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STUDY ON THE SIMULATION DESIGN METHOD ABOUT EXHAUST MUFFLER OF VEHICLE

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Abstract

In this paper, the performance of attenuating sound about automotive exhaust mufflers were simulated with the method of boundary elements, and the flow performance of muffler were simulated with the method of CFD. The results were validated with the experiment data. So it indicated that the simulation model is correct and the method is reasonable. Based on the experiment validation of the transmission loss of muffler, the frequencies of sound should be controlled and the possibility of amelioration were found out for exhaust muffler by the simulation method of boundary element. The reason of causing an attenuation trough for this muffler were analyzed by the acoustic model and sound pressure map analysis methods, and the improvement is put forward with the control measure of long duct standing wave, and the muffler design method and simulation are explored in this process. And the effect of flow excited noise was studied, some useful results were gotten out by the experiment and the simulation.

1. INTRODUCTION

The exhaust noise is an important component of automotive noise and which controlling mainly depends on the exhaust muffler. The muffler has two actions to the engine: the first, it reduces the exhaust noise, this action is called the noise elimination performance; and the second, it increases the resistance of exhaust and makes the engine's performances such as power lower, this action is called the power loss of muffler. The aim of muffler design is to increase the noise elimination performance as possible in the premise condition of guaranteeing the necessary output power.

In the past, a muffler's design depended on the 1-dimension plane wave theory, the experiences of designer and the lot of experiments. It has been proved that plane wave theory isn't correct when calculating at high frequency. Both 1-dimension plane wave theory and experience design can't support the experiment for the complicated muffler design very effectively, especially they need longer design time and higher cost. Compared with above methods, the 3-D numerical method can calculate correctly the more complicated numerical model with computer and can find rapidly the best plan in shorter time, so it can give out the more exact reference parameters for experiment, reduce the cost and accelerate the process of products development.

Based on the theory of sound wave and the principle of muffler, the performances of several mufflers were analyzed with the software of SYSNOISE, FLUENT and FIRE, then the design methods of muffler has been studied and a new design method of muffler was got.

2. THE DESCRIPTION ABOUT EQUATION OF SOUND WAVE

As macrophysics phenomena, the vibration of sound must obey the basic laws of physics, they are the Second law of Newton, the Conservation of mass and the Gaseous state equation. To establish the wave equation of sound, following predigestion and hypothesis must be done first:

1. Medium is ideal and has no glutinosity, there is no energy loss as sound wave diffuse.
2. The transmission process of sound is a process of adiabatic, and has no exchange with outside.
3. Sound pressure P is much smaller than the pressure of medium, particle velocity is much lower than sound velocity, particle displacement is much smaller than the sound wavelength and the increment of medium density ρ is much smaller than static density of medium. All of the hypothesis means that the changes of the matter state is linear. The sound wave equation is as follow:

$$\frac{\partial^2 p}{\partial t^2} = c^2 \nabla^2 p \tag{1}$$

$$\nabla^2 = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}$$

We can suppose $p = p(x,y,z)e^{i\omega t}$, i means a complex number. The above equation can be changed to Helmholtz equation of sound pressure value P(x,y,z):

$$\nabla^2 P + k^2 p = 0 \tag{2}$$

There are three kind of boundary condition in the analysis of mufflers:

1. The first is the pressure condition, it is also named as Dirichlet condition, and particle's velocity is unknown. Normally, pressure is used as the force at the entrance. Helmholtz equation is correct with this condition
2. The second is the velocity of particle, it is also named as Neumann condition, the pressure is unknown. Normally, this condition is used as the force at the entrance or used as a coupling interactional of sound and solid when a vibrating surface radiate noise. With this condition, Helmhtz equation is surly correct:

$$\frac{\partial p}{\partial n} + i\rho\omega\overline{v}_n = 0 \tag{3}$$

Here, $\frac{\partial p}{\partial n}$ is the vertical derivative of vibration surface, $\overline{v}_n$ is the vertical velocity of vibration surface.

3. The third kind is the given admittance, is also named as Robin condition, only the

relation between the particle velocity and pressure. It acts normally on the surface which has the absorption materials. At this boundary, the following equation is correct:

$$\frac{\partial p}{\partial n} + i\rho\omega A_n p = 0 \tag{4}$$

In this, A_n is the admittance acting on the surface. The surfaces met often are rigid surface, A_n=0; absorbing surface completely, $A_n = 1/\rho c$; and the surface with absorbing coefficient α, $A_n = \alpha/\rho c$.

3. THE VALIDATION ABOUT MUFFLER MODEL OF MOTORCYCLE

Motorcycle's muffler usually has a structure of zones, its entrance is small and its exit is large, total 3—5 chambers, usually the exhaust channel is set off center and the airflow sometimes reverse to 180 degree. 1-dimension plane wave theory is not able to calculate such complicated muffler and the 2-dimension axis symmetry model can't be used because of it is not an axis symmetry structure. So 3-dimension method is the only choice. The Indirect Boundary Method (IBM) is suitable because muffler is a complicated lamina structure. Figure 1 is the IBM element model of a motorcycle muffler.

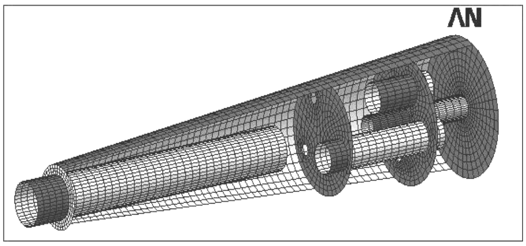


Figure 1. IBM of element model of muffler

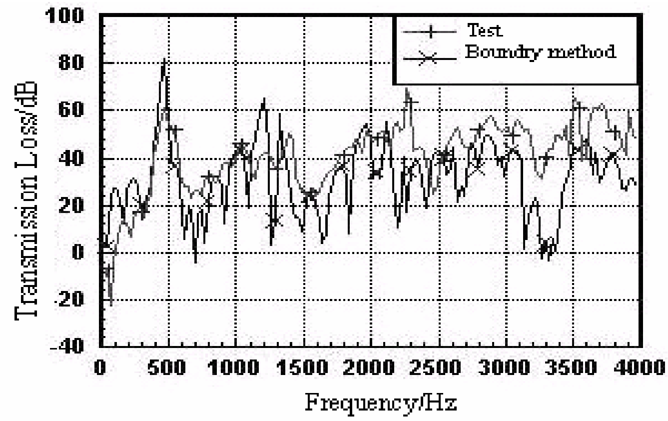


Figure 2. The contrast of calculation and test

Because of the measurement parameters are the sound pressures at the entrance and the exit of muffler, the boundary conditions are unit velocity at entrance, the unclosed exit is looked as a piston radiating in half space. Other walls are looked as rigid walls.

If
$$TL' = 20 \times \log_{10} \frac{p_{in}}{p_{out}} \tag{5}$$

Pin – in pressure, Pout–out pressure

Figure 2 is the result contrast of calculation and test. We can see that calculation tallies basically with test in all frequencies. The reason of undulation with the calculation is that the influence of air glutinosity not is considered in calculating. The wider elements mesh maybe the reason of difference at high frequencies.

4. STUDY ON AMELIORATING THE CHARACTERISTICS OF DECREASING SOUND.

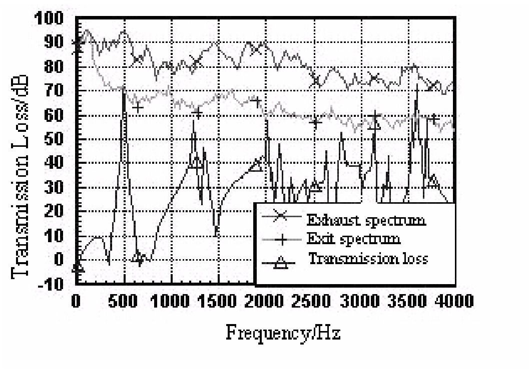


Figure 3. Anechoic characters of muffler

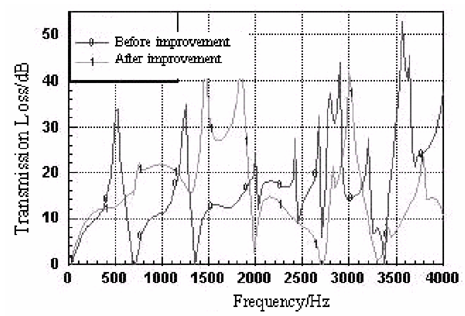


Figure 4. Performances of chamber 1 after and before improving.

The two curves in Fig. 3 are the exhaust frequency spectrum of muffler entrance, the exhaust frequency spectrum of muffler exit and the transfer loss of muffler at 4600r/min speed of motor engine. From Fig. 3 we can know that the anechoic amount need increased.

Showed as Fig. 1, each clapboard in this muffler has two small holes. They can be looked as an orifice plate anechoic structure and has a parallel relation with these three anechoic chambers. Depending on the analogy theory of sound and electricity, when two electric elements is parallel connection, the impedance will decrease, the current will increase and electric power will grow up. Similarly, the parallel connection of two acoustic elements will make the sound impedance decrease, make the volume velocity and sound power increase. It means that the anechoic character will decrease with this structure. This structure’s influence to anechoic character is shown by Fig. 5. The diameter of interpolation pipes has been increased when the small holes were closed, thus the flow area can be guaranteed.

The sound modal and the distribute figure are used to analyze the reason of producing the anechoic valleys in this structure, and the length of interpolation pipes in muffler were adjusted to find the better methods and laws by means of the control method of long standing wave.

The transmission loss in the first anechoic chamber is shown as curve 0 in Fig. 4. It shows that there are two anechoic valleys near 750Hz and 1450Hz. Figure 6 is the presser distribution of these two frequencies in chamber 1. The two curves at right and left are the sketch map of standing wave when air in chamber occurs sympathetic vibration. You can see that it is a half wave length in anechoic chamber when at 750Hz, and a whole wave length at 1450 Hz.

The control principle of long pipe standing wave[2] is mainly use of control the vibration speed of source to control the vibration speed of mass point of sound wave. The max vibration speed of air point will not exceeds the vibration speed of stimulant source that is placed at the wave peak of air point, then the standing wave in pipe will be limited. By contraries, the max vibration speed of air point will not be less the vibration speed of stimulant source and the standing wave will be stimulated. Taking the interpolation pipe as a stimulant source and improving chamber 1 showed as Fig. 9, the result is showed as line 1 in Fig. 4, it is seen that two anechoic valleys at 750Hz and 1450Hz were eliminated. Curve 0 of Fig. 8 is the transmission loss of chamber 2 and by the curve we can know that the improving work ought to be focused on anechoic valley of 2000Hz. Figure 7 is the contrast figure of sound modal and sound pressure at anechoic valley frequency. We can know from Fig. 7 the reason of anechoic valley just is the natural sound sympathetic of the first frequency was stimulated in the chamber. Proving the second chamber with said method (showed in Fig. 9), the result is shown as line 1 in Fig. 8, it is seen that two anechoic valleys at 750Hz and 1450Hz were eliminated, shown as line 1 in Fig. 8.

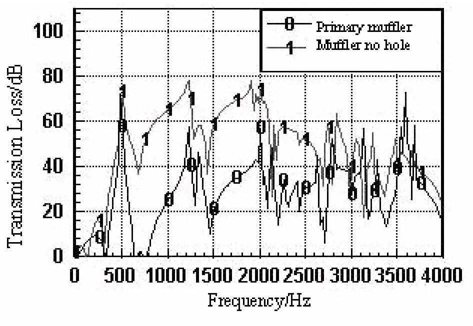


Figure 5. The influence of small holes to anechoic characters

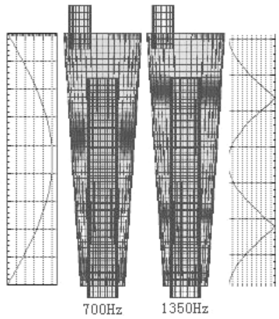


Figure 6. The pressure distribution of two frequencies in chamber 1

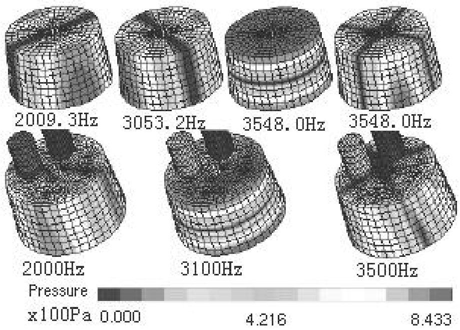


Figure 7. Natural shapes and sound presser of chamber 2

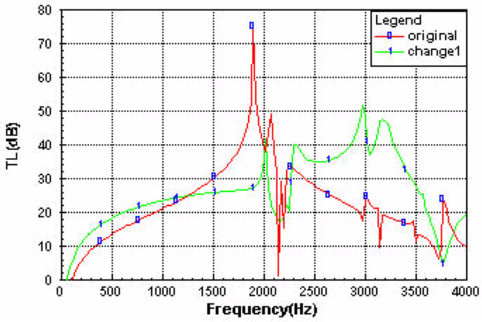


Figure 8. The contrast result of chamber 2 before and after improving

The improvement about the third chamber is same as before two and the whole improvement is shown as plan 1 in Fig. 9.

Because the two small holes of inter clapboard were closed and the flow area of air will decrease, the diameter of pipes a, b, c and d were adjusted. The hole diameter of the old muffler is 0.008m, the flow area of new muffler is lager than the old one, so the flow performance will be better. The anechoic performance improved is shown in Fig. 10. From Fig. 10, we can know that the goal of improving has been achieved. The reduced noise quantity has a evident improvement at frequencies of 350Hz, 750Hz and 1500Hz.

5. THE FLOW EXCITED NOISE IN THE MUFFLER

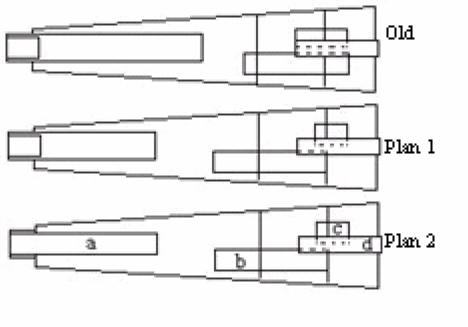


Figure 9. Improvement plan

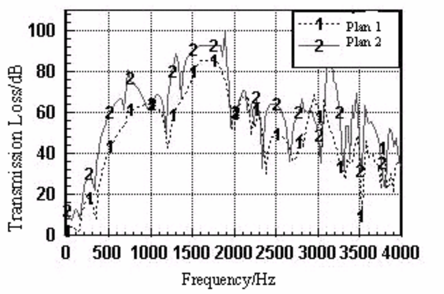
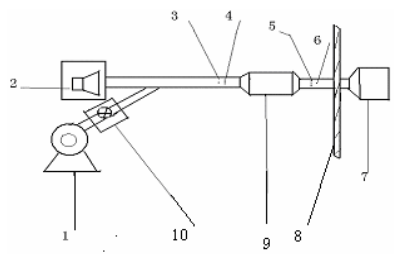


Figure 10. The anechoic performance improved

It is often found that the anechoic amount (Transmission Loss of sound) is smaller when the air speed is higher in the muffler than air speed is lower. This is because of the flow excited noise has been created. Because of the flow excited noise, some differences between test and calculation will exist. So the law of flow excited noise should be studied. To study the flow excited noise, an experiment bench was built as Fig. 11.



1— fan,2—sound source, 3— front sound pressure 4—front air pressure, 5—rear sound pressure 6—rear air pressure, 7—exit, 8—sound insulation 9—muffler, 10—flow valve

Figure 11. Muffler experiment bench

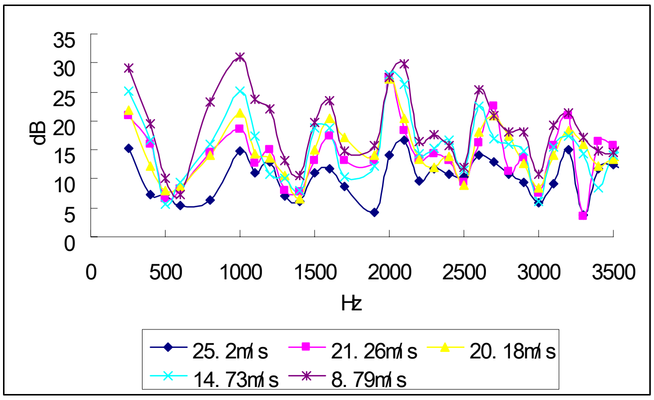
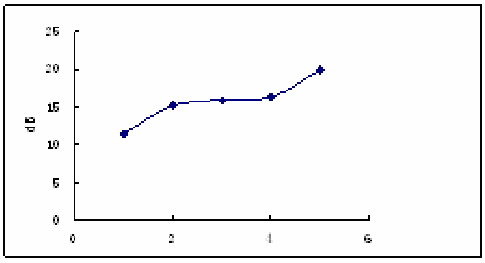


Figure 12. Experiment result of the flow excited noise

The flow excited noise of muffler is the onflow noise (mainly middle and high frequency) when the air flow through the muffler with very high speed and the vibration noise (mainly low frequency) of muffler parts excited by the flow. Figure 12 is the flow excited noise result of test taken out on the above test bench with a car’s muffler. From Fig. 12 and Fig. 13 we can see that the loss of sound pressure is being smaller with the air speed Higher, it means that the effect of flow excited noise is much clear when the speed of air flow is much rapid. From Fig. 14 and Fig. 15, the distributing of velocity has a great difference in muffler and there are several swirls in it. The swirl and the differences of velocity will form the onflow energy in muffler and the swirl noise has been created. The swirl noise will change with the velocity of flow, the higher speed of air, the more violent swirl, and the higher swirl noise. This is why the Transmission Loss of sound is smaller when the velocity of flow is higher.



1—25.2m/s, 2—21.26m/s, 3—20.18m/s, 4—14.73m/s, 5—8.79m/s

Figure 13. Anechoic amount with the velocity of air

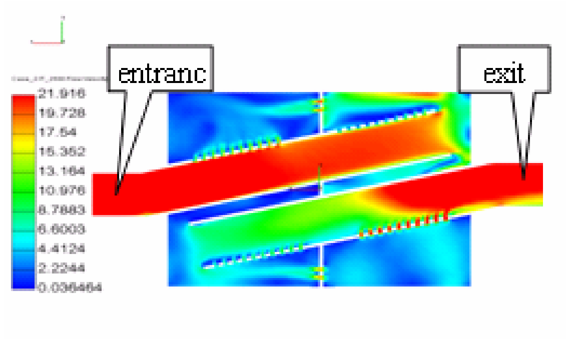


Figure 14. Velocity distributing in the car muffler

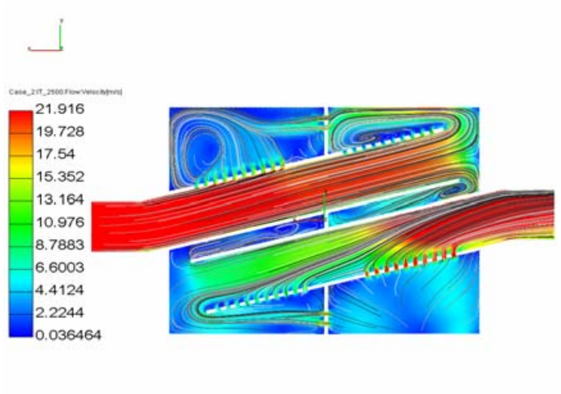


Figure 15. Air velocity trace in muffler

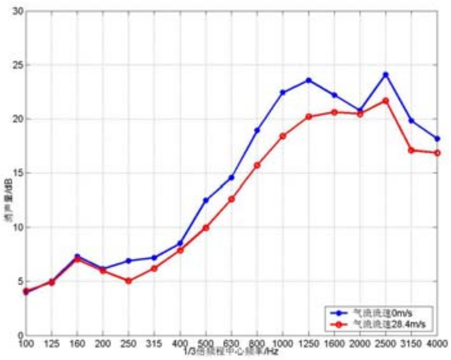


Figure 16. Transmission loss of sound with different velocity of flow

6. CONCLUSIONS

1. The combine method of sound model and distribution figure of sound pressure can be used to analyze the reason of anechoic valleys, you can improve the structure of muffler according it.
2. The control method of long pipe standing way can be used to eliminate anechoic valley. It is very effective for long anechoic structure.
3. The flow excited noise is the reason that the transmission loss of sound becomes less.
4. It still has much work to do for the stimulation of exhaust muffler, this paper only do a elementary research.

REFERENCES

[1] Huan Qibai, the Control of Engineering Noise, Hua Zhong Technology University, 1998.5
[2] Ma Dayou, Noise Control, Science publish, 1987.7
[3] Jian Pang, Robert Rebandt, Greg Knapp, Mohamad Qatu and David Demmith, Flow

Excited Noise Analysis of Exhaust, SAE 2005-01-2352

[4] Frieder H. Kunz, Semi-Empirical Model for Flow Noise Prediction on Intake and Exhaust Systems, SAE paper #1999-01-1654

[5] Masato Sakurai, Relation Between Exhaust Pulsating Flow and Radiation Noise: Development of Exhaust Radiation Noise Simulation Technology, SAE paper#2004-01-0399

[6] Characterization of Rasping noise in Automotive Engine Exhaust Ducts, J. of Sound and Vibration, 2001, 244(1),79-106