SIMULATION AND MODELING OF CYCLOID GEARS USED IN ORIENTATION MECHANISM OF ROBOTS

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Abstract: *Cycloid reducers compared with classical reducers have low axial and radial gauge, high transmission ratio, and a better reliability. Cycloid gearing roller, through its qualities, has an important role in modern mechanical transmissions. The difference between numbers of teeth of the cycloid gear roller can be equal to 1 ($|z1 - z2| \geq 1$), without risk of interference, as a result, can be obtained big gear ratios in accordance with in lower overall dimensions, while the classic gear have the difference between teeth number min $/z1 - z2 /\geq 4$. In this paper, the physical prototype of a new version of roller cycloid reducer is presented. Also one tries to test this variant of cycloid gear on a modern stand trial.*

Keywords:Orientation Mechanism, Cycloid Gears, Experimental Testing, Gear Prototype.

1. INTRODUCTION

The characteristics of cycloid reducers are: compact structures, high exploiting safety and shock absorbing capacity, are less expensive, not noisy, long life functioning and high transmission ratio.

The most essential robot peripheral is the end-effector, or end-of-arm-tooling. Common examples of end effectors include welding devices (such as MIG-welding guns, spot-welders etc.), spray guns and also grinding and debarring devices (such as pneumatic disk or belt grinders, burrs etc.) and grippers (devices that can grasp an object, usually electromechanical or pneumatic). Another common means of picking up an object is by vacuum [9] and [10].

End effectors are frequently highly complex, made to match the handled product and often capable of picking up an array of products at one time. They may utilize various sensors to aid the robot system in locating, handling, and positioning products [6].

The design of robotic mechanisms is a complex process involving geometric, kinematic, dynamic, tolerance and stress analyses [5], [11] and [12].

In the design of a real system, the construction ofa physical prototype is often considered.Regarding the orientation of the gears in the prototyping machine, it is advantageous to keep their axis vertical for smoothness of operation and strength [5].

The orientation mechanisms - OM of industrial robots have two or three output rotations. In the case of OM with cylindrical/conical gears, the kinematic chains are in correspondence with the known planetary mechanisms [12].

While the partially coupled orientation mechanisms with two rotations (OM-2R) are simple, those with three rotations (OM-3R) have a complex structure [7].

Reducing of the speed is a technical goal imposed by the need to adapt relatively high speeds of modern engines at the requirements of effectors which they serve.

2. PROTOTYPE CYCLOID REDUCER’S PRESENTATION

A new variant of cycloid reducer, equipped with a sun gear, which is proposed by the authors, is illustrated in Figure 1; this contains a cycloid gear pair with rollers, consisting of a fix sun gear with internal cycloid teeth 3 and of more eccentric rollers 2. The element H (which contains an eccentric bearing) designates the reducer's input, while the element 1 (on which the rollers 2 are eccentrically articulated) designates the output. In the premise that the reducer uses z2 = 16 rollers (as teeth), then z3 = z2 + 1 = 17 teeth and, implicitly, the reducer accomplish the transmission ratio [1] and [2]:

$$i_0 = i_{13}^H = \frac{\omega_{1H}}{\omega_{3H}} = i_{12}^H \cdot i_{23}^H = (+1)\frac{z_3}{z_2} = \frac{17}{16} = +1{,}0625 \qquad (1)$$

$$i_{H1}^3 = \frac{\omega_{H3}}{\omega_{13}} = \frac{\omega_{HH} - \omega_{3H}}{\omega_{1H} - \omega_{3H}} = \frac{\omega_{3H}}{\omega_{3H} - \omega_{1H}} = \frac{1}{1 - \frac{\omega_{1H}}{\omega_{3H}}} = \frac{1}{1 - i_0} = \frac{1}{1 - (17/16)} = -16 \qquad (2)$$

In the assumption of friction considering, the reducer efficiency ηH1 can be theoretically established through the following relation:

$$\eta_{H1} = \frac{-T_1 \cdot \omega_{13}}{T_H \cdot \omega_{H3}} = \frac{1 - i_0}{1 - i_0 \cdot \eta_0^w} \qquad (3)$$

where T_H and T_1 are the input and respectively output torque, η_0 is the reducer internal efficiency ($\eta_0 = \eta_{13}^H = \eta_{12}^H \cdot \eta_{23}^H = 0{,}995 \cdot 0{,}995 = 0{,}99$) and w is the efficiency coefficient (w = ±1) that models the power circulation in the fixed axis unit associated to the planetary unit[1, 2]:

$$w = \mathrm{sgn}(\omega_{1H} \cdot T_1) = \mathrm{sgn}\left(\frac{\omega_{1H} \cdot T_1}{-\omega_{13} \cdot T_1}\right) = \mathrm{sgn}\frac{\omega_{1H}}{\omega_{3H} - \omega_{1H}} = \mathrm{sgn}\left(\frac{i_0}{1 - i_0}\right) = \mathrm{sgn}\left(\frac{17/16}{1 - (17/16)}\right) = -1 \qquad (4)$$

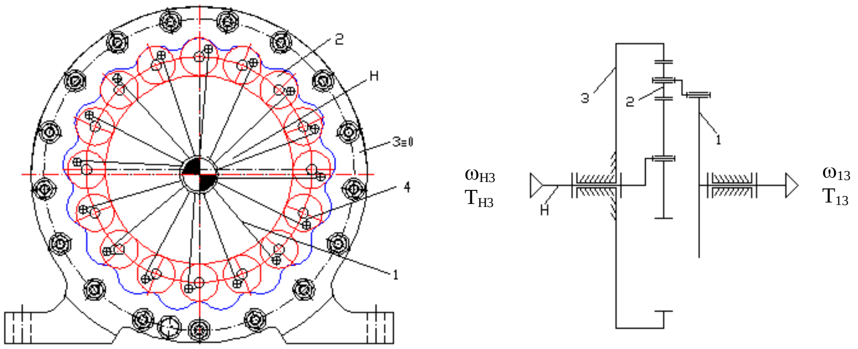

Fig. 1. *The cycloid reducer equipped with eccentric rollers (2) and a sun gear (3)*

3. EXPERIMENTAL TESTING

The experimental testing of cycloidal roller gear was performed on trial stand equipment located in the Transylvania University of Brasov.

The test stand comprises two synchronous machines Siemens, brushless, two Siemens converters, two torque and speed transducers, a tri-axial vibration transducer, a PC type computer system, external data acquisition equipment and a board for serial communication with engine controllers. After entering input data were drawn graphs that represent the torques recorded by transducers. In each graph, the top is illustrated the torque engine and in the bottom is the torque brake [13].

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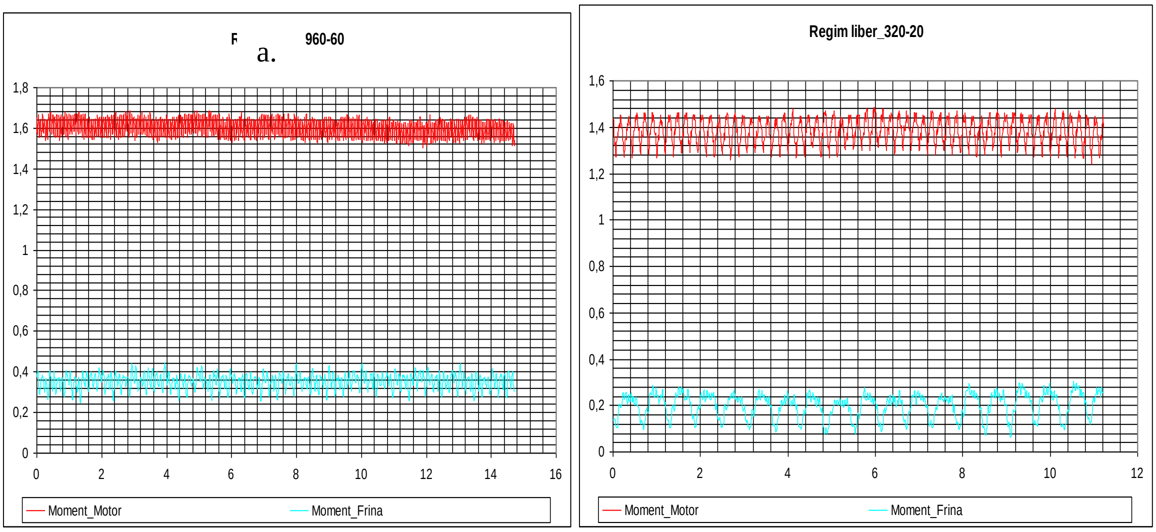
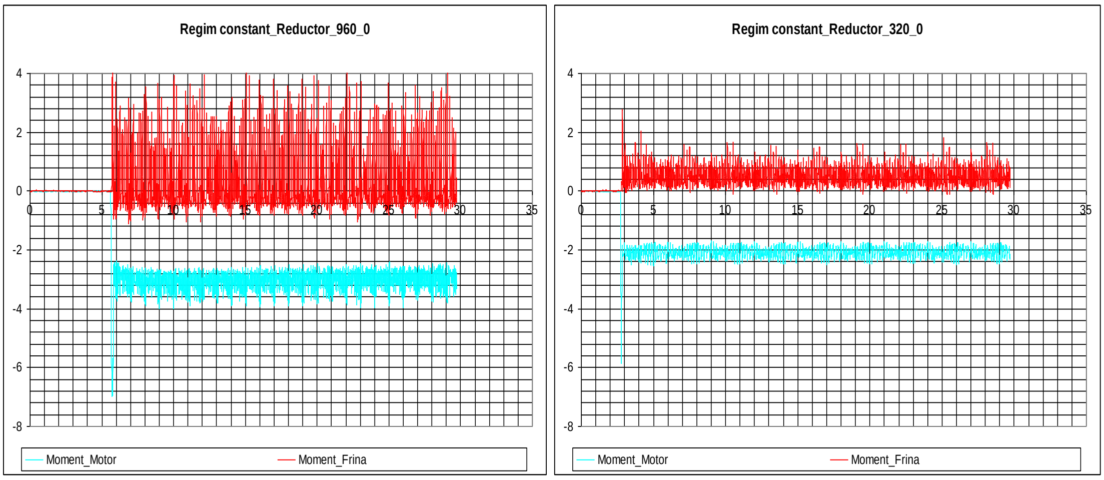


b.

Fig.2. *The torques recorded by transducers, where input and output gear decoupling and coaching of electric - driver machine, with speed of 320 [rpm] and 960 [rpm], and electric - brake machine, with speed of 20 [rpm] and 60 [rpm]*

Based on Figure 2 were obtained the next collated results systematized in the Table 1, where TM1 and TF1 are average torques recorded by the transducer of electric-motor machine and the electric-brake machine, in the case of the decoupling of reducer's input and output.

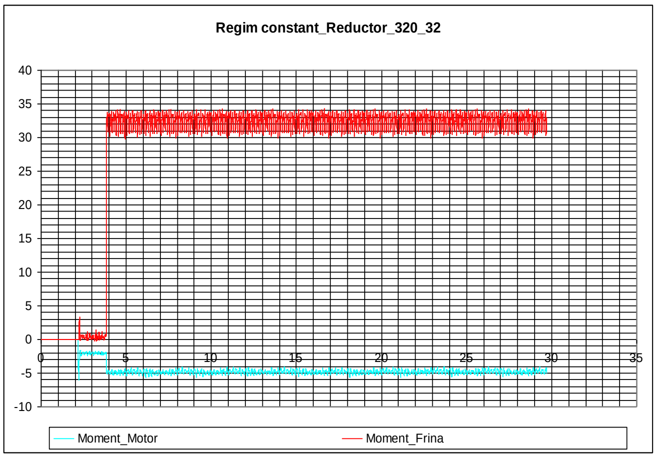


a. b.

Fig.3. *The torques recorded by transducers for: gear engaged, no-load brake and engine trained successively with speeds: a) 320 [rpm] and b) 960 [rpm]*

Table 1

Stand with decoupled gear's input and output			
Brake average speed [rpm]	***Drive average speed [rpm]***	***Average torques at drive transducer T_{M1} [Nm]***	***Average torques at brake transducer T_{F1} [Nm]***
-24.87196937	315.2917041	1.384555654	0.201004452
-59.49240742	960.5306679	1.604687033	0.352477936



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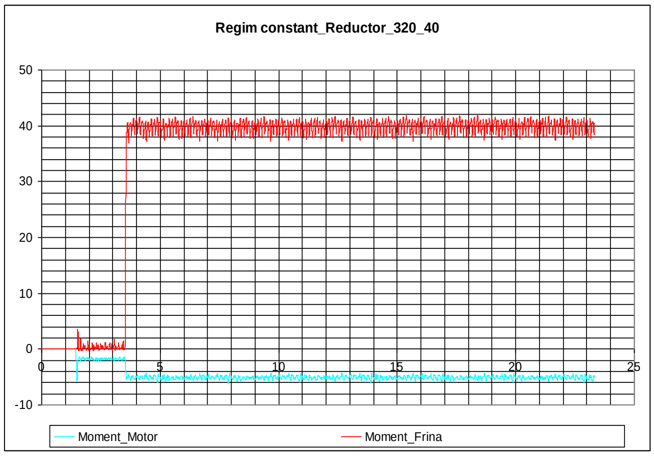

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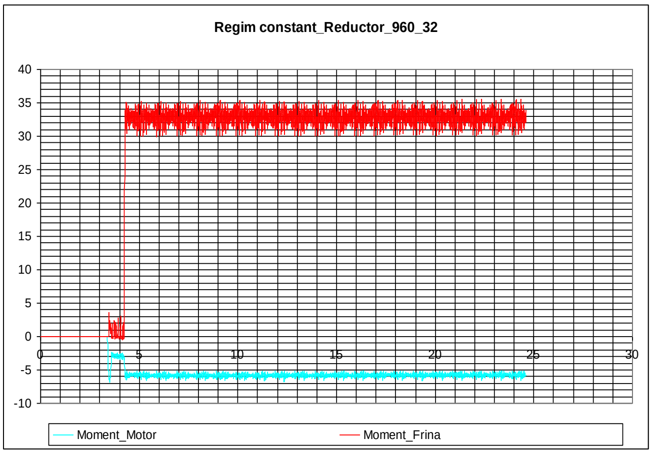

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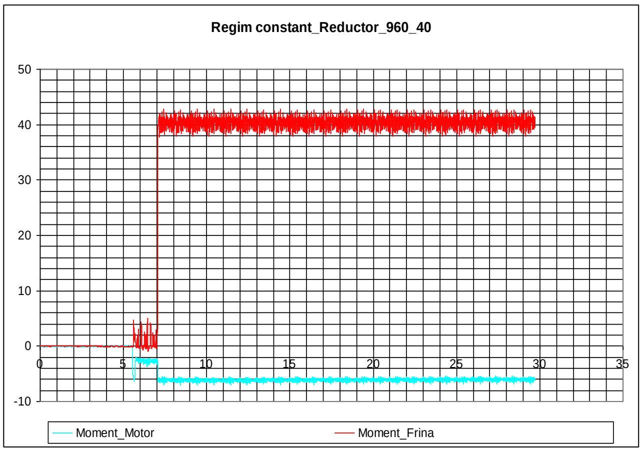

d.

Fig.4. *Torques recorded by transducers for: gear engaged, brake loaded sequentially to 32 Nm (a, c) and 40 Nm (b, d) and engine trained successively with speeds: 320 (a, b) and 960 (c, d)*

In Figure 3 are shown the torques recorded by the transducer adjacent engine and, also, by the transducer adjacent brake for: gear engaged, no-load brake and engine trained successively with speeds: a) 320 [rpm] and b) 960[rpm].

Near by, in Figure 4 are illustrated the torques recorded by transducers for: gear engaged, brake loaded sequentially to 32 Nm (a, c) and 40 Nm (b, d) and engine trained successively with speeds: 320 [rpm] (a, b) and 960 [rpm] (c, d).

After processing data from Figure 3 and 4, one obtained some results collated in Tables 2 and, respectively, 3.

In Table 2.a. are systematized the average values for engine speeds (nM2) and brake (nF2), and, also, for the torques recorded by the transducer adjacent to the engine (TM2) and at the adjacent brake (TF2), in the case in which the motor-gear-brake system works with no load brake (idling on the brake).

In the next Table 2.b., based on results from Tables 1 and 2, a. have been set the average resistant torques of the gear, proper to the dry running of brake (TF3=TF2+TF1) and residual torque engine of gearbox (TM3=TM2-TF3/i/η).

Table 2.a.

Brake average speed [rpm]	***Drive average speed [rpm]***	***Average torques at drive transducer T_{M2} [Nm]***	***Average torques at brake transducer T_{F2} [Nm]***
20.02330412	-319.9920079	-2.072301397	0.493840857
60.01714927	-959.9464764	-3.021674797	0.253289973

Table 2.b.

Brake average speed n_{F2} [rpm]	***Drive average speed n_{M2} [rpm]***	***Average residual engine torques of the gear $T_{M3}=T_{M2}-T_{F3}/(i.\eta)$ [Nm]***	***Average residual engine torques of the gear at no-load brake $T_{F3}=T_{F2}+T_{F1}$ [Nm]***
20.02330412	-319.9920079	-2.017264	0.694844
60.01714927	-959.9464764	-2.973733	0.605766

Table 3

Brake average speed n_{F4} [rpm]	***Drive average speed n_{M4} [rpm]***	***Average torques at drive transducer T_{M4} [Nm]***	***Average torques at brake transducer T_{F4} [Nm]***
20.06838119	-320.0272147	-4.791515959	32.62090036
20.01840823	-319.9954574	-5.081057096	39.93556498
60.15845723	-960.1246776	-5.821692777	32.74974975
60.07087449	-960.0508001	-6.100784185	40.28985173

Residual engine torque of the reducer represent, according to the calculation expression: TM3=TM2-TF3/i/η, the needed torque to drive the gearbox without load (no load to his riding).

For compatibility with common theoretical assumptions, the residual engine torque will be excluded from the calculation of efficiency; in this way, it becomes possible direct comparison between the theoretical value of the yield and its values based on experimental data processing.

On the basis specified in Table 3 were systematized, with the previous tables, the averages values of speeds and torques calculation of yield and, also, the values obtained for the gearbox's yield during the tested regimes.

With calculus, the theoretical yield was obtained for the considered reducer:

$$\eta = (1-i_0)/(1-i_0/\eta_0) = (1-1{,}0625)/(1-1{,}0625/0{,}99) = 0{,}\ 8534 \Rightarrow \boldsymbol{\eta = 85{,}34\%} \qquad (5)$$

The values of yield, based on experimental data in condition of compatibility under theoretical assumptions, are collated in Table 4 and were determined by the expression: η = (TF/TM)/(nM/nF) = (TF/TM)/i; for the four loading regimes, values were obtained:

a) **320** rpm-engine with**32** Nm – brake => **η = 74,190%;**
b) **320** rpm-engine with**40** Nm – brake => **η = 81,956%;**
c) **960** rpm-engine with**32** Nm – brake => **η = 72,827%;**
d) **960** rpm-engine with **40** Nm – brake => **η = 81,323%;**

Table 4

Brake average speed n_F [rpm]	***Drive average speed n_M [rpm]***	***Average engine torques without the residual torque $T_M=T_{M4}-T_{M3}$[Nm]***	***Average residual torques of the gear $T_F=T_{F4}+T_{F1}$ [Nm]***	***Yield $\eta=T_F/T_M/i$***
20.06838119	-320.0272147	-2.7742519	32.821904	74.190%
20.01840823	-319.9954574	-3.0637931	40.136568	81.956%
60.15845723	-960.1246776	-2.8479589	33.102226	72.827%
60.07087449	-960.0508001	-3.1270503	40.642328	81.323%

4. CONCLUSION

Theoretically yield of the gear reducer is determined, in classic premise, taking into account **only** the friction losses of cycloid gear and those from the rotation couples of the radial couplings. Under these conditions, the theoretical yield was obtained for the considered reducer: **η = 85.34%.**

From the above comparison values, can thus be considered that in the manufacturing cycloidal teeth conditions, by copying after tracing, the results obtained through prototype testing confirms satisfactory the theoretical model.

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