SIMULATION AND ANALYSIS OF TRANSMISSION ERROR IN HELICAL NON CIRCULAR GEAR MODEL

Dr. SABAH KHAN

Department of Mechanical Engineering, Faculty of Engineering and Technology, Jamia Millia Islamia, New Delhi-25

ABSTRACT

Gears are one of the most critical components in mechanical power transmission systems. Transmission error is considered to be one of the main contributors to noise and vibration in a gear set. Transmission error measurement has become popular as an area of research. To estimate transmission error in a gear system, the characteristics of involute gears were analyzed using ANSYS. The contact stresses were examined using 2-D FEM models. The bending stresses in the tooth root were examined using a 3-D FEM model.

1. INTRODUCTION

Non Circular gears are used by manufacturers of automatic machines in all applications where variable speed rotary motion is required from a constant speed input. Typical applications include machines in the printing and textile industry where a controlled in-feed is required with a rapid return. Elliptical gears are commonly used in packaging and conveyor applications etc (13). Elliptical gears can be manufactured in a large range of materials, including aluminum, steel, bronze and stainless steel and some company design and manufacture non-standard gears for special applications. Generation of planar and helical elliptical gears by application of rack-cutter, hob, and shaper (1-2).

The most important criteria for proper maintenance and operation of a gear pair is the level of their noise and bearing contact. A main source of gear noise is transmission errors. Traditional methods of gear synthesis that provide conjugate gear tooth surfaces with zero transmission errors and an instantaneous line of contact are not acceptable for the gearing industry due to errors of manufacturing and assembly. Taking into account such errors, the bearing contact is shifted to the edge of the tooth, and transmission errors of an unfavorable shape occur. The new trend of gear synthesis is to localize bearing contact and absorb the transmission errors.
A pinion and gear with perfect involute profiles running with no torque should theoretically run with zero transmission error. However, when these same gears transmit torque, the teeth deflect causing the gear to “lag” behind the pinion. The difference in angular position between where the gear is in relation to the pinion as opposed to where it should theoretically be is what is termed transmission error. This error in position occurs during each mesh cycle, or when each new tooth pair comes in and goes out of contact. This constant variation in the rotational position of the gears is normally converted to linear units (micronches or microns) so that gears of varying sizes can be compared. Even though the error is relatively small, it is these slight variations which can cause noise and vibration. Under normal operating conditions, the main source of vibration excitation is from the periodic changes in tooth stiffness due to non-uniform load distributions from single to double contact zone in each meshing cycle of the mating teeth. This indicates that the variation in mesh stiffness can produce considerable vibration and dynamic loading of gear teeth in mesh.

Predicting the static transmission error (TE) is a necessary condition for the reduction of the noise radiated from the gearbox. To obtain TE the contact problem was seldom included because the nonlinear problem made the model too complicated. This paper deals with estimation of static transmission error including the contact problem and the mesh stiffness variations of spur gears. For this purpose, an FEA numerical modelling system has been developed. For spur gears a two dimensional model can be used instead of a three dimensional model to reduce the total number of elements and the total number of nodes in order to save computer memory. This is based on a two dimensional finite element analysis of tooth deflections (4-5). Two models were adopted to obtain a more accurate static transmission error, for a set of successive positions of the driving gear and driven gear. Two different models of gear pairs have been generated to analyze the effects of gear body deformation and the interactions between adjacent loaded teeth.

It is generally accepted that the noise generated by a pair of gears is mainly related to the gear transmission error. The main source of apparent excitation in gearboxes is created by the meshing process. Researches usually assume that transmission error and the variation in the gear mesh stiffness are responsible for the noise radiated by the gearbox (9-10).

The term transmission error is used to describe the difference between the theoretical and actual relative angular rotations between a pinion and a gear. Its characteristics depend on the instantaneous positions of meshing tooth pairs. Under load at low speeds (static transmission error) these situations result from the tooth deflections and manufacturing errors. In service, the transmission error (7) is mainly caused by:

- **Elastic deformation:** local contact deformation from each meshing tooth pair and the deflections of the teeth because of bending and shearing due to transmitted load.
- **Tooth geometry errors:** including profile, spacing and run out errors from the manufacturing process;
- **Imperfect mounting:** geometric errors in alignment, which may be introduced by static and dynamic elastic deflections in the supporting bearings and shafts.

2. **CALCULATION OF STRESSES USING LEWIS EQUATION**

The stresses are calculated by Lewis formula using standards of AGMA. The various coefficients, such as the dynamic factor, are set at 1.1(11-12). Here analysis of gears with different numbers of teeth is carried out. First, the number of teeth is 16. The meshing helical gear has pitch radii of 60.5mm and a pressure angle of 20 degree. The face width =10 mm, transmitted load is 1020 N.
Then,

\[ P_d = \frac{N}{d} = \frac{16}{121} = 0.132 \text{ m m}^{-1} \]

\[ \sigma_v = \frac{F_t \times P_d \times K_a \times K_s \times K_w}{b \times y \times K_i} \]

\[ = \frac{1020 \times 0.132 \times 1.1 \times 1.1 \times 1.1 \times 1.3}{10 \times 0.5 \times 1.5} = 24.54 \text{ M P a} \]

If the number of teeth is changed to 19 and the load applied is 1050 N then,

\[ P_d = \frac{N}{d} = \frac{19}{121} = 0.148 \text{ mm}^{-1} \]

\[ \sigma_v = \frac{F_t \times P_d \times K_a \times K_s \times K_w}{b \times y \times K_i} \]

\[ = \frac{1050 \times 0.148 \times 1.48 \times 1.1 \times 1.1 \times 1.1 \times 1.3}{10 \times 0.5 \times 1.5} = 28.33 \text{ M P a} \]

The above calculations of the Von-Mises stresses on the teeth were carried out in order to know if they match the results from ANSYS. The results are shown in Table 4.1. This result match in the present study, effective methods to estimate the root bending stress by ANSYS software are proposed. The accuracy of the present method for the bending stresses, 3-D models were built in this chapter. The results with the different numbers of teeth and different load conditions were used in comparison.

<table>
<thead>
<tr>
<th>Number of Teeth</th>
<th>Stress 3D (ANSYS)</th>
<th>Stress (LEWIS EQ.)</th>
<th>Difference in Stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>16</td>
<td>21</td>
<td>24.5</td>
<td>3.5</td>
</tr>
<tr>
<td>19</td>
<td>28.45</td>
<td>28.33</td>
<td>0.12</td>
</tr>
</tbody>
</table>

It can be observed that the difference between the ANSYS stress value and that calculated by Lewis equation is much less than 5% and thus it can be analyzed that the analysis So the analysis of helical elliptical gear using ANSYS software is good enough for stress analysis.

3. **ANALYSIS OF TRANSMISSION ERROR OF HELICAL ELLIPTICAL MODEL**

Under normal operating conditions, the main source of vibration excitation is from the periodic changes in tooth stiffness due to non-uniform load distributions from single to double contact zone in each meshing cycle of the mating teeth. This indicates that the variation in mesh stiffness can produce considerable vibration and dynamic loading of gear teeth in mesh.
3.1 ANALYSIS OF GEARS HAVING 18 TEETH

Fig 3.1 Helical elliptical gear model under meshing condition having 18 teeth

Fig 3.2 Helical elliptical gear model under meshing condition having 18 teeth (meshed)

Fig 3.3 Total deformation of the gear teeth under meshing condition
Fig 3.4 Directional deformation of the gear teeth under meshing condition

Fig 3.5 Equivalent stress of the gear teeth under meshing condition

Fig 3.6 Maximum principal stress of the gear tooth under meshing condition
3.2 ANALYSIS FOR 19 TEETH

![Fig 3.7 Helical elliptical gear model under meshing condition having 19 teeth](image1)

![Fig 3.8 Helical elliptical gear model under meshing condition having 19 teeth (meshed)](image2)

![Fig 3.9 Total deformation of the gear tooth under meshing condition](image3)
Fig 3.10 Directional deformation of the gear tooth under meshing condition

Fig 3.11 Equivalent stress of the gear tooth under meshing condition

Fig 3.12 Total deformation of the gear tooth under meshing condition
TABLE 3.1: Results calculated from meshing condition of gears having 18 teeth

<table>
<thead>
<tr>
<th>Object name</th>
<th>Total deformation (m)</th>
<th>Directional deformation (m)</th>
<th>Equivalent stress(Pa)</th>
<th>Maximum principal stress(Pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum value</td>
<td>0</td>
<td>-5.639e^-7</td>
<td>7.936</td>
<td>4.90e^6</td>
</tr>
<tr>
<td>Maximum value</td>
<td>1.368e^-6</td>
<td>5.3241e^-7</td>
<td>1.0639e^7</td>
<td>-2.25e^6</td>
</tr>
</tbody>
</table>

TABLE 5.2: Results calculated from meshing condition of gears having 19 teeth

<table>
<thead>
<tr>
<th>Object name</th>
<th>Total deformation (m)</th>
<th>Directional deformation (m)</th>
<th>Equivalent stress(Pa)</th>
<th>Maximum principal stress(Pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum value</td>
<td>0</td>
<td>6.0409e^-7</td>
<td>1.3741e^-7</td>
<td>1.1807e^-7</td>
</tr>
<tr>
<td>Maximum value</td>
<td>4.9255e^-6</td>
<td>-2.001e^-6</td>
<td>0</td>
<td>-3.3885e^-6</td>
</tr>
</tbody>
</table>

5.3 CONCLUSION

The FEA model is used to simulate contact between two bodies accurately by verification of contact stresses between two helical elliptical gears in contact and comparison is made with the results of AGMA analysis. The average difference of results between ANSYS and AGMA approach are very small and equal to 1.71%. It was shown that FEA model could be used to simulate bending between two bodies accurately. The transmission error for two sets of meshing helical elliptical gears are analyzed for transmission error.

REFERENCES

3. Igor Zrebski, TadeuszSalanciski “Designing of Non-circular Gear" Warsaw University of Technology, Faculty of Production Engineering; Narbutta Str.8502524 Warsaw, Poland.
5. Jen -Yu Liu and Yen-Chuan Chen, "A Design for the Pitch curve of Noncircular Gears with Function Generation", Jen-Yu Liu is with the Power Mechanical Engineering Department, National Forsoma University, 64 Wen-Hwa Road, Hu Wei 63208, Yunlin, TAIWAN
7. Faydor L. Litvin, "Design and investigation of Gear drives with Noncircular gears applied for speed variation and generation of functions" Department of Mechanical and Industrial engineering, University of Illinois at Chicago, United States.

9. E. Mikhailov, Russia V. Tarabarin, “Models of gears with variable transmission ratio”, Bauman Moscow state Technical University Moscow, Russia.


12. Zeping Wei, “Stresses and Deformations in involute Spur Gears by Finite Element Method” Department of Mechanical Engineering University of Saskatchewan Saskatoon, Saskatchewan.


