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Parametric Modeling of a Cycloid Drive Relative to Input Shaft Angle

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***Abstract*— Creating an equidistant of the shortened epitrochoid curve for a cycloid drive disc is a complex process. By automating this process using a parametric input curve drawing program, the geometry can be precisely defined. In this paper a cycloid drive model is created using such a program written for these purposes. The program input also creates other drive component curves and places them in their current positions relative to the input shaft rotation angle. The model generated from this geometry represents a valuable and time efficient initiate for further stress analyses of cycloid drive components. Automating this process also decreases the possibility of human error and allows for easily changeable input shaft angles.**

***Keywords*— cycloid drive, cycloid gear, parametric modeling, gear position, automated generating**

I. Introduction

Cycloid drives, which are a part of the planetary drive group, have a wide industrial application. This is mainly due to their excellent characteristics, a wide range of gear rations, high efficiency, smooth transmission, high overload capacity, compact overall size, low noise, long and reliable service life as well as suitability for frequent start-stop and reverse duty.

The key component of the cyclo drive is the cycloid disc. The cycloid disc profile is an equidistant of the shortened epitrochoid while the annular sun (central) gear has rollers instead of teeth. The cycloid gear is made with one tooth less than the number of rollers on the central gear (only some newer cycloid drive designs have two teeth less). In practice, most commonly used systems of cyclo drives have two cycloid disks which are rotated by 180º of each.

Basic information about cycloid gearing was covered by Kudrijavcev [1] as well as by Lehmann [2]. Parametric equations for equidistant of trochoid have been developed by Litvin and Feng [3]. Meshing conditions have been covered in detail by Chen, Fang, Li and Wang [4]. Computerized design for generation of surfaces and curves has been developed in [5]. An analytical model has been developed by Blanche and Yang [6] with machining tolerances to minimize backlash and torque ripple. Distribution of loads has been analysed in [7], [8], and [9]. Level of efficiency has been examined by Gorla, Davoli, Rosa, etc. in [10]. The need to cover a wide range of possible analysis on this type of drive starts with being able to create a model which is easily editable.

Stress analyses in the contact of the cycloid disc and the central rollers, for example, are most commonly performed for a few characteristic positions of the gear meshing relative to the input shaft angle. This is the case due to the complex geometry of the cycloid disc which has to be rotated around the central axis by the offset cam and around its own axis, which is a time-consuming process if done manually. By automating the modelling process it is possible to avoid the likelihood of human error as well as shortening the time needed to draw the drive, position the gear meshing in the desired contact position and edit any further iteration.

As a basis, a three-dimensional part created in Autodesk Inventor can be imported into any analysis program as reference geometry. Autodesk Inventor has the capabilities of automating the modelling process, and as such is ideal for the purposes of parametric modelling of whole cycloid drive.

II. Cycloid Drive Parameters and Profile

A cycloid is a curve which is traced by a point located anywhere on a (rolling) circle which rolls along a stationary (basic) circle. The rolling circle has a radius, R_a (1), while the stationary circle's radius is, R_b (2). For defining these radiuses it is necessary to know the pitch circle radius of the central gear, r, as well as the gear ratio, u_{rc}.

The position of the rolling circle in relation to the basic circle and the position of the rolling circle in relation to the rolling curve which traces the cycloid defines the type of cycloid form. The most commonly used tooth profile for cycloid gears is an equidistant of the shortened epitrochoid.

$$R_a = \frac{r}{u_{CR} + 1} \tag{1}$$

$$R_b = r - R_a \tag{2}$$

The equations for x and y coordinates for the equidistant of the shortened epitrochoid are given by functions (3) and (4).

$$x = (R_b + R_b)\cdot\cos\alpha + e\cdot\cos(\alpha+\beta) - q\cdot\cos(\alpha+\phi) \tag{3}$$

$$y = (R_b + R_b) \cdot \sin\alpha + e \cdot \sin(\alpha + \beta) - q \cdot \sin(\alpha + \phi) \quad (4)$$

These equations are given as a function of the angle (α) between the starting and current position of the point of contact of the basic and rolling curve relevant to the center of the base curve. Auxiliary angles, β (5) and ϕ (6), are used in these equations to simplify the calculation of the curve. They are functions of α, the radiuses of the basic and rolling circles and size of eccentricity, *e*.

$$\phi = arctg\left(\frac{\sin\beta}{R_a/e + \cos\beta}\right) \quad (5)$$

$$\beta = \frac{R_b}{R_a} \cdot \alpha \quad (6)$$

III. Parametric Modeling of Cycloid Drive

Parametric modeling of cycloid drives is done in Autodesk Inventor software. Due to easy parametric input a top-down approach to modeling the assembly was adopted. All parameters of the reducer are defined in the assembly file and are connected to the individual parts of the reducer. The way in which the parameters are defined in the assembly is shown in figure 1.

All dependent parameters are also connected through mathematical relations.

After defining parameters individual files of the assembly parts were created: cam shaft, cycloid gear, stationary roller axels, stationary rollers, output shaft with mount, and mobile roller axels and mobile rollers.

With the creation of all the individual parts, parameters which match the assembly parameter names are made. This step is required in order to later connect the script of the part parameters to the assembly parameter code.

Definitely the most time consuming part to model in this entire assembly is the cycloid gear. Its profile is created using the command for parametric function drawing based on equations for drawing the cycloid curve. The drawing of the cycloid gear profile is shown in figure 2.

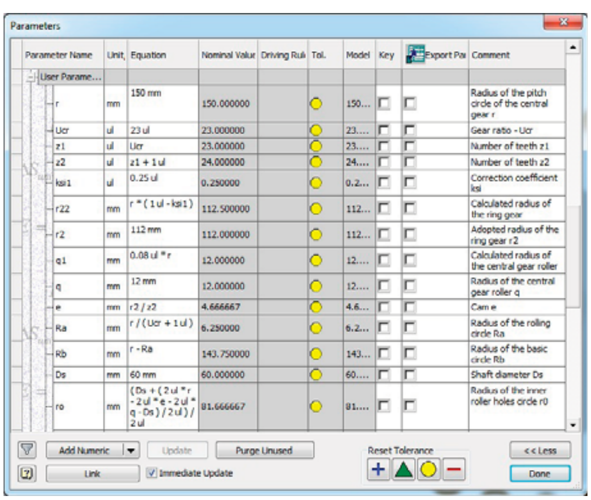

Fig. 1 Parameters definition

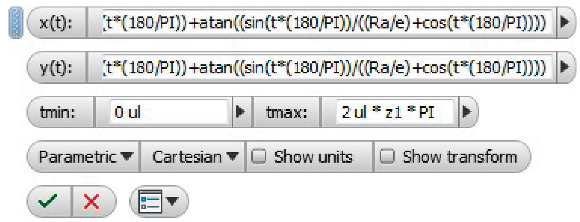

Fig. 2 Cycloid gear profile drawing

After creating the part files and naming the parameters, as they were previously named in the assembly, the parameters are connected. This step is done in the *i-logic* environment. Parameter connecting is done by creating a script for equating parameters from the part and assembly files. Scripts are created using the *Add rule* command. Figure 3 shows the code for connecting parameters of the parts with the previously created assembly parameters.

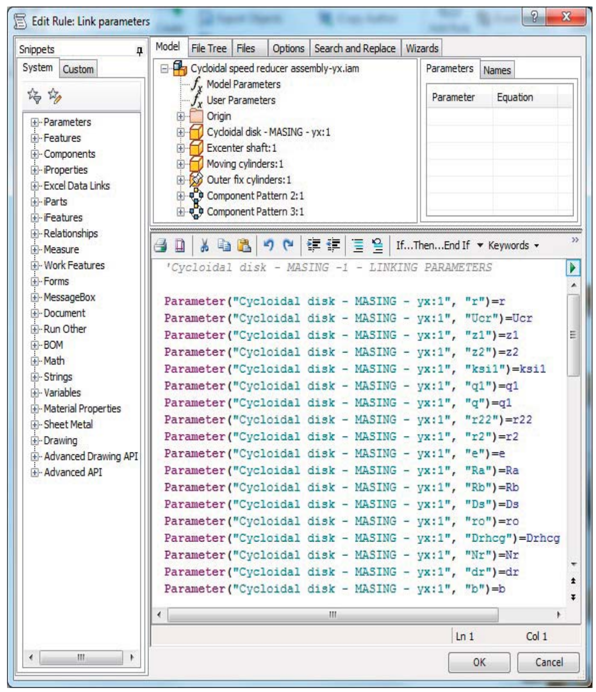

Fig. 3 Rule for linking parameters

Using this approach, a general model of a single-stage cycloid drive was created. Achieving required dimensions and reducer is possible by using values independent of parameters in the dialogs from figure 1. However, in order to speed up this process a form for controlling independent parameters of the cycloid drive was created. The benefits of implementing an input form is that there is no room for error in modifying the model. There are three tabs in the generator window for the independent elements of the cycloid drive.

The first tab is used to set dimensions and shape of the profile of the cycloid gear. The first tab content is shown in figure 4.

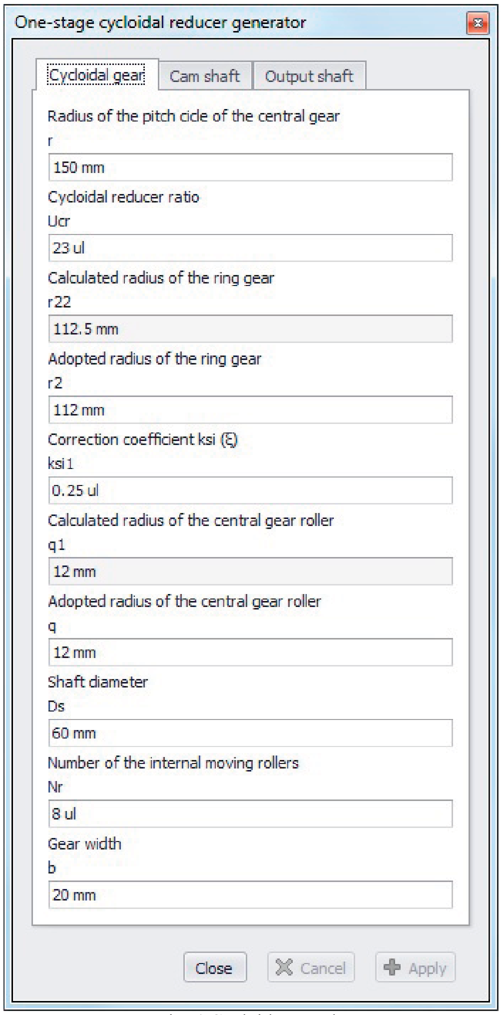

Fig. 4 Cycloid gear tab

The second tab is used for the cam shaft. Cam shaft parameters are length of cam segment, input shaft diameter. The second tab is shown in figure 5.

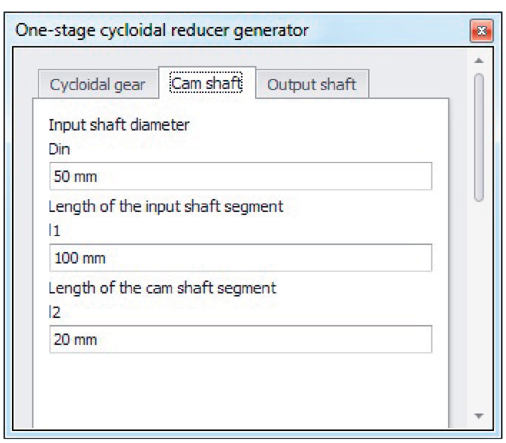

Fig. 5 Cam shaft tab

The third tab is used for setting output shaft parameters and dimensions of the mobile roller mount. In this tab the length of the output shaft and the mounting plate thickness which carries the mobile rollers. The third tab is shown in figure 6.

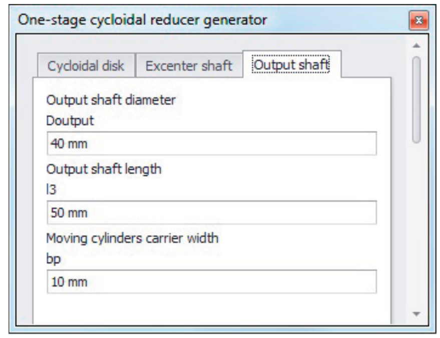

Fig. 6 Output shaft tab

IV. Results

For testing this parametric model three examples were created. Input data for test examples is given in table 1.

TABLE 1. INPUT VALUES FOR A TEST EXAMPLES

Name	Unit	Value Ex. 1	Ex. 2	Ex. 3
Cycloid gear tab				
Radius of the pitch circle of the central gear – r	[mm]	56	92	150
Cycloidal reducer ratio – U_{rc}	[-]	5	11	23
Adopted radius of the ring gear – r_2	[mm]	42	60	112
Correction coefficient - ξ	[-]	0.25	0.35	0.25
Adopted radius of the central gear roller – q	[mm]	4.5	7.5	12
Shaft diameter – Ds	[mm]	20	40	60
Number of the internal moving rollers – N_r	[-]	4	6	8
Gear width – b	[mm]	8	15	20
Cam shaft tab				
Input shat diameter – D_{input}	[mm]	20	30	60
Length of the input shaft segment – l_1	[mm]	40	60	150
Length of the cam shaft segment – l_2	[mm]	20	20	20
Output shaft tab				
Output shat diameter – D_{outout}	[mm]	30	40	80
Output shaft length – l_3	[mm]	80	50	350
Moving cylinders carrier width – b_p	[mm]	6	10	20

Figure 7 shows the model for test example 1. In example 1 the generated cycloid drive has a transmission ratio of 5.

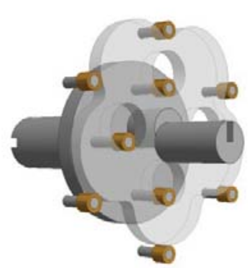

Fig. 7 Test example 1

Figure 8 shows the model for test example 2. In example 1 the generated cycloid drive has a transmission ratio of 11.

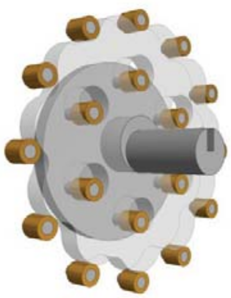

Fig. 8.Test example 2

Figure 9 shows the model for test example 3. In example 1 the generated cycloid drive has a transmission ratio of 23.

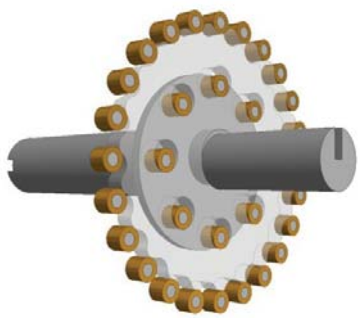

Fig. 9 Test example 3

V. Conclusion

In this paper a parametrization of a single-stage cycloid drive was conducted. A modern approach to CAD modeling was presented which significantly decreases work time. It is very easy to integrate optimization methods in this modeling approach and set a goal function depending on the intended use of the drive.

By solving this problem, the problem of modeling a multistage cycloid drive is practically solved as well. Namely, a multi-stage drive is made by connecting two or more of these drives in the same housing. By generating this model it is also shown that the assembly can be created as well as motion simulations regardless of the drive ratio and overall dimensions of the drive.

The next step in this research would be implementing optimization methods into the design process, as well as FEM analyses of gearing in specific positions relative to input shaft angles.

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