

Reproduction of Engine Torque Fluctuation with Low Inertia and High Stiffness Dynamometer

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ABSTRACT

The dynamometer used in a Virtual and Real Simulator was enhanced to increase the analysis efficiency of issues and events arising from engine torque fluctuation. The contribution of the dynamometer components was clarified using 1D CAE and structural changes were made to expand the reproducible frequency range. As a result, the reproducible torque fluctuation frequencies were expanded, and analysis could be performed using a Virtual and Real Simulator in the operating range where noise and vibration are issues in 4-cylinder and 6-cylinder engines. This bench system can be used to test analyses of mechanisms of drivetrain torsional vibration, such as rattle noise, arising from engine torque fluctuation, without recombining parts. This research can enhance the analysis efficiency of drivetrain torsional vibration issues arising from engine torque fluctuation and contribute to early determination of specifications.

1. Introduction

Electrification continues to accelerate with the goal of enhancing automobile power plant efficiency in order to reduce CO₂ emissions.

Hybrid systems are often used as power sources, most of which are equipped with two motors for driving and power generation. In these systems, the engine is mainly applied to generate the power needed by the driving motor, and is often separated from the tires. This enables to use the efficient rotational speed and torque generation ranges of the engine without being constrained by the driving state and driver demands. When the engine is used in the highly efficient operating range, however, both generated torque and torque fluctuation increase, and the torque fluctuation transmitted to the transmission increases as well. Moreover, a higher compression ratio tends to be set for the engine to increase the thermal efficiency, and torque fluctuation often increases⁽¹⁾. In addition, start from standstill is performed by the motor, so start devices such as a clutch and torque converter are no longer needed, and a structure directly connected with the engine is adopted. The absence of buffer devices in this case leads to an increase in torque fluctuation to the transmission in the

low-load range.

With these changes, rattle noise, etc. produced by gear tooth hitting due to torque fluctuation is becoming an issue. Analysis of these events calls for tests with different combinations of parts, such as replacing the damping device in a completed vehicle, which increases the repeated process time. Therefore, the application of a Virtual and Real Simulator (VRS) was investigated to achieve efficient analysis⁽²⁾.

The dynamometers for conventional VRS have been developed to support transient events such as start from standstill, acceleration and deceleration, beginning with responses to steady-state events. However, as for torque fluctuation, which is a dynamic event, the ranges of torque and frequency that can be generated are limited, and cannot cover the ranges currently used by hybrid vehicles.

Therefore, this research reduced the moment of inertia and increased the stiffness of the dynamometer parts to expand the magnitude and frequency band of torque fluctuation that can be reproduced by the dynamometers. This paper reports the result that oscillation events produced by torque fluctuations in a two-motor hybrid system could be analyzed with greater efficiency.

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2. VRS Architecture and Dynamometer Issues

Figure 1 shows an overview of the VRS. The constructed drive unit VRS uses the actual hardware for the transmission and power control unit (PCU) to be evaluated. An input dynamometer is installed on the transmission input side in place of the engine, and output dynamometers that simulate the vehicle load are installed on the output side. The torque calculated by the engine model and vehicle model is input.

In analysis techniques using a VRS, a verification technique has been proposed that calculates tire torque fluctuation with a model and reproduces the calculation results with equipment⁽³⁾. This enables analysis of event mechanisms by using the VRS to vary the torque fluctuation input to the transmission, and capturing the change in rattle noise. However, this calls for the ability to input torque fluctuation equivalent to that of the engine.

Figure 2 shows the areas of the main combustion order of torque fluctuation that can be generated by dynamometers used by conventional VRS. Engine torque fluctuation includes combustion torque fluctuation arising from the pressure generated by combustion and inertial torque fluctuation produced by the inertia force of reciprocating parts⁽⁴⁾. Inertial torque fluctuation increases proportionally

with the square of the rotational speed, so the inertia force is the main factor in the rotational speed range of 4000 rpm or more in an L4 engine, which makes reproduction a challenge. In addition, even at 4000 rpm or less, the combustion interval is wide, so torque fluctuation is large and the reproducible area is limited.

The 3rd-order is the main component in a V6 engine, and the torque fluctuation frequency is higher than that of the L4 engine, so the torsional resonance of the dynamometer becomes an issue and there are areas over which the excitation torque cannot be properly generated. In addition, since the displacement is large, the torque fluctuation is also large and cannot be reproduced in areas with a high average torque.

To address this issue, it is necessary to reduce the moment of inertia and increase the stiffness of the dynamometer. When examining the specifications, a 1D model of the related parts was used and appropriate moments of inertia and torsional stiffness were set for each part.

3. Low Inertia and High Stiffness Dynamometer for Reproduction of Torque Fluctuation

Figure 3 shows the appearance of the input dynamometer.

Issues such as tooth hitting noise that arise from engine torque fluctuation often occur at 100 Hz or less^{(5), (6)}. The moment of inertia has a large influence at an excitation torque of 100 Hz or less, and in higher frequency bands, the resonant frequency determined by the moment of inertia and the torsional stiffness has an effect. Noise and vibration produced by engine torque fluctuation easily occur at low frequencies, so first the moment of inertia and then the resonant frequency were examined.

3.1. Moment of Inertia Target Setting

In order to examine the excitation torque and resonant frequency, a torsional vibration model of the bench system is considered simply as shown in Fig. 4. In this simple model, the moment of inertia of the dynamometer is J_1 [kgm²], the torsional stiffness of the overall apparatus is k [Nm/rad], the viscous damping coefficient is c [Nms/rad], and the moment of inertia of the test piece is J_2 [kgm²].

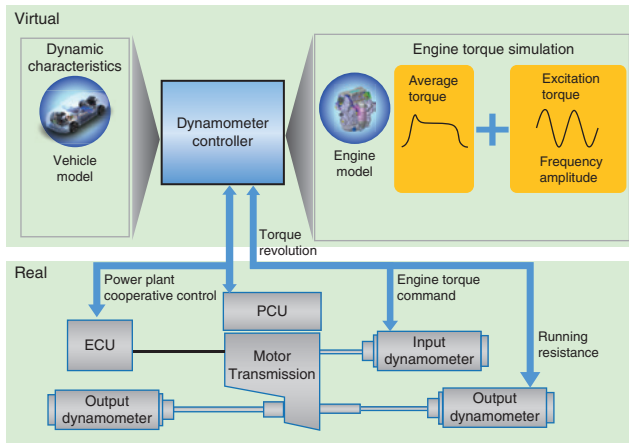


Fig. 1 Overview of VRS

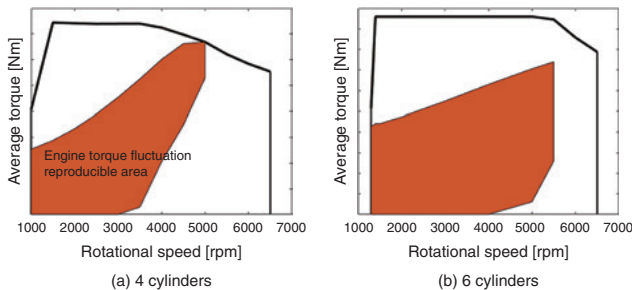


Fig. 2 Conventional specification of bench system

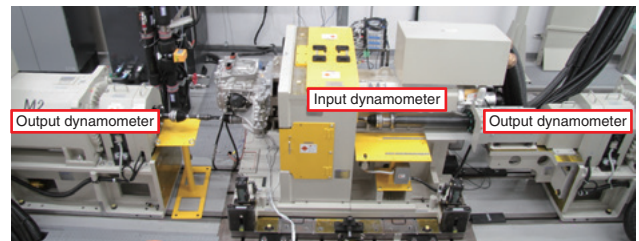


Fig. 3 Appearance of input dynamometer

Here, when $p = \sqrt{k/J_2}$, $\beta = J_2/J_1$, $\zeta = c/2\sqrt{J_2 k}$, the angular displacement θ_1 of the dynamometer and the angular displacement θ_2 of the test piece are expressed as shown in Eq. (1) using the excitation torque T_d [Nm] and excitation frequency ω [rad/s] of the dynamometer.

$$\begin{bmatrix} \theta_1(\omega) \\ \theta_2(\omega) \end{bmatrix} = \frac{T_d/J_1}{\omega^2(\omega^2 - p^2) - \beta p^2 \omega^2 - 2i\zeta p \omega (1 + \beta) \omega^2} \begin{bmatrix} p^2 - \omega^2 + 2i\zeta p \omega \\ p^2 + 2i\zeta p \omega \end{bmatrix} \quad (1)$$

The torque fluctuation T_c input to the test piece at this time is expressed using θ_1 and θ_2 as shown in Eq. (2).

$$T_c = |\theta_1 - \theta_2| \sqrt{k^2 + c^2 \omega^2} \quad (2)$$

Assuming that the excitation frequency ω is sufficiently small relative to the torsional stiffness k , Eq. (2) can be approximated as shown in Eq. (3).

$$T_c = \frac{J_2}{J_1 + J_2} \cdot T_d \quad (3)$$

Likewise, when the engine torque fluctuation is T_e , and the moment of inertia of the crankshaft is J_1' , the torque fluctuation T_c' input to the test piece at that time is as shown in Eq. (3)'.

$$T_c' = \frac{J_2}{J_1' + J_2} \cdot T_e \quad (3)'$$

In order to realize the same torque fluctuation with the dynamometer as that of the engine, T_c and T_c' should be equal, and the demanded moment of inertia can be calculated from Eq. (4).

$$J_1 = \frac{T_d}{T_e} (J_1' + J_2) - J_2 \quad (4)$$

The value of each parameter is as shown in Table 1, so the target moment of inertia of the apparatus was set to 0.2 kgm² or less.

3.2. Resonant Frequency Target Setting

The resonance range is entered under the condition where the torque fluctuation input to the test piece defined by Eq. (2) exceeds the dynamometer excitation torque T_d . Therefore, the target resonant frequency was set so that the resonance range is not entered at the main combustion order frequency of 350 Hz at the upper limit engine speed. When

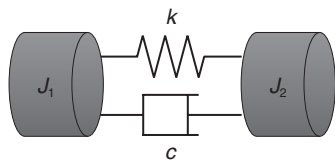


Fig. 4 Torsional vibration model of bench system

Table 1 Calculation condition for requested value of inertia

Excitation torque upper limit of dynamometer [Nm]	1200
Inertia of crankshaft [kgm ²]	0.04
Inertia of test piece [kgm ²]	0.12
Max torque fluctuation of engine [Nm]	600

the frequency ω_1 that exceeds the dynamometer excitation torque is calculated based on Eq. (2) under the condition of viscous damping coefficient $c = 0$, it is as shown in Eq. (5).

$$\omega_1 = \sqrt{\frac{k}{J_2}} \quad (5)$$

The resonant frequency was calculated from Eqs. (5) and (6), and the target resonant frequency was changed to 450 Hz from the current 350 Hz.

$$\omega_n = \sqrt{\frac{(J_1 + J_2)k}{J_1 J_2}} \quad (6)$$

3.3. Examination of Dynamometer Specifications

Dynamometer shaft design should take into account not only the torsional resonance of the dynamometer but also the bending resonance, making it necessary to increase the bearings from the viewpoint of durability to bending vibration. Adoption of a high-stiffness wireless torque meter in order to increase the resonant frequency was also examined, but there is an area where metal parts cannot be positioned so as not to shield the antenna part, and it was necessary to install multiple couplings. For this reason, the initial specifications were as shown in Fig. 5, the total moment of inertia was 0.3 kgm², and the torsional resonant frequency was 270 Hz.

The torsional stiffness k needs to be increased in order to increase the shaft resonant frequency. However, when the dynamometer is considered simply as a cylindrical shape with external diameter D_1 , internal diameter D_2 , shaft length L , material density ρ , and lateral elastic modulus G , the moment of inertia J and torsional stiffness k of the flanges and other parts used in the apparatus can be respectively expressed by Eqs. (7) and (8), and the moment of inertia and torsional stiffness can be seen to have a tradeoff

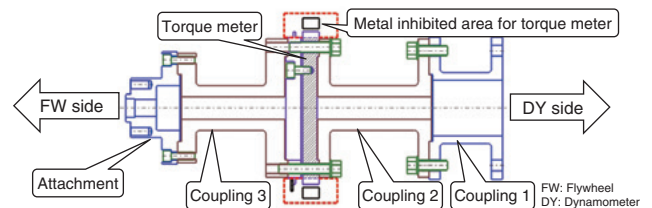


Fig. 5 Initial component of input dynamometer

relationship that depends on the radial dimension.

$$J = \frac{\pi}{32} \rho \cdot L \cdot (D_1^4 - D_2^4) \quad (7)$$

$$k = \frac{G \cdot \pi (D_1^4 - D_2^4)}{32L} \quad (8)$$

In order to reduce the moment of inertia of the rotating part, calculations were performed using the 1D model shown in Fig. 6 to determine the contribution of the torsional stiffness of each part to the 1st-order resonant frequency of the torsional vibration. Figure 7 shows the results.

The torque meter has sufficiently high torsional stiffness and a low contribution to the resonant frequency of the overall system, so the parts configuration was reviewed and specifications making it possible to increase the resonant frequency with low inertia were selected as shown in Fig. 8. In addition, bending vibration was addressed by limiting the mountable weight to 20 kg, which is the upper limit of the test pieces to be attached.

Table 2 compares the specifications with those of the conventional apparatus.

These structural changes raised the torsional resonant frequency of the input dynamometer from 350 Hz to 500 Hz as shown in Fig. 9, expanding the upper limit of the excitation frequency from 280 Hz to 440 Hz.

The adoption of a low-inertia and high-stiffness dynamometer as described above greatly expanded the

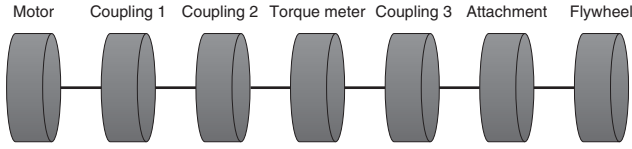


Fig. 6 1D CAE model of input dynamometer

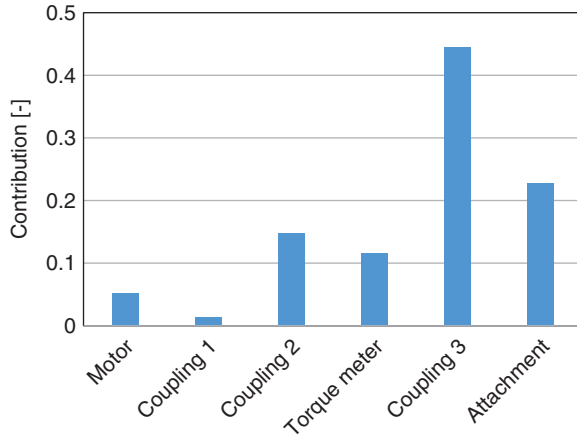


Fig. 7 Contribution of components

Table 2 Comparison of apparatus specifications

	Conventional specification	Proposed specification
Inertia [kgm ²]	0.15	0.2
Stiffness [kNm/rad]	270	485
Max torque [Nm]	500	1600

engine torque fluctuation reproducible area and enabled reproduction of torque fluctuation over the entire range of 3000 rpm or less, in which torsional vibration issues easily occur, for both the 4-cylinder and 6-cylinder engines as shown in Fig. 10.

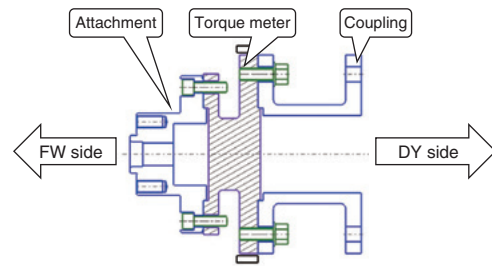


Fig. 8 Final component of input dynamometer

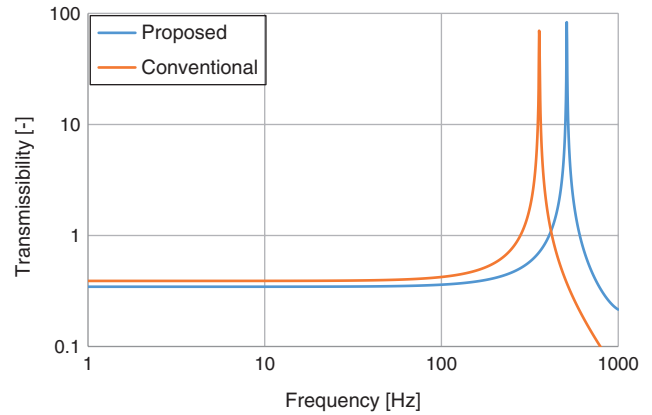


Fig. 9 Comparison for frequency response

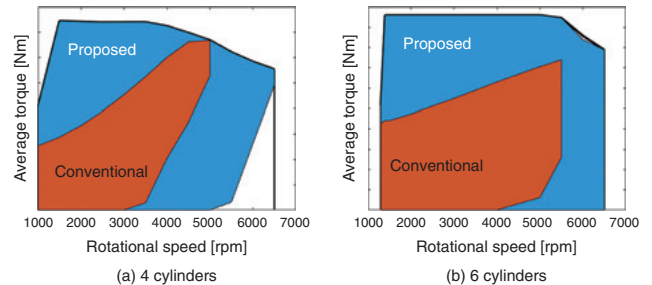


Fig. 10 Proposed specification of bench system

4. Proving Tests

Rattle noise is generated by tooth surface separation of gears on the torque flow at low torque, and by idle gear tooth hitting regardless of the torque. Torque fluctuation in the low torque range could be reproduced even by a conventional dynamometer, but the small excitation torque hindered sufficient parameter studies. In addition, torque fluctuation in the high torque range could not be reproduced.

Tests were conducted to verify that use of the developed dynamometer enables excitation torque parameter studies over the entire torque range, as well as analysis of rattle noise mechanisms without replacing parts. Rattle noise is noise produced when the engine torque fluctuation, which is amplified and damped by a vibration system that has the flywheel (FW) damper spring and drive shaft as springs as shown in Fig. 11, is transmitted to the transmission, internal parts with clearance vibrate, and other parts collide with each other. Therefore, torque fluctuation arising from the FW damping characteristics and vibration of internal parts due to torque fluctuation input were analyzed.

4.1. Analysis of Torque Fluctuation due to Flywheel

Figure 12 shows the frequency analysis results of the crankshaft rotational speed when rattle noise occurs in a two-motor hybrid vehicle.

2nd-order or higher rotational fluctuation normally occurs in a 4-cylinder engine. However, 1st-order rotational

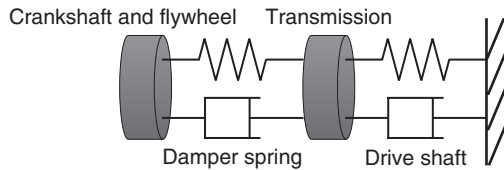


Fig. 11 Overview of torsional vibration model for vehicle

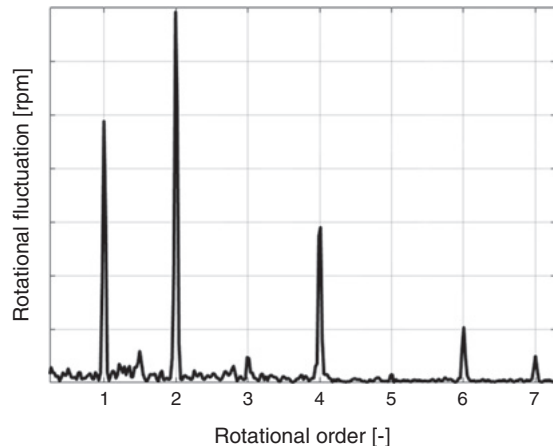


Fig. 12 Rotational fluctuation of crankshaft

fluctuation also occurs, and is thought to be due to subharmonic oscillation. Subharmonic oscillation is an event that occurs when the spring stiffness of the damper spring fluctuates across k_1 and k_2 as shown in Fig. 13.

The target FW has a structure where the hysteresis torque changes when the input torque changes by 30 Nm or more⁽⁷⁾. Figure 14 shows the engine torque waveform on the ECU when rattle noise occurs, and the change in torque is 30 Nm or more, so the high friction area is used. It is thought, therefore, that the transition to the high friction area changes the torsional stiffness of the damper spring as shown in Fig. 15, with the result that subharmonic oscillation occurs.

Next, when the static spring stiffness of the damper spring is k_s [Nm/rad] and the dynamic spring stiffness when the friction changes is k_d [Nm/rad], the FW is modeled as a parallel spring as shown in Fig. 16. At this time, the spring

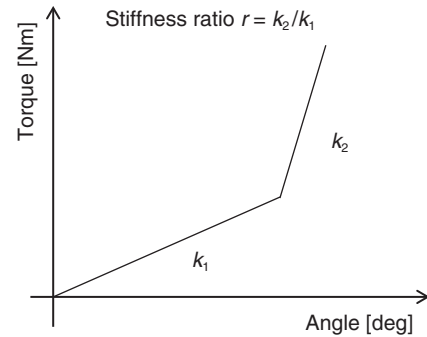


Fig. 13 Overview of damper spring stiffness

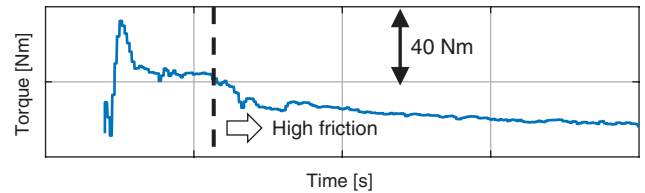


Fig. 14 Time-series data of engine torque

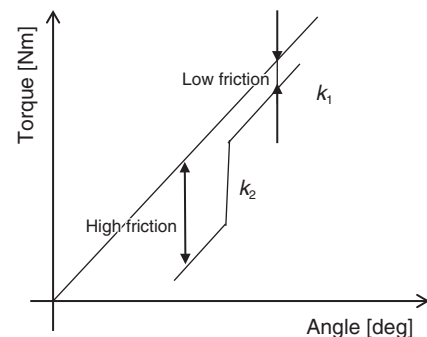


Fig. 15 Stiffness change with friction transition

stiffness k_1 and spring stiffness k_2 are expressed as shown in Eq. (9).

$$\begin{aligned} k_1 &= k_s \\ k_2 &= k_s + k_d \end{aligned} \quad (9)$$

When the spring stiffness at friction part start-up is k_f [Nm/rad] and the friction torque is f [Nm], the dynamic spring stiffness k_d is modeled as shown in Fig. 17. k_d can also be expressed by Eq. (10) using the torsion angle displacement amplitude θ [rad]⁽⁸⁾.

$$k_d = \frac{f}{\theta} \tanh \frac{\theta \cdot f}{k_f} \quad (10)$$

It has been reported that reducing the stiffness ratio r decreases the level of subharmonic oscillation generated⁽⁹⁾. As θ increases, the $\tanh$ term of Eq. (10) becomes asymptotic to 1, so both k_d and the composite spring k_2 decrease. As a result, it is thought that increasing the excitation torque amplitude as well as θ reduce the level of subharmonic oscillation generated.

Figure 18 compares the result of confirming the change in the level of subharmonic oscillation generated

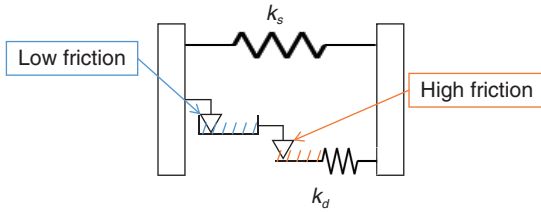


Fig. 16 FW model with parallel spring

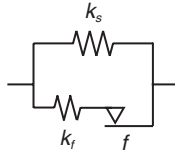


Fig. 17 Damper spring and friction model

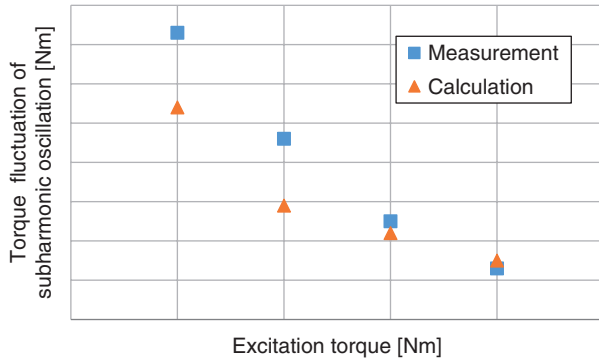


Fig. 18 Torque fluctuation of subharmonic oscillation

in an actual engine when the excitation torque amplitude is increased with the result of CAE analysis incorporating the damper spring model shown in Fig. 16. Both the measurement and CAE show the tendency of the 1st-order torque fluctuation to decrease when the excitation torque amplitude is increased.

The ability to perform torque fluctuation parameter studies made it possible to analyze that the mechanism of subharmonic oscillation generated has nonlinear characteristics due to the friction plate, without recombination to FW with different characteristics.

4.2. Analysis of Internal Part Behavior due to Torque Fluctuation

When rattle noise occurs, reducing backlash and clearance and suppressing part oscillation are effective as countermeasures inside the drivetrain⁽¹⁰⁾. However, as there is significant clearance inside the drivetrain, in order to implement effective countermeasures it is important to identify the clearance sites that are the source of rattle noise. If countermeasure amounts are not sufficient when recombining parts and performing tests, however, the change in the rattle noise level is minute and it is not easy to identify tooth hitting sites.

It was thought, therefore, that tooth hitting sites could be identified by controlling the excitation torque to vary the level of rattle noise generated and comparing the rattle noise to the behavior of drivetrain parts when tooth hitting occurs.

Based on past knowledge, it is clear that events related to part behavior are limited to rotational fluctuation generated by torque fluctuation input, and thrust fluctuation and radial fluctuation due to component force generated from the gear tooth surface. To measure these behaviors, displacement sensors were attached inside the transmission as shown in Fig. 19, and the shaft behavior was measured.

The state in contact with the engine was used as the origin of each part, and the coordinate axes were defined as shown in Fig. 20.

The normal vector n of the plane that passes through the three measurement points A, B, and C of each part is expressed by Eq. (11). The thrust direction displacement S and tilt angle ϕ of each part from the normal vector n are calculated as shown in Eqs. (12) and (13).

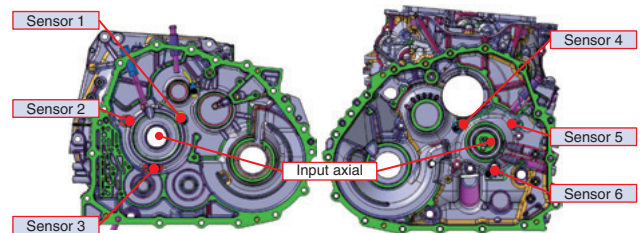


Fig. 19 Positions of displacement sensors

$$\vec{n} = \begin{pmatrix} x_n \\ y_n \\ z_n \end{pmatrix} = \begin{pmatrix} (y_b - y_a) \cdot (z_c - z_a) - (z_b - z_a) \cdot (y_c - y_a) \\ (z_b - z_a) \cdot (x_c - x_a) - (x_b - x_a) \cdot (z_c - z_a) \\ (x_b - x_a) \cdot (y_c - y_a) - (y_b - y_a) \cdot (x_c - x_a) \end{pmatrix} \quad (11)$$

$$S = \frac{x_n x_a + y_n y_a + z_n z_a}{z_n} \quad (12)$$

$$\phi = \tan^{-1} \sqrt{\left(\frac{x_n}{z_n}\right)^2 + \left(\frac{y_n}{z_n}\right)^2} \quad (13)$$

Figure 21 shows an example of the measured time series results. The upper graph shows the acceleration in the film surface direction of the L side cover, the middle graph shows the overdrive gear thrust displacement, and the lower graph shows the input shaft thrust displacement. These results show that each part exhibits large displacement when rattle noise occurs.

Figure 22 shows the frequency analysis result of L side cover acceleration. Unsteady noise such as rattle noise calls for high time resolution, so analysis is performed using the wavelet transformation already in use for rattle noise analysis⁽¹¹⁾. The frequency analysis results reveal a strong rattle noise peak between 500 Hz and 1000 Hz.

The Partial Overall (POA) between 500 Hz and 1000 Hz, which is the frequency band at which rattle noise occurs, was calculated for each behavior under conditions of different excitation torque, and compared with the

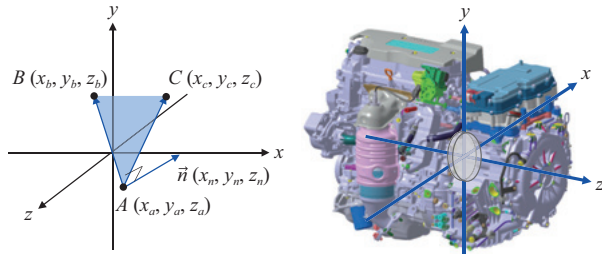


Fig. 20 Definition of coordinate axes

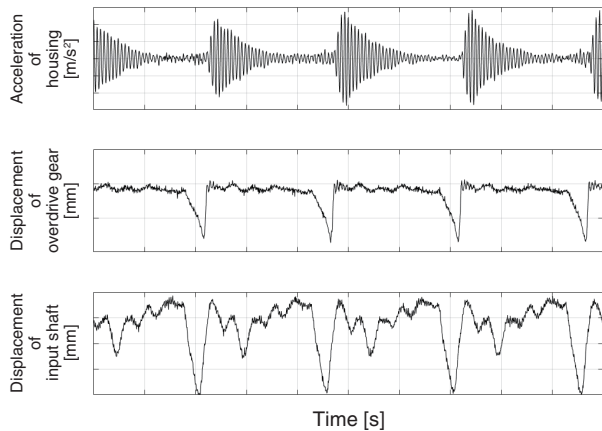


Fig. 21 Results of measurement

Table 3 Correlation coefficient to acceleration

Overdrive gear thrust displacement	0.8994
Input shaft rotation	-0.1645
Input shaft thrust displacement	-0.1303
Input shaft radial angle	0.2105
Overdrive gear radial angle	0.0864

acceleration of the L side cover. As an example, Fig. 23 shows a graph of highly correlated behavior.

Next, Table 3 shows the correlation coefficients between each behavior and acceleration obtained from this scatter diagram.

Of the various behaviors, as overdrive gear thrust displacement and acceleration exhibit a strong correlation, overdrive gear thrust displacement is thought to be the source of the rattle noise.

Next, Fig. 24 shows the test results under the same conditions when changed to a press-fit specification so that overdrive gear thrust displacement does not occur.

Since overdrive gear thrust displacement is suppressed, the generation of rattle noise is also suppressed, and rattle noise generation sites can be identified by varying the excitation torque, without replacing parts.

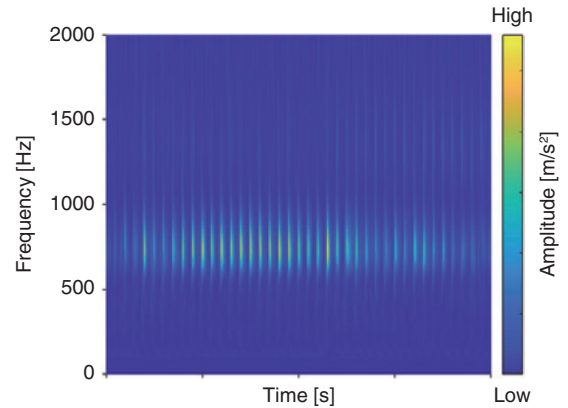


Fig. 22 Result of frequency analysis

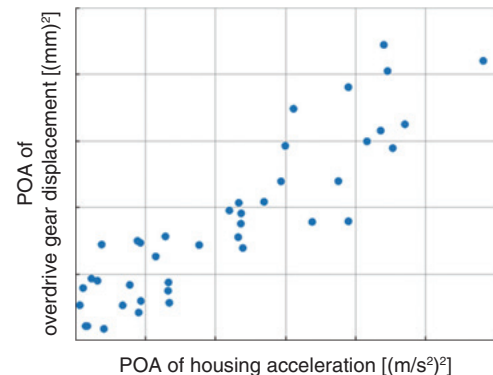


Fig. 23 Relationship of displacement and acceleration

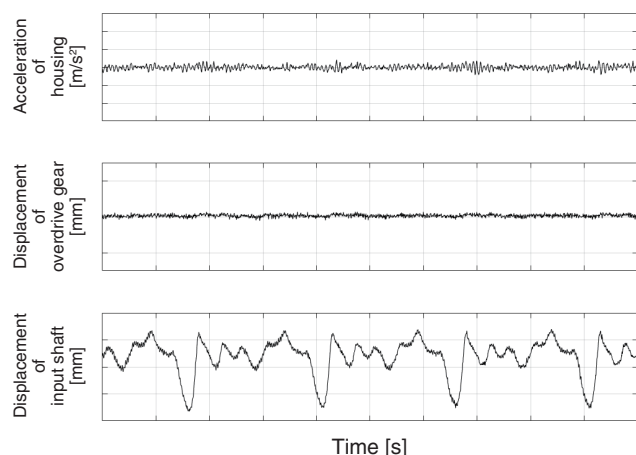


Fig. 24 Result of countermeasure

5. Conclusion

In examination of the dynamometer used in VRS, the structure was changed to reduce inertia and increase stiffness while using 1D CAE to clarify the contribution of the torsional stiffness of each part. This resulted in a moment of inertia of 0.2 kgm^2 and a resonant frequency of 500 Hz, which made it possible to reproduce torque fluctuation over the entire operating range needed for analysis of events arising from torque fluctuation. Thus, a parameter study using a VRS with an expanded excitation torque area verified that the FW damping characteristic mechanism and rattle noise generation sites could be identified without recombination of parts.

This research allowed analysis of drivetrain torsional vibration issues arising from engine torque fluctuation in transmission development by varying the excitation torque, without replacing parts. It is thought that this can contribute greatly to increasing efficiency and early decision on specifications.

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