EPICYCLIC GEAR WITH FRICTION GEARING AND CARDAN MECHANISM

***Jozef Hal'ko*[1]*, Ján Paško*[2]*, Slavko Pavlenko*[3]**
[1,2,3]*Assoc. Prof., Eng. PhD.*
Technnical University of Košice, Faculty of Manufacturing Technologies with the seat in Prešov , Bayerova 1, Prešov, Slovak Republic

Abstract: *The problem of epicyclic gear with cardan mechanism based on friction gearing is presented in the paper. The authors emphasize the possibilities of gear and force ratios in the mechanism.*
Keywords: *epicyclic gear, cardan mechanism, friction gearing, high accuracy transmission.*

1. INTRODUCTION

Regarding the problem of high-accuracy transmissions, which is the part of scientific research at our Department, the principles of epicyclic gear with cardan mechanism were also investigated. The solutions were described in the articles [1] and [2]. Referring to these solutions, the possible solution of the epicyclic gear with cardan mechanism based on friction gearing is presented in the paper.

2. TRANSMISSION MECHANISM WITH FRICTION GEARING

Presented harmonic transmission [1] with cardan mechanism (Figure1) can be also used on the basis of friction gearing between the internal driving gear with friction area along external circumference and the crown wheel with friction area on its internal circumference.

Function of motion transmission from the input to the output is analogous to the previous gearings. The torque is transmitted by friction developed by mutual down-pressure of the both wheels. From the point of view of their capacity, it is assumed that these gearings will be used for low and average capacities. As with the other friction gearings, this transmission can function as safety clutch in case of overloading.

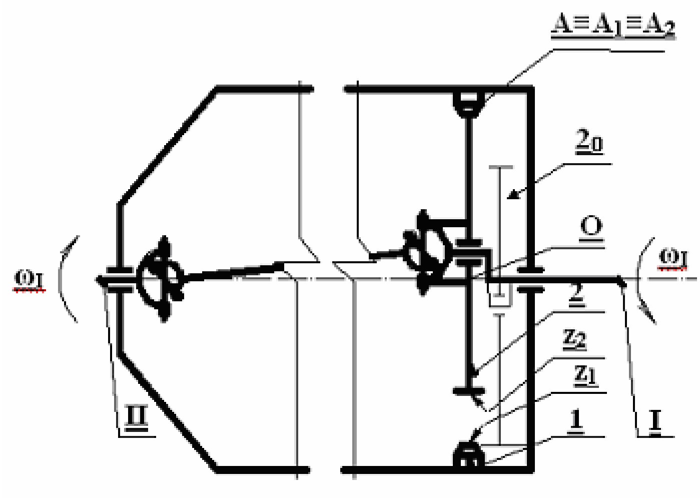

Fig. 1. Harmonic transmission with cardan mechanism

For the gear ratio *u*, the following relation can be derived:

$$u = \frac{\pi d_2}{\left(\pi d_2 - \pi d_1\right)\psi} = \frac{d_2}{\left(d_2 - d_1\right)\psi} = \frac{r_2}{\left(r_2 - r_1\right)\psi}, \tag{1}$$

where d_1 - is the internal diameter of the box crown wheel ($d_1=2r_1$),

d_2 - is the external diameter of the internal wheel ($d_2=2r_2$),

ψ - is the coefficient of slipping .

From the relation (1) it follows that with this gearing it is possible to reach large scale of gear ratios. In this case, the gear ratio is not limited by admissible difference of number of teeth of the both wheels. Technical possibilities in manufacturing of diameters of friction wheels with so little deviations in order to provide the continuous regular functioning of the transmission are considered to be the limiting factor of maximum gear ratio achievement. This is the advantage of the presented friction gearing as compared with the conventional friction gearings. To eliminate the impact of radial compressive force between the wheels it is possible to use supporting second wheel (e.g. with flexible prestressed down-pressure) located on the eccentric of the driving shaft opposite (swiveled through 180^o) the driving gear. This wheel would have a backing function and it does not transmit a torque. Down-pressure of the driving gear can be generated by rotation of the eccentric bushing inlayed for this purpose between the hub of the driving gear and the eccentric journal of the shaft. As with the other friction gearings, there is also a requirement of very precise manufacturing and assembly. To increase the frictional effect, it is possible to use the lining with higher friction coefficient (e.g. ferodo, etc). The assumed efficiency is η = 0,80 – 0,90.

To improve the transmission of circumferential forces and thus the torques it is possible to use the external castellation of friction areas along the circumference of friction wheels (Figure 2a). However, it results in decrease of efficiency because of greater mutual friction of friction areas as compared, for example, with spur friction gearing.

3. FORCE RATIOS ON FRICTION GEARING

The following figure represents the scheme of solution of friction gearing with supporting pressure disc $2'$ located on the eccentric in front of the working friction wheel *2* with the outlined force ratios. Figure 2a represents friction wheel with castellation. Figure 2b represents the friction wheels which are smooth along the circumference.

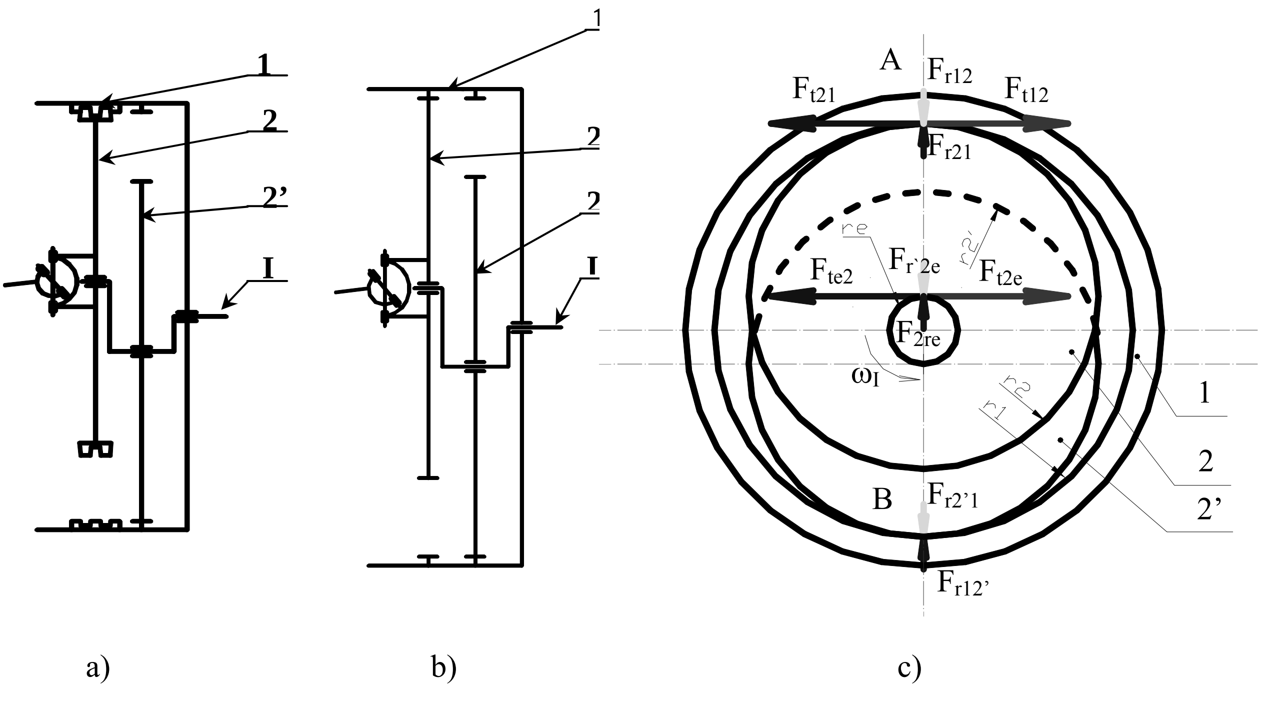

Fig. 2. Force ratios and the scheme of solution of eliminatory radial and centrifugal forces with friction gearing

There affect circumferential (tangential) and radial forces on the gear unit. Circumferential force F_{te2}, by which the eccentric of the shaft on the arm r_e affects the wheel *2,* is as follows:

$$F_{te2} = \frac{M_{ke}}{r_e} \,, \tag{2}$$

where M_{ke} is the input torque. At the same time the following is valid: $F_{te2} = F_{t2e}$.

Circumferential friction force at point A of the mesh on the wheel *2* results from the equilibrium of forces on this wheel:

$$F_{t12} = F_{te2} \tag{3}$$

Then the torque M_{k2} on the wheel *2* with the radius r_2 to the axis 0_2 is as follows:

$$M_{k2} = F_{t12}\ r_2 \tag{4}$$

The torque is then transmitted similarly as with gear transmissions through the cardan to the output shaft. The necessary circumferential friction force when considering the coefficient of friction is as follows:

$$F_T = k\,F_{t12} \tag{5}$$

where k = 3,0 – kinematical transmissions

k = 1,25 - 1,5 – force transmissions

Compressive radial (normal) force F_{r21} for generating of friction force F_t with the coefficient of friction of the friction couple f is as follows:

$$F_{r21} = \frac{F_T}{f} = \frac{k\,F_{t12}}{f} \tag{6}$$

From the stability it follows:

$$F_{r21} = F_{r2e} \tag{7}$$

F_{r2e} - normal force from the wheel *2* to the eccentric of the shaft.

From the above figure it follows that the first shaft bearing bears less radial load equal to the difference of radial compressive forces of the both wheels *Frv*.

$$F_{rv} = F_{r12} - F_{r12}, \tag{8}$$

3. CONCLUSION

With the operations where the fixed coupling in gear transmission is not required it is possible to use the presented technical solution in the form of friction gearing where the teeth will be replaced by friction areas (or, as it can be found in literature, with tooth system, if module m = 0). At the same time, this transmission could function as safety clutch, and greater gear ratios can be achieved when taking into consideration the acceptable slip with the involute or cycloidal transmissions. The requirement of precise production with the adjustment or regulation of radial compressive force is considered to be the disadvantage. In order to reduce the compressive force and to increase the transmission of torques it is possible to construct the circumferential friction areas in the form of circumferential keyways. With greater eccentricities and angular speeds it is also necessary to consider the balancing on the input shaft. With this transmissions, output revolutions are of the opposite direction because of the kinematics of transmission described in the part 4.2. The idea to design a prototype in order to verify and to test it experimentally has been under consideration at our Department.

The presented paper is the partial solution of the grant VEGA 1/2215/05.

4. REFERENCES

[1] HAĽKO, J. – KLIMO, V.: Harmonický prevod s kardanovým mechanizmom. In: *Zborník príspevkov* zo *6. medzinárodnej vedeckej konferencia Nové trendy v prevádzke výrobnej techniky 2003, 20. – 21. november 2003*. Prešov: FVT, 2003, s. 63 - 68, ISBN 80-8073-059-8

[2] HAĽKO, J. – KLIMO, V.: Dvojstupňový viacvýstupový prevod. In: *Sborník zo 45.mezinárodní konference Kateder části a mechanizmů strojů, 7. – 9. září 2004, Blansko, ČR,* Brno: VÚT, 2004, s. 110 – 115, ISBN 80-214-2702-7

[3] HAĽKO, J. – PAVLENKO, S.: Prevody s kruhovým rozmiestnením ozubených súkolesí. In: *Zborník prednášok zo semináru doktorandov FŠT 2003, 18.2. – 19.2.2003, Trenčín.* Trenčín: FŠT TnU AD v Trenčíne, CD-ROM, 2003, 80 s., ISBN 88914-80-9

[4] HAĽKO, J.: Vplyv parametrov na straty v prevodoch s malými a konvenčne používanými modulmi. In: *Zborník referátov z XLIII. Medzinárodnej vedeckej konferencie Katedier*

častí a mechanizmov strojov, Hotel Bieň, Bienska dolina, 3. – 5. September 2002. Zvolen: TU vo Zvolene, 2002, s. 232 – 235, ISBN 80-228-1174-2

[5] HOMIŠIN, J. a kol.: *Základy strojného inžinierstva*. Košice:2002, s.253, ISBN 80-7165-359-4

[6] KRÁL, Š. a kol.: *Časti a mechanizmy strojov II. diel*. Bratislava: STU, 2002, 296 s., ISBN 80-227-1723-1

[7] MALÍK, L. a kol.: *Časti a mechanizmy strojov*. Žilina: Vysoká škola dopravy a spojov v Žiline, SjF, 1992, 411 s., ISBN 80-7100-092-2

[8] MEDVECKÝ, Š. – ČILLÍK, L. – BARYSZ, I. – ŽARNAY, M. – HRČEKOVÁ, A. – BRONČEK, J. – KUČERA, Ľ.: *Základy konštruovania*. Žilina: ES ŽU, 1999, 599s., ISBN 80-7100-547-9

[9] PALKO, A. – NEČEJ, P.: Design of industrial robots based on modular principles. In: *Zborník prednášok zo 7. medzinárodnej konferencie Robotics in Alpe-Adria-Danube Region- RAAD´98.* Bratislava: Ústav teórie riadenia a robotiky SAV, 1998

[10] PEŠÍK, L.: *Částí strojů, stručný přehled, 2. díl.* Liberec: TU, 2002, 150 s., ISBN 80-7083-608-3