

COMPUTER AIDED DESIGNING AND MODELLING OF X-ZERO GEAR DRIVE

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The aim of the publication is the computer aided designing process of the X-zero gear drive. Determination of the geometric parameters and correlations are necessary for the designing process. A computer program has been carried out for the designing process, using which an arbitrary X-zero gear drive could be analysed and modelled. Using this program a concrete geometric gear drive is designed. The CAD models are designed for later connection, geometric and tooth contact (TCA) analysis.

Keywords: X-zero gear drive, modelling, designing, computer program

1. Introduction

Teeth are the complex of the teeth on the offset of the toothed gear. In a wider sense the teeth is the complex of one or more teeth of connection toothed gear. It is appropriate for force and motion transmission. Given transmission ratio is created between the axis of the element pairs. The X-zero gear drive is an extreme case of the X-gear drive, when addendum modification is not used [1, 4–6].

2. Designing of tooth gearing

Given the m modul and the z_1, z_2 number of teeth. Based on Fig. 1 the necessary correlations for the designing are [6]:

1) Centre distance (a):

$$a = \frac{d_1 + d_2}{2} = \frac{z_1 + z_2}{2} m. \quad (1)$$

2) Addendum (h_a):

$$h_a = m. \quad (2)$$

3) Bottom clearance (c):

$$c = c^* m, \quad (3)$$

where: $c^* = 0.25$.

4) Dedendum (h_f):

$$h_f = h_a + c. \quad (4)$$

5) Circular pitch (p):

$$p = m \pi. \quad (5)$$

Table 1. The main parameters of the toothed gearing [1, 4, 6]

| Notation | Nomination | Notation | Nomination |
|------------------|-------------------------------|------------|--|
| d_1, d_2 | Pitch circle diameter | h_f | Dedendum |
| d_{f1}, d_{f2} | Root circle diameter | h_a | Addendum |
| d_{a1}, d_{a2} | Tip circle diameter | h | Whole depth |
| d_{h1}, d_{h2} | Limit circle diameter | p | Circular pitch |
| d_{w1}, d_{w2} | Rolling circle diameter | S_{ax} | Tooth thickness |
| c | Bottom clearance | m | Modul |
| h_w | Working depth | z | Number of teeth |
| a | Centre distance | j_s | Backlash |
| d_{b1}, d_{b2} | Involute base circle diameter | α_0 | Base profile angle ($\alpha_0 = 20^\circ$) |

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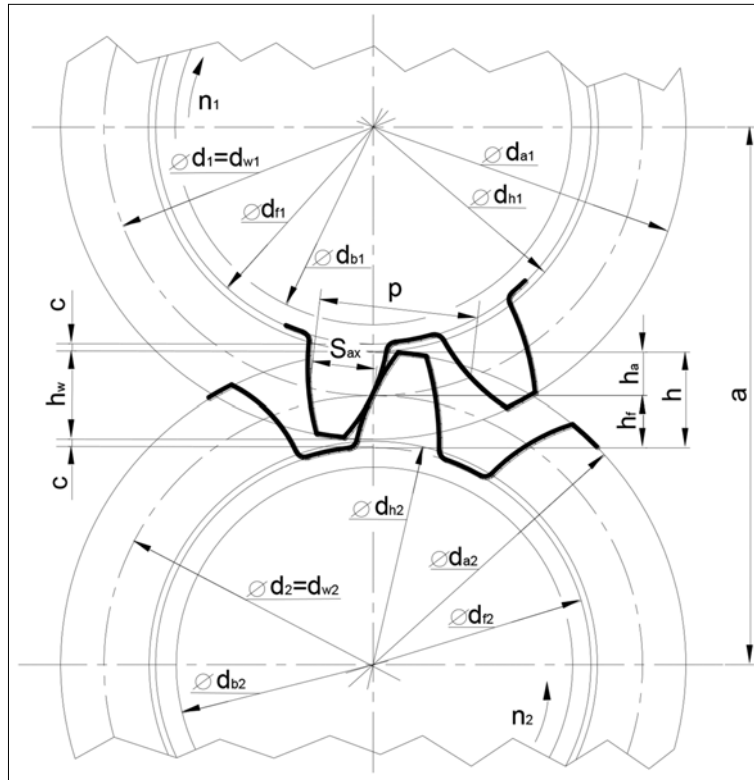


Fig. 1. The parameters of the X-zero gear drive

6) Backlash (j):

$$j = p/20. \quad (6)$$

7) Whole depth (h):

$$h = h_a + h_f. \quad (7)$$

8) Working depth (h_w):

$$h_w = 2m. \quad (8)$$

9) Pitch circle diameters (d_1, d_2):

$$d_{1,2} = z_{1,2} m, \quad (9)$$

10) Tip circle diameters (d_{a1}, d_{a2}):

$$d_{a1,a2} = d_{1,2} + 2h_a = (z_{1,2} + 2)m. \quad (10)$$

11) Root circle diameters (d_{f1}, d_{f2}):

$$d_{f1,f2} = d_{1,2} - 2h_f = (z_{1,2} - 2 - 2c^*)m. \quad (11)$$

12) Involute base circle diameters (d_{ak1}, d_{ak2}):

$$d_{b1,b2} = d_{1,2} \cos \alpha_0. \quad (12)$$

Using the Dudás-type general mathematical model [2] the necessary coordinate system arrangement is

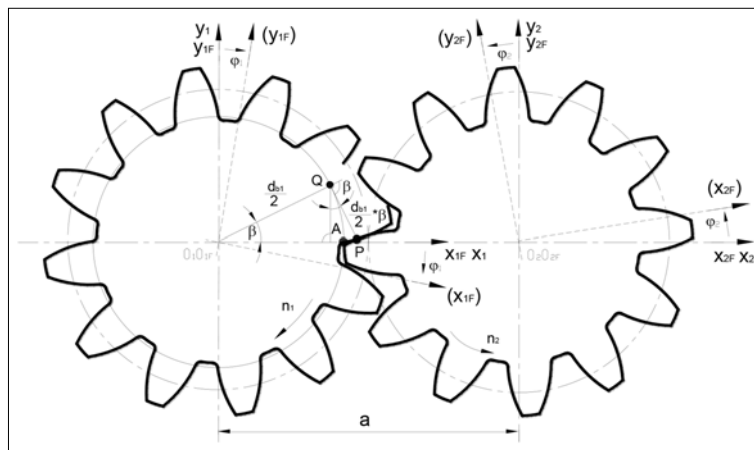


Fig. 2. The necessary coordinate system arrangement for the modelling and connection analysis

created for the modelling and connection analysis of the gear drive (Fig. 2).

The one parametric equation of the toothed gear profile, that is the equation of the circle involute, is on the K_{1F} (x_{1F}, y_{1F}) coordinate system (Fig. 2):

$$\left. \begin{aligned} x_{1F} &= \frac{d_{b1}}{2} (\cos \beta + \beta \sin \beta), \\ y_{1F} &= \frac{d_{b1}}{2} (\sin \beta - \beta \cos \beta). \end{aligned} \right\} \quad (13)$$

We are looking for the curve fixed to K_{2F} connected to $\vec{r}_{1F} = \vec{r}_{1F}(\beta)$ curve, we can use the fact that during the movement of these two curves they overlap each other and taking into consideration that

$$\varphi_2 = i_{21} \varphi_1, \quad (14)$$

we can state that this overlapping can be described by one motion parameter (φ_1) [2, 3, 5]. The transformation matrix between the K_{1F} and the K_{2F} coordinate systems is:

$$M_{2F,1F} = \begin{bmatrix} \cos \varphi_2 \cos \varphi_1 & \cos \varphi_2 \sin \varphi_1 & -a \cos \varphi_2 \\ -\sin \varphi_2 \sin \varphi_1 & +\sin \varphi_2 \cos \varphi_1 & \\ \sin \varphi_2 \cos \varphi_1 & -\sin \varphi_2 \sin \varphi_1 & a \sin \varphi_2 \\ +\sin \varphi_1 \cos \varphi_2 & +\cos \varphi_2 \cos \varphi_1 & \\ 0 & 0 & 1 \end{bmatrix}. \quad (15)$$

The relative velocity vector must be primed in the K_{2F} system. between two surfaces can be defined using the transformation matrix between the K_{1F} and the K_{2F} rotational coordinate systems in the [2, 3, 5]:

$$\vec{v}_{2F}^{(12)} = \frac{d}{dt} \vec{r}_{2F} = \frac{d}{dt} (M_{2F,1F}) \vec{r}_{1F}, \quad (16)$$

$$\frac{d}{dt} M_{2F,1F} = \begin{bmatrix} -i \sin \varphi_2 \cos \varphi_1 - \cos \varphi_2 \sin \varphi_1 & -i \sin \varphi_2 \sin \varphi_1 + \cos \varphi_2 \cos \varphi_1 & ia \sin \varphi_2 \\ -i \cos \varphi_2 \sin \varphi_1 - \sin \varphi_2 \cos \varphi_1 & +i \cos \varphi_2 \cos \varphi_1 - \sin \varphi_2 \sin \varphi_1 & \\ -i \cos \varphi_2 \cos \varphi_1 - \sin \varphi_2 \sin \varphi_1 & -i \cos \varphi_2 \sin \varphi_1 - \sin \varphi_2 \cos \varphi_1 & ia \cos \varphi_2 \\ +\cos \varphi_1 \cos \varphi_2 - i \sin \varphi_1 \sin \varphi_2 & -i \sin \varphi_2 \cos \varphi_1 - \cos \varphi_2 \sin \varphi_1 & \\ 0 & 0 & 1 \end{bmatrix}. \quad (17)$$

Taking into consideration the relation between velocity vectors of the relative motion in K_{1F} and K_{2F} coordinate systems, i.e. [2, 3, 5]:

$$\vec{v}_{1F}^{(12)} = M_{1F,2F} \vec{v}_{2F}^{(12)} \quad (18)$$

in K_{1F} coordinate system, the relative velocity vector is [2, 3, 5]:

$$\vec{v}_{1F}^{(12)} = M_{1F,2F} \frac{dM_{2F,1F}}{dt} \vec{r}_{1F}, \quad (19)$$

where the matrix of the kinematic motion mapping is [2, 3, 5]:

$$P_k = M_{1F,2F} \frac{d}{dt} (M_{2F,1F}). \quad (20)$$

The contact line between the reciprocate meshing elements can be computed solving the contact equation which expresses the I. Law of connection.

$$\vec{n}_{1F} \vec{v}_{1F}^{(12)} = \vec{n}_{2F} \vec{v}_{2F}^{(12)} = \vec{n} \vec{v}^{(12)} = 0 \quad (21)$$

and the vector-scalar function describing the tooth surface, at the same time (Fig. 3) [2, 3, 5, 6].

The equations of the tooth surface of member 2 which can be defined as the mashing surface of the group of contact lines in the K_{2F} coordinate system

$$\left. \begin{aligned} \vec{n}_{1F} \vec{v}_{1F}^{(12)} &= 0, \\ \vec{r}_{1F} &= \vec{r}_{1F}(\beta), \\ \vec{r}_{2F} &= M_{2F,1F} \vec{r}_{1F}, \end{aligned} \right\} \quad (22)$$

where the equation of the normal vector is

$$\left. \begin{aligned} n_{1Fx} &= \frac{1}{2} d_{b1} \beta \cos \beta, \\ n_{1Fy} &= \frac{1}{2} d_{b1} \beta \sin \beta. \end{aligned} \right\} \quad (23)$$

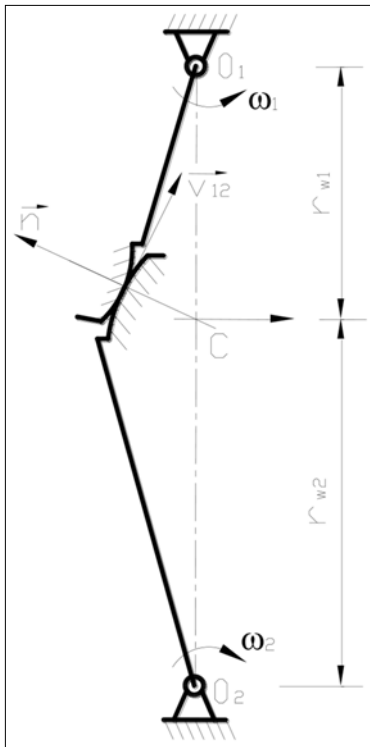


Fig. 3. Planar connection of toothed gears [6]

3. Computer-aided designing

Using the detailed mathematical correlations a computer program has been carried out for the designing of the X-zero gear drive.

The input parameters of the program are the m modul and the z_1, z_2 number of teeth. The main dimensions of the gear drive (centre distance, main circles, etc.) are calculated and the toothed gears are drawn on the screen by the program. The profile points of the toothed gear could be saved into *txt* format and imported into 3D engineering designing software (Fig. 4). Interpolating B spline surface is set on the received profile points. Body element is created from the received involut tooth profile by extrude. It is modelled along the perimeter of the given circle based on the gear number of teeth (Fig. 5).

4. Conclusion

We determined the necessary geometric correlations for the gear designing. The tooth surface of the drive gear which is connected with the power gear is defined by the direct motion mapping [2, 5].

A computer aided program was developed for the designing and modelling of X-zero gear drive. Knowing of the modul and the number of teeth the program is calculated the dimensions of the gear drive and drawn the drawing of gear drive. Importing the received points of the involut profile curve into 3D engineering designing system and setting of interpolating B spline curve into the points the CAD model of the

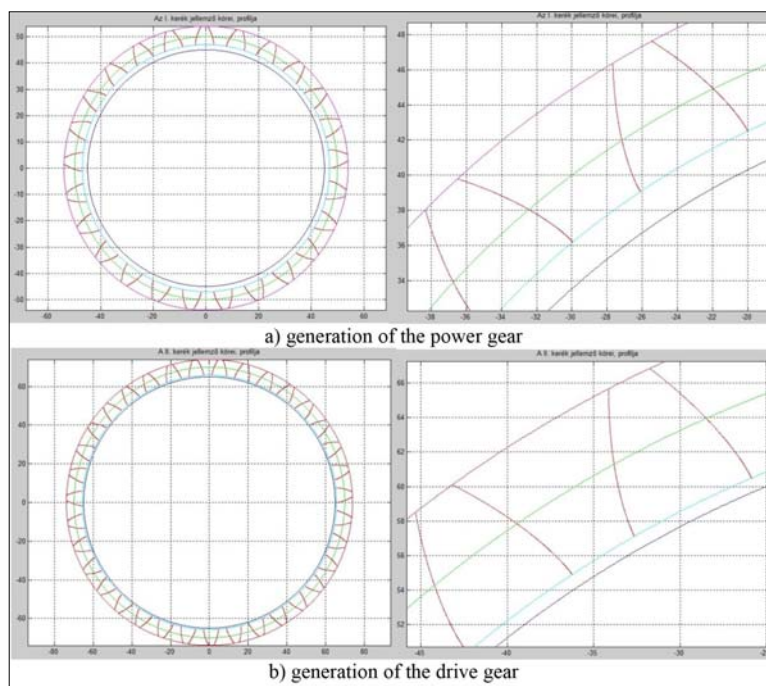


Fig. 4. Generation of a concrete geometric X-zero gear drive



Fig. 5. The CAD model of the X-zero gear drive

gear drive could be designed. This method could be used for the modelling of a concrete geometric gear drive.

Using this CAD models other connection, geometric and tooth contact analysis could be done.

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