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Kinematic calculation of gear reduction gear

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ABSTRACT

In this work based on standard calculations according to kinematic gears, a direct gear calculation is performed. A characteristic of the benefits of gearing is given. The kinematic calculations used when choosing an electric motor for a given kinematic scheme are substantiated. A diagram of the geometric parameters of the angle gear is constructed.

Key words: Kinematic calculation, gearbox, gear.

1. INTRODUCTION

A reducer is a mechanism consisting of gears or worm gears, made in the form of a separate unit and serving to transfer power from the engine to the working machine. The purpose of the gearbox is to reduce the angular velocity and increase the torque of the driven shaft compared to the drive shaft.

When choosing the type of gearbox for the drive of the working body, it is necessary to take into account many factors, the most important of which are: the value and nature of the load change, the required durability, reliability, efficiency, weight and overall dimensions, noise level requirements, product cost, operating costs.

Of all types of gears, gears have the smallest dimensions, mass, cost and friction losses. Toothed gears in comparison with other mechanical gears have great reliability, constant transmission ratio due to lack of slippage, and the possibility of using in a wide range of speeds and gear ratios. These properties provide a large distribution of gears. The disadvantages of gears can be attributed to the requirements of high precision manufacturing and noise when working at significant speeds.In the course project, it is necessary to design a gearbox for the chain conveyor. Chain conveyor is a type of conveyor in which traction is generated by one or two chains. Such conveyors, in comparison with belt conveyors, are capable of transporting goods with high temperature, heavy loads, and they have more productivity.

The gearbox consists of a cast iron housing in which the transmission elements are placed - gear, wheel, bearings, shaft.

2. RESEARCH

To select an electric motor, it is necessary to determine the overall efficiency of the gearbox. The overall efficiency of the gearbox is equal to the product of the efficiency of the series-connected movable links: the gear transmission and two pairs of bearings [2].

$\eta_{общ} = \eta_1 \cdot \eta_2^2$,

гдеη_1 – Efficiency gear pair;

η_2 – Efficiency of one pair of rolling bearings.

Accept$\eta_1 = 0,98$ [1];

$\eta_2 = 0,99$ [1].

Roughly, get: $\eta_{общ} = 0{,}98 \cdot 0{,}99^2 \approx 0{,}96$

We determine the required power on the high-speed shaft of the gearbox according to the formula [2]:

$P_1 = \frac{P_2}{\eta_{общ}}$, kW

WhereP_2 – power on the driven shaft,

$$P_2 = 9 \text{ kW}$$

$$P_1 = \frac{9}{0{,}96} = 9{,}47 \text{ kW}$$

At$P_1 = 9{,}47$kW,и$n_1 = 970\ \text{min}^{-1}$choose motor type 4A1606У3.

$P_{эл} = 11$ kW

$n_{эл} = 965\ \text{min}^{-1}$

$n_{эл} = n_1 = 965\ \text{min}^{-1}$

We will carry out a kinematic calculation of the designed gearbox. The kinematic diagram of the designed gearbox is shown in Figure 1.

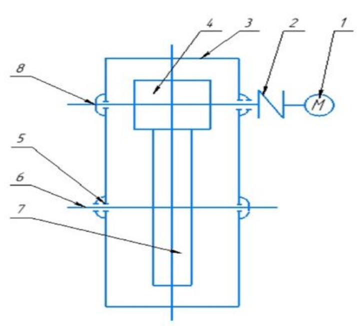

Figure 1: Kinematic diagram of the gearbox

1 - coupling, 2 - electric motor, 3 - gear housing,

4 - drive shaft (shaft - gear), 5 - rolling bearings, 6 - driven shaft, 7 - gear wheel, 8 - bearing cover.

As, since, n_1=$n_з$=970 min^{-1} иu=2,5 – standard
determine the speed of the driven shaft according to the formula [2]
Determine the gear ratio [2]
$n_2 = \frac{n_1}{u}$,
wheren_1 – drive shaft speed, $n_1=970$ min^{-1}
u –gear ratio, $u=2,5$.

$$n_2 = \frac{970}{2,5} = 388\ min^{-1}$$

We find the torque on the shafts by the formula:$T_1 = 9,55 \cdot \frac{P_1}{n_1}$, H·м

$$T_1 = 9,55 \cdot \frac{9470 \cdot 10^3}{965} = 93,23\ H \cdot m$$

$$T_2 = T_1 \cdot U \cdot \eta_{общ}\ , H \cdot m$$

$T_2 = 93,23 \cdot 2,5 \cdot 0,96 = 226,1 H \cdot m$
Determine the angular velocity:
$\omega = \frac{\pi n}{30}, s^{-1}$
Drive shaft angular speed:

$$\omega_1 = \frac{3,14 \cdot 965}{30} = 101,6 s^{-1}$$

The angular speed of the driven shaft:

$$\omega_2 = \frac{3,14 \cdot 388}{30} = 40,6 s^{-1}$$

The results of the kinematic calculation are written in the summary table 1.

Table 1: Kinematic Calculation Results

Design parameters	Shaft numbers	
	1	2
General gearbox efficiency η	η = 0,96	
Shaft speed n, min^{-1}	965	388
Gear ratio U	U=2,5	
Power P, kW	9,47	9
Rotational moment T, H·m	93,23	226,1
Angular velocity, s^{-1}	101,6	40,6

For further design of the gearbox, it is necessary to choose the material of gears with the definition of permissible stresses.
We select the material for the manufacture of gears and gears and determine the permissible stresses.
According to the steel tables, we assign steel to manufacture the gears 40XHwith hardness 269…302-HBheat treated improvement.

$$HB_1 = \frac{(269+302)}{2} = 285,5\ kgf / mm^2$$

According to the composition table of alloy steels, we assign steel to manufacture the gear wheel 40XHwith hardness 235 ... 262 HB with heat treatment improvement.

$$HB_2 = \frac{(235 + 262)}{2} = 248,5\ кгс/мм^2$$

Permissible contact stress for gear teeth is calculated by the formula [3]

$$[\sigma]_{H_1} = K_{HL} \cdot [\sigma]_{HO_1}$$
$$[\sigma]_{H_2} = K_{HL} \cdot [\sigma]_{HO_2}$$

whereK_{HL} – gear teeth longevity coefficient;
$[\sigma]_{HO_1}$and$[\sigma]_{HO_2}$ – permissible contact stresses corresponding to the number of cycles of alternating stresses NH0.

$$K_{HL} = \sqrt[6]{\frac{N_{HO}}{N_1}}$$

whereN_{H0} - the number of cycles of voltage changes corresponding to the endurance limit, according to the table 3.3. [2] accept$N_{H0} = 16,5 \cdot 10^6$;
N_1 – the number of voltage change cycles for the entire service life (operating time) (1.4)
$N_1 = 573 * \omega_1 * L_h$
Where ω_1 – angularvelocityofthecorrespondingshaft
L_h- drive service life (resource),

$$L_h = 15000\ hours$$

$N_1 = 573 \cdot 101,6 \cdot 15000 = 868 \cdot 10^6$
N_{HO}= 16,5*10^6
If N_1> N_{HO}hence$K_{HL} = 1$ (1.6)
According to table 3.1. [2] determine the permissible contact voltage,
corresponding to the number of voltage change cycles N_{H0}.
$[\sigma]_{HO_1}$=1,8*H_{cp} + 67 – for gear
$[\sigma]_{HO_1}$= 1,8*285,5+67=580,9 H/mm^2
$[\sigma]_{HO_2}$=1,8*H_{cp} + 67 – for wheel
$[\sigma]_{HO_2}$= 1,8*248,5+67=514,3 H/mm^2
Substitute the resulting values in the formula: $[\sigma]_{н1}$= 1*580,9=580,9 H/mm^2
Determine the allowable contact stress for the teeth of the wheel, substitute in the formula: $[\sigma]_{н2}$= 1* 514,3 = 514,3 H/mm^2
As $HB_1 - HB_2 = 285,5 - 248,5 = 37$=20…50, then the gear is calculated for strength at a lower value$[\sigma]_H$from received for gear$[\sigma]_{H1}$and wheels$[\sigma]_{H2}$, i.eon less durable teeth [4]
In this way, $[\sigma]_H = 513$ H/mm^2
Define the allowable bending stress for the gear teeth$[\sigma]_{F1}$and wheels$[\sigma]_{F1}$ [2,6]
We accept the coefficient of durability for gears and wheelsK_{FL_1}=1

$$[\sigma]_{F1} = K_{FL1} \cdot [\sigma]_{F01},$$
$$[\sigma]_{F2} = K_{FL2} \cdot [\sigma]_{F02},$$

where $[\sigma]_{F0}$ - permissible bending stress corresponding to the number of cycles of alternating stresses N_{F0}.
K_{FL}- gear ratio;
K_{FL_1}=1; K_{FL_2}=1
Calculations of the gear are listed in table 2.

Table 2: Design Gear Design

Transmission item	steel grade	$D_{пред}$. $S_{пред.}$	Heat treatment	HB_{1cp} HB_{2cp}	σ_B	$[\sigma]_H$	$[\sigma]_H$
					H/mm^2		
Gear	40XH	200 мм 125 мм	Improvement	285,5	920	580,9	220,5
Wheel	40XH	315 мм 200 мм	Improvement	248,5	800	514,3	192

3. RESULT AND DISCUSSED

According to the calculations of the gear transmission and the choice of the electric motor, we will carry out a final study of the basic geometric parameters of the transmission.

Determine the center distance according to the formula [5]

$$a_w = K_a(u+1)\cdot\sqrt[3]{\frac{T_2\cdot 10^3}{\psi_a\cdot u^2\cdot[\sigma]_H^2}\cdot K_{H\beta}}$$

where K_a– вauxiliary factor.

For helical gears K_a= 43

u – gear ratio, u=2,5

T_2- torque on a low-speed shaft when calculating the gearbox, T_2=226,1Н·м

ψ_a- rim width ratio, ψ_a= 0,3

$K_{H\beta}$- coefficient of uneven load along the length of the tooth

$K_{H\beta}$= 1;

$[\sigma]_H$ - permissible contact stress of the wheel with a less durable tooth;

$[\sigma]_H$= 514,3Н/mm^2

$$a_w = 43\cdot(2{,}5+1)\cdot\sqrt[3]{\frac{226{,}1\cdot 10^3}{0{,}3\cdot 2{,}5^2\cdot 514{,}3^2}\cdot 1} = 150{,}3\sqrt[3]{\frac{226100}{495946}} = 127{,}5\ mm$$

According to the value tables, we take aw for the series Ra 40 equal to 130 mm.

Determine the modulus of engagement [2]

$$m_n \geq \frac{2K_m\cdot T_2\cdot 10^3}{d_2\cdot b_2\cdot[\sigma]_F}$$

where K_m - auxiliary ratio, for helical gears

K_m = 5,8;

$b_2 = \psi_a \cdot a_w$ = 0,3 · 130 = 39 mm - gear rim width; $[\sigma]_F$ = 192 Н/mm^2- permissible bending stress of the material of the wheel with a less strong tooth; d_2 - pitch wheel diameter:

$$d_2 = \frac{2\cdot a_w\cdot u}{(u+1)} = \frac{2\cdot 130\cdot 2{,}5}{(2{,}5+1)} = 185{,}7mm$$

$$m_n \geq \frac{2\cdot 5{,}8\cdot 226{,}1\cdot 10^3}{185{,}7\cdot 39\cdot 192} = 2{,}52mm$$

Accepted by a number of standard numbers m_n= 2,75mm

Determine the angle of inclination of the teeth β_{min} for helical gearing [2] $\beta_{min}=\arcsin\frac{3{,}5\cdot m_n}{b_2}$

$b_2 = \psi_a\cdot a_w$-wheel rim width

b_2= 0,3·130 = 39 mm

$$\beta_{min} = \arcsin\frac{3{,}5\cdot 2{,}75}{39} = \arcsin 0{,}2467 = 14{,}2823°$$

Determine the total number of gear teeth and wheels:

Number of Teeth:

$$z_\Sigma = z_1 + z_2 = \frac{2\cdot a_w\cdot \cos\beta_{min}}{m_n}$$

где a_ω - center distance

a_ω=130 mm;

β - tooth angle,

β=14,2823°;

m – engagement module,

m=2,75mm

$$z_\Sigma = \frac{2\cdot 130\cdot 0{,}969}{2{,}75} = 91$$

$z_2 = z_\Sigma - z_1$ = 91-26=65

We specify the actual value of the angle of inclination of the teeth:

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

$$\cos\beta = \frac{(26+65)\cdot 2{,}75}{2\cdot 130} = 0{,}9625$$

β= 15° 18$'$

Determine the number of gear teeth:

$$z_1 = \frac{z_\Sigma}{(u+1)}$$

$$z_1 = \frac{91}{(2{,}5+1)\cdot} = \frac{91}{3{,}5} = 26$$

Determine the actual center distance $u_ф$ and check its deviation Δu from a given u:

$$u_ф = \frac{z_2}{z_1}$$

$$u_ф = \frac{65}{26} = 2{,}5$$

$$\Delta u = \frac{|u_ф - u|}{u}\cdot 100\% \leq 4\%$$

$$\Delta u = \frac{|2{,}5-2{,}5|}{2{,}5}\cdot 100\% \leq 4\%$$

0%< 4%

Determine the actual center distance:

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

$$a_w = \frac{(26+65)\cdot 2{,}75}{2\cdot 0{,}9625} = 130\text{ mm}$$

Determine the actual basic geometric transmission parameters, mm:

Gear pitch and wheel pitch diameters:

$$d_1 = \frac{m_n}{\cos\beta}\cdot z_1$$

$$d_2 = \frac{m_n}{\cos\beta}\cdot z_2$$

Where m– engagement module, m = 2,75 mm;

β - tooth angle, β = 15° 18$'$;

z_1 – gear teeth, z_1 = 26.

$$d_1 = \frac{2{,}75}{0{,}9625}\cdot 26 = 74{,}3mm;$$

$$d_2 = \frac{2{,}75}{0{,}9625}\cdot 65 = 185{,}7mm;$$

The diameters of the tops of the gear teeth:

$$d_{a_1} = d_1 + 2\cdot m_n$$

$$d_{a_1} = 74{,}3 + 2\cdot 2{,}75 = 79{,}8\ mm;$$

The diameters of the tops of the teeth of the wheel:

$$d_{a_2} = d_2 + 2\cdot m_n$$

$$d_{a_2} = 185{,}7 + 2\cdot 2{,}75 = 191{,}2\text{ mm};$$

The diameters of the cavities of the gear teeth:

$$d_{f_1} = d_1 - 2{,}4\cdot m_n$$

$$d_{f_1} = 74{,}3 - 2{,}4\cdot 2{,}75 = 67{,}7mm;$$

Wheel tooth hollow diameters:

$$d_{f_2} = d_2 - 2{,}4\cdot m_n$$

d_{f_2}=191,2 − 2,4 ·2,75 =179,1 mm;

Gear Crown Width:

b_1= b_2+4

b_1= 39 + 4 = 43 mm;

The result of the calculation of the cylindrical gear is written in table 3.

Table 3: The main parameters of the gear spur gear

Design calculation			
Parameter	Value	Parameter	Value
Center distance a_w	130	Tooth angleβ	15° 18′
Engagement modulem_n	2,75	Diameter of pitch circle: Gearsd_1 Wheelsd_2	d_1=74,3 d_2= 185,7
Gear Width: Gearsb_1 Wheelsb_2	b_1=43 b_2= 39		
Number of teeth: Gearsz_1 Wheelsz_2	z_1 =26 z_2=65	The diameter of the circle of vertices: Gearsd_{a_1} Wheelsd_{a_2}	d_{a_1}= 79,8 d_{a_2}=191,2
Type of teeth	Helical gear	The diameter of the circumference of the depressions: Gearsd_{f_1} Wheelsd_{f_2}	d_{f_1}=67,7 d_{f_2}=179,1

According to the calculation of the geometric parameters of a cylindrical gear transmission, a geometric scheme has been developed with the transmission parameter Figure 2.

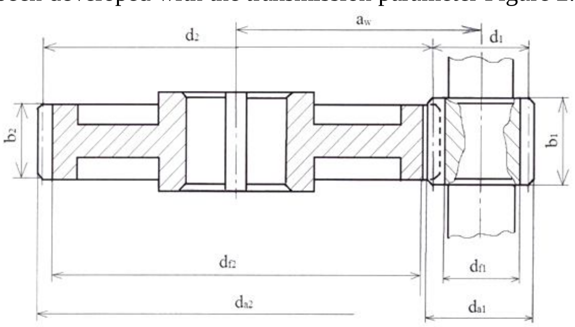

Figure 2: Diagram of transmission geometrical parameters

4. CONCLUSION

Thus, in this article, the geometric parameters of the cylindrical gear transmission of the gearbox are designed, the material from which this gearbox is to be manufactured is selected and the necessary calculations are made for which the cylindrical single-stage gearbox is assembled. These calculations can be used in the engineering design of the manufacture of automobiles, mechanisms and transmission elements. As a result of the calculations and the construction of a single-stage a cylindrical gearbox, you can identify the advantages and disadvantages of this type of gearbox:

Advantages of helical gears:

1. High gearbox efficiency.
2. High load capacity.
3. Low backlash of the output shaft, as a result of which the kinematic accuracy of cylindrical gears is higher than worm gears.
4. Low heat due to high gear efficiency.
5. Reversibility for any gear ratio, in other words, the absence of self-braking.
6. Confident work with uneven loads, as well as with frequent starts-stops.
7. High reliability.
8. Due to the high degree of variability of gears, it is possible to choose a gearbox with the closest gear ratio.

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