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Analysis of Cycloid Drive Dynamic Behavior

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Cycloid drives belong to the latest generation of planetary gear trains. Their dynamic behavior was considered in this paper. On the basis of well-know dynamic models of involute toothed gearing with external and internal toothing, a dynamic model of a single-stage cycloid drive was developed. The differential equations of system motion were written and numerically solved in the MATLAB – SIMULINK software. The system excitation force was acting in the contact of a cycloid gear tooth and a stationary central gear roller. The diagrams are obtained showing the resulting displacements, corresponding velocity and dynamic force in terms of time.

Key words: **planetary gear, cycloid drive, cycloid gear, dynamic behavior, dynamic forces.**

Introduction

CYCLOID drives have many very good operating characteristics, e.g. exceptional long and reliable working life, wide range of reduction ratios, high efficiency, high overload capacity, suitability for frequent start-stop and reversing duty, compact design, reliable operation under dynamic forces, minimal vibrations, low noise, low backlash, high shock load capability, etc. Thanks to the mentioned characteristics, cycloid drives have very wide applications: conveyor systems, presses, extruders, cranes, spinning machines, processing and automotive plants, mixers, food machinery, etc. Model of a single-stage cycloid drive is presented in Fig.1.

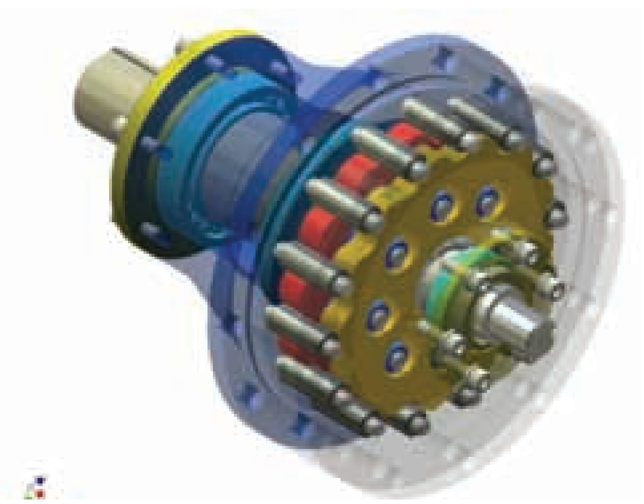

Figure 1. Model of a single-stage cycloid drive

The topological structure of a single-stage cycloid drive with its main elements is presented in Fig.2.

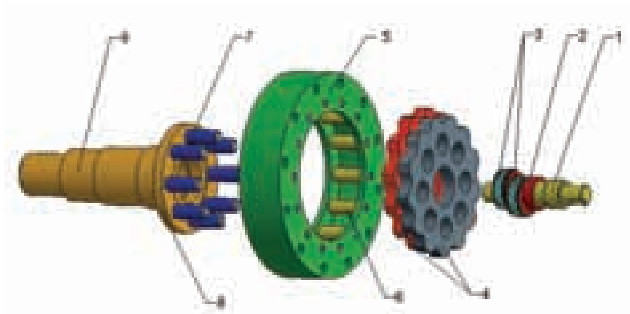

Figure 2. Topological structure of a single-stage cycloid drive (1-input shaft, 2-eccentric cam, 3-needle bearings, 4-cycloid gears, 5- central gear body, 6-central gear rollers, 7-output shaft rollers, 8- output shaft flange, 9-output shaft)

In the operation process of a cycloid drive, dynamic forces occur as well as in other planetary gear trains, [1], [2]. Depending on their character and origin, there are external and internal dynamic forces. External dynamics forces are a consequence of changes in parameters of power and motion at start drive and operating machine. Internal dynamics forces are result of each other interaction between elements in contact. Internal dynamics forces load cycloid drive elements in addition and cause vibrations [1]. On the other hand, vibrations influence the increase of internal dynamics forces. We can say that force is a cause and an implication of vibrations in the same time. The coupled teeth behave as one oscillatory system.

The main causes of internal dynamics forces at toothed gearings are [6], [7], [8]: changing of teeth's deformations in conjugate gear action and teeth shocks, deviations of basic geometric sizes of coupled gears, form deviations of the cycloid gear teeth profile, teeth's wear, etc. In cycloid drives, besides the above mentioned causes, there are also

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some other causes: errors that occur during the production of cycloid gear teeth and other elements of cycloid drive, unequal distribution of load at cycloid gear teeth as well as at central gear rollers and output shaft rollers, elastic deformations of the case and other elements, etc.

Dynamic model of a single stage cycloid drive

On the basis of a very careful analysis of dynamic models of involute toothed gearing with external toothing, as well as dynamic models of planetary gear trains, [9], [10], the dynamic model of a single - stage cycloid drive was developed in this paper and presented in Fig.3.

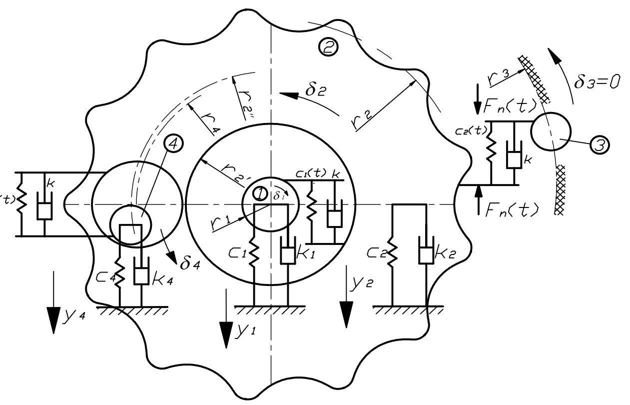

Figure 3. Dynamic model of a single – stage cycloid drive (1-input shaft with the eccentric cam, 2-cycloid gear, 3-central gear roller, 4-output shaft roller)

The cycloid drive elements are connected in their supports in the following way [3]:

- *Input shaft with eccentric cam*: elastic connection of the stiffness c_1 and a damper with the coefficient of damping k_1,
- *Cycloid gear:* elastic connection of the stiffness c_2 and a damper with the coefficient of damping k_2,
- *Central gear roller:* elastic connection of the stiffness c_3 and a damper with the coefficient of damping k_3,
- *Output shaft roller:* elastic connection of the stiffness c_4 and a damper with the coefficient of damping k_4.

The contacts between the corresponding elements of the cycloid drive are described in the following way:

- *Input shaft with eccentric cam – cycloid gear:* elastic connection of the variable stiffness c_1(t) and a damper with the coefficient of damping k ,
- *Cycloid gear – central gear roller:* elastic connection of the variable stiffness c_2(t) and a damper with the coefficient of damping k ,
- *Cycloid gear – output shaft roller:* elastic connection with the variable coefficient of stiffness $c_3(t)$ and a damper with the coefficient of damping k.

The excitation force F_n(t) occurs in the contact between the cycloid gear tooth and the central gear roller. The excitation force is calculated using the following expression [4], [5]:

$$F_n(t) = c \cdot w(t) \cdot b \quad (1)$$

where is:

c - connected teeth stiffness,

$w(t)$ - deformation of the cycloid gear tooth,

b - cycloid gear width.

Forasmuchas deformation's magnitude periodical time's function, the same case is for excitation force. The central gear is fixed and the system has six degrees of freedom as follows: three translations (y_1, y_2, y_3) and three rotations ($\delta_1, \delta_2, \delta_3$).

The geometric dimensions from Fig.3 are:

r_1 – external radius of the eccentric cam,

r_2 – radius of the pitch circle of the cycloid gear,

r_2' – radius of the central hole of the cycloid gear,

r_2'' – radius of the cycloid gear circle with the holes for the output shaft rollers,

r_3 – radius of the pitch circle of the central gear,

r_4 – radius of the output shaft flange with the holes for corresponding rollers.

The total displacement x in the contact between the cycloid gear tooth and the central gear roller (Fig.3) can be expressed as follows:

$$x = r_2\delta_2 - y_2 \quad (2)$$

The result of the excitation force action is a dynamic force which loads the cycloid gear teeth and the central gear rollers:

$$F_e = c \cdot x + b \cdot \dot{x} \quad (3)$$

This dynamic force depends on the total displacement and the adequate velocity.

The kinetic system energy is:

$$\begin{aligned} E_k = &\frac{1}{2} m_1 \dot{y}_1^2 + \frac{1}{2} J_1 \dot{\delta}_1^2 + \frac{1}{2} m_2 \dot{y}_2^2 + \frac{1}{2} J_2 \dot{\delta}_2^2 + \\ &+ \frac{1}{2} m_4 \dot{y}_4^2 + \frac{1}{2} J_4 \dot{\delta}_4^2 \end{aligned} \quad (4)$$

The potential system energy is:

$$\begin{aligned} E_p = &\frac{1}{2} c_1 y_1^2 + \frac{1}{2} c_2 y_2^2 + \frac{1}{2} c_4 y_4^2 + \\ &+ \frac{1}{2} c_1(t) \left[(y_2 - r_2'\delta_2) - (y_1 + r_1\delta_1) \right] + \\ &+ \frac{1}{2} c_2(t) \left[y_2 - r_2\delta_2 \right]^2 + \\ &+ \frac{1}{2} c_3(t) \left[(y_4 + r_4\delta_4) - (y_2 + r_2''\delta_2) \right]^2 \end{aligned} \quad (5)$$

The dissipation system function is:

$$\begin{aligned} \Phi = &\frac{1}{2} k_1 \dot{y}_1^2 + \frac{1}{2} k_2 \dot{y}_2^2 + \frac{1}{2} k_4 \dot{y}_4^2 + \\ &+ \frac{1}{2} k \left[\left(\dot{y}_2 - r_2'\dot{\delta}_2 \right) - \left(\dot{y}_1 + r_1\dot{\delta}_1 \right) \right]^2 + \\ &+ \frac{1}{2} k \left[\dot{y}_2 - r_2\dot{\delta}_2 \right]^2 + \\ &+ \frac{1}{2} k \left[\left(\dot{y}_4 + r_4\dot{\delta}_4 \right) - \left(\dot{y}_4 + r_2''\dot{\delta}_2 \right) \right]^2 \end{aligned} \quad (6)$$

The virtual work of the conservative forces is:

$$\delta \mathrm{A} = F_n(t) r_2 \delta\delta_2 \quad (7)$$

The conservative force is:

$$Q_{\delta_2} = F_n(t) r_2 \quad (8)$$

The signs m_1, m_2 and m_3 stand for the masses of correspondent elements from Fig.3, and J_1, J_2, and J_3 stand for a rectangular moment of inertia of the same elements.

The differential equations of the system motion are:

$$\begin{aligned} &J_1 \ddot{\delta}_1 + c_1(t) \left[(y_1 + r_1\delta_1) - (y_2 - r_2'\delta_2) \right] r_1 + \\ &+ k \left[\left(\dot{y}_1 + r_1\dot{\delta}_1 \right) - \left(\dot{y}_2 - r_2'\dot{\delta}_2 \right) \right] r_1 = 0 \end{aligned} \quad (9)$$

$$\begin{aligned}&m_1\ddot{y}_1 + c_1 y_1 + k_1\dot{y}_1 + \\ &+c_1(t)\left[\left(y_1 + r_1\delta_1\right) - \left(y_2 - r_2'\delta_2\right)\right] + \\ &+k\left[\left(\dot{y}_1 + r_1\dot{\delta}_1\right) - \left(\dot{y}_2 - r_2'\dot{\delta}_2\right)\right] = 0\end{aligned} \tag{10}$$

$$\begin{aligned}&J_2\ddot{\delta}_2 + c_1(t)\left[\left(y_1 + r_1\delta_1\right) - \left(y_2 - r_2'\delta_2\right)\right]r_2' + \\ &+c_2(t)\left[r_2\delta_2 - y_2\right]r_2 + \\ &+c_3(t)\left[\left(y_2 + r_2''\delta_2\right) - \left(y_4 + r_4\delta_4\right)\right]r_2'' + \\ &+k\left[\left(\dot{y}_1 + r_1\dot{\delta}_1\right) - \left(\dot{y}_2 - r_2'\dot{\delta}_2\right)\right]r_2' + \\ &+k\left[r_2\dot{\delta}_2 - \dot{y}_2\right]r_2 + \\ &+k\left[\left(\dot{y}_2 + r_2''\dot{\delta}_2\right) - \left(\dot{y}_2 + r_2\dot{\delta}_2\right)\right]r_2'' = F_n(t)r_2\end{aligned} \tag{11}$$

$$\begin{aligned}&m_2\ddot{y}_2 + c_1(t)\left[\left(y_1 + r_1\delta_1\right) - \left(y_2 - r_2'\delta_2\right)\right] + \\ &+c_2 y_2 + c_2(t)\left[r_2\delta_2 - y_2\right] + \\ &+c_3(t)\left[\left(y_2 + r_2''\delta_2\right) - \left(y_4 + r_4\delta_4\right)\right] + \\ &+k\left[\left(\dot{y}_1 + r_1\dot{\delta}_1\right) - \left(\dot{y}_2 - r_2'\dot{\delta}_2\right)\right] + k_2\dot{y}_2 + \\ &+k\left[r_2\dot{\delta}_2 - \dot{y}_2\right] + \\ &+k\left[\left(\dot{y}_2 + r_2''\dot{\delta}_2\right) - \left(\dot{y}_4 - r_4\dot{\delta}_4\right)\right] = 0\end{aligned} \tag{12}$$

$$\begin{aligned}&J_4\ddot{\delta}_4 - c_3(t)\left[\left(y_2 + r_2''\delta_2\right) - \left(y_4 + r_4\delta_4\right)\right]r_4 - \\ &-k\left[\left(\dot{y}_2 + r_2''\dot{\delta}_2\right) - \left(\dot{y}_4 + r_4\dot{\delta}_4\right)\right]r_4 = 0\end{aligned} \tag{13}$$

$$\begin{aligned}&m_4\ddot{y}_4 + c_4 y_4 + k_4\dot{y}_4 - \\ &-c_3(t)\left[\left(y_2 + r_2''\delta_2\right) - \left(y_4 + r_4\delta_4\right)\right] - \\ &-k\left[\left(\dot{y}_2 + r_2''\dot{\delta}_2\right) - \left(\dot{y}_4 + r_4\dot{\delta}_4\right)\right] = 0\end{aligned} \tag{14}$$

It is a system of the second order of linear parametric differential equations. In the represented dynamic model of the cycloid drive, the stiffnesses in the corresponding contacts and the dynamic force are variable time functions. However, in order to enable the analysis of influence of deviations of geometric dimensions, it is adopted that the mentioned stiffnesses are constant and equal to some average values, and the excitation force stays a variable time function.

$$\begin{aligned}c_1\left(t\right) &= c_{01} = const \\ c_2\left(t\right) &= c_{02} = const \\ c_3\left(t\right) &= c_{03} = const\end{aligned} \tag{15}$$

When the expressions (15) are used in equations (9)-(14), there is the following system of differential equations:

$$\begin{aligned}\ddot{\delta}_1 = &-\frac{c_{01}\cdot r_1}{J_1}\cdot y_1 - \frac{c_{01}\cdot r_1^2}{J_1}\cdot\delta_1 + \frac{c_{01}\cdot r_1}{J_1}\cdot y_2 - \\ &-\frac{c_{01}\cdot r_1\cdot r_2'}{J_1}\cdot\delta_2 - \frac{k\cdot r_1}{J_1}\cdot\dot{y}_1 - \frac{k\cdot r_1^2}{J_1}\cdot\dot{\delta}_1 + \\ &+\frac{k\cdot r_1}{J_1}\cdot\dot{y}_2 - \frac{k\cdot r_1\cdot r_2'}{J_1}\cdot\dot{\delta}_2\end{aligned} \tag{16}$$

$$\begin{aligned}\ddot{y}_1 = &-\frac{c_1 + c_{01}}{m_1}\cdot y_1 - \frac{k_1 + k}{m_1}\cdot\dot{y}_1 - \\ &\frac{c_{01}\cdot r_1}{m_1}\cdot\delta_1 + \frac{c_{01}}{m_1}\cdot y_2 - \frac{c_{01}\cdot r_2'}{m_2}\cdot\delta_2 - \\ &-\frac{k\cdot r_1}{m_1}\cdot\dot{\delta}_1 + \frac{k}{m_1}\cdot\dot{y}_2 - \frac{k\cdot r_2'}{m_1}\cdot\dot{\delta}_2\end{aligned} \tag{17}$$

$$\begin{aligned}\ddot{\delta}_2 = &\frac{r_2}{J_2}\cdot F_n(t) - \frac{c_{01}\cdot r_2''}{J_2}\cdot y_1 - \frac{c_{01}\cdot r_2'\cdot r_1}{J_2}\cdot\delta_1 + \\ &+\frac{c_{01}\cdot r_2' + c_{02}\cdot r_2 - c_{03}\cdot r_2''}{J_2}\cdot y_2 - \\ &-\frac{c_{01}\cdot r_2'^2 + c_{02}\cdot r_2^2 + c_{03}\cdot r_2''^2}{J_2}\cdot\delta_2 + \frac{c_{03}\cdot r_2''}{J_2}\cdot y_4 + \\ &+\frac{c_{03}\cdot r''\cdot r_4}{J_2}\cdot\delta_4 - \frac{k\cdot r_2}{J_2}\cdot\dot{y}_1 - \frac{k\cdot r_2'\cdot r_1}{J_2}\cdot\dot{\delta}_1 + \\ &+k\cdot\frac{r_2' + r_2 - r_2''}{J_2}\cdot\dot{y}_2 - k\cdot\frac{r_2'^2 + r_2^2 + r_2''^2}{J_2}\cdot\dot{\delta}_2 + \\ &+\frac{k\cdot r_2''}{J_2}\cdot\dot{y}_4 + \frac{k\cdot r_2''\cdot r_4}{J_2}\cdot\dot{\delta}_4\end{aligned} \tag{18}$$

$$\begin{aligned}\ddot{y}_2 = &-\frac{c_{01}}{m_2}\cdot y_1 - \frac{c_{01}\cdot r_1}{m_2}\cdot\delta_1 - \\ &-\frac{c_2 - c_{01} - c_{02} + c_{03}}{m_2}\cdot y_2 - \\ &-\frac{c_{01}\cdot r_2'' + c_{02}\cdot r_2 + c_{03}\cdot r_2''}{m_2}\cdot\delta_2 + \frac{c_{03}}{m_2}\cdot y_4 + \\ &+\frac{c_{03}\cdot r_4}{m_2}\cdot\delta_4 - \frac{k}{m_2}\cdot\dot{y}_1 - \frac{k\cdot r_1}{m_2}\cdot\dot{\delta}_1 - \\ &-\frac{k_2 - k}{m_2}\cdot\dot{y}_2 - k\cdot\frac{r_2' + r_2 + r_2''}{m_2}\cdot\dot{\delta}_2 + \\ &+\frac{k}{m_2}\cdot\dot{y}_4 - \frac{k\cdot r_4}{m_2}\cdot\dot{\delta}_4\end{aligned} \tag{19}$$

$$\begin{aligned}\ddot{\delta}_4 = &\frac{c_{03}\cdot r_4}{J_4}\cdot y_4 + \frac{c_{03}\cdot r_4\cdot r_2''}{J_4}\cdot\delta_2 - \frac{c_{03}\cdot r_4}{J_4}\cdot y_4 - \\ &-\frac{c_{03}\cdot r_4^2}{J_4}\cdot\delta_4 + \frac{k\cdot r_4}{J_4}\cdot\dot{y}_2 + \frac{k\cdot r_4\cdot r_2''}{J_4}\cdot\dot{\delta}_2 - \\ &-\frac{k\cdot r_4}{J_4}\cdot\dot{y}_4 - \frac{k\cdot r_4^2}{J_4}\cdot\dot{\delta}_4\end{aligned} \tag{20}$$

$$\begin{aligned}\ddot{y}_4 = &-\frac{c_4 + c_{03}}{m_4}\cdot y_4 - \frac{k_4 + k}{m_4}\cdot\dot{y}_4 + \frac{c_{03}}{m_4}\cdot y_2 + \\ &+\frac{c_{03}\cdot r_2''}{m_4}\cdot\delta_2 - \frac{c_{03}\cdot r_4}{m_4}\cdot\delta_4 + \frac{k}{m_4}\cdot\dot{y}_2 + \\ &+\frac{k\cdot r_2''}{m_4}\cdot\dot{\delta}_2 - \frac{k\cdot r_4}{m_4}\cdot\dot{\delta}_4\end{aligned} \tag{21}$$

Solutions of differential equations for a particular single–stage cycloid drive

The system of differential equations (16-21) was solved by a numerical way using the MATLAB – SIMULINK software for a particular single-stage cycloid drive with the following characteristics:

- Electromotor power: $P_{EM} = 0.25$ kW ,
- Electromotor rotations per minute: $n_{EM} = 1390\ \text{min}^{-1}$,
- Reduction ratio of the single stage cycloid drive: $u = 11$

The input system parameter is the excitation force $F_n(t)$, and the output parameters are: total displacement x in the contact between a cycloid gear tooth and a central gear roller (location where the excitation force $F_n(t)$ occurs), the adequate velocity ($\dot{x}$) and the dynamic force (F_e). The equations are solved by the iterative method with assumption that on the cycloid gear tooth in the first iteration there is only the excitation force $F_n(t)$ acting. All three output system parameters as well as the excitation force are time functions. The values of the coefficients of

damping as well as adequate stiffnesses are adopted on the basis of recommendation from literature [9], [10]:
k = 5000 Ns/m, k_1 = 2500 Ns/m, k_2 = 150 Ns/m, k_3 = 2000 Ns/m, k_4 = 1000 Ns/m, $c_1(t) = c_{01} = 1.3 \cdot 10^8\,\mathrm{N/m}$, $c_2(t) = c_{02} = 2 \cdot 10^7\,\mathrm{N/m}$, $c_3(t) = c_{03} = 1.5 \cdot 10^8\,\mathrm{N/m}$, $c_1 = 2 \cdot 10^9\,\mathrm{N/m}$, $c_2 = 1.6 \cdot 10^8\,\mathrm{N/m}$, $c_3 = 2 \cdot 10^9\,\mathrm{N/m}$, $c_4 = 1{,}6 \cdot 10^8\,\mathrm{N/m}$.

The masses of elements, the rectangular moments of inertia and the adequate radii have the following values: m_1 = 0.17 kg, m_2 = 0.74 kg, m_4 = 0.7 kg, J_1 = 26 kgmm^2, J_2 =1422.3 kgmm^2, J_4 = 762.5 kgmm^2, r_1 = 17.5 mm, r_2 = 62 mm, $r_2' = 20\,\mathrm{mm}$, $r_2'' = 39\,\mathrm{mm}$, r_4 = 39 mm.

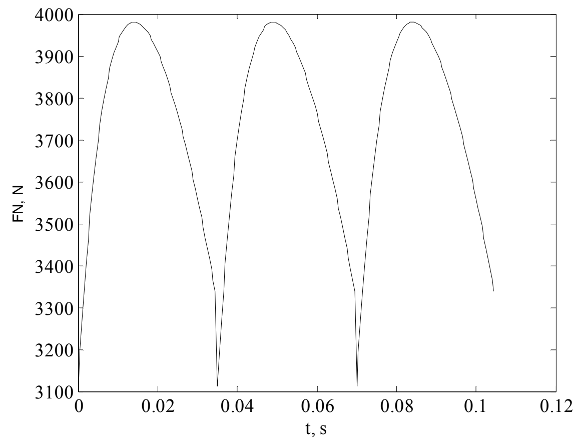

Figure 4. Diagram of the system excitation force F$_n$(t)

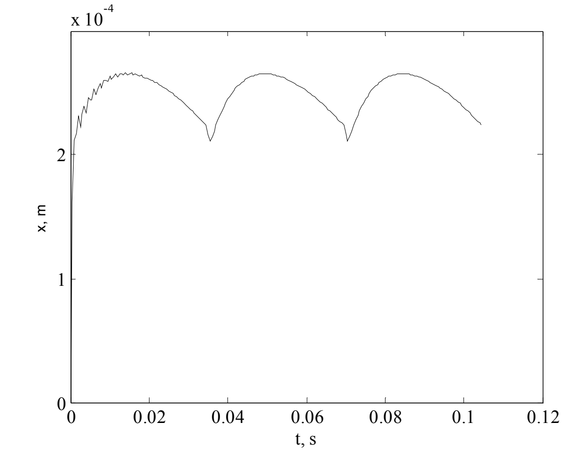

Figure 5. Diagram of the total displacement x

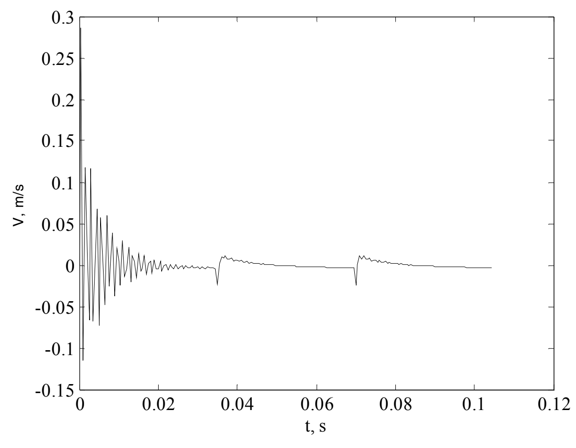

Figure 6. Diagram of the velocity $\dot{x}$

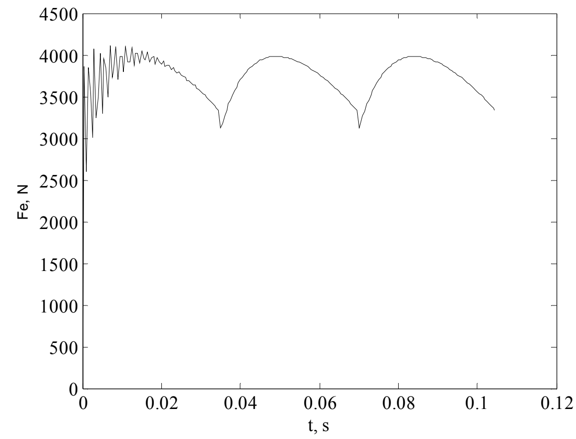

Figure 7. Diagram of the dynamic force F_e

The results presented in this paper concern a single connection (one cycloid gear tooth and one central gear roller are only in contact). This case is the most critical because of the highest values of the dynamic force. The results are represented in Figures 4, 5, 6 and 7.

Vibrations in this system are forced with damping. Analyzing the obtained results in Figures 4, 5, 6 and 7, it is easy to see that vibrations with damping in time disappear very quickly and only forced vibrations remain. This is applied for a case when the values for the coefficients of damping and adequate stiffnesses are selected from a registered range. On the other hand, vibrations with damping exist in the whole period of the contact between the cycloid gear tooth and the central gear roller with very expressed amplitudes.

Conclusions

The dynamic behavior of the cycloid drive is analyzed in this paper. For that purpose, on the basis of well-know dynamic models of involute toothed gearing with external and internal toothing, a dynamic model of a single-stage cycloid drive was developed. Whereas the system has six degrees of freedom (three translations (y_1, y_2, y_3) and three rotations ($\delta_1, \delta_2, \delta_3$)), the defined dynamic model of the cycloid drive is represented with six differential equations of motion. The equations are solved in the MATLAB – SIMULINK software by the iterative method for one particular single – stage cycloid drive.

Forasmuchas the system excitation force $F_n(t)$ is a periodical time function, the same is true for the total displacement, the adequate velocity and the dynamic force. The amplitudes of vibrations with damping are minimized very quickly, so that the changing of the dynamic force is very similar to the changing of the excitation force.

The biggest influence on dynamic operating of the cycloid drive comes from the coefficient of the damping during the contact between the cycloid gear tooth and the central gear roller as well as from it stiffness.

The derived magnitudes of the dynamic force are the starting parameters for analyzing the stress and strain state of the cycloid drive elements under dynamic operating conditions defined in this paper.

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Analiza dinamičkog ponašanja cikloreduktora

Cikloreduktori pripadaju najnovijoj generaciji planetarnih prenosnika snage.U ovom radu se razmatra njihovo dinamičko ponašanje. Na osnovu poznatih dinamičkih modela evolventnih zupčastih prenosnika sa spoljašnjim i unutrašnjim ozubljenjem, razvijen je dinamički model jednostepenog cikloreduktora. Napisane su diferencijalne jednačine kretanja sistema. Iste su rešene numeričkim putem u programskom paketu MATLAB – SIMULINK. Pobudna sila sistema deluje u kontaktu zupca ciklozupčanika i valjka centralnog zupčanika. Kao rezultat, dobijeni su dijagrami rezultujućeg pomeranja, odgovrajuće brzine i dinamičke sile u funkciji vremena.

Ključne reči: **planetarni prenosnik, cikloreduktor, ciklozupčanik, dinamičko ponašanje, dinamičke sile.**

Анализ динамического поведения циклоредуктора

Циклоредукторы принадлежат самой новой генерации планарных передаточных механизмов мощности. В настоящей работе рассматривается их динамическое поведение. На основе известных динамических моделей эвольвентных зубчатых передаточных механизмов со внешним и со внутренним зубчатым зацеплением, развита динамическая модель одноступенчатого циклоредуктора. Здесь приведены и дифференциальные уравнения движения системы, которые решены цифровым путём в программном обеспечении МАТЛАБ-СИМУЛИНК. Возбуждение системы действует в контакте зубца циклошестерни и ролика центральной шестерни, в результате чего получены диаграммы результирующего перемещения, соответствующей скорости и динамической силы в функции времени.

Ключевые слова: Планарный передаточный механизм, циклоредуктор, циклошестерня, динамическое поведение, динамические силы.

Analyse du comportement dynamique chez le réducteur cycloïdal

Les réducteurs cycloïdaux appartiennent à la plus récente génération des engrenages planétaires de la force. Dans ce papier on considère leur comportement dynamique. On a développé le modèle dynamique du réducteur cycloïdal monoétagé à la base des modèles dynamiques connus des engrenages à développante à denture extérieure et intérieure. On a écrit les équations différentielles du mouvement du système. Elles sont résolues par la façon numérique dans le progiciel MATLAB-SIMULINK. La force d'excitation du système agit en contact entre la dent d'engrenage cycloïdal et le rouleau de l'engrenage central. On a obtenu, comme résultat, les diagrammes des déplacements résultants, des vitesses correspondantes et les forces dynamiques en fonction du temps.

Mots clés: **engrenage planétaire, réducteur cycloïdal, engrenage cycloïdal, comportement dynamique forces dynamiques.**