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ON THE FEATURES OF A NEW CYCLOIDAL PLANETARY GEAR USUFUL TO FIT RENEWABLE ENERGY SYSTEMS

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ABSTRACT
The paper deals with a new variant of a planetary cycloid transmission, proposed by the authors, able to accomplish high transmission ratio and high efficiency, useful to fit mechatronic devices of renewable energy systems (RES). Based on a transmission physical prototype and using a modern high-tech stand, theoretical studies and experimental testing on the kinematical and dynamic behavior of the new transmission are presented in the paper.
Keywords: cycloidal planetary gear, kinematics, dynamic features, efficiency, experimental testing.

1. INTRODUCTION
A main parameter that interferes in the speed reducer/amplifier prerequisites is referring to the value of the *kinematical ratio i*. Generally, this ratio increase is accompanied by the composition, in different combinations, of several disadvantages: a) *the efficiency decrease*, b) *the overall dimensions increase*, c) *the complexity increase*, d) *the technological costs increase* etc. [1, 2, 3]. The paper deals with a new variant of a cycloidal transmission proposed by the authors, able to achieve high kinematical ratio and better additional performances: increased efficiency, low complexity and costs, simple technology, decreased overall dimensions. This new variant of cycloidal transmission is illustrated in Figure 1; it contains a cycloidal gear pair with rollers, consisting of a fixed sun gear with internal cycloidal teeth 3 and of more rollers 2. The element *H* designates the reducer's *input* (or amplifier's *output*), while the central element 1 (on which the rollers 2 are eccentrically articulated) designates the reducer's *output* (or amplifier's *input*). In the premise that this transmission acts as *reducer* and uses z_2 = 16 rollers (as teeth), then $z_3 = z_2 + 1 = 17$ teeth and, implicitly, the following *kinematical ratio* is achieved [4]:

$$i_{H1}^{3} = \frac{\omega_{H3}}{\omega_{13}} = \frac{\omega_{3H}}{\omega_{3H} - \omega_{1H}} = \frac{1}{1 - \omega_{1H}/\omega_{3H}} = \frac{1}{1 - i_0} = -16\,;\; i_0 = i_{13}^{H} = i_{12}^{H} \cdot i_{23}^{H} = +1 \cdot \left(+ z_3 / z_2\right) = +1.0625\,, \quad (1)$$

where i_0 is the *internal ratio* of the planetary unit. In the assumption of friction considering, the reducer efficiency η_{H1} can be theoretically established through the following relation:

$$\eta_{H1} = \frac{-T_1 \cdot \omega_{13}}{T_H \cdot \omega_{H3}} = \frac{-T_1 / T_H}{\omega_{H3} / \omega_{13}} = \frac{\overline{i_{H1}^{3}}}{i_{H1}^{3}} = \frac{1 - i_0}{1 - i_0 / \eta_0}\,, \quad (2)$$

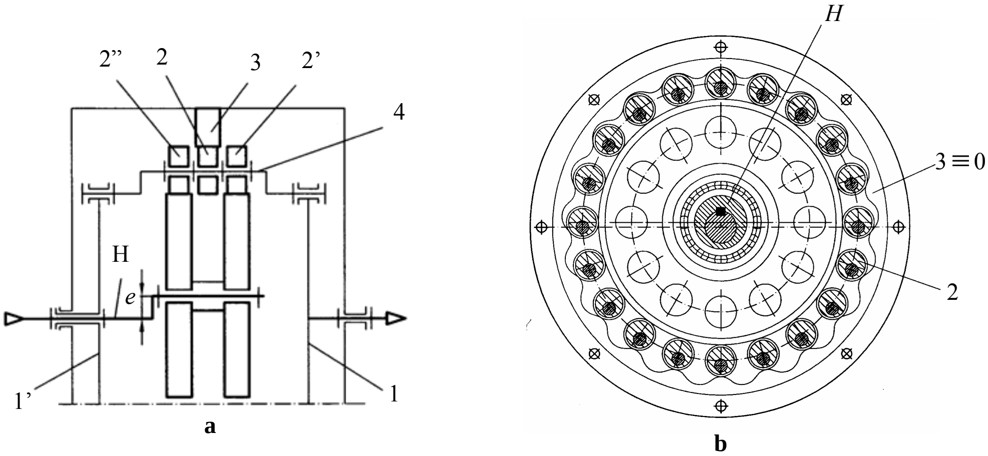


Figure 1. (a) Conceptual scheme, (b) Frontal view of the cycloidal planetary transmission.

where T_H, T_1 are the input and output torque, η_0 is the *internal efficiency* ($\eta_0 = \eta_{13}^H = \eta_{12}^H \cdot \eta_{23}^H$) and w is the efficiency coefficient ($w = -1$, [4]).

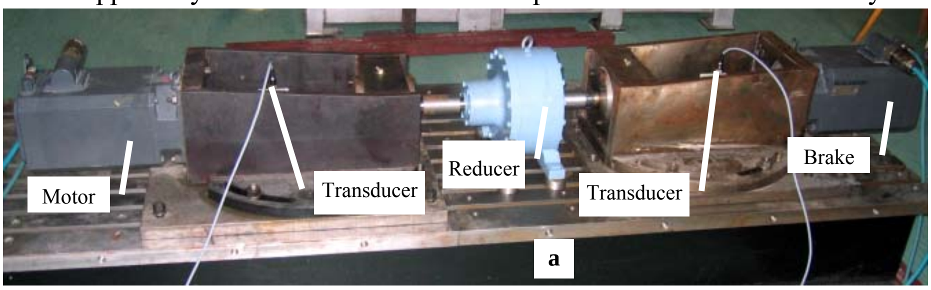
The paper deals with the *efficiency* η_{H1} (see relation 2) experimentally established based on the measurement of the input and output torques (T_H and T_1) and of their corresponding rotational speeds (ω_{H3} and ω_{13}); by means of the obtained *efficiency* η_{H1}, the *internal efficiency* η_0 is thus determined:

$$\eta_{H1} = \frac{1 - i_0}{1 - i_0 \cdot \eta_0^{-1}} \Rightarrow \eta_0 = \frac{i_0 \cdot \eta_{H1}}{i_0 - (1 - \eta_{H1})}. \tag{3}$$

Further on, the paper presents briefly the stand used in experimental testing and the experimental research planning (section 2), the obtained results (section 3), and final conclusions (section 4).

2. PLANNING OF THE EXPERIMENTAL TESTING

Experimental testing of the efficiency of the new cycloidal transmission (Fig. 1) is based on the numerical data supplied by a *mechatronic stand* developed at Transilvania University of Brasov/Romania (Fig. 2).



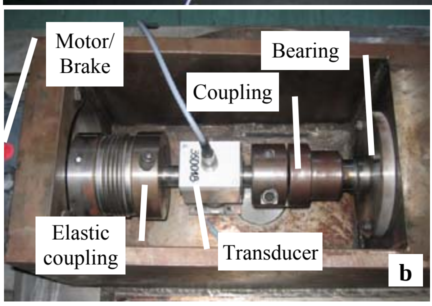


Figure 2. The stand-reducer system: (a) general view, (b) stand sub-components.

This stand has two synchronous servo-motors Siemens 1FT6-105 (Fig. 2,a), which can act as motor or brake, controlled by two Siemens 6SE7023 PWM invertors. The sensorial system (Fig. 2,b) is composed by two T20WN/200 HBM torque-speed transducers. The testing system is controlled by the operator through a PC connected to the servomotor’s controllers and to the acquisition external device HBM Spider 8 [6]. In this stand, a servomotor is acting as motor with speed control, while the other is acting as brake with torque control.

The testing program is based on the following algorithm [5]:

A. There are recorded (experimental data delivered by torque-speed transducers): a) instantaneous torque and speed values of the system motor-reducer-brake, in both cases of idle-running regime ($T_b = 0$) and load running regime ($T_b > 0$); b) instantaneous torque/speed values of each servomotor subsystem (acting as motor), decoupled from the transmission and no load at the output shaft.

B. For each of two previous mentioned working situations a) and b):

– the motor speed was successively commanded at the values: 320; 960 and 1600 [rot/min],
– for each commanded motor speed, in case a), the brake was loaded successively at the following torque values: 0; 32 and 40 [Nm],

– the experimental data (torque and speed) delivered by both stand transducers were recorded and the average experimental values of speed (n_M – motor average speed, and n_B – brake average speed) and of torque (namely T_M and respectively T_B) were established, for each working case.

C. In the cases of load running regime ($T_b > 0$) from the acting situation a):

– there were established the input torque ($T_M = T_H$) and the output torque ($T_B = T_1$) of the transmission,

– the *efficiency* (η_{H1}) was computed in the known *classical assumption*: there are considered only the mechanical energy looses due to the friction in cycloidal gear and in the revolute joints of the reducer radial coupling (see Fig. 1); in this respect, the energy looses from shaft bearing, the energy needed for lubricant moving etc. are neglected.

– finally, the *interior efficiency* (η_0) was established for each distinct operating case, based on the values previously computed of the *efficiency* η_{H1} (see relation 3).

For a correct interpretation of the numerical results obtained from the performed testing and to comply with the classical assumption, it must make evident the mechanical energy losses in the mechatronic stand subcomponents (see Fig. 2,a,b): ***motor*** – elastic clutch – transducer – rigid coupling – bearing – ***reducer*** – bearing – rigid coupling – transducer – elastic clutch – ***brake***.

3. EXPERIMENTAL RESULTS

The records performed on each servomotor subsystem, decoupled from the reducer (stage I), are exemplified in Fig. 3,a for n_{motor} = 960 rot/min and n_{brake} = 60 rot/min. These data, obtained while each servomotor is acting as motor, allow us to identify the energy looses due to the bearing friction (see Fig. 2,b), where T_{M1} and T_{B1} represent the *average torques* recorded by the motor transducer and respectively brake transducer. In the second stage were also recorded the torque and speed values delivered by the transducers, for the case of: *reducer coupled to the servomotor subsystems, no brake load and the motor speed commanded successively to: a) 320, b) 960 and c)1600* [rot/min]. The variation of instantaneous torques is exemplified in the Fig. 3,b in the case of n_{motor} = 960 rot/min. Similarly, in the third stage were recorded the instantaneous torques and speeds for the case of *reducer coupled to the servomotor subsystems, the motor speed commanded successively to: a) 320, b) 960 and c)1600* [rot/min] and the brake successively loaded at *32* [Nm] and *40* [Nm]; the torques' records are exemplified (Fig. 3,c) for n_{motor} = 960 rot/min and T_{brake} = 40 [N·m]. The motor/brake torques' oscillations (Fig. 3) are generated mainly by a certain disequilibrium of the input shaft *H*, combined with certain manufacture inaccuracy of the cycloidal toothing.

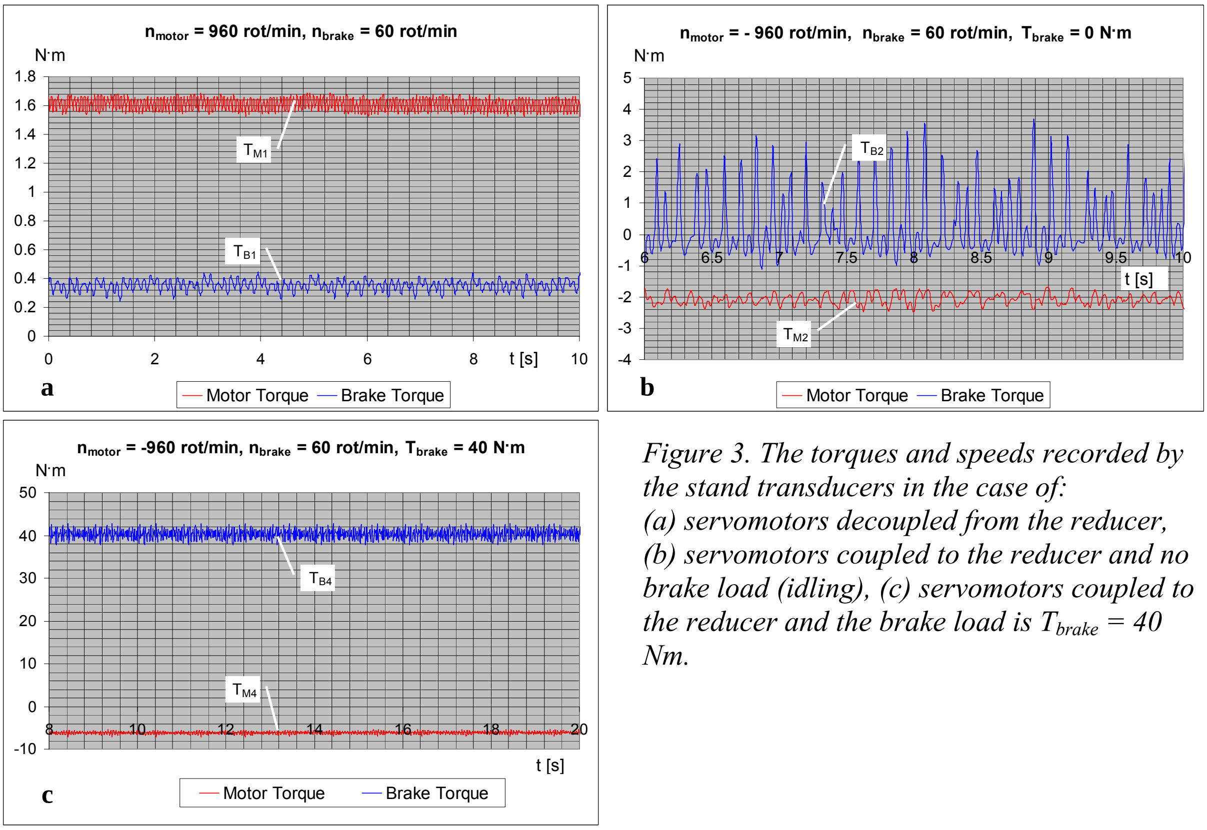


Figure 3. The torques and speeds recorded by the stand transducers in the case of:
(a) servomotors decoupled from the reducer,
(b) servomotors coupled to the reducer and no brake load (idling), (c) servomotors coupled to the reducer and the brake load is T_{brake} = 40 Nm.

Based on results shown in Fig. 3,a and b, for the idling stage, the average residual output torques ($T_{B3} = T_{B2} + T_{B1}$) and average residual input torques ($T_{M3} = T_{M2} - T_{B3}/i_{H1}/\eta_{H1}$) of the reducer were established using an iterative algorithm (Fig. 4) and are systematized in Table 1. The average values of the reducer input/output speeds (namely n_M / n_B) and torques (namely T_M /T_B), together with the efficiency η_{H1} for the tested load-running regime, are also systematized in Table 1.

Table 1. Average residual torques (T_{M3}, T_{B3}) and the reducer experimental efficiency

Average speed [rot/min]		*Average residual torque [N·m]*		*Average in/out torque [N·m]*		*Efficiency* [%]	
Motor n_M	Brake n_B	Motor $T_{M3}=T_{M2}-T_{B3}/(i_{H1}/\eta_{H1})$	Brake $T_{B3}=T_{B2}+T_{B1}$	Input torque $T_M = T_{M4} - T_{M3}$	Output torque $T_B = T_{B4} + T_{B1}$	η_{H1}	η_0
-320.0272	20.0683	-2.013490	0.694845	-2.778025	32.821904	73.843%	97.959%
-319.9954	20.0184	-2.019296	0.694845	-3.061760	40.136569	81.931%	98.719%
-960.1246	60.1584	-2.969479	0.605768	-2.852213	33.102227	72.536%	97.821%
-960.0508	60.0708	-2.975086	0.605768	-3.125697	40.642329	81.267%	98.662%
-1600.1638	100.0300	-4.309854	0.976504	-2.478539	33.061200	83.369%	98.840%
-1600.0849	99.94924	-4.307616	0.976504	-3.152220	40.799834	80.895%	98.630%
					Average values :	**78.973%**	**98.458%**

The efficiency values η_{H1}, established in the condition of complying with the theoretical premises, were computed with the expression $\eta_{H1} = (T_B/T_M)/(n_M/n_B) = (T_B/T_M)/i_{H1}$. Moreover, the internal efficiency η_0 (see relation 3) is derived for each corresponding efficiency value η_{H1}, yielding for all the 6 tested regimes to an average value $\eta_{0med} = 98.458\%$. Comparing the internal efficiency of the tested prototype with the efficiency of the involute gears with milled toothing ($\eta = 96\%–97\%$ [4]), it can be emphasized that the new proposed transmission (when act as reducer) has, beside other advantages (simple technology, reduce costs), a superior energetic performance ($\eta_{0med} = 98.458\%$).

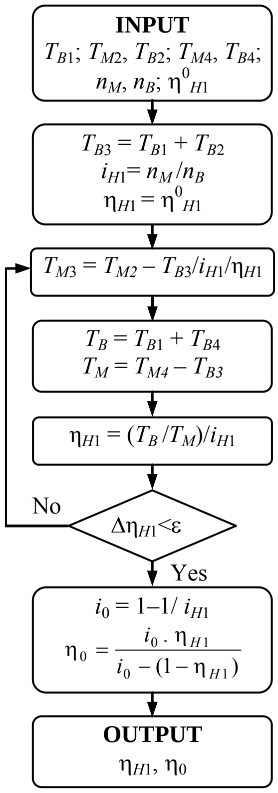


Figure 4. The iterative algorithm.

4. CONCLUSIONS

According to the previous remarks, the following final conclusions can be drawn:

1. A modern mechatronic stand of motor–reducer–brake type with open power flow was used to test the new proposed reducer prototype. This stand includes modern control and monitoring equipments.
2. Several testing running regimes were performed and their experimental torques and speeds were recorded. The average torques and average speeds were established for all the testing regimes.
3. The obtained experimental efficiency (in the conditions of the foremost prototype, where the cycloidal toothing was manufactured using a classical milling machine) attests the energetic superiority of the proposed reducer solution vs. the involute solutions.

5. ACKNOWLEDGMENT

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