NVH Analysis Using Full Vehicle Multi Body Dynamic Model*
-Influence of Constant Velocity Universal Joints on Shudder Vibration–

Vehicle NVH characteristics have been analyzed using a full vehicle simulation model. This model consists of engine, drive train and chassis components that are respectively accurate multi body dynamic models. This paper focuses on a shudder vibration analysis, specifically on the influence of constant velocity universal joints (CVJs) on shudder vibration characteristics.

1. Introduction

In developing automobiles, the mode of NVH problems varies significantly depending on engine and chassis types, with many problems occurring only in completed vehicles. In addition, the NVH problems to be solved during development of a particular car require many hours of work and pose a significant challenge in expediting vehicle design completion. Furthermore, since the NVH problems of a given vehicle are often the result of complex mechanisms, determination of the root causes of each NVH problem through experiments with actual vehicles is very difficult.

Recently, various structural and mechanism analysis techniques have been applied to vehicle development, contributing to the simplification of prototyping and reduction of labor time required for vehicle development. As for NVH problems, examples of analysis techniques applied to idling vibration analysis have been reported. Analysis techniques are used not only for vehicle development but also for automotive component development.

We have developed a full vehicle dynamic model using mechanism analysis to analyze vehicle vibration. The NVH problem we analyzed this time was the shudder vibration that occurs when a vehicle starts and runs straight ahead. This full vehicle dynamic model is a detailed mechanism analysis model that incorporates virtually all the elements usually included in automobiles, including a power train ranging from the engine to tires as well as a steering and suspension system. Using this full vehicle model, we have reviewed the effects of a drive shaft constant velocity joint on shudder vibration. The constant velocity joint model also reflects nonlinear elements in its mechanism simulation. We have verified that this model reproduces the forces that act on the interior of actual constant velocity joints.

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2. Shudder vibration in starting vehicles

Shudder vibration occurring in a starting vehicle is generally defined as the phenomenon in which a car body is excited by resonance between the engine vibration and the induced thrust on a sliding constant velocity joint.

Results of shudder vibration measurements obtained on an actual vehicle are summarized in Fig. 1, where Fig. 1 (a) illustrates vehicles speeds, Fig. 1 (b), torques acting on the drive shaft, and Fig. 1 (c), lateral acceleration below the driver’s seat floor.

The area encircled with a dotted line represents a state where lateral vibration occurred and the vehicle accelerated while vibrating left and right. The conditions that induced shudder on the test vehicle were a time of 2.5 to 3.5 seconds after starting, vehicle speed of 28 to 40 km/h, and drive shaft torque of approximately 700 Nm, and the associated drive shaft speed was in a range of 230 to 330 rpm.

Fig. 2 illustrates the results of tracking analysis with the drive shaft torque values provided in Fig. 1 (c).

As shown in Fig. 2, the major component of the shudder vibration is clearly the drive shaft rotation 3rd order. The drive shaft of the test vehicle is equipped with a tripod type constant velocity joint (hereafter, TJ) on its inboard side and a fixed constant velocity joint (hereafter, EBJ) with eight balls on its outboard side. Because of this, it appears that the inboard side constant velocity joint is responsible for shudder vibration on starting vehicles.

A non-mass-production sliding constant velocity joint was used in our experiment in order to analyze the shudder vibration phenomenon accurately.

3. Full vehicle model

3.1 Power train and chassis model

To theoretically forecast the shudder vibration phenomenon, analyses of engine vibration characteristics and induced thrust on a constant velocity joint, as well as transmission of the engine vibration and the induced thrust to the car body via the mount and suspension, are also necessary. To this end, we developed a detailed full vehicle mechanism simulation model and performed analysis by applying a drive torque from the engine output in order to simulate an induced thrust occurring from rotation of the constant velocity joint.

Fig. 3 schematically illustrates the power train and chassis model used for our analysis work. The engine model simulated the piston and crank motion resulting from the pressure of the fired fuel. The model for the torque-transmitting transmission includes the simulated torque amplification by a torque converter, speed reduction by planetary gearing and motion of a differential. The model for suspension includes a spring mass capable of simulating the suspension resonance characteristics, and a suspension mass capable of simulating the rigidity of the stoppers on a starting vehicle. Furthermore, the tire model accurately simulates tire rigidity, thus allowing the simulation of a complete vehicle running on a road, including friction with the road surface.
The characteristic frequency of the lateral engine motion was approximately 15 Hz. The drive shaft speed was approximately 310 rpm when the acceleration was greatest as shown in Fig. 5, and the frequency of the drive shaft rotation 3rd order component was approximately 15.5 Hz.

Our vehicle model was developed with consideration for not only shudder vibration but also possible application for evaluation of other vehicle NVH performance. Thus, the vehicle model includes detailed simulation of sections not associated with prediction of shudder vibration.

3.2 Drive shaft model
As described in Sec. 2 above, we learned that the drive shaft 3rd order component affects shudder vibration. The test vehicle has the TJ on its inboard side. In order to simulate the drive shaft 3rd order, the constant velocity joint model must be a detailed mechanism simulation model. Fig. 4 shows the constant velocity joint models (TJ and EBJ) that were used in our full vehicle dynamic model. Thus, this configuration provided mechanism simulation-capable inboard and outboard constant velocity joint models. Each of these mechanism models simulated the clearance, contact force and friction between the related components in the CVJs. 3) 4)

4. Verification of analysis accuracy
Fig. 5 shows a comparison between the analysis results for lateral acceleration below the floor at the driver's seat and the actual experiment results. Though the amplitude obtained from the simulation is smaller than that obtained from the experiment, the results of analysis match those of the experiment well.

5. Analysis of shudder vibration phenomenon
5.1 Relation between engine and drive shaft
We found that shudder vibration on a starting vehicle is related to the drive shaft rotation 3rd order component. The speed of a drive shaft that triggers shudder vibration is in the range of 230 to 330 rpm. The frequency of the 3rd order component falls in a range of approximately 11.5 to 16.5 Hz. To investigate the vehicle vibration mode in this frequency range, we performed model analysis with the full vehicle model. Fig. 6 illustrates the lateral direction engine mode. The characteristic frequency of the lateral engine motion was approximately 15 Hz. The drive shaft speed was approximately 310 rpm when the acceleration was greatest as shown in Fig. 5, and the frequency of the drive shaft rotation 3rd order component was approximately 15.5 Hz.
Because of this, shudder vibration seemed to occur from the resonance between the lateral acceleration of the engine and the drive shaft rotation 3rd order component.

For further clarification, we altered the rigidity of the engine mount in the mechanism simulation model, and again performed modal analysis and lateral shudder analysis. Fig. 7 shows an engine vibration mode with reduced engine mount rigidity. The alteration to rigidity can be understood to change the characteristic vibration frequency and mode. Fig. 8 offers a comparison of the magnitudes of shudder vibration between these two modes. In summary, for each particular engine mode, there is a unique vibration frequency and a unique magnitude of shudder vibration.

5.2 Influence of CVJ type

Induced thrust, which is one of the drive shaft rotation 3rd order components, typically occurs with the TJ. When the TJ is run, an induced thrust occurs along the rotational axis of the outer race and that mainly consists of the rotational 3rd order. This induced thrust is a force that results from frictional forces occurring because of the mutual friction of the rotating internal components on the TJ.\(^5\) Therefore, we executed shudder vibration analysis with two sliding constant velocity joints with unique friction characteristics. Fig. 9 summarizes the results of analysis for lateral acceleration below the floor at the driver’s seat. As can be seen from this diagram, it seems that use of a low friction CVJ can alleviate shudder vibration.
5.3 Effect of phase gap with right-hand and left-hand CVJs

The shudder vibration test with an actual car was a forward traveling-only starting and acceleration test. When the vehicle turned, the resultant measurement values greatly varied. Fig. 10 illustrates the running pattern used in the shudder test and the resultant maximum measurements of shudder vibration. The measurement operation was repeated 30 times for each constant velocity joint type. Each CVJ type exhibited measurement variation and had unique vibration characteristics. Notwithstanding, the lowest shudder measurements for both CVJ types were nearly same.

There are right-hand drive and left-hand drive shafts. The right hand and left hand TJs are incorporated via the differential box. The resultant force of the induced thrusts from the two TJs varies depending on the phase gap between both TJs. Fig. 11 (a) shows the definition of phase gap, and Fig. 11 (b) plots induced thrusts resulting from the phase gap angles between the two TJs. As can be seen in Fig. 11(b), when the relative phase of one TJ coincides with that of the other TJ, the resultant force of the induced thrusts theoretically becomes zero, and reaches the maximum level when the phase gap angle is 60 degrees.

With our full vehicle model, the effect of the phase gap between the right-hand TJ and the left-hand TJ was analyzed. As shown in Fig. 12, it was found through simulation that the magnitude of shudder vibration was largest when the phase gap was in a range of 45 to 60 degrees and was smallest when the phase gap was in a range of 100 to 120 degrees. As is apparent from Fig. 12, this trend was validated in the experiment with an actual vehicle. Consequently, we learned that this variation in shudder vibration level occurred when the differential gearing was activated as the vehicle turned and, as a result, phase gap between the right hand and left hand TJs varied.
6. Conclusion

We developed a full vehicle running-capable mechanism simulation model and performed a simulation for vehicular NVH (analysis of shudder vibration).

We identified and verified the relationship between the shudder vibration mechanism on a starting vehicle and engine vibration modes.

We qualitatively and quantitatively clarified the interrelation between induced thrusts and shudder vibration levels on constant velocity joints.

References


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