

The influence of the cycloid disc bearing type on the cycloidal speed reducer efficiency

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Abstract. The robotics industry has experienced a rapid expansion in the last decade. As a result, the market of speed reducers with high gear ratios, high precision and a compact design is growing rapidly. In addition to these characteristics, it is very important for gear trains to have high efficiency. Because of their compact design, cycloidal speed reducers are not exposed to axial forces, which is their important characteristic. This means that radial bearings can be used in their supports. The bearing type has significant effects not only on the design and dimensions of the cycloid speed reducer but also on its efficiency. In this paper, an analysis of power losses for different types of rolling bearings of a cycloid disc has been performed. At the end of the paper, concrete conclusions are presented as well as directions for future research.

1 Introduction

The industrial revolution that we are experiencing nowadays brings advancement and automation of modern equipment and a widespread use of artificial intelligence. People still play a leading role in the production process, but industrial robots are being increasingly used in the process of packaging, welding, testing and assembly in the automatic production chain. Although industrial robots consist of a large number of mechanical and electrical components, high precision speed reducers are one of the most important components. They are mainly planetary, wave and cycloidal speed reducers.

Cycloidal speed reducers meet high requirements such as a compact design, long and reliable service life, high efficiency, very low idle speed when starting off and stopping, low noise level, low vibration, etc. Generally, efficiency is the most important performance characteristic both for reducing energy consumption and increasing productivity of industrial robots.

In the literature, there are a lot of papers dealing with efficiency of cycloidal speed reducers. *Malhotra* [1] was the first to define analytical expressions for calculation of power losses at different locations of cycloidal speed reducers. *Sensingier* [2] used optimisation in order to increase the efficiency of the cycloidal speed reducer. *Mackic* [3] studied the influence of geometric parameters on the cycloid speed reducer efficiency. *Mihailidis* [4]

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presented a new approach to determining efficiency by calculating the friction coefficient at each contact point. *Olejarczyk* analysed the influence of mineral and synthetic oil on efficiency [5]. *Pham, Bednarczyk et al.* [6-8] analysed the effects of machining tolerances on efficiency. In the last few decades, there have been comprehensive studies in the field of defining new concepts of cycloid speed reducers [9,10], modifying the cycloid disc tooth profile [11], dynamic analysis [12, 13] and stress-strain analysis of vital elements [10, 14].

Based on the review of the literature, it can be concluded that researchers have addressed various issues in order to improve the efficiency of cycloidal speed reducers. In this regard, the aim of this paper is to compare power losses for different types of cycloid disc bearings.

Although, due to their compact design, needle bearings are most often used for cycloid discs, this paper gives a comparative analysis of the use of needle, cylindrical and ball bearings.

2 Power losses in cycloidal speed reducers

In general, power losses in cycloidal speed reducers can be divided into load-independent and load-dependent losses which occur between the elements in contact [4, 5, 15]. Apart from the operating conditions and the internal design of the housing, the load-independent losses are mainly related to the viscosity and density of the lubricant, the immersion depth of the parts in the oil bath [16], and the shaft seals (Fig. 1 - h, i). Load-dependent losses can be divided into power losses in [1, 4, 5, 15]:

- the cycloid disc bearing (Fig. 1 – a);
- the contact between the output rollers and holes in the cycloid disc (Fig. 1 – b);
- the contact of the output rollers and their pins (Fig. 1 – c);
- the contact of the central rollers and cycloid disc teeth (Fig. 1 – d);
- the contact between the central rollers and their pins (Fig. 1 – e);
- the input shaft bearings (Fig. 1 – g);
- the output shaft bearings (Fig. 1 – k, j).

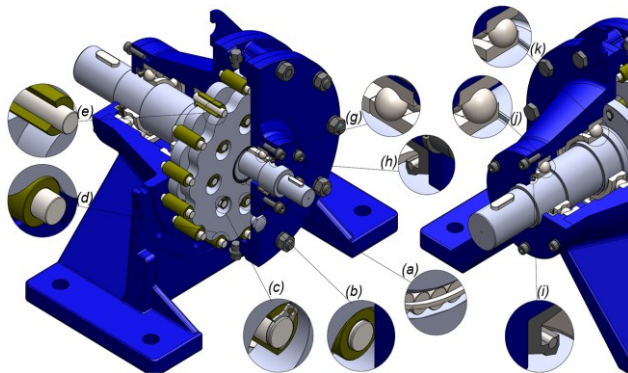


Fig. 1. Power losses in contacts between the cycloid speed reducer elements.

Total theoretical efficiency of cycloid reducer η can be calculated by usage of Malhotra's [1] equation which is based on difference in input torque of cycloid reducer $T_{in}(Nmm)$ and total work of friction forces $W_{loss}(Nmm)$ which is caused by losses in previously defined zones:

$$\eta = \frac{T_{in} \cdot 2\pi - W_{loss}}{T_{in} \cdot 2\pi} \quad (1)$$

This paper studies power losses in the cycloid disc bearing because they have a great effect on the cycloidal speed reducer efficiency. Reducing bearing friction is an absolute imperative both for manufacturers and researchers working in this field. Friction occurs regardless of the type and size of the bearing, i.e. regardless of the characteristics and amount of the lubricant. It is expressed by the frictional torque. The lost energy is mostly converted into heat [17-19].

3 Mathematical model for calculation of the cycloid disc bearing efficiency

Many authors have been engaged in research and development of efficient mathematical models for determination of the frictional torque of rolling bearings, and one of the most acceptable models was proposed by *Palmgren* [20]. He developed simple theoretical and empirical expressions for which the necessary parameters can be found in the manufacturers' catalogues and which have the widest application in engineering practice. The expressions are defined based on laboratory tests carried out under different loads, rotational speeds and using different lubricants.

According to *Palmgren*, the total frictional torque $T_{B\ loss}$ (Nmm) for ball bearings consists of a load-dependent component T_1 (Nmm) and a component dependent on the lubricant viscosity T_v (Nmm), i.e.:

$$T_{B\ loss} = T_1 + T_v \tag{2}$$

For cylindrical roller bearings, in addition to these two components, another frictional torque component is added due to the friction between the rolling elements and the flanges on the rings, i.e. [21,22]:

$$T_{B\ loss} = T_1 + T_v + T_f \tag{3}$$

According to *Schaeffer's* [22] recommendation, the expression (1) can be applied to needle bearings if factors for this type of bearing are used.

The load-dependent frictional torque component is determined by the following expression:

$$T_1 = f_1 \cdot F_\beta \cdot D_{SR} \tag{4}$$

where : f_1 – the factor dependent on the bearing type and relative load, F_β – bearing load which in the case of a cycloidal speed reducer equals the maximum eccentric cam velocity for π rotational speed of the input shaft, i.e. $F_\beta = F_{e\ max}(N)$, $D_{SR} = (d_{cz} + D_{cz})/2$ – the mean diameter of the cycloid disc bearing (mm), d_{cz} – the inner diameter of the bearing (mm), D_{cz} – the outer diameter of the bearing (mm).

Eccentricity force F_e (N) is determined as resultant of horizontal F_{eH} (N) and vertical F_{eV} (N) force component [23,24]:

$$F_e = \sqrt{F_{eH}^2 + F_{eV}^2} \tag{5}$$

Horizontal force component is determined according to equation:

$$F_{eH} = \sum_i F_{Ni} \cdot \sin x_i + \sum_j F_{Kj} \tag{6}$$

where are: F_{Ni} – normal force on i -th central roller (N), x_i – angle between meshing force on i -th ring gear roller and vertical direction ($^\circ$), F_{Kj} – output force on j -th output roller (N).

Vertical force component is determined according to equation:

$$F_{eV} = \frac{T_{in}}{e} \tag{7}$$

where is: e – eccentricity size (mm).

Since the entire amount of lubricant cannot pass between the contact surfaces of the rolling body and the raceway, only a small amount of the lubricant form a hydrodynamic film. The excess lubricant which cannot pass between the rolling surfaces becomes resistance to movement.

Therefore, the component of the frictional torque moment that results from the resistance of the lubricant to the movement of the rolling body is determined according to the expression:

$$\begin{aligned} T_v &= 10^{-7} \cdot f_o \cdot (\nu \cdot n_{in})^{2/3} \cdot D_{SR}^3 & \nu \cdot n_{in} &\geq 2000 \\ T_v &= 160 \cdot 10^{-7} \cdot f_o \cdot D_{SR}^3 & \nu \cdot n_{in} &< 2000 \end{aligned} \tag{8}$$

where: f_o – the factor dependent on the bearing type and lubrication method, ν – kinematic viscosity of the lubricant at operating temperature, and in case of grease, the predominant factor is the viscosity of the base oil at operating temperature (mm^2s^{-1}), n_{in} – input rotational speed (min^{-1}).

The component of the frictional torque resulting from the resistance between the rollers and the flanges of the ring is determined based on the following expression:

$$T_f = f_f \cdot F_a \cdot D_{SR} \tag{9}$$

where: f_f – the factor dependent on the bearing type and lubrication method, F_a – axial bearing load (N).

The overall power loss in the roller bearing $P_{B\ loss}$ (W) is determined based on the expression [22]:

$$P_{B\ loss} = T_{B\ loss} \cdot \frac{n_{in}}{9550} \tag{10}$$

4 Comparative analysis of power losses in the cycloid disc bearing

The following rolling bearings were used in the comparative analysis of power losses in the cycloid disc bearing:

- a needle roller and cage assembly – K 35x40x13;
- a cylindrical roller bearing – NU 1007;
- a deep groove ball bearing – 6007.

Therefore, for the purposes of this research, a single-stage cycloidal speed reducer with the characteristics shown in Table 1 was used. Table 2 shows the dimensions of different types of bearings and the values of the pitch circle radius R_{Oiz} of the output rollers, which depends on the bearing type. Lithium grease LGMT 2 was used as a lubricant. Its kinematic viscosity at a temperature of $40\ ^\circ C$ is $\nu_{40} = 110\ mm^2/s$ [25].

Fig. 2a shows the geometric quantities of the vital elements of the cycloidal speed reducer, while Fig. 2b shows the basic geometric differences between cycloid discs with different

bearing types. It can be noticed that needle cage bearings requires a significantly smaller central hole in the cycloid disc. The size of the central hole in the cycloid disc has a direct effect on the static and dynamic stability of the cycloid disc.

Table 1. Characteristics of the single stage cycloidal speed reducer used for experimental testing.

Characteristics	Value	Characteristics	Value
Input power, P_{EM} (W)	4	Diameter of the central roller base, d_0 (mm)	8
Reducer ratio, u_{CR}	13	Number of output rollers, u	8
Pitch circle radius of the ring gear, r (mm)	86	Output roller diameter, D_{VK} (mm)	14
Eccentric cam, e (mm)	4	Diameter of the output roller base, d_{VK} (mm)	7
Central roller diameter, D_0 (mm)	14	Diameter of the cycloid disc hole, D_{OCZ} (mm)	29

Table 2. Dimensions of different bearing types.

Quantities	K 35x40x13	NU 1007	6007
Pitch circle radius of the output rollers, R_{Oiz} (mm)	47.5	53	53
Inner diameter of the bearing, d_{CZ} (mm)	35	35	35
Outer diameter of the bearing, D_{CZ} (mm)	40	62	62
Bearing width, B (mm)	13	14	14
Dynamic load capacity, C (kN)	15,4	41,5	16,8

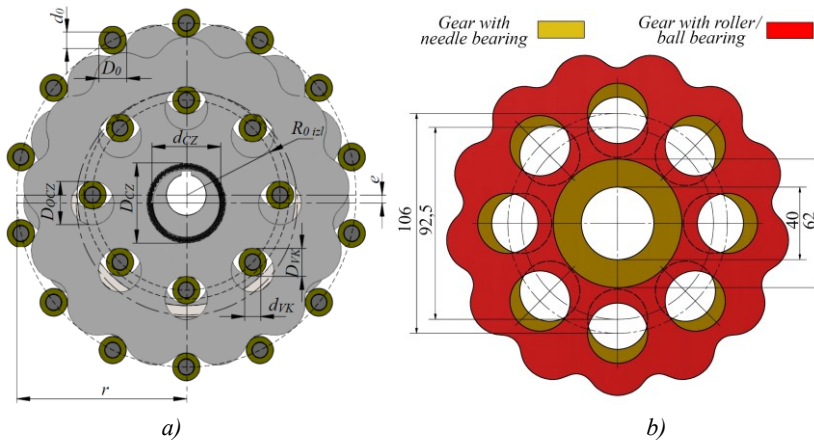


Fig. 2. Geometrical values of the vital cycloid speed reducer elements (a); Dimensions of cycloid discs with different bearing types (b).

From a financial point of view, needle cage bearings are the cheapest roller bearings in relation to the dynamic load capacity, while cylindrical roller bearings are the most expensive. However, since needle cage bearings are placed between the eccentric cam and the cycloid disc, the value of the frictional torque largely depends on the clearances between the contact surfaces, which is not the case with the ball and cylindrical bearings whose rolling bodies are placed between the inner and outer ring.

Fig. 3 shows distribution of the eccentric cam force F_e (N) on the cycloid disc determined according to Lehmann [23]. This distribution also applies to the second cycloid disc which is turned at the angle of 180° to the first one.

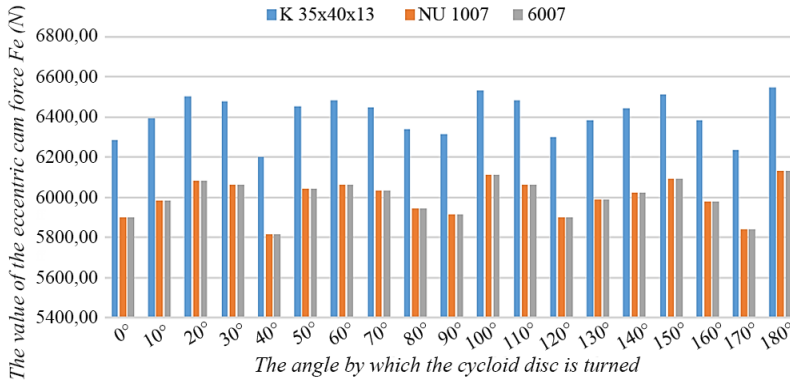


Fig. 3. Distribution of the eccentric cam force F_e (N) for the analysed cycloid speed reducer.

The presented force distribution should be considered approximate because *Lehman's* model [23] does not take into account modifications of the cycloid disc tooth profile and elastic deformations of the elements involved in the load transfer process [26].

In the theoretical case, when there are no clearances in the cycloidal speed reducer, both cycloid discs are in contact with half of the central and output rollers that participate in the load transfer process. However, in reality this is not the case because clearances exist due to manufacturing faults, in order to enable easier assembly and disassembly, better lubrication, etc. The size of the clearances directly affects the number of the rollers in the contact with the cycloid disc - the larger the clearances, the smaller the number of the loaded rollers. This causes changes in the values of the calculated forces.

Fig. 4 shows the dependence of the frictional torque $T_{B\ loss}$ (Nmm) on the rotational speed of the input shaft n_{in} (min^{-1}) for the cycloidal speed reducer with the characteristics given in Tables 1 and 2. The rotational speed of the input shaft varies in the range from 500 to 5000 min^{-1} . As the rotational speed increases, the frictional torque decreases. It reaches the highest values when the train gear starts off.

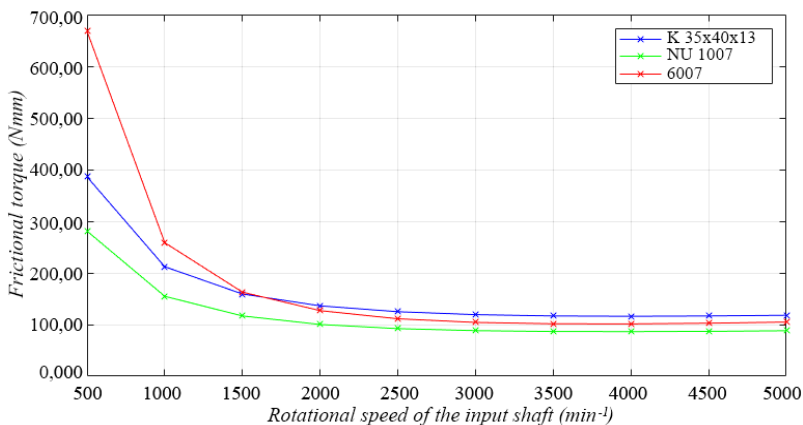


Fig. 4. Dependence of the frictional torque on the rotational speed of the input shaft.

The dependence of the power loss $P_{B\ loss}$ (W) on the rotational speed of the input shaft n_{in} (min^{-1}) is shown in Fig. 5, while the percentage differences are given in Tables 3 and 4. As in the previous simulation, the rotational speed of the input shaft varies in the range from 500 to 5000 min^{-1} . With an increase in the rotational speed, the power losses also increase, and the smallest losses occur when the gear train starts off (except for the ball bearing 6007). The values of power losses are similar for all the three types of bearings, but

the smallest losses occur with the cylindrical roller bearing NU 1007. The power losses are less with the needle cage bearing K 35x40x13 than with the ball bearing 6007 but only at low rotational speeds. As the rotational speed increases, it becomes quite the opposite.

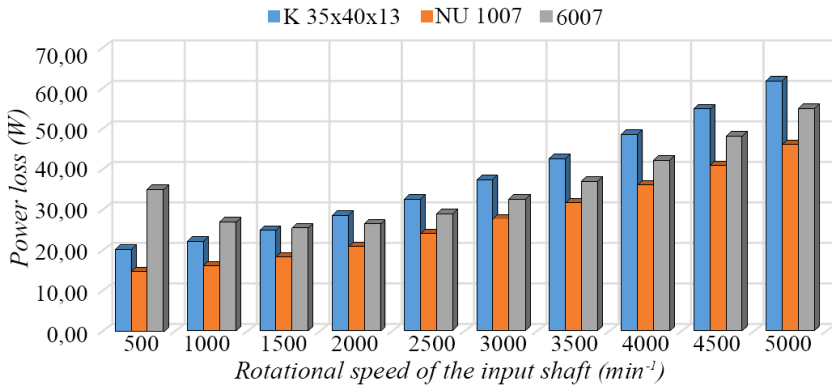


Fig. 5. Dependence of the power loss on the rotational speed of the input shaft.

Table 3. Percentage difference in the power loss between the bearings K 35x40x13 and NU 1007.

$P_{B\ loss} (W)$	500	1000	1500	2000	2500	3000	3500	4000	4500	5000
K 35x40x13	20.201	22.194	25.002	28.518	32.678	37.434	42.750	48.599	54.957	61.804
NU 1007	14.702	16.205	18.324	20.977	24.115	27.702	31.713	36.125	40.922	46.088
Difference	27.22 %	26.98%	26.71 %	26.44%	26.20%	25.99%	25.82%	25.67%	25.54%	25.43%

Table 4. Percentage difference in the power losses between the bearings K 35x40x13 and 6007.

$P_{B\ loss} (W)$	500	1000	1500	2000	2500	3000	3500	4000	4500	5000
K 35x40x13	20.201	22.194	25.002	28.518	32.678	37.434	42.750	48.599	54.957	61.804
6007	34.955	27.062	25.563	26.580	29.100	32.708	37.193	42.430	48.338	54.859
Difference	-42.21%	-17.99%	-2.19%	6.80%	10.95%	12.62%	13.00%	12.69%	12.04%	11.24%

5 Conclusion

This paper presents a comparative analysis of the power loss values in different rolling bearing types of a cycloid disc. The testing was performed on the actual cycloidal speed reducer with the power of 4 kW and the gear ratio of 13. The rotational speed varied in the range from 500 to 5000 min⁻¹.

The analysis of the results shows that the power losses are significantly lower with the cylindrical roller bearing NU 1007 than with the needle bearing K 35x40x13. The losses are 25-27% lower at all rotational speeds. With the ball bearing 6007, power losses occur at lower rotational speeds. With higher rotational speeds of the input shaft (over 1500 min⁻¹), the losses decrease. Therefore, the power losses are 6-11% lower compared to the needle cage bearing K 35x40x13. From the aspect of dynamic loading capacity, the cylindrical roller bearing NU 1007 has an extremely high value of 41.5kN, while the ball bearing 6007 (16.8 kN) and the needle cage bearing K 35x40x13 (15.4 kN) have much smaller values. Therefore, compared to other bearing types, the bearing NU 1007 has a significantly longer service life i.e. it achieves higher rotational speeds or a larger number of operating hours before material fatigue occurs. If we consider dimensions, we can conclude that the needle cage bearing K 35x40x13 requires a significantly smaller central hole in the cycloid disc

compared to the ball bearing 6007 and the cylindrical roller bearing NU 1007 which require a 35.5% larger diameter. The size of the central hole in the cycloid disc has a direct effect on the static and dynamic stability of the cycloid disc, which is the main reason the needle bearings are so widely used. In our future research we are planning to investigate static and dynamic stability of cycloid discs with different types of bearings as well as to perform a comparative analysis.

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