Study on Acoustic and Fluid Characteristics of Exhaust Muffler

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Abstract: - Engine noise is mainly from exhaust noise. Without adding muffler, at a certain speed, the noise up to 100 decibels. Reduce engine exhaust noise, and to high-temperature exhaust gas can be safely and effectively. Muffler as part of exhaust pipe, should ensure that the exhaust flow, resistance and sufficient strength. This research takes an exhaust silencer as a study object, and a muffler sound field model is established by using the finite element pre-processing software Hypermesh. According to the finite element analysis of acoustics and fluid, silencer acoustic performances are analyzed and the muffler transmission loss is calculated. Through the improvement of the first cavity, the transmission loss is compared with the original structure. The results clearly reflect muffler sound field conditions, and provide a basis for the muffler's improving and optimal design.

Key-Words: - Muffler; Exhaust noise; Finite element model; Transmission loss; Acoustic characteristics

1 Introduction

Exhaust noise is mainly due to the high-speed flow of the high-pressure combustion exhaust gas, a kind of aerodynamic noise, including baseband noise, air column resonance noise in evacuation line, vortex noise, sprayer column noise and impact noise, airflow frictional noise and turbulent noise in exhaust pipe inner wall[1], Helmholtz resonance noise, structural radiation noise.

Baseband noise, as the name indicates, is noise when the blast gate of each cylinder of internal combustion engine opens, the gas inside the cylinder ejected out at high speed suddenly, airflow shocks the gas near blast gate inside exhaust passage, producing a pressure upheaval to form a pressure wave, to induce noise[2]. Since each cylinder exhaust is in the phase of the specified periodicity, this noise is a typical low-frequency noise. Frequency is the same as the exhaust frequency. Thus we have the equation

$$f_i = i\frac{Zn}{30\tau} \tag{1}$$

Where i represents Harmonic number, i=1, 2, 3…; Z represents number of cylinders; n represents speed of crankshaft, r/min; τ represents number of stroke[3].

In the engine exhaust system, from the closed exhaust valve, muffler, to atmosphere, consisting of a column one end open and the other closed. The air column is induced periodically by the exhaust noise, generating resonance to bring about air column resonance noise. Thus we have the equation

$$f_i = \frac{(2i-1)c}{4l} \tag{2}$$

Where i represents harmonics times，i=1,2,3…； l represents column length； c represents sound velocity.

When the air flows to the lips of every exhaust gas crossing, it will generate periodical vortex[4]. This vortex will make the gas inside branch pipe bring about pressure fluctuations, generating the noise. The frequency equation is

$$f = S_t\frac{v}{d} \tag{3}$$

Where St represents Si Tuo Hal number, forced exhaust from 1.5-2,free exhaust associated with supercritical pressure ratio; d represents vents feature size; v represents the flow rate of the exhaust gas flowing through the exhaust doors.

In addition, when the high-speed airflow pass through the narrow part, the flow will be faster and bring about exhaust gas vortex. The sound intensity generated by vortex is proportional to airflow velocity to the 8th power, mostly are high frequency. In the free exhaust stage, the air injection will make the blast gate generate strong sprayer column noise. Sprayer column noise is a kind of wideband high-frequency noise. Impact noise is brought about by the unstable air inside the exhaust air duct. Discontinuous surface of air pressure exists near the blast gate. This may result in a shock wave. sprayer column noise and impact noise are continuous broadband noise , their peak frequency is:

$$f = S_t \frac{c}{d} \tag{4}$$

Where c represents sound velocity; St represents Si Tuo Hal number, associated with supercritical pressure ratio; d represents vents feature size[5,6];

When the airflow velocity is quite a few high, the friction is generated between the fluid and the pipe wall. One side ,it create great noise when the airflow spreads to tail tube, bringing about airflow frictional noise; On the other side, eddy current of noise that turbulent gas generates near the evacuation line brings about the gas pressure fluctuation, radiated noise is the turbulent noise. The main peak value is:

$$f = 0.04v / A \tag{5}$$

Where v=velocity through pipe section; A=sectional area.

Helmholtz resonant air space is a resonance system, consisting of a closed volume of the cavity V, through a pipeline with the atmosphere, S is defined as sectional area, L is defined as the length. Resonance frequency is determined by parameter V,S and L. When the engine exhaust valve opens, the cylinder, exhaust passage and evacuation line consist of a Helmholtz resonant air space, with resonance frequency. The frequency similar with the resonance frequency will be fully amplified in this resonant air space. Thus the equation

$$f = \frac{c}{2\pi}\sqrt{\frac{S}{V_k\left(l + \sqrt{S/2}\right)}} \tag{6}$$

Where c represents sound velocity; Vk represents cylinder volume; S represents sectional area of evacuation line; l represents evacuation line length.

Helmholtz resonance noise[7] is irrelevant with engine speed, relevant with the cylinder volume, especially one-cylinder motor, also exists in two-cylinder and three-cylinder engine. But, in four-cylinder and above, due to the interference between each cylinder, this kind of noise is not obvious[8].

The pipe of the exhaust system and muffler device are induced by mechanical vibration and internal fluid pressure wave to bring out vibration, incentive structure will radiate out in sound, forming a structural radiation noise. Structural radiation noise is determined generally by physical dimension, structural style, rigidity and so on, the frequency is relevant with the frequency of thin plate structural vibration[9].

Some types of noise frequency, shown in Table 1.As indicated from Table1, baseband noise and gas injection resonance noise are mainly concentrated in middle and low frequency, and harmonic noise exists. Vortex noise, sprayer column noise and impact noise are mainly concentrated in high frequency without harmonic noise.

Table 1 different types of exhaust noise frequency

speed of（r/min） / noise type (Hz)	4000	7000
baseband noise	33.33×i	58.33×i
air column resonance noise in evacuation line	255.16×(2×i-1)	255.16×(2×i-1)
vortex noise	102857	102857
sprayer column noise and impact noise	30443	30443

2 Acoustic Simulation on Exhaust Muffler

In this part, firstly, the finite element model of muffler is established**.** Then, acoustic properties of muffler is simulated.

2.1 Establish finite element model of muffler

The original muffler is mapped, UG model is established, the UG model is imported to HYPERMESH for dividing grid, then the grid is imported to SYSNOISE for simulation of acoustical property. Based on acoustical theory, the acoustic FEM is adopted, each node pressure of muffler is calculated. Basic steps: finite element model of muffle is established, boundary condition is determined, solution.

Study on the acoustic characteristics of the muffler unit can divide muffler into three acoustic units and analyze their acoustic characteristic separately.

Fig.1 and fig. 2 show the geometric model. The first and third cavity are expansion cavity, the second is resonance cavity. The exhaust flows from left margin, first of all, the cavity of the right margin, the resonance cavity, the third cavity in the left margin, last is exhaust tail tube.

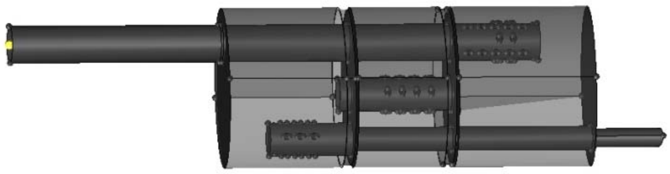
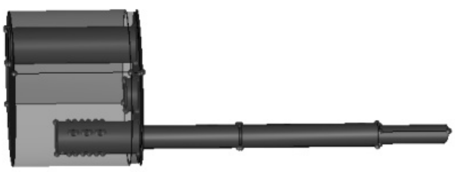


Fig.1 Geometric model of muffler

(2-a.First cavity)

(2-b.Second cavity)



(2-c.Third cavity)

Fig.2 Three cavities geometric model of the muffler

According to the HYPERMESH, the geometric model is analyzed and divided into mesh. Unit weight is determined by the highest calculated frequency f, the higher the calculated frequency, the shorter the maximum unit side length needed. If the total number of units is greater, the cost of computing resources is more. The Eq.7 shows the equation of the maximum unit side length.

$$L_{\max} \leq \frac{1}{6} \cdot \frac{c}{f} \tag{7}$$

According to the motor near field noise spectrum, the calculating frequency range is below 6000Hz, assume the exhaust temperature is 600℃ , the sound velocity c is 591.96m/s , so the maximum unit side length is shorter than 16.44mm.

Fig.3 shows the finite element model of overall muffler with 40848 solid elements and 10110 nodes, the maximum unit side length is 16.0 mm.

Fig.4 shows finite element model of the first cavity with 38285 solid elements and 7999 nodes, the maximum unit side length is 15.0 mm.

Fig.5 shows finite element model of the second cavity with 21490 solid elements and 4624 nodes, the maximum unit side length is 10.0 mm.

Fig.6 shows finite element model of the third cavity with 20560 solid elements and 4679 nodes, the maximum unit side length is 10.0 mm.

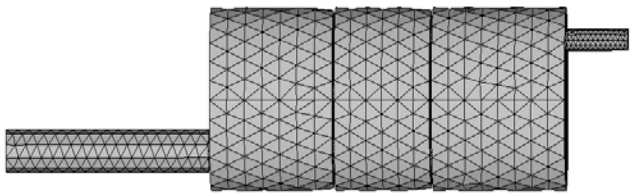


Fig.3 Finite element model of overall muffler

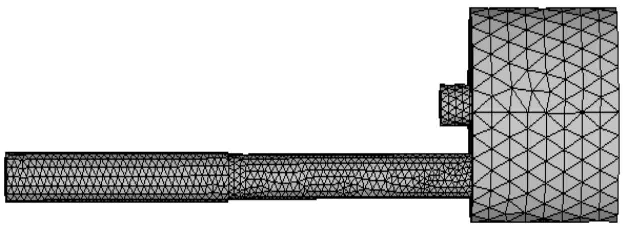


Fig.4 Finite element model of the first cavity

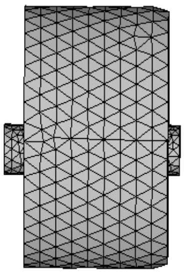


Fig.5 Finite element model of the second cavity

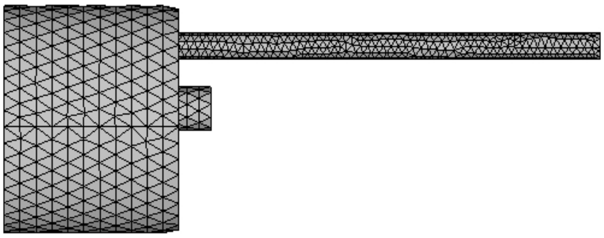


Fig.6 Finite element model of the third cavity

2.2 Simulation on acoustic properties of mufflers

Determine the gas properties in muffler before calculating and regard the exhaust as air approximately. The calculating parameters include density, sound velocity and acoustic impedance, meantime, temperature should be taken into consideration. Assume the exhaust temperature is 600□,set the inlet as unit sound pressure excitation and the outlet as full absorption condition. So the sound absorption coefficient is $\alpha = 1$, outlet resistance is ρc, ρ is the air density at 600□, c is sound velocity at 600□; ρc is intrinsic constant, playing an more important role for sound propagation than ρ and c alone, so is called natural impedance. Sound velocity, density, natural impedance is changed with temperature in different medium. When the medium is air, if the temperature rises, the air density decreased, meantime, sound velocity become faster. For a given wavelength of the acoustic wave, the natural impedance is associated with frequency. The equation is shown as Eq.8

$$A_t = A_{20}\left(\frac{T}{T_{20}}\right)^n = A_{20}\left(\frac{273+t}{293}\right)^n \tag{8}$$

Where T=Kelvin temperature; At=physical quantity at t □; A20= variance at 20 ℃ ;n=temperature exponent

Table 2 shows some temperature exponent of physical quantity

physical quantity	symbol	n value
sound velocity	c	1/2
density	ρ	-1
natural impedance	ρc	-1/2
viscous parameter	η	0.7

Assume the walls of muffler is rigid wall, sound absorption coefficient is zero, set analysis frequency from 20 Hz to 6000Hz, the frequency increases by certain step length of 20 Hz.

Import the grid to SYSNOISE, calculate acoustical property of the whole muffler and each cavity and derive sound pressure data .Draw the transmission loss curve with MATLAB, draw sound pressure nephogram at the frequency that silencing effect is not obvious.

2.2.1 Acoustic characteristic of the whole muffler

Adopt the pressure of the input and output nodes, remove the reflected sound pressure. Fig.7 shows the original transmission loss curve. Fig.8 shows the corresponding one-third transmission loss curve. The passing frequency is at 320Hz and 620 Hz, transmission loss is not obvious and silencing effect is not good at the frequency of 2480Hz，3540 Hz，4120 Hz，4960 Hz，5560 Hz through analyzing the Fig.7. The transmission loss is big when the frequency is above 1000Hz, especially near 2000Hz. Fig.9, Fig.10, Fig.11 and Fig.12 show the whole simulation of sound pressure nephogram at 320Hz 、620 Hz、2480Hz、3540 Hz.

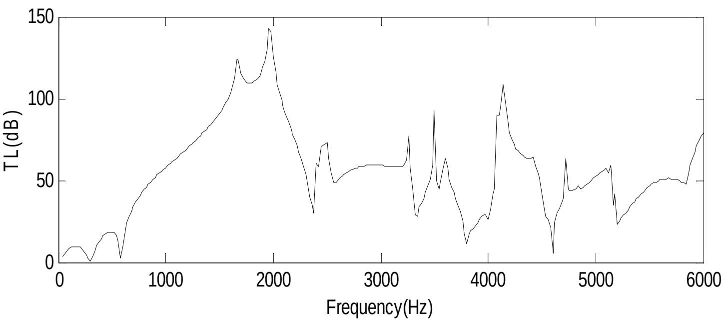


Fig.7 Transmission loss of whole muffler

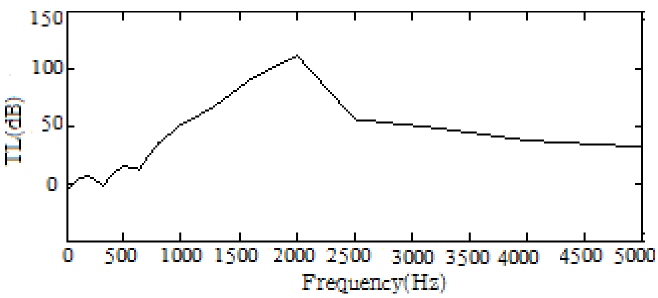


Fig.8 Transmission loss of whole muffler- 1/3oct

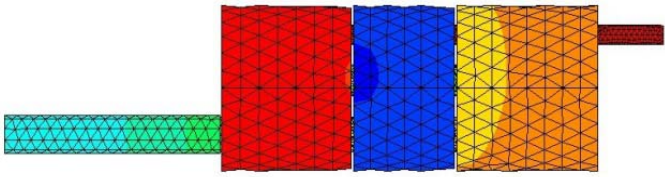


Fig.9 Sound pressure nephogram at 320Hz

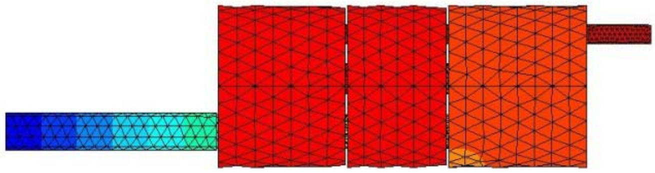


Fig.10 Sound pressure nephogram at 620Hz

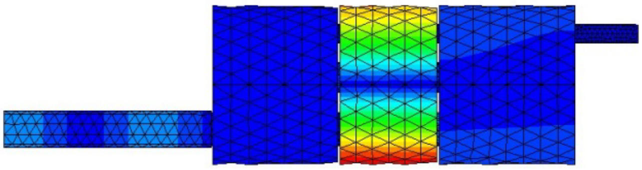


Fig.11 Sound pressure nephogram at 2480Hz

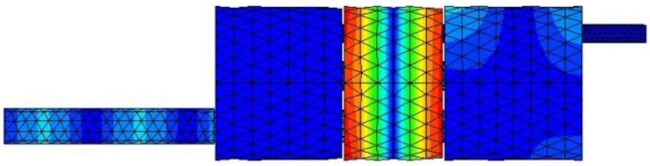


Fig.12 Sound pressure nephogram at 3540Hz

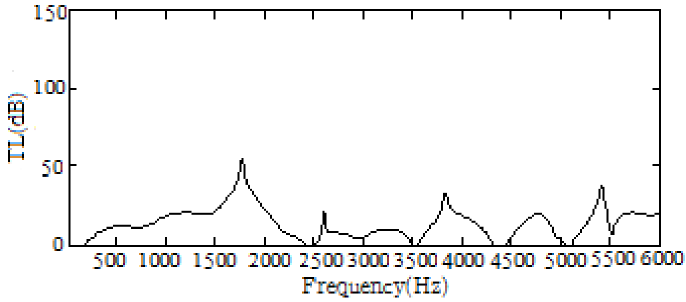


Fig.13 Transmission loss of the first cavity

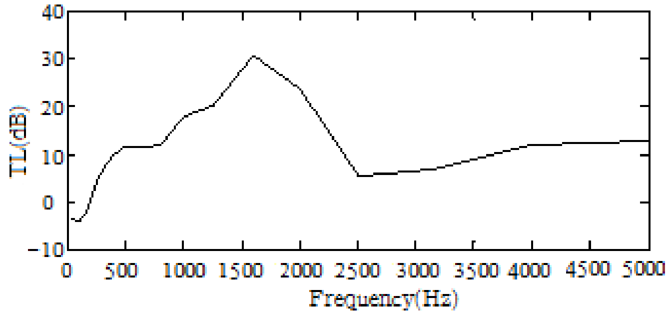


Fig.14 Transmission loss of the first cavity-- 1/3oct

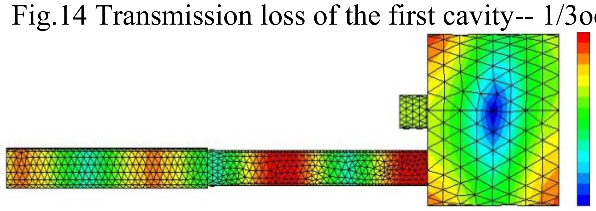


Fig.15 Sound pressure nephogram of the first cavity at 2460 Hz

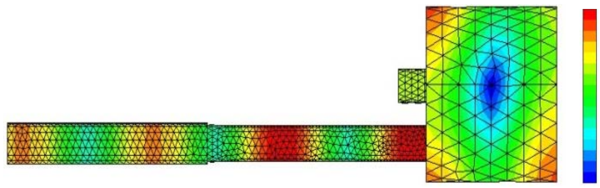


Fig.16 Sound pressure nephogram of the first cavity at 3540 Hz

2.2.2 Acoustic characteristic of the first cavity

Analyze the first cavity alone, adopting the pressure of the input and output nodes, remove the reflected sound pressure, Fig.13 shows the original transmission loss curve. Fig.14 shows the corresponding one-third of the transmission loss curve. The passing frequency is at 2460Hz、3540 Hz、4380 Hz and 5080 Hz, transmission loss is not obvious and silencing effect is not good at the frequency of 5520 Hz through analyzing the Fig.14. Fig.15 and Fig.16 show the whole simulation of sound pressure nephogram at 2460Hz、3540 Hz:

2.2.3 Acoustic characteristic of the second cavity

Analyze the second cavity alone, Adopting the pressure of the input and output nodes, remove the reflected sound pressure, Fig.17 shows the original transmission loss curve. Fig.18 shows the corresponding 1/3 times transmission loss curve. The passing frequency is at 2460Hz、3540 Hz、4100 Hz，5780 Hz; transmission loss is obvious and silencing effect is good near 2000Hz through analyzing the Fig.2-19 and Fig.2-20 show the whole simulation of sound pressure nephogram at 2460Hz、3540 Hz:

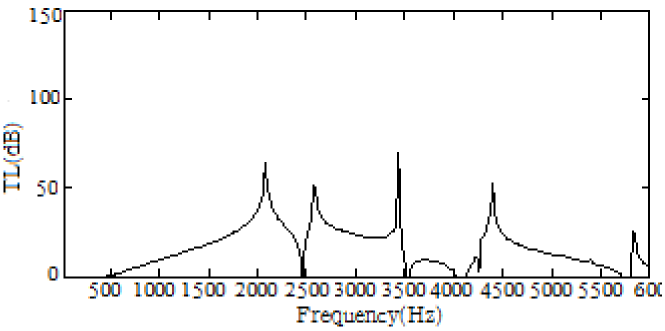


Fig.17 Transmission loss of the second cavity

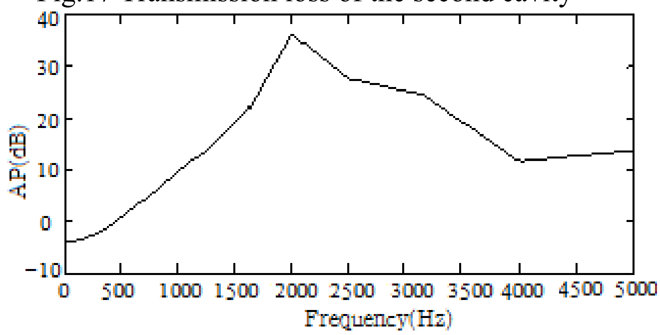


Fig.18 Transmission loss of the second cavity-- 1/3 oct

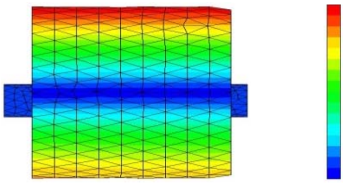


Fig.19 Sound pressure nephogram of the second cavity at 2460 Hz

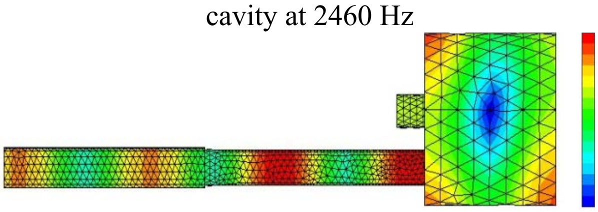


Fig.20 Sound pressure nephogram of the second cavity at 3540Hz

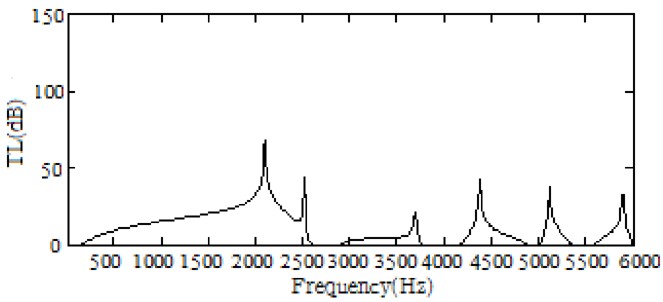


Fig.21 Transmission loss of the third cavity

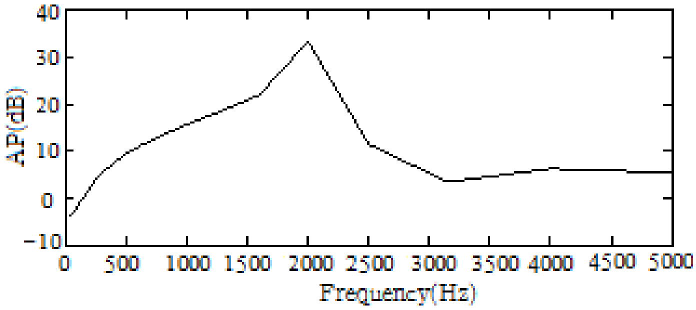


Fig.22 Transmission loss of the third cavity-1/3oct

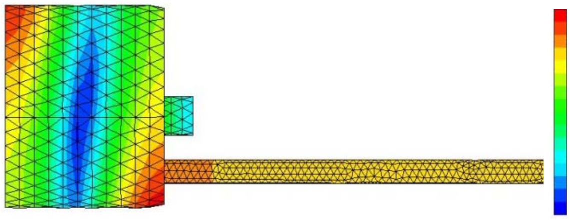


Fig.23 Sound pressure nephogram of the second cavity at 2720Hz

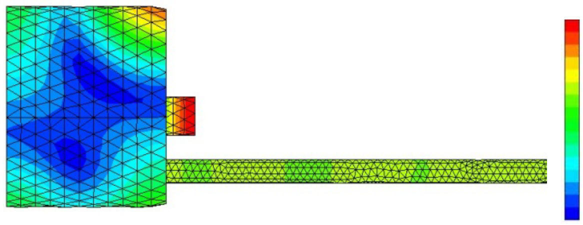


Fig.24 Sound pressure nephogram of the second cavity at 3800Hz

2.2.4 Acoustic characteristic of the third cavity

Analyze the third cavity alone, adopting the pressure of the input and output nodes, remove the reflected sound pressure, Fig.21 shows the original transmission loss curve. Fig.22 shows the corresponding one-third of the transmission loss curve. Transmission loss is not obvious and silencing effect is not good at the frequency of 2720Hz，3800Hz—4160 Hz，4980 Hz，5440 Hz and 6000 Hz through analyzing the Fig.21.

Fig.22 shows that transmission loss is obvious and silencing effect is good. Fig.23 and Fig.24 show the whole simulation of sound pressure nephogram at 2720Hz and 3800 Hz.

3 CFD(Computational Fluid Dynamics) Simulation on Muffler

After the boundary condition setting is finished, and the boundary conditions are initialized, then the count enough iterations is selected. CFD simulation[10] of flow field on muffler is performed, and the convergence calculation results of the post-processing analysis are made. Relevant performance of the muffler on flow field is researched. Fig. 25 shows the residual plots of finite element mesh model to exhaust muffler, its self-convergence is achieved after iteration 1536 steps.

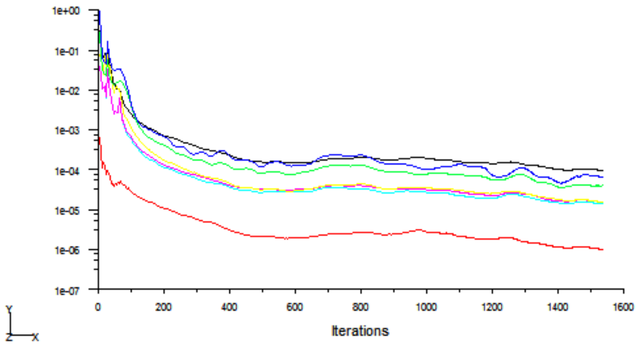


Fig.25 Convergence residual plots of exhaust muffler

Direction and magnitude of air velocity impact spread stream of sound waves, so that the boundary conditions on the surface of the structure changed accordingly, thus affecting the attenuation of acoustic wave propagation in the muffler when, so it reduces the silencing performance of the muffler, while if the flow rate is too high pressure loss muffler will increase the pressure loss is proportional to the square of the flow rate of gas flow inside the muffler. Fig.26 shows internal airflow velocity field of the engine muffler after the nominal operating conditions.

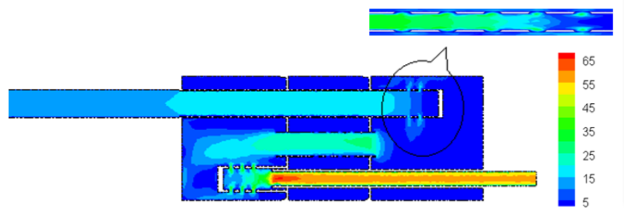


Fig.26 The internal airflow velocity field of the engine muffler after the nominal operating conditions

Engine exhaust flows into the left end of the muffler inlet pipe, velocity is 10 m / s the inlet tube. With the flow of the gas stream, t firstly enters the he rightmost first chamber in which the perforation holes around the tube, the velocity gradient is large, the velocity of the other parts is roughly 5 m / s . From local figure of airflow velocity distribution of muffler perforation holes in the first chamber can be seen that the velocity of around the holes up to 40m / s, the minimum is only 5m / s, the air flow velocity is subtracted from the left hole to the right hole in turn. The airflow along the perforated pipe orifice flows into the second chamber, the flow rate within the buffer tube reach about 20m / s, the flow rate in the chamber more uniform about 5 m / s. Then airflow flows into the third chamber muffler, the variation of flow rate is consistent with the first chamber. In the vicinity of perforated pipe velocity gradients when the airflow flows to the connecting pipe of the muffler pipe, because the e diameter of the connecting pip decreases, the velocity increases to maximum about 65m / s, which makes the muffler outlet flow rate can be 60m / s.

4 Theoretical Calculation of the Muffler Performance and Analyses

The top limit failure frequency of resistance muffler could be calculated by the equation:

$$f_{上} = 1.22\frac{c}{d} \tag{9}$$

Where c represents sound velocity (m/s); d represents interface feature size of the expansion chamber.

The lower limit failure frequency of resistance muffler can calculated by the equation:

$$f_{下} = \sqrt{2} f_0 = \frac{\sqrt{2}c}{2\pi}\sqrt{\frac{S}{lV}} \tag{10}$$

Where c represents sound velocity (m/s);S represents sectional area of airflow channel; V represents expansion chamber volume; l represents expansion chamber length.

The passing frequency of expansion cavity can be calculated by the equation:

$$f_n = \frac{nc}{2l} \tag{11}$$

Where n represents positive integer; c represents sound velocity (m/s); l represents expansion chamber length (m).

Table 3 shows the failure frequency of every cavity,

Table 3 Acoustic characteristic of every cavity

	first	second	third	whole （remove middle partition ）
Types of silencer	Expansion cavity	Resonance cavity	Expansion cavity	Expansion cavity

top limit failure frequency (Hz)	6731.17	-	6731.17	6731.17
lower limit failure frequency (Hz)	1.12	-	1.16	0.42
Passing frequency (Hz)	2466	-	2678.54	923.5
Resonance frequency (Hz)	-	952.23	-	-
Punching rate (%)	8.97	5.12	4.58	-

According to the analysis of the whole muffler and each cavity, the silencing effect is obvious near 2000Hz and 1600Hz. The silencing effect is not obvious near 320Hz and its 2times 8times and so on. Combined with the analysis of main noise frequency, which the silencing effect of the low frequency need to be improved.

According to the basic theory of the muffler, silencer muffler structure and volume is related, the bigger the better volume. But according to the requirements of the manufacturer, the muffler housing shape couldn't be changed, while ensuring that the power loss in less than 5%, sound deadening capacity couldn't be increased by increasing the volume of the muffler, this paper proposed a good effect and low cost solutions which Based on the structure of the first chamber muffler modification . The first chamber is divided into two silencer unit, before muffler defects; the latter half of the chamber is a simple plug-in expansion chamber, which used the ideal cavity 1/2, 1/4, 1/4, 1/2 structure. The improved first cavity is shown in Fig.27. At the same time, the punching rate of the 1/2L inlet tube part is increased from 5% to 10%. The inter mediate tube is increased to 1/4L part in the first cavity, and increased the punching rate. Half-cavity resonance cavity, the purpose is to eliminate the low frequency of 320Hz-500Hz band. The simulation result analyzed the transmission loss of this program between original with optimization is shown in Figure 28.

Fig.27 The improved first cavity structure

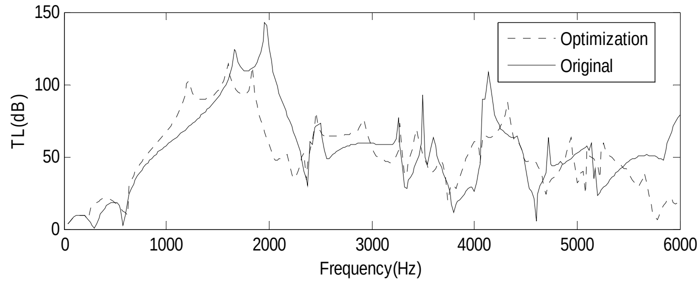


Fig.28 Transmission loss of whole muffler between original with optimization

Figure 28 shows that the improved scheme can obviously improve the low frequency (320Hz) sound deadening performance of the muffler, the sound deadening performance for high frequency are also a small amount of an increase, optimization purposes is achieved.

5 Conclusion

In this paper, finite element model of muffler was established. Then according to the SYSNOISE software, acoustic characteristic of the whole muffler and each cavity were analyzed. Through theoretical calculation of the muffler performance and analyses, the silencing effect is obvious near 2000Hz and 1600Hz. The silencing effect is not obvious near 320Hz and its 2times 8times and so on. The first chamber is improved. The simulation result shows that the improved scheme can obviously improve the low frequency sound deadening performance of the muffler.

Acknowledgments

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