CAE DEVELOPMENT OF PRECESSIONAL DRIVES USING AUTODESK INVENTOR PLATFORM

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Abstract*: The paper presents the modelling and simulation of precessional drives designed in two variants capable of high transmission ratio and torque for one stage compact construction. The constructions were designed in Inventor and also as multi body systems in MotionInventor. The simulations of the drives provide information concerning positions, velocities, accelerations, point trajectories, forces and moments, energies, as well as contact forces at the contact between gear teeth and satellite teeth and other data concerning the system.*
Key words: precessional transmissions, CAD/CAM, simulation

1. Introduction

The engineering methods based on computer permitted to develop a new type of precessional transmissions with multi-couple meshed teeth, which, from the technological point of view, can be manufactured by means of a new method of processing conical teeth with convex-concave profile.

It appeared the necessity of elaboration of new profiles adequate to the sphero-spatial motion of the gears which would ensure high performances to the precessional transmission. Considering the necessity of achieving the transfer function continuity and gear multiplicity some objectives were taken into account. One of them is the integrated methods of design, modelling and simulation using powerful means of creation and management of parametrical models of the mechanical assemblies on the basis of CAD-CAE.

2. General information

The ever-growing requirements, especially, concerning the bearing capacity, the kinematical accuracy and the kinematical possibilities, impose the necessity to develop a new type of planetary gearing with distinct performances. The gearing improving is one solution of the problem. *Novicov-Wildhaber, Symarc* and other gearings have increased considerably the bearing capacity of gear. Another direction of developing gears is the design of new types of mechanical gears.

The creative search of designers has crowned with the elaboration of a new type of gearing – harmonic gear. *W. Musser,* an American engineer, patented the action principle of the harmonic gear in 1959. Starting that year *W. Musser* patented a big number of diverse constructive diagrams for harmonic gears (teeth, friction and tapped gears) and couplings, and demonstrated the possibilities of the new construction principle of mechanical gears. Thus, in 1961, the harmonic drive was produced at one of the American companies, for the first time at industrial scale. Harmonic drive are compact and possess increased bearing capacity; they provide high kinematical accuracy and possibility to transmit motion in sealed mediums, which is one of the basic advantages of harmonic drive. As their disadvantages we can mention reduced reliability of the flexible element (and, thus, of the gear, on the whole), reduced working order at high speeds, and also some technological difficulties.

By the end of the 70s Prof. Ion Bostan designed a new type of gears, new in principle, - precessional planetary gears with multiple gearing (Bostan, 1991). Over 20 years, research was carried out comprising the total range of problems from the idea to the implementation: fundamental theory of the precessional gear-engineering calculation procedures-Know-How manufacturing technologies-applications. *The research results have been published in over 450 scientific papers, in about 150 patents, in 3 monographs and in one design guidebook.*

The absolute multiplicity of the precessional gear (up to 100% of teeth pairs, geared simultaneously, compared to 5-7% in classical gears) provides increased bearing capacity and kinematical accuracy, small dimensions and mass. In addition to the above said, extended kinematical possibilities (±8÷3600 compared to 79÷300 in sinusoidal gears), reduced acoustic emission and solution of all technological problems, as advantages, open new perspectives for precessional planetary gears utilisation in various fields of mechanical engineering (Bostan, 1991). On the whole, precessional planetary gears can be divided in two basic groups:

- power precessional planetary gears;
- kinematical precessional planetary gears.

As the planetary precessional transmission is the new transmission, in the beginning of effectuation the simulation and calculation with programs CAE it is necessary to calculate theoretically base parameters. On fig. 1 is shown kinematical scheme of planetary precessional transmission, but on fig. 2 is shown the design of planetary precessional transmission.

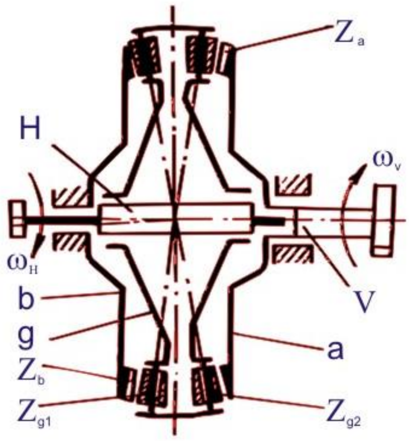

Fig.1. Kinematical scheme of planetary precessional transmission

The main elements of transmission make: crankshaft 1, block satellite 2, fixed wheel gear 3 and mobile wheel gear 4. The Important point this that, having such design (single-stage) is possible to receive transfer rate up to 3600, using designs of double-stage it is possible to receive transfer rate up to 14 million. Obviously that, having such big transfer rate, there are big loadings on a tooth. This problem is solved by that, all

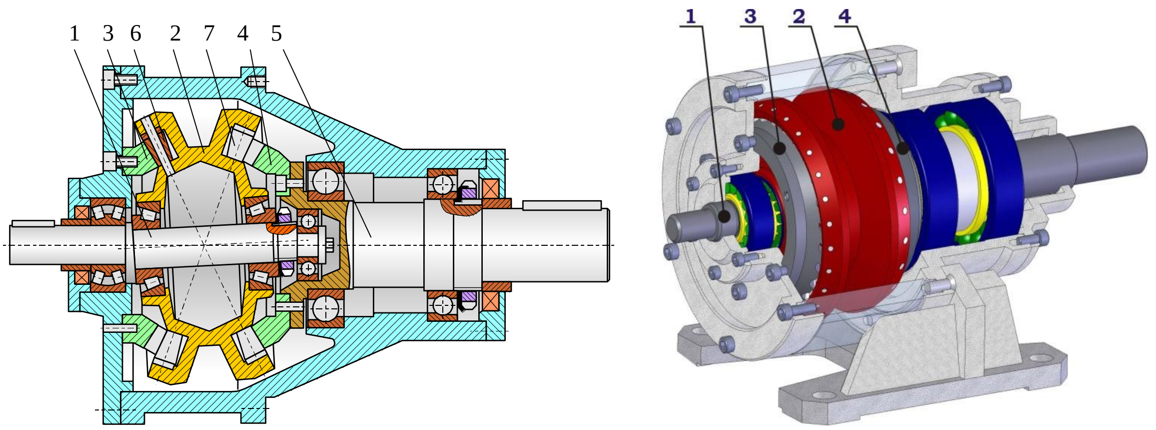

Fig. 2. Design of planetary precessional transmission

teeth it is participate in the gearing, and loading is transferred by half from them. Meaning that, all teeth are in gearing at the big loadings, the kinematic error is small.

Principle of functioning of the planetary precessional transmission the following: crankshaft 1 with inclined section to specify a block satellite 2 spatially spherical movement, block satellite by means of roller crown interaction with fixed wheel gear 3 and mobile wheel gear 4 (having a special structure generated by means of the equations), in turn a mobile wheel gear 4 it is rigidly connected with output shaft of a reducer, transfers the moment and speed of rotation. The direction of rotation of an input shaft and output shaft can be in one or in different directions. It is visible and in calculation of transfer rate if it has positive number we have an identical rotation. The transfer rate of a planetary precessional transmission is defined by the relation (1).

$$i = -\frac{Z_{g_1} Z_a}{Z_b Z_{g_2} - Z_{g_1} Z_a}, \tag{1}$$

where: Z_{g_1}, Z_{g_2} are number of a roller the crown satellite g_1 and g_2; Z_a and Z_b is represented number of a teethes the cog-wheels a and b.

3. Calculation by simulation cae of kinetostatics parameters of precessional transmission

3.1. 3D model elaboration of planetary precessional transmission

Calculation of planetary precessional transmission by a simulation is carried out using the simplified 3D model created in program Motion Inventor 2004+ but (4) from which it is possible to determine and check up dynamic loadings in bearings which have been designed earlier.

Dynamic processes in planetary precessional transmission derive, to a great extent, from the interaction of conical rollers of the satellite crowns with generating surfaces of central wheel teeth. The bearing capacity defined by gear forces (static and dynamic), the noise emission and the transmission vibroactivity, depend on the gear dynamic processes on the whole. With account of these important factors in the elaboration of 3D model of the initial precessional gearing, the linear contour of central wheel teeth profile (fig. 3, a) were designed applying parametric equations:

$$\begin{aligned}
&X_{1E}^m = k_2^m Z_{1E}^m + d_2^m;\\
&Y_{1E}^m = k_1^m Z_{1E}^m - d_1^m;\\
&\qquad Z_{1E}^m = \frac{(k_1^m d_1^m - k_2^m d_2^m)}{k_1^{m2} + k_2^{m2} + 1} -\\
&Z_{1E}^m = \frac{\sqrt{(k_1^m d_1^m - k_2^m d_2^m)^2 + (k_1^{m2} + k_2^{m2} + 1)\cdot(R_D^2 - d_1^{m2} - d_2^{m2})}}{k_1^{m2} + k_2^{m2} + 1},
\end{aligned} \tag{2}$$

Were

$$k_1^m = \frac{X_{1D}^m \left(X_{1D}^m \dot{X}_{1D}^m + Y_{1D}^m \dot{Y}_{1D}^m \right) + Z_{1D}^{m\,2} \dot{X}_{1D}^m}{Z_{1D}^m \left(X_{1D}^m \dot{Y}_{1D}^m - Y_{1D}^m \dot{X}_{1D}^m \right)}; \quad k_2^m = -\frac{k_1^m Y_{1D}^m + Z_{1D}^m}{X_{1D}^m};$$

$$d_1^m = \frac{R_D^2 \cos\beta \, \dot{X}_{1D}^m}{\left(X_{1D}^m \dot{Y}_{1D}^m - \dot{X}_{1D}^m Y_{1D}^m \right)}; \quad d_2^m = \frac{R_D^2 \cos\beta + d_1^m Y_{1D}^m}{X_{1D}^m}.$$

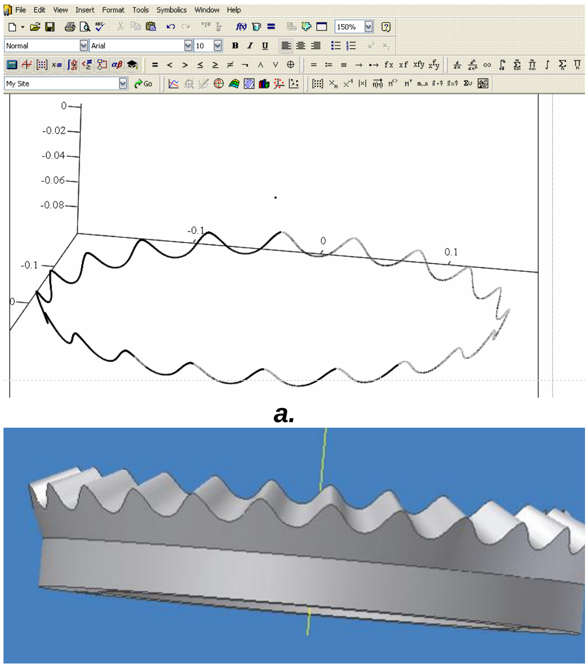

Fig. 3. 3D linear contour of central wheel teeth profile (a), and 3D model of the central wheel (b)

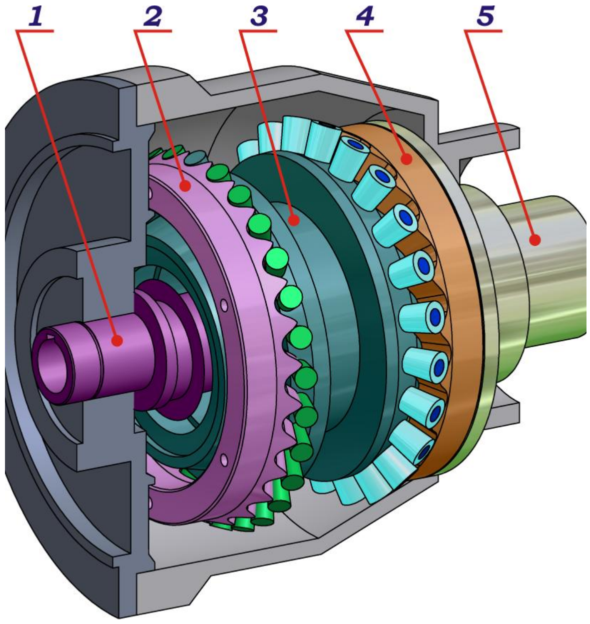

Fig. 4. 3D Model of planetary precessional transmission

The 3D model of the central wheel (fig. 3, b) was designed by using CAD Autodesk Inventor [5]. The 3D model for calculation is shown on fig. 4. It includes a crankshaft 1, a fixed wheel gear 2, the block satellite 3, a mobile wheel gear 4 rigidly connected with output shaft 5.

The dynamic model has been created on the basis of rigid model. As the initial data has been specified speed on input shaft [deg/sec] and the moment of torsion on the output shaft [Nm]. Kinematic joints (fig. 5), have been enclosed according to movements in gearing. The crankshaft and the case of a reducer by means of cylindrical roller bearings are connected by a cylindrical joint to an opportunity of a self-centering. A crankshaft and the block satellite by means of two tapered bearings with pair rotation in points of the appendix of loading on bearings (point - line), pair rotation (revolution) rollers on an axis of the block satellite and 3D contact of rollers to a mobile and fixed wheel gear.

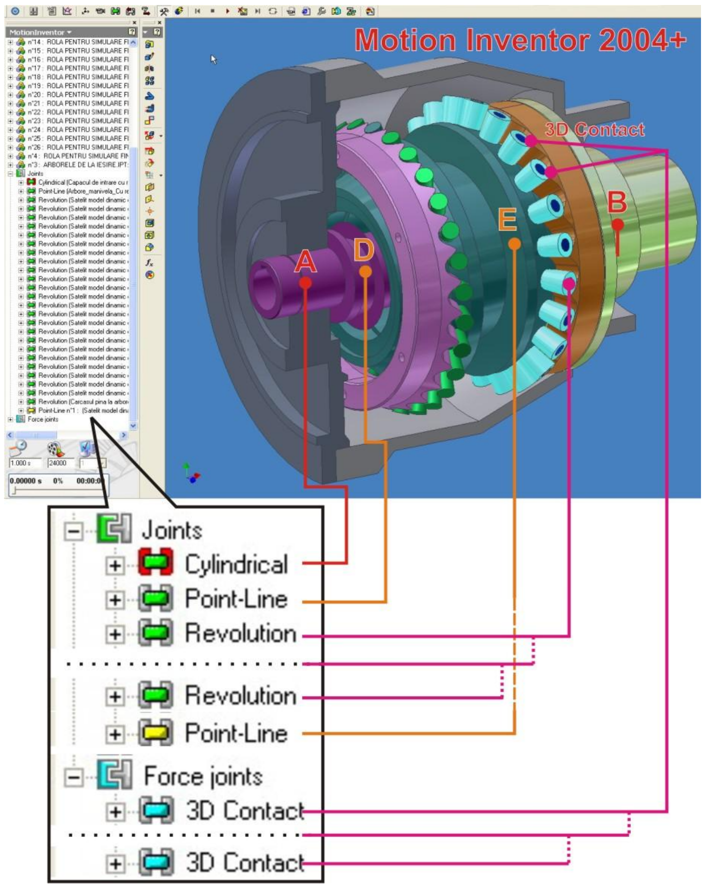

Fig. 5. Kinematics joints in the planetary precessional transmission.

3.2. Precessional transmission kinetostatic simulation

The simulation of model precessional gearings has been created in some stage. At the first stage has been executed the kinematic analysis, from definition of the following parameters: transfer ratio, absolute angular speed of the block satellite, relative angular speed of the block satellite, angular speed on the output shaft. The received results are shown on fig. 6. At the following stage has been executed the kinetostatics analysis with calculation and simulation of total loadings in gearings.

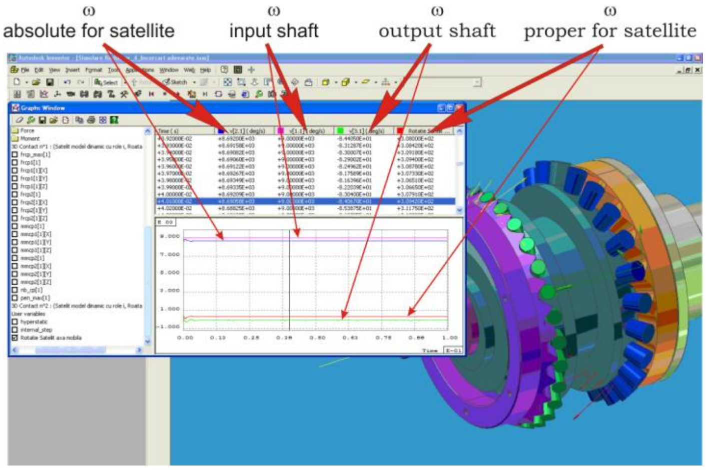

Fig. 6. Kinematic calculation of planetary precessional transmission.

An important advantage of the precessional planetary transmission consists in the insurance of absolute multiplicity (ε=100%) of the central wheel teeth and pinion simultaneous gearing. This advantage provides small dimensions and mass (reduced material consumption) and small cost of the final product.

The vertical line on the diagram (fig. 7) shows the precession phase reported symbolically to the gearing time of the pinion tooth with the central wheel tooth (as reference the time was taken, because as CAE input parameters the angular ω velocity was introduced). On the diagrams one can see the distribution of load among the teeth which corresponds to the respective precession phase (blue colour – distribution of load on the active profile of central wheel tooth, red colour – per one precession cycle). It is possible to state from the diagrams that despite the precession phase, the load among teeth is uniform.

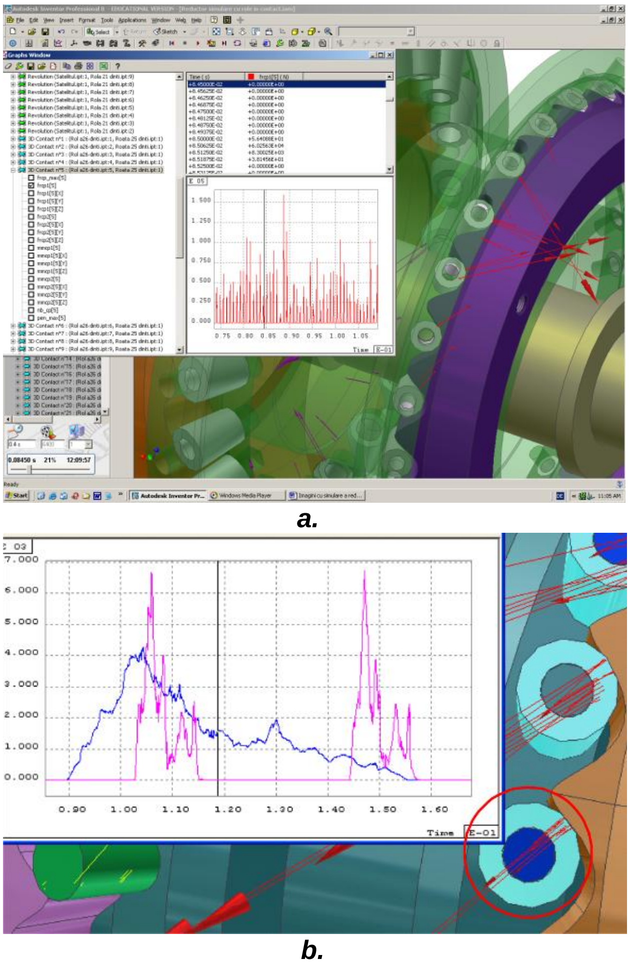

Fig. 7. Normal forces among the teeth and their distribution among simultaneously geared teeth

Due to load transmission from the input shaft to the output shaft by a big number of teeth couples geared simultaneously $\left(\varepsilon = {(Z_4 - 1)}/{2}\right)$ the normal forces at teeth contact are much smaller as in the classical transmissions with involute transmission. The load distribution chart of simultaneously geared teeth is shown in fig. 8.

10. Conclusions

Taking into account the fact that in precessional planetary gearings the $(Z_4 - 1)/2$ teeth couples transmit the load simultaneously we can conclude that the bearing capacity of the precessional gear is much bigger than that of the classical involute gear (in which only *5...7 %* of wheel teeth gear simultaneously).

The elaboration of the planetary precessional transmission dynamic sample (CAE) based on developed 3D model allows verification of kineto-static parameters previously defined.

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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