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차량의 아이들시 실내소음 개선을 위한 소음경로 해석이론의 적용에 관한 연구

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Study on Application of Noise Path Analysis for Improving Interior Noise at the Idle of a Passenger Vehicle

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ABSTRACT

아이들 상태에서 차실 내에 불쾌감을 주는 매우 높은 크기의 소음이 발생하였다. 본 논문에서는 아이들시 문제소음의 원인을 분석하고 저감하는데 필요한 관련이론의 적용과 이에 수반되는 데이터측정, 데이터처리및 데이터 해석과정을 자세하게 기술하였다. 소음진동 측정 및 분석 결과 문제의 소음은 엔진의 회전 2.5차 성분을 가지는 구조기인소음이었다. 아이들시 문제의 실내소음에 영향을 미치는 구조전달경로들의 기여도를 분석하기 위해 소음경로해석시험 및 모드시험을 실시하였다. 시험분석 결과에 따라 기여도가 높은 배기계 및 트랜스미션계의 진동 전달특성을 개선하므로써 만족할 수준의 소음 저감효과를 얻었다.

Key Words: Interior noise(실내소음), Idle(공전), Structureborne noise(구조기인소음), Airborne noise(공기전파음), Noise path analysis(소음경로해석), Modal test(모드시험), Mount(마운트)

1. Introduction

Interior boom noise at idle condition was perceived subjectively at the later developing stage of a passenger vehicle. Since the interior noise quality at the idle of a vehicle is one of a customer's first impressions which influence the customer's purchase of a vehicle, reduction of the interior boom noise at the idle is required. The objective of this paper is to describe a logical and practical approach for improving interior boom at the idle of a passenger vehicle.

The feature of the signal which generated the poor noise quality was defined by subjective assessment and signal processing of the data measured by a binaural measurement system. The boom noise turned out to have the second and a half order component (C2.5) of the engine revolution.

The airborne noise contributions to the interior boom noise of the intake and exhaust systems were investigated by fitting large auxiliary mufflers to the existing intake and exhaust systems.

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Noise path analysis (NPA)[1-4] was used to identify and rank in order of importance structureborne noise paths potentially contributing to the idle boom noise of the vehicle. The analysis included the vertical, lateral and longitudinal forces of engine left/right mounts, a transmission mount, and three exhaust mounts as structureborne noise paths. Countermeasures based on the noise path analysis results significantly reduced the boom noise at the idle of the developing vehicle.

Theory of noise path analysis has been presented in a mathematical form convenient especially for practical engineering use. Data measurement, signal processing procedure and data interpretation have been described in details in connection with application of the noise path analysis.

2. Theory

Noise path analysis (NPA) has been widely used in automotive industries to investigate both structureborne and airborne noise paths contributing to the interior noise of a vehicle (e.g., see Refs. 3-5). The noise path analysis was used in this paper for addressing the interior boom noise at the idle of a vehicle under development.

In this section, the theory of the noise path analysis is presented for the specific angular frequency ω in a form convenient especially for practical engineering application. Since the contributions by airborne noise paths turned out to be negligible for the vehicle under current investigation (see Section 3 below), only structureborne noise paths are included in this theoretical presentation. More complete description of the theory of the NPA can be found in other papers[1-4]. Unless otherwise mentioned, each notation in this theory section denotes a complex number which has magnitude and phase.

An operational relative displacement [$\Delta X_j(\omega)$] across the jth individual path at a specific driving condition is obtained by double-integrating the measured relative accelerations across the jth path:

$$\begin{aligned}\Delta X_j(\omega) &= -\frac{1}{\omega^2}\left[\ddot{X}_b(\omega) - \ddot{X}_c(\omega)\right]_j \\ &= -\frac{1}{\omega^2}\Delta\ddot{X}_j(\omega) \qquad (1)\end{aligned}$$

where $\ddot{X}_b(\omega)$ and $\ddot{X}_c(\omega)$ are the measured accelerations on the body and chassis sides of the jth path, respectively.

The operational force [$F_j(\omega)$] acting on the body side of jth path is obtained by multiplying the relative displacements across the jth path by the measured dynamic stiffness (K_j) of the elastomeric mount corresponding to the jth path:

$$\begin{aligned}F_j(\omega) &= K_j(\omega)\Delta X_j(\omega) \\ &= -\frac{1}{\omega^2}K_j(\omega)\Delta\ddot{X}_j(\omega) \qquad (2)\end{aligned}$$

Sound pressure [P_{kj}] at the k interior location in the vehicle induced by $F_j(\omega)$ through the jth structural path is computed by multiplying the operating force $F_j(\omega)$ by the measured mechanical-acoustic transfer function [$H^a_{kj}(\omega)$]:

$$\begin{aligned}P_{kj}(\omega) &= H^a_{kj}(\omega)F_j(\omega) \\ &= -\frac{1}{\omega^2}H^a_{kj}(\omega)K_j(\omega)\Delta\ddot{X}_j(\omega) \qquad (3)\end{aligned}$$

Here, the mechanical-acoustic transfer function of a vehicle body [$H^a_{kj}(\omega)$] is the sound pressure measured at k interior location induced by unit force applied to the body side of the jth path.

Only magnitude of Eq. (3) is needed to rank the contribution of each structureborne noise path:

$$\begin{aligned}|P_{kj}(\omega)| &= |H^a_{kj}(\omega)||F_j(\omega)| \\ &= \frac{1}{\omega^2}|H^a_{kj}(\omega)||K_j(\omega)||\Delta\ddot{X}_j(\omega)| \qquad (4)\end{aligned}$$

Total sound pressure [$P_k(\omega)$]at the k interior location induced by all structural noise paths is calculated by summing up the contribution of each structureborne noise path:

$$P_k(\omega) = \sum_{j=1}^{N} P_{kj}(\omega) \qquad (5)$$

where index N is the number of all structureborne noise paths.

3. Experiment

A noise quality problem at the idle condition was perceived subjectively as a boom noise at the later developing stage of a passenger vehicle with a five cylinder engine. To define the noise quality problem, measurement with a binaural heading system was made in the front driver seat and at the rear left side seat with idle sweep in a semi-anechoic chamber.

With signal processing of the recorded signature in the time and frequency domains[6], the feature of the signal which produced the boom noise was identified. The boom noise had the second and a half order component (C2.5) of the engine revolution which is induced by the torque variation of engine firing[7,8,9].

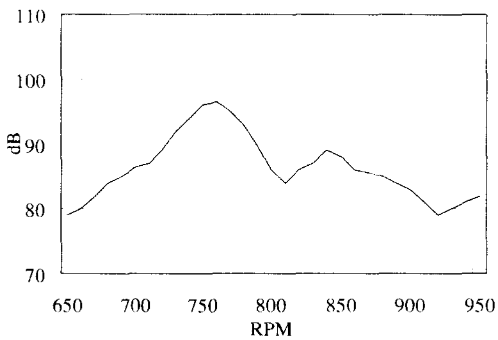


Fig. 1 Sound pressure level (*dB*) of C2.5 component of the baseline vehicle at a driver's left ear

Figure 1 shows the sound pressure level (*dB*) of the C2.5 component at the driver left ear position with idle sweep. The component has sound pressure level of 96 *dB* and frequency of 31.7Hz at 760rpm. All noise measurement data in this paper are unweighted to prevent the distortion of the harmonic patterns in signal processing.

The airborne noise contributions of the intake and exhaust systems to the interior boom of our concern were investigated by fitting large auxiliary mufflers to the intake and exhaust systems. The large auxiliary mufflers have insertion losses of greater than 20 *dB* and are sufficient to effectively eliminate the intake and exhaust orifice airborne noise. Measurements with both the auxiliary intake and exhaust mufflers made little change in the boom noise level at both the front and rear seat microphone positions. This experimental results indicate that the airborne noise contribution of intake and exhaust systems to the idle boom is negligible.

Propeller shaft was then disconnected from other powertrain parts. Disconnecting the propeller shaft had no effect on the idle C2.5 boom noise level in both front and rear seat microphone positions, which indicated that suspension system was also no contributor to the boom noise.

Noise path analysis (NPA)[1,4] was carried out in a semi-anechoic chamber to identify the structureborne noise paths contributing to the C2.5 idle boom. Since the test results showed the similar tendency regardless of seat positions, the data only for the drive left ear position are presented hereafter.

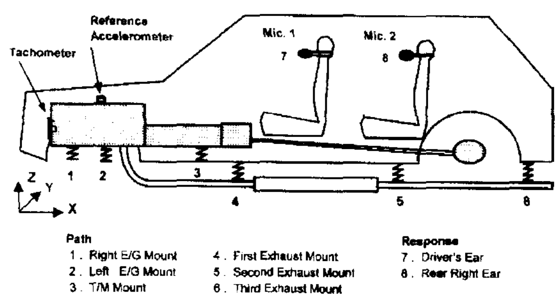


Fig. 2 Measurement setup and measuring positions

Figure 2 shows the measurement setup and the measuring positions. While a tachometer and a reference accelerometer measured the engine revolution and the vertical acceleration of the upper center on the engine block (a reference point), respectively, accelerometers measured the accelerations across each path of the elastomeric engine left/right mounts, transmission mount and three exhaust mounts with idle sweep. All measurements were taken with respect to the same reference point (i.e., the upper center on the engine block). Relative displacement (ΔX) through each path was calculated by double-integrating the C2.5 component of the measured accelerations, according to Eq. (1).

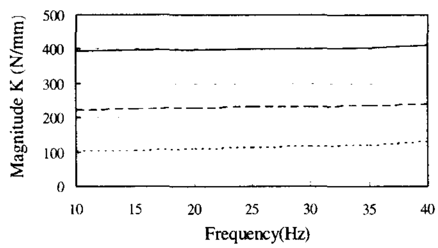
Table 1 Noise path analysis results for the baseline vehicle at 760rpm [31.7Hz (C2.5)]

Path	Direction	Relative Displacement	Dynamic Stiffness	Force	Transfer Function		Sound Pressure	
		$\lvert\Delta X\rvert$ (mm)	$\lvert K\rvert$ (N/mm)	$\lvert F\rvert$ (N)	$\lvert H^*\rvert$ (Pa/N)	dB/N	$\lvert P\rvert$ (Pa)	dB
Right E/G Mount	X	0.056	402	22.51	0.0112	55.0	0.252	82.0
	Y	0.083	121	10.04	0.0071	51.0	0.071	71.0
	Z	0.186	237	44.08	0.0063	50.0	0.278	82.9
Left E/G Mount	X	0.053	405	21.47	0.0122	55.7	0.262	82.3
	Y	0.063	121	7.62	0.0112	55.0	0.085	72.6
	Z	0.201	236	47.44	0.0073	51.2	0.346	84.8
T/M Mount	X	0.061	839	51.18	0.0042	46.4	0.215	80.6
	Y	0.145	255	36.98	0.0139	56.8	0.514	88.2
	Z	0.092	198	18.22	0.0105	54.4	0.191	79.6
1st ExhaustMount	X	0.232	21	4.87	0.0057	49.1	0.028	62.9
	Y	0.732	22	16.10	0.0048	47.6	0.077	71.7
	Z	1.304	42	54.77	0.0153	57.7	0.838	92.4
2nd Exhaust Mount	X	0.059	25	1.48	0.0124	55.8	0.018	59.2
	Y	0.438	28	12.26	0.0061	49.7	0.075	71.5
	Z	0.625	58	36.25	0.0018	39.1	0.065	70.3
3rd Exhaust Mount	X	0.367	25	9.18	0.0022	40.8	0.020	60.1
	Y	0.631	28	17.67	0.0015	37.5	0.027	62.4
	Z	0.815	56	45.64	0.0165	58.3	0.753	91.5

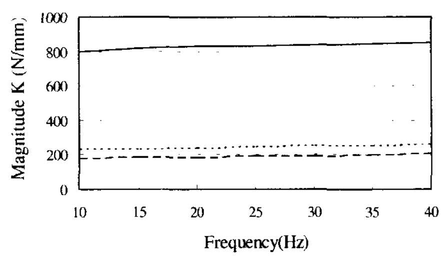
Included in the third column of Table 1 are the magnitudes of the calculated relative displacements of the C2.5 component at the idle of 760rpm (i.e., 31.7Hz) in the vertical, transverse, and longitudinal directions of each mount.

The elastomeric mounts were removed from the vehicle, and the dynamic stiffness (K) of each mount was measured over the frequency range up to 40Hz by using a hydraulic mount test machine. The measured mount stiffness is expressed as a function of frequency in units of Newton per millimeter of displacement.

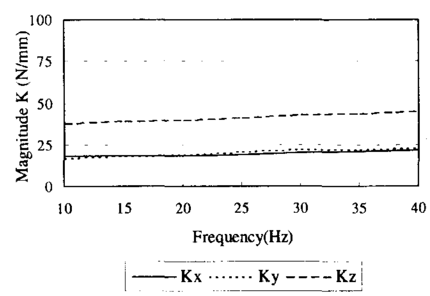
Figure 3 shows the magnitudes of dynamic stiffness in three directions over frequencies up to 40Hz of the engine left mount, the transmission mount, and the first exhaust mount, respectively. The magnitudes change very slightly over the frequency range. The magnitudes of the dynamic stiffness at 31.7Hz of each mount are included in the fourth column of Table 1.



(a) Dynamic stiffness of the engine left mount



(b) Dynamic stiffness of the T/M mount



(c) Dynamic stiffness of the 1st exhaust mount

Fig. 3 Dynamic stiffness-frequency curves for elastomeric mounts

The powertrain-generating forces (F) acting on the body were obtained by multiplying the relative C2.5 displacement (ΔX) by the dynamic stiffness (K) of the individual mount corresponding to

31.7Hz [see Eq. (2)]. The magnitudes of the calculated operational forces are shown in the fifth column of Table 1.

In the next step, the powertrain and suspension system were separated from the body. Artificial force by an impact hammer was applied to the body side of the each mount location in each direction, and the sound pressure at the driver left ear location induced by the artificial force was measured with a microphone. Mechanical-acoustic transfer function (H^a)[10] on the vehicle body was calculated by dividing the measured sound pressure by the applied artificial force for the frequency of interest. The sixth and seventh columns of Table 1 show the magnitude of the body transfer function at 31.7Hz through each path in units of Pa/N and dB/N , respectively.

Finally, structureborne noise path contributions to the noise of our concern were then calculated by multiplying these body acoustic transfer functions (H^a) by the operational forces (F) acting on the body for the specific problem frequency of 31.7Hz, according to Eq. (3). The last two columns of Table 1 present the magnitude of the structureborne noise contribution of each path for 31.7Hz in units of Pa and dB , respectively [see Eq. (4)].

4. Results

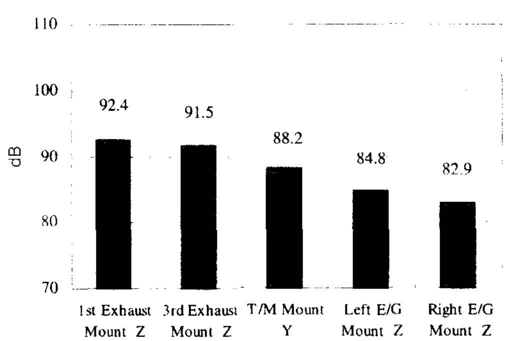


Fig. 4 Ranking of the contribution of each structure-borne noise path

Figure 4 ranks in order of importance the contribution of an individual structureborne noise path to the idle boom from the last column of Table 1. As seen from Fig. 4, the vertical paths of the first and third exhaust mounts are the first and second main dominating factors, respectively, and the transmission lateral mount path is the third main contributor to the idle boom.

The sixth column of Table 1 shows that the body mechanical-acoustic transfer function at 31.7Hz through each path is less than 60 dB/N . Since body transfer function in the order of 50 dB/N to 60 dB/N has been considered good in the vehicle design in the automotive industries, the body transfer function of our vehicle seems to be within reasonable levels.

The fourth column of Table 1 shows that the vertical stiffnesses of the first and third exhaust mounts are normal compared with those of other paths. However, as shown in the third column of Table 1, the vertical relative displacements of the first and third exhaust mounts are very large compared with those of other paths. These large vertical displacements turned out to be induced by the bending resonance of the exhaust system.

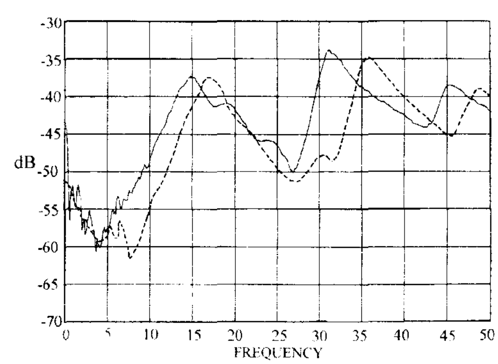


Fig. 5 Comparison of the FRF of exhaust systems before (—) and after (---) modification

The solid line of Fig. 5 shows the frequency response function (FRF) of the baseline (original) exhaust system obtained by modal tests[11] and indicates that the elastic bending mode of the exhaust system exists around 32Hz. The high contributions of the vertical paths of the first and third exhaust mounts thus appear to result from the bending resonance of the exhaust system at 32Hz.

The high contribution of the lateral transmission path seems to be due to both relatively large displacement and stiffness in the lateral direction of the transmission mount (in comparison with those of other lateral paths) as seen in the third and fourth columns of Table 1.

The boom noise can be reduced by changing the vertical bending resonance frequency of the exhaust system. The bending frequency can be changed by either decreasing or increasing the vertical stiffness of the exhaust mounts. When the vertical stiffness of the first exhaust mount was decreased, the first exhaust mount failed in durability test. The vertical stiffness of the first exhaust mount was thus increased from 42 N/mm to 85 N/mm instead. With the stiffer first exhaust mount, the bending resonance frequency of the exhaust system increased from 32Hz to 36.3Hz (see the dashed line of Fig. 5 for the FRF of the modified exhaust system with the stiffer first exhaust mount).

Table 2 Noise path analysis results for the vehicle with the modified exhaust system at 760rpm [31.7Hz (C2.5)]

Path	Direction	Relative Displacement	Dynamic Stiffness	Force	Transfer Function		Sound Pressure	
		$\lvert\Delta X\rvert$ (mm)	$\lvert K\rvert$ (N/mm)	$\lvert F\rvert$ (N)	$\lvert H^{*}\rvert$ (Pa/N)	dB/N	$\lvert P\rvert$ (Pa)	dB
1st Exhaust Mount	X	0.172	25	4.30	0.0057	49.1	0.025	61.8
	Y	0.224	35	7.84	0.0048	47.6	0.038	65.5
	Z	0.217	85	18.45	0.0153	57.7	0.282	83.0
2nd Exhaust Mount	X	0.065	25	1.63	0.0124	55.8	0.020	60.1
	Y	0.052	28	1.46	0.0061	49.7	0.009	52.9
	Z	0.512	58	29.70	0.0018	39.1	0.053	68.5
3rd Exhaust Mount	X	0.092	25	2.30	0.0022	40.8	0.005	48.1
	Y	0.188	28	5.26	0.0015	37.5	0.008	51.9
	Z	0.173	56	9.69	0.0165	58.3	0.162	78.1

Table 2 shows the NPA results of the vehicle with the modified exhaust system. As shown in the third column of the Tables 1 and 2, the upward shift of the bending resonance frequency of the modified exhaust system significantly reduced the relative vertical displacement from 1.304 mm to 0.217 mm for first exhaust mount and from 0.815 mm to 0.173 mm for the third exhaust mount.

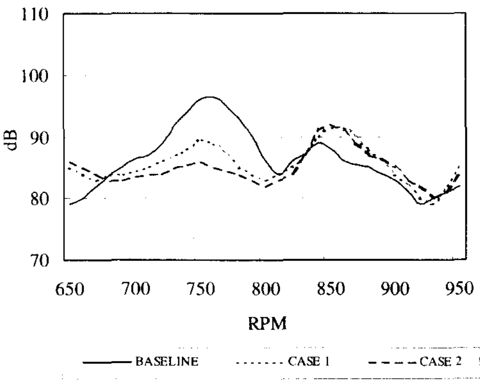


Fig. 6 Comparison of the C2.5 overall sound pressure levels (dB) at a driver's ear position before and after modification

The dotted line (case 1) in Fig. 6 shows the overall C2.5 sound pressure level of the vehicle with the modified exhaust system at a driver's ear position. The modified exhaust system reduces the overall C2.5 boom noise at 760rpm by 7 dB, compared with the baseline exhaust system (the solid line in Fig. 6), but the 7 dB reduction is still insufficient.

To further reduce the boom noise, the lateral stiffness of the transmission mount was decreased from 255 N/mm to 76 N/mm by modifying the transmission mount shape. Table 3 shows the NPA results for the vehicle with the modified transmission mount system. The dashed line (case 2) in Fig. 6 shows the overall C2.5 sound pressure level when both the modified exhaust and transmission systems are installed. The final C2.5 sound pressure level (dashed line of Fig. 6) at 31.7Hz is 85 dB which is 11 dB lower than that of the baseline vehicle (solid line), and the significant peak of the baseline of 31.7Hz at 760rpm is eliminated.

Durability problem and excessive body vibration were not experienced in the driving tests with the low stiffness transmission mount installed on the vehicle.

Table 3 Noise path analysis results for the vehicle with the modified transmission mount system at 760rpm [31.7Hz (C2.5)]

Path	Direction	Relative Displacement	Dynamic Stiffness	Force	Transfer Function		Sound Pressure	
		$\lvert\triangle X\rvert$ (mm)	$\lvert K\rvert$ (N/mm)	$\lvert F\rvert$ (N)	$\lvert H^{*}\rvert$ (Pa/N)	dB/N	$\lvert P\rvert$ (Pa)	dB
Right E/G Mount	X	0.049	402	19.70	0.0112	55.0	0.221	80.9
	Y	0.095	121	11.50	0.0071	51.0	0.082	72.2
	Z	0.167	237	39.58	0.0063	50.0	0.249	81.9
Left E/G Mount	X	0.053	405	21.47	0.0122	55.7	0.262	82.3
	Y	0.088	121	10.65	0.0112	55.0	0.119	75.5
	Z	0.173	236	40.83	0.0073	51.2	0.298	83.5
T/M Mount	X	0.175	85	14.88	0.0042	46.4	0.062	69.9
	Y	0.241	76	18.32	0.0139	56.8	0.255	82.1
	Z	0.126	240	30.24	0.0105	54.4	0.318	84.0

5. Conclusions

The high interior noise at the idle of the developing vehicle was due to structureborne noise paths. Noise path analysis was performed to quantify the contribution of each structureborne noise route to the interior boom noise. The analysis results indicated that the mechanical-acoustic transfer function of the vehicle body were within reasonable levels at the C2.5 idle frequency range. The large vertical displacement of the exhaust system and the high lateral transmission mount stiffness were dominating factors contributing to the idle boom noise. The large displacement of the exhaust system resulted from the bending resonance of the exhaust system. Modifications of the exhaust mount stiffness and transmission mounting shape as countermeasures suggested by the analysis results significantly reduced the idle boom noise of our concern.

Our experimental approach to the NPA application with the hand calculation, instead of commercial NPA package, is helpful for understanding the noise problem in physically clear perspective and also useful for practical engineering applications. This approach can be easily extended to the noise problem at other driving conditions.

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