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Strength Calculation Of Planetary Gear Of The Seed-Removing Pipe

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Abstract: In the article, a planetary gearbox is adopted to ensure the efficiency and compactness of the vas deferent drive. The number of teeth of the sun gear $Z_1=12$, satellite $Z_2=12$ and main gear $Z_3=36$ with a module m=3 mm is checked for compliance with the condition of assembly and proximity to the number of satellites equal to K=4. Verification calculations of the sun gear teeth and the satellite axis for bending are carried out, in which the strength condition is met with a reserve of $\sigma_F=24.93$ MPa$\leq[\sigma_F]=$ 465 MPa – 18.5 times and $\sigma_u=20.26$ MPa$\leq[\sigma_u]=$ 60 MPa – 2.96 times. In addition, taking into account the strength calculations, the dimensional values of the planetary gearbox units of the vas deferent tube are determined.

Introduction:

A planetary gearbox is one of the types of mechanical gearboxes. This type of gearbox, widely used in many industries, is based on a planetary gear. A planetary gearbox is a gear mechanism, the characteristic feature of which is that the axes of some gear wheels are movable (Fig. 1) [1].

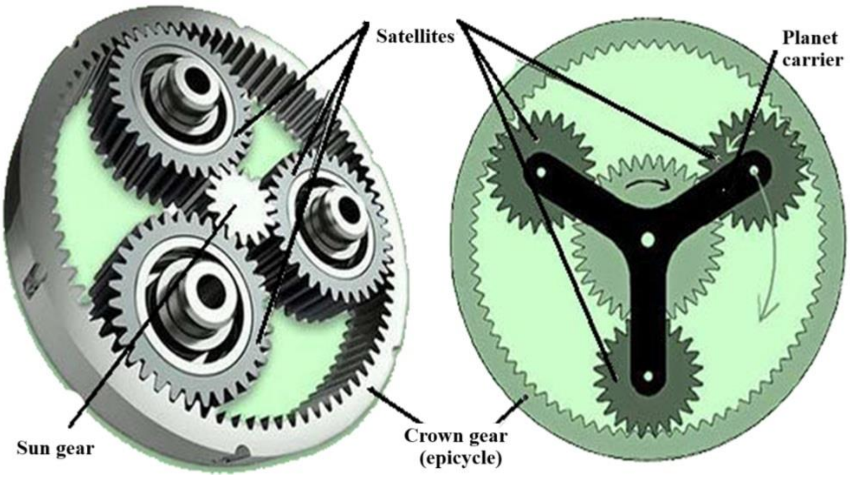

Fig. 1. Planetary gear diagram

The most popular type of planetary gear consists of the following elements:

- Sun gear – a small gear wheel with external teeth, located in the center of the mechanism
- Crown gear (epicycle) – a large gear wheel with internal teeth
- Planet carrier – this part of the planetary gear mechanically connects all the satellites. It is on the planet carrier that the rotation axes of the satellites are mounted.
- Satellites – small gear wheels with external teeth, located between the sun and crown gear. The satellites are in simultaneous engagement with both the sun and crown gear.

Due to their efficiency and compactness, planetary gearboxes have become widespread in mining engineering. A gearbox, like any mechanism, has a leading and a driven link. The leading link in a gearbox is a shaft that receives power from the engine; in a planetary gearbox, this is usually the central gear. The driven link is the output shaft of the gearbox, which sets some mechanism in motion. Considering the new layout of the gearbox elements [2].

It is known that a planetary gearbox, when working with high peripheral speeds, should take into account that during rotation any mechanism creates vibrations that negatively affect the operation of the mechanism as a whole. In addition, each engagement in the transmission, especially when it comes to multi-stage transmissions, negatively affects its power, which in turn depends on the efficiency. The efficiency of a closed planetary transmission without taking into account friction losses and oil mixing is within 0.96-0.98 [3].

Shaft misalignment of units is also a common problem of the lifting and turning mechanisms of electric quarry excavators equipped with planetary gearboxes. Signs of misalignment of electric motors with gearboxes are found on approximately 30% of objects from the surveyed sample. In this case, the spectrum is usually dominated by the second - third component of the electric motor rotation frequency, and the amplitudes of significant harmonics increase in those planes of the spatial position of the unit, where the misalignment of the shaft installation is more pronounced (the so-called "horizontal" misalignment is quite widespread on mining equipment units due to the low qualification of service and repair personnel). Signs of misalignment of the shafts of the first and second stages of planetary gearboxes are much less common, which can be explained by the rapid onset of emergency failure of the units. Destruction and/or jamming of gears in the case of shaft misalignment and general disruption of the geometry of gear engagements in planetary gearboxes develops very quickly, which often does not allow for the timely detection and assessment of diagnostic signs corresponding to this process [4, 5, 6].

The article by P.B. Guericke and A.G. Nikitin [7] summarizes the results of studies of the parameters of vibration generated during operation of planetary gearboxes, widely used in mining equipment, and applies the obtained results to solve the urgent problem of creating a single diagnostic criterion suitable for assessing the actual state and forecasting the degradation processes of the technical condition of planetary gearboxes, taking into account the features of their design and the specifics of the methodology for collecting diagnostic information. The obtained results prove the fundamental effectiveness of the proposed methodological approach for solving the problem of creating algorithms for developing uniform diagnostic criteria for assessing and forecasting the process of changing the technical condition of planetary gearboxes.

In the work of A.M. Pashkevich [8], a new method of engineering calculation of radial-plunger ball reducers was developed, allowing to determine all radial and axial dimensions of reducers, to get rid of undercutting of the central wheel and, consequently, to increase the load capacity of the transmission. The profile of the central wheel after approximation by the simplest lines was described by mathematical dependencies, which formed the basis of the new method.

To ensure the efficiency and compactness of the seed-transfer device drive, we adopt a planetary-type reducer. Further, to ensure compactness and efficiency, we perform a strength calculation of the planetary reducer units.

Initial data of the seed-discharging device planetary gearbox: torque on the drive shaft (pipe) T_t=77.95 N·m; rotation speed of the driven shaft (auger) n_{sh}=1440 rpm; rotation speed of the drive shaft (pipe) n_t=360 rpm; service life of the gearbox is 5 years, 240 working days per year, in three shifts of 8 hours.

1. Determine the gear ratio of the drive

1. Determine the gear ratio of the drive

$$i = \frac{n_{sh}}{n_t} \cdot = \frac{1440}{360} = 4. \tag{1}$$

2. The gear ratio of the planetary gearbox is determined by the following formula:

$$U_{pl} = U_{1H}^{3} = 1 - U_{13}^{H} \text{ hence } U_{1H}^{3} = \frac{Z_3}{Z_1} = 3 \tag{2}$$

3. Select the number of teeth of the sun wheel Z_1 = 12.

4. Determine the number of teeth of the satellite using the formula

$$Z_2 = 0.5 \cdot Z_1 \cdot \left(i_{1H}^{(3)} - 2\right) = 0.5 \cdot 12 \cdot (4-2) = 12. \tag{3}$$

5. From the above number of teeth of the main gear

$$Z_3 = Z_1 + 2 \cdot Z_2 = 12 + 2 \cdot 12 = 36. \tag{4}$$

6. We take the number of satellites (from the condition of balancing the forces in the engagement) K=4.

7. Check the number of satellites according to the neighborhood conditions:

$$\sin\frac{180°}{K} = 0.707 \geq \frac{Z_2 + 2 \cdot h_a^*}{Z_1 + Z_2} = 0.5833 \tag{5}$$

where h_a^*=1.

8. We check the number of teeth of the wheel according to the conditions of collection B, since its value must be an integer (natural) number:

$$B = \frac{Z_1 \cdot U_{1H}}{K} \cdot (1 + p \cdot K) = 9. \tag{6}$$

Here p=0. The condition is met.

9. To install the next satellite, it is necessary to turn the carrier to the following angle:

$$\varphi_H = \frac{360°}{K} \cdot (1 + p \cdot K) = 90° \tag{7}$$

here p=0 and K=4.

10. We select 40XH steel for gear wheels, improved, average hardness HB 280; basic number of stress change cycles [9]:

$$N_{H0} = 2.66 \cdot 10^7. \tag{8}$$

11. We determine the working number of voltage change cycles for the solar wheel for the entire service life t=5•240•3•8=28.8•10^3 h using the formula

$$N_H = 60 \cdot K \cdot n_1^{(H)} \cdot t = 60 \cdot 4 \cdot 1080 \cdot 28.8 \cdot 10^3 = 746.5 \cdot 10^7 \tag{9}$$

here $n_1^{(H)} = n_{sh} - n_t = 1440 - 360 = 1080$ rpm

12. Since $N_H > N_{H0}$, we take the durability coefficient K_{HL}=1 [9].

13. We determine the center distance between the sun wheel and the satellite [9]:

$$a_w \geq K_a \cdot \left((u+1)\sqrt[3]{\frac{T_{sh} \cdot K_{H\beta}}{n_c^{'} \cdot [\sigma_H]^2 \cdot u^2 \cdot \psi_{ba}}} \right) = 49.5 \cdot \left((1+1)\sqrt[3]{\frac{19.487 \cdot 10^3 \cdot 1.2}{3.3 \cdot 550^2 \cdot 1^2 \cdot 0.5}} \right) = 35.68\,mm \quad (10)$$

where K_a=49.5 – for cylindrical spur gear transmissions; $u=Z_2/Z_1$=12/12=1 – gear ratio; $T_{sh} = \frac{T_t}{i_{1H}^{(3)}} = \frac{77.95 \cdot 10^3}{4} = 19.487 \cdot 10^3 \;\; \mathrm{N} \cdot mm$ – screw torque; $K_{H\beta}$=1.2 – load concentration factor; $n_c^{'}=K$–0.7=4–0.7=3.3 – calculated number of satellites; $[\sigma_H] = \frac{\sigma_{H\lim b} \cdot K_{HL}}{[S_H]} = \frac{630 \cdot 1}{1.15} = 550\,MPa$ – permissible contact stress; $\sigma_{H\lim b} = 2 \cdot HB + 70 = 2 \cdot 280 + 70 = 630 MPa$ – limiting value of contact endurance; $[S_H]$=1.15 – safety factor; Ψ_{ba}=0.5 – satellite width factor.

We accept the center distance – a_w=36 mm.

14. We determine the engagement module [9]:

$$m = \frac{2 \cdot a_w}{Z_1 + Z_2} = \frac{2 \cdot 36}{12 + 12} = 3mm \quad (11)$$

15. Determine the width of the wheels [9]:

$$b = \psi_{ba} \cdot a_w = 0.5 \cdot 36 = 18mm \quad (12)$$

16. We perform a verification calculation of the teeth for bending using the values of the coefficients according to GOST 21354-75 [9]:

$$\sigma_F = 2 \cdot Y_F \cdot Y_\beta \cdot Y_\varepsilon \frac{T_{sh} \cdot K_{F\alpha} \cdot K_{F\beta} \cdot K_{Fv}}{n_c^{'} \cdot Z_2 \cdot b \cdot m^2} = 2 \cdot 3.8 \cdot 0.6 \cdot 1 \frac{19.487 \cdot 10^3 \cdot 1 \cdot 1.5 \cdot 1.2}{3.3 \cdot 12 \cdot 18 \cdot 3^2} \approx 24.93\,MPa. \quad (13)$$

17. Compare with the permissible stress:

$$[\sigma_F] = \frac{\sigma_{F\lim b} \cdot K_{FL} \cdot K_{FC}}{[S_F]} = \frac{555 \cdot 1 \cdot 1.5}{1.8} = 465\,MPa. \quad (14)$$

The strength condition σ_F=*24.93*MPa<$[\sigma_F]$=465 MPa is met.

Calculation of planetary gear satellite axes for bending

The most loaded part in planetary gearboxes is the satellite axis, so special attention must be paid to their calculation.

In general, the normal forces acting on the surface of the satellite teeth in the engagement pole of a cylindrical transmission are divided into circumferential F_t, radial F_r and axial S forces. In a straight-toothed cylindrical engagement [10, 11]:

$$F_t = \frac{2M_{kr}}{m \cdot Z_2};\; F_r = F_t \cdot tg\alpha;\; S = F_r : M_u = S \cdot \frac{m \cdot Z_2}{2}, \quad (15)$$

where for our case m=3 mm is the tooth module; $M_{kr}=T_{sh}/K$=19.48/4=4.87 N·m is the torque on the gear shaft; Z_2=12 is the number of satellite teeth; α=20° is the tooth profile angle in the normal section; M_u – is the bending moment created by the axial force S. Then the circumferential force F_t = 270.65 N, the radial force F_r=98.5 N, the axial force S=98.5 N, and the bending moment M_u =1773.15 N·mm.

When calculating the bending stresses arising in the cross section of the axis,

$$\sigma_u = \frac{M_u}{W_u} = \frac{1773.15}{98.17} = 18.06\,MPa \leq [\sigma_u] = 60MPa. \quad (16)$$

where $[\sigma_u]$=60–70 MPa is the permissible bending stress for carbon steel axles; $W_u = \frac{\pi\, d^3}{32} = \frac{3.1415910^3\,mm^3}{32} = 98.17mm^3$ – is the moment of resistance of the satellite axle during bending; d=10 mm is the diameter of the satellite axle.

The bending stability condition σ_u=*18,06* MPa < $[\sigma_u]$ =60 MPa is fulfilled.

CONCLUSIONS

For efficiency and compactness, a planetary gearbox of a reverse scheme is provided in the drive of the seed-removing pipe. In this case, the number of teeth of the sun gear Z1=12, satellite Z2=12 and main gear Z3=36 with a module m=3 mm, are checked for the condition of collection and proximity to the number of satellites equal to K=4.

Verification calculations of the teeth of the sun gear and the satellite axis for bending were carried out, in which the strength condition with a reserve of σ_F=24.93 MPa$\leq$[σ_F]= 465 MPa – 18.5 times and σ_u=20.26 MPa$\leq$[σ_u]= 60 MPa – 2.96 times was met. In addition, taking into account the strength calculations, the dimensional values of the nodes of the planetary gearbox of the seed-removing pipe were determined.

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