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A COMPARATIVE CALCULATION OF CYCLOID DRIVE EFFICIENCY

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Abstract: Determining cycloid drive efficiency is a very complex task that in the most cases requires theoretical analysis, numerical calculation, computational simulations and experimental research. The paper aims to compare the results obtained based on two so far the most recognized and acceptable theoretical models (Malhotra model and Kudryavtsev model). On the one hand, the presented models are very similar (they both take into account the power losses due to friction on the central gear rollers, output rollers and in the cycloid gear bearings). They differ in the methodology of power losses calculation and in the number of contacts between certain cycloid drive elements that are taken into account. The computational simulation was performed in the MATLAB software package on specific examples of classical single-stage cycloid drives.

Keywords: cycloid drive, efficiency, theoretical models

1 INTRODUCTION

Compared to other gearboxes, cycloid reducers cover a wide range of gear ratios, have a very long and reliable service life, high efficiency, almost zero backslash in starting and stopping, low noise, low vibration, mass and overall dimensions. These characteristics make them suitable for industrial application, and certainly one of the most important characteristics is the efficiency.

Thanks to many years of very extensive research, numerous models are available today to predict the value of the efficiency. However, they all start from the hypothesis defined by *Malhotra* and *Kudryavtsev*. Secondly, the application of new efficiency models yields very complex systems of equations whose solutions involve the use of complex algorithms, primarily in defining loads.

Kudryavtsev [1] was the first one who described cycloid reducers in detail and presented a procedure for calculating power losses with simplified forces assumptions. *Malhotra* [2] continued the research of *Kudryavtsev*. He added a larger number of

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contacts that lead to power losses and he defined forces in a different way. *Gorla* et al. [3] presented a modified efficiency model for a new concept of cycloid reducers with results of theoretical and experimental research. *Mačkić* [4] compared the *Malhotra* method with the *Gorla* method for different operating conditions with constant values of friction coefficients and geometric dimensions. *Mihailidis* [5], *Olejarczyk* [6] and *Concli* [7] added power losses resulting from the resistance of the lubricant, sealant, other bearings and other elements of the cycloid reducer. *Matejić* [8] compared *Kudryavtsev*'s method results with results of experimental analysis for different working conditions. *Bednarczyk* [9] dealt with the analysis of load distribution and power losses taking into account the gaps that occur between the gears and the meshing elements. *Olejarczyk* [10] and *Mihailidis* [11] analyzed the efficiency depending on the applied lubricant. *Phama*, *Bednarczyk* and others [12,13] analyzed the dependence of production tolerances on efficiency. A large number of studies are also devoted to the load distribution in the cycloid reducers for the ideal meshing case (when all rollers transfer the load) [1,14], as well as for cases that take into account the gaps between the elements of the cycloid drive [15,16]. Also, researchers are engaged in defining new concepts [3,17], modifying the profile of teeth and lateral clearance [18], dynamic analysis [19,20], analysis of stress-strain state of vital elements [17,21], etc.

Based on the shown literature review, it can be concluded that today, numerous researchers around the world are dealing with various aspects in order to increase the cycloid reducer's efficiency.

Since time has been set as important factor in the product development process, *Malhotra* defined the application of simpler models than it and *Kudryavcev* is justified. Therefore, within this paper, a comparative analysis of the results of efficiency obtained by the method of *Malhotra* and *Kudryavtsev* was conducted. The analysis was performed in the *Matlab* software, in which the existing comprehensive expressions were implemented.

2 MATHEMATICAL MODELS FOR EFFICIENCY CALCULATION

The procedure of cycloid reducer efficiency determining is based on calculation of the total power losses between reducer vital elements. Movement resistance is friction, which occurs due to slipping, or rolling, where the rolling friction is significantly less than sliding. The magnitude of rolling or sliding friction is expressed by the friction coefficient, which depends on the type of material in contact, the quality of treated surfaces, tolerances, gaps and other factors. In Figure 1 the contacts that cause to movement resistance are shown.

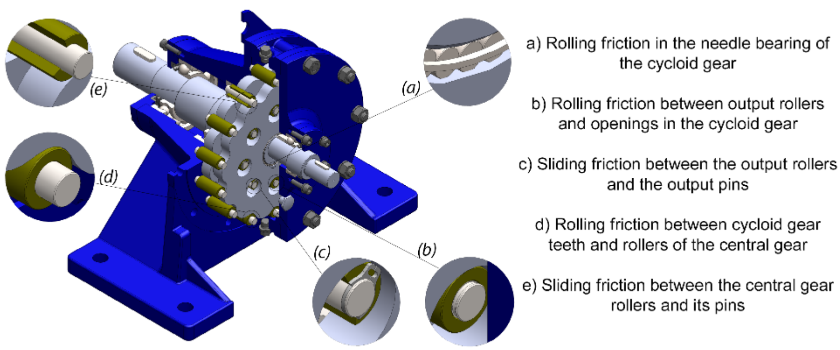


Figure 1. *Cycloid reducer power loses locations*

2.1 *Kudryavcev*'s model

According to *Kudryavcev*, total efficiency is calculated based on equation:

$$\eta = \frac{1-\psi}{1+z_1 \cdot \psi} \tag{1}$$

where are: z_1 – cycloid gear number of tooth (for single-stage reducer $z_1 = u_{CR}$); ψ – total power losses due to friction between cycloid reducer elements.

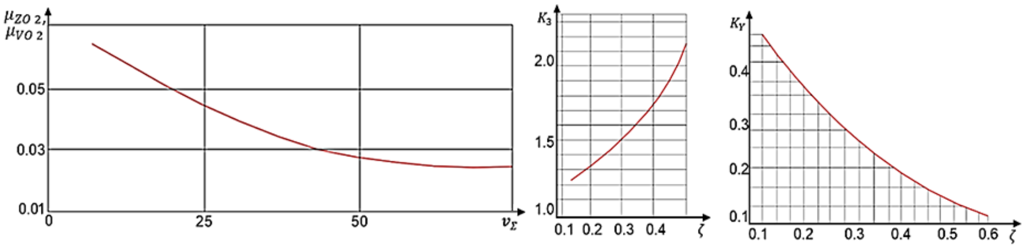
Total power losses ψ present sum of the following: loss due sliding friction between axles and rollers of ring gear ψ_1, loss due sliding between output rollers and its axles ψ_2, loss due rolling friction in eccentric cam bearing ψ_3, as follows:

$$\psi = \psi_1 + \psi_2 + \psi_3 \tag{2}$$

Power loss ψ_1 is calculated by equation:

$$\psi_1 = \frac{K_3 \cdot \mu_{ZO}}{z_2} \tag{3}$$

where are: K_3 – factor which takes into account tooth correction of cycloid gear (Figure 2.b); μ_{ZO} – friction coefficient between ring gear rollers and its axles; z_2 – ring gear roller number.



a) *b)* *c)*

Figure 2. *Diagrams for coefficient determination: a) $\mu_{ZO\,2}$, $\mu_{VO\,2}$; b) K_3; c) K_Y, [1]*

Friction coefficient between ring gear rollers and its μ_{ZO} is equal to lower value between $\mu_{ZO\,1} = (d_0/D_0) \cdot \mu_{s2}$ and $\mu_{ZO\,2}$. Value of the coefficient $\mu_{ZO\,2}$ is determined in dependence of velocity $v_\Sigma, m/s$ defined by equation (4) in accordance with Figure 2.a, where are: d_0 – ring gear axle diameter, mm; D_0 – ring gear roller diameter, mm; μ_{s2} – sliding friction coefficient between ring gear rollers and its axles. All geometric parameters of vital cycloid reducer elements are given in Figure 3.

Value of total rolling velocity in the meshing zone is given by equation:

$$v_\Sigma = 0{,}8 \cdot \frac{r}{1000} \cdot \frac{\pi \cdot (n_{iz} - n_{ul})}{30} \tag{4}$$

where are: r – ring gear dividing circle radius, mm; n_{iz} – output RPM, n_{ul} – input RPM.

Power loss ψ_2 is given by equation:

$$\psi_2 = \frac{30 \cdot P_{VK}}{T_3 \cdot \pi \cdot (n_{iz} - n_{ul})} \tag{5}$$

where are: P_{VK} – power loss due to contact friction between output rollers and cycloid gear, W; T_3 – cycloid gear torque, Nm.

Power loss due to contact friction between output rollers and cycloid gear is given by equation:

$$P_{VK} = \frac{4 \cdot T_3}{\pi \cdot R_{0\,izl}} \cdot e \cdot \frac{\pi \cdot (n_{iz} - n_{ul})}{30} \cdot \mu_{VO} \tag{6}$$

where are: $R_{0\,izl}$ – radius of the cycloid gear holes placement circle, mm; e - eccentricity, mm; μ_{VO} – friction coefficient between output rollers and its axles.

Friction coefficient between ring gear rollers and its axles μ_{VO} is equal to lower value between $\mu_{VO\,1} = (d_{VK}/D_{VK}) \cdot \mu_{s1}$ and $\mu_{VO\,2}$. Coefficient value $\mu_{VO\,2}$ is determined in dependence of velocity $v_{\Sigma}, m/s$ given by equation (7) according to Figure 2.a, where are: d_{VK} – output rollers axle diameter, mm; D_{VK} – output rollers diameter, mm; μ_{s1} - friction coefficient between output rollers and its axles.

Value of total rolling velocity in the contact zone is given by equation:

$$v_{\Sigma} = \frac{\left(\frac{D_{OCZ}}{2} + \frac{D_{VK}}{2}\right)}{1000} \cdot \frac{\pi \cdot (n_{iz} - n_{ul})}{30} \tag{7}$$

where are: D_{OCZ} – cycloid gear hole diameter, mm.

Power loss ψ_3 is given by equation:

$$\psi_3 = 1{,}25 \cdot \frac{T_T}{T_3} \tag{8}$$

where are: T_T – torque moment in bearing, Nm.

Torque moment in bearing is given by equation:

$$T_T = 1{,}3 \cdot \frac{k}{1000} \cdot \left(1 + \frac{d_{CZ}}{d_{kt}}\right) \cdot \frac{1000 \cdot T_3}{r_1} \cdot \sqrt{1 + \left(\frac{4}{\pi} \cdot \frac{r_1}{R_{0\,izl}} - K_Y\right)^2} \tag{9}$$

where are: k – cycloid gear bearing step of rolling resistance (k=0,005 mm); d_{CZ} – eccentric bushing diameter, mm; d_{kt} - cycloid gear bearing rolling body diameter (needle, roller or ball) , mm; r_1– stationary circle radius, mm; K_Y - factor which takes into account tooth correction of cycloid gear (Figure 2.c).

2.2 *Malhotra*'s model

According to the calculation procedure of *Malhotra*, whose conception of the cycloid reducer is identical to the conception of *Kudryavtsev*, the total efficiency is determined based on equation:

$$\eta = \frac{T_{EM} \cdot 2\pi - W_M}{T_{EM} \cdot 2\pi} \tag{10}$$

where are: T_{EM} – input shaft torque, Nm; W_M total friction force work.

The Malhotra's model differs from the *Kudryavtsev* model in the way it is defined and the number of defined power losses, but also in the fact that the *Kudryavtsev* model uses a simplified assumption when analyzing forces. The *Malhotra*'s model uses the assumption that a certain number of rollers is not loaded, so the forces values for these rollers is equal to zero. The forces distribution on the cycloid gear is shown in Figure 3.

Total friction force work W_M represents the integration of the sum of: elementary work due to friction in the cycloid gear bearing, elementary work of rolling friction between output rollers and gear holes, elementary work of rolling friction between gears and ring gear rollers, elementary work of sliding friction between output rollers and its axles and elementary work between the rollers and the axles of the ring gear. The comprehensive equation for calculating the total work W_M is:

$$W_M = \frac{f_{r1} \cdot D_{SR} \cdot z_1}{d_{kt}} \int_0^{\frac{2\pi}{z_1}} F_E(\beta) \cdot d\beta + z_1 \cdot \left(f_{r2} + \frac{\mu_{s1} \cdot d_{VK}}{2}\right) \cdot \int_0^{\frac{2\pi}{z_1}} \sum_{j=1}^{q} F_{Kj}(\beta) \cdot d\beta + \tag{11}$$

$$+(z_1 + 1) \cdot \left(f_{r3} + \frac{\mu_{s2} \cdot d_0}{2}\right) \cdot \int_0^{\frac{2\pi}{z_1}} \sum_{i=1}^{p} F_{Ni}(\beta) \cdot d\beta$$

where are: $f_{r1} = \mu_{r1} \cdot d_{kt}/2$ – cycloid gear bearing step of rolling resistance, mm; μ_{r1} – friction coefficient in cycloid gear bearing; $F_E(\beta)$ – current value of eccentricity force, N; $D_{SR} = (D_{CZ} + d_{CZ})/2$ – middle diameter of cycloid gear bearing, mm; D_{CZ} – outer diameter of cycloid gear bearing, mm; $f_{r2} = \mu_{r2} \cdot D_{VK}/2$ – step of rolling resistance at output roller, mm; μ_{r2} – rolling friction coefficient between output rollers and cycloid gear holes; $F_{Kj}(\beta)$ – current output force on j^{th} output roller, N; q – current number of output rollers meshing with cycloid gear. If number of output rollers u is even, then $q = u/2$, in the opposite case is $q = (u-1)/2$; $f_{r3} = \mu_{r3} \cdot D_0/2$ – distance of ring gear roller rolling resistance, mm; μ_{r3} – friction coefficient between ring gear rollers and cycloid gear; $F_{Ni}(\beta)$ – current value of normal force on i^{th} ring gear roller, N; p – current number of ring gear rollers in meshing. If total number of ring gear rollers is even, then $p = z_2/2$, and in the opposite case is $p = (z_2+1)/2$.

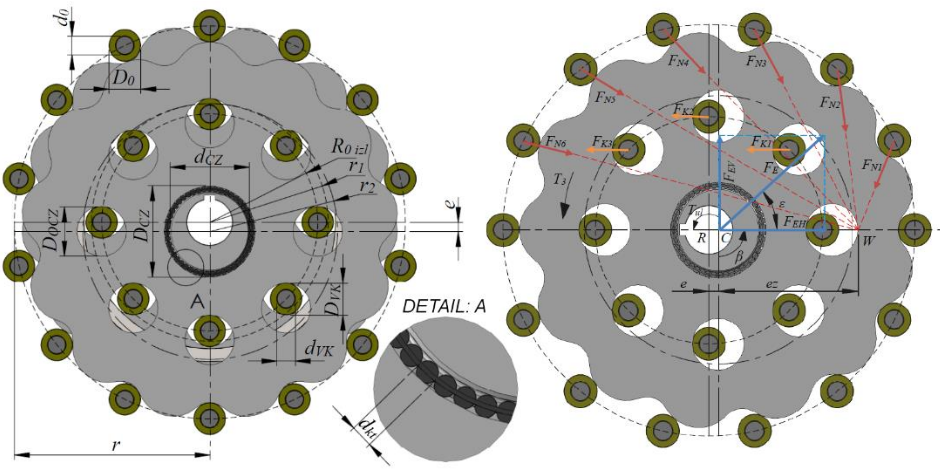


Figure 3. *Geometry and load of cycloid reducer*

3 COMPARATIVE ANALYSES OF EFFICENCY

Comparative analysis of the cycloid reducers efficiency using the *Malhotra*'s and *Kudryavtsev*'s models is based on testing cycloid reducers with different sizes of vital elements that are in function of parameters such as: input power P_{UL}, transmission ratio u_{CR} and input RPM n_{EM}.

Therefore, for the purposes of this research, the values of individual parameters, namely the input power, were taken from the *Sumitomo* catalog $P_{UL} = (2{,}2;\ 3;\ 4;\ 5{,}5;\ 7{,}5;\ 11)\ kW$ and transmission ratios $u_{CR} = (11;\ 13;\ 15;\ 17;\ 21;\ 25)$. Data for standard electric motor RPM's are taken from the *ATB Sever* catalog $n_{EM} = (600;\ 750;\ 1000;\ 1500;\ 3000)\ RPM$.

The values of the sliding and rolling friction coefficients were adopted from references [1,2,4,5,22], and their values are shown in Table 1.

Table 1. *Adopted values of sliding and rolling friction coefficients*

μ_{r1}	$\mu_{r2} = \mu_{r3}$	$\mu_{s1} = \mu_{s2}$	μ_{s3}
0,005	*0,0045*	*0,05*	*0,04*

The computational simulation was performed in the *Matlab* software, and *Lehmann* force equations were used for the *Malhotra*'s model [14].

Efficiency η_{CR} given based on input power P_{ul}, with constant transmission ratio $u_{CR} = 13$ and constant input RPM $n_{ul} = 1500\ RPM$, is given in Figure 4.a Input power is varied from 2,2 kW to 11 kW. According to *Malhotra*, with the increase of input power, the efficiency of cycloid reducers increases from 90,44% to 92,28%, and according to *Kudryavtsev* from 90,19% to 90,79%. The largest difference between the values of the efficiency is 1,56% at input power of 7,5 kW.

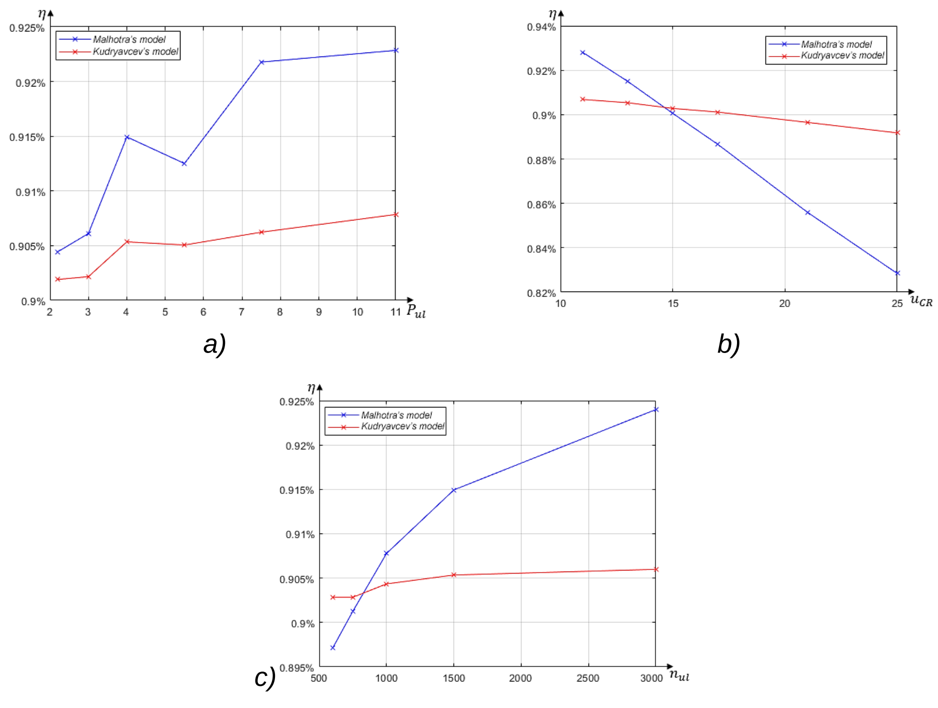


Figure 4. C*omparative results of efficiency for Malhotra's and Kudryavcev's model: a) input power variation; b) transmission ratio variation and c) input RPM variation*

Efficiency dependence η_{CR} from transmission ratio u_{CR} with constant input power $P_{UL} = 4\ kW$ and constant RPM $n_{ul} = 1500\ RPM$ is given in Figure 4.b The transmission ratio was varied in range from 11 to 25. As the transmission ratio increases, the efficiency of the cycloid reducers decreases. According to *Malhotra*, efficiency decreases from 92,80% to 82,84%, and according to *Kudryavcev* from 90,68% to 89,18%. The largest difference between the values of, efficiency is 6,34%.

Efficiency dependence η_{CR} from input RPM n_{ul} with constant power $P_{UL} = 4\ kW$ and transmission ratio of $u_{CR} = 13$ is given in Figure 4.c. The input RPM was varied from 600 to 3000 RPM. According to *Malhotra*, with the increase of the input RPM, the efficiency of the cycloid reducers increases from 89,71% to 92,40%, and according to *Kudryavtsev* from 90,28% to 90,60%. The largest difference between the values of the efficiency is 1,8%.

4 CONCLUSIONS

For the two most known methods, *Kudryavcev* and *Malhotra*, a computational simulation was performed in the *Matlab* software. The simulation consists of three parts: calculation of cycloid reducer geometric dimensions, forces calculation and efficiency calculation based on power losses. All geometric parameters of vital cycloid reducer elements change with the change of input power, input RPM and transmission ratio.

The simulation results show that similar graph shapes are obtained for both models. Deviations range are from 1,56% to 6,34%. Also, the similarities of these models are in the locations of power losses. The advantage of the *Kudryavtsev* model is from the aspect of simplicity, however this model does not take into account the load on the output rollers and ring gear rollers, neither the rolling friction between the output rollers and the holes in the cycloid gear and the rolling friction between the ring rollers and the cycloid gears. In addition, it assumes that all rollers are equally loaded.

According to *Malhotra*, when there are no gaps in the gearbox, both gears are in contact with half of the rollers of the central gear and half of the output rollers that participate in the process of load transmission.

In real working conditions, there are gaps: due to production, easier assembly and disassembly, better lubrication, so less than half of the rollers are loaded. Therefore, the size of the clearance directly affects the number of rollers that are in contact with the cycloid gear, so that as the clearance increases, the number of loaded rollers and efficiency decreases. Therefore, *Malhotra's* model gives more accurate results, although it is much more complex than *Kudryavtsev's*.

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