

Development of SRS Module Damping Structure for Vibration Enhancement and Weight Reduction

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ABSTRACT

Resonance phenomena represent an issue in terms of marketability, and dynamic dampers are employed as one means of responding to this issue. However, because the use of dynamic dampers increases vehicle weight, methods that entail less weight increase are demanded in the area of the steering system. A new SRS module structure able to be employed as a dynamic damper was therefore developed. This new structure reduces vibration without the use of a conventional dynamic damper. The distinctive feature of the new structure is the addition of a floating bush to the SRS module in order to obtain a frequency characteristic suited to the resonance characteristic of the steering system. The use of this structure has balanced increased marketability with reduced weight, and helped to enable the realization of a method of ameliorating steering vibration with a high level of general applicability.

1. Introduction

Honda's desire to reduce fuel consumption and CO₂ emissions generates a demand for the realization of further automotive weight-savings. At the same time, the demand for vibration performance as an aspect of marketability continues to increase.

During the development of steering systems, studies are conducted from the initial stages in an attempt to reconcile weight-savings and performance. However, the realization of vibration performance for vehicle marketability by means of design specs that focus exclusively on stiffness represents a challenge, and in many cases dynamic dampers are ultimately fitted. Methods that modify the vibration characteristic of the response side or reduce vibration input are also available, to make it unnecessary to rely on methods involving dynamic dampers. Methods of modifying the response characteristic include modification of the body configuration and modification of the components of the steering system. In the main, structural optimization studies are conducted using CAE, and response characteristics are modified while attempting to minimize weight increases^{(1),(2)}. However, the reduction of response at the steering resonance frequency without increasing weight presents a high level of technological challenge. While reduction of the spring constant of the engine mount and the suspension bush is

effective as a means of reducing vibration input⁽³⁾, in many cases this method is antagonistic to ride comfort and steering stability, and little beneficial effect can therefore be expected from it in terms of vibration reduction. Because there is a high level of vibration input when cylinders are deactivated in vehicles that employ cylinder idling systems, active engine mounts are employed as a method of reducing the vibration input transmitted to the body⁽⁴⁾. This is, however, a method that involves some cost.

The body, steering hanger, steering column, steering wheel, and Supplemental Restraint System (SRS) module determine the steering vibration characteristic. The relationship between the respective resonance characteristics of these parts determines the vibration characteristic in the completed vehicle. Based on the results of analyses of the resonance characteristics of these parts, the research discussed in this paper investigated a structure that would actively employ the resonance characteristic of the SRS module in order to increase marketability. In concrete terms, by modifying the support structure of the SRS module and using it as a dynamic damper, the research was able to realize a reduction in vibration with extremely minimal increases in weight and cost. This new structure can be employed in multiple vehicle types, and is able to contribute to increased vibration performance for marketability and reduced weight.

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2. Mechanism of Vibration Reduction

2.1. Characteristics of Steering Vibration Phenomena

Three steering vibration phenomena can be indicated as resulting in a decline in marketability. These are idle steering vibration and launch vibration originating in the transmission of engine vibration, and driving vibration originating in the transmission of suspension vibration. These vibration phenomena represent an issue in the area in which the 1st-order resonance frequency of the steering system and the frequency of the input vibration in each case coincide. Today, given the tendency to use an expanded transmission lock-up range in order to increase fuel efficiency, the frequency with which engines operate at low engine speeds is becoming higher than was previously the case. In addition, the generation of an engine vibration component when cylinders are deactivated with the use of cylinder-idling systems is making steering vibration issues more complicated. This demands the realization of increased marketability in a wider range of frequencies than was previously the case.

Figure 1 shows the steering resonance frequency range and the frequency band for each vibration phenomenon. As a result of efforts to balance performance and weight-savings, the 1st-order resonance frequency of the steering system in the complete vehicle exists in the restricted range between 30 and 45 Hz in automobiles ranging from mini-sized to luxury vehicles. Steering vibration becomes an issue at frequencies at which this range and the input vibration for each phenomenon coincide. As the colored section of the figure shows, there is a shared frequency range in which this is the case of around 15 Hz for multiple vehicle types.

2.2. Conception of Using SRS Module as Dynamic Damper

The dynamic dampers used as a method of ameliorating steering vibration are positioned in the steering wheel. The steering wheel generates the greatest amplitude of vibration in the 1st-order resonance mode of the steering system, and the positioning of even a small mass close to

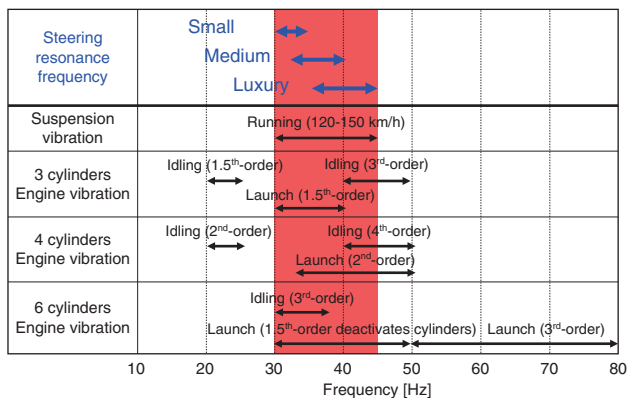


Fig. 1 1st-order resonance frequency of steering system and vibration phenomena

this component is able to produce a vibration reduction effect. However, frequency characteristics must be set for each vehicle model, and the dampers therefore lack general applicability.

The idea of using the mass of an SRS module as a dynamic damper was therefore conceived. This would make it possible to use an increased mass as a dynamic damper in approximately the same position as a conventional damper. SRS modules are safety devices employed as standard, and the main type in use is a floating type, in which the inflator floats by means of a horn spring. This configuration produces a resonance dependent on the mass of the module and the characteristics of the spring used to float the inflator. The resonance characteristic of this floating configuration was analyzed, and the potential for its use as a dynamic damper was verified.

2.3. Evaluation using Standardized Components

The resonance characteristic of the steering system of a complete vehicle was analyzed, and the effect of component parts was compared. In order to simplify the structural components, standardized components with sufficiently increased stiffness were employed (Fig. 2). Figure 3 shows these components when fitted and a comparison of their resonance characteristics.

Configuration D shows the body side of the steering hanger fixed to a rigid plate, and the other components made up of the components used in a complete vehicle, while Configuration A shows this configuration with the steering wheel replaced by a standardized component. In this configuration, steering wheel resonance and the effects of SRS module resonance are excluded. Measurements were conducted using Configuration C, in which the steering wheel was fixed to a rigid plate, and Configuration B, in which the SRS module only was fixed to a rigid plate.

A comparison of Configurations A and D shows that the response level is higher for Configuration A. Based on the results of a comparison of the resonance characteristics of the SRS module and the steering wheel, it was conjectured that the 84 Hz resonance frequency of the steering wheel was sufficiently higher than the 1st-order resonance frequency of the steering system to have no effect in Configuration D. The resonance frequency of the SRS module was 56 Hz, and it was considered that the effect of vibration reduction produced by adding a dynamic damper

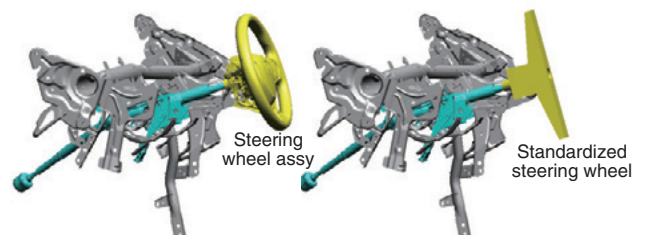


Fig. 2 Standardized components for measurements

to Configuration A was incorporated in Configuration D. It would be possible to expand this vibration reduction effect by optimally setting the resonance frequency of the SRS module.

2.4. Range of Effect of Mass of Dynamic Damper

The necessary mass for the dynamic damper in order to realize a vibration reduction method with a high level of general applicability was determined. In order to help enable consideration of the relationship between the mass of the dynamic damper and its vibration reduction effect, the difference between two resonance frequencies obtained from the equation of motion of a two-degree-of-freedom vibration model was derived.

As shown in Fig. 4, the external force F_1 acts on the main system determined by the mass m_1 and the spring k_1 .

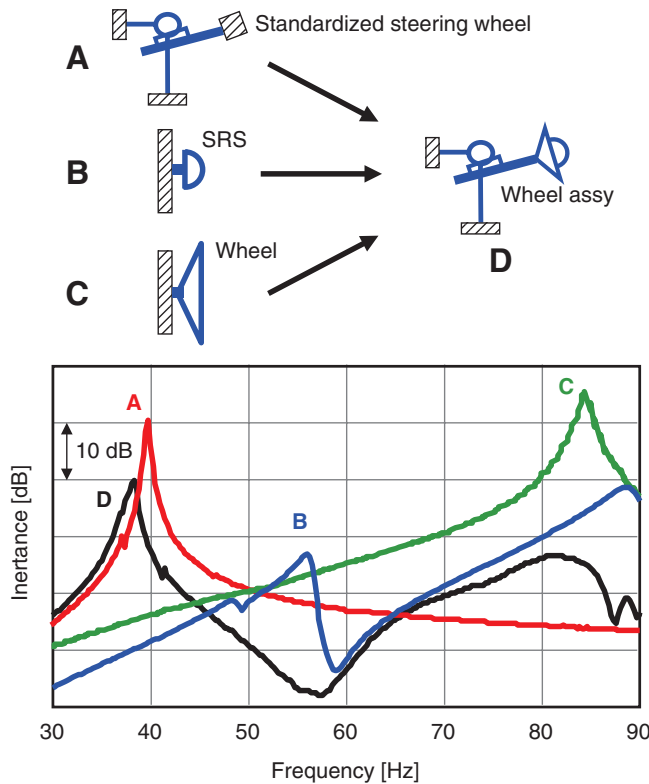


Fig. 3 Comparison using standardized components

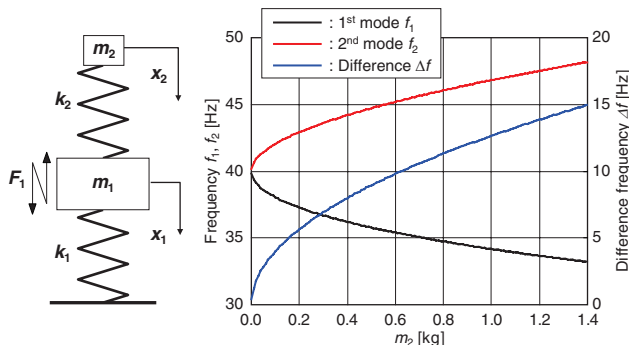


Fig. 4 Effect of dynamic damper and weight ratio

Above the main system is a dynamic damper determined by the mass m_2 and the spring k_2 . Equation (1) represents the equation of motion in this case.

$$\begin{aligned} m_1 \ddot{x}_1 + k_1 x_1 + k_2 (x_1 - x_2) &= F_1 \\ m_2 \ddot{x}_2 + k_2 (x_2 - x_1) &= 0 \end{aligned} \quad (1)$$

Equation (2) expresses the resonance frequencies of the general solution, f , which satisfy this condition.

$$f = \frac{1}{2\pi} \sqrt{\frac{1}{2} \left(\frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2} \pm \sqrt{\left(\frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2} \right)^2 - \frac{4k_1 k_2}{m_1 m_2}} \right)} \quad (2)$$

The respective resonance frequencies are further termed f_1 and f_2 from lower to higher. Here, with the resonance frequency of the main system and the dynamic damper identical at f_0 , and the mass ratio μ termed m_2/m_1 , the difference in width between the two resonance frequencies, Δf , is given by Eq. (3).

$$\begin{aligned} \Delta f &= f_2 - f_1 \\ &= \sqrt{\frac{1}{2} \left((2+\mu)f_0^2 + \sqrt{\mu(4+\mu)}f_0^2 \right)} - \sqrt{\frac{1}{2} \left((2+\mu)f_0^2 - \sqrt{\mu(4+\mu)}f_0^2 \right)} \end{aligned} \quad (3)$$

The resonance frequency of the steering system and the frequency of the dynamic damper, f_0 , was set at 40 Hz, and the mass m_1 , the mass of the dynamic damper subtracted from the total mass, was set at 10 kg. Figure 4 shows results calculated with the mass of the dynamic damper added to the steering wheel considered as m_2 , and the range of effect of the dynamic damper as the difference between the two resonance frequencies in the two-degree-of-freedom vibration model, Δf . In order to reduce steering vibration in a frequency width of 15 Hz, the mass of the dynamic damper would ideally be 1.4 kg. Given that the mass of currently used SRS modules is around 1.2 kg, it was predicted that the use of SRS modules as dynamic dampers would represent an effective vibration reduction method with general applicability.

2.5. Resonance Characteristic of SRS Module and Target Characteristic

The results of unit excitation tests conducted on floating SRS modules indicated the existence of the vertical and horizontal resonance modes shown in Fig. 5.

The resonance characteristics of SRS modules differ significantly due to factors including their design, the position of their center of gravity, and their spring characteristics. Figure 6 shows the vertical and horizontal resonance frequency modes for eight different SRS modules used in different vehicle models. When the resonance frequency of the SRS module is close to the range of the 1st-order steering resonance frequencies in complete vehicles, a vibration reduction effect can be observed, but this effect is small when the frequencies are further separated. In order

for the SRS module to display an optimal steering vibration reduction effect, its resonance frequency should be set in the middle of the range of the 1st-order steering resonance frequencies in complete vehicles.

Given this, it would be necessary to make the resonance frequency lower than that of conventional modules, and a new structure able to stably realize this demand was therefore investigated.

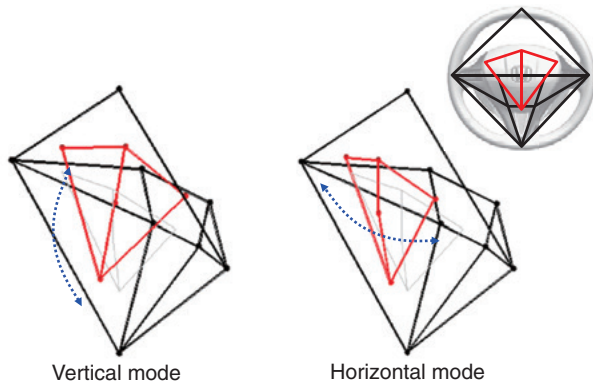


Fig. 5 Shapes of SRS module resonance frequency modes

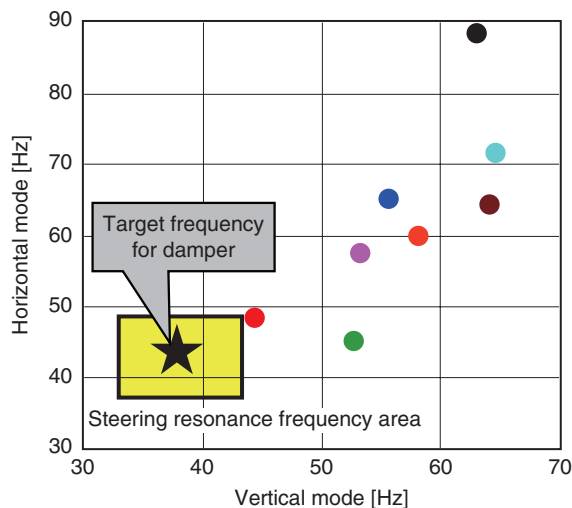


Fig. 6 SRS module resonance frequency

3. Development of SRS Module Damper Structure

3.1. Overview of New Structure

This paper proposes an extension of the conventional concept of the dynamic damper, leading to a method of using an SRS module as a dynamic damper. Figure 7 shows a conceptual diagram of the proposed method. The target resonance characteristic was realized by changing the support structure at the point of connection to the horn that had been used in the existing floating structure. Figure 8 shows the component parts of the developed structure. The

distinctive feature of the new structure is the positioning of a floating bush between the insulators A and B.

3.2. Proposal of Independent Spring Structure

Figure 9 shows the points of change in the developed structure and the concept of the associated spring model. In the conventional structure, the coil spring and the stopper bush are connected, and the resonance characteristic of the structure is determined by the shear spring characteristic (K_r).

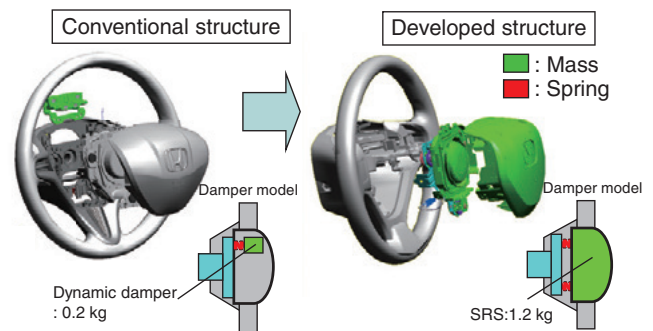


Fig. 7 Models of conventional and developed dynamic dampers

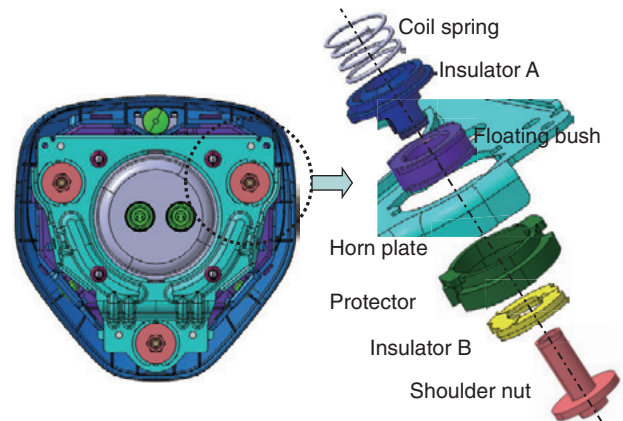


Fig. 8 Structure of SRS module damper

	Structure	Stiffness model
Conventional		
Developed		

Fig. 9 Comparison of conventional and developed structures

of the rubber bush that is pushed by the coil spring. The structure's resonance frequency was higher than the 1st-order steering resonance frequency, and it was necessary to reduce it in order to realize the desired vibration reduction effect. It would have been possible to reduce the resonance frequency by lowering the spring characteristic of the coil spring (K_a), but this would have resulted in the possibility of the horn sounding when the steering tilt and telescopic mechanisms adjusted the position of the steering system.

In order to resolve this issue, a method of using a separate spring that was not dependent on the spring characteristic of the coil spring was considered. A structure in which a spring characteristic, K_r , independent from the coil spring, could be set by positioning a floating bush between the insulators A and B was therefore proposed. Because the reaction force of the coil spring would be received by insulator A and the shoulder nut, it would be possible to reduce the resonance frequency of the SRS module with no effect on the spring characteristic perpendicular to the axis, as in the case of the conventional structure.

Figure 10 shows the relationship between the target for horn function and the target for SRS resonance frequency in relation to the coil spring.

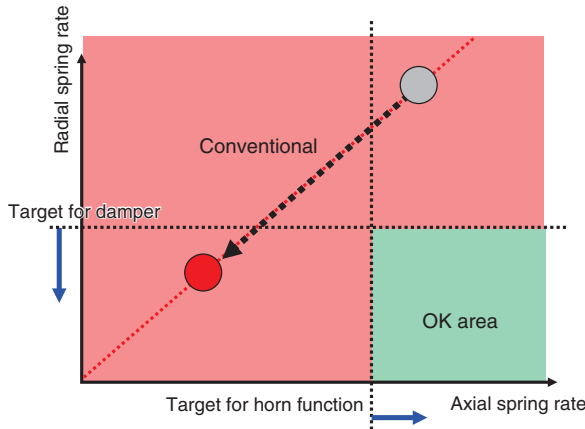


Fig. 10 Compatibility with horn function

4. Test Results

4.1. Results of Unit Verification Tests

The unit resonance characteristic of the developed SRS module was tested, and the results indicated that it realized the target characteristic shown in Fig. 6. Figure 11 shows the results of excitation tests. These results indicate that the peaks of inertance are the vertical and horizontal resonance modes shown in Fig. 5. The developed structure uses a circular floating bush, but vertical and lateral resonance could each be changed by altering the shape of the bush.

Unit tests were conducted in which the spring characteristic of the coil spring was varied in order to verify the viability of the proposed structure. The results of these

tests are shown in Fig. 12. While lowering the spring characteristic of the coil spring in the conventional structure resulted in a similar reduction in the resonance frequency of the SRS module, the frequency did not change in the new structure.

With regard to functioning as an airbag system, it was also demonstrated that the new structure achieved the same level of performance in terms of expansion strength and durability as the conventional structure.

4.2. Results of Vehicle Verification Tests

The results of a study of the steering vibration characteristic in a complete vehicle using the SRS module damper showed that both the vibration reduction effect and the range of frequencies in which the effect occurred were significantly increased with the use of the module against the use of conventional dynamic dampers. Figure 13 shows the vertical steering resonance characteristic. The vibration reduction effect of the new structure was 12 dB against 3 dB for a conventional dynamic damper, and the frequency range in which the effect occurred was approximately four times greater for the new structure at 18 Hz.

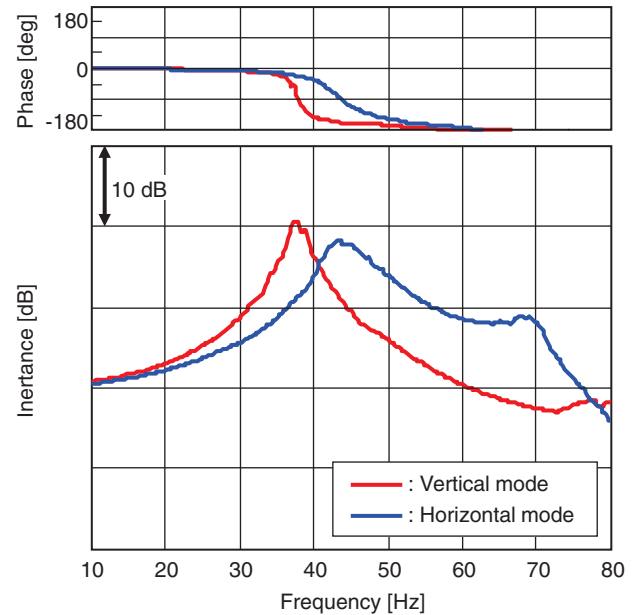


Fig. 11 Resonance frequency of SRS module damper

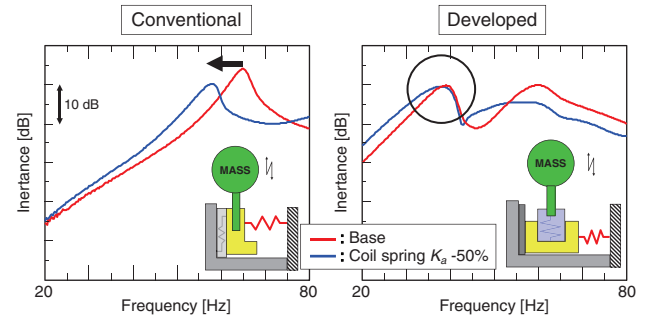


Fig. 12 Verification of stiffness model mechanism

The effect on the three vibration phenomena in completed vehicles was then examined. This study used the same SRS module dampers in different vehicle models.

Figure 14 shows the effect of the developed structure in ameliorating driving vibration and idle steering vibration. Driving vibration occurs when the 2nd-order component of tire rotation and the steering resonance frequency coincide. A vibration reduction effect of 6 dB or more was achieved at vehicle speeds from around 120 to 160 km/h. Strong idle steering vibration is generated when engine vibration components and the steering resonance frequency coincide. In the case of the tested vehicle using a four-cylinder engine, a high level of steering vibration was generated by the 4th-order engine vibration. The use of the SRS module damper resulted in a 10 dB reduction in vibration in the engine speed range set for idling. Similarly, in the case of six-cylinder engines, 3rd-order engine vibration produces significant steering vibration. The use of the developed SRS module resulted in a 12 dB reduction in vibration in the test vehicle.

Figure 15 shows the effect of the developed SRS module in ameliorating launch vibration. In many cases, launch vibration becomes an issue when the vehicle is operating in the low-engine-speed range under high loads for the purpose of increasing fuel efficiency. The use of

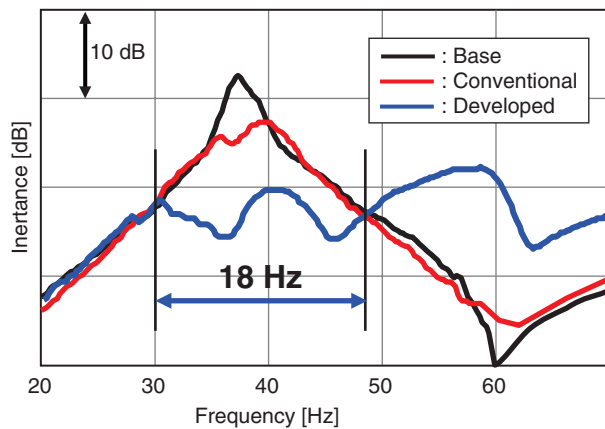


Fig. 13 Effect of SRS module damper in actual car

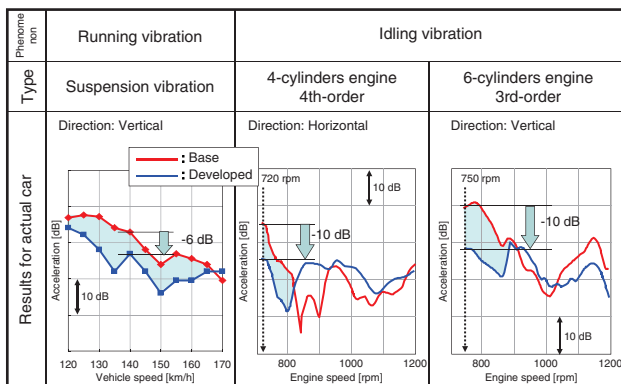


Fig. 14 Measurement results of running vibration and idling vibration

the developed SRS module reduced vibration by 10 dB or more in the vehicle using a four-cylinder engine, and realized a vibration reduction effect from 1200 rpm to 1800 rpm. Some six-cylinder engines are provided with cylinder-idling systems that deactivate the cylinders on one side of the engine. When this is the case, the engine speed and frequency conditions that excite launch vibration are different, and the achievement of vibration reduction for marketability can represent a further challenge. The use of the SRS module realized a reduction of 7 dB in launch vibration in the 1.5th-order range with deactivated cylinders. In addition, an 8 dB vibration reduction was also achieved in the case of 3rd-order vibration with all cylinders activated.

In designing a dynamic damper, it is necessary to consider the range in which the reduction of vibration is desired and the range in which vibration will increase due to an increase in the degree of freedom. In the case of the SRS module damper, by comparison with the vibration characteristic of conventional units, there is an increase in the vibration in the range close to 60 Hz (Fig. 12). However, while a tendency for launch vibration to increase in this range was observed in the cases of the four- and six-cylinder engines, the vibration level was low and its effect minimal in terms of marketability.

The results of the measurements in complete vehicles discussed above indicate the achievement of significant reductions in steering vibration and the realization of the goal of achieving an effect in multiple vehicle models using a single structure. Because the use of this structure makes it possible to realize the target characteristic for steering vibration from the initial stage of development, specifications can be decided on at an early stage.

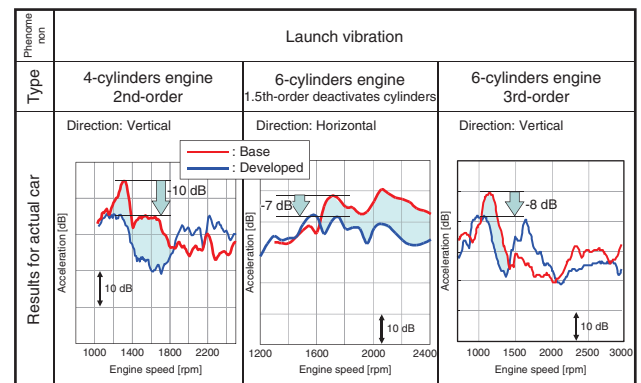


Fig. 15 Measurement results of launch vibration

5. Conclusion

A new method of ameliorating vibration phenomena originating in 1st-order resonance of the steering system was proposed, and a new SRS module structure was developed that realized the following.

(1) SRS module damper structure

A method in which an SRS module was employed as a dynamic damper was proposed, and the SRS module damper structure was realized.

(2) Balance between increased marketability and weight savings

This technology has achieved a balance between increased commercial appeal, cost and vehicle weight reduction.

(3) Early decision on specifications and reduction of development man-hours

The use of a vibration-reducing structure with a high degree of general applicability from the initial stages of the development process makes a conventional dynamic damper unnecessary, and also contributes to a reduction in development man-hours.

This new structure was employed in the new model CIVIC which went on sale in the spring of this year. It is also scheduled to be employed in subsequent models.

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