ON THE COMPUTATIONAL DESIGN METHODS FOR IMPROVING THE GEAR TRANSMISSION PERFORMANCES

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Abstract: The computational design permit to have the possibility of choosing the geometrical gears parameters not in relation with the standard gear rack but in relation with the request of the beneficiary. The parametrical design permit also to study what are the better values of the designing parameters for obtain the optimal solution for the functional parameters of the transmission. The paper present some aspects about the facilities offered by some original computer application, results of the author’s research, to optimise the non-standard gears, considered as a solution for improving the gears transmission quality. It became very easy to design, for the same initial date, symmetric gears, direct or inverted asymmetric gears, find the better for the objective function.

Key words: computational, direct, gear, design, asymmetric

1. INTRODUCTION

Involute gears are the most frequent because of the already known advantages. Asymmetric gears characterized by asymmetric involutes profiles of the tooth are used only in the last few years. Because the great numbers of geometrical parameters, in relation with the classical gears, designing and manufacture those special gears it is more difficult. By using the computational method this inconvenient can be solved without significant supplementary costs [5], [6]. Asymmetric gears, formed of gear wheels with asymmetric involutes profile teeth, add further advantages like an increase in the load capacity, a decrease in weight, a reduction in vibrations, an increase in efficiency and durability [4].

The ratio between the diameter of the base circle of the involutes inactive profile and the diameter of the base circle of the involutes active profile represents the “asymmetry coefficient of the tooth” [3]. In relation with those most important parameter can be defined: the direct asymmetric gear, for which the “direct” profile is the active one and the asymmetry coefficient is higher than 1; the inverted asymmetric gear, for which the “inverted” profile is the active one and the asymmetry coefficient of is lower than 1 and the symmetrical gear with the asymmetry coefficient equals one. It has been called direct profile the profile with a high
gearing angle, and the profile with a low gearing angle it have been called inverted profile. There are also other designing parameters, the angles and the number of the generation gear racks, which influence the geometrical and functional parameters of the designed transmission.

2. APPLICATION AREA

In the following are presented part of results obtained with original applications in Matlab and AutoLisp developed for designing, modelling and studying geometrical and functional parameters of the non-standard spur gears with involutes asymmetrical profiles of the teeth and also for direct design of the symmetrical gears, that have been considered only a particular case of those asymmetrical. All the designed gears that can be obtained represent non-standard gears because the applications use the „direct design” of the spur gears. That offers the possibility of determining first the parameters of the gears, following by the determination of the asymmetric gear rack’s parameters on the base of those of the gears.

The vector of the designing parameters or variables is:

\[
\overline{X} = (x_1, x_2, x_3, x_4, x_5) = (\alpha_{wd}, \alpha_{wi}, f, cr, var),
\]

where: \(\alpha_{wd}\), \(\alpha_{wi}\) are the pressure angle on the direct respectively inverted profile of the gear; “f” called coefficient of modification the gear rack angle determine the difference between \(\alpha_{wd}\) and the gear rack direct profile angle \(\alpha_{dc}\); \(cr = \{1,2\}\) establish design with single or two generation gear rack and the last \(var = \{1,2\}\) is in relation with the using of the gear as direct or inverted gear.

A set of variables generate an asymmetric gear for which can be determined the geometrical parameters. Introducing these parameters in the equation of the involutes and filet profiles result the teeth profiles (figure 1 - with Matlab applications), can be obtained the 2D and 3D models of the gears (figure 2 – with Autolisp applications), some functional parameters mentioned before and on the base of those parameters can be evaluate the performances of the transmission and easy compared with another possibly solutions.

![Fig. 1. Pinion and gear teeth profiles examples](image-url)
The applications developed in AUTOLISP for AUTOCAD permit the rapid representation of the asymmetric gears and the finite element method, being verified in many applications, can be considered as a method for determine but more for verify and confirm the results obtained with the MATLAB application, that is more efficient and easily to use.

3. MATLAB APPLICATIONS FACILITIES

The behaviour of the designed asymmetric gear under the load one can rapid evaluate by the MATLAB applications, many routines that permit to approach the following aspects [2]: mathematical modelling of the teeth and of the meshing of the asymmetric gears; the elasticity, implicitly the rigidity, of the pairs of teeth in contact and the variation for these parameters during the meshing cycle; on the base of the tooth elasticity it can be solved the statically unknown problem of load distribution between the two pairs of teeth in meshing and so it was determined the diagram of variation of the normal force; the relative sliding speed and the variation during the meshing period; the instantaneous power loses, the variation of the instantaneous efficiency and than the medium efficiency for a meshing period ; the bending stress at the bottom of the tooth in relation with the number of the contact point; the variation during the meshing cycle of the bending stress it was so resulting; the contact stress has been determined also for every the contact point and represented the diagram of variation for a meshing cycle.

For all the mentioned parameters have been established the calculus algorithms, the mathematical relation to obtain the values corresponding for a number of contact point of the line of action and from these the significant values.

For example the contact stress has been determined as a function depending on the pressure angle corresponding to the contact point on the pinion active profile:

$$\sigma_{Hjd,i} = \sqrt{\frac{0.3 \cdot E_1 \cdot E_2 \cdot F_{hjd,i} \cdot \tan\alpha_{wd,i} \cdot (1+u)}{(E_1 + E_2) \cdot b \cdot r_{bld,i} \cdot \tan\alpha_{jld,i} \cdot \tan\alpha_{wd,i} \cdot (1+u) - \tan\alpha_{jld,i}}}$$

(2)

where:

$E_1, E_2$ are the elasticity modules of the pinion and gear materials;
\( F_{njd,i} \) (\( F_{njd}, F_{nji} \)) is the normal force on the tooth profile corresponding to the “j” point of contact (for direct gear, for inverted gear);
\( u = z_2 / z_1 \) is the ratio of the number of teeth of the gear and pinion;
\( r_{b1d,i} \) (\( r_{b1d}, r_{b1i} \)) is the radius of the base circle of the active profile of the pinion (for direct gear, for inverted gear);
\( \alpha_{j1d,i} \) (\( \alpha_{j1d}, \alpha_{j1i} \)) is the pressure angle corresponding to the “j” contact point on the pinion active profile.

Another important facilities offered by the developed applications is the rapid comparison between the significant functional parameters corresponding to a number of possible solutions for the same initial date. In the table 1 are given examples of such comparison diagrams [1] that indicate the combination of designing parameters that increase the transmission performances.

**Table 1. Functional parameters comparison diagrams**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( z_1 )</td>
<td>15, 60</td>
</tr>
<tr>
<td>( z_2 )</td>
<td>120 mm</td>
</tr>
<tr>
<td>( n )</td>
<td>1000 rot/min</td>
</tr>
<tr>
<td>( P )</td>
<td>18 kW</td>
</tr>
<tr>
<td>( \alpha_{wd} )</td>
<td>([30^\circ, 32^\circ, 34^\circ, 36^\circ, 38^\circ])</td>
</tr>
<tr>
<td>( \alpha_{wi} )</td>
<td>20(^\circ)</td>
</tr>
<tr>
<td>( f )</td>
<td>([2, 4, 6, 10])</td>
</tr>
</tbody>
</table>

Variation of the maximum bending stress to the pinion in relation with gear rack direct profile angle for different values of the coefficient of asymmetry

Variation of the maximum bending stress to the gear in relation with direct gear rack profile angle for different values of the coefficient of asymmetry

4. AUTOLISP APLICATIONS FACILITIES
The geometric parameters which are necessary to introduce in the first line of the AUTOLISP application, are: the numbers of the teeth for the pinion and for the gear $z_1$, $z_2$; the centre distance $a$; the pressure angles on the direct and inverted profiles $\alpha_{wd}$, $\alpha_{wi}$; the profile angles of the asymmetric gear rack $\alpha_{dc}$, $\alpha_{ic}$; the profile angles on the addendum circles, on the direct teeth profiles, for the pinion and for the gear $\alpha_{a1d}$, $\alpha_{a2d}$; the tip radius of the gear rack $R_1$ (or $R_1$, $R_2$ in the case of generation with two different racks) and the rack shift for the pinion $X_1$. In the table 1 are given for examples the results obtained with finite element method analysis for the asymmetric gears with the numbers of teeth $z_1 = 16; z_2 = 57$, the centre distance $a = 120 \text{mm}$, the mesh angles, $\alpha_{wd} = 40^\circ; \alpha_{wi} = 20^\circ$, the angles of the gear rack profiles equals with the mesh angles $\gamma = 0$, the generation of pinion and gear with one gear rack, for $P = 18 \text{ kW}$, $n = 1000 \text{ rot/min}$, $b = 30 \text{ mm}$ and in table 3 the comparison between gears with different coefficient of asymmetry, different mesh angles on the asymmetric profiles [2].

**Table 2. Stress distribution obtained with FEA**

|---------------|---------------|---------------|---------------|---------------|---------------|---------------|

The stress distribution for the direct asymmetric gear

for point Bd of the line of action AdEd

for point Dd of the line of action AdEd

The stress distribution for the inverted asymmetric gear

for point Bi of the line of action AiEi

for point Di of the line of action AiEi
5. CONCLUSION

The computer applications permit to obtain in very short time many possible solutions for designing a non-standard gear, comparing the performances and establishing the request combination of designing parameters and implicitly the geometrical parameters. Having the 2D model of the gear any design engineer can to obtain the 3D model, as drawing file, and that can be change in initial graphics exchange file. So in the soft of coordinate manufacturing machine, the obtained data will be transformed in machine language and thus all the data necessary for manufacturing are provided.

6. REFERENCES