Experimental Studies about the Effects of Dynamic Loads upon Gear Noise*

By Kantaro Nakamura**

In order to investigate the generating mechanism of gear noise, a power circulating type gear testing machine was installed in an anechoic room. From some preliminary measurements, the effects of operating conditions, accuracies, blank thickness and lubrication on gear noise were studied. The more detailed experiments were successively performed about the generating mechanism, with the test gear which had much involute errors. The largest source is considered to be the load fluctuations in the direction of line of action. Sound intensity is proportional to fluctuating load and square of fluctuating frequency. Tooth profile errors have much effects upon gear noise, especially under the light loads. If separation occurs at a certain speed, noise suddenly increases, and a strange noise of subharmonic frequency is occasionally built up.

1. Introduction

Noise generated from the power transmitting gears has been studied by many investigators and various kinds of mechanism of sound generation have been discussed on many occasions (1)-(8). According to those reports presented up to this time, it is explained that the main causes of gear noise are as follows: impacts of teeth meshing, changes of direction of friction forces, elastic vibrations of gear blanks and other members, lubrication noise, or some other sources. However, these results need yet to be confirmed.

Considering this present situation, the author tried to perform some experimental studies with a gear testing machine in an anechoic room. This report describes how large effects the dynamic load has upon gear noise under the transmitting operations.

2. Testing machine and sound field

A power circulating type of testing machine was used for loading test gears. It consisted of two couples of gears, one of which was mounted on the bed in an anechoic room and another in the next room separated with a thick concrete wall. The test room, as shown in Fig. 1, is constructed with 25 cm thick concrete walls to insulate from external noise, and absorbing materials which consist of 5 cm thick rock wool plates and 70 cm long absorbing wedges filled with glass fibers. The inside absorbing constructions are supported elastically with soft rubbers, so mechanical vibrations or solid-borne noise may be eliminated. The effective volume of the anechoic room is 74.2 m³ and the area of inside floor is 22.5 m².

The testing machine is shown in Fig. 2 and its specifications are given in Table 1. As an induction coupling motor was used, the revolutions

---

* Received 31st January, 1966.
** Research Fellow, Mechanical Engineering Research Laboratory, Hitachi Ltd., Kokubunji.
of wheel were changeable continuously up to 1,450 rpm. The plain bearings were used at the testing side to prevent bearing noise, and a housing which was made of thin sheets covered the testing part of machine.

Acoustic characteristics of sound fields were checked before gear noise was measured. The inverse square law was satisfied at the space around the testing machine between 200 and 5,000 c/sec, as shown in Fig. 3. The ambient noise level was about 11 phons; it was low as electric noise level of measuring apparatus. Sound field around the testing machine is thus deemed as ideal space for measurement of gear noise, because the difference of sound pressure measured in each direction is very small as shown in Fig. 4.

3. Preliminary test

3.1 Measuring apparatus

The condenser microphone was set at the distance of 1 m from the meshing point to the axial direction. Noise spectra were measured with one third octave band spectrometer and recorded with the level recorder; both were made by Brüel & Kjaer. The specifications of test gears are given in Table 2.

![Fig. 2 Power circulating type gear testing machine](image)

![Fig. 3 The inverse square law of sound pressure at the acoustic field in front of the testing machine](image)

![Fig. 4 Pressure distributions at the acoustic field around the testing machine, on the basis of pressures at the point D](image)

<table>
<thead>
<tr>
<th>Prime mover</th>
<th>7.5 kW, L.C. motor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Revolutions of gear shaft</td>
<td>Max. 1,450 rpm</td>
</tr>
<tr>
<td>Torque in the gear shaft</td>
<td>Max. 36 m-kg</td>
</tr>
<tr>
<td>Circulating power</td>
<td>Max. 75 H.P.</td>
</tr>
<tr>
<td>Periphery velocity</td>
<td>Max. 18 m/sec</td>
</tr>
</tbody>
</table>

### Table 2 Test gears

<table>
<thead>
<tr>
<th></th>
<th>S4-A</th>
<th>S4-B</th>
<th>S4-D</th>
<th>S4-E</th>
<th>S-6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Module</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>6</td>
</tr>
<tr>
<td>Number of teeth</td>
<td>60</td>
<td>60</td>
<td>30</td>
<td>30</td>
<td>60</td>
</tr>
<tr>
<td>Pressure angle</td>
<td>20º</td>
<td>20º</td>
<td>20º</td>
<td>20º</td>
<td>20º</td>
</tr>
<tr>
<td>Face width</td>
<td>22 mm</td>
<td>12 mm</td>
<td>40, 30, 20 mm</td>
<td>30 mm</td>
<td>40 mm</td>
</tr>
<tr>
<td>Coefficient of profile shift</td>
<td>0</td>
<td>0</td>
<td>+0.20, +0.32</td>
<td>+0.20, +0.32</td>
<td>0</td>
</tr>
<tr>
<td>Materials</td>
<td>S40C</td>
<td>S40C</td>
<td>S40C</td>
<td>S40C</td>
<td>S40C</td>
</tr>
<tr>
<td>Finishing</td>
<td>Ground</td>
<td>Ground</td>
<td>Ground</td>
<td>Hobbed</td>
<td>Ground</td>
</tr>
<tr>
<td>Natural frequency of gear blank</td>
<td>1,900 c/sec</td>
<td>1,100 c/sec</td>
<td>1,900 c/sec</td>
<td>2,620 c/sec</td>
<td>Ground</td>
</tr>
</tbody>
</table>
3.2 Sound level and sound spectra

Sound level and its spectra were measured in preliminary experiments under various operating conditions. Sound pressure level measured about the test gears S4-A which were finished very precisely is shown in Fig. 5. This indicates that, although sound pressures increase approximately proportional to 0.8—1.2 power of the periphery speed of wheel and linearly proportional to tangential load, yet two or more peaks are found at a certain speed. These peaks seem to be the resonance when the tooth contact frequencies coincide with the natural frequency in the mating gear systems. About the problems mentioned above, particular experiments and analyses are to be studied later.

Sound spectra with respect to static loads are shown in Fig. 6, and sound spectra with respect to rotating speed of wheel are shown in Fig. 7. Each figure shows that the distinguished sound frequencies are tooth contact frequencies, \( f_0 \), and their higher harmonics, \( 2f_0, 3f_0, \ldots \); and these spectra are independent of static loads. It should be noted that the fundamental frequency, \( f_0 \), does not always indicate the highest level, but the spectrum levels between 1 kc/sec and 2 kc/sec are generally higher. Suggestion will be presently made to investigate the generating mechanism.

3.3 Accuracies

The two test wheels, which were finished to have different involute errors, were examined to observe the effects of accuracies of teeth upon noise generated. In this case, however, only a common pinion, which had been finished more precisely, was used for mating with the test wheels. Fig. 8 shows the sound produced by these test gears mentioned above. It indicates that sound pressure is higher as involute errors increase specially at high speed. It should be considered that noise would arise more intensely because of the increase of dynamic loads excited by tooth profile errors.

3.4 Gear blank thickness

It is well-known that the elastic vibration of gear blank may cause gear noise. Two gear blanks of which thickness was different such as 22 mm and 12 mm were tested. The results are shown in Fig. 9.
In the case of a thin wheel, it is recognized that the natural frequencies of the wheel blank were remarkably built up, while in the case of a thick one, teeth meshing frequency and its harmonics were predominant. The sound pressure level of teeth meshing frequency is not so different between the two cases under the same conditions that the difference of overall sound intensity level was not so large—less than 5 dB as far as obtained.

3-5 Lubrications

It abundant oil is supplied to the meshing point, pure noise like the sound of siren is occasionally heard. It may be due to so-called oillipocketing, that is, the lubricating oil fed between engaging teeth is compressed. Fig. 10 shows the noise increase caused by the lubricating oil. As the result of the measurements, noise in question does not depend upon static loads, but depends upon periphery speed, tooth width and amount of oil. The sound spectra indicate that the predominant frequencies of noise are harmonics of teeth meshing frequency and it is noteworthy that the spectrum level between 4 kc/sec and 5 kc/sec is always higher. These frequencies are thought to be determined by the tooth size.

3-6 Sound fields

In order to find noise origins, it is indispensable

![Sound spectra at different microphone positions; test gear S4-A, under the same operating conditions](image)

![Directional patterns of gear noise, for test gear S4-A](image)

![Sound pressure level in dB re 0.0002 μ bar](image)

![Frequency in c/sec](image)

![Sound pressure level in dB re 0.0002 μ bar](image)

![Frequency in c/sec](image)
to measure sound pressure distributions in a free space. On the semi-circle of 0.5 m radius from the meshing point, sound pressure was measured. The results are given in Fig. 11 and Fig. 12. They point out that the sound fields change by wheel rotating speed and yet the maximum pressure is mostly observed at the point C, which is 30 degrees from the axial direction, and the minimum pressure is measured in the plane of wheel. The higher the sound frequencies are, the more sharply the directional patterns are presented. The matters mentioned above are very significant, because the following inference about a noise origin might be derived; the sound would be generated from a dipole as an oscillating disk to the axial direction, that is to say, one noise origin would exist in an axial oscillation of wheel blank. This difficult problem about where the gear noise is radiated from, is to be investigated in the next number.

Table 3  Testing machine and test gears for dynamic load test

<table>
<thead>
<tr>
<th>Type</th>
<th>Power circulation type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power</td>
<td>Max. 700 H.P.</td>
</tr>
<tr>
<td>Speed</td>
<td>Max. 90 m/sec</td>
</tr>
<tr>
<td>Module</td>
<td>5</td>
</tr>
<tr>
<td>Number of teeth</td>
<td>36x x 72z</td>
</tr>
<tr>
<td>Materials</td>
<td>40 mm</td>
</tr>
<tr>
<td>Finishing</td>
<td>Ground</td>
</tr>
</tbody>
</table>

4. Dynamic loads and noise

4.1 Experimental method

As the result of the preliminary measurements, it is considered that noise produced by gear meshing would be much dependent upon dynamic loads. In order to investigate the mechanism of generating sound, some detailed experiments were performed with another back-to-back testing machine, of which specifications together with the specifications of test gears are shown in Table 3. For test gears, involute errors were intentionally given by lapping under a 250 kg tangential load and 300 rpm revolutions of wheel. The involute curves before and after lapping are shown in Fig. 13 (a). It indicates that each tooth has regular involute errors, such as convex near the pitch points and concave at the middle of addendum, and also of dedendum. From this figure, combined involute errors added at each contact point are drawn as Fig. 13 (b).

Stress at tooth root was observed with a wire strain gage, and dynamic loads were calculated by comparing with stress influence curve which was measured under the static condition.

An acceleration of angular oscillation of a wheel was measured with the accelerometer which was set on the side of wheel blank as shown in Fig. 14. Strain near the cantilever root is picked up with semi-conductor gage, and amplified by a low frequency amplifier. The sensitivity of the accelerometer is given in Table 4 which is calibrated on an electro-magnetic exciting table. The natural frequency is 4 570 c/sec and strain is linear propor-

Table 4  Sensitivity of accelerometer

<table>
<thead>
<tr>
<th>Resistance of semi conductor gage</th>
<th>367 Ω</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gage factor</td>
<td>126</td>
</tr>
<tr>
<td>Strain</td>
<td>1.88 x 10^{-19}/cm/sec</td>
</tr>
<tr>
<td>Sensitivity</td>
<td>0.35 mV/cm/sec</td>
</tr>
</tbody>
</table>

Fig. 14  Accelerometer
tional to the acceleration of a fixed end at the frequencies lower than a half of the natural frequency.

Sound pressure at a near field was measured with a half inch condenser microphone covered with thin polyethylene sheet. It was found that decrements of sensitivity of covered microphone were about 2 dB below 5 kc/sec as the result of

![Graphs showing stress cycles at tooth root](image)

(A. Before lapping, B. After lapping)

Fig. 15 Stress cycles at tooth root

![Graphs showing acceleration spectra](image)

(A. Before lapping, B. After lapping)

Fig. 17 Sound spectra and acceleration spectra under the same operating conditions before and after lapping

(Interval of each dot shows tooth meshing period)

Fig. 16 Stress cycles at tooth root (lower) and acceleration cycles of angular vibration (upper), for the test gear after lapping under light loads
calibrations in a reverberation room. Frequency analysis of sound was performed with a heterodyne type of RADIOMETER "Wave Analyzer".

4-2 Dynamic loads
Examples of stress cycle before and after lapping are shown in Fig. 15. At low speed, stress fluctuations are presented and their frequencies are approximately equal to the natural frequency of gear system. Dynamic loads vary with speed and if tooth errors are larger, the dynamic loads grow higher. Fig. 16 shows the oscillograms of tooth stress and accelerations of angular oscillation of wheel. It was obtained with the test gear which had much errors, under a comparatively lower tangential load i.e. 250 kg. At 700 rpm, stress at tooth root suddenly increases, then an alarming noise arises. It is explained as follows: when the inertia forces in relative motions overcome the static load, a break of tooth contact might occur, what is called tooth separation, and very large stress would arise because of extremely short duration of tooth contact.

4-3 Spectra of sound and acceleration
Sound spectra and accelerations of angular vibrations are compared in Fig. 17. It should be noted that every spectrum, obtained at the same test conditions, is closely similar to another. In this figure, it is found that the principal frequency is the number of tooth contact per unit time and its higher harmonics, and the spectrum level at the fundamental frequency of tooth meshing is not always highest, but the maximum spectrum level is presented near the natural frequency of mating gear system. The vibration system of mating gears is constituted by effective masses of pinion and gear $M_p, M_o$, placed on a line of action, and with combined stiffness $K$ of engaging teeth. The natural frequency is given as

$$f = \frac{1}{2\pi} \sqrt{K \left( \frac{1}{M_p} + \frac{1}{M_o} \right)} \quad (1)$$

In the case of test gears which were used in this experiment, the natural frequency becomes 1010 c/sec for the average value of tooth stiffness.

At 1000 rpm a strange noise, which had one third the meshing frequency and its higher harmonics, built up. This phenomenon occasionally occurs at a certain speed near the natural frequency of gear system, in the case of roughly finished gear and under light loads. This noise is considered to be due to irregular meshing, for example, meshing with every second tooth, with every third tooth and so on. These strange phenomena owing to separation are to be studied in another report.

In Fig. 18, sound spectrum level is compared with the spectrum level of acceleration, which is corrected by the characteristic curve of the accelerometer. It indicates that gear noise is in close connection with the relative motions of pinion and gear, and in a certain region of frequencies sound pressure is proportional to the acceleration of angular oscillation of gear.

4-4 Effect of profile errors on sound pressure level
Sound pressure level before and after lapping was measured under various operating conditions and is shown in Fig. 19. Although sound pressure increases with peripheral speed, a resonance breaks out at the speed when tooth contact frequency coincides with the natural frequency of gear system. Besides, several small resonances occur at about half or a third of the natural frequency, and they become clearer as tooth errors are larger.

![Fig. 18 Comparison of spectrum level of sound pressure and acceleration](image-url)
It should be noted that sound pressure increases with the static load for a precise gear, while for a roughly finished gear, any difference of sound pressure under static load can hardly be found as Fig. 19 (b). The reason for these experimental results is considered as follows: if the static load is small, tooth profile errors have larger effects on dynamic loads because the ratio of errors to static deflections is larger. If static deflections of tooth are less than half the combined tooth errors, sound pressure will hardly change with static loads, and if more than twice, sound pressure will be proportional to the tangential load.

As the result of these experiments, it could be concluded that the most effective exciting force to produce gear noise consists of load fluctuations in the direction of line of action.

5. Discussion

Supposing the noise origin exists in a gear blank or other members adjacent to it, and the vibration system is so not complicated between the noise origin and the meshing teeth, some discussions about the generating mechanism will be presented. Sound pressure $p$ radiating from the vibrating member will be proportional to accelerations between two frequencies, which are decided by the area of generating plane, then

$$p = \alpha \omega^2 w$$

where, $w$ is vibration displacement, $\omega$ is angular frequency. The exciting force is thought to be fluctuating loads in the direction of line of action, and if $\beta(\omega)$ is the compliance in this vibrating system, displacement $w$ can be given in the form,

$$w = \beta(\omega)W_d$$

where, $W_d$ means dynamic load, which is proportional to relative displacement $x_b$ between pinion and gear. Then the sound pressure is given as

$$p = \alpha \beta(\omega) \omega^2 x_b$$

If the vibrating system has rather simple constitution, and the fluctuating frequency is lower than the natural frequency of the vibrating system, $\beta(\omega)$ is considered to be constant independently of $\omega$. Thus $p \propto \omega^2 x_b$, which means sound pressure is proportional to the acceleration of angular vibration of wheels. This relation has already been derived in the experiments as explained in the previous chapter.

Standing on a different view point about these experimental results, the following conclusion will be derived; the major source of gear noise would be the fluctuating loads arising in the direction of line of action. From the ratiocination about the generating mechanism of gear noise as mentioned above, the intensity of sound would be proportional to fluctuating loads and square of fluctuating frequency.

In accordance with this ratiocination, the experimental results are looked into again. In Fig. 20, sound pressure level is compared with the fluctuating load level of which the base is the value at 500 rpm. Over the natural frequency of gear system, dynamic loads gradually decrease except higher order resonances, but noise is inclined to increase slowly. Under 1200 rpm, sound pressure level is approximately proportional to $\omega^2 W_d$. But at high speed over 1 200 rpm, that relation no longer holds good. The reason is thought as the acoustic efficiencies of sound radiation decreasing at higher frequencies.
6. Conclusion

As the result of experimental studies about the generating mechanism, the following conclusion was derived.

(1) The largest noise source is load fluctuations in the direction of line of contact.
(2) Sound intensity is proportional to fluctuating load and square of fluctuating frequency, except in the region of higher frequency.
(3) Tooth profile error has much effects upon noise, especially under the light load.
(4) Separation occurs at a certain speed for the roughly finished gear, dynamic load and noise suddenly increase, and a strange noise which has subharmonic frequency is occasionally produced.

7. Acknowledgement

The author would like to express hearty gratitude for the encouragements and helpful suggestions to Dr. T. Nakada, Professor of Tokyo Institute of Technology, and Dr. J. Ishikawa, Professor of the same Institute. Gratitude is also extended for many helpful discussions to Dr. M. Akeyama, Dr. M. Utagawa and other members of Mechanical Engineering Research Laboratory of Hitachi Ltd. All experimental work was done with kind cooperation of Mr. Tanikawa, a member of the same Laboratory.

References

(1) G. Niemann und H. Glaubits: VDI-Z, Bd. 93, Nr. 6 (1953), S. 215.
(2) G. Niemann und M. Unterbogar: VDI-Z, Bd. 101, Nr. 6 (1962), S. 201.
(3) H. Winter: VDI-Z, Bd. 104, Nr. 6 (1962), S. 237.
(4) H. Zink: VDI-Z, Bd. 98, Nr. 8 (1956), S. 297.
(5) H. Glaubits und K. Gosele: ATZ, Bd. 45 (1942), S. 175.

539. 388. 24. 01 : 621. 98. 011 : 621. 891. 2

The Effects of Lubrication on the Press Forming Limits of Sheet Metals*

By Kiyota YOSHIDA**, Kunio MIYAUCHI***, and Hiroshi KOMORIDA****

The results of mathematical analyses or experiments of punch-stretching and deep drawing have been examined to obtain a systematic understanding of lubrication effects on the press forming limits of sheet metals. For this purpose, the relations of lubrication effects to size and shape effects in press forming have mainly been investigated.

The limiting stretching depth depends strongly upon workhardenability of sheet metal, but the correlation between them is weakened by the presence of friction in the round-bottomed punch-stretching rather than in the flat-bottomed one.

In non-axisymmetric punch-stretching, there is the accumulation effect which is caused by the combination of whole and local geometries of a pressed part and is also affected by lubrication. In deep drawing, the lubrication effect on the limiting drawing ratio is complicated and is generally stronger in non-axisymmetric drawing than in axisymmetric one.

1. Introduction

Most of the already-published papers on lub-