TOOTH CONTACT ANALYSIS OF A DESIGNED PLANETARY GEAR DRIVE FOR THE VEHICLE INDUSTRY

Abstract: A planetary gear drive consists of a sun gear, planet pinions and an internal gear. We designed a complex gear system which is usable in the field of the vehicle industry into the automatized robots. The system was designed by GearTeq software which is connected with the SolidWorks designer software. After the assembly and the motion simulations tooth contact analysis (TCA) was made to analyse the normal stresses and the normal deformations on the connecting surface of the planet pinions and the internal gear by different load moments.

Key words: Planetary gear drive, CAD, TCA, normal stress, normal deformation, analysis.

1. INTRODUCTION

The planetary gear drives have two gear systems. The axis of the first system is fixed where the planet gears can rotate around it. The planet carrier can also rotate around it. The axes of the second system are assembled into the planet carrier and their teeth can connect with the first system. These planet pinions can rotate around their axes and the fixed axes of the first system [1, 3-5, 7, 9]. The overall mechanism show a similar motion as the Earth moves around the Sun (two rotation motions around two axes). The sun gear is the central gear which has a fix axes. The planet gears can do two rotation motions parallely. The internal gear is fixed. The planet gears are rotated by the sun gear and they are connected with the internal gear (Figure 1) [1, 3-5, 7, 9].

Fig. 1. The theorem of the planetary gear drive [1]

Considering the function of the gear system the sun gear can be pinion or gear. The planet pinions can also be pinions or gears. The yellow and green axes are not connected. The connection between them depends on the gear ratio (Figure 1) [1, 3-5, 7, 9].

2. THE GEOMETRIC DESIGN OF A GEAR SYSTEM

The geometric design process [3, 6-9] was created by the GearTeq software [2] with which different type of gear pairs can be designed (Figure 2). After knowing of the output geometric parameters the CAD models can be created by SolidWorks software (Figure 3).

Fig. 2. Geometric design by GearTeq software

Fig. 3. The geometric establishment of the designed planetary gear drive

The calculated geometric parameters can be seen on Table 1, 2 and 3. After the assembly and the motion simulations the TCA can be determined.
### Table 1. Geometric parameters of the internal gear

| SYMBOL | VALUE | UNIT | TERM |
|--------|-------|------|------|
| t      | 5.8832 | mm   | Tooth |
| a      | 4     | mm   | Addendum |
| b      | 5     | mm   | Dedendum |
| p      | 12.566 | mm  | Circular Pitch |
| m      | 4     |     | Modular Pitch |
| en     | 20    | deg  | Normal Pressure Angle |
| e      | 20    | deg  | Pressure Angle |
| h      | 0     |      | Helix Angle |
| Gear Data |       |      |       |
| np     | 360   | mm   | Pitch Diameter |
| dp     | 304   | mm   | Pitch Diameter, Normal |
| do     | 80    | mm   | Major Diameter |
| dbn    | 67.658| mm   | Base Diameter, Normal |
| TIF    | 312.317| mm | True Involute Form Diameter |
| ht     | 9     | mm   | Whole Depth |
| dpn    | 67.658| mm   | Base Diameter |
| TIFb   | 312.317| mm | True Involute Form Diameter |
| h      | 0     | mm   | Addendum |
| k      | 0     |      | Addendum Modification Coefficient |
| db     | 285.667| mm | Base Diameter, Normal |
| dbn    | 285.667| mm | Base Diameter, Normal |
| TIF    | 312.317| mm | True Involute Form Diameter |
| h      | 9     | mm   | Whole Depth |
| dp     | 304   | mm   | Pitch Diameter |
| dpn    | 304   | mm   | Pitch Diameter, Normal |
| do     | 314   | mm   | Major Diameter |
| db     | 296   | mm   | Minor Diameter |
| a      | 4     | mm   | Addendum |
| b      | 5     | mm   | Dedendum |
| t      | 5.7312| mm  | Tooth |
| tn     | 5.8832| mm  | Tooth |
| TIF    | 67.658| mm  | Base Diameter, Normal |
| ht     | 9     | mm   | Whole Depth |
| dp     | 304   | mm   | Pitch Diameter |
| dpn    | 304   | mm   | Pitch Diameter, Normal |
| do     | 314   | mm   | Major Diameter |
| db     | 296   | mm   | Minor Diameter |

| SYMBOL | VALUE | UNIT | TERM |
|--------|-------|------|------|
| Pd     | 6,350 |      | Pitch Diameter |
| m      | 4     |      | Modular Pitch |
| en     | 20    | deg  | Normal Pressure Angle |
| e      | 20    | deg  | Pressure Angle |
| h      | 0     |      | Helix Angle |
| C      | 116   |      | Center Distance |
| C      | 15    | mm   | Center Distance Extension |
| C      | 116   | mm   | Center Distance Backlash |
| LWA    | 13.233| mm   | Approach Length |
| SW     | 9.092 | mm   | Recess Length |
| n      | 1.885 |      | Contact Ratio |

**Coarse Pitch Involute 20deg Standard**

- **Pd**: Normal Pitch Diameter
- **m**: Modular Pitch
- **en**: Normal Pressure Angle
- **e**: Pressure Angle
- **h**: Helix Angle
- **C**: Center Distance
- **LWA**: Approach Length
- **SW**: Recess Length
- **n**: Contact Ratio

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**Hunting Data**

- **Ring Gear, Follower01**
- **np**: Number of Teeth
- **dp**: Pitch Diameter
- **do**: Major Diameter
- **db**: Base Diameter, Normal
- **TIF**: True Involute Form Diameter
- **ht**: Whole Depth
- **dp**: Pitch Diameter
- **dpn**: Pitch Diameter, Normal
- **do**: Major Diameter
- **db**: Base Diameter
- **TIF**: True Involute Form Diameter
- **ht**: Whole Depth
- **dp**: Pitch Diameter
- **dpn**: Pitch Diameter, Normal
- **do**: Major Diameter
- **db**: Base Diameter

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**AGMA-Q7 AGMA Quality Class**

- **np**: Number of Teeth
- **dp**: Pitch Diameter
- **dpn**: Pitch Diameter, Normal
- **do**: Major Diameter
- **db**: Base Diameter

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**Pinion Data**

- **Planet, Revolving, Follower01**
- **np**: Number of Teeth
- **dp**: Pitch Diameter
- **dpn**: Pitch Diameter, Normal
- **do**: Major Diameter
- **db**: Base Diameter

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**Pinion RPM**

- **RPM**: Hunting Tooth Frequency
- **F5**: Face Width

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**Hunting Data Ring Gear Follower01**

- **dp**: Pitch Diameter
- **do**: Major Diameter
- **db**: Base Diameter, Normal
- **TIF**: True Involute Form Diameter
- **ht**: Whole Depth
- **dp**: Pitch Diameter
- **dpn**: Pitch Diameter, Normal
- **do**: Major Diameter
- **db**: Base Diameter

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**Hunting Tooth Frequency**

- **6,75pm**: Hunting Tooth Frequency

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**Hunting Mesh Cycle**

- **1,2**: Hunting Mesh Cycle

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**Hunting Common Factors**

- **B0**: Normal Backlash

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**Not Hunting**
Table 2. Geometric parameters of the planet pinions

| SYMBOL | VALUE | UNIT | TERM |
|--------|-------|------|------|
| Mn     | 80,813 | mm  | Measurement Over Pins |
| M     | 80,494 | mm  | Measurement Over Pins-Minimum |
| s     | 0     |     | Span Over Teeth |
| t     | 0     |     | Number of Teeth to Span Over |
| A     | 150,361 | mm  | Base Diameter |
| B     | 150,361 | mm  | Base Diameter, Normal |
| C     | 150,361 | mm  | Circular Pitch |
| D     | 153,335 | mm  | True Involute Form Diameter |
| M     | 0     |     | Addendum |
| b     | 0     |     | Addendum |
| c     | 0     |     | Addendum Modification Coefficient |
| d     | 0     |     | Addendum Modification |
| dbn   | 150,361 | mm  | Base Diameter |
| dw    | 6,667 | mm  | Pin Diameter |
| dp    | 160 | mm  | Pitch Diameter |
| dpn   | 160 | mm  | Pitch Diameter, Normal |
| do    | 168 | mm  | Major Diameter |
| dr    | 150 | mm  | Minor Diameter |
| M1    | 68,891 | mm  | P2 Outer Composite Form Diameter |
| M2    | 68,503 | mm  | P2 Outer Form Diameter |
| M3    | 68,491 | mm  | P2 Form Diameter |
| n     | 0     |     | Span Over Teeth |
| k     | 0     |     | Number of Teeth to Span Over |
| A-M   | 0.10922 | mm | Max Runout |
| B-M   | 0.03048 | mm | Pitch Variation |
| C-M   | 0.04066 | mm | Profile Tolerance |
| D-M   | 0.0635 | mm | Tooth to Tooth Composite Tolerance |
| E-M   | 0.1236 | mm | Total Composite Tolerance |
| B-M   | 0.05842 | mm | Tooth Alignment Tolerance |
| M1    | 80,811 | mm  | AGMA Quality Class |
| M2    | 80,494 | mm  | AGMA Quality Class |
| M3    | 80,494 | mm  | AGMA Quality Class |

Table 3. Geometric parameters of the sun gear

3. TOOTH CONTACT ANALYSIS

The aim of the TCA is to determine and analyse the mechanical parameters into the tooth connection zone by different loads [3, 4]. In our establishment, the sun gear is the pinion that is why it was loaded by different moments. The gear materials are steel (E=210 GPa, ν=0.3, isotropic elasticity). Coordinate systems are defined into the rotation axes...
of the gears and the contact zones between the teeth.

The mesh method is tetrahedrons. Body of influence sizing type is defined into the contact zone to enhance the accuracy of the calculation process. The element size is 0.4 mm into the contact zone.

3.1. TCA between the sun gear and the planet pinion

The sun gear is loaded by different moments (40 – 80 Nm, step: 10 Nm). The effect of the load moment is analyzed on the tooth surface of the planet pinion. The mesh distribution can be seen on Figure 4.

![Fig. 4. The mesh for connection analysis between the sun gear and the planet pinion](image1)

- a) M=40 Nm
- b) M=50 Nm
- c) M=60 Nm
- d) M=70 Nm
- e) M=80 Nm

![Fig. 5. The distribution of the normal stress on the surface of the planet pinion](image2)

The results of the normal stress on the tooth surfaces of the planet pinions can be seen on Figure 5.

![Fig. 6. The results of the normal stress in the function of the moment on the surface of the planet pinion](image3)

The results of the average normal stresses in the function of the moment can be seen on Figure 6. The more the load moment, the more the normal stress on the tooth surface of the planet pinion.

The results of the normal deformations into the ‘x’ direction on the tooth surfaces of the planet pinion can be seen on Figure 7.
The results of the average normal deformations in the function of the moment can be seen on Figure 8. The more the load moment, the more the normal deformation on the tooth surface of the planet pinion.

3.2. TCA between the planet pinion and the internal gear

Considering the gear ratio between the sun gear and the planet pinions, the moments have to be recalculated for the planet pinions since these gears are connected with the internal gear. The calculated moments can be seen on Table 4.

| Sun gear | Planet pinions |
|----------|----------------|
| 50 Nm    | 18 Nm          |
| 60 Nm    | 22.5 Nm        |
| 70 Nm    | 27 Nm          |
| 80 Nm    | 31.5 Nm        |
| 90 Nm    | 36 Nm          |

Table 4. The moments on the sun gear and the planet pinions accordingly the gear ratio

The effect of the load moment is analysed on the surface of the internal gear. The meshing strategy is similar than the previous case (Figure 9).
c) $M = 27\text{ Nm}$

d) $M = 31.5\text{ Nm}$

e) $M = 36\text{ Nm}$

The results of the normal stress on the tooth surfaces of the internal gear can be seen on Figure 10.

The results of the average normal stresses on the tooth surface of the internal gear in the function of the moment can be seen on Figure 11. We got lower stress values since the load moments were lower due to the gear ratio. It is also true the stress is higher if we increase the moment.

The results of the normal deformations ('x' directional) on the tooth surfaces of the internal gear can be seen on Figure 12.
The results of the average normal deformations in the function of the moment can be seen on Figure 12. We got much lower results than in case of the previous analysis. The reason is the gear ratio, the lower moment and the mass. It is also true that increasing the load moment on the planet pinion the normal deformation is also increasing on the tooth surface of the internal gear.

4. CONCLUSION

The vehicle industry is a big filed in the countries that contains two huge fields: vehicle design and vehicle manufacturing. There are more and more vehicles on the roads, consequently the development and the research on this field is actual.

In this study, we designed a complex planetary gear box which is usable in the robotic systems for the vehicle manufacturing.

The geometric parameters was calculated by the help with the GearTeq software. After that, the results could be imported into the SolidWorks three dimensional designer software where the assembly and the motion analysis could be done.

The aim of the TCA is to analyze the mechanical parameters into the tooth connection zone of the gear pairs by different loads. In our case, the load was the moment on the pinions. Firstly, we analyzed the TCA parameters between the sun gear and the planet pinion. Four planet pinions were used around the perimeter of the sun gear. Secondly, we analyzed the same parameters between the planet pinion and the internal gear. In this case, the moments had to be recalculated accordingly the gear ratio from the sun gear, which is the pinion, to the planet pinions, which are intermediary gears. We made diagrams from the results and evaluated the overall analysis. This analysis process is necessary to control the correctness and the function of such gear systems before the real installation into the machines [8].

5. REFERENCES

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