INFLUENCE OF THE FRICTION ON THE CYCLOIDAL SPEED REDUCER EFFICIENCY

M. BLAGOJEVIC[a]*, M. KOCIC[b], N. MARJANOVIC[a], B. STOJANOVIC[a], Z. DORDEVIC[a], L. IVANOVIC[a], V. MARJANOVIC[a]

[a] *Faculty of Mechanical Engineering, University of Kragujevac, 6 Sestre Janjic Street, 34 000 Kragujevac, Serbia*
E-mail: *mirkob@kg.ac.rs*
[b] *Institute Gosa, 35Milana Rakica Street, 11 000 Belgrade, Serbia*
E-mail: *milorad.kocic@institutgosa.rs*

ABSTRACT

Cycloidal speed reducers belong to the new generation of mechanical gears. Cycloidal speed reducers are increasingly used in modern engineering due to their good performances. Friction exists at all points of contact between the moving elements of cycloidal speed reducers, as one of the main causes of power losses in power transmission. In this paper, the friction of the contact of cycloid disc and housing rollers has been taken into account, while the friction at the contact of cycloid disc and output rollers, and at the contact of cycloid disc and central roller bearings are neglected. The effect of friction on the distribution of contact force, the friction torque value is analysed as well as efficiency. This paper presents 2 analytical models for calculating the values of contact force and friction torque. The 2nd model opposed to the 1st, taking into account the phenomenon of friction. The results showed that friction significantly affects the distribution of normal force. Increase of the friction produces the contact forces and friction torque increase, also, while efficiency is decreasing.

Keywords: cycloidal speed reducer, cycloid disc, friction, force analysis.

AIMS AND BACKGROUND

Cycloidal speed reducers have a very wide area of application: at conveyors, food machinery, robots, mixers, recycling machines, automotive plants, steel mills, etc. Their main features are high efficiency, compact design, quite and reliable

* For correspondence.

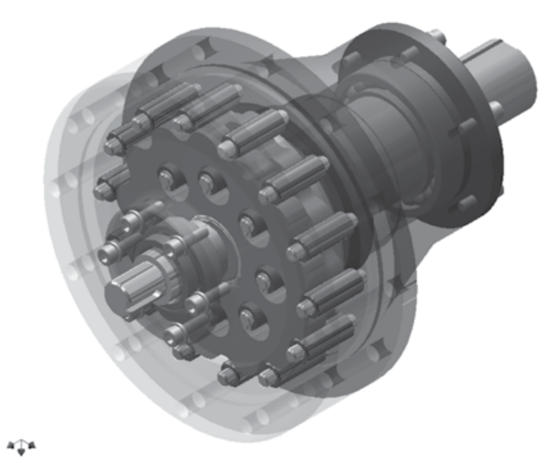


Fig. 1. Single-stage cycloidal speed reducer

operation, high gear ratio, minimum vibration, low noise, low backlash, etc. Single-stage cycloidal speed reducer is shown in Fig. 1.

An exploded view of the one-stage cycloidal speed reducer of traditional design is shown in Fig. 2, where 2 cycloid discs, relatively turned to each other for an angle of 180°, are used for each stage.

The working principle is as follows: eccentric (*2*) is rotating with the same number of revolutions and in the same direction as the input shaft (*1*). Cycloid discs (*4*) are situated on the eccentric through bearings (*3*). Cycloid discs are conjugated with ring gear rollers (*6*) located within the body of the ring gear (*5*). The result of this conjugation is the complex motion of the cycloid discs consisting of a rotation that originates from the eccentric, in the direction of the input shaft rotation and rotation of cycloid discs around their own axes in the opposite direction. Output rollers (*7*) are tightly connected to the output shaft (*9*) by the carrier (*8*) and go through the circular openings of the cycloid discs and transfer their motion to the output shaft.

M. Lehmann[1] described in details the procedure for force distribution at cycloidal speed reducer for theoretical case (when friction does not exist). Malhotra[2]

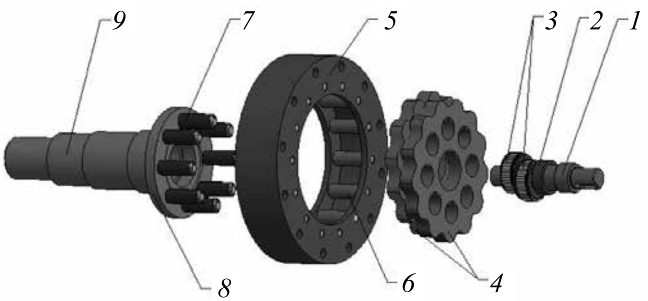


Fig. 2. Exploded view of the single-stage cycloidal speed reducer

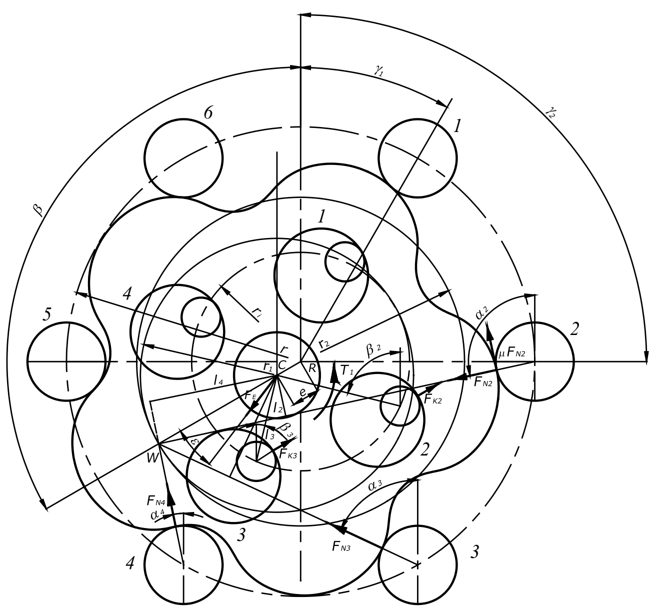
also defined some simplified model for force analysis but without friction. Lixing and Chmurawa[3,4] presented the distribution of loads at cycloidal speed reducer with modified profile of the cycloid disc tooth. Chen et al.[5], Hwang et al.[6] and Chen et al.[7] defined a complete geometric and kinematic analysis of cycloid disc tooth profile. Gorla, Davoli et al.[8] and other give out the results of theoretical and experimental analysis of cycloidal speed reducer efficiency. In Refs 9, 10 and 11 stress and strain state at cycloidal speed reducer elements are presented. Meng et al.[12] defined a mathematical model of transmission performance of 2K-H pin-cycloid planetary mechanism including the friction. Blagojevic et al.[13] presented a new concept of 2-stage cycloidal speed reducer.

The phenomena of friction and wear have a very significant impact on the efficiency and service life of machine elements (Refs 14–17). This paper primarily describes the influence of the friction on the load distribution in cycloidal speed reducer, and then analyses the scope and influence of the friction on the friction torque, the distribution of contact force and the efficiency of the cycloidal speed reducer.

FORCE DISTRIBUTION AT CYCLOIDAL SPEED REDUCER WITHOUT FRICTION

Cycloid disc is the most important element of the cycloidal speed reducer due to its complex geometry, complex stress and strain state. In the 1st step it is necessary to define forces which act upon it. Cycloid disc with contact elements (housing rollers and output rollers) is shown in Fig. 3.

Fig. 3. Cycloid disc in contact with housing rollers and output rollers



Loads on cycloid disc are: F_E – bearing reaction; F_{Ni} – force between housing roller i and cycloid disc; F_{Kj} – force between output roller j and cycloid disc; T_1 – input torque.

The following equations can be expressed, based on Fig. 3:

$$T_1 = F_E\, e \cos(\beta + \varepsilon) \tag{1}$$

$$T_1 = \frac{r_i}{z} \sum_{j=1}^{q} F_{Kj} \sin(\beta_j + \beta) \tag{2}$$

$$-\sum_{i=1}^{p} F_{Ni} \sin\alpha_i + \sum_{j=1}^{q} F_{Kj} \sin\beta - F_E \sin(\beta + \varepsilon) = 0 \tag{3}$$

$$-\sum_{i=1}^{p} F_{Ni} \cos\alpha_i - \sum_{j=1}^{q} F_{Kj} \cos\beta - F_E \cos(\beta + \varepsilon) = 0 \tag{4}$$

$$\sum_{i=1}^{p} F_{Ni}\, l_i - \sum_{j=1}^{q} F_{Kj}\, r_i \sin(\beta_j + \beta) = 0 \tag{5}$$

where e is eccentricity; β – swivel angle of the input shaft; ε – angle between the force and eccentricity direction; r_i – radius of output rollers pitch circle; z – number of teeth of cycloid disk (gear ratio of the cycloidal speed reducer); β_j – angular position of the output roller – j; α_i – angle which force F_{Ni} makes with vertical; l_i – lever arm of force F_{Ni}; p – number of the housing rollers that carry the load; q – number of the output rollers that carry the load.

The values of α_i and l_i are calculated according to Fig. 3, based on the following expressions:

$$\alpha_i = \mathrm{arctg} \frac{\sin\beta + \dfrac{r}{r_2}\sin\gamma_i}{\cos\beta - \dfrac{r}{r_2}\cos\gamma_i} \tag{6}$$

$$l_i = r_1 \sin(\alpha_i - \beta). \tag{7}$$

Angle γ_i (angular position of the housing rollers) is calculated based on the following expression:

$$\gamma_i = \frac{360(2i - 1)}{2(z + 1)} \tag{8}$$

where r is radius of housing rollers pitch circle; r_1 – base circle radius of the cycloid disc; r_2 – base circle radius of the housing rollers.

Forces F_{Ni} and F_{Kj} are proportional to their respective distances from the centre of rotation:

$$\frac{F_{Ni}}{l_i} = \text{const.} \tag{9}$$

$$\frac{F_{Kj}}{\sin(\beta_j + \beta)} = \text{const.} \tag{10}$$

In case when all the power losses in cycloidal speed reducer are neglected, drive torque of one cycloid disk T_1 is calculated on the basis of the expression:

$$T_1 = \frac{T_{EM}}{2} \tag{11}$$

where T_{EM} is the input torque.

The value of torque to the real central disc T_2 is determined as follows:

$$T_2 = T_1 z_2 \tag{12}$$

where z_2 is the number of housing rollers.

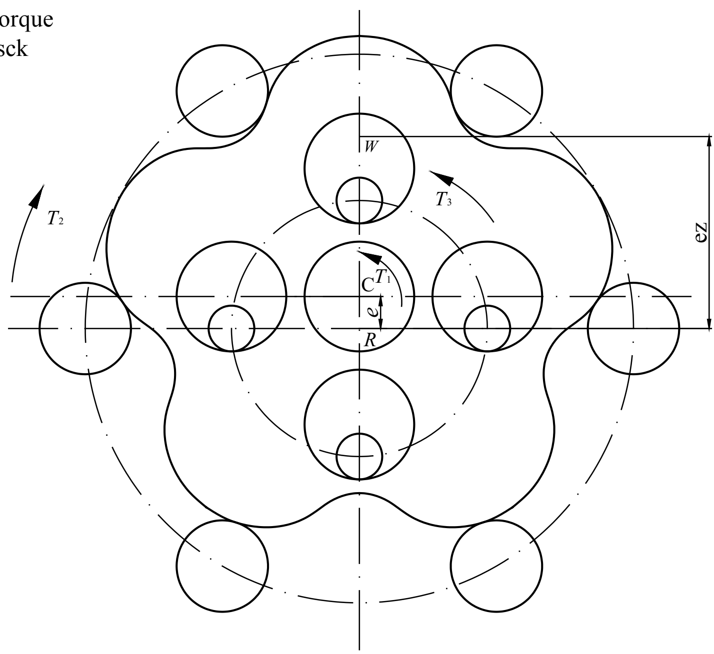
Torque balance equation which operates on cycloid disk has the following form (Fig. 4):

$$T_1 - T_2 + T_3 = 0 \tag{13}$$

where T_3 is the output torque cycloid disk:

$$T_3 = T_2 - T_1 \tag{14}$$

Fig. 4. Distribution of torque that acting on cycloid disck



FORCE DISTRIBUTION AT CYCLOIDAL SPEED REDUCER WITH FRICTION

The influence of the friction on the load distribution at real gear drives is very important. It is the same for the cycloidal speed reducer. There are 3 main areas where friction exists:

- in the contact of cycloid disc and central bearing rollers,
- in the contact of cycloid disc and output rollers, and
- in the contact of cycloid disc and stationary ring gear rollers.

Model with friction in contact of cycloid disc and stationary ring gear rollers is analysed in this paper. Friction in other areas is neglected.

Cycloid disc in contact with output rollers and stationary ring gear rollers (housing rollers) in the presence of friction are presented in Fig. 5.

The following equations can be expressed, based on Fig. 5:

$$-\sum_{i=1}^{p} F_{Ni}\sin\alpha_i + \sum_{j=1}^{q} F_{Kj}\sin\beta - F_E\sin(\beta+\varepsilon) - \mu\sum_{i=1}^{p} F_{Ni}\cos\alpha_i = 0 \tag{15}$$

$$-\sum_{i=1}^{p} F_{Ni}\cos\alpha_i - \sum_{j=1}^{q} F_{Kj}\cos\beta - F_E\cos(\beta+\varepsilon) + \mu\sum_{i=1}^{p} F_{Ni}\sin\alpha_i = 0 \tag{16}$$

$$\sum_{i=1}^{p} F_{Ni}\, l_i - \sum_{j=1}^{q} F_{Kj}\, r_i \sin(\beta_j+\beta) - \mu\sum_{i=1}^{p} F_{Ni}\, r_i \cos(\alpha_i-\beta) = 0 \tag{17}$$

where μ is the coefficient of friction.

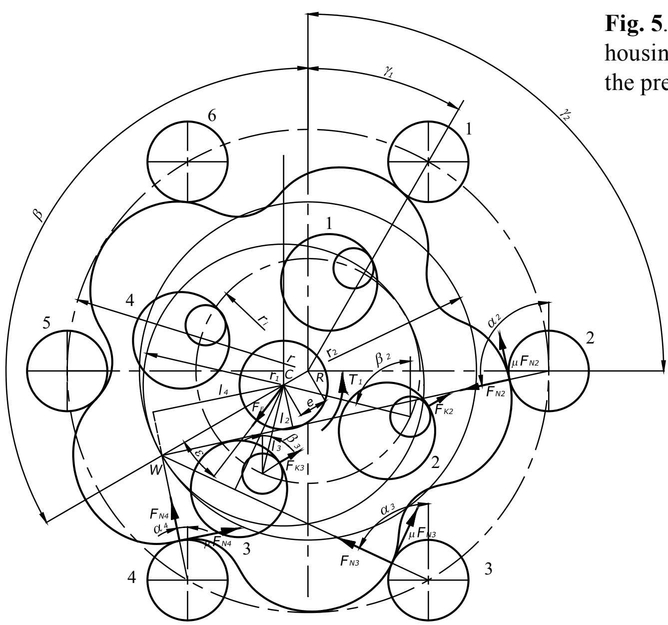
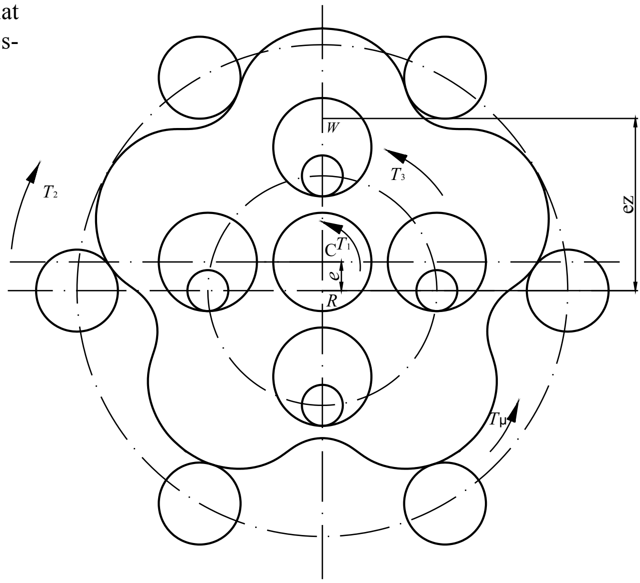


Fig. 5. Cycloid disc in contact with housing rollers and output rollers in the presence of friction

Fig. 6. Distribution of torque that acts on cycloid disk in the presence of friction



The output torque on cycloid disc can be expressed according to the following expression (Fig. 6):

$$T_3 = T_2 - T_1 - T_\mu \tag{18}$$

Friction torque T_μ can be expressed according to the following expression:

$$T_\mu = \mu \sum_{i=1}^{p} F_{Ni} r_i \cos(\alpha_i - \beta) \tag{19}$$

ANALYSIS OF THE INFLUENCE OF THE FRICTION FORCE ON THE LOAD DISTRIBUTION FOR REAL CYCLOIDAL SPEED REDUCER

Analysis of the influence of the friction force on load distribution was done for the real one-stage cycloidal speed reducer with the following characteristics:

- Input power: P_{EM} = 1.1 kW,
- Input number of revolutions: n_{EM} = 1410 min^{-1},
- Gear ratio (number of cycloid disc teeth): $u = 5$ ($z_1 = 5$).

Calculation of the normal forces that occur in contact of cycloidal disc and housing roller F_{Ni} is done for the case whith 3 pairs of teeth (rollers), based on the equations presented in this paper. Depending on the coefficient of friction, the above calculation has been done for 4 cases: when there is no friction and for μ = 0.05; μ = 0.1 and μ = 0.2. The values of the calculated contact forces are given in Table 1.

Table 1. Values of the contact forces as the function of the coefficient of friction

	F_{N1} (N)	F_{N2} (N)	F_{N3} (N)
Without friction	16.1	44.7	50.6
$\mu = 0.05$	100.4	270.6	313.5
$\mu = 0.1$	103.8	279.8	324.1
$\mu = 0.2$	111.4	300.2	347.2

Table 2. Values of friction torque and degree of utilisation cycloidal speed reducer as the function of the coefficient of friction

	T_μ (N mm)	η (–)
Without friction	0	1
$\mu = 0.05$	611.8	0.87
$\mu = 0.1$	1265.2	0.85
$\mu = 0.2$	2714.7	0.77

The values of the torque and the efficiency as a function of the coefficient of friction are given in Table 2.

It is important to note that only the friction in contact of cycloid disc and housing rollers is taken into account, while the friction at all other locations is neglected in calculating the values of contact force and frictional torque.

Figures 7, 8 and 9 show the dependence of the contact force, friction torque and efficiency on the coefficient of friction, respectively .

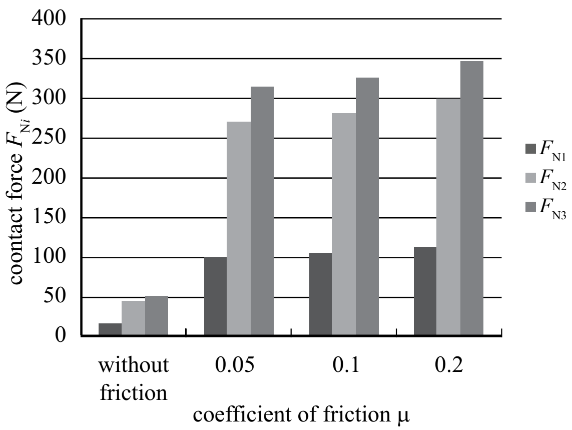


Fig. 7. Dependence of contact force on the coefficient of friction

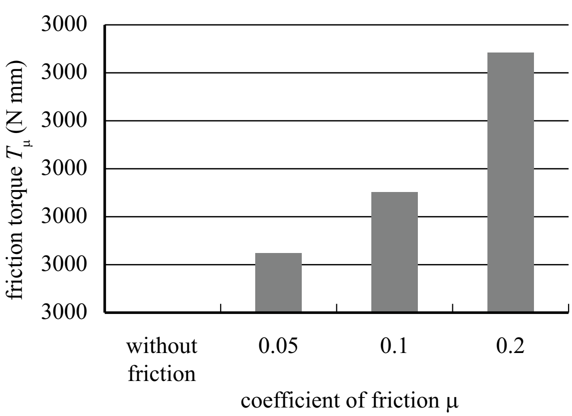
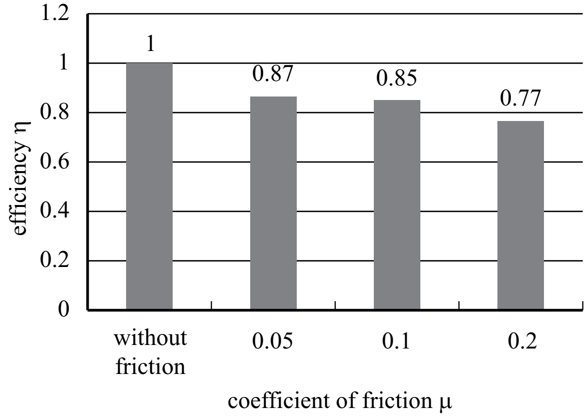


Fig. 8. Dependence of the friction torque on the coefficient of friction

Fig. 9. Dependence of the efficiency on the coefficient of friction of cycloidal speed reducer



By analysing the values of the contact force, given in Table 1 and Fig. 7, it is easy to see that the phenomenon of friction greatly affects the distribution of these forces. The most favourable case is certainly a theoretical coupling case, i.e. the case when the friction between the teeth and rollers of the stationary central gear of the cycloidal speed reducer is ignored. By increasing the coefficient of friction, the contact force recorded a slight growth. A favourable circumstance is certainly the fact that in cycloidal speed reducer sliding friction has mostly been replaced by rolling friction, due to the good coupling conditions and the tooth geometry of the cycloidal disc. The highest value of the contact force is for the case when coefficient of friction is $\mu = 0.2$ ($F_{N3} = 347.2$ N).

As for the friction torque, in case when there is no friction, its value is zero. With friction increase, also the value of friction torque increases, reaching its maximum of $T_\mu = 2714.7$ N mm with maximum value of the coefficient friction of $\mu = 0.2$ (Table 2 and Fig. 8).

The efficiency, as the ratio between output and input power at cycloidal speed reducer, also depends on the presence of the friction in the gear. In the case when friction is neglected in all locations, the value of efficiency is equal to one. This is, of course, only a theoretical case and does not correspond to the real conditions. Table 2 and Fig. 9 show the dependence of the efficiency on the coefficient of friction. The value of the efficiency ranges from $\eta = 1.0$ (without friction) to $\eta = 0.77$ ($\mu = 0.2$).

CONCLUSIONS

Regardless of all advantages that cycloidal gearing has, in terms of its geometric and kinematic properties, the appearance of friction has a very significant impact on the core strength parameter of the cycloidal speed reducer. As a part of this work, the 2 analyitical models are defined for calculation of the values of contact force and friction torque in efficiency of cycloidal speed reducer, as follows:

- Theoretical model (without friction),
- A model that includes frictional phenomenon.

The developed model takes into account only the friction in contact of the cycloid disc and housing rollers, while occurrence of friction in other locations is neglected. Testing of the developed models was performed for calculation of the value of contact force, friction torque and efficiency of a particular cycloidal speed reducer. With increasing the coefficient of friction, contact forces and friction torque are increased, too. However, the friction torque reduces efficiency. Increase in value of the contact forces, also is adversely affected by the stress-deformation state of the elements of the cycloidal speed reducer.

The obtained values of efficiency correspond well to the values from the literature[12], and to the values from manufacturer catalogues, the world most famous cycloidal speed reducer, the Japanese company SUMITOMO[18].

In further researches, in this area, it would be good to develop a more comprehensive mathematical model for calculating the values of the contact forces in contact of the cycloidal disk and housing rollers that would take into account the friction that occurs in the contact of the cycloidal disk and central bearing rollers, and friction at the cycloidal disc between the output rollers.

REFERENCES

1. M. LEHMANN: Calculation and Measurement of Forces Acting on Cycloidal Speed Reducer. Ph.D. Thesis, Technical University Munich, Germany, 1976 (in German).
2. S. K. MALHOTRA, M. A. PARAMESWARAN: Analysis of a Cycloidal Speed Reducer. Mechanism and Machine Theory, **18** (6), 491 (1983).
3. L. LIXING, L. XIN, H. WEIDONG, Q. YUANMEI: Profile Modification and Accurate Force Analysis on Cycloid Drive. In: 4th World Congress on Gearing and Power Transmission, Paris, France, Vol. 3, 1999, 1141–1146.
4. M. CHMURAWA, A. LOKIEC: Distribution of Loads in Cycloidal Planetary Gear (CYCLO) Including Modification of Equidistant. In: 16th Europian ADAMS User Conference, Berchtesgaden, Germany, 2001.
5. B. K. CHEN, T. T. FANG, C. Y. LI, S. Y. WANG: Gear Geometry of Cycloid Drives. Science in China Series E: Technological Sciences, **51** (5), 598 (2008).
6. Y. W. HWANG, C. F. HSIEH: Geometry Design and Analysis for Trochoidal-type Speed Reducers with Conjugate Envelopes. Transactions of the CSME/de la SCGM, **30** (2), 261 (2006).
7. C. CHEN, X. ZHANG, J. ANGELES: Kinematic and Geometric Analysis of a Pure-rolling Epicyclic Train. J. of Mechanical Design (ASME), **129**, 852 (2007).
8. C. GORLA, P. DAVOLI, F. ROSA, C. LONGONI, F. CHIOZZI, A. SAMARANI: Theoretical and Experimental Analysis of a Cycloidal Speed Reducer. J. of Mechanical Design (ASME), **130**, 112604-1 (2008).

9. M. BLAGOJEVIC, N. MARJANOVIC: The Force Distribution on Two Stage Cyclo Speed Reducer with New Concept. In: 9th International Conference on Accomplishements in Electrical and Mechanical Engineering and Information Technology, Banja Luka, 2007, 51–56.
10. M. BLAGOJEVIC: Stress and Strain State of Cyclo Speed Reducer's Elements under Dynamic Loads. Ph.D. Thesis, Faculty of Mechanical Engineering, Kragujevac, Serbia, 2008.
11. M. BLAGOJEVIC, V. NIKOLIC, N. MARJANOVIC, Lj. VELJOVIC: Analysis of Cycloid Drive Dynamic Behavior. Scientific Technical Review, **LIX** (1), 52 (2009).
12. Y. MENG, C. WU, L. LING: Methematical Modelling of the Transmission Performance of 2K-H Pin Cycloid Planetary Mechanism. Mechanism and Machine Theory, **42**, 776 (2007).
13. M. BLAGOJEVIC, N. MARJANOVIC, Z. DJORDJEVIC, B. STOJANOVIC, A. DISIC: A New Design of a Two-stage Cycloidal Speed Reducer. J. of Mechanical Design, **133** (8) (2011).
14. N. MARJANOVIC, B. IVKOVIC, M. BLAGOJEVIC, B. STOJANOVIC: Experimental Determination of Friction Coefficient at Gear Drives. J. of the Balkan Tribological Association, **16** (4), 517 (2010).
15. B. STOJANOVIC, N. MILORADOVIC, N. MARJANOVIC, M. BLAGOJEVIC, A. MARINKOVIC: Wear of Timing Belt Drives. J. of the Balkan Tribological Association, **17** (2), 206 (2011).
16. B. STOJANOVIC, S. TANASIJEVIC, N. MARJANOVIC, L. IVANOVIC, M. BLAGOJEVIC: Wear as the Criterion of Mechanical Transmitters Working Life. J. of the Balkan Tribological Association, **17** (2), 215 (2011).
17. M. BLAGOJEVIC, N. MARJANOVIC, B. STOJANOVIC, Z. DJORDJEVIC, M. KOCIC: Influence of Friction on the Force Distribution at Cycloidal Speed Reducer. In: 12th International Conference on Tribology, Kragujevac, Serbia, 2011, 226–229.
18. SUMITOMO Drive Technologies – Cyclo 6000 Speed Reducer. Catalog (2010).

Received 14 December 2011
Revised 23 March 2012