GEAR TOOTH MODIFICATION FOR NOISE REDUCTION IN AUTOMOTIVE TRANSMISSIONS

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ABSTRACT

A method for gear tooth modification for an automotive transmission is introduced to reduce gear noise that is caused by the impact of gear teeth. The major causes of tooth impact are the elastic deformation of the gear teeth and shafts of the transmission under loading and the errors in the manufacturing of gears. A theoretical shape of an objective tooth profile for avoiding tooth impact is derived from the magnitudes of the elastic deformation of the gear teeth and shafts and from the overall manufacturing errors in gears. The objective tooth profile function is converted to conventional data forms that are usually communicated to commercial gear measuring and inspection instruments. The proposed method applies to the transmission for a four-wheel-drive automobile. The gear noise is reduced to a sound level of eight dB.

Keywords: Gear Tooth Modification, Automotive Transmission, Gear Noise, Tooth Impact, Objective Tooth Profile Function.

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1. INTRODUCTION

Various studies have been carried out for developing technologies for satisfying many customers of automobiles who demand both high performance in terms of power, steering safety and gasoline mileage and luxuries such as convenience, safety, and comfort. To satisfy the requirements of high strength, small size, and lightness of weight for high-powered and efficient engines and high degree of convenience and comport, measures are urgently needed to reduce the vibration and noise of automotive transmissions. The severe noise that is sensed by passengers inside an automobile is the gear whine noise, which is generated by an excitation source of the gear system. This excitation source is the magnitude of the deviation from the ideal tooth profile in the mating gears and has a frequency band between 300 Hz and 3,000 Hz [1]. The gear whine noise is caused by a complicated mechanism, such as tooth impact and double-contact of mating gears due to gear manufacturing errors and deformations under loading. Hence, we investigate the process of generation and methods for the reduction of gear whine noise.

Several optimal modification schemes [2~4], including gear-profile modification and crowning, have been presented. Maruyama et al. [5] first published their work by using the concept of transmission error (TE) to reduce noise and vibration and to increase the tooth strength of automotive transmissions. Du et al. [6] studied the effects of the TE in terms of the variation of the tooth-body stiffness of loaded spur gears. Munro [7] determined that TE that is due to the tooth deformation of a loaded spur gear is the major source of noise and vibration; he expressed the measured TE as a time and frequency domain response. Munro et al. [8] studied the optimal quantities and ranges of the tooth profile modification in the profile direction for avoiding tooth impacts due to the tooth deformations of loaded spur gears. By manufacturing a test model, Kubo [9] investigated the relationship between gear noise and the profile-error curves by measuring TE for the curves that were modularized for four types of transmission gear. By utilizing a gear mesh analysis and a kinematical analysis for the TE that ensues from profile-manufacturing errors, Mark [10] formulated the TE by considering the pitch error. Graber [11] carried out a comparative analysis of both experimental and analytical results for the TE of spur gears and helical gears with several tooth-profile modifications and gearing rates. Lee et al. [12] formulated the tooth-profile modification curve by considering profile-manufacturing errors and the elastic deformation of the spur gear tooth; he also verified the transmission error of the gear system with modified teeth. Choi et al. [13] studied the relationship between the TE that is due to tooth stiffness and the deformation of spur and helical gears under loads. Rao et al. [14] presented a method for minimizing transmission error in helical gears.

By the use of the concept of optimal modification on the plane of action for generating the new tooth profile, an idea of the perfect plane of action for helical gears is introduced. Studies on the design of gear tooth profiles have been actively carried out but most studies focus on designs that are applicable only under restricted conditions. In addition, the theoretical analysis for determining the optimal extent of the tooth profile modification for reducing the gear whine noise is insufficient.

In this paper, by the consideration of the optimal quantities of the gear tooth modification in the profile and lead directions, a method for determining the modification quantity of the objective tooth profile, which is formulated as a B-spline curve, is proposed. In addition, to verify the practicability of the measures for reducing the gear noise, the proposed method is applied to the main transmission and the auxiliary transmission of a four-wheel-drive (4WD) automobile.
2. THE OBJECT OF THE TOOTH MODIFICATION AND THE DETERMINATION OF THE OBJECTIVE TOOTH PROFILE

2.1. The Object of the Tooth Modification

Noises of mating gears include: frictional noise by accuracy of the gear flank; impact noise from torsional vibrations and resonance of the engine; wave noise due to continuous changes of the one-cycle pitch; and interference noise from tooth interference on loaded gears. The most severe noises are the interference noise and the gear whine noise, which is caused by a complicated mechanism, such as tooth impact and double contact of mating gears due to gear manufacturing errors and deformations under loading.

Figure 1 shows the mechanism of generation of tooth impact in the profile direction, which represents a pair of gears in mesh with a contact ratio of 1.0~2.0. The tooth (1) of the pinion ends mesh with the tooth (1) of the gear and the neighboring tooth (2) of the pinion begins to mesh with the tooth (2) of the gear. In this condition, because all loads are acting on the tip of the tooth of the pinion, the tip of the tooth (1) of the pinion rotates less with an increase in the deformation, D, which is caused by loads in the rotational direction of the assembled shaft. However, because the pinion, with the exception of the deformed tooth part, rotates exactly as the shaft, the actual gear mesh is shown by the dashed lines. Therefore, a tooth impact occurs between the tooth tip of the gear and the tooth root of the pinion. In addition, a tooth impact in the lead direction is also caused by torsional deformations of shafts and of the transmission case under loading.

The relationship between the causes of the gear whine noise and the tooth contacts is listed in Table 1. Referring to Table 1, the tooth contact is the tooth tip or the tooth root on one side of the gear that arises from profile errors, i.e., either great pitch errors or deformations of the gear tooth. If the lead errors due to the torsional deformations of both the shafts and the transmission case are great, the tooth contact is one side of the gear face width. In the actual case, because both the profile and the lead errors occur, the tooth contact occurs on both the tooth tip/root on one side of the gear and on one side of the gear face width. Therefore, the objects of the tooth modification to reduce the gear whine noise by causing the tooth contact to occur at the middle of the tooth flank, which is the ideal condition.

To avoid tooth impact in a loaded gear, material needs to be removed at the tip and/or root of the involute tooth profiles. This process is known as tooth modification in the profile and lead directions. The tooth modification in the profile direction, which is the modification of the pressure angle, is considered. The tooth modification curves are determined in light of the manufacturing errors (gear profile errors) and the tooth deformation. The first objective of the tooth modification curves is to set a contact zone for the teeth that are located in the middle of the gear face width. After the range of tooth modifications is defined, the tooth modification curves are designed for locating the objective tooth curves in this range.
Table 1 The tooth contact patterns

<table>
<thead>
<tr>
<th>Causes</th>
<th>Tooth contact patterns</th>
</tr>
</thead>
<tbody>
<tr>
<td>Great pitch errors or deformations of the gear tooth.</td>
<td></td>
</tr>
<tr>
<td>The lead errors from the torsional deformations of both the shafts and the transmission case are great.</td>
<td></td>
</tr>
<tr>
<td>Simultaneous occurrence of both the profile and the lead errors in the actual gearbox</td>
<td></td>
</tr>
<tr>
<td>Modification of the objective tooth.</td>
<td></td>
</tr>
</tbody>
</table>

To avoid tooth impact in a loaded gear, material needs to be removed at the tip and/or root of the involute tooth profiles. This process is known as tooth modification in the profile and lead directions. The tooth modification in the profile direction, which is the modification of the pressure angle, is considered. The tooth modification curves are determined in light of the manufacturing errors (gear profile errors) and the tooth deformation. The first objective of the tooth modification curves is to set a contact zone for the teeth that are located in the middle of the gear face width. After the range of tooth modifications is defined, the tooth modification curves are designed for locating the objective tooth curves in this range.

Figure 2 shows the kinematical relationship of a tooth-modified gear system. Two involute tooth profiles are tangential to the line of action and are in contact at the point Q. The point, T1, which is the point of intersection of the addendum circle of the driven gear and the line of action, is a starting point of the true involute form (TIF) of the driving gear. The point, L1, is the lowest point of single-teeth contact (LPSTC) and the point, H1, is the highest point of single-teeth contact (HPSTC). The point, M, is the mid-point of the teeth contact (MPTC). The point, C1, which is the point of intersection of the addendum circle of the driving gear and the line of action, is an ending point of the TIF of the driven gear. Here, the contact zone is from T1 to C1.

Figure 3 shows the range of tooth modification in the profile direction. This range is defined by several boundary lines, such as the TRUE INVOLUTE, TIF diameter (TIF-Dia), contact diameter (CHAMFER-Dia), \( \overline{T_M}, \overline{MH_2}, \overline{H_2C_2}, \) and \( \overline{H_1C_1} \). Here, the point, T2, on the TIF-Dia is the offset-point of the point, T1, in relation to the gear profile error (E1). The point, H2, is the point of intersection of the perpendicular line through H1 on the TRUE INVOLUTE and the line of symmetry of the boundary-line, \( \overline{T_2M} \). The point, C2, on the CHAMFER-Dia is offset from the point, C1, by the error, E.
Figure 2 Kinematical relationship of a tooth-modified gear system.

Figure 3 Ranges of the tooth modification in the profile direction.

Considering the upper limit of the tooth curve, which is the B-spline curve that passes through the points, T1, L1, M, H1, and C1, and the lower limit of the tooth curve, which is the B-spline curve that passes through the points, T2, L2, M, H2, and C2, the objective tooth curve is defined as the mean value of both the upper limit of the tooth curve and the lower limit of the tooth curve.

The objective tooth curve is formulated as follows.

\[ y(x) = a + bx + cx^2 + dx^3 + ex^4 \]

In Eq. (1), the coefficients a, b, c, d, and e are defined by the positions of the LPSTC, MPTC, and HPSTC. It is desirable to make the top-point of the objective tooth curve lie within 1/3 of the contact length on both sides of the MPTC, because the contact zone must be in the middle of the tooth flank.

When a gear system transmits power, not only gear teeth but also shafts, bearings, and the transmission case are deformed. Hence, the tooth modification in the lead direction, which refers to the modification of the helix angle, must be also considered. The tooth modification curves that consider gear lead errors and shaft deformations are determined. A general method of tooth modification in the lead direction is gear crowning to compensate for the lead error. However, if a gear that is supported on one side is used, the modification of the helix angle to compensate to an extent for the tooth interference in the lead direction, as a result of the shaft deformations, and gear crowning to compensate for the lead error are both considered.

Figure 4 shows the range of tooth modification in the lead direction that considers both the gear lead errors (E3B and E3T) and the shaft deformations (E4B and E4T). This range is defined by several boundary lines, such as the true helix line (TRUE HELX), bottom and top
edges (BOTTOM and TOP), \( B_i W_i \) and \( B_i T_i \), etc. Here, the points, \( W_1 \) and \( F_1 \), are the two trisection points on the TRUE HELIX. The points, \( B_1 \) and \( T_1 \), on the BOTTOM and TOP are offset by \( E3B \) and \( E3T \), respectively. The points, \( B_2 \) and \( T_2 \), on the BOTTOM and TOP are also offset by \( E3B+E4B \) and \( E3T+E4T \), respectively.

Considering the upper limit of the tooth curve, which is the B-spline curve that passes through the points \( B_1 \), \( W_1 \), \( F_1 \), and \( T_1 \), and the lower limit of the tooth curve, which is the B-spline curve that passes through the points, \( B_2 \), \( W_2 \), \( F_2 \), and \( T_2 \), the objective tooth curve is defined as the mean value of both the upper limit of the tooth curve and the lower limit of the tooth curve.

The objective tooth curve is formulated as follows.

\[
y(x) = a + bx + cx^2 + dx^3
\]

In Eq. (2), coefficients \( a \), \( b \), \( c \), and \( d \) are defined by the positions of the two trisection points on the TRUE HELIX. It is also desirable to make the top point of the objective tooth curve lie within \( 1/3 \) of the face width on both sides of the mid-point, because the contact zone must be in the middle of the tooth flank.

Figure 4 Ranges of the tooth modification in the lead direction

2.2. Tooth-Measurement Data-Forms for Ranges of the Objective Tooth Profile Curve

The objective tooth curve is used as the tooth profile of the shaving cutter as a basis for gear-grinding operations. For applying the objective tooth curve to gear machining tools or measuring instruments, it is necessary to convert the ranges of the objective tooth curve into the tooth-measurement data-form. The objective tooth curve is calculated by the intersection points of the cut-off line.

Table 2 Tooth-measurement data-forms.

<table>
<thead>
<tr>
<th>Profile direction</th>
<th>Tooth measurement data</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Top point of CROWNING</td>
</tr>
<tr>
<td></td>
<td>CUT-OFF LINE</td>
</tr>
<tr>
<td></td>
<td>CHAMFER DIA.</td>
</tr>
<tr>
<td></td>
<td>L/H/W/B</td>
</tr>
<tr>
<td></td>
<td>CRNTL</td>
</tr>
<tr>
<td></td>
<td>CRNTR</td>
</tr>
<tr>
<td></td>
<td>CL</td>
</tr>
</tbody>
</table>
3. MEASURES TO REDUCE GEAR WHINE NOISE FOR AUTOMOTIVE TRANSMISSIONS

A flow-chart of the measures to reduce gear whine noise is shown in Figure 5. To verify the application of the proposed method of the tooth modification, the transmission and auxiliary transmission for a 4WD automobile is used as an object of the bench-test. Figure 6 shows a schematic of the test-rig and Figure 7 shows a photograph of the test-rig. The test-rig, which consists of a gasoline engine and transmission under loading, is constructed in a dead room. Sound power spectrum under a steady state is showed in Figure 8. The gear mesh frequency (2100Hz) of the high gear-set in the auxiliary transmission was the major source of the noise.

![Figure 5](image5.png)
Figure 5 Flow-chart of the measures to reduce gear whine noise.

![Figure 6](image6.png)
Figure 6 Schematic of the test-rig
Specifications of the high gear-set in the auxiliary transmission are shown in Table 3. Deformations of the gear teeth and the power shaft of the high gear-set in the auxiliary transmission were calculated by the commercial software of Romax Designer (4). The results are listed in Table 4. The objective tooth profile is improved by the proposed tooth modifications. The converted tooth-measurement data-forms (conventional data-forms of a shaving cutter) are listed in Table 5.

**Table 3 Specifications of the high gear-set in the auxiliary transmission**

<table>
<thead>
<tr>
<th></th>
<th>Module</th>
<th>Pressure angle</th>
<th>No. of teeth</th>
<th>Helix angle</th>
<th>Profile shift coefficient</th>
<th>Face width</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>2.0</td>
<td>17.5</td>
<td>35</td>
<td>31.3975</td>
<td>+0.19</td>
<td>24</td>
</tr>
<tr>
<td>B</td>
<td>2.0</td>
<td>17.5</td>
<td>52</td>
<td>31.3975</td>
<td>-0.19</td>
<td>25.9</td>
</tr>
</tbody>
</table>
Table 4 Calculated deformations of the gear tooth and the power shaft.

<table>
<thead>
<tr>
<th>Deformation of the gear tooth (D)</th>
<th>Deformation of the shaft (C)</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image" alt="Deformation Diagram" /></td>
<td><img src="image" alt="Deformation Diagram" /></td>
<td><em>Accuracy grade: KS-4.</em></td>
</tr>
<tr>
<td><em>F: Load.</em></td>
<td><em>α: Slope of the tilt.</em></td>
<td><em>Error of the profile: 15µ.</em></td>
</tr>
<tr>
<td><em>L: Length.</em></td>
<td></td>
<td><em>Error of the lead: 15µ.</em></td>
</tr>
<tr>
<td>A</td>
<td><em>D: 7.5µ.</em></td>
<td>A</td>
</tr>
<tr>
<td>B</td>
<td><em>D: 8.4µ.</em></td>
<td>B</td>
</tr>
</tbody>
</table>

*Young's modulus: 206 Gpa.*  
*Poisson's ratio: 0.3.*

Table 5 Converted data-forms of the objective tooth profile (conventional data-forms of a shaving cutter).

<table>
<thead>
<tr>
<th>Direction of the gear profile</th>
<th>CL (mm)</th>
<th>CLP (mm)</th>
<th>CLA (mm)</th>
<th>CRNTL (mm)</th>
<th>CRNTR (mm)</th>
<th>CRNP (µ)</th>
<th>PLW (µ)</th>
<th>FA (µ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A 10.69</td>
<td>1.78</td>
<td>1.78</td>
<td>2.78</td>
<td>6.34</td>
<td>5.7±5</td>
<td>7.08</td>
<td>±10.2</td>
<td></td>
</tr>
<tr>
<td>B 10.69</td>
<td>1.78</td>
<td>1.78</td>
<td>4.35</td>
<td>7.91</td>
<td>5.1±4.6</td>
<td>5.2</td>
<td>±4.9</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Direction of the gear lead</th>
<th>FW (mm)</th>
<th>BBL (mm)</th>
<th>BTL (mm)</th>
<th>CRNTL (mm)</th>
<th>CRNTR (mm)</th>
<th>CRNL (µ)</th>
<th>HLW (µ)</th>
<th>HA (µ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A 24</td>
<td>3.0</td>
<td>3.0</td>
<td>8.0</td>
<td>16.0</td>
<td>8.7±3.4</td>
<td>-6.2</td>
<td>±6.2</td>
<td></td>
</tr>
<tr>
<td>B 25.9</td>
<td>3.24</td>
<td>3.24</td>
<td>8.63</td>
<td>17.26</td>
<td>5.4±5.4</td>
<td>0</td>
<td>±11.4</td>
<td></td>
</tr>
</tbody>
</table>

The high gear-set in the auxiliary transmission, including the tooth modification in the profile and lead directions, are newly machined by using the conventional data-forms of a shaving cutter that are listed in Table 5. The machined gear teeth are measured by a commercial gear measuring instrument (Osaka Semitsu, CLP35). The measured results show that the modified tooth profile is in the ranges of objective tooth profiles. In the case of the worst noise condition and major frequency of the gear mesh noise, the sound power spectra are measured before and after the tooth modification. The noise reduction effect of the gear tooth modification is shown in Figure 9. After the tooth modification, noise decreased by eight dB in the case of the gear mesh frequency (2100Hz) of the high gear-set in the auxiliary transmission. There is also no great difference between the measured noise before and after the tooth modification in the case of the 1st harmonic frequency band (4200Hz) of the gear mesh frequency. Therefore, the noise of the gear mesh frequency band is due to the gear tooth impacts and the noise of the harmonic frequency band is due to the gear manufacturing errors. The tooth modification is an effective measure for avoiding gear tooth impacts and reducing the gear noise of the gear mesh frequency band.

(Sound power spectra: solid lines – before the modification; dashed lines – after the modification.)
4. CONCLUSION

To reduce the gear noise due to the tooth impact mechanism that arises from deformations of the gear teeth and the power shaft, the optimal objective tooth profile was analytically determined. In addition, the determined optimal objective tooth profile was converted to the tooth-measurement data-form for applicability to gear measuring and machining instruments. After the high gear-set in the auxiliary transmission was machined, along with the tooth modification in the profile and lead directions by using the conventional data-forms of a shaving cutter, the noise decreased by eight dB. The proposed method for deriving the objective tooth profile applies to not only automotive transmissions but also most industrial and general-purpose transmissions.

REFERENCES


