A PARAMETRIC CAD MODEL FOR AN EXTERNAL HELICAL GEAR

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Abstract: *This article presents how the ISO design method of cylindrical gear wheels with straight or inclined teeth it can be implemented in CATIA graphics software. A precise three-dimensional model is thus obtained, which takes into account the conjugate wheel of the gear but also the possible profile displacements, on which it can be studied the influence of the characteristic mechanical stresses on these machine parts, using the finite element method. The modeling involves providing several input parameters, depending on the design specifics. The proposed model substantially reduces the design time of a new cylindrical gear, which can be successfully used in the design enterprises of these machine parts.*

Keywords: *parametric modelling, involute gear profile, helical gear, external gears*

1. Introduction

Gears are some of the most commonly used transmissions, finding them in most gearboxes for cars, or any technical equipment's aimed at transmitting the rotation movement and torque with a constant transmission ratio. These transmissions are characterized by a fairly high mechanical efficiency of about 0,955 [Budală, 2019]. The most common such transmissions are the spur or helical gear with involute teeth profile, that also called evolvent, thanks to the low technology required for production. Once with helical gear a series of shortcomings of the spur gear have been solved, such as the reduction of noise and vibrations in operation, as a result of ensuring an increased degree of coverage and at the same time an increase in load-bearing capacity.

Modeling and optimizing a cylindrical gear can take quite a long time, 150-200 hours [Jayakiran, 2018]. For this reason the use of a parameterized model for these machine parts is desirable.

Many recent works such as [Shan, 2012], [Ghionea, 2017], [Rosic, 2004] and [Gurav, 2018] propose parameterized CAD models for gears, in various specialized graphics software, especially for straight-tooth cylindrical gears, focusing on the AGMA methodology. This method is one used mainly by manufacturers, much simplified one that does not directly take into account the influence of profile displacements or the characteristics of the conjugate gear wheel.

The present work aims to achieve a parameterized design, with a precise as possible values of the geometric dimensions of the helical gear teeth with involute profile, following the ISO design method, within the CATIA software. A more precise model is required as a result of the fact that the dimensional and shape accuracy directly influences the results of the analysis of the maximum stresses and deformations of any parts, study that can be carried out in CATIA Analysis & Simulation module.

2. Geometry of cylindrical gears

Each gear wheel is characterized in a first instance by an integer number z_n of teeth that actively participate in the gearing process in order to obtain the target gear ratio. In order to be able to engage correctly, it is necessary that the wheels within the same gear stage have the same diametric pitch on the reference circle *p*, which is also called the normal wheel module m_n. The main parameters used to define the geometry of the outer evolventic cylindrical gear are shown in Fig. 1.

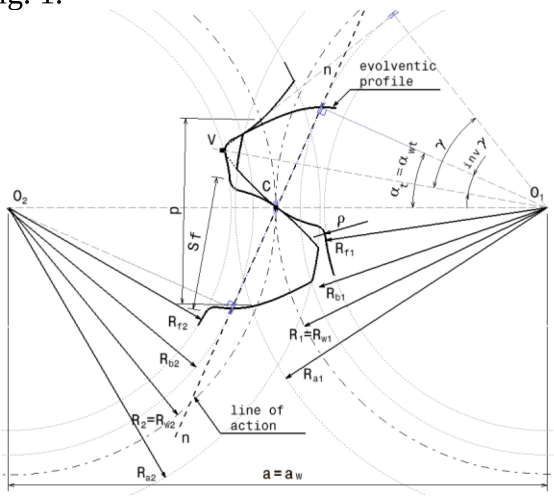

Figure 1 : *Cylindrical gear wheels geometry parameters*

The defining parameter most often used as the input data is the reference distance between the axes *a*. This size can be calculated in the case of cylindrical gears with inclined teeth with Eq.1, what takes into account the total number of gear teeth, the normal modulus of the wheels and the teeth inclination angle on pitch circle *β*.

$$a = \frac{m_n \cdot (z_1 + z_2)}{2 \cdot \cos(\beta)} \qquad [mm] \qquad (1)$$

Although the toothed wheel is defined by a certain number of teeth, according to the technical drawing rules, she is represented through the diameters of its characteristic cylinders, of which is most importantly is the pitch cylinder, described by the *R* circle radius. Depending on this radius, a series of important circles are calculated what characterizes the constructive sizes of the gear wheels such as: base circle R_b, tip circle R_a or root circle R_f.

When we talk about a zero shifted gear, the values of the operating pitch circle radius R_w and reference pitch circle radius *R* overlap. Also, in this particular case, the front real pressure angle α_{wt} is identical to the front angle of the reference profile α_t.

The contact between two flanks of the teeth wheels will constantly evolve along the line of action *n-n*, on which the gear pole *C* is located.

3. Involute spline modeling

The profile of the wheels teeth is represented by mutually winding curves, among which is the involute one. This profile is most often used due to the multiple advantages of which we mention:

- a reduced sensitivity to small changes in center distance;
- relatively low production costs;
- ensuring a minimum slip between teeth flanks;
- allowing the production of teeth with profile shifts in order to optimize transmission capacity and mechanical strength.

The evolution profile is framed between the base circle and the tip circle of the tooth. A radius connection *ρ* described by Eq.2 is made between the base circle and the root circle.

$$\rho = 0{,}38 \cdot m_n \qquad [\text{mm}] \qquad (2)$$

In order to have a full dimensional constraint of the flank profile, within the proposed model, the γ angle corresponding to the peak of the tooth was defined. The V point, which corresponds to this peak, is the intersection of extensions of the two flanks profiles for the same tooth and is placed on the tooth bisector in the front plane according to Fig. 1. In this way, the involute of this *γ* angle is calculated, with which the thickness of the tooth profile on the tip circle will be imposed.

The way of generating the involute approached in CATIA is shown in Fig. 2. This involves obtaining a curve described by a point A, which belongs to the common normal line n-n and which is always tangent to the base circle of the wheel, considered fixed. The line performs a non-slip rolling motion that involves compliance with the condition in Eq.3.

$$TA_0 = TA \qquad (3)$$

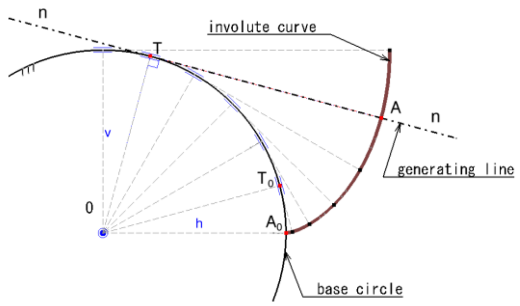

Figure 2 : *Involute curve construction*

4. Constructing and parameterization of the geometric elements of the gear

The modeling process starts by generating the tooth profile. In this sense, an important step is to sketch the profile of the tooth to be extruded. The defining dimensions values of the geometric elements will be calculated with an accuracy of at least five decimal places and are implemented by means of parameterization variables defined with formulas or laws from the Knowledge Advisor CATIA module. Considering how the tooth involute curves is graphical obtain, in the Sketcher module, the circles of the characteristic cylinders of the gear wheels are represented as in Fig. 3. On the circumference of the base circle, a series of points have been established, which are joined by construction lines with the common center of these circles, with respect to the condition that these points are equidistant. Thus, strings of equal lengths are obtained. Through each point on the circumference of the base circle are built tangent segments to this circle. The size of these tangent segments being equal to the length of the resulting circle arc, calculated with Eq.4, in the case of T_1A_1 segment. The following segments will be multiplied by the number of the position related to the segment from the construction.

$$L_b = R_b * \sin(inv(\alpha_{wt})) \quad [\text{mm}] \quad (4)$$

in which:

$$\alpha_{wt} \begin{cases} \alpha_t & if \quad a = a_w \\ arc\cos(\cos(\alpha_t)\cdot \dfrac{a}{a_w}) & otherwise \end{cases} \quad (5)$$

$$\alpha_t = \arctan\left(\frac{\tan(\alpha_0)}{\cos(\beta)}\right) \quad [\text{deg}] \quad (6)$$

$$inv(\alpha_{wt}) = \tan(\alpha_{wt}) - \alpha_{wt}\cdot\frac{\pi}{180} \quad [\text{rad}]\,(7)$$

The pressure angle of the reference rack is denoted by α_0.

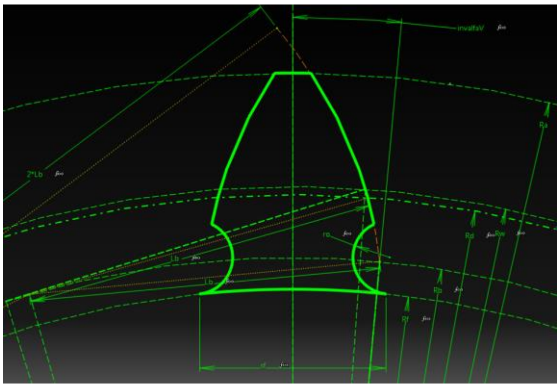

Figure 3 : *Tooth profile sketch*

Another parameter used to constrain the sketch from the Fig. 3 is the s_f, length of the front arc on the root circle of the tooth, the calculation relationship being given by the Eq. 8.

$$s_f = 2\cdot R_f\cdot\left[\frac{\left(\frac{\pi}{2}+2\cdot\chi_{t1}\cdot\tan(\alpha_t)\right)}{z_1}+inv(\alpha_t)-inv(\alpha_f)\right] \quad [\text{mm}]\,(8)$$

where:

$$\alpha_f = \arccos\left(\frac{R_d}{R_f}\cos(\alpha_t)\right) \quad [\text{mm}] \quad (9)$$

$$\chi_{t1} = \frac{0,03(30-\min(z_1,z_2))}{\cos(\beta)} \quad (10)$$

The pinion displacement coefficient of in front plane is χ_{t1}. For most of the parameters defined in the modeling of the gear wheel it was necessary the use of conditioning functions, as shown in Fig. 4. Therefore, by using a programming languages characteristic syntax, the constructive dimensions are established.

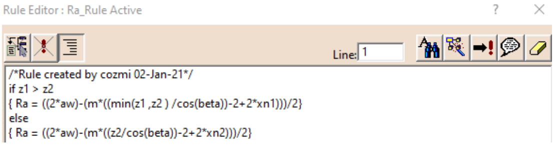

Figure 4 : *Addendum circle radius defining*

The use of rules in parameterization process has been imposed as a result of the fact that the developed model takes into account the shift profile for both the modeled and the conjugated gear wheel and is also able to generate proper spur or helical gear.

4. Results and Conclusions

Figure 5 shows details of the two wheels of a helical gear, consisting of a 17-teeth pinion and 93-teeth driven wheel, with a standardized distance between the axles higher than the center distance, which leads to a positive shifted gear.

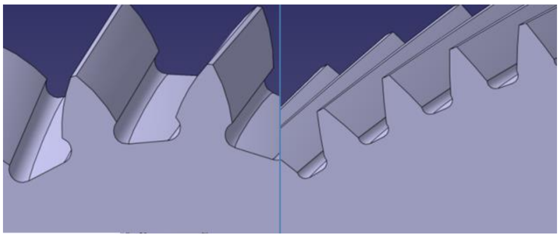

Figure 5 : *Profile detail for positive shifted gear*

By increasing the number of driven wheels teeth with one and keeping all other input parameters, we will get a negative shifted profile for this wheel. For comparison, Fig. 6 shows a detail view of the two new modeled gear wheels profiles. In the case of the driven wheel, there is clearly seen a reduction in tooth thickness foot, concomitantly with the thickening of the tooth head.

The article highlights the possibilities and necessity of using parameterized design for this machine parts.

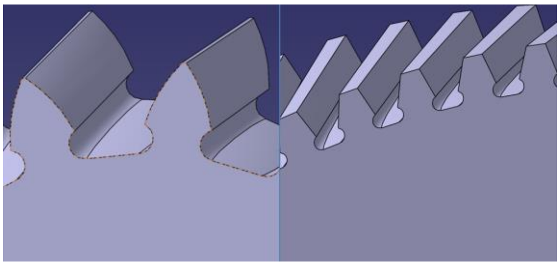

Figure 6 : *Profile detail positive shifted for pinion and negative shifted for the driven wheel*

By simply changing the input parameters a new gear can be studied

A precise gear model, which is taking into account the influence of the shifted profile and the constructive dimensions of the conjugated wheel, has been developed.

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