Journal of Engineering and Applied Sciences 10 (8-12): 243-246, 2015
ISSN: 1816-949X

Method of Loading Capacity Calculation of Bevel Precessional Gear for Pipeline Valve Drives

V. Syzrantsev, K. Syzrantseva and A. Pazyak
Tyumen State Oil and Gas University, Volodarskogo Street 38, 625000 Tyumen, Russia

Abstract: The study considers the problem of loading capacity calculation of bevel precessional gear for pipeline valve drives. By nowadays spiroid gear and worm gear are the basis for industrial engineering of gearboxes. These gearboxes have a lot of disadvantages such as a low efficiency, significant starting torque and low reliability in cold temperature operating conditions. Researchers propose to use gearboxes based on bevel precessional gear. Bevel precessional gear is one of the progressive types of gears with a small shaft angle. Bevel precessional gears allow to increase the loading capacity, reliability and durability of gearboxes to reduce their cost and mass and dimension characteristics and to improve their efficiency and performance criteria. The study consider the problem of determination the loading capacity by the transmitted torque considering multiple contact of teeth and fixed value of the maximum contact stress allowed by the pinion and gear materials. Researchers solve the inverse problem of loading capacity-transmitted torque is calculated at a fixed number of tooth pairs in contact. The results of studying the bevel precessional gear enable to implement original methods of evaluation its loading capacity based on contact stress.

Key words: Gearbox, bevel precessional gear, valve, pipeline, ball valve, transmitted torque, contact stress, loading capacity

INTRODUCTION

Currently, most gearboxes for pipeline valve drives are based on worm (Nabiev, 2010) or spiroid (Goldfarb *et al.*, 2007; Goldfarb, 2010, 2011) gears. Despite their low efficiency (typically, around 0.3) such gears allow implementing an efficient samples of gearboxes. A significant starting torque has a particularly negative impact on reliability of gearboxes in low operating temperatures. An alternative to both Russian and international gearboxes are those based on bevel precessional gear with a small shaft angle (Syzrantsev and Golofast, 2014a, b). Having a high efficiency and a small difference in the number of pinion and gear teeth such gears can be manufactured with a gear ratio 10-65 (Syzrantsev and Golofast, 2014a, b). Figure 1 shows gearbox with bevel precessional gear. Due to multiple contact of teeth, their load capacity significantly exceeds that of other gear types.

Fig. 1: Gearbox with bevel precessional gear

DESCRIPTION OF BEVEL PRECESSIONAL GEAR CONSTRUCTION

Figure 2 shows the design scheme of the proposed coaxial gearbox with bevel precessional gear: pinion 1 with an initial conical surface; gear 2 with ring gears 2 and 3 spaced apart by a value B, initial surfaces of which is a plane; moving pinion 4 with an initial conical surface; carrier 5 with an eccentric shaft sector, located at an angle Σ to the common axis O-O of the coaxial gearbox.

During operation of the coaxial gearbox, pinion 1 is fixed; gear 2 performs a compound motion-rotation about its axis and together with the carrier 5 about the axis O-O. Vertices of the pitch cones of the pinion 1 and ring gear 2 (pitch angle 90°) do not coincide. To exclude planetary motion of the gears in the most loaded (slow-speed) bevel precessional gear (gear coupling), composed of gears with ring gears 3 and 4, it is necessary that the vertex of the pitch cone of ring gear 4 and the vertex of ring gear 3 (pitch angle 90°) not only coincide but are at the crossing

Corresponding Author: V. Syzrantsev, Tyumen State Oil and Gas University, Volodarskogo Street 38, 625000 Tyumen, Russia

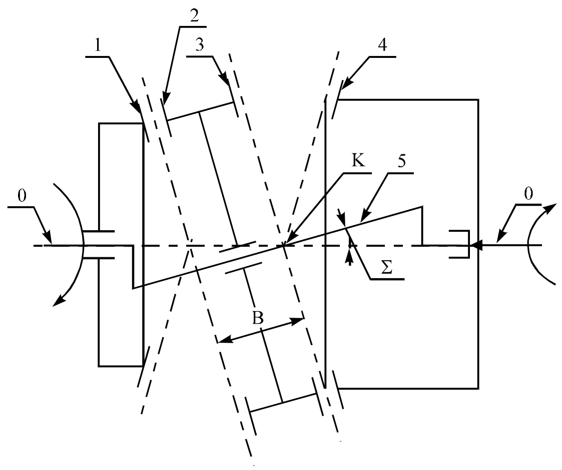

Fig. 2: Scheme of coaxial gearbox based on bevel precessional gear

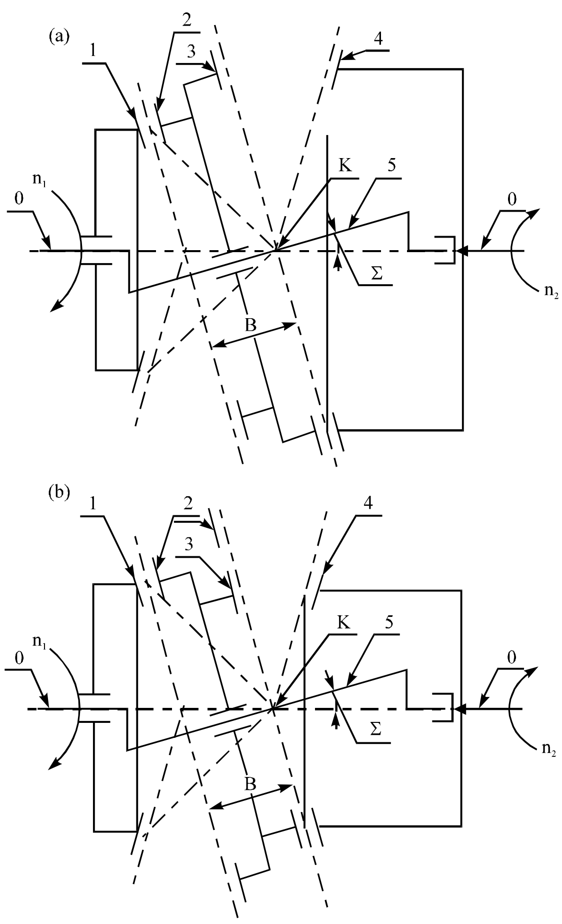

Fig. 3: Coaxial gearbox scheme, gear coupling module is: a) lager or b) smaller than bevel precessional gear

point of the axis of eccentric shaft sector and the axis of the coaxial gearbox O-O. This condition is met by only one value of B calculated by equation:

$$B = \frac{m_n}{2\cdot\cos\beta}\left(\frac{z_2}{tg\varepsilon} - \frac{z_1}{\sin\varepsilon}\right) \tag{1}$$

Where:

m_n = Normal module (mm)
β = Helix angle (°)
z_1 = Number of pinion teeth
z_2 = Number of gear teeth
Σ = Shaft angle (°)

Gear coupling, a bevel precessional gear of coaxial gearbox which is composed of gears with ring gears 3 and 4 in accordance with work (Denisov *et al*., 2014) is designed with number of teeth $z_3 = z_4$. The gear ratio is equal to one. In the gearbox, this gear is a slow-speed and therefore is the most loaded. To increase its load capacity and service life the normal module of this gear is taken as equal, larger or smaller than module m_n for a bevel precessional gear with gear rings 1 and 2 and $z_3 = z_4$ is taken as equal, larger or smaller than z_2 (Fig. 3). Equality of modules and that $z_3 = z_4 = z_2$ provides minimal radial dimensions of coaxial gearbox.

METHOD OF THE LOADING CAPACITY CALCULATION OF BEVEL PRECESSIONAL GEAR

Geometry and kinematics of gears mesh, the process of toothing and optimum parameters of gear-cutting tools and gear-cutting machine are discussed in work by Syzrantsev.

The objective of this work is to determine the loading capacity by the transmitted torque considering multiple contact of teeth and fixed value of the maximum contact stress (σ_{HP}) allowed by the pinion and gear materials. Bevel precessional gear geometry and geometrical parameters of gears mesh are known.

In order to determine the load distribution on each of the tooth pair in contact let us use the approach suggested in work by Ayrapetov. Instead of solving the direct problem of determining the number of tooth pairs n transmitted torque T_1 which is in fact the problem of static indeterminacy of the system with unilateral constraints, the number of which is unknown in advance, we solve the inverse problem of loading capacity-transmitted torque T_1 is calculated at a fixed number of tooth pairs in contact by Eq. 2:

$$\sum_{i=1}^{n} P_i \cdot R_i \cdot \cos\alpha = T_1 \tag{2}$$

Where:
P_i = Concentrated load (N)
R_i = Radius of load action on the tooth pair ith
α = Pressure angle (°)
T_1 = Transmitted torque (Nm)

To determine P_i we use the equation of strain and transition compatibility of teeth in the gearing:

$$W_i = \Delta\text{-}S_i;\ i = \overline{1,n} \quad (3)$$

Where:
S_i = Geometrical separation of tooth-surfaces, mm, calculated by the formula $S_i = \delta S_i\text{-}\delta S_1$ (tooth-surface separation on the first pair δS_1 is «selected» without any tooth deformation by means of rotation the pinion around its axis of rotation)
Δ = The gears approach, a measure of elastic deformation of the gearing
W_i = The deformation of the tooth pair ith

Equation 3 is represented as:

$$P_i = \left(\frac{S_0}{b_i}\right)^{3/2}\left[\left(F_n\text{-}F_i\right)\text{-}\frac{a_i}{S_0}P_i\right]^{3/2},\ i = \overline{1,n} \quad (4)$$

Where:
S_0 and F_i = Correspondingly, the amplitude and function of clearances between the teeth, depending on the geometrical characteristics of the gearing
a_i = The compliance of the tooth pair ith caused by their bending deflection
b_i = The coefficient described the contact compliance of the tooth pair i-th

Equation 4 is transcendent, solved numerically on P_i at a fixed n. Having determined P_i, the contact stresses (σ_{Hi}) that appear in each tooth pair contact i-th can be expressed as:

$$n_\sigma = 0.417\left(r_1/r_2\right)^{0.25};\ \sigma_{Hi} = n_\sigma\left[\left(E/r_1\right)^2 \cdot P_i\right]^{1/3} \quad (5)$$

Where:
E = Elastic modulus of pinion and gear materials
r_1 and r_2 = The main given radii of curvature in profile and longitudinal direction of the tooth line and in the surface contact point of the pinion and gear teeth, respectively

RESULTS

As an example of applying Eq. 2-5, let us consider the load capacity of the bevel precessional gear of a DU-300 ball valve with the following parameters: the number of pinion teeth $z_1 = 64$, the number of gear teeth $z_2 = 65$, the normal module $m_n = 3$ mm, the face width $b_w = 25$ mm. Using the software suit (Kotlikova and Syzrantsev, 2000) to solve Eq. 4 we determined the parameters S_0, F_i and to solve Eq. 5 we determined the main given radii of curvature: $r_1 = 204{,}082$ mm and $r_2 = 3333.0$ mm. After solving Eq. 4 with the number of pairs in contact n = 5, 7, 9 and 11, we determined the values of concentrated load P_i for each tooth pair ith in gear mesh (Fig. 4).

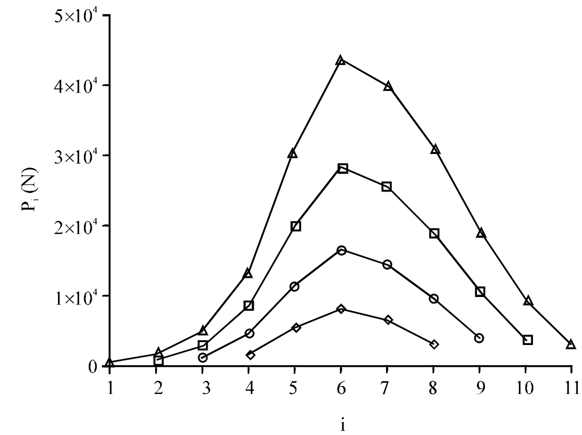

Fig. 4: Load distribution between contact pairs of bevel precessional gear

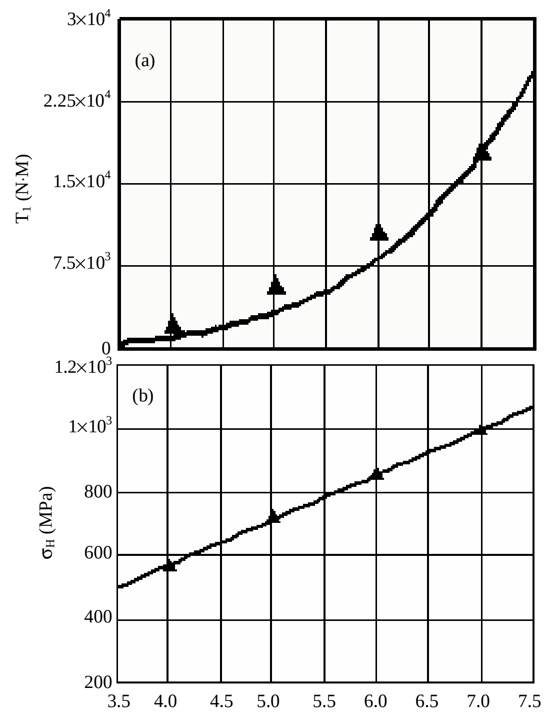

Fig. 5: a) Change of torque T_1 and b) maximum contact stresses depending on the number of contact teeth pairs

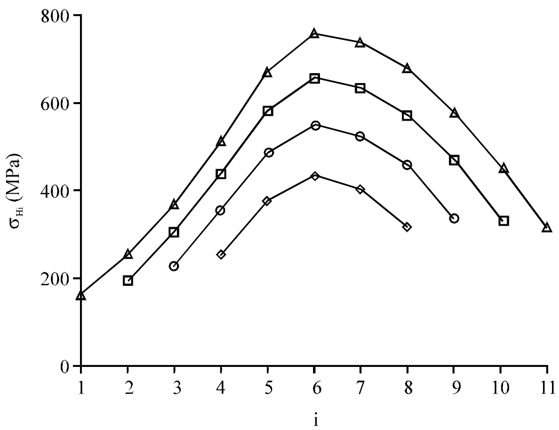

Fig. 6: Contact stresses change in the multiple contact of bevel precessional gear

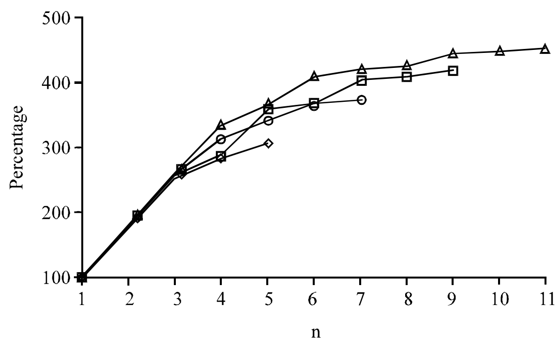

Fig. 7: Load capacity change in gearing depending on «n»

Using Eq. 2 for values n = 5, 7, 9 and 11, the calculated $P_i\ i = \overline{1, n}$ allow determining the transmitted torque T_1 and dependence points $T_1(n)$. The experimental results are shown in Fig. 5a. In result of its approximation we have received the equation:

$$T_1 = n^{5,036} \tag{6}$$

Having calculated Eq. 5 at the known $P_i\ i = \overline{1, n}$ for 5, 7, 9 and 11, let us determine the contact stresses σ_{Hi} in contact pairs. Their maximum value at n = const depending on n is illustrated in Fig. 5b and the load distribution of σ_{Hi} between pairs in contact at n = 5, 7, 9 and 11 is shown in Fig. 6 and 7.

CONCLUSION

The results of studying the bevel precessional gear enable us to implement methods of evaluation its loading capacity based on contact stress. Suppose that for the selected pinion and gear materials the maximum allowable contact stress σ_{HP} = 1000 MPa. By the graph given in Figure 5b let us determine the number of contacting pairs n = 7. After that for n = 7 by Eq. 6 or by the graph given in Fig. 5a let us determine the transmitted torque T_1 = 18026 Nm.

In order to prove the high load capacity of the bevel precessional gear (Fig. 7) demonstrates change of the torque at one-pair contact (100%) and multiple contact gear mesh when n = 5, 7, 9 and 11. It can be seen when n = 11 because loading capacity increases by a factor of 4.5.

REFERENCES

Goldfarb, V.I., 2010. Spiroid gears with small gear ratios. Some problems of design and production. Proceedings of the International Conference on Gears, October 4-6, 2010, Munich, Germany, pp: 429-442.

Goldfarb, V.I., 2011. Spiroid Gearboxes for Pipeline Valves. Veche Publishing House, Moscow, Russia, ISBN 978-5-9533-5882-8, Pages: 222.

Goldfarb, V.I., E.S. Trubachov and S.V. Lunin, 2007. System of hobs unification for gear-wheel cutting of worm-type gears. Proceedings of the 10th International Power Transmission and Gearing Conference, September 4-7, 2007, ASME, Las-Vegas, Nevada, USA., pp: 807-811.

Kotlikova, V. and V. Syzrantsev, 2000. Mathematical and program provision of design of bevel gearing with small shaft angle. Proceedings of the International Conference on Gearing. Transmissions and Mechanical Systems, July 3-6, 2000, Professional Engineering Publishing, UK., pp:13-18.

Nabiev, R.M., 2010. Screw gear motor: A relic of the past. Area Oil Gas, 6: 100-102.

Syzrantsev, V. and S. Golofast, 2014a. Drives of pipelines block valve based on the pan precess gear. Global J. Res. Eng., 14: 15-18.

Syzrantsev, V.N. and S.L. Golofast, 2014b. Pipeline stop valve pan-head processive transmission operator. Sci. Technol. Gas Ind., 1: 64-67.