Design Method of Number of Teeth for Low Noise Gears Considering Human Aural Characteristics*

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Abstract

Recently, increased user demand for quiet vehicle environments has resulted in the need to develop low-noise gears that do not generate noises that are harsh on to the human aural. The principal method for reducing meshing transmission error has been to gear design with as many teeth as possible. However, while these design methods have focused on decreasing gear noise, they have not considered the relationship between gear noise and the background noise of vehicle interiors. In fact, it would appear that no research considering the human aural characteristics associated with gear noise under these conditions has been conducted to date. Furthermore, the problem of gear noise on background noise in vehicle has increased as the use of hybrid engine technology in vehicle has become more pervasive in recent years. The proposed technique for reducing gear noise involves gear design that frequency control of gear noise by changing the number of teeth. The technique described here considers human aural limit and the influence of masking, and how these are affected by the tooth settings of low noise gear. This was achieved by assessing human aural characteristics in the unique acoustic environment of the vehicle interior. Then, the relationship between the number of teeth and the sound pressure level of gear noises was investigated. It was found that the proposed method of varying the number of teeth was effective for designing gears with optimal human aural characteristics.

Key words: Gear Noise, Low-Noise Gear, Frequency Control, Gear Design, Human Aural Characteristics

1. Introduction

Recently, user demands for quiet in vehicles have increased. Demand is also increasing for less gear noise, which is almost a pure tone, that is severely offensive to the human
The frequency of gear noise equals the reciprocal frequencies of the teeth during the meshing cycle, and meshing transmission error becomes a factor in generating noise. A previous method for improving the contact ratio was devised by increasing the number of teeth to decrease the meshing transmission error, which also decreased the gear noise. However, for the common problem of high-frequency resonance, decreasing tooth strength for meshing frequency region expansion resulted from an increased number of teeth. Methods for deciding the best parameters for the number of teeth and the strength performance of helical gears have been reported to solve the problem of decreased tooth strength. On the other hand, a method to reduce the effect of high-frequency resonance by reducing the meshing frequency band region by decreasing the number of teeth has also been reported. However, these techniques only focus on decreasing the gear noise without considering its relationship to the total sound in the vehicle.

Consequently, gear noises, ignored up to now, have become conspicuous whenever the vehicle's interior environment improves, creating a problem: hearing the allophone of high frequencies has become easy, especially in hybrid vehicles, because the engine stops when the motor is running and the background noise in the high-frequency region is reduced. The vehicle interior background noise has changed in this way. Therefore, techniques for reducing the vehicle's interior background noise during operation have recently been proposed to change such background noise and improve the tone quality of the voice recognition performance of the navigation system and the audiovisual apparatus.

However, no research systematically shows the best gear design to adjust to the frequency characteristic changes of such background noise for gear noises. Therefore, a method of reducing gear noise by reducing the meshing transmission error margin (by increasing the number of teeth) is not adopted in this research. This study assumes the case where the vehicle interior background noise is greatly improved with the increased introduction of hybrid vehicles, etc., and focuses on a frequency control technique (decreasing the number of teeth) of the gear noise with consideration of the masking noise (background noise in the vehicle). This technique moves the peak value of the gear noise to the low frequency region level when the background noise in the vehicle is high, even though the absolute value of the meshing transmission error margin of the gear increases. This report discusses a basic technique for setting the number of teeth for low noise gears, based on the human aural limit characteristics of the masking influence.

2. Proposed gear design technique

As defined in the Japanese Industrial Standard (JIS), psychological factors are "sound not good" and "sound not preferable". Noise is physiological and is controlled by psychological factors. In this study, the amount of gear noise projection within human aural limits in the meshing frequency (the amount of gear noise or noise heard by the human aural) determines an index value to identify unpleasant sounds. Figure 1 shows the definition of projective measurements $\Delta A$ and $\Delta A'$ from human aural limits, where $A$ and $A'$ represent gear noise measurements with different meshing frequencies. In Fig. 1 the background noise of a vehicle’s interior level and the gear noise audibility level at meshing frequency. As a result, the number of teeth lowers the level of offensive sounds to aural characteristics. In general, the absolute value $A$ of the gear noise is considered a function of the total contact ratio. Since the number of teeth decreases (the reduction ratio is constant) while the center distance is constant and the total contact ratio decreases, this ratio becomes $A'>A$. However, there is a feature of $\Delta A'<\Delta A$ when the frequency characteristics of the human aural limit level under the vehicle’s interior background noise are larger. This study proposes to make $A'>A$ using fewer teeth. That is, by comparing $\Delta A'$ with $\Delta A$ (instead of $A'$ with $A$) this technique optimizes the number of teeth by considering the differences between the human aural limits and the gear noises of background noise to minimize the gear noise.

level ($\Delta A'$, $\Delta A$).

3. Vehicle interior sound when vehicle is running

Figure 2 shows the spectrum of the interior background noise level of the center of the front seat during the regular running of a small passenger vehicle (front engine and front drive system (FF vehicle) equipped with automatic transmission (AT). The frequency characteristics of the vehicle’s interior background noise levels vary, as shown in Fig. 2, depending on model and velocity. In this study used of the interior of vehicle noise vehicles A, B, and C (velocity is 80 km/h), velocity of vehicle is 80, 100, or 120 km/h, and vehicle sensitivity is low (vehicle model is A), middle (vehicle model is B), or high (vehicle model is C). The lower right gradient of the luxury vehicle’s (vehicle model is C) sound level will increase. However, the spectrum characteristics of the vehicle’s interior background noise levels during regular running generally do not change even if the vehicle model and velocity are different. Tomita et al. (10) confirmed that approximating the sound spectrum of the background noise in the vehicle’s interior is possible because white noise passes through a low-pass filter. The background noise uses the sound in which white noise passed through a low-pass filter cut off frequency 1000 Hz and an attenuation characteristic -24 dB/oct (Fig. 3).

![Diagram](image_url)

Fig. 1 Definition of $\Delta A$ and $\Delta A'$

![Graph](image_url)

(a) Difference of vehicle model (80 km/h)

![Graph](image_url)

(b) Difference of Speed (Vehicle B)

Fig. 2 Examples of background noise in vehicles
4. Calculation of level of human aural limits under vehicle's interior sound environment

4・1 Human aural characteristics (masking)

Sounds clearly heard when the surroundings are quiet are not as easily heard when environmental noise increases. Such masking is a phenomenon of the minimum human aural value rising from obstruction sounds and it equals amount of masking (12). This obstruction sound is specifically referred to as background noise, which is the noise heard when there are no specific "object sounds". In this research, a vehicle’s interior sound was used as background noise, and the gear noise (approximated by the pure tone) was used as object sound to investigate masking. In a related study, Wegel and Lane investigated the masking of a pure tone by another pure tone (13). Fletcher and Zwicker studied the masking of a pure tone by white noise. These studies are based on aural theory (13). However, no research has demonstrated the masking of a pure tone by consecutive spectrum noise, such as a vehicle’s interior sound during regular running. Therefore, this study investigated the masking of a pure tone under a sound environment where white noise first passes through a low-pass filter and then compared the results with Fletcher’s experimental results.

Also, pure-tone masking under the sound environment of a vehicle’s interior sound while running was investigated, and the human aural limit level of the gear noise was calculated. This investigation evaluated gear noise by considering masking.

4・2 Sound level weighing device

A noise meter (Ono Sokki LA-1210) was used for the sound level measurements at A characteristics. Frequency analysis was performed using an FFT analyzer (Ono Sokki CF-350).

4・3 Method of calculating human aural limit level

The amount of projection ΔB was measured by object sound projection detection under background noise. This technique is shown below.

(1) A large pure tone is output and then gradually reduced.

(2) Sound pressure level ΔB+B is not heard. Amount ΔB of the projection from the background noise of the object sound is detected and measured (Figs. 1 or 4). Five subjects measured five times, and their values are the mean. A vehicle’s interior sound under different running conditions was recorded with a DAT data recorder (TEAC Company RD-130TE) at the center of the seat. This sound was used as background noise. For evaluation, the gear noise was assumed to be a pure tone, and its sine wave (100 Hz ratio as 100 Hz-5000 Hz) was generated by a sampling generator to resemble a gear noise. The background noise and a pure tone were synthesized with a mixing amplifier (BOSE 2705 MX) and then output. The experiment was performed through headphones (Sony MDR-Z900).
4.4 Experiment results

Three methods are devised for the output measurement of the pure tone: from the high side, from the low side, and random. The value of projection amount \( \Delta B \) tends to increase in the order of the high side, low side, and then random measurement. In this study measurement method from the high side was adopted for this experiment.

4.4.1 Reproduction technique of sound environment using white noise

The measurement results based on Fig. 3’s sound environment are shown in Fig. 5. The solid line representing Fletcher’s theory is determined based on an individual’s aural characteristics and is not influenced by the frequency characteristics of the background noise \(^{(14)}\). The \( \Delta B \) (projection level, refer to Fig.4) was measured at white noise and at white noise and that passed through a low-pass filter (refer to Fig. 3). The solid line in Fig. 5 is the \( \Delta B \) at frequency characteristics (equivalent to pure tone as meshing frequency) of the white noise experimentally calculated by Fletcher. Therefore, the sees in detail, in the high-frequency area (over 3000 Hz), the \( \Delta B \) of white noise in which background noise is large shows a higher than the white noise that passed through the low-pass filter. In case of masking white noise, it is consider that at not more than 3000Hz critical bandwidth indicate just about constant and at more than 3000Hz critical bandwidth increase with center frequency \(^{(15)}\). For the frequency characteristics of \( \Delta B \) in the actual experiment, no influence to the white noise was indicated, but the same tendency proposed by Fletcher was obtained (refer to Fig. 5). Moreover, when our results were compared with Fletcher’s results, the frequency characteristics seem to correspond well. However, the sound level is smaller overall, probably because detecting pure tones using headphones was easier in this experiment, which assumed a simple anechoic room. According to Fletcher's theory, when the continuous spectrum of white noise is assumed to be background noise, the human aural limit for pure tones can be calculated by adding the solid line value (of Fletcher's calculation) to the noise level \(^{(15)}\). In the next paragraph, the audibility of the gear noise is calculated with the interior noise of vehicle as background noise.
4.4 Vehicle interior sound environment considering the differential between the vehicle models and velocity

An actual interior noise of vehicle was used as background noise to investigate masking under realistic conditions. The Fig.6 shows the relation between the interior noise of vehicle at vehicle models are A, B and C (at the speed of 80km/h). The background noise spectrum is the same as that shown in Fig. 2. The Fig.7 shows the relationship between the background noise (B) for the vehicle model (A and C at the speed 80 km/h) and human aural limit of pure tone (B+ΔB). The human aural limit of gear noise (B+ΔB) can be calculated either by adding the value of ΔB, which does not influence the difference of the background noise, to interior of vehicle background noise level (B), or by applying Fletcher’s theory. The frequency characteristic of the human aural limit of gear noise is decided by the frequency.

5. Calculation of gear noise level

It is necessary to compare the human aural limit spectrum calculated in Section 4 with the sound pressure level spectrum of the gear noise to calculate the gear parameters that comprise the best use of the masking effect. It is necessary to lead the relation of the sound pressure level change in the gear noise to the meshing frequency control (changing the number of teeth). The contact ratio is assumed to be a parameter, and the equation of the sound pressure level in decibels of the gear noise is led. The general type, Eq. (1), can be transformed into Eq. (2) for the contact ratio of the spur gear pair of speed reduction ratio 1 (refer to Fig. 8). Here, \( \varepsilon_v \) is a transverse contact ratio, \( m_v \) is the transverse module, \( \alpha_h \) is the transverse pressure angle, \( \alpha_c \) is the normal pressure angle, \( d_k \) is the tip diameter, \( d_b \) is the base circle, \( z \) is the number of teeth, and 1 and 2 of figure subscript the drive gear and driven gear.

\[
\varepsilon_v = \frac{\sqrt{(d_{k2} / 2)^2 - (d_{k2} / 2)^2} + \sqrt{(d_{k1} / 2)^2 - (d_{k1} / 2)^2} - a \sin \alpha_h}{m_v \cos \alpha_c}
\] (1)
If the pressure angle is assumed to be constant, contact ratio \( \varepsilon_v \) becomes a function only of the number of teeth, \( z \). The contact ratio can be shown by the logarithmic function according to Nakata (16). When the pressure angle is 20°, the contact ratio becomes approximation the formula \( C_1 \ln(z) + C_2 \) (refer to Fig. 9). In case of the helical gear, it is study that the overlap ratio in addition to transverse contact ratio. The overlap ratio is given by Eq. (3):

\[
\varepsilon_\beta = \frac{b \sin \beta}{\pi m_n}
\]

where \( \varepsilon_\beta \) is the overlap ratio, \( b \) is the facewidth, and \( m_n \) is a normal module. And normal module \( m_n \) can be shown as the center distance \( a \) and the number of teeth \( z \). In case of the speed reduction ratio is 1, Eq. (3) can be shown by Eq. (4):

\[
\varepsilon_\beta = z \frac{b \tan \beta}{a \pi}
\]

Overlap ratio \( \varepsilon_\beta \) can be shown by linear functions of the number of teeth \( z \), helix angle \( \beta \), face width \( b \), and center distance \( a \) is suppose to be constant. If all internal contacts \( \varepsilon \) of a helical gear pair are shown by functions of the number \( z \) of teeth, the contact ratio becomes Eq. (5):

\[
\varepsilon = \varepsilon_v + \varepsilon_\beta = C_1 \ln(z) + C_2 z + C_3
\]

\( (C_1, C_2, C_3 : \text{Constant}) \)

On the other hand, the meshing force, which is the source of the vibration of the gear noise, are due to influence the rigidity of gear pair and tooth error, etc. Here, the tooth error is supposed to be constant, and so the focus is the rigidity change of gear pair.

The meshing force \( F \) that causes vibration becomes Eq. (6) when supposing that the change of the tooth surface load distribution by changing the contact ratio and meshing force \( F \) are proportion.

\[
F \propto \frac{1}{\varepsilon_1 + \varepsilon_2}
\]

\( (\varepsilon_1 = 1, 2, 3, \cdots, 0 < \varepsilon_2 < 1) \)

where \( \varepsilon \) is the total contact ratio, \( \varepsilon_1 \) is the contact ratio (natural number), and \( \varepsilon_2 \) is a contact ratio (number below the decimal point, including 0). The variation of meshing number can be shown by supposing that the meshing cycle is \( T_z \), as in Fig. 10. The sound pressure level \( A \) (dB) of the gear noise is shown by Eq. (7) because the sound pressure level \( P \) of the gear noise is proportion in the meshing force.

\[
A(dB) \propto \log(P) \propto \log(F)
\]

\[
A(dB) = C_5 \log\left(\frac{C_4}{\varepsilon_1 + \varepsilon_2}\right)
\]

\( (C_4, C_5 : \text{Constant}) \)

Equation (7) becomes Eq. (8) supposing the contact ratio is not an integer. Here, we assume (supposing) the gear pair for the vehicle and that total contact ratio is the range in approximately 3-4. And the sound pressure level \( A \) (dB) of the gear noise decreases without taking the polar value with the increase in the total contact ratio (19).

In addition, it is consider that the sound pressure level of the gear noise does not become 0 dB. Considering the tooth error \( T_\varepsilon \), it can be approximated to the reciprocal of the total
contact ratio, as shown in Fig. 11. It can also be shown by Eq. (9).

\[ A = \frac{T_e}{\varepsilon} = \frac{T_e}{f(z)} = \frac{T_e}{C_1 \ln(z) + C_2 (z) + C_3} \tag{9} \]

In case of the constant velocity of vehicle, the sound pressure level \( A \) of the gear noise can be shown by the number of teeth \( z \) supposing that the pressure angle \( \alpha \), the helix angle \( \beta \) and center distance \( A \) are constant.

**Fig. 8** Section of gear meshing \(^{(16)}\)

**Fig. 9** Contact ratio of normal gears \(^{(16)}\)

**Fig. 10** Time change of meshing number of teeth

**Fig. 11** Time change of meshing number of teeth

6. **Method of gear design of best teeth**

For example, the gear pair of such the speed reduction ratio is approximately 1.1 as the first meshing gear pair of the AT for the FF (refer to Fig.12) are design of the gear
dimension. The dimension is shown below. To pass the hobbing and the shaving process, it is supposed that the top land is supposed to be $1.5\pm0.03$ mm, and the total coefficient is 0.1 or less. And the transverse contact ratio is set the high by enlarging the addendum as much as possible.

In case of the gear dimension, the ratio between the number of teeth and total contact ratio is shown to Fig.13, in addition the pressure angle is $16^\circ$, the facewidth is 20 mm, the helix angle is $35^\circ$, and the center distance is 125 mm. The result of the transverse contact ratio becomes different for a standard gear based on Fig. 13. Since it becomes easy for a gear with a large module to secure the top land. Consequently, the addendum can be enlarged because it is easier to secure the transverse contact ratio at the gear with fewer teeth. The overlap ratio, which can be approximated by the linear function of the number of teeth, increases with additional teeth. The center distance $a$, between the facewidth $b$, and the helix angle $\beta$, are supposed to be constant, as shown above. It is demonstration that the amount of increase of the overlap ratio is strengthened more than the amount of the decrease of the transverse contact ratio at the relation between the total contact ratio and the number of teeth. As a result, it is almost sufficient to use a linear approximation $(\varepsilon = az + b)$ at the number of teeth of the driving gear is about 35-55. Therefore, for such a number of teeth, the sound pressure level in A dB of the gear noise of the vehicle velocity can be determined by Eq. (9) and is shown by Eq. (10) below:

$$A = \frac{T_\varepsilon}{az + b}$$

(9)

$$A = (a,b: \text{Constant})$$

Especially, the ratio between the sound pressure level of gear noise and number of teeth that is to say the meshing frequency is inversely proportional. It is estimate to show in Fig.14 that ration between the gear noise and human aural limit.

It is thought that, the background noise in the high frequency region becomes especially small, as in hybrid vehicles compared with current vehicles. At this time, the number of teeth, which corresponds to the meshing frequency to which gear noise level $\Delta A$ is minimized, can be derived by increasing the number of teeth. The tooth error margin coefficient $\tilde{T}_e$ is decided by the accuracy of the gear at the finish processing. So, it is estimate that the optimum number of teeth is different by the finish processing method.

![Fig. 12 Cross section of automatic transmission](image)
7. Conclusion

We proposed a technique for the design of the number of teeth in vehicles. It considers the human aural characteristics by controlling the frequency of the gear noise by setting the number of teeth when the frequency characteristics of the vehicle interior background noise and sound level change. In particular, the relation of the sound pressure level of gear noise and the number of teeth in the vehicle transmission were considered. The design technique of the number of teeth is effective if the masking noise level of the high frequency region is low.

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