MADRID
inter.noise 2019
June 16 - 19
NOISE CONTROL FOR A BETTER ENVIRONMENT

Research and application of lumped mass damper on the control of vehicle body panel low frequency vibration and sound radiation

Wan, Yuping[1]
Pang, Jian[2]
Zhang Jie[3]
Yang Liang[4]
Jia Wenyu [5]
State Key Laboratory of Vehicle NVH and Safety Technology, China
Changan Auto Global R&D Center, Changan Automobile Co., Ltd, China
Chongqing, China, 401120

ABSTRACT

This paper introduces the mechanism of vibration transmission and sound radiation of a simplified automobile panel model, reveals the sound and vibration characteristics of the body panel structures, and provides a theoretical support to solve the low frequency panel vibration and booming problems. In this paper, a vehicle liftgate is simplified as a simply-supported rectangular plate. And an analysis model combining the plate and a lumped-mass damper is established. By theoretical analysis and finite element analysis, the influence of the mass damper on the characteristics of the vibration and noise radiation is studied. The paper also analyzes the mechanism of the low frequency interior booming, and provides a control method to attenuate the liftgate vibration and sound radiation by tuning the mass damper. The vehicle testing is consistent with the analysis result, and the vehicle booming is significantly attenuated. The simplified model that is proved by the testing provides a guideline for vehicle interior booming control.

Keywords: Sound and vibration characteristics, Lumped-mass damper, Booming
I-INCE Classification of Subject Number: 43

1. INTRODUCTION

With the customers growing demands on the riding comfort, how to continuously improve sound quality becomes a challenging task for many NVH engineers. Today, the OEMs around world are increasing emphasis on automobile NVH performance development [1, 2, 3].

The automobile body, as a carrier of many other systems and components, consists of beams, pillars, and various panels. The body panels are thin plates similar to speaker membranes. The modal frequencies for the body plates are usually low, from several decades Hz to several hundred Hz. The body is subjected various dynamic loads, such as engine excitation, road excitation. These excitations transfer energy to the body panels (such as front and rear windshields, roof, floor, liftgate, etc.) through the body

1 wanyp@changan.com.cn
2 pangjian@changan.com.cn
3 zhangjie3@changan.com.cn
4 yangliang11.5@163.com

beams and pillars. The excited panels radiate sound to the passenger compartment or are coupled with the body acoustic cavity modes, inducing the interior booming [4]. The booming is a low frequency noise which makes passengers uncomfortable [5]. Therefore, in the early stage of the body development, the booming induced by body panel vibrating should be paid attenuation and verified as early as possible in the body design. One of vehicle body NVH research topics around world is to control the body design to eliminate or reduce the interior booming caused by the body panel structures.

Many scholars and engineers have conducted a lot of research on the booming problem. Doo-Ho Lee studied the booming control method by a hybrid modelling method [6]. He used the modal test data as boundary conditions for acoustic simulation, which greatly improved the accuracy and computational efficiency of the hybrid modelling method, and the results showed the booming was generated by the coupling between the structural mode and acoustic cavity mode. Avnish Gosain solved the booming problem by adding a mass block on the front roof frame [7]. Prasanth B. B and Zhang J. solved the booming problem by strengthening the B pillars and the ceiling structure [8, 9]. Gajanan Tonge solved the booming problem by modifying the cowl grille structure [10]. Gaurav Gupta studied the coupling effectiveness of the structural-acoustic coupling on the booming, and identified the booming was induced by the coupling between the luggage compartment lid mode, the front ceiling mode and the acoustic cavity mode. He solved the booming problem by adding the damping material on the lid and the mass on the front roof frame.

In this paper, we analyze the low frequency booming noise of a vehicle and the vibro-acoustic characteristics of simplified model of liftgate, identifying the effect of lumped-mass damper on the vibro-acoustic characteristics of plate, which provide systematic theoretical explanation of low frequency booming noise optimization.

2. THE BASIC THEORIES

Among rectangular plate bending vibration problems, the case with simply-supported boundary condition is the simplest one and is the only one that analytical solution can be obtained. Consider a thin plate with a lumped mass at (x_C, y_C) and a united force $F_0(t)$ at the point (x_0, y_0).The width, length, and thickness of the plate are a, b and h, respectively, as shown in Figure 1.

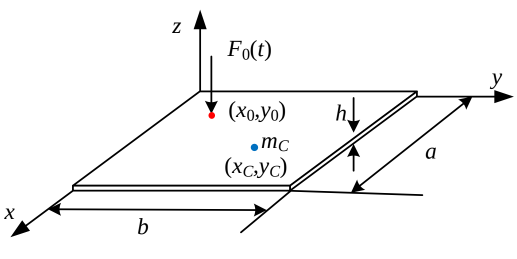

Figure 1: The rectangular plate with a lumped mass

By neglecting the effects of shear deformation and rotatory inertia effects, the governing equation for a uniform isotropic rectangular plate may be written as [11]

$$D\nabla^4 w(x,y,t) + \rho h \ddot{w}(x,y,t) = F_0(t)\delta(x_0,y_0) - m_C\delta(x_C,y_C)\ddot{w}(x,y,t) \quad (1)$$

where, $D = Eh^3/[12(1-\upsilon^2)]$ is the flexural rigidity, $w(x,y,t)$ is the displacement of the plate, E and υ are Young's modulus and the Poisson's ratio, ρ is the density of the plate, m_{C} is the weight of the lumped mass, and $\delta(x,y)$ is Dirac delta function.

The vibration response at a grid of points (x,y) on the surface of the plate using the modal summation method are given as

$$w(x,y,t)=\sum_{m}\sum_{n}q_{mn}(t)\varphi_{mn}(x,y) \tag{2}$$

where, $q_{mn}(t)$ is the modal coordinates, $\varphi_{mn}(x,y)$ is the mnth mode shape function of the structure at point (x,y), which satisfies the natural boundary conditions of the plate.

If simply supported boundaries are assumed, mode shapes can be written as

$$\varphi_{mn}(x,y)=\sin(k_m x)\sin(k_n y) \tag{3}$$

where, $k_m=m\pi/a$ and $k_n=n\pi/b$ are the structural wavelengths along the x and y direction, respectively.

Substitute equation (2) and (3) into (1), we obtain the expression

$$\begin{aligned}&\left(\omega_{mn}^2-\omega^2\right)q_{mn}(t)-4\xi\omega^2\varphi_{mn}(x_C,y_C)\sum_{m'}\sum_{n'}q_{m'n'}(t)\varphi_{m'n'}(x_C,y_C)\\&=\frac{2}{\sqrt{M_p}}F_0(t)\varphi_{mn}(x_0,y_0)\end{aligned} \tag{4}$$

where， $M_p=\rho abh$ is the mass of rectangular plate， $\xi=m_C/M_p$ is the mass ratio, ω_{mn} is represented to the mnth natural frequency of the unloaded plate.

The mass-plate vibration velocity response may be written as

$$v(x,y,t)=\sum_{m}\sum_{n}q_{v,mn}(t)\varphi_{mn}(x,y) \tag{5}$$

where, $q_{v,mn}(t)$ is the mode coordinate of velocity response. The space-average mean square velocity of the plate surface vibration is written as

$$\left\langle\overline{v^2}\right\rangle=\frac{1}{2S}\int_0^a\int_0^b\mathbf{q_v}^{\mathrm{H}}\boldsymbol{\varphi}^{\mathrm{T}}\boldsymbol{\varphi}\mathbf{q_v}\,dxdy \tag{6}$$

where, $\boldsymbol{\varphi}$ is modal shape vectors and $\mathbf{q_v}$ is modal coordinate vectors, respectively.

The total sound power is given by [12]

$$P=\mathbf{q_v}^{\mathrm{H}}\mathbf{A}\mathbf{q_v}=\mathbf{q_v}^{\mathrm{H}}\boldsymbol{\Psi}^{\mathrm{T}}\mathbf{R}\boldsymbol{\Psi}\mathbf{q_v} \tag{7}$$

where， $\mathbf{A}$ is the so-called "radiation resistance matrix", $\boldsymbol{\Psi}$ is an matrix formed by the exact modal shape vectors $\{\boldsymbol{\varphi}_1,\boldsymbol{\varphi}_2\cdots\boldsymbol{\varphi}_K\}^{\mathrm{T}}$, $\boldsymbol{\Psi}$ is given by

$$\boldsymbol{\Psi}=\begin{bmatrix}\boldsymbol{\varphi}(x_1,y_1)\\\boldsymbol{\varphi}(x_2,y_2)\\\vdots\\\boldsymbol{\varphi}(x_R,y_R)\end{bmatrix}_{R\times(M\cdot N)} \tag{8}$$

where $\mathbf{R}$ is an (R×R) matrix, given by

$$\mathbf{R} = \frac{\omega^2 \rho_0 S_e^2}{4\pi c_0} \begin{bmatrix} 1 & \frac{\sin(kR_{12})}{kR_{12}} & \cdots & \frac{\sin(kR_{1R})}{kR_{1R}} \\ \frac{\sin(kR_{21})}{kR_{21}} & 1 & \cdots & \frac{\sin(kR_{2R})}{kR_{2R}} \\ \vdots & \vdots & \ddots & \vdots \\ \frac{\sin(kR_{R1})}{kR_{R1}} & \frac{\sin(kR_{R2})}{kR_{R2}} & \cdots & 1 \end{bmatrix}_{R\times R} \quad (9)$$

Eq. (9) is the so-called "radiation resistance matrix" [13], where ρ_0 is the density of the air, c_0 is the sound speed in the air, S_e is the area of each radiator, and R_{ij} is the distance between the centers of the ith and jth elements.

3. The effect of lumped mass on the vibro-acoustic characteristics of plates

Taking a simply supported steel plate as an example, we calculate its modes and modal frequencies. The effects of lumped-mass on the vibro-acoustic characteristics of the plate are investigated in this subsection. Table 1 provides geometrical and material properties of the steel plate. The location of lumped-mass and excitation force are listed in table 2.

Table 1: Properties of the steel plate and reinforced stiffened beam

Property	Symbol	Value
Length/m	a	0.79
Width/m	b	0.64
Thickness/mm	h_P	1.5
Density /kg/m^3	ρ	7850
Young's Modulus /GPa	E	210
Poisson ratio/-	μ	0.3

Table 2: Point masses employed

	Mass1	Mass2	Mass3
m_C/M_P	5%	10%	15%
$(x_C/a, y_C/b)$	(0.437, 0.585)		
$(x_0/a, y_0/b)$	(0.184, 0.255)		

3.1 The effect of lumped mass on Natural Frequency

The natural frequencies of the plate attached with a lumped mass was analyzed by the ANSYS and shown in table 3. The results are shown that the natural frequencies of plate decreased with the increasing weight of lumped mass, especially for the first mode. The varied ratio of first natural frequency is more than 10 %, and other natural frequency change rates are about 5 %.

Table 3: Natural frequencies of a rectangular plate with a local mass

	Mode shape	Single plate/Hz	Mass 1 /Hz	Varied ratio/%	Mass 2 /Hz	Varied ratio /%	Mass 3 /Hz	Varied ratio /%
1	(1,1)	14.9	13.0	-13.8	12.2	-18.2	12.1	-18.8
2	(2,1)	32.5	31.6	-2.8	31.1	-4.4	30.9	-5.0
3	(1,2)	41.7	40.1	-3.9	39.4	-5.5	39.0	-6.4
4	(2,2)	58.2	57.2	-1.8	54.3	-6.8	53.2	-8.6
5	(3,1)	61.4	59.1	-3.8	58.1	-5.4	58.0	-5.5

3.2 The effect of lumped mass on velocity response

According to Table 2, we change the weight of lumped mass and calculate mean square velocity response of plate surface by equation (6), as shown in Figure 2. The results suggest that the surface velocity response of the plate structure at the first natural frequency has been greatly affected by the lumped mass.

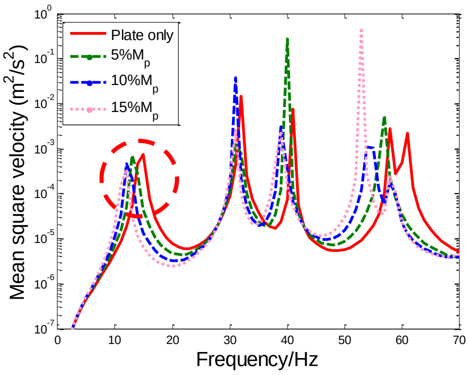

Figure 2: Mean square velocity of plate with a varied point mass

3.3 The effect of lumped mass on sound radiation power

Figure 3 shows the sound radiation power of plate with lumped mass. It is seen that the lumped mass tends to reduce both the vibration and radiation at the fundamental frequency. But the lumped mass obviously increases the radiated sound power for other order modes.

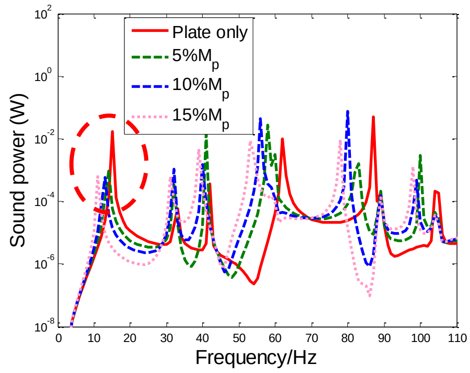

Figure 3: Influences on sound powers of a plate with a varied point mass

4. Case Study: low frequency drumming noise optimization control

Typically, the liftgate assembly is installed on the body using the liftgate hinge, liftgate pole and bolts through the assembly method, keeping the liftgate in a closed state through the liftgate lock assembly. Between the liftgate assembly and the body, we always design some seals and 4-6 bumpers, to avoid abnormal sound or other NVH problems of the vehicle. This fixed installation way of liftgate can be simplified as a simply supported rectangular plate, in order to analyze the NVH problem induced by liftgate vibration.

The interior booming was identified in a SUV by the subjective driving evaluation on a rough road, especially at vehicle speed of 40 km/h at 2nd transmission gear. The booming at the driver seat is louder than the middle seat. Figure 4 shows the tested interior noises. The booming peaks appear at 32 Hz, and the sound pressure level at the diver seat is 5 dB(A) higher than that of the middle seat. By a lot of testing and analysis work, the structural borne sound is confirmed as the major contributor of the booming.

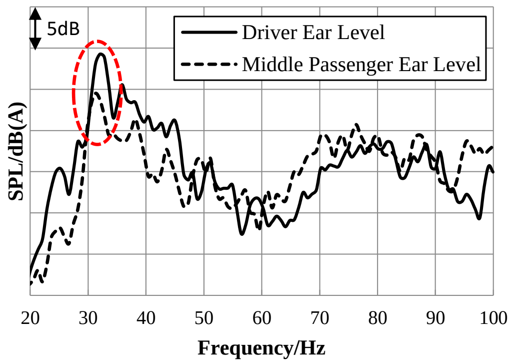

Figure 4: Tested interior sound pressure at vehicle speed of 40km/h

According to “source–path–receive” NVH analysis model, the front and rear suspensions are excited using frequency-sweep excitation separately, and the vibration characteristics of liftgate and interior sound pressure are tested. The results show that the peak of vibration velocity response of liftgate appears at 32 Hz, giving rise to the increasing of the sound pressure response and the booming problem.

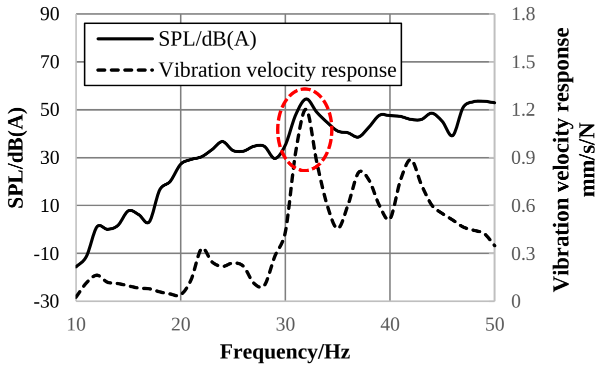

Figure 5: Relationship between liftgate vibration velocity response and sound pressure level

In order to further verify the relationship between the liftgate constraint mode and the booming problem, we added a mass damper on the liftgate and analyzed the effect of the liftgate vibration velocity response on the sound pressure response. Figure 6 shows the effect of the mass damper on the vibration velocity response of the liftgate and the sound pressure response. It is found that the lumped-mass damper tends to reduce the vibration velocity response of the liftgate and the sound pressure level significantly at 32 Hz. The subjective evaluation results also show the booming is reduced obviously.

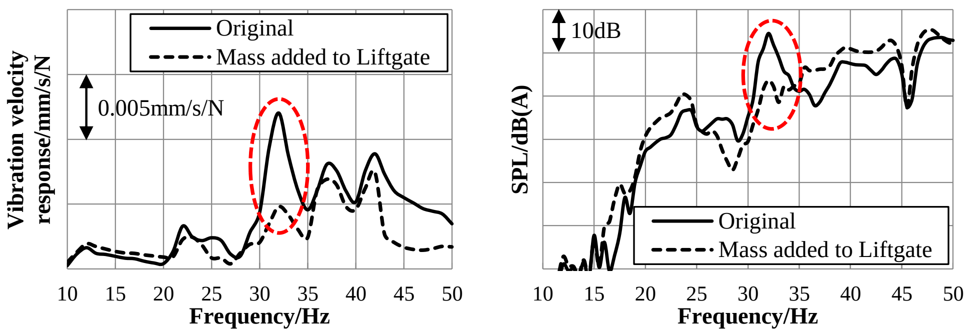

Figure 6: The vibration velocity response of liftgate and sound pressure level comparison by adding masses

Based on the pervious theoretical and structural mode analyses, two mass dampers are added at the location of inner panel of the liftgate, as shown in Figure 7. Figure 8 shows the tested results of the interior noise, which shows that the booming appearing at 32 Hz is reduced by 4 dB(A). The subjective evaluation results also show the annoying booming is significantly reduced, and the drivers don't complain the problem anymore.

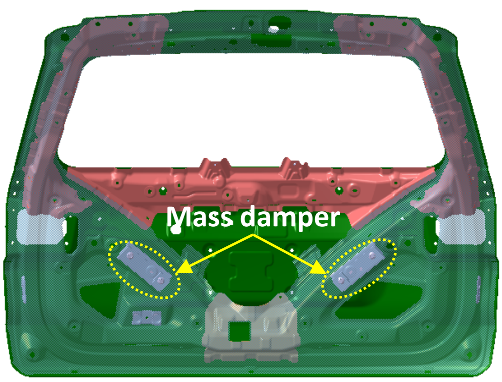

Figure 7: Mass dampers are added on the liftgate

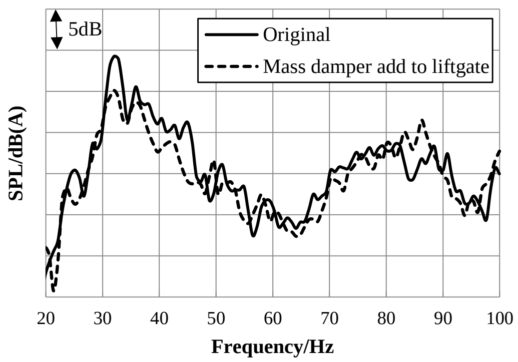

Figure 8: The interior sound pressure comparison by modifying the liftgate

5. Conclusions

The paper analyzed the effect of lumped mass on the vibro-acoustic characteristics of rectangular simply supported plates, and the research results provide guidance to the optimization of the low frequency booming noise. The main conclusions are listed as follows:

a) For the rectangular simply supported plate, when the local mass position is determined, the vibration velocity response of the first order mode is more sensitive to the lumped mass.

b) The local mass tends to reduce the radiation sound power at the first natural frequency, but significantly increase the radiated sound power for the other order modes.

c) After the booming noise is identified, the theoretical analysis is applied on it. Two masses dampers are added on the liftgate and the booming noise is significantly reduced.

6. REFERENCES

1. Qatu, M. S., Abdelhamid, M. K. and Pang, J. Overview of automotive noise and vibration. Int. J. Vehicle Noise and Vibration, 2009, 5: 1-34.
2. Pang, J., Shen, G. and He, H. Automotive noise and vibration-principle and application. Beijing: Beijing Institute of Technology Press, 2006.
3. Pang, J. Vehicle body noise and vibration control. Beijing: China Machine Press, 2015.
4. Tsujiuchi, N., Koizumi, T., Nagao, T., Kido, I. and Hashioka, M. Vibration Transmission Analysis of Automotive Body for Reduction of Booming Noise. SAE: 2011-01-1691.
5. Sung, S., Chao S., Lingala, H. and Mundy, L. Structural-Acoustic Analysis of Vehicle Body Panel Participation to Interior Acoustic Boom Noise. SAE: 2011-01-0496.
6. Gosain, A. and Ravindran, M. Refinement of cabin booming noise in a small passenger car using hybrid numerical-experimental methodology. SAE: 2010-01-0414.
7. Prasanth, B., Wangh, S. and Ranghuvanshi J. Body induced booming noise control by hybrid integrated approach for a passenger car. SAE: 2013-01-1920.
8. Tonge, G. and Agrawal, S. Analysis of open plenum structure for reduction of booming noise. SAE: 2013-01-0636.
9. Zhang, J., Pang, J. and Zhang, J. Experimental and Simulation Analysis of Interior Booming Induced by Vehicle Body Panel Vibration. ICSV23, 2016.
10. Gupta, G., Gautam, R. and Jain, C. P. Study of coupling behaviour of acoustic cavity modes to improve booming noise in passenger vehicles. SAE: 2014-01-1974.

11. Lin, T. R., Pan, J. A closed form solution for the dynamic response of finite ribbed plates. Journal of the Acoustical Society of America, 2006, 119(2): 917-925.
12. Ji, L., Bolton, J. S. Sound power radiation from a vibrating structure in terms of structure-dependent radiation modes. Journal of Sound & Vibration, 2015, 335: 245-260.
13. Elliott, S. J., Johnson, M. E. Radiation modes and active control of sound power. Journal of the Acoustical Society of America, 1993, 94: 2194-2204.