Experimental vibration phenomena of spur gears with tip modifications were studied in this research. Experiments of spur gear pairs with various tip modifications were performed on a universal gear rolling tester under a fixed applied torque and different rotational speeds. The vibration signals acquired by an accelerometer were then analyzed and processed by using several signal processing techniques. Finally, the correlation between the gear vibration magnitudes and the tip modification parameters were studied and discussed based on the experimental results.

Keywords: spur gear, vibration experiments

1. Introduction

Gears are indispensable component in vehicle power transmission. In recent years, due to the rising need of electrical car industry, the issues of reducing gear vibration and noise were once again concerned. Because the power source of an electric car is an electric motor, which has a very low running noise compared to fuel-powered engine. Therefore, the noise of gear meshing becomes relatively obtrusive and troublesome.

Gear would inevitably possessing manufacturing errors and assembly errors. In addition, depending on the tooth surface modifications, the tooth profiles may be different from the standard involute profile. Under the influence of these error interactions, the so-called transmission errors (TE) are likely to be generated during the meshing process. According to Smith’s research, the TE of gear is directly related to the vibration and noise of gear meshing [1]. Therefore, reducing the TE is one of the important goals in gear design.

In practice, in order to avoid tooth tip-interference and to reduce the sensitivity of assembly errors, tooth profile modification (tip-relief/root-relief) and lead modification are often applied on gear tooth profiles. However, if the amount of modification is too large, it may cause an opposite effect. The design parameters of tip-relief modification include the amount of tip-relief and length of tip-relief. The recommended values for the parameters of tip-relief are described in some standards, including ISO 6336, BS 436-2 and AGMA 109.16 [2-4].

In the study of dynamic characteristics of gears, Houjoh et al. analyzed the vibration of gear meshing with contact errors and found out the relationship between gear errors and vibrations [5]. An estimation method of frequency response function and gear meshing excitation with vibration experiments and simulations was also developed [6-7]. In addition, Kahrman et al. also carried out a series of research of the dynamic characteristics of gear meshing with different modification by performing both simulations and experiments [8-10].

In this study, two spur gear pairs with different tip-relief modifications were used in several experiments under a light load and various rotational speeds. The vibration signals were acquired by an accelerometer. Based on the signal filtering theory and Fast Fourier Transform (FFT), the acceleration...
amplitudes at meshing frequencies were estimated from the gear vibration signals. Finally, the effects of tip-relief modifications on the vibration behaviour were discussed.

2. Experimental setup

In this study, the universal gear rolling tester was used to perform the vibration test, as shown in Fig. 1. During the vibration experiments, a PCB accelerometer was mounted on the bearing seat of the driven axle to measure the vertical vibration signals. The vibration signals of spur gear pairs with different tip-relief modifications were acquired by an accelerometer under various meshing conditions of a fixed applied torque and different rotational speeds. The vibration signals were transmitted to the computer through a data acquisition card. Then, the dynamic characteristics of gear meshing were estimated and compared.

![Universal gear rolling tester.](image)

Figure 1: Universal gear rolling tester.

3. Dynamic analysis

3.1 Experimental Planning and Design

As Table 1 shows, there are two pairs of spur gears with different tip-relief modifications in this study: 13L and 13S. Each pair consists of a driving pinion and a driven gear. The numbers of teeth of these gears are all 30. The grade of tooth profile accuracies of the gears are all JIS 0. The naming rules are listed below:

- The first two digits represent the amount of tip-relief. For example, “13S” means that the amount of tip-relief is 13 μm.
- The third letter represents the length of tip-relief. “L” means long tip-relief and “S” means short tip-relief.
- The forth letter in the gear numbers indicates whether there is a lead-crowning applied on the tooth profile. “M” and “U” represent modified and unmodified, respectively.

<table>
<thead>
<tr>
<th>Cases</th>
<th>13L</th>
<th>13S</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear No.</td>
<td>13LU</td>
<td>13SU</td>
</tr>
<tr>
<td>Number of teeth</td>
<td>30</td>
<td>30</td>
</tr>
<tr>
<td>Module</td>
<td>2.5</td>
<td>2.5</td>
</tr>
<tr>
<td>Face width (mm)</td>
<td>24</td>
<td>24</td>
</tr>
<tr>
<td>Pressure angle (degree)</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>JIS grade</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Amount of tip-relief (μm)</td>
<td>13</td>
<td>13</td>
</tr>
</tbody>
</table>

Table 1: Parameters of test gear.
In terms of operating conditions setting, the load of gears were all fixed at 10 N-m due to the limitation of the gear rolling tester. Detailed operating information is listed in Table 2. In Table 2, “13L-5” means that the gear pair of case “13L” was tested at 500 RPM. From the speed limit of the universal gear rolling tester and the time-frequency diagram, the maximum speed should not exceed 1000 RPM and the inherent resonance frequencies of the gear tester are around 100 Hz to 150 Hz, 200 Hz and 450 Hz to 800 Hz. Therefore, the operational speeds in the range of 500 RPM to 800 RPM (i.e., meshing frequency, from 250 Hz to 400Hz) were considered for the dynamic experiments in this study.

<table>
<thead>
<tr>
<th>Length of tip-relief (mm)</th>
<th>4.82</th>
<th>4.82</th>
<th>2.41</th>
<th>2.41</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amount of lead-crowning (μm)</td>
<td>-</td>
<td>12</td>
<td>-</td>
<td>12</td>
</tr>
</tbody>
</table>

Table 2: Operating information of dynamic test.

<table>
<thead>
<tr>
<th>Load</th>
<th>RPM</th>
<th>500</th>
<th>600</th>
<th>700</th>
<th>800</th>
</tr>
</thead>
<tbody>
<tr>
<td>10 N-m</td>
<td>13L-5</td>
<td>13L-6</td>
<td>13L-7</td>
<td>13L-8</td>
<td></td>
</tr>
<tr>
<td>10 N-m</td>
<td>13S-5</td>
<td>13S-6</td>
<td>13S-7</td>
<td>13S-8</td>
<td></td>
</tr>
<tr>
<td>Meshing frequency (Hz)</td>
<td>250</td>
<td>300</td>
<td>350</td>
<td>400</td>
<td></td>
</tr>
<tr>
<td>Rotational frequency (Hz)</td>
<td>8.3</td>
<td>10.0</td>
<td>11.7</td>
<td>13.3</td>
<td></td>
</tr>
</tbody>
</table>

3.2 Results of dynamic experiments

Acceleration signals of gear meshing under various speeds were acquired according to the dynamic experimental setup described in previous section. In order to avoid the high frequency and irrelevant system noise, the first step of signal processing is to apply a low-pass filter to the signal (i.e., low frequencies are passed, and high frequencies are attenuated). Then, the acceleration amplitudes at meshing frequencies were estimated by FFT. The acceleration amplitudes at meshing frequencies can be used as the basis for determining the vibration levels during gear meshing.

Figure 2 and Fig. 3 show the time-domain signals of cases “13L” and “13S” that were low-pass filtered, respectively. The abscissa and ordinate of Figs. 2 and 3 represent the time (in seconds) and acceleration (in meter/second²), respectively. After applying FFT, Fig. 4 and Fig. 5 depict the frequency-domain signals of cases “13L” and “13S”. The abscissa and ordinate of Figs. 4 and 5 represent the frequency (in Hz) and amplitude (in m/s²), respectively. The maximum amplitudes of each rotational speed are at 250/300/350/400 Hz, i.e. the meshing frequencies under each rotational speed. Finally, the acceleration amplitudes at meshing frequencies were recorded for discussion. The meshing frequencies of these experiments with various speeds were summarized in Table 2.
Figure 2: Time-domain vibration signals of cases “13L”.

Figure 3: Time-domain vibration signals of cases “13S”.
Figure 4: Frequency-domain signals of cases “13L”.

Figure 5: Frequency-domain signals of cases “13S”.
Table 3 summarized all acceleration amplitudes at various meshing frequencies of the two testing gear pairs (13L and 13S) under various rotational speeds. Figure 6 depicts the acceleration amplitudes at the meshing frequencies. In Fig. 6, the blue line and red line represent the calculated acceleration amplitudes at meshing frequencies in long tip-relief cases “13L” and short tip-relief cases “13S”, respectively. The results show that the long tip-relief cases “13L” were affected by large tip-relief and has a larger acceleration amplitude at meshing frequencies. However, the short tip-relief cases “13S” are successful in reducing the acceleration amplitude at meshing frequencies by avoiding the interference of tooth-tip. The acceleration amplitudes of cases “13S” of each rotational speed are all obviously smaller than those of cases “13L”.

In addition, a sudden surge of acceleration amplitudes occurred at rotational speed 500 RPM (i.e., meshing frequency, 250 Hz). The reason is that there is an inherent resonance frequency of the gear rolling tester around 250Hz. Therefore, the acceleration amplitude at 500 RPM was obviously higher than those at other rotational speeds. Furthermore, as the rotational speed raising from 600 RPM to 800 RPM, the acceleration amplitudes at meshing frequencies also get higher.

Table 3: Acceleration amplitudes at meshing frequencies under various conditions.

<table>
<thead>
<tr>
<th>Acceleration amplitude at meshing frequency (m/s²)</th>
<th>Load</th>
<th>Gear No.</th>
<th>RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10 N-m</td>
<td>13L</td>
<td>500</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>3.740</td>
</tr>
<tr>
<td></td>
<td></td>
<td>13S</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>2.437</td>
</tr>
</tbody>
</table>

Figure 6: Line graph of acceleration amplitudes at meshing frequencies (Experimental).

4. Conclusion

The purpose of this study is dynamic analysis of spur gear with different tip-relief modifications by low-speed vibration experiments. Firstly, an accelerator was mounted on the bearing seat to acquire the vibration signal during gear meshing on the universal gear rolling tester. After signal processing such as filtering and FFT, the acceleration amplitudes of gear meshing with various conditions and gear pairs were estimated. The experimental results showed that different tip-relief modifications on the gear teeth would cause different vibration phenomena.

The acceleration amplitudes of gear meshing of cases “13L” are higher than those of cases “13S” under all rotational speeds, due to the fact that the length of tip-relief was too large. An inadequate length of tip relief may cause higher noise and vibration. While avoiding the interference of tooth-tip, the cases “13S” are successful in reducing the vibration levels at meshing frequencies. Moreover, the trend of the experimental results is consistent with the simulation results in previous work. The experiments performed in this study can be applied to the dynamic analysis of gears with different profile modifications.
REFERENCES