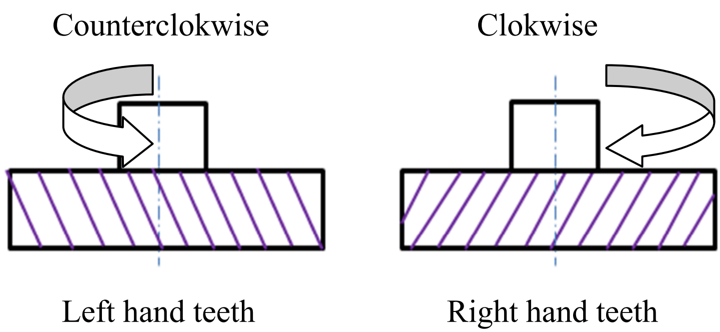
$$a_w = \left(\frac{d_{b1} + d_{b2}}{2} \right) + L_1.$$



Similar measurements are made for the second (slow-moving) stage of the reduction gear box.

The direction of tooth contact line

Place the gear on the table as shown. Look at the arrow. View direction



You can view from either side, but only along the axis of the gear.

Laboratory Work № 3

PARAMETER DETERMINATION OF WORM REDUCTION GEAR

Purpose of the work

1. Studying the gear box design.
2. Determining the main parameters of the worm and worm gear.
3. Acquaintance with the sequence of the worm gear axial position change.
4. Bearing adjustment.

Basic Safety Rules

1. Do not move the gearbox on the table surface while disassembling.
2. Preliminarily, before measurements, install assembly units on special supports.

Construction and description of the worm reduction gear

The reduction gears may have the following arrangements of the worm and worm gear:

– worm under the worm gear-applied for peripheral velocities of the worm up to 4 or 5 ms^{-1} with oil-bath lubrication for the worm, permits high power transmission on the basis of heating but is less favorable with respect to oil leakage. In this case, the maximum permissible oil level shall pass along the centers of the lower rolling bodies of the bearings. If the worm is immersed in the oil by less than 2,5 *m*, where *m* is the worm module, sprinklers are used to bring the oil into engagement;

– worm over the worm gear-used for high-speed drives to avoid extra losses due to the splashing of the oil by the high-speed worm, with oil-bath lubrication for the worm gear;

– worm with a horizontal axis mating with a worm gear having a vertical axis;

– worm with a vertical axis mating with a worm gear having a horizontal axis.

Gear ring *9* (fig. 3.1) is made of materials having antiscouring and antifriction properties (bronzes, brasses or grey cast irons). The gear center is made of steel or cast iron. To support a worm shaft and a worm gear the tapered roller bearing are established. Inner rings of bearings are placed on shafts with interference to prevent turning of necks and their flaring. Outer rings of bearings in housing are installed with slight clearance facilitating axial variation of rings for adjustment of bearing tension and axial position of worm gear.

Sealing rings made of oil-impregnated felt are arranged in bearing cap *5*. The lip-type seals prevent oil from flowing out of reduction gear box and dust and dirt ingress into bearings and engagement. Cuff seals are more reliable. The gear casing 4 and the cover *13* are made of cast iron. These materials are provided with ribs in order to increase rigidity and improve cooling surface. In the lower part of the housing there is a structure closed with oil drain plug *1*. Through this opening the oil is removed from the housing. To measure the oil level, a rod oil – level gauges *2* is used. On the cover *14* of the inspection hole there is an air vent *15* for levelling pressure in the reduction gear and atmospheric pressure. Oil is poured into reduction gear box through inspection hole (hatch), worm forest teeth are checked and contact spots and their value are observed at adjustment of worm gear axial position.

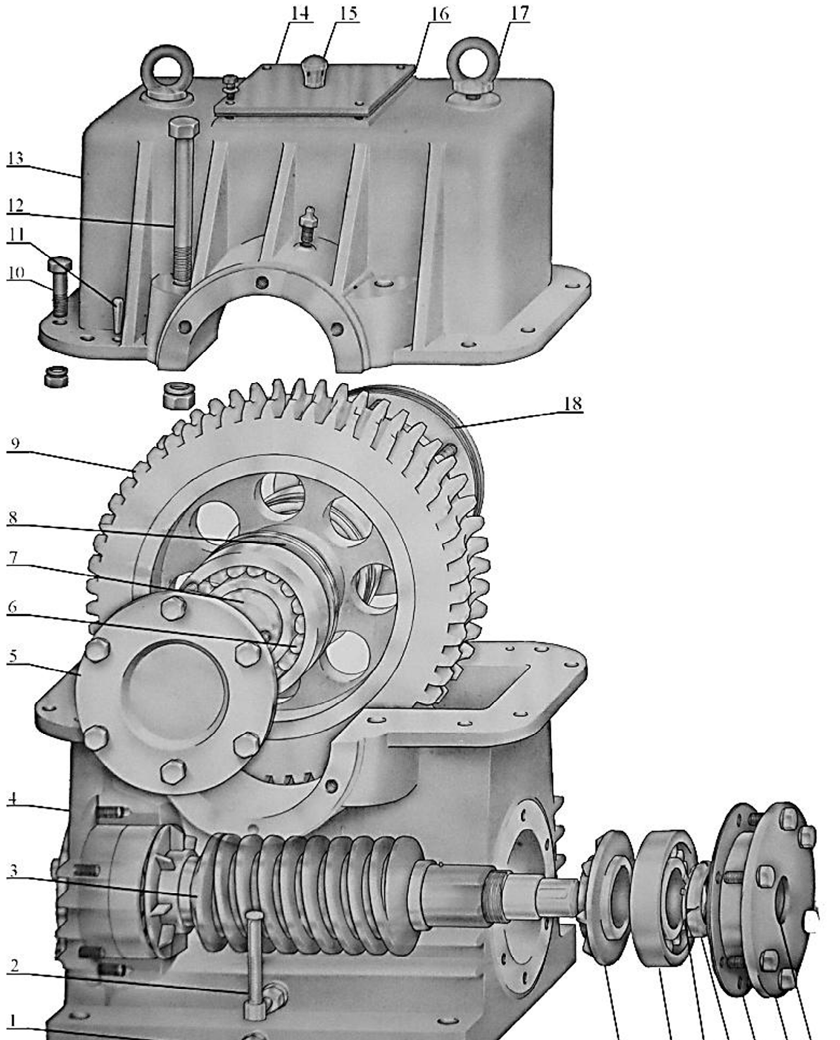


Fig. 3.1. Worm reduction gear:
1 – oil drain plug; *2* – oil-level gauges; *3* – worm; *4* – gear casing; *5* – bearing cap; *6* – tapered roller bearing; *7* – driven shaft; *8* – ring; *9* – worm gear; *10* – bolt for attachment of cover and casing; *11* – pin; *12* – bolt; *13* – gear cover; *14* – oil hole cover; *15* – air vent; *16* – gasket; *17* – eye-bolt; *18* – adjusting washers; *19* – impeller; *20* – multi-pin locking washer; *21* – round splined nut; *22* – felt seal

How to do lab work

1. Measure diameters of low-speed shaft d_t and high-speed shafts d_b, and distances h_1 and h_2 from the base plane of the housing to the upper points of the shafts d_t and d_b. Put down the results of measurements into tab. 3.1.

2. Calculate the center distance from the measurement data according to Equation 4 of the tab. 3.1 of the report. Put down the calculation result in the same table. Compare the design value a_w with the standard value according to GOST 2144-76.

3. Remove plug *1* and drain oil.

4. Remove screws and cap *5* of worm gear shaft bearings.

5. Remove nuts of bolts *10*, *12* and gear cover *13*.

6. Remove worm gear with shaft and bearings. Put the shaft on the support.

7. Remove screws and worm bearing caps. Do not remove bearings from shaft.

8. Remove the worm with bearings and also put on the support.

9. Determine the gear parameters. Put down the results of measurements and calculations into tab. 3.1.

Theoretical information

This does not necessarily mean that all worm-gear drives must be standard. Sometimes, it may prove more convenient to use non-standard centre distances. If so, modified (long or short addendum) worm gears are employed. Alternatively, the same effect may be achieved by varying the transmission ratio or, more specifically, by modifying z_2.

Standard and modified worm gears are cut with the same tools. Since the hob and the worm must have identical dimensions, modification applies to the worm gear a lone. For a specified centre distance a_w, the *tool offset factor* is determined as

$$x = a_w / m - 0{,}5(q + z_2)$$

or, to put it differently,

$$a_w = 0{,}5m\,(q + z_2 + 2x).$$

To avoid undercutting or pointing, the recommended value is $x = \pm 0{,}7$ (or ± 1 on some rare occasions). x – the tool displacement factor.

In an offset worm gear

$$d_{a2} = d_2 + 2m\,(1 + x); \quad d_{f2} = d_2 - 2m\,(1{,}2 - x).$$

Also changed is one design dimension of the worm.

The pitch diameter $d_{w1} = m(q + 2x)$.

All the other dimensions remain unchanged.

The worm-diameter factor q is a function of the module m (z_1 = 1, 2, or 4):

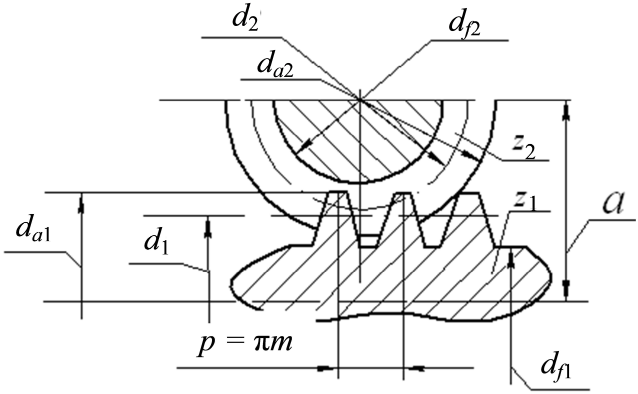
— for m = 1.6 mm, q = 10.0; 12.5; 16.0 or 20.0;
— for m = 2.0; 2.5; 3.15; 4.0; 5.0; 8.0; 10.0; 12.5 mm;
  q = 8.0; 10.0; 12.5; 16.0; 20.0;
— for m = 6.3 mm, q = 8.0; 10.0; 12.5; 14.0; 16.0; 20.0;
— for m = 16.0 mm, q = 8.0; 10.0; 12.5; 16.0;
— for m = 20.0 mm, q = 8.0 or 10.0.

For a worm gear without displacement, the values of parameters q and m required for calculating the dividing diameters of worm d_1 and wheel d_2, the dividing angle of lift γ, the calculated pitch of worm p_1 and other parameters (tab. 3.1 of the report) can be determined by the values a_w, (obtained from measurements) and by the number of teeth z_2:

$$a_w = a = 0{,}5(z_2 + q)m; \quad d_{a1} = m(q + 2).$$

From here

$$q = \frac{0{,}5d_{a_1} \cdot z_2 - 2a_w}{a_w - 0{,}5d_{a_1}}; \quad m = \frac{d_{a_1}}{q + 2}.$$



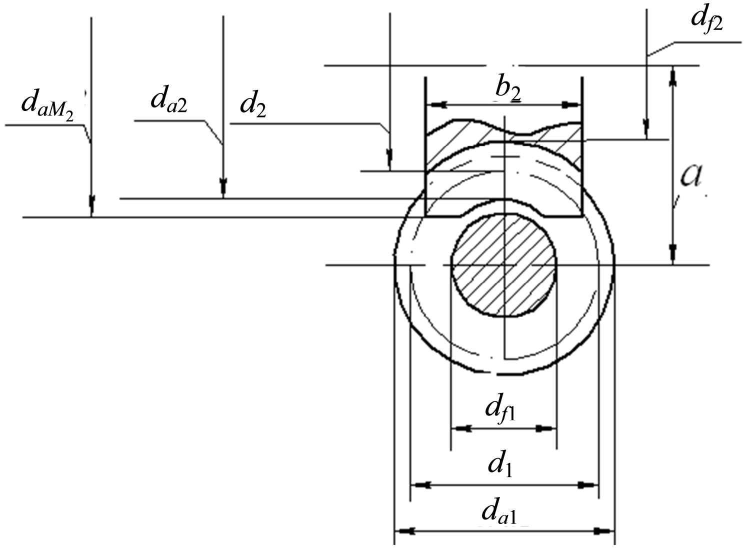


Fig. 3.2. Worm and worm gear parameters

Values q and m will be coordinated with GOST 2144-76:
m = 1.0; 1.25; 1.6; 2.0; 2.5; 3.15; 4.0; 5.0; 6.3; 8.0 – the I row;
m = 1.5; 3.0; 3.5; 6.0; 7.0; 12.0 – the II row;
q = 6.3; 8.0; 10.0; 12.5; 16.0; 20.0 – the I row;
q = 7.1; 9.0; 11.2; 14.0; 16.0; 18.0 – the II row.

Reduction gear assembly. Bearing adjustment. Axial position of worm gear

The parts of reduction gear and assemblies of worm gear and unit are installed in reduction gear box in the order opposite to that in which disassembling is performed. However, while assembling, it is necessary to adjust the pre-tension of the bearings first and then the axial position of the worm gear. Clearance in bearings causes radial and axial run-out and vibration of shafts. In order to eliminate these phenomena, bearings of increased rigidity are selected and their interference is adjusted.

Adjustment of preliminary interference of bearings is carried out by installing under flanges of covers metal gaskets of several individual thicknesses, for example: 0,1; 0,15; 0,2; 0,25 mm, etc., these gaskets are also used to adjust the axial spacing of the worm gear. The difference in thickness of the gaskets makes it possible to shift the rings of bearings and worm gear with accuracy up to 0,05 mm.

Adjustment of bearing tension can be performed separately for worm shaft and worm wheel shaft.

Sequence of bearings pre-tension control

Install worm or worm gear shaft in the housing together with bearings and covers without a set of gaskets so that the support surface of one of the covers is tightly pressed by screws to the housing, and there is a gap between the second cover and the housing.

Measure clearance δ between cover and housing with probe.

Select double set of metal gaskets of total thickness equal to

$$\delta + \Delta_{os},$$

Install packages with thickness of 0,5 $(\delta + \Delta_{os})$ between both caps and casing.

Where Δ_{os} is the permissible axial gap of a shaft. For example, if the shaft neck diameter is d = 30...50 mm, Δ_{os} = 0,05...0,1 mm is allowed.

Tighten the screws of bearing covers. Adjustment of axial position of worm gear is done by repositioning gaskets from one side to the other without change of their total thickness. If, for example, it is necessary to move the wheel by 0,1 mm, then from the right set the gasket is removed by 0,2 mm, and from the left the gasket is removed by 0,1 mm and they are removed in places. The distance between the ends of the covers is not limited, and therefore the previously adjusted pre-tension of the bearings is maintained; Shaft with all parts and cover moves to the left by 0,1 mm.

Axial position control sequence
Worm gear

Before assembling the reduction gear apply thin layer of paint on working surface of worm turns (place worm on support).

Turn the worm so that the wheel turns for full revolution.

Examine the location of contact spots through the inspection hole. Typical location of contact spots is shown in fig. 3.3, *a*, *b*, *c*.

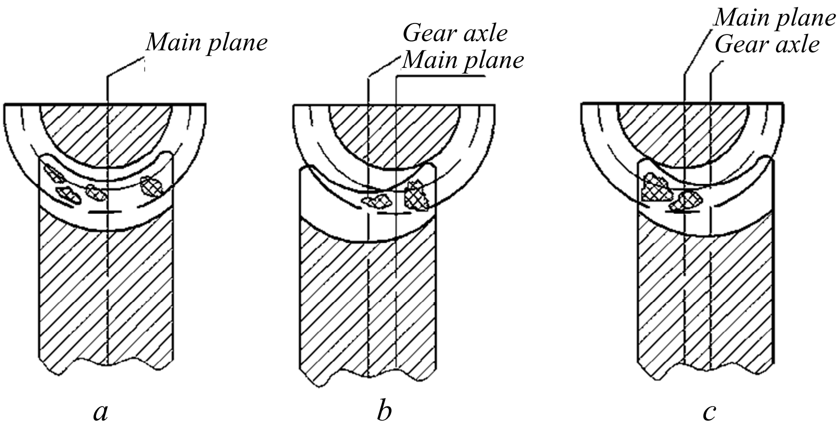


Fig. 3.3. Typical location of contact spots on worm gear teeth

Correct adjustment of gear axial position is controlled by contact spot ("by paint") on teeth.

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Department «Machine Science and Machine Parts»

Laboratory Work № 3

PARAMETER DETERMINATION OF WORM REDUCTION GEAR

Purpose of the work:

1. Studying the gear box design.
2. Determine the main parameters of the worm and worm gear.
3. Acquaintance with the sequence of the worm gear axial position change.
4. Bearing adjustment.

The work was performed by:______________________
Group:______________________
The work was accepted by:______________________

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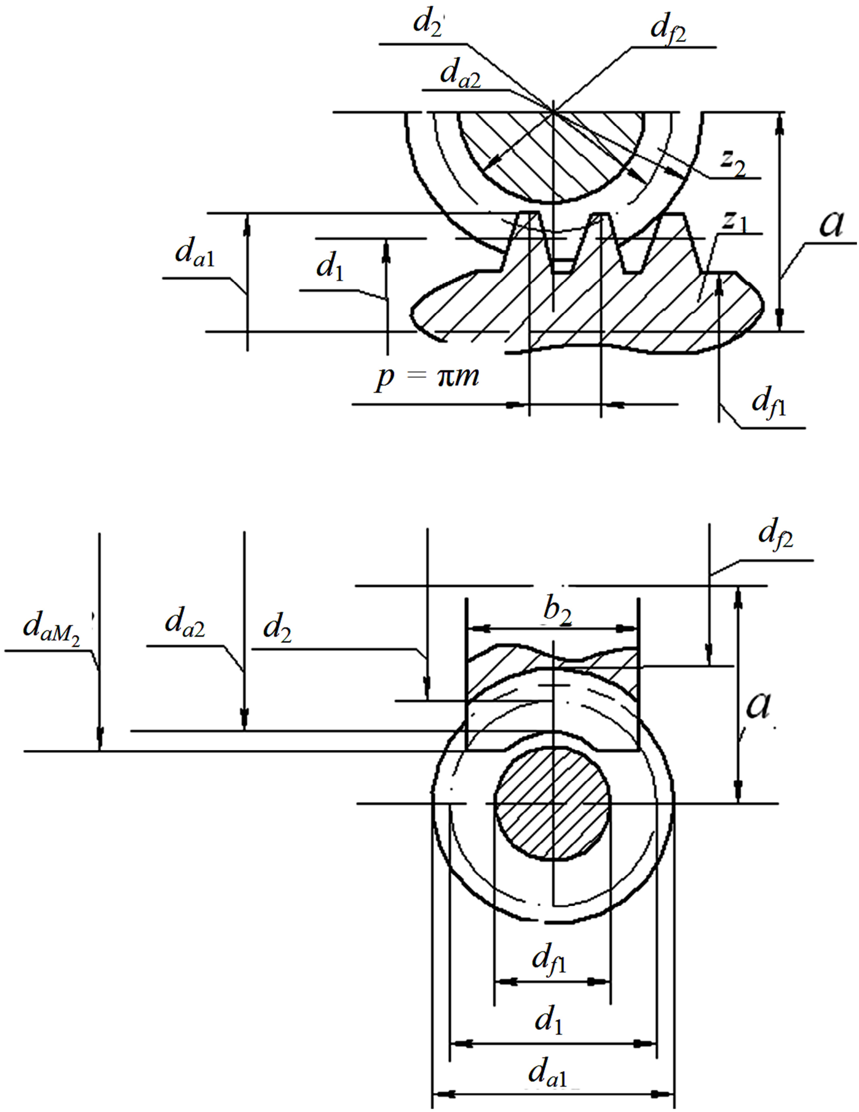

Parameters of the worm and worm gear

Table 3.1

Values of Worm and worm gear

№ P/p	Parameter	Formula	Values	
			Calcu-lation	Mea-sure-ment
1	2	3	4	5
reduction gear				
1	High-speed shaft diameter d_b, mm			
2	Low-speed shaft diameter d_t, mm			
3	Dimensions h_1 and h_2, mm			
4	Centre distance, mm	$a_w = (h_2 - \frac{d_t}{2}) - (h_1 - \frac{d_b}{2})$		
5	Specified Centre distance, mm	$a_w = 0,5(z_2 + q + 2x)m$		
worm				
6	Number of threads	z_1		
7	Diameter factor	$q = \frac{0,5d_{a1} \cdot z_2 - 2a_w}{a_w - 0,5d_{a1}}$		
8	Module, mm	$m = d_{a1} / (q + 2)$		
9	Outside diameter, mm	$d_{a1} = d_1 + 2m$		
10	Reference diameter, mm	$d_1 = mq$		
11	Pitch diameter, mm	$d_{w1} = (q + 2x)m$		
12	Pitch helix angle of the worm threads	$\mathrm{tg}\gamma = \frac{z_1}{q}$ (Accurate to second)		

1	2	3	4	5
13	Root diameter, mm	$d_{f1} = d_1 - 2,4m$		
14	Axial pitch, mm	$p_1 = \pi m$		
15	Threaded length $x = 0$	$b_1 \geq (11 + 0,06z_2)m$ * Z1 is 1 and 2		
* For grinding and milled worms, the value b1 obtained according to the table should be increased by 25 mm with m < 10 mm.				
worm gear				
16	Number of gear teeth	z_2		
17	Reference (= pitch diameter) diameter, mm	$d_2 = mz_2$		
18	Maximum diameter, mm	$d_{aM_2} \leq d_{a2} + \frac{6m}{z_1 + 2}$		
19	Addendum diameter, mm	$d_{a2} = d_2 + 2(1 + x)m$		
20	Face width, mm	$b_2 \leq 0,75d_{a1}$ Z1 is 1 and 2		

Questions for Control

1. Why is the oil level with the lower position of the worm limited to the center of the bearing rolling bodies?
2. Why is it unacceptable to place contact spots on worm gear teeth as shown in fig. 3.3, *b* and 3.3, *c* in?
3. Why do dynamic loads in the transmission increase with reduced rigidity of bearings in supports and with clearance in bearings?
4. What is the reason for the different position of the worm relative to the worm gear? Draw and explain the layout.
5. Why are worm gear crowns made of bronzes?

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