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Calculation of a Drive with a Worm Gear Transmission

Bafoev Bakhrom Botirovich – Probationer teacher
Bukhara Engineering-Technological Institute

Annotation. The article discusses the conceptual requirements of the project, standardization and unification were widely used, as well as worm gears are necessary to work with small gear ratios. It is due to such a gearbox that the smooth operation of the mechanism is ensured and a low noise level is achieved.

Keywords. Technical requirements, dimensions, designation, high-strength smooth operation; Low noise; Large gear ratio of one pair - worm gearboxes with a large gear ratio are much more compact and lighter than equivalent gears, and less material-intensive; compositional development.

To calculate the "Reducer" technological process, it is necessary to determine the type of production. This is necessary to select the type and model of equipment, fixtures, the form of organization of assembly and machining production, etc.

Worm gear calculation

Data: n=30 rpm (wheel speed)

T= 267,5 N·m (torque on the wheel)

u=50 (worm gear ratio)

t =18*· 10^3h (transmission resource)

Worm and wheel materials

Since there are no special requirements in the assignment regarding the dimensions of the transmission, we choose materials with average mechanical characteristics: for the worm - steel 45x, hardened to a hardness of H = 45HRCe, followed by grinding the working surfaces of the coils;

Expected sliding speed:

Vsec = $0{,}45\cdot10^{-3}\cdot n\cdot u\sqrt[3]{T}=0{,}45\cdot10^{-3}\cdot30\cdot 50\cdot\sqrt[3]{267{,}5} = 4{,}3$ m/sec

Based on the expected sliding speed, we select the material of the worm gear gear: group II - tin-free bronzes and brasses.

Permissible stresses

1) Permissible contact stresses

We determine the allowable contact stresses. For group II:

$[\sigma]H = \sigma_{H0}-25v_{ск} = 300 - 25\cdot 4{,}3 = 192{,}5$ МПа

where [σ] H0=300 is the allowable contact stress for worms with hardness ≥ 45HRCe

Life factor $K_{Hl} = \sqrt[8]{\frac{10^7}{N_{HE}}}\le 1{,}15 = \sqrt[8]{\frac{10^7}{0{,}259\cdot10^7}} = 1{,}09 \le 1{,}15$

$N_{HE} = K_{HE}N_k = 0{,}081\cdot 3{,}2\cdot 10^{-7} = 0{,}259\cdot 10^7$ - the equivalent number of loading cycles of the teeth of the worm wheel for the entire service life of the transmission.

$N_k = 60n_2L_h = 60\cdot 30\cdot 18000 = 3{,}2\cdot 10^7$

L_h - transmission time

for mode IV: according to Table 2.15 [Dunaev]$K_{HE} = 0{,}081$

Coefficient taking into account the intensity of wear of the wheel material:

C_V= 1.66 = 1.66 = $0.996 v_{ск}{}^{-0{,}352}\cdot 4{,}3^{-0{,}352}$

Permissible bending stresses

Determine the allowable bending stresses:

We calculate for the material of the teeth of the worm wheel.

$[\sigma]_F = K_{FL}\cdot[\sigma]_{F0};$

$K_{Fl} = \sqrt[9]{\frac{10^6}{N_{FE}}}$ – durability factor

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N_{FE}- the equivalent number of loading cycles of the teeth of the worm gear for the entire service life of the transmission:$N_{FE} = K_{FE}N_k$

$N_k$is the total number of voltage change cycles

N_k= $60n_2L_h$=60· 30 · 18000= $3{,}2 \cdot 10^7$

K_{FE}- coefficient of equivalence. We accept according to the table $K_{FE} = 0{,}016$

$N_{FE} = 0{,}016 \cdot 32{,}4 \cdot 10^6 = 0{,}52 \cdot 10^6$, because <, then we accept$10^6 N_{FE} = 10^6$

$K_{FL} = \sqrt[9]{\frac{10^6}{10^6}}$=1

$[\sigma]_{F0}$ for group II: :$[\sigma]_{F0}$=0,25· $\sigma_T + 0{,}08\sigma_B = 0{,}25 \cdot 460 + 0{,}08 \cdot 700 = 171$

$[\sigma]_F$ =1 MP$\sigma \cdot 171 = 171$

<u>Determine the center distance</u>

$$a_w \geq K_a \sqrt[3]{\frac{K_{H\beta} \cdot T}{[\sigma_H^2]}}$$

We accept an involute worm and;$K_a = 610$

$K_{H\beta}$ −load concentration factor: since the mode is constant, then$K_{H\beta} = 1$

$$a_w \geq 610 \sqrt[3]{\frac{267{,}5 \cdot 1}{192{,}5^2}} = 117{,}8 \text{ мм}$$

The resulting center distance is rounded for a standard worm pair:$a_w = 125$мм

<u>Basic transmission parameters</u>

1. Number of transmission turns -$z_1 = 1$

Number of wheel teeth -$z_2 = z_1 \cdot u = 1 \cdot 50 \approx 50$

2. Let's preliminarily define:

- Transmission module $m = \frac{(1{,}4\ldots1{,}7)a_w}{z_2} = \frac{1{,}55*125}{50} = 3{,}8$ мм
- Worm diameter factor $q = \frac{2a_w}{m} - z_2 = 2 \cdot \frac{125}{3{,}8} - 50 = 15{,}7$

 We accept m=4 ; q=16

3. Offset ratio:

$$x = \frac{a_w}{m} - 0{,}5(z_2 + q) = \frac{125}{4} - 0{,}5(50 + 16) = -1{,}75$$

4. Angle of elevation of the worm coil line:

- on the dividing cylinder:$\gamma = arctg\frac{z_1}{q} = arctg 1/16 = 3{,}57^0$

- on the initial cylinder:$\gamma_w = arctg\frac{z_1}{q+2x} = arctg 1/12{,}5 = 4{,}57^0$

5. Actual gear ratio

$u_ф = \frac{z_2}{z_1} = \frac{50}{1} = 50$. The obtained value does not exceed the error of 5%

<u>Worm and wheel dimensions</u>

1. Pitch diameter:

- Worm: $d_1 = mq = 4 \cdot 16 = 64\ mm$
- Wheels: $d_2 = mz_2 = 4 \cdot 50 = 200\ mm$

2. Top diameter:

- Vitkov: $d_{a1} = d_1 + 2m = 64 + 2 \cdot 4 = 72$ мм
- Wheel teeth: $d_{a2} = d_2 + 2m(1 + x) = 200 + 2 \cdot 4(1 - 1{,}75) = 194\ mm$

3. Diameter of depressions:

- Worm coils:$d_{f1} = d_1 - 2{,}4m = 72 - 2{,}4 \cdot 4 = 62{,}4$мм
- Wheel teeth:$d_{f2} = d_2 - 2m(1{,}2 - x) = 200 - 2 \cdot 4(1{,}2 - 1{,}75) = 204{,}4\ mm$

 Largest wheel diameter:$d_{a2М} = d_{a2} + \frac{6m}{z_1+k} = 194 + \frac{6 \cdot 4}{1+2} = 202$ мм

 K=2 - involute gear

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4. The length of the cut part of the worm:

$$b_1 = (10 + 5{,}5|x| + z_1)m = (10 + 5{,}5 * 1{,}75 + 1)4 = 82{,}5\ mm$$

5. Wheel rim width: =0.355 because =1$b_2 = \psi_a a_w = 0{,}355 \cdot 125 = 44{,}3$ мм, $\psi_a z_1$

2.6 Verification calculation of transmission strength

1. Calculate the design voltage

$$v_{\text{ск}} = \frac{\pi n_1 m(q + 2x)}{60000 * cos\gamma_w} = \frac{3{,}14 * 1500 * 4}{60000 * 0{,}989} = 0{,}25\ m/sec$$

Let's calculate the design stress: [σ]H, where - for involute worms; – load factor

$$\sigma_H = (z_\sigma(q + 2x)/z_2)\sqrt{(\frac{z_2 + q + 2x}{a_w(q + 2x)})^3 KT} \ \le\ z_\sigma = 5350K = K_{H\beta}K_{Hv}$$

The circumferential speed of the worm wheel is: $v_2 = \frac{\pi n d_2}{60000} = 0{,}3$ м/с

Since , then ;$v_2 < 3$ м/с$K_{Hv} = 1$

$K_{H\beta} = 1 + (\frac{z_2}{\theta})^3 \cdot (1 - X)$

θ- the coefficient of deformation of the worm, we select according to the table; X - is a coefficient that takes into account the influence of the transmission operating mode on the running-in of the teeth of the worm wheel and the turns of the worm.$\theta = 225$

Since the loading mode is light IV, then according to Table 2.17 [2, p. 35] we take X = 0.38;

$K_{H\beta} = 1 + (50/225)^3 \cdot (1 - 0{,}38) \Longrightarrow K = 1{,}006$

$\sigma_H = 5350(16 - 3{,}5)/50)\sqrt{(\frac{50+16-3{,}5}{125\cdot(16-3{,}5)})^3 1{,}006 \cdot 267{,}5} = 173{,}8 \ \Rightarrow \le 192{,}5\ Mpa$

transmission efficiency

We determine the transmission efficiency:

Worm gear efficiency

$\eta = tg\gamma_w/tg(\gamma_w + \rho)$,where

γ_w- the angle of elevation of the coil line on the initial cylinder;

ρ - reduced friction angle, it is equal to;$\rho\rho = 1{,}4^0$

$$\eta = \frac{tg4{,}57}{tg(4{,}57 + 1{,}4)} = \frac{0{,}079}{0{,}104} = 0{,}75 * 100\% = 75\%$$

Forces in engagement

1. Circumferential force on the wheel, equal to the axial force on the worm:

$$F_{t2} = F_{a1} = 2 \cdot 10^3 \cdot \frac{T}{d_2} = 2 \cdot 10^3 \cdot \frac{267{,}5}{200} = 2675\ N$$

2. Circumferential force on the worm, equal to the axial force on the wheel:

$$F_{t1} = F_{a2} = 2 \cdot 10^3 \cdot \frac{T}{(d_1 u_\phi \eta)} = 2 \cdot 10^3 \cdot \frac{267{,}5}{(64 \cdot 50 \cdot 0{,}75)} = 222\ N$$

3. Radial force:

$$F_r = F_{t2} 0{,}364 = 2675 \cdot 0{,}364 = 973{,}7\ N$$

$\sigma_F = \frac{KF_{t2}Y_{F2}cos\gamma_w}{1{,}3m^2(q+2x)} \le [\sigma]_F$, where

K - is the load factor, the values of which are calculated in clause 2.6;

Y_{F2}- coefficient of the shape of the wheel tooth, because accept

$$z_{v2} = \frac{z_2}{(cos3\gamma_w)} = \frac{50}{0{,}904} = 55{,}3\ Y_{F2} = 1{,}425$$

$\sigma_F = \frac{1{,}006\cdot 2675\cdot 0{,}800 cos4{,}57}{1{,}3\cdot 4^2(16-3{,}5)} = 8{,}25 \le 171$MPa

All conditions have been met.

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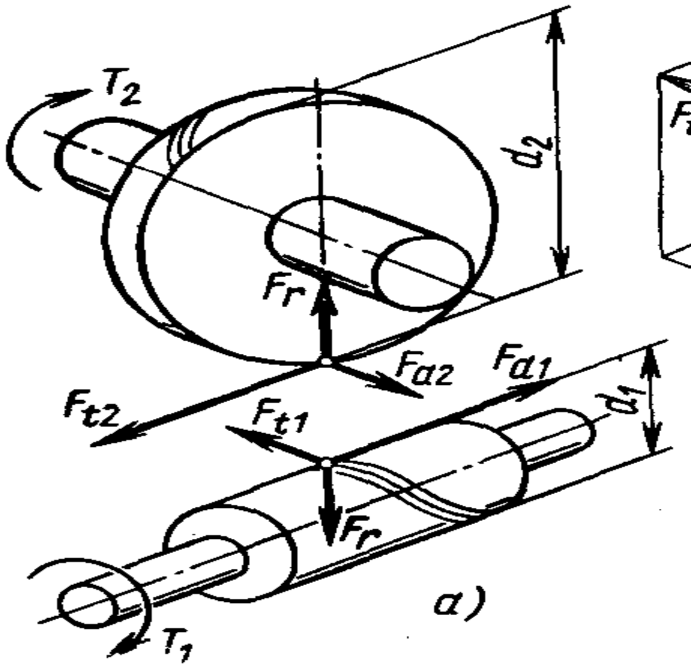

Checking wheel teeth for bending stresses

Conclusion:

As a result of the work done, a worm gear was designed, which is part of the electromechanical drive. A complete calculation of the drive was also made, consisting of a kinematic calculation, calculation of geometric parameters, power and verification calculations. The gearbox is made in a closed cast iron housing. Gearbox parts are made of high quality structural steel.

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