Measurement of Gear Noise Behaviour for Different Microgeometries

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ABSTRACT
Designing machines under consideration of noise emission is more than a convenience aspect. For many applications, it is rather a major design goal. This is especially true for gear meshes, which are an important source of vibration in the transmissions. There are many publications available concerning theoretical aspects of design of micro geometry of gears for vibration reduction. However, an experimental validation and sound comparisons among different theoretical designs can be rarely found.

Therefore, some basic gear micro geometry designs have been carried out for noise reduction. Special attention has been paid in order to preserve the identical main geometry. This provides an excellent basis for a valid comparison of the influences of a theoretically silent design versus standard design. Since spur gears exhibit an inauspicious noise behaviour, measurements have been conducted on this gear type with unmodified micro geometry versus low noise micro geometry. The acceleration measurement results show the advantage of an adequate micro geometry design. As next step, the behaviour of further geometries may be evaluated in comparison to the results shown in this paper.

Keywords: Gear, Noise, Micro Geometry I-INCE Classification of Subjects Numbers:11.1.3, 72.9, 74.8

1. INTRODUCTION

Besides the load carrying capacity, NVH is one of the most important considerations for gearboxes. Excitation induced by gear meshes can be transmitted throughout the gearbox in the form of structure-borne noise. This can finally be transferred into airborne noise on housing surfaces. Under many circumstances, this noise is found to be disturbing and can affect the product quality.

Over the past few decades, many researchers have dedicated themselves to reduce noise excitation in gear meshes. One of the most promising solutions is the modification on the tooth micro geometry. In theory, it can be shown that lower excitation of gear meshes can be archived by means of flank modifications. However, documented experimental validations can be rarely found.

In this paper, experimental validations of the influence of micro geometries on gear mesh excitation shall be presented. The experiments were conducted mainly on spur gears, as they are well-known for their unfavourable noise behaviour. In comparison with helical gears with identical macro geometries, spur gears tend to be a lot more sensitive to the gear mesh excitation due to their discrete leaps between double and single engagements along the line of contact. However, spur gears are sometimes more favourable especially under constructional restrictions, where helical gears may be disadvantageous, i.e. by means of designing the bearings. In many cases, the absence of the axial force in spur gears can reduce further constructional complexities as well as manufacturing cost. Thus, the design of quiet spur gears is practically the compromising solution.
2. THEORETICAL FUNDAMENTALS

The noise behaviour of gearboxes is fundamentally dominated by the excitation in the gear meshes themselves. This excitation is actually the resulting phenomenon, in which many effects participate. These effects have been thoroughly examined by several researchers over the last few decades (see also (1–8)).

Some significant effects can be exemplary mentioned here. The variable mesh stiffness within each engagement defines the relationship between the load and the corresponding mesh deformation. The mesh stiffness itself is a result of the specific tooth pairing stiffnesses. This mesh stiffness can be varied by different gear contact ratios (transverse as well as overlap). Furthermore, the contact ratios can be extended under the loaded condition by means of premature and prolonged engagements, which induces another significant effect for the mesh excitation (see also (9)). The deformation of shafts also affects the mesh excitation and is the result of the interaction among shafts and bearings. Other effects are the directional transition of friction at the pitch point, the asperities of gear wheels, etc.

In addition to the foregoing, the form deviation of gear meshes from their actual involute profile is absolutely one of the non-negligible causes for the mesh excitation. The form deviation consists of manufacturing tolerances as well as modifications, which are specifically applied to the meshes for optimizing purposes like width load distribution, efficiency, and definitely, noise behaviour.

In a stationary state, the deformation of gear mesh $x$ at a specific mesh point, which can be expressed in term of time $t$, due to the load $F$ can be calculated under consideration of form deviation $x_{fi}$ at each contact point as:

$$x(t) = \frac{F - \sum c_i(t) \cdot x_{fi}(t)}{\sum c(t)}$$  \hspace{1cm} (1)

This deformation is also known as the loaded transmission error (LTE). The loaded transmission error characterizes mesh excitation and therefore can be used as prediction value for noise behaviour as stated in (10). Some further examinations can also be found in (11, 12). In order to improve noise behaviour of gears, the progress of the loaded transmission error over teeth engagements needs to be kept as constant as possible (see also (13–15)). That is, fluctuations of the LTE should be minimized.

![Mesh Stiffness Diagram](image)

At points B and D in Figure 1, the transitions between single and double engagement take place. These two discrete points yield a major leap of the mesh stiffness for the raw gear meshes. In the double engagement areas (A-B and D-E), the mesh stiffness is higher because 2 pairs of gears are being engaged. Under constant load, the resulting deformations (LTE) in these areas would be consequently smaller than those in the single engagement area B-D. An example in Figure 1 shows a profile modification, which can be realized by means of a combination of standard modifications, like tip relieves on both pinion and wheel. This modification compensates the small transmission error in the double engagement areas by means of greater deviation $x_f$. Moreover, discontinuities at transition points can also be minimized by the modification.

![Figure 1: Mesh stiffness discontinuity and profile modification (schematically)](image)
3. EXPERIMENTAL SETUP

3.1 FZG Dynamic Test Rig

The experiments were performed on the FZG dynamic test rig with standard centre distance of 140 mm, which is depicted schematically in Figure 2. The test rig makes use of the mechanical power circulation principle, which is commonly utilized in FZG back-to-back test rigs. The test rig consists of two sets of gearboxes, the test gearbox and the drive gearbox. These gearboxes are connected together by two shafts. The loading clutch was installed on one of these shafts, deviding this shaft into two parts.

![Figure 2: FZG dynamic test rig](image)

In order to bring static load into the test rig, the loading clutch shall be decoupled. One of the clutched shafts must then be elastically twisted by means of a lever with adjusted weight. Under this circumstance, engaging the loading clutch shall provide a constant static load torque to the test rig during the test operation, so that high tension can be archived in the system independently from the drive aggregate, which mainly provides speed and some sufficient load, which may lose during the operation due to the efficiency.

The FZG dynamic test rig is equipped with two drive aggregates, in order to cover a wide range of operational speed. The main drive provides high speed range for dynamic measurements, while this can then be decoupled in the slow speed operation, during which the auxiliary drive is used. Loaded transmission error is a good example of a measurable value in the slow speed operation.

3.2 Measurements

In order to acquire a complete information about the noise behaviour of gears, two measurements need to be carried out for each variation. These are:

- measurement of loaded transmission error
- measurement of torsional acceleration

With the measurement of loaded transmission error, the quasi-static excitation of gear meshes can be identified. On the other hand, the measurement of torsional acceleration would retrieve the dynamic characteristics of the gear meshes over a broad range of speed.

Moreover, the gear meshes were applied with 6 load stages described in Table 1 in favour of a good overview of the noise behaviour of gears over different loads.

**Measurement of Loaded Transmission Error**

The objective of the measurement is to identify the intrinsic excitation of the gear meshes under a specific external load. Therefore, a low rotational speed is required. Distortions due to accelerational effects can then be neglected. The measurements were performed at the speed of the pinion at 5 rpm. The angular positions of the pinion and the wheel are recorded by means of high resolution Heidenhain incremental encoders. This digital sensor detects the position of the gear thanks to its optical transmitter and receiver.

The measured loaded transmission error can be defined as:

\[ x = \varphi_1 \cdot r_{b1} - \varphi_2 \cdot r_{b2} \]  

(2)
Table 1: Load stages for the experiments

<table>
<thead>
<tr>
<th>Load Stage</th>
<th>Load on Pinion Nm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>400</td>
</tr>
<tr>
<td>2</td>
<td>666</td>
</tr>
<tr>
<td>3</td>
<td>1000</td>
</tr>
<tr>
<td>4</td>
<td>1334</td>
</tr>
<tr>
<td>5</td>
<td>1600</td>
</tr>
<tr>
<td>6</td>
<td>2000</td>
</tr>
</tbody>
</table>

whereas $r_{b1}$ and $r_{b2}$ are base diameters of pinion and wheel respectively. According to Equation 2, there would be no transmission error, if the gear ratio is ideal ($\varphi_1 = i_{12} \cdot \varphi_2$). An ideal involute profile gear mesh with no external load would satisfy this condition. The transmission error can also be defined by means of an angular expression as follows:

$$\Delta \varphi = i_{12} \cdot \varphi_1 - \varphi_2$$

(3)

Measurement of Torsional Acceleration

The measurement of torsional acceleration enables the detection of the actual excitation within the meshing under the dynamic situation. The torsional acceleration can be measured on either wheel. However, the sensors should be placed directly under the gear wheel, so that no further transmission path can distort the measurement.

For this measurement, three acceleration sensors were attached in the gear wheel in both radial and tangential directions. By combining these measured values mathematically, torsional as well as bending acceleration can then be identified. For instance, the torsional acceleration can be identified by the addition of the radial and tangential sensors depicted in Figure 3.

For this measurement, the speed of the pinion varied between $n = 400$ rpm to $n = 5300$ rpm with speed step at $\Delta n = 20$ rpm. During the measurement, the speed of the pinion was held constant, yielding a quasi-run-up measurement.

3.3 Experimental Variations

The experiments were carried out on two different toothings. The data of their macro geometries are shown in Table 2. Both toothings differ solely from their usable tip diameters.

Table 2: Macro geometries of tested gear sets

<table>
<thead>
<tr>
<th></th>
<th>Unit</th>
<th>Normal Toothing</th>
<th>High Toothing</th>
</tr>
</thead>
<tbody>
<tr>
<td>number of teeth</td>
<td>$z_1/z_2$</td>
<td>43</td>
<td>45</td>
</tr>
<tr>
<td>centre distance</td>
<td>$a$</td>
<td>140</td>
<td>140</td>
</tr>
<tr>
<td>normal module</td>
<td>$m_n$</td>
<td>3.21</td>
<td>3.21</td>
</tr>
<tr>
<td>profile shift</td>
<td>$x_1/x_2$</td>
<td>-0.1747</td>
<td>-0.1984</td>
</tr>
<tr>
<td>width</td>
<td>$b_1/b_2$</td>
<td>39.5</td>
<td>39.5</td>
</tr>
<tr>
<td>root diameter</td>
<td>$d_{F1}/d_{F2}$</td>
<td>127.604</td>
<td>133.874</td>
</tr>
<tr>
<td>usable tip diameter</td>
<td>$d_{Na1}/d_{Na2}$</td>
<td>142.5</td>
<td>148</td>
</tr>
<tr>
<td>transverse contact ratio</td>
<td>$\varepsilon_{a,N}$</td>
<td>1.5</td>
<td>2.0</td>
</tr>
</tbody>
</table>
Within each toothing, experiments were carried out on unmodified as well as flank-modified gear meshes. The unmodified gear meshes shall be used as reference for the flank-modified gear meshes. Tip relief was the used modification, as the profile modifications influence the noise behaviour on spur gears dominantly. A long tip relief was applied on the normal toothing. Due to the lack of single engagement, a long tip relief for the high toothing is not defined. Therefore, a short tip relief was applied. The modifications were designed for the load around 4th load stage (see Table 1). The modification parameters can be found in Table 3.

Table 3: Micro geometries of flank-modified gearmeshes

<table>
<thead>
<tr>
<th></th>
<th>Unit</th>
<th>Normal Tooth</th>
<th>High Tooth</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Pinion</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>tip relief amount</td>
<td>µm</td>
<td>27</td>
<td>15</td>
</tr>
<tr>
<td>tip relief length</td>
<td>mm</td>
<td>4.77</td>
<td>0.97</td>
</tr>
<tr>
<td>% · gₐ</td>
<td></td>
<td>33</td>
<td>5</td>
</tr>
<tr>
<td>modification diameter</td>
<td>mm</td>
<td>138.82</td>
<td>143.36</td>
</tr>
<tr>
<td><strong>Wheel</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>tip relief amount</td>
<td>µm</td>
<td>27</td>
<td>15</td>
</tr>
<tr>
<td>tip relief length</td>
<td>mm</td>
<td>4.77</td>
<td>0.97</td>
</tr>
<tr>
<td>% · gₐ</td>
<td></td>
<td>33</td>
<td>5</td>
</tr>
<tr>
<td>modification diameter</td>
<td>mm</td>
<td>144.46</td>
<td>149.67</td>
</tr>
</tbody>
</table>

4. EXPERIMENTAL RESULTS

4.1 Loaded Transmission Error

The results from LTE measurements are shown in Figure 4 and Figure 5. The frequency spectra were determined after 10 turns of the pinion shaft (after 430 engagements). The frequencies in the diagrams are presented as a function of gear mesh order. The amplitude of each order represents the portion of the harmonic excitation, which participates in the total mesh excitation (see also (16, 17)). Generally, a high amplitude indicates a high participation in the mesh excitation with the frequency corresponding to the mesh order.

The loaded transmission errors of the unmodified normal toothing are shown in Figure 4a. The transmission error grows continuously with increasing load. This characteristic is basically due to its elasticity. With increasing load, the deformation grows higher, which results in a larger transmission error. The spectrum also suggests that the first mesh order is the most dominant.

Figure 4b displays the loaded transmission errors of the flank-modified normal toothing. The first mesh order is the most dominant overall. The loaded transmission error shows its minimum value at the 4th load stage, for which the modification was designed. From this load stage on, the loaded transmission error grows higher in both directions. With the modification, the noise behaviour can be optimised for a certain load range. The deformation of gear mesh can basically be compensated by the modification within this determined load range. However, the quality of the compensation degrades beyond the load range, as the deformation or the modification itself might be insufficient for the compensation below or above the design load respectively. This characteristic is confirmed in Figure 4b.

The modification influences generally the contact ratio under load. The effect of the resulting fluctuation
Figure 4: Spectra of loaded transmission errors from gear meshes with normal toothing.
Figure 5: Spectra of loaded transmission errors from gear meshes with high toothing
of the length of contact can be visualized in the amplitude at the first mesh order. This is also true for the flank modified high toothing, of which the measurement results are presented in Figure 5b. Moreover, the modification tends to reduce the amplitudes at the higher mesh orders. The reason for this phenomenon can be traced back to the reduction of the premature and the prolonged engagements, which consequently means the reduction of impacts along the path of contact.

4.2 Torsional Acceleration

The measurements of the torsional acceleration were carried out over load stages predefined in Section 3.2. An order diagram can be obtained from each single measurement. This diagram can give a comprehensive information of noise behaviour under this specific load. However, the interpretation of order diagrams over load stages can become very complicated. Therefore, a fourier coefficient plot shall be used for further interpretation of measurement results, as it provides scalar values directly derived from the order diagrams. Thus, a precise comparison among different mesh variations over a large load range is feasible.

1.) Construction of order diagram at each specific load

2.) Averaging of Order Spectrum

3.) Averaged Spectrum according to a specific load

Figure 6: Construction of Fourier Coefficient Plot

In order to obtain the fourier coefficient plot, the spectra in each order diagram would have to be averaged over the speed range. Thanks to the averaging, the intrinsic characteristics of the gear meshes could be extracted in the resulting averaged order spectrum. This spectrum provides a solid information about the excitation behaviour in gear meshes. The spectrum at each specific order can be transfered to the fourier coefficient plot in terms of the corresponding load stage. Figure 6 describes the construction of the fourier coefficient plot graphically. For further information, see also (18–20).

The measurement results from the unmodified normal toothing are presented in Figure 7a. The experiments could be carried out until the 4th load stage, at which sensors were repeatedly damaged. The utilized sensors had measurement range up to 500g and should withstand the acceleration of 3000g without damage according to the specification from the manufacturer. This means, there was a high level of acceleration during the measurement apparently. Such high acceleration cannot be obtained from regular engagements. Some effects can theoretically lead to a significant increase in acceleration, for example, flank lift-off in resonances.

The fourier coefficient plot of the flank modified normal toothing in Figure 7b suggests that the first mesh order is the most dominant. The amplitude shows its minimum value at the 4th load stage, for which the flank modification was designed. From this load stage, the amplitudes keep growing continuously.

Similar effects as seen in the unmodified normal toothing can also be observed from the unmodified high toothing. The experiments could be performed without damages until the 5th load stage. The evaluation of the results from load stage 1-4, however, shows that the excitation from the first mesh order has a lesser influence on the total excitation than the one from the second mesh order. Due to the high transverse contact

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ratio ($\varepsilon = 2.0$), the characteristic of the stiffness fluctuation in the mesh changes in comparison to the normal toothing, so that the second mesh order has a major role in the mesh excitation. This phenomenon can also be seen in Figure 7c.

With the predefined tip relief, the flank-modified high toothing shows lower acceleration levels than the unmodified variant. As depicted in Figure 7d, the excitation from the first mesh order exhibits a comparable level to the one from the second mesh order at lower load stages, and keeps up the same level over the whole load stages. The amplitude from the second mesh order tends to develop over the increasing load stages. However, the total acceleration levels are lower than those of the unmodified high toothing.

![Figure 7: Fourier Coefficients Plots of the Averaged Order Spectra of different gear meshes](image)

5. SUMMARY

In this paper, the effects of gears micro geometries on noise behaviour were experimentally evaluated. The experimental results show that, there is a correlation between the loaded transmission error and the torsional acceleration. Thus, it can be said that, the excitation by means of transmission error under load is a major cause for the excitation in gear meshes.

The flank modifications can be used generally in order to reduce the excitation in gear meshes. For a common normal toothing, the optimisation can be done for a specific load range. Overall comparison of the four cases shows that the unmodified high toothing provides significantly lower excitation levels than the unmodified normal toothing. The modified normal toothing reaches a similar or even lower excitation level than the unmodified high toothing for design load only. The modified high toothing provides the lowest excitation level of the tested meshes over a wide load range.
Adequate care has to be taken in comparing the noise behaviour of normal and high toothing, as different mesh orders dominate the total excitation. Especially under the subcritical operation, high excitations can be induced easier by the second mesh order. Under this circumstance, the noise advantage of the high toothing can therefore be affected by this effect.

REFERENCES


