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Modal Analysis of Muffler Based on ANSYS Workbench

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Abstract: This article uses Solidwork software to complete the three-dimensional modeling of the muffler, Using ANSYS Workbench 15.0 Simulation Software, Analyzing change of vibration mode of muffler with or without fluid-structure interaction, Under the fluid-structure interaction, the muffler boundary constraint inlet velocity and outlet pressure modal analysis were used to obtain the finite element analysis results of the first ten natural frequencies, mode characteristics, and muffler stress response of the muffler.
Keywords: muffler, modal analysis, ANSYS Workbench, fluid-structure coupling, natural frequency and mode shape.

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1. INTRODUCTION

A muffler is a device that allows air to pass through while attenuating or obstructing sound propagation. It controls aerodynamic noise and is simple and effective. Muffler is an important part to reduce noise, its performance will directly affect the size of the noise and power loss. The plug-in expansion chamber silencer has a simple structure, is easy to manufacture, and has a large amount of muffler, and is widely used in various types of compressors to control the noise. An expansion chamber muffler is also known as an expansion chamber muffler. An expansion chamber resistance muffler, also known as an expansion chamber muffler, is composed of pipes and chambers. It uses the abrupt expansion or contraction of the pipe section to produce abrupt changes in the acoustic impedance of the pipe, so that sound waves at certain frequencies that propagate in the pipe are reversed in time, and these sound waves cannot be transmitted through the muffler and thus achieve the purpose of clearing the sound[1].

This article examines the plug-in expansion chamber muffler, which, while reducing exhaust noise, is itself subject to various excitations that can generate vibrations. The vibration of the muffler itself not only brings about noise but also affects the muffling quality of the muffler to a certain extent. Therefore, it is necessary to research and improve the structure of the muffler itself.

2. MUFFLER FINITE ELEMENT MODEL

The basic structural parameters of the plug-in expansion chamber muffler obtained in this paper are: the inlet and outlet diameters of the muffler are both 20 mm, the diameter and length of the expansion chamber are 70.34 mm and 40 mm, respectively, and the wall thickness of the muffler is 2 mm[7]. The polybutylene terephthalate has an elastic modulus of 4000 MPa, a Poisson's ratio of 0.39[6], and

a density of 7800 kg/m3. Through the use of SolidWorks software for the establishment of three-dimensional solid model, and in the ANSYS Workbench software for the three-dimensional model cell grid division[2]. The finite element model of the muffler is shown in Figure 1.

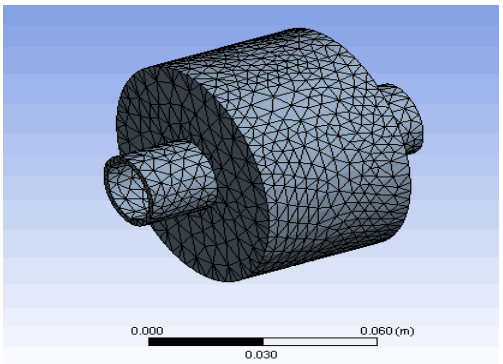


Fig. 1 Muffler finite element model

3. BASIC THEORY OF MODAL ANALYSIS

Modal analysis can determine the vibration parameters (natural frequency and mode shape) of a structural or mechanical part as well as the dynamic analysis of transient dynamics, harmonic response, response spectrum, etc. It is also the starting point of dynamic analysis[3].

For pipeline problems, the amount of fluid pulsation in the pipeline is small compared to the average, and it can be assumed that the fluid is under small disturbance conditions[5]. The average flow rate in the compressor line is generally 10-25 m/s, which is very small compared to the speed of sound (20 °C air velocity is 340 m/s). Therefore, the average flow velocity effect can be ignored. Assuming that the fluid's average density and pressure are constant, the fluid momentum and continuity equations can simplify the acoustic wave equation:

$$\frac{1}{C^2}\frac{\partial^2 P}{\partial t^2}-\nabla^2 p=0 \tag{1}$$

Among them $C=\sqrt{k/\rho_0}$ is the speed of sound in a fluid medium (where: the average density of the fluid; the bulk modulus of the fluid); P is the sound pressure; ∇^2 Laplacian operator.

Equation (1) is multiplied by a imaginary pressure and integrated in the entire fluid domain using the Galerkin method. According to the hypothesis, the relationship between the normal acceleration of the pipeline and the normal pressure gradient of the fluid can be obtained from the fluid momentum equation. Then, the pipeline and fluid are separated by pressure and displacement variables, respectively, and expressed as a matrix, and a discrete wave equation is obtained:

$$\mathrm{M}_e^f\ddot{p}_e+K_e^f p_e+\rho_0 R_e^T\ddot{u}_e=0 \tag{2}$$

In the formula $\mathrm{M}_e^f=\frac{1}{C\int_v NN^T dv}$ is fluid mass matrix ; $K_e^f=\int_v B^T B dv$ is fluid stiffness matrix; $\rho_0 R_e^T=\rho_0\int_s Nn^T\overline{N}^T ds$ is the coupling mass matrix of the fluid-structure coupling interface; N、$\overline{N}$ is the unit shape functions of pressure and displacement, respectively; P_e is the node pressure vector; u_e is the node displacement vector; is the volume of the entire basin; is the interface between fluid and structure; $B=LN^T$ $L=\left[\frac{\partial}{\partial x}\frac{\partial}{\partial y}\frac{\partial}{\partial z}\right]$ (L is a matrix operator) Is the normal vector of the solid interface.

Considering the energy dissipation of the fluid due to damping, the wave equation can be written as:

$$\mathrm{M}_e^f \ddot{p}_e + C_e^f \dot{p}_e + K_e^f p_e + \boldsymbol{\rho}_0 R_e^T \ddot{u}_e = 0 \tag{3}$$

Among them $C_e^f = \frac{B}{C}\int_s NN^T ds$ is the fluid damping matrix, $\boldsymbol{\beta}$ is the boundary absorption coefficient.

At the interface between the fluid and the pipeline, there exists a fluid-pipe coupling effect. The fluid pressure exerts a surface force on the pipeline, and the fluid pressure acting on the fluid-solid interface is added to the dynamic equation of the pipeline. Pipe vibration finite element equation:

$$\mathrm{M}_e^f \ddot{u}_e + C_e^s \dot{u}_e + K_e^f u_e = F_e^s + F_e^f \tag{4}$$

In the formula F_e^s is the load on the pipeline; $F_e^s = R_e P_e$ is the pressure load of the fluid on the pipeline $R_e = \int_e \overline{N} N^T n d(s)$, skill $F_e^f = R_e P_e$ the finite element equation (4) substituted into the pipeline is obtained

$$\mathrm{M}_e^f \ddot{u}_e + C_e^s \dot{u}_e + K_e^f u_e - R_e P_e = F_e^s \tag{5}$$

Equations (2) and (5) give the finite-element discrete equations for the complete fluid-pipe coupling, which can be unified as follows

$$\begin{bmatrix} M_e^s & 0 \\ M_e^{fs} & M_e^a \end{bmatrix} \begin{Bmatrix} \ddot{u}_e \\ \ddot{p}_e \end{Bmatrix} + \begin{bmatrix} C_e^s & 0 \\ 0 & C_e^f \end{bmatrix} \begin{Bmatrix} \dot{u}_e \\ \dot{p}_e \end{Bmatrix} + \begin{bmatrix} K_e^s & K_e^{fs} \\ 0 & K_e^f \end{bmatrix} \begin{Bmatrix} u_e \\ p_e \end{Bmatrix} = \begin{Bmatrix} F_e^f \\ 0 \end{Bmatrix} \tag{6}$$

This is the direct coupling equation between fluid and pipeline, among them, $M_e^{fs} = \boldsymbol{\rho}_0 R_e^T$, $K_e^{fs} = -R_e$ is a mutual coupling item.

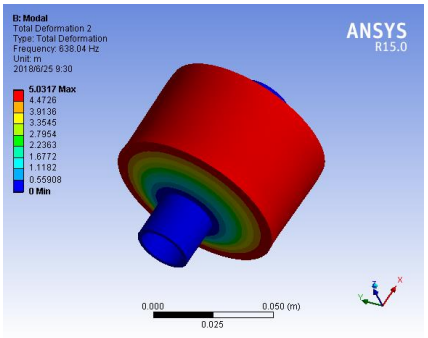
4. MUFFLER MODAL ANALYSIS

The muffler can meet the vibration requirements and achieve a good result. In addition to avoiding resonance due to various stimuli, it is also necessary to minimize its own vibration.

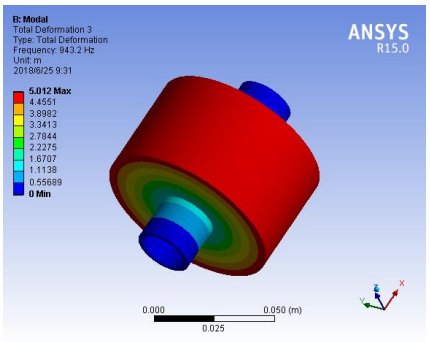
This paper uses the Workbench module in the ANSYS software to calculate the natural frequency and mode shape of the muffler when it is free to vibrate. Inlet flow rate, outlet pressure, no load is applied. Because the low-order vibration mode has a greater influence on the dynamic characteristics of the structure than the high-order mode, this study first analyzes the mode of the muffler, extracts the first ten natural frequencies and modes, and analyzes the modes. The first ten natural frequencies and ten of them are shown in Table 1 and Figure 2, respectively.

Table 1 The first ten natural frequencies of the silencer without fluid

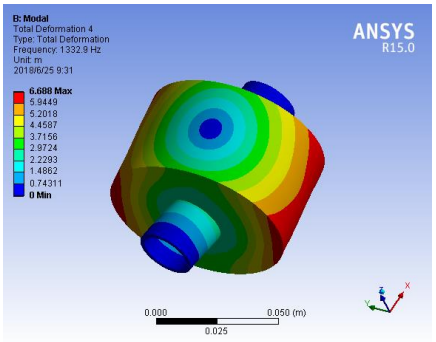
Modal order	Free modal natural frequency/Hz		
1	638.04	6	1622.7
2	943.20	7	3434.8
3	1332.9	8	3435.1
4	1333.2	9	3562.8
5	1621.9	10	3563.8



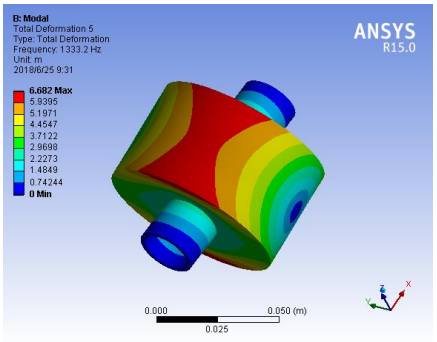
Muffler first vibration mode



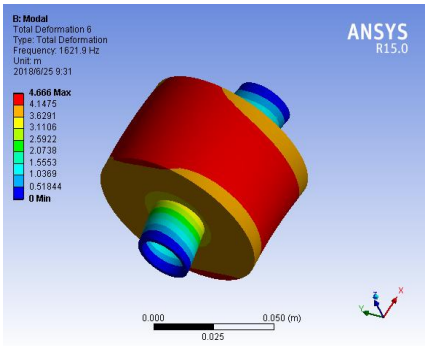
Muffler second vibration mode



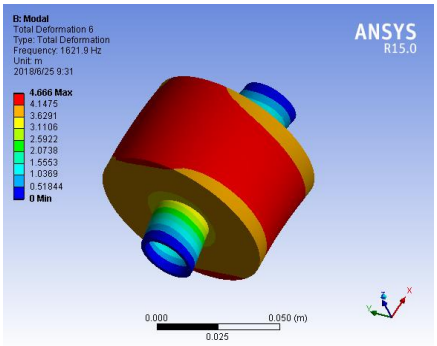
Muffler third vibration mode



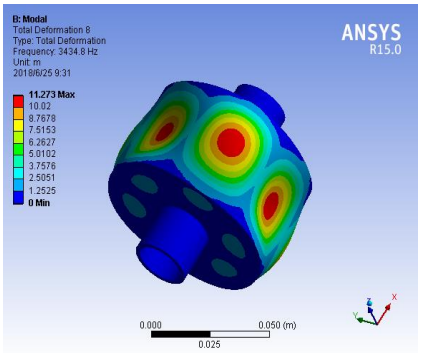
Muffler fourth vibration mode



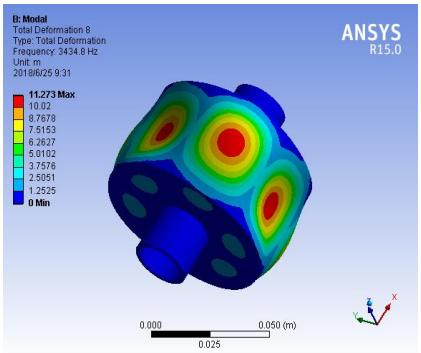
Muffler fifth vibration mode



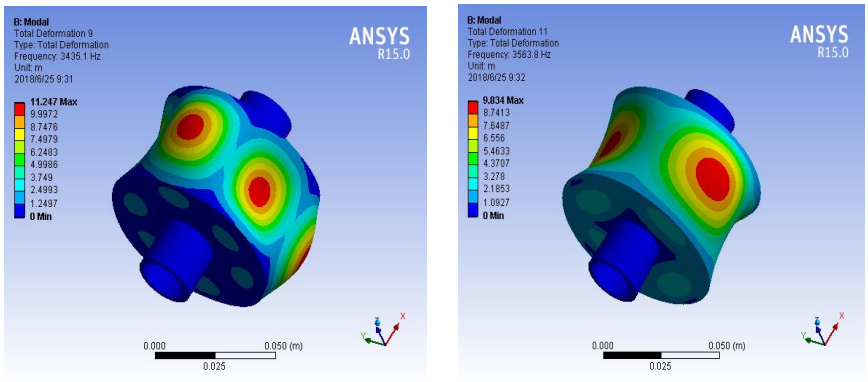
Muffler sixth vibration mode



Muffler seventh vibration mode



Muffler The eighth-order mode



Muffler ninth mode Muffler tenth mode

Figure. 2 The first to the tenth modes of a no-flow state muffler

And use the Workbench module inside the ANSYS software to calculate the natural frequency and mode shape of the muffler in the fluid state. Set boundary conditions, inlet flow rate, and outlet pressure without applying any load. This study first performed a modal analysis of the muffler, extracted the natural frequency and mode shape, and analyzed the mode shape. The natural frequency is shown in Table 2.

Table. 2 The muffler has the first ten natural frequencies of the fluid

Modal order		Free modal natural frequency /Hz	
1	666.69	6	1620.6
2	994.45	7	3626.6
3	1374.6	8	3327.1
4	1374.7	9	3638.0
5	1619.9	10	3639.7

It can be seen from Table 2 that due to the fluid-solid interaction between the oil and the pipeline, the frequency of vibration of the pipeline slightly increases.

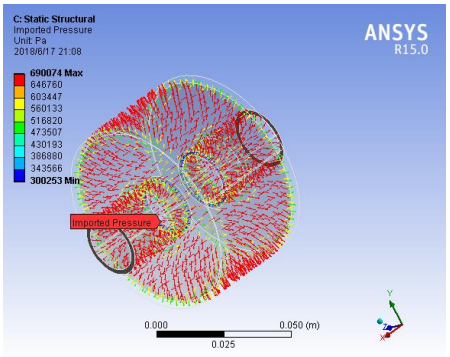


Figure. 3 Pressure of the fluid on the wall of the pipe

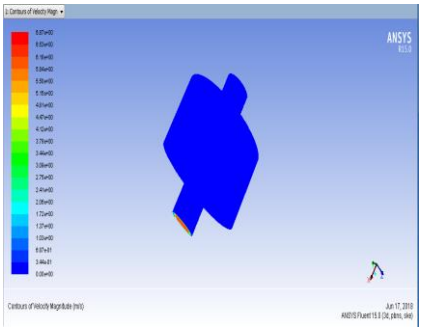


Figure. 4 In-tube flow rate diagram

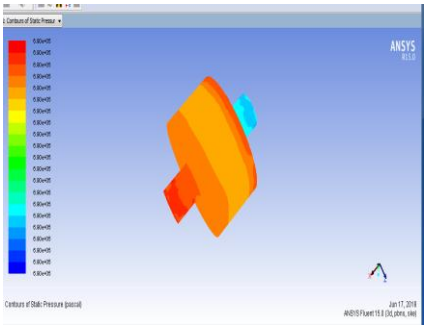


Figure. 5 Pressure cloud

Figure. 3 is a structural pressure cloud diagram of the fluid of the muffler against the pipe wall. It can be seen that the pressure on the pipe wall at the expansion chamber is slightly lower and the pressure on the pipe wall at the inlet and outlet is greater.

Due to the effect of the fluid, coupled at the wall of the tube, the flow velocity at the inlet is higher and the flow velocity to the outlet decreases in turn as the velocity boundary conditions are set at the inlet. Figure 4 shows that the speed at the center reaches its maximum value.

Figure 5 is a pressure cloud diagram of the muffler. The pressure is in a band-like distribution in the muffler, and the distribution is the same as that of the velocity cloud. The pressure at the inlet is the largest, decreases toward the outlet, and the pressure at the outlet is the smallest.

5. CONCLUSION

In this paper, through the modal analysis of the single-cavity expanding muffler, the first ten natural frequencies and corresponding modal characteristics of the muffler with one end fixed at one end are obtained, and the mode of deformation is more obvious for several orders of deformation. Analyze, draw some laws, propose improvement programs, and establish the structure of the muffler and the analysis of the direct fluid-structure coupled finite element model considering the fluid flow in the pipe in ANSYS. The conclusions are as follows:

1) By increasing the wall thickness of the muffler, the deformation of the muffler can be reduced to some extent.

2) For the case where the muffler has a large deformation at one end, by using a non-uniform wall thickness, increasing the wall thickness of the free end pipe can largely reduce the deformation of the free end.

3) The existence of fluid-structure interaction makes the pipe vibration frequency slightly increase. Increasing the restraint properly will increase the vibration frequency of the pipeline, significantly reduce the equivalent stress of the pipeline, and reduce the total deformation of the pipeline, thereby reducing the fatigue damage to the pipeline caused by liquid shock, improving the structural rigidity of the pipeline system, and improving the The natural frequency of the pipe structure keeps it away from the excitation frequency, avoids the occurrence of resonance, and reduces the vibration amplitude of the pipe structure.

4) The purpose of optimizing the layout of the piping system is to change the number and position of the constraints appropriately, and finally reduce the vibration of the pipeline, reduce the local energy loss of the pipeline and reduce the noise.

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