Journal of KSNVE, Vol. 8, No. 6, pp. 1158~1165, 1998.

〈Original Paper〉

Tools to Understand Interior Noise due to Road Excitation in Cars

노면 가진에 의한 실내 소음 해석 방법

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(Received May 14, 1998 : Accepted September 9, 1998)

Key Words : Transfer Path Analysis(전달 경로 해석), Road Noise(노면 가진 소음), Suspension System(현가 장치)

ABSTRACT

Low frequency interior noise in cars is mainly due to structure-borne excitations which are related with road excitation and component vibrations such as suspension and engine mounts. In order to analyze the annoying interior noise, a technique (Transfer Path Analysis) is introduced to find a noise source and the path of that noise. In this study, TPA is reviewed theoretically and applied to investigate the case when the low frequency interior noise at front seat due to road excitations needs to be optimized. The subjective and objective appraisal was performed under the conditions that a testing vehicle traveled on asphalt at 30 km/h, so that the low frequency to be eliminated was detected. The related vibration and noise data for TPA were measured on running and static vehicle. The results reveal that the noise contribution along the z-direction of trailing arm is prominent to low frequency interior noise.

요 약

실내 저주파 소음은 도로가진, 차량 현가 또는 엔진 마운트와 같은 차량 샤시 부품의 진동인 구조 진동에 기인한다. 실내 저주파 소음을 줄이려는 노력의 일환으로 소음 원의 위치 및 소음 전달 경로를 추적하는 기술인 TPA (Transfer Path Analysis) 를 소개 하였다. 본 연구에서는 TPA 기술을 이론적으로 고찰해보고 이를 실제로 적용하여, 도로가진에 의해 앞 좌석에서의 실내 소음에 기여도가 가장 큰 샤시 부품을 찾는 Case Study를 소개하였다. TPA를 적용하기 위한 데이터를 얻기 위해, 도로가진에서 소음으로 연결되는 계측과 실차 운행 상태의 진동 데이터 계측을 실시하였다. 또한 아스팔트 도로에서 시속 30 km/h로 달리면서 관능 평가와 진동 소음 계측 평가 모두를 실시하여 문제가 되는 저주파 실내 소음을 확인하였다. 시험 분석 결과 Trailing Arm을 통해 전달되는 소음이 앞 좌석 저주파 실내 소음에 대한 기여도가 가장 크게 나타났다.

1. Introduction

An interior noise in cars due to road excitation exists in the low frequency range, less than 500 Hz. Such type of noise could be seriously related with driver alertness, communication, comfort, etc. In case when the low frequency interior

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noise under running conditions bothers ride feelings of driver or passengers, it may lead to customer's complain in spite of good overall ratings in dynamic vehicle performances. Therefore, the noise level inside vehicles in most automobile manufactures is still an important subject to be optimized in level since the general levels of vehicle noise have been significantly improved along with developments in vehicle conponents and main body properties[1,3,~6].

In general, there are obstacles in the way of optimizing the interior noise level. First of all, it is necessary to investigate partial noise contributions with respect to the overall interior noise before achieving optimization, but information on partial contributions from direct experimental measurements is not generally available. Moreover, reaching a good result of optimization for low frequency interior noise often demands a compromise in vehicle properties such as ride & handling, comfort, costs, etc.

Recently, TPA (transfer path analysis) has been introduced as a good tool to address the interior noise problem by establishing individual noise source strengths and a partial noise contribution of each of these sources to the overall noise level.

In this study, the TPA will be reviewed theoretically and applied to the case study when the low frequency noise at the front seat due to road excitations is annoying to driver's ears.

2. Mathematical Background

TPA is a method which was developed in automotive industries to understand the contribution of the individual powertrain and drive line mounts to the interior noise[1]. In simple form, TPA is composed of two types of data[2~6] :

(1) Estimation of the operational forces at the attachments.

(2) Calculation by combining the operational forces with mechanical-acoustic transfer functions which convert the forces to sound pressure at the driver's ears.

2.1 Interior Noise

The total interior noise can be described as a linear summation of partial contributions:

$$P = P_1 + P_2 + \ldots\ldots = \sum_{k=0}^{N} P_k \tag{1}$$

where P is total interior noise at driver's ear location and P_k is the k-th partial acoustic pressure contributor to the interior noise.

Each contributor is a product of acting force at the corresponding path and the vibro-acoustical system FRF (Frequency Response Function) matrix, and described as following :

$$p_k = \sum_{i=1}^{M} \sum_{j=1}^{3} H_{ij} f_{ij} \tag{2}$$

where H_{ij} is a mechanical-acoustical transfer function of the i-th attachment in j direction, and f_{ij} is an operating force at the i-th attachment in j direction.

2.2 Force Estimation

There are two ways of estimating operational forces at attachments or mounts. The best and sure way could be to measure forces from fields directly. However, it is practically very hard to obtain field data by using force sensors and therefore, it is recommended to use dynamic stiffness of each mount for estimating forces with measured relative acceleration data under operation conditions.

$$f_{ij} = \frac{1}{\omega^2} K(\overline{\omega}) \{ (\ddot{X}_a)_{ij} - (\ddot{X}_b)_{ij} \} \tag{3}$$

where f_{ij} is the estimated force on the body side of the i-th mount in j direction, $(\ddot{X}_a)_{ij}$ is the operational acceleration on excited side of the i-th mount in j direction, and $(\ddot{X}_b)_{ij}$ is the operational acceleration on body side of the i-th mount in j direction. $K(\omega)$ is the measured complex dynamic stiffness of the corresponding mounting element as a function of frequency.

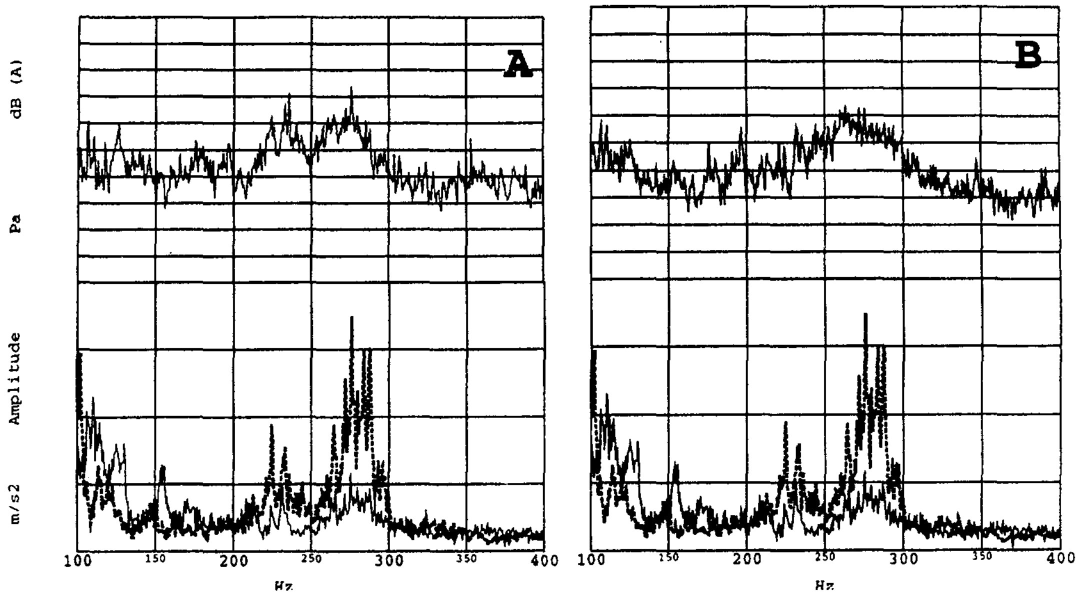

Fig. 1 Road noise

A. Front interior noise(upper) vs wheel vibrations (lower)
B. Rear interior noise(upper) vs wheel vibrations (lower)
Road noise was tested at 30 km/h on asphalt road
Noise level in each grid is 5 dB(A)
Solid line is for front wheel vibration and dotted line is for rear wheel vibration

Second way to estimate operational forces at bushes during running is using transfer function matrix between accelerations and forces at bushes in all directions. It is usually measured from hammer impact testing in the lab.

The formula is following:

$$f_{ij}=[H]^{-1}_{FRF}\{\ddot{X}_b\}_{ij} \tag{4}$$

where $[H]_{FRF}$ is a transfer function matrix from impact testing, f_{ij} is the operational force on the i-th bush in j direction, and $(\ddot{X}_b)_{ij}$ is the acceleration on the i-th bush of body side in j direction.

Although this method could be easier in measurements, it is essential to have an accurate inversion for a better estimation, and it is based on singular value decomposition algorithms in general.

2.3 Mechanical-Acoustical Transfer Function

To obtain the second type of data for the TPA, it is required to measure and find a relation between forces and interior sound pressure which is a mechanical-acoustical transfer function:

$$[H_{MA}]_{ij}=\frac{p_{ij}}{f_{ij}} \tag{5}$$

where $[H_{MA}]_{ij}$ is the same transfer function matrix as described in equation (2). f_{ij} is the force acting on the i-th bush of body side in j direction and p_{ij} is the generated sound pressure in the cabin due to force acting on the i-th bush of body side in j direction.

In general, this is measured in such a way that the body side is not connected to bushes of any of excited side during transfer function measured. The microphones for measuring interior sound pressure are placed at the occupant's ear locations.

3. Case Study

The structure borne noise contributions in a passenger car with 30 km/h speed were investigated to understand the common interior noise problem. After a subjective appraisal on a vehicle with speed of 30 km/h on a smooth road, objective measurements according to the test procedures ES-TY502 and ES-TY506 were made for analysis. Every test was repeated at least three times to check whether the measurements show representative behavior of vehicle.

3.1 Road Noise

First, road noise was subjectively inspected by experienced engineers. The noise in 200~300 Hz was pointed as a target to be removed. Initial tests for road noise measurements were performed to see how interior noise is influenced either by road excitations of front wheel or by those of rear wheel. The performed test conditions were as close to the subjective appraisal as possible.

As the front seat was not comfortable due to road noise for drivers in subjective evaluation, the measurements on front seat were more focused in analysis. In Fig. 1-A, the rear wheel vibration peaks in 200~300 Hz due to road excitation are more closely related with the front seat interior noise frequencies than the front wheel vibration. The sound quality replay proved that the noise in these frequencies was same as an annoying noise pointed by the experts[7]. On the other hand, since the noise in rear seat was evaluated as an acceptable level in subjective appraisal, the detailed investigation was not performed although there were some peaks appeared at rear seat as shown in Fig. 1-B.

3.2 TPA

Since road excitations through rear wheel were closely related with structure borne interior noise, noise contributions to interior noise (200~300 Hz) were investigated mainly on three possible transfer paths : along rear strut (path 1), trailing arm (path 2), and torsional beam (path 3). Figure 2 shows how the rear suspension system is connected to the frame of the tested

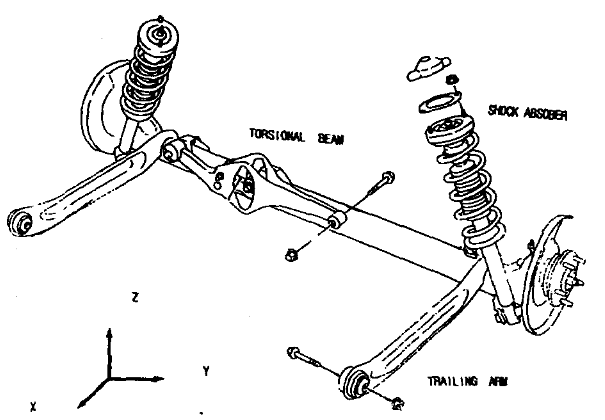

Fig. 2 Rear suspension system

vehicle.

The required measurements for the analysis are composed of three steps. First of all, operational accelerations were measured at three directions of each noise path under operating conditions. Each test was also repeated three times to confirm a repeatability of measured data. Secondly, mechanical-acoustic transfer functions were measured along the three paths on vehicle body sides after separating unsprung mass parts from the test vehicle. Finally, the dynamic stiffness of each connecting mount along the path was estimated. In general, as far as the accuracy is concerned, the dynamic stiffness of each mount should be measured by using a rubber testing machine. Unfortunately, since the testing machine was not available during our tests, all values of dynamic stiffness at each mount were estimated by manufacturing design criterion and educated guesses from experienced engineers. Thus, the results of TPA for this work must be explained and understood along the limitations that the exact nonlinear behavior of rubber mount was not included.

The transfer path analysis was performed by utilizing the TPA Module of LMS CADA-X 3.4.06.

3.3 Results

Figure 3 shows that the front seat noise was not transferred through the 3rd path in all direction since no contribution is estimated. However, Fig. 4 shows that z-direction in the first path is important since the large amount of contribution to front seat noise is observed, but the range of noise frequency is in 300~400 Hz. In Fig. 4, other directions in the first path are

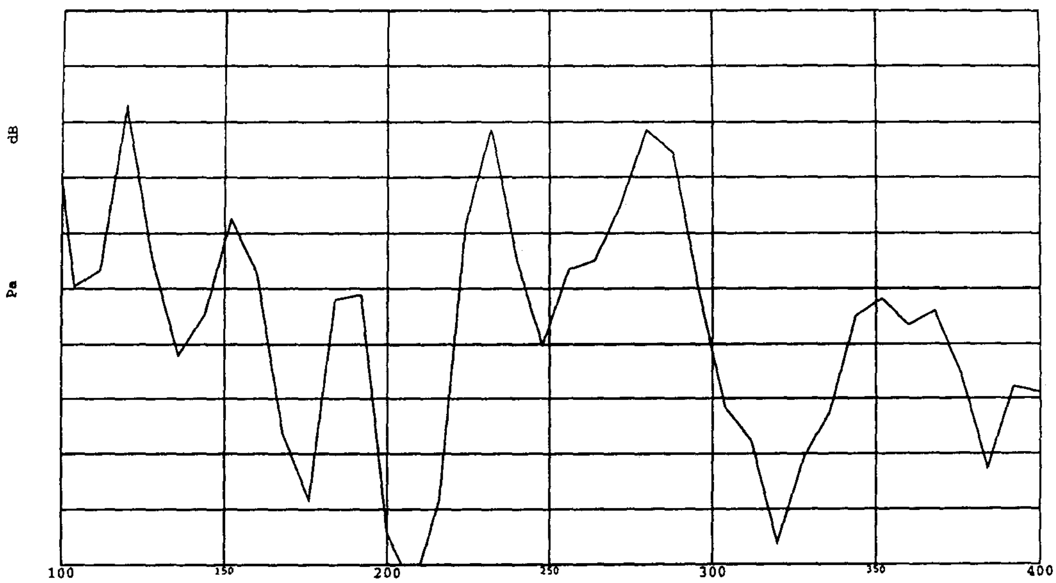

Fig. 3 Contribution of the third path for front seat

——— Calculated total
No contribution from x,y,z directions Noise level in each grid is 5 dB(A)

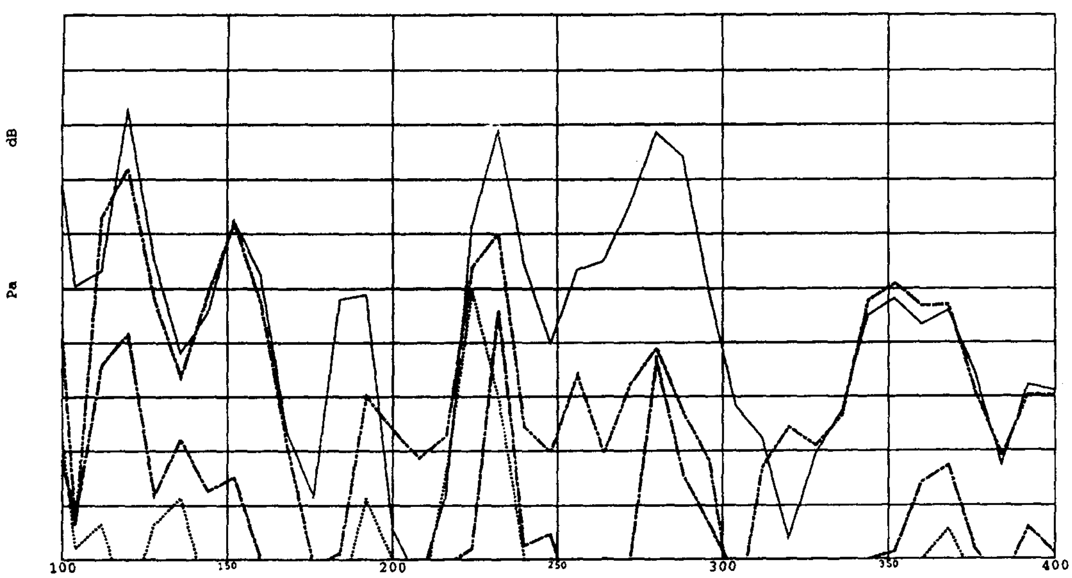

Fig. 4 Contribution of the first path for front seat

——— Calculated total : ··········· Contribution from x direction
—·—·—· Contribution from y direction : —··—··—·· Contribution from z direction
Nose level in each grid is 5 dB(A)

not closely related to front seat noise in the cabin. In a similar way, z direction in the second path is also important since the contribution to overall front seat noise, at 200~300 Hz, is prominent in Fig. 5. Again, as observed in the first path analysis, contributions to overall noise in other directions along the second path are not as important as that in z direction.

Therefore, interior noise in the range of 200~400 Hz is related to only z direction in both of the first and second path.

3.4 Discussion

From both subjective and objective road noise tests, it was noticed that the interior noise in the range of 200~300 Hz bothered driver's ears. Therefore, front seat noise contributions in that range were analyzed intensively in TPA.

Based on the geometry shown in Fig. 2, noise paths are assumed to be x, y and z direction through rear hub from road excitations. x direction (path 2) is trailing arm, y direction (path 3) is torsional beam, and z direction (path 1) is shock absorber.

However, the estimated TPA results on around 100 Hz are quite different from measured values. This could be because of not perfectly estimated characteristics of the rubber mount.

As expected, z direction along the first path was dominantly contributed in interior noise. It was true for front seat position measurements. However, the interior noise bothering passengers in cars is only in the range 200~300 Hz whereas the contributing range to the interior noise along shock absorber is around 300~400 Hz. Thus, shock absorber in this study is not closely related with low frequency interior noise which was pointed out from the subjective evaluation. In other words, shock absorber of this tested car might work properly to any transmitted vibrations due to road excitations. If there were any complains related with 300~400 Hz noise inside, then that should be because of rear suspension components.

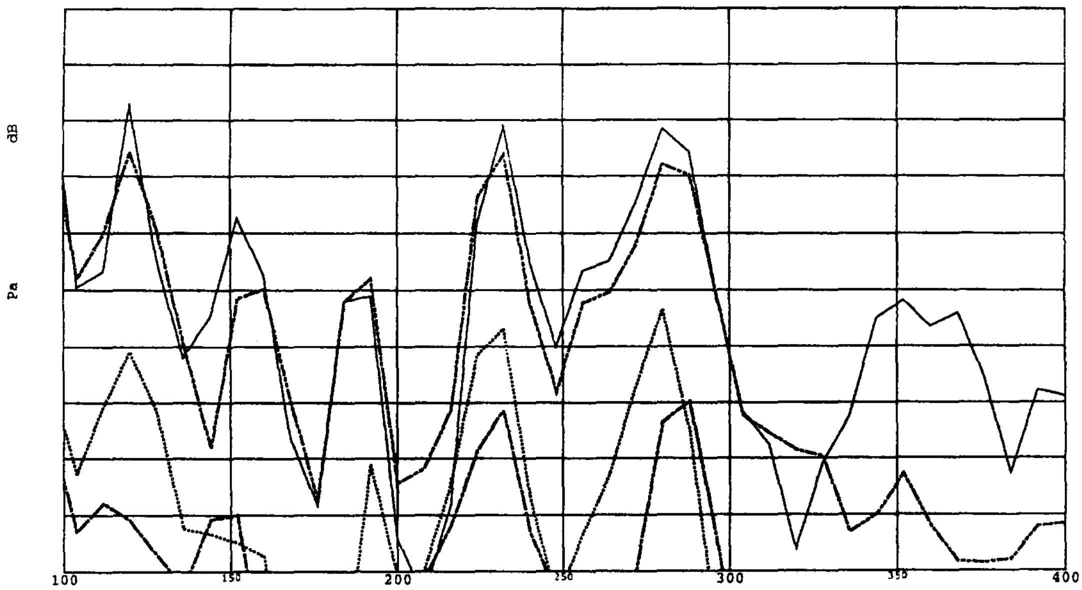

Fig. 5 Contribution of the second path for front seat

———— Calculated total : ·········· Contribution from x direction
—·—·— Contribution from y direction : —··—··— Contribution from z direction
Nose level in each grid is 5 dB(A)

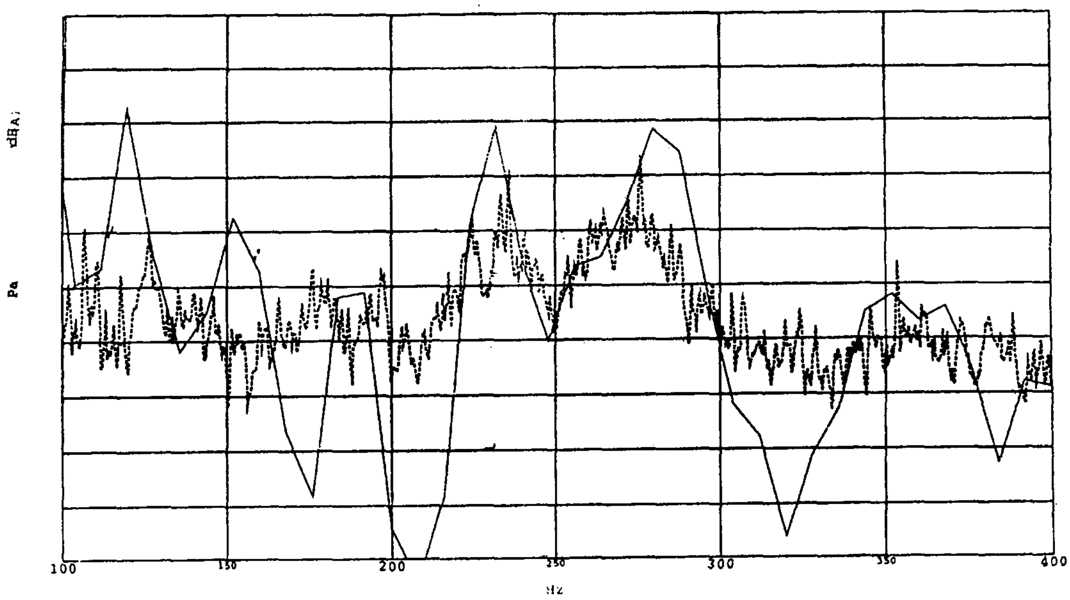

Fig. 6 Estimated vs Measured of front interior noise

———— Calculated total : ………… Measured noise
Nose level in each grid is 5 dB(A)

In this study case, trailing arm is a problem chassis component for low frequency interior noise since TPA shows a z direction of trailing arm is the main path for transferred noise. Specially, all the estimated noise around 200 ~300 Hz is contributed from the trailing arm. Therefore, there are several check points to solve the current situation. Fundamental data such as natural frequencies of components along the second path should be investigated. Also, as a possible and practical solution, any bushing in trailing arm should be re-examined for right performance.

4. Conclusion

TPA was introduced to understand theoretically and quantify the partial contribution of the transferred noise to the specific interior noise due to road excitations.

Front seat interior noise around 200~300 Hz in the test vehicle was noticed in subjective and objective studies. Interestingly, it was closely related with road excitations, especially at rear. The TPA results showed that z direction of a trailing arm was the main noise transfer path of front seat interior noise due to road excitations. On the other hand, z direction in rear strut component showed little relation with low frequency interior noise in the tested vehicle. Torsional beam was not sensitive to low frequency interior noise in this case study.

Acknowledgment

The authors want to express their gratitude to SEMCO (Samsung Electro-Mechanics Co) for this project and Ricardo engineers, Mr. Suggitt and Mr. Bruce for assisting measurements and analysis.

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