

Establishment of NV Design Technology for Power Transfer Unit Gear Noise

Takuya NISHIKOJI*

Kazuhiro SAITOU*

Yoshimi HIRANO*

ABSTRACT

Prediction technology for NV performance of power transfer units installed in four-wheel-drive vehicles was established at the drawing stage.

A finite element analysis model was created for predicting gear noise generated by the mesh of hypoid gears built into power transfer units. A model of hypoid gears that are a source vibration was created by combining hexahedral elements and shell elements in order to compensate for the mass of the gear teeth and the inertia around the rotating axis. The model of the hypoid gear mesh employed a quintuple-node model that takes tooth rigidity into consideration. This approach achieved a peak frequency gap between prediction and measurement within $\pm 5\%$ and peak level gap within ± 2 dB in the 400–1600 Hz frequency range.

The predictive accuracy and time required for calculation are sufficient to make it possible to examine specifications in the early development stages.

1. Introduction

Low fuel consumption, low noise performance, and off-road drivability are being demanded for SUVs and other similar four-wheel drive (4WD) vehicles. A variety of measures are being taken to reduce overall weight as well as cabin noise. In order to simultaneously satisfy these demands, it is important to investigate specifications that could realize target performance at the design stage of a prototype unit through computer-aided engineering (CAE) analysis, and to determine the specifications with consideration of the effects on other functions. In NV marketability development of 4WD vehicles, therefore, it is necessary to perform CAE analysis to achieve the target for housing vibration of the power transfer unit (PTU). The PTU transfers drive power from the transmission to the rear axle, and the hypoid gear mesh in the PTU is taken to be the source of vibration. Figure 1 shows the transmission, PTU, and hypoid gears.

Prediction methods for PTU housing vibration at the design stage include the use of mesh force (MF), which is the excitation force caused by the gear mesh, and the use of gear-to-housing frequency response function ($GtHFRF$) at the vibration evaluation point on the housing. There has been a report, in which MF , derived from a structural analysis model, in combination with a $GtHFRF$ obtained by actual measurement, efficiently reduced gear noise in

an automatic transmission (AT)⁽¹⁾. In another report, design guidelines for gear noise reduction in a rear differential unit in the 400-600 Hz range were decided using CAE analysis⁽²⁾.

In this paper, CAE analysis is used to obtain both the MF and the $GtHFRF$ for analysis of hypoid gear noise in a PTU for a front-wheel-drive-based 4WD vehicle (FF-4WD). Those results were used to predict the PTU housing vibration level and the vibration characteristics. Reflecting the necessity for decoupling the resonance frequency of body-side transmitting parts and the PTU vibration peaks, the target frequency is defined more broadly than before as from 400 to 1600 Hz. On that basis, this paper will introduce an analysis method that has the accuracy to

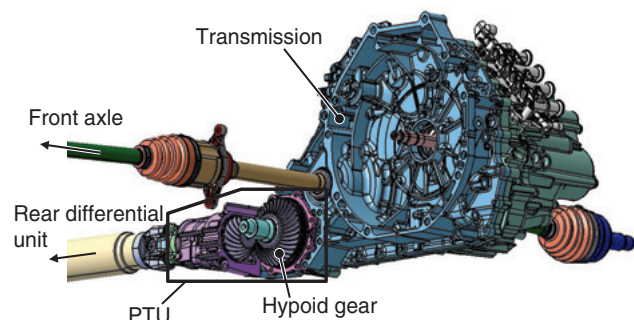


Fig. 1 View of PTU and drive train

* Automobile R&D Center

enable determination of the specification from structural analysis alone, by clarifying the relationship between the design variables that influence housing vibration peaks, the eigenmode, and the operational housing vibration.

2. Development Issues for PTU Gear Noise Marketability

Regarding marketability, in terms of hypoid gear noise, in a PTU for FF-4WD vehicles the target for cabin noise level is achieved by adjusting the peak level and the peak frequency of PTU housing vibration, which is the excitation force input to the vehicle body. The housing vibration level is limited to the set target value or lower, and the housing vibration peak frequencies are decoupled with the resonance frequencies of the propeller shaft and other body-side transmitting parts. The operational vibration level (*OVL*) of the PTU housing that is caused by hypoid gear mesh is shown in Eq. (1) in terms of the *GtHFRF* and transmission error (*TE*) caused by hypoid gear mesh⁽¹⁾. In Eq. (1), *TE* and *k* are real numbers that are not dependent on frequency, while the rest are functions of complex variables.

$$\begin{aligned} OVL &= TE \times \frac{1}{2/k + \phi_1 + \phi_2} \times GtHFRF \\ &= TE \times MF \times GtHFRF \end{aligned} \quad (1)$$

k: Spring constant of teeth

ϕ_1 : Drive-side gear meshing point compliance

ϕ_2 : Driven-side gear meshing point compliance

TE is defined as the relative displacement, advance or delay, that occurs between the tooth surfaces while the gears are meshing. The relationship between the gear path of contact and the *TE* is shown in Fig. 2.

The sound pressure level (*SPL*) of gear noise in a cabin, where PTU hypoid gears are the source of vibration, is

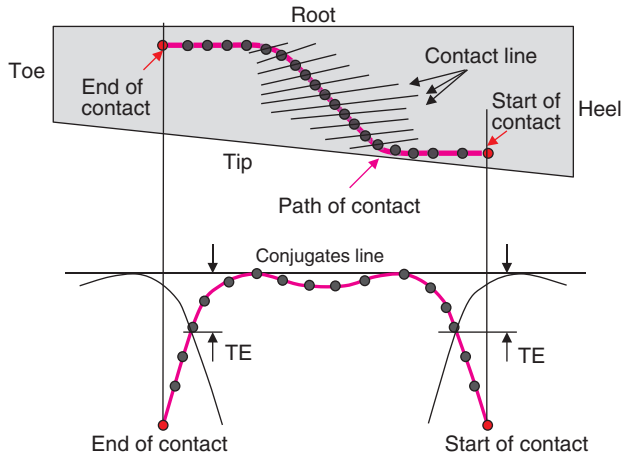


Fig. 2 Transmission error of hypoid gear

expressed in Eq. (2) as the sum of the structure-borne noise and air-borne noise. Here, the structure-borne noise, as expressed by the first term in Eq. (2), is the product of the *OVL* and the acoustic sensitivity between the PTU housing vibration and cabin noise (*ASbVN*). The air-borne noise, as expressed in the second term of Eq. (2), is the product of the near-field volume velocity level around the PTU housing (*NFVVL*) and the acoustic transfer function between the PTU housing and cabin (*ATFbPC*).

$$SPL = \sum_{i=1}^m (OVL \times ASbVN_i) + \sum_{j=1}^n (NFVVL_j \times ATFbPC_j) \quad (2)$$

m: Total number of vibration transfer paths

n: Number of PTU gear noise-radiating surface segments

It is apparent from Eq. (1) that methods for improving the *OVL* include reducing the *TE*, *MF*, or *GtHFRF*. In the past, however, development gave priority to reducing *TE* levels and reducing variation rather than modifying the frequency characteristics of structures. Regarding the reduction of *SPL*, as shown in Eq. (2), in order to reduce the structure-borne noise as discussed above, the main approach in development has been to change the frequency response characteristics of the body-side transmitting parts to adjust the *ASbVN*, thereby satisfying the target values for the system as a whole. Recently, however, market demand for reduction of cabin noise level in 4WD vehicles is so high that the conventional approaches described above are not always sufficient to determine the specifications of the system with positive prospects. Therefore, prediction technology that uses CAE analysis is also necessary for *OVL*, so that frequency characteristics, such as *MF* and *GtHFRF*, can be investigated at the basic layout stage early in development and the specifications required to achieve the targets can be determined on the basis of quantitative judgment.

3. Performance Target of NV Design Technology

In establishing the technology for prediction using CAE analysis, the following targets for predictive accuracy were set, based on the performance sought for this technology as a tool.

- (1) Peak frequency prediction accuracy within $\pm 5\%$ of the frequency in the actual vehicle in order to enable decoupling from the body-side transmitting-part resonance.
- (2) Vibration level prediction accuracy within ± 2 dB of the level of housing vibration necessary to distinguish subjective rating point difference of the cabin noise.
- (3) Considering the natural vibration characteristics of the standard propeller shaft used at Honda, the frequency range that ensures the above predictive accuracy should be from 400 to 1600 Hz.

4. PTU Gear Noise Analysis Model

In order to predict the PTU housing vibration level and corresponding peak frequencies with sufficient accuracy to enable determination of the desired specifications, a finite element analysis model (FEM model) was created. As shown in Eq. (1), calculation of the PTU housing vibration requires MF and $GtHFRF$ to be calculated. In order to accurately calculate the MF in the FEM model for this purpose, it is necessary to have a drivetrain model of the drive side and driven side that includes the mesh points for the PTU hypoid gears. At the same time, in order to accurately calculate the $GtHFRF$, the housing also needs to be modeled so as to include the transmission.

In order to ensure the required prediction accuracy for the frequency range described above, the analysis mesh segment size should be fine to enable expressing the detailed shape of the structure accurately. Then the number of analysis mesh will be increased and the size of model, which influences the calculation time, increases as a whole. Additionally, model calculation time should be considered when determining the parameters of the FEM model. Ideally, the calculation should be completed within several hours to enable efficient countermeasure identification. After taking these various factors into consideration, it was decided to carry out the PTU gear noise analysis using frequency response analysis based on linear eigenvalue analysis in the frequency domain.

4.1. Hypoid Gear Mesh Analysis Model

In order to obtain the MF and $GtHFRF$ of the PTU, an analysis model of the hypoid gear mesh was created. The scope of the analysis model was extended from the mesh points of the hypoid gears inside the PTU to the final gear of the transmission on the drive side, and to the hypoid drive gear in the rear differential through the propeller shaft on the driven side. The main components of the analysis model of hypoid gear mesh are shown in Fig. 3.

When creating the FEM model of hypoid gears, the gear bodies, which were defined by rotating the profile through the root diameter of hypoid gears, were modeled using hexahedral elements in order to achieve a balance between securing the prediction accuracy with regard to bending mode and torsional mode and shortening the calculation time. In order to correct for the mass corresponding to the gear teeth and the inertia around the rotating shaft, the model was configured with a shell element attached to the hexahedral elements described above. The created FEM model is shown in Fig. 4(a).

For the model of the hypoid gear meshing point, on the other hand, a quintuple-node model was configured that takes the rigidity of the hypoid gear teeth into account. A local coordinate system was configured for the purpose of defining the forced displacement input in the direction of the line of action that is determined by the gear dimensions and tooth rigidity. A conceptual drawing of the gear meshing point model that was created is shown in Fig. 4(b). In the

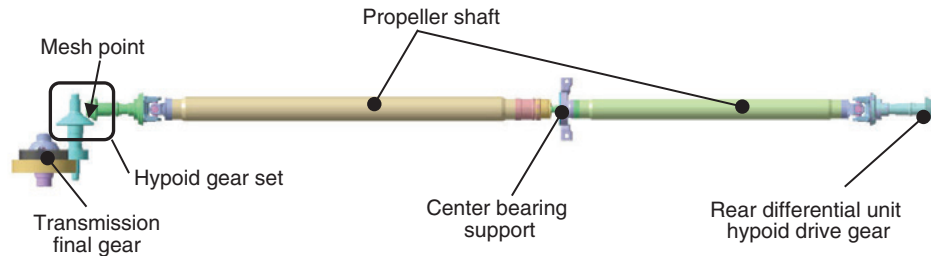


Fig. 3 FEM model for MF analysis

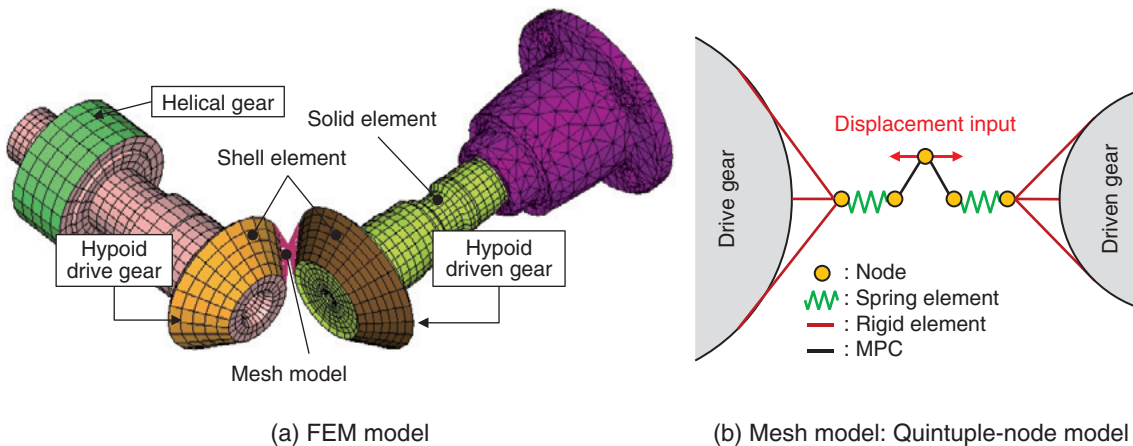


Fig. 4 Hypoid gear mesh model

housing vibration analysis model, which is to be described below, when unit forced displacement input is applied to this gear meshing point, the multi-point constraint force (MPC force) generated at the same meshing point will be the *MF*.

The tooth rigidity used in the created FEM model was identified and defined by comparison of the PTU housing vibration calculation figures derived from the housing vibration analysis, described below, and the measurement results. The peak frequency obtained from the housing vibration analysis by changing the tooth rigidity and from the housing vibration measurement results were compared, with the results shown in Fig. 5. Two types of PTU with differing dimensions of hypoid gears installed in vehicles equipped with V6 engines were used to confirm the relative positioning of their peak frequencies. The PTUs that have respective peak frequencies in housing vibration measurement results at 1100 Hz and 800 Hz were subjected to comparison of estimated peak frequency values obtained when the tooth rigidity was changed. This confirmed that peak frequency obtained by measurement and calculation by using a tooth rigidity value are largely the same. Consequently, it was decided to apply this value for tooth rigidity in vehicles equipped with V6 engines.

4.2. Housing Vibration Analysis Model

In order to calculate the housing vibration that occurs in the PTU when unit forced displacement input is applied to the hypoid gear meshing point, an FEM model of the drive train was created that includes the transmission. The scope of the analysis model was extended from the engine to the rear differential, including the drive shafts, as shown in Fig. 6.

In order to assure prediction accuracy with regard to torsional rigidity, which is important when calculating *MF*, the model was configured to be able to recreate the vibration characteristics of the propeller shaft joints and bearings under actual operating load conditions. As shown in Fig. 7(a), the

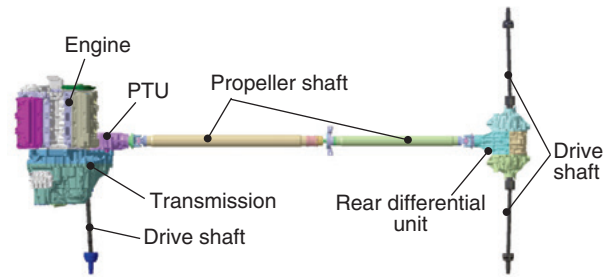


Fig. 6 FEM model for OVL analysis

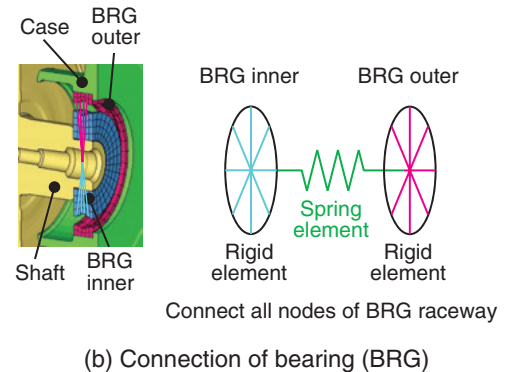
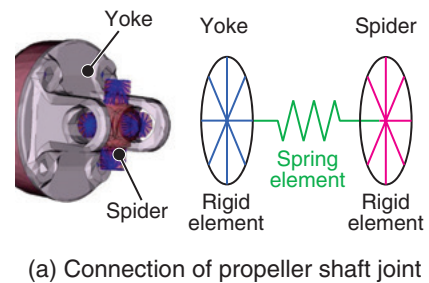


Fig. 7 Parts modeling

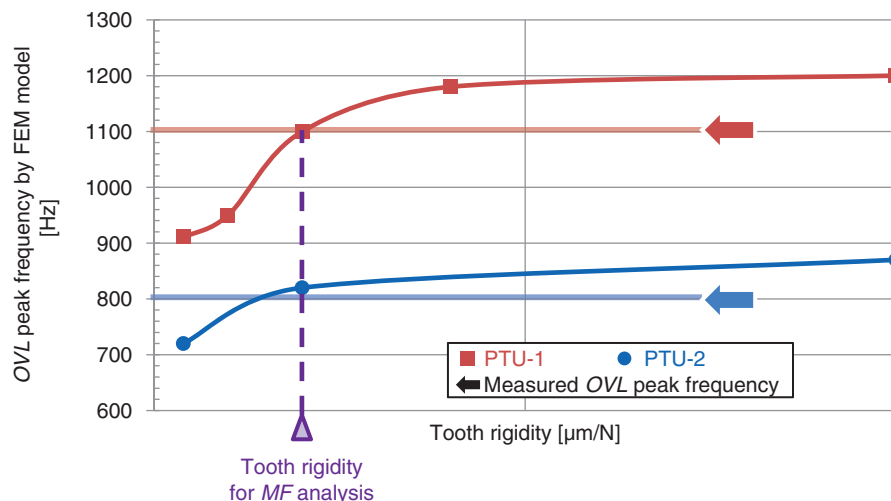


Fig. 5 Correlation between tooth rigidity and OVL peak frequency in calculation

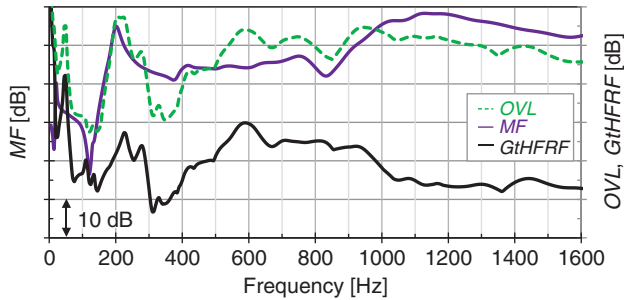
propeller shaft joints have a yoke side and spider side, joined by a spring element. The rigidity value for the spring element used here was identified and configured by correlation with the results of frequency response function (FRF) tests for the joint between yoke side and spider side. As shown in Fig. 7(b), the built-in PTU bearings include inner and outer bearings that were modeled using hexahedral elements and joined using a spring element. The spring element rigidity value used here was derived by static rigidity analysis of the displacement amount with respect to actual operational load in the direction of each load, and that rigidity value was reflected in the model. The FEM models of transmission and PTU housing for *GtHFRF* calculation were constructed along the established guidelines that had verified its prediction accuracy in the past⁽³⁾ and used normally in the gear noise marketability development, and used.

The created FEM modeling method was used to calculate the housing vibration in a PTU applied in vehicles equipped with V6 engines, and the results are shown in Fig. 8. The PTU housing vibration, together with *MF* and *GtHFRF* calculation results, are shown in Fig. 8(a). A model with the drive-side and driven-side mesh points decoupled was also created in order to analyze the factors

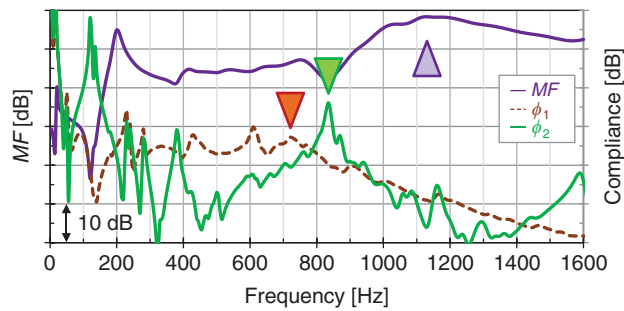
in *MF* peak generation. The results for calculation of ϕ_1 and ϕ_2 compliance are shown in Fig. 8(b). The analysis was carried out with a focus on the *MF* peak that exists in the vicinity of 1200 Hz, which is a resonance frequency of the body-side transmitting parts in the target model. As shown in Eq. (1), the *MF* is the reciprocal of the product of ϕ_1 and ϕ_2 , and it was found, therefore, that the *MF* peak at 1150 Hz is influenced by the 715-Hz mode on the drive side and the 836-Hz mode on the driven side. The calculation results for each mode are shown in Fig. 8(c). In both cases the modes were torsional modes, and it was found that the *MF* peak at 1150 Hz was generated by the torsional mode of the gear on the drive side and the torsional mode of the propeller shaft joint on the driven side.

4.3. Analysis Model Accuracy Verification

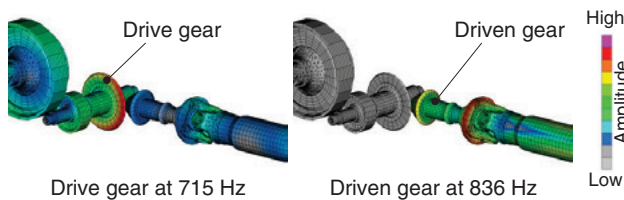
In order to verify whether the FEM model calculation method has the prediction accuracy capable of determining specifications at the layout stage, prior to fabrication of a prototype, a comparison was made of the FEM model calculation values and the vibration measurement results. Since the *MF* cannot be readily measured in an actual vehicle, the FEM model calculation values for *GtHFRF* and PTU housing vibration were compared with the measurement results for verification purposes. Figure 9 shows the results of a comparison made using a PTU that is applied in vehicles equipped with V6 engines. It was confirmed that both the *GtHFRF* shown in Fig. 9(a) and the PTU housing vibration shown in Fig. 9(b) reproduce the waveforms in the vicinity of the peak frequencies within the defined frequency evaluation range.



(a) OVL and *MF*, *GtHFRF*

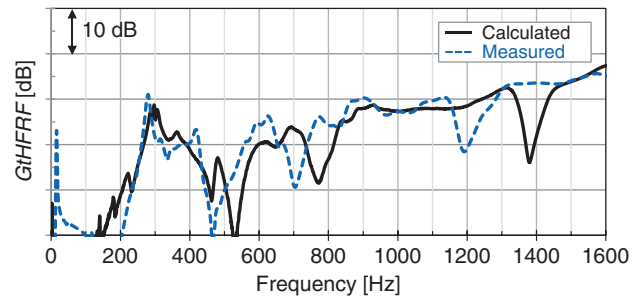


(b) *MF* and compliance

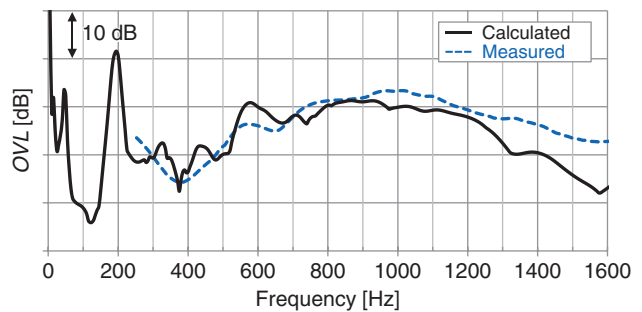


(c) Mode shape

Fig. 8 Calculated results



(a) *GtHFRF*



(b) OVL

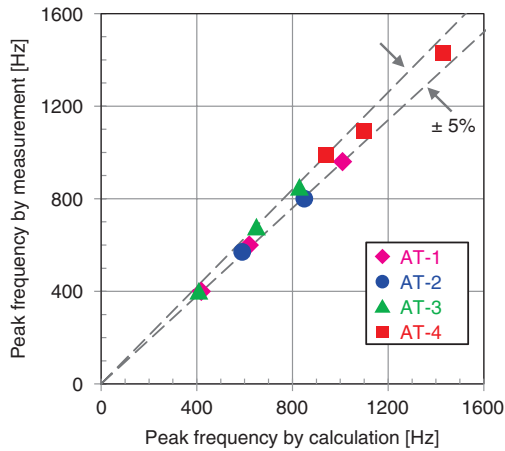
Fig. 9 Comparison of calculated results and measured data

5. PTU Gear Noise Analysis Model Verification

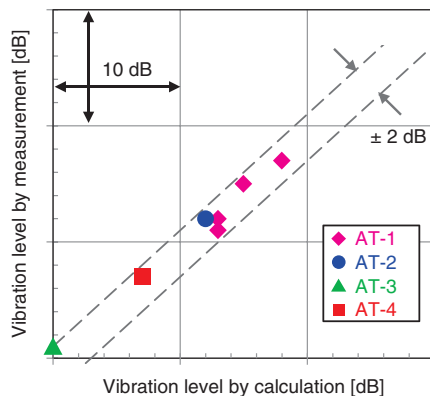
In order to verify the usefulness of a PTU gear noise analysis model in PTU development, a comparison was made of housing vibration calculation values and housing vibration measurement values using differently constructed PTUs. This was done to verify whether or not the target prediction accuracy had been achieved. An investigation was also made of specifications with altered *MF* caused by changes made to the shape of an existing PTU, and a prototype was fabricated. In order to verify the feasibility of investigating specifications using a PTU gear noise analysis model, a comparison was made of housing vibration before and after the shape change between measurement and prediction by calculation.

5.1. Prediction Accuracy Verification

A comparison of housing vibration calculation values and vibration measurement values was made using PTUs with different specifications. Four PTU specifications for automatic transmissions in existing Honda products were used for the comparison, and the correlation between peak frequencies that present PTU gear noise issues and peak vibration levels was confirmed. The peak frequency comparison results are shown in Fig. 10(a) and the peak



(a) Peak frequency



(b) Peak level

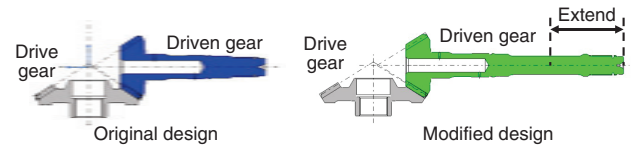
Fig. 10 Verification of prediction accuracy

vibration level comparison results in Fig. 10(b). The Fig. 10 results confirmed that frequency prediction accuracy within $\pm 5\%$ and vibration level prediction accuracy within ± 2 dB was achieved in the defined frequency range of 400 to 1600 Hz, which were set as the prediction accuracy targets for the analysis model of PTU vibration.

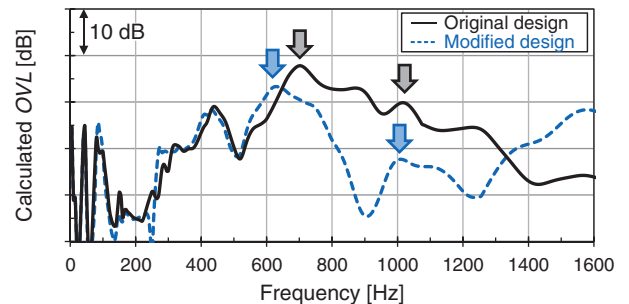
5.2. Verification of Feasibility by Actual Unit Testing

A new specification that changed the *MF* by altering the shape of the driven gear was created, and a comparison was made of the resulting housing vibration calculation values and vibration measurement results. The shapes of the compared gears are shown in Fig. 11(a). Figure 11(b) shows a comparison of the housing vibration calculation results, while Fig. 11(c) shows a comparison of the housing vibration measurement results between the two specifications. When the housing vibration calculation results and housing vibration measurement results from the two specifications are compared in Fig. 11, it is apparent that the vibration peak frequencies and the amount of change in the vibration level in the vicinity of 700 Hz and in the vicinity of 1000 Hz produce the same results.

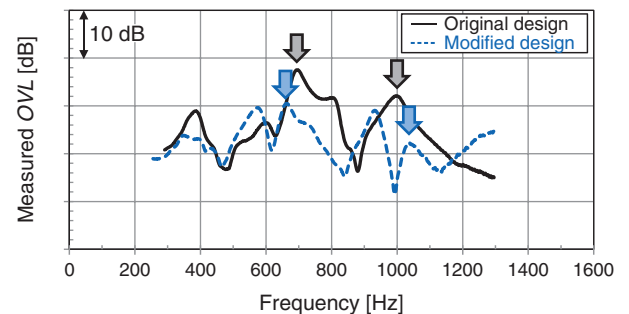
The above verification results confirm that PTU housing vibration calculation results derived using the PTU



(a) Designs for validation



(b) Comparison of calculated OVL



(c) Comparison of measured OVL

Fig. 11 Validation result of application

gear noise analysis model can be used to investigate and determine specifications in the early stages of design.

6. Conclusions

An FEM model was created of the entire drive train, from the engine to the rear differential and including the drive shafts, and the PTU housing vibration caused by hypoid gear mesh was estimated. The model of the hypoid gears was made of hexahedral elements and shell elements. The gear meshing point model was created with a quintuple-node model that takes tooth rigidity into account, and a local coordinate system was configured. A comparison was made of the housing vibration calculation results obtained using the FEM model and the housing vibration measurement results, and the results achieved the following.

- (1) In the frequency range from 400 to 1600 Hz, prediction accuracy achieved a vibration peak frequency error within $\pm 5\%$ and a vibration peak level error within ± 2 dB. This established a technology for quantitative prediction that can be used to judge specifications for PTU layout at an early stage of development.
- (2) Even with specifications involving a shape change, housing vibration calculation results obtained using an FEM model presented waveforms similar to those from actual unit vibration, confirming the usefulness of the analysis model.

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■ Author ■



Takuya NISHIKOJI



Kazuhiro SAITOU



Yoshimi HIRANO