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Low Noise Gear Units Developed by Loading Test Rig in the Anechoic Chamber

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Abstract

Since the noise of the gear unit of the railway vehicle is usually measured by a field test in run- ning situation, it is difficult to separate the noise of a gear unit itself. On the other hand, a bench tester of gear units generally has no loading system. The testing rig at Osaka Steel Works is developed in order to analyze the noise spectrum from gear unit itself under loading condition. It is installed in an anechoic chamber to eliminate the background noise. So we have developed a low noise type gear unit which is changed the tooth profile by using this testing rig.

1. Introduction

With the increase in speed of railway vehicles in recent years, there is growing demand for reduction of noise and vibration of vehicles in operation. Formerly, for evaluating the quietness or vibration characteristics of a driving gear unit, there was no alternative but to subject the completed vehicle to a running test on a commercial line. Besides, it was difficult to measure the noise from the gear unit separately from the other sources of noise. On the other hand, with conventional bench test equipment, the gear unit could not be tested under load. Under these conditions, we developed a loading test rig that allows the analysis of noise from a driving gear unit under load. The loading test rig is installed in an anechoic chamber.

A noise analysis of gear units using the test rig revealed that the noise produced by the meshing of gears was the most prevalent. Therefore, we attempted to reduce the noise from gear units through improvement in the profile of tooth flanks, with tangible results. This paper describes a low-noise gear unit having improved tooth flanks.

2. Development of Loading Test Rig[1)]

2.1 Outline of the loading test rig

The principal specifications of the loading test rig are shown in **Table 1**, and the appearance of the test rig is shown in **Fig. 1**. In response to the ongoing technology development for increasing the speed of the Shinkansen, the test rig is designed such that it permits testing of gear units at vehicle speeds reaching 500 km/h.

In addition, the test rig is provided with the capability to apply the rated torque to the gear unit for evaluating the various characteristics of the gear unit under actual load conditions during vehicle operation at maximum speed. As a result, it has become possible to efficiently evaluate the temperature characteristics, vibration, noise characteristics, and strength of a gear unit under all possible operating conditions—powering, regenerative braking, coasting—prior to the running test using an actual vehicle.

In addition, to accurately evaluate the noise from a gear unit, which is difficult in a conventional running test, the test rig is housed in an anechoic chamber. This has made it possible to accurately examine the sound source and acoustic properties of the gear unit in a space where the level of background noise is sufficiently low.

Table 1 Specifications of the test rig installed in an anechoic chamber

Dimension	W 8.49 m × L 8.39 m × H 7.10 m	
Floor space	71.23 m^2	
Rotating speed	Pinion shaft	Max 10,000 rpm
	Axle	Max 3,000 rpm (eq. 500 km/h)
Motor capacity (continuous)	400 kW	
Dynamo capacity (continuous)	400 kW	
Noise level	Under 40 dB (A) at not operating condition	

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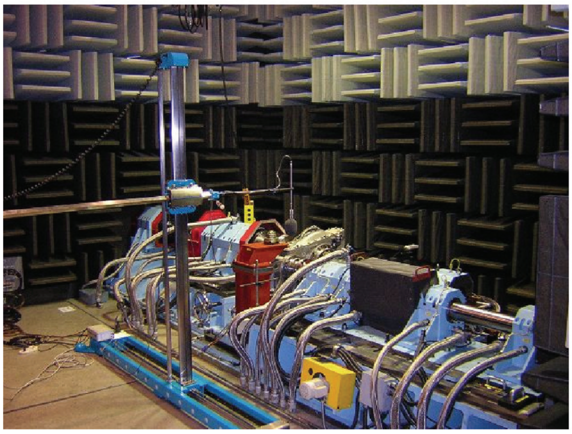

Fig. 1 Test rig installed in an anechoic chamber

To minimize the noise and vibration of the test rig, the driving torque is applied directly to the axle instead of employing a rail wheel system. In addition, sliding bearings are used for the input/output shaft bearings of the test rig. Furthermore, the motor and dynamo are installed outside the anechoic chamber to reduce the level of background noise.

2.2 Concept of reducing vibration and noise of gear units

The construction of a gear unit is shown in **Fig. 2**, and the major causes of noise from a gear unit are shown in **Table 2**. The sources of noise and vibration of a gear unit include meshing of the gears, rolling of the bearings, whirling of the gear coupling, hissing sound of wind, and disturbance from the track. The various types of noise from the above sources directly propagate to the inside and outside of the vehicle. On the other hand, the vibrations that occur in various parts of the gear unit propagate to the bogies and vehicle body via the bearings and gearcase, and thereby produce noise inside the vehicle indirectly.

To reduce the noise and vibration effectively, it is important to clarify their sources and propagation routes. In addition, since the noise and vibration increase through resonance at certain frequencies, it is effective to not only reduce their levels at the sources but also prevent resonance frequencies in the service speed range of trains.

The gear unit has so many resonance frequencies that it is difficult to completely prevent resonance over the entire rotational frequency range. Therefore, it is important to narrow down the frequency range in question.

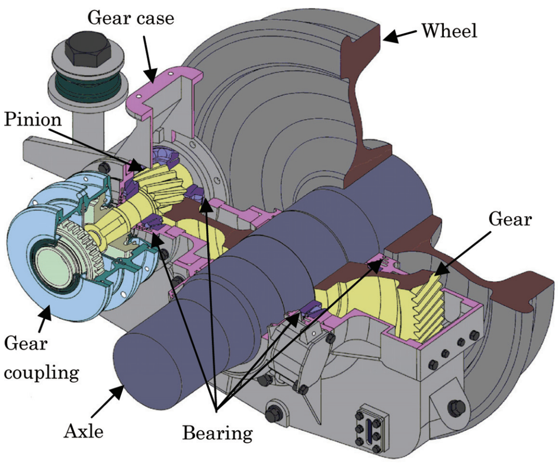

Fig. 2 Structure of the gear unit

Table 2 Main causes of noises of a gear unit

1	Meshing of gear and pinion
2	Rotating of bearings
3	Rotating of gear coupling
4	Wind noise from gear coupling
5	Resonance
6	Others

In a rotation test under load, many test items can be repeatedly measured under stable conditions. Therefore, it can effectively be applied to study and analyze the abovementioned problems.

2.3 Loading test results

Using the loading test rig, we measured the noise and vibration of a gear unit whose specifications are shown in **Table 3** under the test conditions shown in **Table 4**. **Figures 3** and **4** show the levels of noise and vibration acceleration with and without load. As shown in Fig. 3, the level of noise tends to increase with an increase in rotational speed with and without load. However, under load, the noise level reaches a peak at a speed around 4,000 rpm, which is about 15 dB higher than the peak level under no load. Similar peaks in noise level have been observed in a running test using an actual vehicle. As shown in Fig. 4, the vibration acceleration also varies according to the loading condition. Namely, the level of vibration under load is higher than that under no load.

Figure 5 is a three-dimensional graphic representation of the result of a noise frequency analysis. It can be seen that there are noises from components of integer multiples of rotational speed. In particular, the noise of the primary component of meshing is conspicuous.

Figure 6 is a three-dimensional graphic representation of the result of a vibration acceleration frequency analysis. As in the case of noise, the vibration acceleration of the primary component of meshing is especially conspicuous. Thus, the analysis result is similar to that shown in Fig. 5.

Table 3 Specifications of testing gear unit

Gear unit	Module	6
	Pressure angle	26
	Helix angle	20
	Gear ratio	2.7
Coupling type		Gear coupling

Table 4 Testing conditions

Test rig	Loading test rig in the anechoic chamber
Rotating speed at pinion	0 to 6,000 rpm (at fix acceleration)
Loading torque	No loading condition Loading condition (1,100 Nm at dynamo)
Measuring point of noise	1 m over the testing gear unit
Measuring point of vibration	On the surface of upper cover of case

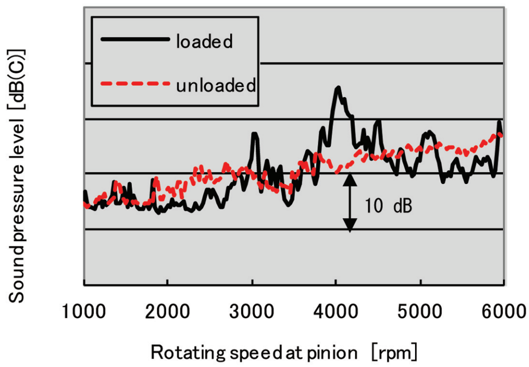

Fig. 3 Difference of sound pressure level in loading and unloading condition (conventional product)

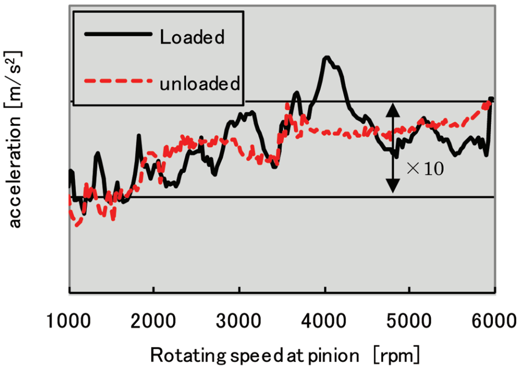

Fig. 4 Difference of acceleration in loading and unloading condition (conventional product)

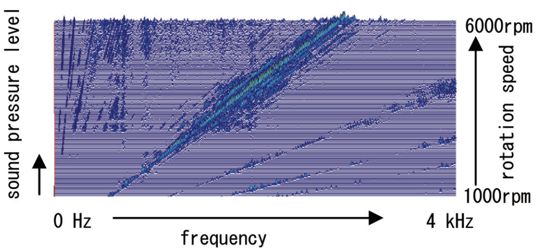

Fig. 5 Frequency analysis of sound pressure level (conventional product)

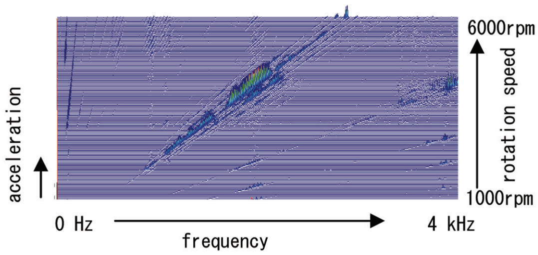

Fig. 6 Frequency analysis of vibratory acceleration (conventional product)

Paying attention to the conspicuousness of the component of meshing in the frequency analysis results, we compared the levels of overall value and primary component of meshing under load in order to confirm the proportion of the meshing component to the overall value. **Figure 7** shows the results for noise, and **Fig. 8** shows the results for vibration acceleration. Under load, the primary component of meshing accounts for almost the overall for both noise and vibration. Thus, it can be judged that the principal components of the measured noise and vibration are ascribable to the meshing of gears.

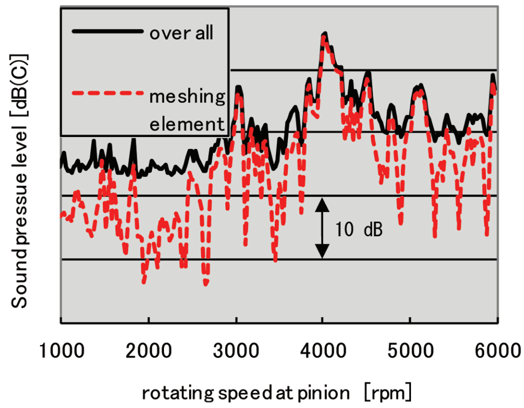

Fig. 7 Comparing the sound pressure level of over all with meshing element (conventional product)

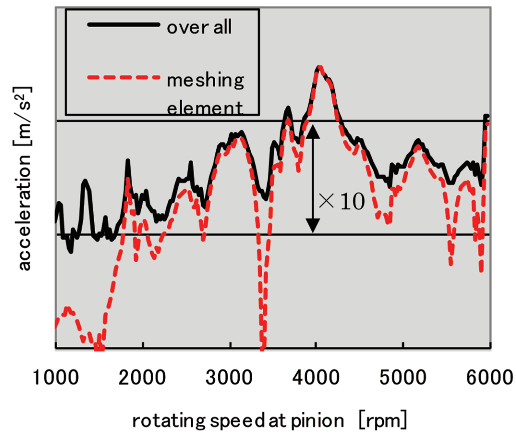

Fig. 8 Comparing the acceleration of over all with meshing element (conventional product)

3. Reduction of Noise by Optimization of Gear Flank Profile

3.1 Content of development

From the study results described in the preceding sections, we found that the noise produced by the gear unit under load was mainly due to the meshing of gears.

The noise produced by the meshing of gears is considered owing to the vibration caused by the contact between the gear flanks during gear rotation. Namely, it is considered that the vibration is caused by the force acting between gear flanks (“exciting force”), which is generated by an uneven rotation (transmission error) during the meshing of gear and pinion. Therefore, considering that reducing the exciting force is effective in lowering the level of noise, we attempted to reduce the gear noise through optimization of the profile of gear flanks.

3.2 Study of optimum gear flank profile

In the conventional schematic representation of the modification of gear flanks, special importance was attached to preventing the edge contact due to gear misalignment. Therefore, the extent of modification was somewhat large. In addition, the flank line and profile modifications were implemented in the same way regardless of the directions of tooth depth and face width. Through an analysis of tooth contacts, we optimized the extent of tooth modification and the profile of gear flank and developed a new method for tooth modification that helps in reducing the exciting force.

3.3 Test results

We fabricated a pinion shaft by tooth modification, which enables the reduction of the exciting force, as described in the preceding subsection, and subjected the gear unit shown in Table 3 (the original pinion shaft was replaced with the new one) to a rotation test under load by the loading test rig in an anechoic chamber.

3.3.1 Results of noise measurement (optimum tooth modification)

Figure 9 is a three-dimensional graphic representation of the noise frequency analysis results. It can be seen that the primary component of gear meshing decreased significantly from the level shown in Fig. 5.

Figure 10 compares the overall noise value with the primary component of gear meshing. The thin line represents the overall values obtained with the conventional tooth profile (see Fig. 7). Since the primary component of gear meshing decreased significantly, the overall value also decreased markedly, and the noise reduced by a maximum of about 15 dB.

3.3.2 Results of vibration measurement (optimum tooth modification)

Figure 11 is a three-dimensional graphic representation of the vibration acceleration frequency analysis results. Both the primary component of gear meshing and the overall value decreased markedly from the level shown in Fig. 6. This result is similar to that of noise. **Figure 12** compares the overall value of exciting vibration with the primary component of meshing. The thin line in the figure represents the overall value obtained with the original pinion shaft (see Fig. 8). The primary component of meshing decreased markedly. The measurement results proved that the optimum tooth modification effectively reduced the levels of noise and vibration of the gear unit.

4. Conclusion

Using the loading test rig, we measured the levels of noise and vibration of a gear unit in an anechoic chamber, and studied measures to reduce the gear unit noise on the basis of the measurement results. The results obtained are as follows.

1) The main component of noise and vibration of the gear unit for the railway vehicle tested under load was the primary component of gear meshing.
2) Through an analysis of tooth contacts, it was possible to reduce the vibration exciting force that causes the noise and vibration of the primary component of gear meshing by decreasing the amount of tooth modification and optimizing the position of modification.
3) We fabricated and evaluated a pinion shaft with an optimum tooth modification profile selected on the basis of the results of an analysis of the vibration exciting force. As a result, the levels of noise and vibration of the primary component of meshing decreased markedly. At the same time, the overall value could also be reduced significantly. Thus, we could confirm that optimizing the tooth modification was effective in lowering the levels of noise and vibration of the gear units.

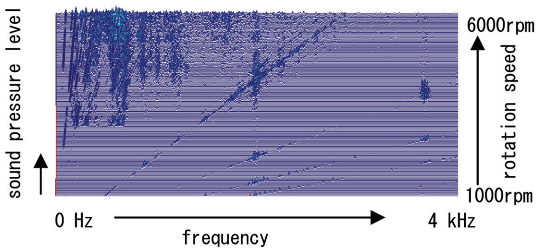

Fig. 9 Frequency analysis of sound pressure level (new type tooth modification

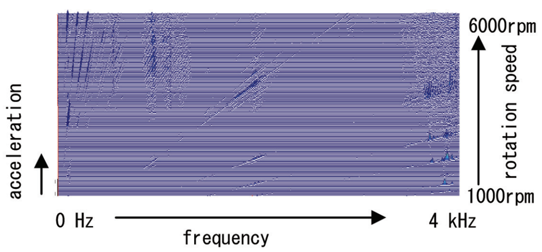

Fig. 11 Comparing the acceleration of over all with meshing element (new type tooth modification)

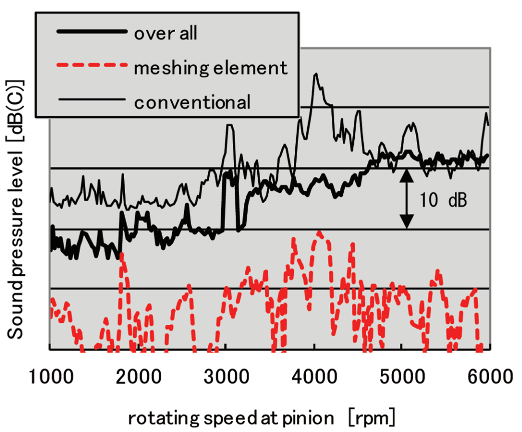

Fig. 10 Comparing the sound pressure level of over all with meshing element (new type tooth modification)

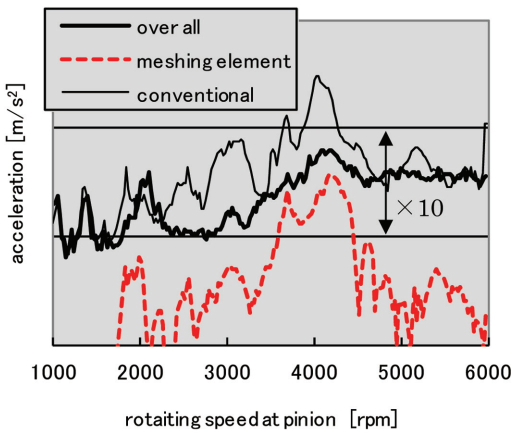

Fig. 12 Comparing the acceleration of over all with meshing element (new type tooth modification)

References

1) Inui, T., Minami, H.: 2004 J-Rail. Development of High-Speed Loading Test Rig for Gear Units of Railway Vehicles. 2004

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