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Noise of Gears

H. Opitz

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Noise of gears

By H. Opitz
University of Aachen

Introduction

In the past there were mainly two important factors in determining the performance of gear units: load capacity and fatigue life. In recent years, however, the noise radiation of running gears is of growing importance. It is now becoming impossible to design gears without considering the significance of this new factor.

While there are many data available in the literature for calculating the permissible load, speed, and fatigue life, no methods have so far been developed for a true prediction of the radiated gear noise. A definite noise level, however, is often subject to guarantee between manufacturer and customer. To meet these requirements two fundamental factors must be resolved. First, both partners must set up the conditions for acceptance tests, and secondly, the gear manufacturer must have the know-how to meet these particular requirements. Investigations were therefore, made to find out the main factors influencing gear noise in order to be able to reduce it. In addition eighty installed gear-units were tested to set up classifications regarding generated noise and operating conditions.

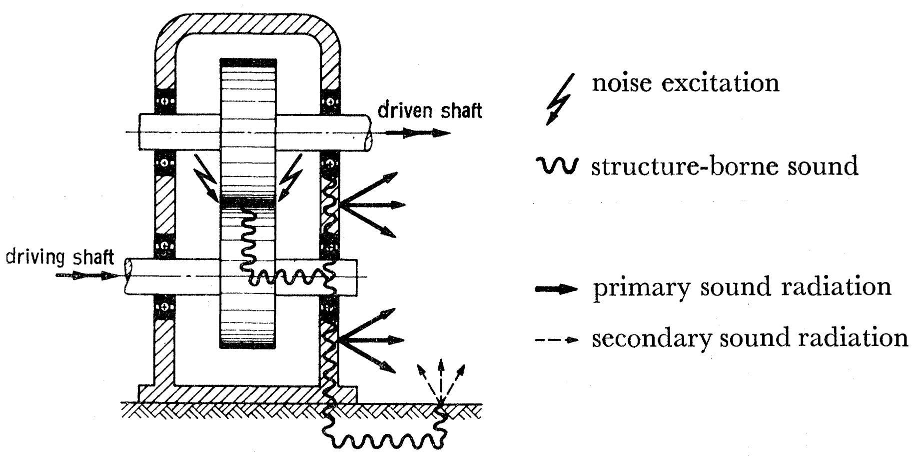

Figure 1. Factors influencing noise level.

In this paper I would like to demonstrate the main factors influencing gear noise and one way of setting up a classification system. These factors controlling gear noise are illustrated in figure 1. In an ordinary gear-unit the region of tooth contact is the source of noise, and consequently the arising sound power is subtracted from the energy flow through the gears. One part of this rather small lost power is immediately radiated by the wheel bodies, the other major part proceeds as structure-borne sound through hubs, shafts, and bearings to the housing walls. There it is radiated as airborne sound. In addition, part of this vibration energy passes through the base, thus causing floor noise-radiation, too. As this secondary sound is not specifically related to the gear unit and depends only on its mounting and floor characteristics, I will go deeper into the details, but confine myself to noise excitation, sound transmission, and radiation by parts of the gear-unit itself.

Noise excitation

The portion of energy separated in the tooth contact area from the mechanical energy flow is in the form of a torsional vibration, and is chiefly responsible for noise excitation. By means of a seismic torsional displacement pick-up, the transmission error of gears caused

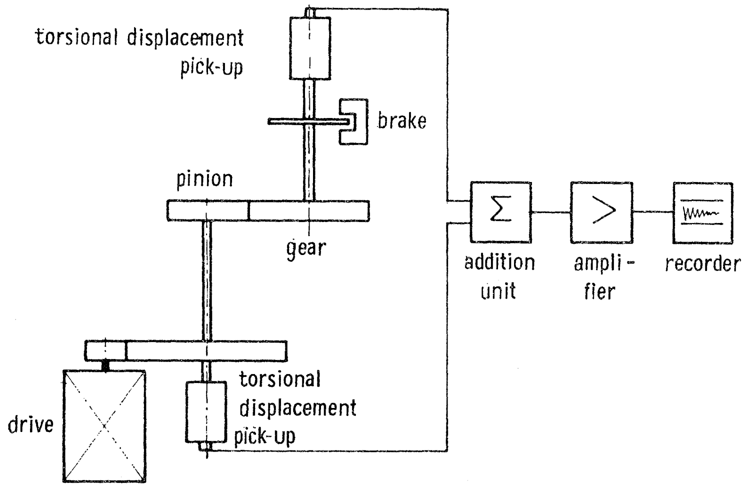

FIGURE 2. Measuring system for transmission error.

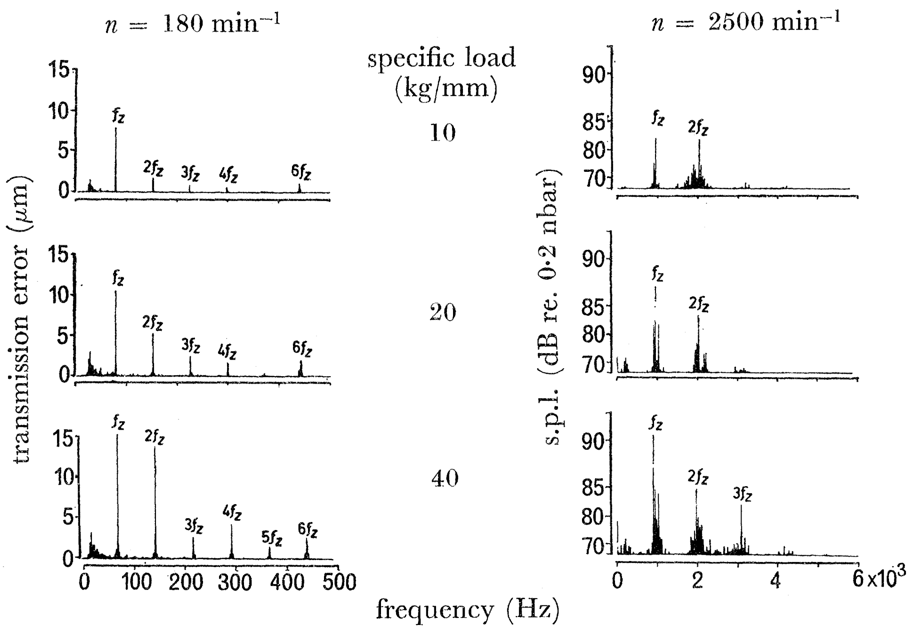

FIGURE 3. Comparison between transmission error and noise.
Gear data: $m = 5$ mm; $z = 25$; $i = 1$; $\beta_0 = 0°$; $b = 10$ mm.

by these torsional vibrations is easily measured. Figure 2 illustrates this measuring system. This device allows an accurate examination of a gear-unit under load, two instruments being used with their housings rigidly mounted to the gear and the pinion shaft. The seismic mass of the pick up is mounted frictionless in the instrument housing which is

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achieved by a cross-spring pivot. Because of its inertia, the seismic mass rotates at the mean rotational speed of the gear, while the instrument housing follows the non-uniform motion of the gear. The relative motion between seismic mass and housing is measured by means of inductive transducers and this indicates the non-uniformity of rotation. For noise investigations only those transmission errors are of interest whose origin lies within the gearing; it is therefore essential to eliminate those errors whose source may be outside, such as drive or brake. This is attained by summing up the two signals in an adder-subtractor unit. This, of course, requires a true measurement of amplitude and phase angle with both instruments. Figure 3 illustrates the relation between transmission error and gear noise. On the left-hand side a narrow-band frequency analysis of the transmission error at different gear loads is shown, on the right the corresponding noise analyses. These analyses demonstrate that the results of both measurements are of similar shape, i.e. both are characterized by mesh frequency and its harmonics.

Influence of load

In addition, these analyses show that the amplitudes of transmission errors as well as of gear noise components rise with increasing specific load. According to the alternating stiffness between single and double contact of teeth, there is a periodic change in tooth deflexion thus disturbing the theoretical tooth engagement. This abrupt change in deflexion causes vibrations which influence the transmission error as well as gear noise.

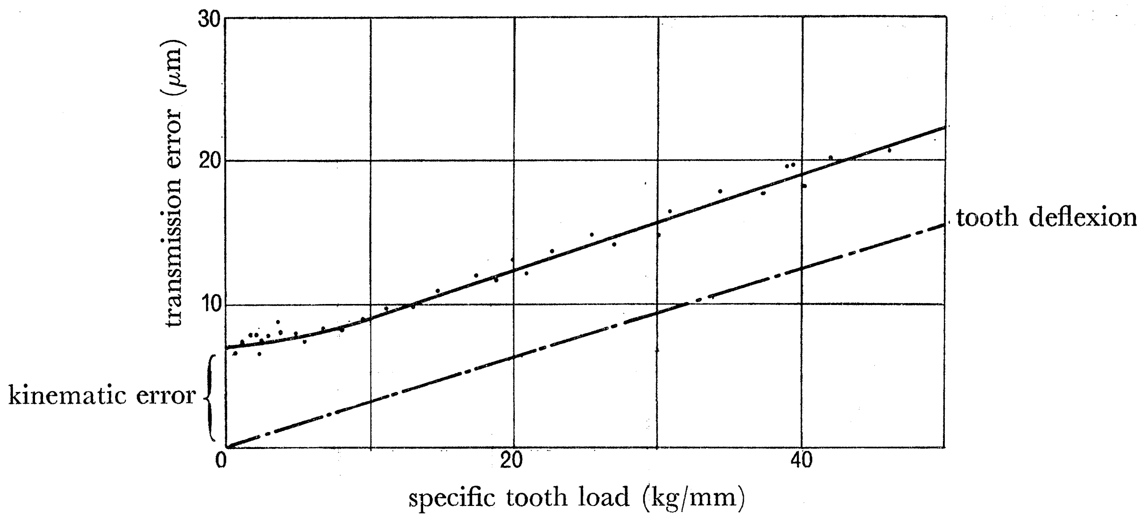

Figure 4. Influence of tooth load on transmission error. Gear data: $m = 5$ mm; $z = 25$; $i = 1$; $\beta_0 = 0°$; $b = 30$ mm; $f_f < 6$ μm; $f_t < 5$ μm.

Figure 4 shows the influence of specific tooth load on the transmission error. The dotted line indicates the deflexion of an ideal pair of teeth at different loads. The full line shows the measured transmission error of a high-quality gear. With the exception of the first part of the curve up to about 8 kg/mm the transmission error runs parallel to increasing tooth deflexion. The distance between both lines corresponds to the kinematic error and represents the difference between ideal and actual flank forms.

In figure 5 the gear noise is plotted as a function of tooth load. It shows a linear relation between sound-pressure level and specific tooth load, similar to the transmission error characteristic presented in figure 4. It may be noticed that the noise level increases on an average 3 dB with doubling the load.

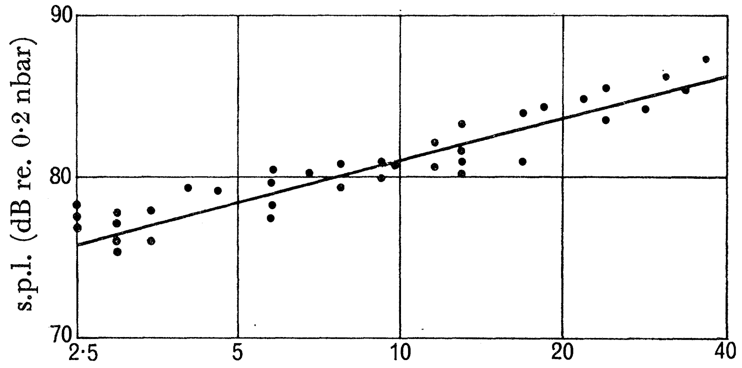

FIGURE 5. Influence of tooth load on noise. Speed range of gears.: $n = 500$ to 3000 rev/min. Gear data: $m = 5$ mm; $z = 25$; $i = 1$; $\beta_0 = 0°$; $b = 30$ mm; $f_f < 6\ \mu$m; $f_t < 5\ \mu$m.

INFLUENCE OF SPEED

The problems dealt with so far only represent the quasi stationary process of tooth engagement. As gear noise, however, results from radiated vibration energy in the mesh region it is necessary to know also the influence of speed. This influence on gear noise is illustrated in figure 6. At a specific load of 9 and 18 kg/mm the diagram gives the relation

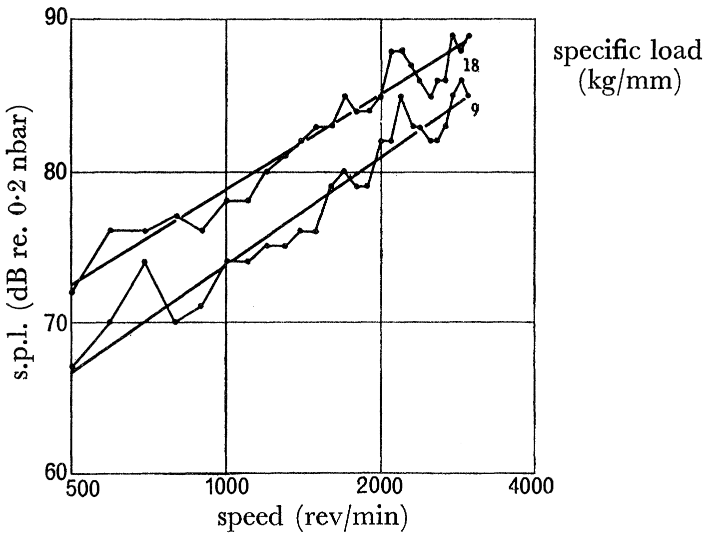

FIGURE 6. Influence of speed on noise. Gear data: $m = 5$ mm; $z = 25$; $i = 1$; $\beta_0 = 0°$; $b = 30$ mm; $f_f < 6\ \mu$m; $f_t < 5\ \mu$m.

between sound-pressure level and speed in the range from 500 to 3000 rev/min corresponding to a circumferential speed of 3 to 18 m/s. On an average the pressure level increases 6 dB when doubling the speed. This increase does not depend on load as both curves demonstrate. The extreme irregularities in the diagram at 700 and 2200 rev/min are caused by the natural frequencies of parts of the test stand.

Influence of face width

As illustrated above, there is an important influence of gear load on gear noise. There are two means to decrease noise excitation: enlarging the face width or extending the contact ratio. By enlarging the face width the specific load is reduced for constant torque. As previously stated, a lower specific load causes lower tooth deflexion, i.e. less noise excitation.

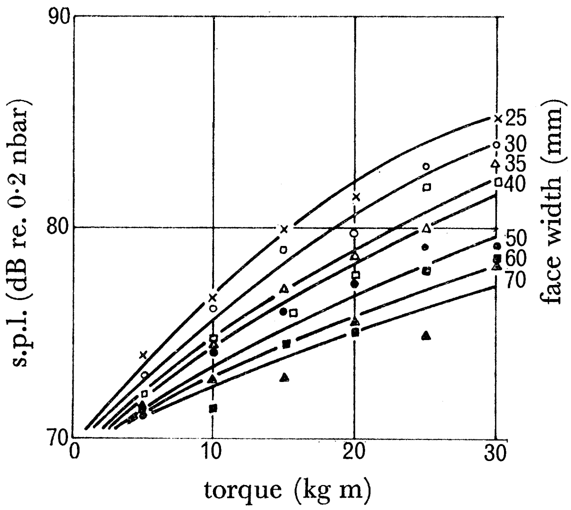

Figure 7. Face width and sound radiation. Gear data: $m_n = 5$ mm; $z = 25$; $i = 1$; $\beta_0 = 20°$; speed range: $n = 2700$ to 3000 rev/min.

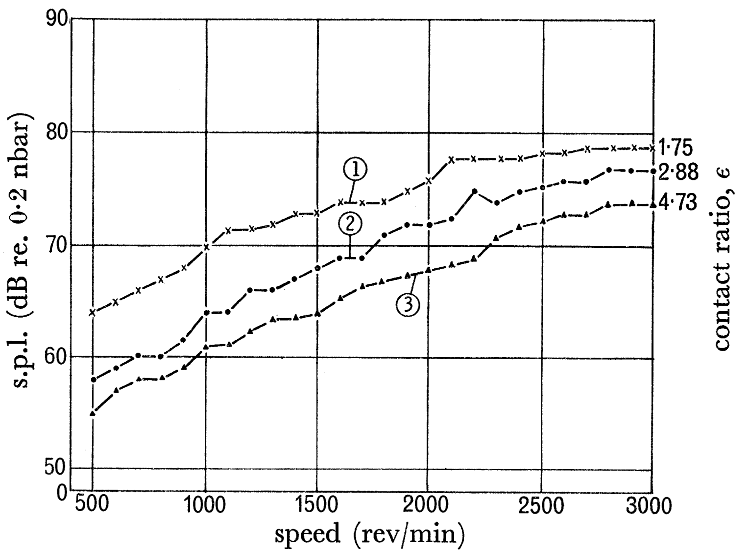

Figure 8. Influence of contact ratio on noise. Gear data: (1) $m = 2$ mm; $z_1 = 48$; $z_2 = 77$; $b = 40$ mm; $\beta_0 = 0°$. (2) $m_n = 2$ mm; $z_1 = 47$; $z_2 = 66$; $b = 40$ mm; $\beta_0 = 10°$. (3) $m_n = 2$ mm; $z_1 = 40$; $z_2 = 65$; $b = 40$ mm; $\beta_0 = 30°$.

In order to investigate this, the face widths of gears were reduced from 70 to 25 mm in steps of 5 mm. The radiated noise of a speed range from 2700 to 3000 rev/min corresponding to a circumferential speed of 17 to 18 m/s was averaged. Figure 7 shows the sound-pressure level against gear torque. All the tests showed an increase of the noise level with enlarged torque. The rate of increase varies, however. The curves of small face widths show a higher gradient than those of wide face widths, corresponding to the different gradients of the deflexion curves. When reducing the face width at constant torque, the noise-level increase rate is of about the same as that found if the torque is enlarged by the corresponding value.

A second reason for varying noise excitation is provided by changing the contact ratio. With larger contact ratio the torsional stiffness of gearing improves in the sense of reduced tooth deflexion. It may also result in avoiding the exclusive contact of only one pair of teeth. Figure 8 gives the relation between contact ratio and noise radiation. The graph shows the sound pressure level as a function of gear speed and contact ratio. It may be seen, for instance, that at a speed of 1500 rev/min the pressure level drops from 73 to 68 dB if the contact ratio is changed from 1·75 to 2·88, if changed up to 4·73 the noise level even drops about 9 dB.

Effect of errors

Not only do the deflexions just described prevent an exact engagement, but so also do errors of gearing resulting from inaccurate manufacturing. There are three kinds of error which mainly influence transmission and noise excitation, those of profile, pitch and alinement. Profile errors are defined as the deviations of a given profile from the ideal one. Pitch errors indicate inaccuracies in spacing between two subsequent teeth and the alinement error determines the deviation of tooth lead across the face width.

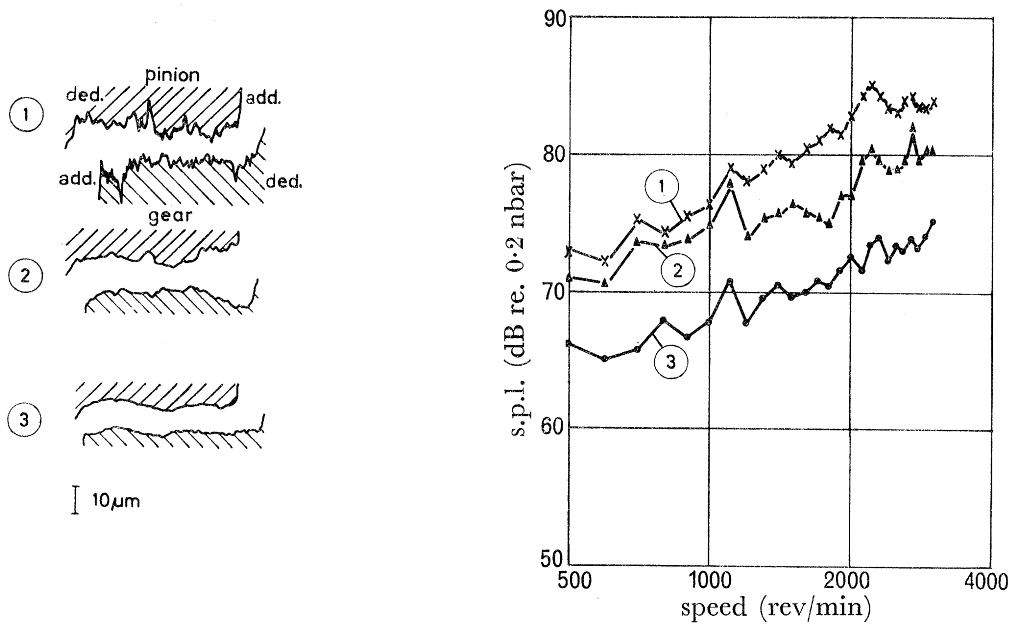

Figure 9. Influence of profile error on noise. Gear data: $m = 2$ mm; $z_1 = 60$; $z_2 = 65$; $\beta_0 = 0°$; $b = 30$ mm; specific load $= 6$ kg/mm.

These errors influence noise excitation of gears. On the left-hand side of figure 9 these involute measurements can be seen. No. 1 is a pair of gears with rough flank surface and a profile error of 15 to 20 μm. Its noise behaviour is characterized by curve 1 on the right-hand side of the figure. Gear no. 2 has a smaller roughness and its profile errors come to about 10 μm. Gear no. 3 was finally lapped and shows a smooth surface with a profile error below 5 μm. As demonstrated by the curves 2 and 3, profile errors have a great influence

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on gear noise. At the same test conditions the noise level of a high-quality gear is about 10 dB lower than that of a gear with high surface roughness.

The second error in gearing having an important influence on noise generation is the pitch error. In figure 10 the sound-pressure level is plotted as a function of specific tooth load for different pitch errors. The error of gear no. 1 is about 50 μm, that of gear no. 2

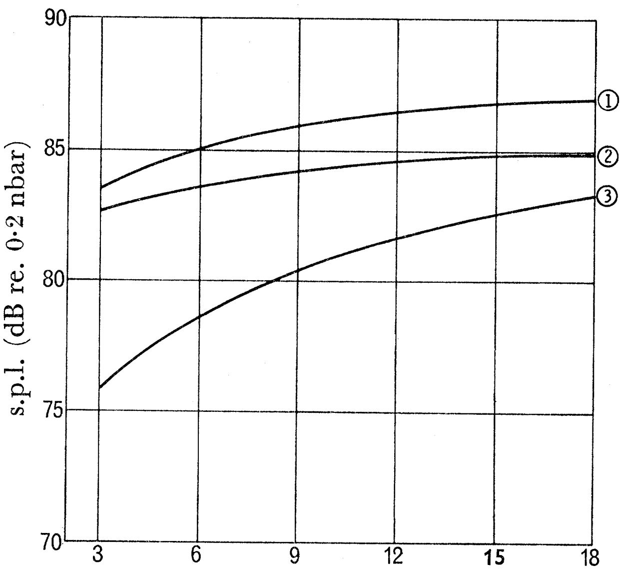

Figure 10. Influence of pitch error on noise. Pitch errors: (1), 50 μm; (2), 40 μm; (3), $<$ 6 μm. Gear data: m = 5 mm; z = 25; i = 1; b = 30 mm; β_0 = 0°.

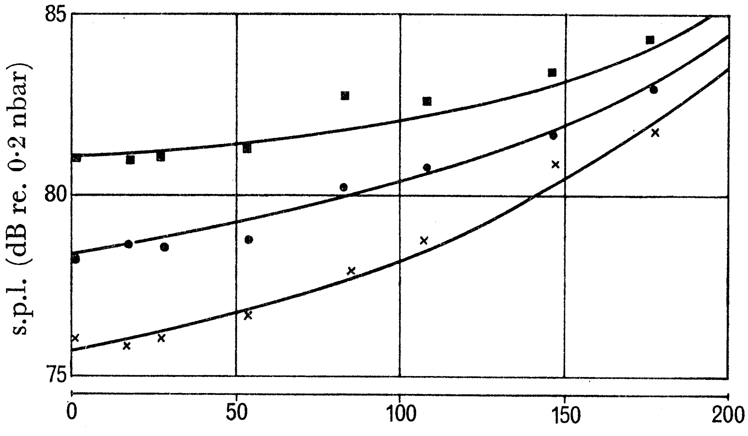

Figure 11. Influence of tooth alinement error on gear noise. Gear data: m = 5 mm; $z_1 = z_2$ = 25; b = 30 mm; β_0 = 0°. Torque (kg m): (×), 5; (●), 10; (■), 15.

is about 40 μm and gear no. 3 has a pitch error of less than 6 μm. Depending on gear load, the noise level radiated by the gear combinations nos. 1 and 2 is about 6 to 12 dB higher than that of the highly accurate gear no. 3. It is interesting to see that the influence of tooth load on the gear noise decreases if pitch errors are present. The different gradient of the curves 1 and 3 is clearly to be noticed. This is due to the inverse influence of pitch errors and tooth deflexions on noise excitation when the specific tooth load increases. In addition, alinement errors have a strong influence on gear noise. If such an error exists in a gear, the power is not transmitted by the total face width. The field of contact is transferred to one or the other end of the teeth and, because of the load concentration at the edge, the tooth deflexion is increased. As shown previously, there is a close connexion between tooth deflexion and noise excitation, so that an increase of the noise level by alinement errors has to be expected. This is confirmed by the figure 11 in which the sound-pressure level is plotted against alinement error for three different torques. When varying the alinement error between 0 and 200 μm/100 mm at an external load of 5 kg m (lower curve) a variation of the noise level of 8 dB is obtained. By increasing the torque this variation is reduced to 6 and 4 dB respectively. This dependency on the external load is explained by the fact that at high torques the misalinement is partly compensated by the tooth deformation corresponding to the theory of ellipticity.

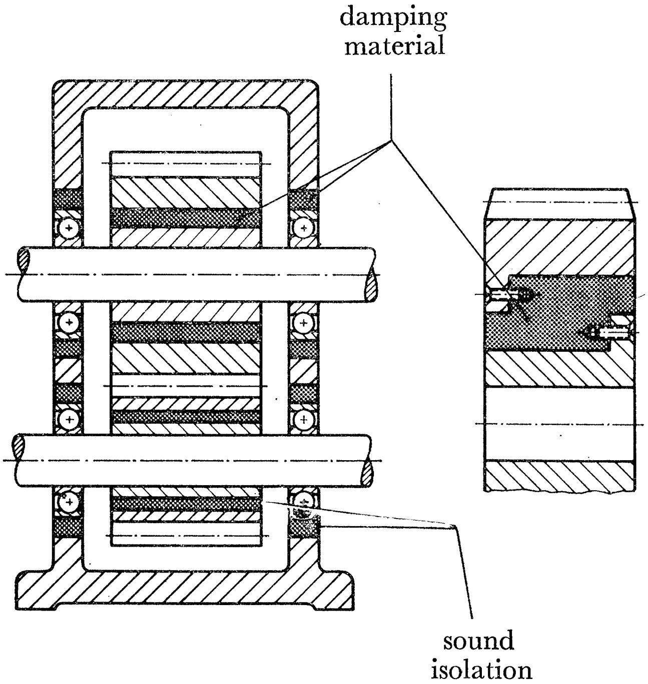

FIGURE 12. Reduction of structure-borne sound by damping material.

Noise transmission and radiation

As noise excitation depends on the geometry and manufacturing accuracy of the gears, a low noise level can be obtained above all by a convenient layout and high quality in production. However, not only the excitation but also the transmission of structure-borne sound affects the noise level. In gear drives the structural vibrations are transferred through the shafts and bearings to the gear box walls where the noise is radiated. By changing the transmission impedance, for instance, by means of damping materials, a sound isolation is obtained thus reducing the noise radiation. The principle application of damping material is shown in figure 12. The points, which are appropriate for the installation of such absorbers are indicated on the left-hand side, whereas on the right a practical example is presented. The structure-borne sound excited by the tooth contact is decreased by the high damping capacity of plastic material between gear rim and hub. The use of such sound isolations in gear drives is very efficient in respect of noise reduction. In highly loaded gears, however, the application of plastics is disadvantageous owing to their mechanical and thermal characteristics. But even in highly loaded gears the transmission of vibrations can be reduced by using suitable types of roller bearings and adapted axial preloads. This can be seen in figure 13, where the noise level generated by the same gear combinations with different bearings is shown. The highest damping effect is achieved by tapered-roller bearings. In comparison with ball bearings the use of this type of bearing results in a noise reduction by 4 to 5 dB. It should be noted that the noise radiation from the bearings themselves does not affect the noise level of the gear unit, since in general the noise level of the bearings is more than 10 dB lower than the total noise level.

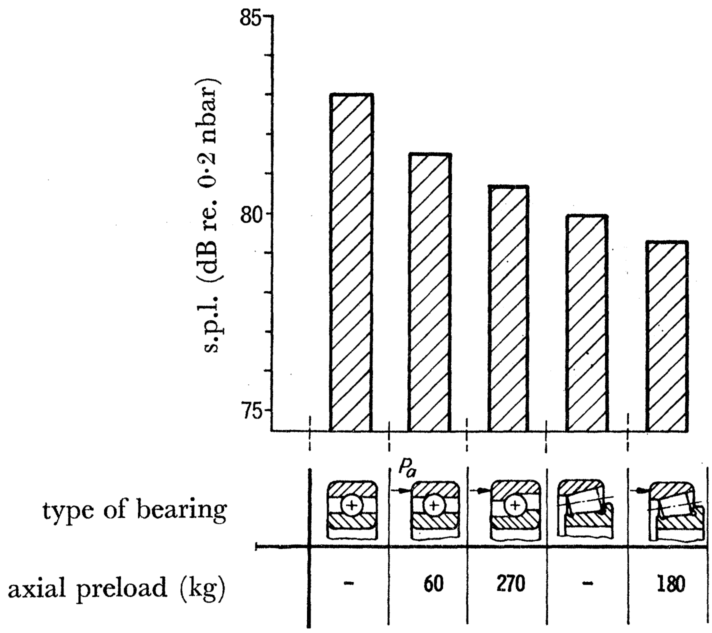

FIGURE 13. Influence of bearings on noise.
Gear data: $m_n = 4{\cdot}5$ mm; $z_1 = 22$; $z_2 = 70$; $b = 125$ mm; $\beta_0 = 10°$.

The noise radiation of the gear box walls depends upon their damping capacity and dynamic behaviour. For instance, resonance effects will occur if the mesh frequency coincides with natural frequencies of the housing. A typical example is shown in figure 14. A passenger car gear generated a marked frequency component of 1000 Hz at certain speeds. It was found that this tone, which is called f_{zv} in the upper narrow band frequency analysis, corresponds to the mesh frequency of the back gear. The frequency response curve of the housing shows two natural frequencies at 1000 and 2500 Hz. After a rib has been added to the left wall of the housing the resonance curve has changed as indicated by the dashed line. The first natural frequency is shifted towards higher frequencies and its amplitude is decreased. The effect of this variation on the noise behaviour is evident from the lower narrow band frequency analysis. It can be seen that the amplitude of the tone f_{zv} drops from 80 to 74 dB.

The damping capacity depends upon wall thickness and material as well as on the design features. Studies on the influence of the gear box proved that the application of cylindrical housings is advantageous for the damping effect. Noise radiation is mainly caused by transverse vibrations of the walls and therefore their resistance to bending is very important. This stiffness is considerably higher for cylindrical shells than for plane walls. In figure 15 two gear units with different boxes are compared. Layout, accuracy and

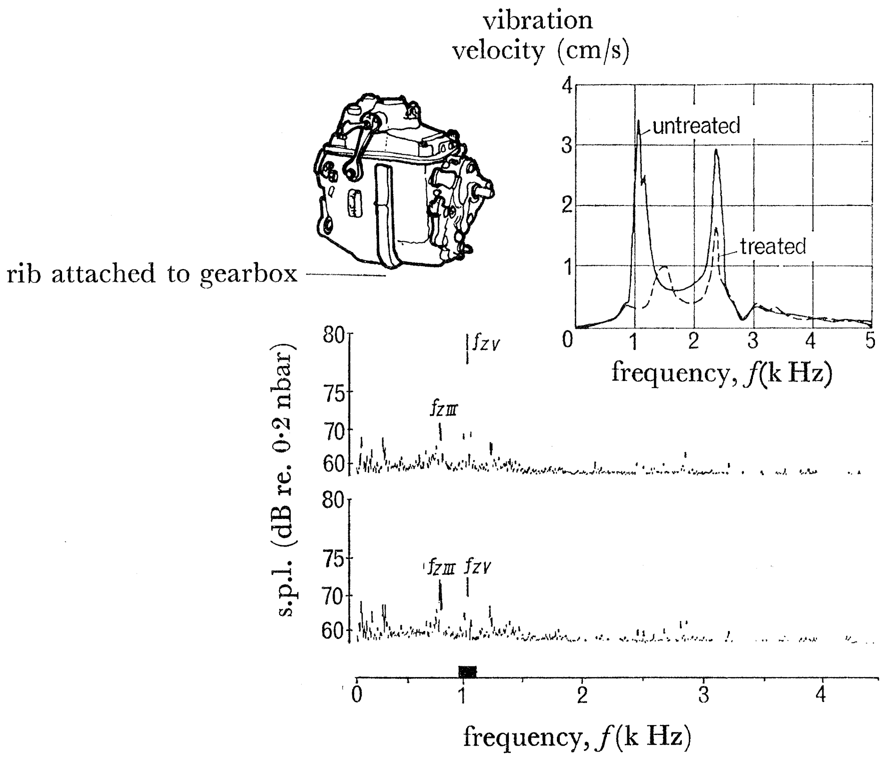

FIGURE 14. Influence of ribbing on vibration and noise.
■, Region of natural frequency of the untreated wall.

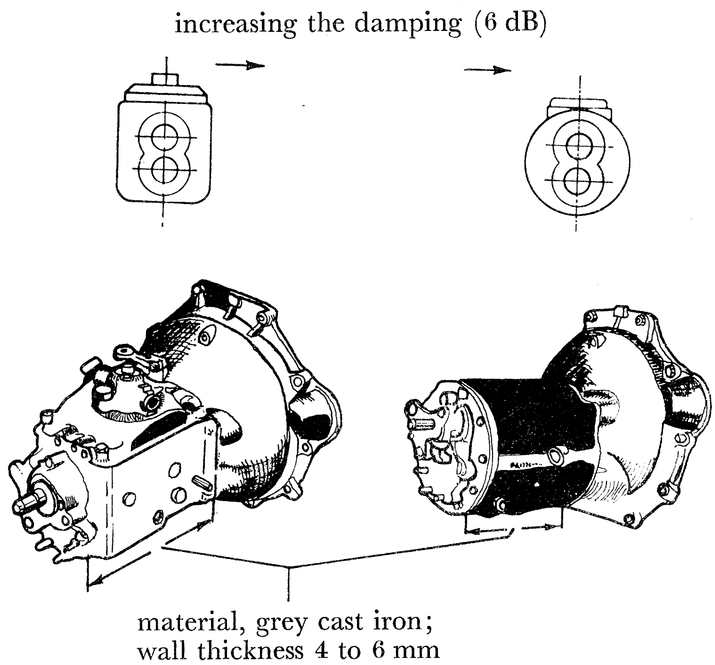

FIGURE 15. Different gear boxes.

material of both gears are the same so that the difference of the generated noise level is caused only by the design variations. It was found that on an average the noise level of the normal box is 6 dB higher than that of the cylindrical box.

Classification

The factors influencing gear noise discussed so far show that the noise radiation depends on running conditions, geometry and manufacturing accuracy. For given running conditions the noise can be affected by the quality of design and manufacturing accuracy. Consequently noise reduction is obtained by increasing the quality of gears and therefore noise radiation may be used for a classification of gear drives. However, the basis of this classification should be present-day manufacturing accuracy. In addition, only those

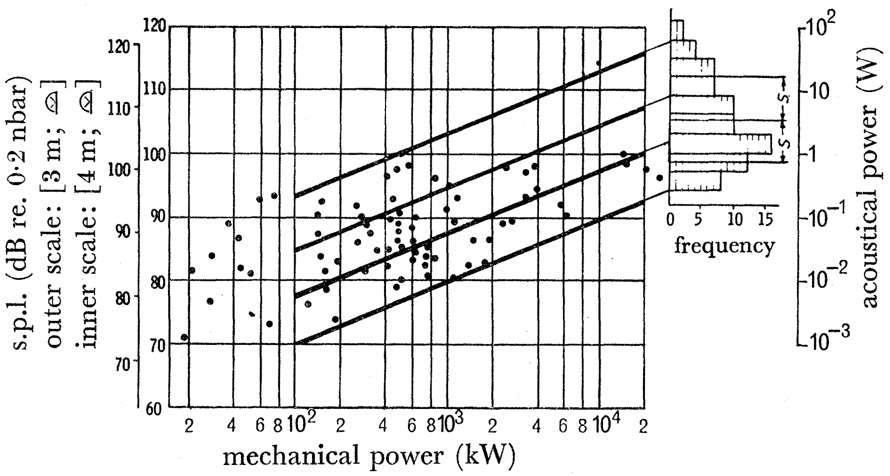

Figure 16. Influence of mechanical power on gear noise.

values should be specified which depend on the gear unit itself. This was achieved by determining the total sound power, thus eliminating the acoustic characteristics of the room and foundation. As gear drives do not radiate uniformly in all directions, the measurements were carried out over a spherical area enclosing the gear. Many investigations in the noise behaviour of a great number of installed gears were carried out in German industry. The results of the measurements are plotted in figure 16. It shows the relationship between mechanical power and sound pressure level. The measurements were carried out under standard condition in order to eliminate the influence of foundations and surroundings. From these special sound levels the acoustical power radiated by the gear box can be calculated and, relating it to the mechanical power, the so-called acoustical efficiency is obtained.

The straight lines in the diagram represent points of equal acoustical efficiency.

With reference to the acoustical efficiency, the quality of gear drives can be rated corresponding to the frequency distribution on the right side of the figure. Five classes were set up as illustrated in figure 17. They are marked *A* to *E*. The different classes may be described as follows.

Class *A*. This noise behaviour cannot be safely obtained, not even by high quality in production. Additional noise-absorbing installations are required.

Class *B*. Result of extremely high manufacturing accuracy.
Class *C*. High manufacturing accuracy required.
Class *D*. Normal manufacturing quality required.
Class *E*. Gear drives with high noise level, easily avoidable by increasing the manufacturing quality.

This classification system for the noise behaviour of gear drives is now widely used as a basis for guarantee discussions between manufacturers and customers.

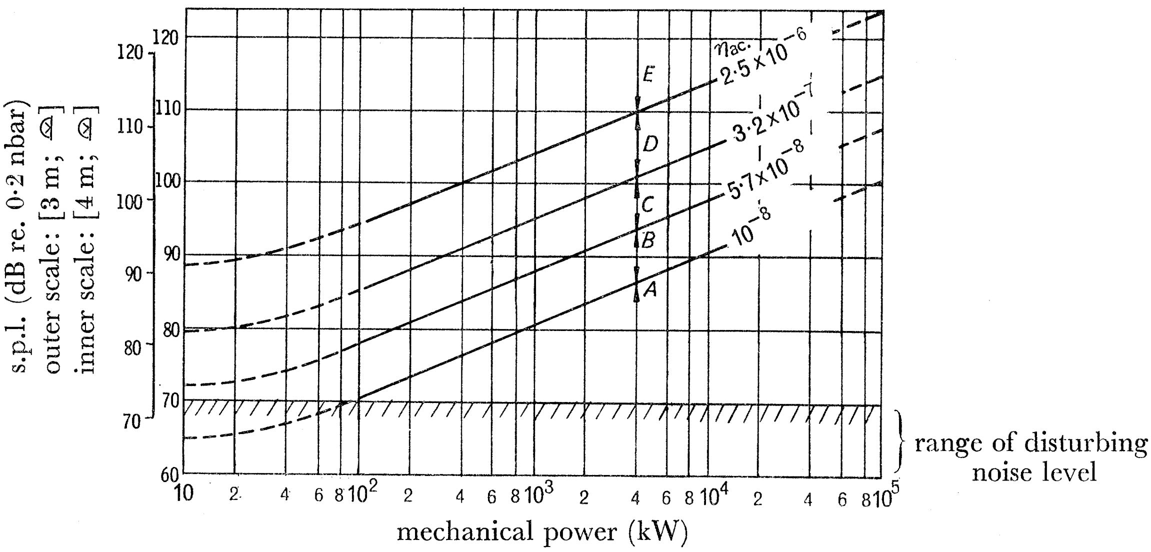

FIGURE 17. Classification (*A* to *E*) of gear noise.

$$\eta_{ac.} = \text{acoustical efficiency} = \frac{\text{acoustical power}}{\text{mechanical power}}.$$

SUMMARY

It has been shown that the excitation in the tooth contact zone, the transmission of structure-borne sound, and the characteristics of the housing are the most important influences on noise behaviour of gear drives. The excitation depends on running conditions, geometry and manufacturing accuracy. For instance, by increasing the gear quality the noise level can be reduced by more than 10 dB in certain cases.

The transmission of structure-borne sound can be diminished by the application of damping materials and suitable types of bearings. Finally the dynamic characteristics and the damping capacity of the gearbox itself affect the noise radiation.

Consequently the production of gear drives with low noise level is a problem of design and manufacturing accuracy. Thus the noise radiation can be used for an appropriate classification of gear units. Such a classification system which is based on the present level of technology has been described.

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