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EVALUATION OF ROAD-INDUCED NOISE OF A VEHICLE USING EXPERIMENTAL APPROACH

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ABSTRACT−In this paper a systematic test procedure for evaluation of road-induced noise of a vehicle and guidelines for each test are presented. Also, a practical application of the test procedure to a small SUV is presented. According to the test procedure, all the tests were performed to evaluate road-induced booming noise that is in low frequency range. First of all the information on characteristics of road-induced noise was obtained through baseline test. Coupling effects between body structure and acoustic cavity of a compartment were obtained by means of modal tests for a structure and an acoustic cavity. Local stiffness of joint areas between chassis system and car-body was determined by test for measurement of input point inertance. Noise sensitivities of body joints to operational forces were obtained through test for measurement of noise transfer functions. Operational deflection shapes made us analyze behaviors of chassis system under running condition and then find sources of noise due to resonance of the chassis system. Finally, Principal Component Analysis and Transfer Path Analysis were utilized to investigate main paths of road-induced noise. In order to evaluate road-induced booming noise exactly, all of tests mentioned above should be performed systematically.

KEY WORDS: Road-induced noise, Test procedure, Transfer path analysis, Principal component analysis, Noise sensitivity

1. INTRODUCTION

As the noise from a power train decreases, road-induced noise plays main role in interior noise of a vehicle. However, it is difficult to improve road-induced boom noise since not only the input force varies with driving speed of a vehicle and road conditions but also vibration energy is transmitted to the car-body through many paths. In general, transmitted force to a car-body is determined by local stiffness and damping of the joint area where the chassis system connects with a car-body. The transmitted forces induce boom noise due to resonance between acoustic cavity and structure of a vehicle. Therefore a lot of tests have been developed to evaluate and improve road-induced noise. For example, one has to perform the baseline test to find road-induced noise characteristics and then all kinds of modal tests are performed to determine modal properties of a cavity and a car-body. In addition, operational deflection shape (ODS) measurement and multifarious tests such as noise transfer function (NTF) measurement and inertance measurement, which are necessary to Transfer Path Analysis (TPA), will be performed. The noise transfer functions give the information about noise sensitivities of body joints to operational forces and inertance functions give the information about local stiffness of joints. The noise contribution of transfer paths such as power train mounts and suspension system mounts are finally obtained by means of TPA. Besides these essential tests, test for measurement of tire non-uniformity which gives information about order characterstics of tire excitation forces can be carried out.

Figure 1 shows the concept of trouble-shooting for road-induced noise. A set of combination of those tests gives the appropriate information about solution of the road-induced noise. Each test has both advantage and disadvantage for finding out characteristics of road-induced noise individually. Since some of those tests give same or similar information, time and efforts can be saved if several tests are performed according to a systematic test procedure. This paper presents a systematic test procedure for evaluation of road-induced noise and methodology for each test and will give some guidelines for those tests in section 2. In addition, for confirming the test procedure, an application to a small SUV is presented in section 3.

2. METHOD OF TESTS AND EVALUATION FOR ROAD-INDUCED NOISE

The test procedure for road-induced noise analysis can be summarized as follows:

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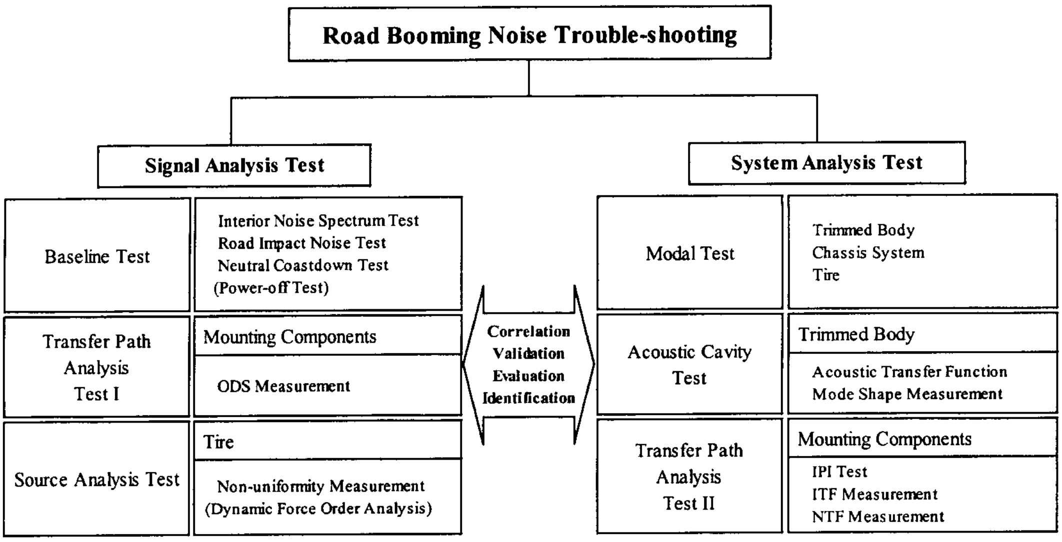


Figure 1. Test and analysis methods for trouble-shooting of road-induced noise.

2.1. Baseline Test

The object of baseline test is to find characteristics of road-induced noise. When the noise levels due to other noise sources such as engine and wind noise are low, the characteristics of the road-induced noise are well represented. Therefore, the experiments have to be performed at relatively low vehicle speed. One set of road-induced noise measurement is performed at a few constant vehicle speeds on rough road. In addition to the constant speed driving tests, the coast-down test is also performed. The coast-down test is performed with the transmission gear on neutral position so that the contamination of the road-induced noise from the influences of the engine and power-train noise may be minimal.

2.2. Modal Test of Acoustic Cavity

In analyzing road-induced noise phenomenon, the acoustic modes of a compartment cavity of a vehicle should be considered. In general, 6 or more acoustic cavity modes exist below 200 Hz for mid-size passenger car. The object of acoustic cavity modal test is to identify the resonance frequencies and to estimate the shapes of the acoustic modes at relatively low frequency range. One difficult thing in estimating the acoustic mode shapes of an enclosed cavity is that the sound pressure should be measured at all points of three-dimensional space. The minimum spatial resolution based on the maximum frequency of interest results in hundreds of measurement points in the interior of the test vehicle. Therefore, to scanning the whole enclosure inside the test vehicle is not considered feasible and the measurement is performed on one vertically extended plane and one horizontally extended plane.

2.3. Modal Test of a Body Structure

The object of body modal test is to find relationship between dynamic characteristics of a car-body and road-induced noise and then to decouple the correlated modes between body structure and acoustic cavity. First of all, one has to check the body structure characteristics based on modal data globally. In general, the bending mode, matchbox mode and torsion mode appear in one-box car. The vibration modes below 35 Hz have little affect on the interior noise but low rigidity of a body structure result in local vibration of panels. The modal test is necessary to confirm linearity and reciprocity of a car-body. To compare frequency response functions, which are measured from various input levels, is a simple method to check the linearity of a system.

2.4. Modal Test of Chassis Systems

The object of the suspension modal test is to find structural coupling effect between the suspension modes and body modes. Since the suspension system undergoes large displacement motion, the modal analysis has little meaning in itself in elastic boundary. However, if the natural frequencies of each part that is consisting of suspension system are much higher than interested frequency range, the suspension system might be considered as a quasi-static system. Also, the natural frequencies of the system are not much different from jounce position to rebound position and mode shapes are same. Joint characteristics and non-linearity of materials such as rubber affect the accuracy of modal test. When the excitation forces are large, non-linearity will be excited too. On the contrary, if the excitation forces are small, the structure cannot be fully excited. As results, the accuracy of test depends on magnitude of the excitation forces. In general, the sine sweep as a force signal is more effective than random or impact for a non-linear system. The selection of excitation position is an important factor in modal test.

2.5. Test for Measurement of Inertance

The object of test for measurement of inertance is to estimate stiffness of the body joints to which chassis components are attached. The stiffness of structures can be estimated by means of their inertance (or accelerance), which is defined as the frequency response functions between the accelerations of structures as responses and the forces as inputs. Particularly, the local stiffness of structure can be estimated by input point inertance (IPI) that makes use of the spectra of acceleration and force measured at an identical position.

2.6. Test for Measurement of Noise Transfer Function

The object of test for measurement of noise transfer functions is to investigate the noise sensitivities of body joints to operational forces. Because the body joints are main transfer paths of road-induced noise, to estimate precisely their noise sensitivities is important. All frequency response functions between the forces measured at chassis-body connection points and acoustical pressures measured at the ear positions of a driver are determined in the three orthogonal directions, X, Y and Z. The acoustic excitation method that is based on the vibro-acoustic reciprocity principle can be used to measure these functions because this method has some advantages compared with the mechanical excitation method using an impact hammer as follows:

(1) The noise transfer functions in all directions including in-plane of a surface can be measured simultaneously;

(2) The effects of stroke noise of an impact hammer can be eliminated;

(3) Even if there is no space for applying an impact hammer, the noise transfer functions can be measured.

2.7. Test for Operational Deflection Shape

As the modal test for chassis system represents only dynamic characteristics of the system, it cannot show dynamic behavior of the system under the driving condition. Therefore, real responses of the system in running can be found through the operational deflection shape test. The vibration modes are extracted at each measurement point under the driving condition. In general, many channels of measuring equipment are required because a lot of measurement points are necessary to describe the geometry of chassis system in detail. To overcome lack of channels, phase synchronized spectrums are measured at selected points. Then operational deflection mode shapes are determined by running mode analysis.

2.8. Test for Transfer Path Analysis

The Transfer path analysis is a method to find the noise contribution of transfer paths such as power-train mounts or suspension system mounts. In order to make use of the Transfer Path Analysis, test for measurement of the input point inertance and noise transfer function are performed in advance. In driving condition, the different wheel inputs due to road excitation are non-coherent and their phase relationships vary continuously. Therefore the Principle Component Analysis needs to be done to separate the uncorrelated inputs into sets of "independent phenomena". The Transfer Path Analysis is then carried out to calculate a set of partial pressures related to a single transfer path and one of the uncorrelated phenomena. The total noise pressure is then calculated by a summation over all phenomena in a RMS way.

3. APPLICATIONS AND DISCUSSION

Subjective appraisal of a test vehicle revealed that low frequency interior noise makes compartment passengers uncomfortable. In order to evaluate the road-induced booming noise in the range below 200 Hz, all tests were performed in accordance with the test procedure.

3.1. Baseline Test

The interior noise data were obtained at two different points inside the test vehicle. One is located at front seat near drivers ear position and another is located at the rear seat. The baseline test was repeated three times to yield an average data. The vehicle speed and road conditions for tests are summarized in Table 1. As the results of test, each constant vehicle speed test shows similar tendency, regardless of the road conditions and the vehicle speeds.

Table 1. Conditions of constant speed test for road-induced noise measurement.

No.	Vehicle speed (km/h)	Road conditions	Remarks
1	40	smooth	Smooth road -Asphalt road Rough road -Concrete road
2		rough	
3	60	smooth	
4		rough	
5	80	smooth	
6		rough	

Table 2. Conditions of coast-down test for road-induced noise measurement.

No.	Vehicle speed (km/h)	Road conditions	Remarks
1	80→30	smooth	TM on "Neutral", averaging 3 runs
2	80→30	rough	

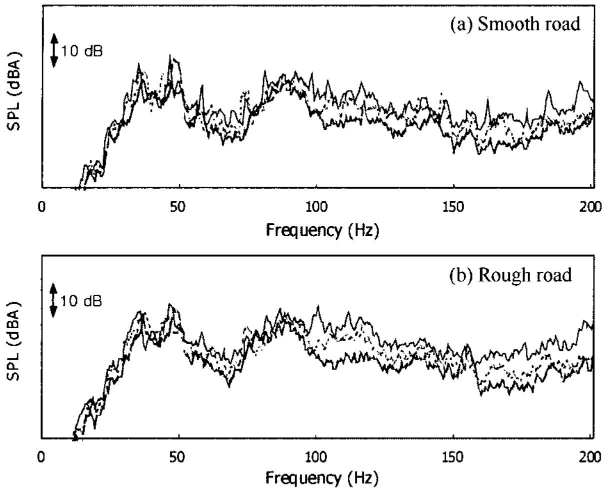


Figure 2. Spectra of sound pressure levels at the three different vehicle speeds on smooth road and rough one. -·-·-, 40 km/h; ------, 60 km/h; ——, 80 km/h.

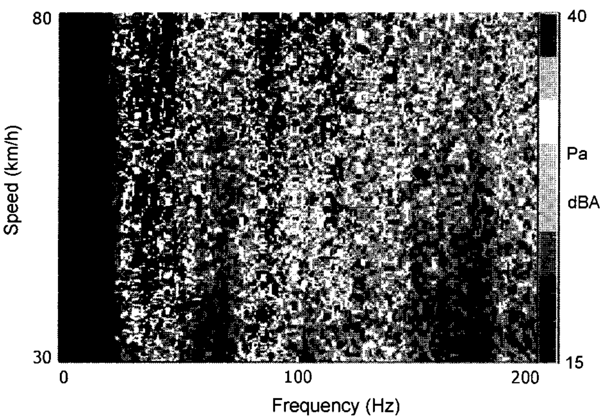


Figure 3. Color map of sound pressure levels measured inside of the test vehicle at coast-down on smooth road.

The sound pressure spectrum obtained from smooth road and rough one are shown in Figure 2. It can be seen that the sound pressure levels obtained from different vehicle speeds and roads share the dominant peaks at three frequencies of 35~40, 45~50, and 87~90 Hz. The coast-down test shows similar tendency to the constant vehicle

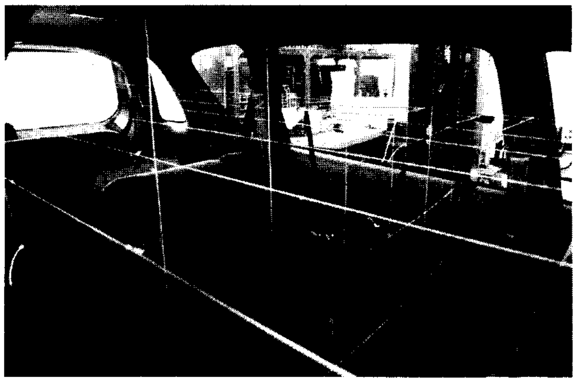


Figure 4. Grids installed on vertical and horizontal planes for measurement points.

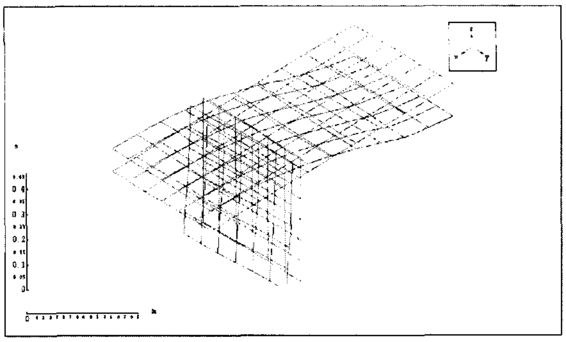
speed tests shown in Figure 3 and both signals obtained from the front seat and the rear one have little difference (see Table 2 to refer to the specifications of the test). Therefore, we can conclude that the road condition is not a critical factor, nor is the vehicle speed as long as the vehicle speed is relatively low in capturing the characteristics of the road-induced noise. From the results of baseline test, the three peaks of the noise below 100 Hz are aimed as targets to be reduced.

3.2. Modal Test of Acoustic Cavity

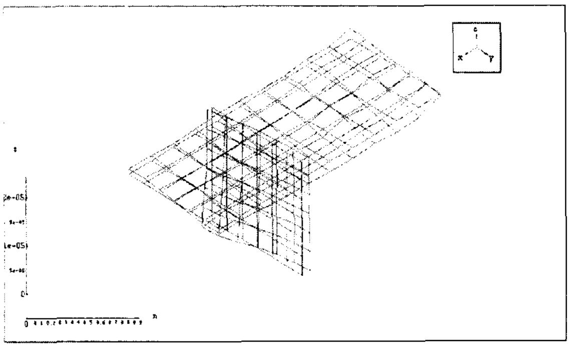
The frequency response functions were measured on two extended planes, one is vertical and another is horizontal. Figure 4 shows the measurement points in both planes. Because the three dimensional acoustic mode shapes of the cavity cannot be easily estimated based on the frequency response functions measured only on these two cross-sectional planes, a numerical method was used to yield the acoustic mode shapes and frequencies of the cavity based on a rough description of the cavity enclosure to be used as a guideline. According to the results of simulation, the first acoustic mode of the cavity was found at around 51 Hz and the second mode was at around 87 Hz. Similarly, the corresponding first mode shown in Figure 5(a) was measured at around 50 Hz and the second acoustic mode shape shown in Figure 5(b) was measured at 89 Hz. Considering the frequencies of the first two acoustic modes below 100 Hz, it is believed that those two modes are related with the two peaks at 45~50 Hz and 87~90 Hz in spectra from the baseline test.

3.3. Modal Test of a Body Structure

The modal test was performed by random input with an electro-magnetic shaker. The frequency response functions were measured using five tri-axial accelerometers at 110 points. Then the 12 modes were extracted within the frequency range of interest, and a global torsion mode of the test vehicle and a local mode of roof are located at 39



(a) First mode at 50.4 Hz



(b) Second mode at 90.0 Hz

Figure 5. Acoustic mode shapes of the compartment measured on the two planes.

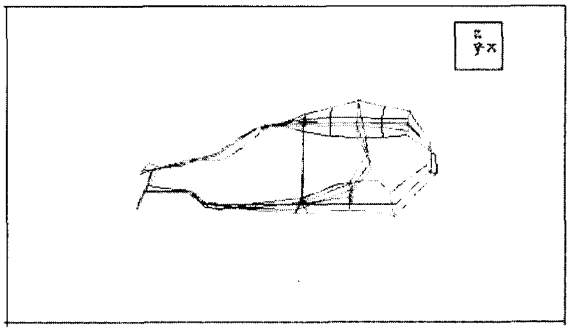
Table 3. Principal motion of the main modes of a body in accordance with frequency.

Mode	Frequency [Hz]	Principal motion
1	22.0	1st Torsion
2	25.2	2nd Bending
3	32.2	1st Bending
4	38.8	Torsion (rear) and bending (front)
5	48.7	Roof (0,1)

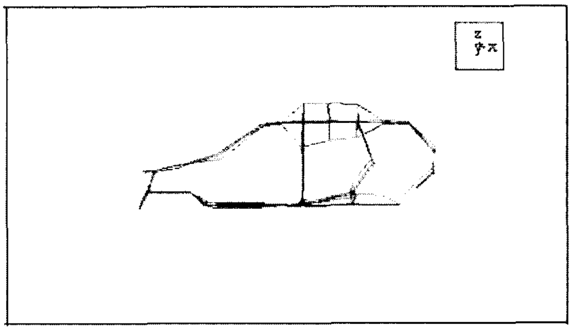
Hz and 49 Hz, respectively. The main modes of the body structure are summarized in Table 3 and two corresponding mode shapes are represented in Figure 6. Comparing the frequencies of those two modes with two acoustic modes below 100 Hz, it is supposed that the structural mode is coupled with acoustic mode at 50 Hz. In addition it is considered that those two modes of a vehicle body have effects on two peaks at 35~40 Hz and 45~50 Hz in spectra of the baseline test.

3.4. Modal Test of Chassis Systems

The test set-up for suspension modal test can be seen in Figure 7. Even though the tire has non-linear characteristics with large damping, it can be considered as a linear elastic part under the inflation condition. Therefore the tire contact point can be selected for an excitation point. The frequency response functions of the front and rear suspension systems were measured at 55 points and 62 points respectively on tire, spring, and sub-frame. Then several modes were extracted within the frequency range of interest below 100 Hz. Especially, some modes of two suspension systems exist at around 35~40 Hz,



(a) Torsion and bending mode shape at 38.8 Hz



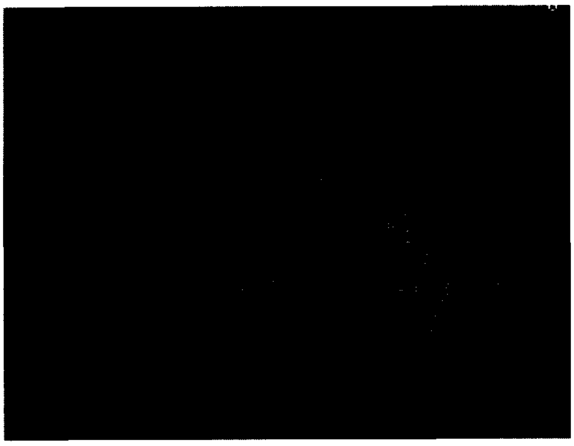
(b) Roof (0,1) mode shape at 48.7 Hz

Figure 6. Mode shapes of the test vehicle.

Figure 7. Modal test set-up for the rear suspension system.

Table 4. Principal motion of the main modes of front and rear suspension systems.

Suspension system	Mode	Frequency [Hz]	Principal motion
Front	1	36.8	Bouncing
	2	46.7	Translation of a tire bending of a x-member
	3	91.1	Bending of a tire
Rear	1	35.1	Bouncing
	2	45	Rotation of a tire about vertical
	3	94.4	Torsion of a tire



(a) Bouncing of a front suspension at 36.8 Hz

(b) Bouncing of a rear suspension at 35.1 Hz

Figure 8. Mode shapes of the suspension systems.

45~50 Hz, and 88~90 Hz. The peaks at the corresponding frequencies of those modes are found in the baseline test, too. The main modes of two suspension systems are summarized in Table 4 and two corresponding mode shapes are represented in Figure 8. Comparing the test results, it is considered that the bouncing of suspension systems, bending of a cross member and elastic modes of a tire have an effect on road-induced noise.

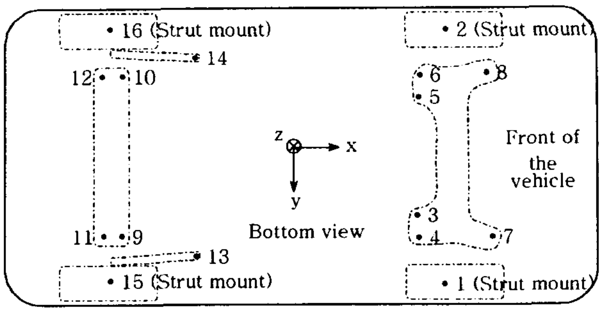


Figure 9. Positions on the vehicle body for measurement of input point inertance.

3.5. Test for Measurement of Inertance

In order to investigate the dynamic stiffness of the suspension-body and chassis-body attachment points, the input point inertance functions were measured at their body side. The forty-eight functions are measured in all directions because there are 16 points that are main transfer paths of road-induced noise in the vehicle as follows:

(1) Four strut mounting points located on the body side between body and suspension systems;

(2) Ten points of the front and rear sub-frames connected to the body;

(3) Two points of the trailing arms connected to the body. All positions and directions of measurement points

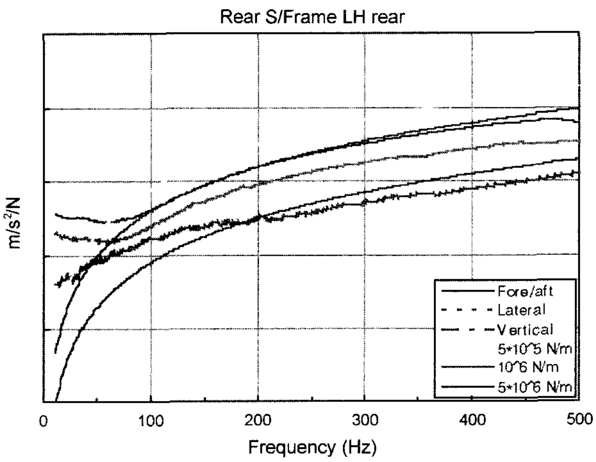


Figure 10. Constant stiffness curves and IPI's measured at a mounting point of a rear sub-frame (position of No. 11 in Figure 9) in 3 directions.

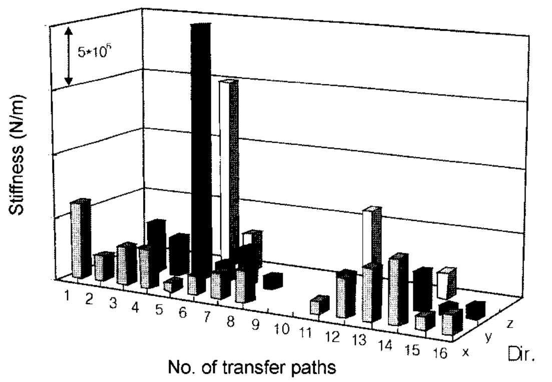


Figure 11. Local stiffness of the mounting points of the chassis systems on a body.

are shown in Figure 9.

As an example, Figure 10 shows the input point inertance spectra measured at the mounting point of a rear sub-frame in three orthogonal directions and the constant stiffness curves such as 5×10^5 N/m, 1×10^6 N/m, and 5×10^6 N/m. By comparing the input point inertances (IPI's) of the mounting points with the constant stiffness curves, the local stiffness of the points is evaluated as shown in Figure 11. Particularly, the point 15 and 16 (rear strut mounting points) show very low stiffness in all directions. From this result, it is guessed that the rear strut mounting points are one of major contribution paths for the road-induced noise.

3.6. Test for Measurement of Noise Transfer Function

The noise sensitivity of structures can be estimated by their noise transfer function (NTF) that is defined as the frequency response function between sound pressure as response and force as input. As the case of the previous test for measurement of the input point inertance, the noise transfer functions were measured at the same points. Because there are some fundamental limitations to using an impact hammer, the acoustic excitation method was used. By way of example, Figure 12 shows the noise transfer functions between driver's ear position and a rear sub-frame attachment point in three directions and A-weighted constant noise transfer function curves, such as NTF-45 dB/N, NTF-50 dB/N, and NTF-55 dB/N. In this case, three peaks affected on the road-induced noise reach up to 55 dB/N. Therefore the noise sensitivities are obtained by ways of the A-weighted constant NTF curves. Figure 13 shows the peak levels of all FRF's in *dB(A)/N* in the 35~40 Hz frequency range. Especially, 13:z and 14:z (trailing links) show very high sensitivity to the booming noise that is caused at the front seat. Thus there is the possibility that these paths have bad effects on the road-induced noise in the 35~40 Hz frequency range.

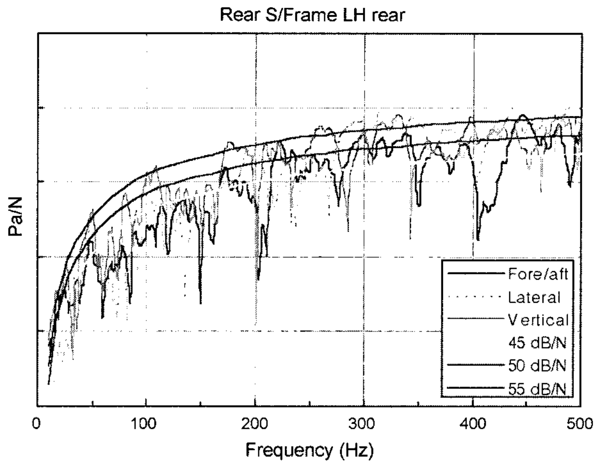


Figure 12. A-weighted constant NTF curves and FRF's measured at the mounting point of a rear sub-frame (position of No. 11 in Figure 9) in 3 directions.

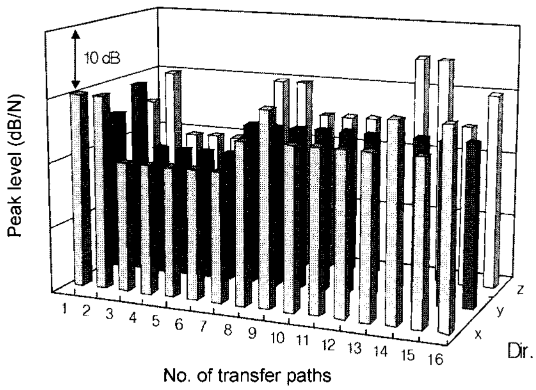


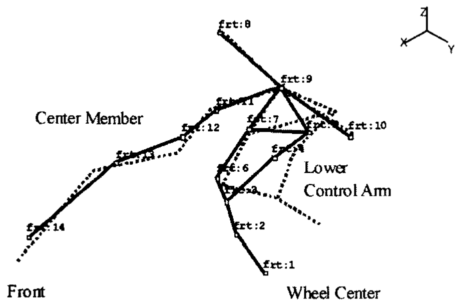
Figure 13. Noise sensitivities of 16 transfer paths to driver's ear position in 35~40 Hz frequency range.

3.7. Test for Operational Deflection Shape

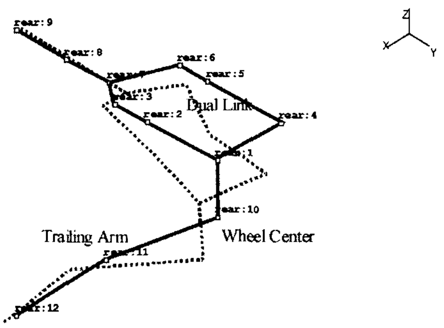
In order to know what kind of phenomenon takes place in chassis systems during operation, test for measurement of operation deflection shapes was carried out. Because the front and rear chassis systems are symmetry, the operating acceleration signals were measured at the half of the systems. The operational deflection shapes of the chassis systems at the several main frequencies are summarized in Table 5. Figure 14 shows the deformed shapes of a front chassis at 44.0 Hz and rear chassis at 35.5 Hz. Comparing these results with those of suspension modal test, the frequencies and their corresponding deformed shapes coincide with the natural frequencies and mode shapes. These results reveal that behaviors of the chassis systems are caused by the resonance of suspension systems. Therefore the peaks that are found in baseline

Table 5. Operational deflection shapes of the chassis systems at several main frequencies.

Chassis system	No.	Frequency [Hz]	Deflection shape
Front	1	36	Fore/aft of lower control arm
	2	44.0	Fore/aft lower control arm
	3	87.5	Fore/aft of lower contrl arm Bouncing
Rear	1	35.5	Fore/aft of dual link Bouncing
	2	43.0	Fore/aft of dual link
	3	92.5	Bending of trailing arm



(a) Front chassis at 44.0 Hz



(b) Rear chassis at 35.5 Hz

Figure 14. Operational deflection shapes of the chassis systems. ——, undeformed shape; ⋯⋯, deformed shape.

test are caused by the resonance of suspensions, which amplifies exciting forces generated from the contacting surfaces of road and tires.

3.8. Test for Transfer Path Analysis

In order to measure the operational forces using the indirect force determination method based on FRF inversion, the operational vibrations (acceleration) were measured at 17 points on the car body as shown in Figure

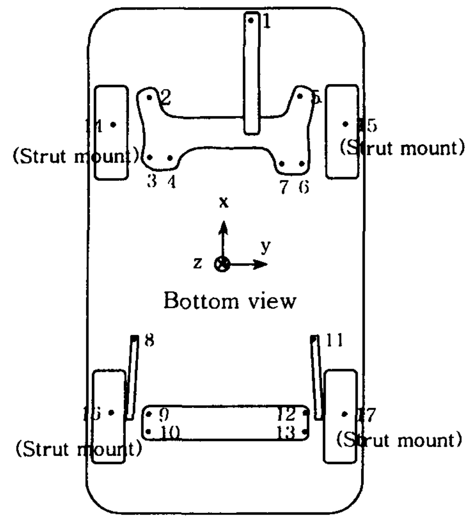


Figure 15. Measurement positions on the vehicle body for transfer path analysis.

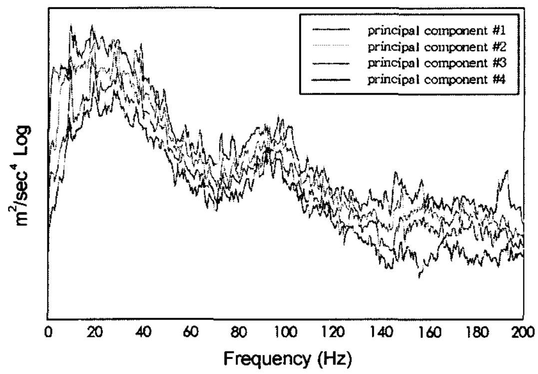


Figure 16. Principal components of four reference signals.

15. To prevent any disturbance, the measurements were carried out at a low constant speed with 60 km/h on asphalt road. The measurements were split up in runs of 17 because of the channel limitation of a digital recorder. Also, only four additional vibration signals, as reference signals, were measured at the center of four wheels in vertical direction during all runs for the same reason. Since the signals of four wheels are non-coherent it is obvious that vibrations of the 17 points are also non-coherent. Therefore, the Principal Component Analysis should be carried out to separate the operational data into sets of "independent phenomena". Because the importance of the four principal components is about the same as shown in Figure 16, all principal components were used to separate the operational vibrations into four sets of the referenced virtual spectra. Then the four sets of referenced operational forces were calculated by multiplying an inverted matrix of flexibility matrix that was measured

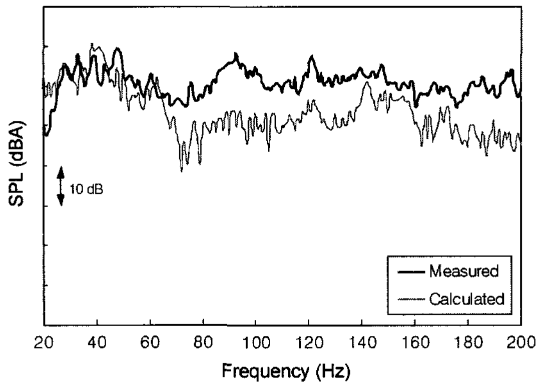


Figure 17. Comparison of the overall sound pressure levels measured at front seat.

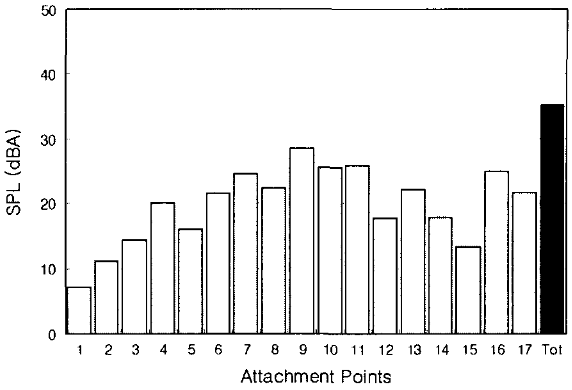


Figure 18. Path contributions to interior sound pressure level at front seat (48.5 Hz).

through the tests for measurement of point inertances together the four sets of referenced operational accelerations. Finally, the four sets of pressures were obtained by means of multiplication of four sets of referenced operational forces with a vibro-acoustic transfer function matrix. When the vibro-acoustic transfer functions of points being measured, the cross coupling between different points were neglected, but the cross coupling between different directions of an identical point were considered. The total sound pressure was then calculated by summation of the four sets of pressure in a root mean square way, since the four signals were independent phenomena.

Comparisons of the calculated and measured interior noise are shown in Figure 17. The result shows a good agreement between the estimation from the Transfer Path Analysis and the direct measurements in 20~70 Hz frequency range. Figure 18 represents the contributions of the noise transfer path to interior noise at front seat in the frequency ranges of interest. By examples, the trailing link mounting points (point 8 and 11), rear sub-frame mounting points (point 10 and 12), and rear strut mounting point (point 16) are the major contribution paths for interior noise around 37.5 Hz of frequency range. The trailing link mounting point (point 11), rear sub-frame mounting points (point 9 and 10) and rear strut mounting point (point 16) are the major contribution paths for interior noise around 48.5 Hz of frequency range, too.

3.9. Summary

Table 6. Estimation of the major paths for road-induced noise by means of each test.

TEST			INDEX	RESULT		
Baseline			Freq. of peaks	35~40	45~50	87~90
Modal	Cavity		Natural freq.	-	50.4	90.0
	Body			38.8	48.7	-
	Susp.	Front		36.8	46.7	91.1
		Rear		35.1	45.0	-
ODS	Chassis	Front	Resonance freq.	36.0	44.0	87.5
		Rear		35.5	43.0	-
IPI			Stiffness	Rear S/M		
NTF			Sensitivity	T/L	Rear S/M T/L Rear S/F	T/L Rear S/M
TPA			Main path	T/L Rear S/M Rear S/F	Rear S/M T/L Rear S/F	Rear S/M

T/L: trailing link, S/M: strut mount, S/F: subframe.

The results of all tests are summarized in Table 6. The major contribution paths investigated by the Transfer Path Analysis coincide with the results of test for measurement of input point inertance and/or noise transfer functions. There are, moreover, good correlations between the suspension modes and the operational deflection shapes of chassis systems since the behaviors of chassis systems are amplified by the resonance of suspension systems. When the local stiffness of a mounting area of the resonating chassis system on the body is weak, this area can be a major contribution path for the road-induced noise.

4. CONCLUSIONS

All tests were carried out according to the test procedure for road-induced noise. Because they have the correlations mutually, it is not necessary to perform the all tests in order to investigate the major contribution paths. For simplification or a quick check of the noise transfer paths, the laboratory tests such as measurements of IPI and NTF can be replaced with the field test such as TPA test. Similarly, for the behavior of suspension systems, the laboratory tests such as modal tests can be used to substitute for the field test such as ODS test.

From results of the tests, several conclusions can be presented as follows:

(1) To examine noise transfer paths, baseline test, TPA test, and ODS test should be carried out under the driving condition;

(2) To obtain information about modification of a body, baseline test, test for measurement of IPI, and test for measurement of NTF should be carried out;

(3) To obtain information about resonance of suspension systems, baseline test and modal test should be carried out.

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