

Study of Suspension System for Reduction of Road Noise

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ABSTRACT

The front strut / rear torsion beam suspension systems employed in the majority of compact cars were considered from the perspective of road noise performance. The low-frequency band from 100 to 200 Hz was the subject of the analysis. Tests using 14 vehicles employing front strut / rear torsion beam suspension systems demonstrated the existence of shared suspension vibration modes. FE model analyses clarified the mechanisms of the respective vibration modes, helping to enable the identification of design guidelines for the enhancement of road noise performance.

1. Introduction

The enhancement of fuel efficiency and the achievement of weight-savings in the vehicle body in order to realize this goal are today two of the most important technological elements in vehicle development. The achievement of weight-savings in the vehicle body and NV performance are normally antagonistic goals, and balancing the two represents a significant issue. Not only are vehicles very prone to be affected by road noise when the tires, wheels, suspension, subframe and other parts are reduced in weight, but there are numerous cases in compact cars in particular in which low-frequency (100 Hz to 200 Hz) road noise in the cabin represents an issue when driving on rough roads.

Numerous parts are implicated in low-frequency road noise, but vibration generated by the natural mode of the suspension is known to have a significant effect on cabin noise^{(1), (2)}. In order to accurately predict road noise, it is necessary to study the characteristics of the suspension with consideration of their combination with the natural frequencies of the tires and the acoustic characteristics of the body⁽³⁾. However, given that the basic specifications of the suspension are decided at an early stage of vehicle development, at which it is not possible to sufficiently predict the characteristics of the tires or the body, techniques for studying the NV performance simply by using the suspension system are also necessary. The research discussed in this paper focused exclusively on the suspension, and identified the vibration modes which made the greatest contribution to road noise and clarified the vibration mechanisms of those vibration modes.

The suspensions focused on in the study were front strut/rear torsion beam systems (Fig. 1), of the type generally used in compact cars.

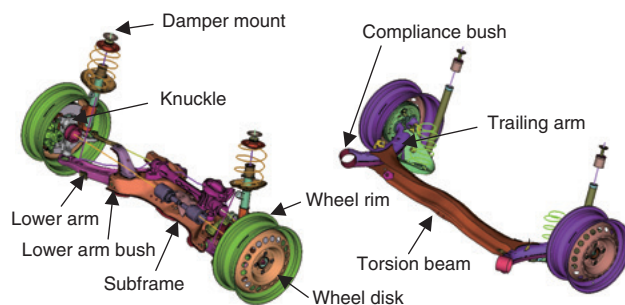


Fig. 1 Front strut and rear torsion beam suspension

2. Research Methodology

The arrangement of the suspension support points and the composition of the parts are virtually identical in front strut/rear torsion beam suspension systems, even when used in different vehicles. In addition, when the analysis is restricted to compact cars, there is no significant difference in the weight or stiffness of the wheels or the suspension parts. It can therefore be conjectured that the distribution of the spring and mass elements that determine the suspension vibration modes are similar for all these vehicles, and that each type of suspension displays vibration modes shared with other suspensions of the same type.

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First, the mechanisms of these vibration modes will be analyzed. Next, the degree to which the mechanisms of the vibration modes are shared will be considered by means of comparative tests using vehicles employing the same suspension type, and factors generating differences in vehicle performance will be analyzed. The results of studies of methods for road noise reduction derived from these vibration mechanisms will further be discussed.

3. Suspension Vibration Mechanisms

Figure 2 shows measurements of the cabin sound pressure level at the position of the driver's ears while driving on road surfaces for the evaluation of cabin road noise in a representative vehicle, and the lateral vibration response of the lower part of the knuckle when the bottom edges of the front wheels undergo lateral excitation. Both sound pressure and vibration display large peaks at close to 120 and 180 Hz.

Figure 3 shows measurements of the sound pressure level at the position of the rear passenger's ears and the vertical vibration response of the compliance bush connecting points when the bottom edges of the rear wheels undergo lateral excitation. Measurements for both sound pressure and vibration display large peaks at close to 120 Hz.

Below, the mechanism of the suspension vibration mode generating these vibration peaks will be classified by means of a parametric study using an FE model.

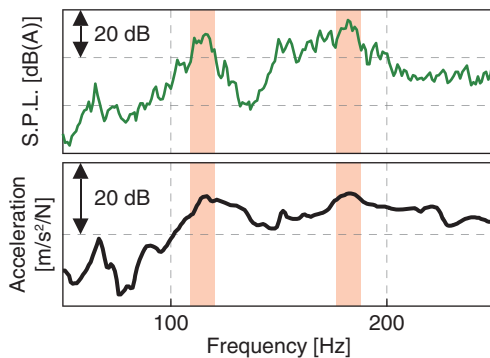


Fig. 2 Road noise and front suspension vibration

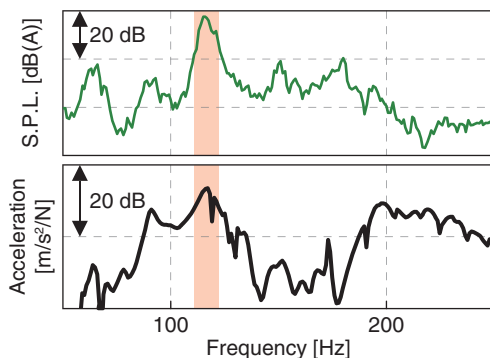


Fig. 3 Road noise and rear suspension vibration

3.1. Vibration Mode of Front Strut Suspension

Figure 4 shows the vibration mode close to 120 Hz. This is a mode in which the damper case, the knuckle, the wheel, and the lower arm rotate basically as a single unit in the direction of the camber. Because the suspension as a whole appears to rotate as a rigid body, this mode was termed “rigid camber mode” in the present research.

Figure 5 shows the vibration mode close to 180 Hz. In this mode, the wheel disk, the hub bearing, the knuckle and other parts undergo elastic deformation, while the wheel rotates in the direction of the camber. For this reason, this mode was termed “elastic camber mode” in the present research.

3.1.1. Vibration mechanism in rigid camber mode

Figure 6 shows the model generated when the rigid camber mode is expressed as a simple spring-mass system.

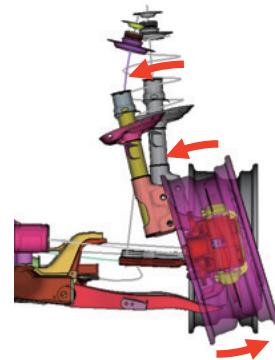


Fig. 4 Rigid camber mode

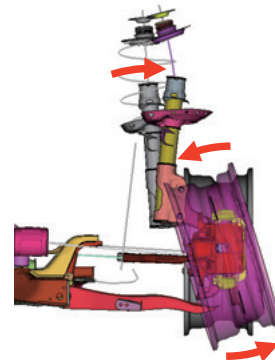


Fig. 5 Elastic camber mode

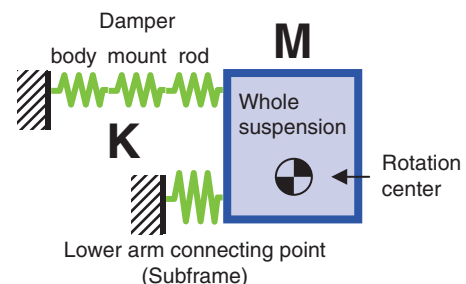


Fig. 6 System concept of rigid camber mode

M represents mass elements, while K represents spring elements. This can be regarded as a configuration in which the large mass of the entire suspension is supported by two springs, the upper section by the damper rod, the damper mount, and the damper connecting points, representing series springs, and the lower section by the stiffness at the lower arm connecting point. These upper and lower springs are the elements that control this mode.

3.1.2. Vibration mechanism in elastic camber mode

Figure 7 shows a simple spring-mass model of elastic camber mode. This can be regarded as a configuration in which the wheel rim, the main mass element, is supported at its center by two rotating springs, the prying stiffness of the hub bearing and the bending stiffness of the wheel disks, which represent the main spring elements. The swan neck section of the knuckle, the damper rod, and the damper mount also have an effect on this vibration mode as spring elements.

3.2. Vibration Mode of Rear Torsion Beam Suspension

Because the heavy, laterally coupled structure is supported by a comparatively flexible compliance bush in torsion beam suspensions, this configuration displays good isolation from vibration input, and offers advantages from the perspective of road noise⁽⁴⁾. However, because the torsion beam functions as a torsion spring connecting the left and right trailing arms, this member is prone to elastic deformation, and has a comparatively significant effect on road noise.

Figure 8 shows the vibration mode at 120 Hz. The U cross-section of a representative automotive torsion beam is open towards the front. While the wheels rotate in the direction of the camber, the center of the beam deforms elastically, generating pitching.

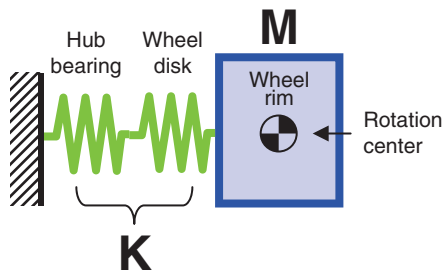


Fig. 7 System concept of elastic camber mode

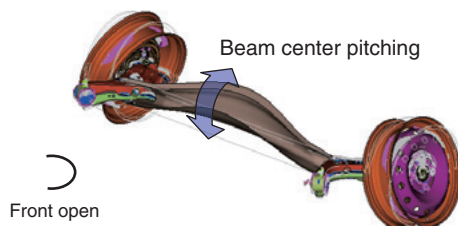


Fig. 8 Torsion beam vibration mode, with front open section

3.2.1. Vibration mechanism in torsion beam elastic deformation mode

As the conceptual diagram shown in Fig. 9 indicates, the wheel rims and the center of the beam represent the mass elements, while the trailing arm torsion, the torsion beam, and the compliance bush are the main spring elements.

As in the front suspension, the wheel rims rotate in the direction of the camber. A force producing vertical bending is therefore applied to the beam. Because the shear center of the beam U cross-section is positioned at the rear of the cross-section when the cross-section is open at the front, twisting occurs at the same time as vertical bending.

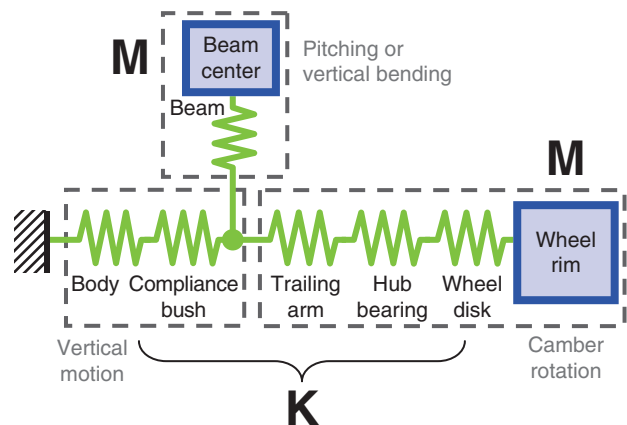


Fig. 9 System concept of torsion beam mode

4. Test Method

Excitation tests were performed on multiple vehicles in order to determine the extent to which the suspension vibration modes discussed in Chapter 3 were shared. All the test vehicles were drawn from the compact car category. Excitation tests were performed on the front strut suspension in ten vehicles, for a total of 25 types when differences in wheel materials and sizes are taken into consideration. Excitation tests were conducted on the rear torsion beam suspension in 14 vehicles. Transfer functions were measured in each section of the suspensions with the lateral direction of the bottom edges of the wheels as the points of excitation, and vibration modes and vibration transfer characteristics at representative points were compared.

5. Test Results

5.1. Results of Comparative Analyses of Front Suspensions

Rigid and elastic camber modes were studied in all the test vehicles. These two modes are believed to exist in almost all front strut suspensions. Figure 10 shows the vibration transfer functions for the lateral direction of the bottom edges of the knuckles in five vehicles. In rigid camber mode, there are minimal differences between the natural frequencies

for each vehicle, but significant differences in their levels of vibration response. In elastic camber mode, there are significant differences between natural frequencies, but minimal differences between levels of vibration response.

Figure 11 shows the relationship between the lateral vibration response of the knuckles and the vertical stiffness of the lower arm connecting point at rigid camber mode frequencies. These results indicate that the vibration of the suspension will decrease, and road noise performance will consequently be enhanced, in proportion to the stiffness of the connecting point of the lower arm, the main spring element.

5.2. Results of Comparative Analyses of Rear Suspensions

Figure 12 shows the arrangement of all measured modes. Each vehicle shares a large number of vibration modes from 100 to 200 Hz, but the vibration modes in which vertical vibration of the compliance bush (the main input point) is high and becomes the major factor in road noise differ with the direction of the opening of the beam U cross-section.

When the U cross-section is open at the front, the vibration mode generates pitching of the center of the beam, as shown in Fig. 8; when the U cross-section is open at the bottom, the vibration mode generates vertical vibration of the entire suspension while the beam bends vertically (Fig. 13).

The vibration response level in this mode was compared in the vertical direction at the body side of the compliance bush connecting point. Figure 14 shows the relationship

between the degree of isolation of the compliance bush from vibration in the vertical direction and vibration on the body side. Significant differences can be observed between each vehicle, and the magnitude of these differences is strongly correlated with the degree of isolation of the compliance bush from vertical vibration. It may be indicated that the greater the degree of isolation of the compliance bush from vibration, i.e., the lower the vertical stiffness of the bush, the lower the degree of vibration on the body side, and hence the better the road noise performance.

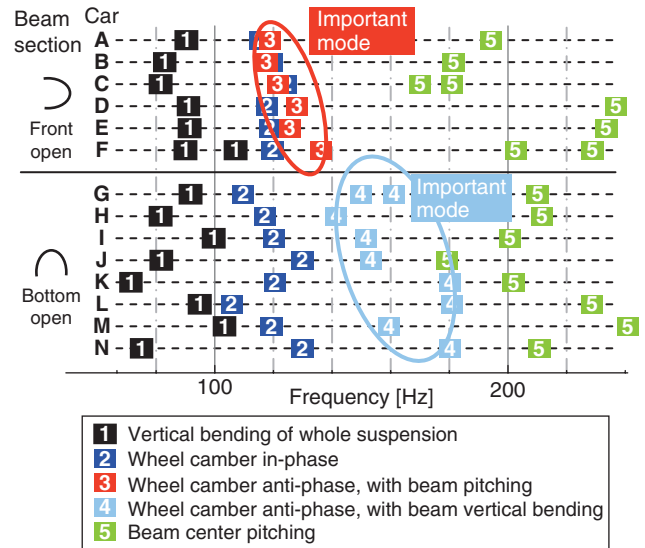


Fig. 12 Rear torsion beam suspension modes

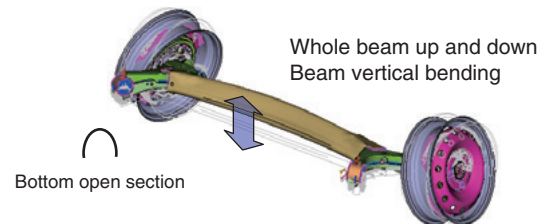


Fig. 13 Torsion beam vibration mode, with bottom open section

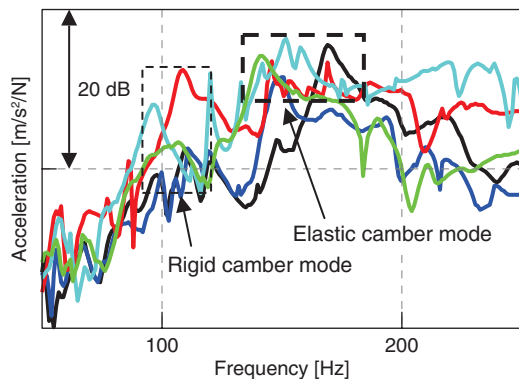


Fig. 10 Camber modes of various cars

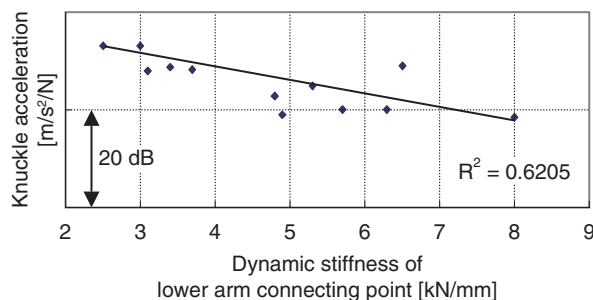


Fig. 11 Lower arm connecting point vertical dynamic stiffness and knuckle acceleration

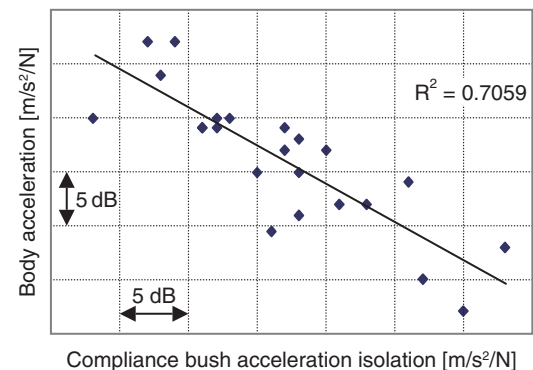


Fig. 14 Compliance bush isolation and body acceleration

6. Study of Methods for Reduction of Road Noise

This chapter will discuss methods for the reduction of the sound pressure level of road noise originating in the suspension vibration modes.

6.1. Front Rigid Camber Mode

Methods of reducing vibration at the lower arm connecting point, a transfer path that makes a significant contribution to cabin noise, were examined. Because in this suspension configuration the large mass rotating in the camber direction is supported by two distant spring elements, the rotational center of the mass can be controlled by means of the ratio between the two springs. It was possible to reduce the vibrational input to the lower arm by reducing the lateral spring of the damper mount in order to lower the rotational center, which had been positioned close to the center of the wheel center in the base state. Figure 15 shows a conceptual diagram, while Fig. 16 shows the results of testing of the measure in an actual vehicle.

6.2. Front Elastic Camber Mode

6.2.1. Correlation between elastic camber mode natural frequencies and cabin noise

As indicated in Section 5.1, in elastic camber mode, there are minimal differences in vibration levels between vehicles, but significant differences between natural frequencies.

A study of the relationship between natural frequencies in the elastic camber mode and cabin noise during lateral excitation of the lower edge of the wheels revealed the loose correlation shown in Fig. 17. The results showed that vibration sound pressure would tend to be lower the lower the natural frequency.

In most cases, a low natural frequency in the elastic camber mode is achieved by means of increasing the weight

of the wheel rims, the main mass element, and a reduction in vibration sound pressure is a natural outcome. However, measurements conducted in the present project showed that almost identical results were obtained when comparing wheels of basically identical mass (Fig. 18), indicating that irrespective of the weight of the wheels, lower natural

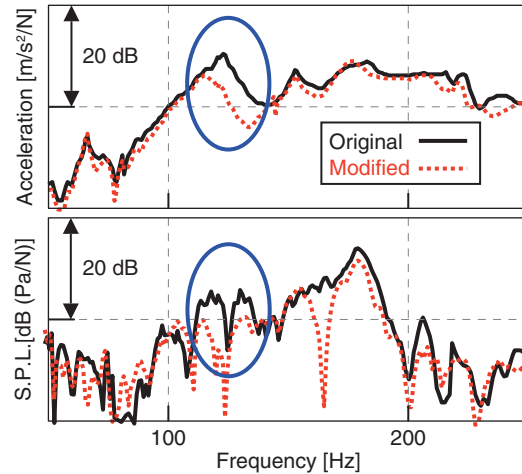


Fig. 16 Enhancement of rigid camber mode by decreasing damper mount lateral stiffness

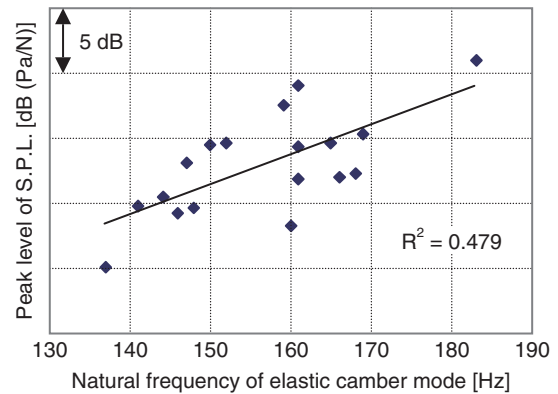


Fig. 17 Elastic camber mode natural frequency and S.P.L.

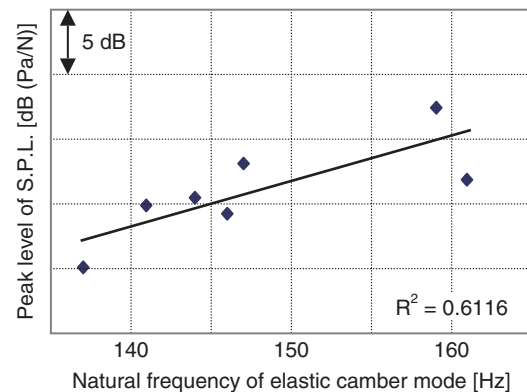


Fig. 18 Elastic camber mode natural frequency and S.P.L. around 8 kg wheel only

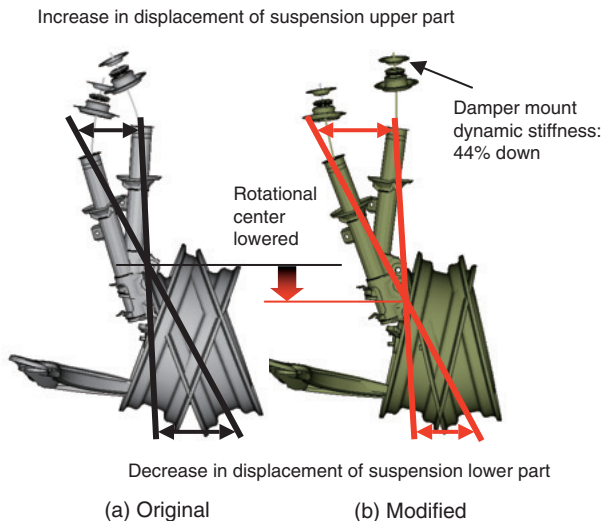


Fig. 15 Lower rotational center of rigid camber mode

frequencies could be considered to tend to enhance road noise performance. Based on these results, it was conjectured that the lower the natural frequency of the suspension elastic camber mode, the less coupled it is with that of the body side response points, and that all the vehicles would display this point of weakness in relation to the body side response points at frequencies close to the elastic camber mode.

6.2.2. Considerations regarding body acoustic sensitivity

This sub-section will discuss acoustic sensitivity from the suspension connecting points, the body side response points, to the sound measurement position in the cabin (body acoustic sensitivity). Because the body acoustic sensitivity is determined by a number of elements, including the inertance of the suspension connecting point, the body framework mode, and the vibration characteristics of the various panels, it is unlikely that the point of weakness in relation to the body side exists in the same frequency band in every vehicle. However, because the basic characteristics of the cabin acoustic mode – one of the elements which determines the body acoustic sensitivity – is determined by the shape and volume of the cabin, it was possible to consider that the acoustic mode peak would occur in similar frequency bands in the present research, which employed vehicles of almost identical size.

Figure 19 shows measurements of cabin acoustic mode and body acoustic sensitivity. The transfer function (sound pressure / volume acceleration) to a microphone set up at the position of the driver's ears when random noise was generated in the same position via a speaker was employed as an index to express the cabin acoustic mode. The average value of body acoustic sensitivity from all the front suspension connecting points was employed as an index to express body acoustic sensitivity. As Fig. 19 shows, the average acoustic sensitivity and the cabin acoustic mode both peak at the same frequencies, between 160 and

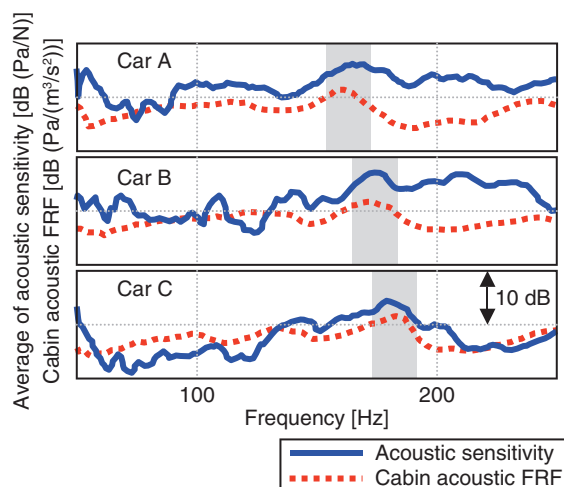


Fig. 19 Cabin acoustic mode and body acoustic sensitivity

180 Hz, close to the elastic camber mode frequency band. Given this, it may be considered that the natural frequency of cabin acoustic mode will also appear in body acoustic sensitivity as a peak. Because this acoustic mode frequency can be predicted when the cabin shape is determined at the initial stage of development, it would be possible to take measures to shift the natural frequency of the elastic camber mode of the suspension.

Figure 20 converts the horizontal axis of Fig. 17 into the difference between the frequencies of elastic camber mode and cabin acoustic mode. It is clear that the correlation becomes higher.

6.2.3. Verification of effect in reducing road noise

The reduction in road noise produced by modifying the natural frequency of the elastic camber mode was studied. The prying stiffness of the bearings and the bending stiffness of the wheel disks, the main spring elements, were modified.

Figure 21 shows results for wheel excitation tests with the original and modified hub bearing stiffness, and

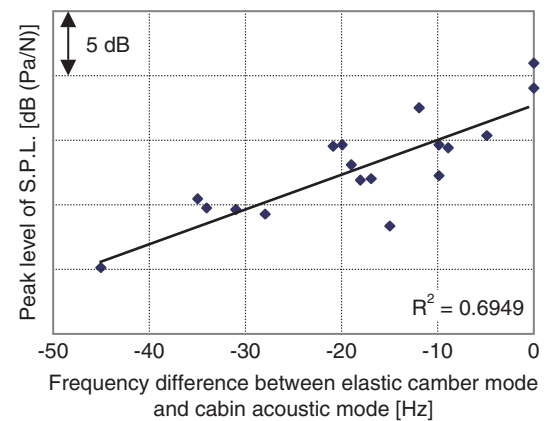


Fig. 20 Frequency difference between elastic camber mode and cabin acoustic mode, and S.P.L.

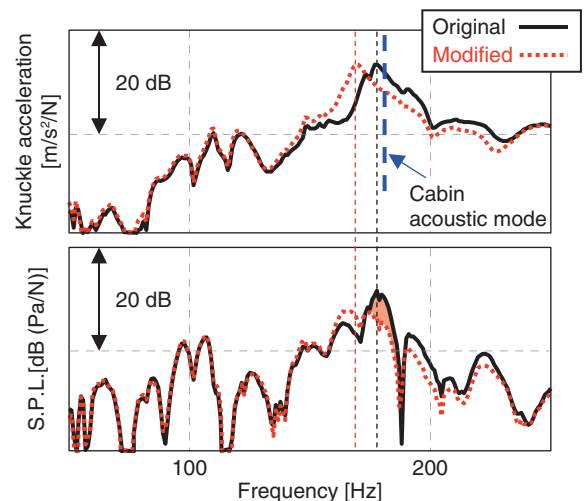


Fig. 21 Enhancement of road noise by decreasing stiffness of hub bearing

Fig. 22 shows results for the original and modified wheel disk stiffness. In both cases, the natural frequency of the elastic camber mode has been lowered, and cabin noise has been ameliorated.

6.3. Rear Torsion Beam Elastic Deformation Mode

As indicated in Section 5.2, reducing the vertical stiffness of the compliance bush is an effective means of enhancing road noise performance. In the present project, the reduction of the vertical vibration of the compliance bush connecting point by modifying the shape of the beam itself was attempted as another method. The study employed an analysis using an FE model of a torsion beam for a mass-production vehicle with a front opening.

As indicated above, in a torsion beam with an opening at the front, the beam twists as it undergoes vertical bending, and the central section displays pitching behavior with the rear extremity of the cross-section as the center. There is therefore significant vertical vibration of the front section of the center of the beam, and this increases the vibration of the connecting point of the compliance bush.

Because the compliance bush connecting point is located on the underside of the rear side frame, which extends longitudinally, from a structural perspective the stiffness of the connecting points is higher in the longitudinal direction than the vertical direction, and body acoustic sensitivity is lower in the longitudinal direction. Therefore, if it is possible to reduce vertical vibration of the suspension even if longitudinal vibration increases to some extent, road noise will be reduced. In order to realize this vibration characteristic, the angle of beam opening was slanted. Figure 23 shows the results, and Fig. 24 shows the difference in motion. The vertical vibration of the compliance bush connecting point has been reduced by modifying the vibration of the center of the beam from the vertical direction to the longitudinal direction. When

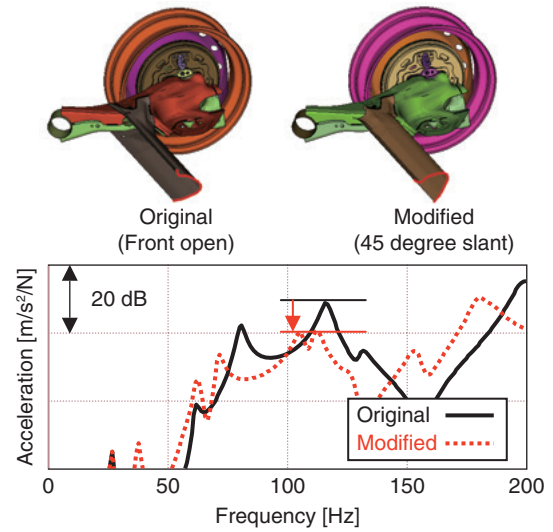


Fig. 23 Enhancement of torsion beam pitching mode

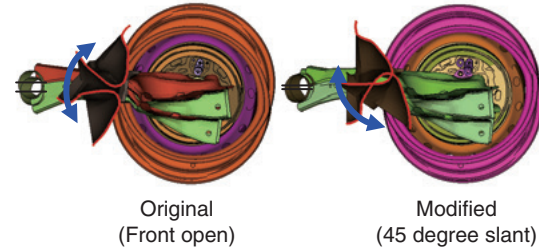


Fig. 24 Enhancement of torsion beam mode shape

the direction of the opening was shifted to the bottom of the beam using the same model, the characteristic bottom opening mode described in Fig. 13 became prominent, and road noise performance was affected. In the analysis discussed here, 45° was the optimum opening angle.

7. Conclusion

The following conclusions were drawn from a parametric study of a suspension FE model and a survey of 14 compact cars.

- (1) It was demonstrated that front strut and rear torsion beam suspensions each possess shared vibration modes that are the chief factors in road noise. The main modes are front rigid and elastic camber modes, and a rear wheel camber mode that involves elastic deformation of the rear torsion beam.
- (2) The main spring and mass elements for control of the vibration mode were identified for each mode.
- (3) The potential for enhancement of road noise performance through the application of suspension vibration mode control derived from the mechanisms of generation of road noise was verified in vehicle tests and computational predictions.

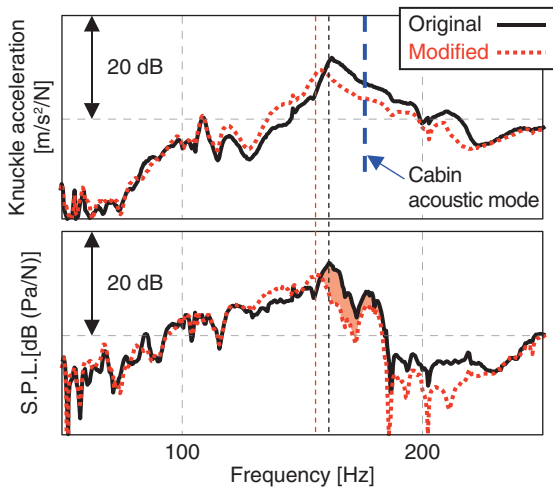
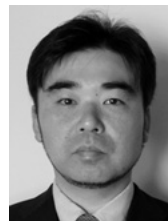


Fig. 22 Enhancement of road noise by decreasing stiffness of wheel disk

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