

MINISTRY OF EDUCATION OF THE REPUBLIC OF BELARUS
Belarusian National Technical University

Department “Machine Science and machine parts”

A. A. Kalina, A. A. Sukhotski

MACHINE PARTS

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of education institution in the field
of mechanical engineering and technologies
as manual for the students of specialties
1-36 01 01 “Technology of engineering”,
1-53 01 01 “Automation of technological processes
and production (by directions)”

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Reviewer:

Department “Mechanics of materials and machine parts” of BSATU
(Head of Department, Candidate of Technical Sciences,

Associate Professor *V. N. Yednach*);

Associate Professor of the Department “Materials Science and Design
of Technical Systems” of BSTU, Candidate of Technical Sciences,

Associate Professor *S. E. Belsky*

Kalina, A. A.

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This manual is intended for technical universities students performing laboratory works
in such disciplines as "Machine parts", "Fundamentals of design and machine parts", "The-
ory of machines and mechanisms", "Technical mechanics", "Applied mechanics". The
manual gives the brief theoretical information, the descriptions of laboratory installations,
methods of work, the recommendations for making reports and the test questions.

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Laboratory Work № 1

BUILDING THE INVOLUTE TEETH BY GENERATION METHOD

Purpose of the work

To familiarize with the generation method of spur gear teeth manufacturing process using a tool rack, as well as to master the skills of calculating their dimensions.

Basic Safety Rules

Use laboratory equipment carefully, avoid it falling on the floor.

Description of the device TMM-42

The schematic diagram and the look of device are shown in fig. 1.1. On the base *1*, a rail *2* and a disk consisting of two circles *3* and *4* are mounted, which are rigidly connected to each other and rotate on a common axis.

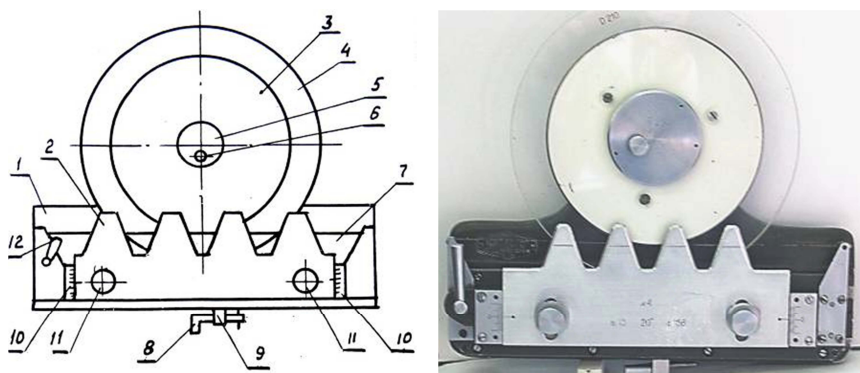


Fig. 1.1. Laboratory device TMM-42:

1 – base; *2* – rack; *3* – reference diameter; *4* – maximum paper diameter; *5* – faceplate;
6 – screw; *7* – carriage; *8* – value; *9* – freewheel lever; *10* – scale; *11* – rack screw

The upper circle *4*, is made of organic glass and serves to install a paper blank on which the teeth are drawn. The paper blank is mounted

on three needles of the upper circle using a removable faceplate 5, fixed with a screw 6. The diameter of the lower circle 3 is equal to the diameter of the reference circle of the gear wheel, the cutting of which is modeled on the device.

The rack 2 together with the carriage 7 can move along the guides of the frame. Such movement is carried out by means of a step-by-step ratchet mechanism actuated by a valve 8.

When the key is pressed, the rail is moved to the left by 4–5 mm and fixed. Using the freewheel lever 9, the rack can be freely moved to the right and left (by hand). In addition to the displacement in the guides, the rack can also be moved in the perpendicular direction – to the center of the disk or away from it, and this shift is counted on the scales 10, and the position of the rack on the carriage is fixed by screws 11.

The movement of the rail in the guides and the disk with the gear blank rotation are kinematically connected. Such connection is achieved with a device made of stretched wire and an eccentric mechanism controlled by a handle 12. In relative motion, the circle 3 (reference diameter) rolls without sliding along the pitch line of the rail. The transparent upper circle allows you to observe the rolling of the reference diameter circle along the pitch line.

How to do lab work

1. Write down the device number and the specified values of m , d_1 in to the report as well as the rack parameters ($\alpha = 20^\circ$, $h_a^* = 1$, $c^* = 0.25$).
2. Calculate the teeth number z_1 and z_2 .
3. Using the freewheel lever 9, move the rail to the rightmost position. Set the rack on the scales 10 to zero marks.
4. Draw a diametrical line on the paper blank. Using a removable faceplate 5 and screw 6, strengthen the workpiece on the disc so that it is brought under the rack, the diametrical line must be located approximately perpendicular to the rail guides, and the center of the workpiece must fall on the central needle of the disc.
5. The teeth contour of the rack must be carefully outlined with a sharpened pencil on a paper blank. Then, by pressing the 8 key, move the rack to the left one step of the ratchet mechanism, and then outline the contour of the teeth again. This must be repeated until the rail comes

to the leftmost position. As a result of rounding method on a paper circle 3 well-defined teeth are obtained (fig. 1.2).

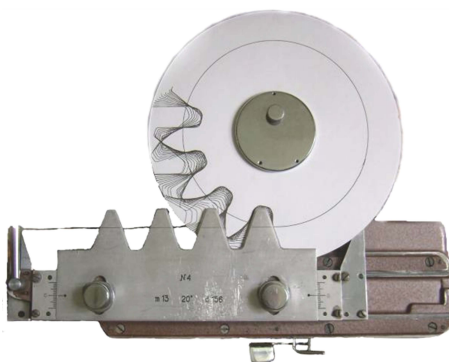


Fig. 1.2. Laboratory device at the beginning of the work

6. Determine the tool displacement factor (profile shift coefficient) x_1 and the tool displacement size Δ_1 . Move the rack away from the center of the gear blank to a distance Δ_1 and fix it with screws 11.

7. Move the rail to the rightmost position again. Turn the handle 12 to the left, turn the disc about 180 so that the clean part of the gear blank is on the right. After that, the handle 12 is moved to the right again and thereby the movements of the rack and disc are connected;

8. Draw the teeth of Corrected Spur Gear by the method described in point 4 (fig. 1.3), then remove the paper circle from the disk.

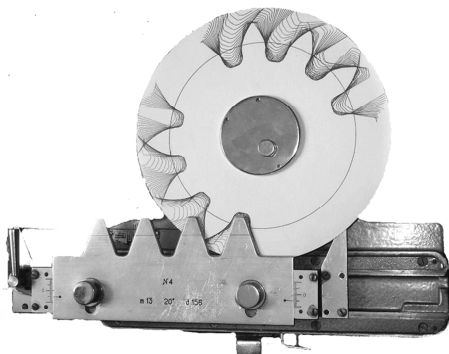


Fig. 1.3. Laboratory device at the final position

9. Calculate the dimensions of Standard and Corrected Spur Gear.
10. Draw the radiuses r_{f1} , r_{b1} , r_1 , r_{w1} и r_{a1} for both Standard and Corrected Spur Gears (fig. 1.4).
11. Measure the teeth thickness S_1 , S_{w1} , S_{b1} , S_{a1} (fig. 1.4).

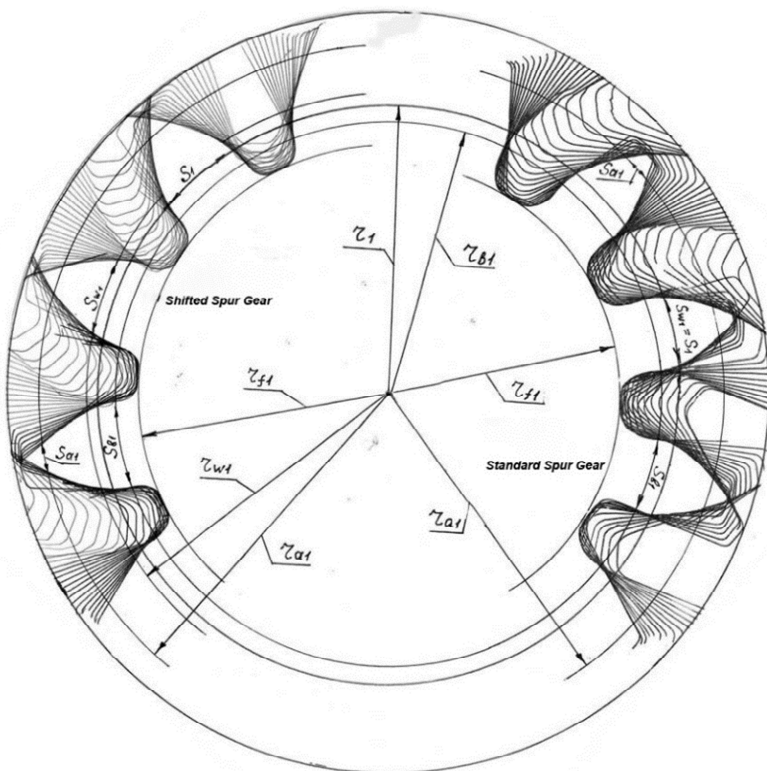


Fig. 1.4. Images of standard and shifted spur gears with marked parameters

Theoretical Information

Determination of standard spur gear parameters

According to international standards, the principal geometric relationships for uncorrected gearing are listed below (fig. 1.5).

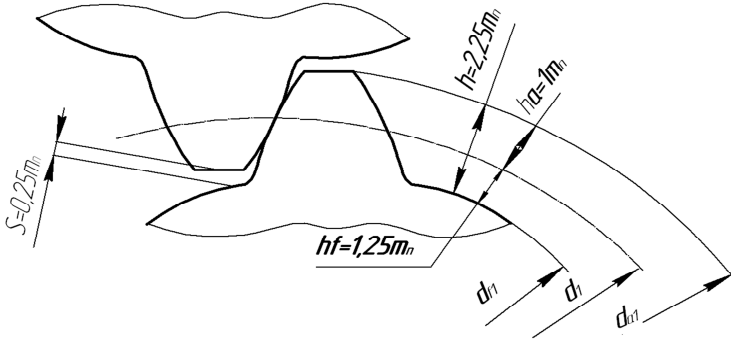


Fig. 1.5

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 :

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

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

Outside (addendum circle) pinion and gear diameters:

$$d_{a1} = d_1 + 2m_n;$$

$$d_{a2} = d_2 + 2m_n.$$

The module $m_n = m_t$ (for spur gear), reference diameter d_1 , standard rack parameters $\alpha = 20^\circ$, $h_a^* = 1$, $c^* = 0.25$ of the gear are given.

The number of teeth of the pinion is

$$z_1 = \frac{d_1}{m}.$$

It is considered that the paired gear 2, as well as the pinion 1, is produced without displacement the rack. Therefore, the working pressure angle is $\alpha_w = \alpha = 20^\circ$.

Pitch radius

$$r_{w1} = r_1 = \frac{d_1}{2}.$$

Base radius

$$r_{b1} = r_1 \cos \alpha,$$

where $\cos \alpha = \cos 20^\circ = 0.9397$.

Outside radius

$$r_{a1} = r_1 + h_a^* m = \frac{m}{2} (z_1 + 2).$$

Root radius

$$r_{f1} = r_1 - (h_a^* + c^*) m = \frac{m}{2} (z_1 - 2.5).$$

Tooth thickness on the reference circle

$$S_1 = \frac{\pi m}{2}.$$

Tooth thickness on the base circle

$$S_{b1} = 2r_{b1} \left(\frac{S_1}{d_1} + \text{inv } \alpha \right),$$

where $\text{inv } \alpha = \text{inv } 20^\circ = 0.014904$.

Tooth thickness on the outside circle

$$S_{a1} = 2r_{a1} \left(\frac{S_1}{d_1} + \text{inv } \alpha - \text{inv } \alpha_{a1} \right),$$

where α_{a1} is tooth profile angle on the outside circle. It can be found from

$$\cos \alpha_{a1} = \frac{r_{b1}}{r_{a1}}.$$

Determination of corrected spur gear parameters

The same parameters as module m , reference diameter d_1 , standard rack parameters $\alpha = 20^\circ$, $h_a^* = 1$, $c^* = 0.25$ of the gear are given.

To calculate the parameters of gear 1 (pinion), you need to know the teeth number z_2 of the paired gear 2 (gear). The teeth number of the 2 gear as an example is taken the same as for the 1 gear but it can be taken any necessary value.

Tool displacement factors x_1 and x_2 can be determined from the condition of no undercut:

$$x_1 = \frac{17 - z_1}{17}; \quad x_2 = \frac{17 - z_2}{17}.$$

Tool displacement size $\Delta_1 = x_1 m$.

Working Pressure Angle, it is determined by the nomogram (fig. 1.6). First you need to find an auxiliary value

$$B = \frac{1000 x_\Sigma}{z_\Sigma} = \frac{1000 (x_1 + x_2)}{z_1 + z_2}.$$

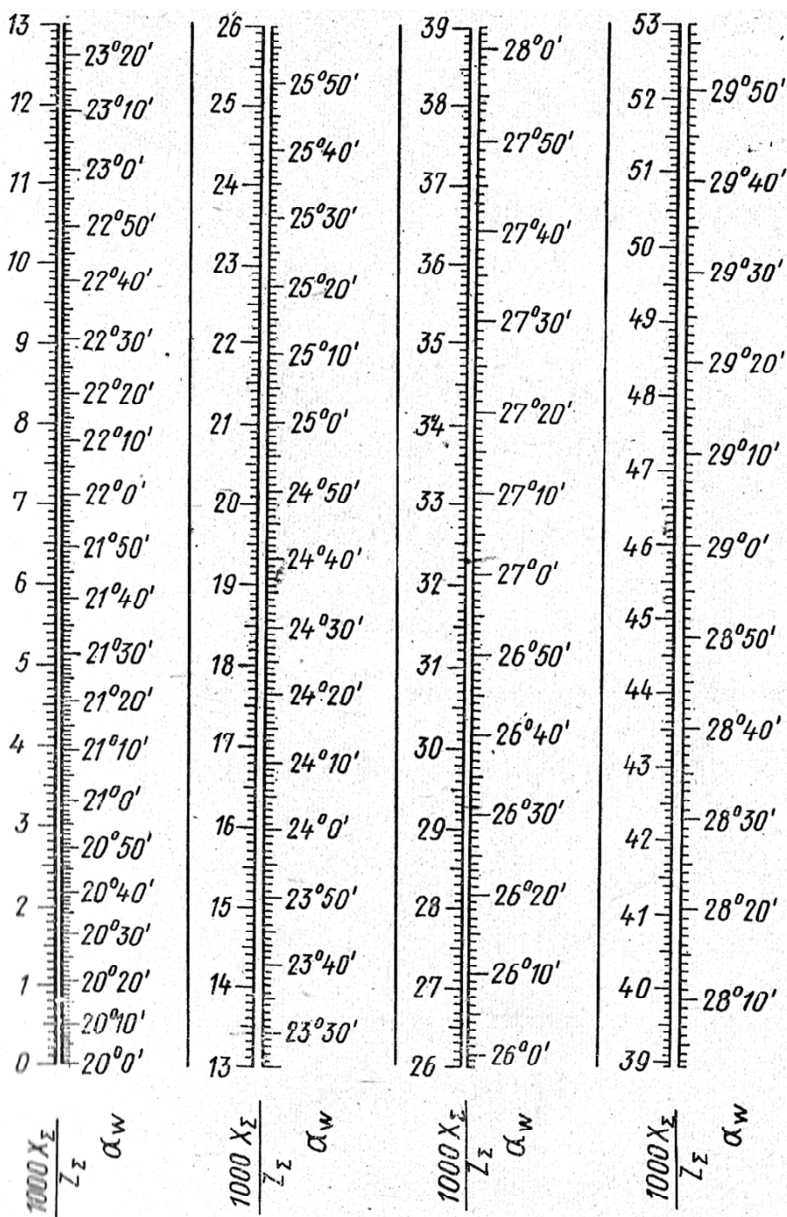


Fig. 1.6. Nomogram α_w determination

Centre distance modification coefficient

$$y = \frac{a_W - a}{m} = \frac{z_1 + z_2}{2} \left(\frac{\cos \alpha}{\cos \alpha_W} - 1 \right).$$

Equalization bias factor

$$\Delta y = (x_1 + x_2) - y.$$

Pitch radius

$$r_{W1} = r_1 \frac{\cos \alpha}{\cos \alpha_W}.$$

Base radius

$$r_{b1} = r_1 \cos \alpha.$$

Outside radius

$$r_{a1} = r_1 + (h_a^* + x_1 - \Delta y) m.$$

Root radius

$$r_{f1} = r_1 - (h_a^* + c^* - x_1) m.$$

Tooth thickness on the reference circle

$$S_1 = \left(\frac{\pi}{2} + 2x_1 \operatorname{tg} \alpha \right) m.$$

Tooth thickness on the working pitch circle

$$S_{W1} = 2r_{W1} \left(\frac{S_1}{d_1} + \operatorname{inv} \alpha - \operatorname{inv} \alpha_W \right).$$

Tooth thickness on the base circle

$$S_{b1} = 2r_{b1} \left(\frac{S_1}{d_1} + \text{inv } \alpha \right).$$

Tooth thickness on the outside circle

$$S_{a1} = 2r_{a1} \left(\frac{S_1}{d_1} + \text{inv } \alpha - \text{inv } \alpha_{a1} \right),$$

where α_{a1} is tooth profile angle on the outside circle. It can be found from

$$\cos \alpha_{a1} = r_{b1} / r_{a1}$$

(or table 1.1).

Involute of any angle φ can be calculated as

$$\text{inv } \varphi = \text{tg } \varphi - \varphi \text{ (in radians),}$$

or can be found from the table 1.2.

Table 1.1

Cos α_a value

Grad Min	36°	37°	38°	39°	40°	41°	42°	43°	44°
0	8090	7986	7880	7771	7660	7547	7431	7314	7193
1	8088	7985	7878	7770	7659	7545	7430	7312	7191
2	8087	7983	7877	7768	7657	7543	7428	7310	7189
3	8085	7981	7875	7766	7655	7541	7426	7308	7187
4	8083	7979	7873	7764	7653	7539	7424	7306	7185
5	8082	7978	7871	7762	7651	7538	7422	7304	7183
6	8080	7976	7869	7760	7649	7536	7420	7302	7181

Continuation of the table 1.1

Grad Min	36°	37°	38°	39°	40°	41°	42°	43°	44°
7	8078	7974	7868	7759	7647	7534	7418	7300	7179
8	8076	7972	7866	7757	7645	7532	7416	7298	7177
9	8075	7971	7864	7755	7644	7530	7414	7296	7175
10	8073	7969	7862	7753	7642	7528	7412	7294	7173
11	8071	7967	7860	7751	7640	7526	7410	7292	7171
12	8070	7965	7859	7749	7638	7524	7408	7290	7169
13	8068	7964	7857	7748	7636	7522	7406	7288	7167
14	8066	7962	7855	7746	7634	7520	7404	7286	7165
15	8064	7960	7853	7744	7632	7518	7402	7284	7163
16	8063	7958	7851	7742	7630	7516	7400	7282	7161
17	8061	7956	7850	7740	7629	7515	7398	7280	7159
18	8059	7955	7848	7738	7627	7513	7396	7278	7157
19	8058	7953	7846	7737	7625	7511	7394	7276	7155
20	8056	7951	7844	7735	7623	7509	7392	7274	7153
21	8054	7949	7842	7733	7621	7507	7390	7272	7151
22	8052	7948	7841	7731	7619	7505	7388	7270	7149
23	8051	7946	7839	7729	7617	7503	7387	7268	7147
24	8049	7944	7837	7727	7615	7501	7385	7266	7145
25	8047	7942	7835	7725	7613	7499	7383	7264	7143
26	8045	7941	7833	7724	7612	7497	7381	7262	7141
27	8044	7939	7832	7722	7610	7495	7379	7260	7139
28	8042	7937	7830	7720	7608	7493	7377	7258	7137
29	8040	7935	7828	7718	7606	7491	7375	7256	7135
30	8039	7934	7826	7716	7604	7490	7373	7254	7133
31	8037	7932	7824	7714	7602	7488	7371	7252	7130
32	8035	7930	7822	7713	7600	7486	7369	7250	7128
33	8033	7928	7821	7711	7598	7484	7367	7248	7126
34	8032	7926	7819	7709	7596	7482	7365	7246	7124
35	8030	7925	7817	7707	7595	7480	7363	7244	7122
36	8028	7923	7815	7705	7593	7478	7361	7242	7120

End of the table 1.1

Grad Min	36°	37°	38°	39°	40°	41°	42°	43°	44°
37	8026	7921	7813	7703	7591	7476	7359	7240	7118
38	8025	7919	7812	7701	7589	7474	7357	7238	7116
39	8023	7918	7810	7700	7587	7472	7355	7236	7114
40	8021	7916	7808	7698	7585	7470	7353	7234	7112
41	8019	7914	7806	7696	7583	7468	7351	7232	7110
42	8018	7912	7804	7694	7581	7466	7349	7230	7108
43	8016	7910	7802	7692	7579	7464	7347	7228	7106
44	8014	7909	7801	7690	7578	7463	7345	7226	7104
45	8013	7907	7799	7688	7576	7461	7343	7224	7102
46	8011	7905	7797	7687	7574	7459	7341	7222	7100
47	8009	7903	7795	7685	7572	7457	7339	7220	7098
48	8007	7902	7793	7683	7570	7455	7337	7218	7096
49	8006	7900	7792	7681	7568	7453	7335	7216	7094
50	8004	7898	7790	7679	7566	7451	7333	7214	7092
51	8002	7896	7788	7677	7564	7449	7331	7212	7090
52	8000	7894	7786	7675	7562	7447	7329	7210	7088
53	7999	7893	7784	7674	7560	7445	7327	7208	7085
54	7997	7891	7782	7672	7559	7443	7325	7206	7083
55	7995	7889	7781	7670	7557	7441	7323	7203	7081
56	7993	7887	7779	7668	7555	7439	7321	7201	7079
57	7992	7885	7777	7666	7553	7437	7319	7199	7077
58	7990	7884	7775	7664	7551	7435	7318	7197	7075

Table 1.2

Involute α value

Grad	Oder	0'	5'	10'	15'	20'	25'	30'	35'	40'	45'	50'	55'
17	0.0	09025	09161	09299	09439	09580	09722	09866	10012	10158	10307	10456	10608
18	0.0	10760	10915	11071	11228	11387	11547	11709	11873	12038	12205	12373	12543
19	0.0	12715	12888	13063	13240	13418	13598	13779	13963	14148	14334	14523	14713
20	0.0	14904	15098	15293	15490	15689	15890	16092	16296	16502	16710	16920	17132
21	0.0	17345	17560	17777	17996	18217	18440	18665	18891	19120	19350	19583	19817
22	0.0	20054	20292	20533	20775	21019	21266	21514	21765	22018	22272	22529	22788
23	0.0	23049	23312	23577	23845	24114	24386	24660	24936	25214	25495	25778	26062
24	0.0	26350	26639	26931	27225	27521	27820	28121	28424	28729	29037	29348	29660
25	0.0	29975	30293	30613	30935	31260	31587	31917	32249	32583	32920	33260	33602
26	0.0	33947	34294	34644	34997	35352	35709	36069	36432	36798	37166	37537	37910
27	0.0	38287	38666	39047	39432	39819	40209	40602	40997	41395	41797	42201	42607
28	0.0	43017	43430	43845	44264	44685	45110	45537	45967	46400	46837	47276	47718
29	0.0	48164	48612	49064	49518	49976	50437	50901	51368	51838	52312	52788	53268
30	0.0	53751	54238	54728	55221	55718	56217	56720	57226	57736	58249	58765	59285
31	0.0	58809	60335	60856	61400	61937	62478	63022	63570	64122	64677	65236	65798
32	0.0	66364	66934	67507	68084	68665	69250	69838	70430	71026	71626	72230	72838
33	0.0	73449	74064	74684	75307	75934	76565	77200	77839	78483	79130	79781	80437
34	0.0	81097	81760	82428	83100	83777	84457	85142	85832	86525	87233	87925	88631
35	0.0	89342	90058	90777	91502	92230	92963	93701	94443	95190	95942	96698	97459

End of the table 1.2

Grad	Oder	0'	5'	10'	15'	20'	25'	30'	35'	40'	45'	50'	55'
36	0.0	09822	09899	09977	10055	10133	10212	10292	10371	10452	10533	10614	10696
37	0.0	10778	10861	10944	11028	11113	11197	11283	11369	11455	11542	11630	11718
38	0.0	11806	11895	11985	12075	12165	12257	12348	12441	12534	12627	12721	12815
39	0.0	12911	13006	13102	13199	13297	13395	13493	13592	13692	13792	13893	13995
40	0.0	14097	14200	14303	14407	14511	14616	14722	14829	14936	15043	15152	15261
41	0.0	15370	15480	15591	15703	15815	15928	16041	16156	16270	16386	16502	16619
42	0.0	16737	16855	16974	17093	17214	17335	17457	17579	17702	17826	17951	18076
43	0.0	18202	18329	18457	18585	18714	18844	18975	19106	19238	19371	19505	19639
44	0.0	19774	19910	20047	20185	20323	20463	20603	20743	20885	21028	21171	21315
45	0.0	21460	21606	21753	21900	22049	22198	22348	22499	22651	22804	22958	23112
46	0.0	23268	23424	23583	23740	23899	24059	24220	24382	24545	24709	24874	25040
47	0.0	25206	25374	25543	25713	25883	26055	26228	26401	26576	26752	26929	27107
48	0.0	27285	27465	27646	27828	28012	28196	28381	28567	28755	28943	29133	29724
49	0.0	29516	29709	29903	30098	30295	30492	30691	30891	31092	31295	31498	31708
50	0.0	31909	32116	32324	32534	32745	32957	33171	33385	33681	33818	34037	34257
51	0.0	34478	34700	34924	35149	35376	35604	35833	36063	36295	36529	36763	36999
52	0.0	37237	37476	37716	37958	38202	38446	38693	38941	39190	39441	39693	39947
53	0.0	40202	40459	40717	40977	41239	41502	41767	42034	42302	42571	42843	43116

Sample design of laboratory work

BELARUSIAN NATIONAL TECHNICAL UNIVERSITY

Department "Machine Science and Machine Parts"

L a b o r a t o r y W o r k № 1

Building the involute teeth by generation method

Purpose of the work:

To familiarize with the generation method of spur gear teeth manufacturing process using a tool rack, as well as to master the skills of calculating their dimensions.

The work was performed by: _____

Group: _____

The work was accepted by: _____

Minsk _____

Table 1.3

Set Value

Equipment number	m , mm	d_1 , mm	α , deg	h_a^*	c^*	z_1	z_2
—	10	100	20	1	0,25	10	14

Determination of standard spur gear parameters

$$1. \ z_1 = \frac{d_1}{m} = \frac{100}{10} = 10.$$

$$2. \ r_{w1} = r_1 = \frac{d_1}{2} = \frac{100}{2} = 50 \text{ mm}.$$

$$3. \ r_{b1} = r_1 \cos \alpha = 50 \cdot 0.9397 = 47 \text{ mm}.$$

$$4. \ r_{a1} = r_1 + h_a^* m = \frac{m}{2} (z_1 + 2) = \frac{10}{2} (10 + 2) = 60 \text{ mm}.$$

$$5. \ r_{f1} = r_1 - (h_a^* + c^*) m = \frac{m}{2} (z_1 - 2.5) = \frac{10}{2} (10 - 2.5) = 37.5 \text{ mm}.$$

$$6. \ S_1 = \frac{\pi m}{2} = \frac{3.14 \cdot 10}{2} = 15.7 \text{ mm}.$$

$$7. \ S_{b1} = 2r_{b1} \left(\frac{S_1}{d_1} + \text{inv } \alpha \right) = 2 \cdot 47 \left(\frac{15.7}{100} + 0.014904 \right) = 16.2 \text{ mm}.$$

$$8. \ \cos \alpha_{a1} = \frac{r_{b1}}{r_{a1}} = \frac{47}{60} = 0.7833; \ \alpha_{a1} = 38^\circ 26', \ \text{inv } \alpha_{a1} = 0.12275.$$

$$9. \ S_{a1} = 2r_{a1} \left(\frac{S_1}{d_1} + \text{inv } \alpha - \text{inv } \alpha_{a1} \right) =$$

$$= 2 \cdot 60 \left(\frac{15.7}{100} + 0.014904 - 0.12275 \right) = 6 \text{ mm}.$$

Determination of corrected spur gear parameters

$$1. x_1 = \frac{17 - z_1}{17} = \frac{17 - 10}{17} = 0.412; \quad x_2 = \frac{17 - z_2}{17} = \frac{17 - 14}{17} = 0.176.$$

$$2. \Delta_1 = x_1 m = 0.412 \cdot 10 = 4.12 \text{ mm}.$$

$$3. B = \frac{1000 x_{\Sigma}}{z_{\Sigma}} = \frac{1000 (x_1 + x_2)}{z_1 + z_2} = \frac{1000 (0.412 + 0.176)}{10 + 14} = 24.5,$$

$$\alpha_W = 25^{\circ}41'.$$

$$4. y = \frac{a_W - a}{m} = \frac{z_1 + z_2}{2} \left(\frac{\cos \alpha}{\cos \alpha_W} - 1 \right) = \frac{10 + 14}{2} (1.04271 - 1) = 0.515.$$

$$5. \Delta y = (x_1 + x_2) - y = (0.412 + 0.176) - 0.515 = 0.073.$$

$$6. r_{W1} = r_1 \frac{\cos \alpha}{\cos \alpha_W} = 50 \cdot 1.04271 = 52.1 \text{ mm}.$$

$$7. r_{b1} = r_1 \cos \alpha = 50 \cdot 0.9397 = 47 \text{ mm}.$$

$$8. r_{a1} = r_1 + (h_a^* + x_1 - \Delta y) m = 50 + (1 + 0.412 - 0.073) 10 = 63.4 \text{ mm}.$$

$$9. r_{f1} = r_1 - (h_a^* + c^* - x_1) m = 50 - (1 + 0.25 - 0.412) 10 = 41.6 \text{ mm}.$$

$$10. S_1 = \left(\frac{\pi}{2} + 2 x_1 \operatorname{tg} \alpha \right) m = \left(\frac{3.14}{2} + 2 \cdot 0.412 \cdot 0.364 \right) 10 = 18.7 \text{ mm}.$$

$$11. S_{W1} = 2 r_{W1} \left(\frac{S_1}{d_1} + \operatorname{inv} \alpha - \operatorname{inv} \alpha_W \right) = \\ = 2 \cdot 52.1 \left(\frac{18.7}{100} + 0.0149 - 0.0326 \right) = 17.6 \text{ mm}.$$

$$12. S_{b1} = 2 r_{b1} \left(\frac{S_1}{d_1} + \operatorname{inv} \alpha \right) = 2 \cdot 47 \left(\frac{18.7}{100} + 0.014904 \right) = 19 \text{ mm}.$$

$$13. \cos \alpha_{a1} = \frac{r_{b1}}{r_{a1}} = \frac{47}{63.4} = 0.7412, \quad \alpha_{a1} = 42^{\circ}3', \quad \operatorname{inv} \alpha_{a1} = 0.16975.$$

$$14. S_{a1} = 2r_{a1} \left(\frac{S_1}{d_1} + \text{inv } \alpha - \text{inv } \alpha_{a1} \right) =$$

$$= 2 \cdot 63.4 \left(\frac{18.7}{100} + 0.014904 - 0.16975 \right) = 4.1 \text{ mm.}$$

Table 1.4

Comparison table of calculations with measurements

Tooth thickness, mm	Standard spur gear		Corrected spur gear	
	calculated	measured	calculated	measured
S_1	15.7	15	18.7	18
S_{W1}	15.7	15	17.6	17
S_{b1}	16.2	15	19	18.5
S_{a1}	6	5	4.1	3

Test questions

1. Formulate the main gear engagement theorem (Willis' theorem).
2. What is the involute of a circle?
3. Write down the equations of the involute of a circle.
4. Specify the main rack parameters.
5. What is the module for a gear?
6. Definition of the outside circle.
7. Write down the formula for minimum profile shift coefficient with the condition of no undercut.
8. Describe the generation method.
9. Describe the forming method.
10. What tools are used in the generation method?
11. In what cases are standard and corrected spur gears used?

Laboratory work № 2

MULTI-STAGE REDACTION GEAR EFFICIENCY DETERMINATION

Purpose of the work

1. Determining the main parameters of multistage reduction gear.
2. Drawing the kinematic diagram of multistage reduction gear.
3. Plotting the graphs: $\eta = f(T_2)$ and $\eta = f(n_1)$.

Basic Safety Rules

1. Start the equipment with the permission of the teacher.
2. The equipment operating, the reduction gear cover must be closed.
3. After finishing the work, disconnect the equipment from the network.

Description of the DPCK TC

The tested reduction gear 8, electric motor (balance) 5 with tachometer 3 and load device 11 (magnetic powder brake) are installed on cast base 1 (fig. 2.1). Input and output shafts of reduction gear are connected to shafts of electric motor and load device by elastic couplings 7 and 9.

The tachometer is connected to the electric motor by coupling 4. Measuring devices are mounted on brackets 2 and 12 which consist of flat springs, indicators 6 and 10, rods rest against springs.

On panel 13 there is toggle switch 17 which turns on and off the electric motor; potentiometer 16 for steplessly adjusting the speed of the motor; toggle switch 15 including a load device and potentiometer 14 for controlling the braking torque T_2 .

Motor stator is installed in rolling bearings so that it can swing relative to the axis coinciding with the armature axis.

The reaction moment generated during the electric motor operation is completely transmitted to the stator and acts in the direction opposite to the rotation of the armature. This is a balance motor.

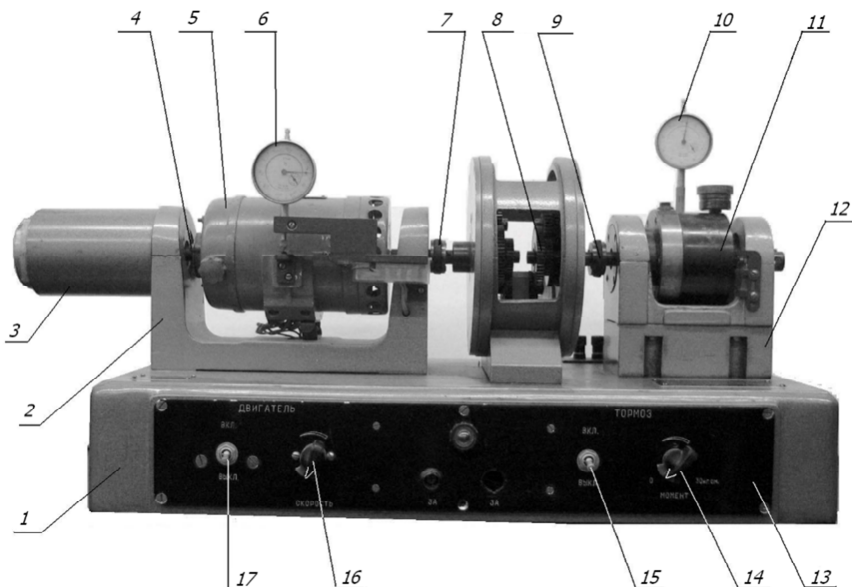


Fig. 2.1. Equipment DPCK TC:

1 – base; 2, 12 – brackets; 3 – tachometer; 4, 7, 9 – couplings; 5 – electric motor; 6, 10 – indicators; 8 – multistage reduction gear; 11 – powder brake; 13 – control panel; 14 – load control hand knob; 15 – toggle switch for load device actuation; 16 – motor speed control knob; 17 – toggle switch for electric motor actuation

As a result of calibrating the electric motor measuring device the value of calibration coefficient is set

$$\mu_1 = 1.6 \frac{\text{N} \cdot \text{mm}}{\text{division}}.$$

Similarly, the calibration coefficient for the braking device is determined:

$$\mu_2 = 23 \frac{\text{N} \cdot \text{mm}}{\text{division}}.$$

Theoretical Information

1. Kinematic study.

Transmission ratio of a reduction gear consisting of n pairs of sequentially engaged wheels (pinion + gear) is equal to the product of transmission ratios of separate pairs of wheels:

$$i_{\text{tot}} = \pm i_1 \cdot i_2 \cdot i_3 \cdot \dots \cdot i_n. \quad (2.1)$$

The Transmission ratio

$$i_{12} = \pm n_1 / n_2 = \pm \omega_1 / \omega_2 = \pm z_1 / z_2;$$

where index 1 – driving link;

index 2 – driven link.

Gear ratio of gear pair (pinion + gear)

$$u = z_{\text{gear}} / z_{\text{pinion}}. \quad (2.2)$$

Outside (addendum circle) pinion and gear diameter (da_1 and da_2), mm,

$$d_a = m(z + 2),$$

where $m = \frac{d_a}{z + 2}$ (is consistent with GOST 9.563-60). (2.3)

Center distance

$$a_w = \frac{m}{2}(z_1 + z_2). \quad (2.4)$$

2. Determining the multistage reduction gear efficiency.

The energy losses in the gears are mainly due to friction losses in the engagement and in the bearings due to oil spraying.

In this case there is no spraying losses, since the reduction gear consists of open gears.

The multistage reduction gear efficiency determination is based on simultaneous and independent measurement of torques T_1 on the input shaft and T_2 on the output shaft of the reduction gear (fig. 2.1).

Multistage reduction gear efficiency can be determined by the equation

$$\eta_{\text{tot}} = \frac{T_2}{T_1 i_{\text{tot}}}, \quad (2.5)$$

where T_1 is the torque on the motor shaft;

T_2 is torque on the output shaft of the reduction gear;

i_{tot} is transmission ratio of reduction gear.

Experiment torque values are determined by dependences

$$T_1 = \mu_1 k_1; \quad (2.6)$$

$$T_2 = \mu_2 k_2, \quad (2.7)$$

where μ_1 and μ_2 are calibration coefficients;

k_1 and k_2 are readings of indicators of measuring devices of electric motor and brake, respectively.

The total efficiency of the mechanism consisting of n pairs of sequentially engaged wheels is equal to the product of the efficiency of separate pairs of wheels:

$$\eta_{\text{tot}} = \eta_1 \cdot \eta_2 \cdot \dots \cdot \eta_n.$$

If the losses in each pair are approximately the same, then

$$\eta_{\text{tot}} = \eta^n,$$

where η is the efficiency of one gear pair (taking into account the losses in bearings).

$$\eta = \sqrt[n]{\eta_{\text{tot}}}.$$

How to do lab work

1. Count the number of pinion and gear teeth, measure the outside diameters da_1 and da_2 . Put down the results of calculations and measurements into table 2.1 of the report.

2. According to table 2.1 of the report, draw a kinematic diagram of the multistage reduction gear, using the conventional symbols shown in fig. 2.2.

3. Start the electric motor and set motor shaft speed to $n = 700 \text{ min}^{-1}$ by turning motor speed control knob 16 (fig. 2.1).

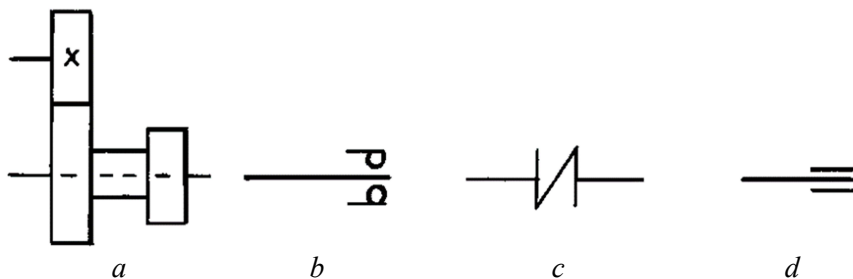


Fig. 2.2. Symbol on diagrams (GOST 2.770-68 and GOST 2.703-68):

- a* – cylindrical gear with straight teeth
(the pinion is fixed on the shaft. The unit rotates freely on the axis);
b – radial ball bearings (general designation);
c – elastic connection of shafts (coupling); *d* – radial bearings

4. Set indicator needles to zero position.

5. Turning the hand knob (14) load the reducer with torques of different magnitude. The motor measuring device indicator readings must be taken at the selected constant motor speed.

6. Put down the indicator readings into table 2.2 of the report.

7. Use formulas (2.5), (2.6) and (2.7) to calculate the values of T_1 , T_2 , and η_{tot} .

8. Using the data in table 2.2 of the report, plot the graphs $\eta = f(T_2)$, $n_1 = \text{const}$.

9. Similarly, perform tests at constant and variable motor speed. Put down the experiment data and the results of the data calculation into table 2.3 of the report.

10. Plot the dependency at $\eta = f(n)$, $T_2 = \text{const}$.

Sample design of laboratory work

BELARUSIAN NATIONAL TECHNICAL UNIVERSITY

Department "Machine Science and Machine Parts"

Laboratory work № 2

**MULTI-STAGE REDUCTION GEAR EFFICIENCY
DETERMINATION**

Purpose of the work:

1. Determining the main parameters of multistage reduction gear.
2. Drawing the kinematic diagram of multistage reduction gear.
3. Plotting the graphs: $\eta = f(T_2)$ and $\eta = f(n_1)$.

The work was performed by: _____

Group: _____

The work was accepted by: _____

Minsk _____

Results of measurements and calculation of reduction gear parameters (table 2.1):

- number of pinion and gear teeth (Z_1 and Z_2);
- outside (addendum circle) pinion and gear diameter (d_{a1} and d_{a2}), mm;
- module m by formula (2.3), mm;
- center distance a_w by formula (2.4), mm;
- transmission ratio i and Gear ratio u ;
- total transmission ratio i_{tot} by formula (2.1).

Table 2.1

Reduction gear parameters

Z_1	Z_2	d_{a1} , mm	d_{a2} , mm	m , mm	a_w , mm	i_{tot}

Kinematic diagram of multistage reduction gear

Table 2.2

Experiment data and calculation results

Values	Experiment number					
	1	2	3	4	5	6
k_1 is number of indicator scale divisions						
$T_1 = \mu_1 k_1$						
k_2 is number of indicator scale divisions						
$T_2 = \mu_2 k_2$						
$\eta_{\text{tot}} = \frac{T_2}{T_1 i_{\text{tot}}}$						



Fig. 2.2. Graph of $\eta = f(T_2)$, $n_1 = \text{const}$

Table 2.3

Experimental data and calculation results

Values	Experiment number					
	1	2	3	4	5	6
k_1 is number of indicator scale divisions						
$T_1 = \mu_1 k_1$						
$\eta_{\text{tot}} = \frac{T_2}{T_1 i_{\text{tot}}}$						



Fig. 2.3. Graph of $\eta = f(n_1)$, $T_2 = \text{const}$

Test questions

1. What are the gear losses and what are the most effective measures to reduce gear losses?
2. What are relative, constant and load losses.
3. How does the transmission efficiency change depending on the transmission power?
4. Why does efficiency increase with increasing the accuracy of gears and gearings?

Laboratory work № 3

STUDYING BELT DRIVE OPERATION

Purpose of the work

1. Plotting the graph of dependence $\varepsilon = f(T_2)$.
2. Determining the belting efficiency and plotting the dependence $\eta = f(T_2)$.

Basic Safety Rules

1. Set the pre-tensioning of the belt in the presence of the teacher.
2. After finishing the experiment disconnect the machine from the mains.

Theoretical Information

A characteristic feature of any belt drive is inevitable true slip because of which the speed of the driven pulley (for the V-belt or rope, the pulley is commonly called a sheave) lags behind the speed of the driven pulley, the heating and wear of the belt occur, and the efficiency of the transmission decreases. The true slip (the relative speed loss) depends on the type and material of the belts, the pre-tensioning force F_0 and the transmitted load T_2 (fig. 3.1).

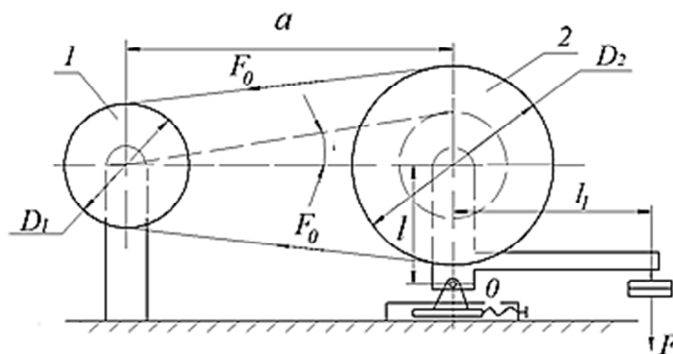


Fig. 3.1. Schematic diagram of the transmission and loader

$$F_0 = \frac{Fl_1}{2l \cos \gamma},$$

where $l = 60 \text{ mm}$;

$l_1 = 150 \text{ mm}$ (fig. 3.1).

The true slip coefficient is determined by to the equation (fig. 3.2)

$$\varepsilon = \left(1 - \frac{D_2 n_2}{D_1 n_1} \right) 100 \%. \quad (3.1)$$

The belt grievie efficiency is determined by the equation

$$\eta = \frac{T_2 n_2}{T_1 n_1}, \quad (3.2)$$

where T_1 is torque on driving pulley, $\text{N}\cdot\text{mm}$;

T_2 is torque on the driven pulley, $\text{N}\cdot\text{mm}$;

n_1, n_2 are speed of driving and driven pulleys (rotation frequency), min^{-1} .

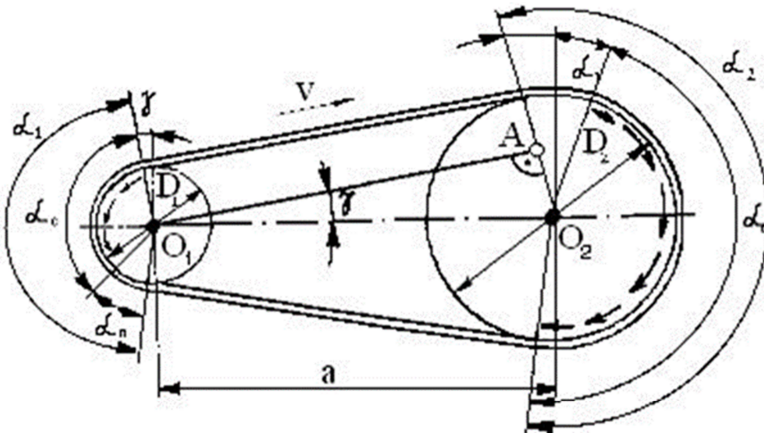


Fig. 3.2. Open-belt two-pulley drive:

a – is the centre distance; α – the angles of contact between the belt and the pulleys;

α_n – the angles at which the belt does not slip; α_c – the angles where the belt slips in relation to the pulley

The operation of a belt drive is always accompanied by slippage because of which the driven shaft loses speed. The angular speed of the driving shaft being ω_1 , the actual speed of the driven shaft determined with allowance for the slip of the belt will be given by

$$i = \omega_1 / \omega_2 = D_2 / (D_1(1 - \varepsilon)); \quad (3.3)$$

$$\omega_2 = \omega_1 D_1(1 - \varepsilon) / D_2 \quad \text{or} \quad n_2 = n_1 D_1(1 - \varepsilon) / D_2;$$

$$v_2 = v_1 (1 - \varepsilon);$$

$$v_1 = \pi d_1 n_1 / 60;$$

$$v_2 = \pi d_2 n_2 / 60,$$

where D_1 and D_2 are the design diameters of the driving and driven shafts, respectively, and ε is the relative speed loss.

The loss of speed which is identified with slippage is an outcome of the complex deformation suffered by the belt when it is in contact with the pulleys. In a flat-belt drive (fig. 3.2), it may be attributed to tensile and shearing deformations. The former occur when the tension in the belt falls from F_1 to F_2 at the driving pulley (as this takes place, each element of the belt shrinks and lags behind the pulley) and rises from F_2 to F_1 at the driven pulley (here, the belt elements elongate and lead the pulley in speed). Called true slip, this loss of speed is directly proportional to the load on the drive and inversely proportional to the belt rigidity.

$$\varepsilon = (F_1 - F_2) / (AE). \quad (3.4)$$

To represent graphically variations in relative speed loss as a function of load, we lay off the relative speed loss and drive efficiency as ordinate and the relative load expressed in terms of the traction factor as abscissa. The traction factor may be defined as

$$\varphi = F_t / (F_1 + F_2) = (F_1 - F_2) / (F_1 + F_2). \quad (3.5)$$

At no load, $\varphi = 0$ and $\varepsilon = 0$. Theoretically, when the load approaches the ultimate value, $\varphi \rightarrow 1$. The curve $\varepsilon = \varepsilon(\varphi)$ may be called the slippage curve. Under light and medium loads, it is close to a straight line in configuration. As the load increases, the amount of slippage rises suddenly and, at $\varphi = \varphi_{\max}$, the belt begins to slip without transmitting any motion (fig. 3.3).

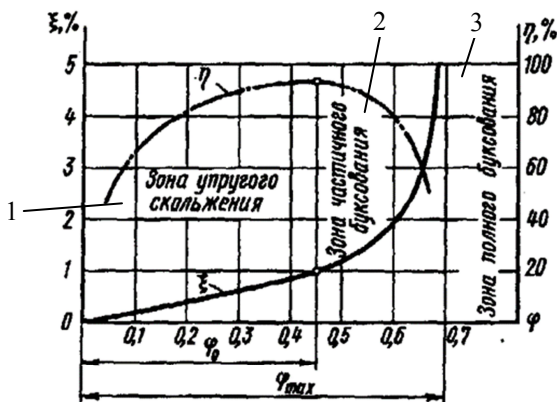


Fig. 3.3. Graphs of belt slippage and drive efficiency of a running belt drive:
 1 – no load; 2 – optimal load; 3 – zone of the full slipping (complete slipping));
 φ_0 – critical value of the traction factor

Other losses that occur during the operation of a belt drive owe their existence to elastic hysteresis when the belt undergoes deformation due to alternate bending and tension, slippage on the pulleys, friction in the shaft bearings, and aerodynamic drag (this applies to both the belt and the pulleys).

Speed losses caused by the bending of the belt on the pulleys are the most substantial. Like the aerodynamic drag and frictional losses in the bearings, they are independent of the applied load and remain unchanged at no load. As a result, the drive efficiency is rather low when the load is light (fig. 3.3). It increases with increasing load, reaching a maximum at φ ranging between 0.4 and 0.6, and then falls sharply again as the belt begins to slip on the pulleys. Because of the considerable belt thickness, large bending deformation, and losses due to radial displacement, V-belt drives are inferior to flat-belt drives in efficiency. Under optimum load, the average efficiency is $\eta = 0.96\text{--}0.97$ for flat-belt drives, $\eta = 0.92\text{--}0.95$ for V-belt drives. These graphs are built to determine the optimal transmission load experimentally.

Description of the testing equipment

The testing equipment (fig. 3.4) consists of a balancing electric motor 2, loading device 8 and the device for producing pre-tensioning of belt 5.

On the cast base there is a bracket on which the electric motor 2 balancing in ball-bearings is mounted. On the electric motor shaft there is a driving two-stage pulley 1. The torque developed by the electric motor is measured by indicator 4 by the deformation of spring 3. In the upper part of the stand there is slider 9 in the guides.

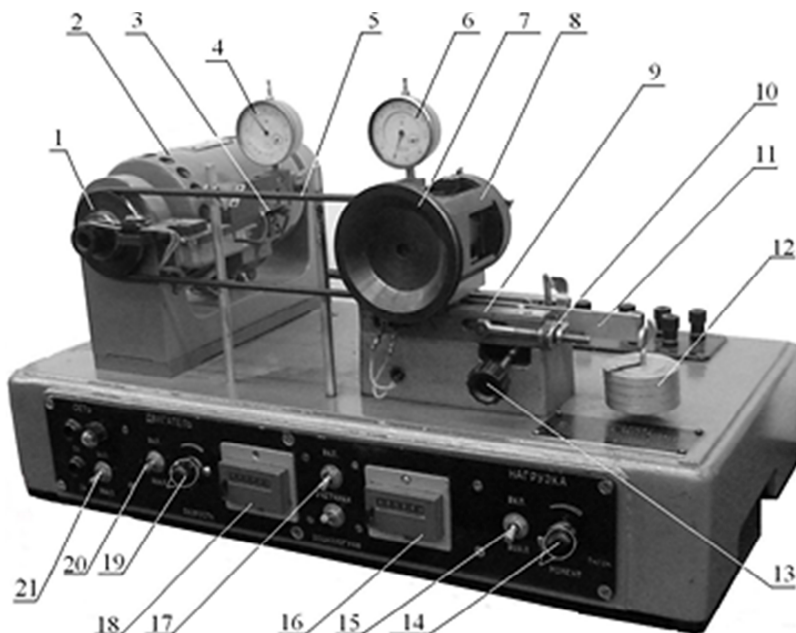


Fig. 3.4. Belt Drive Testing Equipment:

1 – driving pulley; 2 – electric motor; 3 – plate spring; 4, 6 – indicators; 5 – round belt (rope drive); 7 – driven pulley; 8 – powder brake; 9 – slider; 10 – screw moving the slider; 11 – lever; 12 – weights; 13 – screw fixing the slider; 14 – load control hand knob; 15 – load device switch; 16 – driven pulley speed counter; 17 – switch for counters; 18 – driving pulley speed counter; 19 – speed control hand knob; 20 – toggle switch for turning on the electric motor; 21 – toggle switch for turning on the mains

The slider has an axle on which a hinged body is mounted. To this body the loading device is attached the shaft of which is connected to the shaft of two-stage driven pulley 7. Lever 11 is attached to the body. By means of weights 12 a moment is produced relative to the axis in the slider. As a result, the body together with the shaft of the driven pulley can move in the direction from the driving pulley, thereby producing

additional belt tension. Screw 10 is used to move slider 9 with the housing, producing pre-tensioning of the belt. By using the screw 13 the slider is fixed at the selected belt tension. The load torque value is measured by the spring deformation by indicator 6.

How to do lab work

1. Produce a predetermined pre-tension of the belt, suspending certain load 12 to lever 11, then lever handle 10 should be set to a horizontal position which is determined by the coincidence of the marks on the lever and the slider (fig. 3.4).

2. Secure the slider with screw 13.

3. Switch on the installation into the network.

4. Switch on the by toggle switch 21 unit power and switch on electric motor 5 by toggle switch 20.

5. Set the electric motor rotation speed by hand knob 19 (approximately $n_1 = 1100 \text{ min}^{-1}$) and do not change it during the experiment.

6. Start the loading device with switch 15, hand knob 14 is in extreme left position ($T_2 = 0$).

7. Set indicators 4 and 6 to the zero mark of the scale (when the electric motor is running).

8. By pressing the buttons set indicators 16 and 18 to zero.

9. At torque $T_2 = 0$ switch on counters 16 and 18 by means of switch 17 for one minute. Put down the counter readings in to table 3.1 of the report.

10. Repeat the actions in items 8 and 9 for the loads of 10, 15, 20, 25, 30, 35, 40 divisions of the indicator. The torque developed by the electric motor is measured by indicator 4. When calibrating indicators 4 and 6 with the help of calibration attachment the following coefficients are obtained, respectively

$$\mu_1 = 3,35 \frac{\text{N} \cdot \text{mm}}{\text{scale division}};$$

$$\mu_2 = 7,5 \frac{\text{N} \cdot \text{mm}}{\text{scale division}}.$$

11. Put down the measurement data in to table 3.1 of the report.

12. Plot the graphs of ε and η dependence on the load T_2 .

Sample design of laboratory work

BELARUSIAN NATIONAL TECHNICAL UNIVERSITY

Department "Machine Science and Machine Parts"

Laboratory work № 3

STUDYING BELT DRIVE OPERATION

Purpose of the work:

1. Plotting the graph of dependence $\varepsilon = f(T_2)$.
2. Determination of belting efficiency and plotting the dependence $\eta = f(T_2)$.

The work was performed by: _____

Group: _____

The work was accepted by: _____

Minsk _____

Kinematic diagram of the belt drive and load device

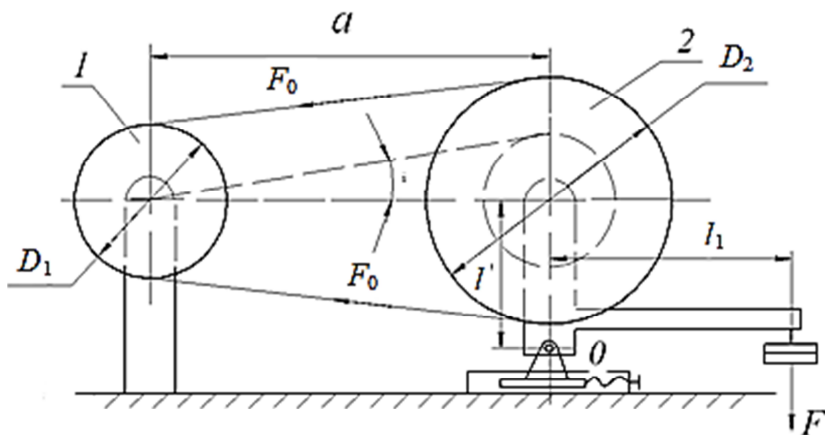


Fig. 3.5. Kinematic diagram of the belt drive and load device

Table 3.1

Measurement results and calculations

Number of indicator scale division		$T_1 = \mu_1 m_1,$ N·mm	$T_2 = \mu_2 m_2,$ N·mm	n_1	n_2	$\varepsilon = 1 - \frac{D_2 n_2}{D_1 n_1}$	$\eta = \frac{T_2 n_2}{T_1 n_1}$
Motor load m_1	Brake load m_2						
	10						
	15						
	20						
	25						
	30						
	35						
	40						

Using the data in table 3.1, plot the graphs of $\varepsilon = f(T_2)$, $\eta = f(T_2)$ (fig. 3.6).

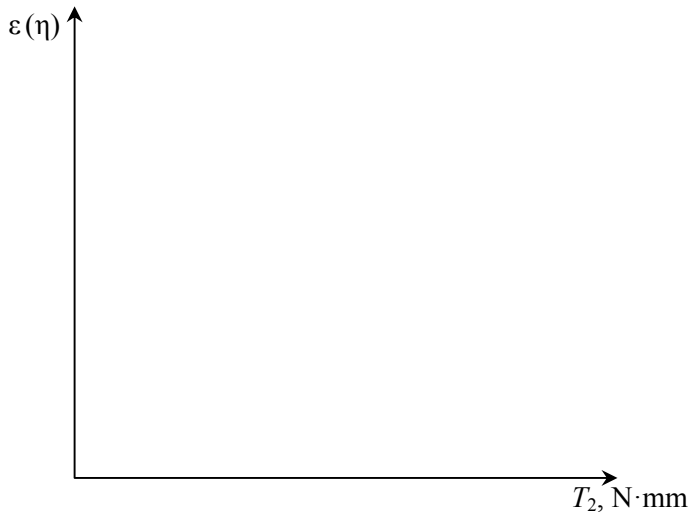


Fig. 3.6. Graphs $\varepsilon = f(T_2)$ and $\eta = f(T_2)$

Test questions

1. Scope, advantages and disadvantages of the belt drive.
2. Geometric and kinematic parameters of friction type belt drives.
3. Belt Tension Methods.
4. Forces acting on the shaft from the belt drive.

Laboratory Work № 4

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 geartrains.

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.

Description of PM-250 reduction gear

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 (fig. 4.1).

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.

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 clearance facilitates 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.

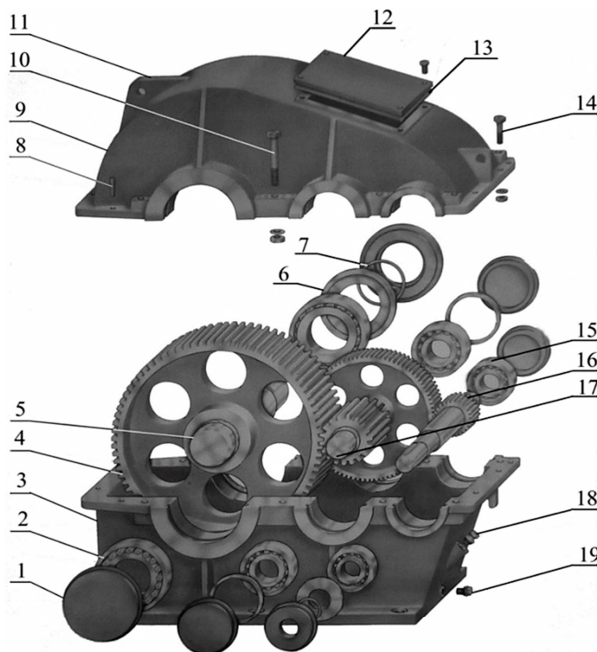


Fig. 4.1. Two-stage helical reduction gear:

- 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 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. 4.2).

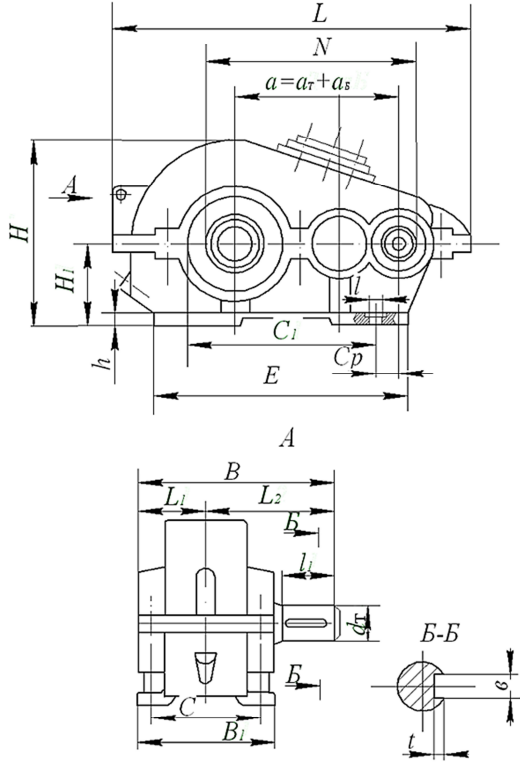


Fig. 4.2. Main dimensions of reduction gear

Put down the results of measurements into table 4.1.

Table 4.1

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°06 34
2.50	99	8°06 34	99	8°06 34	118	10°28 34
3.00	99	8°06 34	99	8°06 34	99	8°06 34

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 (fig. 4.2):

$$a = N - \frac{d_B}{2} - \frac{d_T}{2},$$

where d_B and d_T are diameters of input and output shaft ends.

2. How to disassemble the reducing gear (fig. 4.2):

- 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 table 4.2:

- a) count the number of pinion and gear teeth of high-speed and low-speed stages (fig. 4.1);
- b) measure outside gear and pinion diameters d_a ;
- 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 .

Theoretical Information

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 : $m = p / \pi$. ($m_t = p_t / \pi$; $m_n = p_n / \pi$; $m_x = p_x / \pi$). m_n and m_t are constrained: $m_n = m_t \cos \beta$ (fig.4.3).

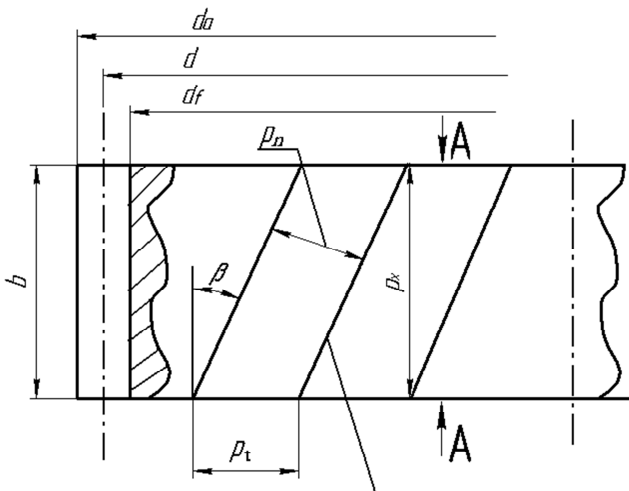


Fig. 4.3. To determine the size of gears:

$p_n = \pi m_n$ – normal pitch; $p_t = \pi m_t$ – circular pitch; $p_x = p_n \sin \beta$ – axial pitch

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 (see fig. 1.5).

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 :

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

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

Outside (addendum circle) pinion and gear diameters:

$$d_{a1} = d_1 + 2m_n;$$

$$d_{a2} = d_2 + 2m_n.$$

Center distance

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

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

The calculated m_n is consistent with the standard.
Center distance

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

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

The value of β calculated from this equation is consistent with those given in the table 4.1.

Centre distance modification coefficient (profile shift coefficient)

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

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

Outside (addendum circle) pinion and gear diameters

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

Sample design of laboratory work

BELARUSIAN NATIONAL TECHNICAL UNIVERSITY

Department "Machines and Machine Parts"

Laboratory Work № 4

**PARAMETER DETERMINATION TWO-STAGE HELICAL
REDUCTION GEAR**

Purpose of the work:

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

The work was performed by: _____

Group No. _____

The work was accepted by: _____

Minsk _____

General view of the reducing gear is shown in the fig. 4.4.

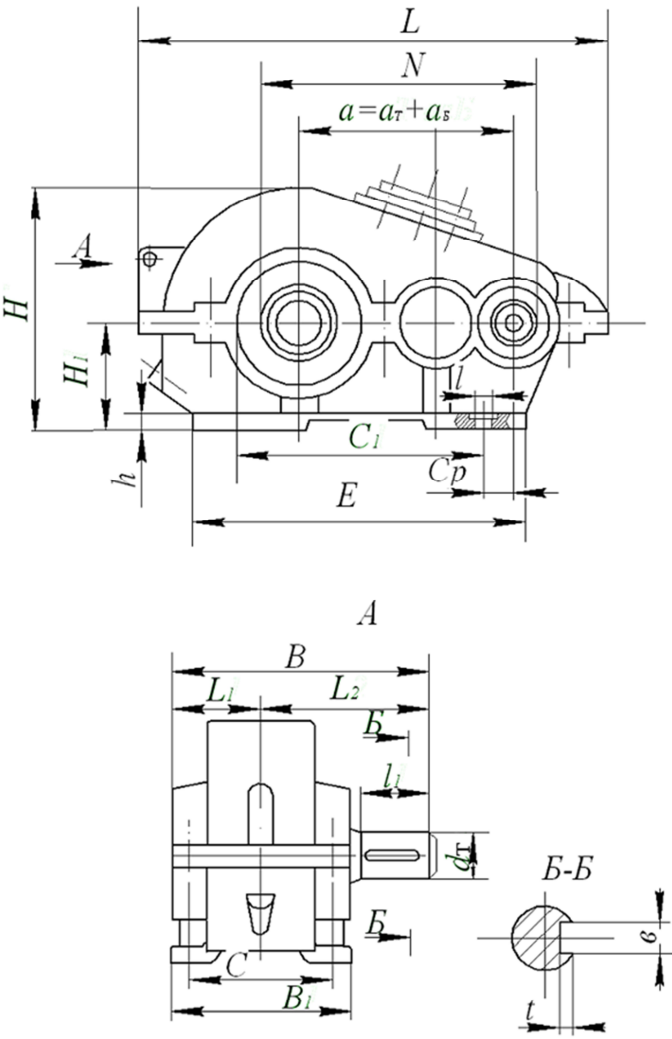


Fig. 4.4. General view of the reducing gear

Table 4.2

The overall and mounting dimensions of reducing gear

Size designation	H	L	B	H_1	h	C_p	C_1	C	E	B_1	L_1	L_2	$a_{w\delta}$	a_{wT}	d_δ	d_T	N
Dimensions, mm																	

Table 4.3

Values of gears and gear trans

Parameter	Formula	Values			
		Fast stage		Slow stage	
		Calc.	Mea-sur.	Calc.	Mea-sur.
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 $U = Z_{\text{gear}} / Z_{\text{pinion}}$					
Gear width factor $\psi_{ba} = b / a_w$					

End of the table 4.3

1	2	3	4	5	6
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_{tot}					

Test 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?

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КАЛИНА Алла Александровна
СУХОЦКИЙ Александр Анатольевич

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