POWER LOSSES ESTIMATION IN PRECESSIONAL GEAR

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INTRODUCTION

The planetary precessional transmissions represent a new principled type of the mechanical transmission. The specific of the relative sphere-spatial movement of the precessional gearing elements makes the sliding friction to persist. The study of the contact between two working surfaces of one gearing is very important, because the duration of the gearing running, its reliability and efficiency depend on the materials behavior, from which the prehension elements are produced. The study of the contact in the kinematical precessional gearing has a major importance, because in the kinematical precessional gearing there is slipping friction, which leads to big power losses and to big heat elimination in the contact zone.

1 ANALYSIS OF POWER LOSSES IN KINEMATICAL PRECESSIONAL TRANSMISSION

Power planetary precessional transmissions have been studied amply. To exclude the sliding friction in the gearing of the power precessional transmission, the tooth-roller gearing has been utilized, which replaced the sliding friction by rolling friction of the gearing. This fact allowed increasing the efficiency, and solution of problems related to fabrication technologies for the crown gears of the satellite block. The utilization of tooth-roller gearing in kinematical planetary precessional transmission is almost impossible because of small dimensions of the teeth. In such case it is reasonable to utilize the tooth-tooth continuous gear. The teeth of the central wheel have nonstandard convex-concave profile described by parametric equations according to the fundamental theory of the precessional gear [1].

The teeth of the planet gear are shaped with a circular arc profile (fig.1). The geometry of the circular teeth profile in normal section is marked by:

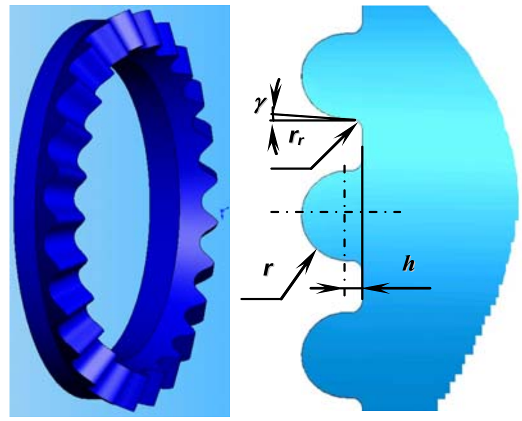
r_d – teeth radius of sphericity;

γ – angle of inclination of flank (technological parameter to ease the removal of the satellite from the cast);

r_r – radius of notch (to reduce the possibility of appearance of tension concentrator at the flank);

h – parameter which is determined by the shape of the central wheel tooth.

In planetary precessional transmissions which represent a rather complicated mechanism the total power losses are determined as the sum of power losses in the friction elements. For this purpose ample analysis



a. *b.*

Fig. 1: The tooth profile of the central wheel (a) and satellite wheel (b)

of the kinematical precessional transmission has been carried out, from the point of view of energy losses. On the basis of the performed analysis a structural model of the kinematical precessional reducer was elaborated (fig.2). It allows the determination of power losses in the friction torque.

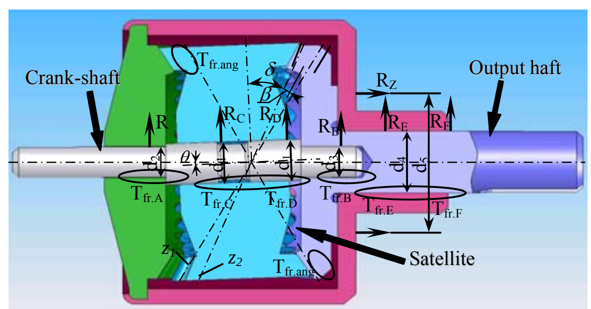


Fig. 2: Structural model of the kinematic precessional reducer

Structural analysis of the kinematic precessional transmission conducted to pointing out basic friction torque, which allows the determination of transmission efficiency. This fact allowed us to estimate quantitatively the power losses sources in the kinematical precessional transmission. The block diagram of power losses in the kinematical precessional transmission is shown in fig. 3.

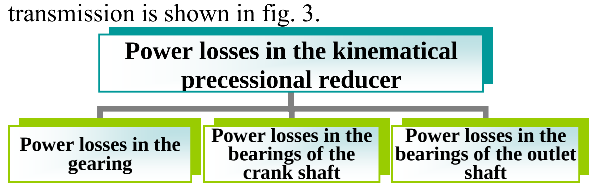


Fig. 3: Block diagram of power losses in the reducer.

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Further on, the study of power losses in each, previously presented, joint is described.

2 ESTIMATION OF THE KINEMATICAL PRECESSIONAL GEAR EFFICIENCY

To determine the losses in the gear it is necessary to carry out the study of the frictional coefficient dependent on:

- Various kinetostatic parameters;
- Geometry of gear elements;
- Material of the friction torque elements.

The sliding friction is predominant in the „tooth-tooth” kinematical precessional transmission. Research on the frictional coefficient demands the estimation of the sliding velocity in the gear, of the forces that drive in the gear and of the lubrication conditions.

The kinematical study of the precessional gear was performed in order to estimate the sliding velocity. A number of analytical calculus of precessional gear sliding velocity for various geometrical parameters has been done by using MATHCAD software: z_1, z_2 =10÷50; δ = 0÷30°; β = 2÷5°; θ = 1,5÷3,5°.

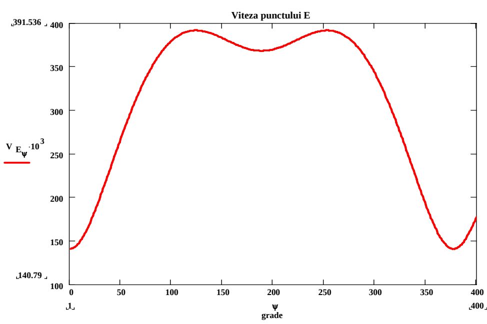

Fig. 4: **The sliding velocity graphic for the number of revolutions of the leading shaft** $n_{a.c.}=3000\ min^{-1}$

For the examined variants experimental investigations will be carried out regarding the friction coefficient of the materials couples. The experiment has been carried out on the laboratory installation „Amsler A 135” presented in the Fig. 5, which is placed in the

Fig. 5: **Laboratory installation** ***„Amsler A 135”***

laboratory of Machine Elements, the Technical University „Gh. Asachi”, Iaşi, Romania. Plastic mass „Hostaform C9021” [2] and steel 40X has been used as testing material.

Efficiency estimation of the precessional gear was carried out as result of ample research on the sliding coefficient of the precessional gear according to the geometry and material of the gear elements, and various kinetostatic parameters.

Efficiency is defined by the relation:

$$\eta_{ang} = \frac{T_{ies}}{T_{ies} + T_{fr.ang}}, \quad (1)$$

$$T_{ies} = \frac{F_t \cdot d_{m4}}{2}, \quad (2)$$

where: $T_{fr.\,ang.}$ gear moment of friction;
T_{ies}– outlet moment of torsion;
F_t – gear tangential force

$$F_t = F_n / cos\alpha_w;$$

F_n – gear normal force;
d_{m4} – average diameter of the gear wheel crown;
α_w – gearing angle (20-70°).

3 Determination of bearings’ mechanical losses

To estimate the losses in the kinematical precessional transmission bearings the following basic notions have been established:

- V_{int}- velocity of the inlet shaft;
- $V_{ieş}$- velocity of the outlet shift ;
- V_{satel}- satellite velocity;
- z_1; z_2- number of teeth of the gear elements.

Satellite velocity is defined by the relation:

$$V_{satel} = -V_{int}/z_2 = -V_{ieş} \cdot i/z_2. \quad (3)$$

Relative velocity of the satellite friction bearing.

$$V_{rel} = V_{int} \pm V_{satel} = V_{int} \pm V_{int}/z_2 = V_{int}(1 \pm 1/z_2), \quad (4)$$

where i – is gear reduction rate.

3.1 ESTIMATION OF THE FRICTION MOMENT IN THE CRANK SECTOR BEARINGS OF THE INLET SHAFT

The shaft diameter was defined (the fitting bore of the sliding bearing) d_1, in order to estimate the friction moment in the crank sector bearings of the inlet shaft. Then, the sliding velocity of the kinematical joint „crank-satellite wheel hub” will be:

$$V_{rel} = V_{int} \pm V_{sat} = \frac{\pi d_1 n_1}{60} \pm \frac{\pi d_1 n_1}{60 \cdot z_2}, m/s \quad (5)$$

where n_1 is the inlet speed.

The friction moment of bearings ***C*** and ***D*** (fig.2) is defined by the relations:

$$T_{frC} = R_C \cdot \mu \cdot {d_1}/{2}, N \cdot mm, \quad (6)$$

$$T_{frD} = R_D \cdot \mu \cdot {d_1}/{2}, N \cdot mm, \quad (7)$$

where μ is the sliding friction coefficient;
R_C – reacting force in bearing ***C***;
R_D – reacting force in bearing ***D.***

The summing friction moment in the sliding bearings of the satellite wheel is defined by the relation

$$T_{fr.sat\Sigma} = T_{frC} + T_{frD}, N \cdot mm. \quad (8)$$

3.2 ESTIMATION OF THE FRICTION MOMENT IN THE INLET SHAFT BEARINGS

For bearing ***A*** of the inlet shaft its diameter is d_2, ***mm.***

Sliding velocity in the bearing A is:

$$V_{al\,A} = \frac{\pi d_2 n_1}{60}, m/s, \quad (9)$$

and the friction moment in bearing ***A*** is:

$$T_{frA} = R_A \cdot \mu \cdot d_2/2, N \cdot mm, \quad (10)$$

where R_A is the reacting force of bearing ***A***

For bearing ***B*** shaft diameter is d_3, ***mm.***

Sliding velocity in bearing ***B*** will be:

$$V_{al\,B} = \frac{\pi d_3 n_1}{60}, m/s, \quad (11)$$

and the friction moment in bearing ***B*** is:

$$T_{frB} = R_B \cdot \mu \cdot d_3/2, N \cdot mm, \quad (12)$$

where R_B is the reacting force of bearing ***B.***

The summing friction moment in the inlet shaft bearings is:

$$T_{fr.int\Sigma} = T_{frA} + T_{frB}, N \cdot mm. \quad (13)$$

3.3 ESTIMATION OF THE FRICTION MOMENT IN THE OUTLET SHAFT BEARINGS

To define the friction moment in the outlet shaft bearings the following geometrical and kinematical parameters have been established:

- Outlet shaft diameter d_4, ***mm.***
- Average diameter of the axial bearing surface d_5, ***mm.***
- Sliding velocity of bearing ***E*** with the diameter d_4 will be:

$$V_{al\,E} = \frac{\pi d_4 n_2}{60}, m/s, \quad (14)$$

where n_2 is the outlet speed.

The friction moment in bearing ***E*** is:

$$T_{frE} = R_E \cdot \mu \cdot d_4/2, N \cdot mm, \quad (15)$$

where R_E is the reacting force in bearing ***E.***

The friction moment in bearing ***F***:

$$T_{frF} = R_F \cdot \mu \cdot d_4/2, N \cdot mm. \quad (16)$$

where R_F is the reacting force in bearing ***F.***

The friction moment in the axial bearing ***Z*** is:

$$T_{frZ} = R_Z \cdot \mu \cdot d_5/2, N \cdot mm. \quad (17)$$

where R_Z is the reacting force in bearing ***Z.***

The summing friction moment in the outlet shaft bearings:

$$T_{fr.ies\Sigma} = T_{frE} + T_{frF} + T_{frZ}, N \cdot mm. \quad (18)$$

3.4 ESTIMATION OF ENERGETIC PARAMETERS IN THE PRECESSIONAL TRANSMISSION BEARINGS

The summing friction moment in the transmission bearings is equal to components sum:

$$T_{fr.\Sigma} = T_{fr.int\Sigma} + T_{fr.cot\Sigma} + T_{fr.ies\Sigma}, N \cdot mm. \quad (19)$$

Then the efficiency of sliding bearings is defined by the relation:

$$\eta_{l.a} = \frac{T_{ies}}{T_{ies} + T_{fr.\Sigma}}. \quad (20)$$

Estimation of power losses in bearings:

In the sliding bearing of the crankshaft:

$$P_{fr.sat} = T_{fr.sat\Sigma} \cdot V_{rel} = T_{fr.sat\Sigma} \cdot V_{ies} \cdot i, \quad (21)$$

where ***i*** is the transmission ratio.

In the sliding bearings of the inlet shaft:

$$P_{fr.int} = T_{fr.int\Sigma} \cdot V_{int} = T_{fr.int\Sigma} \cdot V_{ies} \cdot i. \quad (22)$$

In the sliding bearing of the outlet shaft:

$$P_{fr.ies} = T_{fr.ies\Sigma} \cdot V_{ies}. \quad (23)$$

Summing losses at the time of sliding bearings friction:

$$P_{fr.\Sigma} = P_{fr.sat} + P_{fr.int} + P_{fr.ies}. \quad (24)$$

Efficiency of sliding bearings:

$$\eta_{l.a.} = \frac{P_{ies}}{P_{ies} + P_{fr.\Sigma}}. \quad (25)$$

4 MECHANICAL EFFICIENCY OF KINEMATICAL PRECESSIONAL TRANSMISSION

Having established the estimation relations of power losses in basic kinematical joints, it becomes relatively simple to establish the estimation relation of kinematical precessional transmission efficiency which, on the whole, is equal to the product of the two components (efficiency of sliding bearings and gear efficiency):

$$\eta_{tr} = \eta_{l.a} \cdot \eta_{ang}. \quad (26)$$

On the basis of the obtained relation an algorithm for the estimation of power losses in the kinematical precessional transmissions will be elaborated. Also a series of graphs for various geometrical and kinetostatic parameters will be produced. This thing will be useful to designers who will estimate quantitatively kinematical precessional transmissions at the phase of design.

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