ELABORATION AND DESIGN OF THE PLANETARY PRECESSIONAL TRANSMISSION

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1. INTRODUCTION

A problem for engineering companies is to satisfy the ever-increasing requirements to the transmissions used in majority of industrial machinery and technological equipment related to bearing capacity, compactness, mass and dimensions, low cost of production, to kinematical characteristics, structural compatibility with other aggregates of the equipment, etc. Gearings are considered the most sophisticated components of machines. Machine reliability depends very much on the gearing mechanical transmission operation, in general. The quality indices of traditional gears were increased largely by changing involute gearings, and by creating new gearings.

It becomes more and more difficult to satisfy the mentioned demands by partial updating of traditional transmissions. The target problem can be solved with special effects by developing new types of mechanical transmission - precessional planetary transmissions with multiple gear, that were developed by the authors. Due to these innovative features gearing multiplicity in planetary precessional transmission reaches 100% (in classical transmissions - 3-7%) which provides increased bearing capacity, reduced dimensions and weight, extended kinematical range ± 10 ... ± 3599 (in harmonic transmissions 79 ... 300), high kinematical accuracy, etc. The research team involved in research on precessional planetary transmissions published over 800 scientific articles, obtained about 180 patents, implemented about 20 practical achievements in the field of fine mechanics and specialized technological equipment, in robotic complexes for the exploration of ferro - manganese concretions from the World Ocean bottom (USSR concept), in spaceflight technique, etc.

Know-how in the elaboration of multicouple precessional gear, manufacturing technology and control methods, and a range of precessional transmission diagrams belong to the research team from the Technical University of Moldova [1,2].

- The specific character of sphere-spatial (precessional motions of the precessional transmissions pinion make impossible the utilisation of classical involute teeth profiles. This fact requires the elaboration of new profiles adequate to the sphere-spatial motion of pinion, which would ensure high performances to the precessional transmission. Carrying out on the principle of the transfer function continuity and gear based on the principles of the transfer function continuity and gear multiplicity which aims to:
- the elaboration of the gear mathematics model with account of the peculiarities;
- the analytical description of teeth profiles by a system of parametric equations on spherical surface and normal teeth section for inner and plane gear;
- CAD determination of geometrical and cinematic parameters influence of the gear upon the teeth profiles shape and the justification of their rational limits of variation;
- the elaboration of the theoretical basis evaluation of teeth gear multiplicity in precessional transmissions;
- area definition of gear multiplicity existence by 100% teeth couples.

The production of non-standard teeth profiles requires a new manufacturing technology. In the complexity of problem "*gear-synthesis-profile study- manufacturing*" the elaboration of efficient methods of teeth manufacturing which ensures a maximum productivity and reduced cost while satisfying the requirements related to the gear with precessional motion plays an important role. To solve this problem the following has been done:

- we elaborated the mathematical model of teeth generation which shows the interaction of teeth in precessional gear;
- we investigated the kinematics of the mechanism of method realisation for teeth generation;
- we determined the tool path of motion and the family envelope of the generating surface by using the computer;
- we elaborated and manufactured from metal milling and tooth grinding tools, inclusively their longitudinal modification.

2. KINEMATICAL STRUCTURE

Depending on the structural diagram, precessional transmissions fall into two main types – ***K-H-V*** and ***2K-H***, from which a wide range of constructive solutions with wide kinematical and functional options that operate in reducer and multiplier regime. The kinematical diagram of the precessional transmission ***K-H-V*** (fig. 1,a) comprises five basic elements: planet career ***H***, satellite gear ***g***, central wheel ***b*** with different number of teeth, connection mechanism ***W*** and the body (frame). The roller rim of the satellite gear ***g*** gears internally with the sun wheel ***b***, and their teeth generators cross in a point, so-called the centre of precession. The satellite gear ***g*** is mounted on the planet (wheel) career ***H***, designed in the form of a sloped crank, which axis forms some angle with the central wheel axis θ. Average gear ratio will be:

$$i^{g}_{HV_{med}} = -\frac{z_g - z_b}{z_b}. \qquad (1)$$

For $z_g = z_b + 1, i^{g}_{HV} = -1/z_b$, i.e. the drive and driven shafts have opposite directions.

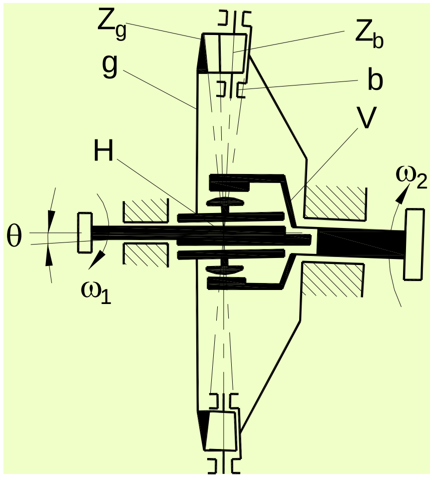
For $z_g = z_b - 1, i^{g}_{HV} = 1/z_b$, i.e. the shafts rotate the same direction. Precessional transmissions *K-H-V* fall under two basic types:

- with satellite wheel fixed to the casing;
- with central wheel fixed to the casing.

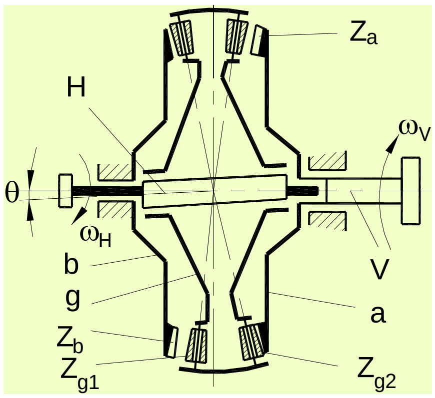
The kinematical diagram of the precessional transmission ***2K-H*** (fig. 1,b) comprises five basic elements: planet career ***H***, satellite gear ***g*** with two crown gears Z_{g1} and Z_{g2}, that gears with the unshiftable ***b*** and movable ***a*** central wheels.

$$i = -\frac{z_{g_1} z_a}{z_b z_{g_2} - z_{g_1} z_a}. \qquad (2)$$

The analysis of this relation demonstrates that precessional transmissions ***2K-H*** provide the fulfilment of a large range of transmission ratios $i = \pm(12...3599)$. It is necessary to point out the series of peculiarities of the precessional transmissions ***2K-H*** that ensure higher performances compared to similar planetary transmissions with cylindrical gears: precessional transmissions do not demand conditions of distance equality between the axis. This factor widens the area of their optimal design; precessional transmission kinematics does not limit the selection of the gear couples modules or of the rollers placement pitch. This factor increases the possibilities of shaping teeth pairs and of the transmission ratios interval; the peculiarities of the designed precessional gears allow increasing in the number of teeth that transmit the load simultaneously and this fact reduces significantly the dimensions and mass for the same loads compared to the traditional involute gearings.



a.



b.

Figure 1. Conceptual diagrams of precesional transmissions that operates efficiently in the multiplication regime.

Based on the carried out analysis a constructive diagram of the precessional planetary transmission was designed, taken as the base of precessional multipliers design. The precessional planetary transmission (Fig. 2) comprises the crank shaft 1 on which the satellite block 2, and the fixed and the movable wheels 3 (the movable wheel is connected to the shaft 5) with nonstandard profile of teeth are installed (Fig. 3) [3]. The satellite block 2 has two crown gears (6 and 7) with the teeth executed as conical rollers mounted on the axle with the possibility of revolving around them. The transmission operates in the multiplier mode, as follows: at the rotation of the input shaft 5 with the gear 4, due to the difference in the number of geared teeth ($Z_4 = Z_7 - 1$, $Z_3 = Z_6 - 1$), the satellite block 2 will

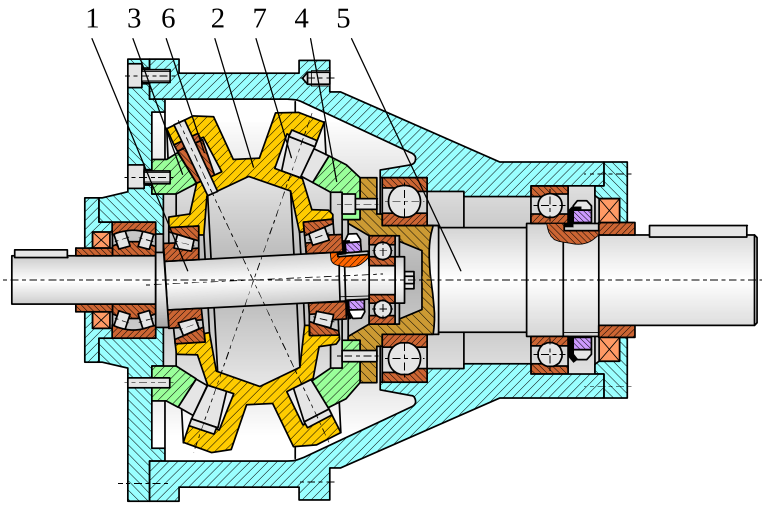


Figure 2. Constructive diagram of the planetary precessional transmission 2K-H.

perform a spherical-spatial motion around the point–centre of precession (the point of intersection of the crown gear roller axes and of the crank shaft axes 1), producing a complete precessional cycle at the rotation of the gear 4 at an angle equal to the angular pitch. Due to its mounting on the sloped side of the crank shaft 1, the precessional motion of the satellite block 2 is transformed into rotational motion of the crank shaft 1 that will produce a complete rotation during a complete precessional cycle of the satellite block.

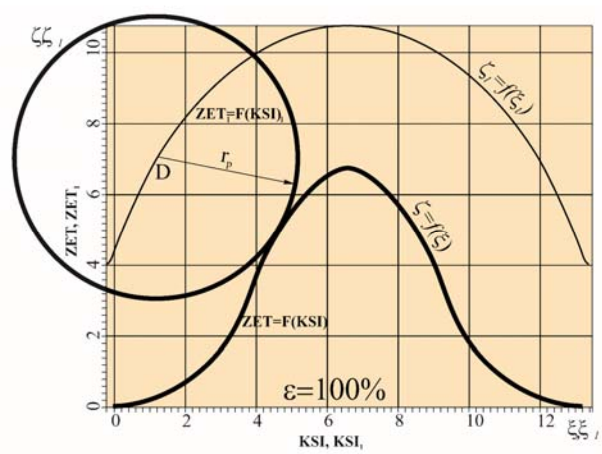


Figure 3. The teeth profile of central wheels.

3. CALCULATION OF PRECESSIONAL GEAR AT CONTACT PRESSURE

Calculation is done for the wheel with the smaller number of teeth Z_4, as it supports higher loads. The Anthology of Inventions [2,4], written by the authors, describes the calculation methodology and the nomographic charts for the selection of values of 5 basic parameters, that influence the teeth profile. The crank shaft pitch angle $\boldsymbol{\theta}$ is recommended in the boundaries ***1,5 – 3º***. Roller taper angle $\boldsymbol{\beta}$, generating angle (pressure angle) $\boldsymbol{\alpha_w}$, the coefficient of proportionality of the associated teeth radiuses $\boldsymbol{\nu}$ and the pitch angle of the roller axis δ are selected in conformity with the nomographic charts [2,4].

Preliminary calculation of the sun gear effective diameter. The basic parameters that need calculation are shown in fig. 4. The calculation is done in the following sequence.

Multiplicity gear $\boldsymbol{\varepsilon}$ is selected from the recommendations [2,4].

The number of teeth $\boldsymbol{Z_\varepsilon}$ that carry simultaneously the load is calculated:

$$Z_\varepsilon = \frac{Z_4 - 1}{2} \cdot \frac{\varepsilon}{100} \tag{3}$$

(as $\boldsymbol{Z_\varepsilon}$ is an integer, the decreasing integral value is selected).

The effective diameter of gear wheels $\boldsymbol{d_m}$ is calculated from the relationship:

$$d_m = 53 \cdot \sqrt[3]{\frac{T_2(1-\nu)\cdot K_{HP} \cdot K_{H\beta} \cdot K_{HV} \cdot \cos(\delta+\theta)}{(\sigma_{HO})^2 \cdot \psi_{bd} \cdot Z_\varepsilon \cdot tg\beta \cdot \cos\alpha_w}}\ ,mm \tag{4}$$

where: K_H is the experimental coefficient [85], that characterises the irregularity of load distribution between the teeth;

$K_{H\beta}$ is the experimental coefficient [2,4], that characterises the irregularity of load distribution on teeth length;

$\boldsymbol{K_H}$ is the experimental coefficient [2,4], that characterises load dynamics;

$\boldsymbol{\psi_{bd}}$ is the coefficient of tooth length [2,4] compared to diameter d_m.

According to the effective diameter d_m the following can be calculated:

– length of gear wheel tooth $\boldsymbol{b_w}$:

$$b_w = \psi_{bd} \cdot d_m,\ mm \tag{5}$$

– length of rollers $\boldsymbol{b_{wr}}$:

$$b_{wr} = b_w + 2,\ mm \tag{6}$$

– rollers' diameter in medium and abut section

$\boldsymbol{d_{mr}}$, $\boldsymbol{d_{rr}}$:

$$d_{mr} = d_m \frac{tg\beta}{cos(\delta+\theta)}, \ mm \qquad (7)$$

$$d_{rr} = d_{mr} + b_{wr} \cdot tg\beta, \ mm \qquad (8)$$

– diameter of rollers' axes $\boldsymbol{d_a}$:

$$d_a = d_{mr} - b_{wr} \times tg\beta - 2\Delta, \ mm \qquad (9)$$

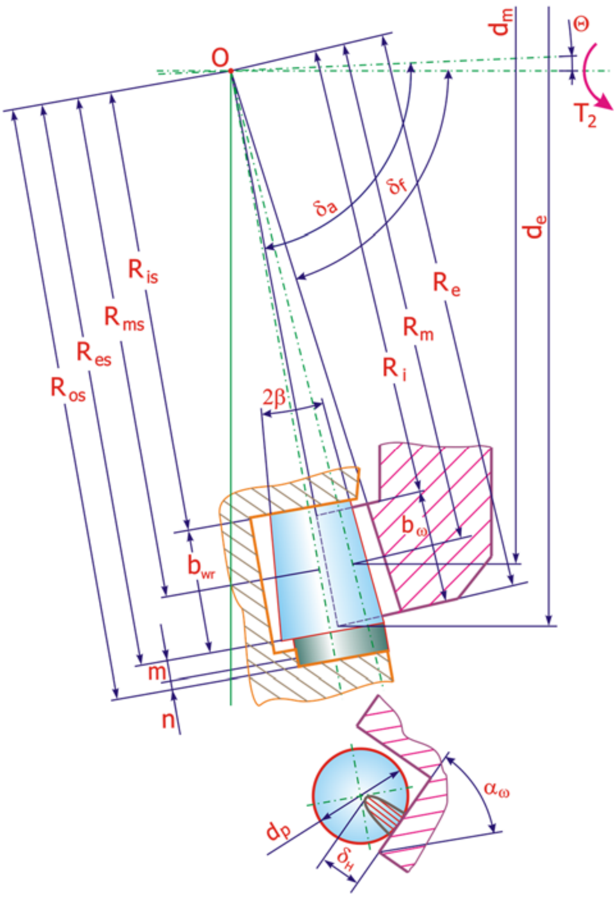


Figure 4. Precessional gear geometry.

where Δ, *mm* is the thickness of roller plate (wall) in minimal section selected in the boundaries *1...5mm*.

Determination of the allowable contact pressures considering the rolling friction

Initially, the tooth-roller friction velocity $\boldsymbol{V_{gl}}$ is determined:

$$V_{gl} = K_1 \cdot ln \left[\frac{f_{max}}{f_{max} - \frac{K}{d_{mr}} - \frac{f \cdot d_a + f \cdot \left(\frac{d_{rr} + d_a}{2} \right) sin\alpha_w \cdot sin(\delta \cdot \beta_i) \cdot cos\delta + K}{d_{mr}}} \right], \qquad (10)$$

where $\boldsymbol{K_1}$ – is the coefficient depending on the working conditions of the upper kinematical coupling [2,4];

$\boldsymbol{f_{max}}$ – maximal value of the friction coefficient;

$\boldsymbol{K}$– rolling friction coefficient;

$\boldsymbol{f}$ – friction coefficient;

Taking into account $\boldsymbol{V_{gl}}$, the allowable contact pressure $\boldsymbol{\sigma_{HOi}}$ is calculated from the relationship:

$$\sigma_{HOi} = \sigma_{HO} \left(0{,}28 + 0{,}72 \cdot e^{-\frac{V_{gl}}{K_3}} \right), \qquad (11)$$

where: $\boldsymbol{K_3}$ – is the coefficient, selected from the recommended ones [2,4].

Determination of gear wheel effective diameter and calculation of the geometrical parameters

The average diameter $\boldsymbol{d_m}$ of the gear wheels is defined by the relationship:

$$d_m = 53 \cdot \sqrt[3]{\frac{T_2 (1-\nu) \cdot K_{HP} \cdot K_{H\beta} \cdot K_{HV} \cdot cos(\delta+\theta)}{(\sigma_{HOi})^2 \cdot \psi_{bd} \cdot Z_\varepsilon \cdot tg\beta \cdot cos\,\alpha_w}}. \qquad (12)$$

And the medium conical distance of the satellite rim with rollers $\boldsymbol{R_{ms}}$ – is defined by the relationship:

$$R_{ms} = \frac{d_m}{2\,cos(\delta+\theta)\left(1 - tg\beta \cdot sin\,\alpha_w \cdot tg(\delta+\theta)\right)}, \qquad (13)$$

Depending on the medium diameter $\boldsymbol{d_m}$ the following can be calculated:

– length of gear wheel tooth $\boldsymbol{b_w}$:

$$b_w = \psi_{bd} \cdot d_m, \ mm \qquad (14)$$

– length of rollers $\boldsymbol{b_{wr}}$:

$$b_{wr} = b_w + 2, \ mm \qquad (15)$$

– rollers' diameter in medium and abut section $\boldsymbol{d_{mr}}$, $\boldsymbol{d_{rr}}$:

$$d_{mr} = d_m \cdot \frac{tg\beta}{cos(\delta+\theta)}, \ mm \qquad (16)$$

$$d_{rr} = d_{mr} + b_{wr} \cdot tg\beta, \ mm \qquad (17)$$

– diameter of rollers' axes $\boldsymbol{d_a}$:

$$d_a = d_{mr} - b_{wr} \cdot tg\beta - 2\Delta, \ mm \qquad (18)$$

where Δ is the thickness of roller plate (wall) in minimal section selected in the boundaries *1...5mm* (calculation is done similarly for another couple of rollers).

Further on, the following can be calculated:

– the medium conical distance of the gear wheel:

$$R_m = \frac{d_m}{2\cos\left(\delta+\theta+ctg\left(\frac{d_{mr}\cdot \sin\alpha_w}{2R_{ms}}\right)\right)}\ mm \quad (19)$$

– exterior and interior conic distances of the roller rim $\boldsymbol{R_{es}}$, $\boldsymbol{R_{is}}$:

$$R_{es} = R_{ms} + \frac{b_{wr}}{2},\ mm;\ R_{is} = R_{ms} - \frac{b_{wr}}{2},mm \quad (20)$$

– exterior and interior conic distances of the gear wheel rim $\boldsymbol{R_{er}}$, $\boldsymbol{R_{ir}}$:

$$R_{es} = R_{mr} + \frac{b_w}{2},\ mm; \quad R_{is} = R_{mr} - \frac{b_w}{2},\ mm \quad (22)$$

– crest and foot cone angles of the teeth $\boldsymbol{\delta_f}$, $\boldsymbol{\delta_v}$

$$\delta_f = 90^\circ - (\delta+\theta+\beta),\ ^\circ;\ \delta_v = \delta_f + 2\theta,\ ^\circ. \quad (23)$$

Verification of teeth resistance at contact pressure. Allowable contact pressures $\boldsymbol{\sigma_H}$ are calculated by formula:

$$\sigma_H = 275\sqrt{\frac{2T_2(1-\nu)K_{HP}K_{H\beta}K_{HV}}{d_m d_{mp} b_w Z_\varepsilon \cos\alpha_w}} < [\sigma, MPa]. \quad (24)$$

Calculation is repeated with the constructive parameters modification, in particular of the *„tooth-roller”* couple to the condition $\boldsymbol{\sigma_{HOi} \geq \sigma_H}$.

Given the specific operation of the micro hydropower plant (24 hours out of 24) and taking into account the fact that the multiplier is overloaded with dynamic tasks, it is strictly necessary that the gear is subject to verification of teeth resistance at contact pressure for each operation system.

4. DESIGN OF PRECESSIONAL REDUCER STRUCTURE

On the basis of the undertaken study, diagram 2K-H was selected for the development of precessional multiplier of the micro hydropower plant. As a result of analysis of a wide range of tooth profiles with different geometrical parameters of gear by using the mathematical modelling package MathCAD 2001 Professional, the optimum tooth profiles were selected with account of their functioning in conditions of multiplication. Also, in MathCAD 2001 Professional software the calculation of geometrical parameters of precessional gear was done (fig. 5).

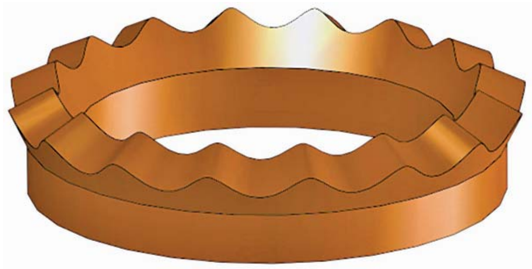
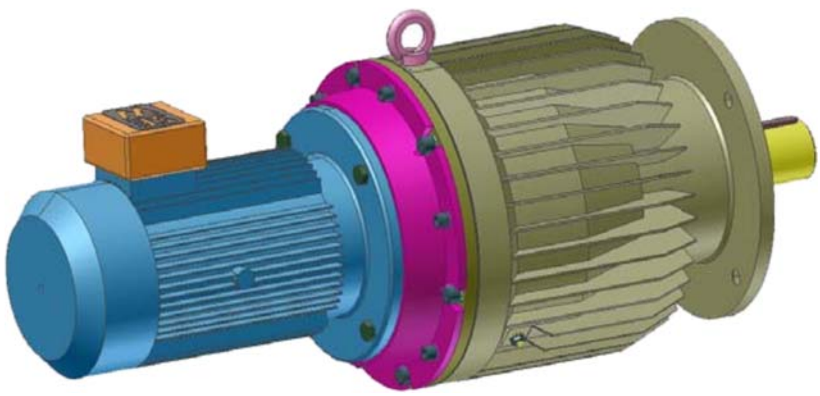


Figure 5. Central wheel with nonstandard profile of teeth.

The structures of precessional reducer were designed in SolidWorks software. The planetary precessional reducer is connected by flange with an electric motor, which allows obtaining a compact module, coaxial with the working machine. The structure from fig. 6 is proposed for planetary precessional reducer functioning in conditions of



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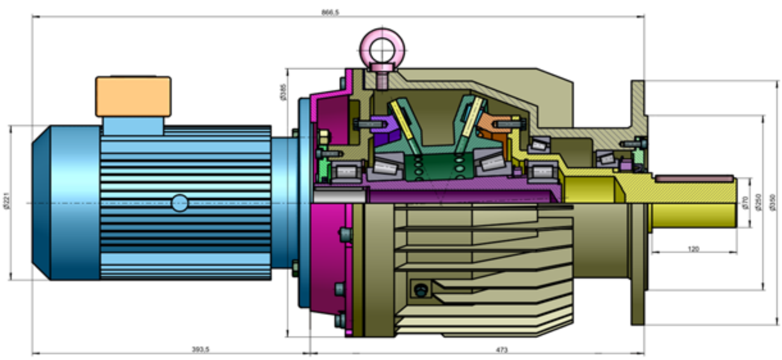

b.

Figure 6. Planetary precessional multiplier: a – general view; b – section view.

lower temperatures. On the Fig. 7 the main components of precessional reducer are shown.

To simulate the multiplier assembly and functioning, the dynamic computerised model of the precessional reducer was developed in AutoDesk MotionInventor. The planetary precessional transmission has reduced dimensions and mass, high lifting capacity and reduced ratio up to i = 144 (based on a two stage diagram) with satisfactory mechanical efficiency.

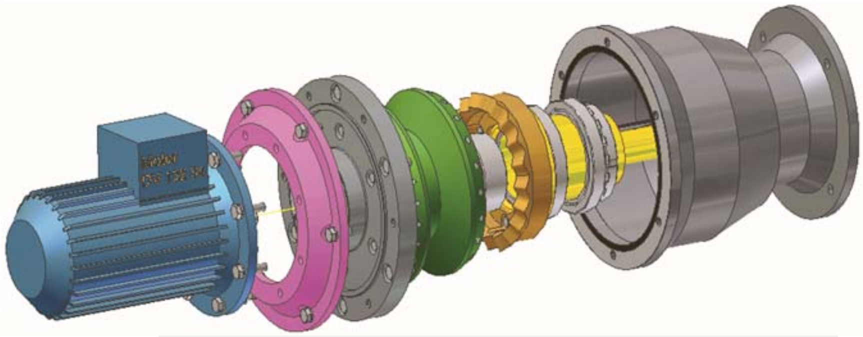

Figure 7. The basic components of precessional gearing reducer.

CONCLUSIONS

Among the characteristics of the estimated results of the research in the field of new and efficient drive development we can enumerate the followings:

- the precessional drives elaborated ensure: high load-carrying capacity; high mechanical efficiency; high kinematic accuracy; low noise level and vibrations;
- costs of drives becomes more attractive as the costs of other equivalent drives.

The structural optimization of the precessional transmissions will allow synthesis of new schematics of precessional transmissions with constant and variable transmission ratio and elaboration of new schematics of precessional transmissions for specific running conditions.

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